Making shipping greener: comparative study between organic fluids and water for Rankine cycle waste heat recovery

Santiago Suárez de la Fuente ª\*, Alistair R. Greig a

Department of Mechanical Engineering, University College London, London, United Kingdom

*\** Roberts Building, 5th floor, Torrington Place, London, WC1E 7JE, United Kingdom

*\* Email address:* santiago.fuente.2011@ucl.ac.uk

# Author biographies

Santiago Suárez de la Fuente is currently a PhD student at University College London (UCL) undertaking research in the development of waste heat recovery systems for the shipping industry. Santiago’s main research interests are marine waste heat recovery systems based on organic Rankine cycles, system optimisation and heat transfer.

Alistair Greig is a senior lecturer at UCL and a professional marine engineer. His research interests cover a broad range of topics including warship design, ship efficiency and marine machinery integration with particular emphasis on advanced craft and the impact of new technology and legislation.

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The largest source of energy loss in ships is found in the propulsion system. This study focuses on the concept of managing waste heat energy from the exhaust gases of the main engine. Using waste heat recovery systems to make shipping more efficient represents a good area of opportunity for achieving the shipping industry’s green objectives. Organic Rankine cycles have been applied in land-based systems before, showing improvements in performance when compared with the traditional Rankine cycle. As marine environmental rules requiring greener vessels and engine thermal efficiency continue to increase, thus reducing the available energy in the exhaust, organic Rankine cycle waste heat recovery systems become a more attractive option.

The proposed waste heat recovery system was modelled using Matlab® with a typical ship installation with a slow speed diesel engine and the waste heat recovery system having been installed after the steam boiler in the exhaust gas system. The energy recovered from the exhaust gas flow is transformed via the thermodynamic cycle - coupled with a generator - into electricity which helps to cover the ship’s demand. The Matlab® code found the highest electric power output varying the waste heat recovery system high pressure, maximising the fuel and CO2 emission savings. Water and four organic fluids were considered and their performance was compared over a range of different engine operating conditions. A representative ship operating profile and a typical marine generator were used to measure CO2 emission reductions. The implications of having flammable organic fluids on-board are also briefly discussed. This work demonstrates that a simple organic Rankine cycle can be more effective than a steam cycle for the same engine operating conditions.

Keywords: waste heat recovery systems; organic fluids, CO2 emissions, optimisation.

# Introduction

Shipping CO2 emissions in 2007 represented around 3.3% of global carbon emissions, and is one of the transportation modes with the greatest growth in energy demand between the years 1990 and 2006, 2.0% for Organisation for Economic Co-operation and Development (OECD) members and 7.3% for non-members (International Energy Agency 2009). If no other policies are implemented and the patterns of social and economic behaviour do not change, future CO2 emissions in shipping could grow 400% by 2050 from the 2007 level (International Maritime Organization 2009a). Shipping therefore represents an opportune, and necessary, sector for CO2 emissions reductions.

Within ships, the majority of energy losses are located in the prime mover. The most frequent option to power a commercial vessel is a slow speed, two-stroke marine diesel (Woodyard 2009; Woud & Stapersma 2012), this is due to its high thermal efficiency which can reach up to 53% (International Maritime Organization 2009a). Waste energy from the engine can be found as heat removed by the cooling system (20-30%), heat escaping from the engine surface due to convection or radiation (1-5%), and finally, a significant proportion (25-45%) through exhaust gases (Woodyard 2009; Balaji & Yaakob 2012; MAN Diesel & Turbo 2012). Normally, waste heat coming from the diesel exhaust gases is found as medium quality heat (i.e. between 230˚C and 650˚C), while heat escaping from the water and air cooling system, and from radiative and convective phenomena are generally low grade quality, i.e. between 30˚C and 230˚C (Dharmalingam *et al*. 2004).

Thermodynamic Waste Heat Recovery Systems (WHRS) absorb part of the waste heat which evaporates a working fluid. The fluid then expands (e.g. inside a turbine) and converts part of the heat energy into mechanical, electrical or cooling work. The use of WHRS in land-based systems has been well covered using low/medium waste heat: in cement processes (Bronicki 2000; Cunningham 2002), solar (Tchanche *et al*. 2011; Vélez *et al*. 2012), geothermal (Mlcak 2004; DiPippo 2004; Bronicki 2010; Öhman & Lundqvist 2013) and, land-based vehicles using internal combustion engines (Dinanno *et al*. 1983; Feng *et al*. 2010; Weerasinghe *et al*. 2010; Dolz *et al*. 2012). The ship WHRS case is a peculiar one when compared to land-based systems: a) The fuels used for vessel propulsion have a higher sulphur content - even when including marine Emission Control Areas – this limits the amount of heat that the WHRS can absorb from the exhaust gas due to sulphuric acid corrosion (Schmid 2004; European Committee for Standardization 2013; International Maritime Organization 2013; Nielsen *et al*. 2014); b) Installing a WHRS on-board a ship will occupy vital space, resulting in compact designs and will also impose additional constraints on the size and mass of components; c) The mass of the WHRS will increase the displacement of the ship and, hence, the power required to move it; d) Due to optimised weather routing, cargo level and scheduling, there is a more varied operating profile and wider range of power demands, affecting the waste heat availability from the ship’s different heat sources; e) When using seawater as a cooling medium, there will be temperature changes as the ship travels around the globe, with cold sink temperatures which can range from -2˚C to 30˚C (Rayner *et al*. 2006).

The work of Alvik *et al*. (2010) showed that WHRS is one of the options which reduces the largest amount of shipping CO2 emissions, just behind slow-steaming and the use of LNG as fuel. The downside is the initial cost of WHRS but, since its CO2 reductions occur from a reduction in fuel use, the impact of the WHRS investment could be reduced in the short to medium term (Theotokatos & Livanos 2012). The preferred thermodynamic WHRS used on-board ships is the water based Rankine Cycle (RC). Some of the reasons (Ringler *et al.* 2009; Chen et al. 2010; Boretti 2012) include: high critical temperature (i.e. 374˚C); the highest latent heat of vaporisation of any substance helps to deliver large power outputs at high efficiencies; there is no chemical decomposition; and at higher temperatures it can be assured that the vapour will not condense inside the expander, eliminating the problem of erosion. Water is a safe working fluid although there are always dangers associated with high pressure steam. Marine RC WHRS have been researched by Hatchman (1991), Tien *et al*. (2007), Theotokatos and Livanos (2012), Benvenuto *et al*. (2014), Dimopoulos *et al*. (2014), Livanos *et al*. (2014) and Sakalis and Frangopoulos (2014). WHRS for shipping based on RC have been manufactured and applied with success (Schmid 2004; MAN Diesel & Turbo 2012). Using the BIMCO EEDI calculator, MAN predicted that the use of WHRS would result in a drop in the Energy Efficiency Design Index (EEDI) by around 9.2% (MAN Diesel & Turbo 2012).

However, as the EEDI pushes for low CO2 emission vessels, the increasing preference of the shipping industry for slow speed steaming, the rise of diesel engine thermal efficiency and other GHG reduction technologies using part of the available waste heat (e.g. exhaust gas scrubbers), it is expected that waste heat quality will be reduced. This makes it harder for the RC to deliver high levels of electric power at a reasonable thermal efficiency. Organic fluids (i.e. those that contain carbon in their molecular structure) using the same cycle proposed by Rankine are good candidates to overcome some of the difficulties found with lower grade waste heat sources (Saavedra *et al*. 2010). The large catalogue of organic fluids permits the creation of tailored designs for a heat source. The use of isentropic or dry organic fluids eliminates the problem of having droplets in the expansion process (Liu *et al*. 2004; Chen *et al*. 2010), see Figure 1. However, organic fluids have lower latent heat than water, which means larger mass flow rates, pump losses and equipment.



