This is a thesis accepted for a Higher Degree of the University of London. It is an unpublished typescript and the copyright is held by the author. All persons consulting the thesis must read and abide by the Copyright Declaration below.

COPYRIGHT DECLARATION
I recognise that the copyright of the above-described thesis rests with the author and that no quotation from it or information derived from it may be published without the prior written consent of the author.

LOANS
Theses may not be lent to individuals, but the Senate House Library may lend a copy to approved libraries within the United Kingdom, for consultation solely on the premises of those libraries. Application should be made to: Inter-Library Loans, Senate House Library, Senate House, Malet Street, London WC1E 7HU.

REPRODUCTION
University of London theses may not be reproduced without explicit written permission from the Senate House Library. Enquiries should be addressed to the Theses Section of the Library. Regulations concerning reproduction vary according to the date of acceptance of the thesis and are listed below as guidelines.

A. Before 1962. Permission granted only upon the prior written consent of the author. (The Senate House Library will provide addresses where possible).

B. 1962 - 1974. In many cases the author has agreed to permit copying upon completion of a Copyright Declaration.

C. 1975 - 1988. Most theses may be copied upon completion of a Copyright Declaration.

D. 1989 onwards. Most theses may be copied.

This thesis comes within category D.

☐ This copy has been deposited in the Library of

☐ This copy has been deposited in the Senate House Library, Senate House, Malet Street, London WC1E 7HU.
METHODS FOR ASSESSING ENVIRONMENTAL, SAFETY AND PERFORMANCE OF ALTERNATIVE REFRIGERANTS

Daniel Henry Colbourne

A thesis submitted for the degree of Doctor of Philosophy in the Department of Mechanical Engineering, University College London

June 2006
DECLARATION

This declares that the content presented in this thesis is the work of the author.

Daniel Henry Colbourne.

Date: 20th June 2006
ABSTRACT

This thesis addresses the general subject of implementation of refrigerants that are alternatives for ozone depleting substances. In particular it covers three topics, which are environmental assessment, system performance and safety aspects related to flammability. The first Part examines the current means of assessing the global warming impact of systems, and proposes an improved approach. The second Part presents a new balancing technique for designing systems that use refrigerants with a temperature glide, as an advancement over the conventional technique for pure refrigerants. The third Part derives a quantitative risk assessment model to be employed for evaluating the safety of hazard of flammable refrigerants.

The concept of “Total Equivalent Warming Impact” is introduced in Part 1, which is currently used by manufacturers, installers and end-users of refrigerating systems to evaluate the contribution to climate change resulting from energy consumption and refrigerant leakage. It was found that existing methods rely on many assumptions, so a thorough approach was proposed to assess the relevance of the assumptions and providing ways of avoiding them. An alternative measure for interpretation of consequences of greenhouse gas emissions is also included. Part 2 investigated the existing approach for rating and balancing system components, which were found to be incompatible when applied to zeotropic refrigerants. Differences in the performance of components using pure and zeotropic refrigerants were identified, and so a new method for component rating and an advanced system balancing technique were developed based on the properties specific to zeotropes. Within Part 3, a framework for calculating ignition frequencies and consequences of a release of hydrocarbon refrigerant is presented. A major element of these calculations is the evaluation of certain quantities of the flammable mixture, and the model developed for this purpose was based on the results from a series of gas dispersion experiments.
ACKNOWLEDGEMENTS

The Author would like to express appreciation to many people. First, my tutor, Dr. K. O. Suen, for his incredible tolerance and stamina, his endless enthusiasm and encouragement, and for making available his inspirational way of thinking, understanding and addressing all matters technical. Also, my parents, Jill and Alan, for their encouragement, support, motivation, financial help, and of course my creation and subsequent existence. Notwithstanding, my brother Rupert, particularly for the invaluable help with coding and moreover the highly amusing observations about things irrelevant to this work. Many colleagues from my previous employer Calor Gas Ltd., including Paul Blacklock and Loretta Powell for their support and encouragement, but in particular Dr. Terry Ritter for his keen interest, enthusiastic discussions, and moral support. Lastly, thanks to a formidable set of friends whom for years have endured endless lectures and discussions whilst having a profound ability to convincingly express fascination in the subjects detailed herein.
TABLE OF CONTENTS

LIST OF TABLES 11
LIST OF FIGURES 13
GENERAL INTRODUCTION 20

PART 1: METHODS FOR EVALUATING THE GLOBAL WARMING IMPACT OF REFRIGERATING SYSTEMS 23
NOMENCLATURE 24
SECTION 1: INTRODUCTION 29
1.1 Environmental burden of refrigerants 29
1.2 Literature review 30
   1.2.1 Existing methodologies for evaluating environmental impacts 30
   1.2.2 Survey of comparative studies 32
   1.2.3 Measures of climatic impacts 34
1.3 Formulation of objectives 37
   1.3.1 Summary of findings 38
   1.3.2 Aims and objectives 39
SECTION 2: DEVELOPMENT OF EMISSIONS MODEL 41
2.1 Modelling approach 41
2.2 Start of life stage 43
   2.2.1 Refrigerant and component manufacture 43
   2.2.2 Construction, assembly and installation 49
2.3 In-use stage 50
   2.3.1 System refrigerant emissions 51
   2.3.2 System energy use emissions 54
   2.3.3 Emissions from servicing and maintenance activities 70
2.4 End of life stage 73
   2.4.1 Dismantling 74
   2.4.2 Disposal of refrigerant and equipment 75
SECTION 3: CLIMATE IMPACTS 76
3.1 Options for evaluating climate change impact 76
3.2 Global warming potential 77
3.3 Global temperature change potential 78
3.4 Total lifetime temperature impact (TLTI) 79
SECTION 4: RESULTS AND DISCUSSION

4.1 Implementation of model
4.2 Breakdown and uncertainty of emissions
4.2.1 Breakdown of emission sources
4.2.2 Uncertainties of emissions
4.3 Sensitivity to transient influences
4.3.1 External conditions
4.3.2 Ageing
4.3.3 Power station emissions
4.4 Influences of equipment design
4.4.1 Pipework and connections
4.4.2 Heat exchanger size
4.4.3 Compressor size
4.4.4 Refrigerant selection
4.5 Effects of material/refrigerant handling and operations
4.5.1 Refrigerant handling practices and recycling
4.5.2 Condenser and system cleaning
4.5.3 Leak checks and repair
4.6 Comparison with BRA method
4.7 Interpretation of global warming
4.8 Use and advantages of methodology

SECTION 5: CONCLUSIONS

5.1 Summary of work
5.2 Key findings
5.3 Limitations and further work

REFERENCES

APPENDICES

Appendix 1: Compilation of sources found to affect emissions

PART 2: DEVELOPING A NEW SYSTEM BALANCING TECHNIQUE FOR MIXTURE REFRIGERANTS
| Table 1: GHG (refrigerant) data from IPCC/TEAP (2005) | 36 |
| Table 2: Manufacturing emissions of refrigerants | 45 |
| Table 3: Component construction data | 46 |
| Table 4: Emissions resulting from raw material manufacture | 48 |
| Table 5: Emissions during construction, assembly and installation | 49 |
| Table 6: Refrigerant leak quantity and leak factor for specific components | 52 |
| Table 7: Comparison of reported annual leak rates | 53 |
| Table 8: Constants for emissions factor projections | 55 |
| Table 9: Constants for equation (36) for two different compressors | 63 |
| Table 10: Constants for fouling equation (45) | 67 |
| Table 11: Studies on effects of refrigerant leakage | 68 |
| Table 12: Details of servicing activities leading to refrigerant emissions | 71 |
| Table 13: Summary of reported venting probabilities | 72 |
| Table 14: Reference input parameters and basic outputs | 83 |
| Table 15: Range of emission contributions from different sources | 85 |
| Table 16: Effect of refrigerant leakage from different component assemblies | 90 |
| Table 17: Refrigerant comparison (a) with fixed system design | 94 |
| Table 18: Refrigerant comparison (b) with fixed system capacity at design conditions | 96 |
| Table 19: Refrigerant comparison (c) with fixed system COP at design conditions | 97 |
| Table 20: Input parameters for refrigerant handling practice | 100 |
| Table 21: Energy calculations according to BRA method | 104 |
| Table 22: Standard calculations according to BRA method | 105 |
| Table 23: Comparison of climate impact for different systems | 107 |
| Table 24: Exhaustive list of sources of emissions and influencing parameters | 124 |
| Table 25: Heat exchanger simulation input values | 186 |
| Table 26: Contribution of TEV inlet temperature to factors affecting capacity | 195 |
| Table 27: Comparison of capacity and temperature variables for balancing | 213 |
| Table 28: Comparison of capacity and reference temperatures for balancing procedure | 222 |
| Table 29: Specific input data for calculation of fictitious rating data | 224 |
| Table 30: Summary of usable data | 235 |
| Table 31: Commercial zeotropic mixtures and selected properties | 242 |
| Table 32: Constants for water property polynomial equation (265) | 245 |
| Table 33: Overview of new balancing technique | 246 |
Table 34: Comparison of equilibrium-points for example balancing exercises  
Table 35: Summary of the ways that main parameters were determined in QRAs  
Table 36: Maximum acceptable risk criteria  
Table 37: Information on mechanical component failures  
Table 38: Frequency of mean monthly temperatures in West London  
Table 39: Summary of test parameters  
Table 40: Information on RAC equipment  
Table 41: Information on instrumentation used  
Table 42: General test schedule  
Table 43: Sample point location and representative volume  
Table 44: Information on test parameters  
Table 45: Summary of influencing variables  
Table 46: Constants/coefficients and indices for pi-groups and for the adjustment factor  
Table 47: Range of applicability for correlations  
Table 48: Input details for dependence of flammable volume-time on number of elements  
Table 49: Reports of incidents involving HC refrigerants  
Table 50: Comparison with maximum acceptable risk criteria  
Table 51: Reference values and variations  
Table 52: Results of parametric evaluation  
Table 53: Reference values and variations  
Table 54: QRA model results for servicing on four applications  
Table 55: Summary of QRA results for stationary RAC equipment  
Table 56: Leak frequencies used in previous studies  
Table 57: Location of sensors and corresponding representative volume ratio  
Table 58: Parameter values and principle results for CO₂ release tests  
Table 59: Test variables and calculated results for Aloisi  
Table 60: Test variables and calculated results for Clodic et al  
Table 61: Test variables and calculated results for Jabbour et al, 2003  
Table 62: Test variables and calculated results for Kataoka et al  
Table 63: Flammable properties of hydrocarbons and selected refrigerants  
Table 64: Data from overpressure measurements and model results
LIST OF FIGURES

Figure 1: Identification of sources and types of GHG emissions 42
Figure 2: Activities at start of life stage causing emissions 43
Figure 3: Data for compressor and compressor oil mass 47
Figure 4: Data for condenser and evaporator mass 47
Figure 5: Data for pipe and valve mass 48
Figure 6: Overview of in-use emission sources and influencing parameters 50
Figure 7: Proportion of refrigerant leaks by mass and number of visits (Radford, 1998) 51
Figure 8: Historical and projected power station CO2 emission factor (DTI, 1998, 2000) 54
Figure 9: Options for different types of performance calculation 57
Figure 10: Changes in heat transfer of fouling over time 66
Figure 11: Comparison of leak effects on COP 69
Figure 12: Activities producing emissions during servicing 70
Figure 13: Activities affecting emissions during end of life process 74
Figure 14: Chain of climate relevant measures 76
Figure 15: Illustration of TLTI, GDI1 and GDI2 interpretation 80
Figure 16: Breakdown of emissions from energy and refrigerant for different stages (based on GWP(100), In-use values for a single year only) 84
Figure 17: Variation of contributions from different emission sources with size of equipment 85
Figure 18: Variation in monthly emissions from energy consumption for different ambient scenarios 86
Figure 19: Influence of various ageing effects on emissions from system energy consumption 87
Figure 20: Influence of change in power station emissions for electricity production 88
Figure 21: Effect of evaporator and condenser size on annual emissions 91
Figure 22: Influence of system sensitivity to charge size on annual emissions from energy consumption 91
Figure 23: Effect of sensitivity to charge size on system performance 92
Figure 24: Effect of compressor size on emissions from system operation 93
Figure 25: Comparison of total emissions for different refrigerants 98
Figure 26: Comparison of emissions for different service and handling practices 99
Figure 27: Influence of condenser cleaning and system cleaning on emissions 100
Figure 28: Influence of condenser and system cleaning on mean application temperature 101
Figure 29: Effect of servicing repair and top-up frequency on emissions 102

Figure 30: Effect of servicing repair and top-up frequency on mean application temperature (design application: -20°C) 103

Figure 31: Comparison of AGTP-time profile for high and low climate impact refrigerant 106

Figure 32: Climate impacts for different system characteristics relative to R22 reference system 108

Figure 33: Climate impacts for different system characteristics relative to R22 and R290 TEWI 109

Figure 34: Stages in the component-system chain 134

Figure 35: Pressure enthalpy diagram for (a) pure refrigerant, and (b) zeotropic refrigerant 145

Figure 36: Graphical representation of (a) condenser, and (b) evaporator characteristic for pure refrigerant (where B is constant) 149

Figure 37: Temperature profiles of counter-flow condenser for reducing condensing temperature with pure refrigerant 151

Figure 38: Effect of apparent capacity rate on effectiveness 153

Figure 39: Effect of apparent capacity rate on basic rating 153

Figure 40: Effect of temperature glide on C_{man}, \varepsilon and B in counter-flow exchanger 154

Figure 41: Temperature profiles for counter-flow condenser 155

Figure 42: Temperature profiles of parallel- and counter-flow condenser for reducing condensing temperature with glide 155

Figure 43: Minimum approach temperature differences for condenser and evaporator 156

Figure 44: Graphical representation of (a) condenser and (b) evaporator characteristic with zeotropic refrigerant 157

Figure 45: Variation of temperature glide (a) with saturation temperature, and (b) with vapour quality at atmospheric pressure for commercial refrigerant blends 158

Figure 46: Correlation for constants (a) c_{G1} and (b) c_{G2} for equation (101) 159

Figure 47: Variation of maximum and minimum glide index over (a) saturation temperature, and (b) saturation pressure 160

Figure 48: Change in condenser basic rating with inlet temperature for fixed (hollow data point) and variable (filled data point) temperature glide 162

Figure 49: Effect of different expansion device inlet temperature on (a) evaporator inlet temperature, and (b) on evaporator characteristic 163

Figure 50: Variation in evaporator basic rating with glide size at different T_{TEV(in)} 164

Figure 51: Variation of evaporator capacity with glide size due to change in TEV inlet temperature from 20°C to 60°C 164
Figure 84: Condenser capacity, against condensing dew-point temperature for a series of discharge superheats 226
Figure 85: Condenser outlet dew-point temperature, against condenser inlet dew-point temperature for a series of desuperheats 226
Figure 86: Condenser capacity against condensing dew-point temperature, for a series of evaporating temperatures 227
Figure 87: Condenser outlet dew-point temperature, against condensing inlet dew-point temperature for a series of evaporating temperatures 227
Figure 88: Compressor characteristic superimposed on new condenser characteristic to obtain condensing unit characteristic 227
Figure 89: Evaporator capacity against evaporating dew-point temperature for a series of condensing outlet dew-point temperatures 228
Figure 90: Evaporator characteristic corrected for condenser inlet 228
Figure 91: Compressor characteristic superimposed on new evaporator characteristic to obtain evaporating unit characteristic 229
Figure 92: System balance condensing dew-point temperature 230
Figure 93: System balance evaporating dew-point temperature 230
Figure 94: Error in $Q_e$ and $Q_c$ at balance points with (a) small size compressor, and (b) large size compressor for conventional and new techniques 231
Figure 95 (a) Layer formation in a quiescent room and, (b) concentration profiles 267
Figure 96: Positioning of air inlet and outlet positions and release locations within a room 270
Figure 97: Sequence of events leading to flammability hazard 276
Figure 98: Partial event tree illustrating variations of conditions 278
Figure 99: Calculation sequence to determine adjustment for selected number of leak sizes 285
Figure 100: Venn diagram indicating occurrences necessary for ignition 289
Figure 101: Time fraction for different infiltration rate ranges 297
Figure 102: Schematic arrangement of the test chamber apparatus 306
Figure 103: Comparison of maximum and mean maximum concentrations ($r^2 = 0.86$) 312
Figure 104: Observations from smoke-only tests 313
Figure 105: Observations from smoke-CO$_2$ tests 314
Figure 106: Concentration-time profiles for reference test (no. 10) 315
Figure 107: Concentration-height profiles for reference test (no. 10) 315
Figure 108: Concentration-height profiles for small release mass 316
Figure 109: Concentration-height profiles for large release mass 316
Figure 110: Effect of released mass on $\dot{C}_{\text{max}}$ ($r^2 = 0.96$) 316
Figure 111: Concentration-time profile for release with no airflow
Figure 112: Concentration profiles for low airflow rate (test no. 28)
Figure 113: Concentration profiles for high airflow rate (test no. 11)
Figure 114: Effect of AHU airflow rate on $\hat{C}_{f_{max}}$ ($r^2 = 0.85$)
Figure 115: Concentration-height profiles for downward air discharge (test no. 14)
Figure 116: Concentration-height profiles for upward air discharge (test no. 15)
Figure 117: Effect of air discharge angle on $\hat{C}_{f_{max}}$ ($r^2 = 0.82$)
Figure 118: Concentration-height profiles resulting from thermal loads (test no. 43)
Figure 119: Effect of post-ventilation period on $\hat{C}_{f_{max}}$ with 3.5 minutes release time.
Figure 120: Effect of release time with 3.5 minute post-ventilation on $\hat{C}_{f_{max}}$
Figure 121: Concentration-height profile for a 7 minute release (test no. 29)
Figure 122: Concentration-height profile for a 14 minute release (test no. 25)
Figure 123: Concentration-height profiles for low release height (test no. 20)
Figure 124: Concentration-height profiles for low release and discharge height (test no. 21)
Figure 125: Effect of release height on $\hat{C}_{f_{max}}$ ($r^2 = 0.96$)
Figure 126: Effect of release height and unit height ($h_u$) on $\hat{C}_{f_{max}}$ ($r^2 = 0.93$)
Figure 127: Effect of release location (horizontal distance from air disch.) on $\hat{C}_{f_{max}}$ ($r^2 = 0.93$)
Figure 128: Configuration of AHU airflows showing discharge and return air paths
Figure 129: Configuration of cabinet air flows
Figure 130: Concentration-height profiles for AHU #2 (test no. 32)
Figure 131: Concentration-height profiles for AHU #3 (test no. 36)
Figure 132: Concentration-height profiles for cabinet #1 (test no. 40)
Figure 133: Concentration-height profiles for cabinet #2 (test no. 41)
Figure 134: Comparison of measured room air speed ($r^2 = 0.92$)
Figure 135: Correlation for $\Pi_2$ ($r^2 = 0.88$)
Figure 136: Correlation for $\Pi_3$ ($r^2 = 0.97$)
Figure 137: Gradient of floor concentration development with time
Figure 138: Correlation between $n$ and mixing effectiveness ($r^2 = 0.74$)
Figure 139: Comparison of calculated against measured concentrations
Figure 140: Comparison of correlation against measurements (overall $r^2 = 0.87$)
Figure 141: Comparison of correlations against data from other studies ($r^2 = 0.78$)
Figure 142: Change in concentration over time, with forced airflow ceased at $t_d$ during release
Figure 143: Effect of release time exceeding airflow time on final floor concentration
Figure 144: Change in concentration with initiation and termination of airflow

Figure 145: Diagram showing basic concept of dispersion models

Figure 146: Flow chart for calculation sequence of flammable quantities

Figure 147: Comparison of plume spread correlations and measurements ($r^2 = 0.43$)

Figure 148: Correlation of friction factor and Reynolds no. for spread velocity ($r^2 = 0.67$)

Figure 149: Schematic diagram of plume model

Figure 150: Illustration of room velocity profile

Figure 151: Dependence of number of radial elements on flammable volume-time

Figure 152: Comparison of height-concentration profile at 120 s interval from model and
   Test no. 9 (no airflow)

Figure 153: Comparison of height-concentration profile at 120 s interval from model and
   Test no. 28 (low airflow)

Figure 154: Comparison of concentrations for reference case (Test no. 10)

Figure 155: Comparison of concentrations for low release height (Test no. 20)

Figure 156: Comparison of concentrations for low release and air disch. height (Test no. 21)

Figure 157: Concentration within individual layers

Figure 158: Cumulative volume of individual layers

Figure 159: Effect of released mass on $V^{t^R}$

Figure 160: Effect of room area on $V^{t^R}$

Figure 161: Effect of room air speed and discharge angle on $V^{t^R}$

Figure 162: Effect of release time and post-ventilation period on $V^{t^R}$

Figure 163: Effect of release height and air discharge height on $V^{t^R}$

Figure 164: Comparison of measured and calculated maximum overpressures

Figure 165: Deviation between measured and calculated maximum overpressures

Figure 166: Examples of room overpressure development

Figure 167: Effect of release mass with constant room size or constant $\hat{C}_{\text{km}}$ on consequence

Figure 168: Effect of room area with constant release mass or constant $\hat{C}_{\text{km}}$ on consequence

Figure 169: Effect of room airspeed on consequence

Figure 170: Effect of release height and airspeed on consequence

Figure 171: Effect of various venting routes on reduction of overpressure

Figure 172: Overall modelling sequence for the combined QRA

Figure 173: $f_{\text{leak}}, P_{\text{SO},}^{*}, P_{\text{V}}^{*}$ and $f^{*}$ for different leak sizes at $\Lambda_{\text{leak}} = 0.4 \text{ h}^{-1}$
Figure 174: Variation of $f_\ast$ with room level for different leak sizes for off-cycle
Figure 175: $f_\ast$ for different leak times for a range of infiltration rates
Figure 176: Overpressure and thermal intensity for different leak times
Figure 177: Integrated consequence-frequencies for different leak times
Figure 178: Overall ignition frequencies
Figure 179: Overall risk factor
Figure 180: Relationship between $\dot{C}_{\text{fmax}}$ and total $V^F t^F$ ($r^2 = 0.82$)
Figure 181: Relationship between $\dot{C}_{\text{fmax}}$ and the proportion of $V^F t^F$ above 1 m
Figure 182: Correlation between $P^F$ and $V^F t^F$ ($r^2 = 1$)
Figure 183: Relationship between $P^F$ and $\dot{C}_{\text{fmax}}$ ($r^2 = 0.82$)
Figure 184: Relationship between mean flammable mass and consequence ($r^2 = 0.99$)
Figure 185: Correlation between $\dot{C}_{\text{fmax}}$ and consequence ($r^2 = 0.83$)
Figure 186: Room orientation and reference directions
GENERAL INTRODUCTION

Background
In 1987 the Montreal Protocol on Substances that Deplete the Ozone Layer was ratified by the majority of industrialised and developing countries, resulting in legislation to phase out their use and production. This initiated a search for suitable replacements for chlorofluorocarbons (CFCs) and hydrofluorochlorocarbons (HCFCs), many of which were used as refrigerants. Since then a number of fluids have become accepted as alternative refrigerants, specifically hydrofluorocarbons (HFCs), hydrocarbons (HCs), carbon dioxide, and various others to a lesser extent. These have been adopted both as single component refrigerants, and as multi-component blends. Physical and chemical properties of these fluids differ from those of the traditional CFC and HCFC refrigerants, and since the design and performance of refrigerating systems is affected by these properties it is necessary to establish means of handling the resulting implications. Additional considerations for system design and performance aspects have arisen due to other legislation relating to environmental and safety matters. The work presented in this study comprise the development of methods that may be employed for assessing certain environmental, performance and safety aspects implicated in the application of these alternative refrigerants.

Motivation and description of work
The author was employed by a producer of alternative refrigerants, and in the course of carrying out technical duties relating to the implementation of these refrigerants, a number of issues affecting the selection and application of fluids were found to be inadequate or open to misuse. A general absence of efforts to develop suitable methods for correctly handling some of these issues was evident, and this provided the intermittent motivation for the author to produce the present study. The three issues dealt with are evaluation of the global warming impact of refrigerating systems, balancing of refrigerating system components that use mixture refrigerants with temperature glide, and investigating the flammability risk associated with use of HC refrigerants. Whilst broadly related, the three subjects handled in this study are diverse and this constituted three separate, self contained parts. As such, each part comprises an independent introduction, literature review, main body of work, conclusion and references, and the content is summarised below.

Extending Methods for Evaluating the Global Warming Impact of Refrigerating Systems.
An approach commonly termed, Total Equivalent Warming Impact (TEWI), is employed to evaluate the contribution of a systems global warming impact resulting from greenhouse gas
emissions from energy use and refrigerant release. The literature shows that whilst existing procedures for evaluating TEWI vary, it is always vague in terms of scope, assumptions and detail. In producing a thorough and more complete approach to estimating emissions, a detailed model was developed to include all possible sources of emissions throughout the equipment lifetime. To this end, influencing parameters were documented, and sub-models developed to accurately describe them. Using the model, the influence of a variety of equipment design features, external conditions, ageing, and repair and maintenance practices were examined in respect to their effect on the global warming contribution of refrigerating systems. Also, the use of the Global Warming Potential (GWP) quantity for TEWI calculations is known to have shortfalls whereby it does not suitably represent the relative impact of emissions of different substances. Therefore, the interpretation of the impact of global warming gases was addressed and alternative measures were adopted in place of GWP.

**Developing a New System Balancing Method for Mixture Refrigerants.**

Conventional methods for testing, rating, data presentation and selection of the main system components (compressor, expansion device, condenser and evaporator) are currently based on a standardised industry practice originally developed for pure refrigerants. System balancing techniques, which utilise performance data, are based on the same premise. Currently, the majority of commercialised alternatives refrigerants are zeotropic mixtures that display a temperature glide during phase-change. This impacts on the performance rating of refrigeration components and as such invalidates the current methods listed previously, particularly for the condenser and evaporator. The consequence of this is introduction of errors in the design of the refrigeration system. Modified methods are proposed to be employed within the testing, rating, data presentation and selection of components, and these changes are integrated into a revised balancing technique so that the behavioural aspects associated with zeotropic refrigerants, and other influences such as pressure loss and superheat, are accounted for.

**Appraising the Flammability Hazards of Hydrocarbon Refrigerants.**

HC refrigerants present fire and explosion hazards due to their flammability. To investigate this, a quantitative risk assessment (QRA) model was developed to evaluate the potential for ignition and consequences when hydrocarbons are employed in stationary refrigeration and air-conditioning equipment. Initially, an approach for calculating ignition frequencies, leakage frequency, probability of flammable concentrations, and associated consequences, being overpressures and thermal radiation was documented. Empirical data for leakage, hardware failure and sources of ignition was provided, as well as data from dispersion experiments was
used to develop sub-models. The findings from the dispersion experiments also allowed a method
to be developed for equipment design and installation features to assist with dispersing leaked
refrigerant. Various data were used to demonstrate the application of the QRA model, and in
particular, to examine the effects that design, installation of equipment and external conditions
have on the frequency of ignition and the associated consequences.
PART 1: METHODS FOR EVALUATING THE GLOBAL WARMING IMPACT OF REFRIGERATING SYSTEMS
NOMENCLATURE

Letters

\( a \) \quad \text{constant}
\( A \) \quad \text{area (m}^2\text{)}
\( AGTP \) \quad \text{absolute global temperature potential (K)}
\( AGWP \) \quad \text{absolute global warming potential (W m}^2\text{ kg}^{-1}\text{ y)}
\( A_{HTF} \) \quad \text{area of HTF-side exchanger surface (m}^2\text{)}
\( A_{sp} \) \quad \text{area for two-phase heat transfer (m}^2\text{)}
\( b \) \quad \text{constant rating (W)}
\( B_{app} \) \quad \text{application basic rating (W K}^{-1}\text{)}
\( c \) \quad \text{constant}
\( C' \) \quad \text{global surface heat capacity (J K}^{-1}\text{ W}^{-2}\text{)}
\( COP \) \quad \text{coefficient of performance (-)}
\( COP_{comp} \) \quad \text{compressor coefficient of performance (-)}
\( Cp \) \quad \text{specific heat at constant pressure (kJ kg}^{-1}\text{ K}^{-1}\text{)}
\( D \) \quad \text{diameter (m)}
\( E \) \quad \text{energy consumption (kW h)}
\( E_{anci} \) \quad \text{energy consumption of ancillary components (kW h)}
\( E_{comp} \) \quad \text{compressor energy consumption (kW h)}
\( F \) \quad \text{radiative forcing (W m}^2\text{ kg}^{-1}\text{)}
\( GDI \) \quad \text{global damage index (-)}
\( GTP \) \quad \text{global temperature potential (kg CO}_2\text{)}
\( GWP \) \quad \text{global warming potential (kg CO}_2\text{)}
\( h \) \quad \text{heat transfer coefficient (W m}^2\text{ K}^{-1}\text{)}
\( H \) \quad \text{specific enthalpy (J kg}^{-1}\text{)}
\( H_1 \) \quad \text{specific enthalpy at evaporator outlet/compressor inlet (J kg}^{-1}\text{)}
\( H_4 \) \quad \text{specific enthalpy at evaporator inlet (J kg}^{-1}\text{)}
\( k \) \quad \text{thermal conductivity (W m}^{-1}\text{ K}^{-1}\text{)}
\( K \) \quad \text{construction emission factor (-)}
\( L \) \quad \text{length (m)}
\( L_{trav} \) \quad \text{travel distance (km)}
$M$ mass (kg)

$M'$ emission mass flux of non-$\text{CO}_2$ gas (kg y$^{-1}$)

$M''$ emission mass flux of $\text{CO}_2$ (kg y$^{-1}$)

$M'''$ total $\text{CO}_2$ equivalent emission mass flux (kg y$^{-1}$)

$m_r$ refrigerant mass flow rate (kg s$^{-1}$)

$n$ index, sensitivity factor

$N$ number

$N_{cmp}$ number of components (-)

$N_{serv}$ number of service/maintenance visits (-)

$N_{start}$ number of compressor starts (-)

$N_{top}$ number of refrigerant top-ups (-)

$p$ pressure (Pa)

$\Delta p$ pressure drop (Pa)

$p_{disch}$ pressure at compressor discharge (Pa)

$p_{suct}$ pressure at compressor suction (Pa)

$p_{vent}$ probability that refrigerant is vented (-)

$Q$ heat flow (W)

$Q_{app}$ application heat load (W)

$Q_c$ condensing capacity (W)

$Q_{c,comp}$ compressor condensing capacity (W)

$Q_{comp}$ compressor capacity (W)

$Q_e$ evaporating capacity (W)

$Q_{e,comp}$ compressor evaporating capacity (W)

$\Delta \bar{R}$ mean variation in leak ratio (-)

$R_{distr}$ leak mass fraction during refrigerant distribution (-)

$Re$ Reynolds number (-)

$R_{oper}$ leak mass fraction during equipment operation (-)

$S_{oil}$ solubility of refrigerant in oil (kg kg$^{-1}$)

$s_w$ wall shear stress (N m$^{-2}$)
\( t \)
\begin{align*}
& \text{time (s, y)} \\
T & \text{temperature (°C)} \\
\Delta t' & \text{reference time (s, y)} \\
T' & \text{variable temperature (°C)} \\
T_{\text{amb}} & \text{ambient temperature (°C)} \\
T_{\text{app}} & \text{application temperature (°C)} \\
T_{\text{disch}} & \text{temperature at compressor discharge (°C)} \\
TLTI & \text{total lifetime temperature impact (K)} \\
\Delta t_{\text{off}} & \text{off-cycle time (s)} \\
\Delta t_{\text{on}} & \text{on-cycle time (s)} \\
t_{\text{op}} & \text{operating time for compressor wear (s)} \\
\Delta T_{\text{set}} & \text{thermostat differential temperature setting (K)} \\
\Delta t_{\text{start}} & \text{compressor start-up time (s)} \\
T_{\text{suc}} & \text{temperature at compressor suction (°C)} \\
T_{\text{w}} & \text{wall temperature (°C)} \\
u & \text{velocity (m s}^{-1}\text{)} \\
U & \text{overall heat transfer coefficient (W m}^{-2}\text{ K}^{-1}\text{)} \\
V & \text{volume (m}^3\text{)} \\
V_{\text{int}} & \text{internal or representative volume (m}^3\text{)} \\
V_{\text{hose}} & \text{refrigerant hose/manifold internal volume (m}^3\text{)} \\
\dot{V}_{\text{HTF}} & \text{HTF volume flow rate (m}^3\text{s}^{-1}\text{)} \\
\dot{V}_{\text{sw}} & \text{swept volume flow rate (m}^3\text{s}^{-1}\text{)} \\
W & \text{power (electrical) (W)} \\
W_{\text{ancil}} & \text{power of ancillary component (W)} \\
W_{\text{comp}} & \text{compressor power (W)} \\
W_{\text{HTF}} & \text{power applied by HTF flow (W)} \\
x & \text{heat exchanger wall/fouling thickness (m)} \\
X & \text{mass fraction of component (-)} \\
Y & \text{year} \\
\end{align*}
Greek and other symbols

$\chi$ component leak factor (v$^{-1}$)

$\varepsilon$ materials emissions factor (kgCO$_2$ kg$^{-1}$)

$\varepsilon_{\text{cprnt}}$ component manufacture emission factor (kgCO$_2$ kg$^{-1}$)

$\varepsilon_r$ refrigerant manufacture emissions factor (kgCO$_2$ kg$^{-1}$)

$\varepsilon_{\text{braz}}$ brazing/welding emissions factor (kgCO$_2$ kg$^{-1}$)

$E$ chemical reaction energy (J mol$^{-1}$)

$\varphi$ reduced charge fraction (-)

$\eta_{\text{cprnt}}$ component (fan or pump motor) efficiency (-)

$\eta_{\text{incm}}$ incineration efficiency (-)

$\eta_{\text{vol}}$ volumetric efficiency (-)

$\lambda$ friction factor (-)

$\mu$ viscosity (Pa s)

$M_{\text{air}}$ molecular mass of air (kg mol$^{-1}$)

$M_{\text{CO}_2}$ molecular mass of carbon dioxide (kg mol$^{-1}$)

$M_{\text{oil}}$ molecular mass of oil (kg mol$^{-1}$)

$M_r$ molecular mass of refrigerant (kg mol$^{-1}$)

$\nu_y$ mean deviation in annual emission factor (kgCO$_2$ kW$^{-1}$ h$^{-1}$)

$\nu_D$ mean deviation in daily emission factor (kgCO$_2$ kW$^{-1}$ h$^{-1}$)

$\theta_y$ coordinate for position of annual emission factor ($^\circ$)

$\rho$ density (kg m$^{-3}$)

$\sigma$ climate sensitivity parameter (-)

$\zeta$ transport emissions factor (kgCO$_2$ km$^{-1}$)

$\tau$ atmospheric lifetime (y)

$T$ time constant (y)

$\xi$ energy emissions factor (kgCO$_2$ kW$^{-1}$ h$^{-1}$)

$\psi$ proportion of refrigerant emissions from refrigerant production (-)

Subscripts

$app$ application
braz  from brazing, welding, etc
chrg  from refrigerant charging
cnst  from equipment construction
cpnt  relating to components
des   design (rated)
disp  from disposal
dist  from refrigerant distribution
dmil  from dismantling
EOL   end of life
evac  of refrigerant evacuated
foul  fouling
HTF   heat transfer fluid
incin of material from incineration
manf  from manufacturing
max   maximum
min   minimum
oil   of compressor oil
oper  from equipment in operation
prod  from production
ptst  from pressure/leak testing
r     refrigerant
re cov of recovered refrigerant
s     saturated
serv  from service and maintenance activities
SOL   start of life
sys   system
Tot   total
trav  from travel
Use   in-use
vent  of vented refrigerant
x     species
SECTION I: INTRODUCTION

1.1 Environmental burden of refrigerants

With the discovery of the link between CFCs and HCFCs and the depletion of the ozone layer, the United Nations Environment Programme (UNEP) formulated the Montreal Protocol in 1987, to phase-out the use and production of these substances. In response to the Montreal Protocol, alternative refrigerants were sought, and this search produced a number of potential substances for applications where only CFCs and HCFCs were previously used: HFCs, HCs, ammonia and carbon dioxide (CO₂). Simultaneously, attention turned to the issue of climate change, and in particular, global warming. Subsequently, the Kyoto Protocol was developed under the UN in 1997, which prescribes the limitation and reduction of emissions of a group of anthropogenic “greenhouse gases” (GHGs): CO₂, nitrous oxide (N₂O), methane (CH₄), HFCs, perfluorocarbons (PFCs) and sulphur hexafluoride (SF₆). Whilst the Kyoto Protocol has not been ratified to date, numerous governments have already published national legislation to meet the Kyoto targets for emissions reduction. In order to quantify the contribution of these gases to climate change, the initial international discussions on climate change in 1990 adopted the use of Global Warming Potential (GWP) of the gas (IPCC, 1990). GWP is a measure of the insulating properties that a gas has on the heat radiating away from the surface of the earth, and is relative to the effect caused by one kilogram of CO₂.

In response to these political developments, GHG emissions associated with refrigeration systems need to be assessed, particularly for comparing and screening alternative refrigerants and systems. Given that conventional vapour compression systems could be responsible for emissions of CO₂ (from power stations) and also leakage of refrigerant (such as HFC’s), the global warming effects of each gas could be added together to evaluate the overall impact of alternatives. Such an environmental assessment technique has developed into a method commonly known as “Total Equivalent Warming Impact” (TEWI), although slight modifications to the method have resulted in other terms, including “Life Cycle Warming Impact” (LCWI) and “Life Cycle Climate Performance” (LCCP).¹ In its most basic form the calculation can be expressed as equation (1).

\[ M^* = \sum \left( M_i' \cdot GWP_i \right) + M^* \]  

¹ Hereafter, the TEWI term will also refer to these other variations, except otherwise stated.
where $M$ is the mass of substance emitted; $M''$ represents the total emissions in terms of CO$_2$ equivalent, for each of the refrigerant species $i$, $M'$ is the amount leaked, and $M''$ is the total mass of CO$_2$ emitted, typically due to energy consumption by the system.

To date, TEWI is used frequently by industry and Government as a tool for comparing the "strength" of GHG emissions of systems and equipment. There is increasing reliance on TEWI, despite many omissions evident in the procedures that are currently employed. Several researchers have questioned the suitability and value of TEWI (e.g., Kruse, 1999) but from surveying the literature, it is known that little attempt has been made to resolve many of the issues. Given the escalating importance of estimation of GHG emissions, there is a clear need for improving the current method.

1.2 Literature review
A literature review was carried out to provide a general overview of the use and application of TEWI and associated methods. In particular, it was used to identify the level of detail and complexity adopted for calculation, means of handling certain variables, sources of input data, etc. Additionally, studies on practical uses of TEWI were considered to indicate the context within which TEWI is normally used, and how the results of comparative assessments are normally handled. Lastly, it was deemed important to understand the basis for quantification of the global warming impact of the gases involved, and any limitations that could affect the interpretation of a TEWI assessment.

1.2.1 Existing methodologies for evaluating environmental impacts
Following the development of the GWP to help compare environmental impact of CFC alternatives (see Section 1.2.3), the TEWI concept was an obvious progression and the evaluation process for alternative refrigerants began to include indirect GHG emissions from energy production required to operate the system. The actual numerical methodology was formalised by a UK industry publication (BRA, 1996), and later within a European Standard on safety and environmental requirements for refrigerating systems (EN 378, 2000). The BRA guidelines provided some background information, and a series of numerical procedures and supporting data, to estimate refrigerant leakage, system energy consumption and corresponding CO$_2$ emission rates. Nevertheless, the guidelines were considered to be rudimentary and lacking the

---

2 Requests for TEWI comparisons have become part of the tendering process for RAC equipment, and as the basis for cost-effectiveness studies commissioned by UK Government departments.
consideration of many influencing parameters (Colbourne and Suen, 1999). Awareness of these peripheral parameters was further highlighted in the UNEP report on alternative refrigerants for ozone depleting substances (ODS) (UNEP, 1999).

Papasavva and Moomaw (1998) proposed a method that extended the conventional TEWI approach, and named it "Life cycle warming impact" (LCWI), which also included emissions associated with refrigerant production, transportation of materials, and refrigerant recycling. They also included for the breakdown of refrigerants in terms of the GWP of decomposition substances; this indirect contribution is now included in the more recent GWP values (IPCC/TEAP, 2005). The description provided by Papasavva and Moomaw was comprehensive, although repeatable use of the method would be difficult due to absence of key calculation procedures and input data. Of importance was an example study of domestic refrigerators, which showed that some of the parameters normally ignored did have a significant impact on the result. Conversely, Aprea et al (1999) present a simplified criterion to determine TEWI that used graphical methods.

More recently, Heikkiä and Fahlén (2003) and Barnabe et al (2003) have proposed life-cycle assessment (LCA) methods to help estimate the emissions associated with the equipment throughout its life. This was used to assist with optimising the design to minimise the environmental impact of the equipment at design stage. Instead of limiting its scope to GHGs, these studies consider other impacts normally included in an LCA (e.g., acidification, volatile organic compounds, etc.), and each of these is subject to weighting to produce a single "overall environmental impact" value. Again, in both studies, the concept is described comprehensively with examples, but data and procedure were largely omitted. Another method proposed by Bodyal and Dubro (2003) used this parameter-weighted approach and included additional measures such as system "efficiency" and safety aspects such as flammability and toxicity. The weighting assigned to each parameter was determined subjectively. The concept and general method of LCA is covered by a series of international standards (e.g., ISO 14040), from which a procedure for a specific application can be developed.

One other variation was a method described by Zhang and Reistad (1998), and named "Total equivalent resource exergy" (TERE). This takes the sum of exergy production for the manufacture and operation of a system, but also includes the exergy production due to the recovery of emitted CO₂ (based on the GWPs of the various GHGs). With this additional CO₂

---

3 A paper published by the present author/tutor resulting from the initial study for this thesis.
recovery element, the method could be used to evaluate the efficiency of a system in its use of natural resources (from a global warming standpoint). Worked examples for some applications were used to demonstrate the difference between the results of TEWI and TERE, the most notable of which was that the influence of manufacture of equipment was much greater using TERE.

1.2.2 Survey of comparative studies
The majority of the literature citing TEWI is normally concerned with performance comparison of different refrigerants and/or systems. Large volumes describe experimental and/or theoretical studies on system energy consumption, which are complimented by presenting a short TEWI comparison. Such exercises normally adopt a simplistic approach where energy consumption is converted to equivalent CO₂ emissions for some operating period, and rarely provide a useful contribution to understanding of the method itself. As such, only those studies that cover additional considerations in terms of TEWI evaluation were reviewed presently.

Perhaps the most widely promoted TEWI studies are a series of comparisons commissioned by the Alternative Fluorocarbons Environmental Acceptability Study (AFEAS), initially in 1991 (Fischer et al), then in 1993 (Fischer et al) and again in 1997 (Sand et al). These were reports concerning several sectors including domestic and commercial refrigeration, car and residential air conditioning, and chillers. A latest report (Sand et al, 2002) repeats the results of the previous studies. Evaluations were carried out to compare the impact of various different refrigerants for a range of different geographical locations, using corresponding local conditions (temperatures, designs, power station emissions, etc.) as inputs to the calculations. Despite the broad range of scenarios, the description of the actual methodology, source data, assumptions and calculations was very limited. Therefore, whilst large amounts of output data were reported, it is questionable as to how useful reports such as these are.

Use of TEWI for comparative purposes is demonstrated in a study by Hwang et al (2004) which compares a low GWP refrigerant (R290) and two HFCs (R404A and R410A). Experiments were conducted to measure the performance in a particular type of system, which were optimised for each refrigerant. The test data showed the COP for R290 was more than 10% above that of the HFCs. However, a 10% cost deficit was assumed for the system using R290 due to flammability mitigation, and this was offset by giving the systems using HFCs a larger condenser. The performance data was re-modelled (causing the HFCs to have superior COP under most conditions), and using an unusually low leakage rate, concluded that the TEWI for R410A was
lower than R290. Since the size of the cost deficit is known by this author to be overestimated (Colbourne, 1999), and that the annual leak is unrealistically low for the equipment used, the emphasis on a 4% lower TEWI is unjustified given the uncertainties associated with the entire TEWI calculation.

A more constructive use of TEWI has been reported by Benstead (1996), where the TEWI for two different supermarket systems were calculated and then independently audited. The retrospective auditing process was useful since it found that initial assumptions about fan power were underestimated and that their assumption of an average ambient temperature leads to overestimation. Another use of TEWI applied to supermarket installations was as a design tool (Davies, 2004). Various system designs were evaluated from a TEWI perspective in order to determine the design that would lead to the least GHG emissions. A type of indirect system using low-GWP refrigerants was found to achieve 26% lower TEWI than the conventional direct expansion approach.

Zhelenzv at al (1996) evaluate different refrigerants for domestic refrigerators, but pay attention to causes of efficiency differences such as component pressure losses. Another use of TEWI was by Aprea and Greco (1998), who carried out thorough tests of a particular system using different refrigerants to enable accurate estimation of the energy contribution. Based on fixed parameters such as leak rate, TEWI for R22 was evaluated, and by working the method backwards, the required minimum system efficiency for R407C was determined if it were to equal the R22 system’s TEWI. The results also showed how the difference in TEWI for R22 and R407C systems varies with ambient conditions and operating mode (i.e., cooling or heating). A computational approach was reported by Gopalnarayanan and Rolotti (1999), where eight different refrigerants were compared using a simulation model to predict energy consumption. A fixed system design was used for all refrigerants, and only compressor swept volume was adjusted to maintain equal refrigerating capacity at rating conditions.

Some other studies have used an extended LCA approach where additional environmental impacts were addressed. Hansen and Gustafsson (1994) and Cintar and Hadfield (2000) both focussed on the manufacture and operation of the compressor in small hermetic systems. The former addressed the manufacture and disposal of several compressor models, which showed similar results for each, not just for GHG emissions, but also for sulphur emissions, water usage and others. It was also shown that by sacrificing a greater impact at design stage (i.e., use of more materials), this could dramatically offset emissions during in-use stage (i.e., less energy
consumption), whilst the scrapping phase had negligible impact. Cintar and Hadfield considered many environmental impacts in comparing compressors for HFC-134a and HC-600a. The analysis showed significant reduction in GHG emissions when using HFC-600a, which was attributed to different compressor design and lubricant types, as well as the refrigerant itself.

Johnson (1998) evaluated HFCs and HCs in car air conditioning to compare TEWI and emissions of volatile organic compounds (VOCs). The analysis used a comparatively thorough “cradle-to-grave” approach, and found that the HC resulted in significantly lower GHG emissions, particularly when refrigerant leak rates were high, and VOC emission were found to be similar. A more comprehensive study was a LCA by Frischknecht (1999) that covered heat pumps, building air conditioning, industrial deep freeze and supermarket refrigeration applications, with particular focus on comparing “natural” (HC, ammonia) and synthetic (HCFC, HFC) refrigerants. Actual products/installations were analysed in terms of individual components and their routes to the end product. A LCA database was used to determine all emissions associated with each component, and other emissions resulting from the system operation. It was found that energy consumption was the dominant factor for most emissions (>50%), and therefore system efficiency was an important consideration. In general, systems using HC and ammonia resulted in less overall emissions when compared to fluorinated refrigerant systems. Again, effort focussed on presenting comprehensive results rather than detailing methodology that prevented a full appreciation of the results.

1.2.3 Measures of climatic impacts

The emission of anthropogenic GHGs results in changes in global surface temperatures, which ultimately leads to climate change, or “an average change in the earth’s weather” (IPCC, 2001). Understanding the physical processes that occur in the atmosphere, on land, and in the oceans is the subject of massive research. In assessing the impact of emissions, it is necessary to have a measure that relates climate impact to a given mass of a particular species, for example, the impact of a 1 kg emission of HFC-134a. As mentioned in the introduction, the internationally agreed measure was GWP, and this has been employed in impact assessments since. However, as implied by its definition (IPCC/TEAP, 2005), GWP is essentially a measure of the property of a species under a given set of conditions, and thus a precursor to the actual climate impact:

---

4 Most of the important research is collated and assessed by the Inter-governmental Panel on Climate Change (IPCC), who publishes report every four years, the latest being the Third Assessment Report (IPCC, 2001a).
An index, describing the radiative characteristics of well-mixed greenhouse gases, that represents the combined effects of the differing times these gases remain in the atmosphere and their relative effectiveness in absorbing outgoing infrared radiation. This index approximates the time-integrated warming effect of a unit mass of a given greenhouse gas in today’s atmosphere, relative to carbon dioxide.

Since GWP was adopted, its suitability to represent the actual impact of the release have been questioned, and extensive research since 1990 has been unable to draw a conclusion on whether any other method is more suitable than GWP (O’Neill, 2000). This is mainly because it is unclear how effectiveness of a measure should be judged. For example, a large number of perceivable physical impacts exist, such as a change in surface (air) temperatures, weather behaviour, or economies, etc., where the size of the change of each impact varies between gases. A further consideration is the time scale of the impact: the rate and the duration of the change, specifically since the impact of an emission may be more sensitive to the rate of surface temperature change, than the absolute change (Godal, 2003). For example, O’Neill notes that the use of GWP to compare two gases (of equivalent emissions) results in different climate responses, i.e., disproportionate rates and magnitudes of change across the different types of impacts. In fact, emissions of two different gases with the same equivalent GWP did not produce the same climate impact response.

In terms of the formulation of GWP, various problems exist with its quantification. First, most common manmade GHGs break down in the atmosphere and therefore have a finite atmospheric lifetime (\( \tau \)), although it varies according to the atmospheric concentration of itself and other gases. CO\(_2\) on the other hand remains chemically stable under normal atmospheric conditions, and therefore has an infinite \( \tau \). Instead of chemical decay within the atmosphere, removal of CO\(_2\) is reliant upon “carbon sinks” such as the ocean and plants that absorb it from the atmosphere, resulting in an apparent \( \tau \approx 500 \) years. Reference to Table 1 shows that for common refrigerants \( \tau \) are less than 50 years. As implied by the definition above, GWP is the ratio of a species’ absolute GWP (AGWP) to the AGWP of CO\(_2\), and AGWP is determined by quantification of the radiative forcing of the gas over some reference, or integration time \( t \).

For gases with a finite \( \tau \), calculation of AGWP using a \( t > \tau \) tends to produce no additional contribution from when \( t = \tau \), whereas AGWP for a gas with infinite \( \tau \) (i.e., CO\(_2\)) always increases with a longer \( t \). Thus for a given species, GWP(\( t \)) tends to reduce as it is evaluated for \( t \) that exceeds the species’ \( \tau \), since its AGWP thereon remains approximately constant whilst the
AGWP of CO₂ continues to rise. This dependency of interpretation of a gas's global warming impact on the largely subjective choice of integration time means that the results of calculations that use GWP are implicitly subjective themselves. Another parameter that introduces uncertainty into GWP is the basis for calculation of radiative forcing ($F$) which is dependent on atmospheric concentrations of other gases. $F$ for a given species is calculated using climate models, but these models were based on an atmosphere that is composed of gases at present day concentrations. Since concentrations of all GHGs will be different at some time in the future, values of $F$ and therefore GWP will be different (Smith, 2003). According to a review of GWP within IPCC/TEAP (2005), the net uncertainty could be as high as ±50%, with longer-t gases tending towards smaller uncertainties.

### Table 1: GHG (refrigerant) data from IPCC/TEAP (2005)

<table>
<thead>
<tr>
<th>Name</th>
<th>Molecular Mass (kg kmol⁻¹)</th>
<th>Radiative forcing, $F$ (W m⁻² ppb⁻¹)</th>
<th>Atmospheric lifetime, $\tau$ (year)</th>
<th>GWP(100) (kgCO₂ kg⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R12</td>
<td>120.9</td>
<td>0.320</td>
<td>1.49 ⋅ 10⁻¹¹</td>
<td>100</td>
</tr>
<tr>
<td>R22</td>
<td>86.5</td>
<td>0.200</td>
<td>1.30 ⋅ 10⁻¹¹</td>
<td>12.0</td>
</tr>
<tr>
<td>R32</td>
<td>52.0</td>
<td>0.110</td>
<td>9.73 ⋅ 10⁻¹²</td>
<td>4.9</td>
</tr>
<tr>
<td>R124</td>
<td>136.5</td>
<td>0.220</td>
<td>9.06 ⋅ 10⁻¹²</td>
<td>6.8</td>
</tr>
<tr>
<td>R125</td>
<td>120.0</td>
<td>0.230</td>
<td>1.08 ⋅ 10⁻¹¹</td>
<td>29.0</td>
</tr>
<tr>
<td>R134a</td>
<td>102.0</td>
<td>0.160</td>
<td>8.27 ⋅ 10⁻¹²</td>
<td>14.3</td>
</tr>
<tr>
<td>R143a</td>
<td>84.0</td>
<td>0.130</td>
<td>8.70 ⋅ 10⁻¹²</td>
<td>52.0</td>
</tr>
<tr>
<td>R152a</td>
<td>66.1</td>
<td>0.090</td>
<td>7.66 ⋅ 10⁻¹²</td>
<td>1.4</td>
</tr>
<tr>
<td>R227ea</td>
<td>170.0</td>
<td>0.260</td>
<td>3.31 ⋅ 10⁻¹¹</td>
<td>34.2</td>
</tr>
<tr>
<td>Ethane</td>
<td>R170</td>
<td>30.1</td>
<td>0.0052</td>
<td>5.98 ⋅ 10⁻¹³</td>
</tr>
<tr>
<td>Propane</td>
<td>R290</td>
<td>44.1</td>
<td>0.0031</td>
<td>3.95 ⋅ 10⁻¹³</td>
</tr>
<tr>
<td>Iso-butane</td>
<td>R600a</td>
<td>58.1</td>
<td>0.0047</td>
<td>4.55 ⋅ 10⁻¹³</td>
</tr>
<tr>
<td>Propylene</td>
<td>R1270</td>
<td>42.1</td>
<td>0.034</td>
<td></td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td></td>
<td>44.0</td>
<td>1.54 ⋅ 10⁻⁴</td>
<td>1.98 ⋅ 10⁻¹⁵</td>
</tr>
<tr>
<td>Ammonia</td>
<td>R717</td>
<td>19.0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

¹ To convert from W m⁻² ppb⁻¹ to W m⁻² kg⁻¹, multiply by $M_{mol}/(5.15 \cdot 10^9 \cdot M_{x})$

In order to overcome some of these limitations several variations of GWP have been proposed. Lashof and Ahuja (1990) suggest using an infinite lifetime integration period, and conversely, Hammond et al (1990) consider the use of one-year integration period appropriate since it accounts for the instantaneous warming effect. The second assessment report of the IPCC (1995)
considers the use of absolute GWP since this has the advantage that the impacts attributed to non-CO$_2$ gases were less dependent of the accuracy of the CO$_2$ evaluation. Wigley (1998) goes back a stage and proposes a Forcing Equivalent Index (FEI), which is based on the rate of change of $F$ required for achieving a specified increase in temperature.

In addition to the parameters closely related to the properties of the GHG (e.g., $\tau$, $F$, and GWP), there are also other indices addressing physical impacts (such as change in sea level) or economic impacts. A new approach to estimate the global temperature change potential (GTP) due to an emission of a GHG is presented by Shine et al (2004). In their work, the radiative forcing effects of a GHG were related to the available thermal mass of the earth’s surface to calculate temperature rise. Taking on the effects of temperature rise, Smith (2003) states that the most important approach to evaluating the impact of GHGs is economic efficiency, suggesting that the cost of impacts is related to the square of the temperature rise integrated over a given timescale. Fuglestvedt et al (2003) also describes this approach, and notes that the index could also include abatement costs so that an optimum or minimum cost for emission reduction could be determined. They refer to several authors who evaluate the damage impact based on an exponent of temperature increase, to a value between 1 and 2. This could be a static (fixed time period) or dynamic approach, so that the time-dependent or accumulated damage costs can be calculated.

Reilly et al (1999) tried to combine scientific and economic aspects in proposing a variable economic index which accounts for particular emission scenarios and costs arising from abatement policies and physical damage caused by climate change. However, the authors point out that the use of such an approach requires a link to the global economy, such as two-way adjustments with interest rates. In evaluating a measure termed “Global Damage Potential” (GDP), Frankhauser (1994) found that for a given GHG, the GDP typically equated to GWP. Conversely, Reilly and Richards (1993) found that the contribution of non-CO$_2$ gases to damage increased with global temperature rise compared to CO$_2$, since CO$_2$ promoted (beneficial) plant growth, thus increasing the size of the carbon sink. Other variations propose an index where abatement costs were evaluated by ensuring a given impact parameter does not exceed a given value. For example, Manne and Richels (2001) estimate a cost index to ensure a given temperature change is not exceeded by a particular year.

1.3 Formulation of objectives
There is a general acceptance of TEWI by RAC equipment manufacturers, installers and end-users wanting to evaluate the environmental impact of systems, and subsequent choices and
designs are influenced by the results. The literature revealed that current methodologies necessitate broad assumptions and interpretations of subsequent results tend to be shallow. Due to the often perceived significance of TEWI calculations suggests that the concept demands deeper consideration. The literature review indicated a number of key areas associated with TEWI calculations that are not normally addressed in detail and therefore it is necessary to develop an approach to resolve these issues.

1.3.1 Summary of findings

There are several key findings drawn from the literature review, as follows.

- **Existing methodologies.** The conventional TEWI concept and procedure, such as that detailed by BRA (1996), is found to be limited in several respects and omit a large number of influencing factors. Whilst the basic formulation is established, written methodologies are scarce and when presented, input assumptions are vague making them open to broad interpretation. This is evident when the vast number of TEWI evaluations found within the published literature that generally have the objective of comparing alternative refrigerants but frequently report conflicting conclusions.

- **Comparative studies.** Few studies detail specific input values, or indeed how they are handled; there were no models described in sufficient detail to enable reported results to be reproduced. Also, many assumptions adopted by the various studies are often without justification, and is particularly the case for choice of annual leakage rates, which for similar systems vary widely and appear to be chosen arbitrarily. Many peripheral parameters (outside the usual system leakage and energy consumption) are omitted on the basis that they are "negligible", although this is rarely justified. Further, when very small differences in comparative results of TEWI calculations (e.g., < 5%) are obtained, they are used to defend the selection of a technology that is found to give the lowest impact.

- **Climate impacts.** The validity of drawing conclusions on this basis is further questioned in light of the uncertainties and subjectivity associated with the determination of GWP, and indeed, its suitability of accurately representing the climate impact of the various gases under consideration. This point is particularly relevant because of the different characteristics of common refrigerants which are short-lived with high radiative forcing, and CO₂ with virtually infinite lifetime and low radiative forcing. To address this, a number of alternative measures
of global warming impact were identified that can be adopted as an alternative where they overcome some of the drawbacks associated with GWP.

The findings from the literature review indicate an absence of an otherwise formal approach to calculating TEWI, in terms of level of detail and selection of influencing parameters. Similarly, comparative results are interpreted in a largely subjective manner, due to neglect of certain influencing factors resulting from misconceived assumptions. The current work is intended to address these aspects by identifying a broad range of influencing parameters and assessing the validity of their inclusion in TEWI calculations, and further, to investigate alternative means of interpreting the global warming impact.

1.3.2 Aims and objectives

The purpose of a TEWI calculation is to evaluate the contribution of an RAC system to global warming, and thereby help identify a suitable refrigerant and/or system that provide the lowest impact. It is not considered necessary to develop a different concept since the current fundamentals of quantifying GHG emissions from individual sources and summing them is sound. Instead, the objective is to seek improvements to better represent the impact of an RAC system. This was achieved by a thorough assessment of factors that could influence GHG emissions, where their relative contributions are evaluated by compiling them within a mathematical model and comparing the outputs under a range of conditions. The model also includes alternatives to GWP so different measures of global warming impact can also be considered.

Selection of influencing parameters

Emissions are currently assumed to result solely from the operation of a refrigerating system, which neglects other stages and activities present during the life of a refrigerating system. Other stages in the life of the equipment (such as manufacture) will also involve energy use and gas emissions, and will therefore contribute to the overall impact to global warming. There are further peripheral sources of emissions that occur solely because of the equipments’ existence. Thus, the boundaries of the assessment should be broadened to consider all emissions for which the refrigerating system is “responsible”. All the parameters should be clearly identified, which was achieved by working through of each process and activity associated with the life of the equipment.

5 The case of automotive air conditioning and transport refrigeration is not considered here.
Development of emissions model

Calculations are currently made using limited input data, so the outputs must be considered general, rather than accounting for the design features of specific equipment. The usefulness of an extended methodology can be enhanced if it accounts for controllable design and operating parameters. This can be achieved by quantifying the additional (peripheral) parameters that influence emissions as discussed. Where a specific parameter is considered to make a significant contribution to the net result, it should be included in the model. It is important to have a basis for determining what a "significant" contribution is, and to clearly state and justify all assumptions and include uncertainty analysis over a range of values about the assumed input value.

Subsequently, the model can be used to answer questions about factors that influence TEWI, such as how it is affected by design, operating and materials handling parameters.

Interpretation of climate change

The use of GWP has limited application because the GWP of most refrigerants is not necessarily equivalent to the GWP of CO₂. Also the concept of GWP has little physical meaning (outside the context of climatology), so a more meaningful measure should be used, and its interpretation be explained. In the first case, AGTP was chosen to represent global warming impact because it has perceivable output (i.e., change in temperature), and it has a clear time element – both of which help quantify consequences of global warming, such as by means of a damage index. Thus, by incorporating alternative measures of climate impact, the results of the emissions model can be reinterpreted in terms of temperature rise and damage.
SECTION 2: DEVELOPMENT OF EMISSIONS MODEL

2.1 Modelling approach

The basic approach to determine TEWI is to estimate the emissions from system leakage and energy consumption, as given in BRA (1996). However, many additional influences may contribute to the total emissions other than those resulting directly from the system operation. Different sources of emissions are illustrated in Figure 1, where the life of the equipment is broken down into three stages: start of life (e.g., production), in-use (e.g., operation, servicing), and end of life (e.g., disposal). For these stages, each source that causes emissions is shown and the type of emission is indicated, i.e., CO₂ from energy production and refrigerant releases. In the model, total lifetime emissions of refrigerant and CO₂ are the sum of emissions for all the elements, as equation (2) and (3) respectively.

\[
M'_{TOT} = \sum_i M'_{SOL,i} + \sum_i M'_{Use,i} + \sum_i M'_{EOL,i}
\]

(2)

\[
M^*_{TOT} = \sum_i M^*_{SOL,i} + \sum_i M^*_{Use,i} + \sum_i M^*_{EOL,i}
\]

(3)

where subscripts SOL, Use, and EOL represent the emissions from start of life, in-use and end of life stages for each element, \(i\), that is a source of emissions.

A mass balance for CO₂ emissions is not necessary, but because any refrigerant produced will either be emitted to the atmosphere or destroyed by incineration a mass balance of refrigerant throughout its life is needed. Equation (4) gives the total mass of refrigerant “consumed” over the life of the equipment; refrigerant recovered at EOL is not included because it is assumed to be recycled for another purpose.

\[
M_{r, Tot} = M'_{Tot} + M_{incin}
\]

(4)

where \(M_{incin}\) is the mass of refrigerant incinerated at EOL.

When evaluating the emissions from any of the possible sources in Figure 1, limiting boundaries must be established in order to determine the accurate relationships between the source and the emissions, whilst avoiding unnecessary detail. For example, emissions associated with the manufacture of a system component may be related to the mass of the material used to make it, such as from energy required by machinery and flue gases from heat production. However, to enable a factory to operate, staff are required and the facilities where they work will produce
emissions, as would the manufacture of machinery used within the factory. Within the boundaries, each source is characterised according to observations based on the authors and others' experience relating to RAC equipment, and the availability of data. Uncertainties are accounted for by identifying a likely range associated with each source, and a maximum and minimum input value associated with the size of the emissions employed. Thus, the output of the model will be a range of maximum and minimum masses.

Figure 1: Identification of sources and types of GHG emissions
2.2 Start of life stage

Start of life stage comprises production of raw materials, component manufacture, assembly, installation and charging with refrigerant. The general process is detailed in Figure 2, where the shaded boxes represent sources of emissions. Emissions of refrigerant and CO$_2$ at SOL comprise emissions from two general sub-stages: component manufacture ($M'_{manf}$) and equipment construction ($M'_{constr}$), equation (2) and (3) respectively.

\[ M'_{SOL} = M'_{manf} + M'_{constr} \]  

\[ M'_{SOL} = M'_{manf} + M'_{constr} \]  

---

2.2.1 Refrigerant and component manufacture

Refrigerant manufacture emissions occur in the production process ($M'_{prod}$) and during distribution and storage ($M'_{dist}$) (equation 7). CO$_2$ emissions also occur in refrigerant production.
and therefore contributes to $M_{\text{manf}}^*$, although the main source is from the production of materials and forming of system components ($M_{\text{r,prod}}^*$) (equation 8).

$$M_{\text{manf}}^* = M_{\text{prod}}^* + M_{\text{dist}}^*$$ (7)

$$M_{\text{manf}}^* = M_{\text{cpl}}^* + M_{\text{r,prod}}^*$$ (8)

**Refrigerant production, storage and distribution**

Manufacture of refrigerants causes emissions associated with energy requirements and leakage of feedstock and other materials. Data have been found for various refrigerants, either as quantities of individual species emitted per unit mass of a given refrigerant, or expressed as total mass of CO$_2$ equivalent. This data is listed in Table 2, where values for all refrigerants are given in CO$_2$ equivalent ($\varepsilon_r$) using GWP(100) for the various species. Several sources had been found for certain refrigerants so the data was summarised by giving a maximum, mean and minimum value. Where only one value of $\varepsilon_r$ was found in the literature, the minimum and maximum values were based on the average variation found for R22 and R134a (i.e., 0.45 and 1.55 respectively), since these were supported by a larger number of sources. Where there was complete absence of data for a refrigerant, values were estimated using the mean ratio of R22 and R134a $\varepsilon_r$ to their GWP, i.e., $\varepsilon_r = 0.12 \times \text{GWP(100)}$ of the refrigerant concerned. This is justified because the manufacturing emissions are largely influenced by plant leakage and therefore is strongly dependent on refrigerant GWP (Campbell and McCulloch, 1998). These data were then used to estimate additional emissions resulting from the total quantity of refrigerant used throughout the system life. To be consistent with the current approach of calculating refrigerant and CO$_2$ emission separately, the production emissions were split between CO$_2$ and other species, but for simplicity these were grouped for the selected refrigerant only, so that refrigerant emissions are converted using its GWP(100). Using the references that provided a breakdown of each species emitted, an average split between CO$_2$ and others was found. In general the larger proportion of GHG contribution is from non-CO$_2$ gases, and the proportion of refrigerant GWP(100) contribution is, $\psi = 0.17$. The mass of refrigerant (equivalent) emitted is therefore calculated from equation (9) and CO$_2$ from equation (10).

$$M_{\text{prod}}' = \psi \cdot \frac{M_{\text{r,Tot}} \cdot \varepsilon_r}{\text{GWP(100)}}$$ (9)
\[ M_{r, prod} = (1 - \psi) \cdot M_{r, Tot} \cdot \varepsilon_r \]  

(10)

where \( M_{r, Tot} \) is the total refrigerant consumed (equation 4).

Table 2: Manufacturing emissions of refrigerants

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>( \text{CO}_2 ) equivalent emissions per unit mass of refrigerant, ( \varepsilon_r ) (kg ( \text{CO}_2 ) kg(^{-1} ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Value</td>
<td>Source</td>
</tr>
<tr>
<td>Min</td>
<td>Mean</td>
</tr>
<tr>
<td>--------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>R12</td>
<td>No data</td>
</tr>
<tr>
<td>R22</td>
<td>205.3 Frischknecht (1999)</td>
</tr>
<tr>
<td></td>
<td>393 ARAP (2004)</td>
</tr>
<tr>
<td>R32</td>
<td>190 (^1) Frischknecht (1999)</td>
</tr>
<tr>
<td>R124</td>
<td>No data</td>
</tr>
<tr>
<td>R125</td>
<td>160 (^1) Frischknecht (1999)</td>
</tr>
<tr>
<td>R134a</td>
<td>38.8 Banks and Sharratt (1996)</td>
</tr>
<tr>
<td></td>
<td>84.5 McCulloch and Campbell (1998)</td>
</tr>
<tr>
<td></td>
<td>72 Frischknecht (1999)</td>
</tr>
<tr>
<td></td>
<td>87 ARAP (2004)</td>
</tr>
<tr>
<td></td>
<td>10 McCulloch and Lindley (2003)</td>
</tr>
<tr>
<td>R143a</td>
<td>120 (^1) Frischknecht (1999)</td>
</tr>
<tr>
<td>R152a</td>
<td>No data</td>
</tr>
<tr>
<td>R227ea</td>
<td>120 Banks et al (1998)</td>
</tr>
<tr>
<td>R404A</td>
<td>136 Frischknecht (1999)</td>
</tr>
<tr>
<td>R407C</td>
<td>142 Frischknecht (1999)</td>
</tr>
<tr>
<td>R410A</td>
<td>173 Frischknecht (1999)</td>
</tr>
<tr>
<td>R170</td>
<td>No data</td>
</tr>
<tr>
<td>R290</td>
<td>1.5 Gover et al (1996)</td>
</tr>
<tr>
<td></td>
<td>0.95 Frischknecht (1999)</td>
</tr>
<tr>
<td>R600a</td>
<td>1.5 McCulloch et al (1998)</td>
</tr>
<tr>
<td></td>
<td>0.9 Gover et al (1996)</td>
</tr>
<tr>
<td>R717</td>
<td>2.0 McCulloch et al (1998)</td>
</tr>
<tr>
<td></td>
<td>2.53 Frischknecht (1999)</td>
</tr>
<tr>
<td>R744 (\text{CO}_2)</td>
<td>1.62 Frischknecht (1999)</td>
</tr>
<tr>
<td>R1270</td>
<td>1.3 SAEFL (1998)</td>
</tr>
</tbody>
</table>

\(^{1}\) Estimated from data for blends. \(^{1}\) Estimated from R22 and R134a \( \varepsilon_r \) and/or GWP data
Refrigerant losses during storage and distribution are assumed to occur at SOL stage for the entire refrigerant used throughout the equipment life; although some of the refrigerant is actually used during In-use or EOL, their production is accounted for here. Emissions due to refrigerant production are calculated from the emitted proportion ($R_{ast}$) of the entire mass of refrigerant used throughout equipment lifetime (equation 11).

$$M'_{d, t} = R_{d, t} \cdot M_{r, t}$$  \hspace{1cm} (11)

Refrigerant emissions occur during storage at the production facility, transfer to containers for transportation, factory storage vessels and from cylinder filling. An analysis of these activities has been made, and it is estimated that a loss of $R_{d, t} = 0.0035$ occurs for refrigerant going into cylinders, and $R_{d, t} = 0.052$ for refrigerant from bulk storage (Ritter, 2004). The higher losses from bulk storage are largely due to leakage from piping circuits at the equipment manufacturers' facility. The value of $R_{d, t}$ is chosen based on the charging method for the equipment.

Table 3: Component construction data

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Values for equation (12)</td>
<td></td>
<td>5159.5</td>
<td>5801.0</td>
<td>17958</td>
<td>112.61</td>
<td>142.91</td>
<td>8933.0</td>
</tr>
<tr>
<td>$a$</td>
<td></td>
<td>9188.5</td>
<td>10295</td>
<td>27861</td>
<td>193.70</td>
<td>410.36</td>
<td>8933.0</td>
</tr>
<tr>
<td>$a$ (upper)</td>
<td></td>
<td>2897.1</td>
<td>3268.9</td>
<td>11575</td>
<td>65.473</td>
<td>49.769</td>
<td>8933.0</td>
</tr>
<tr>
<td>$a$ (lower)</td>
<td></td>
<td>0.8326</td>
<td>0.8275</td>
<td>0.6336</td>
<td>0.4937</td>
<td>0.6554</td>
<td>1.0</td>
</tr>
<tr>
<td>$n$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aluminium</td>
<td></td>
<td>0.1</td>
<td>0.1</td>
<td>0.02</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Copper</td>
<td></td>
<td>0.4</td>
<td>0.4</td>
<td>0.08</td>
<td>0</td>
<td>0</td>
<td>1.0</td>
</tr>
<tr>
<td>Steel</td>
<td></td>
<td>0.5</td>
<td>0.5</td>
<td>0.90</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Brass</td>
<td></td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>1.0</td>
<td>0</td>
</tr>
</tbody>
</table>

Component manufacture

Evaluation of emissions from component production is a complex process and has therefore been simplified by conducting the estimation by mass of primary material. The actual raw materials may be sourced from a number of different countries, and be subject to local processing practices that differ widely (Norgate and Rankin, 2000). Throughout mining, extraction, purification, etc., materials may be transported over a variety of distances. Given the large number of possibilities prior to arrival at the component factory, aggregate emission figures for primary materials have
been used. Both CO₂ and refrigerant emissions are associated with the manufacture of raw materials and production/assembly of actual system components, but are all converted into kgCO₂ equivalent.

![Figure 3: Data for compressor and compressor oil mass](image1)

![Figure 4: Data for condenser and evaporator mass](image2)

For system components, data for production of primary materials is used, and the mass of a given material is dependent upon the mass of the component. Product catalogues were used to obtain the mass of compressors, evaporators, condensers, pipework and valves (and ancillary components). The mass was correlated against a representative internal volume of the component in the form of equation (12), observed to best represent the data.

\[
M_{mat} = a \cdot V_{int}^n
\]

(12)

where \(a\) and \(n\) are a constant and index, listed for the different components in Table 3.

Despite the range of components from different manufacturers, there appears to be a relatively good correlation with the chosen internal volume parameter. The combined data used for these correlations is shown in Figure 3, Figure 4 and Figure 5. In the model, values of internal or representative volume for the various components were calculated during execution of the design routines for the system performance sub-model (Section 2.3.1).
Table 3 also contains approximate proportions of the different material compositions for each component. The composition breakdown is obtained from product catalogues, although compressor data was obtained from physical measurements by Garland (2004). Other construction materials such as plastics and elastomers have been omitted because the proportions are relatively small compared to metals. Correspondingly, their GHG emissions are negligible, although if emissions related to other environmental impact were to be considered, the contribution may be significant (Frischknecht, 1999).

Table 4: Emissions resulting from raw material manufacture

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>7.64</td>
<td>2.06–6.56 1</td>
<td>5.99</td>
<td>-</td>
<td>-</td>
<td>2.1, 5.6, 7.6</td>
</tr>
<tr>
<td>Brass</td>
<td>2.97</td>
<td>-</td>
<td>-</td>
<td>11.4–16.1 1</td>
<td>3.0, 10.2, 16.1</td>
<td></td>
</tr>
<tr>
<td>Copper</td>
<td>-</td>
<td>5.54</td>
<td>13.2</td>
<td>-</td>
<td>3.3–6.2 1</td>
<td>3.3, 7.1, 13.2</td>
</tr>
<tr>
<td>Steel/iron</td>
<td>2.95</td>
<td>1.60–2.78 1</td>
<td>5.46, 1.79</td>
<td>-</td>
<td>-</td>
<td>1.6, 2.9, 5.5</td>
</tr>
<tr>
<td>Oil</td>
<td>-</td>
<td>-</td>
<td>1.6</td>
<td>-</td>
<td>-</td>
<td>1.6, 1.6, 1.6</td>
</tr>
</tbody>
</table>

1 Low- and high-recycled content. 1 Different manufacturing route.

The mass data must be converted to values of equivalent carbon dioxide emissions due to the production and manufacturing process. This was addressed by sourcing emission and energy data for each primary material, and an additional factor is applied to account for the construction and assembly phase. Emissions data was found in several different sources and presented in Table 4. A mean value based on these was used in the model. Where data for recycled material was not
available, it was reduced by a factor of 0.4, which was obtained from the difference between recycled and non-recycled materials from Pira (2001). The mass of CO₂ emissions due to the production of a component \( M'_{\text{cu}} \) is calculated using equation (13).

\[
M'_{\text{cu}} = \sum_i X_i \cdot M'_{\text{cu},i} \cdot \varepsilon_{\text{cu},i} + K \cdot \sum_i M_{\text{cu},i}
\]

(13)

where \( X \) is the composition (as a mass fraction) of a material and \( \varepsilon_{\text{cu}} \) is the emission factor for the component material, from Table 4. \( K \) is the factor for construction emissions and a value of \( K = 3.0 \) was used, which was based on data for compressors from Hansen and Gustafsson (1996).

Table 5: Emissions during construction, assembly and installation

<table>
<thead>
<tr>
<th>Stage</th>
<th>Emission source</th>
<th>Calculation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transport components to factory</td>
<td>Vehicle (fuel)</td>
<td>Equation (20), where ( \zeta = 0.00006 ) kgCO₂ km⁻¹ (Pira, 2001)</td>
</tr>
<tr>
<td>Assembly (or sub-assembly)</td>
<td>Brazing/metal working</td>
<td>Equation (17), where ( \varepsilon_{\text{brac}} = 0.031 ) kgCO₂ kg⁻¹ of system materials (Frischknecht, 1999)</td>
</tr>
<tr>
<td>Pressure/leak tests</td>
<td>Hose/manifold and trace system losses</td>
<td>See refrigerant handling Section 2.3.3</td>
</tr>
<tr>
<td>Transport equipment to site</td>
<td>Vehicle (fuel)</td>
<td>Equation (20), where ( \zeta = 0.00006 ) kgCO₂ km⁻¹ (Pira, 2001)</td>
</tr>
<tr>
<td>Assembly</td>
<td>Brazing/metal working</td>
<td>Equation (17), where ( \varepsilon_{\text{brac}} = 0.031 ) kgCO₂ kg⁻¹ of system materials (Frischknecht, 1999)</td>
</tr>
<tr>
<td></td>
<td>Personal transport</td>
<td>See Transportation Section 2.3.3</td>
</tr>
<tr>
<td>Pressure/leak tests</td>
<td>Hose/manifold and trace system losses</td>
<td>See Charging Section 2.3.3</td>
</tr>
<tr>
<td>Charging</td>
<td>Hose/manifold and losses</td>
<td>See Charging Section 2.3.3</td>
</tr>
</tbody>
</table>

2.2.2 Construction, assembly and installation

The construction and installation process is characterised by a combination of many different activities, resulting in several different sources of GHG emissions. Typically, components will be transported to a factory where the equipment will be constructed to some degree. Pressure and/or leak tests will be made. The parts will then be transported to the installation site and the remaining components will be added to the final assembly. The equipment will normally be subject to a full leak test, and possibly repeated until the system integrity is ensured, and the system will receive a full charge. For smaller systems (e.g., appliances), this will occur only at the manufacturing location. These sources are therefore included in the calculation of refrigerant...
(M'\textsubscript{emst}) and CO\textsubscript{2} (M'\textsubscript{emst}) emissions for construction activities, equation (14) and (15), respectively.

\begin{equation}
M'\textsubscript{emst} = \sum_i M'\textsubscript{ptr,i} + \sum_i M'\textsubscript{chrg,i} \tag{14}
\end{equation}

\begin{equation}
M'\textsubscript{emst} = \sum_i M'\textsubscript{brz,i} + \sum_i M'\textsubscript{trv,i} \tag{15}
\end{equation}

The use of brazing torches produce emissions, and the amount of gas used is related to the material mass for a given installation (equation 16). From Frischknecht (1999), values ranging from 0.018 to 0.043 kg CO\textsubscript{2} per kg of system materials were obtained. The number of visits requiring personal transport is estimated at one person-day per 100 kg of system mass, based on discussions with contractors (Broadhurst, 2002) (equation 17).

\begin{equation}
M'\textsubscript{brz} = \varepsilon\textsubscript{brz} \cdot \sum_i M'\textsubscript{npm,i} \tag{16}
\end{equation}

\begin{equation}
M'\textsubscript{trv} = \xi \cdot L\textsubscript{trv,i} \cdot \sum_i M'\textsubscript{npm,i} \tag{17}
\end{equation}

where \( L\textsubscript{trv} \) is the total distance travelled in km, for the total mass of the system components, and \( \xi \) is the emission from fuel for transportation per distance travelled.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure6.png}
\caption{Overview of in-use emission sources and influencing parameters}
\label{fig:figure6}
\end{figure}

\section*{2.3 In-use stage}

In-use stage emissions occur primarily from energy production for the equipment operation and the leakage of system refrigerant charge, although refrigerant emissions associated with servicing
activities may also be significant (equation 18 and 19). Figure 6 summarises the sources and some key causes of GHG emissions during in-use stage.

$$M'_{\text{Use}} = M'_{\text{oper}} + M'_{\text{serv}}$$  \hspace{2cm} (18)$$

$$M^*_{\text{Use}} = M^*_{\text{oper}} + M^*_{\text{serv}}$$ \hspace{2cm} (19)$$

2.3.1 System refrigerant emissions

The annual system leakage was estimated from system refrigerant mass using equation (20).

$$M'_{\text{oper}} = R_{\text{oper}} \cdot M_{\text{r,des}}$$ \hspace{2cm} (20)$$

where $R_{\text{oper}}$ is the annual leak rate from the system during operation and $M_{\text{r,des}}$ is the “design” refrigerant charge. The design charge is calculated based on the internal volume of the system components, and local refrigerant density at the design, or rated conditions (as described for the model in Section 2.3.2).

Figure 7: Proportion of refrigerant leaks by mass and number of visits (Radford, 1998)

Considerable work has been conducted to establish typical refrigerant leakage rates for various RAC sectors. Efforts have been made to confirm these data by measuring atmospheric concentrations to estimate the annual and total emitted mass of a given refrigerant. Whilst this technique has been shown to correlate well with total estimated emissions for a given species (McCulloch et al, 2003), it does little to help estimate leakage from individual systems. Instead, characterising leak rates from individual components can provide a more accurate indication of
leakage from specific configurations. Radford (1998) presents data on leakage quantities from components in commercial refrigerating systems. Using this and more recent data for the same and different systems from Ayers (2000), a factor \( \chi \) was derived by dividing the proportion of loss by the average number of each component in a typical system from where the original leak data was obtained. The factor allows calculation of annual leak rate for any set of components (equation 21). Here, the first term accounts for the rise in annual leak rate due to the age of the system, and the second term calculates the initial leak rate from the combination of components adjusted for repair frequency, and \( \Delta \overline{R} \) accounts for leak rate uncertainties.

\[
R_{\text{oper}} = \left( 1 + \frac{dR}{dt} \cdot \Delta t' \right) \left[ \frac{1}{N_{\text{serv}}} \cdot \left( \sum_{i} \chi_{i} \cdot N_{\text{cmp},i} \right) \right] \cdot \Delta \overline{R}
\]  

where \( dR/dt \) represents the incremental increase in annual leak rate for the system over the reference time (\( \Delta t' \)). \( N_{\text{cmp}} \) is the number of components of a specific type, and \( N_{\text{serv}} \) is the number of visits per year by a service person to check for and repair leaks. The component leak index \( \chi \) was determined for each component type. \( \Delta \overline{R} \) is the mean variation in annual leak rate based on the literature, which is discussed later in this section. (Note that the theoretical limit of equation 21 is \( R_{\text{oper}} \times N_{\text{serv}} < 1 \) so that the system has a positive charge size at any time.)

**Table 6: Refrigerant leak quantity and leak factor for specific components**

<table>
<thead>
<tr>
<th>Component</th>
<th>Total mass released (Radford, 1998) (kg)</th>
<th>Total mass released (Ayers, 2000) (kg)</th>
<th>Average no. of components</th>
<th>Component leak factor, ( \chi ) (( \text{s}^{-1} ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coil</td>
<td>7071</td>
<td>907</td>
<td>120</td>
<td>7.304 x 10^{-1}</td>
</tr>
<tr>
<td>Pipeing/brazed joint</td>
<td>35576</td>
<td>4563</td>
<td>1700</td>
<td>2.594 x 10^{-1}</td>
</tr>
<tr>
<td>Control valves</td>
<td>7918</td>
<td>1016</td>
<td>3</td>
<td>3.271 x 10^{-2}</td>
</tr>
<tr>
<td>Check valves</td>
<td>5794</td>
<td>743</td>
<td>3</td>
<td>2.394 x 10^{-2}</td>
</tr>
<tr>
<td>Ball valves</td>
<td>1948</td>
<td>250</td>
<td>288</td>
<td>8.384 x 10^{-3}</td>
</tr>
<tr>
<td>Service valves</td>
<td>16655</td>
<td>2136</td>
<td>60</td>
<td>3.441 x 10^{-3}</td>
</tr>
<tr>
<td>EPR valve</td>
<td>8876</td>
<td>1138</td>
<td>5</td>
<td>2.200 x 10^{-2}</td>
</tr>
<tr>
<td>Solenoid valves</td>
<td>3734</td>
<td>479</td>
<td>5</td>
<td>9.257 x 10^{-3}</td>
</tr>
<tr>
<td>Expansion valves</td>
<td>6369</td>
<td>817</td>
<td>120</td>
<td>6.579 x 10^{-4}</td>
</tr>
<tr>
<td>Control lines</td>
<td>5221</td>
<td>670</td>
<td>50</td>
<td>1.294 x 10^{-3}</td>
</tr>
<tr>
<td>Control equipment</td>
<td>1145</td>
<td>147</td>
<td>50</td>
<td>2.838 x 10^{-4}</td>
</tr>
<tr>
<td>Filter</td>
<td>1553</td>
<td>199</td>
<td>4</td>
<td>4.812 x 10^{-3}</td>
</tr>
<tr>
<td>Compressors</td>
<td>3995</td>
<td>512</td>
<td>12</td>
<td>4.126 x 10^{-3}</td>
</tr>
<tr>
<td>Flare joint</td>
<td>3417</td>
<td>438</td>
<td>5</td>
<td>8.471 x 10^{-3}</td>
</tr>
</tbody>
</table>
Table 6 provides the systems' leaked masses and associated leak factors for a variety of components and connection types. These data are based on values obtained from 410 supermarket systems, which received 2381 service calls throughout the year. The average number of service visits was therefore 5.8, rounded to six per year. It should be noted that the correlation provides an average value since the data from Radford and Ayers showed that the leak rate of individual systems actually varied from 0% to 82% per annum. The inclusion of the frequency of service visits in equation (21) is important because when the leaking component is repaired, a proportion of the annual leakage is prevented. This approach is validated by the leak data from Radford, shown in Figure 7 where the leaked mass (top-up) is closely related to the number of service visits to repair a given component. In terms of the incremental increase in leakage with time, an approximate value of $\frac{dR}{dt} = 0.054 \text{y}^{-1}$ was obtained with the data from Ayers.

However, the confidence for this value is low because data was only available for systems of up to four years old, which showed wide scatter. Radford (1993) claims that both refrigerant and lubricant type have been shown to produce significantly different leak rates. For example, HFCs tend to produce higher leak rates than CFCs because of their molecule size and the greater "scouring" ability of polyolester oils. However, since the data in Table 6 is a compilation of various HFC and HCFC refrigerants the correlation does not allow for refrigerant-specific leaks.

Table 7: Comparison of reported annual leak rates

<table>
<thead>
<tr>
<th>Sector/equipment</th>
<th>Reported $R_{\text{oper}}$ (Source)</th>
<th>Equation (21) ($t = 0$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Domestic ref.</td>
<td>1%</td>
<td>1%</td>
</tr>
<tr>
<td></td>
<td>Retail ref.</td>
<td>- Integrals</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Split/dist</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- S/market</td>
</tr>
<tr>
<td>Air cond.</td>
<td>12-20%</td>
<td>10-20%</td>
</tr>
<tr>
<td></td>
<td>- Split/dist</td>
<td>15-22%</td>
</tr>
<tr>
<td></td>
<td>- Chillers</td>
<td>3-10%</td>
</tr>
</tbody>
</table>

Although the data includes losses from refrigerant handling, the relative mass is very small compared to system charge and is therefore neglected for the leak factor.
Verification of this new approach was achieved by comparing equation (21) against data obtained by measurement or estimated leak rates for specific types of equipment. Table 7 summarises leak rates found in the literature, and the results of equation (21). It was found that calculated values of $R_{\text{oper}}$ were generally within the ranges of those cited in Table 7. Referring back to equation (21), the values for $\Delta \bar{R}$ are obtained from the ranges of leak rates reported in Table 7. Across all equipment types, minimum $\Delta \bar{R} = 0.43$, mean $\Delta \bar{R} = 1$ and maximum $\Delta \bar{R} = 1.91$.

2.3.2 System energy use emissions

A significant proportion of the GHG emissions can be related to the energy consumption of the equipment throughout its in-use stage. The energy-related CO$_2$ emissions are calculated from the equipment energy consumption (equation 22).

$$M_{\text{oper}}^* = \sum_{i} \xi_i \cdot E_i$$

(22)

where $\xi$ is the "emission factor", $E$ is the energy consumed by the system, and $i$ represents each set of conditions.

![Figure 8: Historical and projected power station CO2 emission factor (DTI, 1998, 2000)](image)

Within the UK, electricity is generated through a number of different routes including coal, gas, oil, nuclear and a number of renewable sources, each of which results in different levels of GHG emissions per unit of energy produced.7 The average annual kg CO$_2$ per unit of electricity (termed "emission factor", $\xi$ ) changes with the introduction of new, or withdrawal of old power stations. Similarly, the "fuel mix" changes according to total demand at that time, and therefore the

---

7 Note that in countries such as Norway where nearly all electricity originates from hydropower, the power station emissions component is negligible.
emissions factor varies. Such changes in emissions factor should be taken into account because these changes occur throughout the equipment life, and different types of system may operate for different periods throughout the day and year.

Table 8: Constants for emissions factor projections

<table>
<thead>
<tr>
<th>Constant</th>
<th>Lower</th>
<th>Mean</th>
<th>Upper</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_1$</td>
<td>475.12</td>
<td>502.59</td>
<td>583.01</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-0.46972</td>
<td>-0.49632</td>
<td>-0.57566</td>
</tr>
<tr>
<td>$a_3$</td>
<td>0.00011620</td>
<td>0.00012263</td>
<td>0.00014219</td>
</tr>
<tr>
<td>$\nu_Y$</td>
<td>0.0567</td>
<td>0.0631</td>
<td>0.0693</td>
</tr>
</tbody>
</table>

Figure 8 shows how the average annual emissions factor has reduced over the past 25 years (DTI, 1998). Also included are the projected emissions until 2020 (DTI, 2000), where the two data sets represent two alternate economic scenarios. Data for the two scenarios was fit to an equation for use in the model (equation 23). In addition, seasonal and day/night variations are also accounted for, following the approach of Ure (1995) and Beggs (1996) respectively. As with the production of construction materials, electricity generation produces emissions of GHGs other than CO$_2$, but have been converted to CO$_2$ equivalent using GWP(100) for purposes of model simplicity.

$$\xi = (a_1 + a_2 \cdot Y + a_3 \cdot Y^2) + (\nu_Y \cdot \sin \theta_Y) + \nu_D$$  \hspace{1cm} (23)

where the first term represents the mean annual emission factor (see curve fit in Figure 8), $a_1$, $a_2$ and $a_3$ are constants and $Y$ is the year, e.g., 2005. The second and third terms represent the annual and daily variation in emissions respectively, where $\nu_Y$ is the mean deviation of CO$_2$ emissions throughout a year, and $\nu_D$ is the mean deviation for day ($\nu_D = 0.088$ kgCO$_2$ kW$^{-1}$ h$^{-1}$) or night ($\nu_D = -0.088$ kgCO$_2$ kW$^{-1}$ h$^{-1}$). $\theta_Y$ is calculated according to the peak emissions over the annual time frame, using equation (24). Values for $a_1$, $a_2$, $a_3$ and $\nu_Y$ are presented in Table 8, and were obtained from DTI (1998) and DTI (2000).

$$\theta_Y = \frac{\pi}{t_{y,\text{Tot}}} (t_y - 9.107)$$  \hspace{1cm} (24)
where $t_{y,T_{0C}}$ is the total number of time increments in the cycle (i.e., 12 months), $t_Y$ is the time (in months) and the constant 9.107 is the time at which peak $\xi$ occurs.

**Refrigeration system performance**

Due to the potentially high emissions from energy production, it is important to assess system energy consumption in detail. Since comparisons are often made between refrigerants, which perform differently according to the conditions, the model should have a strong dependence on refrigerant properties and should account for transients such as changing ambient temperature and effects of ageing. For any set of conditions, the performance of the refrigeration cycle is calculated and the associated energy consumption is used to estimate the CO$_2$ emissions from electricity production. Also, since calculation of performance demands estimation of the systems' charge size, this was also used for determining refrigerant emissions elsewhere (Sections 2.2.2, 2.3.1, 2.3.3 and 2.4).

It is noted that the performance model was not validated, for two reasons. First, no test data was available where measurements accounted for the various parameters included, such as ageing aspects. Second, the purpose of the model is for comparative purposes and not for prediction of a particular piece of equipment. Nevertheless, the majority of the sub-models are largely empirical which provides confidence in a realistic output.

The model was developed for a single-stage vapour-compression cycle, and was used for five types of performance calculations:

(i) **Steady system performance at design conditions.** Based on specified design conditions, the compressor size (swept volume) is determined, according to the condenser, evaporator and pipework design, condenser heat transfer fluid (HTF) inlet temperature, application temperature, the temperature surrounding the application, and refrigerant selection.

(ii) **Steady system performance at non-design conditions.** Based on specified component designs and the compressor size determined for (i), performance is calculated at non-design ambient conditions, whilst assuming that all dependent temperatures are allowed to float. Non-design conditions such as different condenser HTF inlet temperature and/or
temperature surrounding the application, due to monthly mean day and night temperatures, used to calculate energy consumption over a 12-hour period for each.\(^8\)

(iii) **Non-steady system performance at non-design conditions.** Based on specified component designs and the compressor size determined for (i), performance and 12-hourly energy consumption is calculated for the non-design monthly mean day and night ambient conditions. However, application temperatures are not allowed to float outside a specified range so the system operation is controlled to ensure the application temperature is within a chosen range about a set-point, and this may produce on-off cycling of the compressor. If the ambient conditions are such that the system cannot provide the capacity to achieve the lower application temperature, steady operation occurs as (ii).

(iv) **System performance at design and non-design conditions with ageing.** With each time step throughout the equipments in-use stage, ageing occurs in the form of degradation of component performance and loss of refrigerant. Over time, system capacity deteriorates which affects system equilibrium conditions for steady operation (ii), or cycling behaviour for non-steady operation (iii), and therefore energy consumption.

(v) **System performance with retrofitted refrigerant.** Based on specified component designs and the compressor size determined for (i), performance is calculated according to the properties of a different refrigerant chosen as a retrofit. Thereby the performance of the system under (ii) steady, (iii) non-steady, or (iv) ageing situation can be determined.

---

* A more accurate "bin" type approach was considered, but this would conflict with the inclusion of ageing aspects; although, monthly "bin" data could combine with ageing models, but would result in excessive calculation time.
The various options for performance calculations are indicated in Figure 9. Initially, the design calculations (i) are necessary so that compressor size is obtained. Once the purpose of the simulation has been chosen, the appropriate sub-models and calculation sequence are selected, which include existing or retrofitted refrigerant (v), ageing (iv) or no ageing, and non-design conditions under steady operation (ii) or with on/off cycling (iii). For any of these off-design calculations, a general iterative sequence was employed to determine equilibrium states for steady operation, or the performance at non-equilibrium conditions between the prescribed upper and lower application temperature limits. The general calculation sequence covering all situations is described in two steps: first for the determination of compressor size (i), and secondly for performance at non-design conditions (ii, iii, iv and v). Details for the various component sub-models are provided below, as are the methods for evaluating transient influences such as compressor wear, heat exchanger fouling, and effect of leakage.

For the first step of calculations, a number of input parameters are required which include design temperatures, characteristics of the application heat load and equipment construction details, whilst the compressor size is determined by the model so that its capacity is matched to the heat load at the chosen design conditions. Initially, the saturated compressor suction \((T_{s,suc})\) and discharge \((T_{s,disch})\) temperatures are estimated from the application temperature \((T_{app})\), which in this case is the design \((T_{app,des})\) and ambient \((T_{amb})\) temperatures (based on a 10 K approach).

For a given space to be cooled, the application heat load \((Q_{app})\) is obtained from a basic rating equation (25).

\[
Q_{app}(T_{app}) = b_1 + B_{app} \cdot (T_{amb} - T_{app})
\]  

(25)

where \(b_1\) is a constant and represents heat loads that are independent of ambient temperature, such as electrical devices and occupants. \(B_{app}\) is the application basic rating and \(T_{amb}\) is the temperature surrounding the space. In the case of a refrigerated box \(T_{amb}\) is room temperature, or for an air conditioned space \(T_{amb}\) is the outside temperature, and \(T_{app}\) is the application temperature, inside the cabinet or the room respectively.

From this the mass flow of refrigerant \((\dot{m}_f)\) is calculated from the specific enthalpy difference based on \(T_{s,suc}\) and \(T_{s,disch}\) (equation 26).
\[ m_r = \frac{Q_{\text{app}}}{H_1 - H_4} \]  

(26)

Next, \( m_r \) is used to determine the capacity of the evaporator \((Q_e)\) and capacity of the condenser \((Q_c)\) from the evaporator and condenser sub-models discussed below, as well as refrigerant outlet states for calculation of the succeeding component. Heat transfer and pressure losses associated with interconnecting pipework are also calculated, which are needed for the compressor suction and discharge and TEV inlet states. Using the evaporator and condenser characteristics, revised values for refrigerant states at the compressor suction \((T_{\text{suct}}, P_{\text{suct}})\) and discharge \((T_{\text{disch}}, P_{\text{disch}})\) points are determined. \( m_r \) is then re-calculated from equation (26), where \( H_x \) corresponds to \( H_{\text{suct}} \) and \( H_{\text{disch}} \). \( Q_e \) and \( Q_c \) are subsequently calculated again, and the process iterated until \( Q_e = Q_{\text{app}} \), where the system capacity matches application heat load for the design conditions. Henceforth, the design compressor size (in terms of swept volume, \( V_{sw} \)) is found (equation 27).

\[
V_{sw} = \frac{b_1 + \rho_{\text{app}} \cdot (T_{\text{amb}} - T_{\text{app, des}})}{\eta_{\text{vol}} \cdot \rho_{\text{suct}} \cdot (H_1 - H_4)}
\]  

(27)

where the upper term represents the application heat load, and therefore evaporating capacity (equation 25), \( \eta_{\text{vol}} \) is the volumetric efficiency and \( \rho_{\text{suct}} \) is refrigerant density at the compressor suction. \( V_{sw} \) was converted to cylinder volume (assuming 1400 rpm compressor speed) for calculation of compressor and oil mass in Section 2.2.1.

For the second step of calculations, each set of non-design conditions (ambient temperatures, ageing effects, refrigerant, etc.) are specified whilst maintaining existing system design with the previously determined compressor size (equation 27). Since \( Q_e \) is now unlikely to match \( Q_{\text{app}} \) at the required \( T_{\text{app, des}} \), it is necessary to establish system performance characteristics at the new system equilibrium conditions, or at incremental \( T_{\text{app}} \) between the upper and lower limits specified for the application temperature. In this case, incremental \( T_{\text{app}} \) is interpreted as a floating application temperature, denoted \( T'_{\text{app}} \). Determination of performance characteristics for any of the calculation types (ii, iii, iv, v) follows the same procedure.
Firstly, an initial guess of $T_{s,act} = T_{app,des} - 10$ K and $T_{s,dish} = T_{amb} + 10$ K is made as before, and $\dot{m}_r$ is now estimated from $\dot{V}_{sw}$ (equation 28).

$$\dot{m}_r = \dot{V}_{sw} \cdot \eta_{vol} \cdot \rho_{act} \cdot (H_1 - H_4)$$

(28)

With $\dot{m}_r$, the performance of each component ($Q_e$, $Q_c$, heat transfer and pressure losses associated with interconnecting pipework) is calculated. $Q_e$ is then compared against $Q_{app}$, and if they do not match, $T'_{app}$ is adjusted incrementally ($\delta T'$) until $Q_e$ matches $Q_{app}$ at the new application temperature ($T'_{app} \pm \delta T'$). $\delta T'$ for each new iteration is estimated using values from the prior iteration in equation (29), which is based on the relationship between change in application capacity with temperature. It was found that this convergence approach (as opposed to say, successive guesses of $\dot{m}_r$ proportional to $Q_{app} / Q_e$) was necessary to avoid instability.

$$\delta T' = \left[ \exp\left( \frac{Q_{app}(T'_{app})}{Q_e(T'_{app})} \right) \cdot \frac{T'_{app} - \delta T'}{Q_{app}(T'_{app}) - \delta Q'_{app}} \right]$$

(29)

where $Q_{app}(T'_{app})$ and $Q_e(T'_{app})$ are capacities at $T'_{app}$ from the previous iteration, and $\delta Q'_{app}$ and $\delta T'$ are the incremental change in application heat load and temperature respectively, also from the previous iteration.

If the model is set for steady operation (ii), then $T'_{app}$ is adjusted until $Q_e(T'_{app}) = Q_{app}(T'_{app})$, and energy consumption is calculated on the basis that the system is in constant operation. If non-steady or cycling operation occurs the calculation of energy consumption is determined according to the method described below.

System cycling

It is important to know compressor cycling time, as this has a direct influence on power consumption (although power surge at compressor start-up has been neglected), and indirectly, on degradation of compressor performance. In order to evaluate cycling time, a simple method was developed as follows, based on application heat load characteristics and system performance at off-design conditions, but neglecting pull-down from the initial (ambient) temperature. Cycling occurs due to termination and initiation of a single cylinder compressor, in response to a thermostat which is set to respond to an upper ($T_{app,max}$) and lower ($T_{app,min}$) application temperature, whilst the actual temperature ($T'_{app}$) floats between the two; towards $T_{app,min}$ during
compressor on-cycle and towards \( T_{\text{app,max}} \) during off-cycle. The difference between \( T_{\text{app,max}} \) and \( T_{\text{app,min}} \) is the thermostat setting (\( \Delta T_{\text{set}} \)), and the design temperature is the mean, \( T_{\text{app,des}} = T_{\text{app,min}} + \Delta T_{\text{set}} / 2 \). It is noted that any lag due to the thermal response of the thermostat itself is neglected in this assessment.

As mentioned above, \( \dot{V}_{sw} \) is calculated for design conditions so that \( Q_e \) meet the design load of the application (at full charge and no ageing), thus at design conditions the compressor operates continuously and \( T'_{\text{app}} = T_{\text{app,des}} \). It is assumed that the design conditions represent those that would produce (at least) the greatest demand on the refrigerating system, so under off-design conditions, the system has “excess” capacity. As a result, once \( T_{\text{app,des}} \) is achieved \( Q_e \) exceeds \( Q_{\text{app}}(T_{\text{app,des}}) \), and \( T'_{\text{app}} \) continues to fall from \( T_{\text{app,des}} \) towards \( T_{\text{app,min}} \). In order to calculate the compressor (and ancillary devices) energy consumption, it is necessary to determine the average “instantaneous” power demand at each incremental \( T'_{\text{app}} \), and corresponding pull-down time (between \( T_{\text{app,max}} \) and \( T_{\text{app,min}} \)) for other on/off electrical devices.

The rate of decrease in application temperature – or pull-down rate – during the on-cycle is a function of the thermal mass of the contents of the cooled space, and the excess capacity of the refrigerating system that enables a lower temperature to be achieved (equation 30).

\[
\frac{dT'_{\text{app}}}{dt} = \frac{\Delta Q(T'_{\text{app}})}{(M \cdot C_{\text{p}})_{\text{app}}}
\]

(30)

where \( \Delta Q(T'_{\text{app}}) \) is the difference between \( Q_{\text{app}}(T'_{\text{app}}) \) and \( Q_e(T'_{\text{app}}) \), and \( (M \cdot C_{\text{p}})_{\text{app}} \) represents the thermal capacity of the space.

For each step of \( T'_{\text{app}} \), pull-down time for each \( \Delta T' \) can be obtained by rearranging equation (30), and substituting \( \Delta Q(T'_{\text{app}}) \) for \( Q_e(T'_{\text{app}}) \) and \( Q_{\text{app}}(T'_{\text{app}}) \) from equation (25), as equation (31).

\[
dt = \frac{(M \cdot C_{\text{p}})_{\text{app}}}{Q_e(T'_{\text{app}}) - b_1 + B_{\text{app}} \cdot (T_{\text{amb}} - T'_{\text{app}})} \cdot dT'_{\text{app}}
\]

(31)

Total pull-down time – or on-cycle time (\( \Delta t_{\text{onc}} \)) – requires equation (31) to be integrated over the \( T'_{\text{app}} \) range according to the thermostat setting \( \Delta T_{\text{set}} \) (equation 32).
\[ \Delta t_{\text{onc}} = \Delta t_{\text{start}} + \int_{t_{\text{app, min}}}^{t_{\text{app, max}}} (M \cdot C_p)_{\text{app}} / \Delta Q(T'_{\text{app}}) \cdot dT'_{\text{app}} \]  

(32)

where \( \Delta t_{\text{start}} \) is an additional term which accounts for the extra time at start-up due to degassing of refrigerant absorbed in the oil and flooding of refrigerant into other parts of the system. Whilst this value is dependent upon oil quantity, refrigerant/oil properties and external conditions, a single time of 90 s has been used, as determined by Phillip et al (2000). However, this is known to vary considerably in different types of systems.

Similarly, the off-cycle time (\( \Delta t_{\text{off}} \)) is calculated as with equation (32), but since the refrigerating capacity is absent, the change in \( T'_{\text{app}} \) is dependent only upon the application heat load, and the \( \Delta t_{\text{start}} \) term is not relevant (equation 33).

\[ \Delta t_{\text{off}} = \int_{t_{\text{app, min}}}^{t_{\text{app, max}}} (M \cdot C_p)_{\text{app}} / Q_{\text{app}}(T'_{\text{app}}) \cdot dT'_{\text{app}} \]  

(33)

As required for certain ageing calculations, the number of compressor starts (\( N_{\text{start}} \)) over a given period of time (\( \Delta t' \)) is determined from equation (34).

\[ N_{\text{start}} = \frac{\Delta t'}{\Delta t_{\text{onc}} + \Delta t_{\text{off}}} \]  

(34)

For presentation of results, \( T'_{\text{app}}, Q_{\text{app}}, Q_e, \text{COP}, \text{etc.} \), are given as time-averaged mean (based on \( \Delta t_{\text{onc}} \) and \( \Delta t_{\text{off}} \)) as \( T'_{\text{app}} \) floats between \( T_{\text{app,max}} \) and \( T_{\text{app,min}} \).

System component sub-models

Sub-models for the main system components - compressor, evaporator and condenser - were those described elsewhere in this study (Part 2, Section 3). However, some minor modifications were introduced to address certain factors relevant to the current purpose.

