Feasibility of Using Air Curtains in Urban Buses

Reducing cooling requirements by minimising heat infiltration during boarding/alighting

Aditya Pathak[1,2], Raymond Khoo[1], Aybike Ongel[1]
1, TUM CREATE Ltd.,
1 CREATE Way, #10-02 CREATE Tower,
Singapore

Markus Lienkamp[1,2]
2, Institute of Automotive Technology
Technical University of Munich
Boltzmannstr. 15, 85748 Garching,
Germany

***Abstract*— In tropical countries, the HVAC system of electric vehicles consumes a significant amount of electric power reducing their driving range. This consumption is exacerbated for urban electric buses due to an additional thermal load caused by the infiltration of heat through the doors of the vehicle at bus stops with increased dwelling duration. Therefore, it is of paramount importance to develop methods that minimize thermal loads in the cabin and hence increase the efficiency of the vehicle. One method is to install air curtains that form a jet of air at the door to minimise the heat exchanged between the interior of the vehicle and the outside environment.**

This paper investigates the heat exchanged through the doors of an urban electric bus during dwelling, and the effects of air curtain installation on the cooling requirements of the vehicle. The influence of the air curtains on the heat infiltration was evaluated using Computational Fluid Dynamics (CFD) simulations. These results were used in a thermal cabin model to evaluate the reduction of cooling requirements with the air curtain. The use of air curtains showed a reduction in the cooling requirements by 1.6 kW for a 5.5 m vehicle and 0.5 kW for a 12 m vehicle.

Keywords—Air Curtains; Cooling Requirements; Air Conditioning; Thermal Modelling; Urban Vehicles

I. INTRODUCTION

At TUMCREATE the current research focuses on the development of a new efficient public transport concept for Singapore. The new concept also involves the development of a new modular electric vehicle concept to optimize fleet usage, operational efficiency, improve service quality as well as energy efficiency [1].

For a tropical country like Singapore, the air-conditioning system has a significant auxiliary load on the total energy consumption of the vehicle thereby requiring a larger battery pack. Battery is expected to be the major component cost for EVs in the next 10 years [2]. It is therefore essential to investigate methods of minimizing the HVAC (Heating, Ventilation & Air Conditioning) requirements of the vehicle during the concept development of the vehicle.

For passenger vehicles, the current method of sizing the HVAC system is based on the time taken for the cabin air to reach the desired temperature from a simulated soak test [3]. This method however may not be suitable for larger transit buses as the heat load due to passenger occupancy could be higher than the solar load simulated in the soak test. Also due to the larger internal air volume of buses, the soak temperatures are not as high compared to passenger vehicles. Furthermore the sizing would be influenced by the frequency of bus stops and the dwell duration.

In Singapore the typical interval to the next bus stop is approximately 400 m [4] in residential areas. This implies that buses stop approximately every 4 minutes where doors are open for the duration of the dwell time which allows conditioned air to escape. The HVAC system therefore requires to be suitably sized to maintain the desired cabin temperature between consequent bus stops.

One method of reducing the heat infiltration is the application of air curtains which separates the cabin and environment air zones with a jet of air reducing the amount of warm air entering into the cabin, and hence reducing the cooling requirements.

[5] studied and experimentally measured the heat flows between the cabin and the environment at various temperature differences with and without the use of air walls/air curtains. The analysis, however, did not take into account the sizing of the heating/cooling requirements of the vehicle thus the energy benefits of using the air curtain/air wall were not directly evident. In addition, the results of the experiments are not applicable to vehicles with different sizes and door configurations.

[6] and [7] presented an experimentally validated thermal cabin model that takes into account the various thermal interactions between the cabin and environment. However, the model is limited to passenger vehicles and vans that do not have heat infiltration due to opening and closing of doors during dwelling.

Air curtains and their effectiveness in stationary applications are evident [8],[9], however literature in automotive/transport usage is still limited. The sizing of the cooling requirements and the operation strategy of the urban bus vehicle would strongly influence the suitability of the application of the air curtain. However, currently there is no literature collectively evaluating the reduction of cooling requirements with the use of air curtains. This paper, therefore, proposes an approach to assess the feasibility of using air curtains in urban bus type vehicles of different sizes studying the operational and physical boundaries that influence the suitability of air curtains. Furthermore two different vehicle cases are studied with respect to the reduction of cooling requirements with the use of air curtains.

II. Proposed Approach & Methodology

The proposed method consists of a parametric CFD model to characterize the heat exchanged between the cabin and environment over time under different conditions. The obtained thermal characteristics from CFD then used as an input in the thermal cabin model to evaluate the cooling power requirements, including the resulting cabin temperature with and without the effect of air curtains and other operational aspects such as dwelling time and thermal loads due to solar radiation, ambient load, passenger metabolism and ventilation.

A. CFD Modelling

1) Without Air Curtains

A transient 3D CFD study was conducted in ANSYS FLUENT to approximate the heat gained in the cabin during the opening of the doors. The fluid motion was assumed to be governed by natural convection with a laminar flow as the Grashof number was calculated to be below the critical value. The Grashof number is a ratio of the buoyancy force to the viscous force and is used to categorize the nature of the flow; whether it is free or forced convection and laminar or turbulent in nature. The fluid flow is laminar within the range of $10^3 < Gr < 10^6$ [14].

The Boussinesq approximation was used to simulate the buoyancy caused by the temperature and density difference (between the cabin and environment fluid zones) and therefore to simulate the natural convection. To reduce the computational time, the influence of humidity was ignored and the fluid was assumed to be dry air at 24°C.