Figure 1: Representation of three different types of fluids. Depending on the working fluid's slope in a temperature-entropy diagram, it could be classified as wet fluid when it has a positive slope (e.g. water); Isentropic when its slope is zero (e.g. R11); and Dry when it is negative (e.g. heptane).

ORC systems on-board ships are commonly studied as the bottoming cycle of other WHRS (Fischer 2011; Nielsen *et al*. 2014; Marine Engineers Review 2014); this is due to the strong ORC performance at low temperatures and its simple layout (Liu *et al*. 2004; Chen *et al*. 2010). Opcon Marine manufactures ORC waste heat recovery plants that work for the vessel’s low grade heat sources (i.e. 90-160˚C), such as the engine’s cooling system and steam not used by the vessel (Opcon Energy Systems AB 2012a). These WHRS are installed in *MV Figaro* with a reported saving in fuel consumption of around 5% (Opcon Energy Systems AB 2012b).

Yue *et al*. (2012) studied the possibility of using an isopentane WHRS using the jacket water of a 6,750 kW marine diesel engine as the heat source. The ORC was capable of producing 300 kW at a thermal efficiency of 13.6%. This represented a reduction of around 277 tonnes of heavy fuel oil per year. Using more integral heat management (i.e. using the waste heat available in the charge air cooler, jacket water and exhaust gas system), Larsen *et al*. (2013) performed a comparison between a marine ORC WHRS using the organic fluid R245ca, a typical water-based RC WHRS and a Kalina cycle under the same design point and conditions. The Kalina cycle is a thermodynamic cycle that uses the concepts of the RC and absorption cycle, using a binary working fluid - water and ammonia in different concentrations - in order to attain a better match for the heat source and sink by having a non-isothermal evaporation/condensation, for more details see Zhang *et al*. (2012). It was found that the marine ORC outperformed, in power output, the RC and KC by around 34.4% and 40.6% respectively. In another study, a large catalogue of organic fluids was used by Larsen *et al*. (2013) to recover available heat from the exhaust gas of a low speed two-stroke marine diesel engine. Performance and characteristics of the different WHRS were compared using a multi-objective algorithm with the principle of natural selection. The result showed that if safety is the most important factor, the highest performance was given by R245fa and R236ea. This selection of fluids has its drawbacks since its global warming potential (GWP) is high; meaning that the gas, if it escapes into the atmosphere, will store more energy than a similar mass of CO2 (Myhre *et al*. 2013). When reducing the importance of safety levels, toluene and pentene deliver the best results. The work of Larsen *et al*. focused only on the thermal efficiency of the system and not the power output which dictates the WHRS benefit on CO2 reduction and fuel consumption. Samsung Heavy Industries (Lee & Lee 2013) worked with an ORC WHRS concept using a double loop system which absorbs waste heat from the ship’s exhaust gas, cooling water and scavenge air. The results show that the WHRS could cover up to 73% of the electric demand of a Suezmax tanker while in transit.

The work presented here will focus on the usage of low/medium grade waste heat energy from the exhaust gas to produce electric power. The authors studied how the diesel’s engine load affects the performance and design of the WHRS, and the impact in CO2 reduction when compared to a generator using marine diesel oil, helping to understand the performance differences between marine RC and ORC WHRS.

# Methodology

## Diesel engine and exhaust gas

The engine used to run the simulations was an 87 MW 12 cylinder slow-speed diesel manufactured by MAN Diesel & Turbo. The model was G95ME-C9.5-TII with four turbo-compressors, model MAN TCA88-24. The engine works with marine diesel oil (MDO), with a price of €733[[1]](#footnote-1) per tonne (Bunker Index 2015). The vessel has a steam demand of around 3,800 kg h-1 at any time which will be covered by a waste heat boiler.

The WHRS used the waste heat available from the exhaust gas after it had passed through the turbo-compressors and the waste heat boiler (see Figure 2). Since the fuel contains sulphur in its composition, it was expected that the exhaust gas would contain SOx. Engine manufacturers recommend having a minimum temperature at the exit of the exhaust gas between 160 and 165°C to avoid soot and acid corrosion (Schmid 2004; Huijbregts & Leferink 2004; MAN Diesel & Turbo 2009). This will mean that the WHRS economiser, in this study, could extract energy until the exhaust gas reached a temperature of 165°C.

The exhaust gas was assumed to have similar properties and behaviour to air. Its specific heat at constant pressure was calculated as the average temperature between the temperatures at the turbo compressor’s exit and the minimum possible for the exhaust gas.



Figure 2: Overview of the WHRS inside the ship.

The engine’s loading conditions considered went from 15% to 100% of the engine’s Maximum Continuous Rating (MCR) in steps of 5%.



Figure 3: Engine performance and exhaust gas behaviour. Taken at ISO ambient reference conditions (MAN Diesel & Turbo 2014).

## Working Fluids

The working fluids (Table 1) were selected principally because of their positive performance as identified by Saavedra *et al*. (2010) and their low global warming potential (GWP). A value of 1 is assigned to CO2; larger values than one mean that the gas would store more energy, thus, warming the climate system more than a similar mass of CO2. For example, methane has a GWP100 of 28 which means that in a period of 100 years, it will store 28 times more energy than CO2.

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| *Working Fluids* | *GWP100* | *Latent Heat of Vaporization* | *Auto-ignition Temperature* | *Decomposition Temperature* | *Flash Point* | *Tc* | *Pc* |
|  | (-)***[[2]](#footnote-2)*** | (kJ kg-1)**[[3]](#footnote-3)** | (˚C)**[[4]](#footnote-4)** | (˚C)**[[5]](#footnote-5)** | (˚C) **[[6]](#footnote-6)** | (˚C)**[[7]](#footnote-7)** | (MPa)\*\*\* |
| Water | N/A | 2,257 | - | 2,000 | - | 374 | 22.06 |
| Benzene | N/A | 394 | 562 | 760 | -11 | 289 | 4.91 |
| Toluene | 2.7 | 361 | 536 | 399 | 4 | 318 | 4.13 |
| Heptane | 3.0 | 317 | 223 | 550 | -4 | 267 | 2.74 |
| Hexamethyldisiloxane  (MM) | <10 | 193 | 341 | 300 | -2 | 246 | 1.94 |

Table 1: Working fluids selected with their GWP value in a time interval of 100 years, latent heat, auto-ignition and decomposition temperatures, flash points and, critical temperature and pressure.

The ORC studied here were analysed in their subcritical region. Because of this, the pressure and temperature limits were given by the critical points (shown in Table 1).

Decomposition temperature describes when the fluid beings to decompose chemically. If the conditions in the system exceed this temperature, the useful life of the fluid will be reduced (Saavedra et al. 2010). An important characteristic of the organic fluid with regard to marine safety is the auto-ignition temperature.

## Thermal oil

In order to transfer the waste heat to the organic fluids, a thermal oil circuit was modelled into the code. For some organic fluids this is essential to protect them from thermal decomposition. Aboard ship it is required, due to the low flash points of the organic fluids selected and regulations given by the International Convention for the Safety of Life at Sea (SOLAS) in Chapter II-2, that organic fluids are not permitted in the engine room (International Maritime Organization 2009b). The thermal oil circuit provides flexibility in layout which is particularly useful for retrofits. The thermal oil used was Therminol VP-1 due to its heat stability and low viscosity at the WHRS operational conditions (Solutia Inc. 1999). A detailed study of the thermal oil circuit was out of the scope for this paper.