Compressor performance is highly variable due to the large number of designs and combinations of refrigerants and oils, and to account for such a wide number of parameters would demand very detailed methods and thereby deviate from the current purpose. Whilst the intention is partly to assess differences between refrigerants, it was argued by compressor manufacturers that it is possible to optimise a compressor to suit any given refrigerant and oil combination (Jurgenssen, 2004, Blomhardt, 2002). Therefore, an approach was chosen where such effects are eliminated.
from the direct calculation of compressor power consumption (although the existing method for
calculation of refrigerating capacity and volumetric efficiency are still employed). Subsequently,
the compressor model presented elsewhere (Part 2, Section 3.2) was adapted so that instead of
calculating compressor power \( W_{\text{comp}} \) with the ideal mean pressure approach, it was determined
from compressor refrigerating capacity \( Q_{e,\text{comp}} \) and an empirically derived COP (equation 35).

\[
W_{\text{comp}} = \frac{Q_{e,\text{comp}}}{COP_{\text{comp}}} \quad (35)
\]

where \( COP_{\text{comp}} \) is the coefficient of performance calculated from an empirical correlation based
on the performance of a real compressor as a function of suction \( p_{\text{suct}} \) and discharge \( p_{\text{dusch}} \)
pressures only (equation 36).

\[
COP_{\text{comp}} = \exp (a_1 + a_2 \cdot p_{\text{suct}} + a_3 \cdot p_{\text{dusch}} + a_4 \cdot p_{\text{suct}} \cdot p_{\text{dusch}}) \quad (36)
\]

where \( a_1, a_2, \) and \( a_3 \) are obtained for a specific compressor, and exemplary values from
catalogue data for two different products are provided in Table 9, and are an average of the
various refrigerants for which data was provided. By calculating \( W_{\text{comp}} \) in this manner isentropic
efficiency is dependent mainly upon the compression ratio, and therefore system COP is
ultimately dependent upon the performance of the respective refrigerants within the other
components, such as heat exchangers and pipework.

**Table 9: Constants for equation (36) for two different compressors**

<table>
<thead>
<tr>
<th>Compressor (manufacturer)</th>
<th>2 KC-05.2 (Bitzer)</th>
<th>2 GC-2.2 (Bitzer)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a_1 )</td>
<td>( 8.374 \times 10^{-1} )</td>
<td>( 8.048 \times 10^{-1} )</td>
</tr>
<tr>
<td>( a_2 )</td>
<td>( 3.239 \times 10^{-3} )</td>
<td>( 3.469 \times 10^{-3} )</td>
</tr>
<tr>
<td>( a_3 )</td>
<td>( -5.826 \times 10^{-4} )</td>
<td>( -5.021 \times 10^{-4} )</td>
</tr>
<tr>
<td>( a_4 )</td>
<td>( -5.334 \times 10^{-7} )</td>
<td>( -7.625 \times 10^{-7} )</td>
</tr>
</tbody>
</table>

The total energy drawn by the compressor is the product of \( W_{\text{comp}} \) at the given conditions and
operating time (equation 37).

\[
E_{\text{comp}} = \sum_i W_{\text{comp},i} \cdot \Delta t_{\text{onc},i} \quad (37)
\]

where \( \Delta t_{\text{onc}} \) is the duration of the compressor on-cycle for a given set of conditions, \( i \).
Finally, estimation of the heat that the compressor adds to the refrigerant is assumed to be equal to the compressor power, minus 5% of the heat (Lundqvist, 2004) which is rejected to the environment (equation 38),

\[ Q_{\text{comp}} = (1 - 0.05) \cdot W_{\text{comp}} \]  

Heat exchanger performance was calculated with the evaporator and condenser models detailed elsewhere (Part 2, Section 3.3), and the internal volume of the evaporator and condenser is used to determine component mass for Section 2.2. In addition, the power required for HTF flow across the evaporator and condenser was evaluated. This was considered important since a higher HTF flow rate increases heat transfer coefficient and thereby improves the COP, whilst being offset by the additional power required by the fan or pump. This was calculated using equation (39), which combines the conventional proportionality equation and a correlation for pressure drop across an exchanger.

\[ W_{\text{HTF}} = c \cdot \frac{\dot{V}_{\text{HTF}}}{\eta_{\text{pump}}} \left( \frac{\dot{V}_{\text{HTF}}}{A_{\text{HTF}}} \right)^{1.8} \]  

where \( \eta_{\text{pump}} \) is the efficiency of the pump or fan, and \( c \) is an empirical constant (in \( \text{kJ m}^{-3} \)) for a specific pump or fan in a specific HTF circuit or duct arrangement. \( A_{\text{HTF}} \) is the face area of the exchanger in the direction of HTF flow.

Pressure losses along suction and discharge lines are included in the model, whilst liquid lines are neglected since the pressure drops are known to be insignificant. Conventional single-phase pressure drop correlations are used (equation 40) (Incropera and DeWitt, 1990).

\[ \Delta p = \frac{\lambda \cdot \rho_r \cdot u_r^2}{2 \cdot D} \cdot L \]  

where \( \rho_r \) is refrigerant density, \( u_r \) is mean velocity, \( D \) is inside tube diameter and \( L \) is pipe length. The friction factor is from equation (41) (Incropera and DeWitt, 1990).

\[ \lambda = a \cdot Re^n \]  

where \( Re = \rho_r \cdot u_r \cdot D / \mu_r \), and \( \mu_r \) is refrigerant viscosity. Properties were calculated for local temperatures and pressures at each increment. For \( Re < 20000 \), \( a = 0.316 \) and \( n = -0.25 \), and \( a = 0.184 \) and \( n = -0.20 \) when \( Re > 20000 \).
Similarly, calculations for pipework included finding internal volumes which was used to determine the mass for pipe and valve materials, in Section 2.2.

Electrical devices associated with the refrigerating system should also be added to the calculation. This ancillary equipment includes evaporator and condenser fans, lighting, control circuits, defrost, oil and trace heaters. The energy consumption of these is simply the sum of their electrical rating \( W_{\text{anci}} \) and their respective run times \( \Delta t_{\text{anci}} \) for each device, \( i \) (equation 42).

\[
E_{\text{anci}} = \sum_i W_{\text{anci},i} \cdot \Delta t_{\text{anci},i} \tag{42}
\]

### Transient influences on performance

In addition to the previously described sub-models, three time dependent influences alter performance over time. With continued use of the system, degradation of system performance occurs in the form of compressor wear, exchanger fouling and sub-optimal refrigerant charge. Therefore, for a refrigerating system operating under constant conditions, the COP and capacity will reduce with time.

#### Compressor wear

Compressor wear causes a reduction in volumetric flow of refrigerant and therefore cooling capacity. A lower cooling capacity and (at least) the same motor power results in lower compressor COP (Garland, 2004). Whilst these mechanisms are related to operating time, most are a strong function of the number of compressor starts. Contact surfaces are initially absent of oil, and when oil becomes available, it is initially well mixed with refrigerant making it less effective as a lubricant; subsequently, more surface buffing and changes in geometry occurs. In the performance model, degradation in compressor performance is accounted for by using a simple empirical equation for volumetric efficiency (43) and compressor COP (44), obtained based on the data fitting of Garland’s results.

\[
\frac{dN_{\text{vol}}}{dt} = 1 - \exp \left( a \cdot \sqrt{t_{\text{op}} \cdot N_{\text{start}}} \right) \tag{43}
\]

\[
\frac{dCOP}{dt} = 1 - \exp \left( a \cdot \sqrt{t_{\text{op}} \cdot N_{\text{start}}} \right) \tag{44}
\]
where \( a \) is the same for both equation (43) and (44); for maximum, mean and minimum uncertainty limits, the value of \( a \) is \(-674.5 \times 10^{-3}\), \(-27.0 \times 10^{-3}\), and \(-13.5 \times 10^{-3}\), respectively. 

\( t_{\text{op}} \) is the operating time of the compressor (in years) and \( N_{\text{start}} \) is the number of compressor starts. Although Garland (2004) provides experimentally derived equations for estimating the reduction in flow rate, it requires detailed knowledge of the internal geometry of the compressor. This approach was considered too thorough for the current model, so instead typical values provided by Garland (2004) and Jurgensen (2004) were used to determine \( a \) in equation (43) and (44), for a 10% reduction in \( Q_e \) for 20,000 operating hours and 100,000 starts.

![Figure 10: Changes in heat transfer of fouling over time](image)

**Heat exchanger fouling**

Modifications were made to the existing heat exchanger models to account for the influence of fouling. For condensers and evaporators, refrigerant-side fouling is normally minimal (excepting oil effects) since the internal circuit has to be free of contaminants, and was neglected. However, significant fouling does occur when water, air or other secondary HTFs are used because the conditions are relatively uncontrolled. A number of studies have addressed the issue, and most present formulations to estimate the additional thermal resistance, i.e., increase in \( x/k \), associated with the fouling. Examples of studies of fouling effects on heat transfer coefficient over time are presented in Figure 10.

Whilst some models are very comprehensive, a more simplistic approach was used, as detailed by Butterworth (2002), (equation 45).
\[
\frac{d(x/k)}{dt} = a_1 \cdot Re_{HTF}^{0.88} \cdot \exp \left( -\frac{E}{8.314 \cdot (273 + T_w)} \right) - a_2 \cdot s_w
\]

(45)

where \( a_1 = 8.39 \text{ s m}^{-2} \text{ K W}^{-1} \), \( 8.314 \) is the gas constant, \( T_w \) is wall surface temperature and \( E = 68 \text{ 000 J mol}^{-1} \) and is the chemical reaction activation energy. Whilst \( a_2 \) is provided by Butterworth for oil, the applicability of equation (45) was extended to heat exchangers used in refrigeration by correlating to empirical data reported by Somerscales (1997), producing the values in Table 10. \( s_w \) is the wall shear stress (equation 46).

\[
s_w = c \cdot Re_{HTF}^n \cdot \rho_{HTF} \cdot u^2
\]

(46)

where \( c \) and \( n \) depend on the type of flow (Incropera and DeWitt, 1990). For flow over fins, \( c = 0.332 \) and \( n = 0.5 \) for laminar, and \( c = 0.664 \) and \( n = 0.5 \) for turbulent flow. For flow within tubes, \( c = 0.158 \) and \( n = 0.25 \) for laminar, and \( c = 0.092 \) and \( n = 0.2 \) for turbulent flow.

**Table 10: Constants for fouling equation (45)**

<table>
<thead>
<tr>
<th>Source</th>
<th>Materials</th>
<th>( a_2 ) (Pa s m(^2) K W(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Butterworth. 2002</td>
<td>Steel/oil</td>
<td>(-4.03 \times 10^{-11})</td>
</tr>
<tr>
<td>McAllister et al (Somerscales. 1997)</td>
<td>Copper/water</td>
<td>(-5.44 \times 10^{-13})</td>
</tr>
<tr>
<td>Greiss et al (Somerscales. 1997)</td>
<td>Aluminium/water</td>
<td>(-1.85 \times 10^{-12})</td>
</tr>
<tr>
<td>Gutzeit (Somerscales, 1997)</td>
<td>Aluminium/water</td>
<td>(-2.06 \times 10^{-12})</td>
</tr>
<tr>
<td>Ritter and Suitor (Somerscales, 1997)</td>
<td>Copper/water</td>
<td>(-2.69 \times 10^{-12})</td>
</tr>
<tr>
<td>Lower</td>
<td>-</td>
<td>(-5.44 \times 10^{-13})</td>
</tr>
<tr>
<td>Mean</td>
<td>-</td>
<td>(-1.79 \times 10^{-12})</td>
</tr>
<tr>
<td>Upper</td>
<td>-</td>
<td>(-2.69 \times 10^{-12})</td>
</tr>
</tbody>
</table>

The additional thermal resistance due to fouling after a given duration of operation is accounted by inclusion of the calculation of overall heat transfer coefficient (e.g., equation 47). It was assumed the fouling was equally distributed along the entire HTF surface.

\[
U = \left[ \frac{1}{h_r} + \left( \frac{x}{k} \right)_{foul} + \frac{1}{h_{HTF}} \right]^{-1}
\]

(47)

where \( h_r \) and \( h_{HTF} \) are the refrigerant and HTF heat transfer coefficients respectively.
Leakage effect on performance

When refrigerant leaks the total quantity of refrigerant within the system reduces and thereby starves the evaporator. The resulting effect is increase in temperature lift and loss of cooling capacity, which forces an increase in run time and generally a reduction in COP. The consequence of leakage was integrated into the model by simulating the operating characteristics of an under-charged system. Since lowered refrigerant charge manifest as cessation of phase-change progressively further away from the evaporator outlet, it was simulated by reducing the surface area available for phase-change \( A_{tp} \) in some proportion to the loss of charge (equation 48); the remaining evaporator area provides superheating only.

\[
\frac{A_{tp}(\varphi)}{A_{tp}} = 1 - (1 - \varphi(t))^n; \quad \varphi(t) \leq 1
\]  

where \( \varphi(t) \) is the reduced charge due to leakage at time \( t \) and \( n \) represents the sensitivity of the evaporator to charge reduction. The reduced charge can be expressed as the charge mass at \( t \) as a fraction of the design charge \( (\varphi(t) = M_r(t)/M_r\text{,des}) \), or as the "unleak" fraction \( (\varphi(t) = 1 - R_{oper}(t)) \), and \( M_r\text{,des} \) is the optimum refrigerant charge for the system under design conditions.

Table 11: Studies on effects of refrigerant leakage

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Non-specific</td>
<td>Split a/c</td>
<td>Split a/c</td>
<td>-</td>
</tr>
<tr>
<td>Refrigerant</td>
<td>R407C</td>
<td>Not known</td>
<td>R22</td>
<td>R22</td>
</tr>
<tr>
<td>Evap/cond</td>
<td>Air/air</td>
<td>Air/air</td>
<td>Air/air</td>
<td>Air/air</td>
</tr>
<tr>
<td>Exp. device</td>
<td>TEV, EEV</td>
<td>TEV, Cap</td>
<td>TEV</td>
<td>TEV</td>
</tr>
<tr>
<td>Capacity</td>
<td>3.5 kW</td>
<td>12 kW</td>
<td>11 kW</td>
<td>11 kW</td>
</tr>
<tr>
<td>Charge size</td>
<td>4 kg</td>
<td>3.6 kg</td>
<td>4 kg</td>
<td>3.0 kg</td>
</tr>
<tr>
<td>Ambient</td>
<td>18°C</td>
<td>Not known</td>
<td>35°C</td>
<td>35°C</td>
</tr>
<tr>
<td>Room temp</td>
<td>26°C</td>
<td>Not known</td>
<td>21°C</td>
<td>21°C</td>
</tr>
</tbody>
</table>

To determine realistic values of \( n \), the system model was compared with data from five different studies, details for which are listed in Table 11. Information about any of the tested systems was insufficient to make a direct comparison against the model because many system characteristics were absent. Instead, simulations were made using inputs typical of an air-to-air split air conditioner/heat pump (representative of most of the equipment in the studies). Some results based on the conditions in Table 11 are plotted in Figure 11. By varying values of \( n \), it was
found that nearly all of the reported variations in COP fell within the range calculated by the model when \( n = 1/3 \) and 3/2. These were chosen for the upper and lower limits, with \( n = 3/4 \) being the mean. Two cases lay outside this range, the data from Farzad and O'Neal (1992) for a TEV system implies that the optimum charge was lower than specified, hence an increase in COP with loss of charge. If the data were compared relative to an optimum charge 10% lower, then the data would lie within the modelled range.

Since performance must be evaluated progressively throughout the equipment life, it was necessary to determine the charge deficit at each time increment. It was assumed that the leak rate was constant, so that the reduced charge \( \varphi(t) \) is linearly proportional with time, and that service calls are spaced equally over a year (with the final visit at the end of the year). At each service visit, the service person finds the leak and replenishes the system to its design charge, and since \( \varphi(t) = 1 - R_{\text{oper}}(t) \), the reduced charge at any time throughout the year is estimated from equation (49).

\[
\varphi(t) = \left( \frac{R_{\text{oper}}}{N_{\text{gap}}} - \frac{R_{\text{oper}}}{A'} \right) t
\]  

(49)

where \( R_{\text{oper}} \) is the total leak fraction over one year, \( N_{\text{gap}} \) is the number of service calls per year where the system is topped-up with refrigerant, and \( A' \) is the reference time of one year.

---

**Figure 11: Comparison of leak effects on COP**
2.3.3 Emissions from servicing and maintenance activities

During servicing and maintenance, emissions predominantly originate from refrigerant handling ($M_{serv}'$, equation 50), whilst CO$_2$ emissions ($M_{serv}^{*}$) are minor contributions from transport and possibly assembling replacement components (equation 51).

$$M_{serv}' = M_{charg}' + M_{vent}' + M_{pnt}' + M_{recov}' + M_{evac}'$$ \hspace{1cm} (50)

$$M_{serv}^* = M_{trans}^* + M_{cmsg}^* + M_{brac}^*$$ \hspace{1cm} (51)

---

Figure 12: Activities producing emissions during servicing
Refrigerant handling and other peripheral sources

Throughout the in-use stage of the RAC equipment, regular servicing (and maintenance) is normally provided, and has several purposes: confirming system operation, leak checking, topping-up refrigerant, leak repair, and replacing failed components. During several of these activities, refrigerant emissions may occur because of transferring refrigerant. It is important to account for these losses because studies show they can be significant. For example, Clodic (1998) reports that 60% of emissions from chillers and 16-29% of emissions of supermarket systems occur during refrigerant handling. Similarly, Butler (1993) reports that 30% of emissions are from refrigerant handling activities. In addition to releases of refrigerant, other peripheral GHG emissions occur, such as from transportation to and from the installation. Typical servicing/maintenance procedures are identified in Figure 12, and the specific activities that result in emissions are identified with a shaded box.

The actual emissions associated with servicing activities are strongly dependent upon the behaviour of the service person. Table 12 shows the basis for the calculation of refrigerant mass emitted during a poor and best practice servicing activity, at the stages identified in Figure 12. Each of the sources was modelled as detailed in the following sections.

Table 12: Details of servicing activities leading to refrigerant emissions

<table>
<thead>
<tr>
<th>Activity</th>
<th>Process</th>
<th>Basis of mass of refrigerant emission</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Poor practice</td>
</tr>
<tr>
<td>Topping-up</td>
<td>Charging</td>
<td>Liquid in hoses/ manifold at system pressure</td>
</tr>
<tr>
<td>Repair leak or component</td>
<td>Venting</td>
<td>Release all refrigerant, ( P_{vent} = 1 )</td>
</tr>
<tr>
<td></td>
<td>Recover refrigerant</td>
<td>Vapour in hoses/ manifold at system pressure</td>
</tr>
<tr>
<td></td>
<td>Evacuate system</td>
<td>Vapour in system volume at 1 atm.</td>
</tr>
<tr>
<td></td>
<td>Purge system</td>
<td>2 × system volume at 1 atm.</td>
</tr>
<tr>
<td></td>
<td>Charging</td>
<td>Liquid in hoses/ manifold at system pressure</td>
</tr>
</tbody>
</table>

**Venting**

For various reasons, refrigerant may be vented from a system instead of being recovered. In the model, it is assumed that venting results in the entire refrigerant charge being emitted, and no residual refrigerant in the system, i.e., for a single system \( M'_{vent} = M_{r, des} \) (equation 52).
where \( P_{\text{vent}} \) is the probability that the refrigerant is vented per visit, rather than recovered (see Table 12). \( P_{\text{vent}} \) is interpreted as the percentage of charge that is not recovered over a given population of equipment, and data for typical applications is given in Table 13. Based on this data, the average probability of venting is approximately 0.2.

**Recovery**

Following removal of refrigerant using a recovery machine, interconnecting hoses will still contain vapour that is released once disconnected. Estimating the release employs the same calculation as for charging (equation 55), but the density is of vapour at the recovery pressure, and the probability of not venting \((1 - P_{\text{vent}})\) is included (equation 53).

\[
M'_{\text{recov}} = (1 - P_{\text{vent}}) \cdot M_{\text{sys}} + M'_{\text{charg}} - M'_{\text{evac}}
\]  

(53)

**Evacuation**

The mass of refrigerant released at evacuation stage is the sum of the mass of vapour within the system parts and the mass of refrigerant absorbed in oil (equation 54).

\[
M'_{\text{evac}} = \rho_r \cdot V_{\text{sys}} + M_{\text{oil}} \cdot S_{\text{oil}}
\]

(54)

where both refrigerant vapour density \( \rho_r \) and refrigerant solubility in oil \( S_{\text{oil}} \) are evaluated at room temperature and the pressure at which recovery was terminated (see Table 12). The mass of oil is approximated from equation (12) in Section 2.2.1.

**Table 13: Summary of reported venting probabilities**

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Domestic refrig.</td>
<td>Johnson, 1998</td>
<td>50-75%</td>
<td>20-50%</td>
<td>3-65%</td>
<td></td>
</tr>
<tr>
<td>Retail refrig.</td>
<td>March, 1999</td>
<td>21-43%</td>
<td>20-50%</td>
<td>10-50%</td>
<td>3-6%</td>
</tr>
<tr>
<td>- Integrals</td>
<td>Clodic, 2003</td>
<td>21-43%</td>
<td>20-50%</td>
<td>10-50%</td>
<td>3-6%</td>
</tr>
<tr>
<td>- Split/distrib</td>
<td>Haydock, 2003</td>
<td>5-10%</td>
<td>5%</td>
<td>3-6%</td>
<td></td>
</tr>
<tr>
<td>- Supermarket</td>
<td>Clodic, 1998</td>
<td>5-10%</td>
<td>4-5%</td>
<td>3-6%</td>
<td></td>
</tr>
<tr>
<td>Air conditioning</td>
<td>Johnson, 1998</td>
<td>15-22%</td>
<td>5-10%</td>
<td>4-8%</td>
<td></td>
</tr>
<tr>
<td>- Split/distrib</td>
<td>March, 1999</td>
<td>15-22%</td>
<td>5-10%</td>
<td>4-8%</td>
<td></td>
</tr>
<tr>
<td>- Chillers</td>
<td>Clodic, 2003</td>
<td>3-5%</td>
<td>4-5%</td>
<td>0-100%</td>
<td></td>
</tr>
</tbody>
</table>

72
Refrigerant charging

Refrigerant losses can occur during charging from a cylinder or a production line. The mass of refrigerant released due to disconnection of hoses following charge is equal to that held within the apparatus (equation 55).

\[ M_{\text{chrg}} = \rho_r \cdot V_{\text{hose}} \]  

(55)

where \( V_{\text{hose}} \) is the internal volume of the refrigerant hoses and manifold, typically 0.0007 m³. Refrigerant density \( \rho_r \), assumed to be at room temperature, might be in liquid or vapour phase depending upon charging method (see Table 12).

Energy use

During service and maintenance, CO₂ emissions originate from transportation, and for the construction and fitting of replacement components, which are calculated as SOL. Although energy is consumed by items such as recovery machines, these sources have been ignored presently. In travelling to and from an installation, the mode of transport will produce emissions because of fuel combustion. Therefore, more service calls to a site means the total distance travelled is further and therefore higher emissions, as equation (56).

\[ M_{\text{trav}} = \zeta \cdot L_{\text{trav}} \cdot N_{\text{serv}} \]  

(56)

where \( L_{\text{trav}} \) is the distance travelled per visit (in km) and \( N_{\text{serv}} \) is the number of service visits (i.e., journeys travelled). \( \zeta \) is the CO₂ emission rate for the vehicle used, which is around 0.3 kgCO₂·km⁻¹ for typical petrol cars (AutoLPG, 1998).

2.4 End of life stage

At the end of its life, the equipment is dismantled, first by the refrigerant being recovered or vented, and then the machinery is disassembled. Subsequently, refrigerant is sent for recycling or incineration, as is the oil, whilst the machinery parts (metals) are sent for disposal or recycling. Refrigerant and CO₂ emissions from these activities are as equation (57) and (58) respectively.

\[ M_{\text{sOL}} = M'_{\text{dmlt}} + M'_{\text{disp}} \]  

(57)

\[ M_{\text{sOL}} = M^*_{\text{dmlt}} + M^*_{\text{disp}} \]  

(58)

A generalised process for end of life stage is shown in Figure 13.
2.4.1 Dismantling

The mass of refrigerant vented at end of life is determined in the same way as in Section 2.3.3 (equation 59). Where recovered refrigerant is sent for recycling, the emissions associated with the production of the same mass of refrigerant are deducted from the start of life stage (Section 2.2.2). In addition, the emitted amount calculated for evacuation (Section 2.3.3) was assumed equal to the amount released from the system (and oil) during dismantling.

\[ M_{\text{dml}} = M_{\text{vent}} + M_{\text{recon}} + M_{\text{evac}} \]  \hspace{1cm} (59)

The use of brazing and similar activities used to dismantle the equipment was assumed equal to those calculated from equation (16) for the construction at start of life stage, and again the contribution for transportation is included (equation 60).

\[ M_{\text{dml}} = M_{\text{brz}} + M_{\text{nav}} \]  \hspace{1cm} (60)
2.4.2 Disposal of refrigerant and equipment

Releases occurring as a result of disposal are mainly attributed to incineration of materials, such as oil and refrigerant, which includes both refrigerant and CO₂ emissions (equation 61 and 62).

\[ M'_{\text{disp}} = \sum_i M'_{\text{incin},i} \]  

\[ M^*_{\text{disp}} = \sum_i M^*_{\text{incin},i} \]  

(61)  

(62)

Once refrigerant has been reclaimed, it is tested to determine whether it is of an adequate quality for recycling or whether it should be sent for destruction, which is typically done in a toxic waste incinerator. Normally a small proportion of refrigerant is released un-incinerated and a small proportion of GHG combustion product is released (UNEP, 2002). The energy input is very low due to exothermic burning for most refrigerants. The mass of refrigerant that escapes incineration is simply based on the plant efficiency (equation 63).

\[ M'_{\text{incin}} = (1 - \eta_{\text{incin}}) \cdot M_{r,\text{incin}} \]  

\[ M^*_{\text{incin}} = \eta_{\text{incin}} \cdot M_{r,\text{incin}} \left( \frac{M_{\text{CO₂}}}{M_r} + 10 \cdot \xi \right) \]  

(63)  

(64)

where \( M_{r,\text{incin}} \) is the mass of product sent for incineration, and \( \eta_{\text{incin}} \) is the incineration efficiency of the plant. A typical value for \( \eta_{\text{incin}} = 0.997 \) (UNEP, 2002). The mass of CO₂ emitted from incineration is from equation (64).

Compressor oil is either put into landfill or burned. In the case of burning the oil, the same equation (64) for refrigerant incineration was used, where the molecular mass of oil \( (M_{\text{oil}}) \) is used instead of molecular mass of refrigerant. Typically, \( M_{\text{oil}} \approx 200 \text{ kg kmol}^{-1} \). If the oil goes to landfill, the emissions are assumed zero. Other materials (metals) are either disposed of in landfill or recycled. As with the case of recycled refrigerant, the emissions associated with material production at start of life stage, are altered to represent the lowest emissions factors in Table 4 (Section 2.2.1).
SECTION 3: CLIMATE IMPACTS

3.1 Options for evaluating climate change impact

The introduction and literature review in Section 1 identified that the established method of evaluating the impact of GHG emissions is with the use of GWP; however, the literature revealed that many alternative approaches exist for quantifying the global warming impact. Due to the formulation of GWP, its application in terms of establishing the total effect of emissions of CO₂ and the gases commonly used as refrigerants, introduces many uncertainties. In particular, this includes the difficulty of interpreting the physical consequences of GHG emissions, and further, the choice of a meaningful integration time for which the consequences of CO₂ with an infinite lifetime, and short-lives refrigerants, can be comparable. Due to these issues (discussed in Section 1.2.3), an alternative measure of quantifying the impact of emissions was sought. By adopting an alternative to GWP, results from emissions modelling can be used to compare the significance of the various emission sources against the conventional approach.

![Diagram of climate relevant measures]

Figure 14: Chain of climate relevant measures

An important consideration for the selection of any measure of GHG impact is relevance to the consequence of the emission. Whilst the quantity of emissions are relatively easily measured (or predicted), the chain of events that follows become increasingly sensitive to a very broad range of factors and so are less accurately estimated. In contrast, the emission of GHG, the consequential
events following the release of a GHG have increasing significance on societies. This concept is illustrated in Figure 14, which shows events such as changes in atmospheric concentration to be easily quantifiable, but the ensuing changes to global climate and ultimately how this forces society and economies to be affected, are increasingly speculative.

In this context, the concept of GWP lies around "radiative forcing" since it is a function of atmospheric concentration and chemical properties. In order to shift the result of a TEWI exercise towards greater relevance to society, it is appropriate to employ a measure associated with some form of "climate behaviour" characteristic. Further benefit would be achieved were it possible to overcome the essentially subjective approach of weighting long- and short-lived GHGs through the choice of integration time (as used in GWP calculations). Section 1.2.3 identified a newly developed index (AGTP), which characterises the consequence of an emission in terms of a comprehensible global temperature rise; further, such a measure also permits a damage index to be adopted. On this basis, three new concepts were developed as alternatives to the TEWI/GWP approach; one based on temperature rise, and two representing the subsequent societal impact.

3.2 Global warming potential

For purposes of completion, the calculation of GWP is presented here. The GWP for any species over a given time horizon is the absolute global warming potential (AGWP) of that species relative to the AGWP for CO₂ (equation 65).

\[
GWP(t) = \frac{AGWP_x(t)}{AGWP_{CO₂}(t)}
\]  \hspace{1cm} \text{ (65)}

where AGWP is the absolute GWP at time \( t \). AGWP is obtained from equation (66) for the species, \( x \), and equation (67) specifically for carbon dioxide (Shine et al, 2004).

\[
AGWP_x(t) = F_x \cdot \tau_x \cdot \left[ 1 - \exp \left( -\frac{t}{\tau_x} \right) \right] \]  \hspace{1cm} \text{ (66)}

\[
AGWP_{CO₂}(t) = F_{CO₂} \cdot \left\{ a_0 \cdot t + \sum_{i} a_i \cdot \tau_i \cdot \left[ 1 - \exp \left( -\frac{t}{\tau_i} \right) \right] \right\}
\]  \hspace{1cm} \text{ (67)}

where \( F \) is the radiative forcing and \( \tau \) is the atmospheric lifetime of the species, \( x \); values for common refrigerants are contained in Table 1. Values for \( a_i \) and \( \tau_i \) from equation (67) are from Shine et al, as follows: \( a_0 = 0.1756, a_1 = 0.1375, a_2 = 0.1858, a_3 = 0.2423, a_4 = 0.2589, \tau_1 = 421.093 \text{ y}, \tau_2 = 70.595 \text{ y}, \tau_3 = 21.4216 \text{ y}, \tau_4 = 3.4154 \text{ y}. \)
The "equivalent" global warming impact of an emission of a non-CO₂ gas, relative to CO₂ is calculated from the GWP of the gas (equation 68).

\[ M'' = M'_x \cdot GWP(t)_x \]  
(68)

where GWP is calculated for a given \( t \).

### 3.3 Global temperature change potential

Shine et al derived equations for pulse and sustained emissions of CO₂ and non-CO₂ GHGs. The derivation was based on the additional amount of heat retained at the earth’s surface (over time) due only to the insulating effect of the emitted gas under consideration, and the duration that this excess heat is absorbed into the oceans.\(^9\) Comparison with an energy balance climate change model (EBM) showed that the equation for pulse emissions produced a high error, whilst the equation for sustained emissions was within ±5% for most gases. Therefore, the equations for sustained emissions were used for equations (69) and (70), when \( \tau \neq T \).

\[ AGTP_a(t) = \frac{\tau_a \cdot F_a}{C'} \left\{ T \left[ 1 - \exp\left( -\frac{t}{T} \right) \right] - \frac{1}{(T^{-1} - \tau_a^{-1})} \left[ \exp\left( -\frac{t}{\tau_a} \right) - \exp\left( -\frac{t}{T} \right) \right] \right\} \]  
(69)

\[ AGTP_{CO₂}(t) = \frac{F_{CO₂}}{C'} \left\{ a_0 \cdot T \cdot t - a_0 \cdot T^2 \cdot \left[ 1 - \exp\left( -\frac{t}{T} \right) \right] + \sum \tau_i \cdot a_i \cdot \left[ T \left( 1 - \exp\left( -\frac{t}{T} \right) \right) - \frac{1}{T^{-1} - \tau_i} \left[ \exp\left( -\frac{t}{\tau_i} \right) - \exp\left( -\frac{t}{T} \right) \right] \right] \right\} \]  
(70)

where \( C' \) is the global surface heat capacity, which is interpreted as the specific heat of a 100 m layer of ocean, \( 4.2 \times 10^8 \) J K\(^{-1}\) m\(^2\). \( T \) is a time constant for the climate system and is the product of \( \sigma \times C' \), where \( \sigma \) is a climate sensitivity parameter that relates to the temperature when doubling CO₂ concentrations. \( \sigma \) ranges from 0.4 to 1.2 K m\(^2\) W\(^{-1}\) (typically 0.8, but the sustained emission model is relatively insensitive to \( \sigma \)). \( F \) and \( \tau \) can be found in Table 1, and the values of \( a \) and \( \tau \) (used to calculate the atmospheric "decay" time of CO₂ by means of absorption into carbon sinks) are the same as those from GWP listed in Section 3.2.

Since equations (69) and (70) are based on sustained releases, the actual contribution of a single release is from equation (71).

\(^9\) Obviously, for a few kilogrammes of refrigerant this equates to a very small temperature rise.
\[ AGTP(t)' = AGTP(t) - \sum_{r=1}^{k} AGTP(t_r)' \] (71)

where \( k \) is the number of prior years since the time of the emission.

Following the convention for GWP, the relative global temperature potential (GTP) of a non-CO\(_2\) GHG is found from equation (72).

\[ GTP(t) = \frac{AGTP_{\text{CO}_2}(t)}{AGTP_{\text{CO}_2}(t)} \] (72)

### 3.4 Total lifetime temperature impact (TLTI)

By applying the absolute GTP associated with the emission of 1 kg of GHG, to the calculated quantity of gas emitted for each year that the equipment is responsible for a release, the corresponding temperature rise can be calculated. This yields a novel concept, named Total Lifetime Temperature Impact, offered as an alternative approach to the conventionally used TEWI. However, the formulation of TLTI is somewhat different from that of TEWI in one respect. Whereas TEWI is simply the summation of the product of emissions and GWP, TLTI is a function of the maximum temperature rise experienced as a result of the total emissions. Thus, TLTI is associated with a single point in time subsequent to the emissions, wherever the temperature rise peaks, as equation (73).

\[ TLTI = \max \left[ \sum_i M_i(t) \cdot AGTP(t), + \sum_j M_j \cdot AGTP(t) \right] \quad (t = 0 \to \infty) \] (73)

where \( M \) is the mass of substance emitted: for the refrigerant, \( M_i \) is the amount of each species released, and for the contribution from CO\(_2\), \( M_j \) is the total mass of produced typically due to energy usage of the system.

An example of a TLTI calculation is shown in Figure 15. Total AGTP resulting from emissions of a high global warming refrigerant and CO\(_2\) associated with the systems energy usage (in existence for say 10 years), is plotted over time. The global warming impact of the refrigerant and the additional contribution from CO\(_2\) is represented by the upper line. TLTI corresponds to the maximum total AGTP, as indicated on the Figure. By adopting this approach, the impact associated with a given refrigerating system can be perceived in comprehensible units, and the reliance on choice of integration time is also avoided since the time dimension is implicit within
the TLTI value. It is important to note that TLTI is not "equivalent" to TEWI, in that they do not simply adopt different units. In particular, TLTI brings out the rapid but brief warming effect of refrigerants that have a high radiative forcing but for a limited duration, whilst relying less on the minor warming effect that CO2 may have in hundreds of years in the future.

Figure 15: Illustration of TLTI, GDI1 and GDI2 interpretation

3.5 Global damage index (GDI)
Further to the AGTP concept, characterising the rise in temperature makes it possible to address the global warming impact of emissions at a more relevant point further along the range of climate impact measures (as Figure 14). Since the consequences of global warming are closely related to global temperature change, it is feasible to use AGTP or TLTI to evaluate societal and economic damage. Within the literature review a number of definitions for the concept termed global damage index (GDI) or potential, were identified from Smith (2003), Fuglestvedt et al (2003) and others. "Damage" typically refers to the economic burden on society in order for the population to maintain its quality of life, at some point in the future. As explained in Section 3.1, such a concept is significantly more theoretical, producing results of less certainty, than preceding measures.

For the first GDI, the concept of Fuglestvedt et al (2003) was adopted, and the damage index is calculated from the total temperature rise resulting from the emissions over a chosen integration period, referred to here as $GDI_1$. $GDI_1$ is calculated by integrating the total AGTP to some exponent to the year that is of interest (equation 74), and Fuglestvedt et al suggested the exponent be between $n = 1$ and $n = 2$. 
In this form, \( GDI_1 \) represents a reduced total temperature rise contribution of the emissions within the chosen period, and is shown in Figure 15 (where \( n = 1.5 \)) where \( GDI_1 \) is indicated by the hatched area. On this basis, \( GDI_1 \) is comparable to GWP (and therefore TEWI). The second GDI was established using a different approach where the degree of damage is quantified according to the rate at which global temperature rise occurs, as proposed by Shine et al (2004), Fuglestvedt et al (2003) and others. By interpreting this simply as the average gradient associated with temperature rise over time, an alternative \( GDI_2 \) may be quantified according to the time taken to reach TLTI (equation 75).

\[
GDI_2 = \frac{TLTI}{t(TLTI) - t(0)}
\]  

(75)

where TLTI is from equation (73), \( t(TLTI) \) is the time (year) that TLTI (or maximum AGTP) occurs, and \( t(0) \) is the time (year) of start of life of the equipment.

As with TLTI, \( GDI_2 \) is not “equivalent” to TEWI, unlike \( GDI_1 \) which is proportional to TEWI provided \( GDI_1 \) and GWP have been evaluated over the same integration time. A further similarity between TLTI and \( GDI_2 \), is that it represents the more immediate warming contribution of high radiative forcing and short lived refrigerants. Whilst both GDI concepts represent a more significant interpretation of the consequences of the emissions associated with refrigerating equipment, they are considerably less reliable in terms of representing the likely impact due to the accumulated uncertainties associated with preceding evaluations upon which GDI depends. In addition, an important consideration from the literature review was that the consequence of “individual” emissions is largely dependent upon the conditions and concentrations of gases at that time. In the case of greater preceding emissions, latter emissions of a given quantity will result in more severe additional consequences. In other words, the impact of increasing emissions is believed to cause exponentially worse climatic effects and associated societal penalties. Thus, evaluation of GDI (and TLTI) on an individual basis may under-represent the sum of all emissions.
SECTION 4: RESULTS AND DISCUSSION

4.1 Implementation of model
The model described in Section 2 was used to predict refrigerant and CO₂ emissions resulting from the production, use and disposal of a refrigerating system, and the various consequences measures detailed in Section 3 were used to understand the overall implication of the emissions. To address the objectives of this study (Section 1.3), the model was used to examine the following:

- breakdown and uncertainty of emission sources,
- sensitivity of transient influences,
- effects of system design characteristics on emissions,
- influence of material and refrigerant handling activities, and
- interpretation of consequence of emissions.

Consideration of the first four items should provide a better understanding of the influence of a given parameter within a TEWI calculation. The second, third and fourth items are essentially an in-depth investigation of the uncertainties identified in the first case. The last item will help to provide a broader overview of the consequences of emissions. In order to evaluate different parameters, a reference system has been established and the selected input values (based on mean values within the ranges identified unless stated otherwise) are provided in Table 14, which represents a commercial refrigeration cabinet. Basic outputs for the system performance are also included. The assessment of different parameters addresses variations of these reference inputs.

4.2 Breakdown and uncertainty of emissions
The model was used to generate data for the reference application detailed in Table 14, and for each stage emissions were broken down into those from energy use and from refrigerant release (converted into CO₂ equivalent using GWP(100)).

4.2.1 Breakdown of emission sources
Figure 16 shows a full breakdown of emissions from each element for SOL, In-use and EOL stages. (Data for in-use stage are presented for a one-month period only so that the data for other stages is more visible.) The model was run for three different input settings, corresponding to minimum, mean and maximum values of emissions, and these are indicated on the graph. These settings included parameters for ageing effects, refrigerant handling behaviour, leak tightness and recycling, all of which represent the input ranges for each sub-model as detailed in Section 2.
### Table 14: Reference input parameters and basic outputs

<table>
<thead>
<tr>
<th>Environment</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>External temperature</td>
<td>Ambient 32°C</td>
<td>Ambient temperature steps</td>
<td>2.3 K</td>
</tr>
<tr>
<td>Design ambient</td>
<td></td>
<td>Day/night ambient temp difference</td>
<td></td>
</tr>
</tbody>
</table>

| Application                                      |                  |                  |                  |
| Design refrigerant                               | R22              | Application temperature | -20°C           |
| Refrigerant in use                              | R22              | Thermostat differential setting | 3 K            |
| Application basic rating                        | 250 W K\(^{-1}\) | Application thermal mass | 850 kJ K\(^{-1}\) |
| Application constant rating                     | 0.2              | Operational life     | 8 years          |
| Application heat load source                    | Room             | Fan power constant   | 7 kJ m\(^{-3}\)  |

| System components                                |                  |                  |                  |
| Evaporator diameter                              | 0.009 m          | Condenser diameter | 0.009 m          |
| Total evaporator tube length                     | 60 m             | Total condenser tube length | 80 m           |
| Number evaporator passes                         | 6                | Number condenser passes | 4              |
| Evaporator HTF flow rate                         | 2.5 m\(^{3}\) s\(^{-1}\) | Condenser HTF flow rate | 4.5 m\(^{3}\) s\(^{-1}\) |
| Evaporator int/ext area ratio                    | 50 m\(^{2}\) m\(^{-2}\) | Condenser int/ext area ratio | 60 m\(^{2}\) m\(^{-2}\) |
| Suction pipe diameter                            | 0.02 m           | Discharge pipe diameter | 0.015 m        |
| Suction pipe length                              | 4 m              | Discharge pipe length | 1 m            |

**Refrigerant handling servicing**

|                                            |                  |                  |                  |
| Number service visits per year              | 1                | Recovery fraction | 0.3              |
| Number of top-ups per year                 | 1                | Recovery pressure | 65 kPa           |
| Pressure/leak test gas                     | R22              | Probability of venting when handling | 0.2            |
| Charging phase                              | Liquid           | Frequency of handling per visit | 0.3            |
| Distance travelled per visit               | 75 km            | Material recycling | None            |
| Condenser cleaning                          | No               | Oil and filter drier change | No              |

**Model selection**

|                                            |                  |                  |                  |
| Evap/cond charge sensitivity index (n)      | 1.0              | Annual leak rate increase | 0 % y\(^{-1}\) |
| Include evap/cond fouling                   | Yes              | Include compressor wear | Yes             |

**Basic outputs (design conditions)**

|                                            |                  |                  |                  |
| Refrigerant charge                         | 2.49 kg          | Refrigerating capacity | 14.6 kW         |
| Annual leak rate (first year)\(^{1}\)     | 8.1 %            | Compressor power     | 10.1 kW         |
| Compressor swept volume                    | 58.5 m\(^{3}\) h\(^{-1}\) | Coefficient of performance | 1.47        |
| Evaporating temperature                    | 4.5°C            | Condensing temperature | 48.3°C         |

\(^1\) Ambient for London, UK (BS 5925, 1991). \(^{1}\) Reference values in Table 16.
It was found that the dominant sources of emissions were primarily from operating energy consumption during in-use stage, and then refrigerant leakage from the system and from refrigerant handling during servicing. Energy consumption during SOL, refrigerant releases during servicing and EOL were also significant. SOL refrigerant leakage and energy consumption during servicing and EOL appeared small. Therefore, if a simplified approach was to be used, then SOL refrigerant leakage and energy consumption during servicing and EOL should be omitted.

![Figure 16: Breakdown of emissions from energy and refrigerant for different stages (based on GWP(100). In-use values for a single year only)](image)

The breakdown of emission sources was extended over a range of equipment sizes, and grouped (according to life cycle stages) results are presented in Figure 17, and the values for the In-use stage are for one year only. For the different sized equipment, model inputs were as Table 14, although the application basic rating and thermal mass were changed proportionally, and system component sizes were changed in similar proportions to maintain the same cycle COP as the
reference case. The data shows some variation in proportion to their emissions, particularly as the
capacity of equipment reduces. For example, with the larger capacity system, emissions from
operating energy consumption account for 85% of the total, whereas for the smaller capacity
system, it is about 55% of the total. The two main sources that produce a greater proportion of
emissions for the smaller system are system leakage and refrigerant emissions from servicing.
Thus, it is pertinent to pay more attention to modelling refrigerant emissions for smaller
equipment (particularly to emissions during refrigerant handling), whereas more focus should lie
on modelling energy consumption in larger equipment.

![Figure 17: Variation of contributions from different emission sources with size of equipment](image)

4.2.2 Uncertainties of emissions

The range of proportions of each contribution is shown in Table 15, relative to the total emissions
for the minimum, mean and maximum values, also showing the extremities of the uncertainty.

<table>
<thead>
<tr>
<th>Element</th>
<th>Minimum (relative to max total)</th>
<th>Mean (relative to mean total)</th>
<th>Maximum (relative to min total)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOL. refrigerant</td>
<td>0.0%</td>
<td>0.5%</td>
<td>1.3%</td>
</tr>
<tr>
<td>SOL. energy</td>
<td>0.3%</td>
<td>2.3%</td>
<td>8.6%</td>
</tr>
<tr>
<td>In-use, system leak</td>
<td>2.5%</td>
<td>22.4%</td>
<td>58.1%</td>
</tr>
<tr>
<td>In-use, system energy</td>
<td>35.1%</td>
<td>70.0%</td>
<td>94.7%</td>
</tr>
<tr>
<td>In-use, service refrigerant</td>
<td>0.7%</td>
<td>3.4%</td>
<td>15.5%</td>
</tr>
<tr>
<td>In-use, service energy</td>
<td>0.0%</td>
<td>0.1%</td>
<td>0.1%</td>
</tr>
<tr>
<td>EOL. refrigerant</td>
<td>0.3%</td>
<td>1.2%</td>
<td>6.8%</td>
</tr>
<tr>
<td>EOL. energy</td>
<td>0.0%</td>
<td>0.1%</td>
<td>0.1%</td>
</tr>
<tr>
<td>Total</td>
<td>-</td>
<td>100.0%</td>
<td>-</td>
</tr>
</tbody>
</table>
In the example, the contribution of SOL varies between negligible (<0.5%) to about 10% when the SOL emissions are high and the total of in-use emissions are relatively low. EOL emissions cover a similar range. In-use emissions from energy consumption can vary significantly, from 35% to almost 95% of the total, and although always lower, refrigerant leakage can be from 2% to over 50%. Including releases during servicing, total contribution of refrigerant could account for over 50% of lifetime emissions. Overall, the total variation in TEWI for the given system and environmental conditions ranges from -30% to +140% about the result based on the mean input value for each selected parameter (not the average of the results). With a lower GWP refrigerant (such as HC) this was found to be smaller, within the range -10% to +80%. This strongly identifies the importance of accurate modelling of TEWI and detailed accounting of parameters that influence in-use system leakage, energy consumption and refrigerant emissions from handling. Similarly, the results of any TEWI evaluation should only be presented with the expected uncertainty associated with the detail of input parameters.

4.3 Sensitivity to transient influences
The impact of various parameters was assessed to help understand their influence, and suitability of inclusion within the entire model. Three key time-dependent influences were considered:

- External conditions (ambient temperature variations)
- Ageing effects (fouling, compressor wear and increasing leak rate)
- Power station emissions

Each of these was addressed individually, and then in combination.
4.3.1 External conditions

Figure 18 shows the differences between monthly energy consumption of the compressor for different assumptions: monthly and day/night variations, average monthly variations and a constant ambient. The results show a significant difference between each of these three scenarios, where the total annual emissions from energy consumption with a constant ambient (assumed 27°C) is greatest (2340 kgCO₂), and by varying monthly ambient temperature (1190 kgCO₂) and including changes for day/night temperatures (950 kgCO₂) produces a notable reduction in emissions. As an alternative, the model was tested with an average annual ambient temperature of 10.4°C, which produced an annual energy consumption of 800 kgCO₂. This value is significantly lower than the varied monthly/day/night scenario, confirming that using a constant ambient approach will always be erroneous, and that modelling with incremental temperatures throughout the year is valuable. When the heat load and heat rejection temperatures have a stronger dependence on ambient variations, the errors of using an assumed ambient will be even greater.

![Figure 18: Influence if various ageing effects on emissions from system energy consumption](image)

Figure 19: Influence if various ageing effects on emissions from system energy consumption

4.3.2 Ageing

The influence of heat exchanger fouling, compressor wear and refrigerant leakage on annual emissions from energy consumption is shown in Figure 19. Individually, compressor wear has the smallest impact, but rises almost exponentially with time. This occurs because of the feedback effect of the resulting lower system capacity causing longer running times and therefore more wear. The effect of fouling is seen to have a significant influence, although the degree of fouling is also largely dependent upon the HTF (i.e., water, air, etc.) and local contamination of the HTF (e.g., hard or soft water, salty air, etc.). Typical rate of increasing leakage rate also has a notable effect, but this will be largely dependent upon the sensitivity of the system to leakage (as
discussed later). Both fouling and increasing leakage are seen to produce an increase of annual emissions at a slower rate.

In combination, the influence of compressor wear, fouling and increasing leakage was found to produce emissions about 10% higher in the final year of the equipment life compared to the first year. However, it is also observed that the total increase is less than the sum of the individual contributions, when it would be expected that the contributions would at least be cumulative. This is explained when considering the performance of the system itself. The application temperatures are within the required range when the individual transient influences are accounted for. When all three influences are included the capacity of the system degrades to a point where the required application temperature cannot be achieved, and with a smaller temperature lift, the impact of greater energy consumption is dampened.

![Graph showing influence of change in power station emissions for electricity production](image)

**Figure 20: Influence of change in power station emissions for electricity production**

### 4.3.3 Power station emissions

The effects of changes in CO$_2$ emissions from energy production are shown in Figure 20. With the variable emission factor neglected, the system energy consumption produced constant emissions for each year. When applying anticipated changes in emissions factor, the annual emissions reduce and would be expected from the data in Section 2.3.2. Accounting for annual changes in emissions factor ($\xi$) only (using only the first term in equation 23), it is seen that there is a constant reduction in emissions over the equipment life, where the final emissions are about 6% lower than the first year, and the total lifetime emissions are reduced by about 4%. If monthly variations in emissions factor are introduced (include the second term in equation 23),
the same trend is observed, but the emissions for the starting years are higher by about 3%. This is due to the higher emissions factor occurring at the same as the higher power demand of the system, i.e., during summer months. Including additional day/night variations in emissions factor (full equation 23), the starting emissions are greater still at (5%), because daytime emissions are greater which again correspond to higher daytime power demand. In this example, the difference between detailed accounting for time-dependent emissions factor, and constant emissions factor on total lifetime emissions is negligible. However, this may not be the situation if different load profiles exist, and particularly in the case of heat pump applications.

4.4 Influences of equipment design

It is important to understand how the design of equipment influences GHG emissions, particularly since certain design parameters may lead to lower emissions from one source, but cause more emissions elsewhere. Where trends in emission reduction through certain design parameters are found, these can be used as a basis for improved environmental design of new equipment. The size and type of several components were changed:

- Circuit piping and connections
- Heat exchanger sizes
- Compressor size
- Refrigerant

This will provide additional guidance on how to minimise emissions through design options.

4.4.1 Pipework and connections

The choice of components and connection types used in circuit piping was found to have a notable change on both refrigerant emissions and emissions from energy use, as seen in Table 16 for the five different cases, including the reference systems as Case 1. In Case 2, flare connections are changed to brazed connections, Case 3 is with the service valve removed and swaps the TEV for a capillary tube, and Case 4 has no solenoid valve, or controls (e.g., pressure switch and connecting line). Case 0 represents the worse case, where all pipe connections are flare joints.

Compared to the reference Case 1, equivalent emissions for Case 2 from refrigerant were reduced by 16%, but energy consumption dropped by only 1%, although the mean application temperature is about 0.5 K lower. With Case 3, an additional reduction of about 2% is observed, although it should be noted that an absence of service valve could result in an increase in refrigerant emissions because of additional handling when cutting into the system. Further
emissions from energy consumption are negligible, although there is a slight improvement of application temperature. Case 4 shows an additional 4% reduction in refrigerant emissions, and since it is the situation with minimal components and virtually no mechanical connections, it is considered the best overall reduction in emissions. Again, there are no more emission reductions from energy consumption, but significant benefit is seen in lower application temperature. Where leakage is about twice the rate as the reference case, emissions for Case 0 are 31% for the refrigerant and 3% higher for energy consumption. Consistent with previous trends, application temperature is almost 1K higher.

Table 16: Effect of refrigerant leakage from different component assemblies

<table>
<thead>
<tr>
<th>Component</th>
<th>Case 0</th>
<th>Case 1 (Reference)</th>
<th>Case 2</th>
<th>Case 3</th>
<th>Case 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coil</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Pipework/brazed joint</td>
<td>0</td>
<td>12</td>
<td>18</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Service valves</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Solenoid valves</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>Expansion valves</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Control lines</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>0</td>
</tr>
<tr>
<td>Control equipment</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>0</td>
</tr>
<tr>
<td>Filter</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Compressors</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Flare joint</td>
<td>18</td>
<td>6</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Annual leakage rate</td>
<td>17.9%</td>
<td>8.1%</td>
<td>3.2%</td>
<td>2.8%</td>
<td>1.6%</td>
</tr>
<tr>
<td>Refrigerant contribution (kgCO₂)</td>
<td>13165</td>
<td>10067</td>
<td>8486</td>
<td>8373</td>
<td>7978</td>
</tr>
<tr>
<td>Energy contribution (kgCO₂)</td>
<td>252446</td>
<td>250468</td>
<td>247759</td>
<td>247652</td>
<td>247228</td>
</tr>
<tr>
<td>Total (kgCO₂)</td>
<td>265612</td>
<td>260536</td>
<td>256245</td>
<td>256025</td>
<td>255206</td>
</tr>
<tr>
<td>Mean application temperature (°C)</td>
<td>-20.49</td>
<td>-21.16</td>
<td>-21.46</td>
<td>-21.51</td>
<td>-21.68</td>
</tr>
</tbody>
</table>

4.4.2 Heat exchanger size

Figure 21 shows the refrigerant and energy use contribution to GHG emissions when the exchangers are between 50% and 200% of their original “size”. The size of both condenser and evaporator was changed in two ways: (a) vary tube length, whilst maintaining the same number of circuits and the same HTF flow, and (b) vary tube length and HTF flow proportionally.
With a smaller heat exchanger the corresponding refrigerant charge is smaller and therefore the amount of refrigerant emitted is less (given the same leak rate), so the refrigerant contribution is less. Where the HTF flow rate is reduced with exchange size, the emissions are less because the benefit in lower fan energy consumption outweighs the higher heat transfer coefficient associated with maintaining the higher flow rate. However, where the HTF flow rate increases with exchanger size, the additional power consumption offsets the improvement in heat transfer and total emissions increase. Where HTF flow remains constant with larger exchanger size, the rate of improvement in emissions becomes less, and it can be seen that a further enlargement of the exchanger would begin to increase emissions due to the refrigerant emissions. Obviously, a higher leak rate would produce this reversal in the trend sooner. These observations imply that the size of the condenser and evaporator can be “optimised” when considering the combination of increase of refrigerant emission and decrease in emissions from energy consumption.
Additional emissions due to manufacture of larger components is dwarfed by the savings from lower compressor energy requirement. Changing the size of the exchanger equates to about 20% of the production emissions, but only accounts for less than 1% of the total emissions.

Another aspect associated with exchanger design that influences emissions is the sensitivity of the system to sub-optimal charges. Condensers and evaporators can be designed so that the system performance becomes less sensitive to a reduction in charge size due to leakage. Figure 22 presents results for annual emissions from a system with three different leak rates corresponding to three of the cases detailed in Table 16, where different sensitivity indexes were used.

![Figure 22: Effect of sensitivity to charge size on system performance](image)

For $n = 1/3$, the trend is as would be expected: emissions from energy consumption reduce as leak rate reduces because the system is closer to its optimum charge. However, where $n = 1/17$ and $n = 1/30$, the emissions are actually lower for the high leak rate (Case 0). With reference to the earlier discussion on ageing effects, the same phenomenon is occurring here, in that these conditions lead to significantly reduced capacity, so power consumption is less. This is explained further with Figure 23, where the mean monthly application temperature and corresponding equivalent emissions is plotted for two different leakage conditions (Cases 0 and 4). Where $n = 1/3$, the application temperature is within the required range, and emissions from energy consumption vary with the monthly ambient. However, where the system sensitivity to charge size increases ($n = 1/30$), the emissions reduce as in Figure 22, but also the mean application temperature rises away from its set point, and thus fails to serve its purpose. This effect is more pronounced for the higher leak rate (Case 0). These observations demonstrate the importance of good heat exchanger sizing/design.
4.4.3 Compressor size
Compressor swept volume was varied by setting the design ambient to a different temperature, which gives a larger design heat load and higher condensing temperature. Results are shown in Figure 24 where the design ambient is indicated for the corresponding size of the compressor. Changes in refrigerant emissions increase slightly because of releases of residual refrigerant within the compressor oil, and similarly, material production emissions increase a little, but all are relatively small.

The main reasons for the differences in emissions seen in Figure 24 are due to changes in energy consumption. It is seen that using a compressor with smaller swept volume produces lower emissions. This is because the system balance temperatures are closer to the HTF temperatures, so the compression ratio is less and therefore compressor COP is higher. In addition, a lower refrigerant flow rate reduces pressure drop around the system, again leading to smaller compression ratio, and the lower number of compressor starts also means that less energy is wasted. However, it should be noted that with compressor sizes below 80%, there is a rapid reduction in emissions. As with earlier evaluations, this was found to be partly due to the system possessing insufficient capacity to achieve the required application temperature.

![Figure 24: Effect of compressor size on emissions from system operation](image)

4.4.4 Refrigerant selection
The final assessment of design issues relates to the refrigerant selection, which is particularly important since fluid choice is perhaps the main driver for the popularisation of the TEWI concept. As identified by the Author (Colbourne and Suen, 2000), the subject of comparative
refrigerant performance is complex and controversial and for this reason, only a limited assessment will be carried out here. In addition, the evaluations were simplified by only considering pure refrigerants, so that the complexities associated with composition shift in zeotropes are avoided. Using a similar principle to the cost-based approach described by Douglas et al (1999), the influence of selected refrigerants are based on three different basis for comparison:

a) Fixed system design. Only the refrigerant selection was changed, thus representing different fluids being used in the same system. This could be interpreted as a retrofit type comparison (Table 17).

b) Fixed system capacity at design conditions. Compressor size was changed so the design capacity is fixed for all refrigerants at design conditions, but the rest of the system remains as (a) (Table 18).

c) Fixed system capacity and COP at design conditions. Compressor size was changed as (b), but although compressor COP is the same for all refrigerants, evaporator and condenser size and HTF flow rate (to maintain velocity) were adjusted equally to maintain the system COP at design conditions (Table 19).

Table 17: Refrigerant comparison (a) with fixed system design

<table>
<thead>
<tr>
<th>Parameter</th>
<th>R152a</th>
<th>R134a</th>
<th>R290</th>
<th>R22(ref)</th>
<th>R1270</th>
<th>R143a</th>
<th>R125</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative exchanger size (%)</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Comp. swept volume (m³ h⁻¹)</td>
<td>58.5</td>
<td>58.5</td>
<td>58.5</td>
<td>58.5</td>
<td>58.5</td>
<td>58.5</td>
<td>58.5</td>
</tr>
<tr>
<td>COP at design (⁺⁻)</td>
<td>-</td>
<td>-</td>
<td>1.47</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Mean COP (⁺⁻)</td>
<td>2.44</td>
<td>2.39</td>
<td>2.34</td>
<td>2.15</td>
<td>2.33</td>
<td>2.16</td>
<td>2.04</td>
</tr>
<tr>
<td>Mean system capacity (kW)</td>
<td>11.79</td>
<td>11.93</td>
<td>15.53</td>
<td>16.59</td>
<td>20.21</td>
<td>19.16</td>
<td>18.81</td>
</tr>
<tr>
<td>Time $T_{app} = set$ point (%)</td>
<td>0%</td>
<td>8%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>SOL/EOL</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Refrigerant (kgCO₂)</td>
<td>122</td>
<td>1839</td>
<td>4</td>
<td>2526</td>
<td>3</td>
<td>5524</td>
<td>5753</td>
</tr>
<tr>
<td>Energy (kgCO₂)</td>
<td>3032</td>
<td>3110</td>
<td>3017</td>
<td>3634</td>
<td>3017</td>
<td>3233</td>
<td>3397</td>
</tr>
<tr>
<td>In-Use</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Refrigerant (kgCO₂)</td>
<td>360</td>
<td>5458</td>
<td>12</td>
<td>7675</td>
<td>10</td>
<td>16938</td>
<td>17756</td>
</tr>
<tr>
<td>Energy (kgCO₂)</td>
<td>202880</td>
<td>207690</td>
<td>223581</td>
<td>238351</td>
<td>223958</td>
<td>236968</td>
<td>248654</td>
</tr>
<tr>
<td>Sub total</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Refrigerant (kgCO₂)</td>
<td>482</td>
<td>7296</td>
<td>16</td>
<td>10201</td>
<td>14</td>
<td>22461</td>
<td>23509</td>
</tr>
<tr>
<td>Energy (kgCO₂)</td>
<td>205912</td>
<td>210800</td>
<td>226598</td>
<td>241985</td>
<td>226975</td>
<td>240201</td>
<td>252051</td>
</tr>
<tr>
<td>Total (kgCO₂)</td>
<td>206394</td>
<td>218096</td>
<td>226614</td>
<td>252186</td>
<td>226989</td>
<td>262662</td>
<td>275561</td>
</tr>
</tbody>
</table>
Each table shows variations of heat exchanger size (both evaporator and condenser), compressor swept volume, COP at design, mean COP, mean application temperature, mean refrigerating capacity, and the proportion of time (over the year) that the application achieves design temperature. Mean COP and temperature are time-weighted according to operating conditions. Equivalent emissions for refrigerant and energy consumption are presented for SOL/EOL and In-use stages. Seven refrigerants are included, and are listed in order of decreasing normal boiling point (NBP). The reference refrigerant was R22, which means that the input parameters for comparison (b) and comparison (c) were varied to match the original R22 design. Lastly, it is noted that for each combination—whether the system is designed for a given refrigerant or it is retrofitted—the charge size is optimum and it is assumed that superheat is always adjusted to the prescribed value.

Table 17 shows that the mean COP and capacity approximately follow the NBP of the refrigerant. Lower COPs and higher capacities occur with lower NBP refrigerant, which leads to the higher emissions from energy consumption. This is due to the greater volumetric refrigerating effect of lower NBP refrigerants. However, because the refrigerating capacity of the higher NBP refrigerants is lower, the R152a and R134a systems were rarely able to achieve the average application temperature set point, and consistent with earlier evaluations, this is reflected in the low In-use stage emissions from energy. Equivalent emissions from refrigerant release both for SOL/EOL and In-use stage closely follows the GWP of the fluid, whereas emissions from energy consumption at SOL/EOL stages are virtually equal for all systems. The total lifetime emissions are shown relative to the R22 system in Figure 25. Considering the trends in emissions from in-use stage energy consumption only, it is evident that refrigerant GWP can have a major influence on total emissions as can be seen by the significantly lower values for R290 and R1270, and the maximum emissions for R143a and R125.

For comparison (b), the design evaporating capacity was maintained at a design application load of 14.6 kW. Note that all systems now consistently meet the design application temperature. The consequence of matching compressor size to the heat load for different refrigerants is that high NBP refrigerants suffer degradation of COP at design conditions, although mean annual COP is relatively constant. This is because (for otherwise fixed conditions) increasing condensing temperature degrades cycle efficiency as it approached the critical point. Typically, high NBP refrigerants have lower critical temperatures so raising the condensing temperature means that a higher NBP refrigerant approaches critical temperature more rapidly than the lower NBP refrigerant, and therefore efficiency decays at a higher rate. Since their condensing temperature at
design conditions is intrinsically higher than under usual ambient conditions, the relative
difference between cycle efficiencies of the different refrigerants is amplified, whilst under usual
ambient conditions this influence is less pronounced.

Table 18: Refrigerant comparison (b) with fixed system capacity at design conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>R152a</th>
<th>R134a</th>
<th>R290</th>
<th>R22(ref)</th>
<th>R1270</th>
<th>R143a</th>
<th>R125</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative exchanger size (%)</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Comp swept volume (m³ h⁻¹)</td>
<td>124.2</td>
<td>143.3</td>
<td>67.9</td>
<td>58.5</td>
<td>50.8</td>
<td>59.9</td>
<td>75.7</td>
</tr>
<tr>
<td>COP at design (-)</td>
<td>1.713</td>
<td>1.572</td>
<td>1.668</td>
<td>1.47</td>
<td>1.573</td>
<td>1.326</td>
<td>1.09</td>
</tr>
<tr>
<td>Mean COP (-)</td>
<td>2.14</td>
<td>2.06</td>
<td>2.30</td>
<td>2.15</td>
<td>2.33</td>
<td>2.17</td>
<td>2.04</td>
</tr>
<tr>
<td>Mean system capacity (kW)</td>
<td>16.51</td>
<td>18.85</td>
<td>17.72</td>
<td>16.59</td>
<td>17.37</td>
<td>19.78</td>
<td>24.50</td>
</tr>
<tr>
<td>Mean $T_{app}$ (°C)</td>
<td>-21.70</td>
<td>-21.72</td>
<td>-21.72</td>
<td>-21.70</td>
<td>-21.69</td>
<td>-21.54</td>
<td>-21.52</td>
</tr>
<tr>
<td>Time $T_{app}$ = set point (%)</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>SOL/ Refrigerant (kgCO₂)</td>
<td>124</td>
<td>1874</td>
<td>4</td>
<td>2526</td>
<td>3</td>
<td>5522</td>
<td>5787</td>
</tr>
<tr>
<td>Energy (kgCO₂)</td>
<td>3798</td>
<td>4066</td>
<td>3140</td>
<td>3634</td>
<td>2911</td>
<td>3251</td>
<td>3620</td>
</tr>
<tr>
<td>In-Use Refrigerant (kgCO₂)</td>
<td>354</td>
<td>5414</td>
<td>12</td>
<td>7675</td>
<td>10</td>
<td>16942</td>
<td>17813</td>
</tr>
<tr>
<td>Energy (kgCO₂)</td>
<td>241502</td>
<td>250416</td>
<td>226289</td>
<td>238351</td>
<td>223091</td>
<td>236230</td>
<td>259762</td>
</tr>
<tr>
<td>Sub total Refrigerant (kgCO₂)</td>
<td>478</td>
<td>7287</td>
<td>16</td>
<td>10201</td>
<td>14</td>
<td>22463</td>
<td>23600</td>
</tr>
<tr>
<td>Energy (kgCO₂)</td>
<td>245300</td>
<td>254482</td>
<td>229429</td>
<td>241985</td>
<td>226002</td>
<td>239481</td>
<td>263382</td>
</tr>
<tr>
<td>Total (kgCO₂)</td>
<td>245778</td>
<td>261770</td>
<td>229445</td>
<td>252186</td>
<td>226016</td>
<td>261944</td>
<td>286982</td>
</tr>
</tbody>
</table>

However, emissions from energy consumption are closely correlated to mean COP, but bear no
relationship to the COP at design conditions. The differences in amount of system materials due
to changes in compressor size are seen to have only a negligible impact on overall emissions. As
with (a), equivalent emissions from the refrigerant vary considerably with GWP. Comparison of
total emissions in Figure 25 shows results that are more favourable for R290 and R1270, and all
lower NBP refrigerants have total emissions that remain similar to (a).

Finally, for the comparison (c), the capacity was kept constant and the COP was maintained at
1.47 under design conditions, thus necessitating changes in both exchanger lengths and HTF
volume flow rate. Despite this, it is seen that the mean COP still varies significantly between
refrigerants because of their individual behaviour at off-design temperatures, and in fact, the
mean COP clearly follows the refrigerant NBP. However, there no longer appears to be a
correlation between mean COP and emissions from energy production and this a result of the
neutralising effect that the larger mean capacity of low NBP refrigerant has on compressor
power. Although the capacity matches at design conditions, the system using low NBP
refrigerants becomes significantly oversized at the more usual off design conditions, which leads to the overall system efficiency suffering, and losses due to more frequent cycling. A greater proportion of the contribution of energy consumption for R143a and R125 is due to greater fan power because of larger heat exchangers, and similarly a lower contribution from fan power is evident for the higher NBP refrigerants. It is also noted that R143a and R125 required the additional airflow in order to achieve equal COP at design conditions.

Table 19: Refrigerant comparison (c) with fixed system COP at design conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>R152a</th>
<th>R134a</th>
<th>R290</th>
<th>R22(ref)</th>
<th>R1270</th>
<th>R143a</th>
<th>R125</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative exchanger size (%)</td>
<td>56%</td>
<td>71%</td>
<td>71%</td>
<td>100%</td>
<td>84%</td>
<td>125%</td>
<td>220%</td>
</tr>
<tr>
<td>Comp swept volume (m³ h⁻¹)</td>
<td>166.1</td>
<td>153.2</td>
<td>78.7</td>
<td>58.5</td>
<td>54.2</td>
<td>52.7</td>
<td>49.8</td>
</tr>
<tr>
<td>COP at design (-)</td>
<td>1.47</td>
<td>1.47</td>
<td>1.47</td>
<td>1.47</td>
<td>1.47</td>
<td>1.47</td>
<td>1.47</td>
</tr>
<tr>
<td>Mean COP (-)</td>
<td>1.94</td>
<td>1.96</td>
<td>2.08</td>
<td>2.15</td>
<td>2.19</td>
<td>2.31</td>
<td>2.45</td>
</tr>
<tr>
<td>Mean system capacity (kW)</td>
<td>15.54</td>
<td>17.12</td>
<td>17.26</td>
<td>16.59</td>
<td>17.01</td>
<td>18.81</td>
<td>21.05</td>
</tr>
<tr>
<td>Mean ( T_{\text{app}} ) (°C)</td>
<td>-21.35</td>
<td>-21.67</td>
<td>-21.72</td>
<td>-21.70</td>
<td>-21.66</td>
<td>-21.68</td>
<td>-21.53</td>
</tr>
<tr>
<td>Time ( T_{\text{app}} = \text{set point (°C)} )</td>
<td>96%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
</tr>
</tbody>
</table>

The three comparisons detailed here reveal some key observations. First, it is seen from the analysis that the equivalent emissions can vary significantly depending upon refrigerant selection and system design. It is important to match system and refrigerant in order to assure the required application temperatures. More importantly, the sizing of components should really be done at conditions closer to the average temperatures during normal operation as it is clear that optimising for one ambient can "reverse" the optimisation at a different temperature. To demonstrate this, the emissions were calculated for a design ambient of 11°C (which is near the average temperature over the year) whilst maintaining the same capacity and COP of R22 as in comparison (c), and the results are also shown in Figure 25.

It is seen that the results are somewhat contrary to previous comparisons, where the lower NBP refrigerants have lower emissions than those with high NBP. Further, the emissions for R152a
and R134a are highest in this case that for previous comparisons, whereas the other refrigerants produce lower emissions, including the reference R22. However, it was found from the analysis that this approach (matching capacity and COP at an appropriate ambient) can cause the system to be incapable of achieving the required application temperature at certain times. Also noteworthy is that the mean annual COP for all refrigerants was virtually identical to the design COP; similarly, mean annual capacity was close to the design capacity. Although the mean COP and capacity were nearly identical for all refrigerants, the emissions from energy production for system operation varied significantly with the same ranking as total emissions, where R152a and R134a were 10% and 20% higher than R22 respectively, R290 and R1270 were 5% lower and the others were close to R22. This variation is explained by the additional operating duration for lower pressure refrigerants at the higher ambient temperatures. Finally, any changes in system design to reduce emissions through capacity matching or efficiency enhancement will not be offset by consumption of additional materials.

![Comparison of total emissions for different refrigerants](image)

Figure 25: Comparison of total emissions for different refrigerants

In terms of refrigerant selection, a general observation is that R290 and R1270 always result in lowest emissions when the required application temperature is achieved. This is partly due to favourable thermal properties, which provide good efficiency, and partly because of their low GWP compared to the HFCs.

4.5 Effects of material/refrigerant handling and operations
As with equipment design, it is important to understand the effect that certain material handling and other human operations have on GHG emissions, as a particular measure can lead to lower
emissions from one source, but cause more emissions elsewhere. In order to evaluate the influence of handling and other operations on emissions, the following cases were addressed:

- Poor and best practice during refrigerant handling, and recycling
- Frequency of condenser and system cleaning
- Frequency of checking and servicing leaks

Understanding this process will provide useful insight into optimisation of operational activities to reduce emissions.

![Figure 26: Comparison of emissions for different service and handling practices](image)

4.5.1 Refrigerant handling practices and recycling

Best practice, normal and poor practice for refrigerant handling was characterised by setting the model parameters according to Table 20. Best practice accounts for the refrigerant being used in a manner that aims to minimise emissions. Normal practice is what the author considered as typical for the companies involved, and poor practice represents the case where refrigerant containment is virtually neglected.

The reference system in Table 14 was used to generate emissions for SOL, In-use (for one year), and EOL, and input parameters adjusted according to Table 20. The results are presented in Figure 26. Compared to normal practice, the results for best practice show significant reduction at SOL, in-use servicing and EOL refrigerant handling. Overall, there is a 5% reduction over a 10-year lifetime, although in terms of refrigerant handling only, this increases to 40% reduction. Conversely, poor practice increases total lifetime emissions by almost 20% and in terms of
refrigerant emissions only, 200%. Where a refrigerant with low GWP is employed then there would be negligible effect caused by handling practices.

Table 20: Input parameters for refrigerant handling practice

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Poor practice</th>
<th>Normal practice</th>
<th>Best practice</th>
</tr>
</thead>
<tbody>
<tr>
<td>Recovery pressure</td>
<td>100 kPa</td>
<td>65 kPa</td>
<td>30 kPa</td>
</tr>
<tr>
<td>Frequency of refrigerant handling per visit</td>
<td>0.3</td>
<td>0.3</td>
<td>0.3</td>
</tr>
<tr>
<td>Probability of venting during handling</td>
<td>0.9</td>
<td>0.2</td>
<td>0.1</td>
</tr>
<tr>
<td>EOL recovery fraction</td>
<td>0.1</td>
<td>Refrigerant</td>
<td>0.9</td>
</tr>
<tr>
<td>Leak test gas</td>
<td>Refrigerant</td>
<td>Refrigerant</td>
<td>Inert gas</td>
</tr>
<tr>
<td>Charging phase</td>
<td>Liquid</td>
<td>Liquid</td>
<td>Vapour</td>
</tr>
</tbody>
</table>

Lastly, Figure 26 also presents results for normal practice where materials and refrigerant have been recycled. It is seen that recycling only affects SOL energy emissions, reducing them by about 50%. However, over the total lifetime this accounts for 1-2% of emissions. If a refrigerant with low GWP were used, these differences would be halved since the contribution from refrigerant production emissions at SOL would be minor in the first place. There was also a difference at EOL energy emissions because there is less for incineration, but this again is negligible.

![Figure 27: Influence of condenser cleaning and system cleaning on emissions](image-url)
4.5.2 Condenser and system cleaning

Section 4.3.2 provided results for variation in emissions from the system due to the influence of exchanger fouling and compressor wear. Although both of these cause degradation of system performance and therefore increased emissions, their impact can also be reduced by removing fouling from exchanger surfaces by cleaning, and compressor wear can be lessened by ensuring a clean internal system, by replacing the filter/drier and compressor oil. To quantify these processes, the model was run for different frequencies of maintenance visits where the contribution of fouling is halved whenever the condenser is cleaned and the reduction in volumetric efficiency is reduced proportionally to the number of oil/filter/drier changes. For equation (43) and (44), substitute the constant \( a \) for \( a/(1 + N_{\text{sen}}) \) and for equation (45) assume \((x/k)_{\text{foul}} = 0.5 \cdot (x/k)_{\text{foul}}\) for every visit.

![Figure 28: Influence of condenser and system cleaning on mean application temperature](image)

Figure 27 shows the resultant annual emissions for both energy contribution and total emissions after 8 years of operation, relative to the case for no ageing (i.e., zero years). Data is provided for the case of no cleaning, cleaning the condenser once per year, cleaning the internal system once per year, and for both practices once and twice per year.

As shown here and in Figure 19, the influence of ageing causes a notable increase in emissions. Both condenser and internal system cleaning reduce this effect slightly in terms of emissions from energy use, although total emissions increase above the case for no cleaning because breaking into the system results in increased refrigerant release. Cleaning the system twice in a year produces a minor improvement, but still the effect of additional refrigerant emissions can outweigh the benefits of reduced energy use from minimising compressor wear. Note that if the
refrigerant possesses a small GWP (such as an HC) then there would be value in both condenser and internal system cleaning. The influence of additional emissions from travelling to the site more frequently is negligible. This is seen in Figure 28 also, where the process of maintaining good performance allows the mean application temperature to be closer to the required design temperature.

![Figure 28](image)

**Figure 28: Effect of additional emissions from travelling to the site on emissions**

4.5.3 Leak checks and repair

Loss of refrigerant due to leakage is accounted for in the model by reducing exchanger surface area available for heat transfer in proportion to reduction in charge level. The influence of service repair and top-up in terms of sub-optimal charge effect on emissions is examined here. Six scenarios are considered, where frequency of service for repairing the leak, and topping-up the charge with or without repair are varied. Only topping-up the system means that the system recovers its optimum charge size but the annual leak rate remains the same. Repairing the leak and topping up reduces leak rate and regains optimum charge.

Figure 29 shows the annual equivalent emissions of refrigerant during operation and servicing, from energy consumption and the annual total. In terms of refrigerant emissions, those from operation reduce with more frequent repair. However, a larger number of top-ups increase refrigerant emissions because of releases during refrigerant handling, although this increases when repairs are made also as the refrigerant must be recovered from the system. The emissions resulting from the service person transportation are very small when compared to the equivalent emissions from refrigerant handling during servicing. In terms of emissions from energy consumption, there is a general increase, which seems contradictory since the efficiency of the
system will be improved. If mean application temperature is considered, then this observation can be explained. Figure 30 shows that more frequent servicing and top-ups lead to improved application temperature, which is away from the design value because the loss of charge also causes a reduction in refrigerating capacity. Thus, whilst emissions from energy consumption increase, the purpose of the equipment is better achieved.

Figure 30: Effect of servicing repair and top-up frequency on mean application temperature (design application: -20°C)

From this discussion, it appears that there is a balance between better performance of the equipment, additional refrigerant emissions from handling and whether system energy consumption will be reduced or increase through less system leakage. Thus, operation procedures can be optimised to achieve lowest GHG emissions. These results are strongly dependent upon the sensitivity of the system to reduced charge and the behaviour of the service person when handling refrigerant. In particular, if a refrigerant with negligible GWP were used the situation would be different from the case above because the benefits of servicing and topping-up on energy consumption will never be offset by the negative impact of additional refrigerant emissions.

4.6 Comparison with BRA method
Finally, a comparison was made against the only formalised TEWI methodology currently available, which is published by the BRA (1996). The standard methodology was followed, and all data used in the calculation was taken from the document. All other application data was taken from the reference system data in Table 14. Table 21 reproduces the format for calculating energy consumption, where the power drawn was obtained from running the system model.
(without ageing influences) for the specified ambient temperature. This calculated power consumption data, was then fed into the standard TEWI calculation table from BRA, which is reproduced in Table 22, alongside data generated using the model in this study.

Table 21: Energy calculations according to BRA method

<table>
<thead>
<tr>
<th>Segment name</th>
<th>Utility factor</th>
<th>Ambient</th>
<th>Power drawn</th>
<th>Run time per year</th>
<th>Energy consumed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Summer load - peak</td>
<td>N/a</td>
<td>27°C</td>
<td>9.11 kW</td>
<td>116 h</td>
<td>1057 kWh</td>
</tr>
<tr>
<td>Summer load - base</td>
<td>N/a</td>
<td>22°C</td>
<td>8.71 kW</td>
<td>632 h</td>
<td>5505 kWh</td>
</tr>
<tr>
<td>Autumn/spring load - peak</td>
<td>N/a</td>
<td>17°C</td>
<td>8.26 kW</td>
<td>2154 h</td>
<td>17792 kWh</td>
</tr>
<tr>
<td>Autumn/spring load - base</td>
<td>N/a</td>
<td>12°C</td>
<td>8.25 kW</td>
<td>2834 h</td>
<td>23381 kWh</td>
</tr>
<tr>
<td>Winter load - peak</td>
<td>N/a</td>
<td>7°C</td>
<td>8.08 kW</td>
<td>2234 h</td>
<td>18051 kWh</td>
</tr>
<tr>
<td>Winter load - base</td>
<td>N/a</td>
<td>2°C</td>
<td>7.96 kW</td>
<td>788 h</td>
<td>6272 kWh</td>
</tr>
<tr>
<td>Total (compressor)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>72057 kWh</td>
</tr>
<tr>
<td>Fans – condenser</td>
<td>100 %</td>
<td>N/a</td>
<td>1.31 kW</td>
<td>8760 h</td>
<td>6570 kWh</td>
</tr>
<tr>
<td>Fans – evaporator</td>
<td>100 %</td>
<td>N/a</td>
<td>0.75 kW</td>
<td>8760 h</td>
<td>11738 kWh</td>
</tr>
<tr>
<td>Total (ancillary)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>18308 kWh</td>
</tr>
</tbody>
</table>

Comparison between the results of the BRA method and the model in the present study show that BRA produces significantly larger overall TEWI, where even the model calculations using the maximum settings were 20% lower. For the refrigerant contribution, the modelled values are between one half and seventeen times the BRA value, as reflected in the in-use and the retirement or EOL emission values. This difference is attributed to the inclusion of the numerous peripheral release sources that the model accounts for, and the range in annual leak rates. The modelled energy consumption is 25% to 40% lower than that estimated by the BRA method. The main reason for this is that the model includes for compressor cycling and reduction in application heat load in response to variations in ambient conditions, despite including additional emissions due to SOL, servicing and EOL, and a reduction of power station emissions over time. Also of interest is the proportion of total emissions due to refrigerant release: the BRA method gives less than 1% whereas the model gives up to 10%. This comparison suggests that to provide a more accurate representation of TEWI, the BRA method should at least include detailed system energy use calculations, and improved assessment of refrigerant releases such as during refrigerant handling.
### Table 22: Standard calculations according to BRA method

<table>
<thead>
<tr>
<th>Input parameter</th>
<th>Values for BRA (1996)</th>
<th>Values for present study</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Minimum</td>
<td>Mean</td>
</tr>
<tr>
<td>Refrigerant name</td>
<td>R22</td>
<td>R22</td>
</tr>
<tr>
<td>Refrigerant charge size</td>
<td>2.49 kg</td>
<td>2.49 kg</td>
</tr>
<tr>
<td>Annual energy Compressors</td>
<td>72057 kWh</td>
<td>17977 kWh</td>
</tr>
<tr>
<td>Annual energy Ancillary</td>
<td>18308 kWh</td>
<td>1259 kWh</td>
</tr>
<tr>
<td>System operating lifetime</td>
<td>8 years</td>
<td>8 years</td>
</tr>
<tr>
<td>Refrigerant GWP</td>
<td>1700 kgCO₂</td>
<td>1700 kgCO₂</td>
</tr>
<tr>
<td>Leakage factors</td>
<td></td>
<td></td>
</tr>
<tr>
<td>L1 (annual leak rate)</td>
<td>0.05</td>
<td>0.022</td>
</tr>
<tr>
<td>L2 (annual purge factor)</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>S1 (annual serv release)</td>
<td>0.0025</td>
<td>0.026</td>
</tr>
<tr>
<td>S2 (catastrophic failure)</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Recovery efficiency</td>
<td>0.95</td>
<td>1.0</td>
</tr>
<tr>
<td>Electricity CO₂ emission factor</td>
<td>0.53 kgCO₂</td>
<td>See 2.3.2</td>
</tr>
<tr>
<td>Direct effect</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Refr loss (operation)</td>
<td>1.05 kg</td>
<td>0.55 kg</td>
</tr>
<tr>
<td>Refr loss (retirement)</td>
<td>0.12 kg</td>
<td>0.04 kg</td>
</tr>
<tr>
<td>Total refrigerant loss</td>
<td>1.17 kg</td>
<td>0.59 kg</td>
</tr>
<tr>
<td>Refrigerant loss CO₂ equivalent</td>
<td>1990 kgCO₂</td>
<td>1068 kgCO₂</td>
</tr>
<tr>
<td>Indirect emissions CO₂ equivalent</td>
<td>383150 kgCO₂</td>
<td>242573 kgCO₂</td>
</tr>
<tr>
<td>Total equivalent warming impact</td>
<td>385140 kgCO₂</td>
<td>243642 kgCO₂</td>
</tr>
</tbody>
</table>

### 4.7 Interpretation of global warming

Section 3 detailed alternative measures for evaluating climate change impact, rather than depending upon the GWP concept. The values of TLTI, GDI1 and GDI2 were obtained from the emissions model, and compared against results for TEWI. Calculations were made for a reference system, and four other systems with differing characteristics: (1) high efficiency and high leakage, (2) high efficiency and low leakage, (3) low efficiency and high leakage, and (4) low efficiency and low leakage. These were adopted to help illustrate the differences in global warming impact according to different measures when a refrigerant with a high \( F \) and \( \tau \) (R22) is compared against a low \( F \) and \( \tau \) refrigerant (R290).

A first example is shown in Figure 31 where a new reference system (Table 14) is calculated for R22 and R290 to show the difference between a high and low climate impact refrigerant. The contribution from energy consumption and direct effect of refrigerant are separated for the R22 system, but only the total has been shown for R290. It is seen that the temperature rise is rapid.
soon after the start of the system installation and within 15 years the maximum temperature rise, or TLTI, is reached. Although the leak rate was $R_{oper} = 12\%$, the contribution of the R22 emissions accounts for over one-third of the temperature rise for the duration of the system life. Following this peak, the impact on temperature drops rapidly and within 70 years all influence from R22 emissions have ceased. However, the contribution from energy consumption has only decayed by about one-quarter of its maximum. By comparison, AGTP of the refrigerant for the R290 system is negligible and TLTI of that system is therefore a function of energy consumption only.

![Figure 31: Comparison of AGTP-time profile for high and low climate impact refrigerant](image)

Table 23 presents full results for TEWI, TLTI, GDI1 and GDI2 for the reference system with each refrigerant. Using GWP(100), TEWI of the R22 system is 10,653 kgCO₂ in comparison to 6,867 kgCO₂ for R290, being 36% lower. This is a close comparison to TLTI, which indicates the R290 system to be 39% lower than the R22 system. For the global damage indices, the R22 system GDI1 is 33% higher than GDI1 when R290 is used. Most significant is the difference in GDI2, where the R290 GDI2 is 64% less than the R22 value.

The comparison between the different measures was extended by evaluating four different systems, each representing different combinations of efficiency and leakage. The model input conditions are based on a reference system, although for the high efficiency system, heat exchanger sizes were doubled, and half size for the low efficiency system. In addition, a low leakage system is assumed to have annual $R_{oper} = 5\%$, whereas the high leakage system has annual $R_{oper} = 20\%$. The data in Table 23 shows a broad variation of values for each climate.
change impact. As expected, the highest value of TEWI is for the low efficiency, high leakage system, and the lowest value of TEWI are for the high efficiency, low leakage system. When comparing data for the other measures, the sign is the same.

Table 23: Comparison of climate impact for different systems

<table>
<thead>
<tr>
<th>System characteristics</th>
<th>TEWI (kgCO₂)</th>
<th>TLTI (K)</th>
<th>GDI1 (K³ y⁻¹)</th>
<th>GDI2 (K y⁻²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference efficiency</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Reference leakage</td>
<td>R22</td>
<td>10653</td>
<td>3.09 × 10⁻¹⁹</td>
<td>8.46 × 10⁻²⁷</td>
</tr>
<tr>
<td></td>
<td>R290</td>
<td>6867</td>
<td>1.82 × 10⁻¹⁹</td>
<td>5.63 × 10⁻²⁷</td>
</tr>
<tr>
<td>High leakage</td>
<td>R22</td>
<td>10341</td>
<td>3.06 × 10⁻¹⁹</td>
<td>8.94 × 10⁻²⁷</td>
</tr>
<tr>
<td></td>
<td>R290</td>
<td>7559</td>
<td>2.01 × 10⁻¹⁹</td>
<td>6.50 × 10⁻²⁷</td>
</tr>
<tr>
<td>Low leakage</td>
<td>R22</td>
<td>9363</td>
<td>2.64 × 10⁻¹⁹</td>
<td>7.51 × 10⁻²⁷</td>
</tr>
<tr>
<td></td>
<td>R290</td>
<td>7039</td>
<td>1.87 × 10⁻¹⁹</td>
<td>5.84 × 10⁻²⁷</td>
</tr>
<tr>
<td>Low efficiency</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>High leakage</td>
<td>R22</td>
<td>12734</td>
<td>3.90 × 10⁻¹⁹</td>
<td>1.07 × 10⁻²⁶</td>
</tr>
<tr>
<td></td>
<td>R290</td>
<td>7124</td>
<td>1.89 × 10⁻¹⁹</td>
<td>5.94 × 10⁻²⁷</td>
</tr>
<tr>
<td>Low leakage</td>
<td>R22</td>
<td>11478</td>
<td>3.19 × 10⁻¹⁹</td>
<td>8.64 × 10⁻²⁷</td>
</tr>
<tr>
<td></td>
<td>R290</td>
<td>6904</td>
<td>1.83 × 10⁻¹⁹</td>
<td>5.67 × 10⁻²⁷</td>
</tr>
</tbody>
</table>

The impact using TEWI, TLTI, GDI1 and GDI2 are shown for the different system characteristics, relative to the R22 reference system in Figure 32. Initial general observations are that the impact of R290 is always lower than that of the R22 system, and for a given system efficiency, the difference is greater with a high leak. Subsequently, TEWI of the R290 system remains similar regardless of leak rate, whilst there is considerable variation between TEWI of the different R22 systems. TLTI closely follows the impact indicated by TEWI, although it is generally a little lower than TEWI for the R290 systems, and slightly higher for the R22 systems with high leakage. GDI1 always implies a higher relative impact than TEWI regardless of system or refrigerant, except in the one case of a low efficiency R22 system with low leakage. Finally, GDI2 follows the pattern of TLTI where it has a higher relative impact for R22 systems with large leakage, but an appreciably lower impact for all R290 systems. The variations between the different measures of global warming impact with differing system characteristics are examined further by comparing the relative variation of each against that of TEWI. To evaluate the data from Table 23 in this context, it is presented in Figure 33 as the relative change in impact for each system, compared to the change of TEWI (e.g., \( \frac{TLTI}{TLTI_{ref}} \pm \frac{TEWI}{TEWI_{ref}} \)), where different trends emerge depending upon the measure.
In terms of TLTI, comparison of results for systems using R290 shows that the relative impact is almost identical to that of TEWI regardless of leak rate and efficiency. The R22 systems have TLTI greater relative to TEWI for the high leak cases and less than TEWI for low leakage, although the variation is more pronounced for the low efficiency systems. When comparing TLTI for the CO₂ contribution only of the different systems (not shown), the relative change against TEWI was also negligible, as with R290. These observations demonstrate that TLTI consistently emphasises the rapid global warming contribution of a refrigerant’s high $F$ and $\tau$.

Initial observations for GDI1 are that the relative change in impact is nearly always greater than TEWI regardless of refrigerant, although for a given system efficiency the contribution is less when leak rates are low. Similarly, the impact relative to TEWI is greater when system efficiency is low. In general, these findings indicate that GDI1 amplifies the contribution of both refrigerant and CO₂ emissions, which is expected given that GDI1 is a direct escalation of AGTP. The one exception is the low leakage, low efficiency R22 system where the relative GDI1 is less than TEWI, which suggests that the contribution of the refrigerant is still given a greater weighting than CO₂.

GDI2 produces similar results to TLTI for R290 systems where there is negligible difference between GDI2 and TEWI, which is consistent with its direct dependence on TLTI. However, results for the R22 systems reveal somewhat different results since GDI2 demonstrates a lesser impact than TEWI for three of the four cases. Of these the high efficiency, low leakage system exhibits a very large reduction relative to TEWI, and conversely the low efficiency, high leakage system is the only case where the relative impact is greater than TEWI. These observations
suggest that GDI2 dampens the contribution of CO$_2$, but overall there is a more distinct indication of which system designs are favourable in terms of global warming impact.

![Graph showing climate impacts for different system characteristics relative to R22 and R290 TEWI](image)

**Figure 33: Climate impacts for different system characteristics relative to R22 and R290 TEWI**

To summarise, this Section has presented examples where the proposed alternative measures for evaluating the climate change impact of refrigerating systems, using a high $F$ and $\tau$ refrigerant, and a low $F$ and $\tau$ refrigerant have been illustrated. A newly derived measure known as AGTP was employed to develop various climate impacts because (a) for most refrigerants it avoids reliance upon accuracy of evaluating CO2, (b) less reliance upon selection of appropriate time horizons, (c) it is simple to extend to a damage index, and (d) it's meaning is comprehensible to non-climatologists. These indices are TLTI, GDI1 and GDI2 as detailed in Section 3. Through evaluating selected systems, the relative climate impact using these measures was found to be similar, although they tended to emphasise the effect of refrigerant over the CO2 contribution. Lastly, one crucial feature is that the choice of how global warming impact is interpreted can change the rating order of different system. In a practical sense, a TEWI comparison of two different system designs (including different refrigerant choice) may imply one of the options produces least global warming impact, whilst use of TLTI, GDI1 or GDI2 may demonstrate the other option has the smallest impact. This disparity can be used advantageously, in that greater confidence is gained when an option produces the lowest impact with all four measures.

4.8 Use and advantages of methodology

The comprehensive method presented here can be of use in two respects. The first main use is for accurate evaluation of different designs of equipment, whether it is refrigerant or component
selection. This approach can be extended to GHG emission optimisation of equipment design, installation and optimisation, for manufacturers, installers, service companies and system operators. Secondly, it can be used for technology assessment purposes, where the overall impact of prescribing certain measures within policies can be evaluated in terms of reduction of CO₂ and CO₂ equivalent emissions.

With the increasing attention paid to achieving reduced environmental impact, industry is under increasing pressure to employ equipment and processes that are low in GHG emissions. For example, "high street brand" companies are increasingly specifying high efficiency, low-GWP cooling equipment, and for servicing companies to have strict refrigerant containment work procedures in place. The approach detailed in the present study would allow those involved to analyse designs or procedures and demonstrate a low impact, or to help determine an optimum design or procedure. Examples of this could include:

- best distribution of funds for system components and materials,
- most suitable production and installation procedures,
- preferred refrigerant selection,
- optimum design of equipment for anticipated temperature and load profiles,
- favourable service and maintenance regime,
- most effective route for disposal of refrigerant and equipment.

Using a more transparent approach such as this would help eliminate bias from "generalised" results being presented as actuality.

Studies used for policy reasons, such as estimation of national GHG emissions for RAC sectors (e.g., March 1999, Haydock et al, 2003) employ a very simplistic approach to estimate impacts. These generally draw upon studies that are more specific, where systems are evaluated for different refrigerant selection, and associated generic system designs (e.g., AFEAS, 1997, etc.). Evaluations using the model show such a wide variation in TEWI results for a single system, that a few small changes in assumptions can easily reverse the rating of two different refrigerants or system types. This suggests that using the basic TEWI methodologies as detailed within such reports is erroneous and are unlikely to lead to valid findings. Consequently, a more analytic approach that employs statistical frequencies for occurrence of different influencing assumptions should be use to produce a range of TEWI results for a given sector. If a sector can be broken down into a range of different designs, practices, etc., a thorough and reliable TEWI evaluation can be made across the range.
SECTION 5: CONCLUSIONS

5.1 Summary of work
The current investigation relates to emissions of GHGs from RAC systems and their effect on climate change. Review of the literature found that the current calculation procedures as used by manufacturers, installers and end-users, vary in name, scope and detail. Most calculation procedures are left open to many assumptions, and reported evaluations tend to employ assumptions that are either not stated, or made to suit some predetermined conclusion.

To address the various shortfalls, a detailed emissions model was constructed, which included numerous sources of emissions associated with the equipment lifetime; an exhaustive list of these are listed in Appendix 1 (Table 24). The following aspects examined in terms of their effect on overall lifetime emissions: size and range of equivalent emissions from different sources, influence of external conditions, effects of system ageing, lifetime emissions from power stations, design and selection of equipment and components, refrigerant selection, repair and maintenance of system and topping up of refrigerant, behaviour during servicing and refrigerant and material handling. In addition, the sensitivity of assumptions was examined by considering the range of anticipated emissions from a particular source.

Lastly, because GWP is known to have shortfalls in terms of accurately describing the implication of the overall TEWI value, three alternative measures were adopted to interpret the impact of GHGs emissions. The impact resulting from eight different system designs were compared.

5.2 Key findings
Under certain circumstances, emissions associated with equipment production and servicing activities can represent a significant proportion of total emissions. Omission of these aspects in the standardised approach may result in significantly different TEWI. Assumptions about operating parameters also have a large influence on the output, for example, load profile and sensitivity of performance to charge.

Accurate evaluation of TEWI for a particular system requires detailed knowledge of environmental conditions and how the system behaves under the anticipated range of conditions, system control strategy, design of evaporator, condenser, compressor and pipework, knowledge of service and refrigerant handling processes and behaviour, and information of production of
materials, component/assembly manufacturing and disposal procedures. These revealed a number of important observations to be considered in assessments of RAC equipment, as summarised below.

- **Distribution of emission size.** The majority of the contribution (35 – 95%) normally occurs from system operating energy and the second most significant (3 – 58%) is normally leakage from the system. Refrigerant emissions from servicing activities during in-use stage may also be of similar size. The contribution of EOL and SOL may vary between <1% to over 10% of the total. Proportions of the total contribution attributed to different sources vary according to equipment size, where relative contribution from energy consumption and system leakage increase with larger equipment (67% for 4 kW to 91% at 58 kW).

- **Equipment design.** Component selection has an influence on annual leakage rates (2 – 20%), and as a consequence annual energy consumption (±5%) and application temperature (up to 20 K). An increase of evaporator and condenser sizes can reduce emissions (by 10% when doubling the size) from energy consumption or increase (by 50%) by halving their size. Using larger exchangers also raises the system refrigerant charge, in which case greater refrigerant emissions can also offset the benefits of improved system efficiency. Increasing SOL emissions through larger exchanger size has a negligible influence on total emissions (<1%), as does recycling equipment materials (1 – 2%). The sensitivity of the system to sub-optimal charge can affect both emissions from energy production and application temperature; if the system can achieve design temperatures, emissions increase with higher leakage and sensitivity to charge level (±20%), whereas high sensitivity may result in greater emissions for lower leakage rates, but applications temperature moves further from the design value. A smaller compressor (in terms of swept volume) always helps reduce emissions, although the influence of less materials, oil and refrigerant charge is minor (<1%). However, below a certain size, the compressor becomes too small to match the capacity under all conditions. The effect of refrigerant selection influences emissions in two respects. Contribution from refrigerant emissions is proportional to the refrigerant GWP. Contribution from energy consumption is largely dictated by refrigerant properties, but also the basis for comparison is considered crucial. The ranking between refrigerants can reverse when the basis is changed (±40%). Similarly, comparison refrigerants on a different basis will result in some refrigerants not achieving design application temperatures.
- **External influences.** The evaluation of energy consumption over a more precise range of ambient temperatures provides increasingly accurate results for subsequent emissions (±100%). Anticipated changes in the UK CO₂ emissions per kWh of electricity result in a lower lifetime emissions from energy consumption, up to 5%.

- **Equipment handling.** The manner in which installation, service and maintenance personnel carry out refrigerant handling activities affects total emissions, up to ±30%. Increased frequency of maintenance can help to reduce emissions, energy consumption and maintain application temperature. Where a refrigerant has a higher GWP, changing compressor oil and replacing filter/driers can cause an increase in total emissions. More frequent top-ups may be less beneficial than repair plus topping-up because system leakage is higher, but repairing increases servicing refrigerant releases which can offset the benefit of lower leak rate if refrigerant handling practices are lax. Where a low GWP refrigerant is used, topping-up is the favoured approach. Effects of ageing (compressor wear, exchanger fouling, increase in leak rate over time) can affect emissions from energy consumption by up to ±10%, but also result in the inability to achieve design application temperatures.

- **Interpretation of consequence of emissions.** Comparison based simply on GHG emissions is pointless unless they are considered in parallel with the state of application temperatures. An important conclusion drawn from evaluating the influence of refrigerant handling activities, is that a balance has to be sought, between the benefits associated with improved system performance (e.g., topping-up the charge) and the increase in emissions when servicing. The higher the refrigerant GWP, the more importance this trade-off becomes. Similarly, by using a refrigerant with very low GWP, the impact associated with peripheral refrigerant emissions can be neglected and so servicing activities that improve performance ought to be as frequent as possible.

If a simplistic approach such as BRA (1996) were employed, a significant over-prediction in the contribution of energy-related emissions (up to 40%), and under-prediction of high-GWP refrigerant contributions by a factor of 10 occurred. Finally, the findings reported in Section 4 help identify a number of design and operation features that can assist in minimising GHG emissions over equipment lifetime.

The present study has produced a number of outputs that are considered to be original contributions to this subject:
A thorough procedure for TEWI evaluation has been formalised, using a detailed modelling approach, which can demonstrate the influence of various equipment design parameters on total emissions. This was achieved by identifying an exhaustive list of different sources of emissions within the lifetime of equipment and calculations to evaluate them, and a detailed breakdown of GHG emissions from different sources during the lifetime of the equipment, along with uncertainty ranges was presented.

Part of the emissions model employed correlations produced to approximate masses of system components and thereby estimating emissions associated with their manufacture, including a factor to account for the construction of the system. Using a system model for in-use energy consumption calculations, the influence of ageing effects was incorporated by accounting for degradation of compressor performance, exchanger fouling and the effect of leakage. A comprehensive empirical leakage model was presented, based on number and selection of different components within the circuit, and the frequency of repair of the leaks. For peripheral sources, refrigerant emissions during handling procedures for production, servicing and decommissioning were characterised. In terms of indirect emissions due to power consumption, a correlation for estimating variable emissions of CO$_2$ per kWh of energy consumed according to the year, time of year and part of the day, was established along with upper and lower ranges.

Novel alternative means to evaluate the impact of GHG emissions from a refrigeration system were presented, based on absolute global temperature change due to an emission of a particular GHG; TLTI, GDI1 and GDI2. A comparison of these different interpretations of impacts from GHG emissions using several different system designs was offered.

Omissions and errors associated with the BRA (1996) method were identified, and the resulting uncertainty in using the BRA method presented.

The importance of the present study is reflected by considering the current demands for reducing environmental burden of equipment through national, European and international legislation, and environmental policies of businesses. Information provided by this study can assist manufacturers and users of RAC equipment to improve the quality of their environmental assessments when considering different options. Further, it helps focus attention towards the aspects of equipment design, construction and operation that can give most benefit when trying to reduce impact due to GHG emissions. In particular, there exists an extensive choice of refrigerants whose GWP ranges from zero to several thousand, and properties that dictate a wide range of efficiencies. The findings of this study thereby assist with identifying the most suitable refrigerant options.
5.3 Limitations and further work

The first consideration is that sub-models associated with the overall emissions model could always be made more accurate, and observations in the differences in model output when including "peripheral" aspects (such as fouling) confirms this. In particular, improved models should include:

- More accurate models for emissions from manufacturing and equipment installation.
- Better characterisation of system leakage, including the effects of external temperatures and operating conditions on leakage.
- Improved system performance model, particularly with respect to transient effects, also including effect of evaporator frosting and subsequent defrost strategy.
- Ability to account for differences in associated system designs, such as indirect circuits, multiple evaporators/compressors, and thermal storage.
- Other than simple on/off cycling, the effect of other control features such as pressure switches, defrost timing, evaporator/head pressure regulating valves should be accounted for.
- Ancillary system components such as desuperheaters and suction-liquid heat exchangers that may produce changes in energy consumption should be included.
- Inclusion of a more precise "bin" type approach for improved ambient temperature ranges and calculation of load profiles.
- Extension of the model to include for blended refrigerants, which could be useful for predicting changes in impacts due to preferential leakage of refrigerant components with different GWP, and effect on system performance.
- Specifying actual components used within a particular system, and also validating the performance model using measurements from that, or a similar system.

A major addition to the model would be to integrate a cost model. This would enable cost-effectiveness analysis of different emission reduction measures. This would help to pinpoint areas of system design or operation where expenditure should focus, and it would enable economic optimisation to achieve maximum emission reductions.

As a final observation, this study has shown that the use of TEWI for the usual purpose of comparing refrigerants has major uncertainties associated with it, and results can be highly sensitive to assumptions. Therefore, the most reliable approach for conducting a TEWI study would be to make it retrospective. Following the study of Benstead (1996), two different types of
systems should be installed and operated over a given lifetime, although a statistically significant
number of each system types would have to be used. Production and installation would be closely
monitored and evaluated, as would the EOL stage activities. Moreover, the performance (and
corresponding local conditions) of each system within the population should be recorded as well
as all service and maintenance activities. Full evaluation of individual systems within each
population would provide a genuinely meaningful comparison, and additionally, could be used
also to help validate models such as this.
REFERENCES


Banks, R. E., Sharrat, P. N. Environmental impacts of the manufacture of HFC-134a. Department of Chemical Engineering, UMIST, Manchester. 1996.


March. UK Emissions of HFCs, PFCs and SF6 and potential emission reduction opportunities. For Global Atmosphere Division, DETR, London. 1999.


Sand, J. R., Fischer, S. K., Baxter, V. Comparison of TEWI of alternative fluorocarbon refrigerants and technologies in residential heat pumps and air conditioners. Oak Ridge National Laboratory. For Alternative Fluorocarbons Environmental Acceptability Study (AFEAS) and Department of Energy. USA. 2002.