The geometry of the cabin was simplified to that of a cuboid having an adjacent fluid zone representing the environment. The vehicle length and door width were parameterized to evaluate the effect of vehicle length and door width on the heat exchanged with the environment. Four vehicle lengths were simulated from 4 m up to a maximum length of 18 m with different number of doors and door widths. Through the design of experiments the effect of the door to vehicle length ratio was evaluated. Table 1 shows the cases simulated using CFD.

The temperature for the exterior environment zone was set as 32°C and 24°C for the cabin interior. All walls were specified to have a no slip boundary condition and the door surface was set to interior type to allow the fluid to flow across the door.

Table I: shows the experimental design for the CFD simulations

Simulated Cases		
Cabin dimensions	***Door to vehicle length ratio***	***Door widths***
18m x 2.6m x 2.8m	0.07, 0.13	1x1.3m, 3x1.2m
10m x 2.6m x 2.8m	0.13, 0.25	1x1.3m, 2x1.25m
7m x 2.6m x 2.8m	0.186	1x1.3m
4m x 2.6m x 2.8m	0.325	1x1.3m

To simplify the CFD model, only the effect of heat exchanged between the cabin and environment was simulated. It was therefore assumed that there is no cooling of the cabin when the door of the vehicle is open. Additionally the effect of door positions was neglected and the influence of different ambient temperatures was not studied due to limited computational time and resources.

The transient simulation was run for a flow time of 60 s. A large time step can result in instability and inaccuracy of the numerical simulation. The time step was therefore adjusted such that the cell convective courant number was below a value of 40. The courant number is a measure of the stability and convergence of the numerical solution and should be within a range of 20-40 [13] to ensure a stable and accurate solution. Similarly a mesh independent study was conducted to find the mesh element sizes to ensure an accurate solution.

The measured output from the simulation was the volume-averaged temperature of the cabin interior at each time step. Using the average temperature rise of the cabin at each time step, the average heat gain was calculated using eqn. 1.

$$\dot{Q} = m * c_p * \frac{\Delta T}{time\ step} \quad (1)$$

2) CFD Simulation with Air Curtains

To simulate the influence of air curtains, the geometry of the cabin interior was modified to consist of an air curtain unit at the top of the door region. The geometry and technical specification of the air curtain was taken from a catalogue. The inlet nozzle was specified with an inlet velocity of 8 m/s as per the technical specification of the air curtain unit and the outflow boundary condition was selected for the outlet boundary as depicted in Fig 1. It was further assumed that the air curtain blows air vertically downwards at a temperature of 24 °C irrespective of the cabin temperature.

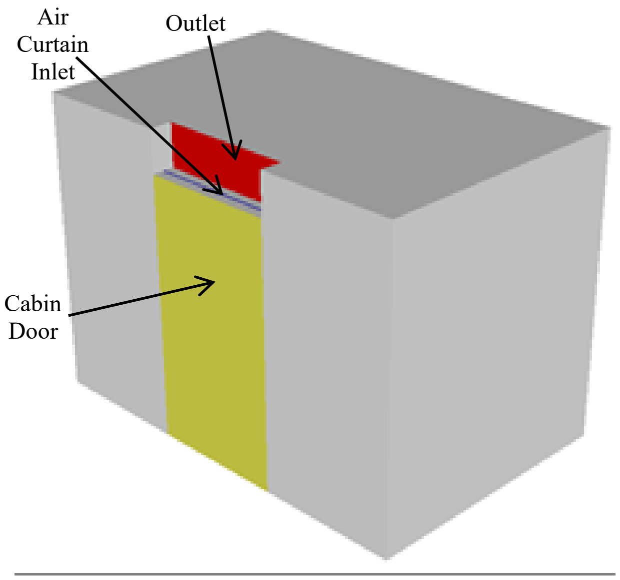

Fig 1: Segment of the cabin internal volume indicating the key boundary regions.

The flow with the air curtains was assumed to be turbulent as the Reynolds number was above the critical value of 2300, the k-ε model with full buoyancy effects was used. It was further assumed that the air curtain device is operational before the doors are open therefore the jet of air would be formed prior to the opening of doors. The simulation was hence divided into two stages: 1. obtaining a steady state solution for the formation of the air curtain jet (fig 2), 2. simulating the transient characteristics of heat flow across the open doors. The boundary condition of the cabin door is therefore switched from a wall to an interior (open) condition after the first step.

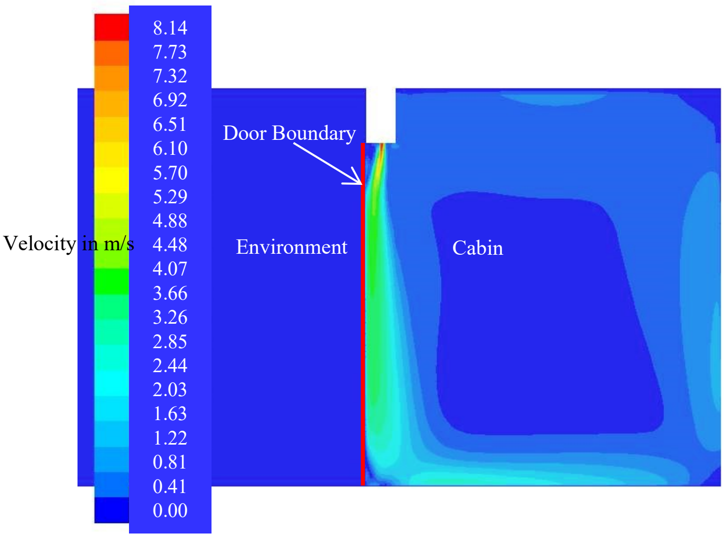

Fig 2: Velocity contours of the air curtain during closed doors.

The transient simulation for the second stage is computed for 60 s of flow time while monitoring the volume-averaged temperature rise of the cabin at each time step. The average heat gain in the cabin is calculated using equation 1.