## Waste heat recovery system

The model iterated the high pressure level in steps of 10 kPa up to the critical pressure (*Pc*) of each working fluid at the various engine loads. The iterative process stopped when critical pressure was reached or when the working fluid saturation temperature was higher than the value set for the expander’s inlet temperature(*T1*) in Table 2. The purpose of the iterative process was to find the WHRS maximum electrical power output for a given engine load.

For this study, the thermodynamic cycle used an economiser, an expander connected to an electrical generator, a condenser with a sea pump, and a second pump for the working fluid. A recuperator is installed after the pump and expander if the expander outlet temperature (*T2*) is 10% greater than that of the pump (*T4*).



Figure 4: WHRS layout including recuperator and thermal oil circuit.

### Heat Exchangers

The heat exchangers were counter-flow with a constant pinch point temperature difference of 5 K. These were defined as the difference between the saturation temperature and the heat source or sink temperature.

|  |  |  |
| --- | --- | --- |
|  |  | (1) |

A minimum of 5 K was chosen for the approach temperature (*Tap*), defined in a counter-flow heat exchanger as the temperature difference between the cold fluid inlet and hot fluid outlet, and cold fluid outlet and hot fluid inlet (Shah & Sekulic 2003). The *Tap* constraint was inside the temperature range of a plate or spiral tube heat exchangers (Tranter 2011; Sun *et al*. 2013).

The economiser and condenser were subdivided into three regions (Figure 5) in order to accurately measure the energy balance in processes where there were two states of the fluid present (i.e. liquid and vapour).

* Subcooler: The region between saturated liquid and the working fluid subcooled region. In the case of the condenser, this region did not exist due to the restrictions imposed.
* Evaporator: Region that takes the working fluid from the saturated liquid state to the saturated vapour.
* Superheater: Located between the superheater state of the working fluid and the saturated vapour state.

Figure 5: Representation of the three regions presented in the water’s economiser.

Starting with the economiser, the heat flow available from the exhaust gas is defined as follows:

|  |  |  |
| --- | --- | --- |
|  |  | (2) |

With the pinch point temperature difference and the saturation temperature of the working fluid (equation ( 1 )), it is possible to ascertain the heat coming into the subcooler:

|  |  |  |
| --- | --- | --- |
|  |  | (3) |

Equating the heat leaving the exhaust gas to that absorbed by the working fluid, it is possible to determine the mass flow rate required by the subcool region, assuming that there is no recuperator:

|  |  |  |
| --- | --- | --- |
|  |  | (4) |

Where *h4a* is the high pressure enthalpy at the saturated liquid point of the working fluid, and *h4* is the working fluid at the entrance of the heat exchanger. is used here as the maximum mass flow rate possible which falls inside the restrictions set by the pinch point temperature and the exhaust gas exit temperature. A lower mass flow rate than that set by will not be able to absorb all the energy available at the subcooler region, but will still be inside the restriction limits. With equations (2) and (3) is possible to determine the mass flow rate required to take the working fluid from its liquid saturated point to the expander inlet enthalpy:

|  |  |  |
| --- | --- | --- |
|  |  | (5) |
|  |  | (6) |

The logic of the algorithm will not take into consideration mass flow rates that do not comply with the restriction shown in equation (6) which guarantees that there is no temperature crossover inside the heat exchanger. is used as the working fluid mass flow rate. From this point onwards the will be written as .

The total heat absorbed by the working fluid can be calculated as follows:

|  |  |  |
| --- | --- | --- |
|  |  | ( 7 ) |

For the condenser, the heat rejected to the seawater when there is not a recuperator is given by:

|  |  |  |
| --- | --- | --- |
|  |  | ( 8 ) |

The recuperator temperatures were restricted by the following expression:

|  |  |  |
| --- | --- | --- |
|  |  | ( 9 ) |

Where the recuperator efficiency (*ηr*) is given by the following equation assuming constant specific heat:

|  |  |  |
| --- | --- | --- |
|  |  | ( 10 ) |

### Expander and pump

The power output of the expander is given by:

|  |  |  |
| --- | --- | --- |
|  |  | ( 11 ) |

Where *h2s* is the enthalpy at low pressure after an isentropic expansion.

The power input of the pump is given by:

|  |  |  |
| --- | --- | --- |
|  |  | ( 12 ) |

Where *h4s*is the enthalpy at high pressure after an isentropic compression.

## WHRS Performance

The net power () is the addition of all the power flows of the WHRS. The thermal efficiency is given by:

|  |  |  |
| --- | --- | --- |
|  |  | ( 13 ) |

is the heat absorbed by the economiser or heat exchanger (per Figure 4 and depending if there is a thermal fluid circuit or not), and it is less or equal than the total heat available in the exhaust gas. *h4(r)* stands for *h4* when the plant does not have a recuperator and *h4r* when it does. The electric power output is given by:

|  |  |  |
| --- | --- | --- |
|  |  | ( 14 ) |

Table 2 shows the different constraints and conditions imposed to the different parts of the WHRS model. These values remain constant unless the contrary is stated. Assumptions made for the thermodynamic simulations were that the electrical efficiency of the WHRS’s generator and pumps was 90%, while the thermal efficiency of the expander and working fluid pumps was 75%; all efficiencies were assumed to be constant at off-design conditions. Also, it was assumed that there were no pressure changes except for the expander and pump; no heat losses or leakages and it was a steady-state operation. The recuperator efficiency was set to 90% which guaranteed a recuperator *Tap* of more than 4 K, inside the capabilities of a plate heat exchanger. Saavedra *et al*. (2010) studied the effects of changing the sink temperature while keeping constant the inlet temperature and waste heat mass flow rate. They found that changing the sink temperature by 55°C, there was a 35% drop in power output when using n-heptane: 37% for water and 39% for toluene. In this paper, a constant inlet sink temperature of 9°C was assumed. This value is close to the mean annual ocean temperature (Rayner 2003; Rayner *et al*. 2006).

|  |  |  |  |
| --- | --- | --- | --- |
| ***Equipment*** | ***Variable*** | | ***Value*** |
| *Expander* |  |  | |
|  | Inlet temperature, *T1* (°C) | 0.975*TH,i* | |
|  | *ηexp* | 75% | |
|  | Exit pressure, (kPa) | 10 | |
| *Condenser* |  |  | |
|  | *Tap* (°C) | > 5 | |
|  | *ΔTpp,C* (°C) | 5 | |
|  | Inlet Sink Temperature, *TS,i* (°C) | 9 | |
| *Recuperator* |  |  | |
|  | *ηr* | 90% | |
| *Pump* |  |  | |
|  | Inlet temperature, *T3* (°C) | Liquid saturation temperature at low pressure | |
|  | *ηp* | 75% | |
| *Heat Exchanger/Economizer* |  |  | |
|  | *Tap* (K) | > 5 | |
|  | *ΔTpp,H* (°C) | 5 | |
|  | Inlet Source Temperature, *TH,i* (°C) | Per Figure 3 | |
|  | Minimum exit Source Temperature, *TH,o\_min* (°C) | 165 | |

Table 2: Operating parameters for the thermodynamic cycles.

## Marine Diesel Generator

In order to measure the CO2 benefit of a marine WHRS connected to the vessel grid, a simple model of a diesel generator was used. The generator was a Wärtsilä 16V32 genset that works with MDO and has a fuel consumption of 176 g kWh-1 (Wärtsilä 2013). It was assumed that the electricity generated by the WHRS will reduce the genset’s work load, hence its fuel consumption:

|  |  |  |
| --- | --- | --- |
|  |  | ( 15 ) |

Where *SFOCg* is the fuel consumption of the diesel generator in g kWh-1 and *CF* is the carbon conversion factor set at 3.206 Tonne of CO2 per tonne of MDO is used (International Maritime Organization 2009c).