APPENDICES

Appendix 1: Compilation of sources found to affect emissions

Table 24: Exhaustive list of sources of emissions and influencing parameters

<table>
<thead>
<tr>
<th>Stage</th>
<th>Activity</th>
<th>Source (and type)</th>
<th>Influencing parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOL</td>
<td>Material production</td>
<td>Material processing facilities (CO₂)</td>
<td>Material type, component types and mass, recycling</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Refrigerant process plant (CO₂, R)</td>
<td>Refrigerant type, lifetime used mass, recycling/EOL incineration</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Refrigerant distribution (R)</td>
<td>Lifetime used mass, means of distribution</td>
</tr>
<tr>
<td>SOL</td>
<td>Components</td>
<td>Manufacture, assembly (CO₂)</td>
<td>Factory “efficiency”</td>
</tr>
<tr>
<td></td>
<td>Installation</td>
<td>Purging/leak testing (R)</td>
<td>Type of gas, Hose/ manifold size</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Charging (R)</td>
<td>Hose/ manifold size</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Construction/assembly (CO₂)</td>
<td>Equipment size, no. connections</td>
</tr>
<tr>
<td>SOL</td>
<td>Transport</td>
<td>Equipment movement (CO₂)</td>
<td>Mass of equipment</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Site visits (CO₂)</td>
<td>Size of installation</td>
</tr>
<tr>
<td>In-use</td>
<td>Operation</td>
<td>Compressor energy (CO₂)</td>
<td>Power station emission factor, component and refrigerant selection/design, (efficiency, pressure losses, heat transfer, application thermal mass, application temperature, ambient temperatures, compressor wear, exchanger fouling, system charge sensitivity, servicing/maintenance frequency</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Ancillary device energy (CO₂)</td>
<td>Power station emission factor, component selection, system cycling times</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Refrigerant leakage (R)</td>
<td>Component numbers, type, repair frequency</td>
</tr>
<tr>
<td>EOL</td>
<td>Servicing/maintenance</td>
<td>System venting (R)</td>
<td>Behaviour</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Refrigerant recovery (R)</td>
<td>Use of recovery machine, hoses/ manifold</td>
</tr>
<tr>
<td></td>
<td></td>
<td>System evacuation (R)</td>
<td>Oil charge, solubility</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Charging (R)</td>
<td>Hose/ manifold size</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Purging/flushing/leak test (R)</td>
<td>Type of gas, Hose/ manifold size</td>
</tr>
<tr>
<td>EOL</td>
<td>Transport</td>
<td>Site visits (CO₂)</td>
<td>Distance, visit frequency</td>
</tr>
<tr>
<td></td>
<td>Decommissioning</td>
<td>System venting (R)</td>
<td>Behaviour</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Refrigerant recovery (R)</td>
<td>Use of recovery machine, hoses/ manifold</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Dismantling (CO₂, R)</td>
<td>Oil charge, solubility</td>
</tr>
<tr>
<td></td>
<td>Transport</td>
<td>Equipment movement (CO₂)</td>
<td>Mass of equipment</td>
</tr>
<tr>
<td></td>
<td>Incineration</td>
<td>Plant emissions (CO₂, R)</td>
<td>Mass of refrigerant, mass of oil, incineration efficiency</td>
</tr>
<tr>
<td></td>
<td>Disposal</td>
<td>Material/refrigerant production</td>
<td>Recycling</td>
</tr>
</tbody>
</table>

124
PART 2: DEVELOPING A NEW SYSTEM BALANCING TECHNIQUE FOR MIXTURE REFRIGERANTS
NOMENCLATURE

Letters

\(a\) a constant (-)

\(A\) area (m\(^2\))

\(A_{TEV}\) area of TEV orifice (m\(^2\))

\(A_{tot.des}\) total design surface area of heat exchanger (m\(^2\))

\(b\) a constant

\(B\) basic rating (kW K\(^{-1}\))

\(B_{cond}\) condenser basic rating (kW K\(^{-1}\))

\(b_{dsh}\) basic rating for desuperheating region (kW K\(^{-1}\))

\(B_{evap}\) evaporator basic rating (kW K\(^{-1}\))

\(Bo\) boiling number (-)

\(b_{sc}\) basic rating for subcooling region (kW K\(^{-1}\))

\(b_{sh}\) basic rating for superheat region (kW K\(^{-1}\))

\(c\) a constant

\(C\) capacity rate (kW K\(^{-1}\))

\(c_d\) discharge coefficient (-)

\(\dot{C}_{HTF}\) HTF capacity rate (kW K\(^{-1}\))

\(\dot{C}_{max}\) maximum capacity rate (kW K\(^{-1}\))

\(\dot{C}_{min}\) minimum capacity rate (kW K\(^{-1}\))

\(COP\) coefficient of performance (-)

\(C_p\) specific heat at constant pressure (kJ kg\(^{-1}\) K\(^{-1}\))

\(C_{P_{HTF}}\) specific heat of HTF (kJ kg\(^{-1}\) K\(^{-1}\))

\(C_p^r\) refrigerant apparent specific heat (kJ kg\(^{-1}\) K\(^{-1}\))

\(\dot{C}_r\) refrigerant apparent capacity rate (kW K\(^{-1}\))

\(c_{TEV}\) TEV constant (-)

\(C_v\) specific heat at constant volume (kJ kg\(^{-1}\) K\(^{-1}\))

\(D\) tube diameter (m)

\(D_o\) diameter of outside tube (m)
$F_1$ function to account for mixture effects in condensation (-)

$F_2$ function to account for mixture effects in condensation (-)

$F_{JR}$ parameter for mixture heat transfer coefficient (-)

$F_{me}$ mixture correction factor for heat transfer coefficient (-)

$Fr$ Froude number (-)

$F_{tp}$ two-phase enhancement factor (-)

$F_{mn}$ mixture correction factor for heat transfer coefficient (-)

$F\{x\}_1$ condensing heat transfer vapour quality factor (-)

$F\{x\}_2$ condensing heat transfer vapour quality factor (-)

$g$ gravitational constant (m s$^{-2}$)

$G$ mass flux (kg s$^{-1}$ m$^{-2}$)

$G_v$ mass flux of vapour (kg s$^{-1}$ m$^{-2}$)

$h$ heat transfer coefficient (W m$^{-2}$ K$^{-1}$)

$H$ enthalpy (kJ kg$^{-1}$ K$^{-1}$)

$H_{disch}$ enthalpy at compressor discharge (kJ kg$^{-1}$ K$^{-1}$)

$h_{HTF}$ HTF heat transfer coefficient (W m$^{-2}$ K$^{-1}$)

$h_{l-v}$ heat transfer coefficient between the liquid interface and the liquid (W m$^{-2}$ K$^{-1}$)

$h_{l-v}$ heat transfer coefficient between the liquid interface and the vapour (W m$^{-2}$ K$^{-1}$)

$h_{ph}$ pool-boiling heat transfer coefficient (W m$^{-2}$ K$^{-1}$)

$h_r$ refrigerant heat transfer coefficient (W m$^{-2}$ K$^{-1}$)

$H_{suc}$ enthalpy at compressor suction (kJ kg$^{-1}$ K$^{-1}$)

$k$ thermal conductivity (W m$^{-1}$ K$^{-1}$)

$\dot{m}$ mass flow rate (kg s$^{-1}$)

$\dot{m}_{HTF}$ HTF mass flow rate (kg s$^{-1}$)

$\dot{m}_r$ refrigerant mass flow rate (kg s$^{-1}$)
$n$  an index (-)
$N$  number (-)
$N_{crt}$  number of tubes or circuits (-)
$n_G$  temperature glide index (-)
$N_H$  number of elements based on enthalpy (-)
$N_{TU}$  number of transfer units (-)
$Nu$  Nusselt number (-)
$N_x$  number of elements based on vapour quality (-)
$p$  pressure (Pa)
$\Delta p$  pressure difference/drop (Pa)
$p_0$  saturation pressure at 0°C (Pa)
$p_{atm}$  atmospheric pressure (Pa)
$p_{cr,mvc}$  critical pressure of more volatile component (Pa)
$p_{me}$  ideal mean effective pressure (Pa)
$p_{mf}$  pressure of mimicking fluid (Pa)
$Pr$  Prandtl number (-)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$q$</td>
<td>heat flux (kW m$^{-2}$)</td>
</tr>
<tr>
<td>$Q$</td>
<td>rate of heat flow, capacity (kW)</td>
</tr>
<tr>
<td>$Q_{e,\text{comp}}$</td>
<td>compressor condensing capacity (kW)</td>
</tr>
<tr>
<td>$Q_{e,\text{cond}}$</td>
<td>condenser capacity (kW)</td>
</tr>
<tr>
<td>$Q'_{e,\text{cond}}$</td>
<td>condenser capacity at standard rating conditions (kW)</td>
</tr>
<tr>
<td>$Q_{e,c/u}$</td>
<td>condensing unit condensing capacity (kW)</td>
</tr>
<tr>
<td>$Q_{e,e/u}$</td>
<td>evaporating unit condensing capacity (kW)</td>
</tr>
<tr>
<td>$Q_{dsh}$</td>
<td>capacity of desuperheating region (kW)</td>
</tr>
<tr>
<td>$Q_{e,\text{comp}}$</td>
<td>compressor evaporating capacity (kW)</td>
</tr>
<tr>
<td>$Q_{e,c/u}$</td>
<td>condensing unit evaporating capacity (kW)</td>
</tr>
<tr>
<td>$Q_{e,e/u}$</td>
<td>evaporating unit evaporating capacity (kW)</td>
</tr>
<tr>
<td>$Q_{e,\text{evap}}$</td>
<td>evaporator capacity (kW)</td>
</tr>
<tr>
<td>$Q'_{e,\text{evap}}$</td>
<td>evaporator capacity at standard rating conditions (kW)</td>
</tr>
<tr>
<td>$Q_{e,\text{TEV}}$</td>
<td>TEV evaporating capacity (kW)</td>
</tr>
<tr>
<td>$Q_{sc}$</td>
<td>capacity of subcooling region (kW)</td>
</tr>
<tr>
<td>$Q_{sh}$</td>
<td>capacity of superheating region (kW)</td>
</tr>
<tr>
<td>$Q_{sp,l}$</td>
<td>capacity of single-phase liquid region (kW)</td>
</tr>
<tr>
<td>$Q_{sp,v}$</td>
<td>capacity of single-phase vapour region (kW)</td>
</tr>
<tr>
<td>$Q_{tot}$</td>
<td>total heat exchanger capacity (kW)</td>
</tr>
<tr>
<td>$Q_{tp}$</td>
<td>capacity of two-phase region (kW)</td>
</tr>
<tr>
<td>$R_{C}$</td>
<td>capacity rate ratio (-)</td>
</tr>
<tr>
<td>$R_{cyl}$</td>
<td>clearance volume ratio (-)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number (-)</td>
</tr>
<tr>
<td>$R_{\text{surf}}$</td>
<td>ratio of outside (HTF) to inside (refrigerant) surface area (-)</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>( T )</td>
<td>temperature (°C)</td>
</tr>
<tr>
<td>( T_{bp} )</td>
<td>bubble-point temperature (°C)</td>
</tr>
<tr>
<td>( T_c )</td>
<td>condensing temperature (°C)</td>
</tr>
<tr>
<td>( \Delta T_{c(m)} )</td>
<td>condenser approach temperature difference (K)</td>
</tr>
<tr>
<td>( T_{c,mp} )</td>
<td>condensing mid-point temperature (°C)</td>
</tr>
<tr>
<td>( T_{cold} )</td>
<td>temperature of cold fluid stream (°C)</td>
</tr>
<tr>
<td>( T_{disch} )</td>
<td>discharge temperature (°C)</td>
</tr>
<tr>
<td>( T_{dp} )</td>
<td>dew-point temperature (°C)</td>
</tr>
<tr>
<td>( \Delta T_{dis} )</td>
<td>condenser desuperheat (K)</td>
</tr>
<tr>
<td>( T_e )</td>
<td>evaporating temperature (°C)</td>
</tr>
<tr>
<td>( \Delta T_{e(m)} )</td>
<td>evaporator approach temperature difference (K)</td>
</tr>
<tr>
<td>( T_{e,mp} )</td>
<td>evaporating mid-point temperature (°C)</td>
</tr>
<tr>
<td>( \Delta T_G )</td>
<td>temperature glide (K)</td>
</tr>
<tr>
<td>( \Delta T_G' )</td>
<td>adjusted temperature glide (K)</td>
</tr>
<tr>
<td>( T_{hot} )</td>
<td>temperature of hot fluid stream (°C)</td>
</tr>
<tr>
<td>( T_{HTF} )</td>
<td>HTF temperature (°C)</td>
</tr>
<tr>
<td>( \Delta T_{(m)} )</td>
<td>approach temperature difference (K)</td>
</tr>
<tr>
<td>( \Delta T_{(m)}' )</td>
<td>approach temperature difference at standard reference conditions (K)</td>
</tr>
<tr>
<td>( \Delta T_{min(m)} )</td>
<td>minimum approach temperature difference (K)</td>
</tr>
<tr>
<td>( \Delta T_{lp} )</td>
<td>saturated temperature difference due to pressure loss (K)</td>
</tr>
<tr>
<td>( T_r )</td>
<td>refrigerant temperature (°C)</td>
</tr>
<tr>
<td>( T_s )</td>
<td>temperature at saturation line (°C)</td>
</tr>
<tr>
<td>( \Delta T_{sc} )</td>
<td>condenser subcooling (K)</td>
</tr>
<tr>
<td>( \Delta T_{sh} )</td>
<td>evaporator superheat (K)</td>
</tr>
<tr>
<td>( T_{sh(out)} )</td>
<td>evaporator outlet superheat temperature (°C)</td>
</tr>
<tr>
<td>( \Delta T_{suct} )</td>
<td>suction superheat (K)</td>
</tr>
<tr>
<td>( T_{TEV(in)} )</td>
<td>refrigerant temperature into TEV (°C)</td>
</tr>
<tr>
<td>( T_{wall} )</td>
<td>temperature of tube wall surface (°C)</td>
</tr>
</tbody>
</table>
\( u \) velocity (m s\(^{-1}\))
\( U \) overall heat transfer coefficient (W m\(^{-2}\) K\(^{-1}\))
\( V_{sw} \) compressor swept volume flow rate (m\(^3\) s\(^{-1}\))
\( W_{comp} \) compressor power (kW)
\( W_{fr} \) frictional power (kW)
\( x \) vapour quality (-)
\( x_{e(in)} \) evaporator inlet vapour quality (-)
\( X_n \) Lockhart-Martinelli parameter (-)
\( Y_l \) mole fraction of the more volatile component in liquid (-)
\( Y_v \) mole fraction of the more volatile component in vapour (-)
\( z \) distance, length (m)
\( z_{crit} \) length of single refrigerant circuit (m)

**Greek**

\( \beta \) contact angle (°)
\( \varepsilon \) effectiveness (-)
\( \phi_{pp} \) two-phase multiplier (-)
\( \gamma \) ratio of specific heats (-)
\( \eta_{vol} \) volumetric efficiency (-)
\( \lambda \) friction factor (-)
\( \lambda_l \) liquid-only friction factor (-)
\( \lambda_v \) vapour-only friction factor (-)
\( \mu \) viscosity (Pa s)
\( M \) molar mass (kmol kg\(^{-1}\))
\( \tau \) temperature parameter (K\(^{-1}\))
\( \vartheta \) void-fraction parameter defined in text (-)
\( \rho \) density (kg m\(^{-3}\))
\( \sigma \) surface tension (N m\(^{-1}\))
\( \nu \) void fraction (-)
\( \zeta \) correction factor (-)
Subscripts

1 condition at evaporator outlet/compressor inlet, or numerical reference
2 condition at compressor outlet/condenser inlet, or numerical reference
3 condition at condenser outlet/TEV inlet, or numerical reference
4 condition at TEV outlet/evaporator inlet, or numerical reference

bp bubble-point
c condensing
cond condenser
comp compressor
c/u condensing unit
dp dew-point condition
dsh desuperheat
e evaporating
evap evaporator
e/u evaporating unit
fr frictional
g gravitational
HTF heat transfer fluid
(in) inlet condition
l liquid
M momentum
max maximum
min minimum
(out) outlet condition
s saturation
sc subcooling
sh superheat
suct suction
TEV TEV
tp two-phase
v vapour
Y mole fraction of the more volatile component
SECTION 1: INTRODUCTION

1.1 New refrigerants

With the phase-out of CFCs and HCFCs a large number of substances have been proposed as alternatives, including HFCs, PFCs and HCAs (UNEP, 2002). Very few of these new substances possess thermodynamic properties that match the prohibited fluids. Therefore, refrigerant producers have mixed these substances to obtain likened properties, which are intended to provide similar system operating characteristics.¹⁰

There are no particular technical reasons for developing refrigerant blends, except for ease of application and use in existing systems. Historically, refrigerant mixtures have been used in niche applications and experimental investigations for many years (e.g., Hasleden and Klimek, 1957) but it is only since the beginning of the phase-out of CFCs that zeotropic blends have become commonplace commercially. It is seen that over three-quarters of commercially available refrigerants (UNEP, 2002 – Refrigerants Annex) are non-azeotropic refrigerant mixtures. This fact highlights the importance of studies in this area.

Zeotropic mixtures differ from pure and azeotropic refrigerants in that they exhibit a “temperature glide” or temperature variation when changing phase. This temperature glide is caused by the components of the mixture possessing different vapour pressures. When the mixture begins to change phase, there is preferential boiling (or condensation) of one of the components with the lower (or higher) vapour pressure. The remaining mixture increases (or reduces) in temperature until phase-change is complete, and this temperature glide typically ranges from 0 K to 12 K at normal operation temperatures for most commercially available zeotropes. The effect of this temperature glide has added a new consideration to the design and operation of refrigeration components and systems.

1.2 Overview of refrigeration component-system chain

The impact of refrigerant temperature glide on refrigeration components and combination of components can be observed from a system’s conceptual beginning to its physical operation, termed the component-system chain. Figure 34 provides an overview of the stages under

¹⁰ A mixture, or blend, of two or more refrigerants can be categorised into those that exhibit a temperature glide (zeotrope), and those that do not (azeotrope). A zeotrope is defined as: a blend composed of two or more refrigerants whose equilibrium vapour and liquid phase compositions are not the same. An azeotrope composes of two or more refrigerants whose equilibrium vapour and liquid phase compositions are the same at a given pressure (ISO 817).
consideration. There are two distinct phases within this chain, which are generally the responsibility of the product manufacturer, and the utilisation engineers respectively. It is necessary to ensure a proper flow of product information throughout the chain, and that a robust mechanism for this transfer of information is in place. Without this flow of information, the final operation of a system is unlikely to meet the specifications of the initial design requirements.

The various stages of the component-system chain are described in Figure 34:

- **Purpose.** A product manufacturer identifies a need for a product. No specific technical considerations with respect to temperature glide are needed here.

- **Design.** The design stage is where the purpose of the component is assessed and expressed as a visualisation of how the purpose will be achieved. Since the design of a component will be based on specified operating conditions (such as temperature), it is evident that the component design stage relates to refrigerant temperature glide.

- **Construction.** This is simply a fabrication of the design where no consideration to temperature glide is necessary.
- **Performance testing/rating.** The main purpose of testing is for rating. For components in common use published industry standards usually exist that describe methods for performance testing. Operating temperatures are implicit in most component performance assessments and therefore the impacts of temperature glide should also be noted here.

- **Catalogue data.** Test results and standardised models are applied to generate information that can be used to select or assess a component's ability to perform in a particular environment. The information normally comes in the form of catalogue data. Interpretation of the catalogue data is necessary and therefore the rating conditions must work from a particular basis, such as a reference temperature, hence the implication of temperature glide at this stage.

- **Selection.** An utilisation engineer applies the component to a design and eventually the installation of a system. Selection of a component, chosen from catalogue data to suit the system operating conditions, is normally based on a number of parameters, including refrigerant temperature, implying that temperature glide is a consideration here.

- **Balancing.** Following a selection of various components that form a system an evaluation technique is required to determine the system characteristics. From the balancing process it is determined whether the choice of components was suitable to achieve the desired overall system operating conditions, meeting certain design specifications. Both the balancing and selection stages are integrated as it may form an iterative process of re-selection. Both stages are therefore dependent upon consistent design parameters, which include reference temperatures, implicating the temperature glide.

- **Installation.** This stage is the physical linking of the various components with the system according to the design specification and therefore there is no implication for temperature glide.

- **Commissioning.** Once a system has been installed and charged with refrigerant, it will be set into operating mode and controls, etc., adjusted accordingly. In order to achieve this, measurements must be made which include temperature measurement and by implication, temperature glide. However, since there is no numerical procedure, this stage can be neglected.

The main focus of this study is the transfer of data throughout the component-system chain, particularly in performance testing, catalogue data, selection and balancing stages, and the characteristics of zeotropic refrigerants that influence these.
1.3 Literature review

Information on conventional methods for performance testing, catalogue rating, component selection and balancing technique, normally found in industry codes and standards, should be well understood. Since the key issue is the effect that zeotropic refrigerants and the associated temperature glide have on these methodologies, a general understanding of the behaviour of these refrigerants is also necessary.

1.3.1 Performance testing, rating, selection and balancing

Research literature on component performance, testing, rating and selection is rare since traditional practices and standards are used, and until recently it has not been necessary to change them. Databases from standardisation organisations (such as BSI) were searched for standards relating to expansion devices, compressors, condensers and evaporators. Selection of components generally makes use of catalogue data that has been developed according to the rating procedures. A review of selected manufacturers literature provided information on the parameters required to select specific components. For example, nominal air cooled evaporator capacities are based on entering air temperature and the refrigerant saturation temperature (usually referred as the approach temperature difference, $\Delta T_{(in)}$) and correction factors are given for converting standard dry capacities into nominal capacities for “real” or wet conditions.11

Compressor

ISO 917 details a method for conducting measurements on single stage positive displacement refrigerant compressors, although it could be used similarly for most other types of refrigeration compressors. Several different test methods are offered by the standard, of which two must be used simultaneously to determine capacity, and the results averaged. ASHRAE 23 specified similar test methods for reciprocating compressors. The method for presenting compressor rating data is from ISO 9309. Data are presented for conditions according to standard superheat values. For each compressor, the data should be fitted to a polynomial equation, to interpolate for fixed increments in tables.

A selection of compressor catalogues (Prestcold, undated) provided data in both tabular and graphical formats. For each model, it’s capacity, motor power and motor current was provided

---

11 The reasons for testing units under dry conditions and then applying the correction factors is so that steady-state conditions can be achieved for testing purposes. Whilst water vapour is condensing or subliming on exchanger surfaces a transient process is occurring that does not allow for a constant measurement.
according to evaporating and condensing temperature. There was a separate set of data provided for each refrigerant, and based on specific subcooling and suction temperatures. Correction factors and charts were offered for adjusting compressor capacity rating in the case of non-standard subcooling and superheat settings. Selection is by matching the estimated evaporating and condensing temperatures of the system with the cooling capacity.

**Expansion device**

Despite the variety of expansion devices available, only one standard was found (ASHRAE 17), for testing and rating of thermostatic expansion valves (TEVs) which require two test arrangements. The first is to determine flow rates for static superheat conditions, where inlet and outlet pressures are applied to a fixed orifice size corresponding to various liquid temperatures. The second test is for the measurement of mass flow against the degree of orifice opening. Capacity rating is obtained from mass flow and refrigerant enthalpies based on certain rating temperatures. TEV catalogue data (Danfoss, undated) showed that valves are selected based on tabulated capacities defined by nominal evaporating, condensing and liquid entering temperatures for various pressure drops across the valve orifice.

**Condenser**

There are standards for performance testing of a variety of different condensers, for example, EN 1117 describes test procedures for liquid-cooled condensers. It specifies a fixed set of test conditions under which temperature and flow rate measurements are made for both refrigerant and the heat transfer fluid (HTF). Using this data, a standard capacity is determined from which values for other non-standard conditions could be obtained.

Catalogue data for various refrigerant-air finned-tube condensers was reviewed (Searle, undated). Condenser selection was based on tabulated capacities for dry conditions, which are dependent upon a number of variables including the approach temperature difference. The approach temperature difference used for selection purposes is the difference between the condensing temperature and ambient temperature. The actual capacity for non-standard conditions was obtained by applying correction factors to the tabulated capacity.

**Evaporator**

Evaporator standards are similar in principle to condenser standards. EN 1118 describes performance testing and rating methods for various types of liquid coolers. Again, fixed test conditions are specified for measurements leading to calculation of the standard capacity. Since
the superheat is difficult to maintain at a prescribed value, tests are made across a number of
superheat values (whilst maintaining other specified conditions) and the data interpolated to
determine evaporator capacity at the specified superheat. Non-standard capacities were found
from a general rating equation. Additionally, ASHRAE-25 and EN 328 provide test methods for
air-cooling evaporators that adopt the same basis as EN 1118.

Evaporator selection data (Searle, undated) for horizontal refrigerant-air finned-tube type were
reviewed. The tabulated rated capacity was based on a single fixed evaporating temperature,
adjusted for humid operating conditions, and the actual capacity was found using correction
factors. Correction factors were supplied, to account for evaporating temperature, refrigerant
type and temperature difference (also applicable to wet coils).

Further rating methods
Whilst evaporator and condenser standards provide a basic methodology for estimating heat
exchanger capacity to conditions other than the test conditions, it only applied to variations of
\( \Delta T_{\text{in}} \). Hence, effects of different construction and operational parameters such as fin spacing,
air flow rate, refrigerant type and air humidity on capacities were not accounted for. Since
manufacturers generally offer a wide range of components, performance testing for all possible
configurations would be a costly exercise. Manufacturers therefore tend to produce their own
semi-empirical models to determine component performance across a range of geometries and
operating conditions (Atkins, 1999). Although the details of these models are generally
confidential, some published literature does exist on the development of such rating methods.

Stoneham et al (1980) proposed a number of generalised correlations to approximate evaporating
heat transfer coefficient for various refrigerants under non-standard conditions where only a few
variables are needed, although the errors are significant through using such a generalised
approach. These formulae were incorporated into a model for extending catalogue data by Trott
catalogue cooling coil data through correlating simple exchanger configurations and fluid
properties. It allowed for more accurate extrapolation of coil performance data to other operating
conditions. Studies for condensers is sparse, although a paper by Holladay (1984) devised a
method of rating combined desuperheater/condenser capacity as a function of refrigerant and
water temperatures, and water flow rates. Kayansayan (1994) considered the effect of airside coil
configuration and correlated effects of various fin geometries on heat exchanger performance.
Following commercialisation of mixture refrigerants, some discussion on use of mixtures in certain components, typically the compressor has been published. Murphy et al (1998) discussed the potential errors when compressors are selected according to bubble, dew or “mid-point” temperatures. Errors manifest when a compressor is selected using a reference temperature different to that for which it was rated. Hundy (2000) and Hundy and Vittal (2000) discussed the same problem, but also offer a correction method to adjust the capacity for a non-standard reference temperature. This problem was also addressed in a practical sense (Jansen and Bekes, 2000) where performance of compressors with a pure and mixture refrigerant was tested in a calorimeter. When performance of the pure and mixture refrigerant was compared using dew-, bubble- and mid-point temperatures, the rated capacity could be higher or lower than that of the pure refrigerant, depending on the conditions.

A theoretical study of Bobbo et al (1995) considered the choice of reference temperatures for all system components and proposed that – at least for comparison purposes – the thermodynamic mean temperature (defined as enthalpy divided by entropy) be used. The only discussion on heat exchanger rating in respect to use of zeotropes was a study by Ntalles (1997). The conventional basic rating method was applied, but an ‘excess’ capacity introduced to account for the differences resulting from the temperature glide. All rating values were based on the evaporator outlet temperature.

Balancing
The conventional component balancing method is detailed in several textbooks; ASHRAE (1998), Dossatt (1991), Trane (1988), and Stoecker and Jones (1982). The method described was identical in each, and none specified whether they apply to pure only or both pure and mixture refrigerants. Since most of these sources were published prior to the commercialisation of mixture refrigerants, it was assumed that they were intended for pure refrigerants only. Gosney and Suen (2001) also detailed the conventional balancing method, and offered solutions to problems involving system pressure losses. Discussion with Suen (2000) confirmed the methods (Gosney and Suen, 2001) applied only to refrigerants without temperature glide. Only two further references specifically covered balancing. One (Page, 1989) detailed the conventional method and subsequently described how it could be applied to computer programmes and spreadsheets. Trott (1981) did the same, although included formulas to calculate component performance at non-standard catalogue conditions. In all of these sources, balancing was based on a single saturation “evaporating” and/or “condensing” temperature.
1.3.2 Implications of zeotropic refrigerants

With the search for alternative refrigerants, a large amount of studies related to mixture refrigerants has been published. Whilst a minotia of publications addresses the topics under consideration, findings on related subjects may provide additional insight into the behaviour of zeotropic refrigerants that could impact on their effect on various stages in the component-system chain. Therefore studies on heat exchanger analysis, heat transfer characteristics, composition shift and overall system performance are of interest.

Analysis of heat exchangers with mixture refrigerants

The two main approaches for heat exchanger analysis are the logarithmic mean temperature difference (InΔT), and the Number of Transfer Units-Effectiveness (NTU-ε) method. Both approaches are based on several assumptions that are not necessarily valid when a phase-change fluid is used. One is constant overall heat transfer coefficient \( (U) \) and specific heat capacity \( (C_p) \) throughout the fluid path. Liu (1996) revised formulations for InΔT based on these variations. Additionally, NTU-ε method assumes that phase-change fluids possess infinite capacity rates, and is not valid for zeotropes. Granyrd and Conklin (1990) derived new formulae for NTU-ε for phase-change with finite capacity rate, and later Conklin and Vineyard (1991) utilised the revised relations to assess measurements on heat exchangers with high temperature glide. The revised formulae showed good agreement with measurements when pressure loss in the heat exchanger was low. These studies imply that use of either InΔT or NTU-ε conventional methods will introduce error to analysis with zeotropes when applied to an entire heat exchanger, unless they are suitably modified.

Much recent work on mixture refrigerants focussed on their positive or negative impacts on system performance. One aspect in particular, is the potential optimisation of a system – through improved heat exchanger performance – achieved with “glide-matching”. Mulroy et al (1994) investigated this issue, and on the basis of a linearised temperature glide (and linear temperature profile of a single-phase heat transfer fluid) found entropy generation could be reduced (i.e., by obtaining smaller temperature differences). Measurements from a system using a tertiary mixture (blended to linearise temperature glide) were presented to support the argument. Venkatarathnam et al (1996) discussed the effect of mixture properties and specific glide characteristics in relation to temperature profiles along heat exchangers. Later, Venkatarathnam and Murthy (1999) considered the effect of changing refrigerant composition to show that using wide-glide binary mixtures produced better glide matching.
Heat transfer in mixture refrigerants

It is known that the presence of two or more refrigerant components has implications on heat transfer coefficient \((h)\), and hence exchanger performance. Earlier, Bivens and Yokozeki (1993) explained the theoretical reduction of \(h\), and attributed it to resistance of mass transfer of each species as the mixture changes phase. A generalised formula was proposed for re-calculating \(h\) of any mixture. Recently, Hewitt (1999) provided an overview of research progress into \(h\), including for mixture refrigerants. Based on review studies by Wang and Chato (1995), Shin et al (1997) and Stephan (1995), the consensus for the most accurate correlation for evaporation was that of Jung and Radermacher (1993a and 1993b). Similarly, review studies on condensation were carried out by Gaibel et al (1994) and Kenny et al (1994). However, a more recently correlation by Shao and Granryd (1998, 2000) showed more favourable accuracy.

Composition shift

Other than the presence of a temperature glide and an effect on \(h\), another characteristic of zeotropic refrigerants is separation of species within the parts of the system where refrigerant exists in two-phases. This results in localised “hold-up” of certain species, thereby causing the circulating composition to differ from the refrigerants' original charged composition resulting in a possible departure from the intended operation. Measurements of variations of refrigerant composition around a system were reported by Chen and Kruse (1995) and Sumida et al (1995), and generalised models were developed in order to predict the composition shift. Chen and Kruse (1995) found the accuracy of the model to be highly dependent upon the choice of void fraction model, and the Hughmark correlation was shown to provide results that matched measurements best.

1.4 Formulation of objectives

Initially, the issue of increasing application of zeotropic refrigerants was presented, as was an explanation of the flow of performance data through the component-system chain. In light of the fundamental differences between the properties of conventionally used pure refrigerants, and the recently commercialised zeotropic refrigerants, possible incompatibilities within the current methods may exist. The review of the current practices within the component-system chain and the behaviour particular to zeotropic refrigerants indicate the possibility of incompatibilities, and therefore it is necessary to define an approach to examine and resolve these issues.
1.4.1 Summary of findings

There are several key findings drawn from the literature review, as follows:

- **Component rating and catalogue data.** Presently no industry standards or manufacturers information relate specifically to the characteristics of mixture refrigerants. A small number of studies present options for converting component capacity data from one reference temperature to another, where a glide is present. Whilst most standards require data to be formatted in a specific manner, catalogue information appeared to be inconsistent in terms of reference conditions. Use of correction factors appeared to be the most popular format for data conversion. These observations highlight the need for a standardised approach, especially with the introduction of refrigerants with temperature glide.

- **System balancing.** This conventional method is based on pure refrigerants only and accounts for a limited number of parameters, and neglecting others such as pressure loss effects. Considerable manipulation of data may be required to convert data from published reference temperature to one consistent with other component data, so that system balancing is possible. Evidently, there is a demand to improve the balancing technique and to ensure the supply of consistent data for this purpose.

- **Implications of zeotropic refrigerants.** The literature contained findings relating to a variety of aspects associated with zeotropes. The heat exchanger analysis techniques such as NTU-ε were shown to provide accurate results when correctly revised for zeotropic refrigerants. Different mixtures exhibit a range of glide profiles (variation of saturation temperature with vapour quality), and this can have an impact on cycle efficiency. To optimise heat exchanger performance, glide matching techniques are used to reduce approach temperature difference. The studies on evaporation and condensing heat transfer coefficients showed degradation with mixtures and provided information on favoured correlations for use in heat exchanger models. Refrigerant composition shift was shown to occur in systems particularly where mixtures have a large glide.

Little exploration of the influence of temperature glide on testing, rating and selection of components was found. As such the understanding and recent developments in peripheral subjects can provide understanding to assist with the current investigation. The review of industry practices for rating and catalogue information provided a base from where to start developing techniques for accounting for the behaviour of zeotropic refrigerants.
1.4.2 Aims and objectives

This study links the two themes covered previously: zeotropic refrigerants and their integration into the component-system chain stages. Presently, current methods employed within the component-system chain are primarily based on the use of pure refrigerants. Transition from a pure to zeotropic refrigerants calls for revision of the current methods employed with respect to the different behaviour exhibited by zeotropes.

The examination of the procedures within the various design stages found that the flow of information between stages is generally reliant on standard reference conditions for development and interpretation of performance data. Since the difference between pure and mixture refrigerants are primarily manifest as a temperature glide, the validity of currently defined reference conditions is questioned. The stages within the component-system chain that rely on choice of reference condition are: component performance testing/rating, catalogue data presentation and selection, and system balancing.

Since each component-system stage is linked to the next, intuitively there must be consistency in the reference conditions and interpretation methods applied throughout. Following an assessment of each stage, associated methods need to be modified if necessary. Adoption of improved methods should lead to improved accuracy in the design of a refrigerating system.

The general approach to developing modified or new methods requires the performance of the primary components to be evaluated, first in terms of pure refrigerants, and then re-evaluation by introduction of the properties of zeotropic refrigerants. Initially adopting a simplistic analysis should identify any differences in performance that are inconsistent with current rating, selection and balancing practices. This will indicate where further work would be required to revise the existing methods.

The general sequence of activities for investigating these issues is as follows:

- **Differences between pure and zeotropic refrigerants.** The basic differences between pure and zeotropic refrigerants are discussed, particularly in terms of inherent properties associated with rating, selection and balancing activities. In addition, the primary refrigeration system components are introduced, and general characterisation of their performance is described.
- **Further analysis of zeotropic refrigerants in heat exchangers.** The focus was on the condenser and evaporator and their behaviour with and without refrigerants with temperature glide. Initially a general analysis is used, and the findings examined further with detailed modelling. This will help to conclude how reference states should be handled and leads to new rating methods.

- **Assessment and modification of balancing technique.** On the basis of the changes identified for component rating methods, a modification to the existing balancing technique is proposed. The procedure of the revised balancing technique is illustrated in a worked example, which also acts to check whether the new methods achieve an improvement in accuracy.

Lastly, it is noted that the scope of the current study omits certain considerations to maintain focus. In particular, only direct expansion systems are addressed and interconnecting components such as pipework, valves, receivers and accumulators are ignored. In terms of mixtures, the occurrence of composition shift and any influences associated with oils is neglected. The components and systems under consideration are assumed to operate under steady conditions so transient operation and influences of any controls are not under consideration.
SECTION 2: IMPLICATIONS ON COMPONENT PERFORMANCE

2.1 Initial consideration of impact of glide on component performance

The first activity is to clarify the basic differences in the properties of pure and zeotropic refrigerants. The distinctive behaviour of zeotropic refrigerants is observed with respect to a simple refrigeration cycle, as shown on the pressure-enthalpy diagrams in Figure 35. The cycle is illustrated on both diagrams, where the refrigerant vapour enters the compressor at point (1), discharged at (2), cools, condenses, and flows into the expansion device at (3), and is expanded into the evaporator at (4), where it changes to vapour and reaches (1) again. This basic process occurs with both refrigerant types. However, by following the cycle with respect to the lines of constant temperature, a key difference is manifest. Whilst assuming isobaric flow from (2) to (3) and (4) to (1), the zeotrope in Figure 35(b) shows the condensation and evaporation process to occur with varying temperature, and this is termed the temperature glide ($\Delta T_g$).

![Figure 35: Pressure enthalpy diagram for (a) pure refrigerant, and (b) zeotropic refrigerant](image)

Assuming the two cycles operate at identical pressures, the pure refrigerant changes phase at constant temperature, or in other words, the saturated refrigerant temperature into the condenser ($T_{2,s}$) also known as the dew-point ($T_{c,dp}$), is equal to the saturated temperature at the outlet ($T_{3,s}$) also known as the bubble-point ($T_{c,bp}$). Similarly, with the evaporator the inlet temperature ($T_a$) equals the bubble-point ($T_{e,bp}$), which is the same as the saturated temperature at the evaporator outlet ($T_{1,s}$) or dew-point ($T_{e,dp}$). Conversely, the corresponding points for the zeotrope show a higher inlet temperature at the condenser compared to the outlet (i.e., $T_{c,dp} > T_{c,bp}$), and a lower inlet evaporator temperature than its outlet (i.e., $T_a < T_{e,dp}$). Since for a pure
refrigerant $T_{dp} = T_{bp}$, the subscripts $dp$ and $bp$ are neglected and a single saturation temperature is normally referred to.

Therefore, defining the performance of a component according to a single reference temperature is appropriate for a pure refrigerant, but inexact for a zeotropic refrigerant without further definition of the reference temperature. For the four main system components the performance measure of interest are cooling and/or heating capacity. Therefore, it is important to identify how component capacity (or definition of capacity) is affected by selection of reference temperature, and to identify a means of choosing the proper reference temperature to apply consistently. With regards to the condenser and evaporator, both are considered to be similar in operation and therefore it was chosen to analyse them together in the following sections.

2.2 Compressor

ISO 917 details methods for measuring performance and ISO 9309 suggests how performance data are fitted to a non-functional polynomial equation to assist with generation of catalogue data. The same formula is used for both evaporating capacity ($Q_{e,comp}$) (equation 76) and input power ($W_{comp}$) (equation 77).

$$Q_{e,comp} = c_1 + c_2 \cdot T_e + c_3 \cdot T_e^2 + c_4 \cdot T_e^3 + c_5 \cdot T_c + c_6 \cdot T_c^2 + c_7 \cdot T_c^3 + c_8 \cdot T_e^2 \cdot T_c^2 + c_9 \cdot T_e \cdot T_c^2 + c_{10} \cdot T_c^3$$

$$W_{comp} = c_1 + c_2 \cdot T_e + c_3 \cdot T_e^2 + c_4 \cdot T_e^3 + c_5 \cdot T_c + c_6 \cdot T_c^2 + c_7 \cdot T_c^3 + c_8 \cdot T_e^2 \cdot T_c^2 + c_9 \cdot T_e \cdot T_c^2 + c_{10} \cdot T_c^3$$

where $c_1 \ldots c_{10}$ are specific to $Q_{e,comp}$ and $W_{comp}$, and two reference temperatures, $T_e$ is saturated temperature at suction pressure, and $T_c$ is saturated temperature at discharge pressure.

Condensing capacity of the compressor ($Q_{c,comp}$) is normally assumed to be the sum of $Q_{e,comp}$ and $W_{comp}$ (although not specified in the standards). It is noted that for balancing purposes, $Q_{e,comp}$ is the variable of interest, not $W_{comp}$.

These equations were originated for pure refrigerants, so it is necessary to consider their application with a zeotrope, and the associated impact of $\Delta T_G$ on reference temperatures.

However, because equations (76) and (77) are simply curve-fits, the actual behaviour of the compressor is obscured. Therefore another approach was to examine the conventional functional
calculation method for compressor performance. Compressor performance is described by inlet temperature and pressure of the superheated vapour \((T_1\text{ and } p_1)\) and the outlet conditions \((T_2\text{ and } p_2)\). Based on these conditions, the compressor evaporating capacity \((Q_{e,\text{comp}})\) is given by equation (78).

\[
Q_{e,\text{comp}} = \dot{m}_r \cdot (H_1 - H_4)
\]  

(78)

where \(\dot{m}_r\) is mass flow of refrigerant \(H_1\) and \(H_4\) are specific enthalpy of refrigerant exiting and entering the evaporator respectively. Mass flow rate of the refrigerant is defined in equation (79).

\[
\dot{m}_r = \eta_{\text{vol}} \cdot \dot{V}_sw \cdot \rho_1
\]  

(79)

where \(\eta_{\text{vol}}\) is volumetric efficiency of compressor, \(\dot{V}_sw\) is swept volume rate of compressor, and \(\rho_1\) is density of superheated vapour entering the compressor. Compressor work, or the energy applied to the refrigerant whilst passing through the compressor \((W_{\text{comp}})\) is expressed in equation (80).

\[
W_{\text{comp}} = \dot{m}_r \cdot (H_2 - H_1)
\]  

(80)

where \(H_2\) is specific enthalpy of the superheated vapour exiting the compressor.

Since the capacity of the compressor is not reliant upon temperature differences to produce a particular capacity, then the \(\Delta T_G\) itself has no fundamental influence on its performance.

Similarly, according to the standards ISO 917 and ISO 9309, there are no parameters within the test or rating methods that apply to refrigerants in such a way that a \(\Delta T_G\) could affect the results. However, in terms of equations (78), (79) and (80), reference points must be specified for obtaining specific properties, implying that there must be some consideration as to which reference temperatures are used to define the performance. The same is borne out with the standards where the rating is provided as a function of saturated temperature. Thus a distinction should be made between \(T_{\text{evp}}\) and \(T_{\text{bp}}\). Since the compressor operation "sees" the evaporating and condensing dew-point temperatures, these points are considered to be the preferred choice as the reference. There should be no further considerations here with regards to temperatures and therefore \(\Delta T_G\).

2.3 Expansion device

A TEV was considered in this study since it is the most commonly used expansion device.
The rating procedure for TEVs given in ASHRAE 17 requires that refrigerant mass flow and orifice aperture be used to determine its evaporating capacity ($Q_{e,TEV}$) for any desired operating condition. Calculation of capacity is given by equation (81), essentially being the product of refrigerant mass flow rate and evaporating enthalpy difference.

$$Q_{e,TEV} = m_r \cdot (H_1 - H_a) \cdot \zeta$$

(81)

where $\zeta$ is a correction factor dependent upon the amount of subcooling and superheating.

Evidently, determination of capacity (from equation 81) does not rely directly on reference temperature. The valve capacity is approximate due to the self-modulating behaviour within a wide operating range. The effect of $\Delta T_G$ does not impact on the selection methods except in terms of the reference temperature. This is also the conclusion of Castle et al (1999) where this aspect was investigated experimentally. Based on discussion of the choice of reference temperature for the compressor, to be consistent the dew-point temperature corresponding to evaporating and condensing pressure should be used for valve selection.

2.4 Condenser and evaporator

Determination of condenser and evaporator capacity is currently obtained for reference conditions detailed in standards EN 1117 and EN 1118, respectively. For presentation of data, the capacity ($Q_{c,cond}$ or $Q_{e,evap}$) is re-calculated for other non-standard conditions according to some approach temperature difference (equations 82 and 83).

$$Q_{c,cond} = \frac{Q'_{c,cond}}{\Delta T_{c(in)}} \cdot \Delta T_{e(in)}$$

(82)

where $Q'_{c,cond}$ is capacity at standard rating conditions, $\Delta T_{c(in)}$ is the standard approach temperature difference between condenser HTF and refrigerant inlet temperatures corresponding to the saturated discharge pressure, and $\Delta T_{e(in)}$ is the non-standard temperature difference.

$$Q_{e,evap} = \frac{Q'_{e,evap}}{\Delta T_{e(in)}} \cdot \Delta T_{e(in)}$$

(83)

where $Q'_{e,evap}$ is the capacity at standard rating conditions, $\Delta T'_{e(in)}$ is the standard temperature difference between HTF and refrigerant inlet temperature corresponding to the saturated suction pressure, and $\Delta T_{e(in)}$ is the approach temperature difference at non-standard conditions.
For both heat exchangers, the capacity is a function of a temperature difference between HTF and the refrigerant. The existence of a $\Delta T_c$ implies an impact on both $\Delta T'_{\text{in}}$ and $\Delta T_{\text{(in)}}$, compared to the case of a pure refrigerant, and therefore equations (82) and (83) may no longer be applicable to zeotropes. Therefore, clarification of the implications of zeotropic refrigerants is required, and will be evaluated presently. In particular, the use of equations (82) and (83) will be investigated in more detail, with the goal of identifying potential modifications to existing rating and catalogue presentation methodologies.

![Graphical representation of (a) condenser, and (b) evaporator characteristic for pure refrigerant (where $B$ is constant)](image)

**Figure 36: Graphical representation of (a) condenser, and (b) evaporator characteristic for pure refrigerant (where $B$ is constant)**

### 2.4.1 Analysis of heat exchanger rating with pure refrigerants

Taking the general rating equations (82) and (83), the standard performance parameters ($Q' / \Delta T'_{\text{in}}$) are termed the basic rating ($B$). By rearranging equations (82) and (83), and substituting $Q' / \Delta T'_{\text{in}}$ for $B$, provides the general formula for the basic rating of the condenser (equation 84) and evaporator (equation 85) respectively.

$$B_{\text{cond}} = \frac{Q_{\text{c,cond}}}{T_{\text{r(in)}} - T_{\text{HTF(in)}}} \quad (84)$$

$$B_{\text{evap}} = \frac{Q_{\text{c,evap}}}{T_{\text{HTF(in)}} - T_{\text{r(in)}}} \quad (85)$$

where $T_{\text{HTF(in)}}$ and $T_{\text{r(in)}}$ (i.e., $T_{\text{e}}$ and $T_{\text{c}}$) are the inlet temperatures for the HTF and saturated refrigerant respectively. The graphical interpretation of equation (84) and (85) are in Figure 36, indicating the linear relationship between $Q$ and $\Delta T_{\text{(in)}}$. The use of the basic rating approach is

149
employed to generate catalogue data from performance tests and extrapolation of catalogue data to application design conditions.

Thus, the foundation for the basic rating should be understood if it is to be adapted for zeotropic refrigerants. If equations (84) and (85) are rearranged, they are found to have similarity with the NTU-\(e\) method (equation 86).

\[
Q = e \cdot \dot{C}_{min} \cdot (T_{hot(in)} - T_{cold(in)}) \tag{86}
\]

where the inlet temperatures of the hot \(T_{hot(in)}\) and cold \(T_{cold(in)}\) fluids relate directly to the inlet temperatures of the refrigerant and HTF, where \(B\) equates to the term \(e \times \dot{C}_{min}\), being the product of exchanger effectiveness and minimum capacity rate of the two fluid streams (equation 87).

\[
B = e \cdot \dot{C}_{min} \tag{87}
\]

For a single-phase fluid, capacity rate is the product of the mass flow rate \(\dot{m}\) and its specific heat \(C_p\) as equation (88).

\[
\dot{C} = \dot{m} \cdot C_p \tag{88}
\]

When equation (88) is applied to fluid undergoing changing phase \(C_p\) is infinite, thus \(\dot{C}\) is infinite. In equation (87), \(\dot{C}_{min}\) refers to the fluid stream that possesses the lower capacity rate of the two. Since \(\dot{C}\) is infinite for a pure refrigerant, \(\dot{C}_{min}\) must belong to the HTF \(\dot{C}_{HTF}\).

Further, for a heat exchanger with fixed HTF mass flow rate \(\dot{m}_{HTF}\), \(\dot{C}_{min}\) is constant. The exchanger effectiveness term in equation (87) is normally presented as a function of the non-dimensional number of transfer units \(N_{TU}\) (equation 89) and capacity rate ratio \(R_c\), being the ratio of \(\dot{C}_{min}\) and the maximum capacity rate \(\dot{C}_{max}\) (equation 90), i.e., that of the pure refrigerant.

\[
N_{TU} = \frac{U \cdot A}{\dot{C}_{min}} \tag{89}
\]

\[
R_c = \frac{\dot{C}_{min}}{\dot{C}_{max}} \tag{90}
\]

Consequently \(R_c \to 0\), which reduces the calculation of effectiveness to equation (91).
\[ \varepsilon = 1 - \exp(-N_{TU}) \] (91)

where \( A \) is the exchanger surface area and \( U \) is the overall heat transfer coefficient. If \( U \) is assumed to be constant regardless of \( T_{r(in)} \), \( N_{TU} \) will be constant and therefore so will \( \varepsilon \); thus it follows that the supposition of constant basic rating is valid.

A graphical representation of the heat exchanger capacity with saturated refrigerant temperature is shown in Figure 36. It is seen that reducing the temperature difference by shifting \( T_{r(in)} \) (i.e., \( T_e \) or \( T_c \)) reduces capacity until the point at which \( T_{HTF(in)} = T_{r(in)} \) and \( Q = 0 \). The relationship between capacity and approach temperatures may also be shown through the visualisation of refrigerant and HTF temperature profiles. Figure 37 illustrates the effect of a lowering condensing temperature in a counter-flow heat exchanger in three stages, with the area between the profiles representing the capacity. To the left is the case of a wide \( \Delta T_{(in)} \) where a large quantity of heat is transferred throughout the entire distance of the heat exchanger. Next, the central diagram shows a medium \( \Delta T_{(in)} \) where a reduced quantity of heat is transferred. Lastly, to the right, nearly equal approach temperatures give miniscule \( \Delta T_{(in)} \) where negligible heat is transferred. Whilst the shape of the refrigerant temperature profile remains constant over the exchanger distance, the shape of the HTF converges towards the refrigerant logarithmically (assuming \( U \), \( A \) and \( C_{pHTF} \) remain constant).

2.4.2 Analysis of heat exchanger for refrigerants with temperature glide

The NTU-e basis was used to investigate the influence of \( \Delta T_G \) on exchanger rating. In the first place the definition of \( B \) is considered, latterly the approach is used to address overall exchanger
behaviour. Each of the parameters in equation (86) was considered with respect to the properties of a zeotrope. An overview of equations (86) to (91) identifies the quantification of the refrigerants $C_p$ as the main difference between pure and zeotropic mixtures, since the existence of a refrigerant temperature rise/fall during phase-change implies a finite $C_p$. If $C_p$ is finite, then equation (88) suggests that $C_p'$ of a zeotrope is also finite, and therefore $\dot{C}_{min}'$ cannot automatically be assumed to belong to the HTF and $B$ becomes dependent on $\dot{C}_{min}'$.

Determination of $C_p$ was achieved through transposition of exchanger energy balances. Using the conventional formula for sensible heat gain of a fluid under steady flow gives equation (92).

$$Q = \dot{m} \cdot C_p \cdot (T_{(out)} - T_{(in)})$$ (92)

Equation (93) represents the conventional heat balance for steady flow undergoing phase-change.

$$Q = \dot{m} \cdot (H_{(out)} - H_{(in)})$$ (93)

Combining equations (92) and (93) provides a definition of the "apparent" specific heat of the refrigerant ($C_{p}'$) (equation 94).

$$C_{p}' = \frac{H_{(out)} - H_{(in)}}{T_{(out)} - T_{(in)}}$$ (94)

If equation (94) is applied to a pure refrigerant, then $T_{(out)} = T_{(in)}$ and therefore $C_{p}' = \infty$.

However, for a zeotrope undergoing phase change where $T_{(out)} - T_{(in)} = \Delta T_G$, then $C_{p}' < \infty$.

Further, substituting equation (94) into equation (88) provides the "apparent" capacity rate of the refrigerant ($\dot{C}_{p}'$) (equation 95).

$$\dot{C}_{p}' = \dot{m} \cdot \frac{H_{(out)} - H_{(in)}}{T_{(out)} - T_{(in)}}$$ (95)

Similarly, if equation (95) is applied to a pure refrigerant, $\dot{C}_{p}' = \infty$. It is noted that equations (94) and (95) only provide mean values, representing a linear glide with respect to enthalpy or quality change. Since most mixtures exhibit a non-linear glide, $C_{p}'$ and $\dot{C}_{p}'$ are more accurately defined with respect to local vapour quality; equations (96) and (97), respectively.

$$C_{p}' = \left(\frac{dH}{dT}\right)_x$$ (96)
\[ \dot{C}' = \dot{m}_{\text{f}} \left( \frac{dH}{dT} \right)_{x} \quad (97) \]

Equation (95) demonstrates that a greater $\Delta T_G$ gives low $Cp'$ and $\hat{C}'$. Now that $\dot{C}' < \infty$, the capacity rate ratio (equation 90) must be greater than zero, and therefore exchanger effectiveness is dependent on $R_{\hat{C}}$; thus equation (91) is no longer valid. With $R_{\hat{C}} > 0$, effectiveness is dependent on exchanger geometry, and for parallel-flow and counter-flow (for concentric tube), Incropera and DeWitt (1990) offer equations (98) and (99) respectively, and for cross-flow, equation (100).

\[ \varepsilon = \frac{1 - \exp \left( -N_{\text{TU}} \cdot (1 + R_{\hat{C}}) \right)}{1 + R_{\hat{C}}} \quad (98) \]

\[ \varepsilon = \frac{1 - \exp \left( -N_{\text{TU}} \cdot (1 - R_{\hat{C}}) \right)}{1 - R_{\hat{C}} \cdot \exp \left( -N_{\text{TU}} \cdot (1 - R_{\hat{C}}) \right)} \quad (99) \]

\[ \varepsilon = 1 - \exp \left( \frac{N_{\text{TU}}^{0.22}}{R_{\hat{C}}} \cdot \left( \exp \left( -R_{\hat{C}} \cdot N_{\text{TU}}^{0.72} \right) - 1 \right) \right) \quad (100) \]

![Figure 38: Effect of apparent capacity rate on effectiveness](image)

![Figure 39: Effect of apparent capacity rate on basic rating](image)

Interpretation of the revised NTU-$\varepsilon$ formulations revealed that with different $\Delta T_G$, $\hat{C}'$ can be higher or lower than $\hat{C}_{\text{HTF}}$, meaning that $\hat{C}_{\text{mn}}$ could be either $\hat{C}'$ or $\hat{C}_{\text{HTF}}$. More importantly, different $\Delta T_G$ will affect both $N_{\text{TU}}$ and $R_{\hat{C}}$, where a greater $\Delta T_G$ lessens $N_{\text{TU}}$ and $R_{\hat{C}}$. 

153
To illustrate the differences between a pure and zeotropic refrigerant, effectiveness is shown as a function of $C'$ for the three exchanger geometries (from equations 98, 99 and 100). The results are shown in Figure 38, where calculations assumed linear glide and constant $U$ and $C_{HTF} = 1$ kW K$^{-1}$. With a smaller $C'$ (i.e. larger $\Delta T_G$), $\varepsilon$ always tends towards unity, where the counter-flow geometry converges on $\varepsilon = 1$ first and the parallel has $\varepsilon < 1$ until $C' \to 0$ kW K$^{-1}$.

Introducing $C' = \dot{C}_{min}$ to the values from Figure 38 (as equation 87), Figure 39 shows how the basic rating varies with $\dot{C}'$ due to $\Delta T_G$, when $C_{HTF} = 1$ kW K$^{-1}$. Whilst with a pure refrigerant $B$ is constant (due to $\dot{C}' = \dot{C}_{max}$), there is a wide variation for a zeotropic refrigerant, depending upon the geometry. As with effectiveness, the counter-flow exchanger gives the highest $B$ and parallel-flow the least.

![Figure 40: Effect of temperature glide on $C_{min}$, $\varepsilon$ and $B$ in counter-flow exchanger](image)

Figure 40 shows explicitly, the effect of $\Delta T_G$ size on $\dot{C}_{min}$ and $\varepsilon$ and subsequently, $B$. The dashed line indicates $R_c = 1$; to the left $\dot{C}' > \dot{C}_{HTF}$ and $\dot{C}' < \dot{C}_{HTF}$ to the right. When the glide is small, $C'$ is relatively large, and because $\dot{C}_{HTF}$ is constant regardless of glide $\dot{C}_{min}$ is also constant.

As the glide increases $C'$ gradually reduces from infinity until at $\Delta T_G \approx 6$ K, it falls below $\dot{C}_{HTF}$, and the trend of $\dot{C}_{min}$ therefore begins to reduce. With zero glide, $\varepsilon$ is high due to $R_c =$
0, but as $C' \rightarrow C_{HTF}$, $\varepsilon$ also reduces until $R_c = 1$, and then as $C'$ falls below $C_{HTF}$, $\varepsilon$ increases. The combined effect of $\varepsilon \times C_{min}$ shows a smooth reduction in $B$ with increasing glide. The implication is that for a condenser or evaporator using azeotrope, the basic rating is (theoretically) lower than if a pure refrigerant was used. Since $B$ is dependent upon $\Delta T_G$, there are additional implications since it is known that the glide tends to vary with saturation pressure (discussed later).

![Figure 41: Temperature profiles for counter-flow condenser](image1)

In analysing the characteristics of a heat exchanger using azeotrope, the behaviour of the temperature profiles provides insight. Previously, it was shown that with a pure refrigerant, heat exchanger characteristics meant that when $\Delta T_{(in)} \rightarrow 0$, then $Q \rightarrow 0$ (Figure 36). With a zeotropic refrigerant, the analysis is complicated due the existence of a $\Delta T_G$.

![Figure 42: Temperature profiles of parallel- and counter-flow condenser for reducing condensing temperature with glide](image2)
Possible refrigerant and HTF temperature profiles are illustrated in Figure 41, which indicates three categories for the temperature gradients depending upon size of glide: (i) $\dot{C}' > \dot{C}_{HTF}$ for small $\Delta T_G$, (ii) $\dot{C}' = \dot{C}_{HTF}$ for medium $\Delta T_G$, and (iii) $\dot{C}' < \dot{C}_{HTF}$ for large $\Delta T_G$, for the example of a counter-flow condenser. The temperature profile for a pure refrigerant ($\dot{C}' = \infty$) is also included for comparison. It is the latter category (iii) that is of most interest, since it implies a greater departure from constant $B$, and it contradicts the conventional exchanger characteristic in that it suggests $\Delta T_{(in)}$ must be greater than zero when $T_{HTF(in)} = T_{HTF(out)}$, i.e., $Q = 0$.

This matter is investigated further by establishing the consequence of having incrementally lower $T_{(in)}$, as in Figure 37 for pure refrigerants. For azeotrope, Figure 42 illustrates a reduction of $T_{(in)}$ and hence $\Delta T_{(in)}$ (for both parallel and counter-flow type condensers), assuming a fixed $\Delta T_G$ and $T_{HTF(in)}$. Initially (left) the wide $\Delta T_{(in)}$ results in a large capacity and correspondingly a large HTF temperature rise. Lowering $T_{(in)}$ reduces capacity and correspondingly lessens the HTF temperature rise, but $T_{HTF(out)}$ is still above $T_{r(out)}$. $T_{(in)}$ is reduced further, but must cease when $\Delta T_{(out)} = 0$ K, and this corresponds to a minimum approach temperature difference ($\Delta T_{min(in)}$). $T_{r(out)}$ cannot be less than the temperature of the HTF stream at any point within the exchanger. In this final case fluid stream temperatures converge, $Q$ should approach zero; although this seems paradoxical since some temperature difference between the fluid streams exists.

![Figure 43: Minimum approach temperature differences for condenser and evaporator](image_url)
On the basis that the capacity should be zero when $\Delta T_{\text{(out)}} = 0$ K, then for the phase-change region only, $\Delta T_{\text{min(in)}}$ must be equal to $\Delta T_G$. This situation applies to both parallel- and counter-flow heat exchangers, and although not shown, to cross-flow exchangers as well. Essentially, the difference between the HTF inlet and refrigerant inlet temperatures is the offset due to the $\Delta T_G$, as shown graphically in Figure 43. On this basis, the graphical representation of the condenser and evaporator characteristic in Figure 36 is modified by shifting the $T_{\text{HTF(in)}}$ for the amount of the $\Delta T_G$ (Figure 44). Presently, the basic rating is assumed constant – resulting in the linear relationship between $Q$ and $\Delta T_{\text{(in)}}$ – although the earlier discussion implied this may not be correct, and therefore demands further consideration.

![Graphical representation of (a) condenser and (b) evaporator characteristic with zeotropic refrigerant](image)

**Figure 44:** Graphical representation of (a) condenser and (b) evaporator characteristic with zeotropic refrigerant

### 2.4.3 Implications of temperature glide behaviour

The prior analysis assumed linear temperature glide with respect to vapour quality and constant overall heat transfer coefficient. Observation of temperature glides of real mixtures often deviate from this linear form, and similarly $U$ is also known to vary. Initially, characteristics of the temperature glide were addressed, and in particular two properties were identified for further investigation: variation of $\Delta T_G$ with saturation pressure, and "shape" of $\Delta T_G$ with respect to vapour quality.

### Properties of the temperature glide

An appreciation of the glide behaviour of zeotropes was gained through examining the $\Delta T_G$ properties of various commercialised refrigerants. Details of zeotropic refrigerants were taken from UNEP (2002), and properties calculated using the subroutines of Refprop (Lemmon et al, 2015).
A list of 51 refrigerants and their compositions and selected properties are listed in Table 31 of Appendix 2.

First, total $\Delta T_G$ (i.e., over the entire phase envelope) was calculated for a range of pressures that corresponded to saturated dew-point temperatures from $-40^\circ C$ to $+60^\circ C$. The results are presented in Figure 45 (a); each refrigerant has not been identified individually because the purpose of the graph is to illustrate the general trend of change in $\Delta T_G$ with saturation temperature. Towards the lower temperature range some glides reach up to 12 K, but as saturation temperature approaches the critical point $\Delta T_G \rightarrow 0$.

About 10% of the mixtures exhibit a $\Delta T_G$ that does not follow this trend (although they will eventually reach zero at the critical point), and these are shown as dashed lines in Figure 45 (a). These fluids were identified to be two sets of mixtures. The first set was two R32/R125 blends, which have small and relatively constant $\Delta T_G$ (~0.5 K and ~1.5 K) over the temperature range, but peak in size around $0 - +20^\circ C$. The second set was three R134a/R142b blends that had a trend contrary to the others where $\Delta T_G$ tended to increase with higher saturation temperature, and their glide also ranged around 0.5 K to 1.5 K. Although the relatively wide range of these blends' partial pressures would lead to a large $\Delta T_G$, these observations suggest that the mixtures exhibit azeotropy at compositions close to those of the refrigerants composition. They are therefore not considered to be "true" zeotropes and are subsequently neglected from the following analysis.
Given the generally steady trend in $\Delta T_G$ with saturation pressure seen with most refrigerants, it is useful to characterise it. The variation in $\Delta T_G$ was found to follow saturation pressure exponentially (equation 101).

$$\Delta T_G(p_s) = c_{G1} \cdot \exp(c_{G2} \cdot p_s) \tag{101}$$

where $p_s$ is the saturation pressure, and $c_{G1}$ and $c_{G2}$ are constants. This exponential trend provided a good fit with the refrigerant data. In correlating the data, it was found that $c_{G1}$ was comparable to $\Delta T_G$ except at very low pressures. After evaluating different pressures, the closest fit included $\Delta T_G$ at standard atmospheric pressure, thus $c_{G1} = \Delta T_G(p_{atm})$, as Figure 46 (a).

![Figure 46: Correlation for constants (a) $c_{G1}$ and (b) $c_{G2}$ for equation (101)](image)

Similarly, Figure 46 (b) also shows an approximate linear correlation between the constant $c_{G2}$ and $\Delta T_G(p_{atm})$, as equation (102); since the purpose of this exercise is to obtain a general appreciation of $\Delta T_G$ characteristics, this was accepted despite the scatter.

$$c_{G2} = a_1 \cdot \Delta T_G(p_{atm}) + a_2 \tag{102}$$

Lastly, by including the relationship between $c_{G1}$ and $c_{G2}$ to $\Delta T_G(p_{atm})$, equation (101) may be rewritten to characterise the change in temperature glide with saturation pressure, based on a chosen reference $\Delta T_G(p_{atm})$ (equation 103).

$$\Delta T_G(p_s) = \Delta T_G(p_{atm}) \cdot \exp \left( [a_1 \cdot \Delta T_G(p_{atm}) + a_2 ] \cdot p_s \right) \tag{103}$$
where $a_1 = 0.0000143$, and $a_2 = -0.00036$.

The second property is the "shape" of the glide, or the variation in local saturation temperature with vapour quality as the zeotrope moves between dew- and bubble-points. Figure 45(b) shows the variety of glide shapes exhibited by the different refrigerant blends, which include convex, linear and concave behaviour.

For all binary mixtures (at any pressure), those containing a greater proportion of the higher pressure component leads to a concave glide shape, whilst equal mass fractions tend towards linearity. Apart from possessing smaller $\Delta T_C$, a mixture of components with similar saturation vapour pressures tend to a linear shape over a wider range of compositions. The variation of saturation temperature (at constant pressure) with vapour quality ($T_x$) was found to follow the form of equation (104), by employing a power-law relationship.

$$T_x(p_x) = T_{dp}(p_x) - \Delta T_C(p_x) \cdot (1 - x)^{n_G}$$  \hspace{1cm} (104)

where $T_{dp}$ is the saturated dew-point temperature, $x$ is the local vapour quality, and $n_G$ is an index dictating the shape of the glide for a given refrigerant: $n_G > 1$ produces a concave shape, $n_G = 1$ is linear, and $n_G < 1$ for a convex glide, see Figure 45(b).

![Figure 47: Variation of maximum and minimum glide index over (a) saturation temperature, and (b) saturation pressure](image-url)
Given the trend of $n_G$ indicated in Figure 47(a), it was correlated with temperature and for consistency with equation (104), $T_{dp}$ was chosen as the correlating variable. An appropriate value of $n_G$ can be estimated from equation (105).

$$n_G = a_1 \cdot T_{dp}(p_s) + a_2$$

where for a given refrigerant, $a_1$ is between -0.015 and 0.002, and $a_2$ is between 2.5 and 0.6, depending upon the desired shape, although correct selection of the constants will cause $n_G = 1$ when $T_{dp} = T_c$.

### Influence of glide properties on basic rating

With reference to the discussion in Section 2.4.2, basic rating for a pure refrigerant is constant, but when a zeotrope is used it is affected by the size of the glide (e.g., Figure 40). A variation of $\Delta T_G$ affects $\dot{C}_p$, and therefore $B$. Previous sections showed that $\Delta T_G$ varies with saturation temperature, which implied additional variation of $B$ over a range of evaporating or condensing temperatures. An example of the influence of the variable $\Delta T_G$ is shown in Figure 48, which was calculated for a counter-flow condenser. The filled data-points refer to the variable $\Delta T_G$ (equation 103), and the hollow data-points were calculated for a fixed $\Delta T_G$ regardless of saturation pressure.

For a pure refrigerant ($\Delta T_G(p_{sat}) = 0$ K), $B$ is constant under all conditions. For a refrigerant with a small glide (say, $\Delta T_G(p_{sat}) = 2$ K), there was only a small variation of $B$ with saturation
temperature and whether the variation of glide is accounted for or not has little effect on $B$. However, in the case of a large glide (say, $\Delta T_G(p_{atm}) = 8$ K), it is seen that a significant difference in $B$ can occur when the variation of glide size with saturation temperature is included. This trend amplifies the importance of accounting for a variable $\Delta T_G$, when the glide is large. For both variable and fixed glide cases, $B$ reduces as $T_{c(in)} \rightarrow T_{HTF(in)}$ (i.e., $\Delta T_{c(in)}$ gets smaller).

![Graph](image)

**Figure 48**: Change in condenser basic rating with inlet temperature for fixed (hollow data point) and variable (filled data point) temperature glide

**Particular influence on evaporator**

The condenser was used as an example because it exhibits complete phase-change, as opposed to the evaporator, where if the refrigerant enters the evaporator at vapour quality between 0 and 1, the temperature glide is bisected. As such, the local refrigerant temperature is between bubble- and dew-point, and therefore the $\Delta T_G$ experienced by the evaporator is smaller than the $\Delta T_G$ for a complete phase-change. The temperature at which liquid refrigerant enters the expansion device ($T_{REV(in)}$) determines the evaporator inlet quality ($x_{e(in)}$), and therefore the size of the partial temperature glide, i.e., $\Delta T_{G,e} = T_{e,dp} - T_{e(in)}$. Earlier it was shown that the size of $\Delta T_{G,e}$ had implications on $B$, and it therefore follows that $B_{evap}$ must be dependent upon $T_{REV(in)}$. This is illustrated in Figure 49(a) where different $T_{REV(in)}$ are shown on a pressure-enthalpy diagram, and the corresponding evaporating inlet temperatures are shown for a fixed evaporator pressure.
Considering the basic rating equation (85), the evaporator characteristic should be re-plotted, and this is shown in Figure 49(b) where a different curve is presented for each $T_{TEV(in)}$.

The implications of changing $T_{TEV(in)}$ can also be evaluated in terms of the heat exchanger analysis. A higher $T_{TEV(in)}$ and subsequent smaller $\Delta T_{G,e}$ will result in higher $C'_e$ ultimately changing $B_{evap}$. It is also noted that because local refrigerant temperature rarely changes linearly with $x$, the variation in $B_{evap}$ will depend upon the glide shape. These implications were addressed by evaluating evaporator characteristics with the different refrigerants from UNEP (2002).

It was noted that whilst a change in $x_{e(in)}$ affects $\Delta T_{G,e}$, refrigerant heat transfer coefficient ($h_r$) is also reliant upon vapour quality, and therefore, a change in $T_{TEV(in)}$ may also influence $U$; both of these ($\Delta T_{G,e}$ and $U$) affect evaporator capacity. Whilst a higher $T_{TEV(in)}$ will raise $T_{e(in)}$ and therefore reduce $Q_e$, the shift in $x_{e(in)}$ implies a higher $U$, because local $h_r$ typically increases with $x$ resulting in larger $Q_e$. The degree of variation $Q_e$ depends on the relative influence of $\Delta T_{G,e}$ and $U$. In fact, a shift in $x_{e(in)}$ also influences pressure loss along the refrigerant path which impacts on the overall temperature glide, but this aspect was neglected for the time being.

![Figure 49: Effect of different expansion device inlet temperature on (a) evaporator inlet temperature, and (b) on evaporator characteristic](image-url)
For the analysis, variation in $U$ was accounted for by simply adjusting a mean value ($\overline{U}$) according to $x_{e(in)}$. The evaporator model (detailed later in Section 3.2) was used to obtain local values of $U$, which were used to calculate the area-weighted $\overline{U}$ for a range of $x_{e(in)}$ under different sets of input conditions, geometries and refrigerants. Most situations showed an upward trend in $\overline{U}$ with $x_{e(in)}$. For the purposes of this exercise, a simple linear formula was used to represent the variations in $\overline{U}$ (equation 106).

$$\overline{U}(x_{e(in)}) = \overline{U}(0) \cdot (1 + a \cdot x_{e(in)})$$

where $\overline{U}(0)$ is the mean overall heat transfer coefficient when $x_{e(in)} = 0$. The value of $a$ is largely dependent upon the input condition, geometry and refrigerant, but was typically found to be between 0.05 - 0.1, and from 0 to 0.3 in extreme cases.

![Figure 50: Variation in evaporator basic rating with glide size at different $T_{TEV(in)}$.](image1)

![Figure 51: Variation of evaporator capacity with glide size due to change in TEV inlet temperature from 20°C to 60°C.](image2)

The NTU-ε method detailed above was used to calculate the capacity of a counter-flow evaporator using various refrigerants. $Q_*$ at a fixed $T_{c,dp} = 0°C$ was evaluated twice, for $T_{TEV(in)}$ at 20°C and 60°C. Other inputs included $C_{HTF} = 1$ kW K$^{-1}$, $T_{HTF(in)} = +5°C$, and $\overline{U}(0) = 200$ W m$^{-2}$ K$^{-1}$, and $a$ in equation (106) was fixed at 0 or 0.3. Some results are shown in Figure 50, where $B_{evap}$ is plotted against the full $\Delta T_G$ at 0°C for each refrigerant. For any $T_{TEV(in)}$, there is a general trend of reducing $B_{evap}$ as the full $\Delta T_G$ gets larger. The scatter is partly due to the
variation of refrigerants' glide shape, but it is also influenced by the gradient of the bubble-line curve, which dictates the change of $x_{e(i)}$ with $T_{TEV(i)}$.

When $\overline{U}$ is constant (filled data-points), there is a minimal difference between $B_{\text{evap}}$ at $T_{TEV(i)} = 20^\circ C$ and $60^\circ C$ at smaller $\Delta T_G$, but an increasing difference occurs with larger $\Delta T_G$. With $T_{TEV(i)} = 20^\circ C$, the variation in $B_{\text{evap}}$ with $\Delta T_G$ is smooth, whilst greater scatter is seen for $T_{TEV(i)} = 60^\circ C$. This is attributed to the increasing influence of the gradient of the bubble-line, as this moves the point of evaporator glide intersection further away from $T_{e, bp}$, thereby bringing out the specific glide characteristics of each refrigerant. When the contribution of heat transfer coefficient is introduced by varying $\overline{U}$ with $x_{e(i)}$ (hollow data-points), the difference between $B_{\text{evap}}$ at the two $T_{TEV(i)}$ is large throughout the range of $\Delta T_G$. Again, when $T_{TEV(i)} = 20^\circ C$, the change in $B_{\text{evap}}$ over the range of refrigerants is relatively smooth, and follows the corresponding data for constant $\overline{U}$. However, when $T_{TEV(i)} = 60^\circ C$, $B_{\text{evap}}$ is on average consistently greater than $T_{TEV(i)} = 20^\circ C$ across all refrigerants, although significantly more scatter is seen that all other cases. This is to be expected given that the stronger dependence of the gradient of the bubble-line does not only affect the resulting $\Delta T_{G,e}$, but $x_{e(i)}$ and therefore sensitivity to $\overline{U}$ as well.

The data is also presented in Figure 51 in terms of the relative difference between $Q_e$ over the two $T_{TEV(i)}$ for both constant and variable $\overline{U}$. With constant $\overline{U}$, $Q_e$ for $T_{TEV(i)} = 20^\circ C$ and $60^\circ C$ is seen to reduce by about 15% as the refrigerants' $\Delta T_G$ → 10 K, whereas a variable $\overline{U}$ initially produces a 10% higher $Q_e$ at $\Delta T_G \approx 0$ K, but then eventually shows a 5% reduction in $Q_e$. Again, the scatter associated with the data is a result of the refrigerants' different $n_G$ and bubble-line curves.

Several important conclusions can be drawn from the discussion above. Primarily, evaporator characterisation is dependent upon the inlet condition of the refrigerant, and for any specific refrigerant, this is dictated by the inlet temperature to the expansion device. A refrigerant with a larger temperature glide typically exhibits a greater variation in $Q_e$ for a given change in $T_{TEV(i)}$, although the shape of the glide, and the gradient of the bubble-line have a some
influence. Assuming a fixed $U$, a higher $T_{TEV(in)}$ normally leads to a reduction of $Q_e$ on account of the approach temperature difference (i.e., $T_{HTF(in)} - T_{e(in)}$) being smaller. However, if $U$ increases with higher $T_{TEV(in)}$, the variation of $Q_e$ due to changing $\Delta T_G$ is offset to an extent, and if $U$ has as strong dependency on $x$, this offset may negate any reduction in $Q_e$. In fact, for a refrigerant with a given set of glide characteristics used within a certain evaporator the behaviour of $\bar{U}(x_{e(in)})$ may be such that $Q_e$ becomes effectively independent of $T_{TEV(in)}$.

2.5 Findings relevant to component-system chain and additional considerations

A number of findings were made, which confirmed the necessity to clarify suitable methods used within the component-system chain when using zeotropic refrigerants. The foregoing review of the main system components with respect to the use of zeotropic refrigerants provided an indication of areas where component capacity and performance rating differ compared to a pure refrigerant.

Performance testing, rating and presentation of data for compressor and TEV requires some reference value to be specified that correspond to the inlet and outlet state, and because of the $\Delta T_G$, it follows that the choice of reference state must be redefined when a zeotrope is used. Due to the $\Delta T_G$, the conventional definition of approach temperature difference for evaporator and condenser is not suitable. The size of the temperature glide changes with saturation temperature which affects the basic rating and the capacity, meaning that these parameters must also be included in the component characterisation. In the case of the evaporator, a change in $T_{TEV(in)}$ and therefore inlet quality changes $\Delta T_G$ which also influences the approach temperature difference. Consequently, an alternative rating method is needed and definitions of reference states should be identified.

A key aspect of the study is to establish suitable reference states for the rating and presentation of data of each component, and the subsequent system balancing. When considering each component in isolation, it was generally concluded that the dew-point was the most suitable choice. However, when components are matched during the balancing procedure, it is important to ensure that consistent reference states are used as each must "see" the same reference state as the adjoining component. In the conventional technique, evaporating and/or condensing temperatures that correspond to the saturated evaporating/condensing pressures are used. Neglecting temperature
and pressure changes within interconnecting piping and ancillary components, this provides a consistent basis for reference states of a pure refrigerant.

In terms of a zeotrope, saturated evaporating and condensing temperatures may be associated with the dew-point, bubble-point or somewhere between, such as the mid-point temperature (half way between bubble- and dew-points). Whilst the bubble-point would be suitable for the condenser, it is not directly experienced by the evaporator. Similarly, because the saturated temperature at evaporator inlet is variable, the choice of mid-point would be inappropriate because it would change as the performance of the evaporator changed. The only reference state that would remain consistent for all components is the dew-point, and therefore this is adopted hereon.

Several parameters have so far been neglected which individually or in combination, may impact on the interpretation of exchanger performance.

- **Superheat, desuperheat and subcooling.** Inclusion of sensible heat changes to the refrigerant was omitted, despite being nearly always present in real condensers and evaporators. These aspects may contribute to changes in the heat exchanger capacity and thus introduce a departure from conclusions from the NTU-ε analysis which assumed phase-change only.

- **Pressure loss.** This is of significance in the case of phase-change fluids since a sufficiently high pressure loss will cause a marked change in saturation temperature through the change in saturation pressure. The effect on temperature profiles when using zeotropic refrigerants are such that a pressure loss in the condenser would effectively increase the temperature glide, but reduce it in the evaporator. This would affect heat exchanger performance.

The most appropriate approach to evaluate the cumulative effect is considered to be combining them in the form of a heat exchanger simulation.
SECTION 3: CHARACTERISATION OF COMPONENTS WITH ZEOTROPES

3.1 Approach for improved characterisation
Initially, it was proposed to use dew-point temperatures as the new reference state. However, it was also noted that this conclusion was based on simple analysis of evaporator and condenser performance, with certain parameters omitted. Therefore it was deemed necessary to further evaluate their performance using a more detailed modelling approach. The simulation results will help to clarify the appropriate definition of reference states, and in addition, identify the functional dependency of capacity on relevant parameters that can be used for the rating method. Further, performance data is required in a later section for the balancing exercise, so these models can be used to generate fictitious data. This is also needed for the compressor, so a basic model was similarly produced to generate fictitious performance data. 12

The following sections first detail the performance models for the compressor and then the condenser and evaporator, where a generic heat exchanger model is described, and correlations specific to either condenser or evaporator are given separately. The models are then used to investigate the influence of various aspects such as glide shape, pressure loss and sensible heat transfer, and from this conclusions are drawn relating to definition of reference states. Lastly, possible options for revised characterisation methods are presented.

3.2 Simulation of compressor performance
Generation of compressor performance data requires information on discharge superheat, and since conventional compressor catalogue does not normally provide this, using a simulation approach is the only means by which it can be obtained. It is important to ensure that the data are based on the newly established reference states for zeotropic refrigerants.

3.2.1 General modelling approach
A semi-empirical model was chosen representing a conventional single-cylinder reciprocating machine. Existing catalogue data was employed to generate the required coefficients. It must be noted that it was not necessary to use a particularly accurate approach provided the results were conceptually sound and consistent with normal compressor operation.

12 The compressor, condenser and evaporator models were also utilised by cycle models in Part 1.
Relevant inputs for the model are swept volume flow rate, saturated evaporating temperature, suction pressure, saturated condensing temperature, discharge pressure, evaporator superheat, and suction superheat; saturated condensing and evaporating temperatures correspond to dewpoint. The output data required for system balancing are evaporating capacity, compressor power or condensing capacity, and discharge superheat.

The model programme was written in Visual Basic (VB), and where necessary, refrigerant property data was called from the subroutines within the Refprop database (Lemmon et al, 2002).

3.2.2 Semi-empirical model

The total evaporating capacity of the compressor is obtained from enthalpy balance, equation (107).

\[ Q_{\text{comp}} = \dot{V}_w \cdot \eta_{\text{vol}} \cdot \rho_{1,\text{nuc}} \cdot (H_{1,\text{nuc}} - H_4) \]  

where \( \rho_{1,\text{nuc}} \) is suction density, and \( (H_{1,\text{nuc}} - H_4) \) is specific enthalpy difference between the evaporator inlet and the compressor suction. \( \eta_{\text{vol}} \) is the volumetric efficiency and was calculated with a modified formula based on Gosney (1982) (equation 108). The original formula was intended for a single refrigerant, so the initial constant was adapted to account for differences resulting from the use of other refrigerants. This was achieved by correlating the initial constant with the molecular mass \( M \) and saturation pressure at 0°C \( (p_0) \) of the different refrigerants.

\[ \eta_{\text{vol}} = c_1 + c_2 \cdot M \cdot \left[ R_{\text{vir}} - R_{\text{vir}} \cdot \left( \frac{p_2}{p_1} \right)^{\frac{1}{\gamma}} \right] + c_3 \cdot \left[ \left( \frac{p_2}{p_1} \right)^{\frac{1}{\gamma}} - 1 \right] + (c_4 + c_5 \cdot p_0) \]  

where \( R_{\text{vir}} \) is the “apparent” clearance volume ratio, \( p_1 \) and \( p_2 \) are suction and discharge pressures respectively, and \( \gamma \) is the isentropic index, defined as the ratio of specific heats at suction conditions \( (C_{p_{1,\text{nuc}}}/C_{v_{1,\text{nuc}}}) \). The constant \( c_1 \) is a function of the specific compressor geometry and the remaining \( c_2 \ldots c_5 \) relate to the refrigerant type; all were obtained from correlating catalogue data.

Determination of compressor power \( (W_{\text{comp}}) \) was also based on Gosney (1982), which adopts the ideal mean effective pressure term \( (p_{\text{ime}}) \). The power was calculated as a linear function of volumetric flow and \( p_{\text{ime}} \), and includes a constant value to account for frictional effects \( (W_f) \) (equation 109).
\[ W_{\text{comp}} = W_f + b \cdot P_{\text{ime}} \cdot (V_w \cdot \eta_{\text{vol}}) \]  

where \( b \) is an empirical constant, and the product \( P_{\text{ime}} \cdot (V_w \cdot \eta_{\text{vol}}) \) is the ideal compressor work; \( P_{\text{ime}} \) is from equation (110).

\[ P_{\text{ime}} = \left( \frac{\gamma}{\gamma - 1} \right) \cdot p_1 \cdot \left[ 1 + R_{\text{refl}} - R_{\text{refl}} \cdot \left( \frac{p_1}{p_2} \right)^{1/2} \right] \cdot \left[ \frac{p_1}{p_2} - 1 \right] \]  

(110)

The condensing capacity, \( Q_{\text{comp}} \), was calculated as the sum of \( Q_{\text{e,comp}} \) and \( W_{\text{comp}} \).

Determination of the discharge superheat was achieved by calculating discharge temperature from local enthalpy and pressure, using the property routines within Refprop. Whilst discharge pressure is known, determination of discharge enthalpy \( (H_{2,\text{disch}}) \) was based on an energy balance across the compressor, which assumed 95% contributes to the heat of compression and 5% loss to the surroundings (equation 111).

\[ H_{2,\text{disch}} = H_{1,\text{nct}} + \frac{0.95 \cdot W_{\text{comp}}}{V_w \cdot \eta_{\text{vol}} \cdot \rho_{1,\text{nct}}} \]  

(111)

where the lower term represents refrigerant mass flow rate. Discharge superheat was based on the difference between discharge temperature and dew-point temperature corresponding to discharge pressure, i.e., \( \Delta T_{\text{dsh}} = T_{2,\text{disch}} - T_{2,\text{dp}} \).

3.2.3 Determination of constants

The constants in equations (108) and (109) were obtained from catalogue data, and in addition, because the actual clearance ratio was not given in the product information, an “apparent” value \( R_{\text{refl}} \), was also determined from the catalogue data.

Information was used for several different compressors from manufacturers’ catalogue (Bitzer, undated). For each compressor, capacities and power was given for several different refrigerants: R22, R134a, R404A and R407C. For equation (108), the constant \( c_1 \) ranged from 5 to 40 as compressor size increased. The best-fit for the remaining constants were: \( c_2 = -0.093 \), \( c_3 = 0.184 \), \( c_4 = 0.785 \), and \( c_5 = 0.00027 \). For equation (109), \( b = 1.24 \), and \( W_f \) varied according to compressor choice, but generally represented 5 - 10% of \( W_{\text{comp}} \) depending on the operating.
conditions. Lastly, $R_{\text{opt}}$ was approximated (since it is not provides in manufacturers data) by adjusting its value in equation (108) and equation (110) until the highest regression constant ($r^2$) was achieved for all refrigerants in any one compressor model, and ranged from 0.10 – 0.15. This value was then used for determination of all the constants mentioned above.

The results produced by the model were compared against the original catalogue data and $Q_{e\text{,comp}}$ was found to match to within ±9%, and $W_{\text{comp}}$ to within ±6%. Judging these errors against those reported by other authors provide an indication to the acceptability of the approach detailed. As an example, Popovic and Shapiro (1995) report errors of greater than ±5% of the measured power and ±10%, and in some cases greater than −20%, for the capacity.

### 3.3 Simulation of condenser and evaporator performance

A general model was developed to simulate the performance of a horizontal evaporator and condenser, of parallel- and counter-flow geometries. The construction of the exchangers was based on a multi-circuit concentric tube design, using a liquid HTF. The objective of this exercise was to generate fictitious data that is principally identical to what would be expected as if measured from a standard calorimeter test.

#### 3.3.1 General modelling approach

Inputs relating to the physical construction of the exchanger are: number of refrigerant/water circuits, individual circuit tube length, refrigerant tube internal diameter, HTF tube external diameter, and ratio of outside (HTF) to inside (refrigerant) surface area. Inputs for the refrigerant conditions vary between condenser and evaporator. The following pressures and temperature were fixed for the condenser: dew-point temperature corresponding to inlet pressure, superheat above saturation temperature corresponding to inlet pressure, subcooling temperature drop, HTF inlet temperature, and HTF mass flow rate. Correspondingly for the evaporator, dew-point temperature corresponding to outlet saturation pressure, TEV inlet temperature and pressure, superheat temperature rise, HTF inlet temperature, and HTF mass flow rate were fixed. Lastly, the model allowed $\Delta T_G$ characteristics to be specified, as detailed in Section 2.4.3.

It is noted that it was deemed not necessary to validate the models against experimental data because the trends in performance behaviour are the desired outputs. Output data of interest were: incremental refrigerant and HTF temperatures, total condenser or evaporator capacity, and refrigerant saturation temperatures corresponding to condenser inlet pressure or evaporator outlet.
pressure. These input and output values were subsequently used to assess the effect of zeotropic refrigerants on heat exchanger characterisation.

Numerical approach
Based on a set of specified temperatures, calculation of condenser and evaporator capacity was based on local heat transfer for both single- and two-phase regions. For the single-phase regions, the size of each element was based on the enthalpy change (equation 112).

\[ \delta H = \frac{H_{(out)} - H_{(in)}}{N_H} \]  

(112)

where the term \( H_{(out)} - H_{(in)} \) is the enthalpy difference over the entire single-phase region, and \( N_H \) is the chosen number of elements. Within the two-phase region, the elements were based on vapour fraction (i.e., linked directly to enthalpy change) (equation 113).

\[ \delta x = \frac{x_{(out)} - x_{(in)}}{N_x} \]  

(113)

where the term \( x_{(out)} - x_{(in)} \) is the vapour fraction over the entire two-phase region, and \( N_x \) is the chosen number of elements. Within the condenser, equation (113) is effectively the reciprocal \( 1/N_x \). For the evaporator, equation (113) corresponds to \((1 - x_{e(in)})/N_x\), and inlet vapour quality was found in the normal way (equation 114).

\[ x_{e,in} = \frac{H_{1,s} - H_3}{H_{1,s} - H_{4,s}} \]  

(114)

where the enthalpies \( H_{1,s} \) and \( H_{4,s} \) correspond to saturated temperature at local evaporator inlet pressure, and \( H_3 \) at the TEV inlet conditions.

Refrigerant properties were taken from the Refprop database whilst properties of HTF were obtained from the equations in Appendix 1. Extraction of properties from Refprop was via a link to its relevant subroutines and the equations in Appendix 1 were included in the programming code. To acquire different glide size and shape normally require reformulation of refrigerant composition, but this also results in modifying other thermodynamic and transport properties. To avoid confusion in this respect, an approach that used a full fictitious \( \Delta T_G \) with reference to the dew-point temperature was adopted, where size (equations 103 and 104) and shape (equation 105) of the glide is dictated by the inputs, so that the effects of glide can be isolated and studied.
The local HTF temperature \( T_{HTF,j} \) is obtained from the energy balance (equation 115).

\[
T_{HTF,j} = T_{HTF,(in)} + \sum_{i=1}^{N_j} \frac{Q_i}{m_{HTF} \cdot C_{p_{HTF}}}
\]  

(115)

where \( N_j \) is the number of elements preceding element \( i \), \( Q_i \) is the capacity of each of those elements, and the sign depends upon whether it is heating (condenser) or cooling (evaporator).

For either heat exchanger, the total capacity \( Q_{tot} \) comprises the heat transfer associated with the refrigerants' two-phase region \( Q_{sp} \), and the single phase regions, which may include liquid \( Q_{sp,l} \) and/or vapour \( Q_{sp,v} \) (equation 116).

\[
Q_{tot} = Q_{sp,l} + Q_{sp,v} + Q_{tp}
\]  

(116)

Calculation of heat transfer between fluid streams of each element was based on equation (86) for the NTU-\( \epsilon \) method.

\[
Q_i = \epsilon_i \cdot C_{min,i} \cdot \Delta T_{(in),i}
\]  

(117)

where \( \Delta T_{(in),i} \) refers to the temperature difference between the two fluids within each element, \( \epsilon \) was calculated as equations (98) or (99) according to its geometry, and \( C_{min} \) was determined from the HTF or refrigerant stream (Section 2.4). The total capacity of the single-phase and two-phase regions of a heat exchanger was based on the sum of its elements (equation 118 and 119, respectively).

\[
Q_{sp} = \sum_{i=1}^{N_{htf}} Q_{sp,i}
\]  

(118)

\[
Q_{tp} = \sum_{i=1}^{N_{htf}} Q_{tp,i}
\]  

(119)

To clarify, \( \dot{C}_{HTF,j} \) was obtained as equation (88) but adapted to account for the number of circuits (equation 120), and \( \dot{C}_{r,j} \) from (97), again adapted for the number of circuits and local change in refrigerant temperature (equation 121).

\[
\dot{C}_{HTF,j} = \frac{\dot{m}_j}{N_{crt}} \cdot C_{p_{HTF,j}}
\]  

(120)
where $N_{crt}$ is the number of refrigerant circuits.

Here, it is important to note that since pressure losses along the refrigerant flow path were accounted for, there will always be a difference in refrigerant inlet and outlet temperature even with a single-component refrigerant. Therefore calculation of $\dot{C}_{r,i}$ from equation (121) is applicable for all refrigerants.\textsuperscript{13} Since $\varepsilon$ is a function of NTU, calculation of local $U$ was required. Neglecting thermal resistance of construction materials $U$ was obtained with the conventional equation (122).

\[ U_i = \left( \frac{1}{h_{r,i}} + \frac{1}{R_{surf} \cdot h_{HTF,i}} \right)^{-1} \] (122)

where $R_{surf}$ is the ratio of external to internal surface area, $h_{HTF}$ is the HTF-side heat transfer coefficient, and $h_r$ is the local refrigerant-side heat transfer coefficient, as determined from the relevant Section 3.3.2 or 3.3.3.

Ultimately, the solution of the heat exchanger model is implicit and convergence of $Q_{tot}$ was achieved through successive estimates of refrigerant mass flow based on determining total exchanger area. Thus, $\dot{m}_r$ was revised in proportion to the over- or under-estimation of total exchanger area, until calculated area matched the total design area ($A_{tot,des}$) (equation 123).

\[ \dot{m}_r = \dot{m}_r(0) \cdot \left( \frac{NH + Nx}{A_{tot,des} / \sum_i A_i(0)} \right)^{1/3} \] (123)

where $\dot{m}_r(0)$ and $A(0)$ refer to the mass flow and total area of the prior iteration, and the index $^{1/3}$ was used to provide stability to iterative process. Convergence of calculated area to within 0.01\% of $A_{tot,des}$ was usually achieved after some 50 iterations. Since $\dot{m}_r$ is unknown, an initial guessed value was required. The first $\dot{m}_r(0)$ was estimated by evaluating equation (86) in an

\textsuperscript{13} The exception to this is a unique case in the evaporator only, where the temperature gradient corresponding to the pressure loss along the refrigerant flow path is the negative of the temperature glide gradient. Thus, each incremental decrease in temperature due to reducing pressure is equal to each incremental rise in temperature due to glide, thereby cancelling each other out causing isothermal flow.
energy balance for the entire heat exchanger, whilst assuming \( \dot{C}_{\text{min}} = \dot{C}_{\text{HTF}} \), \( \dot{C}_r = \infty \), and an average \( U = 100 \, \text{W m}^{-2} \, \text{K}^{-1} \) (equation 124).

\[
\dot{m}_r(0) \approx \frac{\Delta T_{\text{in}} \cdot \dot{C}_{\text{HTF}}}{H_{d_p,s} - H_{b_p,s}} \left( 1 - \exp \left[ -\frac{100 \cdot A_{\text{tot,des}}}{\dot{C}_{HTF}} \right] \right)
\]

(124)

where \( H_{d_p,s} - H_{b_p,s} \) is the enthalpy difference between saturation points at constant pressure.

The basic calculation sequence was as follows:

- With \( \dot{m}_r(0) \), determine local \( \dot{C}_{\text{min}} \) (equations 120 and 121).
- Calculate effectiveness of each element with equation (125) based on capacity from local enthalpy balance (equation 118).
- From \( \varepsilon \) and \( \dot{C}_{\text{min}} \), calculate NTU; for parallel flow (equations 126) or for counter-flow (equation 127) (transposed from equations 98 and 99).
- Determine surface area of element (equation 128) and for pressure loss calculation, the tube length of each \( \delta \times \) or \( \delta \times \) element (equation 129).
- Sum the calculated area for each element, including single- and two-phase regions and revise \( \dot{m}_r \) (equation 123).

\[
\varepsilon_i = \frac{Q_i}{\dot{C}_{\text{min}i} \cdot \Delta T_{\text{in}i}}
\]

(125)

\[
N_{\text{TU},i} = \frac{\ln[1 - \varepsilon_i \cdot (1 + R_{\dot{C}_i})]}{1 + R_{\dot{C}_i}}
\]

(126)

\[
N_{\text{TU},i} = \frac{1}{R_{\dot{C}_i} - 1} \cdot \ln \left( \frac{\varepsilon_i - 1}{\varepsilon_i \cdot R_{\dot{C}_i} - 1} \right)
\]

(127)

\[
A_i = \frac{N_{\text{TU},i} \cdot \dot{C}_{\text{min}i}}{U_i}
\]

(128)

\[
\delta z_{\text{tot},i} = \frac{A_i}{\pi \cdot D}
\]

(129)

This procedure is elaborated on in Figure 52 for the condenser and in Figure 53 for the evaporator.
### 3.3.2 Condenser heat transfer and pressure loss

Throughout the literature there exist a range of different correlations for condensing heat transfer coefficient. Most were developed specifically for condensation of pure fluids, although some have been shown to provide relatively accurate results in the case of mixtures. Shao and Granrud (2000) provide a correlation specifically for mixture refrigerants, which directly takes account of the $\Delta T_G$; this suits the approach adopted for this model where fictitious glide characteristics are specified.

![Figure 52: Calculation sequence for condenser capacity](image-url)
Condensing heat transfer coefficient for pure (and azeotropic mixtures) – where $\Delta T_G = 0$ – is obtained from the conventional equation (130).

$$h_{tc} (\Delta T_G = 0) = \frac{Nu \cdot k_l}{D}$$  \hspace{1cm} (130)

where $k_l$ is the liquid thermal conductivity and $Nu$ is the Nusselt number. The correlation for Nusselt number (equation 131) was derived for pure and zeotropic mixtures.

$$Nu = c_1 \cdot Pr_l^{1/3} \cdot \left( \frac{H_{2,s} - H_{3,s}}{Cp_l \cdot (T_s - T_{wall})} \right)^{1/6} \cdot Re_v^{c_2}$$  \hspace{1cm} (131)

where $Pr_l$ is liquid Prandtl number, the term $H_{2,s} - H_{3,s}$ is the saturation enthalpy difference, $Cp_l$ is liquid specific heat and the term $T_s - T_{wall}$ is the temperature difference between saturation and the tube wall surface ($Q / [h_{tc,l} \cdot A_t]$).

When the vapour Reynolds number $Re_v < 24000$, the constants $c_1 = 15.9$ and $c_2 = 0.15$; where $Re_v > 24000$ the constants $c_1 = 0.084$ and $c_2 = 0.67$. The vapour Reynolds number is according to equation (132).

$$Re_v = \left( \frac{G_v \cdot D}{\mu_l} \right) \cdot \left( \frac{\rho_l}{\rho_v} \right)^{0.5}$$  \hspace{1cm} (132)

where $D$ is the tube inside diameter, $\mu_l$ is liquid viscosity and $\rho_l$ and $\rho_v$ are liquid and vapour densities respectively. The vapour mass flux ($G_v$) is calculated from the mass flow of refrigerant vapour (equation 133).

$$G_v = x \cdot \dot{m}_v \cdot \frac{4}{\nu \cdot \pi \cdot D^2}$$  \hspace{1cm} (133)

where the void fraction ($\nu$) is calculated from a model developed by Smith (equation 134).

$$\nu = \left\{ 1 + \frac{\rho_v}{\rho_l} \cdot \frac{1 - x}{x} \cdot \left[ 0.4 + 0.6 \cdot \left( \frac{x \cdot (\rho_l + \rho_v) + 0.4 \cdot (1 - x)}{x + 0.4 \cdot (1 - x)} \right)^{1/2} \right] \right\}^{-1}$$  \hspace{1cm} (134)

Shao and Granryd identified an inequilibrium of components between the liquid and vapour phase which results in resistance in mass transfer, and therefore offered a compensatory
correction factor. Heat transfer coefficient for a mixture – when $\Delta T_G > 0$ – is from equation (135).

\[
h_{r,e}(\Delta T_G > 0) = \frac{1}{1 + F_1 + F_2} \cdot h_{r,e}(\Delta T_G = 0)
\]

(135)

where $F_1$ and $F_2$ represent the effects of the liquid-vapour inequilibrium (equations 136 and 137 respectively).

\[
F_1 = x \cdot F(x)_1 \cdot \left( \frac{Cp_v}{Cp'} \right)_x \cdot \left( \frac{h_{r,e}(\Delta T_G = 0)}{h_{l-v}} \right)_x
\]

(136)

\[
F_2 = (1 - x) \cdot F(x)_2 \cdot \left( \frac{Cp_l}{Cp'} \right)_x \cdot \left( \frac{h_{r,e}(\Delta T_G = 0)}{h_{l-l}} \right)_x
\]

(137)

where $Cp'$ is the apparent specific heat as already detailed in equation (94), and the functions $F(x)_1$ and $F(x)_2$ are as equations (138) and (139).

\[
F(x)_1 = 1.1 \cdot x \cdot (1 - \exp[-10 \cdot (1 - x)])
\]

(138)

\[
F(x)_2 = 0.9 \cdot (1 - x)
\]

(139)

$h_{l-v}$ and $h_{l-l}$ are the heat transfer coefficient between the liquid interface and vapour, and the liquid interface and liquid, respectively, and are approximated from equations (140) and (141).

\[
h_{l-v} = \frac{k_v}{D} \cdot \left( 0.023 \cdot Re_v^{0.8} \cdot Pr_v^{0.3} \right)
\]

(140)

\[
h_{l-l} = \frac{k_l}{D} \cdot \left( 0.023 \cdot Re_l^{0.8} \cdot Pr_l^{0.3} \right)
\]

(141)

where $k_v$ is vapour thermal conductivity, and Reynolds number and Prandtl number are calculated in the usual way from vapour ($Re_v$, $Pr_v$) or liquid ($Re_l$, $Pr_l$) properties only.

Pressure loss occurs along the flow path, and it is important to account for this since it affects local saturation temperature, and therefore influences the glide. The pressure loss within each element was calculated using the correlation developed specifically for condensation pressure loss of pure and mixture refrigerants, proposed by Shao and Granryd (2000). Normally, the total two-phase pressure loss is the sum of frictional ($fr$), momentum ($M$) and gravitational ($g$) components (equation 142).
\[ \frac{dp}{dz} = \left( \frac{dp}{dz} \right)_p + \left( \frac{dp}{dz} \right)_M + \left( \frac{dp}{dz} \right)_g \]  
(142)

For horizontal flow, the gravitational component is not applicable so \((dp/dz)_g = 0\). Also, the contribution from momentum has a positive effect, but in comparison to the frictional component it is reported to be miniscule, so \((dp/dz)_M = 0\). Equation (143) was used to calculate the frictional two-phase pressure loss, which is determined from the vapour-only pressure gradient, adjusted by a two-phase multiplier \((\phi_p)\).

\[ \left( \frac{dp}{dz} \right)_p = \phi_p \cdot \left( \frac{dp}{dz} \right)_{fr,v} \]  
(143)

where \((dp/dz)_{fr,v}\) is the frictional pressure loss of a vapour-only flow (equation 144).

\[ \left( \frac{dp}{dz} \right)_{fr,v} = - \frac{2 \cdot \lambda_v \cdot (x \cdot G)^2}{\rho_v \cdot D} \]  
(144)

where the vapour only friction factor \((\lambda_v)\) is calculated from equation (145).

\[ \lambda_v = 0.045 \cdot \left( \frac{x \cdot G \cdot D}{\mu_v} \right)^{-0.2} \]  
(145)

The two-phase multiplier (equation 146) was correlated to the Lockhart-Martinelli parameter by Shao and Granryd using data for both pure and zeotropic refrigerants (equation 147).

\[ \phi_p = 1 + 0.8 \cdot X_p^{0.1} \]  
(146)

\[ X_p = \left( \frac{1-x}{x} \right)^{0.9} \cdot \left( \frac{\mu_l}{\mu_c} \right)^{0.1} \cdot \left( \frac{\rho_c}{\rho_l} \right)^{0.5} \]  
(147)

3.3.3 Evaporator heat transfer and pressure loss

For the present study, the correlation of Jung and Radermacher (1993) was selected upon the recommendation of others who reviewed the various options within the literature (e.g., Hewitt, 1999). The two-phase heat transfer coefficient of the evaporating mixture \((h_{re})\) is from equation (148).\(^{14}\)

\(^{14}\) Note that the following equations differ slightly from the citation due to the published version containing errors.
\begin{equation}
K^* = (F_{JR} \cdot F_{un}) \cdot h_{pb} + (F_{me} \cdot F_{ip}) \cdot h_l
\tag{148}
\end{equation}

where $F_{JR}$ is dependent upon the Lockhart-Martinelli parameter (equation 149 or 150).

\begin{equation}
F_{JR} = 4048 \cdot X_{tu}^{1.22} \cdot Bo^{1.13}, \text{ for } X_{tu} < 1
\tag{149}
\end{equation}

\begin{equation}
F_{JR} = 2 - 0.1 \cdot X_{tu}^{-0.28} \cdot Bo^{-0.33}, \text{ for } 1 < X_{tu} < 5
\tag{150}
\end{equation}

$Bo$ is the boiling number as defined in equation (151).

\begin{equation}
Bo = \frac{q}{G \cdot (H_{1,s} - H_{4,s})}
\tag{151}
\end{equation}

where the heat flux is calculated for the element from a previous iteration, $q = \frac{Q_{ip,i}}{A_i}$, $G$ is the mass flux, and the term $H_{1,s} - H_{4,s}$ is the saturated enthalpy difference. The parameter $F_{un}$ is a mixture correction factor (equation 152).

\begin{equation}
F_{un} = \left\{ \left[ 1 + (b_2 + b_3) \cdot (1 + b_4) \right] \cdot (1 + b_5) \right\}^{-1}
\tag{152}
\end{equation}

where $b_2 \ldots b_5$ are constants, defined in equations (153), (154), (155), (156), and (157).

\begin{equation}
b_2 = (1 - Y_t) \cdot \ln \left[ \frac{1.01 - Y_f}{1.01 - Y_v} \right] + Y_i \cdot \ln \left( \frac{Y_i}{Y_v} \right) + (Y_v - Y_f)^{1.5}
\tag{153}
\end{equation}

\begin{equation}
b_3 = 0, \text{ for } Y_i < 0.01
\tag{154}
\end{equation}

\begin{equation}
b_3 = (Y_v / Y_f)^{0.1} - 1, \text{ for } Y_i > 0.01
\tag{155}
\end{equation}

\begin{equation}
b_4 = 152 \cdot (p_s / p_{cr,mvc})
\tag{156}
\end{equation}

\begin{equation}
b_5 = 0.92 \cdot (Y_v - Y_f)^{0.001} \cdot (p_s / p_{cr,mvc})^{0.66}
\tag{157}
\end{equation}

where $Y_i$ and $Y_v$ are the mole fractions of the more volatile component in the liquid and the vapour respectively. $p_s$ is the local saturation pressure and $p_{cr,mvc}$ is the critical pressure of the more volatile component (MVC).

