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Energy Consumption in HVAC System and Occupants' Thermal Comfort Optimization Using BIM-Supported Computational Approach

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Abstract: Presently, buildings are vastly responsible for total energy consumption and GHG emissions in Hong Kong with indoor comfort being a major concern for the building industry. Comfort involves control of temperature, humidity, air velocity and indoor air quality interacting with occupants. Building owners are becoming increasingly aware of the importance of comfort with minimal energy consumption for the building occupants. As such engineers and experts are being challenged to design systems that provide a comfortable and energy efficient environment to solve this problem. The main aim of this study is to evaluate the indoor built environment of the IAS lecture theatre at the Hong Kong University of Science and Technology (HKUST). which is served by VAV type air-conditioning system and to investigate the effect of layout of diffusers on occupant thermal comfort in CFD as well as analyze energy consumption and CO2 emission by the system in eQuest. This study presents how Building Information Modelling (BIM) and Computational Fluid Dynamics (CFD) can be utilized to address thermal comfort and energy efficiency in buildings by optimizing HVAC system to provide well controlled environment. Indoor temperature, air velocity, CO_2 concentration, Predicted Vote Mean (PMV) and Predicted Percentage of Dissatisfied (PPD) were selected as the factors of evaluation with lightings, space cooling and other equipment being simulated in eQuest for energy analysis. Multiple scenarios with different numbers of occupants at different locations were simulated to assess the thermal comfort range which found the current design of the HVAC system could not sustain the high internal loading from the occupants when either all the seats were occupied, or only the back zone was occupied with the front seats vacant. In this paper, alternative method has been proposed for HVAC layout to address the simulated result of uneven temperature distribution which indicated design problem of overheating at the back rows and over cooling at the front rows, to achieve the desired comfort range with optimized energy consumption.

Keywords: BIM; thermal comfort; energy consumption, computational fluid dynamics, HVAC optimization.

1. Introduction

Human comfort in an optimized energy efficient built environment [McQuiston and Parker, 1982] is an integral part of sustainable design particularly in a densely populated and warm humid climatic zone [2] like Hong Kong where people spend 85% of their times indoor during daytime [3]. Therefore, it is imperative to create a good indoor climate for their wellbeing. Generally, the prospects for improving building environmental performance occur at the early design phase or preconstruction stages as the possibility of making changes at this point is thought to be best [Azhar and Brown, 2009]. Extraction of building information directly from a model is made possible by BIM, to perform various building performance analyses such as daylighting, building energy use, indoor air quality (IAQ) and thermal comfort. Occupant thermal comfort affected by temperature, humidity, and airspeed, is one of the crucial design criteria for building sustainability [2]. Lecture theatres, or large rooms, with unpredictable occupancy patterns, present a special design challenge for professionals [5]. The classic approach of measuring occupant thermal comfort by means of Predicted Mean Vote (PMV) and Predicted Percentage of Dissatisfied (PPD) has been introduced by Fanger [6]. However, acquisition of data for the calculations of PMV and PPD requires time and specific instrument which may not always be available. Researchers have been looking for alternative tools for the prediction of thermal comfort and the recent findings show that Computational Fluid Dynamics (CFD) has become a popular tool. The installed ventilation systems must be designed to respond to the fluctuating fresh air demand but without excess supply of conditioned fresh air. The demand-controlled air-conditioning system, from which fresh air is supplied to meet its demand based on the levels of CO_2 in the room, provides a feasible solution that is being widely used [5]. According to the Hong Kong EPD guidelines there is a requirement of the CO_2- concentration not exceeding 0.1 % in non-residential buildings [7]. IAQ can therefore not be ignored as the regulations must be accommodated. Moreover, escalating energy costs in more recent times have caused increased interest in efficiency of operation. The wide use of HVAC systems also leads to up to 40% of the total energy demand in buildings [8]. Poor design of the control system accounts for this high degree of energy consumption. Many of these systems could not precisely measure the demand for cooling energy, leading to over-cooling and wastage of energy. Therefore, there is a need to optimize the design of the HVAC system while increasing the occupant thermal comfort.

Previous studies have examined factors influencing occupant thermal comfort with the use of airflow analysis. However, most research hardly explored the effect of layout of diffusers by BIM integrated CFD and energy simulation. This paper gives an overview of the current HVAC system configuration, its design specification, and operational criteria of the theatre. The primary objectives of the research are:

1. to evaluate the effect of the air distribution system in the lecture theatre towards the occupant thermal comfort through CFD simulation.
2. to propose alternative method for HVAC layout design which can provide optimum air flow to prevent uneven temperature and reduce the buildup of indoor contaminant in the audience area.
3. to analyze the existing energy consumption pattern with eQuest [9] which can be helpful in future to recommend further modification or newer methods for better optimization.

Before proceeding to the design stage, building a 3D BIM model of the lecture theatre is of the utmost importance to visualize the real scenario. Therefore, validation of simulated results to correspond to reality in terms of temperature, wind speed and CO_2 concentration is required. On-site measurements and observation will be carried out for the validation process. The findings and conclusion drawn from this study should have a significant contribution to the future of HVAC system design with the use of BIM. In long term, it could save time and cost in the design process while attaining the desired comfort range with optimized energy consumption.