B. Thermal Cabin Modelling

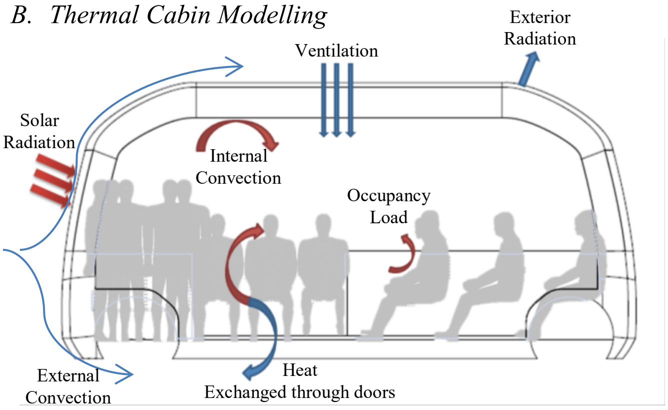

Fig 3: Thermal heat loads in the cabin

The thermal cabin model was implemented to quantify the total cooling requirements of the vehicle. As the analysis was carried out in the early concept phase, a simplified approach was taken to be able to simulate different vehicle parameters. A 1-D lumped element model was therefore implemented in MATLAB-Simulink which evaluates the mean temperature of a single air zone and the variation of the temperature due to the interaction with various heat loads such as internal and external convection with the cabin surfaces, solar radiation, ventilation, longwave radiation, sensible heat gain due to passengers and heat gained from opening and closing of doors. For electric vehicles, the heat transmitted from the battery pack and motors to the cabin is negligible [7] and was therefore not taken into account.

The cabin geometry was simplified to be of a cuboid shape and was therefore divided into 6 sides: 2 horizontal surfaces which represented the roof and floor and 4 vertical surfaces representing the sides, front and back of the vehicle body. Each surface has 2 additional convection connections: the inner facing sides having a convection connection to the interior air zone while the outer facing side having a convection connection to the ambient air.

Using the lumped element method, a network of the thermal interactions could be built. One benefit of using a resistance network is that each element can have a connection in series therefore heat conduction through different materials can be simulated. In the simulation model, the body surfaces consisted of an Aluminum layer connected to an insulation layer of Glass foam material while the glazing surfaces were assumed to be comprised of glass without any insulation.

As the radiation and thermal properties of the glazing and body materials are different, the side, front, and back surfaces were given an additional parallel connection to the interior air zone and environment. A variable Body-to-Glass area ratio was used to be able to vary and calculate the area representing glazing and body surfaces given the overall dimensions of the cabin. It was assumed that the side facing panels had a Body-to-Glass area ratio of 0.5 while the front and rear facing surfaces having a ratio of 0.2 and 0.8 respectively. A ratio of 1 would imply no presence of glazing surfaces.

The long wave radiation from the outer facing surfaces (T_s) of the vehicle was calculated using equation 2.

$$\dot{Q}_{rad} = \varepsilon_s \sigma A_s \left(T_s^4 - T_{sky}^4\right) \quad (2)$$

Where σ is the Stefan Boltzmann constant and ε is the emissivity of the surface. A_s is the surface area, T_s is the surface temperature and T_{sky} is the sky equivalent temperature calculated as per [12].

The incident direct and diffuse solar irradiance (I_s) for horizontal and vertical surfaces facing North, East, South and West directions was calculated as per [12] for clear sky conditions in Singapore from sunrise till sunset as depicted in Fig 4. As the side facing surfaces had a larger area of glazing compared to the front and rear facing surfaces, it was assumed that the vehicle is facing North (the side surfaces facing East and West) as this configuration would result in the highest solar radiation transmitted into the cabin during the duration of the day as shown in Fig 4.

The solar radiation incident on a surface is dependent on the material properties. For opaque materials the incident radiation is either absorbed by the surface or reflected while for transparent materials, part of the radiation is transmitted through surface. The solar radiation absorbed and transmitted by the body and glazing surfaces was added as source terms applied to each surface using the equations 3 and 4.

$$\dot{Q}_{sol} = \alpha_s * I_s * A_s \quad (3)$$

$$\dot{Q}_{sol} = \tau_s * I_s * A_s \quad (4)$$

Where α_s is the absorptivity of the surface and τ_s is the transmissivity of the surface and I_s is the solar irradiance incident at each surface. For simplicity it was assumed that all the transmitted solar radiation is absorbed by the interior thermal mass.

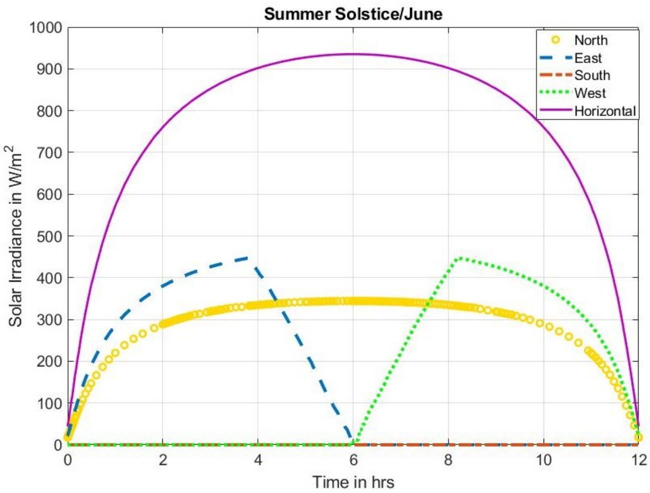

Fig 4: Direct solar irradiance on the horizontal and vertical surfaces facing different directions

Additional thermal source terms were added for the metabolic/occupant load, where each passenger was assumed to transmit 100 W to the interior air mass. The thermal load due to the heat infiltration (also referred to as heat exchanged) through the open doors calculated from the CFD results (Fig 6) was implemented as an intermittent thermal source term that repeated every 4 mins in the simulation. The signal was scaled based on the length and door width of the vehicle using the correlation found in Fig 5.