The pool boiling component of the heat transfer coefficient ($h_{pb}$) is from equation (158).
The local heat transfer coefficients, $h_1$ and $h_2$, are based on the assumption of the individual components only. This is calculated using the Stephan and Abrahamson correlation (159).

$$h = \frac{207 \cdot k_l \cdot \left( \frac{q \cdot b_1}{k_i \cdot T_s} \right)^{0.475} \cdot \left( \frac{Pr}{Pr_l} \right)^{0.581}}{b_1 \cdot \left( \frac{2 \cdot \sigma}{g \cdot (\rho_l - \rho_v)} \right)^{0.5}}$$ (159)

where $b_1$ is from equation (160) and $T_s$ is the local saturation temperature.

$$b_1 = 0.0146 \cdot \beta \cdot \left( \frac{2 \cdot \sigma}{g \cdot (\rho_l - \rho_v)} \right)^{0.5}$$ (160)

where $\sigma$ is refrigerant surface tension, and the contact angle, $\beta = 35^\circ$. The remaining terms in equation (148) are found from equation (161), (162), and the liquid-only heat transfer coefficient ($h_l$) is equation (163).

$$F_{me} = 1 - 0.35 \cdot (Y_v - Y_f)^{1.56}$$ (161)

$$F_{lg} = 2.37 \cdot (0.29 + 1/X_a)^{0.85}$$ (162)

$$h_l = 0.023 \cdot \frac{k_l}{D} \cdot \left( \frac{G \cdot (1-x) \cdot D}{\mu_l} \right)^{0.8} \cdot Pr_l^{0.4}$$ (163)

Although the model for evaporating heat transfer coefficient (equations 148 to 163) is for refrigerant mixtures, it may also be used for calculation of pure refrigerants. This is achieved by setting the composition of the MVC as $Y = 1$, and in doing so, the mixture correction factors reduce to unity, i.e., $F_{un} = 1$ and $F_{me} = 1$. The acceptability of this approach was confirmed in Jung and Radermacher (1993).

With reference to the method of adopting fictitious $\Delta T_G$ within the heat exchanger models, it can be seen that this approach is suitably handled by the chosen correlation for condensation $h_r$, where refrigerant apparent specific heat is employed (see equations 136 and 137). In contrast, the chosen correlation for evaporation $h_e$ relies on mixture compositions instead of glide characteristics. This was found to be the case for other such correlations (e.g., ASHRAE, 1997, Jung and Radermacher, 1989, Gaibel et al, 1994). To maintain consistency, the approach of
fictitious $\Delta T_G$ was integrated into the evaporation correlation using a method devised as part of this study. This method was based on linking fictitious compositions of the mixture MVC to the selected glide characteristics, and is detailed below.

![Diagram of calculation sequence for evaporator capacity](image)

Figure 53: Calculation sequence for evaporator capacity

From equations (152) to (161), it is seen the two mixture correction factors ($F_{un}$ and $F_{me}$) are a function of the mole fractions of the MVC within the mixture. Examination of the relationship
between MVC mole fractions and glide characteristics - $\Delta T_G$ and shape, $n_G$ - of several binary mixtures, indicated that properties could be indirectly linked to the calculation of $h_{r,e}$.

Subsequently, for a chosen glide characteristic the composition of the MVC in liquid ($Y_l$) and vapour ($Y_v$) phases over the range $x = 0 \rightarrow 1$ can be estimated.

First, the mole fraction in liquid phase at $x = 0$ is chosen, which is also equal to the vapour phase composition at $x = 1$. As mentioned in Section 2.4, when $n_G = 1$, the composition of a binary mixture was observed to be approximately equal for both components, so $Y_l(x = 0) \approx 0.5$ (assuming similar molar mass) based on analysis data for a number of refrigerant combinations. A larger $n_G$ indicates a lower proportion of MVC, whereas a smaller $n_G$ implies mixture is richer in the MVC.

As the relative size of the vapour space increases, a higher proportion of the MVC migrates to the vapour phase, so as $x \rightarrow 1$, $Y_l$ reduces. This shift in composition was linked to the size and shape of glide, and could be approximated with a power-function (equation 164).

$$Y_l(x = 1) = Y_l(x = 0) - a_Y \cdot \left( \frac{\Delta T_G}{n_G} \right)^{n_Y} \quad (164)$$

where $a_Y = 0.07$ and $n_Y = 0.53$ based on the refrigerants listed in Table 31 of Appendix 2, and a greater difference in the components' partial pressures leads to a smaller $a_Y$ and $n_Y$. For a correct solution of equation (164), $a_Y$ and $n_Y$ may require adjustment to ensure that $Y_l(x = 1) > 0$, always.

The corresponding vapour phase compositions are also needed. Since $Y_v(x = 1) = Y_l(x = 0)$, it is necessary to estimate the equilibrium vapour composition at $x = 0$, and it was found that the variation in $Y_v$ over $x = 0 \rightarrow 1$ provided a close correlation with $Y_l$ over $x = 0 \rightarrow 1$ (equation 165).

$$Y_v(x = 0) = Y_v(x = 1) + \left[ Y_l(x = 0) - Y_l(x = 1) \right] \cdot n_{n_Y} \quad (165)$$

where the index was found to be about $n_{n_Y} = 1.4$ for most of the assessed refrigerants.
Finally, the liquid and vapour compositions at local vapour qualities were estimated using the same approach as for local saturation temperature \( T_x \), equation (104), so equation (165) was modified accordingly for liquid (equation 166) and vapour (equation 167) phases.

\[
Y_l(x) = Y_l(x = 0) - \left[ Y_l(x = 0) - Y_l(x = 1) \right] \left( 1 - [1 - x]^{\alpha_l} \right) \quad (166)
\]

\[
Y_v(x) = Y_v(x = 0) - \left[ Y_v(x = 0) - Y_v(x = 1) \right] \left( 1 - [1 - x]^{\alpha_v} \right) \quad (167)
\]

These formulas were subsequently used in the calculation of evaporator heat transfer coefficient.

The pressure loss for the mixture refrigerant within the two-phase regions has been accounted for using the correlation of Jung and Radermacher (1989, 1993) as recommended by others (e.g., Wang and Chato, 1995), equation (168).

\[
\frac{dp}{dz} = \frac{2 \cdot \lambda_l \cdot G^2}{D \cdot \rho_l} \cdot \frac{1}{\Delta x} \left( \int_{\phi_p} \phi_p \cdot dx \right) \quad (168)
\]

where \( \lambda_l \) is the liquid-only friction factor (equation 169), and \( \phi_p \) is a two-phase multiplier (equation 170), which is defined differently from condensation (i.e., equation 146).

\[
\lambda_l = 0.046 \cdot Re_l^{-0.2} \quad (169)
\]

\[
\phi_p = 12.82 \cdot X_n^{-1.47} \cdot (1 - x)^{1.8} \quad (170)
\]

Local void fraction is required for various heat transfer and pressure drop calculations. ASHRAE (1997) and Chen and Kruse (1995) give a selection of different models. Chen and Kruse compared results from computations against measurements with R23/R152a and found the Hughmark correlation was most suited, so it was selected presently. Equation (171) is the basic formula for void fraction \( \nu \).

\[
\nu = a \left( \frac{1}{(x / \rho_v) + (1-x) / \rho_l} \right) \quad (171)
\]

where \( a \) is obtained from equation (172).

\[
a = 0.7266 - 3.482 \times 10^4 \cdot \theta - 0.845 \frac{0.06}{\theta} + 0.0601 \cdot \theta^{1/3} \quad (172)
\]

where the coefficient \( \theta \) is from equation (173).
Reynolds and Froude number ($Fr$) are from equations (174) and (175), respectively.

$$Re_v = \frac{D \cdot G}{\mu_i + \nu \cdot (\mu_v - \mu_t)}$$  \hspace{1cm} (174)

$$Fr = \frac{1}{g \cdot G} \left( \frac{G \cdot x}{(\nu / \alpha) \cdot \rho_v} \right)^2$$  \hspace{1cm} (175)

Where the mass flux ($G$) is calculated in the usual way (equation 176).

$$G = \frac{4 \cdot \dot{m}_r}{\pi \cdot D^2}$$  \hspace{1cm} (176)

Evidently, determination of $\nu$ is implicit and therefore required the calculation to iterate through equations (171) to (176) until input and output values of $\nu$ converged. The calculation was initiated by setting $\nu = x$.

### 3.3.4 Single-phase heat transfer coefficient

Single-phase heat transfer coefficient was required for superheat, desuperheat, and subcooling regions, or the HTF. It is assumed that fully developed laminar or turbulent flow exists. For a laminar region ($Re < 2300$) where there is a uniform surface heat flux (as it assumed in the model), the Nusselt number ($Nu$) is according to equation (177) (Incropera and De Witt, 1990).

$$Nu = 4.36$$  \hspace{1cm} (177)

For turbulent flow ($Re > 2300$), the accepted Dittus-Boelter correlation (Incropera and De Witt, 1990) for Nusselt number is employed (equation 178).

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^n$$  \hspace{1cm} (178)

where the value of $n$ depends on the whether the fluid is heating ($n = 0.4$) or cooling ($n = 0.3$). Reynolds and Prandtl numbers are found in the usual way (equation 179 and 180).

$$Re = \frac{\rho \cdot u \cdot D}{\mu}$$  \hspace{1cm} (179)

$$Pr = \frac{C_p \cdot \mu}{k}$$  \hspace{1cm} (180)
Bulk velocity \( u \) in equation (179) is determined for in-tube flow (equation 181) or from the flow area between the two tubes for a concentric tube (equation 182) in the case of using a HTF.

\[
\begin{align*}
\frac{u}{\rho} &= \frac{4}{\pi \cdot D^2} \\
\frac{u}{\rho} &= \frac{4}{\pi \cdot (D_o^2 - D^2)}
\end{align*}
\]  

(181)  

(182)

Heat transfer coefficient is from equation (183).

\[
h = \frac{Nu \cdot k}{D}
\]  

(183)

Although pressure loss of the HTF is irrelevant to the purpose of the model, it is important to the single-phase regions of refrigerant flow. The conventional approach detailed in Incroprea and De Witt (1990) was used.

\[
\frac{dp}{dz} = \frac{\lambda \cdot \rho \cdot u^2}{2 \cdot D}
\]  

(184)

where the friction factor is determined by Reynolds number (equation 185).

\[
\lambda = a / Re^n
\]  

(185)

where for laminar flow \( a = 64 \) and \( n = 1 \), for \( Re \leq 20000 \), \( a = 0.316 \) and \( n = 0.25 \), and for \( Re > 20000 \), \( a = 0.184 \) and \( n = 0.2 \); throughout, the pipe surface was assumed to be smooth.

| Table 25: Heat exchanger simulation input values |
|---|---|---|
| Inputs | Condenser | Evaporator |
| Refrigerant dew-point temperature | +20 to +60°C (40°C) | +10 to -40°C (0°C) |
| Superheat | 0 to 60 K (0 K) | 0 to 10 K (0 K) |
| TEV inlet temperature | 0 to 5 K below \( T_{bp} \) (0 K) | 20 to 60°C (30°C) |
| Temperature glide, \( \Delta T_G(p_{\text{sat}}) \) | 0 to 10 K (10 K) | 0 to 10 K (10 K) |
| Glide gradient; \( a_1, a_2 \) in equation (103) | as equation (103) (0, 0) | as equation (103) (0, 0) |
| Glide shape, | \( 0.25 < n_G < 4 \) \( (n_G = 1) \) | \( 0.25 < n_G < 4 \) \( (n_G = 1) \) |
| HTF mass flow rate | 0.4 kg s\(^{-1}\) | 0.3 kg s\(^{-1}\) |
| Exchanger length | 3 m | 3 m |
| Number of refrigerant passes | 1 to 10 (6) | 1 to 10 (6) |
| Geometry | counter, parallel | counter, parallel |
3.4 Observations from component performance simulation

The condenser and evaporator models were used to examine the effects associated with zeotropic refrigerants on their performance, which were highlighted in Section 2.5. The following presents detailed discussions of the component behaviour based on the simulation of the heat exchangers, generated to represent the data expected from calorimeter testing. The model inputs were based on typical commercial heat exchangers, and are listed in Table 25, and when a given variable is not being examined it was fixed according to the value in parentheses. In terms of fluid choice, water was selected as the HTF, and refrigerant properties (other than glide) were based on a blend of 50% R290 and 50% R600a by mass.

Influence of approach temperatures and variable heat transfer coefficient

The discussion in Section 2.4.2 considered the behaviour of fluid temperature profiles to understand as the refrigerant dew-point temperature gets gradually closer to the HTF temperature and to explain the existence of zero virtual capacity whilst a temperature difference is present due to the glide. Since the condenser exhibits full phase-change of the refrigerant (as opposed to the evaporator) this will be used in the first assessment.

![Temperature Profiles](image)

*Figure 54: Counter-flow condenser, \( C_r > C_{HTF} \)  Figure 55: Counter-flow condenser, \( C_r \approx C_{HTF} \)

Figure 54 shows temperature profiles for a large \( \Delta T_{c(m)} \) of 30 K. Since \( C_r > C_{HTF} \) the HTF temperature change is larger and the temperature profiles taper together as HTF approaches the refrigerant inlet. When \( \Delta T_{c(m)} \) was reduced to 20 K, a point is reached where \( C_r \approx C_{HTF} \), resulting in an equidistant temperature profile throughout the exchanger length (Figure 55).
Figure 56 illustrates the temperature profile where $\Delta T_{e(in)}$ was reduced to 10.1 K, resulting in a situation where $\dot{C}_v' \ll \dot{C}_{HTF}$. Observation of the data-points corresponding to equally incremented changes in vapour quality (i.e. spacing of the data points) shows a pronounced shift of heat transfer towards the refrigerant inlet. Since the temperature difference is minimal towards the refrigerant outlet, the majority of $\Delta T_G$ is "pushed" towards the refrigerant inlet end of the heat exchanger with the greatest $\Delta T$ with majority of heat transfer occurring within one-third of the exchanger. Extrapolating this behaviour until $\Delta T_{e(in)}$ is reduced to its minimum when $T_{e(out)} \rightarrow T_{HTF(out)}$, all heat transfer is pushed to towards the refrigerant inlet (Figure 56) and eventually no heat transfer can occur when $\Delta T_{e(in)} = \Delta T_G$. The entire $\Delta T_G$ will be forced to the point at which the refrigerant enters of the exchanger. Simultaneously, the refrigerant mass flow necessary to give up an infinitesimally small quantity of heat converges on zero $m_r$.

Figure 56: Counter-flow condenser with very small approach, $C_v' \ll C_{HTF}$

Parallel-flow condensers were found to behave with identical manner, where the refrigerant enters at the same point as the HTF where the initial temperature difference is greatest produces the majority of heat transferred. As both fluids progress along the exchanger length, the temperature difference reduces until they meet at the exchanger exit. This behaviour similarly applies to the evaporator; a $\Delta T_{e(in)} \rightarrow 0$ K, $m_r$ and $Q_e$ reduce and the majority of the heat transfer occurs at the refrigerant inlet where the temperature difference is greatest. Eventually when $T_{e,dp} \rightarrow T_{HTF(out)}$, $Q_e \rightarrow 0$. 

188
Figure 58: Characteristic line for condenser with pure and zeotropic refrigerant

For comparison, exchanger temperature profiles using a pure refrigerant are shown in Figure 57, where $\Delta T_{c(m)}$ was gradually reduced until a minimal temperature difference was achieved as in Figure 56. Obviously, the refrigerant does not exhibit a change in temperature, and as a result, a near-even distribution of the equally incremental $x$ data-points is seen which indicates that a similar amount of heat is transferred across the total exchanger length, even as $Q \rightarrow 0$.

Figure 59: Influence of glide shape on temperature profiles for small $\Delta T_{c(m)}$

The simulation was also used to illustrate the condenser characteristics using a pure refrigerant and a zeotrope (Figure 58). An initial observation is that neither characteristics are linear, which equates to a non-constant basic rating. However at high $\Delta T_{c(m)}$ a pure refrigerant gives an
approximately constant basic rating. For the zeotropic refrigerant, the curvature tends to span over a wider range, giving a variable basic rating.

Effect of non-linear glide shape on temperature profiles
The influence of the glide shape on temperature profile and exchanger characteristic was addressed. Figure 59 illustrates how the shape of the glide affects the temperature profiles. It can be seen that as the shape moves from concave \( n_G = 3 \) to linear to convex \( n_G = \frac{1}{2} \), the distribution of the \( x \) elements are pushed towards the higher \( \Delta T \) at the refrigerant inlet. Where \( n_G = \frac{1}{2} \), a significant proportion of \( \Delta T_G \) and heat transfer occurred within one-tenth of the heat exchanger. Conversely, when \( n_G = 3 \), there is a broader distribution of heat transfer.

The variation in glide shape produces a change in capacity despite a fixed \( \Delta T_{c(\text{in})} \). Shown in Figure 60 is the relative capacity of the condenser for a range of shapes over a series of \( \Delta T_{c(\text{in})} \).

For individual glide shapes, the influence on capacity is greatest at small \( \Delta T_{c(\text{in})} \). For convex glide shapes the capacities are always larger than that of the linear case, whilst concave shape always produces a lower capacity. In the case of the evaporator the opposite is true concerning the impact of glide shape, i.e. when \( n_G < 1 \), the capacity is greater than when \( n_G = 1 \).

\[ \text{Figure 60: Variation of capacity with } \Delta T_{c(\text{in})} \text{ for different } n_G \]

Effect of sensible heat transfer
In reality a proportion of the condenser or evaporator capacity is due to sensible heat transfer, and this is normally small compared to the two-phase region, and because \( h \) is lower a relatively large exchanger area is required. On this basis, it is useful to assess the condenser and evaporator
performance in light of sensible heat transfer, and is illustrated in Figure 61, which shows condenser temperature profiles with 30 K desuperheat and 3 K subcooling. The high use of surface area is confirmed where the sensible heat transfer is seen to use approximately one-third of the exchanger area (distance), whilst representing a much smaller proportion of the total heat transferred. A similar effect was found with the evaporator. The implication is that \( Q \) reduces disproportionately and thus influences basic rating.

Sensible heat transfer also influences the minimum approach temperature difference. As demonstrated previously, \( \Delta T_{in} \) may be gradually reduced until \( \Delta T_{min(in)} \) is reached, and this corresponds to \( \Delta T_c \) when considering phase-change region only. If a condenser comprises subcooling \( \Delta T_{sc} \) then an additional offset is produced for \( \Delta T_{c,min(in)} \) (equation 186). Similarly with the evaporator, existence of superheat \( \Delta T_{sh} \) will also offset \( \Delta T_{c,min(in)} \) (equation 187).

\[
\Delta T_{c,min(in)} = \Delta T_G + \Delta T_{sc}
\]

(186)

\[
\Delta T_{c,min(in)} = \Delta T_{G,e} + \Delta T_{sh}
\]

(187)

![Figure 61: Condenser temperature profile with single-phase heat transfer](image)

It is noted that whilst the presence of desuperheat \( \Delta T_{sh} \) within the condenser does introduce a further offset, the definition of \( \Delta T_{in} \) is based on the inlet saturation temperature \( T_{c,dp} \), so in this context it does not directly affect \( \Delta T_{c,min(in)} \).
Figure 62 shows various condenser characteristics based on $T_{c, dp}$, to illustrate the influence of $\Delta T_{sc}$ or $\Delta T_{dbh}$. In terms of approach temperature, it is seen that without subcooling, $\Delta T_{c, min(m)} = \Delta T_G$, i.e. $10$ K, but when subcooling is included, $\Delta T_{c, min(m)} = \Delta T_G + \Delta T_{sc}$, i.e. $13$ K.

Regarding the effect on condenser capacity, inclusion of desuperheat or subcooling or both produces an overall reduction for a given $\Delta T_{c(m)}$. The greatest reduction occurs when both $\Delta T_{sc}$ and $\Delta T_{dbh}$ are included. It is also seen that the actual shape of the overall characteristic becomes more linear.

![Figure 62: Condenser characteristic affected by single-phase heat transfer](image)

**Effect of pressure loss**

Pressure loss along the refrigerant flow path produces a reduction in saturation pressure and thus a change in local saturation temperature. The simulation was used to observe the subsequent effect of relatively large $\Delta p$ (for clarity) on local saturation temperature ($T_r$), Figure 63. The dashed lines represent $T_{dp}$ and $T_{bp}$ that correspond to the inlet pressure. $\Delta p$ along the flow path causes these saturation temperatures to fall, $T_{c, dp}$ (local) and $T_{c, bp}$ (local), and the corresponding local $T_r$ is shown passing from the inlet $T_{c, dp}$ to the outlet $T_{c, bp}$. Whilst $T_r$ lowers as a result of $\Delta T_G$, an additional reduction occurs simultaneously as pressure falls (in this example, equivalent to about $5$ K in $T_{c, dp}$). The majority of the contribution from $\Delta p$ is seen to occur in the first half of the condenser, where the high vapour quality produces the greater effect. The effect is opposite in the evaporator; whilst $\Delta p$ reduces local saturation pressure $\Delta T_G$ is partially neutralised, rather than complimenting the glide. For both condenser and evaporator, the effective shift in
$\Delta T_c$ alters $\dot{C}'$, and therefore $B$, thus this phenomenon should be accounted for within the revised exchanger characteristic.

Figure 63: Effect of pressure drop on refrigerant temperature in condenser

Figure 64 presents a condenser characteristic (for two-phase only), where the reference state corresponds to $T_{dp}$ at condenser inlet pressure ($T_{c,dp(in)}$) and also $T_{dp}$ at outlet pressure ($T_{c,dp(out)}$). With $\Delta T_{c(in)}$ increasing, $Q_c$, $m_r$ and thus $\Delta p$ increases which is manifest in a greater differential between $T_{c,dp(in)}$ and $T_{c,dp(out)}$. The existence of two possible characteristics indicates the importance of defining the appropriate reference $T_{dp}$ for rating purposes. Because the condenser is ultimately to be linked to the compressor discharge, then the correct location for the reference $T_{dp}$ has to be the condenser inlet, i.e., $T_{c,dp(in)}$. Similarly, the evaporator has to be linked to the compressor suction and therefore the correct location for the evaporator $T_{dp}$ must be the evaporator outlet, $T_{e,dp(out)}$.

A further complication is introduced when considering the effect of $\Delta p$ within the sensible heat region. In the case of desuperheat, $T_{dp}$ at the onset of condensation will be lower than that implied by the $T_{dp}$ that corresponds to the pressure of the superheated vapour at the inlet of the condenser. Nevertheless, to be consistent with the reference condition at the compressor outlet, $T_{c,dp(in)}$ must accommodate the single-phase pressure drop. Similarly, the evaporator $T_{dp}$ corresponding to outlet pressure will be lower than $T_{dp}$ at $x = 1$ due to $\Delta p$ in the superheat.
region. Therefore, $T_{e,dp(m)}$ includes the contribution of single-phase pressure drop within the superheat region.

![Graph](image)

*Figure 64: Effect of pressure loss on condenser characteristic (two-phase only)*

**Aspects unique to the evaporator**

Additional consideration is required for the evaporator in terms of the effect of intersection of $\Delta T_G$ due to its dependency on $T_{TEV(m)}$. The simulation was used to understand the consequences of this factor.

![Graph](image)

*Figure 65: Effect of different inlet temperature on evaporator capacity*

Figure 65 illustrates two sets of temperature profile data, where refrigerant enters the TEV at +20°C and also at +60°C. The former condition produces a lower $x_e(m)$ than the latter, and this is reflected in the size of their respective $\Delta T_G$, although it must be cautioned that the full $\Delta T_G$ is
obscured by differing contribution from pressure drop. The evaporator $\Delta T_G$ is greater when $T_{TEV(in)} = +20^\circ C$, thereby producing a larger $\Delta T_{(in)}$ and hence higher $Q_e$ which is evidenced by the lower HTF outlet temperature.

Table 26: Contribution of TEV inlet temperature to factors affecting capacity

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Low $T_{TEV(in)}$</th>
<th>High $T_{TEV(in)}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Contribution</td>
<td>Effect on $Q_e$</td>
</tr>
<tr>
<td>$x_e(m)$</td>
<td>$\rightarrow 0$</td>
<td>$\rightarrow -$</td>
</tr>
<tr>
<td>$T_{ei}$</td>
<td>lower</td>
<td>increase</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>smaller</td>
<td>increase</td>
</tr>
<tr>
<td>$U$</td>
<td>smaller</td>
<td>reduce</td>
</tr>
<tr>
<td>$\Delta T_G$</td>
<td>Larger</td>
<td>reduce</td>
</tr>
</tbody>
</table>

Apart from $\Delta T_G$ intersection, there are other factors associated with the evaporator inlet conditions that can cause a variation of $Q_e$. Since $h_r$ varies with vapour quality, a higher $T_{TEV(in)}$ changes $U$ and therefore $Q_e$. Further, as $x_e(in) \rightarrow 1$, higher $\Delta p$ occurs and therefore amplifies the corresponding reduction in local $T_r$. Contrary to the condenser, this manifests as a smaller $\Delta T_{G,e}$ across the evaporator. A summary of these effects is detailed in Table 26; increasing $T_{TEV(in)}$ lessens $\Delta T_{G,e}$ due to both effects of intersection and higher $\Delta p$, whilst average $U$ may be greater or smaller which would manifest as a variation of $Q_e$.

Figure 66: Effect of different inlet temperature on evaporator capacity
An example of the net effect on $Q_e$ due to the different contributions from $T_{TEV}(in)$ can be observed in Figure 66. With smaller $\Delta T_{e(in)}$, a $T_{TEV}(in) = 60^\circ C$ produces a $Q_e$ less than when $T_{TEV}(in) = 30^\circ C$, although at larger $\Delta T_{e(in)}$ the case is reversed as the greater $Q_e$ occurs when $T_{TEV}(in) = 60^\circ C$. This cross-over represents the point at which the positive influence of lower $T_{e(in)}$ and $\Delta p$ dominate over the negative influence of lower $U$ and larger $\Delta T_{G,e}$.

3.5 Proposed characterisation methods

The purpose of this section is to offer possible alternative rating methods for when components use zeotropic refrigerants. Existing methods mainly use linear line-fits to describe the performance of evaporators and condensers, and polynomial equations for the compressor. Characterisation methods can go further than this by adopting more complex models. Figure 67 is an overview of the range of possible options available for component characterisation, and the degree of complexity typically indicates its accuracy. An approach based on functional line-fitting and approximate correlation with specific variables was chosen as a compromise between simplicity and accuracy. Fictitious test data obtained from the simulation models in Section 3.2 were used to assist in the development of the new methods.

![Figure 67: Range of performance prediction methods for component rating](image)

3.5.1 Compressor

The general expressions for evaporating and condensing capacities as a function of $T_{e,dp}$ and $T_{e,dp}$ are given in equation (188) and (189) respectively.
As discussed, it is also necessary to characterise the degree of discharge superheat \( \Delta T_{dsh} \), also as a function of the operating conditions (equation 190).

\[
\Delta T_{dsh} = f\left(T_{e,dp}, T_{d,dp}\right) \tag{190}
\]

Applying these general expressions to the established form (equations 76 and 77) leads to equation (191) for evaporating capacity, and equation (193) for compressor power. Although the compressor capacity is normally calculated from

\[
Q_{e,comp} = f\left(T_{e,dp}, T_{d,dp}\right) \tag{188}
\]

\[
Q_{c,comp} = f\left(T_{e,dp}, T_{d,dp}\right) \tag{189}
\]

\[
W_{comp} = c_1 + c_2 \cdot T_{e,dp} + c_3 \cdot T_{e,dp}^2 + c_4 \cdot T_{e,dp}^3 + c_5 \cdot T_{e,dp} \cdot T_{c,dp} + c_6 \cdot T_{c,dp}^2 + c_7 \cdot T_{c,dp}^3 + c_8 \cdot T_{c,dp} \cdot T_{e,dp}^2 + c_9 \cdot T_{e,dp} \cdot T_{d,dp} + c_{10} \cdot T_{d,dp}^3 \tag{191}
\]

\[
Q_{c,comp} = c_1 + c_2 \cdot T_{e,dp} + c_3 \cdot T_{e,dp} + c_4 \cdot T_{e,dp}^2 + c_5 \cdot T_{e,dp} \cdot T_{c,dp} + c_6 \cdot T_{c,dp}^2 + c_7 \cdot T_{c,dp}^3 + c_8 \cdot T_{c,dp} \cdot T_{e,dp}^2 + c_9 \cdot T_{e,dp} \cdot T_{d,dp} + c_{10} \cdot T_{d,dp}^3 \tag{192}
\]

\[
W_{comp} = c_1 + c_2 \cdot T_{e,dp} + c_3 \cdot T_{e,dp} + c_4 \cdot T_{e,dp}^2 + c_5 \cdot T_{e,dp} \cdot T_{c,dp} + c_6 \cdot T_{c,dp}^2 + c_7 \cdot T_{c,dp}^3 + c_8 \cdot T_{c,dp} \cdot T_{e,dp}^2 + c_9 \cdot T_{e,dp} \cdot T_{d,dp} + c_{10} \cdot T_{d,dp}^3 \tag{193}
\]

where the coefficients \( c_1 \ldots c_{10} \) are specific to \( Q_{e,comp} \), \( Q_{c,comp} \), and \( W_{comp} \) and obtained from reduction of measured data, \( T_{e,dp} \) is saturated dew-point temperature corresponding to suction pressure, and \( T_{c,dp} \) is saturated dew-point temperature corresponding to discharge pressure.

Although not currently available from catalogue data, \( \Delta T_{dsh} \) is known to vary according to operating conditions, and the same form was also found to be appropriate in this case (equation 194).

\[
\Delta T_{dsh} = c_1 + c_2 \cdot T_{e,dp} + c_3 \cdot T_{e,dp} + c_4 \cdot T_{e,dp}^2 + c_5 \cdot T_{e,dp} \cdot T_{c,dp} + c_6 \cdot T_{c,dp}^2 + c_7 \cdot T_{c,dp}^3 + c_8 \cdot T_{c,dp} \cdot T_{e,dp}^2 + c_9 \cdot T_{e,dp} \cdot T_{d,dp} + c_{10} \cdot T_{d,dp}^3 \tag{194}
\]

where the coefficients \( c_1 \ldots c_{10} \) are obtained from reduction of measured data.
3.5.2 Expansion device

The only influence identified was with respect to selection of the appropriate reference conditions for determining refrigerant enthalpies. Therefore, it was concluded to maintain the existing characterisation method, although to ensure consistent definitions, i.e., equation (81).

3.5.3 Heat exchangers

Of interest to the performance testing, rating, catalogue presentation and selection stages are the following:

- \( \Delta T_{\text{min(in)}} \) is greater than zero, since it is offset by \( \Delta T_G \) and \( \Delta T_{sc} \) in the condenser and \( \Delta T_{sh} \) in the evaporator.
- The properties of \( \Delta T_G \) and the other parameters associated with the exchanger configuration mean that the condenser and evaporator characteristic is not normally linear, so \( B \) is rarely constant.
- Sensible heat transfer within the condenser affects the capacity which is important because compressor discharge superheat varies with \( T_{e,dp} \) and \( T_{e,dp} \), thus the condenser rating should be a function of \( \Delta T_{sh} \).
- Pressure drop in the heat exchangers require that the reference states must be correctly chosen to ensure they are consistent with the compressor inlet and outlet conditions: \( T_{c,dp(in)} \) and \( T_{e,dp(out)} \) must be employed in component rating.
- Evaporator capacity is subject to variation according to \( T_{TEV(in)} \) due to intersection of the \( \Delta T_G \), thus \( T_{TEV(in)} \) should be accounted for in evaporator rating.

As a basis for the revised rating method, two general expressions for heat exchanger capacity can be made. For a fixed geometry condenser, the capacity is a function of three parameters (equation 195).

\[
Q_{c,\text{cond}} = f \left( T_{c,dp(in)}, \Delta T_{sh}, \Delta T_{sc} \right)
\] (195)

Similarly, the capacity of a fixed geometry evaporator can also be expressed as a function of two operating parameters (equation 196).

\[
Q_{e,\text{evap}} = f \left( T_{e,dp(out)}, T_{TEV(in)} \right)
\] (196)
The conventional rating approach for both condenser and evaporator, based on the NTU-\(\varepsilon\) method, was chosen to develop a revised rating method. A general rating method was first derived for any heat exchanger using a zeotrope, which was then adapted to account for particular characteristics of the condenser or evaporator. Through observing the trends in \(\varepsilon\) and \(R_C\), it was found that \(\ln(\varepsilon)\) could be accurately approximated as a linear function of \(\ln(R_C)\) over a typical range of \(\Delta T_{in}\), although the gradient when \(\dot{C}_{\text{min}} = \dot{C}_{HTF}\) was distinctly different from when \(\dot{C}_{\text{min}} = \dot{C}_r\). This was found to be applicable to the three main exchanger geometries, except for the unusual situation of \(R_C \rightarrow 0\) in any exchanger type (i.e., a very small \(\Delta T_G\)). In response to this, it was deemed necessary to develop two general rating equations applicable to each case of \(\dot{C}_{\text{min}} = \dot{C}_{HTF}\) or \(\dot{C}_{\text{min}} = \dot{C}_r\).

Initially, the original calculation of exchanger capacity were simplified (equation 197) where the capacity is for the two-phase region only.

\[
O_q = B \cdot \Delta T_{in} \tag{197}
\]

\(B\) is the product of \(\varepsilon\) and \(\dot{C}_{\text{min}}\). In the first case, the derivation is for \(\dot{C}_{\text{min}} = \dot{C}_{HTF}\) (equation 198).

\[
B = \varepsilon \cdot \dot{C}_{HTF} \tag{198}
\]

The approximate linear relationship between \(\ln(\varepsilon)\) and \(\ln(R_C)\) as noted above, was substituted into equation (198), to give equation (199).

\[
B = \alpha \cdot R_C^n \cdot \dot{C}_{HTF} \tag{199}
\]

where \(R_C\) is the ratio of \(\dot{C}_{HTF}\) to \(\dot{C}_r\), and \(\dot{C}_r\) was shown (Section 2.4) to be equal to \(Q_q / \Delta T_G\) when based on an entire two-phase heat exchanger (from equation 95). Equation (199) can be rewritten as equation (200).

\[
B = \alpha \left( \frac{\dot{C}_{HTF} \cdot \Delta T_G}{Q_q} \right) \cdot \dot{C}_{HTF} \tag{200}
\]

Substituting equation (200) back into equation (197) and rearranging, provides an explicit expression for the basic rating (equation 201).
Correspondingly, by substituting equation (201) into the initial rating equation (197), a final formula for capacity is found (equation 202).

\[ Q_p = a^{1/(n+1)} \cdot \hat{C}_{HTF} \cdot \frac{\Delta T_G}{\Delta T_{in}} \]  

(202)

For the second case when \( C_{mn} = \hat{C}_r' \), swapping \( \hat{C}_r' \) for \( \hat{C}_{HTF} \) in equation (198) gives equation (203).

\[ B = \varepsilon \cdot \hat{C}_r' \]  

(203)

Using the same power-law expression for \( \varepsilon \) and \( R_c \) as above, and substituting into equation (203), gives equation (204).

\[ B = a \cdot R_c^n \cdot \hat{C}_r' \]  

(204)

Now that \( R_c = \hat{C}_r' / \hat{C}_{HTF} \), and again using \( \hat{C}_r' = Q_p / \Delta T_G \) for the entire two-phase heat exchanger, equation (204) was rewritten as equation (205).

\[ B = a \left( \frac{Q_p}{\hat{C}_{HTF} \cdot \Delta T_G} \right)^n \cdot \frac{Q_p}{\Delta T_G} \]  

(205)

Substituting equation (205) back into equation (197) and rearranging, provides an explicit expression for the basic rating (equation 206).

\[ B = \frac{1}{a^{1/n}} \cdot \left( \frac{\Delta T_G}{\Delta T_{in}} \right)^{(n+1)/n} \cdot \hat{C}_{HTF} \]  

(206)

Finally, by substituting equation (206) into the initial rating equation (197), a final formula for capacity is obtained for when \( \hat{C}_{mn} = \hat{C}_r' \) (equation 207).

\[ Q_p = a^{-1/n} \cdot \hat{C}_{HTF} \cdot \Delta T_G^{(n+1)/n} \cdot \Delta T_{in}^{1/n} \]  

(207)

When using equations (202) and (207) for characterisation of a condenser or an evaporator, it is useful to know the minimum number of performance data needed to determine the values of \( a \) and \( n \). Assuming that over the intended operating range \( \hat{C}_{mn} = \hat{C}_{HTF} \) always, then a minimum
of two $Q - \Delta T_{(in)}$ measurements would be needed, and the same applies if $\dot{C}_{\text{min}} = \dot{C}'$ over the intended range of operating conditions.

A small $\Delta T_A$ implies a larger $C'_r$, which suggests that generally $\dot{C}_{\text{min}} = \dot{C}_{\text{HTF}}$ for most small glide refrigerants. If $\dot{C}'_r \approx \dot{C}_{\text{HTF}}$ over the application range, three $Q - \Delta T_{(in)}$ measurements instead of four would be required. This is because the value of $a$ in equations (202) and (207) is the same; the form $\varepsilon = a \cdot R^c$ implies the values of $\varepsilon$ for both $\dot{C}_{\text{min}} = \dot{C}_{\text{HTF}}$ and $\dot{C}_{\text{min}} = \dot{C}'_r$ converge when $R_c = 1$. Nevertheless, since the power-law formula is an approximation, it would be pertinent to determine $a$ and $n$ from a larger number of data-points.

**Condenser**

If equations (202) and (207) are to be employed for a rating method, they should be modified to account for the sensible heat transfer regions and pressure drop. First, the generic rating equations were adapted to account for the contribution of subcooling. Normally, $\Delta T_{\text{sc}}$ is assumed to remain constant over the entire operating range, and given that the liquid refrigerant $C_p$ is usually constant over a typical temperature range implies $Q_{\text{sc}}$ is proportional to the refrigerant mass flow. Assuming that the refrigerant exhibits a proportional change in saturated enthalpy difference over the expected operating $T_{c,dp}$, the contribution of $Q_{\text{sc}}$ can be approximated from equation (208).

$$Q_{\text{sc}} = b_{\text{sc}} \cdot Q_{\text{dp}}$$

(208)

where $b_{\text{sc}}$ represents the basic rating of the subcooled region for a fixed $\Delta T_{\text{sc}}$. Although it was stated that subcooling is assumed to be constant, equation (208) could be easily modified to account for variations by changing $b_{\text{sc}}$ proportionally, since $b_{\text{sc}} / \Delta T_{\text{sc}} \approx \text{constant}$.

Similarly, the capacity of the desuperheat region was handled independently. Unlike subcooling, $\Delta T_{\text{dsh}}$ is subject to variation because of compressor discharge conditions, and therefore $Q_{\text{dsh}}$ is not solely dependent upon refrigerant mass flow. Further, as $\Delta T_{\text{dsh}}$ varies the surface area available for two-phase heat transfer is affected, and therefore the overall condenser capacity for the chosen $\Delta T_{c(in)}$ will differ. It was found that for either a fixed $\dot{C}_{\text{min}}$ or a fixed $\Delta T_{\text{dsh}}$ there was
a close correlation between the respective \( f(\Delta T_{c,\text{(in)}}) \) term in equation (202) or (207) and \( Q_{dsh} \) (equation 209).

\[
Q_{dsh} = a_{dsh} \cdot f(\Delta T_{c,\text{(in)}})^{\eta_{dsh}}
\]  

(209)

where \( a_{dsh} \) was found to vary proportionally with \( \Delta T_{dsh} \), and \( \eta_{dsh} \) was roughly constant (for each \( C_{min} \) case). By including a linear variation of capacity with \( \Delta T_{dsh} \), and putting \( \Delta T_{\text{c,in}}^{*} = \Delta T_{c,\text{(in)}}^{1/(n-1)} \) for when \( C_{min} = \dot{C}_{HTF} \), and putting \( \Delta T_{\text{c,in}}^{*} = \Delta T_{c,\text{(in)}}^{1/2} \) for when \( C_{min} = \dot{C}' \), a general rating formula was produced for desuperheat capacity (equation 210), which was adopted to enable the integration of desuperheat into the condenser rating.

\[
Q_{dsh} = b_{dsh} \cdot \Delta T_{dsh} \cdot (\Delta T_{\text{c,in}}^{*})^{\eta_{dsh}}
\]  

(210)

where \( b_{dsh} \) (obtained from \( a_{dsh} + \Delta T_{dsh} \)) and \( \eta_{dsh} \) are specific to each \( C_{min} \) case.

The final parameter for inclusion in the condenser rating is pressure drop. Given that the general rating formulas (equation 202 and 207) are dependent upon \( \Delta T_{G} \), it is convenient to adjust \( \Delta T_{G} \) accordingly. Thus, the effective temperature glide (\( \Delta T'_{G} \)), which is the difference between isobaric \( \Delta T_{G} \) and equivalent change in saturation temperature (\( \Delta T_{c,\Delta p} \)) (equation 211), can be applied to equations (202) and (207).

\[
\Delta T'_{G} = \Delta T_{G} - \Delta T_{c,\Delta p}
\]  

(211)

A basic formula for estimating \( \Delta T_{c,\Delta p} \) was deduced from relating \( \Delta p \) with capacity, and the subsequent shift along the saturation curve. From Bernoulli, \( \Delta p \propto u'^{2} \) assuming refrigerant properties do not change significantly over the anticipated operating range. Similarly, for a fixed geometry velocity can be substituted for mass flow, so \( \Delta p \propto m'^{2} \). Ignoring any variation in enthalpy difference, the pressure drop can therefore be approximated as a function of \( Q_{c} \) (equation 212).

\[
\Delta p_{c} = c_{1} \cdot Q_{c}^{2}
\]  

(212)

Over a limited range of operating conditions, the equivalent change in \( T_{c,\Delta p} \) produced by \( \Delta p_{c} \) can be estimated since \( \Delta T_{\Delta p} / \Delta p \approx \) constant, as was observed for most refrigerants. Therefore, applying this to equation (212) relates change in saturation temperature to capacity (equation 213).
Using equation (213) within the general rating formulas would result in an implicit equation, so instead it was chosen to adopt a non-functional line-fit to be substituted for $Q_c$. Taking the approach of the conventional basic rating method, and a simplified version of equation (210), provides an estimation of $Q_c$ (equation 214).

$$Q_c = \Delta T_{c(m)} \cdot (b_1 + b_2 \cdot \Delta T_{dsh})$$

(214)

It is noted that the contribution of the subcooling region is neglected, since liquid flow is known to produce a minimal $\Delta p$ relative to vapour or two-phase flows. Lastly, substituting equation (214) into (213) provides a means of predicting the corresponding change in $T_{c,dp}$ as a function of $\Delta T_{dsh}$ and $\Delta T_{c(m)}$ (equation 215).

$$\Delta T_{c,dp} = b_{c,dp} \cdot \left( \Delta T_{c(m)} \cdot [b_1 + b_2 \cdot \Delta T_{dsh}] \right)^2$$

(215)

where $b_{c,dp} = c_1 \times c_2$.

In addition to capacity rating, the dew-point temperature corresponding to the outlet pressure of the condenser must also be provided to meet the requirements for system balancing. The equivalent pressure loss is expressed as a temperature difference ($\Delta T_{c,dp}$) (equation 216).

$$T_{c,dp(out)} = T_{c,dp(in)} - \Delta T_{c,dp}$$

(216)

The final rating equation is obtained by substituting the various adjusted parameters from equation (211) ($\Delta T_G \rightarrow \Delta T_G'$) and equation (208) for subcooling, and adding equation (210) for desuperheat, in equations (202) and (207). This provides general rating formulas for: (1) when $\dot{C}_{min} = \dot{C}_{RTF}$ (equation 217), and (2) when $\dot{C}_{min} = \dot{C}_r$ (equation 218).

$$Q_{c,\text{cond.1}} = b_{dsh,1} \cdot \Delta T_{dsh} \cdot \left( \Delta T_{c(m)} \right)^{n_{dsh,1}} + B'_{c,1} \cdot \left( \Delta T_G \right)^{n_{T_G}} \cdot \Delta T_{c(m)}^{1/n_{T_G}}$$

(217)

$$Q_{c,\text{cond.2}} = b_{dsh,2} \cdot \Delta T_{dsh} \cdot \left( \Delta T_{c(m)} \right)^{n_{dsh,2}} + B'_{c,2} \cdot \left( \Delta T_G \right)^{n_{T_G}} \cdot \Delta T_{c(m)}^{1/n_{T_G}}$$

(218)

where the desuperheat basic ratings $b_{dsh,1}$ and $b_{dsh,2}$ are found from rearranging equation (210) for each $\dot{C}_{min}$ case, and the basic ratings $B'_{c,1}$ and $B'_{c,2}$ are essentially constants, calculated from equation (219) and (220) respectively.
\[ B'_{\text{c,1}} = (1 + b_{\text{sc,1}}) \cdot a_c^{1/(n+1)} \cdot C_P \cdot \dot{m}_{\text{HTF}} \] (219)

\[ B'_{\text{c,2}} = (1 + b_{\text{sc,2}}) \cdot a_c^{-1/n} \cdot C_P \cdot \dot{m}_{\text{HTF}} \] (220)

For clarity the temperature difference, \( \Delta T_{\text{c(in)}} = T_{\text{c,dp(in)}} - T_{\text{HTF(in)}} \).

The use of degree of desuperheat as a variable also implies that more data is required. In Section 3.5.3, it was noted that a minimum of two or three \( Q_c - \Delta T_{\text{c(in)}} \) measurements would be needed, depending upon the variation of \( C_{\text{min}} \). However, since \( \Delta T_{\text{sh}} \) is a dependent variable, at least one additional measurement would be necessary for each \( \Delta T_{\text{c(in)}} \) in order to determine \( b_{\text{sh}} \). The constants associated with pressure drop calculations demand that the pressure is known for both inlet and outlet conditions. Lastly, constants associated with subcooling can be found from enthalpy balances.

**Evaporator**

As with the condenser, equations (202) and (207) are to be modified to account for the influences of evaporator inlet condition and \( \Delta T_{\text{c}} \) (equation 196) as well as pressure drop. Additionally, because equations (202) and (207) are based on the two-phase region only it is again pertinent to adapt the rating method further to integrate the sensible heat transfer from the superheat region.

Consistent with the subcooling region of the condenser, superheat capacity \( Q_{\text{sh}} \) was found to relate directly to the two-phase capacity (equation 221).

\[ Q_{\text{sh}} = b_{\text{sh}} \cdot Q_{\text{tp}} \] (221)

where \( b_{\text{sh}} \) represents the basic rating of the superheat region for a fixed \( \Delta T_{\text{sh}} \). Whilst the use of a TEV or similar expansion device implies a fixed superheat, equation (221) could be easily modified to account for variations, since \( b_{\text{sh}} / \Delta T_{\text{sh}} \approx \text{constant} \).

Use of equations (202) and (207) allow the influence of variable \( \Delta T_{\text{c,e}} \) to be integrated into the rating method. Therefore, \( \Delta T_{\text{c,e}} \) must be determined and two approaches are proposed based on the knowledge of \( T_{\text{TEV(in)}} \) and \( T_{\text{c,dp}} \), and the glide properties specific to the refrigerant in use. The most straight-forward approach is to employ a non-functional line-fit, and for all of the mixtures
evaluated good accuracy was obtained with a linear relationship when it contained a smaller proportion of MVC (equation 222).

\[ \Delta T_{G,e} = a_1 + a_2 \cdot T_{TEV'(in)} + T_{e,dp(in)} \cdot (a_3 + a_4 \cdot T_{TEV'(in)}) \quad (222) \]

where \( a_1 \) to \( a_4 \) are determined from a refrigerant's property data, and \( T_{e,dp(in)} \) represents the dew-point at a pressure corresponding to the evaporator inlet.

Conversely, \( \Delta T_G \) of zeotropes whose composition is dominated by MVC tend to be independent of \( T_{e,dp} \) due to the vertical \( p - T \) gradient of the bubble-line, so \( \Delta T_G \) normally exhibits a non-linear relationship with \( T_{TEV'(in)} \). If this is the case then a second-order polynomial equation (223) can be used instead of equation (222).

\[ \Delta T_{G,e} = a_1 + a_2 \cdot T_{TEV'(in)} + a_3 \cdot T_{TEV'(in)}^2 \quad (223) \]

where \( a_1 \) to \( a_3 \) are also determined from refrigerant property data.

Alternatively, the approach detailed in Section 2.4.3 can be employed, where \( \Delta T_{G,e} \) is estimated from \( x_{e,in} \). Determination of \( x_{e,in} \) may be done in the usual way from enthalpies (equation 114), or by using a non-functional linear relationship based on a temperature drop across the expansion device (equation 224).

\[ x_{e,in} = a_1 + a_2 \cdot (T_{TEV'(in)} - T_{e,dp(in)}) \quad (224) \]

where \( a_1 \) and \( a_2 \) are from refrigerant property data.

Equation (222), (224), and the calculation for \( \Delta T_{e,in} \) require knowledge of the saturated conditions at evaporator inlet, whilst the rating data should be based on outlet conditions (\( T_{e,dp(out)} \)). Therefore it is important to establish the equivalent saturated temperature change across the evaporator due to pressure loss (\( \Delta T_{e,dp} \)). This was handled in the same way as the condenser pressure drop. As with equation (211), the effective temperature glide (\( \Delta T_G' \)) can be determined from equation (225).

\[ \Delta T_G' = \Delta T_{G,e} - \Delta T_{e,dp} \quad (225) \]
Similarly, the derivation of $\Delta T_{e, \Delta p}$ followed the same sequence as for the condenser; equation (226) instead of (212), and equation (227) instead of (213).

$$\Delta p_e = c_1 \cdot Q_e^2$$  \hspace{2cm} (226)

$$\Delta T_{e, \Delta p} = c_1 \cdot c_2 \cdot Q_e^2$$  \hspace{2cm} (227)

As with equation (214), total $Q_e$ was obtained only as a function of a single temperature difference, rather than accounting for sensible heat regions separately (equation 228).

$$Q_e = (T_{HTF(in)} - T_{e, dp(out)}) \cdot b_1$$  \hspace{2cm} (228)

Lastly, substituting equation (228) into (227) provides a means of predicting the corresponding change in $T_{e, \Delta p}$ (equation 229), and thus the inlet dew-point (equation 230).

$$\Delta T_{e, \Delta p} = b_{e, \Delta p} \cdot \left( T_{HTF(in)} - T_{e, dp(out)} \right)^2$$  \hspace{2cm} (229)

where $b_{e, \Delta p} = c_1 \times c_2 \times b_1^2$.

$$T_{e, dp(in)} = T_{e, dp(out)} - \Delta T_{e, \Delta p}$$  \hspace{2cm} (230)

The final rating equation is obtained by substituting the various adjusted parameters from equation (225) ($\Delta T_o \rightarrow \Delta T_{o'}$) and equation (221) for the superheat region, into equations (202) and (207). This provides general rating formulas for: (1) when $\hat{C}_{min} = \hat{C}_{HTF}$ (equation 231), and (2) when $\hat{C}_{min} = \hat{C}_{e}'$ (equation 232).

$$Q_{e, evap, 1} = B'_{e, 1} \cdot (\Delta T_{o'}^{n_1(n+1)} \cdot \Delta T_{e(in)}^{n_1(n+1)})$$  \hspace{2cm} (231)

$$Q_{e, evap, 2} = B'_{e, 2} \cdot (\Delta T_{o'}^{n_2(n+2)} \cdot \Delta T_{e(in)}^{n_2(n+2)})$$  \hspace{2cm} (232)

where the basic ratings $B'_{e, 1}$ and $B'_{e, 2}$ are essentially constants, calculated from equation (233) and (234) respectively.

$$B'_{e, 1} = (1 + b_{sh, 1}) \cdot a_e^{1/(n_1 - 1)} \cdot C_{pHTF} \cdot \dot{m}_{HTF}$$  \hspace{2cm} (233)

$$B'_{e, 2} = (1 + b_{sh, 2}) \cdot a_e^{-1/n_2} \cdot C_{pHTF} \cdot \dot{m}_{HTF}$$  \hspace{2cm} (234)

For clarity, the temperature difference, $\Delta T_{e(in)} = T_{HTF(in)} - (T_{e, dp(out)} - \Delta T_{o'})$. 

206
As with the condenser, determination of the constants and indices for the new rating equations necessitate additional performance data-points. Therefore, a minimum of two or three $Q_e - \Delta T_{e(m)}$ measurements would be needed, depending upon the variation of $\dot{C}_{\text{min}}$. Unlike the condenser though, additional measurements to accommodate sensible heat contributions are unnecessary since $\Delta T_{\text{th}}$ is assumed constant. The constants associated with pressure drop calculations demand that both inlet and outlet pressures are known.
SECTION 4: SYSTEM BALANCING

4.1 Introduction
It is the aim of this section to identify the differences associated with balancing system components when moving from a pure refrigerant to one with a $\Delta T_o$ and ultimately devise a revised balancing technique to improve the accuracy of matching system capacity with the application requirements. Both pure and mixture refrigerants are in common use and therefore the balancing technique should be suitable for both.

In addition, other system operating parameters such as heat exchanger pressure loss and variation of compressor discharge superheat should also be incorporated into the new balancing scheme. Based on the principles of system balancing for refrigerants with temperature glide, a general methodology should be detailed to allow the utilisation engineer to apply it.

4.2 Balance technique theory
Normally when balancing a refrigerating system, the compressor and condenser are combined to produce a characteristic for the sub-system usually referred as the condensing unit. The condensing unit is then combined with the evaporator to form a complete system (given that the TEV is ignored, as noted previously). Combining these components is done on the basis of defined refrigerant reference state temperatures and heat transfer capacities. For a specific component, capacities at corresponding reference temperatures are obtained from component catalogue data.

For the methodology described in this section, the following are assumed to apply:

- **Basic system.** Only a simple system – using a single stage of compression and only four fundamental components (evaporator, condenser, compressor and expansion device) – is considered.
- **Expansion device.** The type of expansion device is restricted to TEV.
- **Heat and pressure loss.** It is assumed that there are no losses to the environment from the system, except for those accounted for within the component rating, i.e., no heat transfer or pressure losses associated with interconnecting pipework, etc.
- **Liquid subcooling.** The subcooling at the condenser outlet is fixed with the same value for each component. Component ratings must be based on the same subcooling for compressor,
condenser and evaporator. To simplify the current exercise, it is assumed that $\Delta T_{sc} = 0$ K always.

- **Reference state temperature.** The rating reference temperature for all components must be fixed.
- **HTF temperatures.** The condenser and evaporator HTF inlet temperatures and mass flow rates are fixed.

For conventional system balancing using a pure refrigerant, the component characterisation are expressed in functional form: evaporator capacity (equation 235), condenser capacity (equation 236), and compressor evaporating (equation 237) and condensing (equation 238) capacities.

$$Q_{e,\text{evap}} = f(T_e)$$  
(235)

$$Q_{c,\text{cond}} = f(T_c)$$  
(236)

$$Q_{e,\text{comp}} = f(T_e, T_c)$$  
(237)

$$Q_{c,\text{comp}} = f(T_e, T_c)$$  
(238)

The condenser and the compressor are combined to form the condensing unit sub-system using their respective $Q_c$ and $T_c$ variables. Its capacity is characterised in the form of equation (239), which shows that the change in evaporating temperature is dependent of the change of condensing unit condensing capacity ($Q_{c,c,u}$) with $T_e$. $T_e$ is implicitly dependent on the characteristic.

$$Q_{c,c,u}(T_e) = f(T_e)$$  
(239)

The graphical interpretation of this is shown in Figure 68.

![Graphical interpretation of equation (239)](image-url)
Since $Q_{e,comp}$ is a function of the same two variables as $Q_{c,comp}$, the equivalent evaporating capacity of the condensing unit ($Q_{e,c,u}$) can be expressed in the same terms (equation 240). This represents the variation in $Q_{e,c,u}$ with $T_e$, whilst $T_c$ is now implicit.

$$Q_{e,c,u}(T_e) = f(T_e)$$  \hspace{1cm} (240)

Finally, the condensing unit (equation 240) is matched to the evaporator (equation 235) to determine the equilibrium conditions of the complete system. The balance point for $T_e$ and $Q_e$ is found when the expression in equation (241) is satisfied.

$$\frac{Q_{e,c,u}(T_c)}{f(T_c)} - \frac{Q_e}{f(T_e)} = 0$$  \hspace{1cm} (241)

After obtaining the balance $T_e$ and $Q_e$, the corresponding equilibrium $T_c$ and $Q_c$ can also be determined. This is achieved in the same manner as conversion of the condensing unit characteristic, where compressor data is used to find the equivalent $T_c$ and then $Q_c$.

4.3 Balance technique for zeotropic refrigerants

In order to apply the balancing theory to components operating on zeotropic refrigerants, the influence of $\Delta T_G$ on individual component characterisation must be known. To start with, the desuperheat and pressure loss aspects are ignored at this stage (referred to as the interim balancing technique), so the heat exchanger inlet and outlet conditions are equivalent, i.e., $T_{dp(out)} = T_{dp(in)}$.

Functionally the compressor capacity is expressed as before in equations (188) and (189). For the condenser, $Q_{c,cond}$ is based on equation (195), and ignoring $\Delta T_{dbh}$ and $\Delta T_{sc}$ it reduces to a function of $T_{c,dp}$ only, equation (242).

$$Q_{c,cond} = f(T_{c,dp})$$  \hspace{1cm} (242)

The evaporator capacity has been shown to vary with inlet conditions, the evaporator characteristic in equation (196) is reduced to a function of $T_{e,dp}$ and $T_{EIV(out)}$, equation (243).
\[ Q_{e,\text{evap}} = f(T_{e,dp}, T_{\text{TEV'}(m)}) \]  \hspace{1cm} (243)

However, assuming \( \Delta T_{sc} = 0 \) K, then \( T_{\text{TEV'}(m)} \) in equation (243) can be substituted for the condensing dew-point temperature, since \( T_{c,dp} = T_{\text{TEV'}(m)} + \Delta T_{c} \) (equation 244).

\[ Q_{e,\text{evap}} = f(T_{e,dp}, T_{c,dp}) \]  \hspace{1cm} (244)

The avoidance of \( T_{\text{TEV'}(m)} \) (or \( T_{c}(m) \) or \( T_{c,dp} \)) simplifies the procedure.

Following the conventional procedure, the condenser and the compressor are initially combined to form the condensing unit. Condensing unit condensing capacity (\( Q_{c,c,u} \)) is expressed as equation (245), and as usual, the evaporating temperature (\( T_{e,dp} \)) is implicit within the condensing unit characteristic.

\[ Q_{c,c,u}(T_{e,dp}) = f(T_{c,dp}) \]  \hspace{1cm} (245)

Because \( Q_{e,\text{comp}} \) is a function of both \( T_{e,dp} \) and \( T_{c,dp} \), \( Q_{e,c,u} \) can be converted so that the evaporating capacity of the condensing unit (\( Q_{e,c,u} \)) can also be determined (equation 246). This process is consistent with the procedures associated with the conventional technique.

\[ Q_{e,c,u}(T_{e,dp}) = f(T_{e,dp}) \]  \hspace{1cm} (246)

Normally, at this stage the evaporator would be balanced with the condensing unit to form the complete system. However, from equation (244) it is known that \( Q_{e,\text{evap}} \) is dependent upon an additional variable, being \( T_{c,dp} \). Since \( T_{c,dp} \) is implicit in equation (246), it follows that \( Q_{e,\text{evap}} \) cannot be matched directly to \( Q_{e,c,u} \). In order to match the evaporator to the condensing unit, this additional variable must somehow be integrated into the evaporator characteristic. The solution was achieved through the development of an additional sub-system. Since \( Q_{e,\text{comp}} \) is a function of the same variables as \( Q_{e,\text{evap}} \), the two components can be combined to produce a new sub-system, termed the "evaporating unit." 15 The evaporating unit evaporating capacity (\( Q_{e,e,u} \)) is expressed

---

15 Note that the compressor characteristic has been used twice, once to produce a condensing unit and now for an evaporating unit. It should be understood that the compressor characteristic is simply employed to fix a relationship between capacities of the various components, and should not be interpreted in a physical sense.
in equation (247), and an example of the graphical form of equation (247) is provided in Figure 69.

\[ Q_{e,u}(T_{c,dp}) = f\left(T_{e,dp}\right) \]  

(247)

As with the condensing unit, the compressor data can be used to convert \( Q_{e,u} \) into the condensing capacity of the evaporating unit \( (Q_{e,u}) \).

\[ Q_{e,u} = Q_{c,u} \]

Comparison of equation (246) and equation (247) implies that the condensing unit and the evaporating unit can be matched to produce a system, whose equilibrium conditions are determined when the expression in equation (248) is satisfied.

\[ \frac{Q_{e,u}(T_{c,dp})}{f\left(T_{e,dp}\right)} - \frac{Q_{c,u}(T_{c,dp})}{f\left(T_{e,dp}\right)} = 0 \]  

(248)

As with the conventional balancing technique, once \( Q_{e} \) and \( T_{e,dp} \) are found, they can be used to determine the corresponding \( Q_{c} \) and \( T_{c,dp} \) at equilibrium conditions through manipulation of compressor performance data.

Table 27 lists the key stages for the conventional balancing technique, and the newly developed interim technique for \( \Delta T_{G} \). Capacities for the respective stages are identified in terms of the dependent temperature variables, and the Table summarises the main differences between the two techniques. Note that the ‘and’ term means that both variables are needed to define the capacity, whereas the ‘or’ term means that the variables are interchangeable so either can be used.
Table 27: Comparison of capacity and temperature variables for balancing

<table>
<thead>
<tr>
<th>Component/system</th>
<th>Temp. variables</th>
<th>Capacity variables</th>
<th>Temp. variables</th>
<th>Capacity variables</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>$T_e$ and $T_c$</td>
<td>$Q_{e,comp}$, $Q_{c,comp}$</td>
<td>$T_{e,dp}$ and $T_{c,dp}$</td>
<td>$Q_{e,comp}$, $Q_{c,comp}$</td>
</tr>
<tr>
<td>Condenser</td>
<td>$T_c$</td>
<td>$Q_{c,cond}$</td>
<td>$T_{c,dp}$</td>
<td>$Q_{c,cond}$</td>
</tr>
<tr>
<td>Evaporator</td>
<td>$T_e$</td>
<td>$Q_{e,evap}$</td>
<td>$T_{e,dp}$ and $T_{c,dp}$</td>
<td>$Q_{e,evap}$</td>
</tr>
<tr>
<td>Condensing unit</td>
<td>$T_e$ or $T_c$</td>
<td>$Q_{e,ext}$, $Q_{c,cond}$</td>
<td>$T_{e,dp}$ or $T_{c,dp}$</td>
<td>$Q_{e,ext}$, $Q_{c,cond}$</td>
</tr>
<tr>
<td>Evaporating unit</td>
<td>$-$</td>
<td>$-$</td>
<td>$T_{e,dp}$ or $T_{c,dp}$</td>
<td>$Q_{e,ext}$, $Q_{c,ext}$</td>
</tr>
<tr>
<td>System</td>
<td>$T_e$ or $T_c$</td>
<td>$Q_e$, $Q_c$</td>
<td>$T_{e,dp}$ or $T_{c,dp}$</td>
<td>$Q_e$, $Q_c$</td>
</tr>
</tbody>
</table>

4.4 New balance technique for zeotrope with desuperheat and pressure loss

As discussed in Section 3.4.3, the heat exchanger pressure loss and compressor discharge superheat/condenser desuperheat should be accounted for. It is important to first clarify how their presence impacts on the logic of the balancing concept.

![Figure 70: Refrigeration cycle with pressure loss and superheat](image_url)
Examination of the effects of $\Delta p$ in heat exchangers showed that the corresponding change in saturation pressure affects $\Delta T_G$. However, $\Delta p$ across the condenser causes a difference between inlet and outlet $T_{c,dp}$. and consequently affects the interpretation of $Q_{e,\text{evap}}$. To ensure consistency, $T_{c,dp}$ determination of $Q_{e,\text{evap}}$ must be linked to the correct reference state as used by the condenser and compressor; this implies that the condenser $\Delta p$ characteristic must become a parameter in the balancing technique.

The second consideration is condenser $\Delta T_{\text{dsh}}$, which was previously shown to be dictated by the compressor performance. Observations from condenser simulations showed that $\Delta T_{\text{dsh}}$ had a significant effect on $Q_{c,\text{cond}}$ (and $\Delta p$). It was shown that for a specific $T_{c,dp(in)}$, the $\Delta T_{\text{dsh}}$ influences condenser $\Delta p$ with the secondary effect of reducing saturation temperature leading to a change in $\Delta T_G$ and $T_{c,dp(out)}$. In this respect the amount of compressor $\Delta T_{\text{dsh}}$ must be implicated in balancing when using a zeotrope.\(^{16}\)

A refrigeration cycle has been plotted on a pressure-enthalpy chart in Figure 70, showing a basic processes with evaporator and condenser $\Delta p$ and compressor $\Delta T_{\text{dsh}}$. Evaporator and condenser $T_{dp}$ that correspond to the inlet and outlet pressures are also shown. The pressure that the refrigerant enters the condenser corresponds to a specific dew-point temperature ($T_{c,dp(in)}$). The refrigerant cools, and reduces in pressure until it reaches a saturation point. During the condensing process, the refrigerant temperature reduces as a result of $\Delta p$ but also due to further $\Delta p$. At the end of the condensing process the bubble-point is reached and (as subcooling is neglected), this point is also $T_{TEV}(in)$\(^{\footnote{16}}\). As discussed earlier $T_{TEV}(in)$ in the evaporator characteristic was substituted for the corresponding dew-point (equation 244); to ensure consistency this reference point should ultimately be expressed in terms of $T_{c,dp(in)}$. To do this, knowledge of the condenser $\Delta p$ is needed, which has to be identified from the condenser characteristics. Since the evaporator rating is independent of the condenser, the evaporator capacity is initially rated according to $T_{c,dp(out)}$. Following expansion, the refrigerant enters the evaporator at $T_{e(in)}$, as determined by $x_{e(in)}$ and glide properties. However, this condition is of no

\(^{16}\) It is noted that this parameter does not only affect systems with temperature glide, but also those with pure refrigerants as the overall condenser capacity is changed regardless of the existence of temperature glide.
A direct consequence to the balancing procedure, since it is already implicit within the evaporator characteristic. Evaporation then follows where further $\Delta p$ occurs, coupled with $\Delta T_G$ and $\Delta T_{sh}$, and the refrigerant eventually exits the evaporator at a superheated condition. The exit pressure corresponds to a dew-point temperature which is designated $T_{e,dp(out)}$.

![Figure 71: New condenser characteristic](image)

In order to account for pressure losses and compressor discharge superheat in the condenser, the definitions of the condenser characteristic must first be modified. Since there are now two values of $T_{c,dp}$ corresponding to its inlet and outlet pressure, of which the former was selected for the reference condition. Thus taking equation (195) and ignoring $\Delta T_{sc}$ gives equation (249).

$$Q_{c,cond} = f(T_{c,dp(out)}, \Delta T_{dbh})$$  \hspace{1cm} (249)

The existence of the additional desuperheat variable, the condenser characteristic implies that the condenser now exhibits several characteristic lines corresponding to the different values of $\Delta T_{dbh}$ (see Figure 71).

![Figure 72: Discharge superheat characteristic](image)
Condenser rating can only be based on fixed increments of $\Delta T_{\text{dsh}}$ as the degree of desuperheat produced by a compressor at the rating stage is unknown. Therefore the compressor $\Delta T_{\text{dsh}}$ must also be characterised, as previously given in equation (190). Since compressor $T_{c,dp} = T_{c,dp(\text{in})}$ of the condenser and compressor $T_{e,dp} = T_{e,dp(\text{out})}$ of the evaporator, then equation (190) can be rewritten as equation (250). This characteristic is shown graphically in Figure 72.

$$\Delta T_{\text{dsh}} = f\left(T_{e,dp(\text{out})}, T_{c,dp(\text{in})}\right) \quad (250)$$

Similarly, the compressor capacity should also be defined with these new reference temperatures so altering equations (188) and (189) gives equations (251) and (252) respectively.

$$Q_{e,\text{comp}} = f\left(T_{e,dp(\text{out})}, T_{c,dp(\text{in})}\right) \quad (251)$$

$$Q_{c,\text{comp}} = f\left(T_{e,dp(\text{out})}, T_{c,dp(\text{in})}\right) \quad (252)$$

![Figure 73: Modified condenser characteristic](image)

The next stage is to combine the condenser and compressor to form a condensing unit. However, now that $Q_{c,\text{cond}}$ is dependent on $\Delta T_{\text{dsh}}$ this is not initially possible as seen with the dissimilar variables when comparing equations (249) and (252). The solution to this problem is to utilise the compressor $\Delta T_{\text{dsh}}$ data to convert condenser data rated for a specific $\Delta T_{\text{dsh}}$ to capacities that correspond instead to an equivalent evaporating temperature. In other words, for a fixed set of compressor operating temperatures, $\Delta T_{\text{dsh}}$ can be determined, and the corresponding $Q_{c,\text{cond}}$ obtained for each of the $T_{c,dp(\text{in})}$ and $T_{e,dp(\text{out})}$ increments. From this, a new condenser characteristic can be produced that is a direct function of both the reference temperatures of the compressor. Thus equation (249) becomes equation (253) when $T_{e,dp(\text{out})}$ is substituted for $\Delta T_{\text{dsh}}$;
the graphical interpretation of modified condenser characteristic is provided in Figure 73, and can be compared with Figure 71.

\[ Q_{c, \text{cond}} = f\left(T_{c, \text{dp}(\text{in})}, T_{e, \text{dp}(\text{out})}\right) \]  

(253)

Following this conversion, both the condenser characteristic (equation 253) and the compressor condensing capacity characteristic (equation 252) use identical variables \( Q_{c, \text{cond}}, T_{c, \text{dp}(\text{in})} \) and \( T_{e, \text{dp}(\text{out})} \). Now the modified condenser characteristic can be matched to the compressor, where the balance points correspond to the same \( T_{e, \text{dp}(\text{out})} \) of each component (equation 254).

\[ Q_{c, \text{cond}}(T_{c, \text{dp}(\text{in})}) = f\left(T_{e, \text{dp}(\text{out})}\right) \]  

(254)

The compressor and condenser characteristics and corresponding condensing unit characteristic are shown in Figure 74. Equation (254) is an expression for this condensing unit characteristic where the condensing temperature is implicit, and by default, \( \Delta T_{\text{c}, \text{sh}} \) must also be implicit.
Now with reference to the evaporator, by adopting equation (196) it is known that $Q_{e,\text{evap}}$ is a function of both $T_{T_{E3}^{\text{in}}}$ and $T_{e,\text{dp(out)}}$, although as with the interim technique, $T_{T_{E3}^{\text{in}}}$ becomes $T_{c,\text{dp}}$. However, because the condenser $\Delta p$ is unknown, the existing evaporator characteristic is not suitable because its current condensing reference temperature corresponds to the condenser outlet pressure ($T_{c,\text{dp(out)}}$). For the time-being, the evaporator characteristic is redefined in equation (255) and shown graphically in Figure 75.

$$Q_{e,\text{evap}} = f(T_{e,\text{dp(out)}}, T_{c,\text{dp(out)}}) \quad (255)$$

Following the procedure for the interim technique, the next stage is to link the evaporator and compressor to produce an evaporating unit. Comparison of equation (255) and the compressor evaporating capacity characteristic (equation 251) reveals a disagreement with respect to the reference condensing temperature. The compressor rating uses $T_{c,\text{dp(in)}}$, whereas the evaporator rating assumes $T_{c,\text{dp(out)}}$. In order to account for this, two options are available: either the evaporator or the condensing unit must be re-rated with respect to the condenser pressure drop. These can be achieved by substituting the evaporator rating condition of $T_{c,\text{dp(out)}}$ with the equivalent value of $T_{c,\text{dp(in)}}$, or substituting the condensing unit rating condition $T_{c,\text{dp(in)}}$ with the equivalent value of $T_{c,\text{dp(out)}}$, respectively for the condenser.

![Condenser pressure loss characteristics](image)

*Figure 76: Condenser pressure loss characteristics*

Ultimately, the favoured option is the former, since this will maintain consistency in the component reference points, i.e., the dew-point corresponding to the pressure at the compressor discharge. Initially, to determine the condenser pressure loss, data must be available. As described in Section 5.3, $\Delta p$ is implicit in the revised rating method and is therefore a function of
\[ T_{c,\text{dp}(m)} \] and \( \Delta T_{\text{dbh}} \). Accordingly, the equivalent \( \Delta p \) can be expressed in terms of the difference between in inlet and outlet saturation temperatures, where it is influenced by \( \Delta T_{\text{dbh}} \) (equation 256); a graphical representation is given in Figure 76.

\[
T_{c,\text{dp(out)}} = f(T_{c,\text{dp(in)}}, \Delta T_{\text{dbh}})
\]

(256)

In its present format this \( \Delta p \) data cannot be applied to the conversion of the evaporator data, since the evaporator characteristic in independent of \( \Delta T_{\text{dbh}} \). In equation (256), the absent variable is \( T_{c,\text{dp(out)}} \), indicating that the condenser \( \Delta p \) characteristic needs to be converted from a function of \( \Delta T_{\text{dbh}} \) to \( T_{c,\text{dp(out)}} \). Following the earlier discussion on the conversion of condenser capacity, the same logic can be applied here. It is known that for each set of operating temperatures, the compressor will produce a corresponding \( \Delta T_{\text{dbh}} \). Therefore, the equivalent evaporating temperature is determined from the compressor \( \Delta T_{\text{dbh}} \) data at the specified \( T_{c,\text{dp}(in)} \). Figure 77 illustrates the revised condenser pressure loss characteristic resulting from this conversion.

\[
Q_{c,\text{evap}}
\]

(256)

Figure 77: Converted condenser pressure loss characteristics

Figure 78: Modified evaporator characteristics

219
With this converted condenser ΔP characteristic, the existing evaporator characteristic can be modified so that \( Q_{e,\text{evap}} \) is expressed as a function of \( T_{c,\text{dp}(in)} \) rather than \( T_{c,\text{dp}(out)} \). Equation (255) becomes equation (257) and the graphical representation is as Figure 78.

\[
Q_{e,\text{evap}} = f(T_{c,\text{dp}(in)}, T_{c,\text{dp}(out)})
\]

This characteristic is now not only specific to the selected evaporator, but also specific to the selected condenser and compressor. Now that \( Q_{e,\text{evap}} \) is a function of the same reference temperatures as \( Q_{e,\text{comp}} \) (equation 251), it is simple to link the two to produce an evaporating unit (equation 258).

\[
Q_{e,\text{evap}}(T_{c,\text{dp}(in)}) = f(T_{c,\text{dp}(out)})
\]

This balancing exercise is illustrated in Figure 79, where it is seen that the condensing temperature becomes implicit in the evaporating unit characteristic, and indirectly, \( \Delta T_{\text{bb}} \).

The final stage is to link the condensing unit and evaporating unit to obtain the overall system balance points, and this can be carried out using condensing or evaporating capacity as a basis.
Using the former, $Q_{c,c/u}$ is a function of $T_{c,dp(m)}$ (equation 254) can be converted to evaporating capacity as a function of $T_{e,dp(out)}$, as explained previously for the conventional and interim techniques, to give equation (259).

$$Q_{e,c,u}(T_{e,dp(out)}) = f(T_{e,dp(out)})$$  \hspace{1cm} (259)

Thus, the evaporating unit and condensing unit are balanced to provide the system equilibrium evaporating temperatures and evaporating capacity (equation 260); a graphical interpretation of this is illustrated in Figure 80.

$$\frac{Q_{e,c,u}(T_{e,dp(m)})}{f(T_{e,dp(out)})} - \frac{Q_{e,c,u}(T_{e,dp(m)})}{f(T_{e,dp(out)})} = 0$$  \hspace{1cm} (260)

Similarly, with these balance points known, the corresponding balance points for the condensing temperature and condenser capacity can be determined (equation 261); this is also shown graphically in Figure 81.

$$\frac{Q_{c,c,u}(T_{c,dp(out)})}{f(T_{e,dp(m)})} - \frac{Q_{c,c,u}(T_{c,dp(out)})}{f(T_{e,dp(m)})} = 0$$  \hspace{1cm} (261)

Subsequently, values for $T_{c,dp(out)}$ and $\Delta p_{dsh}$ can be obtained.

![Figure 81: Condenser balance point for condensing and evaporating unit](image)

The key stages in the new balancing technique (revised for $\Delta p$ and $\Delta T_{dsh}$) are listed in Table 28, with the corresponding temperature and capacity variables. The interim technique for $\Delta T_G$ only is also included in Table 28 for comparison. It is seen that the revised methodology is more complex, since in addition to the evaporating unit, accounting for $\Delta p$ and $\Delta T_{dsh}$ demands
additional stages for data conversions. Otherwise, the capacity and temperature variables are essentially the same as for the interim technique.

### 4.5 Clarification of new procedures

A summary of this new procedure with all relevant stages of data manipulation is detailed in Table 33 (Appendix 3) where the relevant equations are identified. The technique detailed here presents a standardised procedure for balancing system components. It can be used for both pure and zeotropic refrigerants. In addition, the final technique can be applied regardless of whether rating data is available for all of these additional parameters since the relevant steps necessary to account for a specific parameter can be neglected if the relevant data is unavailable.

#### Table 28: Comparison of capacity and reference temperatures for balancing procedure

<table>
<thead>
<tr>
<th>Component</th>
<th>Segment</th>
<th>for $\Delta T_G &gt; 0$ K. $\Delta p = 0$ Pa</th>
<th>for $\Delta T_G &gt; 0$ K. $\Delta p &gt; 0$ Pa</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>(interim technique)</td>
<td>(full new technique)</td>
</tr>
<tr>
<td></td>
<td>Temp. variables</td>
<td>Capacity variables</td>
<td>Temp. variables</td>
</tr>
<tr>
<td>Compressor</td>
<td>$T_e,dp$ and $T_c,dp$</td>
<td>$Q_{e,comp}$</td>
<td>$T_{e,dp}(out)$ and $T_{c,dp}(in)$ and $\Delta T_{dbh}$</td>
</tr>
<tr>
<td>Condenser</td>
<td>$T_c,dp$</td>
<td>$Q_c$</td>
<td>$T_c,dp$ and $\Delta T_{dbh}$</td>
</tr>
<tr>
<td>Evaporator</td>
<td>$T_e,dp$ and $T_c,dp$</td>
<td>$Q_e$</td>
<td>$T_{c,dp}(out)$ and $T_{e,dp}(out)$</td>
</tr>
<tr>
<td>Condensing unit</td>
<td>$T_e,dp$ or $T_c,dp$</td>
<td>$Q_{c,e/u}$</td>
<td>$Q_{c,e/u}$</td>
</tr>
<tr>
<td>Evaporating unit</td>
<td>$T_e,dp$ or $T_c,dp$</td>
<td>$Q_{c,e/u}$</td>
<td>$Q_{c,e/u}$</td>
</tr>
<tr>
<td>System</td>
<td>$T_e,dp$ or $T_c,dp$</td>
<td>$Q_e, Q_c$</td>
<td>$Q_e, Q_c$</td>
</tr>
</tbody>
</table>

Finally, it is noted that the choice of reference temperatures (i.e., $T_{c,dp}(in)$ and $T_{e,dp}(out)$) is beneficial from a practical sense. When making field measurements, the easiest readings to take tend to be suction and discharge pressures at the compressor. Neglecting $\Delta p$ in suction and discharge lines, these measurements enable dew-point temperatures to be estimated, thereby providing a direct interpretation of the real balance-points.
SECTION 5: ILLUSTRATIVE EXAMPLE OF NEW TECHNIQUES

5.1 Introduction
The intention of this section is to work through the methodology by means of a practical example. There are three purposes for this example. The first is to assist the user in following the technique; a practical example helps to convey its application to the user. Secondly, it is used to validate the methodology; simulated component catalogue data was used, and the results of the exercises are then checked against component simulation data for the conditions at the respective balance points. Capacities from the balancing exercise should equal those obtained from the direct simulation at balance temperatures if the proposed technique is valid. Lastly, the accuracy of the new method can be compared against existing techniques to establish whether the new technique provides a benefit over the existing techniques; a comparison can be made against the results obtained using the conventional approach where references temperatures are interpreted as suggested in the literature. For example, Hundy (2000) and Murphy et al (1998) describe using both dew-point conditions and mid-point conditions for compressor rating, and Atkins (1999) uses mid-point for evaporator and condenser rating. The definition of mid-point temperature for the condenser \( T_{c,mp} \) and evaporator \( T_{e,mp} \) are according to equations (262) and (263) respectively.

\[
T_{c,mp} = \frac{(T_{c,bp} + T_{c,dp})}{2} \tag{262}
\]

\[
T_{e,mp} = \frac{\left(1 - x_{e(on)}\right) \cdot (T_{e,bp} + T_{e,dp})}{2} \tag{263}
\]

By substituting \( T_{dp} \) or \( T_{mp} \) for pure refrigerant evaporating and condensing temperatures in the component performance data, the exercise can be worked through using the conventional balancing technique. Comparison of the results against those from the new technique can be used to judge the improvement in accuracy.

5.2 Generation of component data
Eight examples were used to cover a range of system characteristics that the new balancing technique was intended to account for, specifically small and large \( \Delta T_G \), heat exchanger \( \Delta p \), and a range of \( \Delta T_{(in)} \). Table 29 shows two different heat exchanger combinations were used with small and large \( \Delta T_G \) (systems 1 and 2, and 3 and 4), each with (a) small size and (b) large size compressor. Condenser and evaporator \( \Delta p \) were varied by tube length and number of refrigerant
circuits. For each component design, corresponding models were used to generate fictitious performance data as for a catalogue.

Table 29: Specific input data for calculation of fictitious rating data

<table>
<thead>
<tr>
<th>Component</th>
<th>Variable</th>
<th>System 1 (high $\Delta T_g$, high $\Delta \rho$)</th>
<th>System 2 (high $\Delta T_g$, low $\Delta \rho$)</th>
<th>System 3 (low $\Delta T_g$, high $\Delta \rho$)</th>
<th>System 4 (low $\Delta T_g$, low $\Delta \rho$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>Temp. glide at 0°C</td>
<td>10 K</td>
<td>10 K</td>
<td>5 K</td>
<td>5 K</td>
</tr>
<tr>
<td>Compressor</td>
<td>(a) Swept volume</td>
<td>15 m³ h⁻¹</td>
<td>15 m³ h⁻¹</td>
<td>15 m³ h⁻¹</td>
<td>15 m³ h⁻¹</td>
</tr>
<tr>
<td></td>
<td>(b) Swept volume</td>
<td>35 m³ h⁻¹</td>
<td>35 m³ h⁻¹</td>
<td>35 m³ h⁻¹</td>
<td>35 m³ h⁻¹</td>
</tr>
<tr>
<td>Condenser</td>
<td>Desuperheat</td>
<td>30 K – 90 K</td>
<td>30 K – 90 K</td>
<td>30 K – 90 K</td>
<td>30 K – 90 K</td>
</tr>
<tr>
<td></td>
<td>No. circuits</td>
<td>3</td>
<td>4</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Circuit length</td>
<td>8.0 m</td>
<td>3.5 m</td>
<td>8.0 m</td>
<td>3.5 m</td>
</tr>
<tr>
<td>Evaporator</td>
<td>No. circuits</td>
<td>6</td>
<td>12</td>
<td>6</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>Circuit length</td>
<td>3.0 m</td>
<td>2.0 m</td>
<td>3.0 m</td>
<td>2.0 m</td>
</tr>
</tbody>
</table>

The refrigerant thermodynamic and transport properties were calculated for a mixture of 50% R290 and 50% R600a by mass, although an artificially imposed glide was used as explained in Section 3. For the two refrigerant characteristics, the glide at $T_{ip} = 0°C$ was $5 K$, and $10 K$ and both had a glide shape with index $n_G = 3$.

Compressor rating data was based on 5 K evaporator superheat and no subcooling. Both compressors had a clearance ratio of 0.15. Similarly, the condenser rating data was also based on 0 K subcooling, and HTF inlet temperature was +20°C, for a water flow rate of 0.4 kg s⁻¹. The inside tube, containing refrigerant had a diameter of 0.010 m, and the surrounding tube containing HTF was 0.025 m diameter and total tube length and the number of circuits differed.

The evaporator used a similar construction, with an inside refrigerant tube diameter of 0.015 m, and the HTF again passed through a 0.025 m tube. The HTF mass flow rate was 0.35 kg s⁻¹, and entered at +15°C, and rating data was again based on 5 K superheat, consistent with the compressor. Neither condenser nor evaporator used extended heat transfer surfaces. Lastly, it was assumed that all components were directly connected, to negate the possibility of temperature and pressure changes within interconnecting piping and ancillary components.
5.3 Illustration of worked example

The worked example was carried out using the data for System 1(a), although the results for all eight systems are provided.

Stage 1: Compressor characteristic data

Compressor condensing capacities, and discharge superheat catalogue data was obtained (Figure 82 and Figure 83). It is noted that the reference temperatures are not designated for \((in)\) or \((out)\) conditions, which is because they do not necessarily refer to the heat exchangers. Logically though, since the compressor suction is connected to the evaporator outlet, its \(T_{e,dp} \equiv T_{e,dp(out)}\) of the evaporator, and similarly because the compressor discharge in connected to the condenser inlet, its \(T_{c,dp} \equiv T_{c,dp(in)}\) of the condenser.

Stage 2: Condenser characteristic data

For the selected condenser, \(Q_c\) capacity and equivalent \(\Delta p\) catalogue data is obtained (Figure 84 and Figure 85). The \(\Delta p\) data is presented in terms of inlet and outlet dew-point temperatures \((T_{c,dp(in)}\) and \(T_{c,dp(out)})\).
Stage 3: Evaporator data

For the selected evaporator, $Q_{e,\text{evap}}$ catalogue data is obtained as a function of $T_{e,\text{dp(out)}}$ for a series of $T_{e,\text{dp(out)}}$ (Figure 89).

Stage 4: Condenser characteristic data matched to compressor superheat

The condenser capacity is to be matched to the compressor, but reference to $Q_{c,\text{comp}}$ (Figure 82) and $Q_{c,\text{cond}}$ (Figure 84) are a function of different variables. Therefore, the condenser characteristic lines in must be modified so that the capacity is also given as a function of $T_{e,\text{dp(out)}}$ (rather than $\Delta T_{\text{dbh}}$). This is achieved through use of the compressor $\Delta T_{\text{dbh}}$ data (Figure 83), so for a given condenser data-point in Figure 84, the corresponding $T_{e,\text{dp(out)}}$ is taken from Figure 83.

Once all data-points have been converted, a new condenser characteristic is produced where $Q_{c,\text{cond}}$ is presented against $T_{e,\text{dp(in)}}$ for a series of $T_{e,\text{dp(out)}}$ (Figure 86), which requires interpolation. It is noted that $\Delta T_{\text{dbh}}$ is implicit in the new condenser characteristic. This same conversion procedure is also carried out for the condenser pressure drop characteristic, since the data is needed for a series of $T_{e,\text{dp(out)}}$ instead of $\Delta T_{\text{dbh}}$ (Figure 87).
Stage 5: Condensing unit characteristic

Now that both $Q_{comp}$ and $Q_{c,comp}$ data are a function of the same variables ($T_{e,dp(ouw)}$ and $T_{c,dp(in)}$), they can be matched to provide a condensing unit characteristic. The condenser characteristic is superimposed on the compressor data, although it is converted to be a function of $T_{e,dp(ouw)}$ for a series of $T_{c,dp(in)}$. Where the component characteristics of the same $T_{c,dp(in)}$ intersects, indicates the condensing capacity of the condensing unit (Figure 88).

Figure 86: Condenser capacity against condensing dew-point temperature for a series of evaporating temperatures

Figure 7: Condenser outlet dew-point temperature against condensing inlet dew-point temperature for a series of evaporating temperatures

Stage 5: Condensing unit characteristic

Now that both $Q_{comp}$ and $Q_{c,comp}$ data are a function of the same variables ($T_{e,dp(ouw)}$ and $T_{c,dp(in)}$), they can be matched to provide a condensing unit characteristic. The condenser characteristic is superimposed on the compressor data, although it is converted to be a function of $T_{e,dp(ouw)}$ for a series of $T_{c,dp(in)}$. Where the component characteristics of the same $T_{c,dp(in)}$ intersects, indicates the condensing capacity of the condensing unit (Figure 88).

Figure 88: Compressor characteristic superimposed on new condenser characteristic to obtain condensing unit characteristic
Stage 6: Evaporator characteristic matched to condenser inlet

The next two components to be balanced are the evaporator and compressor. However, before this can be achieved, the evaporator inlet condition has to be determined. Given that \( Q_{e,\text{evap}} \) is presently based on \( T_{c,\text{dp}(\text{out})} \) (i.e., into the TEV, \( \sim T_{\text{TEV}(\text{in})} \), Figure 89), whilst \( Q_{e,\text{comp}} \) is based on \( T_{c,\text{dp}} \equiv T_{c,\text{dp}(\text{in})} \) (Figure 82), the evaporator characteristic needs to be adjusted so that reference states are consistent.

The data within the modified condenser pressure drop characteristic (Figure 87) is therefore used to convert the evaporator characteristic from \( T_{c,\text{dp}(\text{out})} \) to \( T_{e,\text{dp}(\text{in})} \). This is achieved as follows.

From the evaporator data (Figure 89), a \( T_{c,\text{dp}(\text{out})} \) and \( T_{e,\text{dp}(\text{out})} \) coordinate is selected and applied to the converted condenser pressure drop data (Figure 87) for which the corresponding \( T_{c,\text{dp}(\text{in})} \) is determined. For the chosen \( T_{c,\text{dp}(\text{out})} \) and \( T_{e,\text{dp}(\text{out})} \), the \( Q_{e,\text{evap}} \) value from Figure 89 is plotted against the original \( T_{e,\text{dp}(\text{out})} \) and the identified \( T_{c,\text{dp}(\text{in})} \). This procedure is followed for the entire evaporator data set, and Figure 90 shows the resulting modified evaporator characteristic.

![Figure 89: Evaporator capacity against evaporating dew-point temperature for a series of condensing outlet dew-point temperatures](image1)

![Figure 90: Evaporator characteristic corrected for condenser inlet](image2)
Stage 7: Evaporating unit characteristic

As with the condenser, the evaporator is matched with the compressor to form the evaporating unit. Both compressor evaporating capacity characteristic (Figure 82) and the modified evaporator characteristic (Figure 90) are now a function of the same reference states \( T_{c,dp(in)} \) and \( T_{e,dp(out)} \), and therefore they can be linked. Figure 91 shows \( Q_{e,comp} \) superimposed on \( Q_{e,evap} \) and the evaporating unit characteristic corresponds to the intersection of their respective \( T_{c,dp(in)} \).

![Figure 91: Compressor characteristic superimposed on new evaporator characteristic to obtain evaporating unit characteristic](image)

Stage 8: Convert condensing and evaporating unit capacity characteristic

Once the characteristic for both condensing and evaporating units are obtained, the balance points for the total system can be determined. This is achieved by identifying the intercept of the units’ capacities. However, it is noted that the condensing unit characteristic is currently in the form of \( Q_{c,c/u} \) over a range of \( T_{c,dp(out)} \) (with implicit \( T_{c,dp(in)} \)) (Figure 88), whilst evaporating unit characteristic is \( Q_{e,e/u} \) a function of \( T_{e,dp(out)} \) (Figure 91). To balance the system, the capacity of one unit must be converted to the capacity of the other. Using the compressor data, the condensing unit characteristic is converted to the equivalent evaporating capacity at the corresponding \( T_{c,dp(out)} \) and \( T_{c,dp(in)} \). Similarly, the opposite conversion can also be made, where \( Q_{e,e/u} \) is converted to \( Q_{c,c/u} \), using the compressor characteristic data. It is noted that this is the same process is required for the conventional balancing process.
Stage 9: Balance condensing unit with evaporating unit

The final stage is relatively straightforward. The evaporating unit and condensing unit are matched twice: once for $Q_e$ as a function of $T_{e,dp\{in\}}$, and for $Q_e$ as a function of $T_{e,dp\{out\}}$. Thus, the intercepts of these provide the system equilibrium $Q_c$ and $Q_e$ and corresponding $T_{c,dp\{in\}}$ and $T_{e,dp\{out\}}$ (Figure 92 and Figure 93). The numerical values of these balance points are provided in Appendix 4, Table 34.

![Figure 92: System balance condensing dew-point temperature](image1)

![Figure 93: System balance evaporating dew-point temperature](image2)

5.4 Comparison of results

The accuracy of the new technique was assessed by comparing the results of the eight system balance exercises, against the output of the component models based on $T_{c,dp\{in\}}$ and $T_{e,dp\{out\}}$. In addition, the conventional procedure for system balancing was also used to find equilibrium points in the normal way, where the dew-point and mid-point temperatures were used as reference states (Section 5.1). The results are listed in Appendix 4 (Table 34). An overview of the "error" associated with each approach is presented in Figure 94, for the systems using (a) small size compressor and (b) large size compressor. The error is calculated as the ratio of capacity returned by the respective technique to the modelled capacity (which is taken as the correct value).

In Figure 94, it can be seen that the new balancing technique generally concurs with the modelled capacities to within ±1% for the condenser, and within ±2% for the evaporator. Whilst the new
technique is deemed to be conceptually correct, minor disagreement with the modelled values is expected, due to successive linear interpolation between performance data-points. With reference to Figure 88, Figure 91, Figure 92, and Figure 93, it is observed that the gradient of the evaporating capacity characteristic has more curvature than the condensing characteristic, which is consistent with the greater error associated with the evaporator capacity due to linear interpolation being less accurate. Nevertheless, given that new technique produces capacities at system equilibrium conditions consistent with those of the model over the broad range of system configurations suggests that it achieves its purpose.

Figure 94 shows that unlike results from the new technique, capacities estimated with the conventional method using $T_{dp}$ or $T_{mp}$ are always over-predicted compared to the models. Also, the evaporating temperature is always higher than that predicted by the new technique (see Table 34, Appendix 4). A higher evaporating temperature means a smaller $\Delta T_{e(in)}$, and so lower $Q_e$ would normally be expected according to the evaporator characteristic; this is contrary to results of the balancing exercise. The fact that the conventional method overestimates $Q_e$ for a smaller $\Delta T_{e(in)}$ demonstrates that the error is actually more severe than implied by Figure 94.

Furthermore, the overestimation of capacity is worse where mid-point temperature is used considering that $\Delta T_{e(in)}$ based on $T_{e,mp}$ implies a larger $Q_e$ than when calculated with $T_{e,dp}$. Ultimately, use of the conventional balancing technique using either $T_{dp}$ or $T_{mp}$ is expected to dramatically underestimate the capability of the system to meet the heat load.
With respect to the condenser, over-prediction of $Q_e$ is proportionally less than with $Q_e$, although the difference between equilibrium condensing temperatures are negligible across all approaches. Since the errors associated with the condenser are smaller, this suggests that most focus should be directed towards accurate evaporator characterisation. Finally, it was noted that under no circumstances does the use of the conventional balancing technique, using $T_{dp}$ or $T_{mp}$ provide better accuracy than the new technique. Neither can a preference be made in favour of adoption of mid-point or dew-point when used in the conventional balancing technique.

As mentioned earlier, the example systems (Table 29) represent varying sizes of $\Delta T_G$, $\Delta p$, $\Delta T_{e(m)}$ and $\Delta T_{c(m)}$. It is useful to review the results in respect of these differences, to identify any aspects demanding further attention. Observation of errors from the new technique does not show any distinct trends across any of the range of variables. For the conventional technique, comparison of the results for high and low $\Delta T_G$ do not reveal any consistent differences either.

Generally, the greater errors are for $Q_e$ when $\Delta T_G$ is large, depending upon whether $T_{dp}$ or $T_{mp}$ is employed, although low $\Delta T_G$ tends to result in a greater error for $Q_e$. The systems with high $\Delta p$ consistently (except one) produce significantly greater error than those with low $\Delta p$, since a larger $\Delta p$ causes a wider shift in saturation temperatures and therefore over-prediction of capacities; consistently, errors were greater with the large size compressor, which implicitly causes higher $\Delta p$. A higher refrigerant mass flow equates to an increase in capacity, thereby producing larger $\Delta p$ and simultaneously depressing $T_{e,dp}$ and elevating $T_{c,dp}$. Quantitatively, the errors in capacity generated in these examples range from about $\pm 1\%$ to $+15\%$ for $Q_e$ and $\pm 20\%$ for $Q_e$, with the largest errors being associated with the conventional technique using $T_{dp}$ or $T_{mp}$ for a system with large size compressor. The over-prediction of capacity observed in both approaches of dew- and mid-point using the conventional technique is consistent with the fact that it does not account for these parameters ($\Delta T_G$, $\Delta p$ and $\Delta T_{ash}$) when operating under non-standard conditions (e.g., reduction in capacity with $\Delta p$ and variable $\Delta T_{ash}$). With this it is evident that an improvement in accuracy of prediction of equilibrium conditions will always be achieved when using the new technique.
SECTION 6: FINAL SUMMARY AND CONCLUSION

6.1 Summary of work
The objective of this study was primarily to develop a means of integrating the characteristics of zeotropic refrigerants (which exhibit a temperature glide) and associated effects of pressure drop and sensible heat transfer in heat exchangers, into design activities within the component-system chain. Several stages within this chain necessitate the generation, manipulation and interpretation of individual component performance data to enable correct selection and design of the system. Of particular relevance to this are the performance testing/rating, catalogue data, selection and system balancing stages.

Particular attention was paid to the behaviour of condenser and evaporator performance when using a zeotrope. Results from a detailed performance simulation were used to clarify the implications of zeotropes. Accordingly, new rating methods and selection of suitable reference state temperatures were proposed. In terms of system balancing, the conventional technique was adapted to produce a new technique. Finally, use of the new technique was illustrated by means of a worked example.

6.2 Key findings and originality
The literature review revealed there were virtually no studies relating to component characterisation and system balancing when using zeotropes. The review of component rating standards and catalogues showed no consistency in the choice of suitable reference states. An assessment of current methods of performance rating and catalogue data indicated that there were inconsistencies between these methods and the parameters necessary to handle zeotropic refrigerants correctly. In particular, it was established that the condenser and evaporator were affected in terms of choice of rating reference states, as was the general logic for the basis of rating and balancing.

It was found that the conventional definition of approach temperature difference was not valid since the existence of the glide prevents the saturated refrigerant inlet temperature from reaching HTF inlet temperature, which is contrary to the premise for standardised methods, thus the necessity for a revised characterisation method. Analysis of evaporator performance also identified the dependency of its capacity on the inlet condition of the refrigerant due to the intersection of the temperature glide.
Observation of simulation results confirmed that as refrigerant-HTF inlet temperature difference of a condenser or evaporator approached that of the temperature glide, exchanger capacity would theoretically converge on zero (despite the existence of a temperature difference). It was also found that capacity was also largely affected by the shape (up to ±200%) and size of the glide (up to ±100%) especially at small approach temperature differences. Simulations also showed condenser capacities to have a strong dependency on desuperheat (30% at 30 K) and this identified the need to include it within the condenser rating method. Pressure loss through exchangers was also found to affect component rating due to the resulting effect on reference state temperature (±20% at high approach temperature differences), and it was concluded that it should be the condensing dew-point temperature corresponding to the condenser inlet pressure, and the evaporating dew-point temperature corresponding to the evaporator outlet pressure.

The revised balancing technique was adopted to account for the various parameters associated with a zeotrope identified to affect component performance. The new technique was compared with the existing approaches (using mid-point and dew-point) where up to 20% over-prediction in component capacity at the balance conditions was observed. These findings demonstrate the improved accuracy associated with the revised techniques.

Various alterations to the production, use, and manipulation of performance data within the component-system have been proposed, and are detailed below, and summarised in Table 30.

- **Considerations for component performance testing rating.** At the outset, the performance testing of the component should ensure that it accounts for all the parameters relevant to latter stages of the component-system chain. These parameters are detailed in Table 30 (Items 1, 2 and 3).

- **Considerations for component catalogue data.** The catalogue data should provide capacity data according to Table 30 (Item 1). Whether a range of data is provided, or just at rating conditions with correction factors, it must be based on the correct reference temperature as detailed in Table 30 (Item 2). Options exist for the characterisation, or extension of standard rating data: it can be presented for an entire range of conditions, or generated from a characterisation method such as those proposed earlier. These options are detailed for the respective components in Table 30 (Item 4).

- **Considerations for component selection.** Guidelines for correct component selection should be provided in catalogue information. Components should be selected according to the correct reference temperatures and all corresponding performance parameters in Table 30 (Item 3).
Considerations for system balancing. To ensure accurate balancing, the new technique should be employed. Component data according to Table 30 (Item 5) should be available to complete the full system balance.

It is noted that the new methodologies are more complex than those found in the literature, and therefore working through the relevant procedures would be a more time consuming.

Table 30: Summary of usable data

<table>
<thead>
<tr>
<th>Item</th>
<th>Compressor</th>
<th>TEV</th>
<th>Condenser</th>
<th>Evaporator</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Capacity</td>
<td>$Q_{e,\text{comp}} \cdot Q_{c,\text{comp}}$</td>
<td>$Q_{e,\text{TEV}}$</td>
<td>$Q_{c,\text{cond}}$</td>
<td>$Q_{e,\text{evap}}$</td>
</tr>
<tr>
<td>2. Reference temperature</td>
<td>$T_{e,\text{dp(out)}} \cdot T_{c,\text{dp(m)}}$</td>
<td>$T_{e,\text{dp(out)}}$</td>
<td>$T_{c,\text{dp(m)}}$</td>
<td>$T_{e,\text{dp(out)}} \cdot T_{c,\text{dp(out)}}$</td>
</tr>
<tr>
<td>3. Other data required</td>
<td>$\Delta T_{\text{dsh}}$</td>
<td>$\Delta T_{\text{dsh}} \cdot T_{c,\text{dp(out)}}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4. Characterisation method</td>
<td>Double-quadratic equation, correction factors</td>
<td>Equation (81) and correction factors</td>
<td>Full data, rating formula, correction factors</td>
<td>Full data, rating formula, correction factors</td>
</tr>
<tr>
<td>5. Data required for balancing</td>
<td>$Q_{e,\text{comp}} \text{ vs. } T_{e,\text{dp(out)}} \cdot T_{c,\text{dp(m)}}$</td>
<td>None</td>
<td>$Q_{c,\text{cond}} \text{ vs. } T_{c,\text{dp(m)}} \cdot \Delta T_{\text{dsh}} \cdot T_{c,\text{dp(out)}}$</td>
<td>$Q_{e,\text{evap}} \text{ vs. } T_{e,\text{dp(out)}} \cdot T_{c,\text{dp(out)}}$</td>
</tr>
</tbody>
</table>

The importance of the present study can be considered with respect to the fact that the majority of new refrigerants are zeotropic mixtures, which are gradually replacing the existing pure and azeotropic mixtures. It was shown in Part 1 that close matching of a system’s cooling capacity with the application heat load improves efficiency, which is becoming an increasingly common prerequisite for new equipment due to the cost of energy, and environmental legislation. Given that the new methodologies assist with better determination of system capacity, a closer match with the application can be made, thereby promoting better efficiency.

6.3 Limitations and further work

Several aspects were neglected in the development of the new methodologies, which could be incorporated to improve applicability. These are explained below.
- **Expansion devices.** Other expansion devices, such as capillary tube or short tube orifices may be non-adiabatic and, and its capacity does not automatically adjust according to the demand. To include this type of expansion device within the balancing technique requires additional stages and manipulation of data, as well as a suitable rating method.

- **Ancillary components.** The new balancing technique assumes that compressor, TEV, condenser and evaporator are directly connected. However, since most systems employ interconnecting piping, a variety of valves, liquid receivers and other components, their influence on reference state temperatures (due to pressure losses) and rated capacities of the main components (due to heat transfer) should be accounted for.

- **Composition shift.** Any zeotropic refrigerant will be subject to a variation in circulating composition, and this affects refrigerant properties and consequently, capacities, glide characteristics and definitions of reference state. In addition to the characteristics of the mixture, the degree of composition shift is largely dependent upon system configuration, which makes it difficult to account for at the component rating and balancing stages. Nevertheless, integration of the influence of composition shift into both component characterisation and system balancing would provide additional benefits.

- **Influence of oil.** When the oil and refrigerant are mutually soluble, there can be an impact on refrigerant behaviour and properties. A combination of a zeotropic refrigerant and soluble oil complicate the issues, due to preferential solubility of the different refrigerant components within the oil. Evidently, to account for oil effects within the rating and balancing techniques would provide a more comprehensive approach.

It would be advantageous to produce a computational code to automate the manipulation of rating data and ultimately determine system equilibrium conditions. As such, inclusion of the influence of ancillary components would be simplified (for the user). More importantly, the code could be extended to “re-rate” various components that are currently presented in catalogues with insufficient detail.
REFERENCES


Prestcold. PL, K, L, LG and R-type compressor data. Copeland Corporation Ltd, Reading, Undated.


Searle. MDR-type air-cooled condenser data. KS and KM-type unit coolers. Searle Manufacturing Ltd, Fareham, Undated.


Appendix 1: Information on refrigerant mixtures

Table 31 is a list of commercial refrigerant mixtures taken from UNEP (2002), including their compositions as mass fractions. Saturation pressure (at 0°C dew-point temperature), the corresponding temperature glide, and mean glide shape index ($n_G$) were calculated from Refprop (Lemmon et al, 2002).