2. Materials and Methods

The methodology has been illustrated in Figure 1. Collected data for the research are shown in Table A1 and Figure A1 in Appendix A. There are two air handling units (AHU 01 and 02) below the theater which supply air to 29 diffusers near the floor, 6 on the ceiling on the left zone and remaining 29 diffusers on the right zone respectively. The air to the middle zone is supplied by both AHUs and the supply air duct is located under the middle zone (row 6). It has a VAV type HVAC system programmed to deliver a specific quantity of air to the space. The system has no heating coil in it and operates with free cooling. It detects if the return air temperature is more than 24^{o}C, then the feedback sensor automatically lowers it down to 19^{o}C. The theatre is equipped with displacement ventilation where the diffusers are located under each row of seat on the vertical pane through which air is supplied directly to the occupied zone at low velocity forming a layer of warm air above the occupied zone and internal heat loads and contaminants being carried away by the return air [2]. This system is best suited for rooms where the room heights are more than 3 meters, hence appropriate for spaces like IAS theatre.

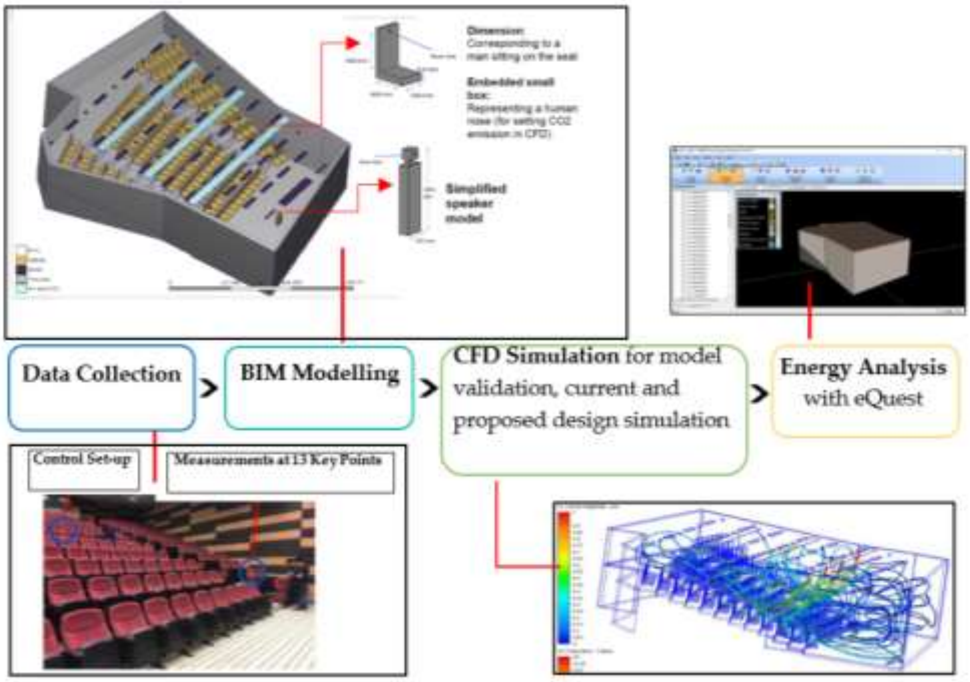
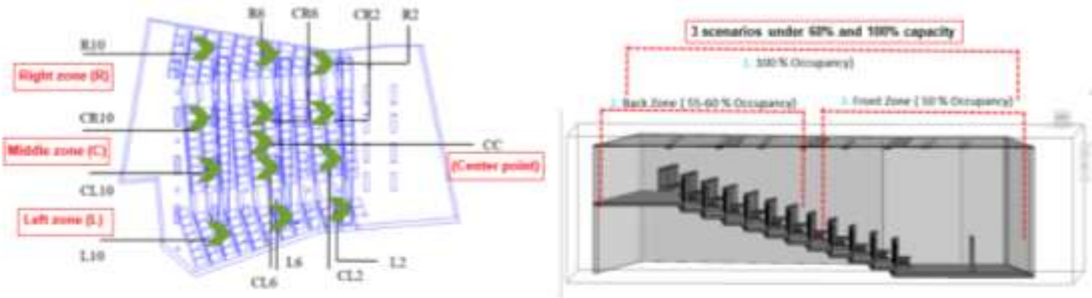


Figure 1. The first step was collection of information in two phases – field validation and scenario analysis. BIM model was created with Autodesk Revit using data from the lecture theatre which included seats, lighting fixtures, diffusers, seats with nose boxes and 1 human at the front zone as speaker which was then imported into Autodesk CFD [10] for scenario analysis with materials and boundary conditions assigned accordingly. The BIM model was exported to .xml file and converted to "DOE-2 File" under the ASHRAE defined climate zone for conducting energy analysis in eQuest.

According to ASHRAE, if area < 500 m², then one center point is enough for field validation. Although the area of the theatre is 241.043 m², for more accuracy of the verification, measurements were taken at 13 key points in alternative rows including center point in Row 6 [Figure 2(a)] to ensure continuous environmental change monitoring during point measurement. The location points were chosen to be uniformly distributed in the lecture theatre to study the spatial effects. The parameters that were measured are air velocity, temperature, and humidity.