The air conditioning system in most buses is run on recirculation mode and therefore the cabin air is recirculated. Fresh air is however required to be introduced to maintain passenger comfort and safe levels of CO_2 concentrations within the cabin. This additional ventilation air needs to be cooled to provide a continuous supply of cold air to the cabin.

A recirculation ratio (r) was therefore specified to calculate the temperature and humidity ratio of the mixture of recirculated air and fresh air. It was assumed that the amount of recirculated air exhausted to the environment was equal to amount of fresh air introduced. The additional cooling requirement for fresh air ventilation was calculated using the psychometric chart to calculate the enthalpies and using equation 5.

$$\dot{Q}_{vent} = \dot{m}_{ven} * r * (e_o - e_i) \quad (5)$$

$$\dot{Q}_{cooling} = \dot{m}_{ven} * (e_o - e_i) + (m_a c_a) * \left(\frac{T_i - T_{comf}}{t_c}\right) \quad (6)$$

Where m_{vent} is the mass flow rate of the supply air, m_a is the mass of interior air, c_a is the specific heat of the interior air, T_i is the temperature of the interior air, T_{comf} is the target comfort temperature, t_c is the pull down constant, e_o and e_i is the enthalpy of air at the outlet and inlet of the evaporator. To calculate the enthalpy at the exit of the evaporator, the fluid temperature leaving the evaporator was assumed to be equal to the coil temperature and at 100 % relative humidity.

The pull down time constant was calculated using equation 7 and represents the time required for the system to exponentially reach 63 % of the target temperature.

$$t_c = \frac{t_p}{\ln |T_0 - T_{comf}|} \quad (7)$$

Where t_p is the pull down time, and T_0 is the initial temperature. The pull down time is the time required to reach the comfort temperature within 1°C [11].

The transfer of heat due to long wave radiation, the conduction and convection of heat through the vehicle body to the and the solar radiation absorbed by the exterior surfaces were summed together and referred to as the ambient load. The solar radiation transmitted through the windows and absorbed by the interior surfaces was summed and referred to as the solar load.

Two cases were simulated in the thermal cabin model both with and without the use of air curtains. The first case was the vehicle concept being developed by TUMCREATE with a preliminary length of 5.5 m with a door to vehicle length ratio of 0.325 and the second Case of current existing single decker buses of 12 m length with a door to vehicle length ratio of 0.2. The simulation was carried out using the parameters listed in table 2 to evaluate the effects of relative vehicle length and the use of air curtains on the change in cooling power requirements of the vehicle.

Table II: shows the parameters of the vehicle cabin simulated

Variables	Vehicle Parameters		
	Symbols	*Case 1*	*Case 2*
Cabin Dimensions	LxWxH	5.5m x 2.6m x 2.8m	12m x 2.6m x 2.8m
Door to vehicle Length Ratio		0.325	0.2
Ambient Temperature	°C	32°C	32°C
Ambient Relative Humidity		80%	80%
Initial Temperature	°C	24°C	24°C
Pull-down time	s	600	600
Location		Singapore	Singapore
Passengers		20	90
Dwell Duration	s	60s	60s
Frequency of Door Openings	s	240s	240s
Air Recirculation Ratio	r	80%	80%
Body-to-Glass area ratio	Sides Front Rear	0.5 0.2 0.8	0.5 0.2 0.8
Glazing Transmissivity	τ_s	0.4	0.4
Surface Absorptivity	α_s -body α_s -glass	0.8 0.4	0.8 0.4
Body Material		Aluminum	Aluminum
Glazing Thickness		3mm	3mm
Body thickness		3mm	3mm
Insulation Material		Glass Foam	Glass Foam
Insulation Thickness		30mm	30mm

III. Results

1) CFD Results of Heat Infiltration

The CFD results of the volume-averaged cabin temperature were numerically differentiated with respect to flow time to calculate the rate of change of cabin temperature. The rate of change of temperature in vehicles with higher door to vehicle length ratio (large doors and shorter vehicle lengths) is larger therefore higher heat is exchanged per unit volume compared to longer vehicles with smaller doors as shown in figure 5.

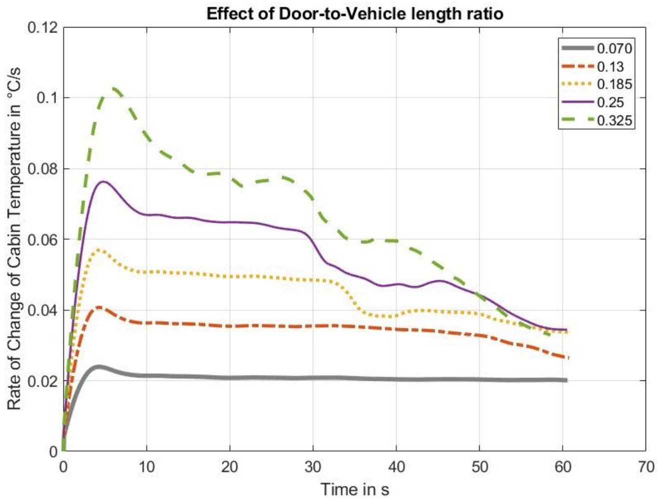

Fig 5: Effect of Door to vehicle length ratio of the rate of change of cabin temperature

For smaller door to vehicle length ratios, the rate of change of cabin temperature is observed to be near constant from 10 to 50 s as there is a larger proportion of colder air available. The temperature difference with the environment and therefore the rate of change of cabin temperature reduce gradually. For larger ratios, the rate of change of cabin temperature decreases as a result of the average cabin temperature increasing and the temperature difference with the environment decreasing.