Table 31: Commercial zeotropic mixtures and selected properties

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Component</th>
<th>Component mass fraction</th>
<th>$p$ at $T_{dp}$ = 0°C (kPa)</th>
<th>$\Delta T_G$ at $T_{dp}$ = 0°C (K)</th>
<th>Glide shape index, $n_G$ (-)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-401A</td>
<td>R22 R152a R124</td>
<td>0.53 0.13 0.34</td>
<td>343</td>
<td>6.1</td>
<td>0.95</td>
</tr>
<tr>
<td>R-401B</td>
<td>R22 R152a R124</td>
<td>0.61 0.11 0.28</td>
<td>368</td>
<td>6.0</td>
<td>0.88</td>
</tr>
<tr>
<td>R-401C</td>
<td>R22 R152a R124</td>
<td>0.33 0.15 0.52</td>
<td>280</td>
<td>5.6</td>
<td>1.14</td>
</tr>
<tr>
<td>R-402A</td>
<td>R125 R290 R22</td>
<td>0.60 0.02 0.38</td>
<td>660</td>
<td>0.8</td>
<td>1.17</td>
</tr>
<tr>
<td>R-402B</td>
<td>R125 R290 R22</td>
<td>0.38 0.02 0.60</td>
<td>621</td>
<td>1.3</td>
<td>1.07</td>
</tr>
<tr>
<td>R-403A</td>
<td>R290 R22 R218</td>
<td>0.05 0.75 0.20</td>
<td>643</td>
<td>2.2</td>
<td>0.81</td>
</tr>
<tr>
<td>R-403B</td>
<td>R290 R22 R218</td>
<td>0.05 0.56 0.39</td>
<td>669</td>
<td>0.1</td>
<td>0.48</td>
</tr>
<tr>
<td>R-404A</td>
<td>R125 R143a R134a</td>
<td>0.44 0.52 0.04</td>
<td>613</td>
<td>0.5</td>
<td>0.70</td>
</tr>
<tr>
<td>R-406A</td>
<td>R22 R600a R142b</td>
<td>0.55 0.04 0.41</td>
<td>336</td>
<td>9.1</td>
<td>0.94</td>
</tr>
<tr>
<td>R-407A</td>
<td>R32 R125 R134a</td>
<td>0.20 0.40 0.40</td>
<td>559</td>
<td>5.3</td>
<td>0.93</td>
</tr>
<tr>
<td>R-407B</td>
<td>R32 R125 R134a</td>
<td>0.10 0.70 0.20</td>
<td>617</td>
<td>3.1</td>
<td>0.78</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R-407C</td>
<td>R32</td>
<td>R125</td>
<td>R134a</td>
<td>0.23</td>
<td>0.25</td>
</tr>
<tr>
<td>R-407D</td>
<td>R32</td>
<td>R125</td>
<td>R134a</td>
<td>0.15</td>
<td>0.15</td>
</tr>
<tr>
<td>R-407E</td>
<td>R32</td>
<td>R125</td>
<td>R134a</td>
<td>0.25</td>
<td>0.15</td>
</tr>
<tr>
<td>R-408A</td>
<td>R125</td>
<td>R143a</td>
<td>R22</td>
<td>0.07</td>
<td>0.46</td>
</tr>
<tr>
<td>R-409A</td>
<td>R22</td>
<td>R124</td>
<td>R142b</td>
<td>0.60</td>
<td>0.25</td>
</tr>
<tr>
<td>R-409B</td>
<td>R22</td>
<td>R124</td>
<td>R142b</td>
<td>0.65</td>
<td>0.25</td>
</tr>
<tr>
<td>R-410A</td>
<td>R32</td>
<td>R125</td>
<td>R32</td>
<td>0.50</td>
<td>0.50</td>
</tr>
<tr>
<td>R-410B</td>
<td>R32</td>
<td>R125</td>
<td>R32</td>
<td>0.45</td>
<td>0.55</td>
</tr>
<tr>
<td>R-411A</td>
<td>R1270</td>
<td>R22</td>
<td>R152a</td>
<td>0.02</td>
<td>0.88</td>
</tr>
<tr>
<td>R-411B</td>
<td>R1270</td>
<td>R22</td>
<td>R152a</td>
<td>0.03</td>
<td>0.94</td>
</tr>
<tr>
<td>R-412A</td>
<td>R22</td>
<td>R218</td>
<td>R142b</td>
<td>0.70</td>
<td>0.05</td>
</tr>
<tr>
<td>R-413A</td>
<td>R218</td>
<td>R134a</td>
<td>R600a</td>
<td>0.09</td>
<td>0.88</td>
</tr>
<tr>
<td>R-414A</td>
<td>R22</td>
<td>R124</td>
<td>R600a</td>
<td>R142b</td>
<td>0.51</td>
</tr>
<tr>
<td>R-414B</td>
<td>R22</td>
<td>R124</td>
<td>R600a</td>
<td>R142b</td>
<td>0.50</td>
</tr>
<tr>
<td>R-415A</td>
<td>R22</td>
<td>R152a</td>
<td>R600a</td>
<td>R142b</td>
<td>0.82</td>
</tr>
<tr>
<td>R-415B</td>
<td>R22</td>
<td>R152a</td>
<td>R600a</td>
<td>R142b</td>
<td>0.25</td>
</tr>
<tr>
<td>R-416A</td>
<td>R134a</td>
<td>R124</td>
<td>R600</td>
<td>0.59</td>
<td>0.40</td>
</tr>
<tr>
<td>R-417A</td>
<td>R125</td>
<td>R134a</td>
<td>R600</td>
<td>0.47</td>
<td>0.50</td>
</tr>
<tr>
<td>R-418A</td>
<td>R290</td>
<td>R22</td>
<td>R152a</td>
<td>0.02</td>
<td>0.96</td>
</tr>
<tr>
<td>--------</td>
<td>------</td>
<td>------</td>
<td>-------</td>
<td>------</td>
<td>------</td>
</tr>
<tr>
<td>R22</td>
<td>R22</td>
<td>R124</td>
<td>R600</td>
<td>0.50</td>
<td>0.47</td>
</tr>
<tr>
<td>R23</td>
<td>R32</td>
<td>R134a</td>
<td></td>
<td>0.05</td>
<td>0.22</td>
</tr>
<tr>
<td>R23</td>
<td>R125</td>
<td>R143a</td>
<td></td>
<td>0.20</td>
<td>0.36</td>
</tr>
<tr>
<td>R32</td>
<td>R125</td>
<td>R134a</td>
<td>R600</td>
<td>0.10</td>
<td>0.42</td>
</tr>
<tr>
<td>R32</td>
<td>R125</td>
<td>R143a</td>
<td></td>
<td>0.10</td>
<td>0.45</td>
</tr>
<tr>
<td>R32</td>
<td>R125</td>
<td>R143a</td>
<td>R134a</td>
<td>0.02</td>
<td>0.41</td>
</tr>
<tr>
<td>R32</td>
<td>R125</td>
<td>R143a</td>
<td>R134a</td>
<td>0.10</td>
<td>0.33</td>
</tr>
<tr>
<td>R32</td>
<td>R134a</td>
<td></td>
<td></td>
<td>0.25</td>
<td>0.75</td>
</tr>
<tr>
<td>R32</td>
<td>R134a</td>
<td></td>
<td></td>
<td>0.30</td>
<td>0.70</td>
</tr>
<tr>
<td>R125</td>
<td>R125</td>
<td>R22</td>
<td></td>
<td>0.70</td>
<td>0.30</td>
</tr>
<tr>
<td>R125</td>
<td>R134a</td>
<td></td>
<td></td>
<td>0.35</td>
<td>0.40</td>
</tr>
<tr>
<td>R125</td>
<td>R134a</td>
<td>R290</td>
<td>R22</td>
<td>0.42</td>
<td>0.06</td>
</tr>
<tr>
<td>R125</td>
<td>R152a</td>
<td>R227ca</td>
<td>R22</td>
<td>0.40</td>
<td>0.05</td>
</tr>
<tr>
<td>R134a</td>
<td>R142b</td>
<td></td>
<td></td>
<td>0.80</td>
<td>0.20</td>
</tr>
<tr>
<td>R134a</td>
<td>R142b</td>
<td></td>
<td></td>
<td>0.81</td>
<td>0.19</td>
</tr>
<tr>
<td>R134a</td>
<td>R142b</td>
<td></td>
<td></td>
<td>0.88</td>
<td>0.12</td>
</tr>
<tr>
<td>R170</td>
<td>R290</td>
<td></td>
<td></td>
<td>0.06</td>
<td>0.94</td>
</tr>
<tr>
<td>R290</td>
<td>R600a</td>
<td></td>
<td></td>
<td>0.50</td>
<td>0.50</td>
</tr>
<tr>
<td>R600a</td>
<td>R600</td>
<td></td>
<td></td>
<td>0.50</td>
<td>0.50</td>
</tr>
</tbody>
</table>
Appendix 2: Calculation of HTF properties

Water was chosen arbitrarily as the HTF used in the heat exchanger models. Property data from Incropera and De Witt (1990) was fitted to 6th-order polynomial equations to enable their rapid calculation at selected temperatures within the code (equation 264).

\[
\{ \rho, \mu, Cp, k \} = c_1 \cdot \tau^6 + c_2 \cdot \tau^5 + c_3 \cdot \tau^4 + c_4 \cdot \tau^3 + c_5 \cdot \tau^2 + c_6 \cdot \tau + c_7 
\]  

(264)

where \( \{ \rho, \mu, Cp, k \} \) refers to the selected property, and \( c_1, \ldots, c_7 \) are specific to each property and are listed in Table 32. \( \tau \) is a function of the absolute temperature according to equation (265).

\[
\tau = \frac{1000}{273.15 + T_{HTF}} 
\]

(265)

where \( T_{HTF} \) is the local HTF temperature.

Comparison of equation (264) with the original data showed an error of less than \( \pm 0.1\% \).

---

### Table 32: Constants for water property polynomial equation (265)

<table>
<thead>
<tr>
<th>Constant</th>
<th>Property ( { \rho, \mu, Cp, k } )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density ( (\rho) )</td>
<td>Viscosity ( (\mu) )</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>( c_1 )</td>
<td>415.370</td>
</tr>
<tr>
<td>( c_2 )</td>
<td>-7901.99</td>
</tr>
<tr>
<td>( c_3 )</td>
<td>62462.2</td>
</tr>
<tr>
<td>( c_4 )</td>
<td>-262581</td>
</tr>
<tr>
<td>( c_5 )</td>
<td>619076</td>
</tr>
<tr>
<td>( c_6 )</td>
<td>-775931</td>
</tr>
<tr>
<td>( c_7 )</td>
<td>404721</td>
</tr>
</tbody>
</table>
Appendix 3: Overview of new balancing technique

Table 33 gives a summary of the stages required for the new balancing technique. The second equation listed in stages 1, 2 and 3 refer to the proposed numerical form of the characteristic expressions

Table 33: Overview of new balancing technique

<table>
<thead>
<tr>
<th>Stage</th>
<th>Procedure</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Compressor data</td>
<td>Use compressor characteristic data for:</td>
<td>( (251, 191) )</td>
</tr>
<tr>
<td></td>
<td>( Q_{c, \text{comp}} = f (T_{c, dp(i out)}, T_{c, dp(i in)}) )</td>
<td>(252, 192)</td>
</tr>
<tr>
<td></td>
<td>( \Delta T_{\text{data}} = f (T_{c, dp(i out)}, T_{c, dp(i in)}) )</td>
<td>(250, 194)</td>
</tr>
<tr>
<td>2. Condenser data</td>
<td>Use condenser characteristic data for:</td>
<td>( (249, 217/218) )</td>
</tr>
<tr>
<td></td>
<td>( Q_{c, \text{cond}} = f (T_{c, dp(i out)}, \Delta T_{\text{in}}) )</td>
<td>(256, 215)</td>
</tr>
<tr>
<td></td>
<td>( T_{c, dp(i out)} = f (T_{c, dp(i in)}, \Delta T_{\text{in}}) )</td>
<td></td>
</tr>
<tr>
<td>3. Evap data</td>
<td>Use evaporator characteristic data for:</td>
<td>( (255, 231/232) )</td>
</tr>
<tr>
<td></td>
<td>( Q_{e, \text{evap}} = f (T_{e, dp(i in)}, T_{e, dp(i out)}) )</td>
<td></td>
</tr>
<tr>
<td>4. Condenser charac. matched to compressor desuperheat</td>
<td>Taking the relevant compressor characteristic data (equation 252) and the relevant condenser characteristic data (equation 249), new condenser characteristics are determined by matching the condenser capacity to desuperheat. This provides a condenser characteristic that is a function of both condensing temperature and evaporating temperature:</td>
<td>( (253) )</td>
</tr>
<tr>
<td></td>
<td>( Q_{c, \text{cond}} = f (T_{c, dp(i out)}, T_{c, dp(i in)}) )</td>
<td></td>
</tr>
<tr>
<td>5. Condensing unit charac.</td>
<td>Applying the new condenser characteristic (equation 253) to the compressor characteristic (equation 252) provides evaporating temperature equilibrium points which, when linked, provide a single condensing unit characteristic:</td>
<td>( (254) )</td>
</tr>
<tr>
<td></td>
<td>( Q_{c, \text{evap}} (T_{c, dp(i in)}) = f (T_{c, dp(i in)}) )</td>
<td></td>
</tr>
<tr>
<td>6. Evaporator charac. matched to condenser inlet</td>
<td>Using the compressor desuperheat characteristic (equation 250) with the condenser ( \Delta p ) conversion characteristic (equation 256) the corresponding value of ( T_{c, dp(i in)} ) is found for ( T_{c, dp(i in)} ) for the condenser and compressor. Using the evaporator characteristic (equation 255) a new set of evaporator data is obtained from selecting an evaporating capacity based on evaporating temperature and the value of ( T_{e, dp(i in)} ) that corresponding to the ( T_{c, dp(i in)} ):</td>
<td>( (257) )</td>
</tr>
<tr>
<td></td>
<td>( Q_{e, \text{evap}} = f (T_{e, dp(i in)}, T_{c, dp(i in)}) )</td>
<td></td>
</tr>
</tbody>
</table>

246
<table>
<thead>
<tr>
<th>Stage</th>
<th>Procedure</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>7. Evaporating unit charac.</td>
<td>Applying the new evaporator characteristic (equation 257) to the compressor characteristic (equation 251) provides condensing temperature ( T_{c,p_{in}} ), equilibrium points which, when linked, provide a single evaporating unit characteristic: [ Q_{r,s}(T_{c,p_{in}}) = f(T_{r,p_{in}}) ]</td>
<td>(258)</td>
</tr>
<tr>
<td>8. Convert unit capacity charac.</td>
<td>Using the compressor characteristic (equation 252) and the condensing unit characteristic (equation 254), convert the condensing unit capacity from condensing capacity to evaporating capacity: [ Q_{c,s}(T_{c,p_{in}}) = f(T_{c,p_{in}}) ], or Using the compressor characteristic (equation 251) and the evaporating unit characteristic (258) convert the capacity variable from evaporating capacity to condensing capacity: [ Q_{c,s}(T_{c,p_{in}}) = f(T_{r,p_{in}}) ]</td>
<td>(258)</td>
</tr>
<tr>
<td>9. Balance condensing unit with evaporating unit</td>
<td>Apply the condensing unit evaporating capacity (equation 258) to the evaporating unit evaporating capacity (equation 259) to obtain the system equilibrium point: [ \frac{Q_{c,s}(T_{c,p_{in}})}{f(T_{c,p_{in}})} - \frac{Q_{r,s}(T_{r,p_{in}})}{f(T_{r,p_{in}})} = 0 ], or Apply the condensing unit condensing capacity (equation 254) to the evaporating unit condensing capacity (equation 259) to obtain the system equilibrium point: [ \frac{Q_{c,s}(T_{c,p_{in}})}{f(T_{c,p_{in}})} - \frac{Q_{r,s}(T_{r,p_{in}})}{f(T_{r,p_{in}})} = 0 ]</td>
<td>(260)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(261)</td>
</tr>
</tbody>
</table>
Appendix 4: Results of worked examples for balancing

Table 34 present the main results from the different balancing exercises referred to in Section 5.

<table>
<thead>
<tr>
<th>Example system</th>
<th>Balance technique</th>
<th>Evaporator</th>
<th>Condenser</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$T_{e,dp}$ (°C)</td>
<td>$Q_e$ (kW)</td>
<td>Error (%)</td>
</tr>
<tr>
<td>System 1 (a); small compressor</td>
<td>model</td>
<td>-10.46</td>
<td>3.26</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-10.46</td>
<td>3.31</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-9.74</td>
<td>3.47</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-8.54</td>
<td>3.63</td>
</tr>
<tr>
<td>System 2 (a); small compressor</td>
<td>model</td>
<td>-13.73</td>
<td>2.91</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-13.73</td>
<td>2.98</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-13.06</td>
<td>3.09</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-13.03</td>
<td>3.09</td>
</tr>
<tr>
<td>System 3 (a); small compressor</td>
<td>model</td>
<td>-10.19</td>
<td>3.45</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-10.19</td>
<td>3.41</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-9.07</td>
<td>3.62</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-7.28</td>
<td>3.87</td>
</tr>
<tr>
<td>System 4 (a): small compressor</td>
<td>model</td>
<td>-13.06</td>
<td>3.10</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-13.06</td>
<td>3.12</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-12.24</td>
<td>3.25</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-12.28</td>
<td>3.25</td>
</tr>
<tr>
<td>System 1 (b); large compressor</td>
<td>model</td>
<td>-18.93</td>
<td>4.15</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-18.93</td>
<td>4.33</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-16.05</td>
<td>4.92</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-16.63</td>
<td>4.81</td>
</tr>
<tr>
<td>System 2 (b); large compressor</td>
<td>model</td>
<td>-21.62</td>
<td>3.94</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-21.62</td>
<td>3.97</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-19.66</td>
<td>4.41</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-20.18</td>
<td>4.31</td>
</tr>
<tr>
<td>System 3 (b); large compressor</td>
<td>model</td>
<td>-18.91</td>
<td>4.34</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-18.91</td>
<td>4.40</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-15.03</td>
<td>5.19</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-16.13</td>
<td>4.99</td>
</tr>
<tr>
<td>System 4 (b); large compressor</td>
<td>model</td>
<td>-21.18</td>
<td>4.07</td>
</tr>
<tr>
<td></td>
<td>new</td>
<td>-21.18</td>
<td>4.11</td>
</tr>
<tr>
<td></td>
<td>mid-pt</td>
<td>-20.29</td>
<td>4.37</td>
</tr>
<tr>
<td></td>
<td>dew-pt</td>
<td>-19.61</td>
<td>4.49</td>
</tr>
</tbody>
</table>
PART 3: APPRAISING THE FLAMMABILITY HAZARDS OF HYDROCARBON REFRIGERANTS
NOMENCLATURE

Letters

\( a \) \quad \text{a constant}

\( A \) \quad \text{area (m}^2\text{)}

\( A_d \) \quad \text{area of air discharge vent (m}^2\text{)}

\( A_{\text{eff}} \) \quad \text{effective area (m}^2\text{)}

\( A^F \) \quad \text{flammable area (m}^2\text{)}

\( A_f \) \quad \text{area of room floor (m}^2\text{)}

\( A_{Rm} \) \quad \text{area of room floor (m}^2\text{)}

\( A_v \) \quad \text{vent area (m}^2\text{)}

\( A_{vp} \) \quad \text{vent panel area (m}^2\text{)}

\( A_{\text{wall}} \) \quad \text{wall area (m}^2\text{)}

\( b \) \quad \text{a constant}

\( Br \) \quad \text{Bradley number (-)}
\( C \) concentration (kg m\(^{-3}\))
\( \overline{C} \) mean concentration (kg m\(^{-3}\))
\( C_{\text{actm}} \) actual measured concentration (kg m\(^{-3}\))
\( c_d \) discharge coefficient (-)
\( \overline{C}_d \) mean concentration with airflow (kg m\(^{-3}\))
\( C_f \) floor concentration (kg m\(^{-3}\))
\( \overline{C}_f \) mean floor concentration (kg m\(^{-3}\))
\( C^F \) flammable concentration (kg m\(^{-3}\))
\( \overline{C}_{\text{calc}} \) calculated mean floor concentration (kg m\(^{-3}\))
\( C_{f,\text{max}} \) maximum floor concentration (kg m\(^{-3}\))
\( \overline{C}_{f,\text{max}} \) maximum mean floor concentration (kg m\(^{-3}\))
\( C_{\text{max,des}} \) maximum design concentration (kg m\(^{-3}\))
\( \overline{C}_{r-d} \) mean concentration without airflow (kg m\(^{-3}\))
\( \overline{C}_{\text{rm}} \) homogenous room concentration (kg m\(^{-3}\))
\( C_{\text{surr}} \) concentration of surroundings (kg m\(^{-3}\))
\( \Delta C_{\text{wind}} \) wind pressure coefficient (-)
\( D \) diameter (m)
\( E \) an event (-)
\( f \) frequency (y\(^{-1}\))
\( f^* \) frequency of ignition (y\(^{-1}\))
\( f_{\text{fatal}} \) fatality frequency (y\(^{-1}\))
\( f_{\text{leak}} \) leak frequency (y\(^{-1}\))
\( g \) gravitational acceleration (m s\(^{-2}\))
\( g' \) reduced gravity (m s\(^{-2}\))
\( h \)  
height (m)

\( H \)  
enthalpy (J kg\(^{-1}\))

\( h_d \)  
height of air discharge (m)

\( h_{\text{eff}} \)  
effective height (m)

\( H_{\text{mol}} \)  
molar enthalpy (J mol\(^{-1}\))

\( h_r \)  
height of release (m)

\( h_{\text{rm}} \)  
room height (m)

\( h_{\text{therm}} \)  
height of thermal source (m)

\( I \)  
thermal intensity (s (kW m\(^{-2}\))\(^{1.3}\))

\( I_{\text{occ}} \)  
thermal intensity to room occupants (s (kW m\(^{-2}\))\(^{1.3}\))

\( K \)  
deflagration coefficient (Pa m s\(^{-1}\))

\( L/F/I. \)  
lower flammability limit (kg m\(^{-3}\))

\( m \)  
mass (kg)

\( m_{\text{flam}} \)  
mass of burned mixture/combustion products (kg)

\( m^F \)  
flammable mass (kg s\(^{-1}\))

\( \bar{m}^F \)  
mean flammable mass (of refrigerant) (kg s\(^{-1}\))

\( \bar{m}^F_{r,a} \)  
mean flammable mass of refrigerant and air mixture (kg s\(^{-1}\))

\( m_r \)  
mass of refrigerant (kg)

\( \dot{m}_r \)  
mass flow rate of release (kg s\(^{-1}\))

\( m_{r,a} \)  
mass of refrigerant and air (kg)

\( m_{r,e} \)  
entrained mass of refrigerant (kg)

\( m_{r,\text{max}} \)  
max allowable mass of refrigerant (kg)

\( m_{r,\text{rm}} \)  
mass of refrigerant within room (kg)

\( m_{\text{rm,sol}} \)  
total mass of refrigerant and air within room (kg)

\( m_v \)  
vented/ventilated mass (kg)
\( n \) an index (-)

\( N \) number of items (-)

\( N_{\text{conn}} \) number of connections (-)

\( N_{dp} \) number of spatial elements in descending plume (-)

\( N_{dr} \) number of room radial elements (-)

\( N_{E} \) number of events in reference time (-)

\( N_{\text{leak}} \) number of leak sizes (-)

\( N_{\text{mol}} \) number of moles (-)

\( N_{\text{occ}} \) number of occupants (-)

\( N_{\text{soi}} \) number of sources of ignition (-)

\( N_{\text{soi},\text{f}} \) number of sources of ignition at floor level (-)

\( N_{T} \) number of outside temperature increments (-)

\( N_{\text{wind}} \) number of wind speed increments (-)
\( p \) pressure (Pa)
\( P \) probability (-)
\( P_{\text{atm}} \) atmospheric pressure (Pa)
\( P_{\text{avail}} \) probability of availability (-)
\( PD \) potential difference (V)
\( Pe \) Peclet number (-)
\( P^{F^*} \) probability of ignition (-)
\( P_{\text{fail}} \) probability of mechanical failure (-)
\( P_{\text{fkal}} \) probability of fatality (-)
\( \Delta P_{\text{fri}} \) friction pressure difference (Pa)
\( P_{\text{sos}}^{F} \) probability of source of ignition (-)
\( P_{\text{v}}^{F} \) probability of flammable volume (-)
\( P_{\text{leak}} \) leak probability (-)
\( P_{\text{local}} \) probability of local leakage (-)
\( P_{\text{max}} \) maximum pressure (Pa)
\( \Delta P^o \) overpressure (Pa)
\( \Delta P_{\text{rm}}^o \) room overpressure (Pa)
\( \Delta P_{\text{cp}}^o \) overpressure necessary to remove vent panel (Pa)
\( P_{\text{perc}} \) percolation probability (-)
\( P_{\text{sta}} \) static pressure (Pa)
\( \Delta P_{\text{stack}} \) stack pressure difference (Pa)
\( P_{\text{Sys}} \) probability of leak from a system (-)
\( \Delta P_{\text{tot}} \) total pressure difference (Pa)
\( P_{\text{vel}} \) velocity pressure (Pa)
\( P_{\text{cp, fix}} \) vent panel fixing pressure (Pa)
\( \Delta P_{\text{wind}} \) wind pressure difference (Pa)
\( q \) heat flux (W m\(^{-2}\))
\( Q \) thermal load, output (kW)
\( q_{\text{occ}} \) heat flux radiated to occupants (kW m\(^{-2}\))
\[ r \quad \text{radius (m)} \]
\[ Re \quad \text{Reynolds number (-)} \]
\[ Ri \quad \text{Richardson number (-)} \]
\[ Ri_c \quad \text{local Richardson number for cloud/spreading plume (-)} \]
\[ S \quad \text{a set of events (-)} \]
\[ t \quad \text{time (s)} \]
\[ T \quad \text{temperature (°C)} \]
\[ t' \quad \text{reference time (s)} \]
\[ T_a \quad \text{air temperature (°C)} \]
\[ t_d \quad \text{post-ventilation period (s)} \]
\[ t_{d,min} \quad \text{min post-ventilation period (s)} \]
\[ \Delta t_E \quad \text{event interval time (s)} \]
\[ t_{fail} \quad \text{time until repair of failure (s)} \]
\[ t^F \quad \text{flammable time (s)} \]
\[ t_{flam} \quad \text{combustion/burning time (s)} \]
\[ T_{flam} \quad \text{flame temperature (°C)} \]
\[ t_{leak} \quad \text{leak time (s)} \]
\[ t_{leak,cst} \quad \text{catastrophic leak time (s)} \]
\[ t_{mech} \quad \text{operating time of mechanical ventilation (s)} \]
\[ t_{occ} \quad \text{occupancy time (s)} \]
\[ t_{pl} \quad \text{plume descent time (s)} \]
\[ t_r \quad \text{release time (s)} \]
\[ t_{res} \quad \text{residence time (s)} \]
\[ T_{Rm} \quad \text{room temperature (°C)} \]
\[ t_{sprd} \quad \text{floor spreading time (s)} \]
\[ t_{therm} \quad \text{presence time of thermal sources (s)} \]
\[ t_{tot} \quad \text{total time (s)} \]
\[ t_{unit} \quad \text{operating time of RAC unit airflow (s)} \]
\( u \) velocity (m s\(^{-1}\))
\( u_{\text{fri}} \) friction velocity (m s\(^{-1}\))
\( UFL \) upper flammability limit (kg m\(^{-3}\))
\( u_{\text{flam}} \) burning velocity (m s\(^{-1}\))
\( u_{\text{flam,lam}} \) laminar flame velocity (m s\(^{-1}\))
\( u_{\text{f,spr}} \) spread velocity across floor (m s\(^{-1}\))
\( u_p \) plume velocity (m s\(^{-1}\))
\( \bar{u}_{\text{flam}} \) mean room air speed (m s\(^{-1}\))
\( u_{\text{ss}} \) sonic velocity (m s\(^{-1}\))
\( u_{\text{wind}} \) wind speed (m s\(^{-1}\))
\( V \) volume (m\(^3\))
\( \dot{V} \) volume flow rate (m\(^3\) s\(^{-1}\))
\( \dot{V}_a \) air volume flow rate (m\(^3\) s\(^{-1}\))
\( \dot{V}_d \) discharge air volume flow rate (m\(^3\) s\(^{-1}\))
\( V_e \) entrained volume (m\(^3\))
\( V_F \) flammable volume (m\(^3\))
\( \bar{V}_F \) mean flammable volume (m\(^3\))
\( V_{\text{flam}} \) volume of burned mixture (m\(^3\))
\( \dot{V}_{\text{infil}} \) infiltration volume airflow rate (m\(^3\) s\(^{-1}\))
\( \dot{V}_{\text{mech}} \) mechanical ventilation volume airflow rate (m\(^3\) s\(^{-1}\))
\( V_p \) plume volume (m\(^3\))
\( \dot{V}_p \) plume volume flow rate (m\(^3\) s\(^{-1}\))
\( \dot{V}_r \) refrigerant volume flow rate (m\(^3\) s\(^{-1}\))
\( V_{\text{Rem}} \) room volume (m\(^3\))
\( \dot{V}_{\text{therm}} \) volume airflow rate from thermal sources (m\(^3\) s\(^{-1}\))
\( \dot{V}_{\text{unit}} \) unit volume airflow rate (m\(^3\) s\(^{-1}\))
\( V_v \) vented/ventilated volume (m\(^3\))
\( W_{\text{vp}} \) vent panel plane density (kg m\(^{-2}\))
mass fraction (-)

mass fraction of burned mixture/combustion products (-)

mass fraction of flammable refrigerant/air mixture (-)

mass fraction of refrigerant/air mixture (-)

distance (m)

burning distance (m)

distance to occupant (m)

entrainment coefficient (-)

local layer entrainment coefficient for cloud/spreading plume (-)

plume entrainment coefficient (-)

mixing exchange ratio (-)

integrated thermal intensity-frequency (s m^-2 W y^-1)

integrated overpressure-freq. (Pa y^-1)

integrated risk factor (-)

mixing effectiveness (-)

expansion ratio (-)

time fraction (-)

time fraction of infiltration rate (-)

time fraction of mechanical ventilation (-)

time fraction of occupying time fraction (-)

time fraction of on-cycle operation (-)

time fraction of thermal source (-)

time fraction of unit airflow (-)

fuel radiative fraction (-)

safety factor (-)
\( \lambda \) friction factor (-)

\( \Lambda \) room air changes \((h^{-1})\)

\( \Lambda_{50} \) reference infiltration air changes at 50 Pa \((h^{-1})\)

\( \Lambda_{infl} \) infiltration air changes \((h^{-1})\)

\( \Lambda_{mech} \) mechanical ventilation air changes \((h^{-1})\)

\( \Lambda_{unit} \) unit airflow air changes \((h^{-1})\)

\( M \) molar mass \((kg \ kmol^{-1})\)

\( \mu \) viscosity \((Pa \ s)\)

\( \Pi \) pi-group for dimensional analysis

\( \theta \) air discharge angle \((rad)\)

\( \rho \) density \((kg \ m^{-3})\)

\( \rho_a \) density of air in room \((kg \ m^{-3})\)

\( \rho_{flam} \) density of burned mixture/combustion products \((kg \ m^{-3})\)

\( \rho_r \) density of gas at room conditions \((kg \ m^{-3})\)

\( \rho_v \) density of mixture at vent \((kg \ m^{-3})\)

\( T \) turbulence factor (-)

\( \tau_{fri} \) fiction shear stress \((N \ m)\)

\( \zeta \) incidence ratio (-)

\( \sigma \) deflagration outflow interaction parameter (-)

\( \omega \) mass fraction of element (-)

\( \psi \) refrigerant retention ratio (-)

\( \xi \) angle adjustment factor (-)
Subscripts

0  condition at origin
h  height level element
i  environmental condition
in  into element
j  dispersion temporal/spatial element
k  airflow source
l  leak size element
leak  leakage
max  maximum value
min  minimum value
out  out of element
r  refrigerant

Superscripts

*  ignition
$F$  flammable
mol  molar
SECTION 1: INTRODUCTION

1.1 Hydrocarbon refrigerants

Hydrocarbons (HCs) are increasingly being used as alternatives to fluorinated refrigerants such as HCFCs and HFCs because of environmental issues.\(^{17}\) However, as they are flammable, this raises a safety issue not normally associated with conventional refrigerants. A substantial release of flammable refrigerant into a room that has not been specifically designed to accommodate flammable concentrations – such as an office environment – could result in fire and/or explosion. Safety standards are written as a step towards achieving an acceptable safety level when using HCs, but they should be used in conjunction with a risk assessment to further help mitigate the associated hazards.

1.1.1 Safety standards and codes

There are safety standards that provide guidance on the use and application of refrigerants, and these include specific requirements for flammable refrigerants. However, they do not necessarily provide insight into risk assessment methodology, refrigerant leakage, dispersion and consequences of ignition; instead, they offer practical guidance on best industry practice.

Within Europe, the generic standard for safety of refrigerating systems is EN 378: 2000. This classifies refrigerants according to their flammability and toxicity, and system design and installation requirements are given for each refrigerant classification. Flammable refrigerants are classified according to certain properties: a fluid with a low Lower Flammability Limit (LFL) and a large heat of combustion (HOC) is designated “A3” and those that have higher LFLs and small HOCs are “A2”. R290 and R600a are examples of A3, and R152a and R32 are examples of A2. The standard specifies the maximum quantity of refrigerant permitted in a location, whether to employ ventilation, refrigerant detection and/or other safety features, and details requirements for various system components. Equipment using flammable refrigerants must be devoid of sources of ignition. For systems located within occupied spaces, the maximum refrigerant mass per circuit is specified, for example, 2.5 kg of an A2, and 1.5 kg of an A3 refrigerant in publicly occupied spaces. However, to avoid relatively large quantities of refrigerant being released into small spaces, an additional limit is specified in terms of a maximum mean concentration. This is termed the “practical limit” and corresponds to 20% of the LFL, being approximately 8 g m\(^{-3}\) for HCs. For example, a room of 100 m\(^2\), the maximum refrigerant charge within a single refrigerant

\(^{17}\) This is addressed in detail in Part 1.
circuit would be 0.8 kg, whereas a system in a 250 m$^3$ public occupied room would have its charge limited to 1.5 kg (rather than 2.0 kg).

Other than the generic standard, there are specific product standards such as EN 60335-2-24 and EN 60335-2-89, which cover domestic and commercial refrigerators respectively. Both specify a maximum charge of 150 g of any flammable refrigerant regardless of flammability classification, room size and occupancy type; if this amount is exceeded, then EN 378 must be used. Also, industry codes such as ACRIB (2000) and IOR (2000) contain practical guidance on safety measures.

1.1.2 Risk assessment for refrigerating systems

The concept of “risk assessment” can be applied qualitatively (i.e., “what if?”), or quantitatively (i.e., determining the probability/frequency of the “what if”). In either case, the first stage is to conduct a qualitative assessment, so that potential hazards are identified, which in the case of a flammable refrigerant is ignition of a release. For this to occur, three criteria must be satisfied: a release of refrigerant, mixing of refrigerant and air to within the flammable range, and the presence of an “active” source of ignition (SOI).

For a release of flammable refrigerant into air, locally there will always be a gradient from 100% refrigerant to pure air, so it follows that a flammable concentration will always occur. An active SOI is one that has sufficient energy to initiate combustion between the flammable material and the oxygen in the air. This energy may be in the form of heat (e.g., from an electric heater), a spark (e.g., from an arc between electrical contacts) or a chemical reaction. Thus, the presence of a flammable mixture at the same time and location as the active SOI may result in ignition. Minimisation of this occurrence is necessary to reduce the risk.

Consideration of the construction, installation and operation of typical refrigeration and air conditioning (RAC) equipment provides an indication of the reduction measures available. A leak-tight refrigerant circuit contains refrigerant and is therefore prevented from mixing with air and reaching SOI. Containment can be achieved in most situations through proper design and installation, although unforeseen, external and human behavoural influences can lead to leakage. Leakage issues are well documented but whilst containment should be a priority during design, it cannot be employed as the sole mitigation technique to prevent ignition.

261
Whilst the potential SOI associated with equipment can be eliminated, the SOI within an occupied space cannot be controlled, so the only option is to minimise the size (or volume) and duration of a flammable concentration; this concept is core to the study.

1.2 Literature review
This literature review draws on studies relating to two main areas: risk assessment methodology for flammable gases, and more specifically, the behaviour of a release of flammable gas within a confined space.

1.2.1 Risk assessment of flammable gas
Given the common use of flammable materials, literature on appropriate quantitative risk assessment (QRA) methodology is extensive, although only a selection of the most relevant are discussed presently. A set of publications that provide comprehensive information on the subject were published by the Dutch Government. The "Purple Book" (CPD, 1999) details the general approach and formulae for risk calculations, and calls upon the "Red Book" (CPD, 1988) for calculating probabilities, the "Yellow Book" (CPD, 1997) for methods and data on release and dispersion, and the "Green Book" (CPD, 1992) for methods and data on consequences of ignition.

Due to widespread use of HCs as fuel gas (e.g., liquefied petroleum gas, LPG), QRA is frequently used and various models have been developed. Khan and Abbasi (1999 and 2000) present a QRA method (as used for their commercial software) for addressing the release of HCs in refineries. It was designed for major discharges of flammable material, and employs a number of models selected according to the type and location of the release. These deal with dispersion of an outdoor cloud and subsequent fire and/or explosion. The model outputs frequencies and severity of consequences. Ditali et al (2000) describe a similar approach (also as commercial software) but focused on LPG storage vessels. Both methods use a simple steady flow discharge equation and liquid spread model for the initial release, and an outdoor dispersion model to determine flammable concentrations. The consequence is limited to radiation emitted from the resulting fire. Some results are given, but no empirical data on aspects such as leakage and ignition sources was given. An industrial QRA model developed by Barry (1992) addressed failure probability of components within a LPG plant, and also presents an approach for calculating ignition frequency at time steps from initiation of the release.

Whilst these studies focus on outdoor releases, the only study found on indoor HC releases was Lee (2002). Although it mainly deals with consequences of toxicity in the case of a spill the
methodology was still relevant since calculations addressed discharge, evaporation and dispersion of the release, and estimation of fatality from a given concentration; thus analogous to fatality from explosion effects. Rew et al (2000) evaluated ignition probability models for LPG releases from off-site storage vessels, and compared the results with a newly developed model. Calculations of the existing and new methods relied on various coefficients dependent on location type (urban, rural and industrial, indoor/outside) and time of day/night. Ignition probability was estimated for a range of locations and size of flammable clouds.

1.2.2 Risk assessment of flammable refrigerants

To date, many QRAs that address the use of flammable refrigerants in different types of stationary RAC equipment have been published, as well as some for vehicle air conditioners and transport refrigeration. The latter were not considered relevant and therefore not discussed.

Several QRAs were based on the approaches detailed in the Purple Book (CPD, 1999). Adopting its principles, Blom-Bruggeman et al (1996a) published a generic method specific to the context of RAC equipment, covering both flammable refrigerants (such as HCs) and ammonia. Subsequent studies by Blom-Bruggeman et al (1996b) and Elbers and Verwoerd (1997) addressed risk of fatality from HCs used in bulk milk tanks and domestic heat pumps respectively. Normal operation, service/maintenance and charging activities were included in both, and the methodology employed was based on a fixed set of installation characteristics and environmental conditions. Evaluation of leak frequencies, release size, dispersion of the release and fire/explosion consequences was made with separate commercially available models designed for industrial process plant. Values used for ignition probabilities were constant and releases inside rooms were assumed to disperse evenly, whilst more sophisticated models were employed for releases outside. A more recent study from the same group (van Blanken and Verwoerd, 2001) assessed a commercial freezer cabinet in a number of different locations. The approach was similar to those detailed above, although results of ignition tests using cigarettes were included, and an explosion from within the compressor compartment was modelled. Publications including van Gerwen and Jansen (1994), van Gerwen and Koffijberg (1995) and van Gerwen and Verwoerd (1996) reiterate the results of these studies.

A series of QRAs for a range of commercial refrigeration equipment, including supermarket chest freezers, integral supermarket display cases, and drinks chillers in public houses were produced by ADL (1998). These were based on detailed fault tree analysis specific to the equipment and installation, assuming three activity modes (normal operation, defrost and maintenance). They
called on separate release size, dispersion and fire/explosion software, and leak frequencies were from consultation with manufacturers and contractors. Dispersion modelling used a simplistic approach based on one fixed set of environmental conditions, and SOI were specified according to the application.

One of the earliest works relating to QRA of flammable refrigerants is Nowacki (1995), which summarised a basic methodology. Software was provided, which calculates societal risk (i.e., risk to the local population) as a function of refrigerant mass and characteristics of the surrounding area (such as population density). Historical data for LPG incidents was used to estimate the number of injuries and fatalities associated with an installation of a given mass of flammable material. Thus, the approach is basic and takes no account of specific characteristics of the equipment or the environmental conditions.

Wolfer and Seiler (1999) and Wolfer (1999) report on a QRA for three different applications: a domestic heat pump, an indirect central supermarket system and an integral display cabinet. Probability calculations were made according to Blom-Bruggeman et al (1996a) and a pre-published draft of EN TR 14739 (2004). Whilst the report contains lengthy discussion on the characteristics of each application, there was minimal input data; descriptions of the calculation procedures were brief and the dispersion was treated simplistically. One interesting aspect of the study was a comparison of risk associated with the heat pump versus that of a gas heating system.

Recently, the standards committee responsible for EN 378 has published a report (EN TR 14739: 2004) on the methodology for conducting a QRA on domestic refrigerators. The methodology is simplistic in that basic empirical data are given for use in basic calculation procedures. However, unlike most other references it considers stages other than in-use, such as transportation of appliances, servicing and disposal. For each stage comprehensive flow charts are given. Given that the purpose of the document is methodology, it contains to no results.

A comprehensive study by Goetzler et al (1998) addressed flammable fluorocarbon refrigerants in residential split air conditioners. Part of the study reported on extensive ignition test conducted on common devices to determine which were potential SOI, and a number of leak simulation tests to characterise refrigerant concentrations. Conventional fault tree analysis was used to estimate frequency of fires in different household locations.
Overall, these studies cover a wide range of RAC equipment and charge sizes and most address operation and servicing activities separately. Results comprised different forms of risk, including occurrence of fire, explosion, secondary fire, injury, fire damage, although the majority use frequency of fatality. Frequencies for fire during normal operation vary considerably from $10^{-9}$ to $10^{-4}$ year$^{-1}$, whereas fatality is typically $10^{-4}$ to $10^{-8}$ year$^{-1}$. It was not possible to make comparisons between the results because each was based on unique equipment and applications. Specific details of each are provided in a summary in Appendix 1 (Table 55).

1.2.3 General gas dispersion in confined spaces
The subject of gas dispersion is vast, and therefore it was necessary to establish the main characteristics relevant to this study, and useful information was found in Britter (1988), CCPS (1996) and Woodward (1998). Also, the mixing process is complex and dependent upon a number of factors, so knowledge of release characteristics and surrounding environment is essential. First, releases can be split into two categories: releases to the open atmosphere (outside) and released within confined spaces (inside). Of relevance to this study are releases within confined spaces, so most literature related to outdoor dispersion was not covered. Given that all refrigerants of interest have a density greater than air, studies on light and neutrally buoyant releases were also neglected. Further, releases are also differentiated in terms of instantaneous (or puff), and continuous releases.

A leak initially produces a jet or plume for which “aspect ratio” – or the size of the plume to the room dimensions – is an important factor. The difference between low and high aspect ratios are characterised by dominant mixing in the descent of a plume as opposed to mixing occurring only at ground level in the case of a low aspect ratio release. There are two basic mechanisms by which mixing occur. One is diffusion which is caused by random motion of particles across a concentration differential. The second mechanism is convective mixing, comprising buoyancy and externally driven mixing.

Most experiments have shown the rate of convective mixing, caused by density difference and/or external air movement, to be proportional to a (vertical) “stratification parameter” (Britter, 1988), termed the Richardson number ($Ri$). This is used to characterise the work required to mix a stable stratified layer and it depends upon the environment and the nature of the release such as the buoyancy or externally induced convection. Normally both diffusion and convection are present and a measure of their relative impacts is the Peclet number ($Pe$). When $Pe/Ri > 6$, the effect of diffusion is negligible (Britter, 1988). Another significant parameter (particularly for
low aspect ratio releases) is the texture of the surface below the cloud. A rough surface causes drag so the horizontal or spreading velocity decreases, which inhibits mixing.

Some investigations considered the influence of room objects on dispersion. Experiments by Leach and Bloomfield (1973) indicated that objects such as furniture had a negligible effect on mixing. Britter (1988) confirmed that experiments with room obstacles generally show that they divert the direction of the gas flow and in doing so increase turbulence, leading to a higher rate of mixing. The exception is where structures channel the flow thereby reducing the exposed area and thus mixing rate. A small number of tests by Clodic (1997) used a 100 mm barrier laid across the room floor. Contrary to the conclusions of others, it was found that floor concentrations were higher than that without the barrier, comparable to a release into a room with a smaller floor area.

The influence of household furniture was further investigated by Jabbour and Clodic (2002) and Jabbour et al (2003). Tests demonstrated refrigerant accumulated beneath a sofa, although elsewhere the refrigerant dispersed as would be expected. Also noted was the influence of a thick carpet which was found to "accumulate" refrigerant whilst better mixing occurred elsewhere. The carpet increased the surface roughness, which corresponds to observations by Britter (1988).

1.2.4 Releases into quiescent rooms

There were a number of studies relating specifically to releases into "still" enclosures, absent of forced airflow. An early study by Baines and Turner (1969) addressed the development of a single plume by entrainment in an enclosed space of high aspect ratio. A generalised model was described that accounts for density difference, release velocity and diameter, shown to correlate closely with measurements. An illustration of the dispersion process is shown in Figure 95(a), where the plume impinges on the floor surface to produce overturning or continuous pushing up of the preceding layer, creating stratified layers. As the layer depth increases, the mixture is "re-entrained" into the plume as it passes through the layer. Whilst the observations of Baines and Turner apply to a downward release of a dense refrigerant, a paper by Baines et al (1990) provided insight into the situation where a release is pointed upward thereby forming a "fountain". By comparison with the downward release (as Figure 95a) considerable turbulence developed when the plume reversed its direction with the consequence of more effective mixing.

Various authors reported on continuous releases into (full-scale) rooms. Fletcher and Johnson (1986) continuously released CO₂ into an unventilated room, and their results mirrored those of
Baines and Turner (1969). At any time during the release, concentration at the floor was highest, and gradually reduced with height up to the leak point, with the concentration profiles represented by a parabolic relation with height (dotted curve in Figure 95b). A release from low level produced a much higher floor concentration since the distance for the plume to descend was insufficient to establish sufficient buoyancy driven mixing and entrainment. A release of lower density gas tended to produce lower concentrations and to an extent that a homogenous concentration occurred when the release point was above three-quarters of the room height.

More detailed work by Marshall (1983) showed the profile remained constant over time (when forced airflow was absent) whilst continued gas flow caused the overall concentration within the region to increase; this is consistent with Britter’s work (1998). Following Fletcher and Johnson (1986), Marshall also demonstrated concentrations were sensitive to the direction of the leak especially where the release was close to the stratified layer. With a release from the ceiling, the release direction had a lesser influence on floor concentration. Positioning of leak source at different lateral locations within the room had negligible effect. Further experiments by Marshall indicated that releases into single rooms were representative of releases into complex structures such as two or more rooms linked by an open passage, or rooms with irregular geometries.

![Figure 95](image)

*Figure 95 (a) Layer formation in a quiescent room and, (b) concentration profiles*

The concept of Baines and Turner (1969) was adapted by Cleaver et al (1994) – in the form of a non-dimensional correlation – to describe the behaviour of gas build-up in quiescent rooms according to release characteristics and room dimensions. From experiments with positively and negatively buoyant gases, it was observed that jet length, momentum and buoyancy determined the mixing rate, and formation of a stratified layer. In line with Marshall’s work, measurements
revealed two types of gas layer, as seen in Figure 95(b): one of homogenous concentration and another where concentration reduced from that of the first layer to zero (at the height of the source). Additionally, high velocity releases tended to mix well, leading to a near-homogenous room concentration.

Rabkinkov (1988) presented concentration measurements for a number of continuous releases into a room from different heights over several time increments. The results were consistent with Cleaver et al (1994), and Fletcher and Johnson (1986). In contrast to the approach of Cleaver et al, a diffusion theory-based model was developed, which showed good agreement with the measurements.

In a study specific to releases of flammable refrigerants, Kataoka et al (2000a, 2000b) considered the case of a rapid release from an air-conditioner. The dependency of the development of floor-level concentration on several variables was evaluated for a high aspect ratio situation, including release height, velocity and floor area. Again, velocity was shown to have a significant effect on mixing rate (especially at low level), and floor concentrations reduced as leak height and gas density increased, which results generally agreed with Cleaver et al (1994), Marshall (1984) and Fletcher and Johnson (1986). Room tightness was generally found to affect floor concentrations, since even a room with minimum fabric leakage produced lower concentrations than a completely sealed room. Using measurements, the concentration-height profile was characterised with a parabolic equation, so floor concentration could be estimated if floor area and release height and mass were known.

A series of refrigerant leak simulation tests were reported by Clodic and Cai (1996, 1997) and Clodic (1997). R290 and R600a were released through a capillary tube from freezers and portable air-conditioners, over a range of leak rates, heights and positions within the room. A clear relationship was found between release height and maximum floor concentration, although in contrast to some other studies, the results did not show an obvious correlation between increasing release flow rate and floor concentration for either refrigerant. Releases from non-central lateral positions (particularly at room corners) produced higher floor concentrations. Following cessation of the release the denser R600a was found to reside as a stratified layer for longer periods than R290.

Zhao et al (2002) addressed the dispersion of a jet – independent of the room and airflow conditions – and presented a model to characterise local concentrations. The steady flow equation
for sonic flow was used to estimate leak exit velocity from different parts of a refrigeration circuit. The concentration field was calculated with a Gaussian model based on a constant entrainment (or “turbulence”) coefficient.

In the event of a release of a volatile liquid, vapourisation occurs rapidly. Generally, the rate of evaporation is in the order of 10 m³/hr⁻¹ m⁻² for a heat transfer coefficient of about 30 W m⁻² K⁻¹, where denser and more conductive floor material or lower normal boiling point increases boiling rate (CCPS, 1996). Other influencing factors include air velocity and ground surface geometry. Because of the rapid evaporation rates, dispersion of a vapourising pool is often treated as a cloud release (Woodward, 1998). Kataoka et al (2000a, 2000b) supports this, where concentration measurements following a liquid refrigerant release showed subsequent room concentration to be almost homogenous.

1.2.5 Releases into rooms with ventilation
Several studies address releases into rooms given forced airflow which is analogous to the various convection sources that cause turbulent motion within rooms; from external wind (i.e., infiltration), mechanical ventilation, and thermal convection from heat sources.

In addition to experiments under still conditions, Fletcher and Johnson (1986) also studied releases of CO₂ into a ventilated room whilst varying release location relative to air discharge position. With small release rates and moderate ventilation rates, a homogenous gas concentration was generally observed regardless of release position. Ventilation from floor upwards – C(in) to X(out) in Figure 96 – and counter-flow to the plume flow when release at high level (Figure 96) produced rapid mixing. Upwards ventilation with entering air at low level and the outlet at mid-height – C(in) to Y(out) in Figure 96 – reduced the rate of mixing. Downwards ventilation parallel to the plume direction – A(in) to Z(out) in Figure 96 – produced a homogenous mixture from halfway between the release point and the air outlet. Similarly a release at mid-height gave a homogenous mixture from halfway between the release point and the air outlet.

A higher mixing rate was therefore produced when airflow is counter to plume flow but removal of gas is more effective where a parallel flow configuration is used, although this leads to a higher concentration within the layer. Fletcher and Johnson presented a modified decay-equation to predict the change in room concentration, and included a factor for imperfect mixing between the inlet air and the room air. This factor varied according to room size (for example it was shown that mixing was more effective in smaller rooms), ventilation inlet location and airflow rate.
Recirculation fans were tested and resulted in near-perfect mixing. Also the effect of ventilation when a stratified layer already existed within a room was investigated. Comparison with results for airflow in parallel to the plume — e.g., release at high level with $A_{\text{in}}$ and $Z_{\text{out}}$ in Figure 96 — shows the effectiveness of forced airflow on mixing was diminished. This occurs because the energy needed to overcome the inertia presented by the negative buoyancy of the dense layer is greater.

![Figure 96: Positioning of air inlet and outlet positions and release locations within a room](image)

Effect of ventilation was also assessed by Marshall (1984). In a similar way to Fletcher and Johnson, Marshall looked at the effect of ventilation in relation to the release source location. Where the two flows were parallel — e.g., release at high level and $A_{\text{in}}$ to $Z_{\text{out}}$ in Figure 96 — the concentration-height profile was similar to the unventilated case, although more evenly distributed within the layer. In all cases, higher airflow rates reduced the maximum floor concentration. Ventilation counter to the release — $C_{\text{in}}$ to $X_{\text{out}}$ in Figure 96 — resulted in a similar concentration profile as was observed for the quiescent room, where two distinct layers develop. Cross-room ventilation was also considered and measurement showed lowest concentrations were obtained when the outlet vent was at low level (in the case of a dense gas) irrespective of the inlet position. Similarly with a dense gas, an outlet vent at high level resulted in negligible mixing. The dependence of floor concentrations on release velocity lessened with increasing ventilation rates. In line with the findings of Leach and Bloomfield, it was found that much greater ventilation rates were required to initiate mixing of an already stratified layer, than if the ventilation was operating during the release.
Leach and Bloomfield (1973) conducted similar mixing experiments with pre-mixed air and CO₂, i.e., with negligible buoyancy, and a second case with pure CO₂, under a range of ventilation rates. Mixing of pure CO₂ was less effective than with the neutrally buoyant CO₂/air mixture since the higher density gave the release greater inertia. However, with a low release rate, a moderately low air change guaranteed homogenous room concentrations. It was observed that the position of the ventilation outlet affected the development of a stratified layer below the outlet duct. A model was developed to describe the process, which showed good agreement with experiments. Measurements also supported similar observations of a parabolic variation of concentration with release height at low ventilation rates, but this concentration-height profile approached linear as airflow increased. The study also found that velocity of the discharge source also affected floor concentration.

Heiselberg (1991) measured concentrations of CO₂ for a constant gas flow under a range of ventilation rates. Increasing airflow gave near-homogenous concentrations. Also noted was a peak in concentration at release source height. This result seemed inconsistent with other studies but it can probably be explained by the CO₂ and air being premixed prior to the release into the room, which would carry it across the centre of the room, i.e., where the high concentrations were observed. Above 1.5 h⁻¹ air change, the concentration distribution was no longer influenced by the increasing airflow rate. By varying the locations of the air inlet and the source of the release, it was found that these factors were important with respect to the density of the gas. For example, a ventilation source at low level when a high-density gas is released produced better mixing because of the greater gravity induced convection within the room.

Clodic and Cai (1996) conducted tests with extract ventilation and air conditioning unit fan operating. These showed significantly lower concentrations at floor level compared to the same tests with no forced airflow. More recent tests by the same group (Jabbour and Clodic, 2002, Jabbour et al, 2003) measured floor concentrations occurring from a release of different flammable refrigerants with extract ventilation, unit airflow and vents within the room fabric. These also demonstrated that presence of higher airflow rates dispersed refrigerant more effectively than without forced airflow. Positioning of mechanical ventilation ducts was changed between the outlet at low and high level. Contrary to findings from other studies, these tests showed similar peak concentrations regardless of outlet position, especially when releases were made close to the floor. Lastly, the effect of natural ventilation openings on the wall at floor level resulted in lower peak concentrations, whereas elevated openings made no difference.
In terms of thermally induced convection, Cleaver et al (1994) reported that the air movement generated by a domestic radiator was sufficient to prevent stratified layers occurring. As described in CCPS (1996) and Woodward (1998), the effects of thermally induced convection sources (e.g., solar radiation) increase mixing rate in the case of outside releases. No studies were found that specifically considered this aspect in confined spaces.

1.3 Formulation of objectives

HCs (and other flammable refrigerants) are being employed in a variety of different RAC applications and since this is relatively new the approaches used to address flammability hazards are limited. Considering the areas covered by the literature review, it is evident that current safety guidance neglects to account for many aspects associated with the behaviour of flammable gases. A number of key areas relevant to the flammability risk were identified from the literature and these should be accounted for when developing improved methods for addressing the safety requirements of relevant RAC equipment.

1.3.1 Summary of findings

There were a number of subjects related to the flammability hazard of HC refrigerants and the main observations follow.

- **Risk assessment of flammable gas and refrigerants.** Most QRA methods are for very large outdoor releases of flammable gas, and have little relevance to flammable refrigerants. However there have been a number of QRAs focussing specifically on the RAC situation, and an overview of the way in which various parameters were dealt with is detailed in Table 35. For ignition probability, mostly a single value was used regardless of the environment, and only differentiated between instant and delayed ignition.18 Between two and three different leak sizes were assumed and data came from discussions with service people, or using methods normally employed for industrial plant. Dispersion of refrigerant was accounted for by dispersion tests specific to the equipment being evaluated, or from basic dispersion models mainly used for external releases of large gas clouds. Similarly, consequences including explosion and/or fire were mainly evaluated using methods developed for industrial situations.

- **Gas dispersion in confined spaces.** Only a limited number of published documents relate to the specific conditions of interest, being finite releases of dense gas into confined space with various forms of airflow. Those that are relevant generally demonstrate many similarities in

18 Instant ignition is where material is ignited whilst being release, and delayed ignition follows cessation of the release.
their analysis and conclusions, which provide confidence in terms of interpretation of mixing phenomenon in confined spaces. Several papers also related specifically to refrigerants. Mixing of a release is dependent upon many factors, and it is therefore important to precisely define the conditions. This includes establishing the release as instantaneous or continuous, ventilated or unventilated and aspect ratio. Generally in terms of a release of dense gas, a high velocity, elevated, upwards directed source would promote good mixing. Conversely, a downward, low velocity release near to ground level will inhibit mixing, equating to higher floor concentrations. Volatile liquid releases generate turbulence as they vapourise, which normally mixes more effectively than a gaseous release. A dense release into a room mixes with the ambient air, and upon reaching the floor develops a stratified layer. Disagreement exists in characterising the concentration-height profile; some studies indicate two distinct layers, whereas others demonstrate a continuous parabolic gradients, but probably due to differences in release and environmental conditions. There is a clear relationship between developed concentration, release rate and airflow rate. Above a certain ventilation rate, homogenous concentrations are achieved. The direction of the airflow discharge relative to the direction of the release is significant in terms of mixing. The presence of additional objects within a space will generally result in improved dispersion except where channelling occurs to confine the release further.

It transpires that studies either focus specifically on carrying out a QRA, or investigate gas dispersion, and little attempt has been made to address both aspects simultaneously.

1.3.2 Aims and objectives
QRA evaluations are typically based on models for industrial and large-scale situations that lose relevance when applied to non-industrial applications. The review highlights the necessity for a more detailed approach to the evaluation of risks associated with the use of HC refrigerants, and to establish more thorough guidance for design and installation practices and guidance relating to the factors that affect gas dispersion.

The QRA approach is considered to be the most suitable for addressing the influence of design and installation characteristics of equipment. The review indicated a more concise approach to QRA in RAC equipment is required where a broader range of parameters need to be accounted for. This includes a proper assessment of SOI, an improvement in evaluating leakage, and thorough characterisation of dispersion of refrigerant. It is also important to account for different operating situations, including cycling and safety controls. In addition to normal operation, the
method should also be capable of handling servicing and storage activities. The work was carried
out in several stages.

Table 35: Summary of the ways that main parameters were determined in QRAs

<table>
<thead>
<tr>
<th>Reference</th>
<th>Parameter</th>
<th>SOI</th>
<th>Leakage</th>
<th>Dispersion</th>
<th>Consequences</th>
<th>Stages</th>
</tr>
</thead>
<tbody>
<tr>
<td>Goetzler et al (1998)</td>
<td>Probability of each being active</td>
<td>Small and catastrophic, anecdotal frequency for catastrophic and small leak</td>
<td>Experiments of floor concentrations</td>
<td>Based on ignition experiments</td>
<td>Operating (for different locations within a residence), servicing</td>
<td></td>
</tr>
<tr>
<td>Wolfer and Seiler (1999)</td>
<td>Constant (delayed, instant)</td>
<td>Instant/constant, catastrophic/small</td>
<td>Decay equation</td>
<td>Assuming explosion or fire outcome according to situation</td>
<td>Operating, servicing</td>
<td></td>
</tr>
<tr>
<td>Elbers, Verwoerd, van Blanken, van Gerwen (1996)</td>
<td>Constant (delayed, instant)</td>
<td>1%, 10%, 100% of full bore rupture, and frequencies from industrial plant</td>
<td>Commercial “Release” model (mainly decay equation)</td>
<td>Explosion severity based on commercial model prediction</td>
<td>Operating, servicing</td>
<td></td>
</tr>
<tr>
<td>ADL (1998)</td>
<td>Constant (delayed, instant) based on estimate</td>
<td>Catastrophic leak, small leak, anecdotal frequencies</td>
<td>Maximum flammable volume from total release</td>
<td>Explosion or fire severity based on commercial software</td>
<td>Operating, defrosting, servicing</td>
<td></td>
</tr>
</tbody>
</table>

Probability models

Section 2 presents the development of the overall QRA model. The first step was to break down
the events that ultimately lead to ignition of refrigerant and subsequent effects. This included
identifying failure modes and local environmental characteristics that affect the likelihood of
ignition and the severity of the resulting events. A general framework for calculating frequency of
ignition and its consequences was developed. Similarly, a set of numerical limits were determined
in order to judge the significance of the calculated risk. Individual probability models were
produced to calculate the components of the overall risk; leak frequency, SOI probability and
probability of a flammable concentration. Each of these component probability models relies on
physical data to represent the mentioned events.
Dispersion of refrigerant leaks

Suitable data on dispersion of refrigerant leaks was largely absent so it was necessary to conduct suitable experiments, and use this to develop a dispersion model in Section 3. RAC equipment and installations are reviewed to identify the characteristics that would influence gas dispersion, and experiments were designed accordingly. Concentrations from simulated leaks were measured and the data was subsequently employed to produce a number of correlations and incorporated into the dispersion model.

Consequence

Ignition of a flammable mixture produces fire and explosions that determine the likelihood of fatality and other consequences. Section 4 presents additional sub-models to calculate the severity of the consequences of thermal intensity and overpressure to be employed by the fatality probability models.

Evaluation of risk

In Section 5, outputs from the entire risk model are presented, and attention is paid to the influence of the various factors on calculated risk. An exhaustive examination of these various factors helps identify the effectiveness of risk-reducing measures that can be applied to the design and installation of RAC equipment.
SECTION 2: GENERAL RISK ANALYSIS

2.1 Fundamental risk model

The purpose of this section is to analyse the sequence of events, and quantify the individual events that lead to a hazardous outcome. Prior to any numerical assessment, a qualitative analysis was carried out where the sequence of events, from component failure to consequence, was identified. The possible occurrence of each event leading up to the hazard must be interpreted in a way that allows it to be evaluated numerically, thereby accounting for physical conditions that affect the outcome of the event. Subsequently, probability and frequency equations must be formulated for each of the components leading to, and influencing the consequences. This also requires the sourcing of empirical data suitable for applications within the scope of the QRA.

Figure 97: Sequence of events leading to flammability hazard
The basic sequence of events leading to an undesirable outcome is: component failure – creation of a flammable mixture – ignition of the mixture – damage. Figure 97 shows the sequence, identifying the possible outcomes of each discrete event. Also indicated are the frequency and probability calculation requirements corresponding to specific groups of events. The first output of interest from a QRA is the frequency of ignition – expressed as the number of occurrence of an event per unit time, for a specific RAC equipment design, within a particular environment. With reference to Figure 97, an ignition event is coincidence of three fundamental events:

- occurrence of a leak,
- development of a flammable-refrigerant/air mixture at a specific location, and
- co-existence of a source of ignition being “active” within the flammable mixture.

These events have their individual probabilities or frequencies, and the frequency of ignition is essentially the product of these. The ignition frequency of a single leak under any one set of conditions \( f_{\text{leak},i} \) is calculated from equation (266), where \( f_{\text{leak},i} \) and \( P_{\text{t,i}} \) are leak frequency and probability of ignition of a flammable mixture by an active SOI, respectively. \( P_{\text{t,i}} \) is evaluated with respect to a vertical element/region \( h \) within the room so that SOI are linked to local formation of a flammable mixture (in light of the relative buoyancy of the gas), and \( f_{\text{leak},i} \) is of a specific leak size in term of duration \( \ell \) for a set of conditions.

\[
f_{\text{t,i}} = f_{\text{leak},i} \cdot \sum_{h=1}^{3} P_{h,i}^{t} \tag{266}
\]

where \( i \) represents each set of operating and environmental conditions and \( h \) refers to low, medium or high region within the room. \( f_{\text{leak},i} \) is largely related to the construction of the system and compressor operating mode (on- or off-cycle), and is normally based on empirical (and anecdotal) data. The size of a flammable mixture (in terms of volume) and its duration is influenced by many parameters, including charge size, leak duration and airflow conditions, arising from infiltration, convection by thermal sources, evaporator and/or condenser fans, and various forms of mechanical ventilation. Thus \( P_{h,i}^{t} \) is primarily dependent on environmental conditions and equipment, and requires modelling of a release to determine relevant characteristics, and physical data for the number, location and active time of local SOI.

Description of the sub-models used to quantify \( f_{\text{leak},i} \) and \( P_{h,i}^{t} \) are given later.
To account for the dynamic nature of equipment operation and/or associated environment, $f^*_i$ was evaluated for each set of conditions. Figure 98 identifies the possible variations for each parameter, and $f^*_i$ was therefore evaluated for the combination that applied, such as compressor operating mode and alternate or multiple airflow types. Consequently, the overall ignition frequency ($f^*$) is the sum of the individual ignition frequencies for all leak sizes, weighted with the corresponding time fraction for each combination of conditions – compressor operating mode ($\phi_{\text{op}}$) and each airflow type – that is present (equation 267).

$$f^* = \sum_{i=1}^{N_{\text{leak}}} \left( \sum_{k=1}^{5} f^*_{i,k} \cdot \phi_{i,k} \right)$$  \hspace{1cm} (267)

with $\phi_1$ for $\phi_{\text{op}}$, $\phi_2$ for $\phi_{\text{unit}}$, $\phi_3$ for $\phi_{\text{therm}}$, $\phi_4$ for $\phi_{\text{ann}}$, and $\phi_5$ for $\phi_{\text{mech}}$. It is noted that for RAC equipment that comprises a number of refrigerant circuits, the total ignition frequency for the equipment is the summed $f^*$ for every circuit.

![Figure 98: Partial event tree illustrating variations of conditions](image-url)
Figure 98 shows that the compressor may be operating or not (op), and for each operating mode, there are a range of different leak sizes. During any individual leak event, the dispersion of the release is affected by air movement due to any combination of unit airflow (unit), mechanical ventilation (mech), thermal sources (therm) and some infiltration rate (infil). A subsequent flammable mixture may be present at high, middle and/or low level within the room, where an active SOI may be present.

Ignition of refrigerant may result in fire and/or explosion, where the subsequent heat and/or pressure wave can cause damage and injury to occupants. For this study, intensity of thermal radiation and overpressure (within the room) were used to evaluate consequence. Where the consequence was of sufficient severity, the individual risk of fatality (fatalities associated to the installation) was estimated. Societal risk is not addressed specifically since for relatively small releases the consequence normally affects only those within the room occupied by the equipment.

Frequency of fatality ($f_{fatal}$) is the sum of individual fatality frequencies resulting from each $f^*_{li}$ (equation 268).

$$f_{fatal} = \sum_{i=1}^{N_{total}} \left( \sum_{j=1}^{N_{leak}} \left( f^*_{lj} \cdot p_{fatal}(f^*_{lj}) \right) \cdot \phi_{occ,i} \cdot N_{occ,i} \right)$$  \hspace{1cm} (268)$$

where $N_{occ,i}$ and $\phi_{occ,i}$ are the number of occupants and their residence time fraction, respectively. $p_{fatal}(f^*_{lj})$ is the probability of fatality due to thermal intensity ($I_{occ}$) and/or room overpressure ($\Delta p_{Rm}$), resulting from ignition under the conditions corresponding to $f^*_{lj}$.

In order to represent the overall risk impact posed by an installation, the use of integrated thermal intensity-frequency (equation 269) and integrated overpressure-frequency (equation 270) were proposed and used. They represent the time- and severity-weighted consequence of ignition.

$$x_{Iocc} = \sum_{i=1}^{N_{total}} \left( \sum_{j=1}^{N_{leak}} f^*_{lj} \cdot I_{occ,i,j} \right)$$  \hspace{1cm} (269)$$

$$x_{Iapo} = \sum_{i=1}^{N_{total}} \left( \sum_{j=1}^{N_{leak}} f^*_{lj} \cdot \Delta p^0_{Rm,i,j} \right)$$  \hspace{1cm} (270)$$

For convenience $x_{Iocc}$ and $x_{Iapo}$ were combined to form an overall dimensionless "risk factor" (equation 271). The formulation of equation (271) was chosen simply to account for the combined damage or injury caused by both consequences occurring simultaneously.
\[ X = \sqrt{(a_1 \cdot \chi_i)^2 + (a_2 \cdot \chi_{spo})^2} \]  

(271)

where constants \( a_1 \) and \( a_2 \) are included to provide a weighting for each type of consequence.

Determination of \( a_1 \) and \( a_2 \) was considered to be largely subjective as different parties may rate one form of injury less significant than another. Here, the values \( a_1 = 0.075 \, \text{s}^{-1} \, (\text{kW}^{-1} \, \text{m}^3)^{1/3} \, \text{y} \) and \( a_2 = 1 \, \text{kPa}^{-1} \, \text{y} \) were chosen because 0.075 represents the relative difference between the acceptable limits of \( \chi_{iocc} \) and \( \chi_{spo} \) as discussed in the next section.

2.2 Context and risk acceptance criteria

It is necessary to put the calculated risks into context if equipment is to be judged as "safe" or "unsafe". This was achieved by establishing a set of limits to compare against the outcome of the QRA, indicating an acceptable or unacceptable risk considering the following measures:

- frequency of ignition,
- frequency of fatality,
- severity of consequence, and
- integrated consequence-frequency, or "risk factor".

Ignition risk was interpreted as fire risk, since ignition may lead to a secondary fire; as discussed in Section 4, addressing the likelihood of a secondary fire from ignition is complex, so frequency of fire is equated directly to ignition frequency. It is useful to compare the risk of RAC equipment against risks from other household and small commercial appliances. Based on statistics of fires in the UK (Collier and Watson, 1998) and product population data (Waterston, 2003), the following frequencies of building fires originating from an appliance, were obtained: refrigerators (non-HC) \( 1.13 \times 10^{5} \, \text{y}^{-1} \), gas cookers \( 8.71 \times 10^{4} \, \text{y}^{-1} \), gas central heating \( 4.03 \times 10^{5} \, \text{y}^{-1} \), washing machines \( 1.31 \times 10^{4} \, \text{y}^{-1} \), televisions \( 2.65 \times 10^{5} \, \text{y}^{-1} \), and other audio/visual \( 8.08 \times 10^{6} \, \text{y}^{-1} \). Causes were electrical faults or a release of flammable gas depending on the type of equipment. These fire risks are evidently considered "acceptable" by society, and therefore may be used as a reference point for acceptability of the ignition risk of flammable refrigerants. The lowest (rounded) value for all the appliances listed in the data was used as the acceptance limit (Table 36), so it is favourable for \( f^* \) to be lower than these appliance fire frequencies.

In terms of severity of a consequence, limits corresponding to those that could cause a fatality were chosen. Alternatively, a lower value based on some degree of injury or physical damage could be adopted. For fatality risk, the Health and Safety Executive (HSE) recommend values for
“negligible” risk to an individual: member of the public $f_{\text{fatal}} = 1 \times 10^{-6} \text{ y}^{-1}$, and service industry worker $f_{\text{fatal}} = 1 \times 10^{-5} \text{ y}^{-1}$ (HSE, 2001). Calculated $f_{\text{fatal}}$ should be lower than that specified by the HSE (Table 36).

Similarly, the acceptance limits for $X_{\text{locx}}, X_{\text{apo}}$ and $X$ were derived according to likelihood of fatality and the tolerances specified by the HSE. $X_{\text{locx}}$ and $X_{\text{apo}}$ are conceptual parameters, and it is not possible to directly adopt empirical values for the acceptance limits. Instead, limiting values of $X_{\text{locx}}$ and $X_{\text{apo}}$ were determined from the lowest integrated consequence-frequency, as derived from the limits for fatality frequencies and the probability of fatality associated with a chosen consequence. From equation (269) and (270), the formula for $X_{\text{locx}}$ and $X_{\text{apo}}$ can be reduced to equation (272) and (273) by assuming one single leak size and a single set of conditions.

$$X_{\text{locx}} = f^* \cdot I_{\text{occ}}$$  \hfill (272)

$$X_{\text{apo}} = f^* \cdot \Delta \rho_{\text{rm}}$$  \hfill (273)

The frequency of fatality for a given consequence is the product of ignition frequency and the probability of fatality associated with a given consequence (equation 274).

$$f_{\text{fatal}} = f^* \cdot P_{\text{fatal}}$$  \hfill (274)

Equation (274) can be transposed for $f^*$, and then substituted into equation (272) and (273), to give equation (275) and (276).

$$X_{\text{locx}} = \frac{f_{\text{fatal}}}{P_{\text{fatal}}} \cdot I_{\text{occ}}$$  \hfill (275)

$$X_{\text{apo}} = \frac{f_{\text{fatal}}}{P_{\text{fatal}}} \cdot \Delta \rho_{\text{rm}}$$  \hfill (276)

$P_{\text{fatal}}$ can be calculated from the severity of a chosen consequence as detailed later in Section 4, which is normally achieved using a logarithmic formula. Replacing $P_{\text{fatal}}$ in equations (275) and (276) with the appropriate formula gives equations (277) and (278).
\begin{align*}
\chi_{\text{occ}} &= \frac{f_{\text{fatal}}}{a + b \cdot \ln I_{\text{occ}}} \cdot I_{\text{occ}} \\
\chi_{\Delta p} &= \frac{f_{\text{fatal}}}{a + b \cdot \ln \Delta p_{\text{rm}}} \cdot \Delta p_{\text{rm}}
\end{align*}

(277) 

(278)

where the constants \(a\) and \(b\) are specific to each consequence.

Using equations (277) and (278), acceptance limits for \(\chi_{\text{occ}}\) and \(\chi_{\Delta p}\) can be determined according to a chosen value of \(f_{\text{fatal}}\), which as previously mentioned was taken from HSE (2001). The severity of the consequence – \(I_{\text{occ}}\) and \(\Delta p_{\text{rm}}\) – necessary to cause fatality varies, and since equations (277) and (278) imply non-linear dependence of \(\chi_{\text{occ}}\) and \(\chi_{\Delta p}\) on the consequence it was necessary to evaluate over an appropriate range of \(I_{\text{occ}}\) and \(\Delta p_{\text{rm}}\). For thermal intensity, the lowest value that can cause fatality is \(I_{\text{occ}} \approx 1050 \text{ s (kW m}^2\text{)}^{4/3}\) and the lowest overpressure is \(\Delta p_{\text{rm}} \approx 250 \text{ kPa}\), and so equations (277) and (278) were evaluated for a series of incrementally larger \(I_{\text{occ}}\) and \(\Delta p_{\text{rm}}\). Subsequently, the smallest value of \(\chi_{\text{occ}}\) and \(\chi_{\Delta p}\) calculated from the range of \(I_{\text{occ}}\) and \(\Delta p_{\text{rm}}\) was selected as the acceptance limit.

Table 36: Maximum acceptable risk criteria

<table>
<thead>
<tr>
<th>Criteria</th>
<th>To members of public</th>
<th>To service industry workers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency of ignition (fire) ((f^*))</td>
<td>&lt; (1 \times 10^{-5}) y(^{-1})</td>
<td>&lt; (1 \times 10^{-5}) y(^{-1})</td>
</tr>
<tr>
<td>Thermal intensity ((I_{\text{occ}}))</td>
<td>(&lt; 1050 \text{ s (kW m}^2\text{)}^{4/3})</td>
<td>(&lt; 1050 \text{ s (kW m}^2\text{)}^{4/3})</td>
</tr>
<tr>
<td>Overpressure ((\Delta p_{\text{rm}}))</td>
<td>(&lt; 250 \text{ kPa})</td>
<td>(&lt; 250 \text{ kPa})</td>
</tr>
<tr>
<td>Frequency of fatality ((f_{\text{fatal}}))</td>
<td>(&lt; 1 \times 10^{-6}) y(^{-1})</td>
<td>(&lt; 1 \times 10^{-5}) y(^{-1})</td>
</tr>
<tr>
<td>Integrated thermal intensity-freq. ((\chi_{\text{occ}}))</td>
<td>(&lt; 5.6 \times 10^{-3} \text{ s (kW m}^2\text{)}^{4/3}) y(^{-1})</td>
<td>(&lt; 5.6 \times 10^{-2} \text{ s (kW m}^2\text{)}^{4/3}) y(^{-1})</td>
</tr>
<tr>
<td>Integrated overpressure-freq. ((\chi_{\Delta p}))</td>
<td>(&lt; 4.2 \times 10^{-4}) kPa y(^{-1})</td>
<td>(&lt; 4.2 \times 10^{-3}) kPa y(^{-1})</td>
</tr>
<tr>
<td>Risk factor ((X))</td>
<td>(&lt; 5.9 \times 10^{-4})</td>
<td>(&lt; 5.9 \times 10^{-3})</td>
</tr>
</tbody>
</table>

Using \(f_{\text{fatal}}\) for “members of the public”, the minimum \(\chi_{\text{occ}}\) was found to be \(5.6 \times 10^{-3} \text{ s (kW m}^2\text{)}^{4/3}\) y\(^{-1}\), which resulted from \(I_{\text{occ}} \approx 3000 \text{ s (kW m}^2\text{)}^{4/3}\), and the minimum \(\chi_{\Delta p}\) was \(4.2 \times 10^{-4}\) kPa.
y\(^{-1}\) corresponded to \(\Delta P_{rm} \approx 330\) kPa. The limits are included in Table 36, and equivalent values for "service industry worker" is also listed. Lastly, for calculation of the overall risk factor \(X\), the value of \(a_1\) in equation (271) is obtained from the ratio \(X_{esp} / X_{loc} = 0.075\). Evaluating equation (271) when \(a_1 = 0.075\) s\(^{-1}\) (kW m\(^{-1}\))\(^{y}\) and \(a_2 = 1\) kPa\(^{-1}\) y, the acceptance limit of \(X\) for both activities was calculated and listed in Table 36.

2.3 Leakage frequency

Reviewed QRAs employ leak frequencies which vary by size of leak, activity and equipment type. A compilation of leak frequencies from the literature (Table 56, Appendix 1) show values extending from \(3 \times 10^9\) y\(^{-1}\) to \(2 \times 10^1\) y\(^{-1}\) for normal operation. Some studies adopt a single frequency to represent all leaks, whereas others differentiate between small and large leak sizes. Relying on these reported values for the QRA model was not deemed appropriate due to the inconsistency between equipment types, but more so, because most were simply based on interviews with service technicians and manufacturers. Further, considering the strong dependency of risk upon leak characteristics (as detailed later), it is important to account for a variety of releases anticipated for a given equipment construction. This requirement led to the development of a new leak frequency model.

The model employed leak data ranging from the smallest leak mass flow rates to largest ("catastrophic") leak, and the data suited to the equipment under consideration in terms of number and types of connections. Statistical data on leakage is currently scarce; however, permanent refrigerant monitoring is employed on systems for leak detection. This type of data was considered a useful source for characterising leak frequency. A paper by Radford (1998) provides detailed information on such measurements, upon which the frequency model was derived.

There are two possible events for a system that contains refrigerant: the system leaks \((E_{\text{leak}})\), or it doesn't. The probability of \(E_{\text{leak}}\) is the ratio of leak events, to the total events within the set \((S)\), which comprises leak events and non-leak events (equation 279).

\[
P(E_{\text{leak}}) = \frac{N(E_{\text{leak}})}{N(S)}
\]

(279)

The number of events \((N)\) can interpreted either on a time basis or a leak initiation basis and it is important to differentiate between these two definitions. A time-based probability can be related
directly to mass flow rate of a leak and therefore this approach was employed for dispersion calculations (whereas an initiation-based leak frequency cannot). Equation (279) is presented in terms of the ratio of time that a system leaks to total time (equation 280).

\[ P_{\text{leak}} = \frac{t_{\text{leak}}}{t_{\text{tot}}} \]  
(280)

If the total time \((t_{\text{tot}})\) is set at one year and the duration of the leak \((t_{\text{leak}})\) is within the same period, the leak probability is a frequency \((f_{\text{leak}})\). When equation (280) is applied for a range of leak sizes (assumed to correspond to a selected range of measured concentrations at specific locations, \(\bar{C}\)), \(f_{\text{leak}}\) of a given leak size per connection (or joint) can be obtained from equations (281) and (282).

\[ t_{\text{leak}}(\bar{C}) = \frac{N_{E,\text{leak}}(\bar{C}) \cdot \Delta t_E}{N_{\text{conn}}} \]  
(281)

\[ f_{\text{leak}}(\bar{C}) = \frac{N_{E,\text{leak}}(\bar{C}) \cdot \Delta t_E}{N_{\text{conn}} \cdot t'} \]  
(282)

where \(N_{\text{conn}}\) is the total number of the connections, \(N_{E,\text{leak}}(\bar{C})\) is the number of leak events that result in a measured concentration range per year, and \(\Delta t_E\) is the event interval time. A further consideration is the shortest time of a catastrophic leak where the rate of refrigerant release is largely independent of leak aperture; the flow rate is dictated by internal characteristics of the system. Thus, equation (281) does not necessarily apply in some cases and instead \(t_{\text{leak,catastrophic}}\) is determined and fixed based on experiment. Further, \(f_{\text{leak}}\) differs according to the compressor operating mode, so separate values for on-cycle and off-cycle conditions are required.

As discussed earlier, a large number of leak sizes increments \((N_{\text{leak}})\) need to be evaluated from smallest to catastrophic leaks. In order to obtain a smooth variation of leak data over the range of chosen leak sizes, the empirical \(f_{\text{leak}}\) and \(t_{\text{leak}}\) was correlated in the form of polynomial equation (283).

\[ f_{\text{leak},i} = a_1 \cdot t_{\text{leak},i}^2 + a_2 \cdot t_{\text{leak},i} + a_3 \]  
(283)

In determining \(a_1\), \(a_2\) and \(a_3\) in equation (283) direct polynomial fitting cannot be used due to the fact that the number of incremental leak sizes assigned \((N_{\text{leak}})\) may be different to the number of leak sizes in the empirical data. In order to evaluate the frequency at any other number of leak
sizes, the coefficients need to be determined specifically for the selected number of leak size increments. The procedure adopted is detailed in Figure 99. Empirical data for leak frequencies/times (presently, for small, medium and catastrophic sizes) was adjusted by using an assumed coefficient $b$ (in equation 284), giving: $f_{\text{leak},j} = b \cdot f_{\text{meas},j}$ and $t_{\text{leak},j} = b \cdot (f_{\text{meas},j} \cdot t')$, except with a catastrophic leak, when $t_{\text{leak},j} = t_{\text{leak},\text{cat}}$. Once the data is fitted ($a_1$, $a_2$ and $a_3$ found), it must be checked to ensure the total leak time of all leak sizes is equal to the empirical leak time, using equation (284). Equation (284) is evaluated for the range of $t_{\text{leak}}$, and if not satisfied, $b$ is adjusted, and the process reiterated.

$$\sum_{i=1}^{N_{\text{leak}}} t_{\text{leak},i} = [t_{\text{leak},\text{cat}} + b \cdot t'] \sum_{i=1}^{N_{\text{leak}}-1} f_{\text{leak},i} \text{ meas} \quad \text{(284)}$$

Note that since $P_{b,j}^T$ is a function of $t_{\text{leak}}$, therefore comparison of $f_{\text{leak}}^*$ for different operating modes would be valid only if identical incremental leak rates are used.

Choose inputs: $N_{\text{leak}}$, $N_{\text{com}}$, $t_{\text{leak},\text{cat}}$, initial guess for $b(N_{\text{leak}}) = 3/N_{\text{leak}}$

Calculate small, medium and catastrophic leak frequencies using empirical leak frequencies and $N_{\text{com}}$, e.g., for small leak size: $f_{\text{leak}}^* (\text{sml}) = f_{\text{leak}} (\text{sml}, N_{\text{com}} = 1) \cdot N_{\text{com}}$

Similarly, calculate small, medium and catastrophic semi-empirical leak times, e.g., for small leak size: $t_{\text{leak}}^* (\text{sml}) = t_{\text{leak}} (\text{sml}) \cdot t'$

Calculate, total semi-empirical: $f_{\text{leak}}^* (\text{tot}) = f_{\text{leak}}^* (\text{sml}) + f_{\text{leak}}^* (\text{med}) + f_{\text{leak}}^* (\text{cat})$

Adjust semi-empirical leak frequencies and times according to the selected number of leak sizes ($N_{\text{leak}}$), e.g., for small leak size: $f_{\text{leak}}^* (\text{sml}) = b(N_{\text{leak}}) \cdot f_{\text{leak}}^* (\text{sml}), t_{\text{leak}}^* (\text{sml}) = f_{\text{leak}}^* (\text{sml}) \cdot t'$, although catastrophic leak time is fixed, i.e., $t_{\text{leak}}^* (\text{cat}) = t_{\text{leak},\text{cat}}$

Determine constants, $a_1$, $a_2$, and $a_3$ for quadratic equation using the adjusted semi-empirical $f_{\text{leak}}^*$ and $t_{\text{leak}}^*$ for small, medium and catastrophic leaks.

Choose incremental leak times for QRA, $t_{\text{leak},j}$, and use quadratic equation to calculate corresponding incremental leak frequencies, $f_{\text{leak},j}$, and use to calculate the sum of incremental leak frequencies: $f_{\text{leak}}^* (\text{tot}) = \sum_{j=1}^{N_{\text{leak}}} f_{\text{leak},j}$

Does $f_{\text{leak}}^* (\text{tot}) = 1$?

Yes

Adjust number leak of sizes coefficient: $b(N_{\text{leak}}) = [b(N_{\text{leak}}) \cdot (f_{\text{leak}}^* (\text{tot}))^{1/2}]$

No

$Figure 99$: Calculation sequence to determine adjustment for selected number of leak sizes
The leak frequencies used in the model are subject to a final adjustment to account for the locality of leaks. To address leaks that only occur from a portion of the system within a specific location – say indoors – the values obtained from equation (283) are then tailored as equation (285).

\[ f_{\text{leak}} = f_{\text{leak}} \cdot P_{\text{local}} \]  

(285)

where \( P_{\text{local}} \) is the probability of the leak occurring locally. Considering an occupied room, \( P_{\text{local}} \) would be unity for integral equipment and <1 for split-type equipment.

For application of the model (equations 281 to 284) it is necessary to obtain empirical values. Data on refrigerant leakage was made available (Ayers, 2000), which had been gathered from monitoring 410 supermarket installations throughout the UK. Refrigerant concentrations were measured at 30-minute intervals, and quantities leaked and source locations recorded. Out of the 108 million samples, 1,533,239 leak events were detected and measured; 1,435,473 were regarded as ‘small’ leaks, 86,882 were ‘medium’ and 10,884 were ‘catastrophic’ (categorised according to measured concentrations). Using equation (282), and taking the average number of connections per supermarket system as 700 (excluding those on pre-fabricated components such as coils), correspondingly leak frequencies per connection are: small leaks, \( f_{\text{leak}} (N_{\text{conn}} = 1) = 2.85 \times 10^{-6} \text{ y}^{-1} \); medium leaks, \( f_{\text{leak}} (N_{\text{conn}} = 1) = 1.73 \times 10^{-5} \text{ y}^{-1} \); and catastrophic leaks, \( f_{\text{leak}} (N_{\text{conn}} = 1) = 2.16 \times 10^{-6} \text{ y}^{-1} \). To apply these derived frequencies to a system of specific design, they are multiplied by the number of connections (\( f_{\text{leak}} = f_{\text{leak}} (N_{\text{conn}} = 1) \cdot N_{\text{conn}} \)).

It is believed that adoption of these data addressed some of the concerns highlighted for existing frequencies: it is based on measurements, and given the large sample set it accounts for a wide range of installation- and service-quality and a large number of different types of system components and connections. It is noted that further work would enable adjustment of \( f_{\text{leak}} \) according to the vulnerability of different components to leak.\(^{19}\)

These data, particularly for catastrophic leaks, apply only to systems that have their compressor operating (on-cycle), since none of these installations completely shut down. A system in off-cycle has a catastrophic \( f_{\text{leak}} \) less than on-cycle because many mechanisms that cause leakage are not present, such as vibration, pressure and temperature changes. The only situation found to

\(^{19}\) The approach in Section 2.3.1 of Part 1 could be extended for weighting leak frequencies according to the likelihood of leakage from different components, for example, flared versus brazed joints.
have the potential for failure during off-cycle was a combination of fatigue and stress corrosion cracking under severe environmental conditions (Guyoncourt and Fennell, 2000), leading to an estimated catastrophic leak frequency of $1 \times 10^8 \text{ y}^{-1}$ for a split air-conditioner, or $f_{\text{leak}}(N_{\text{conn}} = 1) = 1 \times 10^{10} \text{ y}^{-1}$. The former is comparable to the historical figure of $f_{\text{leak}} = 3.0 \times 10^8 \text{ y}^{-1}$ recommended for LPG tanks (Roberts, 1980), which was considered analogous. The above supermarket measured frequencies for small and medium leaks were assumed the same for both on and off-cycles since leak duration is typically longer than typical cycling times. In general, off-cycle $f_{\text{leak}}$ appears to be about $10^3$ lower than for on-cycle. Referring to equation (285), analysis of the sources of leaks from Radford (1998) and Ayers (2000) show that 28% of the leaks occurred from components normally located indoor (e.g., expansion device, etc), so $P_{\text{leak}} = 0.28$ to account for indoor leaks from "split" equipment, otherwise for integral equipment, $P_{\text{leak}} = 1$.

Alternatively, Goetzler et al (1998) assumes a 50% split between indoor and outdoor parts, i.e., $P_{\text{leak}} = 0.5$.²⁰

Servicing $f_{\text{leak}}$ is greater than for normal operation since the technician could be intentionally breaking in to the system. Goetzler et al (1998) recommended catastrophic $f_{\text{leak}} = 1.0 \times 10^{-3} \text{ y}^{-1}$ for equipment normally having 20 to 30 connections (Table 56), thus $f_{\text{leak}}(N_{\text{conn}} = 1) = 4.0 \times 10^{-5} \text{ y}^{-1}$. This frequency is about 20 times greater than that of normal on-cycle operation. Alternatively, comparing the data in Table 56 shows that servicing $f_{\text{leak}}$ is typically between 1 and 3 times that of total $f_{\text{leak}}$ for normal operation. Thus, small and medium $f_{\text{leak}}$ were assumed proportionally higher compared to normal operation.

### 2.4 Probability of ignition

In the review of QRAs, probability of an active SOI surrounded by a mixture at a flammable concentration ($C_F$) normally involved two fixed values: one for immediate ignition and one for delayed ignition, e.g., $8 \times 10^{-3}$ and $2 \times 10^{-3}$ respectively (Blom-Bruggeman et al, 1996a). This is typical of industrial risk assessment, where single generic values are used because the exact SOI characteristics for large areas of land are unknown. The probability of immediate ignition is applied to ignition during the release, which typically results in a less severe consequence than delayed ignition. The exception is Goetzler et al (1998) who employ a constant probability specific to each location of system parts, between $1 \times 10^{-4}$ and $2 \times 10^{-2}$. Woodward (1998) detailed

---

²⁰ Again, the approach in Section 2.3.1 of Part 1 could be employed to determine this probability more precisely.
the model used in a HSE code, which is for a flammable cloud passing over urban, rural or industrial land (equation 286).

\[ P^{F*} = 1 - (1 - P^F)^{A^* A^F} \]  

(286)

where \( P^F \) is the probability of having a local \( C^F \), \( A \) is the total area covered by the cloud at some point in time, and \( A^F \) is the total area covered by the portion of the cloud at \( C^F \), until it disperses to below LFL. This approach is not applicable for releases into rooms; considering the case of a flammable cloud that entirely covers the room floor, \( P^{F*} = P^F = 1 \). For a cloud of varying volume and duration this is counter-intuitive, and further, it cannot account for different populations and characteristics of SOI. A more suitable approach was offered by Rew et al (2000), being location and time specific. \( P^{F*} \) (at a given time) is predicted on the assumption that the likelihood of active SOI increases over time consistent with expansion of the cloud (equation 287).

\[ P^{F*}(t) = 1 - \left( 1 - \frac{P^F_{soi} \cdot t_{soi}}{\Delta t + t_{soi}} \right) \cdot \exp \left( \frac{-P^F_{soi} \cdot t_{r0}}{\Delta t + t_{soi}} \right) \]  

(287)

where \( P^F_{soi} \) is the probability of the presence of an active SOI within a potentially flammable cloud, \( t_{r0} \) is the time since initiation of the release, \( t_{soi} \) is the active time of the SOI, and \( \Delta t \) is the average time interval between active SOI events. Rew et al give empirically derived values for \( P^F_{soi}, \ t_{soi} \), and \( \Delta t \) according to different locations and time of day. Whilst this approach better account for the local situation compared to equation (286) and values are provided for selected indoor locations, it still lacks the ability to account for SOI specific to a room, and it is vague in how it handles the duration of the flammable volume.

To address the shortcomings of the methods currently in use (with respect to the present purpose), a new formula was developed where the characteristics of each individual SOI and each release scenario may be accounted for. For a given space-time, ignition could potentially occur when an active SOI is present at the same time and spatial position as \( C^F \). To illustrate the quantification of \( P^{F*} \), the Venn diagram in Figure 100 shows the existence of a potential SOI event, the active SOI event and the \( C^F \) event. The coincidence of these leads to ignition, as indicated by the shaded area. To determine \( P^{F*} \), it is evident that two component probabilities are needed: probability of an active SOI at the same time as \( C^F \), and probability of \( C^F \) at the same spatial position as the potential SOI.
Occurrence of SOI events

Within most domestic and commercial occupied spaces, an active SOI would be an electrical spark of at least 20 mJ and a flame or surface at a temperature of at least 460°C. SOI may be present on the RAC equipment or elsewhere within an occupied room, a storage room, and tools used for servicing activities. For a single active SOI ($E_{sol}$) assumed to occur for some duration randomly within a reference period ($S$), the probability can be given by equation (288).\(^{21}\)

$$P(E_{sol}) = \frac{N(E_{sol})}{N(S)} \quad (288)$$

Similarly, if $C^F$ occurs randomly over a given duration ($E^F$) within the same reference period ($S$), the probability of $C^F$ being present is equation (289).

$$P(E^F) = \frac{N(E^F)}{N(S)} \quad (289)$$

As $E_{sol}$ and $E^F$ are independent, the probability occurring simultaneously is the sum of both probabilities (equation 290).

$$P(E_{sol}^F) = \frac{N(E_{sol})}{N(S)} + \frac{N(E^F)}{N(S)} \quad (290)$$

\(^{21}\) If knowledge of the temporal behaviour of the SOI and/or flammable concentrations is known, alternative methods can be applied to obtain a more precise representation of probability.
For each potential SOI, the number of events $N(E_{soli})$ is represented by the active time of a single event within the reference period, and $N(E_{soli})$ is the time in which $C^F$ is present. It is assumed that this flammable time ($t^F$) is continuous, i.e., the leak produces constant flow until the refrigerant charge is depleted. Therefore, equation (290) can be written as equation (291), showing the probability of an active SOI within a flammable volume within a given reference volume under a given set of conditions resulting from a certain leak size.

$$P_{soli,h,l,i}^F = P_{avail} \left( \frac{t_{soli} + t_{i,h,l,i}^F}{t'} \right)$$  \hspace{1cm} (291)

where $P_{soli,h,l,i}^F \leq 1$, and $t'$ is the reference period. $P_{avail}$ is included to account for the availability of the SOI. For example, electrical components on equipment may be protected against the ingress of refrigerant ($P_{avail} = 0$), although for various reasons the integrity of the protection may fail ($0 < P_{avail} < 1$). Otherwise, for permanent SOI, $P_{avail} = 1$. In general, active SOI events arising from faulty electrics are neglected, and it is assumed that any active SOI is immediately available to $C^F$, i.e., no time lag occurs as gas migrates through housings, etc. Also, an external fire is not considered as a SOI because ignition of refrigerant would make a negligible contribution to the existing consequence (Ritter and Colbourne, 1998).

Availability of any SOI is largely dependent upon its purpose. Electrical components associated with RAC equipment should be protected according to safety standards, but a recent survey of electrical components in hazardous areas of petrochemical plants (Murdoch, 2003) revealed that 14% of protective enclosures were ineffective due to degradation and poor repair, thus $P_{avail} \approx 0.14$. In addition, for servicing activities, Goetzler et al. (1998) state that 15% of visits require brazing activities ($P_{avail} = 0.15$) and 40% of service technicians may smoke ($P_{avail} = 0.4$).

Further, servicing activities may sometime be carried out by technicians who may not be suitably trained in handling flammable refrigerants. This implies a difference in use of tools that have SOI, therefore $P_{avail}$ should be altered accordingly.

It is also noted that leakage data from Ayers (2000) showed that once initiated, leaks rarely produce constant release rates, particularly with small leaks. Variations in operating and environmental conditions – typically cyclic temperature and pressure changes – terminate and resume leaks owing to contraction and expansion of joints, etc. As such, $t^F$ is likely to be intermittent and therefore $P_{soli}^F$ would realistically be smaller than equation (291).
Occurrence of flammable volume events

The probability of having a flammable volume \( P_{F} \) is based on the coincidence of the active SOI being present in the same spatial location as \( C_{F} \). As with \( P_{SOI} \), most of the literature does not define \( P_{F} \) specifically, but use a probability based on the likelihood of \( C_{F} \) occurring following a release. Again, this approach is not sufficiently precise since any release will always produce \( C_{F} \) at some location within the space; it would only be relevant if linked directly to a SOI at a given point in space and time. Instead, the approach adopted assumed the SOI are located randomly within the same reference volume that \( C_{F} \) may exist. To obtain \( P_{F} \), it is necessary to determine the total volume of the flammable mixture \( V_{F} \) following a release (as quantified by the dispersion model in Section 3.3).

The probability model assumes two possible outcomes: a \( C_{F} \) occurs within a given location \( E_{F} \) representing a flammable volume, or it is free from a \( C_{F} \), whether it comprises a mixture outside the LFL – UFL or is absent of refrigerant entirely. The probability of \( E_{F} \) occurring is the ratio of the number of flammable volume events to the total possible events for the flammable volume to occupy \( S \), where the number of events is interpreted as total volumetric elements (equation 292).

\[
P(E_{F}) = \frac{N(E_{F})}{N(S)}
\]  

If the reference volume is discretised the number of events \( N(E_{F}) \) is the number of elements at \( C_{F} \), and \( N(S) \) is the total elements within the reference volume. The probability of the flammable volume is therefore (equation 293).

\[
P_{F}^{i,j} = \frac{V_{F}^{i,j}}{V_{h}^{i}}
\]  

where the reference volume \( V_{h} \) corresponds to the horizontal levels chosen for grouping local SOI \( V_{h} = A_{Rm} \cdot h_{Rm} / N_{h} \), and \( V_{F}^{i,j} \) is the flammable volume existing within that reference volume. For the present study, three reference volumes \( N_{h} = 3 \) were chosen because it is convenient to identify SOI normally located at low, middle of high levels. Following the initiation of a release, \( V_{F}^{i,j} \) changes over time, which would demand an integral approach to determine
Outputs from the dispersion model (Section 3.3) show $V^F$ to increase somewhat linearly until the end of the release, and is thereon approximately constant within the decaying cloud and eventually falls to zero in a relatively short duration. This trend in behaviour of $V^F$ was found in most cases, and therefore it was considered appropriate to maintain the basic $P^F_i$ formulation (equation 293) by employing an averaged flammable volume with respect to flammable time ($\bar{V}^F$).

Two additional parameters must be accounted for, the proportion of systems that leak their charge and the likelihood that an active SOI successfully ignite $C^F$. Various sources report system leakage data in the form of “annual leak rate”, which is defined as the ratio of total mass leaked from a given population of systems to the total refrigerant mass within that population, or “refrigerant bank”. This ratio is interpreted as the probability of a system to release its charge ($P_{sys}$). Failure to ignite a $C^F$ is referred to as “percolation” (Rew7 and Spencer, 1998), and is explained by the presence of small pockets of unmixed gas or air within a cloud. Including $P_{sys}$ and percolation probability ($P_{perc}$) the probability of flammable volume becomes (equation 294).

$$P^F_{i,h,l,j} = \left( \frac{\bar{V}^F_{h,l,j}}{A_{gm} \cdot h_h} \right) \cdot P_{sys} \cdot P_{perc} \tag{294}$$

$P_{sys}$ is normally determined for each equipment category, e.g., small integral commercial refrigeration units, or air-conditioning chillers. $P_{perc}$ was estimated from Jabbour et al (2002) where 48 tests were conducted to try to ignite R290 and R600a whilst at $C^F$ in a small room. Despite measurements showing $C^F$ at the SOI position, 27% of the attempts did not result in ignition, thus $P_{perc} = 0.73$.

Whilst $\bar{V}^F$ is partly dependent on refrigerant charge, the actual final leaked amount may be less than the total charge because a proportion is retained within system. Therefore, the mass used to calculate various flammable quantities is the system charge reduced by the appropriate proportion ($\psi$). Measurements of catastrophic R290 leaks (PR 100374, 2000) showed the mass released was less than the charged amount, and was due to vapour residing in the system and

---

22 Evaluation of this is addressed in detail within Section 2.3.1 of Part 1, which can be employed to determine $P_{sys}$ according to system design.
being absorbed in compressor oil. After a 3.5 minute catastrophic leak, 30% of the charge was retained under static conditions or off-cycle, and on average, 18% for on-cycle; so \( \psi = 0.30 \) and \( \psi' = 0.18 \) respectively. Additionally, Kataoka (1999) made similar tests by measuring the mass of R22 removed from an air conditioner using a large recovery machine. The system contained 1.0 kg of R22 and 0.4 kg of mineral oil. A catastrophic leak was simulated after cessation of the compressor, the system was found to retain 5% of refrigerant. Conducting the same test but with leaving the refrigerant to soak for increasing longer time before the leak was simulated, showed the retained mass proportion increased linearly, to 15% after the system had been static for 20 hours. The differences in values between PR 100374 (2000) and Kataoka (1999) may be attributed to system internal volumes, and different solubility of R290 and R22 in mineral oil.

**Occurrence of SOI and flammable concentration**

Lastly, it is necessary to combine the SOI and \( V^F \) probabilities to determine the ignition probability indicated in Figure 100. To simply the derivation, it is convenient to employ the compliment of the preceding events: probability of no flammable volume (\( P^F = 1 - P^F_v \)), and the probability of the SOI not being active within the flammable volume (\( P^F_{so} = P^F \cdot (1 - P^F_{so}) \)). The probability of no flammable volume and the SOI not being active within the flammable volume is the sum of \( P^F \) and \( P^F_{so} \) (equation 295).

\[
P^F_{h,i,j} = (1 - P^F_{V,h,i,j}) + P^F_{V,h,i,j} \cdot (1 - P^F_{so,h,i,j})
\]

(295)

where \( P^F_{V,h,i,j} \) and \( P^F_{so,h,i,j} \) are from equations (294) and (291), respectively. Conversely, the probability of having a SOI being active only within the flammable volume is the complement of \( P^F_{h,i,j} \) (equation 296).

\[
P^F_{h,i,j} = 1 - \left[ (1 - P^F_{V,h,i,j}) + P^F_{V,h,i,j} \cdot (1 - P^F_{so,h,i,j}) \right]
\]

(296)

It is expected that most SOI would produce several active events within the reference period, \( t' \).

It is held that any \( P^F_{h,i,j} \) event is independent of the others since any active SOI event is random and therefore occurs irrespective of any other SOI event. Hence, the total probability of all active events for a single SOI is the complement of the product of all non-ignition events associated with the given SOI (equation 297).

\[
P^F_{h,i,j} = 1 - \left[ P^F_{h,i,j}(E_{soi,1}) \cdot P^F_{h,i,j}(E_{soi,2}) \cdot \ldots \cdot P^F_{h,i,j}(E_{soi,N}) \right]
\]

(297)
Substituting equation (295) into (297) gives equation (298) for the total number of active events $N_e$ for each individual SOI.

$$P_{h,l,i}^{F} = 1 - \left[ (1 - P_{v,h,l,i}^{F}) + P_{v,h,l,i}^{F} \cdot (1 - P_{soi,h,l,i}) \right]^{N_e}$$ (298)

Within any control volume, there may be several different SOI, of which some may be identical to each other. The total probability of all SOI is the sum of the individual probabilities for each active SOI (equation 299).

$$P_{h,l,i}^{F} = \sum_{N=1}^{N_{soi}} P_{h,l,i}^{F} (N_{soi})$$ (299)

This probability is then used in equation (266).

### 2.5 Time fractions

Time fractions represent the proportion of time that the occupancy, operating mode and airflow types occur as the ratio of residence time to some reference time (equation 300).

$$\phi_i = \frac{t_{res,i}}{t'}$$ (300)

where $t_{res}$ is the annual operating or residence time for the parameter under consideration at condition $i$. For compressor operation ($\phi_{op}$), occupancy ($\phi_{occ}$), thermal sources ($\phi_{therm}$), forced airflow from the RAC unit ($\phi_{unit}$) and mechanical ventilation ($\phi_{mech}$), $t_{res}$ is based on the operating or residence times, $t_{op}$, $t_{occ}$, $t_{therm}$, $t_{unit}$ and $t_{mech}$, respectively.

Unit airflow or mechanical ventilation may be initiated by a controller that responds to a refrigerant detector or pressure transducer. Whilst these control methods are operating as intended, the residence time is effectively the reference time, thus $t_{res} = t'$. As far as $V^F$ and $t^F$ are concerned, these airflow are always present, whether they are operating constantly or started part-way through a release and thus for a given set of conditions, $\phi_{unit,i}$ or $\phi_{mech,i} = 1$. However, such unit airflow and mechanical ventilation are subject to unexpected failure due to mechanical faults or incapability to detect refrigerant, and this was accounted for by the failure probability ($P_{fail}$) and subsequent down-time ($t_{fail}$), i.e., the time taken for completion of the repair and the airflow resumed. $P_{fail}$ is based on the number of failures of components leading to cessation of airflow, out of a total population within the reference time. Thus within $t'$, the average time that
the airflow is absent is $P_{\text{fail}} \times t_{\text{fail}}$, and the total time that the airflow is present is therefore equation (301).

$$t_{\text{res,0}} = t_{\text{res(0),r}} - P_{\text{fail,r}} \cdot t_{\text{fail,r}} \quad (301)$$

where $t_{\text{res(0)}}$ is the intended operating or residence time for the airflow. In general, the cause of airflow failure may include duct blockage, fan motor burn-out, faulty positioning or degradation of refrigerant detectors, or malfunction of electrical circuits or controls. Evidently $P_{\text{fail}}$ can vary widely depending upon the type of equipment, manufacturer, quality of installation, etc. Based on a limited number of sources, example data is provided in Table 37. $t_{\text{fail}}$ depends upon the degree of inconvenience (if at all) to those associated with the equipment. For example, failure of airflow from an office air conditioner during hot weather may result in a short downtime (of days), whereas a faulty refrigerant sensor may only be corrected at the next inspection (possibly months later).