(a) (b)

Figure 2. (a) Field validation phase - division of the theatre into 3 zones with 13 key points to measure the parameters (temperature, air velocity). 3 minutes were given for each point with reading at every 30s interval to monitor the change in velocity and temperature and to evaluate the temporal effects. (b) Scenario analysis phase - with full and half occupancy under 60% and 100% capacity. Measurement of CO_2 level with relative humidity were taken in the 13 points with each point at 10s reading.

Measurements of the flow rates in the AHU room were recorded directly coming from the return air, supply air and fresh air ducts [Appendix A] (Table A1). The required cooling capacity of the theatre is not very high as the usage rate is low given that only conferences will be carried out in the theatre occasionally, so the HVAC system will only operate when bookings are made. Even if there are conferences, seats do not fill up more than 50-60% as such the HVAC system always operates at 60% capacity.

As the supply air duct runs under row 6 which is at the center, the supply flow rate of the diffusers of other rows was calculated in accordance with the distance from row 6 which is assumed as the total supply flow rate. There are 6 rows above and 5 rows below row 6 excluding row 1 since row 1 does not have diffusers. Since the distance between alternate rows are the same,

$$\text{Geometric Factor} = \frac{\text{Total number of rows from row 5 to the furthest row} - \text{number of rows from Row x to Row 6}}{\text{Total number of rows from Row 6 to the furthest}} \quad (1)$$

For example, geometric factor for row 7 is (6-1)/6 = 5/6. The flow rate of a diffuser in row 6, y, is found by the following equation:

$$y = \frac{\text{Total supply flow rate of (AHU 01/ AHU 02)}}{\text{Total number of diffusers powered by (AHU 01/AHU 02)}} \quad (2)$$

For example, y for AHU 01 will be = $\frac{\text{Total supply flow rate of AHU 01}}{\text{Total number of diffusers powered by AHU 01}}$

So,

Flow rate of a diffuser in a particular row = y × geometric factor (3)

Velocity of a diffuser in a particular row = y × Geometric factor × Surface area of the diffuser (4)

Velocity of diffusers under 60% and 100% fan capacity can be determined by the above equations and applied in CFD boundary conditions settings. Furthermore, CO_2 concentration of the supply air is calculated by the mass conservation theorem. As we know, total supply air is the sum of fresh air and return air, therefore,

$$C_f\, Q_f + C_r\, Q_r = C_s\, Q_s, \quad (5)$$

where C_f = CO_2 concentration of fresh air, Q_f = flow rate of fresh air, C_r = CO_2 concentration of return air, Q_r = flow rate of return air, C_s = CO_2 concentration of supply air, Q_s = flow rate of supply air. C_f, C_r, Q_f, Q_r and Q_s were obtained by measurement in the AHU room with which C_s was calculated and applied in CFD settings.

To simulate the mixing of two fluids in CFD, scalar boundary was used to track their relative concentrations such as for the occupants, scalar boundary condition was set to 40,000 as exhaled air for occupants is 40000ppm (= 4% in air) [11]. Supply air velocity, temperature and CO_2 concentration were set for every diffuser [Appendix A] (Table A1).

For energy analysis, two primary scenarios with 200 occupants (100% Occupancy) and 100 occupants (50% occupancy) were selected. The sources of internal heat gains included: occupants (sensible and latent heat gain), lights (sensible heat gain only) and equipment [12].

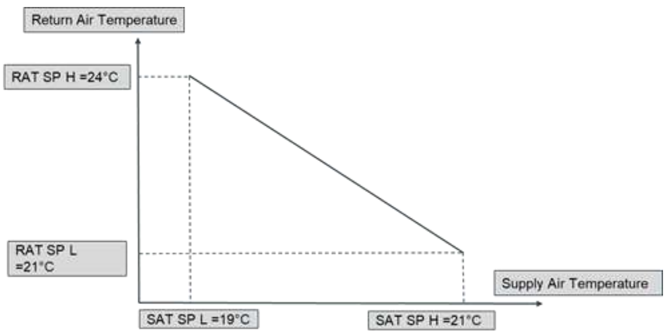


Figure 3. Cooling schedule showing the function of feedback sensor automatically adjusting the supply air temperature according to the measured return air temperature. When the return air temperature is more than 24°C, supply air is cooled to 19°C and the supply of chilled water is stopped when the RAT is less than 21°C. With reference to the DDC guide, proportional control is applied in

cooling set point [13] as such 50% of the throttling range should be taken as the set point sensor value. In this case, the throttling range is the average of return air temperatures i.e., 22.5°C, thus 22.5°C should be taken as the set point.

As the theatre does not operate 24/7 whole year, the daily levels of cooling capacity, occupancy, lighting vary. Therefore, daily, and weekly schedule for cooling, occupancy, and light are required in eQUEST to calculate the energy consumption. Equipment, lights, the HVAC system are on during the opening period. For lighting and equipment schedule, ratio of 1 is set during working hours; otherwise, ratio of 0.05 is set. During closing hours, the HVAC system is off while occupancy is zero. For cooling schedule, the HVAC system of the theatre is designed to follow the relation shown in Figure 3. Since free air cooling is used in winter, spacing cooling is off from November to March. Natural ventilation, infiltration, and solar radiation through windows are absent since there are no openings in the lecture theatre. The standards provided by EMSD HK were used for setting up the parameters which adopts an "Operating Schedule A: offices" according to PB BEC [14, 15]. See Table A2 in Appendix A for details.