## Operating hours

It was assumed that the WHRS would be installed in a ship in operation for 6,480 hours per year (i.e. it is in use around 75% of the time) for all its useful life, assumed in this paper to be 20 years. The operating profile, shown in Figure 6, is similar to that found for a 14,000 twenty-foot equivalent unit container ship (MAN Diesel & Turbo 2012).



Figure 6: Ship's annual operating profile while at sea in percentage and hours per year.

The vessel’s annual CO2 reductions are found by multiplying, for each engine’s loading condition, the *CS* per the hours operated:

|  |  |  |
| --- | --- | --- |
|  |  | ( 16 ) |

Where *n* is the number of observations for the engine’s MCR, which for this paper is 18, and *t* is the time spent at each of the engine’s loading conditions as given in Figure 6.

## Model Validation

The code was validated against results presented by Butcher and Reddy (2007) and Saavedra *et al.* (2010). The validation process indicated that the WHRS model had an agreement to within 1.4% of the results obtained by Butcher and Reddy (2007). This difference is caused mainly by visual measurement errors when reading the mass flow rates and pinch point temperature graphs from the paper; data tables were not available. When the model is compared against the work of Saavedra *et al*. (2010), the difference was 0.95%, a small difference due to the varying approaches used to calculate the thermodynamic properties of the working fluids. Saavedra *et al*. integrated the Peng-Robison Stryek Vera equation of state, while in this work the data was extracted from NIST Refprop 9.0

## Software and databases

The model is programmed in Matlab®. In particular, tools from NIST Refprop 9.0 (Lemmon *et al*. 2010) for the working fluids, CoolProp (Bell *et al*. 2014) for the thermal oil and the code developed by Sharqawy and Mistry (2010) for seawater properties are used.

# RESULTS AND DISCUSSION

## WHRS performance at constant engine loading

In the first section, the engine loading was set at 75% MCR. The thermal efficiency and electrical power output were analysed for different working fluids.

Figure 7 presents the net electrical power output change with respect to the system’s high pressure (*P1*). For all the working fluids, the electrical net power output increases as *P1* decreases. Since *T1* and *Tpp\_H* are fixed, as *P1* increases the latent heat and *h1* are reduced, also decreasing the mass flow rate and power output. The minimum and maximum pressure values are limited by the constraints given by equations (4)-(6). Benzene is the working fluid that produces the largest electrical power and has the highest thermal efficiency (Table 3). Water was found to reject the most energy absorbed from the exhaust gas to the sink because the RC cannot use a recuperator since the expander and pump exit temperatures are the same.

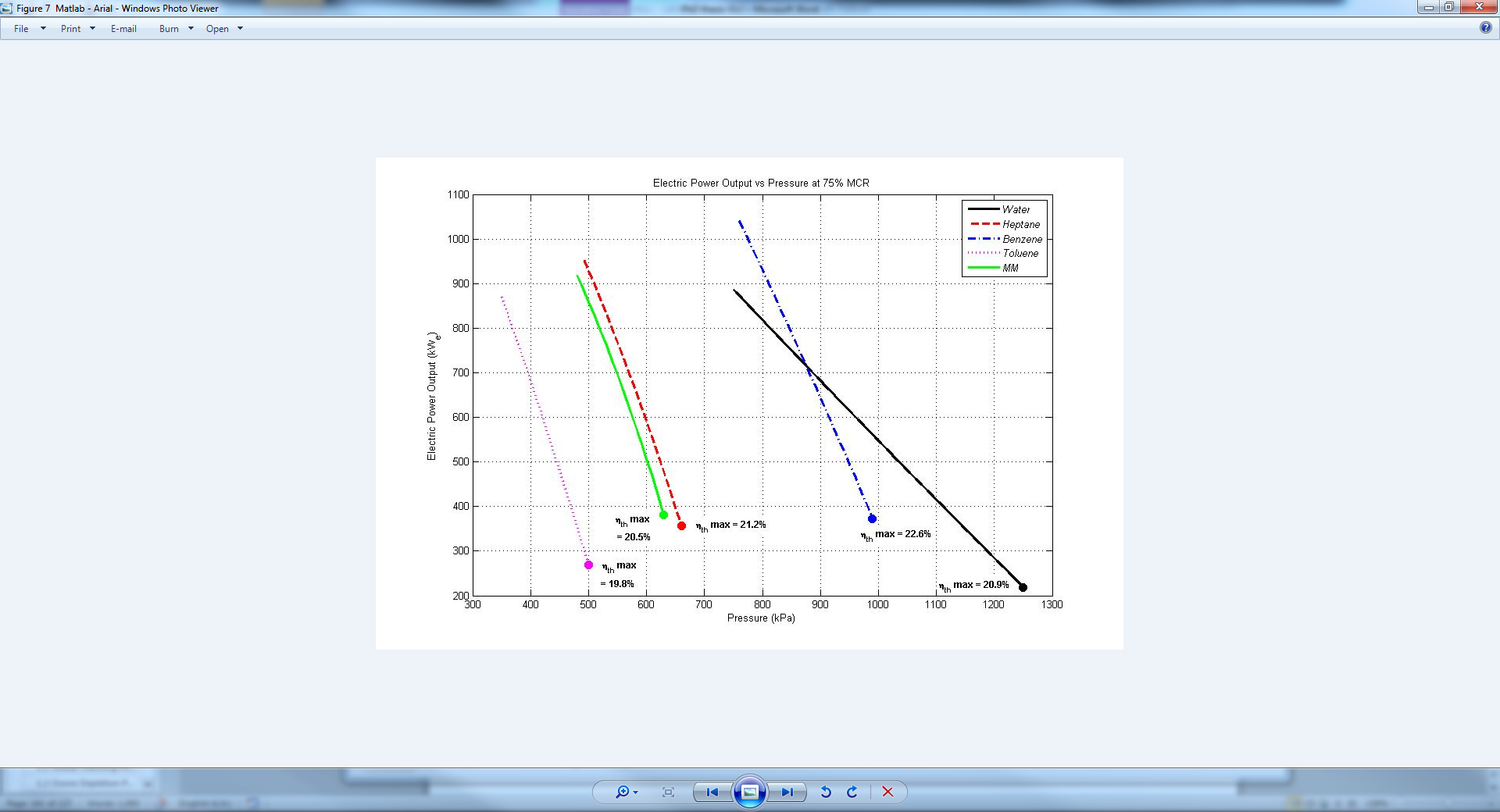


Figure 7: WHRS electrical net power output at 75% MCR with the WHRS maximum thermal efficiencies shown.

|  |  |  |  |
| --- | --- | --- | --- |
| **Working Fluids** | **(kWe)** | ***P***  **(kPa)** | ***ηth***  **(%)** |
| *Water* | 886 | 750 | 19.0 |
| *Heptane* | 959 | 490 | 20.4 |
| *Benzene* | 1,040 | 760 | 22.1 |
| *Toluene* | 869 | 350 | 18.7 |
| *MM* | 918 | 480 | 19.8 |

Table 3: Data at the maximum net electric power output for different working fluids.