Fig 6 shows the heat load due to infiltration of air through the doors, calculated using equation 1. For a 5.5m long vehicle, with a door to vehicle length ratio of 0.325, it can be seen that the heat exchanged when using air curtains shows a high peak of 5 kW for 2-3 s; which occurs due the cold cabin air being pushed out to the environment when the door opens. The overall heat energy exchanged however (represented by the area under the curve) was calculated to be 2.4 times lower with the use of air-curtains.

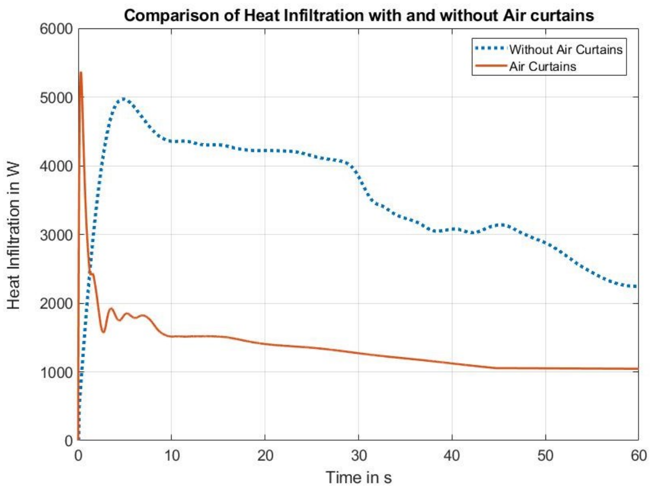

Fig 6: Heat infiltration in the cabin with and without the use of air curtains

2) *Thermal Cabin Modelling Results*

The heat loads of the vehicle were first calculated without the consideration of the heat infiltration through the open doors. Fig 7 shows the different contribution of the heat loads in the cabin. From the simulation it was deduced that the maximum total heat load in the vehicle is 8.2 kW. The cooling power required to cool the cabin down to the desired temperature within 600 s was calculated to be 9.4 kW.

The ambient load in Fig 7 is around 3 kW which reduces to 2 kW during the peak heat load. This occurs as the cabin temperature increases from 20 °C to 24 °C shown in Fig 9.

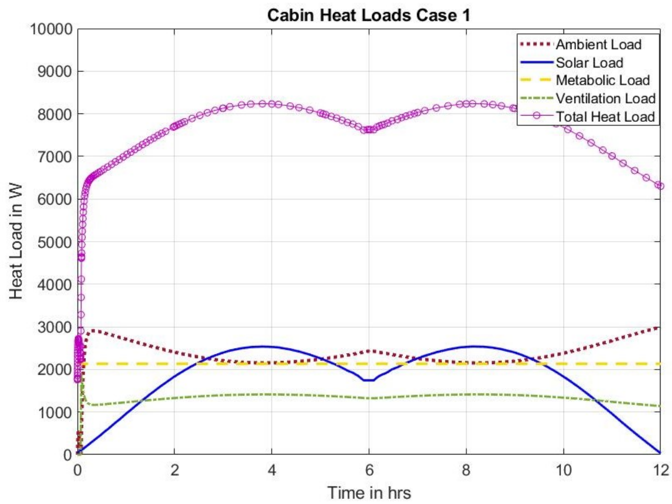

Fig 7: Cabin heat loads for Case 1

In the second Case of the longer vehicle, the total peak heat load in the cabin was 24 kW as shown in Fig 8. The cooling requirement was calculated to be 25 kW for the cabin temperature to reach 24 °C within 600 s. For Case 2 it can be seen that the largest contribution to the total heat load is from passenger occupancy compared to ambient load for Case 1.

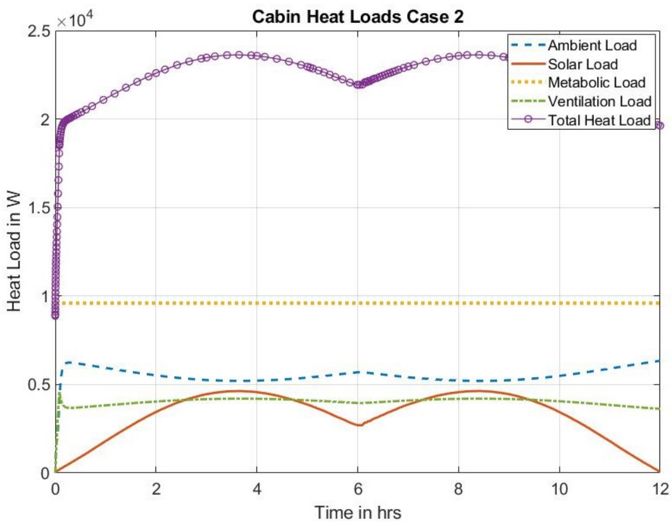

Fig 8: Cabin heat loads for Case 1

In Fig 9 it is observed that for Case 1, the temperature increases from the initial value of 24 °C to 31 °C until the cooling is switched on. The cabin temperature then decreases to 20 °C as a result of the cooling. When the heat load in the cabin is not at the maximum value of 8.2 kW, there is sufficient cooling power available to cool the cabin to 20 °C. Similarly for Case 2, the initial temperature increases from 24 °C to 35 °C and 21 °C when the cooling is switched on.