3. Results

3.1. Results from CFD Analysis

3.1.1. Field Validation Simulation

Table 1. Comparison of measured and simulated flow field for Velocity and Temperature

	Measured Velocity (m/s)	Simulated Velocity (m/s)	Measured Temperature (Deg C)	Simulated Temperature (Deg C)
Average	0.2	0.1	22.8	22

The CFD model validation [Table 1] was carried out using collected data assuming the theatre has zero occupants. Temperature and air velocity of 13 points were compared between the results from CFD simulation and the actual measurement. The measured velocity is a little higher due to the presence of humans at that time. However, if it is omitted then the air flow range can comply with the actual value. According to ASHRAE [16], velocity < 0.2 is proved as excellent class. Hence, the air velocity in the lecture theatre falls under excellent class. The average difference between measured and simulated temperature is 3.5%. In practice, less than 5% difference would be acceptable. Therefore, it can be concluded the model validates the physical measurements for further simulation.

3.1.2. 100% occupancy scenario simulation results

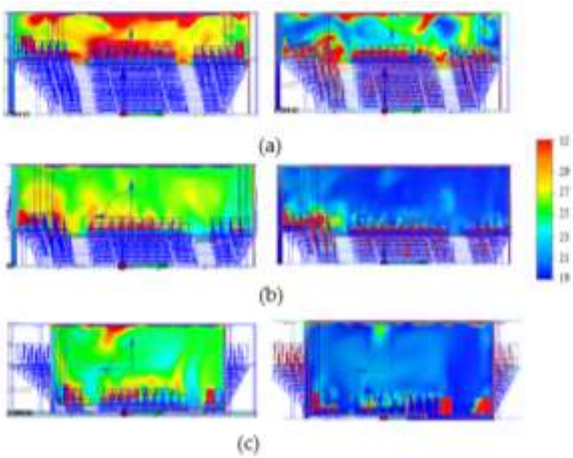


Figure 4. Comparison of temperature flow field between 60% (left) and 100% (right) HVAC capacity (a) Back zone (b) Middle zone (c) Front zone. A clear increasing trend from the bottom to the top row

is observed. The back zone due to absence of diffusers, sees the highest temperature scale (more than 27°C) which sharply differs from the range of standard temperature range 20-25°C [16]. In the front, middle zone and center point the temperature difference is minimal whereas at speaker zone it rises little. Only the temperature level at the speaker zone remains in 23~24°C range. The largest difference between the top row and the bottom row is 2.1°C which is considered a huge difference in a room and the difference could cause thermal discomfort to occupants.

CFD output shows the lack of supply air grilles at the back which results in a dead zone where the air movement is low. This zone has the highest values for the local mean age of air as air spends longer time in this region [Figure 5].

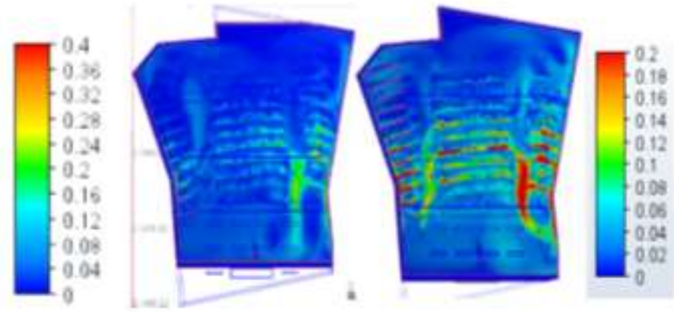
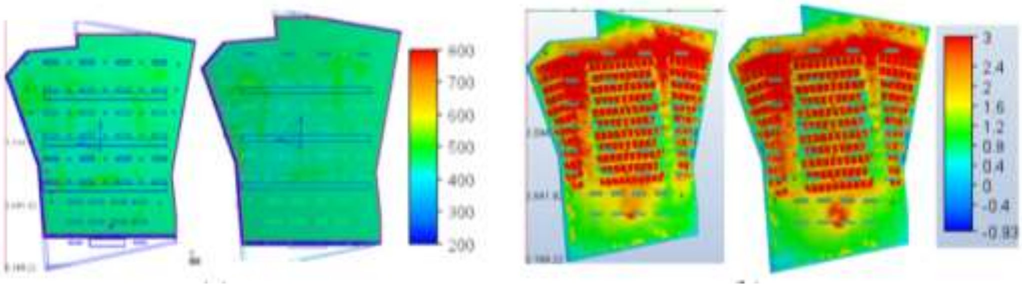


Figure 5. Comparison of velocity flow field between 60% (left) and 100% (right) HVAC capacity. Under 100% fan capacity, the airflow increases and reaches those zones which were not receiving any air flow before. However, the back zone remains as a dead zone. At the front zone with speaker, the air flow dropped to 33% which indicates the speaker will receive less airflow even after keeping the fan capacity at full speed.