In Figure 8 it is shown that all working fluids increased their thermal efficiency as the high pressure increased. This behaviour is caused by equation (13) which measures thermal efficiency with respect to the heat absorbed, and not by the total heat available at the exhaust gas. As the pressure increases, so does the saturation temperature, reducing the heat absorbed by the cycles. So, although the thermal efficiency increases, the total power output declines as seen in Figure 7. At low pressures, the saturation point is reached at lower temperatures, allowing the working fluid to absorb more heat from the waste heat source. The power demand from the pumps begin to dominate as the pressure increases, causing a reduction in the thermal efficiency gradient. The behaviour of the WHRS thermal efficiency, as measured in this work, is also discussed by Liu *et al*. (2004) but instead of using the pressure as independent variable they used the saturation temperature. Benzene delivered a maximum thermal efficiency of 22.1% while toluene achieved the lowest at 18.7%.

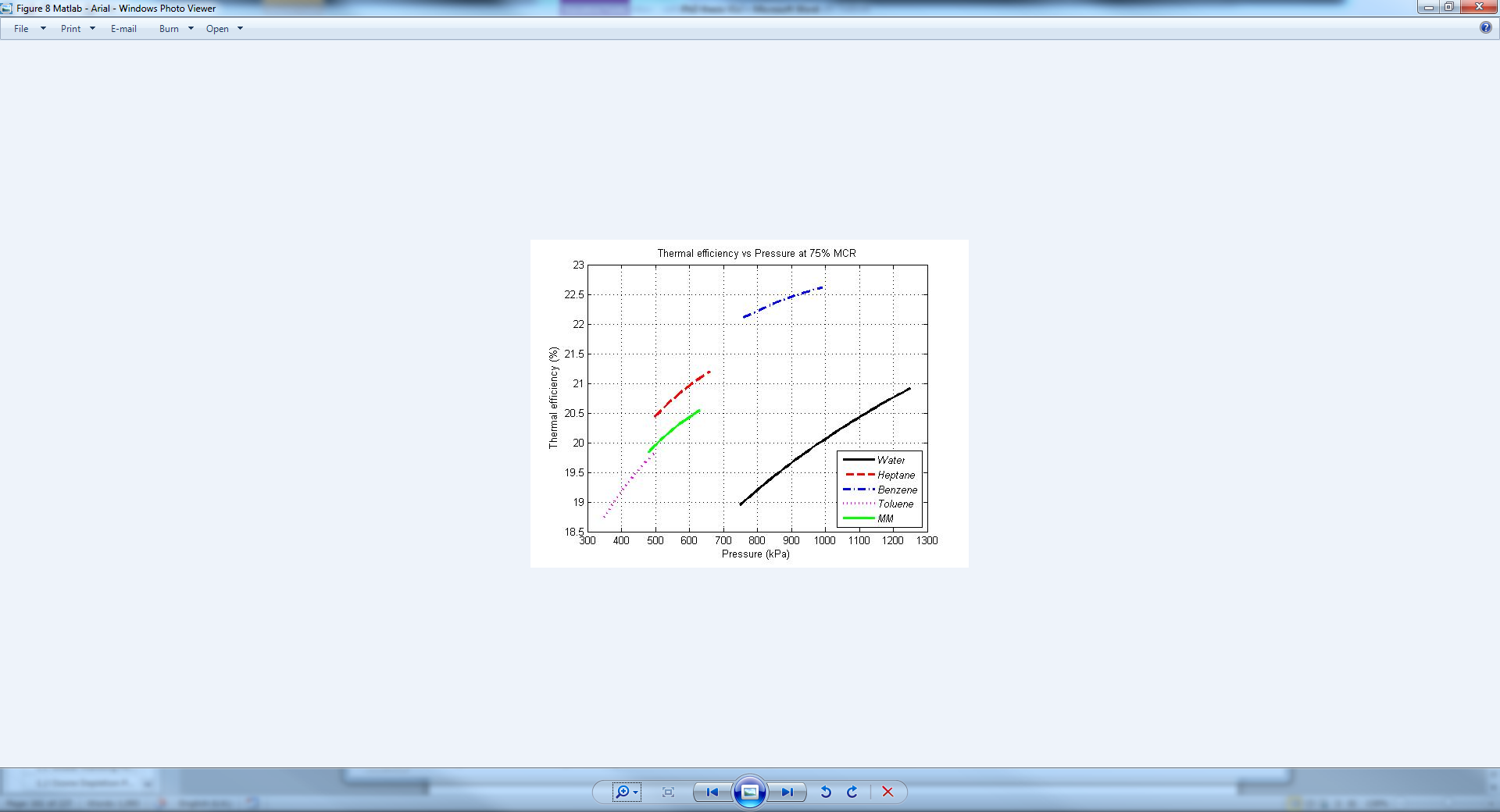


Figure 8: Thermal efficiency achieved by the working fluids at 75% MCR.

The ORC operational lines seen in Figure 8 are shorter than those for the water system. This is caused by the thermal oil’s temperature gradient between the heat exchanger’s entrance and exit. The restrictions affected designs with higher *P1* which tend to have higher thermal efficiencies.

## WHRS performance at different engine loadings

This section presents the results obtained when the operating conditions of the ship’s engine were changed, as shown in Figure 3. The values used are at the maximum electric net power output obtained for every engine load and working fluid.