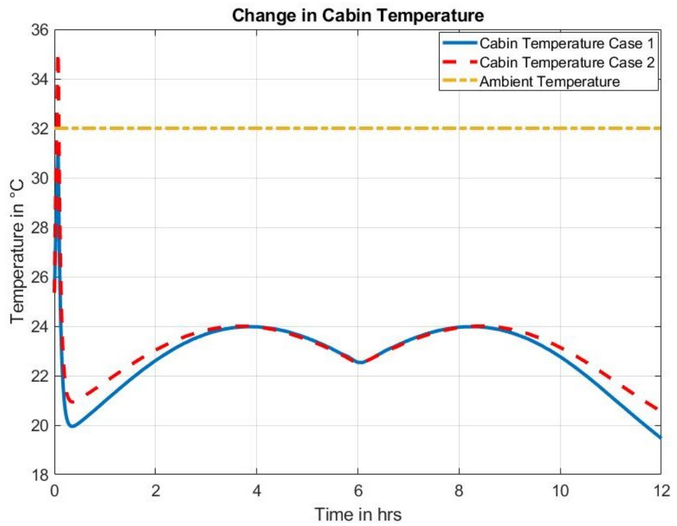

Fig 9: Variation of cabin temperature for Case 1 and Case 2 without the use of air curtains

Fig 10 shows the heat loads with the additional heat infiltration (heat exchanged) due to the frequent door opening and closing for Case 1. It can be seen that there is a spike of 6 kW of heat load every 240 s when the door opens. Consequently, the ambient load decreases when the doors are open and the ventilation load increases. This is because the cabin temperature increases therefore reducing the heat transmitted to the environment and increasing the recirculated air temperature and consequently the ventilation load.

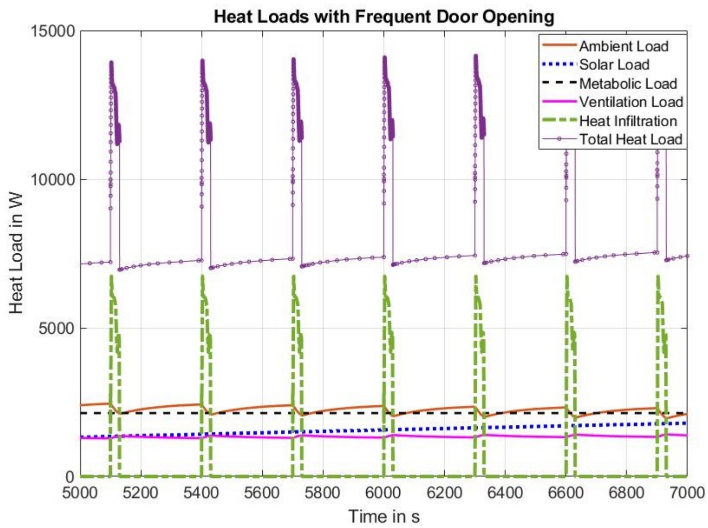

Fig 10: Cabin heat loads for Case 1 with intermittent opening of doors

For Case 2, a similar trend is observed as shown in Fig 11. The heat infiltration however, only comprises of a small fraction of the total heat load in comparison to Case 1 shown in Fig 10.

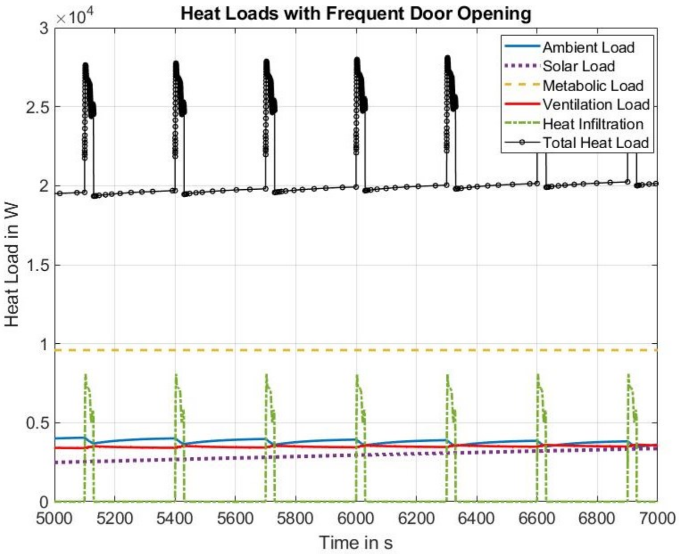

Fig 11: Cabin heat loads for Case 2 with intermittent opening of doors

Fig 12 shows the effects of the door opening and closing on the cabin temperature for both with and without the use of air curtains. After every 5 minutes when the door opens, the temperature spikes up by 1.3 °C when the air curtain is not in use. During the high thermal loads, it can be seen that the cabin temperature reaches 24 °C just before the door opens. The HVAC, however, cannot provide a continuous temperature of 24 °C required for passenger comfort. The cooling requirement therefore requires to be increased to be able to continuously maintain the desired temperature of 24 °C. With the use of air curtains, the temperature fluctuates by 0.5 °C. Additional cooling power is required as well to maintain the desired temperature of 24 °C at the peak heat loads.

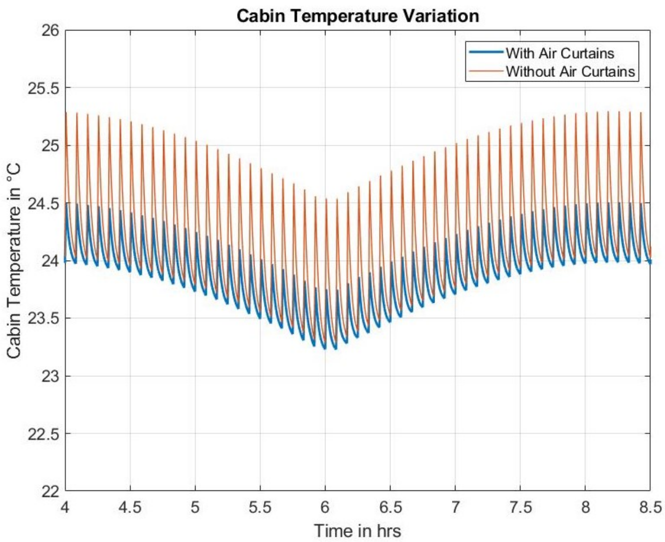

Fig 12: Cabin temperature for Case 1 with and without the use of air curtains

For Case 2, the cabin temperature fluctuates by 0.25 ºC with the use of the air curtains, and 0.55 °C without the air curtains as shown in Fig 13. As the cabin in Case 2 has a larger volume of air and a lower relative heat exchange load with the environment, the fluctuation of temperature is less compared to that of Case 1.