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
<th>Probability</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blockage of flues, ducts, etc.</td>
<td>13% per annum of gas appliance flues – probability of a blocked RAC unit duct assumed about half.</td>
<td>$P_{\text{fail}} = 0.070$</td>
<td>Davies and Pearson (1999)</td>
</tr>
<tr>
<td>Fan and fan motor failure</td>
<td>Air conditioning manufacturer reported 0.2% per year.</td>
<td>$P_{\text{fail}} = 0.002$</td>
<td>Aloisi (2000)</td>
</tr>
<tr>
<td>Failure of some part of electrical/control circuits.</td>
<td>Air conditioning manufacturer reported 5% per year.</td>
<td>$P_{\text{fail}} = 0.050$</td>
<td>Aloisi (2000)</td>
</tr>
<tr>
<td>Failure of detection equipment to initiate airflow, alarm, etc.</td>
<td>Statistics on failure of smoke detectors (except where batteries had been removed) show 13.8% failed to detect for various reasons.</td>
<td>$P_{\text{fail}} = 0.138$</td>
<td>Collier and Watson (1998)</td>
</tr>
</tbody>
</table>

Another implication of unit airflow is its functional link to the compressor operation. For example, if the evaporator or condenser airflow fails, low or high pressures can be generated, which may cause the compressor to cut out. To account for this, calculation of $\phi_{\text{op}}$ employed equation (301) to calculate $t_{\text{res}}$, where $t_{\text{res(0)}}$ is the intended compressor operating time, and $P_{\text{fail}}$ and $t_{\text{fail}}$ are those for unit airflow.
Infiltration rates vary over time, so evaluation of $\phi_{\text{infil}}$ requires attention. For a room, infiltration rates ($\Lambda_{\text{infil}}$) were calculated from the fabric leakage ($\Lambda_{0}$) at the standard reference 50 Pa air pressure differential ($\Delta p$). As local weather conditions change over the year, different $\Delta p$ across the room fabric occur, leading to a range of $\Lambda_{\text{infil}}$ and an associated range of $\phi_{\text{infil}}$. To calculate $\phi_{\text{infil}}$, the corresponding $t_{\text{res}}$ must first be found and this is based on the frequency of weather conditions leading to each $\Delta p$. The duration of the different weather conditions resulting in a specific $\Delta p$ is the sum of the wind speed and outside temperature time increments that lead to that $\Delta p$. An $\Lambda_{\text{infil}}$ is calculated for each $\Delta p$, and the range of these (between minimum and maximum $\Lambda_{\text{infil}}$) is divided equally into a number of sub-ranges ($\Delta\Lambda_{\text{infil}}$). Thus $\phi_{\text{infil}}$ for each incremental $\Delta\Lambda_{\text{infil}}$, is based on the time that weather conditions that lead to each $\Delta\Lambda_{\text{infil}}$ are present, $t_{\text{res}}(\Lambda_{\text{infil}})$, (equation 302). \(^2\)

$$\phi_{\text{infil}} = \frac{t_{\text{res},i}(\Lambda_{\text{infil},i})}{t}$$ (302)

For a given room, $\Lambda_{\text{infil}}$ was calculated using the method detailed in BS 5925: 1991. First, the total pressure difference between the internal and external wall is required ($\Delta p_{\text{tot}}$), being the sum of the stack pressure difference ($\Delta p_{\text{stack}}$) produced by the differential air density between inside and outside, and the pressure imposed by external wind ($\Delta p_{\text{wind}}$) (equation 303).

$$\Delta p_{\text{tot},i} = \Delta p_{\text{stack},i} + \Delta p_{\text{wind},i}$$ (303)

$\Delta p_{\text{stack}}$ was calculated using equation (304), and $\Delta p_{\text{wind}}$ with equation (305).

$$\Delta p_{\text{stack},i} = g \cdot \frac{h_{\text{RM}}}{2} \cdot (\rho_{\text{out},i} - \rho_{\text{in},i})$$ (304)

$$\Delta p_{\text{wind},i} = \Delta c_{\text{wind}} \cdot \frac{\rho_{\text{out},i} \cdot u_{\text{wind},i}^2}{2}$$ (305)

where $h_{\text{RM}}$ is room height, $u_{\text{wind}}$ is wind speed, $\rho_{\text{out}}$ and $\rho_{\text{in}}$ are air densities and the wind pressure coefficient $\Delta c_{\text{wind}}$ was fixed at 0.75 for convenience.

\(^2\) This approach is similar to the "bin" method used for ambient temperatures.
Both outside air density (due to temperature) and wind speed vary independently throughout the year. Knowing this, the year is discretised and $\Delta p_{\text{tot}}$ is calculated for each range of wind speeds for each monthly temperature. Mean monthly temperatures for one UK location, and corresponding frequencies are provided in Table 38. Data in BS 5925 is given for the cumulative proportion of time that the wind speed differs from the average. A polynomial was used to fit the data for inland areas of the UK, which was then differentiated, so that equation (306) could be employed to quickly estimate the wind speed for a given range of time fractions.

$$
\phi(u') = \int_{u'_1}^{u'_2} a_1 + a_2 \cdot u' + a_3 \cdot u'^2
$$

(306)

where $\phi$ is the time fraction, and $a_1 = -0.3361$, $a_2 = -0.5308$, and $a_3 = 0.3156$. $u'$ is the wind speed relative to the time-averaged ($\phi = 0.5$) wind speed (equation 307).

$$
u' = \frac{\bar{u}_{\text{wind}}(\phi)}{\bar{u}_{\text{wind}}}
$$

(307)

where $\bar{u}_{\text{wind}}$ is the mean ($\phi = 0.5$) annual wind speed (1.8 m s$^{-1}$ for West London).

![Figure 101: Time fraction for different infiltration rate ranges](image-url)
Selecting a fabric leakage (in terms of $\Lambda_{50}$), equation (382) was used to estimate $\Lambda_{\text{infl}}$ for the range of calculated $\Delta p_{\text{tot}}$. Based on a range of calculated $\Lambda_{\text{infl}}$ from minimum to maximum values, a set of equally sized $\Delta\Lambda_{\text{infl}}$ elements were determined. The time fraction of each $\Delta\Lambda_{\text{infl}}$ was found by calculating the total time ($t_{\text{rez.}}(\Lambda_{\text{infl}},z)$) that a given outside temperature and wind speed were present, that leads to each $\Delta\Lambda_{\text{infl}}$ (equation 302). As an example, a $\Lambda_{50}$ of 14 h$^{-1}$ was selected and for ten increments, $\Delta\Lambda_{\text{infl}}$ was calculated for a 40 m$^2 \times 3$ m high room. The results are presented in Figure 101, showing the lower $\Delta\Lambda_{\text{infl}}$ (0.4 h$^{-1}$ to 1.6 h$^{-1}$) occur most often.
SECTION 3: DISPERSION OF REFRIGERANT LEAKS

3.1 Dispersion experiments
The probability calculations in Section 2.4 require the dispersion of a release to be both spatially and temporally characterised. For the range of conditions relevant to this study, the data available from existing literature was insufficient in scope and of minimal practical use. To remedy this, a number of experiments were conducted to obtain data suitable for the development of a dispersion model based on a purpose-built test chamber, which included some initial smoke visualisation tests. This section details experimental work, qualitative observations and formulation of empirical correlations for application to the dispersion model.

3.1.1 Experiment rationale
A rigorous assessment of the physical variables likely to be encountered in typical RAC applications was made to develop a test programme. The relevant characteristics associated with various RAC equipment was interpreted in generic terms so that the results could be extended to other equipment. To minimise the number of tests, certain variables were fixed (typically those that cannot be controlled by equipment design) according to a "realistic worst case", since it is conventional in safety assessments not to underestimate the severity of a hazard. A wall-mounted split air conditioner air-handling unit (AHU) was employed for the majority of the experiments since it is easily adjusted for the majority of characteristics of interest. In addition, other equipment was employed for verification purposes to determine whether initial observations could be extrapolated. Table 39 gives a summary of the variables expected to have an influence on the dispersion of released refrigerant, and an indication of the values selected for the tests.

Refrigerant mass
Mass of refrigerant is closely linked to type, design and capacity of RAC equipment, and from the literature a larger release raises concentrations. Equation (308) represents the conventional method for specifying maximum allowable mass of HC within a space, being a function of LFL. Normally, safety standards adopt a safety factor of four ($\varphi = 0.25$, ASHRAE 15, 2001) or five ($\varphi = 0.2$, EN 378, 2000).

$$m_{r,\text{max}} = \varphi \cdot LFL \cdot V_{\text{hm}}$$  \hspace{1cm} (308)

Evaluation of equation (308) using $\varphi = 0.2$, HCs (whose LFL range from 0.038 kg.m$^{-3}$ to 0.045 kg.m$^{-3}$) give 7 - 9 g m$^{-3}$ of room volume, known as the Practical Limit. EN 378 (2000) also specifies an upper limit for HCs, which is 1.5 kg in publicly occupied spaces (e.g., a shop), and
2.5 kg in privately occupied spaces (e.g., an office). Accordingly, a range of refrigerant masses will be evaluated up to approximately 2.5 kg, but predominantly 1 – 1.5 kg since this corresponds to the upper limit for the test chamber.

Table 39: Summary of test parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Selected for tests</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment type</td>
<td>3 AHU, 2 refrigerators</td>
</tr>
<tr>
<td>Refrigerant release mass</td>
<td>Up to 2.5 kg</td>
</tr>
<tr>
<td>Choice of gas/refrigerant</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>Room air movement</td>
<td>Variable flow rates from unit fan</td>
</tr>
<tr>
<td>Air discharge height</td>
<td>Variable</td>
</tr>
<tr>
<td>Air discharge direction</td>
<td>Horizontal, upwards, downwards</td>
</tr>
<tr>
<td>Post-ventilation time</td>
<td>Variable</td>
</tr>
<tr>
<td>Refrigerant release height</td>
<td>Variable</td>
</tr>
<tr>
<td>Refrigerant release location</td>
<td>Beneath fan, other points in room</td>
</tr>
<tr>
<td>Refrigerant release time</td>
<td>Variable</td>
</tr>
<tr>
<td>Room size</td>
<td>135 m³</td>
</tr>
<tr>
<td>Floor surface</td>
<td>Course carpet</td>
</tr>
<tr>
<td>Room tightness</td>
<td>Fully sealed</td>
</tr>
<tr>
<td>Thermal internal convection</td>
<td>None, people and PCs</td>
</tr>
<tr>
<td>Room air temperature</td>
<td>21°C</td>
</tr>
<tr>
<td>Room obstacles</td>
<td>None, people and PCs</td>
</tr>
<tr>
<td>Release phase</td>
<td>Vapour</td>
</tr>
<tr>
<td>Refrigerant release velocity</td>
<td>&lt; 0.1 m s⁻¹</td>
</tr>
<tr>
<td>Refrigerant release direction</td>
<td>Downward</td>
</tr>
</tbody>
</table>

Room size

Due to practical constraints, only a single room size could be tested. The literature review highlighted studies that showed that for a release of mass corresponding to a fixed practical limit, a larger floor area tended to produce higher concentrations (e.g., Fletcher and Johnson, 1986, CCPS, 1996, Kataoka et al, 2000b). Fletcher and Johnson also comment on the effect of room size when ventilation is used, pointing out that for a specific airflow rate, mixing is less effective for larger rooms. In terms of room geometry, several studies (e.g., Marshall, 1984 and Cleaver et al, 1994) found that room shape and ceiling height had negligible influence on floor concentrations particularly in a still environment. Thus, a relatively large space was favoured for the experiments. A test chamber of 6.7 m × 6.7 m × 3.0 m in height was constructed.
Floor surface

Conventional dispersion models for large releases outside have a dependence upon terrain geometry, where an uneven surface tends to poorer mixing (CCPS, 1996, Woodward, 1998). Assuming this applies equally to floor surfaces in a confined space, a relatively coarse carpet was employed.

Room obstacles

From the literature, it is known that obstacles do not generally inhibit mixing (e.g., Leach and Bloomfield, 1973 and Britter 1988), except in rare occasions where channels are created or when there is a barrier at floor level. The number of possible types of obstacles and combinations would massively increase the number of tests. Accordingly, it was decided not to account for obstacles.

Air temperature

The effect of air temperature is (and humidity) largely neglected in the literature. However, since measurements are generally interpreted in terms of relative density, or buoyancy of the release, their effects are implicit. A higher air (and release) temperature has the effect of reducing buoyancy of the refrigerant. Given that the application of RAC equipment is normally in occupied spaces, a normal design temperature of 21°C and 50% relative humidity was used.

Room air movement

The literature identifies room air movement as a major influence on dispersion. Air movement is generated by movement of the gas within the room, and may also be produced by infiltration, forced airflow and thermal sources (addressed below). Since RAC equipment often utilises an integral fan to discharge air from the evaporator or condenser into the room, it was important to understand its effect on dispersion. Integral fans were tested under varying airflow rates, discharge velocities and directions, and positioning within the room. External ventilation (into and extracted from the room) was not employed.

A further consideration is the operation period for the fan. During on-cycle a fan will normally be running and switched off during off-cycle, although a loss of refrigerant lowers suction and discharge pressures which may cause the evaporator and/or condenser fans to be terminated. With this in mind it is important to consider the whether the fan needs to operate continuously, which is decided according to the possibility of refrigerant leakage during the periods that the
system if off. Further, due to the rapid change in system pressures, temperatures and vibration following compressor start-up and shut-down, the possibility of a leak developing at these times is likely to increase. Given that a catastrophic leak may occur after compressor cessation, there is a case for prolonging fan operation following shutdown; this concept of protracted airflow is termed the post-ventilation period. As with compressor termination, the effect of compressor starting may also initiate a leak, but for this situation where the leak may begin at the same time as the fan, airflow will be underdeveloped and therefore less effective dispersion. Similarly, initiation of airflow by a pressure sensor or refrigerant detector would have the same effect. Therefore, to simulate this worse case airflow, tests were conducted by initiating airflow simultaneously with the release.

Room tightness
The tightness of the room influences infiltration, which introduces airflow that is difficult to measure. Despite typical infiltration rates being small airflow rates of less than 0.5 h⁻¹ have been shown to influence floor concentration (Leach and Bloomfield, 1973, Fletcher and Johnson, 1986, Marshall, 1984). Therefore, a gas-tight sealed room was considered essential to avoid erroneous interpretation of measurements. Sealing the room also prevents migration of refrigerant through cracks or porosity of the walls.

Thermal convection
Cleaver et al (1994) stated that thermally induced convection from warm surfaces could produce significant mixing, which implies that heat sources should be removed from the room. Heat transfer through walls creates a differential temperature across the air layer at the wall, and thus convection. To avoid this, the test chamber was located within a larger space which was maintained at the same temperature. Instrumentation within the room was not expected to generate any noticeable heat output.

Height of airflow source (fan)
From Marshall (1984) and Fletcher and Johnson (1986), the height of the ventilation inlet and outlet are critical to mixing, particularly with respect to the position of the release source. Since the fan height varies widely with equipment type, it is considered an important variable. Appropriate values were selected according to typical installation heights for equipment, covering floor to ceiling mounted.
Airflow discharge direction

Whilst most condensing units discharge air in a single direction, many AHU have a multi-directional discharge vent. According to Marshall (1984) and Fletcher and Johnson (1986), airflow direction relative to the direction of the release is a significant factor in the amount of mixing. Therefore upward, horizontal or downward air discharges from the AHU were assessed.

Gas/refrigerant

Several HCs are likely to be used in air-conditioning equipment: R290, R600a and R1270, and mixtures thereof. Since the modification of test facilities and purchase of specialist instrumentation to mitigate the potential for ignition when dealing with flammable gases imposes excessive cost, a non-flammable mimic was used. Carbon dioxide was chosen based on similar density to R290 and R1270 at room conditions, and the large number of studies that successfully employed it as an alternative to LPG in dispersion tests (e.g., Leach and Bloomfield, 1973, Fletcher and Johnson, 1986, Cleaver et al, 1994, Kataoka et al, 2000a, 2000b).

Release rate

From various studies (Fletcher and Johnson, 1986, Cleaver et al, 1994), it is known that that a higher release flow rate produces higher concentrations. For a finite charge, the mass flow rate of the refrigerant (assumed to be constant) dictates the release time, which implies that the shortest time for the total charge to be leaked will produce higher concentrations. The theoretical minimum release time, $t_{r,min}$, occurring from a full-bore pipe shear whilst the system is operating can be estimated from the mass flow rate and refrigerant charge ($m_r$), (equation 309)

$$t_{r,min} \approx \frac{m_r \cdot \Delta H}{Q}$$  \hspace{1cm} (309)

where $\Delta H$ is refrigerant enthalpy difference across the evaporator and $Q$ is refrigerating capacity. From reviewing manufacturer data, typical HC charge sizes equate to 0.1 – 0.2 kg kW\(^{-1}\) of evaporator capacity. Evaluating equation (309) for several HCs found $t_{r,min}$ to be around one minute. However, under real conditions a loss of refrigerant stops the compressor, thus slowing release rate.

Another approach is to consider the refrigerant flow through a leak hole of a given size. For example, the reports of Blom-Bruggeman et al (1996a, 1996b) use the steady flow equation to calculate mass flow of refrigerant through a full bore pipe rupture, but results in very rapid release times, far shorter than equation (309). Other literature reports on tests to determine
catastrophic leak time, including Kataoka (1999) who gave a time of 4 minutes from an air conditioner and Blackwell at al (2004) measured 196 s using 0.39 kg of CO₂ and 146 s for 0.42 kg of R152a for a car air conditioning system. A model for catastrophic releases from chemical plant was reported by Woodward (1999) that takes account of internal frictional losses as the fluid flow through pipework and fittings. This was tested by using inputs appropriate for a refrigerating circuit but was found to return unrealistic results.

For this work a number of catastrophic leakage experiments were made (PR100374, 2000) and found that the refrigerant released from an air conditioner ceased after about 3 minutes, approximately consistent with the results from Kataoka et al (1999) and Blackwell at al (2004). Therefore, a minimum release time of 210 s was considered reasonable for the RAC equipment presently under consideration. Whilst this release time represents a catastrophic leak, prolonged releases also occur. Although the effect of a longer release time for a given mass is a lower floor concentration, a release duration that exceeds the post-ventilation period has the potential to produce higher concentrations. In this situation, it was important to understand the influence of post-ventilation airflow on subsequent mixing. Note that although leak time cannot be controlled by design, it was tested over a range of values because the worse case changed with different post-ventilation periods.

Release velocity
Studies by Baines and Turner (1969), Marshall (1984) and Cleaver et al (1994) considered the effect of release velocity. In all, it was shown that a higher velocity produced improved mixing, particularly where the release point is close to the floor. Generally, a leak from a refrigeration circuit will be at high velocity due to the large pressure difference. However, in some cases the leak may impinge on the inside surface of an enclosure causing the release into the room to have negligible momentum. Examples of this include releases inside a refrigerated compartment or within AHU casing. Despite the observation of Marshall that the effect of release velocity is inhibited when the room is ventilated, it was considered important to employ low velocity releases throughout the tests. From the literature, velocity influences are minimal below 0.1 m s⁻¹, so a diffuser device was employed to ensure this was not exceeded.

Two-phase releases also occur from refrigerant circuits, but in light of the discussion of liquid releases in the literature review, a two-phase release is expected to produce more favourable mixing so vapour only releases were considered. Quantification of the models in Woodward
(1999) showed that for a liquid R290 release into the test chamber would have an expansion rate equivalent to 2 – 3 room volume changes per hour, sufficient for rapid dispersion of the gas.

Release direction
The effect of release orientation was reported by Baines and Turner (1969), Baines et al (1990) and Barnett (1991), which indicated that a downward direction leads to higher floor concentrations. In the case of very low velocity releases though, the effect of orientation becomes minor since the momentum of the plume is small compared to buoyancy forces. For the purposes of the tests, only a downward release was used.

Release height
Literature consistently reported that the mean concentration within a plume reduces with distance from the source, which is reflected by floor concentrations. Observations of RAC equipment reveal that leaks could occur from a variety of heights, demanding testing over a similar range. However, some leak positions could be ignored because they are likely to result in thorough mixing anyway, as seen in the measurements of Goetzler et al (1998). Such positions included the case of a release occurring from within the airflow duct or directly in the air discharge, and releases from above the air discharge.

Release location
Most studies that addressed ventilation present results based on releases over a range of heights but largely ignore lateral positioning. In the case of a split-system installation, piping is often fitted along walls and ceilings to the evaporator, thus the potential for a release at positions other than directly below the fan. Accordingly, it was necessary to assess release sources from positions around the room away from the source of airflow.

3.1.2 Apparatus set-up and experimental procedure
A general scheme of how the test chamber was set up, according to the discussion of variables, is illustrated in Figure 102. The CO2 and associated control equipment was located outside the room, as was the CO2 analyser. After warming and modulating the CO2 according to the flow meter it was passed through a tube to a diffuser located within the room. Above the diffuser was the AHU with the respective air discharge and inlet grilles, and the fan was initiated and stopped using an externally located switch. Sampling points drew the CO2/air mixture from various points

24 The experiments were carried out within environmental chambers at BRE laboratories, Watford, UK.
in the room and the resulting concentrations were determined with the CO₂ analyser. Temperature and velocity measurements were also made at various locations within the chamber.

The test chamber (6.7 m × 6.7 m × 3.0 m in height) provided an overall empty volume of 135 m³. The additional room contents such as the air-handler, tri-pods and instrumentation did not contribute to the depletion of room volume by any more than 0.5 m³ and the impact on room volume was considered negligible. To ensure the room was airtight, it was fully sealed internally with polyethylene sheeting. A decay test was conducted on the room each day to check tightness over the duration of a normal test period, which exhibited no noticeable decay in concentration.

![Figure 102: Schematic arrangement of the test chamber apparatus](image)

High purity CO₂ was used to avoid practical problems associated with the presence of moisture and other impurities as well as property calculations. To ensure sufficient gas be released within the specified time, four cylinders were placed within a warm water bath. The outlet of the cylinder was connected to a heated regulator to prevent freezing and to ensure a discharge temperature close to ambient. The outlet of the regulator was piped to a pressure transducer and an analogue flow meter and then to the diffuser located within the room. The regulator was controlled manually so that a constant flow rate was indicated on the flow meter. A diffuser was used to provide a mean gas exit velocity of between 0.05 m s⁻¹ and 0.1 m s⁻¹ across the anticipated range of mass flow rates. The construction of this diffuser was primarily of a plastic funnel with 10 mm inlet diameter and 190 mm outlet diameter and 200 mm long. The funnel was stuffed with insulating fibre to even radial velocity and course gauze used to hold in the fibre. The smoke used for visualisation tests was generated using a “Smokefactory Sky Walker” fog generator.
The RAC equipment comprised three AHUs and two refrigerated cabinets, as detailed in Table 40. The Impact 100 unit was chosen for the majority of the tests for several reasons. Apart from an AHU being more versatile, the Impact 100 had a manual unidirectional air discharge grille that could be easily redirected. In addition, the velocity into the unit was considerably lower than the other wall mounted AHU, which helped mitigate the influence of turbulence from the return air.

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Wall mounted AHU #1</th>
<th>Wall mounted AHU #2</th>
<th>Ceiling AHU cassette #3</th>
<th>Open front display cabinet #4</th>
<th>Refrigerated storage cabinet #5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Name</td>
<td>Impact 100</td>
<td>PKH 3</td>
<td>PLH 4</td>
<td>ZM</td>
<td>Gastronorm</td>
</tr>
<tr>
<td>Dimension of air outlet</td>
<td>1.4 m x 0.5 m</td>
<td>1.4 m x 0.3 m</td>
<td>1.5 m x 0.6 m</td>
<td>0.6 m x 1.3 m</td>
<td>2.1 m x 0.7 m</td>
</tr>
<tr>
<td>Nominal Capacity</td>
<td>9.9 kW †</td>
<td>7.7 kW †</td>
<td>9.7 kW †</td>
<td>1.7 kW</td>
<td>0.4 kW</td>
</tr>
<tr>
<td>Nominal Charge</td>
<td>0.96 kg ‡</td>
<td>1.30 kg ‡</td>
<td>1.68 kg ‡</td>
<td>0.80 kg</td>
<td>0.35 kg</td>
</tr>
<tr>
<td>Test regime</td>
<td>Extensive</td>
<td>Selected</td>
<td>Selected</td>
<td>Selected</td>
<td>Selected</td>
</tr>
</tbody>
</table>

† At defined standard conditions. ‡ With standard 7m interconnecting pipe.

**Instrumentation**

A list of instrumentation is given in Table 41. Cleaver (1999) suggests a minimum of 6 sample points with 2 measuring continuously. It should be noted that there is a time lapse between actual events and gas analyser measurements due to the time taken to draw the air/gas through the sample tubes. Positioning of sample points and the thermistors/anemometers was in accordance with the coordinates in Table 57 orientated according to Figure 186 (Appendix 2).

Concentration data was obtained in the form of a voltage output from gas analysers. Calibration required voltages that corresponded to known concentrations across the normal operating range; these were obtained with calibration gases at standard atmospheric concentration (0.0007 kg m⁻³), 0.009 kg m⁻³ and 0.018 kg m⁻³. Output voltages were converted to actual measured concentration \( (C_{\text{actm}}) \) using equation (310).

\[
C_{\text{actm}} = a_1 \cdot PD + a_2
\]  

(310)

where \( a_1 \) and \( a_2 \) are constants (obtained from a daily calibration routine) and \( PD \) is the output voltage. Given that a background concentration of CO₂ exists, an “absolute” concentration was...
calculated by subtracting the actual measured concentration measured during the tests from the measured concentration prior to the beginning of the test, \( C_{\text{actm}(0)} \). The absolute concentration, \( C \), was obtained from equation (311)

\[
C = C_{\text{actm}} - C_{\text{actm}(0)}
\]

(311)

where \( C_{\text{actm}(0)} \) was always about 0.0007 kg m\(^{-3}\).

Table 41: Information on instrumentation used

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Instrumentation details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Concentration</td>
<td>2 × Leybold Binos 1 infra-red analysers. 0 - 0.5 % (by volume) 5 × ADC infra-red analysers, 2 with a 0% to 3% range. and three with a 0 % to 5 % range (one fitted to a multi-point selector unit to give four intermittent sampling points). Sampling point lines leading to analysers. The concentration is sampled at an interval of 5 seconds.</td>
</tr>
<tr>
<td>Flow rate</td>
<td>Flow meter to measure gas (CO(_2)) flow rate for simulated refrigerant release (analogue)</td>
</tr>
<tr>
<td>Cylinder weight.</td>
<td>Salter Digital Scale for measuring gas release from cylinder.</td>
</tr>
<tr>
<td>Air velocity and</td>
<td>22 DANTEC combined spherical omni-directional hot-film anemometer and thermistor sensors connected to a DANTEC 54N10 multi-channel flow analyser for measurement of air speed and dry bulb air temperature. Both velocity and temperature measurements are recorded every 200 seconds, although velocity is monitored continuously over an integrated time period to provide the relative turbulence intensity.</td>
</tr>
<tr>
<td>temperature.</td>
<td></td>
</tr>
<tr>
<td>Data acquisition</td>
<td>Grant Instruments (Cambridge) Ltd Squirrel Meter/Logger 1000</td>
</tr>
</tbody>
</table>

Procedure

The choice of test conditions was largely determined on an evolving basis, where results were analysed following each test to help decide whether further investigation of each variable was necessary. This approach was employed so that use of resources could be optimised. The initial step was to obtain a visual appreciation of the behaviour of a release, which was achieved by filming smoke tests. Following this carbon dioxide releases were made and concentrations measured.

During the first set of visualisation tests the smoke was released from various points about the unit and the room. This provided a basic appreciation of how the AHU moved air within the room. The second set of smoke tests was aimed towards visualising the effect of the airflow on the refrigerant release. In order to counteract the neutral buoyancy of the smoke, it was mixed with a flow of carbon dioxide within a duct prior to its release so as to give the smoke an artificially increased bulk density. Observations were used to determine the best positioning for
sampling points so that the highest concentrations were measured. The carbon dioxide release tests all followed the procedure below.

i. Initiate ventilation to remove excess carbon dioxide from previous test, so that concentration was at the background level.

ii. Position air handler and gas release diffuser according to test schedule.

iii. Check positioning of sampling point lines and anemometers.

iv. Weigh CO₂ cylinder and note mass.

v. Switch off room ventilation system and seal dampers.

vi. Initiate recording of data loggers.

vii. Wait several minutes for room air movement to cease.

viii. Switch on air handler fan from remote position.

ix. Wait several minutes for room airflow to reach steady state.

x. Open gas-regulating valve to obtain prescribed flow rate.

xi. After prescribed release duration terminate gas flow.

xii. After prescribed post-ventilation period switch off air handler fan.

xiii. Monitor room concentration until near-steady state is achieved.

xiv. 'Clean' room of CO₂ by initiating room ventilation system.

xv. Weigh CO₂ cylinder and note mass.

For the concentration measurements, consistency of results were achieved by the following procedure conducted each day: visual check for room sealing, calibrate carbon dioxide analysers, and check carbon dioxide analyser dryers and replace crystals if necessary. In addition to these, a decay test on the room was conducted daily to check for room air tightness.

Test schedule

Based on the earlier discussion, specific combinations of variables were chosen. Initially, a reference test was established so that repeatability could be checked and to act as a basis for comparison of results. The variables selected for the reference conditions are as follows: wall mounted Impact 100 AHU, with horizontally discharged airflow at 0.26 m³ s⁻¹ from a height of 2.35 m and approximately 1 kg of CO₂ released from 1.95 m directly below the AHU over 210 s. Variables were adjusted throughout the testing, and the final schedule listed in Table 42; full details are provided in Table 58.
Table 42: General test schedule

<table>
<thead>
<tr>
<th>Purpose (variable)</th>
<th>Test no.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smoke visualisation (reference, with airflow, airflow + CO₂)</td>
<td>1, 2, 3</td>
</tr>
<tr>
<td>Decay/tightness (still)</td>
<td>4, 5, 7</td>
</tr>
<tr>
<td>Decay/tightness (with air movement)</td>
<td>6, 8</td>
</tr>
<tr>
<td>No air movement</td>
<td>9</td>
</tr>
<tr>
<td>Reference test</td>
<td>10, 24, 30</td>
</tr>
<tr>
<td>Air flow rate</td>
<td>11, 12, 26, 27, 28</td>
</tr>
<tr>
<td>Post-ventilation period and release time</td>
<td>13, 25, 29</td>
</tr>
<tr>
<td>Air discharge direction</td>
<td>14, 15</td>
</tr>
<tr>
<td>Release mass</td>
<td>16, 17</td>
</tr>
<tr>
<td>Horizontal release location</td>
<td>18, 31</td>
</tr>
<tr>
<td>Fan and release height</td>
<td>19, 20, 21, 22, 23</td>
</tr>
<tr>
<td>Change to AHU #2 (reference, airflow rate, release height)</td>
<td>32, 33, 34</td>
</tr>
<tr>
<td>Change to AHU #3 (reference, reference, release height)</td>
<td>35, 36, 37</td>
</tr>
<tr>
<td>Change to cabinet #1 (reference, release mass, release mass)</td>
<td>38, 39, 40</td>
</tr>
<tr>
<td>Change to cabinet #2 (reference, release height)</td>
<td>41, 42</td>
</tr>
<tr>
<td>Thermal manikins and PCs</td>
<td>43</td>
</tr>
</tbody>
</table>

Interpretation of data

Measured data from each test was converted into a usable format so it could be employed to identify trends associated with the selected variables. Three concentrations were extracted from the measurements as shown below, and were treated according to the explanations.

Mean room concentration. In order to estimate the mass of refrigerant within the room from the release, the mean room concentration was multiplied by room volume. At the end of each test, although the room air was mixed, final analysis of the data showed that fully homogenous distribution of CO₂ did not always occur, so the mean room concentration ($\bar{C}_{Rm}$) was calculated using volume weighting for each sampling location (equation 312).

$$\bar{C}_{Rm} = \left( \sum_{i=1}^{10} C_i \cdot V_i \right) / V_{Rm}$$

(312)

where $V_i$ is the representative volume for a specific sampling point, as Table 43. Room refrigerant mass was then calculated using equation (313).

$$m_{r, Rm} = \bar{C}_{Rm} \cdot V_{Rm}$$

(313)
A mass balance was used to check the amount released from the cylinder against the amount present in the room. The mass calculated from equation (313) was compared against the difference in cylinder mass before and after the release (see Table 58), which showed good agreement, within ±5%. For the purposes of interpretation of the results, the mass based on room concentration was employed instead of by cylinder weight, since it corresponds directly to the measured concentrations, which are of interest in this study.

Table 43: Sample point location and representative volume

<table>
<thead>
<tr>
<th>Sample point</th>
<th>Location (key)</th>
<th>( \frac{A_i}{A_{Rm}} )</th>
<th>Ratio to define representative volume for sample point, ( V_i / V_{Rm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>C₁</td>
<td>Floor level, back (0 m. back)</td>
<td>0.129</td>
<td>0.135</td>
</tr>
<tr>
<td>C₁₀</td>
<td>Floor level, beneath unit (0 m. front)</td>
<td>0.129</td>
<td>0.139</td>
</tr>
<tr>
<td>C₆</td>
<td>Floor level, middle of room (0 m. mid)</td>
<td>0.337</td>
<td>0.135</td>
</tr>
<tr>
<td>C₈</td>
<td>Ceiling level, middle of room (3 m. mid)</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>C₁</td>
<td>Floor level, left of unit (0 m. LH front)</td>
<td>0.138</td>
<td>0.061</td>
</tr>
<tr>
<td>C₂</td>
<td>Floor level, left at mid room (0 m. LH mid)</td>
<td>0.129</td>
<td>0.350</td>
</tr>
<tr>
<td>C₃</td>
<td>Floor level, left at back room (0 m. LH back)</td>
<td>0.138</td>
<td>0.180</td>
</tr>
<tr>
<td>C₇</td>
<td>1 m above floor, middle (1 m. mid)</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>C₉</td>
<td>1 m above floor, beneath unit (1 m. front)</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>C₄</td>
<td>1 m above floor, back of room (1 m. back)</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Maximum floor concentration. One of the key areas of interest in this part of the study is the maximum concentration \( C_{f,\text{max}} \) resulting from a release. \( C_{f,\text{max}} \) is the highest concentration measured (typically at floor level) for the duration of each test, and is obtained from checking each complete set of sampled data. The exception is directly within or below the release, since it represents the concentration of the plume only. Concentrations from this sampling point were only included in the analysis from the time after the release had stopped. In all cases, the results showed that \( C_{f,\text{max}} \) occurred within a few seconds of the time of cessation of the release.

Maximum mean floor concentration. There are possible problems associated with the approach of using \( C_{f,\text{max}} \) as a representative concentration. Given the limited number of sampling points and the erratic nature of changes in concentration over time – which lead to high localised concentrations – erroneous interpretation could occur. Therefore, a more robust measure of
maximum concentration is required, i.e., an averaged value of all sampling points at each time
increment, of which the highest value is taken. This value is defined as the maximum mean floor
concentration, denoted $\overline{C}_{f,max}$. This is believed to provide a better indication of the degree of
stagnation and was supported by closer correlations. In quantifying the mean floor concentration
($\overline{C}_f$) each measurement at floor level was averaged, with respect to an allocated representative
floor area (Table 43), as equation (314).

$$\overline{C}_f = \frac{1}{A_{Rm}} \left( \sum_{i=1}^{6} C_i \cdot A_i \right)$$

(314)

where $A_i$ is the representative area for a specific sampling point, and was calculated according to
the radial distance from the release, thus differing values depending upon whether the release was
at the room wall (beneath AHU in Figure 186) or the centre. Other averaging techniques were
tested (for example, simple numerical mean, etc.) but were found to result in more scatter.

![Figure 103: Comparison of maximum and mean maximum concentrations ($r^2 = 0.86$)](image)

Assuming symmetry, where sampling points were not located (on the right-hand side of the room
with respect to the outlet direction of the fan), values from the left-hand side of the room were
used as approximation. A relationship between $\overline{C}_{f,max}$ and $C_{f,max}$ was found (Figure 103), and
the best-fit was according to equation (315).

$$C_{f,max} = 1.28 \cdot \overline{C}_{f,max}$$

(315)
3.1.3 Interpretation of Results
This section presents the interpretation of the smoke/visualisation tests and the results of concentration measurements. The complete set of experimental conditions and results are provided in Table 58 (Appendix 2), and selected data is correlated to characterise the influence of variables on room concentrations. Observations and trends assisted in developing the QRA and the associated empirical formula and for establishing design requirements to minimise flammable concentrations, and for the flammable volume calculations. The following examines the effect of important variables in terms of effect on $\overline{C}_{f_{\text{max}}}$. To help understand the dispersion process, results are presented in three formats: time-concentration profiles for each sampling point, height-concentration profiles at pre-set time intervals (where height and direction of unit airflow, refrigerant release and mean concentration are indicated), and overall trends in $\overline{C}_{f_{\text{max}}}$. Any observed trends are clarified with best-fit curves, and the associated regression coefficient ($r^2$) is also provided to indicate the quality of the correlation. Prior to presenting the results, qualitative information gained from visualisation tests is discussed.

![Diagram](image)

*Figure 104: Observations from smoke-only tests*

Visualisation test results
Visualisation tests were conducted in two stages: initially with neutrally buoyant smoke, and secondly with a mixture of smoke and carbon dioxide to increase the bulk density of the smoke. The smoke only tests provide observation of the airflows generated by the air handler, whereas the smoke/CO$_2$ tests indicated the effect of the unit airflow on the dense gas release.
The main observations from the smoke-only releases were that a release within the air stream or near the inlet grille drew the smoke directly into the return air stream, causing localised mixing in the discharge jet. This then tended to mix with the room air within the far side of the room. Similarly when the release was made from beneath the AHU it was seen that a large proportion of the smoke was rapidly drawn upwards to the AHU air inlet grille, again discharging a jet of smoke, which then dilutes towards far side of the room. These observations are illustrated in Figure 104. At all fixed fan speeds the air handler managed to rapidly move smoke to virtually all points within the room.

For smoke/CO$_2$ releases, similar events to the smoke only tests were observed. Observations generally showed three distinctly different mixing processes occur simultaneously: entrainment of air into the gas plume during its decent, gas entrainment into the return airflow to the inlet grille, and mixing caused by discharge airflow near floor level. A release from beneath the unit appeared to draw the dense smoke up towards the inlet grille. The main difference though was that the amount of smoke/CO$_2$ entrained directly into the AHU inlet grille from the diffuser was notably reduced. There was also a more pronounced 'spreading' effect of the smoke, that falls towards the floor and then across towards the extremities of the room. These effects are influenced by the higher density of the plume. As the cloud flows across the floor, the return air entrains a proportion of the smoke in a counter-flow manner. The discharge and dispersion of the entrained air in the AHU discharge airflow is as with the smoke only case. These are illustrated in Figure 105. Releases from beside the unit behaved in a similar manner to those from beneath the unit.

These visualisation tests are useful in two respects. Firstly, they offer an indication of the regions where higher concentrations are expected. The dense smoke is seen to 'land' at two locations:
directly beneath and in front of the diffuser, and towards the far wall when the airflow was sufficient to entrain it into the AHU, and throw it across the room. The placing of sample points at these locations is therefore important. Similarly, from the entrainment theory mentioned in the literature review, the velocity close to the gas/air interface is a key factor, which provides an indication as to necessary positioning of anemometers. The second benefit from these observations is the contribution towards interpretation of the concentration measurements.

![Concentration-time profiles for reference test (no. 10)](image1)

![Concentration-height profiles for reference test (no. 10)](image2)

**Figure 106: Concentration-time profiles for reference test (no. 10)**

**Figure 107: Concentration-height profiles for reference test (no. 10)**

**Reference test**

Initially, the results for a reference test (no. 10) are presented to act as a basis for comparing the influence of different variables in other experiments. The reference test was repeated on two more occasions (nos. 24 and 30) to check consistency of the results, and all three test results were within ± 15% of the average. Figure 106 shows a concentration-time profile for the reference test, and the legend refers to sampling positions in Table 57 (Appendix 2). General characteristics were observed from this data: immediately following the start of the release, concentrations at the floor, close to the plume, tend to rise quickly, with a more gradual rise at points further away. Concentrations at 1 m and 3 m tend to have a smoother and slower increase. A key observation is the erratic changes in floor concentrations, which is a particular characteristic of releases with forced airflow. From the cessation of the release all concentrations tend towards the mean room concentration and the profiles become much smoother.
Figure 106 shows the development of floor, mid height and ceiling concentration at one minute interval from initiation of the release until a relatively steady state exists. In this test, it was seen that following the initiation of the release gas falls towards the floor, but also a proportion is directed towards ceiling level (60 s). As the release continues all concentrations increase (180 s) until the supply of gas ceases at 210 s and a more homogenous room concentration develops (240 s to 360 s). During this period, floor concentration reduces as it is mixed with air in the upper parts of the room and approaches the homogenous concentration, $\bar{C}_{\mu m}$.

**Effect of released mass**

The effect of the released mass upon the concentration has been evaluated, as shown in Figure 108, where a lower mass, based on the same release time, reduces $\bar{C}_{f_{\text{max}}}$. This trend was
explained by entrainment theory. When a release occurs, the rate of entrainment is governed by its buoyancy and momentum. Volume flux of the release has a minimal influence on plume velocity and the quantity of entrained air is proportionally less for a higher released mass. Thus, a higher mass flow results in a higher concentration at the floor, giving a higher density, meaning that it requires more work to produce the same amount of mixing. This effect was observed by comparing Figure 108 and Figure 109, where the low release mass is seen to have an almost homogenous concentration within the room throughout the release duration except near the start. Conversely, with a large release mass, the concentration within the upper part of the room is significantly less than that at the floor, representing the residual quantity of gas in a dense cloud.

A linear relationship between release amount of gas and $C_{f,\text{max}}$ is seen in Figure 110. This is contrary to the case of no airflow (e.g., Cleaver et al, 1994, Kataoka et al, 2000b) where there is an exponential increase in $C_{f,\text{max}}$ with mass. It follows that where room airflow sufficient to produce very effective mixing, the concentration $C_{f,\text{max}} \approx C_{R,m}$ – at least until the effect of the airflow ceases to provide sufficient mixing to larger released masses, where the trend is expected to err towards the exponential trend as if there were no airflow. Finally, it is logical for the intercept of the line-fit line to be set at zero since no concentrations can develop without a release.

![Figure 111: Concentration-time profile for release with no airflow](image)

**Figure 111: Concentration-time profile for release with no airflow**

**Effect of room air movement**

A number of experiments were conducted to help understand the dependency of mixing on airflow rates. Different AHU fan speed settings were used (low, medium and high) and in addition, a voltage transformer was used to reduce the supply voltage to further lower the flow rates.
not provide additional reductions in $\overline{C_{f\text{,}max}}$. Also noteworthy is that where no forced airflow was used, mean air speed was never below about $0.03 \text{ m s}^{-1}$. This was believed to be due to a combination of residual air movement within the room, and the generation of air movement from the displacement of air by the descending plume of CO$_2$. Observations of the strong dependence of mixing on airflow rate were consistent with all the studies in the literature review, and in particular these effects were demonstrated experimentally in terms of inward discharge and extract ventilation, by Leach and Bloomfield (1973) and Fletcher and Johnson (1986).

![Figure 114: Effect of AHU airflow rate on $\overline{C_{f\text{,}max}}$ ($r^2 = 0.85$)](image)

Where air conditioners have multi-directional airflow, this can impact on the effectiveness of the mixing. Figure 115 and Figure 116 present results where the air discharge duct was rotated $\pm 30^\circ$ about the horizontal (taken as $90^\circ$ to the release direction). With a downward jet, the mixing is improved, whereas an upward jet results in higher floor concentrations than with a horizontal discharge. This is attributed to the downward airflow creating a higher air speed near the stratifying cloud at floor level. This effect is seen in Figure 115, where the floor concentration has much lower values than when the discharge is horizontal (Figure 107). Worth noting, is that this set of conditions resulted in a unique situation where the maximum concentration is encountered at 1 m above the floor, rather than at the floor. However, because the concentrations at 1 m elsewhere within the room were lower, $\overline{C_{f\text{,}max}}$ was still at floor level.

The effect of upward directed flow, as seen in Figure 116, results in much higher floor concentrations than the downward or horizontal case. The upward airflow is seen to produce a more stratified layer and the floor concentration changes dramatically with time (jumping by 1% within several seconds). Examining the air speed measurements show that although the mean
room velocity is similar in all cases, the local velocity near the floor is much higher in the case of the downward directed duct. Figure 117 shows the dependency of $\bar{C}_{f,max}$ on the duct discharge angle.

**Figure 115:** Concentration-height profiles for *downward air discharge* (test no 14.)

**Figure 116:** Concentration-height profiles for *upward air discharge* (test no 15.)

By extrapolating these observations, it is anticipated that discharge air originating from floor level will make $\bar{C}_{f,max}$ almost independent of discharge angle, since both discharge and return airflows are already present in the region where stratification may occur.

**Figure 117:** Effect of air discharge angle on $\bar{C}_{f,max}$ ($r^2 = 0.82$)

Normally, occupied spaces tend to contain thermal sources such as appliances and people. In order to observe the influence of thermal convection currents on a dense gas release, an
experiment was conducted with items designed to imitate the effect of thermal load from people and computers. A typical office worker population (4 per 45 m<sup>3</sup>) was located within the test chamber and a release made from the centre of the room. Heat output from people (at sedentary work) was mimicked with thermal manikins, set at 100 W and each PC with 200 W output. Each was located at approximately 2.2 m spacing from each other.

![Figure 118: Concentration-height profiles resulting from thermal loads (test no. 43)](image)

The room concentrations resulting from the release are illustrated in Figure 118. Initially, the floor concentration increased rapidly as if under quiescent conditions, but once the release stopped, floor concentration began to reduce, whilst the concentration at 3 m increased towards $c_{Rm}$, unlike a situation with no or minimal forced airflow (e.g., Figure 112), where ceiling concentration remained near zero. This difference is due to the mixture at low level being entrained upwards by thermal convection. Mean room temperature was 21.7°C with maximum variations of ±1.6 K – a range similar to other tests without thermal sources (except for fan motors). Evidently, only minor differences in local air temperature are needed to generate sufficient air movement to disperse a large refrigerant release.

**Effect of post-ventilation period**

The reference test used a post-ventilation period that began at the same time as the release, and finished at the same time as the cessation of the release, i.e., simultaneously for 3.5 minutes. The relationship between mixing and the post ventilation period was evaluated by stopping the fan prematurely. Figure 119 shows that a shorter post-ventilation period (relative to release time) has
a disproportionate effect in increasing $\bar{C}_{f,\text{max}}$. In fact by stopping the airflow halfway through the release produced virtually no difference in $\bar{C}_{f,\text{max}}$, than if no forced airflow was present at all.

Refrigerant leaks will not normally result in an instantaneous release, so the leak may persist for longer than 3.5 minutes of a catastrophic release. The results presented inFigure 120 show that when the release overshoots the post-ventilation period, significantly higher $\bar{C}_{f,\text{max}}$ develops. This is consistent with the case of reducing post-ventilation time. In both situations there is a lack of air movement, thereby inhibiting mixing. Also noteworthy is the effect of doubling the release duration has very little impact on increasing the floor concentration. This difference is understood to be due to the compensatory effect of the longer time available for entrainment of a specific released mass. Examination of the height-concentration profiles helps to further clarify the situation.

In Figure 121, it is seen that initially during the post-ventilation period the CO₂ is well-mixed throughout the height of the room. After cessation of the air discharge at 210 s, the continuing release causes a rapid increase in concentration mainly at floor level that appears to be negligible variation from the end of the release at 420 s to 1140 s. These results suggest that a longer post-ventilation period to provide continued forced air speed may be necessary where situations of extended release times would otherwise develop unacceptably high concentrations.
A release of 14 minutes followed the same pattern (Figure 122) although this was slower particularly within the last five minutes where the concentration hardly changed. This suggests that after a certain time a prolonged release reaches a maximum floor concentration and therefore cloud height increases. Another observation is that the mid- and ceiling level concentrations are approximately twice as high for the 7 minute release compared to the 14 minute release. This is attributed to the higher mass flow (for the 7 minute release) being as well mixed as the longer release, thus developing an initial concentration twice as high.

Figure 121: Concentration-height profile for a 7 minute release (test no. 29)

Figure 122: Concentration-height profile for a 14 minute release (test no. 25)

Figure 123: Concentration-height profiles for low release height (test no. 20)

Figure 124: Concentration-height profiles for low release and discharge height (test no. 21)
Effect of release and air discharge position.

The effect of different release heights is known to be a significant factor when there is no forced convection, as observed from the literature review. Tests were conducted to help determine this effect when there is forced airflow. Figure 123 shows the concentration-height profiles for a release height of 0.4 m. Comparing this to the reference test (Figure 107) it is seen that significantly greater concentrations occur at the floor. Also, concentrations at the upper part of the room, 0.004 kg m$^{-3}$, are half that of the reference test. A trend showing the dependence of $\bar{C}_{f,\text{max}}$ on the release height is shown in Figure 125 that suggests an exponential increase as the release position approaches the floor. This trend is explained by the descending plume having less entrainment with the surroundings if released closer to the floor, and once at floor level the concentration and therefore the mixture density is higher than if it had been released closer to the ceiling, making it less easy to disperse. Given that high room airspeed has a major influence on the plume’s rate of entrainment it is believed that the dependence of $\bar{C}_{f,\text{max}}$ on release height is stronger than under quiescent conditions.

![Graph 125: Effect of release height on $\bar{C}_{f,\text{max}}$ ($r^2 = 0.96$)](image1)

![Graph 126: Effect of release height and unit height ($h_r/h_d$) on $\bar{C}_{f,\text{max}}$ ($r^2 = 0.93$)](image2)

The effect of release height was investigated further by lowering air discharge height, thereby moving the high discharge velocity closer to the spreading plume. Comparing Figure 123 and Figure 124 where air discharge is near the release height, concentration profiles are seen to develop differently. When the air discharge is close to the floor there is a relatively slower rise in floor concentration and it is closely followed by the concentrations in the upper parts of the room, eventually moving towards $\bar{C}_{f,m}$ by the end of the release. This event is explained by the low-
level air stream rapidly entraining the gas close to the floor and then continuing up the far wall towards the ceiling.

Figure 126 shows additional results that account for the unit at differing heights, where data is displayed for various ratios by varying both the release height to air discharge height, \( h_r / h_d \). Other than the effect of the actual release height, this also indicates improved mixing of the refrigerant as the release position approaches the air discharge. These observations are explained by a combination of higher air speed promoting entrainment with the spreading plume. Where both release and air discharge is at a medium height \( h_r / h_d \approx 0.8 \) a comparable result to the cases at high and low heights was found. This implies that the high local airspeeds near the air discharge and return ducts creates significant amounts of mixing when in close proximity to the release source. As with the earlier discussion, the dependence of dispersion upon air speed rather than volumetric flow rate is highlighted. It also follows that if air speed is lowered, the benefit of having an air discharge position close to the release source lessens. Similarly, with air speed increased, its effect on mixing would be sufficient for \( \bar{C}_{f,\text{max}} \rightarrow \bar{C}_{\text{Rm}} \), again negating the effect of the proximity of air discharge to release source.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure127.png}
\caption{Effect of release location (horizontal distance from air disch.) on \( \bar{C}_{f,\text{max}} \) (\( r^2 = 0.93 \))}
\end{figure}

A number of tests were conducted to determine the effect of the release occurring at different locations within the room. The alternate locations were on the wall beside the end of the unit and in the opposite corner from the air discharge to represent a location as far a distance from the original leak position as possible. Figure 127 shows change in \( \bar{C}_{f,\text{max}} \) with respect to horizontal distance of the release from the centre of the AHU. It appears that the location has a significant effect on the concentration when the release point was located at the opposite corner of the room.
This is deemed to be a result of lower local air speed near the descending the plume – analogous to the case of lowering release height beneath the air discharge. Similarly, in the absence of airflow $C_{f_{max}}$ would become independent of the (horizontal) release location, and if high air speeds were present throughout the room, $C_{f_{max}} \rightarrow C_{r_{max}}$ regardless of release location. On this basis it may suggest that for any given set of conditions, there could be a mean air speed that would result in a “peak” sensitivity of air discharge and release position on $C_{f_{max}}$

![Figure 128: Configuration of AHU airflows showing discharge and return air paths](image)

**Equipment type**

Two additional AHUs and two refrigerated display cabinets were also tested. The results from these tests cannot be directly correlated alongside those for AHU #1 because their configurations differed, particularly in terms of airflow characteristics and positioning. Primarily, the purpose of testing different equipment was to establish whether the trends observed with AHU #1 could be applied across a range of equipment so that the findings may be considered generic. A schematic diagram of the airflow characteristics of the three AHUs and the refrigerated cabinets are provided in Figure 128 and Figure 129 respectively. The arrow pointing away from the unit represents the discharge air and those pointing into the unit, the return, or inlet airflows.

The physical construction of AHU #2 differed slightly from AHU #1 in that the direction of air discharge appears to be spanned across some 60° rather than discharging directly in one direction. Another difference between them was that its inlet grille was positioned below the discharge duct, whereas the former was opposite with the discharge at the top. AHU #3 differed in a number of ways; other than being located at the ceiling rather than the wall, it has four downward-pointing discharge grilles around the perimeter of the unit, with the inlet in the centre.
In comparing AHU #1 performance in Figure 107 against AHU #2 in Figure 130 where the conditions were identical to the reference test, AHU #2 produced better mixing than in the reference case. Despite AHU #2 having a lower volume flow rate than AHU #1, the concentration at cessation of the release was closer to $\overline{C}_{\text{r.m.s.}}$, and $\overline{C}_{\text{f,max.}}$ was lower. This difference is considered to be a result of two factors. First, the discharge duct area of AHU #2 was proportionally less (for the lower volume flow rate), which produced a higher mean air speed within the room. The second is that the geometry of discharge duct itself partially spanned downwards, ranging from about 0° to about 30° below the horizontal. Air speed measurements supported both of these observations, showing a significantly higher air speed closer to the floor, consistent with the test of AHU #1 with downward directed discharge.

![Figure 129: Configuration of cabinet air flows](image)

![Figure 130: Concentration-height profiles for AHU #2 (test no. 32)](image)

![Figure 131: Concentration-height profiles for AHU #3 (test no. 36)](image)
AHU #3 also demonstrated good mixing characteristics. In Figure 131 it is seen that the highest concentration develops above the floor and eventually approaches $\overline{C}_{zm}$. The high air speed measured close to the floor, partially a result of the downward air discharge, is responsible for the rapid mixing. A peripheral air discharge and central return duct is believed to cause the high concentrations at 1 m, where the entrained gas is drawn back up across the sampling points within the middle of the room. Interestingly, this is the only other situation where the maximum local concentration is at 1 m instead of at floor level, in addition to AHU #1 when discharge airflow is in downward direction.

Both refrigerated cabinets were floor-mounted (Figure 129). Cabinet #1 is an open fronted display cabinet with the condensing unit located beneath the storage space, with the airflow being produced by the condenser fan, drawing air through from the rear of the unit and discharging at the front. The Cabinet #2 has a top-mounted condensing unit that is only open to the top, again with a fan drawing air down through the condenser and then discharging it out in an upwards direction. In comparison to the AHUs, the most important difference is the low air velocity at discharge, due to a relatively large duct area.

![Concentration-height profiles for cabinet #1 and cabinet #2](image.png)

Figure 132: Concentration-height profiles for cabinet #1 (test no. 40)  
Figure 133: Concentration-height profiles for cabinet #2 (test no. 41)

The floor concentrations resulting from a release beside the Cabinet #1 were relatively high, as shown in the profiles in Figure 132. This is because air discharge velocity – and therefore mean room air speed – is low, and the release source being at low level. However, whilst a high concentration developed at the beginning of the release, the concentration at floor level halves by the end of the release and post-ventilation period. This suggests that whilst the air velocity is low,
the effect of air movement still contributes to mixing, similar to the case where thermal sources were used.

Table 44: Information on test parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Observation for lower $\overline{C}_{f,\text{max}}$</th>
<th>Test range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant release mass</td>
<td>Lower mass</td>
<td>0.3 – 2.0 kg</td>
</tr>
<tr>
<td>Room air movement</td>
<td>High air speed, counter-flow to release plume</td>
<td>From integral fan only. $u_{R_{\text{m}}} = 0$</td>
</tr>
<tr>
<td>Air discharge height</td>
<td>Low, close to release</td>
<td>0.2 – 2.5 m</td>
</tr>
<tr>
<td>Air discharge direction</td>
<td>Downward</td>
<td>Horizontal. $\pi / 3$ rad. $2 \cdot \pi / 3$ rad</td>
</tr>
<tr>
<td>Post-ventilation time</td>
<td>Longer duration</td>
<td>0 – continuous</td>
</tr>
<tr>
<td>Refrigerant release height</td>
<td>High, close to air discharge and return</td>
<td>0.2 – 2.5 m</td>
</tr>
<tr>
<td>Refrigerant release location</td>
<td>Close to air discharge and return</td>
<td>Beneath fan, other points in room</td>
</tr>
<tr>
<td>Refrigerant release time</td>
<td>Longer duration</td>
<td>210 – 840 s</td>
</tr>
<tr>
<td>Room size</td>
<td>Smaller per unit mass released $^a$</td>
<td>36 – 135 m$^3$</td>
</tr>
<tr>
<td>Room tightness</td>
<td>High infiltration $^1$</td>
<td>Fully sealed</td>
</tr>
<tr>
<td>Thermal internal convection</td>
<td>Increased thermal source output</td>
<td>None</td>
</tr>
<tr>
<td>Room air temperature</td>
<td>Negligible $^1$</td>
<td>Fixed (20°C - 22°C)</td>
</tr>
<tr>
<td>Room obstacles</td>
<td>Can reduce $C_f$ $^1$</td>
<td>Not tested</td>
</tr>
<tr>
<td>Release phase</td>
<td>Liquid $^1$</td>
<td>Vapour</td>
</tr>
<tr>
<td>Refrigerant release velocity</td>
<td>High $^1$</td>
<td>0.03 – 0.17 m s$^{-1}$</td>
</tr>
<tr>
<td>Refrigerant release direction</td>
<td>Upward $^1$</td>
<td>Downward</td>
</tr>
</tbody>
</table>

$^1$ Based on literature review.

As with Cabinet #1, Cabinet #2 also produced relatively high floor level concentrations probably due to low air speeds within the room, Figure 133. Following cessation of the release, the floor level concentration begins to reduce slightly, whilst the mid and ceiling level concentrations continue to increase. With AHU #1, when $h_r / h_d \rightarrow 1$, $\overline{C}_{f,\text{max}}$ was mainly independent of release height, whereas Cabinet #1 produced a much higher concentration than Cabinet #2, despite both their $h_r / h_d \approx 0.8$. If the airspeed associated with both cabinets is compared to that of AHU #1, it is found to be much lower, approaching that of a still room. Therefore the influence of air discharge height becomes less relevant, and this results in a stronger dependency of $\overline{C}_{f,\text{max}}$ on release height.
Concluding remarks

Certain variables have been identified that characterise the development of floor concentration due to a gas release from RAC equipment. It was found that $C_{f,\text{max}}$ always occurred soon after cessation of the release (for example, Figure 106 and Figure 111). Once $C_{f,\text{max}}$ was reached, $C_f$ tended to reduce gradually towards a homogenous mixture. The conditions leading to improved mixing and lower floor concentrations are:

i. smaller refrigerant charge, bearing in mind that it should be limited so that it does not exceed the LFL when evenly mixed within the room,
ii. high mean room air speed, achieved by increasing fan airflow rate or reducing outlet area,
iii. directing air discharge downwards,
iv. having a release source at high level, by positioning refrigerant containing parts as far above the floor as possible,
v. installing points of potential release close to air discharge/inlet,
vi. using a longer post-ventilation period.

A summary of the key test parameters and the corresponding conditions favourable for reducing $C_{f,\text{max}}$ are listed in the first and second columns of Table 44, and additional parameters not tested, but where detailed assessments were found in the literature review are also included. The third column lists the range of each variable tested. Despite using these techniques as an aid to reduce $C_{f,\text{max}}$, local concentrations above the LFL will always be present in the event of a leak. However, by incorporating appropriate measures based on the observed trends, the flammable region can evidently be minimised. This observation applies to different equipment when the dispersion characteristics and conditions were comparable or similar.

3.2 Development of correlation for floor level concentration

It was necessary to develop a correlation so that the concentration following a release could be predicted quantitatively. The correlation should be applicable to as broad range of RAC equipment as possible. An advantage of a generic formula is that it can be used to determine numeric design limits for RAC equipment to minimise the occurrence of concentrations exceeding the LFL following a refrigerant release.
The floor concentration following a release is a function of various characteristics associated with
the design and installation of that equipment. Some of these were identified as independent
variables, and some as dependent variables. The independent variables are those that can be fixed
within the design and construction of the equipment. Dependent variables are those that cannot be
controlled by equipment design. For example, refrigerant release velocity is dependent, whereas
release height is an example of an independent variable. Although room size is a dependent
variable it is assumed to be controlled through correct installation practice, and therefore is
handled as an independent variable. Where a dependent variable is considered, its value was fixed
according to what would produce highest $C_{f,\text{max}}$. A correlation was developed for two separate
cases: firstly, determination of $C_{f,\text{max}}$ resulting from a catastrophic release, and secondly,
determination of $C_f$ development and $C_{f,\text{max}}$ for a prolonged release.

3.2.1 Maximum floor concentration from catastrophic release

Within the literature, several methods have been employed to estimate concentrations from the
release of gases. These include: computational fluid dynamics (CFD), zonal decay models
(ZDM), Gaussian (concentration field probability) models, non-dimensional correlations, and
purely empirical correlations. It is considered that simplicity, ease of use, and generic application
are prerequisites for the chosen approach. This directs towards a non-dimensional correlation
being most suitable, and this approach has also been successfully employed for a similar purpose
elsewhere (i.e., Cleaver at al, 1994).

**Dimensional analysis**

The procedure to develop the correlation initially required identification of the variables that
significantly affect $C_{f,\text{max}}$. Each was assigned a quantifiable variable that could be employed for
the correlation. From the list in Table 44, the key dependent variables are listed in Table 45.

Some of the variables in Table 45 were combined to reduce the number of non-dimensional
groupings. A reduced gravity ($g'$) is defined to represent the buoyancy of the refrigerant relative
to the air, equation (316).

$$g' = g \cdot \frac{\rho_r - \rho_a}{\rho_a}$$  \hspace{1cm} (316)

---

Note that the correlations presented here differ slightly from those in Colbourne and Suen (2003b) due
to improvements in data analysis and availability of new validation data.
where \( g \) is gravitational acceleration, \( \rho_r \) is density of the released gas, and \( \rho_a \) is the density of air (based on room temperature and atmospheric pressure).

Table 45: Summary of influencing variables

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Selected variable</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Concentration</td>
<td>Max mean floor concentration</td>
<td>kg m(^{-3})</td>
</tr>
<tr>
<td>Quantity of released refrigerant</td>
<td>Mass</td>
<td>kg</td>
</tr>
<tr>
<td>Choice of refrigerant</td>
<td>Density</td>
<td>kg m(^{-3})</td>
</tr>
<tr>
<td>Room air movement</td>
<td>Volume flow rate</td>
<td>m(^3) s(^{-1})</td>
</tr>
<tr>
<td>Air discharge height</td>
<td>Height</td>
<td>m</td>
</tr>
<tr>
<td>Air discharge direction</td>
<td>Duct angle</td>
<td>rad</td>
</tr>
<tr>
<td>Air discharge area</td>
<td>Duct area</td>
<td>m(^2)</td>
</tr>
<tr>
<td>Refrigerant release height</td>
<td>Height</td>
<td>m</td>
</tr>
<tr>
<td>Room size</td>
<td>Area, Height</td>
<td>m(^2), m</td>
</tr>
<tr>
<td>Room air</td>
<td>Density</td>
<td>kg m(^{-3})</td>
</tr>
</tbody>
</table>

Since entrainment of a release into the air is normally a function of velocity, air volume flow rate from the unit and duct discharge area were combined to obtain the mean room air speed (\( \bar{u}_{Rm} \)).

Other studies in the literature demonstrate the relevance of \( \bar{u}_{Rm} \), for example, the experiments by Leach and Bloomfield (1974), where the air speed was adjusted by altering air inlet vent area, whilst maintaining volume flow rate resulted in different concentrations. The equation for calculating \( \bar{u}_{Rm} \) given by Etheridge and Sandberg (1996) was compared against averaged air speeds from the experiments. Figure 134. In averaging the local air speeds, different weightings were applied to each sensor and the best correlation was obtained with based on the relative volumes listed in Table 57 (Appendix 2). The sensors at 2 m and 3 m above floor level were neglected because the small number of sensors at these positions was considered insufficient to fully account for the variations in air speed at those horizontal planes. More so, it was deemed acceptable to neglect these since most mixing occurs close to floor level. The comparison showed measured air velocities correlated well with the calculated \( \bar{u}_{Rm} \) using equation (317) of Etheridge and Sandberg, where the discharge coefficient, \( c_d = 0.45 \).

\[
\bar{u}_{Rm} = \frac{\dot{V}_d}{\sqrt{c_d \cdot A_d \cdot \frac{V_{Rm}}{Rm}^{1/3}}}
\]  

(317)
where, $V_d$ is volume flow rate of air discharged from the duct, \( A_d \) is outlet area of duct discharge and \( V_{rn} \) is room volume.

\[
\frac{\bar{C}_{f,\max}}{(A_{rn} \cdot h_{rn})^{-1} \cdot m_r} = \Pi_1
\]  

\[
\frac{g' \cdot h_r}{\bar{u}_{rn}^2} = \Pi_2
\]  

\[
\frac{h_d}{h_r} = \Pi_3
\]

Figure 134: Comparison of measured room air speed ($r^2 = 0.92$)

A number of non-dimensional groupings were established according to the Buckingham pi-theorem, for the independent variables identified in Table 45. Initially, only measurements obtained for AHU #1 were used. There are six variables, which comprise three fundamental dimensions (mass, length and time) meaning that three repeating variables are required for the dimensional analysis, leaving three pi-groups to be evaluated. The pi-groups are formed as follows (equations 318, 319 and 320).

First, $\Pi_1$ represents the ratio of maximum floor concentration to mean room concentration, termed mixing effectiveness ($\varepsilon$); as $\varepsilon$ nears unity, homogenous mixing is observed. Secondly, $\Pi_2$ follows the definition of the Richardson number ($Ri$) which is commonly applied to dense gas release phenomena, and represents the ratio of the potential energies of density difference to air/mixture turbulence. There are several numerical definitions of $Ri$ in use, but in this case, it
represents the degree of air stability within the cloud that initially develops below the release point. A larger $R_i$ indicates a better cloud stability, i.e., less air movement and a more defined decent of the plume due to its higher density and/or longer fall. Thus, in isolation a smaller $R_i$ concurs with better mixing. Thirdly, $\Pi_3$ is the ratio of air discharge height to gas release height.

\[
\Pi_2 = a_1 \cdot \Pi_1^{n_1}
\]  
(321)

\[
\Pi_3 = a_2 \cdot \left( \frac{\Pi_2}{a_1 \cdot \Pi_1^{n_1}} \right)^{n_2}
\]  
(322)

\[
\Pi_1 = a_3 \cdot \left[ \frac{\Pi_1}{\Pi_2/(a_2 \cdot \{a_1 \cdot \Pi_1^{n_1}\})^{n_2}} \right]^{n_3}
\]  
(323)

Combining equations (318), (319) and (320) produced the overall correlation (equation 324).

\[
\overline{C}_{f,\text{max}} = 0.0913 \left( \frac{m_r}{V_{\text{Fr}m}} \right) \left( \frac{g' \cdot h_r}{(\overline{u_{\text{Fr}m}} \cdot \overline{f})^2} \right)^{0.49} \left( \frac{h_d}{h_r} \right)^{1.63}
\]  
(324)

The conventional equation for $R_i$ is employed here, but with the inclusion of the discharge angle adjustment factor: equation (319) becomes equation (325).
It is noted that despite a fixed $\bar{u}_R$, the experiments showed that $C_{f,max}$ changed with air discharge angle. Because insufficient data was available to integrate this variable using dimensional analysis, an adjustment factor ($\xi$) was introduced and applied to the mean room air speed to account for the effect of non-horizontal air discharge (equation 326). It includes the relative height of air discharge to room height since it is expected that the influence of discharge angle on mixing is less apparent as the height of air discharge approaches floor level, as discussed in Section 3.1.

$$\xi = \exp \left[ \frac{h_d}{h_R} \cdot (a_4 \cdot \theta + a_5) \right]$$ (326)

The angle of the discharge, $\theta$, (given in radians) is defined relative to the vertical plane. For example, $\theta = 90^\circ = \frac{\pi}{2}$ rad for a horizontal discharge and $\theta = 60^\circ = \frac{\pi}{3}$ rad for a downward jet. The value of the constants $a_4$ and $a_5$ are also included in Table 46.

<table>
<thead>
<tr>
<th>Pi-group/equation</th>
<th>Constant</th>
<th>Index</th>
</tr>
</thead>
<tbody>
<tr>
<td>(321)</td>
<td>$a_1 = 0.060$</td>
<td>$n_1 = -0.5799$</td>
</tr>
<tr>
<td>(322)</td>
<td>$a_2 = 0.123$</td>
<td>$n_2 = -1.72$</td>
</tr>
<tr>
<td>(323)</td>
<td>$a_3 = 5.611$</td>
<td>$n_3 = 1.60$</td>
</tr>
<tr>
<td>(326)</td>
<td>$a_4 = -1.214, a_5 = 1.464$</td>
<td>-</td>
</tr>
</tbody>
</table>

Substituting equation (325) into (324) and rearranging for $\varepsilon$ (as equation 318) gives equation (327).

$$\varepsilon = 0.091 \cdot \sqrt{Ri} \cdot \left( \frac{h_d}{h_r} \right)^{1.63}$$ (327)

The mean maximum floor concentration is therefore (equation 328).

$$C_{f,max} = \varepsilon \cdot C_{Rm}$$ (328)

where $C_{Rm}$ is the mean room concentration ($m_0/V_{Rm}$). Since the experiments always showed the mean maximum concentration to occur at floor level, $C_{f,max}$ cannot be lower than the homogenous concentration so equation (328) is only valid when $\varepsilon \geq 1$. Lastly, it is important to
note that where no air movement is present \((\bar{u}_{Rm} \approx 0)\) equations (327) and (328) give unrealistically high values of \(C_{f, max}\). From the measurements, it was found that the air speed in a quiescent room was consistently in the range \(0.03 \text{ - } 0.05 \text{ m s}^{-1}\), regardless of the duration that it was left undisturbed. The formula of Kataoka et al (2000a) was based on a quiescent room (equation 329), so this was used to determine the effective minimum air speed.

\[
C_{f, max} = \frac{1.5}{1.64} \frac{m_{r}}{h_{r} \cdot \sqrt{A_{Rm}}} 
\]  
(329)

Minimum air speed was found by substituting equation (329) into equations (327) and (328) and after putting \(h_{d} = h_{r}\), rearranging for \(\bar{u}_{Rm}\) (equation 330). The derived value of \(\bar{u}_{Rm, min}\) compared favourably with \(\bar{u}_{Rm}\) from the measurements without forced airflow. It is also noted that the introduction of \(A_{Rm}\) implies \(C_{f, max}\) has a dependence on room size when forced airflow is absent.

\[
\bar{u}_{Rm, min} = 0.1 \frac{m_{r}^{3/2}}{h_{Rm}^{3/2}} \frac{g'}{A_{Rm}} 
\]  
(330)

When there is no forced airflow, \(\bar{u}_{Rm}\) in equation (324) becomes \(\bar{u}_{Rm, min}\) in equation (330).

3.2.2 Floor concentration from non-catastrophic releases

Whilst evaluation of \(C_{f, max}\) is important for predicting the worse case (i.e., catastrophic release) concentrations, it is also useful to predict the concentrations resulting from slower releases. This is particularly relevant when considering the case of a release continuing after forced airflow has ceased. Therefore, an additional set of empirical formulae was developed for leaks lasting more than 210 s. To do this it was necessary to characterise the concentration development over time,
and a simple time-concentration model was used to extrapolate concentration change over an extended release time, as illustrated in Figure 137.

A good correlation was observed between $1/C_f$ and time, in the form of equation (331), although at the start of the release (typically for the first 50 - 100 s) the concentration-time profile tended to be erratic (e.g., Figure 106 and Figure 111), and did not necessarily match the smooth curve fit, nor any other pattern.

$$\frac{1}{C_f(t)} = a \cdot t^n$$  (331)

where $a$ and $n$ are obtained from equations (332) and (333), and the subscript 1 represents the start and subscript 2 the end of the release at $C_f = C_{f_{\text{max}}}$.

$$n = \frac{\ln(1/C_{f,1}) - \ln(1/C_{f,2})}{\ln t_1 - \ln t_2}$$  (332)

$$\ln a = \ln \left(\frac{1}{C_{f,2}}\right) - n \cdot \ln t_2$$  (333)

Substituting equations (332) and (333) into (331) gives equation (334), characterising the development of floor concentration with time.

$$\frac{1}{C_f(t)} = \exp \left[ \ln \left(\frac{1}{C_{f,1}}\right) - n \cdot \ln t_{r,1} \right] \cdot t^n$$  (334)

If $C_{f_{\text{max}}}$ for a catastrophic leak is calculated, the increase in concentration as the leak continues past 210 s could then be estimated, because it will be approximated by the curve prescribed by equation (331), as seen in Figure 137. Taking equation (334), and setting $C_{f,2} = C_{f_{\text{max}}}(210)$ and $t_{r,2}$ as $t_r(210)$ for a catastrophic release, it was rearranged to find $C_f$ when $t_r > 210$ s (equation 335).

$$C_f(t_r) = \frac{1}{\exp \left[ \ln \left(\frac{t_r(210)^n}{C_{f_{\text{max}}}(210)}\right) - n \cdot \ln t_r \right]}$$  (335)
Based on the mass that is released after 210 s, \( \bar{C}_{f,\text{max}}(210) \) is calculated from equation (328), but using \( \bar{C}_{Rm}(210) \) instead of \( \bar{C}_{Rm} \), as obtained from equation (336), which gives the mean room concentration had the release stopped at 210 s.

\[
\bar{C}_{Rm}(210) = \left( t_r(210) \cdot \frac{m_r}{t_r} \right) \cdot \frac{1}{v_{Rm}}
\]  

(336)

If equation (335) is to be used for conditions other than those in the experiments, the unknown variable \( n \) must be found (since this cannot be determined from equation (332)). The value of \( n \) was determined for each experiment by fitting equation (334) to the experimental data (for the latter two-thirds of the release time) until \( \bar{C}_{f,\text{max}} \) is reached. The data showed that when \( \varepsilon \approx 1 \), \( dC_f / dt \) was approximately constant, and therefore \( n \approx 1 \). With larger \( \varepsilon \), the curvature of \( dC_f / dt \) increased, so \( n < 1 \). This decay in the rate of increase of floor concentration is attributed to changes in the rate of mixing near floor level as a result of the density of the mixture at the floor approaching that of the descending plume, inhibiting the rate of mixing. From the data set, a reasonable correlation between \( n \) and \( \varepsilon \) was established (Figure 138), from which equation (337) was derived.

\[
n = \frac{1}{\sqrt{\varepsilon}}
\]  

(337)

A comparison against the measurements from tests 25 and 29 are shown in Figure 139, where the release time exceeded 210 s. For experiment no. 25, the correlation replicates the data reasonably well, although whilst the shape of the concentration development is comparable for experiment no. 29, the numerical fit is poor. One observation is that for the first stage when the fan is
operating (to 210 s), predicted concentrations are lower than the measured. Equation (327) gives $\varepsilon = 1.7$, which supports the observation of good mixing in the first stage, whilst the second stage (after 210 s) where the airflow had been terminated, the model follows the shape of the curve but the increase in $\overline{C_f}$ is over predicted. This is probably attributed to residual air momentum following cessation of the fan producing better mixing than if the room air was "still" from the point at which the fan terminated. In addition, the difference in $\overline{C_f}$ during either stage could be due to non-constant mass flow of gas during the experiment due to the fact that it was manually modulated.

![Figure 139: Comparison of calculated against measured concentrations](image)

3.2.3 Verification of correlations

The correlations for catastrophic and non-catastrophic releases were verified. First, the accuracy of the correlations was checked against the measurements from which they were initially derived. This was achieved using all the data for AHU #1, and data for the other equipment is also shown (Figure 140); in both sets, good overall agreement was found.

General discrepancies between predicted and measured values was attributed to the difference between of directly measured mass of CO$_2$ released and mass calculated from homogenous concentration within room and the effect of modulating the release of CO$_2$ into the room causing a variable mass flow rate. Overall, the comparisons imply that the methods described above are applicable to range of different units.
Second, in order to validate both the general correlation for catastrophic releases (equation 328) and the correlation for non-catastrophic releases (equation 335) they were compared against data reported by other studies. The literature review identified a number of related studies, and the measured $\bar{C}_{f,max}$ data from these is presented in Figure 141, against the developed correlations. Note that in some instances, some of the data needed for the correlations was absent, so assumptions had to be made. Where $C_{f,max}$ was reported instead of $\bar{C}_{f,max}$, the adjustment in equation (315) was used. All of the data associated with Figure 141 is listed in Appendix 3 (Table 59, Table 60, Table 61 and Table 62).

![Figure 140: Comparison of correlation against measurements (overall $r^2 = 0.87$)](image1)

![Figure 141: Comparison of correlations against data from other studies ($r^2 = 0.78$)](image2)

The comparison showed a good match, where the average value of calculated $\bar{C}_{f,max}$ was 16% higher than the measured values (shown by the best-fit), and about 80% of the calculated values were within ± 50% of measurements. These tolerances are considered to demonstrate good accuracy of the correlations. By comparison, Cleaver et al (1994) found their model (for slow releases in quiescent rooms) predicted mean floor concentrations to within a factor of 2 for 95% of their experimental results. Duijm et al (1996) compared the results of two separate dispersion models against six different sets of field and wind tunnel test data. The combined results for mean relative bias (MRB) of maximum concentrations (considered to be a good indicator by the authors') show that the Britter and McQuaid models gave MRB of -0.99 to -0.55 and -0.54 to -0.12 for the Gaussian Plume Model. Using the same analysis on the current experimental data (Figure 140) gives an MRB of -0.007 and the validation data (Figure 141) gives an MRB of -
0.014. Since the optimum value is zero, these results indicate that the correlation developed here demonstrates good accuracy.

Note that although the results from Clodic et al are presented in Figure 141, they were not included in the best fit (or the $r^2$ value). For this data-set, it is seen that calculated $C_{f_{max}}$ was generally much higher than the measurements, as a result of a capillary tube being used to release the refrigerant. The cross-sectional area of the capillary tube is small (0.004 m diameter – compared to 0.2 m for the diffuser), leading to a very high release velocity, thus not comparable to those used for the correlations. Further, many of the releases were made at a much lower level (0.05 m).

Similarly, some of the tests of Aloisi employed a capillary tube, and the over-predicted data-points are also seen in Figure 141. Examination of the data showed that at low $h_r$, calculated $\bar{C}_{f_{max}}$ was 10 – 20 times greater than measured. As $h_r$ increases, the difference reduced to a factor of 5 at the highest release. This implies that $\bar{C}_{f_{max}}$ is significantly more sensitive to release velocity as $h_r$ approaches floor level. This effect of release height and velocity was similarly observed by Baines and Turner (1969), Marshall (1984) and Cleaver et al (1994). However, where high airflow rate was present the differences were smaller, since the local air velocity encourages a rate of entrainment approaching that experienced with a high initial release velocity. Although both these data sets, based on releases from capillary tube, are inconvenient from a validation point of view, they do provide confidence in the “safety” of the correlations since most releases will realistically occur as high velocity leaks, therefore designs are likely to overestimate the “real” $\bar{C}_{f_{max}}$.

Other aspects that may have contributed to deviation between measured and calculated results include uncertain assumptions for absent values (such as air duct outlet height and discharge dimensions). As mentioned earlier, the correlations assume a constant release flow rate, and given that the methods described in some of the studies suggest a variable release rate, additional discrepancies would be expected. Additional reasons for deviation was attributed to positioning of sampling points, sampling intervals and of course incorrect reporting of data (for example in some cases, results presented $\bar{C}_{max}$ as being less than $\bar{C}_{R_m}$, which is a contradiction).
Based on the range of experimental conditions and those used in the validation studies, Table 47 lists what is considered to be the acceptable ranges for correlations, on the basis that actual $\overline{C}_{f_{\text{max}}}$ would not exceed that of equation (328).

Table 47: Range of applicability for correlations

<table>
<thead>
<tr>
<th>Variable</th>
<th>Acceptable application range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant release mass</td>
<td>&lt; 2.0 kg</td>
</tr>
<tr>
<td>Room air movement</td>
<td>$\bar{u}_{\text{re}} = 0.02 - 0.4 \text{ m s}^{-1}$</td>
</tr>
<tr>
<td>Air discharge height</td>
<td>0.2 m - 3 m</td>
</tr>
<tr>
<td>Air discharge direction</td>
<td>$\pi / 3 - 2 \cdot \pi / 3 \text{ rad}$</td>
</tr>
<tr>
<td>Post-ventilation time</td>
<td>90 s - 840 s</td>
</tr>
<tr>
<td>Refrigerant release height</td>
<td>0.2 m - 3 m</td>
</tr>
<tr>
<td>Refrigerant release location</td>
<td>Anywhere in room</td>
</tr>
<tr>
<td>Refrigerant release time</td>
<td>210 s upwards</td>
</tr>
<tr>
<td>Room size</td>
<td>&lt; 200 m$^3$</td>
</tr>
<tr>
<td>Room tightness</td>
<td>Any tightness</td>
</tr>
<tr>
<td>Room air temperature</td>
<td>10°C - 30°C</td>
</tr>
<tr>
<td>Room obstacles</td>
<td>Normal furniture distribution</td>
</tr>
<tr>
<td>Release phase</td>
<td>Liquid and vapour</td>
</tr>
<tr>
<td>Refrigerant release velocity</td>
<td>&gt; 0.03 m s$^{-1}$</td>
</tr>
<tr>
<td>Refrigerant release direction</td>
<td>Any</td>
</tr>
<tr>
<td>Refrigerant density</td>
<td>$\rho_r &gt; \rho_a$</td>
</tr>
</tbody>
</table>

3.3 Use of correlations for design purposes

In addition to using the correlations developed in Section 3.2 for the flammable volume modelling (Section 3.4), it is also beneficial to employ them for the purpose of aiding design of RAC equipment for HCs. Although this is not strictly part of the QRA, it is an appropriate starting point for determination of equipment design and installation features, with the implication that $P^{F*}$ is maintained at reasonably low values. Three concepts are addressed in this section:

- ensuring that $\overline{C}_{f_{\text{max}}}$ does not exceed the LFL in the case of a catastrophic release,
- ensuring that post-ventilation period is sufficient to avoid $\overline{C}_{f_{\text{max}}} > \text{LFL}$ in the case of a prolonged leak, and
- extending the application of the correlations to control of airflow for the same purpose.
3.3.1 Maximum concentration from catastrophic leak

The first stage is to determine a maximum charge amount (or minimum room size) according to the installation location or system needs. To be consistent with current practice, the amount of refrigerant is calculated using the approach detailed in EN 378 (equation 308), where the safety factor $q$ is normally 0.2. Departure from EN 378 is possible where $q$ is set at a value greater than 0.2, provided it is less than 1 otherwise mixing to below the LFL may not be possible. Conversely, for a fixed refrigerant charge, a minimum room volume is obtained from transposing equation (308).

The maximum tolerable concentration for design purposes $C_{\text{max,des}}$, should not exceed the LFL. However, to account for the regional fluctuation in floor concentrations, the LFL should be reduced by the factor based on equation (315). So $\overline{C}_{\text{f,max}}$ should not exceed $C_{\text{max,des}}$ (equation 338).

$$C_{\text{max,des}} = \frac{LFL}{1.28}$$  (338)

The variables that determine $\overline{C}_{\text{f,max}}$ (i.e., $m_r$, $V_{\text{ref}}$, $h_r$, $h_d$, $V_a$, $A_d$, $\xi$, and $\rho_r$ by changing refrigerant) were adjusted through alterations in design so that the corresponding $\overline{C}_{\text{f,max}} \leq C_{\text{max,des}}$ in the case of a catastrophic release.

3.3.2 Maximum concentration from non-catastrophic leaks

For fixed conditions, a catastrophic leak is known to produce highest $\overline{C}_{\text{f,max}}$ compared to a prolonged leak. If equipment has been designed to employ forced airflow to ensure $\overline{C}_{\text{f,max}}$ from a catastrophic leak does not exceed $C_{\text{max,des}}$, it is necessary for the airflow to be present for at least the same duration of the leak (i.e., 210 s). However, if the leak time exceeds the duration of the airflow (say $> 210$ s) then a $\overline{C}_{\text{f,max}}$ higher than $C_{\text{max,des}}$ may develop. One solution to prevent $\overline{C}_{\text{f,max}} > C_{\text{max,des}}$ is to ensure constant forced airflow (regardless of whether the compressor is operating, or if there is a deficit of refrigerant), but this implies increased energy use. An alternative method is to determine a minimum post-ventilation period ($t_{d,\text{min}}$) so that $C_{\text{max,des}}$ is not exceeded.

343
To reiterate, if for a catastrophic leak, \( t_d = t_r = 210 \) s, then \( \bar{C}_{f,\text{max}} \) is determined according to equation (328). However, if \( t_r > t_d \), the additional leak time in absence of airflow may result in an increase in rate of increase of \( C_f \) with time, as illustrated in Figure 142 (also shown earlier for tests 25 and 29). Conversely, if \( t_r \gg t_d \), then the mass flow of the release may be sufficiently small that \( \bar{C}_{f,\text{max}} = \bar{C}_r \) regardless of whether airflow is present or not. This implies that there is a "critical" leak time that will lead to a peak \( \bar{C}_{f,\text{max}} \), corresponding to a specific \( t_d \), thus a suitable \( t_d \) must be determined to ensure that neither a catastrophic or prolonged release would lead to \( \bar{C}_{f,\text{max}} > C_{\text{max,des}} \). Characterising this trend enables the minimum post-ventilation period \( t_{\text{d,min}} \) to be determined. In Figure 142, \( t_{\text{d,min}} = t_d \), which is the time corresponding to the intercept of the two concentration curvatures that results in \( C_{\text{max,des}} = \bar{C}_{f,\text{max}} \) at time \( t_r \).

Assuming only two discreet stages where the first results in a concentration rise to the end of the post-ventilation \( (\Delta\bar{C}_f(t_d)) \) and a second concentration rise to the end of release when forced airflow is absent \( (\Delta\bar{C}_f(t_{r-d})) \). Therefore, the final maximum concentration is the sum of these (equation 339), given that \( t_d \leq t_r \).

\[
\bar{C}_{f,\text{max}}(t_r) = \Delta\bar{C}_f(t_d) + \Delta\bar{C}_f(t_{r-d})
\]

(339)

where \( \Delta\bar{C}_f(t_d) \) and \( \Delta\bar{C}_f(t_{r-d}) \) are obtained from equation (335), and \( \bar{C}_r \) is determined from equation (336) for their respective portions of release time.

Figure 142: Change in concentration over time, with forced airflow ceased at \( t_d \) during release.
Since \( \Delta \overline{C}_f(t_d) \) and \( \Delta \overline{C}_f(t_d - t_r) \) are both implicit functions of \( t_r \), equation (339) cannot be solved directly for \( t_{d\text{min}} \). Required \( t_{d\text{min}} \) is when the concentration at the end of the release \( \overline{C}_{f\text{max}}(t_r) \leq C_{\text{max,des}} \), according to equation (338). In practice, equation (339) is evaluated by increasing \( t_d \) for a range of \( t_r \). The results of an example exercise are shown in Figure 143: when \( t_r = t_d \), the final concentration is below \( C_{\text{max,des}} \), but when \( t_r > t_d = 210 \text{ s} \), \( \overline{C}_{f\text{max}}(t_r) \) is above \( C_{\text{max,des}} \). As \( t_d \) was increased, \( \overline{C}_{f\text{max}}(t_r) \) falls, and in this example \( \overline{C}_{f\text{max}}(t_r) \) can no longer exceed the \( C_{\text{max,des}} \) (of 0.04 kg m\(^{-3}\)) provided \( t_{d\text{min}} > 510 \text{ s} \).

In general, a longer \( t_{d\text{min}} \) is needed when \( \overline{C}_{f\text{max}} \) from equation (328), when forced airflow is not included, is significantly higher than the LFL. Conversely, \( t_{d\text{min}} \) has a shorter duration when other variables such as installation height, are sufficient to keep floor concentration below the LFL. It is important for the user to conduct a proper evaluation. Firstly, if it is found that a long \( t_{d\text{min}} \) is necessary, the design and installation parameters associated with the equipment in question could be revised, leading to a smaller \( t_{d\text{min}} \). Second, a short \( t_{d\text{min}} \) helps minimise energy consumption of the fan motor.

![Figure 143: Effect of release time exceeding airflow time on final floor concentration](image)

### 3.3.3 Application to control of airflow

Forced airflow may be activated or terminated by pressure transducers or refrigerant detectors. RAC equipment sometimes employs pressure transducers that upon measuring a predetermined value may terminate the compressor, condenser and/or evaporator fans. Since a loss in refrigerant
charge normally results in a reduction of evaporating and condensing pressures, a certain charge
deficit may cause the pressure transducers to cease the unit forced airflow. Were the relationship
between refrigerant charge and system operation parameters known, the duration from the start of
a leak to the point at which the airflow terminates could be estimated.

Figure 144 illustrates the low rate of increase of $C_f$ due to high $u_{R_m}$ - line (a) - followed by the
sudden increase in the $C_f$ as the controls detect the loss of refrigerant through system operation
parameters at time ($t_{detect}$), switching off the fans thus terminating the air flow. Conversely, a
similar control could be employed to benefit dispersion of a release, where initiation of forced
airflow is activated. This is also illustrated in Figure 144 - line (b) - where a leak initially
produced a rapid rise of $C_f$, until the controls detect the absence of charge at $t_{detect}$, at which
point forced airflow would be initiated thereby increasing $u_{R_m}$, and subsequently lowering the
rate of $C_f$ rise, thus lowering $C_{f_{max}}$. Similarly forced airflow could be initiated on demand of a
refrigerant detector. A detector (typically located at floor level for a dense gas) would be set to
initiate emergency airflow at a predetermined concentration ($C_{f_{detect}}$) - line (c). At the time when
first $C_f = C_{f_{detect}}$, the controls would initiate airflow resulting in lower $C_{f_{max}}$ (Figure 144).

Determination of $C_{f_{max}}$ for all of these schemes use the same numerical procedure as that for
evaluating the use of post-ventilation period, for example, evaluation of $t_{detect}$ instead of $t_{d_{min}}$.

![Figure 144: Change in concentration with initiation and termination of airflow](image-url)
3.4 Development of flammable volume/time models

To evaluate the probability of ignition (via $P_{\text{ign}}^P$ and $P_{y}^P$) and the consequence models, it is necessary to estimate the size and duration of the flammable volume for any set of conditions, and to this end a dispersion model was developed.

Different options for modelling the dispersion process were discussed in Section 3.2. Of these, CFD, ZDM and Gaussian methods are viable, and have been used for similar purposes elsewhere (Woodward, 1998). Considering the objectives, CFD is unsuitable as it requires dedicated software, large computing power and generally long calculation times. Gaussian models demand extensive experimental data, and are more suited to large-scale unbounded releases. The ZDM approach is based on discretising a space into sub-volumes and handling gas/air flows within each, as separate, homogenously mixed enclosures, and number of elements and calculation complexity can be selected as desired. Given this versatility, ZDM was the preferred option.

To satisfy the needs of the probability and consequence models, the desired model outputs were flammable volume, flammable mass and associated flammable times. With respect to the release and dispersion characteristics identified in the literature (Section 1.2.3), observations from the experimental results suggest that the design of the model account for: indoor situation only, instantaneous and continuous finite releases, isoenthalpic vapour only, negatively buoyant (denser than air), low momentum and mixing mainly driven by convective and buoyancy-induced air flow.

3.4.1 General modelling approach

The entire dispersion process was modelled by evaluating three different stages of the dispersion process: descending plume, spreading plume and decaying cloud. Each stage was independently discretised into spatial elements. Figure 145 shows the overall concept of the model, where the direction of flow (hollow arrows) and interfacial entrainment (filled arrows) are indicated. The plume descends from some height, entraining surrounding air until it reaches the floor (first stage). The plume then spreads across the floor creating a layer over the entire floor (second stage). Further addition of material from the descending plume pushes up the preceding layer, forming a series of successive layers at certain time intervals that depend on the spreading time (from the plume base to the wall, $t_{\text{spred}}$). Entrainment occurs between the individual layers. When the release stops, spreading ceases and thereon the mixture is considered as a decaying cloud.
(third stage). This sequence was evaluated from initiation of the release to the time that the concentration within every element was below LFL.\textsuperscript{26}

![Diagram showing basic concept of dispersion models](image-url)

**Figure 145: Diagram showing basic concept of dispersion models**

For each stage the volume was discretised so that the dimensions of each element corresponded to a fixed time step. For the descending plume stage, the vertical dimension of the element was based on local plume velocity and the diameter of the element varied with time due to entrained material. With the spreading plume stage, the horizontal radial dimension was dictated by local spreading velocity. With the decaying cloud stage that exhibits no lateral movement, the radial dimensions of the spreading plume stage were adopted for convenience. For both the spreading and decaying cloud stages, similarly the vertical dimension of the elements varied with time due to entrained material. Since no new material is introduced once the release stops, the number of layers thereon remains constant for duration of the decaying cloud stage.

For simplicity the room geometry was assumed circular, with the release originating from the centre of the room, moving axi-symmetrically outwards in a radial direction until the flow reached the wall at a distance of $\sqrt{A_{\text{rm}}/\pi}$. Converting to a circular room was considered acceptable since both Marshall et al (1984) and Kataoka et al (2000b) found that room geometry, whether square, rectangular, partially separated or other irregular shapes, had a minor influence on concentration development. The size of the time steps ($\delta t$) was governed by the duration that

\textsuperscript{26} Although the value of LFL and UFL are known to vary with ambient temperature, humidity, atmospheric pressure, and method of ignition, it is assumed that LFL and UFL are constant.
a concentration front took to move from the base of the descending plume to the wall ($t_{sprd}$) and the number of spatial elements selected for the spreading plume ($N_{dr}$). Local radial spreading velocity across the floor was assumed constant for the duration of the release (supported by floor levels air speed measurements), hence $t_{sprd}$ remained constant for the release duration, as do the radial dimension of each floor element. $\delta t$ was obtained by dividing the total plume spreading time ($t_{sprd}$) by the number of radial elements ($N_{dr}$), the choice of which is addressed later (equation 340).

$$\delta t = \frac{t_{sprd}}{N_{dr}}$$

(340)

As with the spreading plume, the time taken for a front to travel from the release source, down the descending plume to the floor (called $t_{pl}$) was divided by $\delta t$ to determine the number of spatial elements ($N_{dp}$). $t_{pl}$ was calculated from the release height and local velocity of the descending plume, which was assumed constant for the release duration. Again, because plume centreline velocity changes with distance, the vertical dimension of each element differed. At each time step, entrainment of refrigerant and air mixture occurs between adjacent volumetric elements. The volume flux of material transferring across element boundaries was calculated according to the basic formula (equation 341) from entrainment theory (e.g., Baines and Turner, 1969).

$$\frac{dV}{dt} = A \cdot \alpha \cdot u_{fn}$$

(341)

where $A$ is the area of the interface and $u_{fn}$ is a velocity relative to the boundary, termed friction velocity. $\alpha$ is the entrainment coefficient and represents the ratio of actual volume flow across the boundary to the total possible flow rate, or conventionally defined: the ratio of the velocity of entrained material across the boundary to the relative velocity between elements. Evaluation of $u_{fn}$ and $\alpha$ was handled in different ways according to the particular stage of the dispersion process.
The overall calculation sequence was based on an iterative procedure, as summarised in Figure 146. Based on a set of input conditions, for each stage the mean floor concentration \((\bar{C}_{f,\text{calc}})\) was calculated over the duration of the release. However, for the spreading plume, calculation of \(u_{fn}\) and \(\alpha\) each requires an unknown constant, \(a(u_{fn})\) and \(a(\alpha)\) respectively, to be used later. In order to determine \(a(u_{fn})\) and \(a(\alpha)\), the dispersion model must be iterated for a set of conditions where \(\alpha\) is already established, and \(u_{fn}\) can be subsequently obtained. Thereafter, \(a(\alpha)\) can be employed for the conditions where \(\alpha\) is not known. The literature indicates a consensus on the value of entrainment coefficient for a descending \((\alpha_d \approx 0.1)\) and a spreading plume \((\alpha_c \approx 0.05)\) for quiescent conditions (e.g., Baines and Turner, 1969). Given the
descending plume and spreading velocities are known, an initial set of calculations for floor concentrations were made using certain selected input conditions, except $\bar{u}_{Rm}$ was set to a minimum value according to a quiescent room.

Using an initial guess for $a(u_{fr})$ and $a(\alpha)$, the concentration of each floor element ($C_{f,j}$) was used to calculate the mean floor concentration at each time step (equation 342), for $t \to t_r$.

$$\bar{C}_{f,calc}(t) = \left( \sum_{j=1}^{N_{dr}} C_{f,j}(t) \cdot A_{f,j} \right) / \sum_{j=1}^{N_{dr}} A_{f,j}$$  \hspace{1cm} \text{(342)}$$

where $A_{f,j}$ is the floor area covered by an element. For each $\delta t$, the result from equation (342) was compared against the corresponding $\bar{C}_f$ as determined from the empirical formulae (equation 335). The constant $a(\alpha)$ was then adjusted accordingly, and the calculation reiterated until $\bar{C}_{f,calc}(t) = \bar{C}_{f}(t)$ was satisfied, when the calculation proceeded to the next time step.

Once the final calculation for $t = t_r$ was complete, the time-averaged $\alpha_c$ for the spreading plume was compared against the established value ($\approx 0.05$) for quiescent conditions. $a(u_{fr})$ was then obtained accordingly and the entire set of calculations reiterated, until the mean calculated $\alpha_c$ was satisfied. $a(u_{fr})$ was then fixed for the final iteration of the entire model with the introduction of forced airflow conditions. Subsequently, empirical $\bar{C}_f(t)$, and $\bar{C}_{f,calc}(t)$ were recalculated for the new $\bar{u}_{Rm}$, which requires only one iteration since the constant $a(u_{fr})$ was now fixed.

For the decaying cloud, $\alpha_c$ was similarly used to calculate $\bar{C}_{f,calc}(t)$. However, it was not possible to determine $a(\alpha_c)$ according to $\bar{C}_f(t)$ since the empirical correlation is not applicable once the release has stopped. Instead, $a(\alpha_c)$ was fixed at the value determined for the final time step of the spreading plume (i.e., at $t_r$), and used for the duration of the decaying cloud stage.

The calculation sequence ceased once the concentration within every element was below LFL.

Recalling the purpose of the model, values taken during the final iteration from each time step when $\bar{C}_{f,calc}(t) = \bar{C}_{f}(t)$ were used to calculate the various flammable quantities. Within each room reference volume (corresponding to the three room height elements, $h$) the flammable
volume-time, \((V^F t^F)_h\), and the flammable mass-time, \((m^F t^F)_h\), are calculated by summing the flammable volume of all the spatial elements, \(j\), at each time step, \(i\), until all the mixture is below the LFL (equations 343 and 344, respectively).

\[
(V^F t^F)_h = \sum_{i=1}^{n} \left( \sum_{j=1}^{3} V^F_{\text{out,h},i,j} \cdot \delta t \right)
\]

(343)

\[
(m^F t^F)_h = \sum_{i=1}^{n} \left( \sum_{j=1}^{3} V^F_{\text{out,h},i,j} \cdot C^F_{\text{out,h},i,j} \cdot \delta t \right)
\]

(344)

where \(V^F_{\text{out}}\) is the exit volume of each flammable element whether in the descending plume, spreading plume or decaying cloud, and \(C^F_{\text{out}}\) is its corresponding concentration. The total flammable volume-time and flammable mass-time are the addition of all three room height elements (equation 345 and 346, respectively).

\[
V^F t^F = \sum_{h=1}^{3} (V^F t^F)_h
\]

(345)

\[
m^F t^F = \sum_{h=1}^{3} (m^F t^F)_h
\]

(346)

Lastly, the mean flammable volume \((\overline{V}^F)\), and mean flammable mass of the refrigerant \((\overline{m}^F)\), are determined by dividing the totals from equation (345) and (346) by the total flammable time, \(t^F\).

\[
\overline{V}^F = \frac{1}{t^F} \sum_{h=1}^{3} (V^F t^F)_h
\]

(347)

\[
\overline{m}^F = \frac{1}{t^F} \sum_{h=1}^{3} (m^F t^F)_h
\]

(348)

Summing the refrigerant mass within all elements (as \(V_{\text{out,j}} \times C_{\text{out,j}}\)) gives the total mass released up to a time \(t\) provided infiltration and extract ventilation are absent.

### 3.4.2 Spreading velocity

It is important to determine the spreading velocity of the plume, as it is needed to calculate the dimensions of spatial elements, and because it influences entrainment in that it is needed for
calculation of friction velocity. From the literature, two formulas were found for estimating
velocity of the flow of a gas release across a flat surface. CCPS (1996) details an approach of
Fay, which is similar to that of Sykes et al (1999) and when transposed, gives equation (349).

\[ u_{\text{spread},j} = \left( \frac{4 \cdot g' \cdot m_r \cdot t_{\text{spread}} / \rho_r}{t_j} \right)^{1.4} \cdot t_j \]  

(349)

Alternatively, Ditali et al (2000) give a formula that is also transposed for velocity (equation
350).

\[ u_{\text{spread},j} = \frac{g' \cdot m_r \cdot t_j^2}{2 \rho_r \cdot r_j^3} \]  

(350)

where \( r \) is the radial distance from the release source at time \( t \).

When equations (349) and (350) were compared against the test data, it was found that there was
a poor correlation (Figure 147), possibly due to not accounting for release height, air movement,
etc. In addition, as Sykes et al (1999) point out, the velocity of a spreading plume varies with the
height of the layer above floor level, which implies it is important to specify spatial location of
the velocity.

![Figure 147: Comparison of plume spread correlations and measurements \( (r^2 = 0.43) \)](image1)

![Figure 148: Correlation of friction factor and Reynolds no. for spread velocity \( (r^2 = 0.67) \)](image2)

To address these points, a new correlation was developed based on Bernoulli. To include the
influence of the release characteristics, the room air speed should also be considered since it may
act against or contribute to the momentum of the moving front. The new approach addressed the
static, velocity and frictional forces acting on the moving front. First, the static pressure imposed by the front was calculated from equation (351).

$$\Delta p_{sta, j} = g \cdot h_{eff} \cdot (\bar{\rho}_0 - \rho_a)$$

(351)

where $\bar{\rho}_0$ is the mean density of the mixture at the base of the plume; this is used instead of local density because it avoids excessive iterations of the entire model to obtain the local concentration. The effective cloud height $h_{eff}$ is based on an ideal homogenised cloud at $C_{f, max}$ (equation 352).

$$h_{eff} = \frac{m_j / C_{f, max}}{A_{lm}}$$

(352)

Second, the velocity pressure was approximated according to the difference between the plume front and the opposing force imposed by the room air (equation 353).

$$\Delta p_{vel,j} = \frac{1}{2} \cdot (\bar{\rho}_0 \cdot u_{sprd,j}^2 - \rho_a \cdot w_{pm}^2)$$

(353)

Thirdly, the frictional pressure was obtained from the conventional formula for flow across a flat surface (Douglas et al., 1986), equation (354).

$$\Delta p_{fr,j} = 8 \cdot \lambda_j \cdot \bar{\rho}_0 \cdot u_{sprd,j}$$

(354)

where the friction factor $\lambda_j$ was obtained from (equation 355).

$$\lambda_j = c \cdot Re_j^n$$

(355)

where the Reynolds number $Re_j$ was calculated according to a modified version of the conventional formula (equation 356).

$$Re_j = \frac{\bar{\rho}_0 \cdot u_{sprd,j}}{\mu} \cdot \sqrt{\frac{r_{f,j}^2 + h_r^2}{\lambda_j}}$$

(356)

where $\mu$ is viscosity of the mixture and was approximated to 0.00002 Pa s. The length modification ($\sqrt{r_{f,j}^2 + h_r^2}$) was included because a much closer correlation was achieved with the direct distance from the release source height, instead of simply using $r_{f,j}$. Finally, the local spreading velocity $u_{sprd,j}$ was obtained by iterating through equations (351) to (356) until equation (357) was satisfied.

$$\Delta p_{fr,j} + \Delta p_{sta, j} + \Delta p_{vel,j} = 0$$

(357)

354
The unknowns in equation (355) were found by applying the data from the experiments in Section 3.3.1 to equations (354) to (356). The data consisted of a combination of both floor level anemometer measurements for the duration of the release, and time differences between initial concentration measurements from different floor sampling points following the first positive concentration measurement directly below the release point. Both data sets provided approximately similar results. The correlation between \( \lambda \) and \( \text{Re} \) is shown in Figure 148, and for equation (355) the best-fit gave \( c = 37570 \), and \( n = -1.1 \). Typically for flow over a flat surface, \(-1 < n < 0\) (Douglas et al, 1986). The scatter was attributed to the measurements being of inadequate quality. Since the experiments were not specifically designed for obtaining the plume spreading velocity, it is believed that errors were introduced by insufficient frequency of sampling times, anemometers providing unidirectional air speeds and the influence of near walls on flow patterns. Nevertheless, the proposed approach is better than the formulae found in the literature, and therefore it was employed in the dispersion model. Further, the new approach can be developed when more reliable data becomes available.

### 3.4.3 Descending plume model

When an aperture develops within the refrigerant circuit, the leak is assumed to be released at a constant mass flow rate that is determined based on the released mass and the leak time (equation 358).

\[
\dot{m}_r = \frac{m_{sys} \cdot \psi}{t_r} 
\]

(358)

where \( m_{sys} \) is the design charge size of one circuit of the RAC equipment, \( \psi \) is a refrigerant retention factor (Section 2.4), and \( t_r = t_{\text{leak}} \).

The leak produces a high velocity jet, which may impinge on a surface or flow directly into an open space in any direction. Ultimately, whether through impingement or entrainment, momentum gives way to buoyancy forces and a relatively low velocity plume is formed that falls towards the floor. As it travels downwards, it entrains more air from the ambient thereby increasing in volume flow rate \( (\dot{V}_{p,\text{out}} > \dot{V}_{p,m}) \) and decreasing mean radial concentration \( (\overline{C}_p) \), density and centreline velocity \( (u_p) \). In addition, some mixture is entrained out of the plume into the surroundings. This process can be followed in Figure 149. The basis of the plume descent model uses the
fundamental formula for evolution of a plume in a quiescent environment, as developed by Baines and Turner (1969) (equation 359).\(^\text{27}\)

\[
\frac{dV_{e,j}}{dt} = \pi \cdot D_j \cdot \alpha_p \cdot u_{fn,j} \cdot \delta z_j
\]  

(359)

where subscript \(e\) denotes entrainment and \(D_j\) is the local plume diameter.