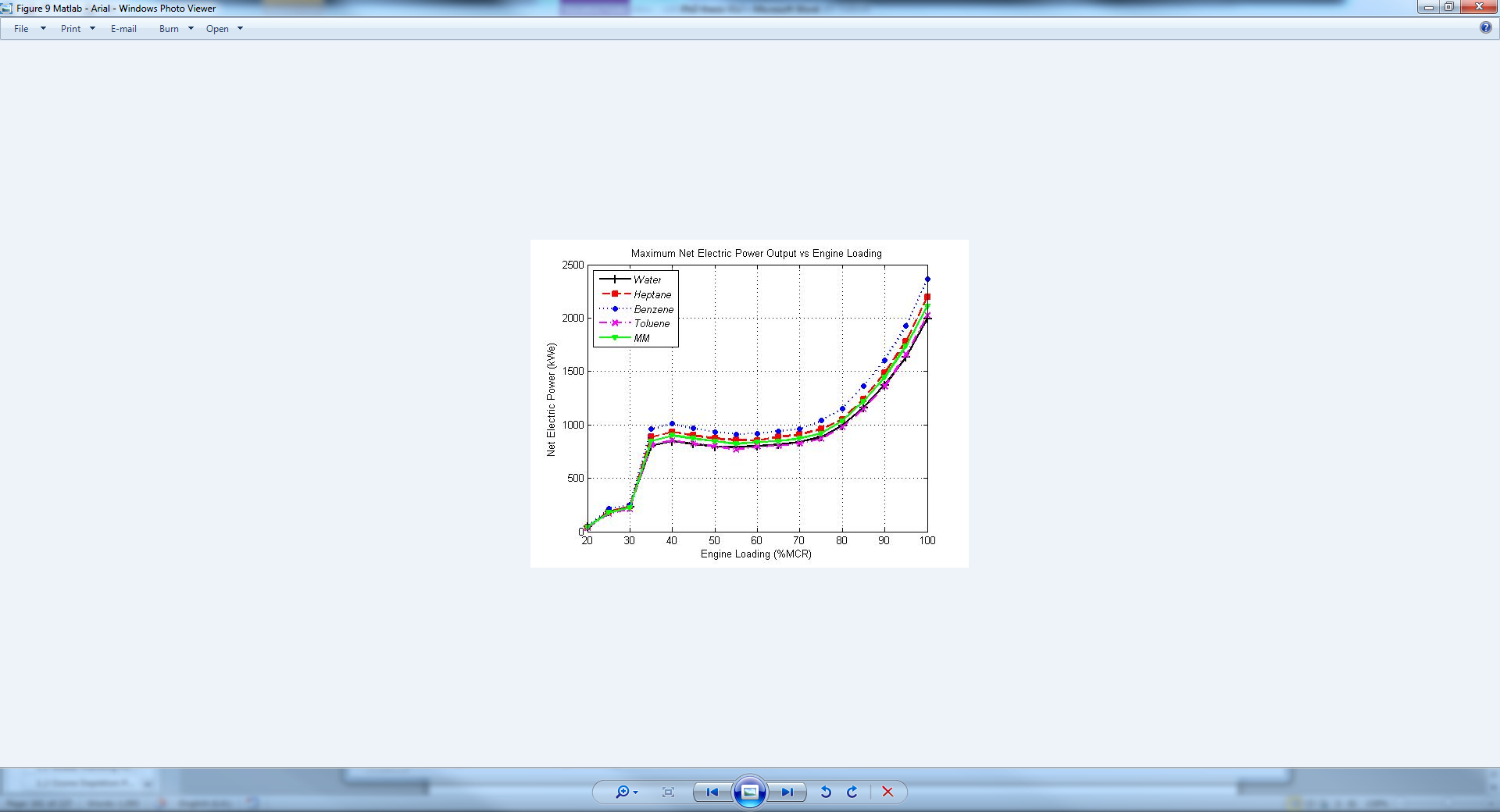


Figure 9: Maximum net electric power output at different engine loadings.

As can be seen in Figure 9, the electric power output tends to increase with the engine loading. After the 75% region, a higher growth in net power output for all the working fluids is observed. There is also a rapid increase in power output after the 30% MCR when the turbos start to operate. There are mainly two reasons for this: a) the exhaust mass flow rate is increasing with the engine loading, deriving more waste heat to absorb, and b) the exhaust gas temperature increase requires a higher *P1* which translates into a larger energy drop within the expander.

The maximum CO2 emission reduction per hour is achieved by benzene at 100% MCR with 1.4 tonnes of CO2. A RC WHRS in the same engine loading saved 1.2 tonnes of CO2. This represents reductions of 3.1% for benzene and 2.6% for water when compared to the engine’s own fuel consumption given at almost 14 tonnes of MDO per hour. The biggest marine ORC benefit compared to a water-based RC is found at high engine loading, while at low load they have similar performances.

The thermal efficiencies for the different marine WHRS are not only strongly influenced by the heat quality of the vessel’s exhaust gas, but also by heat availability. From Figure 3 it is seen that at 35% MCR, there is a temperature rise of more than 42°C in the exhaust gas temperature compared to the 30% MCR due to the turbo operation, and the highest temperature of the engine profile. At this point all five working fluids have their maximum thermal efficiency (see Figure 10) with benzene at 23.60% and water being the lowest at 19.5%. Beyond this point, the temperature and thermal efficiency find their highest points at 100% MCR. Interestingly, even though the exhaust gas temperature at 100% MCR is 22˚C lower than at 35%, the working fluids’ thermal efficiencies for benzene WHRS at 35% and 100% MCR are very similar at 23.64% and 23.60% respectively. The reason behind this is that the system has at these points the same *P1*, but for the 100% MCR case *T1* is lower due to cooler temperature of exhaust gas/thermal oil. This reduces the enthalpy change between the evaporator and superheater (see Figure 5), and per equation (13) the thermal efficiency increases.

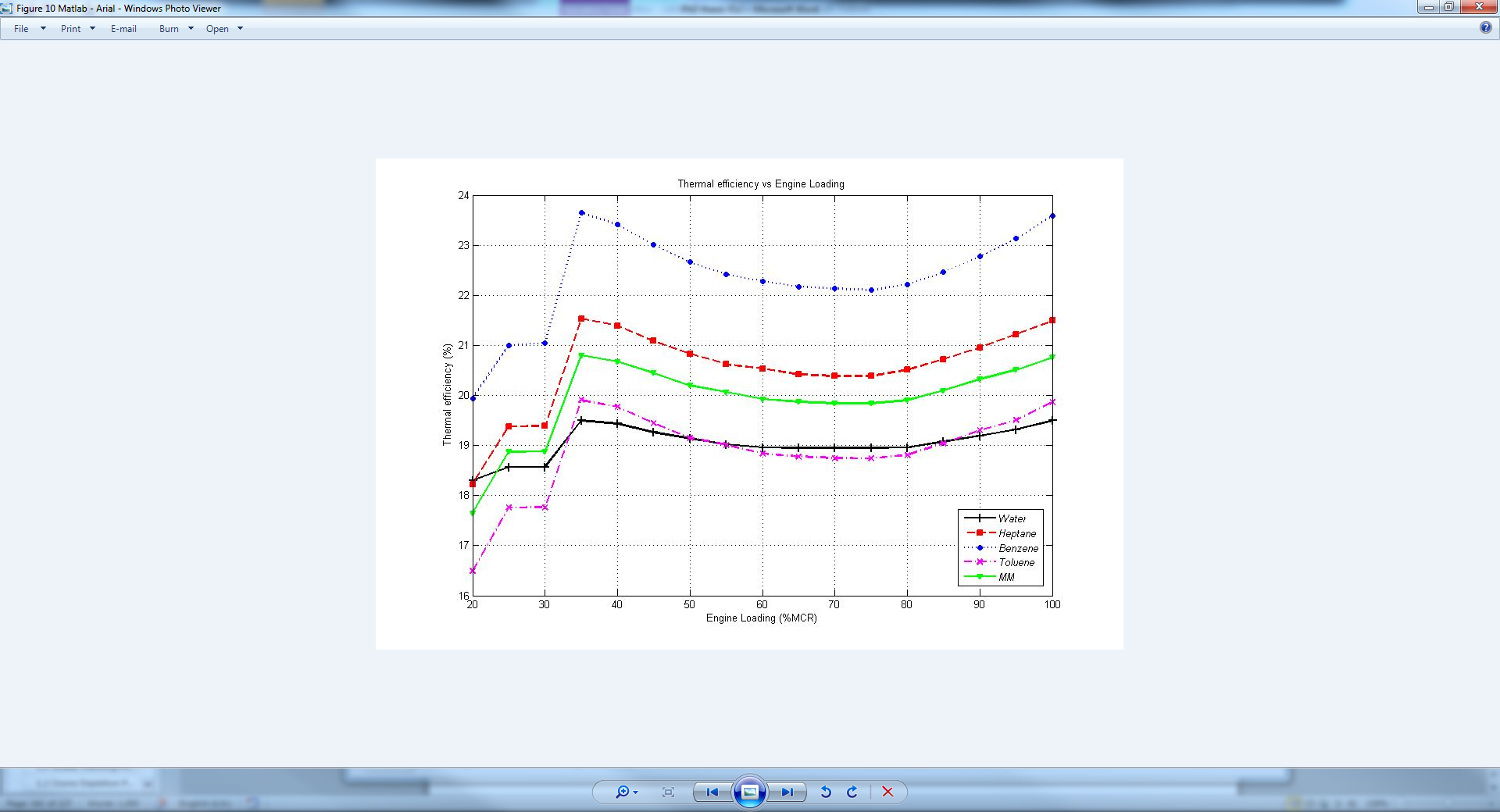


Figure 10: Working fluids’ thermal efficiencies at the maximum net electric power output for different engine loadings.

## Mass flow rate and an approximation to sizing

Figure 11 shows the large difference between organic fluids and water with respect to their mass flow rate. Organic fluids have a low latent heat and require a small amount of energy to vaporise, forcing larger mass flow rates to absorb the waste heat available (Maizza & Maizza 1996; Chen *et al*. 2010). In all scenarios, water had the lowest mass flow rate, while MM showed the largest - more than seven times larger than water at 100% MCR.