(a) (b)

Figure 6. (a) Comparison of CO_2 flow field between 60% (left) and 100% (right) HVAC capacity. The approximate range for CO_2 level under both capacities is 430 - 470 ppm with the lowest being at the speaker zone and the highest at the back. (b) Comparison of PMV flow field between 60% (left) and 100% (right) HVAC capacity. PMV is observed to be the highest in the back zone. For 60% fan capacity, it reaches over +2 whereas in 100% fan capacity it decreases slightly below +2.

The overall level of CO_2 is evenly distributed across the occupied zones and falls under excellent class according to HKIAQ guide [7]. According to ASHRAE, standard PMV values range should be -0.5 to +0.5 and Predicted Percentage Dissatisfied (PPD) should be less than 10% to comply with the optimal thermal comfort in indoor environments [16]. The simulation shows poor performance of comfort as all the PMV values are above 0.5 indicating hot feeling. Even after increasing fan power to 100%, the PMV does not improve. PPD for all zones are well above 10% even after full fan capacity showing poor performance of the air distribution system.

3.1.3. 50% occupancy scenario simulation results at front

Under both fan capacities, the center point receives considerably higher air flow than other occupied zones due to being closer to the supply duct. But the back zone remains a dead zone with no significant airflow [Figure 7(a)] as the hot air from the front is attracted to the back although no occupants are present. Even though the temperature decreases with the increase of fan capacity to 100%, the back zone continuously remains the warmest zone [Figure 7(b)]. The PMV value ranges from 0.8 at the front zone with cooler sensation to exceeding +2 at the back zone making occupants feel very hot [16]. The central zone stays at a range of -0.4 ~ 0.4. The PPD range complies with ASHRAE standard at the occupied zones except for the back zone, where it exceeds 20% which indicates that more than 40 occupants will be feeling discomfort. The CO_2 level stays in the acceptable range indicating excellent class under both fan capacities.

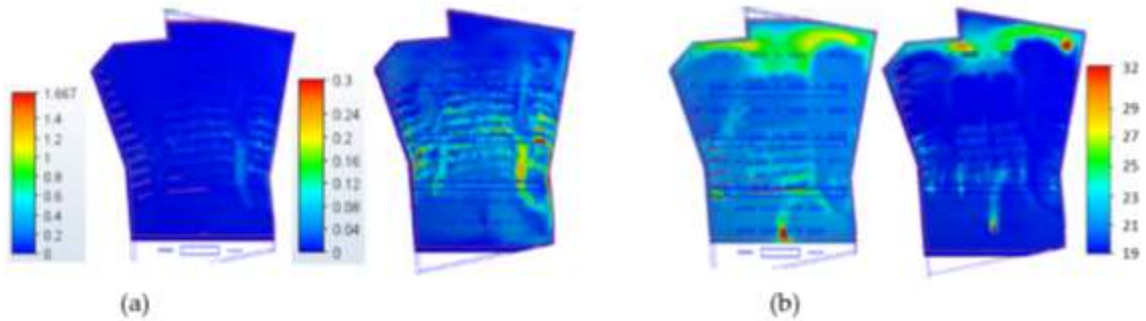


Figure 7. (a) Comparison of velocity flow field (m/s) with 60% (left) and 100% (right) fan capacity (b) Comparison of temperature flow field (m/s) with 60% (left) and 100% (right) fan capacity showing uneven distribution. The temperature at the back is seen to be higher than the front. The range for 60% HVAC capacity is from 20.6- 23.8°C and for 100% HVAC capacity is from 19°C - 21.1°C.

Similar simulation was performed with 50% occupancy at back indicating similar trend of temperature distribution with overheating at the back and over cooling at the front. Among the 3 scenarios of different occupants' locations, occupants sitting at the front work the best in the current HVAC design where temperature only a little below 20°C is observed at the middle rows and the speaker position in 100% HVAC capacity. In terms of temperature, 60% HVAC capacity works better than 100% HVAC capacity in this case.

In conclusion, despite increasing the fan capacity to 100%, the temperatures in three zones do not comply with the standard ranges of thermal comfort. Therefore, more optimized design is needed to achieve a comfortable indoor environment. The hot sensation at the back rows indicated design problem in the current HVAC system where temperature reaches 28°C which is 3°C higher than comfort level. The largest temperature difference under 60% operating capacity is 6°C between the back and the speaker position when all the occupants are sitting at back which is not acceptable within a room. As a result, higher cooling capacity and better ventilation are required. Overcooling at the front rows is another design problem where around 20°C is recorded at the front rows. The PMV of these rows are also below zero which is slightly cooler.

3.1.4. Proposed HVAC design simulation

At present, the supply duct runs below the center of the theatre which is why the maximum flow rate was calculated from Row 6. Traditionally adding an air diffuser at the back and air outlet at the front above the speaker position to dissipate the hot air without moving to the back would be proposed, however, these methods can come at a high cost. The proposed method is to relocate the main air supply duct at the back zone and calculate the maximum flow rate from the back most row (i.e. row 11 shown in [Figure 8]) and follow the pervious equations (1-4) earlier mentioned.