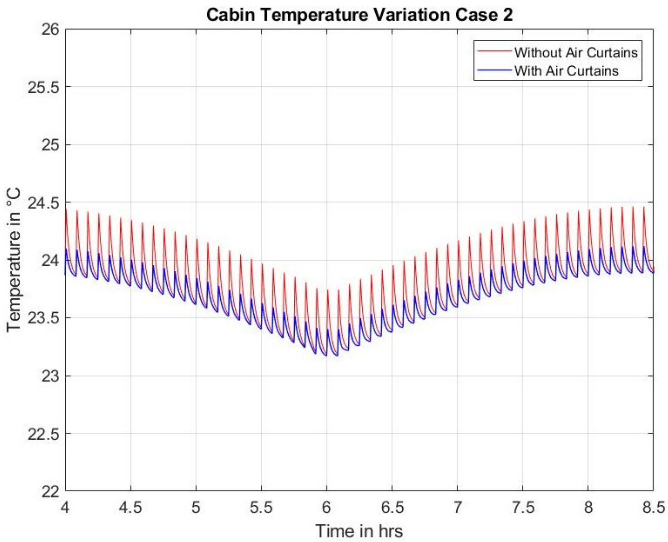

Fig 13: Cabin temperature for Case 2 with and without the use of air curtains

Due to the fluctuating heat loads and the resulting cabin temperatures, the cooling requirements also oscillate with the opening and closing of doors. To operate below the target temperature of 24 °C when the doors are open, the peak cooling requirements increases from 9.2 kW to 10.2 kW with the use of air curtains and to 11.8 kW without the use of air curtains; therefore reducing the cooling requirement by 1.6 kW as shown in Fig 14.

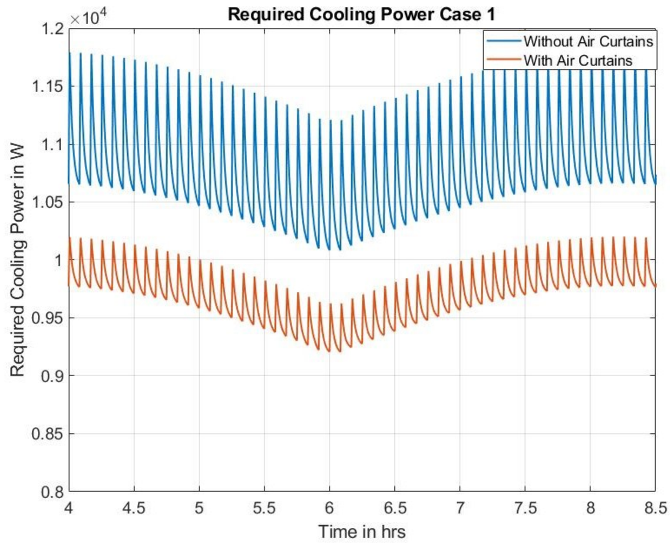

Fig 14: Cooling power required for Case 1 with and without use of air curtains

As seen in Fig 13, the existing cooling power of 25 kW is not sufficient to maintain the cabin temperature at 24 °C. The new cooling power requirement was thus calculated to be 26.1 kW when the air curtain is used and 26.6 kW without the use of air curtains. The use of air curtains thus decreases the cooling power requirement by 500 W as shown in Fig 15.

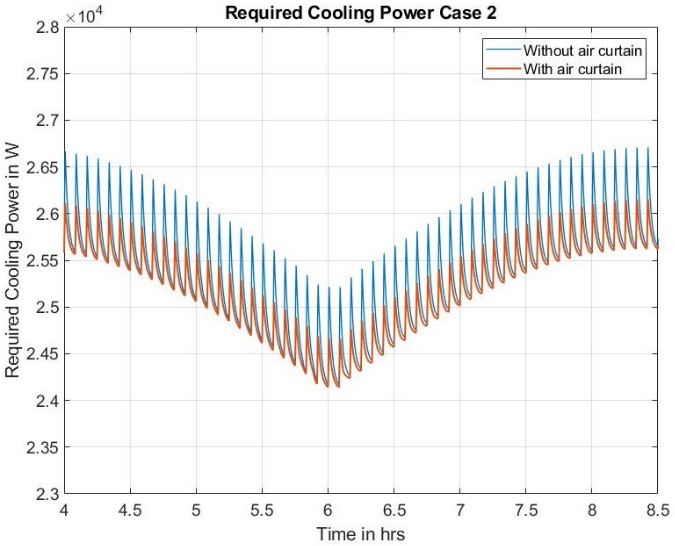

Fig 15: Cooling power required for Case 2 with and without use of air curtains

Fig 16 shows the breakdown of the different heat loads as a percentage of the heat energy flowing into the cabin over the duration of daytime (from sunrise to sunset). As the heat load due to door opening and closing is intermittent, its contribution to the overall energy entering the cabin is less compared to the other heat loads which occur continuously in the simulation. It can be seen that the heat infiltration comprises of 6 % of the heat energy applied on the cabin for Case 1 and 2 % for Case 2. For both cases the use of air curtains reduced the heat infiltration contribution by half.