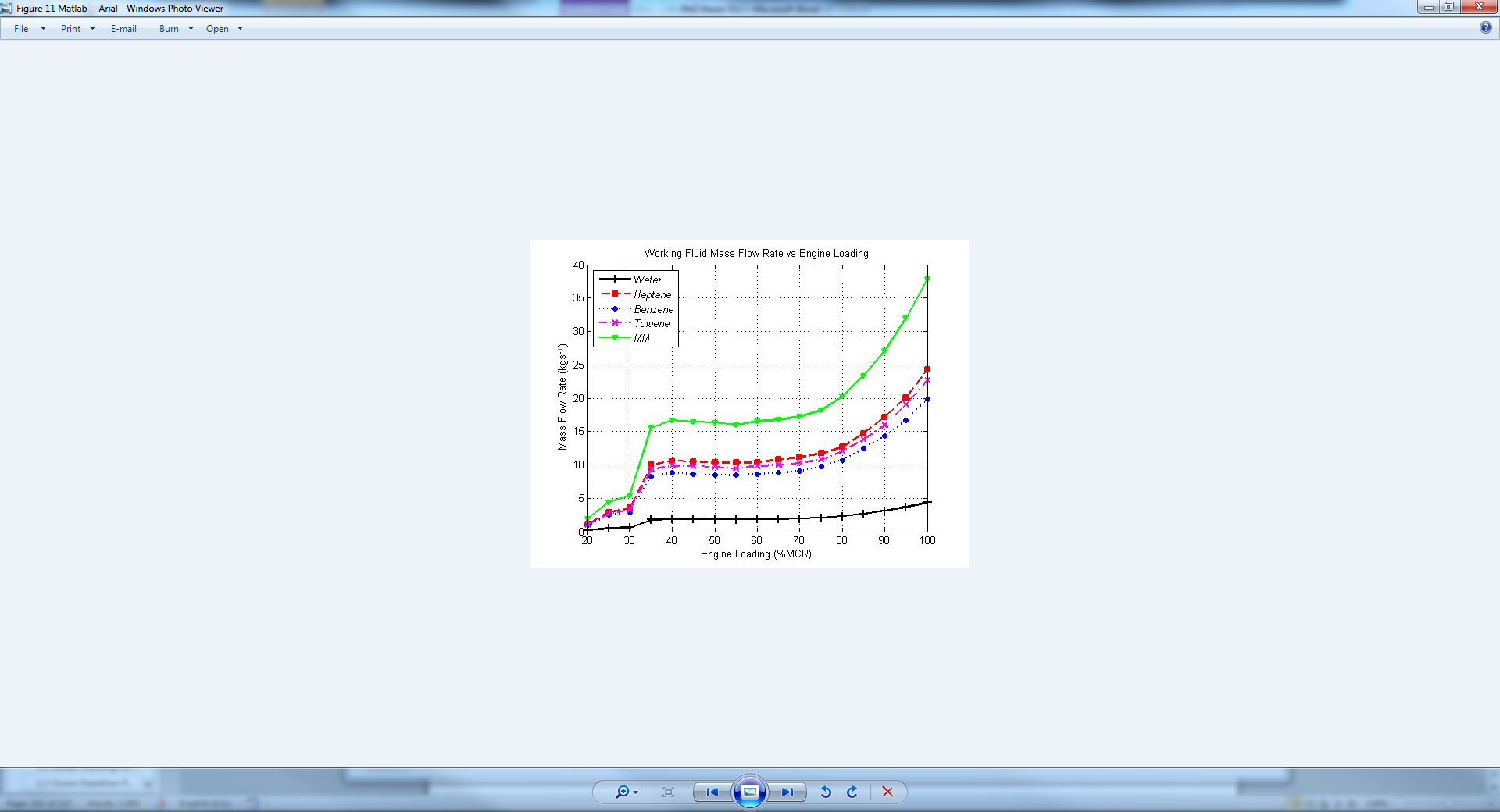


Figure 11: Comparison of the working fluid’s mass flow rate for different Diesel engine loads.

With the systems’ mass flow rate, the amount of working fluid required could be calculated. This could be an important factor in the total system’s weight on-board, but also defines the tank size. The volumetric flow rate, on the other hand, can give the first indication of size of other WHRS equipment such as heat exchangers and pipes. The working fluid density changes along the WHRS circuit where the lowest density is seen after the expansion process (point 2 in Figure 4) and the highest after the pump (point 4 in Figure 4). Table 4 shows that after the turbine, the organic vapours are denser than water, meaning that the volume flow rate for organic fluids is much closer to that of water. For example, the MM’s mass flow rate is more than seven times larger than water, while in comparing the volumetric flow rate at point 2, the difference is reduced to around 30%. For the high pressure side, the roles are reversed: now water is denser than the organic fluids and, in converting to volumetric flow rate, the difference is shown to be even larger than in the case of the mass flow rate. Using the same fluids as before, the difference between MM’s volumetric flow rate is above nine times what the RC requires.

While the scope of the paper does not include the study of heat exchangers, costs and piping, some basic comparison can be made in order to give a deeper understanding of how the volume flow rate affects the sizing and pressure losses of the different WHRS. With the numbers given in Table 4, the RC system brings the most compact plant without sacrificing performance. The ORC systems could have the same equipment size as a RC plant – if the working fluid velocity falls inside the recommended limits[[8]](#footnote-8)- but with a substantial increase in fluid velocity, especially after the pump. An increase in velocity would bring a larger pressure drop which in turn will reduce the plant’s power output (Shah & Sekulic 2003; Sinnott 2005). If the fluid velocity is maintained equal for the different systems, the WHRS with largest volumetric flow rate will require larger pipe diameters and cross-sectional areas for the heat exchangers, having an impact on plant mass, volume and cost.

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| **Working Fluids** | **(kg s-1)** | ***2*** | | ***4*** | | |
| ***ρ***  **(kg m-3)** | **(m3 s-1)** | ***ρ***  **(kg m-3)** | | **(x10-3 m3 s-1)** |
| *Water* | 4.4 | 0.07 | 63.9 | 990.16 | 0.4 | |
| *Heptane* | 24.2 | 0.29 | 84.5 | 671.04 | 3.6 | |
| *Benzene* | 19.9 | 0.25 | 78.1 | 879.13 | 2.3 | |
| *Toluene* | 22.6 | 0.27 | 82.7 | 843.34 | 2.7 | |
| *MM* | 37.8 | 0.45 | 83.7 | 747.70 | 5.1 | |

Table 4: WHRS' mass flow rate at 100% MCR. The table also shows the different system densities and volumetric flow rates after the turbine (point 2) and the pump (point 4).

## Safety on-board

Having on-board flammable fluids such as the organic fluids shown in Table 3 presents a major hazard which must be assessed and the vessel design adapted to provide the highest safety level possible. As mentioned before, SOLAS prohibits the presence of low flash point fluids inside the engine room, meaning an ORC WHRS must be located in a separate compartment. Other important safety measures for highly flammable fluids can be drawn from the IGF[[9]](#footnote-9) draft since LNG flash point is around -188˚C (Burel *et al*. 2013), much lower than the organic fluids used in this paper. The IGF recommends the use of double-walled piping, ventilation for the enclosed space to the atmosphere, gas detection and firefighting systems according to the fluid used. Equally important are personnel training, risk management (e.g. Enclosed Space Management system, see Allan 2015) and assessments, and safety procedures in case of fire or leakages (Suárez de la Fuente *et al*. 2015).