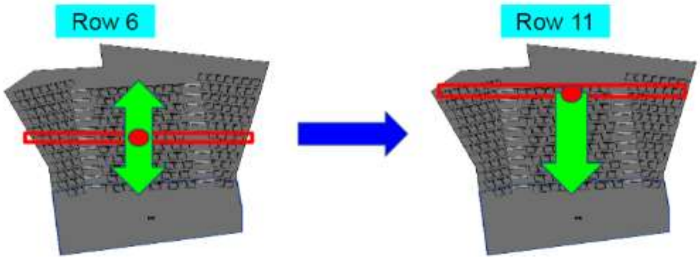


Figure 8. Relocation of supply duct from row 6 to row 11 for proposed layout. The supply duct is assumed to be supplying air flow at the back zone under 100% fan capacity to achieve the required air flow range at the back rows. The new design layout will reduce the cool air supply at the front while increasing the supply at the back rows.

$$\text{Geometric Factor} = \text{Total number of rows below row 11} - \frac{\text{Number of rows from Row x to Row 11}}{\text{Total number of rows below row 11}} \quad (6)$$

For example, row 10 = 10–1/10 = 9/10. According to the equation (2) y will be calculated to find the flow rate of a diffuser at row 11. Total supply flow rate would be the same as the original design. The velocity of each diffuser will be calculated by equation (4).

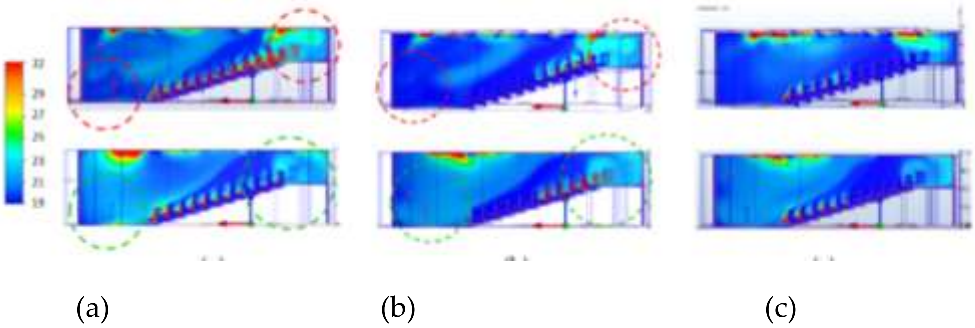


Figure 9. Temperature flow field (Deg C) - Comparison of current layout (top) and proposed layout (bottom) with 100% Fan Capacity (a) with 100% occupancy (b) 50% occupancy at back (row 7 – 12) (c) 50% occupancy at front (row 1 – 6). The range of temperature in the new design is 20°C - 23.5°C which is under the excellent class.

The problem of high temperature at the back and the uneven temperature distribution in the original design have been resolved by the new design. The temperature reduction between the original and new design varies from 3°C to 8°C. The largest reduction is 8°C located at the middle rows. Temperature has been substantially reduced to 23°C from 28°C at the back and the simulations show an even color in light blue throughout the theatre.

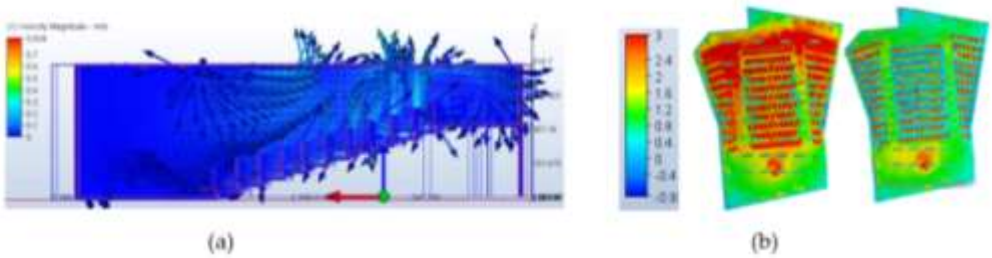


Figure 10. (a) Velocity flow field at 100% occupancy under proposed design. Airflow is seen moving upward due to displacement ventilation and jetting out from the relocated supply duct at the back before being exhausted via the outlets. Velocity sharply increases at the back zone indicating this zone is getting more air flow than before. (b) PMV Flow Field - Comparison of current layout (left) and proposed layout (right) with 100% occupancy under 100% fan capacity. PMV range decreases significantly close to 0 (neutral) in all scenarios with all the zones falling under -0.5 ~ +0.5.

The observed air flow is below 0.1 m/s showing even airflow distribution which is excellent class according to HKEPD standard. A decrease in velocity is also recorded at the front rows which may be caused by the alteration of the main supply air duct from row 6 to row 11. Range of CO_2 level is 424ppm- 450ppm for all scenarios. A reduction of CO_2 level of 11ppm is observed at the front rows while other rows remain unchanged. There is even comfort sensation throughout the theatre and most of the zones fall under 10% PPD which is excellent with 20% and 16% reductions recorded on left and right zones.

3.2. Results from Energy Analysis with eQuest

The difference between total annual electric consumption of 200 and 100 occupants is 25% and such large difference is the result of the substantial drop of chilled water consumption. Light is the only type of consumption that does not increase with the level of occupancy since the light schedule is always 1 ratio when the theatre is occupied in both occupancy scenario. Occupants may be scattered across the theatre for which whole room may need to be illuminated by turning on every light. The equipment schedule is set to vary with the level of occupancy. There are three overhead projectors, computers and an AV room which may also emit significant heat. The equipment load should increase with the level of occupancy since students may bring their own electronic equipment. Since equipment consumes energy and releases heat when being used, it is one of the major contributors of electricity consumption in the theatre.