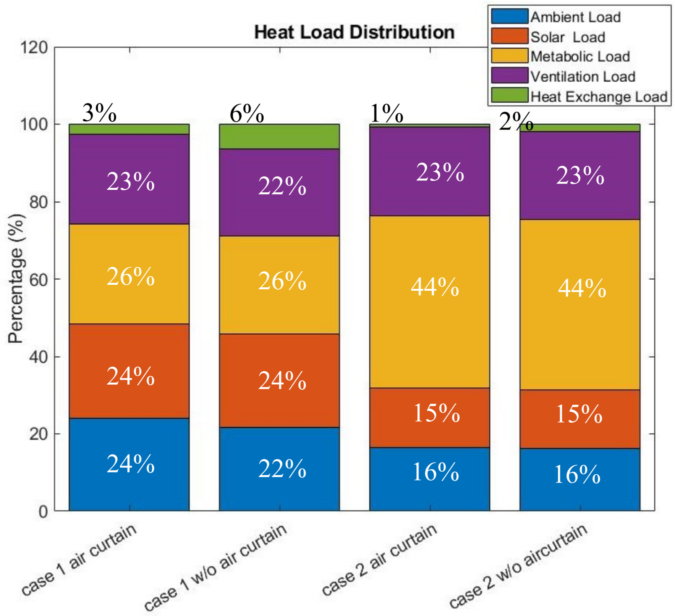

Fig 16: shows the contribution of each heat loads relative to the total heat energy interacting with the cabin

IV. Disussion

CFD Simulations of the infiltration of heat into the vehicle showed that cabins with a larger door to vehicle length ratio have a higher rate of change of cabin temperature. The influence of vehicle length (volume) and door width must therefore be taken into account when calculating the heat exchanged through open doors.

The use of air curtains to reduce the heat infiltration was shown to be an effective method. For the simulated configuration with 8 m/s inlet velocity, the air curtains reduced the heat exchanged by 2.4 times. The influence of the air curtain air velocity, temperature and nozzle angles has not been considered in this study. Optimization of these factors could lead to a higher effectiveness of the air curtains.

The results from the cabin thermal simulation indicated an increase in the cooling requirement for vehicles with frequent opening and closing of doors. Without the use of air curtains, the cabin temperatures fluctuate by 1.3 °C for Case 1 and 0.55 °C for Case 2. Consequently, the cooling requirement increases by 2.6 kW for Case 1 and 1.6 kW for Case 2. The use of air curtains showed a reduction in the cooling requirements by 1.6 kW for Case 1 and 0.5 kW for Case 2.

It was calculated that the heat exchanged contributes to 6 % of the total energy entering the cabin for Case 1 (smaller vehicle with a wide door) compared to 2 % for Case 2 (long vehicle with narrow doors). The use of air curtains in smaller vehicles could therefore be more effective in reducing the cooling requirements than for long vehicles with less doors/shorter door widths.

The benefit of using the air curtains is observed only in reducing the peak cooling requirements of the vehicle. If the peak heat loads only seldom occur then the use of air curtains would not be advantageous as the cooling system would have sufficient cooling power to achieve comfortable cabin temperatures despite the additional heat load due to opening and closing of doors. In such a case, the use of air curtains may result in an increase of energy consumption due to the added fan power and additional weight. The implementation of the air curtains would only be beneficial in reducing the cooling requirements if the sizing is done to meet the nominal cooling requirements.

The operational strategy of the public transport system would also have a large influence on the feasibility of the air curtains. In this paper the dwell time simulated was 60 s. For shorter dwell times, the heat load due to door opening and closing would comprise of a smaller proportion of the total heating loads. The reduction of cooling power requirement of the vehicle using air curtains would subsequently also be smaller.

The effect of different ambient temperatures was not investigated as the focus of this paper was limited to the investigation of average weather conditions in Singapore. For countries with warmer climates, the reduction of cooling power using air curtains could be more profound as the heat infiltration through the doors could be up to 16 kW [5].

From a perspective of thermal comfort, the use of air curtains could be beneficial in the reduction of 'hot spots' in the cabin. The method used in this paper employs a lumped element model and therefore the entire cabin is modelled as having a single value of temperature. In reality there would be large temperature gradients within the cabin as the area near the doors where the heat is exchanged would be warmer compared to the regions near the cooling vents. The use of air curtains therefore could reduce the temperature gradients due opening and closing of doors at bus stops.

Selection of the HVAC components would have a large influence on the feasibility of the air curtains as the components used in the refrigerant cycle can have a wide operating range and therefore the ability to operate at a higher cooling capacity. Component selection therefore requires to be further investigated to quantify the reduction in energy consumption and improvement of vehicle range due to the reduction of the cooling requirement with the use of air curtains.

V. Conclusion and Future Work

This paper presented a method to investigate the effects of heat infiltration due to frequent door opening and closing on the cooling requirements of vehicles. The use of air curtains was further investigated to assess the reduction in the cooling requirements.

It was found that the use of air curtains resulted in the decrease of cooling requirement for urban vehicles. The reduction was more significant for smaller vehicles with wide doors compared to longer vehicles with narrow doors.

As the results showed that the heat infiltration constitutes only a small percentage of the total cooling requirement, the effects of air curtains at higher ambient temperatures requires to be further investigated as the heat infiltration through the door is larger at higher ambient temperatures.

The use of air curtains can reduce the temperature gradient within the cabin when the doors are open. It would therefore be of future interest to study the increase in thermal comfort levels of passengers with the use of air curtains.

The method proposed in this paper was to evaluate the suitability of the air curtains at an early phase, a more detailed approach will be implemented in the future by including the effects of humidity in the cabin model and coupling the simulation with a refrigerant cycle simulation to calculate the energy consumption and to accurately capture the transient behavior of the evaporator and the resulting cabin temperatures.

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