The safety requirements for the marine ORC WHRS selected will increase the initial and maintenance cost, but performance may also be affected. For example, depending where the ORC compartment is installed there will be an increase in pump power, for both seawater and working fluid, due to the existence of a larger distance to the heat source and sink. The minimisation of cost and performance loss due to safety requirements should be assessed at the early stages of vessel design.

## Short and long term marine WHRS benefits

Using a container ship operating profile (Figure 6), it is possible to assess the environmental benefits of installing a marine WHRS. The overall advantage of installing a marine WHRS is a vessel with reduced fuel consumption and GHG emissions.

Figure 12 shows that the vessel operator will reduce CO2 and other GHG emissions which, with the potential future introduction of carbon taxes to shipping, could also be financially beneficial. A marine WHRS could bring, at the minimum, savings in the range of 3,300 tonnes of CO2 per year and around 65,500 tonne of CO2 in the ship’s operational life.

Figure 12 also demonstrates the advantage of using an ORC WHRS: benzene WHRS has an advantage of more than 16% over the traditional RC which will translate over the vessel’s lifetime to a reduction in fuel worth 11,000 tonnes of CO2 when compared to the RC. The CO2 reductions for water and toluene are almost identical.

It is important to compare system complexities and characteristics in order to distinguish which system is better. As seen in Figure 11, toluene has the largest mass flow rate which implies from the previous section that the WHRS will require more space inside the vessel. For some ship types which are weight-driven, such as bulk carriers, this may not be an issue. On the other hand, a toluene expander would be simpler than one designed for water since toluene is a dry fluid, not requiring stages in order to avoid condensation inside the expander.



Figure 12: CO2 emission reductions achieved by the WHRS under different operating conditions in one year.

These results can be seen as over-optimistic due to the assumption of having an adaptable WHRS. But they do follow the trend seen in the work of Alvik *et al*. (2010), achieving high CO2 emission reductions which is in line with shipping industry’s green objectives. One of the main caveats of installing an WHRS is the initial cost. Quoilin *et al*. (2013) show that the cost of an ORC WHRS module of the size proposed in this paper should be less than £1,640 per kWe[[10]](#footnote-10), while for an RC WHRS the cost can range between £688 and £862 per kWe (BCS Incorporated 2008). However, the WHRS payback period can be less than three years for any of the cycles tested (Suárez de la Fuente *et al*. 2015). Thus, the installation of a WHRS is an economically-viable green technology.

Finally, this study has demonstrated that under the same operating conditions and restrictions, ORC WHRS can reduce fuel consumption and environmentally harmful gases to a greater extent than the traditional water-based RC. This behaviour is also seen in the work of Saavedra *et al.* (2010).

# CONCLUSIONS

This paper has compared the performance of a WHRS based on the RC using five different working fluids, water and four organic fluids with the aim of achieving the highest CO2 emission reduction by covering part of the vessel’s electrical demand - normally supplied by a diesel generator - while also supplying the vessel’s steam requirement.

The authors first presented the results obtained by modifying the high pressure level of different WHRS for the vessel’s slow speed diesel main engine at 75% MCR. It was possible to distinguish that an ORC WHRS - except in the case of Toluene - could offer a larger power output at higher thermal efficiency than a RC WHRS. The results also showed that the maximum power output did not occur when the WHRS was operating at its maximum thermal efficiency.

In the second part of the simulations, the objective was to find how the RC and ORC WHRS behaved under different ship engine loading conditions without fixing the WHRS design. It was seen that the best performance of the ship’s WHRS was located in the higher engine loading region (i.e. above 75% MCR); this is caused by the amount and quality of the waste heat in the engine exhaust gas. Moreover, it was observed that an ORC WHRS could save up to a maximum of around 77,300 tonnes of CO2 in the ship’s operating life, a 16.7% improvement over what can be delivered by water based RC. Furthermore, it was demonstrated that an ORC WHRS (except for toluene) can outperform the traditional water RC WHRS over the full range of the ship’s loading conditions. Still, it is important to consider that the large mass flow rates of organic fluids will require more space inside the ship, and in addition there will be an installation cost increase due to the flammable and toxic characteristics of such fluids.

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# Nomenclature

|  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **Symbol** | | **Name** | | | | **Units** | | **Symbol** | **Name** | | | | | **Units** | |
| *CS* | | Carbon Savings | | | | Tonne of CO2 h-1 | | *t* | Time | | | | | h | |
| *CF* | | Carbon Factor | | | | - | | *T* | Temperature | | | | | K | |
| *Cp* | | Specific heat | | | | kJ kg-1 K-1 | |  | Volume flow rate | | | | | m3 s-1 | |
| *h* | | Enthalpy | | | | kJ kg-1 | |  | Power | | | | | kW | |
|  | | Mass flow rate | | kg s-1 | | | **Greek Symbol** | **Name** | | | | **Units** | | | |
| *P* | | Pressure | | kPa | | | *ε* | | Ratio of mechanical power returned | | - | | | |
|  | | | | Heat transfer rate | | | kW | *ρ* | | | Density | | | | kg m-3 | |
| *SFOC* | | | | Specific Fuel  Oil Consumption | | | g kWh-1 | *η* | | | Efficiency | | | | % | |

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Subscripts and superscripts** | | **Name** | **Subscripts and superscripts** | **Name** | |
| *a,b* | Thermal oil circuit section | | *o* | | Outlet/Out |
| *ap* | Approach | | *min* | | Minimum |
| *C* | Sink | | *n* | | Observations |
| *co* | Condenser | | *p* | | Pump |
| *c* | Critical | | *pp* | | Pinch Point |
| *e* | Electrical | | *r* | | Recuperator |
| *eq* | equipment | | *rvap* | | Reduced saturated  vapour point |
| *ev* | evaporator | | *S* | | Seawater/Sink |
| *exp* | Expander | | *Sat* | | Saturation |
| *g* | Generator | | *sc* | | Subcooler |
| *gen* | Generated | | *sh* | | Superheater |
| *H* | Source | | *t* | | Net |
| *h* | High pressure | | *th* | | Thermal |
| *ha* | Heat absorbed | | *v* | | Vessel |
| *i* | Inlet/In | | *wf* | | Working fluid |
| *l* | Low pressure | |  | |  |

1. Average price for a tonne of MDO between July 2013 and October 2014. [↑](#footnote-ref-1)
2. For Toluene: (Forster *et al*. 2007); for Heptane: (Environmental Protection Agency 2013) ; and for MM: (Environment Agency 2011). [↑](#footnote-ref-2)
3. Taken from (Moore *et al*. 2014) at 101,325 Pa. [↑](#footnote-ref-3)
4. Taken from (Yaws 1997). [↑](#footnote-ref-4)
5. For Water: (Bilgen *et al*. 1977), for Benzene: (Brioukov *et al*. 1999), for Toluene: (Hnat *et al*. 1982), for heptane:(Appleby *et al*. 1947) , and for MM: (Heberle *et al*. 2013). [↑](#footnote-ref-5)
6. Taken from (Yaws 1997). [↑](#footnote-ref-6)
7. Critical points were obtained from (Lemmon *et al*. 2010). [↑](#footnote-ref-7)
8. The fluid’s velocity limits have the purpose of restricting fouling and corrosion, while also avoiding stagnant flows, inside the heat exchangers (Hesselgreaves 2001; Shah & Sekulic 2003; Sinnott 2005; Rodriguez & Smith 2007; Shen *et al*. 2014). [↑](#footnote-ref-8)
9. International Code of Safety for Ships using Gases or other Low Flashpoint Fuels. [↑](#footnote-ref-9)
10. An exchange rate of €1.37 to £1; and $1.53 to £1 was used (European Central Bank 2013; Board of Governors of the Federal Reserve System 2015). [↑](#footnote-ref-10)