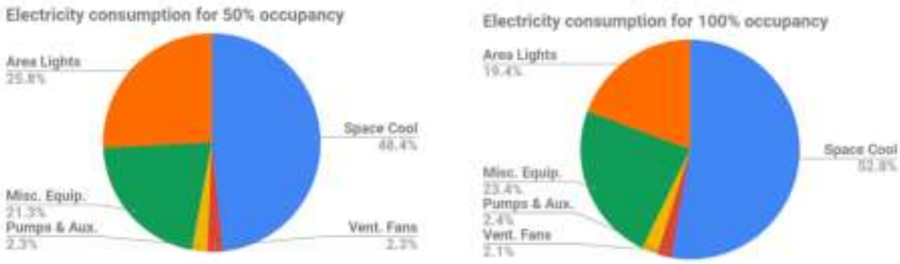


Figure 11. Comparison of Electricity consumption with 50% (left) and 100% occupancy (right) where space cooling is found to be the major contributing source for both 100 and 200 occupants respectively. Space cooling uses 45% more electricity when the occupancy increases to 100%, which means electricity consumption due to space cooling rises 0.9% for every 1% increase in occupancy. The second highest usage of energy is lighting which consumes 7090 kWh electricity in both scenarios. When the level of occupancy increases from 50% to 100%, electricity consumption from equipment rises to 46.5%, which is the highest among different types of energy use. It implies that 1% increase in occupancy causes the rise of electricity consumption of the equipment by 0.93%.

Since space heating is not required for buildings in Hong Kong and there are no refrigerators, water boiling machines and exterior lighting in the lecture theatre, the electric consumptions for refrigeration, space heating, heat pump supply, and exterior lighting are all 0. Electricity consumption of space cooling fluctuates significantly in a year. Since free air cooling is allowed in the winter, space cooling is off from November-March. In contrast, space cooling peaks in the hot season. In July, chilled water consumption, ventilation fans, and chilled water reach their highest consumption which result in higher electricity bill [Figure 12]. For the electricity bills, references from China Light and Power Co Ltd. (CLP), an electric company providing electricity in Hong Kong are used [17].

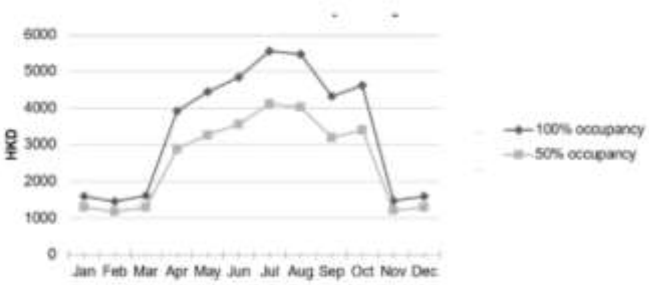


Figure 12. Electricity bill for a year with 100% and 50% occupancy showing the highest bill occurs in July as need for space cooling rises during hot weather. According to simulation, 18300kg CO2 is emitted annually due to electricity consumption and $40882 is the annual electricity fee for 100% occupancy in the theatre. For 50% occupancy, 13765kg CO2 is emitted and the annual electricity fee costs $30751.

4. Discussion

From the CFD simulations of the theatre, limitation of the existing HVAC system was discovered such as 8-h average time should have been given for measurement for validation as established by HKIAQ which was not possible due to time limitation. For both 50% and 100% occupancy scenario, average CO_2 level and air velocity are acceptable. Thermal comfort for 50% occupancy at the front is better than at the back in terms of floor temperature, the average temperature on nose level and the average vertical temperature difference. Therefore, it is recommended for the IAS theatre that all occupants sit at the front to experience a higher thermal comfort level if the occupancy level is equal to 50%. For both occupancies, the temperature at the back is undesirable due to poor ventilation performance. Reducing the level of occupancy at the front has an insignificant effect on the occupant thermal comfort level at the back. Therefore, modification to the layout of air diffusers is recommended as a remedy to the problem. The simulation result shows that the modification has made great improvement in terms of temperature, air movement, and CO_2 level. Large reductions in PPD are also observed between the original and the new design. The lecture theatre fulfils the requirement of an excellent class according to HKIAQ. However, theaters having different HVAC layouts may present different results. Hence, engineers and designers are recommended to utilize the CFD tool integrated with BIM which can apply different air-supply layouts and parameters including size, location, shape of the supply inlets, easily to evaluate the air distribution for the indoor environment, and compute PMV and PPD faster. This practice will ensure the decisions to redesign HVAC layouts according to the occupant behaviour and predict future patterns for both air quality and thermal without comprising both.