For a circular plume with similar density to the (quiescent) surroundings, \(\alpha_p\) was reported to be constant at around 0.08 to 0.10 (Baines and Turner, 1969), where the plume centreline velocity \(u_p\) was used as \(u_{fn}\). CCPS (1996) gives \(\alpha_p = 0.9\) for a high density gas release from an outdoor vent stack, whilst Thomas (2000) stated that \(\alpha_p\) is not constant but dependent upon plume density (although no values or correlations were provided). In the current study it is necessary to account for the influence of forced airflow within the room and therefore – consistent with the approach of Etheridge and Sandberg (1996) – \(\alpha_p\) was fixed (at 0.10) and \(u_{fn,j}\) was calculated according to relative velocities of the plume centreline and the cross-wind (i.e., \(\bar{u}_{rn}\)), termed the “excess velocity”. Conventionally, the friction velocity (equation 360) is a function of the fluid boundary friction stress (equation 361).

\[
u_{fn} = \sqrt{\tau_{fn} / \rho}
\]  

(360)

\(^{27}\) Zhao et al (2002) presented a more detailed Gaussian approach to determining concentration profiles within a jet. It was not employed here because it assumed only a high velocity jet and could therefore lead to an underestimation of concentrations.
\[
\tau_{fi} = \mu \cdot \frac{du}{dh} \tag{361}
\]

where \( \frac{du}{dh} \) is the velocity gradient with respect to distance \( h \) from a boundary. By substituting equation (360) into equation (361), the friction velocity was obtained (equation 362).

\[
u_{fi,j} = a(u_{fi}) \cdot \sqrt{\frac{\mu}{P} \cdot \frac{du}{dh}} \tag{362}
\]

where \( a(u_{fi}) \) was included to account for effects of turbulence as a result of the absence of a solid boundary, and was determined through an iterative process discussed in Section 3.4.1 (Figure 146). \( \frac{du}{dh} \) was based on the combined local plume centreline velocity and the mean room airspeed (i.e., \( u_{p,j} + \bar{U}_{rm} \)) as well as twice the radial distance from the centreline (i.e., \( 2 \times D_{j}/2 \)). \( u_{p,j} \) was calculated using the solution derived by Etheridge and Sandberg (1996) (equation 363).

\[
u_{p,j} = 0.437 \cdot \left( \frac{g' \cdot \bar{V}_{r}^{1/3}}{\alpha_{p}^{2/3} \cdot z_{j}^{1/3}} \right) \tag{363}
\]

where \( \bar{V}_{r} \) is the release volume flow rate of the refrigerant at room temperature and atmospheric pressure (equation 364).

\[
\bar{V}_{r} = \dot{m}_{r} / \rho_{r} \tag{364}
\]

The local plume diameter was equation (365).

\[
D_{j} = 2.69 \cdot \alpha_{p} \cdot z_{j} \tag{365}
\]

Equation (359) assumed that entrainment occurs only into the plume, whereas it is known from both initial visualisation tests and from concentration measurements (Section 3.1) that material from the plume is also entrained into room air. This is handled with a gas exchange coefficient \( \beta \) in equations (366) and (367) for volume flow, and equations (368) and (369) are corresponding mass flow of refrigerant into, and out of the plume respectively.

\[
\frac{dV_{in,j}}{dt} = \beta_{j} \cdot \frac{dV_{e,j}}{dt} \tag{366}
\]

\[
\frac{dV_{out,j}}{dt} = (1 - \beta_{j}) \cdot \frac{dV_{e,j}}{dt} \tag{367}
\]

357
where $C_{\text{sur},j}$ is the mean concentration within the local surroundings, and $C_{\text{out},j}$ is the mean concentration at the outlet of an element (equation 370).

$$C_{\text{out},j} = \frac{m_{\text{out},j}}{V_{\text{out},j}}$$

The total volume and mass balance for each element is equation (371) and (372) respectively.

$$\frac{dV_{\text{out},j}}{dt} = \frac{dV_{\text{in},j}}{dt} + \frac{dV_{\text{e, in},j}}{dt} - \frac{dV_{\text{e, out},j}}{dt}$$

$$\frac{dm_{\text{e, out},j}}{dt} = \frac{dm_{\text{e, in},j}}{dt} + \frac{dm_{\text{e, in},j}}{dt} - \frac{dm_{\text{e, out},j}}{dt}$$

Equation (341) states that entrainment is proportional to the velocity difference between the plume centreline and the environment. Therefore, $\beta$ was simply assumed proportional to the relative velocities between the plume centre line velocity and the room mean airspeed, as expressed by equation (373).

$$\beta_j = \frac{u_j}{u_j + \bar{u}_R}$$

Given that plume centreline velocity is normally an order of magnitude greater than $\bar{u}_R$, the majority of material is entrained into the plume as seen in equations (366) and (367).

The output from equations (371) and (372) provide values of volume and mass for inclusion in the calculation of $\bar{V}^F$ (equation 347) and $\bar{m}_r^F$ (equation 348).

3.4.4 Spreading plume model
The work of Baines and Turner (1969), Cleaver et al (1994) and others showed that when a plume within a quiescent room reaches floor level, the flow spreads laterally; further entraining air until a layer is created. Depending upon the local momentum and buoyancy, the layer can be fully mixed (e.g., when $Ri < 10$) becoming homogenous, otherwise a second stratified layer that graduates from the concentration of the homogenous layer to zero, is created above it. With
forced airflow, similar two-layer patterns are demonstrated experimentally and in various studies detailed in the literature.

Based on these characteristics, the plume spread model was based on a multi-layer approach where an initial floor layer is formed. As it spreads, the flow entrains material from above and subsequently dilutes. As the descending plume continues to the floor, the new denser front displaces the existing diluted mixture thereby creating an upper layer. This process continues until the release ceases and no new layers are formed. Each successive layer entrains material from the layer above it.

As with the descending plume, entrainment theory was also used to calculate exchange of material across the interface of two layers, although adapted in respect of the stability of the lower layer. The properties of the lower layer were accounted for with the inclusion of the local Richardson number to produce the conventional formula for entrainment of dense clouds (e.g., Britter, 1998) (equation 374).

\[
\frac{dV_{e,j}}{dt} = \frac{A_j \cdot \alpha_{e,j} \cdot u_{fni,j}}{Ri_{e,j}}
\]  

(374)

where \(A_j\) is the area of the interface between the two layers. The review of Kumar et al (2003) identified from most previous studies on cloud dispersion that entrainment was dominated by the horizontal interfaces, so entrainment into any “vertical” moving front (when existing) was neglected. The local Richardson number \(Ri_{e,j}\) is defined in equation (375).

\[
Ri_{e,j} = \frac{g' \cdot h_j}{u_{fni,j}^2}
\]  

(375)

where \(g'\) is based on the local density. The local friction velocity \(u_{fni,j}\) is handled in a similar manner as with the descending plume. Following the discussion for descending plume calculations, \(u_{fni,j}\) for the lower/upper layer interface also employed equation (362), where the velocity gradient \((du/dh)\) is the difference between layer centre velocities with respect to distance (height) across the boundary (from equation 376). The local centreline velocity was based on interpolation between \(u_{sprd,j}\), \(\bar{u}_{lm}\) and zero velocity at the ceiling. However, since the spreading plume flow was assumed to be counter to the room airflow (due to return airflow back to the AHU), a combined velocity is employed \((\bar{u}_{lm} + u_{sprd,j})\). With this, interpolation of velocity
relative to height was achieved by using a simple power-law equation (376) (as for wind speed relative to height).

\[ u(h) = a \cdot (h_{Rm} - h)^n \]  

(376)

where \(a\) and \(n\) are determined from \(u = 0\) at \(h = h_{Rm}\), \(u = \bar{u}_{Rm}\) at \(h = h_d\), and \(u = \bar{u}_{Rm} + u_{syed,j}\) at \(h = 0\), as illustrated in Figure 150. Although using \(h_d\) as a reference height implies that the air discharge centreline should be the corresponding velocity, such high air speeds would only be present within a localised region; \(\bar{u}_{Rm}\) is considered to represent the average lateral air speed. In the case of a non-horizontal directed discharge duct, \(h\) is calculated as the mean height of the discharge centreline across the room – or the average height of the lowest layer \(h_j\), whichever is the highest.

A similar approach was validated experimentally by Brown et al (1997) for unbounded (outside) releases. It is known that the density of layers influence the velocity profile and this is handled comprehensively by Sykes et al (1999). The effect was seen to be minor, so it has been neglected here.

\[ \text{Figure 150: Illustration of room velocity profile} \]

Evaluation of entrainment coefficient for a spreading plume or cloud \((\alpha_c)\) within the context of transient or "static" stratified layers formed by a small release in a confined space was not found in the literature; most addressed large-scale outside releases. For the surface of a dense cloud, Gopalakrishnan and Sharan (1997) reported \(\alpha_c\) to be proportional to the change in cloud mean density as it disperses over time. Tickle (1996) considered dispersion of a high density gas.
travelling down a slope, and by correlating a model with measurements found $\alpha_e$ ranged from 0.026 – 0.045. Baines and Turner (1969) and Etheridge and Sandberg (1988) proposed $\alpha_e = 0.04$ to 0.05 for a plane or spreading plume under quiescent conditions. Britter (1988) summarised various correlations for $\alpha_e$, again determined from large outdoor releases. These were considered for use in the model and their generalised forms are presented as equations (377) and (378).

$$\alpha_e = a(\alpha_e) \cdot R_{ic}^{-1}$$  \hspace{1cm} (377)

$$\alpha_e = a(\alpha_e) / (1 + b \cdot R_{ic})$$  \hspace{1cm} (378)

As cited by Britter (1988), for equation (377) Kanatha et al gives $a(\alpha_e) = 5$, from Lofquist $a(\alpha_e) = 1.4$, and McQuaid, $a(\alpha_e) = 2.85$ (although in calculating $R_{ic}$, McQuaid has the reduced gravity as a function of mean density, $g' = g \cdot (\bar{\rho} / \rho_a - 1)$, instead of the conventional formulation). For equation (378), Havens and Spicer give $a(\alpha_e) = 0.4$, and $b = 0.125$. The various constants were derived under differing ranges of $R_{ic}$ and for different release conditions, such as outdoor ground source plume and outdoor elevated plumes; none of these were derived for the present indoor release case, so each were assessed within the model. Subsequently, equation (378) was selected, but $a(\alpha_e)$ was adjusted to suit the empirical data for the indoor case. Lastly, equation (377) produced instability within the overall model when applied to the wide range of $R_{ic}$ values normally experienced.

The volume and mass balances for each element within each layer was handled consistent with the descending plume, so equations (366), (367), (368) and (369) also apply for the spreading plume, as with the calculation of the exchange parameter, $\beta$. In equation (366), $C_{surj}$ represents the concentration within the element above the interface. However, because the presence of mechanical (extract) ventilation or infiltration may remove material from the room an additional term is required to account for an overall loss of refrigerant mass. Assuming that room internal pressure is maintained, the volume of air removed will be replenished (with air containing no refrigerant), so the volume based equation (371) still holds. On the other hand, as air is removed from the room there is a net loss of refrigerant mass, so equation (372) was adapted accordingly (equation 379).

$$\frac{dm_{r,\text{out},i}}{dt} = \frac{dm_{r,\text{in},j}}{dt} + \frac{dm_{r,e,\text{in},j}}{dt} - \frac{dm_{r,e,\text{out},j}}{dt} - \sum_{k=1} \omega_{r,j,k} \cdot \frac{dm_{r,v,k}}{dt}$$  \hspace{1cm} (379)

361
where \( dm_{r,v,k} / dt \) is the mass flow of the refrigerant out of each element due to the particular type of ventilation (where the subscript \( k \) corresponds to the types in Section 2.1), denoted by (equation 380).

\[
\frac{dm_{r,v,k}}{dt} = \frac{C_{Rm}(t)}{t} \cdot \frac{dV_{r,v,k}}{dt} \tag{380}
\]

In using the term for mean room concentration, equation (380) assumes that the ventilation removes a homogenous gas/air mixture, an approach adopted here for simplicity. \( \omega_{j,k} \) in equation (379) represents the mass fraction of material removed from each element, and is simply assumed to correspond to the mass fraction of the given element to the entire content of refrigerant within the room at any given time (equation 381).

\[
\omega_{j,f}(t) = \frac{m_{r,j}(t)}{\sum_j m_{r,j}(t)} \tag{381}
\]

The output from equations (371) and (379) provide values of volume and mass for inclusion in the calculation of \( \bar{V}_F \) (equation 347) and \( \bar{m}_F \) (equation 348).

### 3.4.5 Cloud decay model

At the end of the release, if any \( C_{out,j} \) are above the LFL then the decaying cloud model was employed, since a flammable volume still exists. The cloud decay model was similar to that for the spreading plume in terms of discretisation and entrainment equations. At cessation of the release, the descending plume was assumed to "disappear", and the remaining layers of the spreading plume become "static", and no new layers are formed. Although there is no longer horizontal flow across the floor, a velocity gradient still exists due to \( u_{Rm} \), so entrainment continues. The rate of entrainment is again calculated with the appropriate equations (366, 367, 368 and 369), although adapted in two ways. First the velocity profile is re-evaluated for \( u_{f,j} = 0 \), since lateral movement of the mixture is assumed to have ceased. Second, as mentioned in Section 3.4.1, it was not possible to calculate \( \bar{C}_f(t) \) from the correlations, so calculation of \( \bar{C}_{f,calc}(t) \) required \( \sigma(\alpha_c) \) to be fixed at the value determined for the final time step of the spreading plume (i.e., at \( t_r \)). This produced comparable results to the approach of Gopalakrishnan and Sharan (1997), where \( \alpha_c \) is extrapolated over time according to the change in local density. Since horizontal flow within the cloud has ceased, each parcel remains at the same lateral position as from time \( t_r \). The calculation sequence continued until all \( C_{out,j} < \text{LFL} \).
The main output from the cloud decay model is calculation of volume and mass within the elements – again using equations (371) and (379) – for inclusion in general equations for determining $\overline{V}^F$ (equation 347) and $\overline{m}^F$ (equation 348), except where $\bar{C}_{\text{f,calc}}$ is based on fixed $\alpha(a_c)$, and no further refrigerant is added to the layers.

3.4.6 Airflow models

Characterising room airflow is important for dispersion modelling, and this section summarises determination of $\bar{u}_{\text{Rm}}$ for the different forms of airflow. From Section 2.1, the four sources of airflow under consideration are: unit airflow, mechanical ventilation, infiltration, and thermal convection. Whilst airflow from infiltration and thermal sources are independent of the operation of the refrigerating system, unit airflow and mechanical ventilation can be linked to the occurrence of a leakage. Modelling of airflows within rooms can be very complex, but for the purposes of this study, a relatively simplistic approach was be taken, and models that are more complicated can be applied at a later stage if necessary. In addition to the airflow types listed above, movement of people within an environment increase air speeds, but because of the complexity of this problem, it was neglected.

Unit airflow

For airflow from RAC equipment, volume flow rates are specific to the equipment being considered. Generally, airflow rate is a function of the thermal load ($Q$), typically $\dot{V}_{\text{unit}} \approx 0.03 \cdot Q$ to $0.3 \cdot Q$, depending upon its function, as for an evaporator or condenser. Reference to Table 40 provides some examples of selected equipment.

Mechanical ventilation

Mechanical airflow rate is governed by the application and may be used for a number of purposes including heat and odour removal, or extraction of leaked refrigerant. (Note that calculation of $\bar{u}_{\text{Rm}}$ for mechanical ventilation using extract vent only was treated as if it were infiltration because air speed is dictated by the room fabric leakage, and not on inlet vent area.) Air speed from mechanical ventilation that employs a specific inlet duct was calculated as with unit airflow and $A_i$ is the duct area (rather than the outlet duct). Specifically, mechanical ventilation for buildings other than dwellings requires a minimum airflow, which is 5 l s$^{-1}$ per person or 3 l s$^{-1}$ per m$^2$ in the UK (BS 5925: 1991). Alternatively, refrigeration safety standards provide options
for machinery rooms, including a flow rate based on refrigerant charge \( V_{\text{mech}} = 0.014 \cdot m^2 r^{2/3} \) or a nominal \( \Lambda_{\text{mech}} = 10 \text{ h}^{-1} \) (EN 378: 2000).

**Infiltration**

Infiltration occurs in all buildings due to air leakage through fabric, caused by an imposed pressure difference \( \Delta p \) between the internal and external environment, and calculation of infiltration rate was detailed in Section 2.5. For an imposed \( \Delta p_{\text{tot}} \) other than the standard reference \( \Delta p_{50} = 50 \text{ Pa} \), infiltration rate was calculated with equation (382). Note that \( \Lambda_{50} \) is halved on the assumption that inflow occurs through half of the room fabric surface, instead of the total surface applicable to when the room is pressurised to determine \( \Lambda_{50} \).

\[
\dot{V}_{\text{infl}} = \frac{\Lambda_{50}}{2} \left( \frac{\Delta p_{\text{tot}}}{50} \right)^{0.7} \frac{V_{\text{tot}}}{3600}
\]  

(382)

Combining equation (382) with the steady flow equation, the room air speed due to infiltration is obtained (equation 383).

\[
\bar{u}_{\text{infl}} = \dot{V}_{\text{infl}} \cdot \sqrt{\frac{2 \cdot 50 / \rho g}{\Lambda_{50} \cdot 3600 / 2} / c_d \cdot V_{\text{tot}}^{1/6}}
\]  

(383)

where the discharge coefficient \( c_d \) is 0.6. For infiltration, a survey by Orme et al (1998) provided \( \Lambda_{50} \) data for dwellings and non-domestic buildings in several countries. In the UK \( \Lambda_{50} \) ranged from 3 \text{ h}^{-1} to 30 \text{ h}^{-1}, giving a statistical mean value of 14 \text{ h}^{-1}. Using equation (383) and the data within the example calculations in Section 2.5 (Figure 101), the range of \( \Delta \Lambda_{\text{infl}} \) for the specified room produce \( \bar{u}_{\text{infl}} \) of 0.015 to 0.03 m s\(^{-1}\) (for infiltration only).

**Thermal convection**

To evaluate airflow due to thermal sources, an approach from Etheridge and Sandberg (1996) was adopted. Assuming an equal heat output per object, the equivalent volume flow rate of warm air per object is calculated from equation (384).

\[
V_{\text{therm}} = 2.23 \cdot \sqrt{\pi} \cdot (\alpha \cdot (h_{\text{tot}} - h_{\text{therm}}))^{5/3} \cdot (2.8 \times 10^{-5} \cdot Q^{1/3})
\]  

(384)

where \( Q \) is the average heat input into the room per object, \( h_{\text{therm}} \) is the height of the object, and \( \alpha = 0.083 \). To calculate \( \bar{u}_{\text{therm}} \), the discharge area in equation (317) is substituted by the effective area of all thermal sources: \( N_{\text{therm}} \times A_{\text{eff}} \), where \( N_{\text{therm}} \) is the number of items. The value of

364
$A_{efr}$ was found to be $0.56 \text{ m}^2$ per object based on the measured air velocities in test no. 43. In practice, the number of thermal sources depends on the occupants and purpose of the room (e.g., an office with occupants, PCs, monitors and other electronic equipment).

**Multiple sources of airflow**

Where airflows originate from different sources simultaneously, RMS values of $\bar{u}_{Rm}$ were used as Etheridge and Sandberg (1996) (equation 385). Also the correlations in Section 3.3 require a corresponding height for the discharged air, a $\bar{u}_{Rm}$-weighted RMS was used to determine equivalent $h_d$ (equation 386).

$$\bar{u}_{Rm} = \sqrt{\sum_k \bar{u}_{Rm,k}^2} \quad (385)$$

$$h_d = \sqrt{\sum_k \left(h_d \cdot \bar{u}_{Rm,k}\right)^2 / \sum_k \bar{u}_{Rm,k}^2} \quad (386)$$

**3.4.7 Model verification**

Two separate exercises were carried out to verify the dispersion model. These were evaluating sensitivity to the number of elements and consistency with measurements.

**Table 48: Input details for dependence of flammable volume-time on number of elements**

<table>
<thead>
<tr>
<th>Run number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Release mass (kg)</td>
<td>0.50</td>
<td>1.50</td>
<td>1.50</td>
<td>1.50</td>
<td>0.50</td>
<td>1.50</td>
</tr>
<tr>
<td>Room area (m$^2$)</td>
<td>20</td>
<td>45</td>
<td>45</td>
<td>45</td>
<td>20</td>
<td>45</td>
</tr>
<tr>
<td>Room height (m)</td>
<td>2.5</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>2.5</td>
<td>3</td>
</tr>
<tr>
<td>Leak time (s)</td>
<td>240</td>
<td>240</td>
<td>240</td>
<td>500</td>
<td>500</td>
<td>500</td>
</tr>
<tr>
<td>Release height (m)</td>
<td>1.5</td>
<td>1.5</td>
<td>1.5</td>
<td>0.5</td>
<td>0.5</td>
<td>2</td>
</tr>
<tr>
<td>Unit height (m)</td>
<td>1.5</td>
<td>1.5</td>
<td>2.5</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Airflow (m$^3$ s$^{-1}$)</td>
<td>0.014</td>
<td>0.038</td>
<td>0.113</td>
<td>0.131</td>
<td>0.049</td>
<td>0.019</td>
</tr>
</tbody>
</table>

The number and dimensions of the elements within the respective sub-models was dictated by the choice of radial elements, $N_{dr}$. Conventionally, the preferred number of elements is obtained by comparing outputs for an increasing number of elements until the successive change is negligible. For this exercise six different sets of input conditions were chosen (listed in Table 48, plus...
general inputs: R290, airflow only from horizontal discharge angle, 0.12 m² duct area). From these, a range of total flammable volume-times \((V^F t^F)\) was generated for \(N_{dr}\) from 10 to 250.

Figure 151 shows the results for the six different conditions, as a function of increasing \(N_{dr}\). In general, \(V^F t^F\) changed erratically – up to about ±20% of the average value – until \(N_{dr} \approx 50\). By \(N_{dr} \approx 100\), the variation in \(V^F t^F\) had smoothed and after \(N_{dr} \approx 150\), \(V^F t^F\) remained approximately constant. Between \(N_{dr} \approx 150\) to 250, the deviation was typically within ±1% of the final values, although a little higher for run 6, (±3%). Compromising calculation time and desired precision \(N_{dr} = 150\) was considered appropriate.

![Figure 151: Dependence of number of radial elements on flammable volume-time](image)

Consistency of the output with measurements was assessed by comparing concentration-height profiles. This is relevant since flammable volume has a strong dependence on the vertical concentration profile. Comparing calculated and measured \(C_f(t)\) was not necessary because the model was already calibrated against the empirical correlations. Rather than checking against the entire set of experiments, five were selected to represent a wide range of conditions:

- No airflow (Figure 152) – Test no. 9
- Low airflow (Figure 153) – Test no. 28
- Reference case (medium airflow) (Figure 154) – Test no. 11
- Low release height (Figure 155) – Test no. 20
- Low release and air discharge height (Figure 156) – Test no. 21
The results of the comparisons are shown as concentration-height profiles at 120 s and 210 s (240 s for test no. 9) from the start of the release. The results from the model are shown as dashed lines, and the measurements are solid lines.

Overall, calculated concentration-height profiles were generally consistent with measurements, particularly at the floor and 1 m. The greatest differences occurred at the ceiling, where the predicted concentrations were always around half the measured values, although offset by slightly higher floor concentrations. This was possibly because the model does not directly account for entrainment of the release into the return airflow, which is then discharged to high level. However, because there was only one sampling point at 3 m, the measurement may be misrepresenting the average concentration at that height. Given this uncertainty, the deviation in concentrations at 3 m was not considered a significant problem. In addition, the concentrations at ceiling height were always well below the LFL, so $C_F$ would not occur in that region anyway.
Calculated (or interpolated) concentrations at 1 m were sometimes lower than the measurements within the initial period of the release. Towards the latter part of the release, calculated and measured concentrations converge. In general, these comparisons provide confidence in the accuracy of the model.

### 3.4.8 Application

The model was used to provide an insight into the dispersion process, and how certain variables affect its output. The results were used initially to describe the behaviour of a typical release, and secondly to illustrate the effect of selected variables on $V_F t_F$. The main variables that were considered are: mean room air speed, released mass, release time, post ventilation time, room area, release height, and air discharge height.
Behaviour of modelled dispersion

The behaviour of the dispersion of a release was analysed with an example – a 1 kg release of R290 over 250 s from a height of 2 m into a 3 m × 40 m² room, with an airflow rate of 2 h⁻¹ room air changes (giving \( \bar{u}_{m} = 0.08 \text{ m s}^{-1} \)), also discharged from 2 m in a horizontal direction.

Figure 157: Concentration within individual layers

Figure 158: Cumulative volume of individual layers

Figure 157 shows how concentrations vary between layers over time. The square data points illustrate the development of \( \bar{C}_f \) up to \( \bar{C}_{f_{\text{max}}} \) at each time interval (i.e., \( t_{\text{grad}} \)) for each layer, and how the concentration in the layer, say layer (10) reduces with time as weaker mixture from the layer above, say layer (9) is entrained. It is seen that at the start of the release, the material within the layers tended to mix rapidly within about 30 s, taking up to about 60 s towards the latter part of the release due to a lessening density difference between the descending and spreading plume. Dilution of the final two layers (10 and 11) created toward the end of the release occurred less rapidly and concentration tended to decay slowly over time. This was because of floor level spreading velocity becoming absent, causing a drop in friction velocity.

Also in Figure 157 is the mean concentration of the flammable elements – \( \bar{C}_F = \bar{m}_F(t)/\bar{V}_F(t) \) – which falls from about stoichiometric concentration (~ 0.07 kg m⁻³) towards the LFL. The downward trend of \( \bar{C}_F \) is expected since more of the elements approach the LFL over time, whilst accumulating material to produce a larger volume. Throughout the flammable time, the
proportions of flammable to total mixture volume and the flammable mass to release mass both vary in a similar manner as $C_{V\text{F}}$.

Figure 158 illustrates the cumulative change in layer volume, where the upper line (1) represents the total volume of the entire refrigerant/air mixture. It is seen that as each layer was created, the volume of the layer above it approaches zero, as its contents were gradually entrained into the next layers beneath. This entrainment produced dilution in successive layers as described for Figure 157. Once the release stopped the existing layers cease to be entrained into lower layers and instead increase in volume as a result of continued entrainment from the upper part of the room. Eventually layer (11) will completely fill the room with the mixture, as its concentration approaches $C_{Rm}$.

**Influence of selected variables**

The influence of certain variables associated with the design and installation of equipment on $V^F t^F$ was addressed, and calculations were based on a set of reference conditions, similar but not identical to the reference test no. 10: $m_r = 1.0 \text{ kg}$, $A_{Rm} = 40 \text{ m}^2$, $h_{Rm} = 3 \text{ m}$, $h_r = 2 \text{ m}$, $h_d = 2 \text{ m}$, $A_d = 0.115 \text{ m}^2$, $\Lambda = 1 \text{ h}^{-1}$, $\Theta = \pi / 2$, $t_d = \infty$, and the refrigerant was R290. It is noted that low airflow was selected so as to emphasise the effects of other variables that may not otherwise create a noticeable change in $V^F t^F$.

![Figure 159: Effect of released mass on $V^F t^F$](image1.png)  ![Figure 160: Effect of room area on $V^F t^F$](image2.png)

Figure 159 shows the influence of released mass on $V^F t^F$ from two respects: where the room area was constant whilst mass was varied, and where room area was simultaneously increased...
with mass so that \( \bar{C}_{Rm} \) is constant (whilst adjusting \( A_d \) to maintain \( \bar{u}_{Rm} \)). With constant room area, an increase in mass produced an almost exponentially larger \( V^{F}t^{F} \), although it was not until \( m_{r} > 0.8 \text{ kg} \) that the gradient changed and much greater values were achieved; the change of gradient coincided with \( \bar{C}_{f,max} > \text{LFL} \). On the other hand, if room area is correspondingly increased, the change in \( V^{F}t^{F} \) was almost linear with mass and the transition of \( \bar{C}_{f,max} \) to above the LFL (0.5 kg) does not demonstrate the change in gradient seen with fixed room area.

Similarly, the effect of changing room area (and \( A_d \)) with fixed mass is presented in Figure 160, and included again for comparative purposes is the data for constant \( \bar{C}_{Rm} \). Enlarging room area causes \( V^{F}t^{F} \) to decrease exponentially until about 50 m\(^2\). At this point, the gradient changed and \( V^{F}t^{F} \) became relatively constant with additional area. Again, this transition occurs when \( \bar{C}_{f,max} \approx \text{LFL} \). As with the previous example, maintaining fixed \( \bar{C}_{Rm} \) by simultaneously increasing mass produced a linear rise in \( V^{F}t^{F} \) with room area.

Air movement is created by the release displacing air, and forced airflow from the AHU. Neglecting the contribution of the release, the influence of unit airflow – in terms of \( \bar{u}_{Rm} \) – on \( V^{F}t^{F} \) was addressed (Figure 161). Also included was data for an upward and downward directed air discharge. For horizontal air discharge (\( \theta = \pi / 2 \)), there was a rapid reduction in \( V^{F}t^{F} \) until \( \bar{u}_{Rm} \) reaches approximately 0.05 m s\(^{-1}\), at which point a more gradual fall occurs.
After about 0.1 m s\(^{-1}\), the effect of air speed seems to have negligible impact on \(V^F\). This was consistent with the behaviour of \(C_{f,max}\) found in the experiments. Where the air discharge was directed downwards (\(\theta = \pi / 3\)), \(V^F\) was distinctly lower, particularly at lower \(\bar{u}_{rms}\). Again, the same trend was observed, where the reduction in \(V^F\) with \(\bar{u}_{rms}\) became more gradual, and then almost constant above \(\bar{u}_{rms} = 0.06\) m s\(^{-1}\). Similarly, an upward directed discharge also produced similar behaviour but required higher \(\bar{u}_{rms}\) to achieve the same effect. In all three cases, the initial transition from high to low \(V^F\) occurred when \(C_{f,max} \approx \text{LFL}\). The second transition concurred with the mean concentration at the base of the plume falling below \(\text{LFL}\). Thus the entire flammable volume occurred only within the plume, and thereon higher \(\bar{u}_{rms}\) had little influence on \(V^F\).

\(V^F\) for a range of release times (210 s upwards) is shown in Figure 162, where airflow was continuous (\(t_d = \infty\)). In addition, results were also generated for three different post-ventilation periods (which could also be interpreted as time for airflow termination due to equipment controls). With continuous airflow, increasing \(t_r\) showed \(V^F\) to fall exponentially until \(t_r > 300\) s, when longer release times had no further effect. Termination airflow at 315 s and 210 s caused a minor increase of \(V^F\) with a 210 s release, because the flammable cloud persisted for longer. But with longer \(t_r\), the difference disappeared because \(\bar{u}_{rms}\) has a lesser influence on dispersion. For \(t_d = 105\) s, a distinct rise in \(V^F\) was observed because forced airflow was absent for half of the release, so thereon only minimal \(\bar{u}_{rms}\) was present to disperse the mixture to below \(\text{LFL}\). For longer \(t_r\), with \(C_{f,max} < \text{LFL}\), so the \(t_d\) ceased to influence \(V^F\).

Figure 163 illustrates changes in \(V^F\) with release height (hollow data-points), but also where air discharge height was equal to the release height (\(h_d = h_r\), filled data-points). To elaborate on the effect of these parameters, additional data for a higher airflow was included. For \(h_d = 2\) m, lowering \(h_r\) produced a steady rise in \(V^F\), which applied to both \(\bar{u}_{rms}\), although of course \(V^F\) was always smaller for the higher \(\bar{u}_{rms}\). As found previously, the gradient of \(V^F\) flattens as \(C_{f,max} < \text{LFL}\), and when \(\bar{u}_{rms} = 0.08\) m s\(^{-1}\), a second step change was observed corresponding to the mean plume concentration at floor level being < LFL. When \(h_d = h_r\), a
different tendency was observed; for $\bar{u}_{Rm} = 0.03 \text{ m s}^{-1}$ a small $V^{F}\tau^{F}$ occurs at low $h_{d}$ and $h_{r}$, but elevating $h_{d}$ and $h_{r}$ increases $V^{F}\tau^{F}$ until about 2 m where above this $V^{F}\tau^{F}$ began to reduce. At low $h_{d}$ and $h_{r}$, the high local airspeed caused rapid mixing within the developing cloud near the floor thereby offsetting the high $\overline{C}_{f,max}$ that would normally be associated with a low level release. As $h_{d}$ and $h_{r}$ were raised the high discharge velocity became less effectual. Above 2 m the release occurred from a sufficiently high level to offset the absence of airflow near to the floor resulting in a fall in $V^{F}\tau^{F}$. With $\bar{u}_{Rm} = 0.08 \text{ m s}^{-1}$, the airflow was always sufficient to ensure $\overline{C}_{f,max} < \text{LFL}$, so $V^{F}\tau^{F}$ remained small regardless of $h_{d}$ and $h_{r}$.

![Figure 163: Effect of release height and air discharge height on $V^{F}\tau^{F}$](image)

The above analysis has revealed several important points. Depending upon the conditions $V^{F}\tau^{F}$ varies widely, and small changes in certain variables can produce a dramatic effect. Altering one single variable that affects $\overline{C}_{f,max}$ will also influence $V^{F}\tau^{F}$ in the same direction but was rarely directly proportional. Where the conditions are such that $\overline{C}_{f,max} < \text{LFL}$, $V^{F}\tau^{F}$ “jumps” to smaller values and was thereon subject to minimal changes.
SECTION 4: CONSEQUENCE ANALYSIS

4.1 Selection of consequences

The two selected forms of direct consequence of ignition were thermal intensity ($I$), which can be used to quantify degrees of burns to people (or likelihood of secondary fire), and overpressure ($\Delta p^o$) that causes damage to building structure and injury to people. An outcome of high $I$ and/or $\Delta p^o$ includes damage to buildings, their contents, injury to people and fatalities.

As mentioned in Section 2.1, the outcome is limited to fatality of room occupants, due to either consequence. A further consequence of ignition is a secondary fire, either caused by direct impingement of a sustained jet flame on a combustible material, or ignition of materials due to sufficient $I$ (termed "flashover"). With regards to an ignited jet or plume, its velocity (at $C_F$) is generally greater than flame speed and so burning is unlikely to be sustained. The necessary $I$ for flashover varies considerably with construction and furniture materials. Data collated by Babrauskas et al (2003) showed flashover to occur over a range of $I = 125 - 4100$ s (kW m$^2$)$^{1/3}$, including different foam insulation, wood and gypsum based materials. Other easily combustible materials such as loose paper or hair may ignite at even lower $I$. ADL (1998) uses different probabilities of a secondary fire as a function of the frequency of the prior consequence: from a jet fire $f^* \times 10^{-2}$, from a cloud fire $f^* \times 10^{-3}$, and $f^* \times 10^{-2}$ from an explosion. The choice of these factors was not explained and they seem to be arbitrary. Therefore, without additional data to determine the probability of the presence of various combustible materials, it would not be meaningful to estimate the likelihood of secondary fires from flashover. On this basis it was assumed that the frequency of a secondary fire equates to ignition frequency, for simplicity. The probability of fatality directly caused by a secondary fire is ignored.

Models to calculate $I$ and $\Delta p^o$ as a result of igniting flammable material within a room are detailed below.\(^{28}\) These models demand flammable property data of refrigerants such as heat of combustion, and these are normally quoted for stoichiometric concentrations. However, the results of the dispersion model demonstrated that the mean concentration within flammable volumes were rarely at stoichiometric, generally lower. Flammable properties at concentrations

---

\(^{28}\) The overpressure model applies to a flammable mixture in an (empty) room. The case of ignition of a mixture within a compartment (such as a refrigerated cabinet, or condensing unit housing) has been ignored for simplicity. However, recent incident investigations (e.g. Blades and Colbourne. 2004) demonstrated that this is a possible outcome, and should be addressed in future studies.
other than stoichiometric differ considerably and therefore for the models described below properties at a mean concentration corresponding to \( \bar{m}^F / \bar{V}^F \) were employed. Data for flammable properties at stoichiometric concentrations are provided in Appendix 4, along with methods used to approximate non-stoichiometric properties.

4.2 Thermal radiation model

When a flammable substance burns, the heat energy is partially given off as thermal radiation. A “dose” of thermal radiation may be sufficient to cause burns to individuals nearby. The severity of burns is a function of heat flux and its duration, and this dose is termed thermal intensity (equation 387).

\[
I_{occ,i} = q_{occ,i}^{4/3} \cdot t_{flam}
\]  

(387)

where \( t_{flam} \) is the time taken for complete combustion, as calculated below for overpressures (equation 395). The heat flux radiated onto an occupant (\( q_{occ} \), in kW m\(^{-2}\)) was calculated with a correlation for a “fireball” from Hymes et al (1996) (equation 388). Other correlations available for “pool fires” were not considered relevant, given the assumption of vapour-only releases as discussed previously.

\[
q_{occ,i} = \Gamma \cdot \left( \frac{\Delta H(CF) \cdot \bar{m}_F^F}{t_{flam}} \right) \cdot \frac{\zeta}{4 \cdot \pi \cdot z_{occ}^2}
\]  

(388)

where the incidence ratio \( \zeta \) is influenced by the size and shape of the flammable volume: \( \zeta = 0.2 \) for flames from a plume, and \( \zeta = 0.3 \) for fireballs (Hymes et al, 1996). \( \Gamma \) is the radiative fraction of the fuel, or the proportion of energy from the burning material that translates to thermal radiation; the data presented by Beyler (2002), gives \( \Gamma \approx 0.3 \). \( \Delta H(CF) \) is the HOC at the corresponding mean concentration based on the average flammable mass over the flammable time, (\( \bar{m}_F^F / \bar{V}^F \)) and \( t_{flam} \) was the time for combustion. \( z_{occ} \) is the distance between the fire and the occupant which is impractical to estimate, but assuming occupants are evenly distributed, \( z_{occ} = 0.5 \cdot \sqrt{A_{occ} / \pi} \), where \( A_{occ} \) is the average area occupied per person. Normally, a term for heat absorption in the atmosphere (by water and carbon dioxide) is included in equation (388), but since the distances are relatively small, this factor was neglected.

At around \( I_{occ} = 115 \) s (kW m\(^{-2}\))\(^{4/3}\) pain will be felt on exposed skin, becoming more severe with higher \( I_{occ} \). Above \( I_{occ} = 1050 \) s (kW m\(^{-2}\))\(^{4/3}\) fatalities are expected and death is almost
guaranteed at 6500 s (kW m^2)^{4/3}. The probability of fatality associated with a dose of thermal radiation was calculated using data from Hymes et al (1996), by equation (389), when 0 < P_{\text{fatal}} < 1.

\[ P_{\text{fatal}}(I_{\text{occ}}) = a + b \cdot \ln I_{\text{occ}} \]  

(389)

where \( a = -3.79 \) and \( b = 0.54 \).

### 4.3 Overpressure model

Modelling an explosion from ignition of a flammable mixture is complex, and the most accurate methods typically employ CFD codes. As with estimation of flammable quantities, use of CFD was not considered suitable so a more simplistic lumped approach was used.

When a flammable mixture is ignited, the exothermic reaction raises the temperature of the combustion products, translating to an increase in pressure and/or volume. Under ideal conditions, where ignition is at the centre of a spherical stoichiometric mixture, the maximum pressure or volume can be reliably determined from the gas law (Harris, 1983). Transposing the gas law, a non-dimensional expansion ratio (\( E \)) is obtained (equation 390).

\[ E = \frac{N_{\text{mol}}^{\text{flam}}}{N_{\text{mol}}^{\text{vF}}} \cdot \frac{273 + T_{\text{flam}}}{273 + T_a} \]  

(390)

where \( T_a \) is the initial (ambient) temperature of the flammable volume, and \( T_{\text{flam}} \) is the maximum adiabatic flame temperature. \( N_{\text{mol}}^{\text{flam}} \) and \( N_{\text{mol}}^{\text{vF}} \) are the number of moles in the burned and unburned mixture, respectively. Assuming a confined (fixed) volume that is completely filled with the mixture, the maximum rise in pressure is the product of initial (atmospheric) pressure (\( P_{\text{atm}} \)) and \( E \). The pressure developed within the space (\( \Delta P^o \)) is the difference between the initial and maximum pressure (equation 391).

\[ \Delta P^o = P_{\text{atm}} \cdot (1+E) - P_{\text{atm}} \]  

(391)

Conversely, if the flammable mixture were in an unconfined volume (i.e., constant pressure) its volume after combustion would be equation (392).

\[ V_{\text{flam}} = V^F \cdot E \]  

(392)

Although for the situation under consideration, the flammable material is within a confined space, that \( V^F < V_{\text{flam}} \) implies both a pressure rise, and a volumetric change of the burning mixture.
within the room. As a result the overpressure within the room ($\Delta p_{Rm}$) will be less than equation (391). Assuming a fixed room volume, and substituting equation (391) into the gas law, $\Delta p_{Rm}$ was calculated from the ratio of mass of flammable mixture ($\bar{m}_F^{\text{R}} = m_F^w + m_F^g$) and total air mass plus refrigerant mass within the room ($m_{Rm,\text{tot}}$) (equation 393).

$$\Delta p_{Rm} = p_{\text{atm}} \cdot E \cdot \frac{\bar{m}_F^{\text{R}}}{m_{Rm,\text{tot}}}$$  \hspace{1cm} (393)

Harris (1983) and others show that the pressure and/or volume of a flammable mixture increase exponentially following ignition until a maximum pressure ($p_{\text{max}}$) is reached, and that development of $\Delta p^o$ at time ($t$) after ignition follows the "cube-law" (equation 394).

$$\Delta p^o(t) = \frac{K \cdot p_{\text{atm}} \cdot U_{\text{flam}}^3 \cdot t^3}{V_{F}^{\text{R}}}$$  \hspace{1cm} (394)

where the constant $K$ is dependent upon the characteristics of the explosion and was found by rearranging equation (394), and setting $\Delta p^o = \Delta p_{\text{max}}^o = (p_{\text{max}} - p_{\text{atm}})$ and $t$ as the combustion time, $t_{\text{flam}}$ was estimated for the flame travel distance ($z_{\text{flam}}$) from the point of ignition to the point at which combustion ceases (equation 395).

$$t_{\text{flam}} = \frac{z_{\text{flam}}}{U_{\text{flam}}^\text{lum}}$$  \hspace{1cm} (395)

If ignition occurs at the centre of the volume, $z_{\text{flam}}$ is the mean distance from the centre to the boundary of $V_F^\text{R}$, or alternatively if ignited at some distant point, say in the room corner, $z_{\text{flam}}$ is the furthest distance between the boundaries of $V_F^\text{R}$. The burning velocity ($U_{\text{flam}}$) is based on laminar flame velocity ($U_{\text{flam}}^\text{lum}$) which is a property of the gas, and for HCs it is around $0.4 \text{ m s}^{-1}$ (Appendix 4). With a rise in pressure or volume, the flame front accelerates and under ideal conditions, flame speed increases proportionally with $E$. However, the influence of room geometry, obstacles and air movement creates turbulence which further accelerates the flame front. Therefore, an additional turbulence factor ($T$) was included in the determination of burning velocity (equation 396).

$$U_{\text{flam}} = U_{\text{flam}}^\text{lum} \cdot E \cdot T$$  \hspace{1cm} (396)

Correlations exist for estimating $T$ for "congested" situations (e.g., chemical plant), but a literature review revealed nothing for relatively empty spaces such as occupied rooms. Instead, a
general value of $T$ was obtained by adjustment until optimum fit of model results with published experimental data was obtained. The average ratio $\frac{\Delta p_{\text{calc}}}{\Delta p_{\text{meas}}}$ approached unity when $T = 2.94$, and this also corresponded with the best fit ($r^2 = 0.83$). Conventionally, $T$ is defined as the ratio of the turbulent flame front area, to that of a laminar (and spherical) flame front.

Equation (393) is valid for a completely sealed enclosure, but realistically, most occupied spaces have paths in the room fabric through which developed pressure will escape. As $\Delta p_{\text{pm}}$ rises, material may be vented from the room, and this occurs in two stages. Initially, any openings such as gaps in the room fabric, ventilation ducts, etc., allows the higher pressure gas to escape from the room. Secondly, certain barriers that comprise the room construction may eventually give way at a given $\Delta p_{\text{pm}}$, thereby enlarging the venting area. In both cases, the exhausted material results in a lower rate of pressure rise. Using the steady flow equation (397), the mass of material vented through an opening of a given area ($A_v$) can be estimated, assuming that the velocity of the vented material ($u_v$) is less than sonic velocity ($u_{ss}$).

$$\frac{dm_v(t)}{dt} = c_d \cdot A_v \cdot \sqrt{2 \cdot \Delta p_{\text{pm}}(t) \cdot \bar{\rho}_v(t)}$$

(397)

where $\bar{\rho}_v$ is the density of vented gas, which may comprise room air, combustion products and gas/air mixture. For simplicity, $\bar{\rho}_v$ was calculated according to the mass fraction of the gas components at time $t$. Under certain conditions, the vented gas may approach $u_{ss}$, in which case equation (398) was used, i.e., $u_{ss} = u_v$.

$$\frac{dm_v(t)}{dt} = c_d \cdot A_v \cdot u_{ss} \cdot \bar{\rho}_v(t)$$

(398)

The overpressure required to blow out vulnerable barriers within the room fabric was determined using a method for explosion vent panels. Hochst and Leuchel (1998) conducted experiments on the blow-out of doors and windows, and found that such equations for explosion vent panels could be applied. Both Anthony (1977) and Razus and Krause (2001) reviewed a number of different approaches. In general, all studies report that the blow-out pressure is related to a number of specific parameters: fixing force of the vent panel ($P_{vp,fix}$), plane density of the panel ($W_{vp}$), burning velocity and a ratio of wall to vent area ($A_{wall} / A_{vp}$). Razus and Krause found that the correlation of Molkov provided the best results compared with experiments. An improved approach was developed by Molkov et al (2003, 2004) that suggested equation (399).
\[ \Delta p_{vp} = p_{vp,fix} + a \cdot (u_{\text{flam}} \cdot \sigma)^2 \cdot \frac{A_{\text{wall}} \cdot W_{vp}}{V_{Rm}^{1/3}} \] (399)

where the coefficient \( a = 0.00043 \). \( \sigma \) is a "deflagration outflow interaction" parameter, from equation (400) for HC-air mixtures, representing the inertia of the panel.

\[ \sigma = 1.75 \left[ \frac{1 + 10 \cdot V_{Rm}^{1/3} \cdot (1 + 0.5 \cdot \sqrt{Br})}{1 + p_{vp,fix} / p_{atm}} \right]^{0.4} \] (400)

where \( Br \) is the Bradley number (equation 401).

\[ Br = \frac{A_{vp} \cdot u_{sv}}{V_{Rm}^{2/3} \cdot u_{\text{flam,lam}} - (E - 1)} \] (401)

Thus, once \( \Delta p_{Rm} > \Delta p_{vp} \), the area \( A_{vp} \) becomes available for venting material. However, as gas exits the room \( \Delta p_{Rm} \) is affected, and this is influenced by the quantity of high temperature combustion products that no longer contribute to a pressure rise. So, throughout the combustion process the change in \( \Delta p_{Rm} \) was based on the total mass of material present in the room, and the corresponding change in average room temperature \( \bar{T}_{Rm} \) (equation 402).

\[ \Delta p_{Rm}(t) = \Delta p_{Rm}(0) \cdot \frac{m_{Rm,loc}(t) \cdot [273 + \bar{T}_{Rm}(t)]}{m_{Rm,loc}(0) \cdot [273 + \bar{T}_{Rm}(0)]} \] (402)

where \( (0) \) indicates initial conditions, and \( \bar{T}_{Rm} \) is based on the mass of each mixture, and assumed similar specific heats (equation 403).

\[ \bar{T}_{Rm}(t) = \frac{m_{Rm,loc}^{F}(t) \cdot [273 + T_{Rm}^{F}] + m_{\text{flam}}(t) \cdot [273 + T_{\text{flam}}] + m_{r,o}(t) \cdot [273 + T_{r,o}] - 273}{m_{Rm,loc}(t)} \] (403)

where \( m_{Rm,loc} \) is the sum mass of unburned flammable mixture \( m_{r,a}^{F} \), burned mixture \( m_{\text{flam}} \) and non flammable refrigerant/air mixture \( m_{r,o} \), whilst accounting for the proportion having escaped through vents (equations 404, 405 and 406). The mass of mixture burned by time \( t \), is the product of mixture density and volume obtained by the transposition of equation (394).

\[ m_{\text{flam}}(t) = \rho_{\text{flam}} \cdot K \cdot (u_{\text{flam}} \cdot \sigma)^3 - X_{\text{flam}}(t) \cdot m_{r}(t) \] (404)
\[
\bar{m}^F_{r,a}(t) = \left( m^F_{r,a}(0) - \frac{\rho \cdot u \cdot t}{E} \right)^3 - X^F_{r,a}(t) \cdot m_v(t)
\]

where \( X \) is the mass fraction of the given mixture within the room at time \( t \). As unburned flammable material is vented, the explosion ceases before the anticipated burning time (as equation 395), and therefore a revised \( t_{flam} \) was determined as \( \bar{m}^F_{r,a}(t) \to 0 \). This would be relevant in the calculations of \( I_{occ} \), reducing the intensity.

\[
m^F_{r,a}(t) = m^F_{r,a}(0) - X^F_{r,a}(t) \cdot m_v(t)
\]

As mentioned earlier, the model was tested against experimental data for vented explosions as cited in Razus and Krause (2001) for methane/air and propane/air mixtures, as well as small hexane clouds from DeHaan et al (2001). Hjertager et al (1998) also reported data for vented explosions from stratified layers, but no information on flammable quantities or distribution of concentrations was provided so it was not possible to utilise the information. Both measured and calculated \( \Delta p^o_{Rm} \) is shown in Figure 164, and all relevant data for the measurements are provided in Appendix 5. The different data-sets from Razus and Krause correspond to the different gases and room sizes by which they compiled the measurements.

Following the adjustment of \( T \) (to 2.94), the model predicted maximum \( \Delta p^o_{Rm} \) to within \( \pm 100\% \) for 40 of the 45 measurements, and within \( \pm 50\% \) for 31 of the measurements, which were
considered good agreements. As seen in Figure 164, three of the conditions caused the model to over predict by 4 – 6 times, which was inconsistent with the remaining data. It was concluded that this relatively simple model provided a sufficiently accurate estimation of overpressures for the purposes of this QRA.

Four examples of $\Delta p_{Rm}^o$ development over time are given in Figure 166, which were based on a flammable R290 mixture in a $40 \text{ m}^2 \times 3 \text{ m}$ room. By far the greatest $\Delta p_{Rm}^o$ is for a fully confined room that is entirely filled (120 m$^3$) with a stoichiometric mixture. By introducing a vent panel (equivalent to 5% or the total wall area), there is a major reduction in $\Delta p_{Rm}^o$ as the excess pressure is rapidly exhausted, noting also that the burn time is reduced because some of the flammable material was vented before it was burned. A more realistic situation is where a 24 m$^3$ mixture (at a concentration between LFL and stoichiometric) was ignited, and produced $\Delta p_{Rm}^o$ proportionally lower than the completely flammable room. Again, by introducing a vent panel (refer to right-hand scale), $\Delta p_{Rm}^o$ is further reduced; this example shows a temporary reduction in $\Delta p_{Rm}^o(t)$ at the point where the vent gives way, shortly after $\Delta p_{Rm}^o(t)$ began to rise again.

![Figure 166: Examples of room overpressure development](image)

To interpret the severity, any $\Delta p_{Rm}^o$ below about 3 kPa is unlikely to cause any physical damage to persons or property. Persons exposed to overpressure (alone) above about 250 kPa would begin to suffer fatalities. As $\Delta p_{Rm}^o$ approaches 450 kPa, death is almost guaranteed. Probability of fatality due to overpressures was calculated using data from Beyler (2001) in equation (407), when $0 < p_{fatal} < 1$. 381
\[ P_{\text{f used}}(\Delta \rho_{\text{Rm}}) = a + b \cdot \ln \Delta \rho_{\text{Rm}} \] (407)

where \( a = -3.62 \) and \( b = 0.76 \).

### 4.4 Application

Some of the results from the dispersion model were used to generate \( I_{\text{occ}} \) and \( \Delta \rho_{\text{Rm}}^o \), had the flammable mixture been ignited. Certain assumptions were made: \( z_{\text{flam}} \) was based on a central ignition point, an occupancy of 1 per 10 m\(^2\) of floor area, and initially the room fabric was resistant to overpressure (i.e., no venting). The results are shown in Figure 169 to Figure 168, where \( I_{\text{occ}} \) was indicated with solid data-points and \( \Delta \rho_{\text{Rm}}^o \) with hollow data-points.

![Figure 167: Effect of release mass with constant room size or constant \( \tilde{C}_{\text{Rm}} \) on consequence](image1)

![Figure 168: Effect of room area with constant release mass or constant \( \tilde{C}_{\text{Rm}} \) on consequence](image2)

Figure 167 and Figure 168 give the results for the effect of released mass (for constant room area and fixed \( \tilde{C}_{\text{Rm}} \)), and room area (for constant mass and fixed \( \tilde{C}_{\text{Rm}} \)), respectively. In Figure 167 it was seen that larger \( m_r \) always increased the severity of the consequence, although for fixed \( \tilde{C}_{\text{Rm}} \), raising \( m_r \) above about 1 kg produced little further change in \( I_{\text{occ}} \) and \( \Delta \rho_{\text{Rm}}^o \). Compared the resulting change in \( V^F \cdot t^F \) (Figure 159) the sensitivity of \( I_{\text{occ}} \) and \( \Delta \rho_{\text{Rm}}^o \) on room area seem to be less pronounced. This was attributed to the fact that whilst \( V^F \cdot t^F \) increases, it is only due to an extended \( t^F \) whereas the "instantaneous" \( \bar{m}^F \) (which dictates \( I_{\text{occ}} \) and \( \Delta \rho_{\text{Rm}}^o \)) becomes
almost constant. The same tendency of minimal change in $I_{occ}$ and $\Delta p_{Rm}^o$ at larger room areas was seen in Figure 168 where room size is varied (compared to the change in $V^{F_i}T^F$, Figure 160).

The effect of varying room airspeed is shown in Figure 169, where air discharge angles were upwards and downwards – the data for horizontal airflow was not included for clarity. In general it was seen that the results for both $I_{occ}$ and $\Delta p_{Rm}^o$ follow a similar pattern, which also follows $V^{F_i}T^F$ when compared with the results in Section 3.4.8. However, $I_{occ}$ appeared to be slightly more sensitive to the variation in $\bar{u}_{Rm}$ than $\Delta p_{Rm}^o$. Similarly, a downward directed air discharge ($\theta = \pi/3$) significantly reduces the severity of both consequences. Figure 170 provides similar data for different release height (where $h_d$ was constant at 2 m). Elevating the position of $h_r$ always resulted in a reduction of $I_{occ}$ and $\Delta p_{Rm}^o$. Although not included here, the results for $h_d = h_r$ also followed the trend found for $V^{F_i}T^F$.

![Figure 169: Effect of room airspeed on consequence](image1)

![Figure 170: Effect of release height and airspeed on consequence](image2)

Regarding thermal intensity, the most relevant independent factor is the distance $z_{occ}$ (as equation 388), which was considered to be a function of population density. Under any set of conditions, a more populated environment produced a rise in $I_{occ}$ proportional to the mean target distance to the power $\frac{1}{2}$ (as equation 387).
Throughout, the severity of the consequence was mostly proportional to $V F t F$ due to the dependency on $m F$. When explosions develop high overpressures, the values indicated above are unlikely to occur due to the venting phenomenon. For example, most occupied rooms are constructed with windows, doors and gaps in the fabric that could act to vent an explosion. To investigate the differences in consequence associated with venting, the conditions used to provide the results in Figure 169 to Figure 168 were used to recalculate $\Delta p r m$ when a number of vent situations were employed. Results are given in Figure 171 as the ratio of vented to unvented (or confined) $\Delta p r m$ for four different venting situations, using $m F$ as the reference variable. The different forms of venting were the presence of gaps in the fabric that normally lead to infiltration, an open door, and windows that occupy 3% and 8% of the rooms’ total wall area.

![Figure 171: Effect of various venting routes on reduction of overpressure](image)

With gaps corresponding to an infiltration rate of $\Lambda_{50} = 14 \text{ h}^{-1}$ (equivalent to a total vent area of about 0.1 m$^2$ in a 3 m $\times$ 40 m$^2$ room), $\Delta p r m$ was reduced by about 10% of that of the confined space. An open 2 m$^2$ door, reduced $\Delta p r m$ to approximately 10 – 40% of unvented $\Delta p r m$, but as $m F$ exceeded about 0.1 kg, the vented $\Delta p r m$ reduced to below 5%. In the case of windows, using typical values of $p_{wp, fix} = 4.5 \text{ kPa}$ and $W_{wp} = 7.5 \text{ kg m}^2$, the variation of vented $\Delta p r m$ was different from the open (door, infiltration) vents. An initial resistance is offered by the window fixing, so that regardless of the severity of the explosion, the presence of any sized window would not permit reduction in overpressure below that 4.5 kPa (corresponding to $m F \approx 0.05 \text{ kg}$, as indicated in Figure 171). Above this value, plus a minor addition associated with the inertia of the glass (panel), a considerable reduction in $\Delta p r m$ occurred. In the case of a large window area (8%
of the total wall surface), an exponential fall in $\Delta p_{Rm}^{\circ}$ was seen, until it approached a few percent of the confined $\Delta p_{Rm}^{\circ}$ at the largest $m^F$. Where a smaller window was present (3% of the total wall surface), a similar reduction occurred, but varied by up to 60% of the original value. This is due to the vent area being insufficient to relieve the majority of the developed pressure under certain circumstances.

Comparing the minimum $I_{occ}$ and $\Delta p_{Rm}^{\circ}$ to achieve $P_{fatal} > 0$, against the results in Figure 169 to Figure 168, showed that few conditions produced a consequence severe enough to cause fatality. Only 0.5% of the cases present a probability of fatality greater than zero, and these were due to thermal intensity only. If the occupancy density in the calculations were doubled (to 1 person per 5 m$^2$), then the number of situations for potential fatality rose to 8%. Were the severity of injury due to $I_{occ}$ considered, around 10% of the cases could lead to second-degree burns ($> 700 \text{ s (kW m}^{-2})^{4/3}$, Hymes et al, 1996). None of the calculated $\Delta p_{Rm}^{\circ}$ – with or without a means of venting – would lead to death. Were injury from $\Delta p_{Rm}^{\circ}$ considered (Beyler, 2001), lung damage ($> 85 \text{ kPa}$) could potentially occur in about 5% of the (unvented explosion) cases, and eardrum damage ($> 35 \text{ kPa}$) in around 30% of cases. Data is also available to evaluate the damage to parts of buildings (e.g., Harris, 1983).
SECTION 5: APPLICATION OF OVERALL MODEL

5.1 Uses for the QRA model

The individual components of the entire QRA model were detailed in previous sections. Each sub-model was linked into a single program module, and compiled in a sequence according to Figure 172. A single set of inputs were required, and shared amongst relevant calculations. Outputs were generated for several purposes: evolution of the overall risk showing the influence of each component, parametric assessment of key input variables, example calculations for selected RAC equipment, and lastly derivation of a simplified risk calculation method.

Evolution of the risk calculation is illustrated with output data from each probability or frequency sub-model, and the intermediate stages where these are combined. The parametric evaluation showed how the risk is affected by changing parameters associated with the equipment design, the installation environment and aspects associated with its operation. Example cases are given for four different equipment types, considering the variation of risk when equipment is designed according to a safety standard.

Prior to these exercises, the issue of validating the QRA model was considered. In general, it is virtually impossible to empirically validate a QRA since not only is comprehensive historical data needed (which should include detailed records of equipment design, installation conditions and incidents), but there must be a large number of installations of identical design, operating and environmental conditions. Unfortunately, with an emerging technology (such as HC refrigerants), such data is scarce. Validation becomes more difficult when considering that the majority of consequences are relatively minor (e.g., small jet/flash fire) and of no concern to the relevant authorities, making reporting unlikely. Nevertheless, information about all known incidents has been collated for consideration (Table 49). Initial observations from Table 49 were that most accidents occurred either during servicing or as a direct result of the servicing activity. Also, about half of the incidents are associated with equipment that was retrofitted with HCs, and therefore not designed appropriately. Nevertheless, in absence of market data for most of the equipment types above, it was not possible to compare the incidence of reported ignition events for a known population, against risk frequencies from the model.
Another point of interest is comparison of results produced by the present QRA, against the results of QRA exercises reported in the literature. With reference to the studies discussed in Section 1.2.1, it was decided that a direct comparison was not possible, either because the application was outside the present scope, or because insufficient input details were provided for a precise representation of the equipment and installation characteristics. The latter was confirmed by applying a range of assumed input values to the QRA model, to find that it output frequencies that varied over several orders of magnitude for the single application. However, in conducting this exercise it was found that the range of modelled risk frequencies for a given application was typically lower than the ignition/fire and fatality frequencies of the published QRAs. This observation implies that the more general QRA methods previously employed
elsewhere overestimate the risk, which was attributed primarily to the manner by which assumptions are often made within the context of a safety assessment. Normally, an assumed quantity represents an educated guess at an "average" value in absence of better data, but with safety assessments the conventional approach is to skew assumptions towards values that would lead to a more severe outcome. Thus, where existing QRAs adopt assumptions for inputs or calculation methods, they tend to employ "worse case" values. Since the current study sought empirical data and correlations to overcome many of the assumptions used within published QRAs, the choice of exaggerated values for variables that could increase the risk was avoided.

Whilst the present study does rely on a limited number of least favourable suppositions (i.e., vapour-only, low velocity leak), the series of worse-case assumptions adopted at each stage in the calculation sequence (in most of the reported QRAs) compounds the overestimation. It is the avoidance of this effect that makes the risk estimates from the present QRA significantly lower than the literature.

Table 49: Reports of incidents involving HC refrigerants

<table>
<thead>
<tr>
<th>Description</th>
<th>Equipment</th>
<th>Cause</th>
<th>SOI</th>
<th>Effect</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Home, Australia</td>
<td>Domestic fridge, retrofitted</td>
<td>Leaked from joint that was made with epoxy resin.</td>
<td>Thermostat</td>
<td>Fire and overpressure sufficient to distort fridge</td>
<td>Workplace Services, 2000</td>
</tr>
<tr>
<td>Café, Australia</td>
<td>Commercial fridge, retrofitted</td>
<td>Overcharged system known to have a leak, condenser split when charging.</td>
<td>Spark from plug when pulled from socket</td>
<td>Flash fire, burn injuries</td>
<td>Workplace Services, 2000</td>
</tr>
<tr>
<td>Cellar, Switzerland</td>
<td>Domestic heat pump</td>
<td>Leak from evaporator into water circuit and exited via air vent</td>
<td>Service person’s cigarette lighter, used when looking for leak</td>
<td>Temporary jet flame</td>
<td>Afjei, 1999</td>
</tr>
<tr>
<td>Workshop, Algeria</td>
<td>Domestic fridge</td>
<td>Broke into system unaware of flammable refrigerant</td>
<td>Brazing torch</td>
<td>Fire and minor damage</td>
<td>Mariager, 2001</td>
</tr>
<tr>
<td>Workshop, Philippines</td>
<td>Mobile air conditioner, retrofitted</td>
<td>Release during servicing</td>
<td>Not known (possibly lighting cigarette)</td>
<td>Small flash fire/jet flame</td>
<td>Private communication, 2002</td>
</tr>
<tr>
<td>Home, England (3 cases)</td>
<td>Domestic fridge</td>
<td>Leaked into refrigerated compartment</td>
<td>Thermostat</td>
<td>Overpressure sufficient to distort fridge</td>
<td>WMFS, 2003</td>
</tr>
</tbody>
</table>
5.2 Evolution of risk calculation

The probability models in Section 2 were employed to illustrate the results of the component probability calculations, and the evolving overall risk frequencies. To do this, a set of reference conditions was chosen and the results from the probability and frequency calculations (based on results produced with the dispersion model in Section 3 and consequence models in Section 4) were presented. Calculations were based on an installation according to the following: 1 kg of R290 in a system comprising 30 joints, where \( h_r \) and \( h_d \) are both at 2 m. Following a leak, none of the charge was retained within the system. It operated for 12 hours per day, and during on-cycle, \( \dot{V}_d = 0.033 \, \text{m}^3 \, \text{s}^{-1} \) (the low value—equivalent to \( \lambda = 1 \, \text{h}^{-1} \)—was chosen for illustrative purposes); the forced airflow remained constant in the event of a leak, and mechanical equipment faults were ignored. The room was 40 m\(^2 \) x 3 m high and infiltration \( \Lambda_{so} = 14 \, \text{h}^{-1} \), where the corresponding \( \Lambda_{\text{refl}} \) were as Figure 101. There were four occupants present for 24 hours per day, although airflow from thermal sources was neglected. Within the lower (0 m - 1 m) and middle (1 m - 2 m) room levels, there were five SOI and two SOI, respectively, and all SOI were active for 1 s, four times each day in a random manner. Other probabilities associated with the release were: \( P_{\text{leak}} = 1 \), \( P_{\text{perc}} = 0.73 \), \( P_{\text{avail}} = 1 \) and \( P_{\text{sys}} = 1 \). For the consequence, the room was totally confined (except for the infiltration opening of 0.084 m\(^2 \)), and ignition always occurred at the centre of the flammable volume.

Following the sequence in Figure 172, leak frequencies and leak times were characterised, corresponding flammable quantities were obtained from the dispersion model, and then the component probabilities — \( P_{\text{soi}}^F \), \( P_{\text{v}}^F \) and \( P^F_{*} \) — were calculated, followed by the severity of consequences and associated \( \chi_{\text{loc}} \), \( \chi_{\text{ap}} \), and \( \chi \). Information on system operation, occupants and weather conditions were used to establish the relevant time fractions.

\( f_{\text{leak}} \) and \( t_{\text{leak}} \) were calculated for 15 different sizes (as Section 2.3) for both on- and off-cycle. For each leak, \( \dot{V}_F \), \( \bar{m}_F \) and \( t_F \) were generated for both on-cycle (when \( \dot{V}_d = 0.033 \, \text{m}^3 \, \text{s}^{-1} \)) and off-cycle (\( \dot{V}_d = 0 \)), and for each cycling mode the calculation were carried out for (additional) airflow from infiltration for the range of \( \Lambda_{\text{refl}} \). Subsequently, \( P_{\text{soi}}^F \), \( P_{\text{v}}^F \), \( P^F_{*} \) and finally \( f^* \) were obtained for each separate combination of conditions: cycling mode, leak size, forced airflow, and infiltration rate. Results for \( f_{\text{leak}} \), \( P_{\text{soi}}^F \), and \( P_{\text{v}}^F \) at the lowest infiltration rate (\( \Lambda_{\text{refl}} = 0.4 \, \text{h}^{-1} \), chosen to clarify variations) are presented in Figure 173, for both on- and off-cycle over
a range of leak times, or leak sizes (on the basis that a shorter leak of the same mass produces a "larger" leak); the data does not include adjustment with $\phi_{infl/3}$.

Generally for longer leak times ($t_{leak} > 2000$ s), $f_{leak}$ for a given leak size was similar for both on- and off-cycle. The observed deviation at short leak times was expected given that their duration would normally be shorter than compressor cycling intervals for many equipment types. The trend in $P_{soli}^F$ behaves in a similar fashion. For longer leak times, the similarity between on- and off-cycle is due to the fact that the released is dispersed within $t_{leak}$, so $t_F = t_{leak}$. This is consistent for all leak sizes during on-cycle since the associated forced airflow is sufficient to rapidly disperse the release even for the most catastrophic leak. Conversely, for $t_{leak} < 1000$ s, the conditions during off-cycle cause a flammable cloud such that $t_F > t_{leak}$, and $P_{soli}^F$ tends to increase towards a catastrophic release.

Another factor that affects $P_{soli}^F$ is the positioning of the SOI - as $t_{leak}$ increases, the region occupied by $V_F$ gets smaller and gradually shifts closer to the origin of the release, i.e., upwards towards $h_r$. Consequently, the possibility of ignition becomes more dependent on the SOI within the middle level. This is seen in Figure 174, which plots the ratio of local ignition frequency in the lower and middle vertical room elements to the total frequency ($f_{n}^* / f^*$) for the different leak sizes (the upper element is ignored since $C_F$ never occurs there) during off-cycle only. It is seen at shorter leak times, the ignition frequency is mainly attributed to the lower level, and this is because the corresponding large mass flow rate creates a stratified layer on the floor, thus large
As $t_{\text{leak}}$ gets larger, the greater proportion of the ignition frequency occurs at the middle level, and the contribution at the lower level falls off. The distribution is equal at about $t_{\text{leak}} \approx 5 \times 10^3$ s, and the entire $f^*$ shifts to the middle level at about $t_{\text{leak}} \approx 10^4$ s, when $C^F$ cease to exist in the lower level (so for the lower level, $P_{s\text{oil}}^F = 0, P_{o}^F = 0$). A similar trend in the distribution of $f^*$ was seen for on-cycle, although the transition from $f^*$ dominating the low level to the middle level occurred at shorter leak times since the higher airflow disperses larger releases more rapidly before they reach floor level. These observations demonstrate the importance of establishing the correct position of the SOI and $V_F$. Also shown in Figure 174 is the ratio of total $f^*$ to the “lumped” ignition frequency which was calculated assuming a single element, i.e., $P_{s\text{oil}}^F$ and $P_{o}^F$ were calculated for the total $V^F$. The results further support the importance of discretising the room volume, as in the present case the lumped approach overestimates $f^*$ for a catastrophic leak by 40% and underestimates $f^*$ for the longest leak time by 90%.

![Figure 174: Variation of $f^*$ with room level for different leak sizes for off-cycle](image)

![Figure 175: $f^*$ for different leak times for a range of infiltration rates](image)

The overall variation in $f^*$ with $t_{\text{leak}}$ is also shown in Figure 173. For longer leak sizes, it is seen that $f^*$ reduces slightly from about $t_{\text{leak}} \approx 10^3$ s and longer, and this applies to both on- and off-cycle whilst having similar values. This trend is expected when observing the component probabilities, $f_{\text{leak}}, P_{s\text{oil}}^F$, and $P_{o}^F$, for each leak size. For shorter, more catastrophic leaks, a deviation between on- and off-cycle $f^*$ occurs, although under both conditions, individual $f^*$
are significantly greater than when $t_{\text{leak}} > 10^3$ s. These higher $f^*$ follow the higher values of the component $P_{\text{oc}}^F$, and $P_F^F$ which are due to the larger $V_F^F$ resulting from the catastrophic leak conditions.

Next, the effect of different infiltration rates was considered. Figure 175 shows how $f^*$ (not adjusted for $\phi_{\text{infil}}$) varies with $\Lambda_{\text{infil}}$, for a number of different leak sizes up to 800 s (since the trend thereafter is obvious). At low $\Lambda_{\text{infil}}$ and short leak times, high $f^*$ occur and reduce rapidly with greater $\Lambda_{\text{infil}}$. Eventually, further increase in $\Lambda_{\text{infil}}$ has negligible impact on $f^*$ as it remains approximately constant thereafter. The infiltration rate at which this transition to constant $f^*$ gets lower as leak time is prolonged. This is explained by reference to Figure 161 where once a particular $\vec{u}_{\text{Rm}}$ has been exceeded, $V_F^F t_F^F$ thereon remains nearly constant and Figure 162 where $V_F^F t_F^F$ also remains constant for longer leak times. Since $P_{\text{oc}}^F$, and $P_F^F$ are strongly dependent upon $V_F^F t_F^F$ and $t_F^F$, it follows that the corresponding $f^*$ will follow this behaviour. In a practical sense, where high infiltration rates are normally present, evaluation of a single long leak time for a range of high $\Lambda_{\text{infil}}$ may be acceptable. This approach is supported for the present example because higher $\Lambda_{\text{infil}}$ occur less often (Figure 101). Lastly, a notable difference is seen in $f^*$ between on- and off-cycle operation, where a constant $f^*$ occur at lower $\Lambda_{\text{infil}}$ (for the same leak time) with off-cycle on account of the associated $\vec{V}_d$ giving higher $\vec{u}_{\text{Rm}}$.

From the data-sets as above, $I_{\text{oc}}$ and $\Delta P_{\text{Rm}}^o$ were calculated as shown in Figure 176. Both $I_{\text{oc}}$ and $\Delta P_{\text{Rm}}^o$ are seen to reduce with longer leak time due to smaller $V_F^F$ and $t_F^F$ and likewise, values for releases during on-cycle are smaller than for off-cycle. A transition in the gradient occurs when there is no longer flammable cloud development at the floor and thereon $I_{\text{oc}}$ and $\Delta P_{\text{Rm}}^o$ are similar regardless of operating mode. Under conditions that lead to highest $I_{\text{oc}}$ ($\approx 100$ s (kW m$^{-2}$)), ignition may cause minor burns and may be sufficient to produce secondary ignition of easily combustible materials. The highest $\Delta P_{\text{Rm}}^o$ would damage windows and light panels but is insufficient to injure occupants. Ignition of a release longer than 500 s would produce $I_{\text{oc}}$ that is unlikely to be felt by occupants, and $\Delta P_{\text{Rm}}^o$ would not damage occupants or property. Under no circumstances would the consequences lead to fatality. (The values below $10^{-4}$
s (kW m\(^{-3}\))\(^{43}\) were ignored because of negligible practical significance – for example, \(I_{\text{occ}} = 10^4\) s (kW m\(^{-3}\))\(^{43}\) would be comparable to holding a cigarette lighter at arms length for one second.)

**Figure 176: Overpressure and thermal intensity for different leak times**

**Figure 177: Integrated consequence-frequencies for different leak times**

Considering the wide variation of \(I_{\text{occ}}\) and \(\Delta p_{\text{RM}}^o\) due to ignition of small and large leaks, the value of \(f^*\) does not necessarily reflect the "risk" because it ignores the severity of the consequence of the ignition event. Thus, by employing the product \(f^* \times I_{\text{occ}}\) and \(f^* \times \Delta p_{\text{RM}}^o\) as \(X_{\text{loc}}\) and \(X_{\Delta p}\) respectively, the risk is better represented since it accounts for both likelihood of ignition and the subsequent severity of the consequence. Figure 177 shows these values for the current example. As with the variation of \(f^*\) with \(t_{\text{leak}}\), the catastrophic leaks produce a notably higher risk than the longer leaks, although the range over which \(X_{\text{loc}}\) and \(X_{\Delta p}\) differ is far greater than \(f^*\). The range over which \(f^*\) varies is in the order 10\(^3\), whilst for \(X_{\text{loc}}\) and \(X_{\Delta p}\) the variation is about 10\(^{10}\); whilst being subjective, this is believed to provide a better representation of the difference in risk between an occasional temporary flame, and a more likely explosion event capable of secondary fire, building damage and serious injury.

For the overall ignition frequencies, individual \(f^*\) was adjusted for \(\phi_{\text{inf}}\), summed for all infiltration rates, and these were adjusted to account for 50% on-cycle (\(\phi_{\text{oc}} = 0.5\)) and 50% off-cycle time fractions. The resulting \(f^*\) for on-cycle, off-cycle and the summed total \(f^*\) is presented in Figure 178 for the range of leak sizes. Compared to Figure 173 and Figure 175 for
$\Lambda_{\text{rep}} = 0.4 \text{ h}^{-1}$, the variation in total $f^*$ is seen to be smoother, where there is less of a distinction over the range of leak sizes, due to the addition of frequencies for the higher infiltration rates, since these are known to have more consistent $f^*$ as seen in Figure 175. However, despite the evening effect of the contribution of higher infiltration rates to the overall $f^*$, the previously observed elevated $f^*$ associated with catastrophic leaks during off-cycle still dominates.

![Figure 178: Overall ignition frequencies](image)

![Figure 179: Overall risk factor](image)

The overall risk factor as a function of leak size is shown in Figure 179, again for off-cycle, on-cycle and the total $X$ (handled the same as Figure 178). This data follows the previously identified characteristics of $X_{\text{on}}$ and $X_{\text{off}}$ in Figure 177, where on- and off-cycle $X$ are similar with long $t_{\text{leak}}$, and that the off-cycle $X$ dominates the total for catastrophic releases. Most importantly, the variation of $X$ with leak size compared to $f^*$ (in Figure 178), implies a relatively greater risk for catastrophic leaks than for "pinhole" leaks.

For the installation as a whole, on-cycle $f^* = 3.2 \times 10^9$, off-cycle $f^* = 4.9 \times 10^9$, and the (sum) total $f^* = 8.1 \times 10^9$. The comparable values for on- and off-cycle imply that the risk (per operating time, since $\phi_{\text{op}} = 0.5$) is similar regardless of operating mode. However, in terms of risk factor, on-cycle $X = 3.4 \times 10^9$, off-cycle $X = 3.6 \times 10^8$, and total $X = 3.9 \times 10^8$. Thus using $X$ for comparative purposes shows that the risk during off-cycle accounts for about 90% of the total risk, and therefore represents a more hazardous situation per unit of time. Thus $X$ is
considered to better characterise the extent of the risk associated with the range of conditions encountered.

Table 50: Comparison with maximum acceptable risk criteria

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Example results</th>
<th>Limits (members of public)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency of ignition (fire) ( f^* )</td>
<td>( 8.1 \times 10^{-3} ) y(^{-1} )</td>
<td>&lt; ( 1 \times 10^{-5} ) y(^{-1} )</td>
</tr>
<tr>
<td>Thermal intensity (( I_{occ} ))</td>
<td>( 79 ) s (kW m(^2) (^{4/3} ))</td>
<td>&lt; ( 1050 ) s (kW m(^2) (^{4/3} ))</td>
</tr>
<tr>
<td>Overpressure (( \Delta p_{\text{pm}} ))</td>
<td>( 15.4 ) kPa</td>
<td>&lt; ( 250 ) kPa</td>
</tr>
<tr>
<td>Frequency of fatality (( f_{\text{fatal}} ))</td>
<td>( 0 ) y(^{-1} )</td>
<td>&lt; ( 1 \times 10^{-6} ) y(^{-1} )</td>
</tr>
<tr>
<td>Integrated thermal intensity-freq. (( X_{\text{loc}} ))</td>
<td>( 1.6 \times 10^{-7} ) s (kW m(^2) (^{4/3} ) y(^{-1} ))</td>
<td>&lt; ( 5.6 \times 10^{-3} ) s (kW m(^2) (^{4/3} ) y(^{-1} ))</td>
</tr>
<tr>
<td>Integrated overpressure-freq. (( X_{\text{op}} ))</td>
<td>( 3.7 \times 10^{-8} ) kPa y(^{-1} )</td>
<td>&lt; ( 4.2 \times 10^{-4} ) kPa y(^{-1} )</td>
</tr>
<tr>
<td>Risk factor (( X ))</td>
<td>( 3.9 \times 10^{-8} )</td>
<td>&lt; ( 5.9 \times 10^{-4} )</td>
</tr>
</tbody>
</table>

Finally, the results were compared against the various criteria for determining the significance of the calculated risk measures (Table 50). In terms of \( f^* \), \( X_{\text{loc}} \), \( X_{\text{op}} \) and \( X \), calculated values are in the order of \( 10^3 \) to \( 10^4 \) times lower than the maximum tolerable values for members of the public (from Table 36). \( f_{\text{fatal}} \) is zero and \( I_{occ} \) and \( \Delta p_{\text{pm}} \) are around 10% of the maximum values.

5.3 Parametric assessment of input variables

A parametric evaluation was conducted to indicate how certain variables would affect risk, based on a fixed set of input (reference) values for a split air conditioning system in an office environment, similar to the situation detailed in Section 5.2. Values associated with the equipment design, operation and the installation environment were addressed, and relevant inputs listed in Table 51.

Several inputs were not subject to change, either because their influence on the overall risk is known to be negligible or approximately proportional. Additionally, it was assumed that there was no synchronisation between occupancy, cycling, SOI events or thermal sources. The fixed inputs were as follows: location of components as split type system, \( P_{\text{leak}} = 0.21 \); system leak probability, \( P_{\text{as}} = 0.2 \); unit airflow discharge area, 0.12 m\(^2\); number of SOI on unit = 10; unit SOI active time is 1 s; number of events of each unit SOI = 10 per 24 h; number of SOI = 6
(low), 4 (mid), 2 (high); unit airflow fault probability = 0.10; unit airflow fault repair time = 7 days; height of thermal sources is 1.5 m; mechanical ventilation fault probability = 0.10; mechanical ventilation fault repair time is 30 days; termination of unit operation (compressor and forced airflow) occurs when 50% of charge is lost.

**Table 51: Reference values and variations**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Reference</th>
<th>Additional range tested</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant type</td>
<td>R290</td>
<td>R600a, R1270</td>
</tr>
<tr>
<td>Refrigerant mass</td>
<td>1 kg</td>
<td>0.5 kg, 1.5 kg</td>
</tr>
<tr>
<td>Number refrigerant circuits</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>Release height</td>
<td>2 m</td>
<td>1 m</td>
</tr>
<tr>
<td>Airflow rate</td>
<td>1 h⁻¹</td>
<td>2 h⁻¹</td>
</tr>
<tr>
<td>Airflow discharge height</td>
<td>2 m</td>
<td>1 m</td>
</tr>
<tr>
<td>Airflow discharge direction</td>
<td>Horizontal</td>
<td>Downward, upward</td>
</tr>
<tr>
<td>No of connections</td>
<td>30</td>
<td>10</td>
</tr>
<tr>
<td>Unit SOI ( P_{\text{total}} )</td>
<td>0.14</td>
<td>1</td>
</tr>
<tr>
<td>Room size</td>
<td>3 m × 40 m²</td>
<td>3 m × 20 m², 3 m × 60 m²</td>
</tr>
<tr>
<td>Mech vent (airflow/discharge height)</td>
<td>0 h⁻¹/-</td>
<td>3 h⁻¹/2.5 m, 3 h⁻¹/0.5 m</td>
</tr>
<tr>
<td>Mech vent detector initiation</td>
<td>-</td>
<td>20% of LFL (as EN 378)</td>
</tr>
<tr>
<td>Window size</td>
<td>4 m²</td>
<td>0 m²</td>
</tr>
<tr>
<td>Fabric leakage</td>
<td>14 h⁻¹</td>
<td>7 h⁻¹, 28 h⁻¹</td>
</tr>
<tr>
<td>No. SOI events/24 h (low, mid, high)</td>
<td>2, 4, 4</td>
<td>4, 8, 8</td>
</tr>
<tr>
<td>SOI active time</td>
<td>1 s</td>
<td>100 s</td>
</tr>
<tr>
<td>Post-ventilation period</td>
<td>0 s</td>
<td>300 s</td>
</tr>
<tr>
<td>Unit airflow initiated by refl. detection</td>
<td>-</td>
<td>20% of LFL (0.008 kg m⁻³)</td>
</tr>
<tr>
<td>Unit airflow initiated by refrigerant loss</td>
<td>-</td>
<td>25% of charge lost</td>
</tr>
<tr>
<td>Number of thermal sources</td>
<td>0</td>
<td>20</td>
</tr>
</tbody>
</table>

Table 52 presents results in terms of total ignition frequency, maximum thermal intensity, maximum overpressure and overall risk factor; integrated thermal intensity-frequency and integrated overpressure-frequency were not included as they are encompassed within \( X \). (Maximum \( I_{\text{acc}} \) and maximum \( \Delta p_{\text{run}} \) refer to the highest value calculated over all conditions within a single QRA run.) No results for fatality frequency were presented as \( f_{\text{fatal}} = 0 \) for all of the conditions. The difference in \( f^* \) and \( X \) for the different variables in the range listed in Table 51 compared to the reference case, were expressed as logarithmic change, i.e., \( \ln \Delta f^* = \)
1 - \ln f^*/\ln f^* \{\text{ref}\} \text{ and } \ln \Delta X = 1 - \ln X/\ln X\{\text{ref}\}, \text{ in percentages. This presentation assists interpretation since many of the variations were several orders of magnitude in difference.}

Differences in values of \( I_{oc} \) and \( \Delta \rho_{rm} \) were expressed as percentages in the usual way, i.e.,

\[
\Delta I_{oc} = 1 - I_{oc}/I_{oc}\{\text{ref}\}, \text{ and } \Delta \Delta \rho_{rm} = 1 - \Delta \rho_{rm}/\Delta \rho_{rm}\{\text{ref}\}.
\]

Several general observations were made from Table 52. First, the behaviour of the variation in \( f^* \) is always followed by \( X \), although it is rarely proportional. Under most situations, a large change in \( f^* \) and \( X \) was consistent with a large change in maximum \( I_{oc} \) and \( \Delta \rho_{rm} \). Further, variations in maximum \( I_{oc} \) and \( \Delta \rho_{rm} \) were consistent with each other (and with maximum \( \bar{C}_{f_{\text{max}}} \)), and the direction was always followed by \( f^* \) or \( X \) except when the conditions meant that maximum \( I_{oc} \) and \( \Delta \rho_{rm} \) remained constant. Overall, there were never contradictions amongst these risk measures, for example, a reduction in \( f^* \) and increase of \( X \) did not occur.

### Table 52: Results of parametric evaluation

<table>
<thead>
<tr>
<th>Variation for range in Table 51</th>
<th>( f^* ) ((\text{y}^{-1}))</th>
<th>( \ln \Delta f^* ) (%)</th>
<th>( I_{oc}, \Delta \rho_{rm} ) ((\text{skW/m}^2, \text{kPa}))</th>
<th>( \Delta I_{oc}, \Delta \rho_{rm} ) (%)</th>
<th>( X ) ((-))</th>
<th>( \ln \Delta X ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference</td>
<td>6.9×10^{-10}</td>
<td>0.0</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>7.0×10^{-10}</td>
<td>0.0</td>
</tr>
<tr>
<td>Refrigerant, R600a</td>
<td>5.1×10^{-10}</td>
<td>-1.4</td>
<td>1.3, 3.1</td>
<td>-8, -4</td>
<td>5.0×10^{-10}</td>
<td>-1.6</td>
</tr>
<tr>
<td>Refrigerant, R1270</td>
<td>1.8×10^{-10}</td>
<td>-6.3</td>
<td>0.4, 1.1</td>
<td>-63, -67</td>
<td>7.4×10^{-11}</td>
<td>-10.7</td>
</tr>
<tr>
<td>( m_r = 0.75 \text{ kg} )</td>
<td>7.6×10^{-11}</td>
<td>-10.4</td>
<td>0.0, 0.1</td>
<td>-96, -98</td>
<td>1.5×10^{-12}</td>
<td>-29.3</td>
</tr>
<tr>
<td>( m_r = 1.75 \text{ kg} )</td>
<td>2.3×10^{-9}</td>
<td>5.7</td>
<td>3.3, 13.2</td>
<td>198, 313</td>
<td>7.2×10^{-9}</td>
<td>11.0</td>
</tr>
<tr>
<td>No. circuits = 2</td>
<td>5.0×10^{-11}</td>
<td>-12.4</td>
<td>0.0, 0.1</td>
<td>-96, -98</td>
<td>2.1×10^{-12}</td>
<td>-27.6</td>
</tr>
<tr>
<td>( h_r = 1 \text{ m} )</td>
<td>1.8×10^{-9}</td>
<td>4.5</td>
<td>2.6, 11.3</td>
<td>138, 255</td>
<td>4.1×10^{-9}</td>
<td>8.4</td>
</tr>
<tr>
<td>( h_d ) and ( h_r = 1 \text{ m} )</td>
<td>2.8×10^{-9}</td>
<td>6.7</td>
<td>2.6, 12.7</td>
<td>138, 298</td>
<td>6.4×10^{-9}</td>
<td>10.5</td>
</tr>
<tr>
<td>( \dot{V}_d ), ( \Delta_d = 2 \text{ h}^{-1} )</td>
<td>6.0×10^{-10}</td>
<td>-0.7</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>6.6×10^{-10}</td>
<td>-0.3</td>
</tr>
<tr>
<td>Airflow ( \theta_d = 2 \text{ rad} )</td>
<td>8.4×10^{-10}</td>
<td>0.9</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>9.1×10^{-10}</td>
<td>1.2</td>
</tr>
<tr>
<td>Airflow ( \theta_d = 1 \text{ rad} )</td>
<td>6.0×10^{-10}</td>
<td>-0.7</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>6.6×10^{-10}</td>
<td>-0.3</td>
</tr>
<tr>
<td>Unit ( C_{\text{detect}} = 20% \text{LFL} )</td>
<td>2.9×10^{-10}</td>
<td>-4.1</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>1.2×10^{-10}</td>
<td>-8.5</td>
</tr>
<tr>
<td>Post-vent ( t_d = 300 \text{ s} )</td>
<td>6.9×10^{-10}</td>
<td>0.0</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>7.0×10^{-10}</td>
<td>0.0</td>
</tr>
<tr>
<td>( N_{\text{con}} = 10 )</td>
<td>2.9×10^{-10}</td>
<td>-4.1</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>2.3×10^{-10}</td>
<td>-5.3</td>
</tr>
</tbody>
</table>

397
Regarding equipment design, refrigerant has a distinct influence where R600a produces a slight reduction in risk (despite a higher $C_{f_{max}}$) but a more notable improvement occurs with R1270. Both refrigerants have a higher LFL than R290 which accounts for a portion of the lower risk, and R1270 has a lower density than both and therefore easier to disperse. However, a more thorough comparison of different refrigerants should account for the variation in system charge given equal refrigerating capacity. This is particularly important with respect to the sensitivity to released mass on risk (reflected by the trends in $V_{F_{T}}$, Figure 159). Splitting the charge in half through the use of two independent refrigerant circuits is also beneficial. In this case, the risk is calculated as usual but with half the refrigerant charge, and the final result is doubled to account for the two systems (circuits) existing concurrently. In this case, the risk associated with a smaller charge in two separate circuits is less than a single system with twice the charge, indicating that the contribution of charge size to the risk is greater than that of the increased $f_{leak}$ associated with having two systems.

<table>
<thead>
<tr>
<th>Variation for range in Table 51</th>
<th>$f^*$ (y$^{-1}$)</th>
<th>$\ln \Delta f^*$ (%)</th>
<th>$I_{occ} \cdot \Delta P_{Rm}^R$ (skWm$^{-2}$kPa)</th>
<th>$\Delta I_{occ} \cdot \Delta P_{Rm}^R$ (%)</th>
<th>$X$ (-)</th>
<th>$\ln \Delta X$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unit $P_{avoll} = 1$</td>
<td>6.9 x 10$^{-10}$</td>
<td>0.0</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>7.0 x 10$^{-10}$</td>
<td>0.0</td>
</tr>
<tr>
<td>Unit $P_{avoll} = 1$, $h_d = 1$ m</td>
<td>9.9 x 10$^{-9}$</td>
<td>12.6</td>
<td>1.1, 3.2</td>
<td>138, 298</td>
<td>9.1 x 10$^{-8}$</td>
<td>16.4</td>
</tr>
<tr>
<td>$A_{Rm} = 20$ m$^2$</td>
<td>4.4 x 10$^{-10}$</td>
<td>8.8</td>
<td>6.6, 37.6</td>
<td>500, 1078</td>
<td>3.0 x 10$^{-8}$</td>
<td>17.8</td>
</tr>
<tr>
<td>$A_{Rm} = 60$ m$^2$</td>
<td>2.8 x 10$^{-12}$</td>
<td>-26.1</td>
<td>0.0</td>
<td>-97, -99</td>
<td>7.5 x 10$^{-14}$</td>
<td>-43.4</td>
</tr>
<tr>
<td>$\Lambda_{mech} = 3h^1$, $h_{mech} = 2.5$m</td>
<td>3.0 x 10$^{-11}$</td>
<td>-14.8</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>6.8 x 10$^{-12}$</td>
<td>-22.0</td>
</tr>
<tr>
<td>$\Lambda_{mech} = 3h^1$, $h_{mech} = 0.5$m</td>
<td>2.3 x 10$^{-11}$</td>
<td>-16.0</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>6.5 x 10$^{-12}$</td>
<td>-22.2</td>
</tr>
<tr>
<td>Window $A_w = 0$ m$^2$</td>
<td>6.9 x 10$^{-10}$</td>
<td>0.0</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>7.0 x 10$^{-10}$</td>
<td>0.0</td>
</tr>
<tr>
<td>$\Lambda_{so} = 7$ h$^1$</td>
<td>1.4 x 10$^{-9}$</td>
<td>3.4</td>
<td>2.5, 9</td>
<td>126, 182</td>
<td>3.2 x 10$^{-9}$</td>
<td>7.2</td>
</tr>
<tr>
<td>$\Lambda_{so} = 28$ h$^1$</td>
<td>2.4 x 10$^{-11}$</td>
<td>-15.9</td>
<td>0.1</td>
<td>-96, -97</td>
<td>1.1 x 10$^{-12}$</td>
<td>-30.5</td>
</tr>
<tr>
<td>$N_{G_F} = 4, 8, 8$</td>
<td>9.8 x 10$^{-10}$</td>
<td>1.7</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>2.8 x 10$^{-9}$</td>
<td>1.7</td>
</tr>
<tr>
<td>$t_{sol} = 100$ s</td>
<td>1.4 x 10$^{-9}$</td>
<td>3.3</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>3.9 x 10$^{-9}$</td>
<td>3.3</td>
</tr>
<tr>
<td>25% loss of $m_r$ starts</td>
<td>6.0 x 10$^{-10}$</td>
<td>-0.7</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>5.2 x 10$^{-10}$</td>
<td>-1.4</td>
</tr>
<tr>
<td>$V_d$</td>
<td>3.6 x 10$^{-10}$</td>
<td>-3.1</td>
<td>1.1, 3.2</td>
<td>0.0</td>
<td>3.5 x 10$^{-10}$</td>
<td>-3.3</td>
</tr>
<tr>
<td>$N_{therm} = 20$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
The effects of both $h_r$ and $h_d$ on risk also follow the behaviour of $V^F t^F$, shown earlier in Section 3.4.8. A lower $h_r$ increases risk but simultaneously lowering $h_d$ does not reverse this, as implied by Figure 163. This is due to the influence of large $V^F t^F$ occurring during off-cycle; if on-cycle were longer the benefit of lowering $h_d$ would become more apparent since a correspondingly larger proportion of on-cycle $V^F t^F$ would be accounted for. This dependency on cycling modes is equally relevant when unit airflow is doubled. Assuming a system operating constantly, a doubling of the associated airflow would lead to a much lower $V^F t^F$ (as Figure 161), which would translate to a significant reduction in risk. However, with the system in off-cycle, an absence of forced airflow can produce $V^F t^F$ that is typically several times larger than the difference between $V^F t^F$ for low and high airflow rates. Thus, unless the contribution from off-cycle $V^F t^F$ is reduced by a small time fraction (minimal off period), it would dwarf the benefit offered by a higher airflow during on-cycle.

Similarly, it is seen in Table 52 that introducing a post-ventilation time ($t_{d,min}$, as calculated in Section 3.3) had negligible effect on reducing risk. This is due to the fact that the risk during off-cycle is unaffected by post-ventilation, and during on-cycle the post-ventilation only provides additional airflow for (in this example) 300 s following the start of the release. Without post-ventilation, it was stated previously that the fans continue to operate until 50% of the charge is lost. Reference to Section 3.3.2 and 3.3.3 shows that the rate of floor concentration ($\bar{C}_f$) development for large leaks is highest during the start of the release, whilst for longer leaks $\bar{C}_f(t) \rightarrow \bar{C}_{\text{run}}(t)$. As such for catastrophic leaks, the effect of continued airflow for the first 50% of $t_{\text{leak}}$ mitigates against the development of much high $\bar{C}_f$ anyway, and so the additional post-ventilation will provide little extra benefit. For longer leaks, $\bar{C}_f$ is low anyway. Assuming the risk is proportional to $V^F t^F$, the insignificant benefit associated with post-ventilation is reflected in Figure 162. Alternatively, where the unit airflow is initiated (or allowed to continue) by the pressure switch or (more effectively) the refrigerant detector, a notable reduction in risk is evident, and this is particularly evident when recalling that these control schemes introduce forced airflow in both on- and off-cycle mode, the latter receiving most benefit in terms of a greater reduction in $V^F t^F$ and therefore risk.
Concerning the integrity of the system, a reduction in the number of connections leads to a lower risk and this is purely due to lower leak frequency. In terms of the integrity of equipment electrical components (SOIs), two examples were given where the electrics are completely unprotected \( P_{\text{total}} = 1 \) – the first is with the SOI at unit height (2 m), and the second where the SOI are located lower down (1 m). Whilst the presence of additional SOI would imply an increase in risk, the results from the first case (unit SOI at 2 m) show negligible difference compared to the reference example. This is on account of the positioning of the SOI being above the region occupied by the flammable mixture. Conversely, with the second case where SOI are closer to the floor, the effect was to significantly increase risk.

Factors other than those related to the equipment design also have significant effects on risk. It is seen that room size has major influence, which again is implied by previous results for \( V^F t^F \) (Figure 160). The use of mechanical ventilation also reduces risk, but positioning of the inlet duct only has minor influence. The amount of room fabric leakage (or \( A_{S0} \)), also dictates the risk because of its effect on dispersion in terms of airspeed, regardless of the operating mode. Existence of thermal sources also has a similar benefit. Increasing the active time of room SOI produced a slight increase in risk, and more so when the number of room SOI events doubled.

Other parameters not included in the evaluation can be commented on according to the formulation of the model. In particular, if all components were within the room \( P_{\text{total}} = 1 \), then the risk would increase proportionally from the risk for \( P_{\text{total}} = 0.21 \) (i.e., \( 1/0.21 \)). The same logic applies to a change of system leak probability \( P_{\text{sys}} \), and the number of SOI (not on the equipment). A lower fault probability or repair time for either unit airflow for mechanical ventilation will obviously lead to a lower risk. Similarly, initiating airflow from system pressures that correspond to a smaller leaked mass, or the detector set to a lower concentration would also benefit risk reduction.

Based on this exercise, general guidance for risk reduction was developed. On the assumption that a manufacturer cannot dictate the environment within which the equipment is installed, the following are recommendations for design:

- Refrigerant selection for density close to air, and a high LFL (e.g., R1270)
- Small refrigerant charge
- Multiple independent circuits
- Piping and components located at highest position, and close (or above) air discharge
- High airflow rate and small discharge duct area
- Air discharge from floor level, or if at high level, discharge directed downwards
- Minimum number of connections, and the majority to be positioned within outdoor unit (if applicable)
- Unit SOI with good protection against ingress of gas, and positioned as high as possible
- Airflow initiated with pressure transducer but preferably with a refrigerant detector

5.4 Example calculations for selected RAC equipment
To demonstrate the intended use of the QRA, four “case” studies were prepared. Initial input conditions were based on the minimum required by EN 378: 2000, and then re-evaluated with certain (practicable) changes to the equipment design intended to improve safety. Since equipment design does not affect the room characteristics and environmental conditions, these were not changed. In addition, certain equipment characteristics were also fixed; specifically, it was assumed that the refrigerant charge was already minimised, the selected refrigerant was used on the basis that it was thermodynamically preferable, and using multiple circuits would be too costly. For all applications, room temperature were 21°C, average output of thermal sources was 150 W and their height was 1.5 m. Retention of refrigerant was 0.18 for on-cycle and 0.3 for off-cycle, and during on-cycle, airflow ceased once 50% of the charge had been lost. Unit airflow fault probability was 0.10, the repair time was 7 days, and no mechanical ventilation was used. All SOI, both on the unit and within the room were 1 s, and $P_{\text{fault}}$ was 0.14, and 1, respectively.

Window linear density, $W_{vp} = 10 \text{ kg m}^{-2}$ and the fixing force was equivalent to $p_{vp,fix} = 1 \text{ kPa}$.

Values for other input parameters for the four applications are provided in Table 53, along with the results from the QRA model. Variables adjusted to improve safety are identified in square parentheses, and similarly for the corresponding results.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Integral chiller cabinet in shop</th>
<th>Split freezer cabinet in shop</th>
<th>Portable A/C in office</th>
<th>Split A/C in office</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>R600a</td>
<td>R290</td>
<td>R290</td>
<td>R1270</td>
</tr>
<tr>
<td>$m_r$</td>
<td>0.5 kg</td>
<td>0.8 kg</td>
<td>0.4 kg</td>
<td>1.5 kg</td>
</tr>
<tr>
<td>Location, $P_{\text{leak}}$</td>
<td>1</td>
<td>0.281</td>
<td>1</td>
<td>0.281</td>
</tr>
<tr>
<td>Leakage, $P_{\text{sys}}$</td>
<td>0.05</td>
<td>0.20</td>
<td>0.05</td>
<td>0.15</td>
</tr>
<tr>
<td>$h_r$</td>
<td>0.5 m [1 m]</td>
<td>0.5 m</td>
<td>0.5 m</td>
<td>2.5 m</td>
</tr>
<tr>
<td>Unit $h_d$</td>
<td>- [1 m]</td>
<td>2.0 m [0.5 m]</td>
<td>0.5 m</td>
<td>2.5 m</td>
</tr>
<tr>
<td>Parameter</td>
<td>Integral chiller cabinet in shop</td>
<td>Split freezer cabinet in shop</td>
<td>Portable A/C in office</td>
<td>Split A/C in office</td>
</tr>
<tr>
<td>-----------------------------------------</td>
<td>----------------------------------</td>
<td>--------------------------------</td>
<td>------------------------</td>
<td>---------------------</td>
</tr>
<tr>
<td>Unit airflow $\theta$</td>
<td>horiz.</td>
<td>horiz.</td>
<td>upward</td>
<td>horiz. [down]</td>
</tr>
<tr>
<td>Unit $\dot{V}_d$</td>
<td>- [0.08 m$^3$ s$^{-1}$]</td>
<td>0.12 m$^3$ s$^{-1}$</td>
<td>0.06 m$^3$ s$^{-1}$</td>
<td>0.23 m$^3$ s$^{-1}$</td>
</tr>
<tr>
<td>Unit $A_d$</td>
<td>- [0.2 m$^2$]</td>
<td>0.6 m$^2$ [0.3 m$^2$]</td>
<td>0.1 m$^2$ [0.06 m$^2$]</td>
<td>0.4 m$^2$ [0.3 m$^2$]</td>
</tr>
<tr>
<td>Unit $N_{dwr}$</td>
<td>4 per 24 h</td>
<td>4 per 24 h</td>
<td>4 per 24 h</td>
<td>4 per 24 h</td>
</tr>
<tr>
<td>$H_{Rm} \times A_{Rm}$</td>
<td>3 m x 30 m$^2$</td>
<td>3 m x 30 m$^2$</td>
<td>3 m x 20 m$^2$</td>
<td>3 m x 40 m$^2$</td>
</tr>
<tr>
<td>$N_{vol}$ (low, mid, high)</td>
<td>20, 5, 5</td>
<td>20, 5, 5</td>
<td>10, 2, 2</td>
<td>20, 2, 2</td>
</tr>
<tr>
<td>$N_{dwr}$ (low, mid, high)</td>
<td>8, 4, 4 per 24 h</td>
<td>8, 4, 4 per 24 h</td>
<td>4, 2, 2 per 24 h</td>
<td>4, 2, 2 per 24 h</td>
</tr>
<tr>
<td>Window $A_v$</td>
<td>3 m$^2$</td>
<td>3 m$^2$</td>
<td>2 m$^2$</td>
<td>4 m$^2$</td>
</tr>
<tr>
<td>On-cycle $t_{op}$</td>
<td>18 h per 24 h</td>
<td>18 h per 24 h</td>
<td>8 h per 24 h</td>
<td>8 h per 24 h</td>
</tr>
<tr>
<td>Loss of $m_r$ to stop $\dot{V}_d$</td>
<td>50%</td>
<td>50%</td>
<td>50%</td>
<td>50%</td>
</tr>
<tr>
<td>$N_{occ}$</td>
<td>8</td>
<td>8</td>
<td>3</td>
<td>6</td>
</tr>
<tr>
<td>Occupancy $t_{occ}$</td>
<td>12 h per 24 h</td>
<td>12 h per 24 h</td>
<td>8 h per 24 h</td>
<td>8 h per 24 h</td>
</tr>
<tr>
<td>Post-ventilation $t_d$</td>
<td>- [240 s]</td>
<td>-</td>
<td>- [240 s]</td>
<td>-</td>
</tr>
<tr>
<td>$C_{detect}$ to start $\dot{V}_d$</td>
<td>-</td>
<td>- [20% LFL]</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Loss of $m_r$ to start $\dot{V}_d$</td>
<td>-</td>
<td>-</td>
<td>- [25%]</td>
<td>-</td>
</tr>
<tr>
<td>$A_{So}$</td>
<td>20 h$^{-1}$</td>
<td>20 h$^{-1}$</td>
<td>10 h$^{-1}$</td>
<td>10 h$^{-1}$</td>
</tr>
<tr>
<td>Thermal source $t_{therm}$</td>
<td>12 h per 24 h</td>
<td>12 h per 24 h</td>
<td>8 h per 24 h</td>
<td>8 h per 24 h</td>
</tr>
<tr>
<td>$N_{therm}$</td>
<td>10</td>
<td>10</td>
<td>9</td>
<td>18</td>
</tr>
</tbody>
</table>

$f^*$ (y$^{-1}$)                         | $2.0\times10^9$                  | $3.6\times10^8$                | $2.3\times10^{10}$     | $2.4\times10^9$    |

Max $I_{occ}$ (s (kW m$^{-1}$ $y^{0.5}$)) | 0.9 [0.8]                     | 2.2 [2.2]                     | 1.6 [1.6]              | 1.4 [0.1]          |

Max $\Delta p_{Rm}$ (kPa)                | 8.6 [5.9]                      | 26.6 [26.6]                    | 4.9 [4.9]              | 8.5 [0.4]          |

$X$ (*)                                  | $2.1\times10^9$               | $8.9\times10^8$               | $2.3\times10^{10}$     | $3.6\times10^9$    |

$1$ Given in volume flow rate instead of air changes because they are equipment specific.
The integral chiller cabinet used natural convection for the condenser, so forced airflow was introduced by employing a condenser fan. For a small unit, using a pressure sensor or refrigerant detector is too costly, so post-ventilation was used to continue airflow for an additional 240 s. Piping connections were reduced and the height of the lowest refrigerant containing part was raised to 1 m by rearranging pipework. The results indicate that these measures could lessen the risk (in terms of ignition frequency and risk factor) by a factor of 10, and the maximum severity of the consequences was also reduced slightly.

For the split freezer cabinet, the condenser air discharge was positioned much closer to the floor, and the discharge velocity was increased by halving the outlet duct area. A refrigerant sensor was positioned at floor level, which initiated the condenser air (if not already operating) if the concentration exceeded 20% of the LFL. Additionally, the number of connections and SOI were reduced by locating those that could be, to the outside unit. The effectiveness of these measures was slightly less than with the chiller cabinet, where both ignition frequency and risk factor were about five times lower. The maximum consequence was unchanged.

In the case of the portable air conditioner, the amount of possible alterations is limited because of compromising its ease of use, for example, a floor based appliance is easier to manoeuvre than a ceiling mounted unit. Other than modifying the piping, only the air discharge duct was made smaller and post-ventilation was used. Consequently, only a minor reduction in ignition frequency and risk factor occurred, where the altered equipment has about half that of the original design, and similarly, there was virtually no change in the maximum overpressure and thermal intensity.

A similar approach to the split freezer cabinet was used for the split air conditioner, where additional piping connections and SOI were located within the outdoor unit. Also, the air discharge duct was given a smaller area whilst being directed downward, and in the case of a leak during on- or off-cycle, a pressure sensor would initiate the airflow. With these additional features, a reduction in ignition frequency by a factor of 15 was achieved whilst the risk factor was lowered by some 200 times. This is a result of the major reduction in severity of maximum overpressure and thermal intensity too.

In addition to the exercise above, calculations were also made for servicing activities. For each application, the values in Table 53 were used, except where interference from a service technician would normally cause them to change. These included the variations listed below.

- Cycling time is assumed to be split equally between on and off mode, i.e., \( \phi_{op} = 0.5 \).
Release height is at 0.5 m to account for leaks originating from service manifolds, hoses and cylinders, in addition to the actual refrigerant circuit.