Similarly, results from the energy analysis for 50% and 100% occupancy by eQuest showed that space cooling has the highest electricity consumption in both scenarios. According to HK Climate Change Report 2015, the average carbon emission of a person in Hong Kong is 6000kg in a year [18]. Emission of 82,200kg per year from the theater will be equal to emission from 14 people in Hong Kong annually. There are more than thousands of buildings larger than the lecture theatre in Hong Kong and their total amount of CO_2 emission would be a lot. This proves that space cooling is responsible for a huge amount of CO_2 released. Therefore, improving the HVAC system is one of the best ways to reduce cooling energy consumption. Revising the layout without increasing the supply air flow rate is an energy-efficiency approach to improve the thermal comfort level and indoor air quality for the occupant. However, the supply air flow rate is automatically calculated by energy simulation models, the user has little freedom in changing the flow parameter making it is difficult to assess the amount of energy saved by lowering the supply flow rate using the energy simulation tools. Generally, modifying the layout without changing the supply flow rate has the potential of improving the thermal comfort level and indoor air flow without using additional energy to increase the supply air flow rate as well as concurring less cost. Therefore, the possibility of further modification to the layout should be studied in the future to implement such proposed method in other general use cases.

5. Conclusion

Public places such as theaters, offices, or other commercial buildings with unpredictable occupancy patterns, would not be comfortable without year-round control of the indoor environment. The thermal comfort of the building occupants with good indoor air quality are essential design factors for sustainable buildings and present special design challenges for professionals. BIM based simulation is considered as a crucial factor during the pre-design stage as well as in operational phase, to develop a sustainable building design. Contemporarily, researchers only carry out one type of simulation for each study. To study occupant thermal comfort level and the energy efficiency of the HVAC system, an integrated approach of BIM, CFD, and energy simulation was adopted in this project by analyzing different scenarios with Autodesk CFD and eQUEST to obtain a more comprehensive result within a reasonable amount of simulation time and show how the occupant can enjoy a high level of thermal comfort and indoor air quality after the modification.

Contributions: Quazi Samira Rahman, Tsz Hin Jeffrey Luk, Chun Fai Siu and Helen H. L. Kwok conceptualized the study. Helen H. L. Kwok worked on the methodology. Quazi Samira Rahman, Tsz Hin Jeffrey Luk, Chun Fai Siu and Helen H. L. Kwok worked on validation of the model. Quazi Samira Rahman, Tsz Hin Jeffrey Luk, Chun Fai Siu performed formal analysis. Jack C. P. Cheng provided the resources. Quazi Samira Rahman, Tsz Hin Jeffrey Luk and Chun Fai Siu worked on data curation and writing—original draft preparation. Helen H. L. Kwok worked on writing—review and editing. Quazi Samira Rahman, Tsz Hin Jeffrey Luk and Chun Fai Siu worked on visualization. Weiwei Chen and Jack C. P. Cheng worked on supervision and project administration.

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Appendix A

Table **A1.** Data collection from IAS Lecture Theatre at the HKUST

Data Type	Value
Height	14.2 ft
Area	2509.1 square ft (240.6 m^2)
Occupancy	200
Lights	26 Nos 52W round downlights
	32 Nos 56W double parabolic fluorescent luminaire
Diffusers	70 for displacement ventilations, 1 ceiling air inlet and 2 ceiling air outlets.
CO_2 Concentration	462 ppm (average)
Air Velocity	0.2 m/s (average)
Temperature	22.1^{o}C (average)
Relative Humidity	74% (average)
Total Air Flow Rate	2.83 m^3/s (60% fan capacity)
	4.73 m^3/s (100% fan capacity)

Figure A1. HVAC system arrangement in IAS lecture theatre

Table A2. Parameters used for Energy Analysis in eQuest

Data Type	Value
Climate Zone for Hong Kong [ASHRAE]	2A (Hot-Humid)
Operating schedule	9:00 AM – 5:30 PM, Monday-Friday, closed on weekends and holidays
Area per person	13ft^2
Sensible heat gain for seated occupant	245 Btu/h-person
Latent heat gain for seated occupant	155 Btu/h-person
Total light level	3144 W
Concrete wall thickness	0.656 ft
Operation of lights in closing hour	5%
Operation of equipment in closing hour	20%
Equipment Power Density	0.93W/ft^2
Required minimum OA flow/person	17 cfm
Required minimum supply rate ratio	0.3
Typical motor efficiency	0.86
Typical fan efficiency	0.6
Typical impeller efficiency	0.77
Typical chilled water design temperature	7^{o}C
Typical chilled water design temperature difference	5.5^{o}C

Table A3. Boundary Conditions for CFD Simulation

Data Type	Value
Air inlet scalar condition	0
CO_2 inlet scalar condition	1
Density for 400 ppm ambient air	1.2047g/cm3
Conductivity for 400 ppm ambient air	0.024W/m-K
Density for pure CO_2	1.773g/cm3
Conductivity for pure CO_2	0.0146W/m-K
Diffusivity coefficient for air and CO_2 mix	0.16 cm^2/s
Total Heat Generation for human	100W
Scalar boundary condition for nose level	40000
Iterations	1000
Thermal comfort factor	1 CLO value
Metabolic Rate (ASHARE standards) [2]	60

Relative humidity for field validation	69%
Relative humidity for scenario analysis	74%

Table A4. Results from Energy Analysis

Data Type	Value
Carbon Emission from theatre	0.50 kg CO_2/kWh
Total annual electric consumption (100 Occupants)	27530 kWh
Total annual electric consumption (200 Occupants)	36600 kWh
Total annual chilled water consumption (100 Occupants)	8068 kWh
Total annual chilled water consumption (200 Occupants)	127755 kWh

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