- Additional SOI for service tools, specifically a brazing torch (used twice for ten minutes), sparker, and vacuum and recovery pumps.
- Some equipment electrical devices become unprotected thereby increasing availability of SOI, so $P_{\text{avail}} = 0.43$.
- Leak frequencies for both on- and off-cycle increase (as discussed in Section 2.3), for which a factor of ten was used, and off-cycle $f_{\text{leak}}$ was set as the value for on-cycle.

The risk calculations were based on a single two-hour service visit in one year, the only occupant is the service person and SOIs elsewhere in the room are ignored. The results of the calculations in terms of $f^*$ and $X$ are provided in Table 54, for the four different (unadjusted) applications detailed in Table 53. Also given are $f^*$ and $X$, relative to the respective values under normal operation.

<table>
<thead>
<tr>
<th>Application</th>
<th>Ignition frequency $f^*$ ($y^{-1}$)</th>
<th>$f^<em>$(serv)/$f^</em>$(norm)</th>
<th>$X$ (-)</th>
<th>$X$(serv)/$X$(norm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Integral chiller shop</td>
<td>1.2×10^{-7}</td>
<td>60</td>
<td>9.5×10^{-8}</td>
<td>45</td>
</tr>
<tr>
<td>Split freezer cabinet</td>
<td>1.0×10^{-6}</td>
<td>29</td>
<td>1.9×10^{-5}</td>
<td>21</td>
</tr>
<tr>
<td>Portable A/C</td>
<td>7.0×10^{-8}</td>
<td>308</td>
<td>6.6×10^{-8}</td>
<td>284</td>
</tr>
<tr>
<td>Split A/C</td>
<td>8.6×10^{-7}</td>
<td>352</td>
<td>1.4×10^{-5}</td>
<td>378</td>
</tr>
</tbody>
</table>

In all of the examples, the risk is seen to be higher for servicing activities compared to normal operation. This result is expected due to higher leak frequencies, more vulnerable SOI and larger flammable volumes caused by releases (in some cases) occurring from positions closer to the floor. For the chiller and freezer, the increased risk is relatively small compared to the air conditioners, and this is explained as follows. When the a/c is in on-cycle, there is usually a high room air speed, which offsets the corresponding high on-cycle $f_{\text{leak}}$ by producing much smaller flammable volumes, whilst both chiller and freezer produce no, or low very low air speeds in on-cycle. As such, the result of increasing the chiller or freezer off-cycle $f_{\text{leak}}$ to equal that at on-cycle is comparable to simply increasing the on-cycle time to $\phi_{\text{op}} \approx 1$, since the resulting $V^F t^F$
is about the same in either mode. However, the air conditioners produce a relatively large $V^F t^F$ during off-cycle which is normally offset with the low corresponding $f_{\text{leak}}$, so increasing off-cycle $f_{\text{leak}}$ greatly overwhelms the contribution of the on-cycle $V^F t^F$. This effect is amplified in the case of the split unit because of the lower release height.

In general, the differences between operating and servicing $f^*$ and $X$ are consistent with the review of incidents (Table 49) where very few apply to the operating mode. Nonetheless, by examining the contribution of each different variable that contributes to a higher risk during servicing can help identify preferred behaviour and use of tools that would minimise the hazard.

![Figure 180: Relationship between $\hat{C}_{f_{\text{max}}}$ and total $V^F t^F$ ($r^2 = 0.82$)](image1)

![Figure 181: Relationship between $\bar{C}_{f_{\text{max}}}$ and the proportion of $V^F t^F$ above 1 m](image2)

**5.5 General risk dependency on maximum concentration**

Intuitively, it is expected that higher $\bar{C}_{f_{\text{max}}}$ leads to a greater risk. In order to address this, results for $\bar{C}_{f_{\text{max}}}$ and subsequently $f^*$ were generated over a range of input parameters so they could be used to identify any general tendencies. First, the entire set of $V^F t^F$ results (Figure 159 to Figure 163) from the dispersion model was plotted against $\bar{C}_{f_{\text{max}}}$ to identify any general relationship. The data is shown in Figure 180 where a scattered s-curve is observed that is bisected by the LFL of R290. When $\bar{C}_{f_{\text{max}}}$ is above LFL ($\geq 0.041 \text{ kg m}^{-3}$) a high $V^F t^F$ always occur (i.e., $> 5000 \text{ m}^3 \text{ s}$), and below the LFL ($\leq 0.034 \text{ kg m}^{-3}$) only small $V^F t^F$ occur (i.e., $< 500 \text{ m}^3 \text{ s}$). This observation provides a useful indication that a small $V^F t^F$ is almost guaranteed.
if the conditions lead to $\overline{C}_{f,\text{max}} < \text{LFL}$. This was consistent with the earlier recommendation that equipment should be designed to maintain $\overline{C}_{f,\text{max}}$ to below about $\text{LFL} + 1.28$ to avoid local concentrations reaching LFL (Section 3.3).

Figure 182: Correlation between $P^* F^*$ and $V F^*$

Figure 183: Relationship between $P^* F^*$ and $C_{f,\text{max}} (r^2 = 0.82)$

The influence of SOI positioning was considered since the probability calculations in Section 2.1 require the room be divided into three levels of equal volume to account for the locality of SOI. In Section 5.2 (Figure 174) it was seen that the greater proportion of $f^*$ shifted from the lower level to the middle level as leak times got longer, which was commensurate with an overall reduction of $f^*$. Plotting the ratio of $V F^* F^*$ existing within the middle level (above 1 m and below 2 m) to total $V F^* F^*$ indicated when it is important to consider SOI positioned above floor level (Figure 181). For many situations a lower $\overline{C}_{f,\text{max}}$ is consistent with a greater proportion of $V F^* F^*$ occurs at the higher level. Although it was found that the greater part of $V F^* F^*$ and $f^*$ did occur within the middle level under these conditions, these corresponded to very low $\overline{C}_{f,\text{max}} (< \text{LFL} + 2)$. Such low $\overline{C}_{f,\text{max}}$ were due to long leak times and/or high $\bar{u}_{\text{rel}}$, consequently resulting in minute $f^*$ relative to $f^*$ from catastrophic leaks. However, under certain conditions there is a contrary tendency for an increasing proportion of $V F^* F^*$ to occur in the middle level as $\overline{C}_{f,\text{max}}$ rises. This is generally when $\overline{C}_{f,\text{max}} < \text{LFL}$ and very large flammable clouds develop that already occupy the entire lower level.
was determined from the same \( V^F t^F \) and \( \overline{C}_{f_{\text{max}}} \) data using other simple values: \( P_{\text{avail}} = 1 \), \( P_{\text{ps}} = 1 \), \( P_{\text{perc}} = 0.73 \), \( t_{\text{sol}} = 1 \text{ s} \), \( N_{\text{sol}} = 1 \), and \( N_{\text{e}} = 1 \). Figure 182 shows the linear relationship between \( P^{F*} \) and \( V^F t^F \), which was found to apply generally provided \( t_{\text{sol}} < 100 \text{ s} \), \( N_{\text{sol}} < 30 \). Given the linearity with \( V^F t^F \), \( P^{F*} \) was justifiably plotted against \( \overline{C}_{f_{\text{max}}} \) (Figure 183). As with the observations of \( \overline{C}_{f_{\text{max}}} \) and \( V^F t^F \), similar generalisations are made: where \( \overline{C}_{f_{\text{max}}} > \text{LFL} \) always \( P^{F*} > 2 \times 10^{-4} \), below LFL, \( P^{F*} < 1 \times 10^{-4} \), and the range \( 1 \times 10^{-4} < P^{F*} < 2 \times 10^{-4} \) generally sits on the LFL. Assuming a constant \( \dot{f}_{\text{leak}} \) the trends in \( P^{F*} \) are considered to apply equally to \( f^* \).

\[ \begin{align*}
\Delta p^0_{\text{Rm}}(\text{kPa}) & \quad \text{ or } L_{\text{occ}}(\text{kW m}^{-2})^\gamma \\
\overline{m}^F(\text{kg}) & \quad 1 \times 10^{-3} \quad 1 \times 10^{-1} \quad 1 \times 10^{0}
\end{align*} \]

\[ \begin{align*}
\Delta p^0_{\text{Rm}}(\text{kPa}) & \quad \text{ or } L_{\text{occ}}(\text{kW m}^{-2})^\gamma \\
\overline{C}_{f_{\text{max}}}(\text{kg m}^{-3}) & \quad 0 \quad 0.05 \quad 0.1 \quad 0.15
\end{align*} \]

*Figure 184: Relationship between mean flammable mass and consequence \((r^2 = 0.99)\)  
*Figure 185: Correlation between \( \overline{C}_{f_{\text{max}}} \) and consequence \((r^2 = 0.83)\)*

Trends in \( I_{\text{occ}} \) and \( \Delta p^0_{\text{Rm}} \) with \( \overline{m}^F \) as the selected flammable quantity were plotted in Figure 184, and these also produced a linear relationship; this is expected since both \( I_{\text{occ}} \) and \( \Delta p^0_{\text{Rm}} \) are directly dependent upon the available energy within a given mass of mixture. As with \( P^{F*} \), the relationship between \( I_{\text{occ}} \) and \( \Delta p^0_{\text{Rm}} \) and \( \overline{C}_{f_{\text{max}}} \) was plotted (Figure 185). It is seen that below the LFL, the consequence is often insignificant \((< 1 \text{ kPa and } < 1 \text{ s (kW m}^{-2})^\gamma\)) yet beginning to exceed 50 kPa and 100 s \((\text{kW m}^{-2})^\gamma\) at \( \overline{C}_{f_{\text{max}}} \) slightly above the LFL.
SECTION 6: CONCLUSIONS

6.1 Summary of work
Observations from a literature review were used to identify the tasks necessary to enable a more comprehensive QRA method of RAC equipment; more precise leak characterisation, probability calculations for localised conditions, behaviour of the dispersion process of refrigerants leaks (focussing on the influence of airflows), consequence analysis, and criteria for judging the acceptability of risk. Initially, a hazard analysis was used to derive a methodology for calculating ignition and consequence frequencies, and an acceptability criteria based on the chosen risk measures. Methods were derived for characterising the likelihood of leakage, active sources of ignition and flammable concentrations. The latter were based on characterisation of room concentrations from a refrigerant leak, for which empirical correlations were developed using measurements from a series of experiments. In the event of ignition, consequences were quantified by modelling thermal intensity and room overpressures.

Overall risk of RAC equipment was found to be influenced by a large number of variables and the QRA accounts for around 50 of these, which can be adjusted according to the design, construction and behaviour of the equipment and the local environment. The QRA model is broadly generic within the range of characteristics associated with RAC equipment normally used in domestic and commercial contexts provided the conditions are within the applicability limits of the sub-models. Examining QRA model outputs led to identification of design features that can minimise the risk associated with RAC equipment.

6.2 Key findings and originality
Several important findings were made, particularly with respect to the dispersion of a leak of flammable refrigerant. For any specific application, no single design feature can be practically used to guarantee low risk because other combinations of factors may result in the opposite. However, the QRA model did indicate a number of techniques – preferably used in combination – that can contribute to improved safety. The risk associated with equipment already conforming to safety standards can vary widely, and checking various design features can lead to a marked reduction in risk, by a factor of up to $10^2$ or more.

Although not exactly comparable, the QRA model was used to "re-calculate" published QRAs which use the conventional approach; under most situations the re-calculated risk was significantly lower by up to a factor of $10^3$. This was due to the current method having greater
precision on account of the wide use of empirical data and discretisation of conditions, thereby avoiding the compounding effect of numerous worse case assumptions. Also, the published QRAs presented results typically for one or two fixed set of environmental conditions thereby negating any anticipated changes over time. Moreover, certain combinations of equipment and environmental characteristics could potentially create an unexpectedly high risk situation, and by adopting very general assumptions, and neglecting particular influences, a high risk situation could be overlooked.

Throughout the preparation of this study, there have been several important developments.

- **Evaluation of risk.** Several important measures of the “risk” were identified: frequency of ignition, frequency of fatality, maximum thermal intensity and overpressure, and additionally, the newly developed measures of integrated thermal intensity-frequency, integrated overpressure-frequency, and in combination, a generalised “risk factor”. The calculation of these risk measures was based directly on leakage characteristics, flammable concentration and SOI probabilities, and time fractions of transient conditions. The criteria for establishing whether results constituted a “safe” or “unsafe” risk were determined from knowledge of the risk associated with equipment having socially “acceptable” safety record, and tolerances specified by the national safety authorities, for example an ignition frequency lower than $10^{-6}$ y$^{-1}$ are considered to pose a “negligible” risk.

- **Leak frequency.** Existing QRAs have employed anecdotal data on leak frequency and normally categorised in terms of “catastrophic” leak and a “small” leak. As part of this study, data for released quantities and leaks duration was made available. This data was employed to develop a method for calculating leak frequencies and durations over a wide range of leak sizes as a function of the number of connections or joints associated with a system.

- **Probability models.** Existing QRA calculate ignition probabilities based on large-scale outdoor releases. Instead, an approach was developed to account precisely for all known SOI associated with the RAC equipment and others within a given room.

- **Concentrations from leakage.** A number of full-scale dispersion experiments were conducted, allowing the concentration development to be studied, whilst accounting for the influence of forced airflow and other parameters normally associated with RAC equipment. For example, by increasing mean air speed from 0.05 m s$^{-1}$ to 0.2 m s$^{-1}$ can reduce maximum floor concentrations from twice the LFL to 0.25 × LFL. Empirical correlations were developed to estimate concentrations arising from catastrophic leaks (where the entire charge is released in 3 minutes) and releases over longer durations. Complimentary to their application in the dispersion model, these formulæ can be employed for evaluating the design
of RAC equipment in terms of maximum concentrations resulting from leaks, and used iteratively to establish equipment design that avoids flammable floor concentrations.

- Modelling dispersion of a leak. Quantification of flammable volume, flammable mass and flammable time was achieved through a numerical model that addressed the descending plume stage, spreading plume and decaying cloud parts.

- Calculation of severity of consequences. Consequences of ignition were identified as thermal intensity and room overpressure. In particular, the explosion process (leading to overpressure) was evaluated with a newly developed model of burning of a partially flammable cloud in a confined and/or vented space. Comparison with experimental data showed good agreement, typically within ±100% of the maximum overpressure.

- Determination of risk. The influence of variables associated with equipment design, external conditions and operation, on the risk measures was assessed with the QRA model. The observations were used to establish general guidance for the design of equipment to minimise the risk, providing additional requirements to those specified in the safety standards.

Whilst the significance of the contribution offered by this study is largely subjective, considering the scale of the related applications implies a potentially far-reaching influence. International agreements on ozone depleting and global warming gases (typically fluorocarbons) mean that many refrigerants used historically or indeed recently introduced, are becoming increasingly restricted. One option for RAC equipment manufacturers is to build systems that use environmentally benign gases, which include HCs (UNEP, 2002). Currently, about one-quarter of new domestic refrigerators produced globally use R600a. Within Europe an increasing number of producers of a variety of small and medium-sized commercial cabinets have adopted R290, and manufacturers of split and integral air conditioners offer systems with R290 and R1270. As regional legislation further restricts fluorinated refrigerants, it is likely that the uptake of HCs will accelerate. Furthermore, developing countries and countries in economic transition are adopting HCs (in addition to other alternative refrigerants). For domestic and commercial RAC sectors, estimates based on UNEP (2002) suggest around 150 million appliances are produced globally each year, and some 20% of these may soon be using flammable refrigerants. It is evident from these numbers that a significant reduction in potential incidents could be achieved by using a more comprehensive approach to safe design.

6.3 Limitations and further work

Use of the QRA model is limited to the range of conditions that can be reliably accounted for and the accuracy that the variables can be represented, as well as correct interpretation of input.
of RAC equipment in terms of maximum concentrations resulting from leaks, and used iteratively to establish equipment design that avoids flammable floor concentrations.

- **Modelling dispersion of a leak.** Quantification of flammable volume, flammable mass and flammable time was achieved through a numerical model that addressed the descending plume stage, spreading plume and decaying cloud parts.

- **Calculation of severity of consequences.** Consequences of ignition were identified as thermal intensity and room overpressure. In particular, the explosion process (leading to overpressure) was evaluated with a newly developed model of burning of a partially flammable cloud in a confined and/or vented space. Comparison with experimental data showed good agreement, typically within ±100% of the maximum overpressure.

- **Determination of risk.** The influence of variables associated with equipment design, external conditions and operation, on the risk measures was assessed with the QRA model. The observations were used to establish general guidance for the design of equipment to minimise the risk, providing additional requirements to those specified in the safety standards.

Whilst the significance of the contribution offered by this study is largely subjective, considering the scale of the related applications implies a potentially far-reaching influence. International agreements on ozone depleting and global warming gases (typically fluorocarbons) mean that many refrigerants used historically or indeed recently introduced, are becoming increasingly restricted. One option for RAC equipment manufacturers is to build systems that use environmentally benign gases, which include HCs (UNEP, 2002). Currently, about one-quarter of new domestic refrigerators produced globally use R600a. Within Europe an increasing number of producers of a variety of small and medium-sized commercial cabinets have adopted R290, and manufacturers of split and integral air conditioners offer systems with R290 and R1270. As regional legislation further restricts fluorinated refrigerants, it is likely that the uptake of HCs will accelerate. Furthermore, developing countries and countries in economic transition are adopting HCs (in addition to other alternative refrigerants). For domestic and commercial RAC sectors, estimates based on UNEP (2002) suggest around 150 million appliances are produced globally each year, and some 20% of these may soon be using flammable refrigerants. It is evident from these numbers that a significant reduction in potential incidents could be achieved by using a more comprehensive approach to safe design.

### 6.3 Limitations and further work

Use of the QRA model is limited to the range of conditions that can be reliably accounted for and the accuracy that the variables can be represented, as well as correct interpretation of input.
values. As with any numerical model, extending the range of applicability and raising “apparent” accuracy can be achieved through improving sub-models, and their integration into the overall model. Similarly, wherever empirical correlations or data have been employed, there is always scope for improvement with additional data, and for a wider range of variables.

The following list suggests options for improvements.

- **RAC equipment construction.** Currently, the QRA can handle a limited number of design and construction features associated with equipment. Some equipment differs from the generic representation established earlier, such as when the entire system is located within housing, where piping and components are routed around a room, and where there are multiple or shifting air discharge ducts. Therefore, a broader range of equipment characteristics could be accounted for if more comprehensive correlations were developed from new measurements, and the appropriate input options provided.

- **Probability calculations.** It is possible to develop more sophisticated probability models, which could be employed where variables that impact on the behaviour of leakage, dispersion, and activation of SOI simultaneously are linked (such as occupancy, on-cycle times and presence of SOIs). Further, ageing of equipment will increase the frequency of leakage, mechanical faults and integrity of SOI protection. This aspect has not been addressed, although given that such failures are known to change with time, risk is likely to increase over equipment lifetime. A proper assessment of ageing would again be drawn from empirical data. Also, a better understanding of the likelihood of flammable concentrations reaching SOI within their casing or housing would also be valuable, rather than assuming flammable concentrations are always immediately available.

- **Release characteristics.** Currently, leaks are assumed as constant mass flow rate, although it is known that leaks often occur intermittently and variation of conditions can lead to changes in the leak aperture and subsequently mass flow. Better characterisation of leak patterns is needed, and how equipment operation behaves in response to leakage (e.g., how release rates vary and the relationship between charge deficit, pressures and system controls). Also leakage is known to be influenced by quality of materials and workmanship, which could also be characterised.

- **Dispersion modelling.** Additional experimental data is required for effects of leaks from various equipment components (rather than a diffuser for worse-case). More comprehensive data and modelling are required on dispersion behaviour due to the effect of multiple sources of airflow occurring simultaneously within the room. Airflows from RAC equipment are normally at a higher or lower temperature than the ambient, and it is likely that the resultant
difference in air buoyancy affects dispersion characteristics, also demanding more comprehensive evaluation of infiltration for rooms within buildings.

- **Uncertainty analysis.** It would be beneficial in terms of interpretation of results to include some indication of the uncertainty associated with each part of the QRA, and accordingly, the final risk measures.

- **Consequence.** Ignition within either RAC equipment housing or other SOI-containing enclosures can produce high overpressures, shock waves and flying fragments; including these elements would improve consequence analysis. Depending upon its geometry, ignition may be extinguished if the flame front is prevented from reaching a larger flammable mixture by small apertures. Physical examination of SOI-containing equipment and associated measurements would be needed for this.

- **General validation.** It would be useful to conduct a series of experiments where, under different equipment/room configurations, a "realistic" release is made and potential SOI are activated. The measurements for all intermediate parameters (such as release rates, air speeds, concentrations, overpressures, heat flux, etc) can be compared against the output of the various sub-models, ultimately to help validate the overall QRA model. Additionally, the model could be adapted to permit inputting of measurement data from leak simulation tests (as obliged by certain safety standards) so that dispersion modelling could be calibrated for the equipment under consideration.

Lastly, of interest to equipment producers, is the capital and operational costs of including certain design features that could reduce risk. Therefore, addressing this cost effectiveness would provide useful and practical information.
REFERENCES


Hymes, I., Boydell, W., Prescott, B. Thermal radiation: Physiological and pathological effects. Institution of Chemical Engineers, Rugby. 1996.


Private communication regarding mobile air conditioner fire incident in the Philippines, 2002 (details withheld upon request).


## APPENDICES

### Appendix 1: Data from published QRAs

**Table 55: Summary of QRA results for stationary RAC equipment**

<table>
<thead>
<tr>
<th>Reference</th>
<th>Application</th>
<th>Refrigerant</th>
<th>Activity</th>
<th>Consequence</th>
<th>Frequency (y⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Goetzler et al (1998)</td>
<td>Split system residential heat pump</td>
<td>R32/R134a</td>
<td>operation</td>
<td>fire on AHU/Cond U</td>
<td>3.2×10⁻⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation</td>
<td>fire in cellar/garage</td>
<td>8.9×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation</td>
<td>fire in attic</td>
<td>8.2×10⁻⁹</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation</td>
<td>fire in closet</td>
<td>9.5×10⁻⁹</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation</td>
<td>fire in room</td>
<td>9.0×10⁻¹⁰</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service</td>
<td>fire</td>
<td>2.6×10⁻⁶</td>
</tr>
<tr>
<td>Woller and Seiler (1999)</td>
<td>domestic heat pump</td>
<td>R290</td>
<td>operation</td>
<td>fatality</td>
<td>8.3×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service</td>
<td>fatality</td>
<td>2.0×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service</td>
<td>injury</td>
<td>7.0×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td>indirect s/m system</td>
<td>R1270</td>
<td>operation</td>
<td>fatality</td>
<td>8.5×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service</td>
<td>damage</td>
<td>7.0×10⁻⁷</td>
</tr>
<tr>
<td>Elbers and Verwoerd (1997)</td>
<td>domestic heat pump</td>
<td>R290</td>
<td>operation (7.5 m³ room)</td>
<td>occupant fatality</td>
<td>1.8×10⁻⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation (50 m³ room)</td>
<td>occupant fatality</td>
<td>8.0×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation (7.5 m³ room)</td>
<td>public fatality</td>
<td>2.1×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service (7.5 m³ room)</td>
<td>fatality</td>
<td>3.8×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service (50 m³ room)</td>
<td>fatality</td>
<td>3.0×10⁻⁷</td>
</tr>
<tr>
<td>van Blanken and Verwoerd (2001)</td>
<td>ice cream freezer</td>
<td>R290</td>
<td>operation</td>
<td>fatality within 2 m</td>
<td>1.1×10⁻⁶</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation</td>
<td>fatality within 3 m</td>
<td>7.2×10⁻⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation</td>
<td>fatality within 5 m</td>
<td>2.5×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation</td>
<td>fatality within 8 m</td>
<td>1.8×10⁻⁸</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service</td>
<td>fatality</td>
<td>2.3×10⁻⁷</td>
</tr>
<tr>
<td>van Gerven and Verwoerd (1996)</td>
<td>bulk milk tank</td>
<td>R290</td>
<td>operation (50 m³ room)</td>
<td>occupant fatality</td>
<td>6.9×10⁻⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation (100 m³ room)</td>
<td>occupant fatality</td>
<td>1.0×10⁻⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation (50 m³ room)</td>
<td>public fatality</td>
<td>1.2×10⁻⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service (50 m³ room)</td>
<td>fatality</td>
<td>9.0×10⁻⁷</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service (100 m³ room)</td>
<td>fatality</td>
<td>4.0×10⁻⁷</td>
</tr>
<tr>
<td>Reference</td>
<td>Application</td>
<td>Refrigerant</td>
<td>Activity</td>
<td>Consequence</td>
<td>Frequency ($y^{-1}$)</td>
</tr>
<tr>
<td>-----------</td>
<td>-----------------------</td>
<td>-------------</td>
<td>---------------------------</td>
<td>----------------------</td>
<td>----------------------</td>
</tr>
<tr>
<td></td>
<td>domestic heat pump</td>
<td>1.5 kg R290</td>
<td>operation (20 m$^3$ room)</td>
<td>occupant fatality</td>
<td>$2.3 \times 10^{-7}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>operation (80 m$^3$ room)</td>
<td>occupant fatality</td>
<td>$1.0 \times 10^{-7}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service (20 m$^3$ room)</td>
<td>public fatality</td>
<td>$2.6 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service (80 m$^3$ room)</td>
<td>fatality</td>
<td>$5.0 \times 10^{-7}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>fatality</td>
<td>$4.0 \times 10^{-7}$</td>
</tr>
<tr>
<td></td>
<td>Cold store</td>
<td>5.0 kg R290</td>
<td>operation</td>
<td>Fire and overpressure</td>
<td>$1.3 \times 10^{-5}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service</td>
<td>Fire and overpressure</td>
<td>$1.0 \times 10^{-9}$</td>
</tr>
<tr>
<td></td>
<td>Chest freezer</td>
<td>0.1 kg R290</td>
<td>operation</td>
<td>Fire and overpressure</td>
<td>$6.7 \times 10^{-7}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Fire and overpressure</td>
<td>$8.4 \times 10^{-6}$</td>
</tr>
<tr>
<td></td>
<td>Multideck freezer</td>
<td>0.4 kg R290</td>
<td>operation</td>
<td>Fire and overpressure</td>
<td>$2.6 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Fire and overpressure</td>
<td>$4.2 \times 10^{-7}$</td>
</tr>
<tr>
<td></td>
<td>Chiller cabinet</td>
<td>1.5 kg R290</td>
<td>operation</td>
<td>Fire and overpressure</td>
<td>$6.6 \times 10^{-9}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>service</td>
<td>Fire and overpressure</td>
<td>$9.6 \times 10^{-6}$</td>
</tr>
<tr>
<td></td>
<td>Remot draft chiller</td>
<td>0.2 kg R600a</td>
<td>operation</td>
<td>Fire and overpressure</td>
<td>$4.8 \times 10^{-8}$</td>
</tr>
<tr>
<td></td>
<td>Shelf draft chiller</td>
<td>0.1 kg R600a</td>
<td>operation</td>
<td>Fire and overpressure</td>
<td>$4.8 \times 10^{-8}$</td>
</tr>
<tr>
<td></td>
<td>Bottle cooler</td>
<td>0.3 kg R600a</td>
<td>operation</td>
<td>Fire and overpressure</td>
<td>$4.8 \times 10^{-8}$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Source</th>
<th>Equipment details</th>
<th>Leak frequency ($y^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>TR 14739 (2004)</td>
<td>Domestic fridge during operation</td>
<td>$2.9 \times 10^{-9}$</td>
</tr>
<tr>
<td></td>
<td>Domestic fridge during transport</td>
<td>$7.1 \times 10^{-6}$</td>
</tr>
<tr>
<td>Jansen and van Gerwen (1996)</td>
<td>Transport refrigeration. full bore hole</td>
<td>$5.5 \times 10^{-5}, 2.4 \times 10^{-6}$</td>
</tr>
<tr>
<td></td>
<td>Transport refrigeration. 10% full bore</td>
<td>$1.8 \times 10^{-4}, 8.0 \times 10^{-5}$</td>
</tr>
<tr>
<td></td>
<td>Transport refrigeration. 1% full bore</td>
<td>$5.5 \times 10^{-4}, 2.4 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td>Transport refrigeration. servicing</td>
<td>$1.0 \times 10^{-3}$</td>
</tr>
<tr>
<td>Elbers and Verwoerd (1997)</td>
<td>Heat pump full bore rupture</td>
<td>$6.0 \times 10^{-5}$</td>
</tr>
<tr>
<td></td>
<td>Heat pump small leakage</td>
<td>$6.0 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td>Heat pump servicing</td>
<td>$3.8 \times 10^{-4}$</td>
</tr>
<tr>
<td>Wolfer and Seiler (1999)</td>
<td>Supermarket chiller continuous</td>
<td>$5.7 \times 10^{-5}$</td>
</tr>
<tr>
<td></td>
<td>Supermarket chiller spontaneous</td>
<td>$6.3 \times 10^{-6}$</td>
</tr>
<tr>
<td></td>
<td>Heat pump continuous</td>
<td>$7.0 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td>Heat pump spontaneous</td>
<td>$2.9 \times 10^{-4}$</td>
</tr>
<tr>
<td>Goetzler et al (1998)</td>
<td>Split heat pump, total</td>
<td>$9.6 \times 10^{-3}$</td>
</tr>
<tr>
<td></td>
<td>Split heat pump, total catastrophic</td>
<td>$4.6 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td>Split heat pump, servicing</td>
<td>$1.0 \times 10^{-3}$</td>
</tr>
</tbody>
</table>

Table 56: Leak frequencies used in previous studies
<table>
<thead>
<tr>
<th>ADL (1998)</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Cold stores, total</td>
<td>$4.5 \times 10^3$</td>
</tr>
<tr>
<td>Cold stores, servicing</td>
<td>$1.4 \times 10^5$</td>
</tr>
<tr>
<td>Supermarket chillers, total</td>
<td>$2.0 \times 10^4$</td>
</tr>
<tr>
<td>Supermarket chillers, servicing</td>
<td>$4.4 \times 10^3$</td>
</tr>
<tr>
<td>Chest freezer, total</td>
<td>$9.3 \times 10^6$</td>
</tr>
<tr>
<td>Multi-deck freezer, total</td>
<td>$1.2 \times 10^4$</td>
</tr>
<tr>
<td>Remote draft chiller, total</td>
<td>$6.0 \times 10^4$</td>
</tr>
<tr>
<td>Shelf draft chiller, total</td>
<td>$4.0 \times 10^3$</td>
</tr>
<tr>
<td>Bottle cooler, total</td>
<td>$2.0 \times 10^{-5}$</td>
</tr>
</tbody>
</table>
### Appendix 2: Data for dispersion experiments

#### Table 57: Location of sensors and corresponding representative volume ratio

<table>
<thead>
<tr>
<th>Location (m)</th>
<th>Sampling point</th>
<th>Ratio to define representative volume for anemometer location (AL), $V_r / V_{ref}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x=3.35$</td>
<td>$y=3.35$</td>
<td>$z=0$</td>
</tr>
<tr>
<td>$x=3.35$</td>
<td>$y=3.35$</td>
<td>$z=1$</td>
</tr>
<tr>
<td>$x=3.35$</td>
<td>$y=3.35$</td>
<td>$z=2$</td>
</tr>
<tr>
<td>$x=0.5$</td>
<td>$y=0.5$</td>
<td>$z=0$</td>
</tr>
<tr>
<td>$x=3.35$</td>
<td>$y=0.5$</td>
<td>$z=3$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=0.5$</td>
<td>$z=0$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=0.5$</td>
<td>$z=1$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=3.35$</td>
<td>$z=0$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=3.35$</td>
<td>$z=1$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=6.2$</td>
<td>$z=0$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=6.2$</td>
<td>$z=1$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=6.2$</td>
<td>$z=2$</td>
</tr>
<tr>
<td>$x=6.2$</td>
<td>$y=6.2$</td>
<td>$z=3$</td>
</tr>
<tr>
<td>$x=3.35$</td>
<td>$y=6.2$</td>
<td>$z=0$</td>
</tr>
<tr>
<td>$x=3.35$</td>
<td>$y=6.2$</td>
<td>$z=1$</td>
</tr>
<tr>
<td>$x=0.5$</td>
<td>$y=6.2$</td>
<td>$z=0$</td>
</tr>
<tr>
<td>$x=0.5$</td>
<td>$y=3.35$</td>
<td>$z=0$</td>
</tr>
</tbody>
</table>

*Figure 186: Room orientation and reference directions*
Table 58: Parameter values and principle results for CO\textsubscript{2} release tests

| Test no | Unit type | Unit position | Unit base (m) | Unit airflow (m\textsuperscript{3} \textpersecond) | Airflow direction | Fan run time (s) | Fan disch. (m) | Mean room velocity (m \textpersecond) | Release position | Release height (m) | Release time (s) | Mass, cylinder (kg) | \( C_{f_{\text{max}}} \) (kg m\textsuperscript{3}) | Mass, room (kg) | \( C_{f_{\text{max}}} \) (kg m\textsuperscript{3}) |
|---------|-----------|---------------|--------------|-----------------|------------------|----------------|---------------|-------------------------------|----------------|----------------|----------------|----------------|-------------------|-------------------|-----------------|-------------------|
| 9        | none      | n/a           | 1.95         | 0               | n/a              | 0              | 1.95          | 0.048                         | beneath         | 1.95           | 240             | 1.00               | 0.008             | 1.07              | 0.046           | 0.041              |
| 10       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.198                         | beneath         | 1.95           | 210             | 1.40               | 0.007             | 0.94              | 0.018           | 0.009              |
| 11       | AHU#1     | Wall          | 2.05         | 0.35            | horizontal       | 210            | 2.35          | 0.304                         | beneath         | 1.95           | 210             | 1.25               | 0.007             | 0.93              | 0.009           | 0.007              |
| 12       | AHU#1     | Wall          | 2.05         | 0.52            | horizontal       | 210            | 2.35          | 0.427                         | beneath         | 1.95           | 210             | 1.50               | 0.008             | 1.08              | 0.010           | 0.009              |
| 13       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.267                         | beneath         | 1.95           | 210             | 1.36               | 0.007             | 0.98              | 0.038           | 0.033              |
| 14       | AHU#1     | Wall          | 2.05         | 0.26            | downwards        | 210            | 2.35          | 0.228                         | beneath         | 1.95           | 210             | 1.18               | 0.008             | 1.07              | 0.012           | 0.008              |
| 15       | AHU#1     | Wall          | 2.05         | 0.26            | upwards          | 210            | 2.35          | 0.226                         | beneath         | 1.95           | 210             | 1.18               | 0.008             | 1.03              | 0.026           | 0.016              |
| 16       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.220                         | beneath         | 1.95           | 210             | 0.38               | 0.003             | 0.39              | 0.004           | 0.003              |
| 17       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.268                         | beneath         | 1.95           | 210             | 2.00               | 0.015             | 2.04              | 0.031           | 0.024              |
| 18       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.240                         | beside end      | 1.95           | 210             | 1.00               | 0.008             | 1.05              | 0.028           | 0.013              |
| 19       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.268                         | beneath         | 1.30           | 210             | 1.30               | 0.010             | 1.31              | 0.038           | 0.029              |
| 20       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.219                         | beneath         | 0.40           | 210             | 1.25               | 0.008             | 1.11              | 0.083           | 0.073              |
| 21       | AHU#1     | Wall          | 0.50         | 0.26            | horizontal       | 210            | 0.80          | 0.248                         | beneath         | 0.40           | 210             | 1.18               | 0.008             | 1.14              | 0.024           | 0.012              |
| 22       | AHU#1     | Wall          | 1.40         | 0.26            | horizontal       | 210            | 1.70          | 0.196                         | beneath         | 1.30           | 210             | 1.25               | 0.009             | 1.26              | 0.045           | 0.019              |
| 23       | AHU#1     | Wall          | 1.40         | 0.26            | horizontal       | 210            | 1.70          | 0.265                         | beneath         | 0.40           | 210             | 1.32               | 0.009             | 1.21              | 0.106           | 0.064              |
| 24       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.264                         | beneath         | 1.95           | 210             | 1.28               | 0.009             | 1.15              | 0.032           | 0.015              |
| 25       | AHU#1     | Wall          | 2.05         | 0.26            | horizontal       | 210            | 2.35          | 0.215                         | beneath         | 1.95           | 210             | 1.30               | 0.007             | 1.03              | 0.030           | 0.027              |
| 26       | AHU#1     | Wall          | 2.05         | 0.11            | horizontal       | 210            | 2.35          | 0.097                         | beneath         | 1.95           | 210             | 1.30               | 0.010             | 1.31              | 0.033           | 0.033              |
| 27       | AHU#1     | Wall          | 2.05         | 0.14            | horizontal       | 210            | 2.35          | 0.138                         | beneath         | 1.95           | 210             | 1.40               | 0.009             | 1.24              | 0.044           | 0.038              |
| 28       | AHU#1     | wall          | 2.05         | 0.11            | horizontal       | 210            | 2.35          | 0.079                         | beneath         | 1.95           | 210             | 1.35               | 0.009             | 1.21              | 0.061           | 0.049              |
| Test no | Unit type | Unit position | Unit base (m) | Unit airflow (m³ s⁻¹) | Airflow direction | Fan run time (s) | Fan disch. (m) | Mean room velocity (m s⁻¹) | Release position | Release height (m) | Release time (s) | Mass, cylinder (kg) | Mass, room (kg) | $\bar{C}_{r_m}$ (kg m⁻³) | $\bar{C}_{f_{max}}$ (kg m⁻³) |
|---------|-----------|---------------|---------------|------------------------|-------------------|----------------|----------------|--------------------------|----------------|----------------|----------------|-----------------|----------------|----------------|----------------|----------------|
| 29      | AHU#1     | wall          | 2.05          | 0.26                   | horizontal        | 210            | 2.35          | 0.202                    | beneath        | 1.95           | 420            | 0.93            | 0.007           | 0.90           | 0.29           | 0.025           |
| 30      | AHU#1     | wall          | 2.05          | 0.26                   | horizontal        | 210            | 2.35          | 0.199                    | beneath        | 1.95           | 210            | 1.15            | 0.008           | 1.11           | 0.027           | 0.012           |
| 31      | AHU#1     | wall          | 2.05          | 0.26                   | horizontal        | 210            | 2.35          | 0.200                    | room corner    | 1.95           | 210            | 1.03            | 0.010           | 1.30           | 0.046           | 0.030           |
| 32      | AHU#2     | wall          | 2.05          | 0.23                   | horizontal        | 210            | 2.05          | 0.297                    | beneath        | 1.95           | 210            | 1.24            | 0.008           | 1.13           | 0.019           | 0.010           |
| 33      | AHU#2     | wall          | 2.05          | 0.17                   | horizontal        | 210            | 2.05          | 0.255                    | beneath        | 1.95           | 210            | 1.21            | 0.009           | 1.16           | 0.022           | 0.013           |
| 34      | AHU#2     | wall          | 2.05          | 0.23                   | horizontal        | 210            | 2.05          | 0.323                    | beneath        | 0.40           | 210            | 1.13            | 0.008           | 1.03           | 0.056           | 0.029           |
| 35      | AHU#3     | ceiling       | 2.50          | 0.38                   | horizontal        | 210            | 2.50          | 0.263                    | beneath        | 2.20           | 210            | 0.60            | 0.005           | 0.62           | 0.007           | 0.005           |
| 36      | AHU#3     | ceiling       | 2.50          | 0.38                   | horizontal        | 210            | 2.50          | 0.281                    | beneath        | 2.20           | 210            | 2.02            | 0.013           | 1.72           | 0.017           | 0.015           |
| 37      | AHU#3     | ceiling       | 2.50          | 0.38                   | horizontal        | 210            | 2.50          | 0.234                    | wall           | 0.40           | 210            | 1.04            | -               | -              | -               | -               |
| 38      | Cabint#1  | floor         | 0.30          | 0.35                   | horizontal        | 210            | 0.30          | 0.254                    | beside end     | 0.20           | 210            | 1.24            | 0.005           | 0.72           | 0.020           | 0.008           |
| 39      | Cabint#1  | floor         | 0.30          | 0.19                   | horizontal        | 210            | 0.30          | 0.133                    | beside end     | 0.20           | 210            | 2.31            | 0.017           | 2.30           | 0.093           | 0.079           |
| 40      | Cabint#1  | floor         | 0.30          | 0.19                   | horizontal        | 210            | 0.30          | 0.132                    | beside end     | 0.20           | 210            | 0.80            | 0.005           | 0.67           | 0.052           | 0.038           |
| 41      | Cabint#2  | floor         | 2.00          | 0.05                   | horizontal        | 210            | 2.00          | 0.048                    | in cabinet     | (1.4)          | 210            | 0.80            | 0.007           | 0.97           | 0.095           | 0.084           |
| 42      | Cabint#2  | floor         | 2.00          | 0.05                   | horizontal        | 210            | 2.00          | 0.047                    | cond unit      | 2.10           | 210            | 0.70            | 0.005           | 0.67           | 0.018           | 0.012           |
| 43      | none      | n/a           | 1.95          | n/a                    | n/a               | 0              | 1.95          | 0.066                    | mid room       | 1.95           | 210            | 1.10            | 0.008           | 1.08           | 0.056           | 0.043           |
Appendix 3: Validation data from other sources

Notes for the Tables:
- Maximum concentrations were generally converted to $C_{f_{\text{max}}}$ using equation (315).
- Equipment abbreviations: "PAC" – portable air conditioner, "SAC" – split air conditioner, "Dehum" – dehumidifier
- Position abbreviations: "mid" – middle/centre of room, "corner" – a corner of the room, "below" – beneath unit, "beside" – beside unit, "wall" – against wall. "nr wall" – closer to wall than to the centre.
- No information was provided on unit position or unit base height in Clodic et al
- The first calculated $C_{f_{\text{max}}}$ was for a catastrophic release ($t_{r,\text{cat}} = 210$ s) and the second calculated $C_{f_{\text{max}}}$ was the conversion to the specified leak time.
- Room volume for Jabbour et al was 26 m$^3$.
- The data from Kataoka et al (c) was from CFD calculations, but was considered acceptable for this exercise.

Table 59: Test variables and calculated results for Aloisi
[a – Aloisi, September 1999, b – Aloisi, February 2000, c – Aloisi, May 1999]

<table>
<thead>
<tr>
<th>Test no</th>
<th>Unit type</th>
<th>Gas type</th>
<th>Unit base (m)</th>
<th>Unit air flow ($\text{m}^3 \text{s}^{-1}$)</th>
<th>Air outlet ($\text{m}^2$)</th>
<th>Fan run time (s)</th>
<th>Fan disch. (m)</th>
<th>Release position</th>
<th>Release height (m)</th>
<th>Release time (s)</th>
<th>Mass (kg)</th>
<th>Room volume ($\text{m}^3$)</th>
<th>$C_{f_{\text{max}}}$ ($\text{kg m}^{-3}$)</th>
<th>Calc $C_{f_{\text{max}}}$ ($\text{kg m}^{-3}$)</th>
<th>Calc $C_{f_{\text{max}}}$ ($\text{kg m}^{-3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 (a)</td>
<td>PAC</td>
<td>R290</td>
<td>0.10</td>
<td>0.039</td>
<td>0.050</td>
<td>180</td>
<td>0.38</td>
<td>nr wall, below</td>
<td>0.5</td>
<td>240</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0110</td>
<td>0.0082</td>
<td>0.0108</td>
</tr>
<tr>
<td>2 (a)</td>
<td>PAC</td>
<td>R290</td>
<td>0.50</td>
<td>0.039</td>
<td>0.050</td>
<td>180</td>
<td>0.78</td>
<td>nr wall, below</td>
<td>0.5</td>
<td>240</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0106</td>
<td>0.0262</td>
<td>0.0328</td>
</tr>
<tr>
<td>3 (a)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.10</td>
<td>0.042</td>
<td>0.058</td>
<td>180</td>
<td>0.28</td>
<td>nr wall, below</td>
<td>0.5</td>
<td>240</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0153</td>
<td>0.0072</td>
<td>0.0096</td>
</tr>
<tr>
<td>4 (a)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.50</td>
<td>0.042</td>
<td>0.058</td>
<td>180</td>
<td>0.78</td>
<td>nr wall, below</td>
<td>0.5</td>
<td>240</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0209</td>
<td>0.0261</td>
<td>0.0327</td>
</tr>
<tr>
<td>Test no</td>
<td>Unit type</td>
<td>Gas type</td>
<td>Unit base (m)</td>
<td>Unit air flow (m^3 s^-1)</td>
<td>Air outlet (m^2)</td>
<td>Fan run time (s)</td>
<td>Fan disch. (m)</td>
<td>Release position</td>
<td>Release height (m)</td>
<td>Release time (s)</td>
<td>Mass (kg)</td>
<td>Room volume (m^3)</td>
<td>Calc C_{f,max} (kg m^-3)</td>
<td>Calc t_{r,cat} (kg m^-3)</td>
<td>Calc C_{f,max} t_r (kg m^-3)</td>
</tr>
<tr>
<td>---------</td>
<td>-----------</td>
<td>----------</td>
<td>---------------</td>
<td>--------------------------</td>
<td>-----------------</td>
<td>-----------------</td>
<td>----------------</td>
<td>----------------</td>
<td>----------------</td>
<td>----------------</td>
<td>-----------</td>
<td>-----------------</td>
<td>--------------------------</td>
<td>--------------------------</td>
<td>---------------------------</td>
</tr>
<tr>
<td>5 (a)</td>
<td>SAC</td>
<td>R290</td>
<td>1.50</td>
<td>0.042</td>
<td>0.048</td>
<td>180</td>
<td>1.5</td>
<td>nr wall, below</td>
<td>0.5</td>
<td>240</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0223</td>
<td>0.0444</td>
<td>0.0547</td>
</tr>
<tr>
<td>1 (b)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.05</td>
<td>0.036</td>
<td>0.058</td>
<td>180</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>194</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0248</td>
<td>-</td>
<td>0.0157</td>
</tr>
<tr>
<td>2 (b)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.05</td>
<td>0.036</td>
<td>0.058</td>
<td>180</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>240</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0280</td>
<td>-</td>
<td>0.0259</td>
</tr>
<tr>
<td>3 (b)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.05</td>
<td>0.036</td>
<td>0.058</td>
<td>180</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>450</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0238</td>
<td>-</td>
<td>0.0379</td>
</tr>
<tr>
<td>4 (b)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.05</td>
<td>0.036</td>
<td>0.058</td>
<td>180</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>720</td>
<td>0.35</td>
<td>36.3</td>
<td>0.0140</td>
<td>-</td>
<td>0.0665</td>
</tr>
<tr>
<td>1 (c)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.05</td>
<td>0.010</td>
<td>0.058</td>
<td>180</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>282</td>
<td>0.23</td>
<td>36.3</td>
<td>0.0313</td>
<td>0.0441</td>
<td>0.0367</td>
</tr>
<tr>
<td>2 (c)</td>
<td>Dehum</td>
<td>R290</td>
<td>0.05</td>
<td>0.056</td>
<td>0.058</td>
<td>282</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>282</td>
<td>0.23</td>
<td>36.3</td>
<td>0.0067</td>
<td>0.0084</td>
<td>0.0081</td>
</tr>
<tr>
<td>3 (c)</td>
<td>PAC</td>
<td>R290</td>
<td>0.05</td>
<td>0.042</td>
<td>0.058</td>
<td>282</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>282</td>
<td>0.23</td>
<td>36.3</td>
<td>0.0033</td>
<td>0.0111</td>
<td>0.0103</td>
</tr>
<tr>
<td>4 (c)</td>
<td>PAC</td>
<td>R290</td>
<td>0.05</td>
<td>0.042</td>
<td>0.058</td>
<td>180</td>
<td>0.38</td>
<td>nr wall, beside</td>
<td>0.5</td>
<td>282</td>
<td>0.23</td>
<td>36.3</td>
<td>0.0076</td>
<td>0.0111</td>
<td>0.0103</td>
</tr>
</tbody>
</table>

Table 60: Test variables and calculated results for Clodic et al
<table>
<thead>
<tr>
<th>Test no</th>
<th>Unit type</th>
<th>Gas type</th>
<th>Unit air flow ($m^3/s$)</th>
<th>Air outlet ($m^2$)</th>
<th>Fan run time (s)</th>
<th>Fan disch. (m)</th>
<th>Release position</th>
<th>Release height (m)</th>
<th>Release time (s)</th>
<th>Mass (kg)</th>
<th>Room volume ($m^3$)</th>
<th>Calc $\overline{C}_{f,\text{max}}$ (kg m$^{-3}$)</th>
<th>Calc $\overline{C}<em>{f,\text{max}}$ ($t</em>{r,\text{out}}$ (kg m$^{-3}$))</th>
<th>Calc $\overline{C}_{f,\text{max}}$ ($t_r$ (kg m$^{-3}$))</th>
</tr>
</thead>
<tbody>
<tr>
<td>9 (a)</td>
<td>n/k</td>
<td>R600a</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>mid</td>
<td>0.05</td>
<td>384</td>
<td>0.13</td>
<td>16</td>
<td>0.0542</td>
<td>0.8346</td>
<td>0.4842</td>
</tr>
<tr>
<td>10 (a)</td>
<td>n/k</td>
<td>R600a</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>mid</td>
<td>0.05</td>
<td>192</td>
<td>0.13</td>
<td>16</td>
<td>0.0542</td>
<td>0.8346</td>
<td>0.9049</td>
</tr>
<tr>
<td>3 (a)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>mid</td>
<td>0.05</td>
<td>192</td>
<td>0.13</td>
<td>16</td>
<td>0.0329</td>
<td>0.8346</td>
<td>0.9049</td>
</tr>
<tr>
<td>7 (a)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.30</td>
<td>mid</td>
<td>0.30</td>
<td>192</td>
<td>0.13</td>
<td>16</td>
<td>0.0233</td>
<td>0.1391</td>
<td>0.1489</td>
</tr>
<tr>
<td>5 (a)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.70</td>
<td>mid</td>
<td>0.70</td>
<td>192</td>
<td>0.13</td>
<td>16</td>
<td>0.0110</td>
<td>0.0596</td>
<td>0.0631</td>
</tr>
<tr>
<td>6 (a)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>1.50</td>
<td>mid</td>
<td>1.50</td>
<td>192</td>
<td>0.13</td>
<td>16</td>
<td>0.0082</td>
<td>0.0278</td>
<td>0.0290</td>
</tr>
<tr>
<td>12 (a)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>corner</td>
<td>0.05</td>
<td>384</td>
<td>0.13</td>
<td>16</td>
<td>0.0576</td>
<td>0.8346</td>
<td>0.4842</td>
</tr>
<tr>
<td>13 (a)</td>
<td>n/k</td>
<td>R600a</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>corner</td>
<td>0.05</td>
<td>384</td>
<td>0.13</td>
<td>16</td>
<td>0.0813</td>
<td>0.8346</td>
<td>0.4842</td>
</tr>
<tr>
<td>14 (a)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>corner</td>
<td>0.05</td>
<td>384</td>
<td>0.13</td>
<td>16</td>
<td>0.0452</td>
<td>0.8346</td>
<td>0.4842</td>
</tr>
<tr>
<td>2 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.00</td>
<td>0.04</td>
<td>0</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>480</td>
<td>0.24</td>
<td>60</td>
<td>0.0466</td>
<td>0.1303</td>
<td>0.0659</td>
</tr>
<tr>
<td>3 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.00</td>
<td>0.04</td>
<td>0</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>360</td>
<td>0.24</td>
<td>60</td>
<td>0.0480</td>
<td>0.1303</td>
<td>0.0835</td>
</tr>
<tr>
<td>4 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.00</td>
<td>0.04</td>
<td>0</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>576</td>
<td>0.24</td>
<td>60</td>
<td>0.0548</td>
<td>0.1303</td>
<td>0.0567</td>
</tr>
<tr>
<td>6 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.01</td>
<td>0.04</td>
<td>360</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>360</td>
<td>0.24</td>
<td>60</td>
<td>0.0055</td>
<td>0.0339</td>
<td>0.0238</td>
</tr>
<tr>
<td>7 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.01</td>
<td>0.04</td>
<td>360</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>360</td>
<td>0.24</td>
<td>60</td>
<td>0.0548</td>
<td>0.1303</td>
<td>0.0835</td>
</tr>
<tr>
<td>8 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.00</td>
<td>0.04</td>
<td>576</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>576</td>
<td>0.24</td>
<td>60</td>
<td>0.0192</td>
<td>0.0339</td>
<td>0.0175</td>
</tr>
<tr>
<td>9 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.01</td>
<td>0.04</td>
<td>360</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>360</td>
<td>0.24</td>
<td>60</td>
<td>0.0329</td>
<td>0.0339</td>
<td>0.0238</td>
</tr>
<tr>
<td>15 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.01</td>
<td>0.04</td>
<td>360</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>360</td>
<td>0.24</td>
<td>60</td>
<td>0.0822</td>
<td>0.0218</td>
<td>0.0160</td>
</tr>
<tr>
<td>16 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.00</td>
<td>0.04</td>
<td>0</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>1440</td>
<td>0.24</td>
<td>60</td>
<td>0.0302</td>
<td>0.1303</td>
<td>0.0266</td>
</tr>
<tr>
<td>17 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.01</td>
<td>0.04</td>
<td>360</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>360</td>
<td>0.24</td>
<td>60</td>
<td>0.0218</td>
<td>0.0160</td>
<td>0.0266</td>
</tr>
<tr>
<td>18 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.00</td>
<td>0.04</td>
<td>0</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>720</td>
<td>0.24</td>
<td>60</td>
<td>0.0411</td>
<td>0.1303</td>
<td>0.0472</td>
</tr>
<tr>
<td>19 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.01</td>
<td>0.04</td>
<td>720</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>720</td>
<td>0.24</td>
<td>60</td>
<td>0.0411</td>
<td>0.0339</td>
<td>0.0151</td>
</tr>
<tr>
<td>Test no</td>
<td>Unit type</td>
<td>Gas type</td>
<td>Unit air flow (m³ s⁻¹)</td>
<td>Air outlet (m²)</td>
<td>Fan run time (s)</td>
<td>Fan disch. (m)</td>
<td>Release position</td>
<td>Release height (m)</td>
<td>Release time (s)</td>
<td>Mass (kg)</td>
<td>Room volume (m³)</td>
<td>Calc $\bar{C}<em>{f</em>{\text{max}}}$ (kg m⁻³)</td>
<td>Calc $C_{f_{\text{max}}}$ (kg m⁻³)</td>
<td>Calc $\bar{C}<em>{f</em>{\text{max}}}$ (kg m⁻³)</td>
</tr>
<tr>
<td>---------</td>
<td>-----------</td>
<td>----------</td>
<td>------------------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>------------------</td>
<td>-----------------</td>
<td>------------</td>
<td>-----------------</td>
<td>-------------------------------</td>
<td>-------------------------------</td>
<td>-------------------------------</td>
</tr>
<tr>
<td>20 (b)</td>
<td>PAC</td>
<td>R290</td>
<td>0.00</td>
<td>0.04</td>
<td>0</td>
<td>0.28</td>
<td>corner</td>
<td>0.28</td>
<td>360</td>
<td>0.24</td>
<td>60</td>
<td>0.0822</td>
<td>0.1303</td>
<td>0.0835</td>
</tr>
<tr>
<td>8 (c)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>mid</td>
<td>0.05</td>
<td>192</td>
<td>0.13</td>
<td>16</td>
<td>0.0110</td>
<td>0.0246</td>
<td>0.0256</td>
</tr>
<tr>
<td>11 (c)</td>
<td>n/k</td>
<td>R290</td>
<td>0.00</td>
<td>n/k</td>
<td>0</td>
<td>0.05</td>
<td>mid</td>
<td>0.05</td>
<td>192</td>
<td>0.13</td>
<td>16</td>
<td>0.0247</td>
<td>0.8346</td>
<td>0.9049</td>
</tr>
</tbody>
</table>

Table 61: Test variables and calculated results for Jabbour et al. 2003
<table>
<thead>
<tr>
<th>Test no</th>
<th>Gas type</th>
<th>Airflow source</th>
<th>Airflow (m³ s⁻¹)</th>
<th>Air outlet (m³)</th>
<th>Fan run time (s)</th>
<th>Fan disch. (m)</th>
<th>Release position</th>
<th>Release height (m)</th>
<th>Release time (s)</th>
<th>Mass (kg)</th>
<th>Calc $\bar{C}_{f,\text{max}}$ (kg m⁻³)</th>
<th>Calc $t_{r,\text{cut}}$ (kg m⁻³)</th>
<th>Calc $C_{r,\text{max}}$ (kg m⁻³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>239</td>
<td>0.150</td>
<td>0.0777</td>
<td>0.0627</td>
<td>0.0568</td>
</tr>
<tr>
<td>4</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>308</td>
<td>0.150</td>
<td>0.0560</td>
<td>0.0627</td>
<td>0.0468</td>
</tr>
<tr>
<td>5</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>169</td>
<td>0.150</td>
<td>0.0596</td>
<td>0.0627</td>
<td>0.0740</td>
</tr>
<tr>
<td>7</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>256</td>
<td>0.383</td>
<td>0.1174</td>
<td>0.1600</td>
<td>0.1375</td>
</tr>
<tr>
<td>8</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>406</td>
<td>0.383</td>
<td>0.0993</td>
<td>0.1600</td>
<td>0.0967</td>
</tr>
<tr>
<td>106</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>224</td>
<td>0.172</td>
<td>0.0939</td>
<td>0.0718</td>
<td>0.0684</td>
</tr>
<tr>
<td>107</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>232</td>
<td>0.172</td>
<td>0.1011</td>
<td>0.0718</td>
<td>0.0666</td>
</tr>
<tr>
<td>108</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.68</td>
<td>mid</td>
<td>0.54</td>
<td>208</td>
<td>0.172</td>
<td>0.1084</td>
<td>0.0718</td>
<td>0.0724</td>
</tr>
<tr>
<td>109</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.60</td>
<td>mid</td>
<td>0.50</td>
<td>111</td>
<td>0.086</td>
<td>0.0704</td>
<td>0.0391</td>
<td>0.0640</td>
</tr>
<tr>
<td>110</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.037</td>
<td>0</td>
<td>0.60</td>
<td>mid</td>
<td>0.50</td>
<td>54</td>
<td>0.043</td>
<td>0.0524</td>
<td>0.0195</td>
<td>0.0559</td>
</tr>
<tr>
<td>9</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.048</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>248</td>
<td>0.150</td>
<td>0.0470</td>
<td>0.0568</td>
<td>0.0501</td>
</tr>
<tr>
<td>10</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.048</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>151</td>
<td>0.150</td>
<td>0.0524</td>
<td>0.0568</td>
<td>0.0728</td>
</tr>
<tr>
<td>111</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.048</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>222</td>
<td>0.172</td>
<td>0.0506</td>
<td>0.0651</td>
<td>0.0625</td>
</tr>
<tr>
<td>112</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.048</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>254</td>
<td>0.172</td>
<td>0.0524</td>
<td>0.0651</td>
<td>0.0565</td>
</tr>
<tr>
<td>113</td>
<td>R600a</td>
<td>0</td>
<td>0.00</td>
<td>0.048</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>264</td>
<td>0.383</td>
<td>0.0524</td>
<td>0.1451</td>
<td>0.1221</td>
</tr>
<tr>
<td>114</td>
<td>R600a</td>
<td>0</td>
<td>0.05</td>
<td>0.048</td>
<td>275</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>275</td>
<td>0.383</td>
<td>0.0397</td>
<td>0.0312</td>
<td>0.0275</td>
</tr>
<tr>
<td>115</td>
<td>R600a</td>
<td>0</td>
<td>0.05</td>
<td>0.048</td>
<td>294</td>
<td>2.20</td>
<td>wall</td>
<td>2.20</td>
<td>294</td>
<td>0.383</td>
<td>0.0235</td>
<td>0.0376</td>
<td>0.0316</td>
</tr>
<tr>
<td>116</td>
<td>R600a</td>
<td>0</td>
<td>0.05</td>
<td>0.048</td>
<td>294</td>
<td>2.20</td>
<td>wall</td>
<td>2.20</td>
<td>227</td>
<td>0.383</td>
<td>0.0325</td>
<td>0.0396</td>
<td>0.0380</td>
</tr>
<tr>
<td>117</td>
<td>R600a</td>
<td>0</td>
<td>0.05</td>
<td>0.048</td>
<td>222</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>222</td>
<td>0.383</td>
<td>0.0181</td>
<td>0.0312</td>
<td>0.0304</td>
</tr>
<tr>
<td>11</td>
<td>R600a</td>
<td>0</td>
<td>0.05</td>
<td>0.048</td>
<td>222</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>315</td>
<td>0.383</td>
<td>0.0759</td>
<td>0.1451</td>
<td>0.1069</td>
</tr>
<tr>
<td>12</td>
<td>R600a</td>
<td>0</td>
<td>0.05</td>
<td>0.048</td>
<td>222</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>315</td>
<td>0.383</td>
<td>0.0759</td>
<td>0.1451</td>
<td>0.1069</td>
</tr>
<tr>
<td>Test no</td>
<td>Gas type</td>
<td>Airflow source</td>
<td>Air flow (m³ s⁻¹)</td>
<td>Air outlet (m²)</td>
<td>Fan run time (s)</td>
<td>Fan disch. (m)</td>
<td>Release position</td>
<td>Release height (m)</td>
<td>Release time (s)</td>
<td>Mass (kg)</td>
<td>C_f,max (kg m⁻³)</td>
<td>C_f,max t_r (kg m⁻³)</td>
<td>Calc C_f,max t_r (kg m⁻³)</td>
</tr>
<tr>
<td>---------</td>
<td>----------</td>
<td>----------------</td>
<td>------------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>------------------</td>
<td>-----------------</td>
<td>-----------</td>
<td>-----------------</td>
<td>-----------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>51</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>232</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>232</td>
<td>0.300</td>
<td>0.0548</td>
<td>0.0374</td>
<td>0.0353</td>
</tr>
<tr>
<td>52</td>
<td>R290</td>
<td>mech vent</td>
<td>0.08</td>
<td>0.031</td>
<td>267</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>267</td>
<td>0.450</td>
<td>0.0589</td>
<td>0.0326</td>
<td>0.0294</td>
</tr>
<tr>
<td>53</td>
<td>R290</td>
<td>mech vent</td>
<td>0.08</td>
<td>0.031</td>
<td>267</td>
<td>1.77</td>
<td>wall</td>
<td>1.00</td>
<td>267</td>
<td>0.550</td>
<td>0.0274</td>
<td>0.0205</td>
<td>0.0195</td>
</tr>
<tr>
<td>76</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>422</td>
<td>2.43</td>
<td>mid</td>
<td>0.60</td>
<td>422</td>
<td>0.280</td>
<td>0.0960</td>
<td>0.0652</td>
<td>0.0405</td>
</tr>
<tr>
<td>75</td>
<td>R290</td>
<td>mech vent</td>
<td>0.11</td>
<td>0.031</td>
<td>453</td>
<td>2.43</td>
<td>mid</td>
<td>0.60</td>
<td>453</td>
<td>0.605</td>
<td>0.0973</td>
<td>0.0510</td>
<td>0.0354</td>
</tr>
<tr>
<td>74</td>
<td>R290</td>
<td>mech vent</td>
<td>0.21</td>
<td>0.031</td>
<td>922</td>
<td>2.43</td>
<td>mid</td>
<td>0.60</td>
<td>922</td>
<td>1.050</td>
<td>0.0754</td>
<td>0.0482</td>
<td>0.0315</td>
</tr>
<tr>
<td>82</td>
<td>R290</td>
<td>mech vent</td>
<td>0.11</td>
<td>0.031</td>
<td>482</td>
<td>2.43</td>
<td>wall</td>
<td>0.60</td>
<td>482</td>
<td>0.605</td>
<td>0.0699</td>
<td>0.0510</td>
<td>0.0344</td>
</tr>
<tr>
<td>81</td>
<td>R290</td>
<td>mech vent</td>
<td>0.21</td>
<td>0.031</td>
<td>675</td>
<td>2.43</td>
<td>wall</td>
<td>0.60</td>
<td>675</td>
<td>1.050</td>
<td>0.0493</td>
<td>0.0482</td>
<td>0.0344</td>
</tr>
<tr>
<td>87</td>
<td>R290</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>308</td>
<td>2.43</td>
<td>wall</td>
<td>1.80</td>
<td>308</td>
<td>0.403</td>
<td>0.0384</td>
<td>0.0467</td>
<td>0.0378</td>
</tr>
<tr>
<td>88</td>
<td>R290</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>543</td>
<td>2.43</td>
<td>wall</td>
<td>1.80</td>
<td>543</td>
<td>0.410</td>
<td>0.0452</td>
<td>0.0448</td>
<td>0.0268</td>
</tr>
<tr>
<td>90</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>685</td>
<td>2.43</td>
<td>wall</td>
<td>1.80</td>
<td>685</td>
<td>0.490</td>
<td>0.0096</td>
<td>0.0311</td>
<td>0.0195</td>
</tr>
<tr>
<td>86</td>
<td>R290</td>
<td>mech vent</td>
<td>0.09</td>
<td>0.031</td>
<td>724</td>
<td>2.43</td>
<td>wall</td>
<td>1.80</td>
<td>724</td>
<td>0.700</td>
<td>0.0192</td>
<td>0.0213</td>
<td>0.0182</td>
</tr>
<tr>
<td>93</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>760</td>
<td>2.43</td>
<td>wall</td>
<td>2.20</td>
<td>760</td>
<td>0.555</td>
<td>0.0137</td>
<td>0.0284</td>
<td>0.0187</td>
</tr>
<tr>
<td>94</td>
<td>R290</td>
<td>mech vent</td>
<td>0.09</td>
<td>0.031</td>
<td>652</td>
<td>2.43</td>
<td>wall</td>
<td>2.20</td>
<td>652</td>
<td>0.770</td>
<td>0.0137</td>
<td>0.0186</td>
<td>0.0182</td>
</tr>
<tr>
<td>47</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>264</td>
<td>1.77</td>
<td>mid</td>
<td>0.60</td>
<td>264</td>
<td>0.300</td>
<td>0.0740</td>
<td>0.0374</td>
<td>0.0328</td>
</tr>
<tr>
<td>48</td>
<td>R290</td>
<td>mech vent</td>
<td>0.08</td>
<td>0.031</td>
<td>242</td>
<td>1.77</td>
<td>mid</td>
<td>0.60</td>
<td>242</td>
<td>0.450</td>
<td>0.0658</td>
<td>0.0326</td>
<td>0.0307</td>
</tr>
<tr>
<td>49</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>238</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>238</td>
<td>0.300</td>
<td>0.0493</td>
<td>0.0374</td>
<td>0.0348</td>
</tr>
<tr>
<td>50</td>
<td>R290</td>
<td>mech vent</td>
<td>0.08</td>
<td>0.031</td>
<td>257</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>257</td>
<td>0.450</td>
<td>0.0631</td>
<td>0.0326</td>
<td>0.0299</td>
</tr>
<tr>
<td>34</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>269</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>269</td>
<td>0.300</td>
<td>0.0596</td>
<td>0.0635</td>
<td>0.0538</td>
</tr>
<tr>
<td>35</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.06</td>
<td>0.031</td>
<td>427</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>427</td>
<td>0.450</td>
<td>0.0542</td>
<td>0.0534</td>
<td>0.0359</td>
</tr>
<tr>
<td>36</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>307</td>
<td>1.77</td>
<td>wall</td>
<td>1.00</td>
<td>307</td>
<td>0.300</td>
<td>0.0451</td>
<td>0.0614</td>
<td>0.0477</td>
</tr>
<tr>
<td>Test no</td>
<td>Gas type</td>
<td>Airflow source</td>
<td>Air flow (m³ s⁻¹)</td>
<td>Air outlet (m²)</td>
<td>Fan run time (s)</td>
<td>Fan disch. (m)</td>
<td>Release position</td>
<td>Release height (m)</td>
<td>Release time (s)</td>
<td>Mass (kg)</td>
<td>Calc $C_{f_{\text{max}}}$ (kg m⁻³)</td>
<td>Calc $C_{f_{\text{max}}}$ $t_{r,\text{calc}}$ (kg m⁻³)</td>
<td>Calc $C_{f_{\text{max}}}$ $t_r$ (kg m⁻³)</td>
</tr>
<tr>
<td>--------</td>
<td>---------</td>
<td>----------------</td>
<td>------------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>---------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>----------------</td>
<td>-----------</td>
<td>---------------------------------</td>
<td>---------------------------------</td>
<td>---------------------------------</td>
</tr>
<tr>
<td>37</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>244</td>
<td>1.77</td>
<td>wall</td>
<td>1.00</td>
<td>244</td>
<td>0.300</td>
<td>0.0488</td>
<td>0.0614</td>
<td>0.0556</td>
</tr>
<tr>
<td>38</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>469</td>
<td>1.77</td>
<td>wall</td>
<td>1.80</td>
<td>469</td>
<td>0.450</td>
<td>0.0343</td>
<td>0.0505</td>
<td>0.0326</td>
</tr>
<tr>
<td>24</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>220</td>
<td>1.77</td>
<td>mid</td>
<td>0.60</td>
<td>220</td>
<td>0.300</td>
<td>0.0596</td>
<td>0.0635</td>
<td>0.0615</td>
</tr>
<tr>
<td>25</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.06</td>
<td>0.031</td>
<td>440</td>
<td>1.77</td>
<td>mid</td>
<td>0.60</td>
<td>440</td>
<td>0.450</td>
<td>0.0686</td>
<td>0.0534</td>
<td>0.0353</td>
</tr>
<tr>
<td>26</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>222</td>
<td>1.77</td>
<td>mid</td>
<td>0.60</td>
<td>222</td>
<td>0.300</td>
<td>0.0668</td>
<td>0.0635</td>
<td>0.0611</td>
</tr>
<tr>
<td>27</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.06</td>
<td>0.031</td>
<td>535</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>535</td>
<td>0.450</td>
<td>0.0415</td>
<td>0.0534</td>
<td>0.0317</td>
</tr>
<tr>
<td>28</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.06</td>
<td>0.031</td>
<td>405</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>405</td>
<td>0.450</td>
<td>0.0506</td>
<td>0.0534</td>
<td>0.0370</td>
</tr>
<tr>
<td>29</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.06</td>
<td>0.031</td>
<td>258</td>
<td>1.77</td>
<td>wall</td>
<td>0.60</td>
<td>258</td>
<td>0.450</td>
<td>0.0668</td>
<td>0.0534</td>
<td>0.0476</td>
</tr>
<tr>
<td>31</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>318</td>
<td>1.77</td>
<td>wall</td>
<td>1.00</td>
<td>318</td>
<td>0.300</td>
<td>0.0415</td>
<td>0.0614</td>
<td>0.0466</td>
</tr>
<tr>
<td>30</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.05</td>
<td>0.031</td>
<td>442</td>
<td>1.77</td>
<td>wall</td>
<td>1.00</td>
<td>442</td>
<td>0.450</td>
<td>0.0415</td>
<td>0.0388</td>
<td>0.0271</td>
</tr>
<tr>
<td>32</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>394</td>
<td>1.77</td>
<td>wall</td>
<td>1.80</td>
<td>394</td>
<td>0.450</td>
<td>0.0343</td>
<td>0.0505</td>
<td>0.0359</td>
</tr>
<tr>
<td>33</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.05</td>
<td>0.031</td>
<td>425</td>
<td>1.77</td>
<td>wall</td>
<td>1.80</td>
<td>425</td>
<td>0.600</td>
<td>0.0451</td>
<td>0.0271</td>
<td>0.0222</td>
</tr>
<tr>
<td>17</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>233</td>
<td>2.43</td>
<td>mid</td>
<td>0.60</td>
<td>233</td>
<td>0.300</td>
<td>0.1264</td>
<td>0.1061</td>
<td>0.0982</td>
</tr>
<tr>
<td>18</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.06</td>
<td>0.031</td>
<td>281</td>
<td>2.43</td>
<td>mid</td>
<td>0.60</td>
<td>281</td>
<td>0.450</td>
<td>0.1047</td>
<td>0.0892</td>
<td>0.0737</td>
</tr>
<tr>
<td>19</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>193</td>
<td>2.43</td>
<td>wall</td>
<td>0.60</td>
<td>193</td>
<td>0.300</td>
<td>0.0596</td>
<td>0.1061</td>
<td>0.1130</td>
</tr>
<tr>
<td>20</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.06</td>
<td>0.031</td>
<td>295</td>
<td>2.43</td>
<td>wall</td>
<td>0.60</td>
<td>295</td>
<td>0.450</td>
<td>0.0668</td>
<td>0.0892</td>
<td>0.0714</td>
</tr>
<tr>
<td>21</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.01</td>
<td>0.031</td>
<td>258</td>
<td>2.43</td>
<td>wall</td>
<td>1.80</td>
<td>258</td>
<td>0.400</td>
<td>0.0325</td>
<td>0.0505</td>
<td>0.0449</td>
</tr>
<tr>
<td>22</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.02</td>
<td>0.031</td>
<td>296</td>
<td>2.43</td>
<td>wall</td>
<td>1.80</td>
<td>296</td>
<td>0.450</td>
<td>0.0361</td>
<td>0.0568</td>
<td>0.0467</td>
</tr>
<tr>
<td>23</td>
<td>R600a</td>
<td>mech vent</td>
<td>0.05</td>
<td>0.031</td>
<td>441</td>
<td>2.43</td>
<td>wall</td>
<td>1.80</td>
<td>441</td>
<td>0.600</td>
<td>0.0542</td>
<td>0.0452</td>
<td>0.0325</td>
</tr>
<tr>
<td>70</td>
<td>R290</td>
<td>nat vent</td>
<td>0.00</td>
<td>0.025</td>
<td>0.60</td>
<td>0.60</td>
<td>mid</td>
<td>0.60</td>
<td>140</td>
<td>0.115</td>
<td>0.0877</td>
<td>0.0436</td>
<td>0.0591</td>
</tr>
<tr>
<td>44</td>
<td>R290</td>
<td>nat vent</td>
<td>0.00</td>
<td>0.025</td>
<td>0.60</td>
<td>0.60</td>
<td>mid</td>
<td>0.60</td>
<td>215</td>
<td>0.150</td>
<td>0.0685</td>
<td>0.0568</td>
<td>0.0558</td>
</tr>
<tr>
<td>Test no</td>
<td>Gas type</td>
<td>Airflow source</td>
<td>Air flow (m³ s⁻¹)</td>
<td>Fan run time (s)</td>
<td>Fan disch. (m)</td>
<td>Release position</td>
<td>Release height (m)</td>
<td>Release time (s)</td>
<td>Mass (kg)</td>
<td>$\bar{C}_{f,max}$ (kg m⁻³)</td>
<td>$\text{Calc} \bar{C}<em>{f,max}$ $t</em>{r,cut}$ (kg m⁻³)</td>
<td>$\text{Calc} \bar{C}_{f,max}$ $t_r$ (kg m⁻³)</td>
<td></td>
</tr>
<tr>
<td>---------</td>
<td>----------</td>
<td>----------------</td>
<td>-------------------</td>
<td>------------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>------------------</td>
<td>-----------------</td>
<td>-----------</td>
<td>-----------------</td>
<td>-----------------------------</td>
<td>-----------------</td>
<td></td>
</tr>
<tr>
<td>71</td>
<td>R290</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>132</td>
<td>0.115</td>
<td>0.0493</td>
<td>0.0436</td>
<td>0.0618</td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>R290</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>232</td>
<td>0.150</td>
<td>0.0411</td>
<td>0.0568</td>
<td>0.0527</td>
<td></td>
</tr>
<tr>
<td>46</td>
<td>R290</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>204</td>
<td>0.150</td>
<td>0.0466</td>
<td>0.0568</td>
<td>0.0581</td>
<td></td>
</tr>
<tr>
<td>73</td>
<td>R290</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>1.80</td>
<td>wall</td>
<td>1.80</td>
<td>361</td>
<td>0.327</td>
<td>0.0260</td>
<td>0.0413</td>
<td>0.0303</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>R600a</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>0.60</td>
<td>mid</td>
<td>0.60</td>
<td>207</td>
<td>0.150</td>
<td>0.0921</td>
<td>0.0568</td>
<td>0.0574</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>R600a</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>0.60</td>
<td>mid</td>
<td>0.60</td>
<td>252</td>
<td>0.150</td>
<td>0.0759</td>
<td>0.0568</td>
<td>0.0495</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>R600a</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>208</td>
<td>0.150</td>
<td>0.0379</td>
<td>0.0568</td>
<td>0.0572</td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>R600a</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>1.00</td>
<td>wall</td>
<td>1.00</td>
<td>242</td>
<td>0.224</td>
<td>0.0433</td>
<td>0.0509</td>
<td>0.0462</td>
<td></td>
</tr>
<tr>
<td>72</td>
<td>R290</td>
<td>nat vent</td>
<td>0.00</td>
<td>0</td>
<td>0.60</td>
<td>wall</td>
<td>0.60</td>
<td>412</td>
<td>0.410</td>
<td>0.0548</td>
<td>0.1553</td>
<td>0.0935</td>
<td></td>
</tr>
<tr>
<td>79</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>583</td>
<td>mid</td>
<td>0.60</td>
<td>583</td>
<td>0.410</td>
<td>0.0699</td>
<td>0.0955</td>
<td>0.0475</td>
<td></td>
</tr>
<tr>
<td>80</td>
<td>R290</td>
<td>mech vent</td>
<td>0.05</td>
<td>0.048</td>
<td>580</td>
<td>mid</td>
<td>0.60</td>
<td>580</td>
<td>0.410</td>
<td>0.0137</td>
<td>0.0242</td>
<td>0.0166</td>
<td></td>
</tr>
<tr>
<td>85</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>611</td>
<td>wall</td>
<td>0.60</td>
<td>611</td>
<td>0.410</td>
<td>0.0617</td>
<td>0.0955</td>
<td>0.0460</td>
<td></td>
</tr>
<tr>
<td>84</td>
<td>R290</td>
<td>mech vent</td>
<td>0.05</td>
<td>0.048</td>
<td>592</td>
<td>wall</td>
<td>0.60</td>
<td>592</td>
<td>0.410</td>
<td>0.0137</td>
<td>0.0242</td>
<td>0.0165</td>
<td></td>
</tr>
<tr>
<td>89</td>
<td>R290</td>
<td>mech vent</td>
<td>0.04</td>
<td>0.031</td>
<td>682</td>
<td>2.43</td>
<td>2.43</td>
<td>682</td>
<td>0.490</td>
<td>0.0096</td>
<td>0.0311</td>
<td>0.0196</td>
<td></td>
</tr>
</tbody>
</table>

Table 62: Test variables and calculated results for Kataoka et al

[a - Kataoka, 1999, b - Kataoka et al, 2000b, c - Kataoka, 2001]
<table>
<thead>
<tr>
<th>Test no</th>
<th>Gas type</th>
<th>Air flow (m³ s⁻¹)</th>
<th>Fan run time (s)</th>
<th>Fan disch. (m)</th>
<th>Release position</th>
<th>Release height (m)</th>
<th>Release time (s)</th>
<th>Mass (kg)</th>
<th>Room volume (m³)</th>
<th>$\bar{C}_{f,\text{max}}$ (kg m⁻³)</th>
<th>$t_{r,\text{cat}}$ (kg m⁻³)</th>
<th>Calc $\bar{C}_{f,\text{max}}$ $t_r$ (kg m⁻³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3 (a)</td>
<td>R290</td>
<td>0.202</td>
<td>0</td>
<td>1.80</td>
<td>n/k</td>
<td>1.80</td>
<td>240</td>
<td>1.22</td>
<td>18</td>
<td>0.0589</td>
<td>0.0669</td>
<td>0.0669</td>
</tr>
<tr>
<td>4 (a)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>n/k</td>
<td>1.80</td>
<td>240</td>
<td>1.22</td>
<td>18</td>
<td>0.0781</td>
<td>0.2339</td>
<td>0.2198</td>
</tr>
<tr>
<td>5 (a)</td>
<td>R290</td>
<td>0.000</td>
<td>240</td>
<td>1.80</td>
<td>wall</td>
<td>1.80</td>
<td>240</td>
<td>0.22</td>
<td>18</td>
<td>0.0284</td>
<td>0.0419</td>
<td>0.0393</td>
</tr>
<tr>
<td>6 (a)</td>
<td>R290</td>
<td>0.017</td>
<td>240</td>
<td>1.80</td>
<td>wall</td>
<td>1.80</td>
<td>240</td>
<td>0.25</td>
<td>18</td>
<td>0.0288</td>
<td>0.0480</td>
<td>0.0451</td>
</tr>
<tr>
<td>1 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.10</td>
<td>mid</td>
<td>0.10</td>
<td>602</td>
<td>0.62</td>
<td>17</td>
<td>0.7509</td>
<td>2.1472</td>
<td>0.8600</td>
</tr>
<tr>
<td>2 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.10</td>
<td>mid</td>
<td>0.10</td>
<td>1560</td>
<td>1.30</td>
<td>17</td>
<td>1.0430</td>
<td>4.4877</td>
<td>0.7858</td>
</tr>
<tr>
<td>3 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>1560</td>
<td>1.30</td>
<td>17</td>
<td>0.3546</td>
<td>0.8975</td>
<td>0.2175</td>
</tr>
<tr>
<td>4 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>1560</td>
<td>1.30</td>
<td>17</td>
<td>0.4380</td>
<td>0.8975</td>
<td>0.2175</td>
</tr>
<tr>
<td>5 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>1560</td>
<td>1.30</td>
<td>17</td>
<td>0.4589</td>
<td>0.8975</td>
<td>0.2175</td>
</tr>
<tr>
<td>6 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.18</td>
<td>mid</td>
<td>0.18</td>
<td>1114</td>
<td>1.30</td>
<td>17</td>
<td>0.4380</td>
<td>0.8975</td>
<td>0.2175</td>
</tr>
<tr>
<td>7 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.90</td>
<td>mid</td>
<td>0.90</td>
<td>1114</td>
<td>1.30</td>
<td>17</td>
<td>0.2503</td>
<td>0.4986</td>
<td>0.1812</td>
</tr>
<tr>
<td>8 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>1114</td>
<td>1.30</td>
<td>17</td>
<td>0.4380</td>
<td>0.8975</td>
<td>0.2759</td>
</tr>
<tr>
<td>9 (b)</td>
<td>R134a</td>
<td>0.000</td>
<td>0</td>
<td>0.10</td>
<td>mid</td>
<td>0.10</td>
<td>1114</td>
<td>1.30</td>
<td>17</td>
<td>1.0847</td>
<td>4.4877</td>
<td>1.0526</td>
</tr>
<tr>
<td>10 (b)</td>
<td>CO2</td>
<td>0.000</td>
<td>180</td>
<td>0.10</td>
<td>mid</td>
<td>0.10</td>
<td>180</td>
<td>0.15</td>
<td>17</td>
<td>0.2886</td>
<td>0.5178</td>
<td>0.5920</td>
</tr>
<tr>
<td>11 (b)</td>
<td>CO2</td>
<td>0.017</td>
<td>0</td>
<td>0.10</td>
<td>mid</td>
<td>0.10</td>
<td>180</td>
<td>0.15</td>
<td>17</td>
<td>0.0144</td>
<td>0.0112</td>
<td>0.0114</td>
</tr>
<tr>
<td>1 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.07</td>
<td>15</td>
<td>0.0776</td>
<td>0.0505</td>
<td>0.0460</td>
</tr>
<tr>
<td>2 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.07</td>
<td>15</td>
<td>0.0794</td>
<td>0.0505</td>
<td>0.0460</td>
</tr>
<tr>
<td>3 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.13</td>
<td>15</td>
<td>0.1371</td>
<td>0.0864</td>
<td>0.1274</td>
</tr>
<tr>
<td>3a (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.15</td>
<td>15</td>
<td>0.0000</td>
<td>0.1037</td>
<td>0.0945</td>
</tr>
<tr>
<td>4 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.25</td>
<td>15</td>
<td>0.1714</td>
<td>0.1728</td>
<td>0.1576</td>
</tr>
<tr>
<td>5 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.23</td>
<td>15</td>
<td>0.0379</td>
<td>0.0498</td>
<td>0.0446</td>
</tr>
<tr>
<td>Test no</td>
<td>Gas type</td>
<td>Air flow (m³ s⁻¹)</td>
<td>Fan run time (s)</td>
<td>Fan disch. (m)</td>
<td>Release position</td>
<td>Release height (m)</td>
<td>Release time (s)</td>
<td>Mass (kg)</td>
<td>Room volume (m³)</td>
<td>Calc $\overline{C}_{f,\text{max}}$ (kg m⁻³)</td>
<td>Calc $\overline{C}_{f,\text{max}}$ $t_r$ (kg m⁻³)</td>
<td>Calc $\overline{C}<em>{f,\text{max}}$ $t</em>{r,\text{cal}}$ (kg m⁻³)</td>
</tr>
<tr>
<td>---------</td>
<td>----------</td>
<td>------------------</td>
<td>-----------------</td>
<td>---------------</td>
<td>-----------------</td>
<td>-----------------</td>
<td>-----------------</td>
<td>-----------</td>
<td>-----------------</td>
<td>--------------------------------</td>
<td>--------------------------------</td>
<td>--------------------------------</td>
</tr>
<tr>
<td>6 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>0.22</td>
<td>15</td>
<td>0.0379</td>
<td>0.0419</td>
<td>0.0396</td>
</tr>
<tr>
<td>7 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>0.68</td>
<td>154</td>
<td>0.0379</td>
<td>0.0415</td>
<td>0.0379</td>
</tr>
<tr>
<td>8 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>0.93</td>
<td>154</td>
<td>0.0451</td>
<td>0.0564</td>
<td>0.0515</td>
</tr>
<tr>
<td>9 (c)</td>
<td>R32</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>1.09</td>
<td>15</td>
<td>0.6083</td>
<td>0.7536</td>
<td>0.6870</td>
</tr>
<tr>
<td>10 (c)</td>
<td>R32</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>3.41</td>
<td>154</td>
<td>0.2680</td>
<td>0.7456</td>
<td>0.6676</td>
</tr>
<tr>
<td>11 (c)</td>
<td>R32</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>3.26</td>
<td>15</td>
<td>0.3871</td>
<td>0.6261</td>
<td>0.5921</td>
</tr>
<tr>
<td>12 (c)</td>
<td>R32</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>3.26</td>
<td>15</td>
<td>0.3956</td>
<td>0.6261</td>
<td>0.5921</td>
</tr>
<tr>
<td>13 (c)</td>
<td>R32</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>10.20</td>
<td>154</td>
<td>0.2829</td>
<td>0.6195</td>
<td>0.5662</td>
</tr>
<tr>
<td>14 (c)</td>
<td>R152a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.26</td>
<td>15</td>
<td>0.2052</td>
<td>0.1825</td>
<td>0.1664</td>
</tr>
<tr>
<td>15 (c)</td>
<td>R152a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.83</td>
<td>154</td>
<td>0.0945</td>
<td>0.1815</td>
<td>0.1625</td>
</tr>
<tr>
<td>16 (c)</td>
<td>R152a</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>0.88</td>
<td>15</td>
<td>0.1080</td>
<td>0.1686</td>
<td>0.1595</td>
</tr>
<tr>
<td>17 (c)</td>
<td>R152a</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>2.49</td>
<td>154</td>
<td>0.0918</td>
<td>0.1512</td>
<td>0.1382</td>
</tr>
<tr>
<td>18 (c)</td>
<td>R600a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.07</td>
<td>15</td>
<td>0.0737</td>
<td>0.0505</td>
<td>0.0460</td>
</tr>
<tr>
<td>19 (c)</td>
<td>R600a</td>
<td>0.000</td>
<td>0</td>
<td>0.50</td>
<td>mid</td>
<td>0.50</td>
<td>240</td>
<td>0.23</td>
<td>154</td>
<td>0.0356</td>
<td>0.0501</td>
<td>0.0448</td>
</tr>
<tr>
<td>20 (c)</td>
<td>R600a</td>
<td>0.000</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>0.22</td>
<td>15</td>
<td>0.0356</td>
<td>0.0421</td>
<td>0.0398</td>
</tr>
<tr>
<td>21 (c)</td>
<td>R600a</td>
<td>0.000</td>
<td>240</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>0.69</td>
<td>154</td>
<td>0.0333</td>
<td>0.0417</td>
<td>0.0381</td>
</tr>
<tr>
<td>22 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>240</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>1.22</td>
<td>15</td>
<td>0.0974</td>
<td>0.2339</td>
<td>0.2212</td>
</tr>
<tr>
<td>23 (c)</td>
<td>R290</td>
<td>0.017</td>
<td>240</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>1.22</td>
<td>15</td>
<td>0.0776</td>
<td>0.2339</td>
<td>0.2212</td>
</tr>
<tr>
<td>24 (c)</td>
<td>R290</td>
<td>0.017</td>
<td>240</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>0.35</td>
<td>15</td>
<td>0.0487</td>
<td>0.0663</td>
<td>0.0627</td>
</tr>
<tr>
<td>25 (c)</td>
<td>R290</td>
<td>0.000</td>
<td>240</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>1.18</td>
<td>154</td>
<td>0.0577</td>
<td>0.0718</td>
<td>0.0657</td>
</tr>
<tr>
<td>26 (c)</td>
<td>R290</td>
<td>0.171</td>
<td>240</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>1.18</td>
<td>154</td>
<td>0.0505</td>
<td>0.0133</td>
<td>0.0129</td>
</tr>
<tr>
<td>Test no</td>
<td>Gas type</td>
<td>Air flow (m³ s⁻¹)</td>
<td>Fan run time (s)</td>
<td>Fan disch. (m)</td>
<td>Release position</td>
<td>Release height (m)</td>
<td>Release time (s)</td>
<td>Mass (kg)</td>
<td>Room volume (m³)</td>
<td>Calc $\bar{C}_{f,max}$ (kg m⁻³)</td>
<td>Calc $C_{f,max}$ $t_{r,cat}$ (kg m⁻³)</td>
<td>Calc $C_{f,max}$ $t_r$ (kg m⁻³)</td>
</tr>
<tr>
<td>--------</td>
<td>----------</td>
<td>------------------</td>
<td>------------------</td>
<td>----------------</td>
<td>-----------------</td>
<td>-------------------</td>
<td>------------------</td>
<td>----------</td>
<td>-----------------</td>
<td>-------------------------------</td>
<td>---------------------------------</td>
<td>-------------------------------</td>
</tr>
<tr>
<td>27 (c)</td>
<td>R32</td>
<td>0.000</td>
<td>240</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>15.20</td>
<td>154</td>
<td>0.3722</td>
<td>0.9231</td>
<td>0.8438</td>
</tr>
<tr>
<td>28 (c)</td>
<td>R32</td>
<td>0.171</td>
<td>0</td>
<td>1.80</td>
<td>mid</td>
<td>1.80</td>
<td>240</td>
<td>15.20</td>
<td>154</td>
<td>0.3530</td>
<td>0.2079</td>
<td>0.1994</td>
</tr>
</tbody>
</table>
Appendix 4: Properties of flammable refrigerants

Table 63 provides flammable properties of various substances used within this Part 3. The data for HCs was from Woodward (1998) and from Richard (2003) for fluorocarbons. The density constant was obtained from density data generated using Refprop (McLinden et al, 1998).

Table 63: Flammable properties of hydrocarbons and selected refrigerants

<table>
<thead>
<tr>
<th>Chemical name</th>
<th>Iso-butane</th>
<th>Propane</th>
<th>Propene</th>
<th>Methane</th>
<th>Hexane</th>
<th>Difluoro-methane</th>
<th>Difluoro-ethane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant number</td>
<td>R600a</td>
<td>R290</td>
<td>R1270</td>
<td>R50</td>
<td>-</td>
<td>R32</td>
<td>R152a</td>
</tr>
<tr>
<td>Mol mas (kg mol(^{-1}))</td>
<td>58.1</td>
<td>44.1</td>
<td>42.1</td>
<td>16.0</td>
<td>86.2</td>
<td>52.0</td>
<td>66.1</td>
</tr>
<tr>
<td>LFL (kg m(^{-3}))</td>
<td>0.045</td>
<td>0.038</td>
<td>0.043</td>
<td>0.032</td>
<td>0.039</td>
<td>0.27</td>
<td>0.121</td>
</tr>
<tr>
<td>UFL (kg m(^{-3}))</td>
<td>0.203</td>
<td>0.172</td>
<td>0.173</td>
<td>0.097</td>
<td>0.277</td>
<td>0.52</td>
<td>0.436</td>
</tr>
<tr>
<td>(C_{\text{stoic}}) (kg m(^{-3}))</td>
<td>0.074</td>
<td>0.072</td>
<td>0.077</td>
<td>0.062</td>
<td>0.076</td>
<td>0.37</td>
<td>0.208</td>
</tr>
<tr>
<td>(H) (kJ kg(^{-1}))</td>
<td>45580</td>
<td>46320</td>
<td>45805</td>
<td>50020</td>
<td>44700</td>
<td>44700</td>
<td>9400</td>
</tr>
<tr>
<td>(H_r^{\text{mol}}) (kJ mol(^{-1}))</td>
<td>134600</td>
<td>103900</td>
<td>20400</td>
<td>74900</td>
<td>167300</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>(T_{\text{flam}}) (K)</td>
<td>2400</td>
<td>2394</td>
<td>2506</td>
<td>2225</td>
<td>2400</td>
<td>2300</td>
<td>2325</td>
</tr>
<tr>
<td>(u_{\text{flam,law}}) (m s(^{-1}))</td>
<td>0.37</td>
<td>0.43</td>
<td>0.48</td>
<td>0.35</td>
<td>0.44</td>
<td>0.06</td>
<td>0.23</td>
</tr>
<tr>
<td>Density constant (a)</td>
<td>0.515</td>
<td>0.354</td>
<td>0.322</td>
<td>-0.802</td>
<td>0.629</td>
<td>0.450</td>
<td>0.571</td>
</tr>
</tbody>
</table>

The density of a refrigerant and (dry) air mixture at room temperature and standard atmospheric pressure is (equation 408)

\[ \rho = 1.2 + a \cdot C \]  \hspace{1cm} (408)

where \(C\) is the refrigerant concentration (kg m\(^{-3}\))

For flammable mixtures at concentrations other than stoichiometric, properties were estimated using the methods below.

Heat of combustion

The heat of combustion was calculated with equation (409), where only the quantity of refrigerant or oxygen available for stoichiometric combustion (depending upon which was in excess) was included, i.e., the energy from excess fuel or oxygen was discounted.

\[ H = \frac{1}{M_r \cdot N_r^{\text{mol}}} \left( \sum_i N_{1,i}^{\text{mol}} \cdot H_{1,i}^{\text{mol}} - \sum_i N_{2,i}^{\text{mol}} \cdot H_{2,i}^{\text{mol}} \right) \]  \hspace{1cm} (409)
where \( M \) is the molar mass, \( N^{\text{mol}} \) is the number of moles, \( H^{\text{mol}} \) is the molar enthalpy of formation. Subscript 1 represents unburned species and subscript 2 the combustion products.

(For the combustion products, carbon dioxide \( H^{\text{mol}} = 393800 \, \text{kJ kmol}^{-1} \) and water vapour \( H^{\text{mol}} = 242000 \, \text{kJ kmol}^{-1} \).)

**Adiabatic flame temperature**

The adiabatic flame temperature was found from the following energy balance (equation 410).

\[
\sum_i X_{1,i} \cdot \Delta H_{1,i} (T_a) + H_{1,i} = \sum_i X_{2,i} \cdot C_{p2,j} \cdot (T_{\text{flam}} - T_a) \tag{410}
\]

where \( X_1 \) is the mass fraction of the reactants, and \( X_2 \) the mass fraction of the combustion products, \( \Delta H_1 \) is the enthalpy of reactants (at initial air temperature), \( H_1 \) is the heat of combustion, and \( C_{p2} \) is the specific heat of combustion products.

**Laminar flame velocity**

Flame velocity is known to be a function of the density, thermal conductivity and specific heat of the initial mixture, as well as the combustion rate. Across the flammable range, these properties vary little, so \( u_{\text{flam, lam}} \) is adjusted according to the combustion rate only. Since the combustion rate is related to expansion ratio, and therefore \( T_{\text{flam}} \), \( u_{\text{flam, lam}} \) is approximated from the variation in \( T_{\text{flam}} \) (equation 411).

\[
u_{\text{flam, lam}} (C) = \frac{\Delta T_{\text{flam}} (C_{st} \cdot C)}{\Delta T_{\text{flam}} (C_{st} \cdot C_{FL})} \tag{411}
\]

where \( \Delta T_{\text{flam}} \) refers to the difference between the stoichiometric \( T_{\text{flam}} \) and the value at the reference concentration, or the concentration of the flammable limit \( C_{FL} \) adjacent to the reference concentration.
## Appendix 5: Data for overpressure calculations

### Table 64: Data from overpressure measurements and model results

<table>
<thead>
<tr>
<th>Ref No</th>
<th>Gas</th>
<th>$\bar{m}$ (kg)</th>
<th>$\bar{V}$ (m$^3$)</th>
<th>$V_{fix}$ (m$^3$)</th>
<th>$\bar{A}$ (m$^2$)</th>
<th>$W_{vp}$ (kgm$^2$)</th>
<th>$P_{vp, fix}$ (kPa)</th>
<th>$\Delta p^o_{meas}$ (kPa)</th>
<th>$\Delta p^o_{calc}$ (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>hex</td>
<td>0.136</td>
<td>0.2</td>
<td>20.7</td>
<td>1.0</td>
<td>1.25</td>
<td>0</td>
<td>5.2</td>
<td>5.5</td>
</tr>
<tr>
<td>2</td>
<td>hex</td>
<td>0.139</td>
<td>0.2</td>
<td>20.7</td>
<td>1.0</td>
<td>1.25</td>
<td>0</td>
<td>5.9</td>
<td>5.7</td>
</tr>
<tr>
<td>3</td>
<td>hex</td>
<td>0.138</td>
<td>0.2</td>
<td>20.7</td>
<td>1.0</td>
<td>1.25</td>
<td>0</td>
<td>4.7</td>
<td>4.7</td>
</tr>
<tr>
<td>4</td>
<td>hex</td>
<td>0.158</td>
<td>0.3</td>
<td>20.7</td>
<td>1.0</td>
<td>1.25</td>
<td>0</td>
<td>4.7</td>
<td>5.0</td>
</tr>
<tr>
<td>5</td>
<td>hex</td>
<td>0.153</td>
<td>0.3</td>
<td>20.7</td>
<td>1.0</td>
<td>1.25</td>
<td>0</td>
<td>5.8</td>
<td>4.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.16</td>
<td>0</td>
<td>50</td>
<td>140</td>
</tr>
<tr>
<td>2</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.36</td>
<td>0</td>
<td>50</td>
<td>64</td>
</tr>
<tr>
<td>3</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.10</td>
<td>0</td>
<td>10</td>
<td>140</td>
</tr>
<tr>
<td>4</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.10</td>
<td>0</td>
<td>20</td>
<td>219</td>
</tr>
<tr>
<td>5</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.20</td>
<td>0</td>
<td>50</td>
<td>195</td>
</tr>
<tr>
<td>6</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.20</td>
<td>0</td>
<td>10</td>
<td>84</td>
</tr>
<tr>
<td>7</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.20</td>
<td>0</td>
<td>20</td>
<td>148</td>
</tr>
<tr>
<td>8</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.20</td>
<td>0</td>
<td>50</td>
<td>111</td>
</tr>
<tr>
<td>9</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.30</td>
<td>0</td>
<td>10</td>
<td>43</td>
</tr>
<tr>
<td>10</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.30</td>
<td>0</td>
<td>20</td>
<td>77</td>
</tr>
<tr>
<td>11</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.30</td>
<td>0</td>
<td>50</td>
<td>59</td>
</tr>
<tr>
<td>12</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.05</td>
<td>0</td>
<td>32</td>
<td>200</td>
</tr>
<tr>
<td>13</td>
<td>R50</td>
<td>0.06</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.10</td>
<td>0</td>
<td>16</td>
<td>100</td>
</tr>
<tr>
<td>14</td>
<td>R50</td>
<td>3.05</td>
<td>49.1</td>
<td>49.1</td>
<td>0.8</td>
<td>3.46</td>
<td>0</td>
<td>0</td>
<td>12</td>
</tr>
<tr>
<td>15</td>
<td>R50</td>
<td>2.08</td>
<td>33.5</td>
<td>33.5</td>
<td>0.5</td>
<td>2.57</td>
<td>0</td>
<td>0</td>
<td>15</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>R50</td>
<td>0.16</td>
<td>2.4</td>
<td>2.4</td>
<td>0.5</td>
<td>0.26</td>
<td>0</td>
<td>14</td>
<td>6</td>
</tr>
<tr>
<td>2</td>
<td>R50</td>
<td>0.15</td>
<td>2.3</td>
<td>2.3</td>
<td>1.0</td>
<td>0.37</td>
<td>0</td>
<td>0</td>
<td>6</td>
</tr>
<tr>
<td>3</td>
<td>R50</td>
<td>0.15</td>
<td>2.3</td>
<td>2.3</td>
<td>1.0</td>
<td>0.16</td>
<td>0</td>
<td>0</td>
<td>21</td>
</tr>
<tr>
<td>4</td>
<td>R50</td>
<td>0.15</td>
<td>2.3</td>
<td>2.3</td>
<td>1.0</td>
<td>0.09</td>
<td>0</td>
<td>0</td>
<td>44</td>
</tr>
<tr>
<td>5</td>
<td>R50</td>
<td>0.04</td>
<td>0.7</td>
<td>0.7</td>
<td>1.0</td>
<td>0.19</td>
<td>0</td>
<td>0</td>
<td>3</td>
</tr>
<tr>
<td>6</td>
<td>R50</td>
<td>0.04</td>
<td>0.7</td>
<td>0.7</td>
<td>1.0</td>
<td>0.08</td>
<td>0</td>
<td>0</td>
<td>15</td>
</tr>
<tr>
<td>7</td>
<td>R50</td>
<td>0.04</td>
<td>0.7</td>
<td>0.7</td>
<td>1.0</td>
<td>0.04</td>
<td>0</td>
<td>0</td>
<td>35</td>
</tr>
<tr>
<td>8</td>
<td>R50</td>
<td>2.01</td>
<td>30.4</td>
<td>30.4</td>
<td>0.5</td>
<td>1.33</td>
<td>0</td>
<td>1</td>
<td>21</td>
</tr>
<tr>
<td>9</td>
<td>R50</td>
<td>2.03</td>
<td>30.4</td>
<td>30.4</td>
<td>1.0</td>
<td>2.74</td>
<td>0</td>
<td>2</td>
<td>22</td>
</tr>
<tr>
<td>10</td>
<td>R50</td>
<td>2.05</td>
<td>30.4</td>
<td>30.4</td>
<td>1.0</td>
<td>1.33</td>
<td>0</td>
<td>4</td>
<td>54</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>R290</td>
<td>0.07</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.20</td>
<td>0</td>
<td>10</td>
<td>100</td>
</tr>
<tr>
<td>2</td>
<td>R290</td>
<td>0.07</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.40</td>
<td>0</td>
<td>10</td>
<td>32</td>
</tr>
<tr>
<td>3</td>
<td>R290</td>
<td>0.07</td>
<td>1.0</td>
<td>1.0</td>
<td>0.5</td>
<td>0.60</td>
<td>0</td>
<td>10</td>
<td>14</td>
</tr>
<tr>
<td>4</td>
<td>R290</td>
<td>14.70</td>
<td>203.8</td>
<td>203.8</td>
<td>1.0</td>
<td>21.58</td>
<td>0</td>
<td>0</td>
<td>3</td>
</tr>
<tr>
<td>Ref</td>
<td>No</td>
<td>Gas</td>
<td>$\bar{m}_F$ (kg)</td>
<td>$\bar{V}_F$ (m$^3$)</td>
<td>$V_{R_m}$ (m$^3$)</td>
<td>$\frac{V_{R_m}}{0.5^{\frac{1}{3}}}$ (m)</td>
<td>$A_v$ (m$^2$)</td>
<td>$W_{vp}$ (kgm$^{-2}$)</td>
<td>$P_{vp,fat}$ (kPa)</td>
</tr>
<tr>
<td>-----</td>
<td>----</td>
<td>-----</td>
<td>-----------------</td>
<td>-----------------</td>
<td>-----------------</td>
<td>----------------------</td>
<td>-------------</td>
<td>-------------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>5</td>
<td>6</td>
<td>R290</td>
<td>14.70</td>
<td>203.8</td>
<td>203.8</td>
<td>1.0</td>
<td>17.30</td>
<td>0.00</td>
<td>0</td>
</tr>
<tr>
<td>6</td>
<td>7</td>
<td>R290</td>
<td>0.80</td>
<td>11.0</td>
<td>11.0</td>
<td>1.0</td>
<td>1.36</td>
<td>0.00</td>
<td>5</td>
</tr>
<tr>
<td>7</td>
<td>8</td>
<td>R290</td>
<td>2.44</td>
<td>30.4</td>
<td>30.4</td>
<td>0.5</td>
<td>0.58</td>
<td>0.00</td>
<td>40</td>
</tr>
<tr>
<td>8</td>
<td>9</td>
<td>R290</td>
<td>0.06</td>
<td>0.7</td>
<td>0.7</td>
<td>0.5</td>
<td>0.02</td>
<td>0.00</td>
<td>65</td>
</tr>
<tr>
<td>9</td>
<td>10</td>
<td>R290</td>
<td>0.06</td>
<td>0.7</td>
<td>0.7</td>
<td>0.5</td>
<td>0.03</td>
<td>0.00</td>
<td>65</td>
</tr>
<tr>
<td>10</td>
<td>11</td>
<td>R290</td>
<td>0.06</td>
<td>0.7</td>
<td>0.7</td>
<td>0.5</td>
<td>0.07</td>
<td>0.00</td>
<td>86</td>
</tr>
<tr>
<td>11</td>
<td>12</td>
<td>R290</td>
<td>0.06</td>
<td>0.7</td>
<td>0.7</td>
<td>0.5</td>
<td>0.10</td>
<td>0.00</td>
<td>86</td>
</tr>
<tr>
<td>12</td>
<td>13</td>
<td>R290</td>
<td>0.00</td>
<td>0.0</td>
<td>0.0</td>
<td>0.5</td>
<td>0.03</td>
<td>0.00</td>
<td>10</td>
</tr>
<tr>
<td>13</td>
<td>14</td>
<td>R290</td>
<td>0.07</td>
<td>0.8</td>
<td>0.8</td>
<td>0.5</td>
<td>0.29</td>
<td>0.00</td>
<td>0</td>
</tr>
<tr>
<td>14</td>
<td>15</td>
<td>R290</td>
<td>3.31</td>
<td>35.0</td>
<td>35.0</td>
<td>1.0</td>
<td>1.00</td>
<td>0.00</td>
<td>0</td>
</tr>
<tr>
<td>15</td>
<td>16</td>
<td>R290</td>
<td>3.31</td>
<td>35.0</td>
<td>35.0</td>
<td>0.5</td>
<td>1.00</td>
<td>0.00</td>
<td>0</td>
</tr>
</tbody>
</table>

441