Dynamic modelling and performance prediction of a recovery fresh air heat pump for a generic bus

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Abstract—**Air circulation inside the buses' cabin seems to negatively affect the spread of contagious diseases, such as the COVID-19 virus and raises valid health concerns over using public transportations. Employing all-fresh air and avoiding to recirculate it could help with lowering the exposure time and the virus density in buses; however, it makes the heating more challenging, especially in Electric buses. Here a Baseline and a proposed Recovery Heat Pump (BHP and RHP, respectively) systems in a generic single decker bus were modeled to investigate their dynamic performance and the cabin's conditions using 100% fresh air. Simulink and Simscape toolbox from MATLAB (R2020a) were used to build up the real-time model by integrating an HP system with a cabin sub-model. The cabin is modeled using a moisture air network and is coupled with the HP to exchange heat with the refrigerant through the condenser. For both cases, 100% fresh air flows through the condenser into the cabin. In BHP the evaporator is exposed to 100% cold fresh air, while in RHP the warm air from the cabin passes through the evaporator before being vented outside. Both cases were studied for different ventilation modes in low and medium occupancy levels. Results indicate that RHP shows superior performance compared with BHP. Under the studied operational conditions, RHP reduced the power requirement, warm-up time, and operation time by 36%-6% (at most- at least), 57%-7%, and 39%-13%, respectively.**

*Keywords—**Heat pump; Waste heat recovery; Bus; Dynamic modelling; COVID-19; contagious diseases.***

I. Introduction

Nowadays, using public transportation has become more challenging due to the growing concerns on their potential to spread the infections from person to person while a large number of people travel in a limited space, especially during contagious diseases outbreaks such as the global coronavirus (COVID-19) pandemic [1, 2]. On top of that, the required power for heating in electric vehicles, where the energy storage capacity is limited, should be kept at a minimum to prevent driving range loss [3, 4]. It is therefore important to adopt efficient system architectures to meet the safety requirements for the people on-board and reduce energy consumption. In recent years Heat Pumps (HPs) have attracted considerable attention for their high coefficient of performance and low energy consumption, making them a reasonable substitution for electric heaters in electric buses, especially for mild-cold climates. In HPs, it is a common practice to employ recirculated air with a proper mixing ratio with fresh air to reduce heating loads. Zhang et al. [5] showed that utilizing recirculated air can reduce the heating demand by 46.4–62.1% when the ambient temperature is -5°C and -20°C, for a passenger electric vehicle. Pan et al. [6] also evaluated that utilizing recirculated air can reduce heating energy by 33-57%. However, since we are now living in unprecedented times of the COVID-19 pandemic, the transport operators should take proper measures to lower the possibility of the virus spreading and put the safety of the drivers and passengers first. One way could be running HP and air conditioning only in all-fresh-air mode and avoid recirculating air inside the cabin.

Moreover, a rising number of numerical and experimental studies have been conducted on Heating, Ventilation, and Air Conditioning (HVAC), for passenger cars [7, 8, 9] and to less extend on buses [10, 11, 12]. Among them, numerical studies have mainly investigated the thermal conditions of the cabin through detailed CFD [8, 13] and lumped-parameter models [14, 15, 16, 17], or the performance of HVAC units. For example, a thermal model of the cabin in passenger vehicles was developed by Marcos et al. [14] and later Torregrosa-Jaime et al. [15] studied the cabin of a mini-bus by a lumped-parameter transient thermal model. On top of that, some researchers have studied the cabin coupled with HVAC unit. For instance, Schaut and Sawodny [16] adopted a lumped-parameter cabin sub-model coupled with a resistance heater (heating mode)/evaporator (cooling mode), representing the HVAC unit while the heating/cooling power and airflow rate were adjusted by a blower. Recently, Afrasiabian et al. [17] proposed a dynamic model to analyze the impact of the occupancy level in a generic bus. They proposed a coupled model of a Vapor

Compression Cycle (VCC) for a refrigeration system and the cabin's thermal network to predict the dynamic performance of an AC system and real-time cabin's condition.

In this paper, we built up a real-time model by coupling two sub-models (HP and Cabin) on the Simscape platform within the Simulink environment. Two heating systems where modeled where the first one was a Baseline Heat Pump (BHP) and 100% fresh air was employed on its both heat exchangers, namely; evaporator and condenser. The second one was a Recovery Heat Pump (RHP) where the warm air from the cabin was vented outside after passing through its evaporator. Then we used these models to predict and compare the cabin's condition (temperature and humidity), warm-up time, operation time, and required power by the heating unit under different volumetric flow rates of the adopted fan and blower.

II. METHODOLOGY

In this study, we used Simscape toolbox and Simulink from MATLAB (R2020a) to build up the BHP and RHP models for a generic single decker bus. One VCC sub-model representing the HP system as demonstrated in Fig. 1, and one cabin sub-model of a generic bus as shown in Fig. 2, were integrated to form a coupled master-model. A two-phase network was used to establish R134a flow as the phase-changing refrigerant in a conventional VCC that is composed of an evaporator, accumulator, compressor, condenser, and an expansion valve. In the evaporator the low pressure-temperature refrigerant evaporates and absorbs heat from i) 100% cold fresh air in BHP or ii) a mixture of fresh and cabin's air before being vented out in RHP. Then the heat is released from the high pressure-temperature refrigerant to the cabin through the condenser. Both the evaporator and condenser here are generic cross-flow heat exchangers, sized according to the compressor's capacity and the operational conditions. The compressor is represented by a controlled mass flow source that increases the pressure and temperature of the refrigerant. The refrigerant mass flow rate reads as:

$$\dot{m}_r = \eta_v \, \rho_{r,suc} \, \dot{V}_{comp} \tag{1}$$

where $\dot{V}_{comp}$ is the compressor's displacement volume, η_v is the volumetric efficiency, and ρ_r is the refrigerant density at the suction line. Compressor work (w_{comp}) is a function of isentropic work and efficiency (η_{is}) as:

$$w_{comp} = w_{comp,is}/\eta_{is} \tag{2}$$

In BHP, the condenser's blower and the evaporator's fan are fed with all-fresh-air. In RHP 100% fresh air is sent into the cabin through the condenser while the warm air from cabin passes through the evaporator before venting outside, as shown in Fig. 3. The power demand (w) by the fan and blower varies as the airflow rate (V) changes as:

$$w_{V2} = \left(\frac{V_2}{V_1}\right)^3 w_{V1} \tag{3}$$

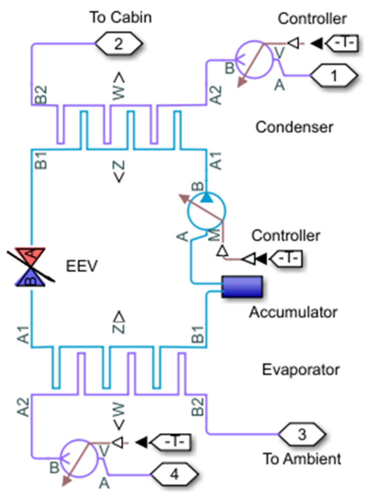

Fig. 1: VCC's architecture showing the refrigerant lines in blue and the moist air lines in purple.

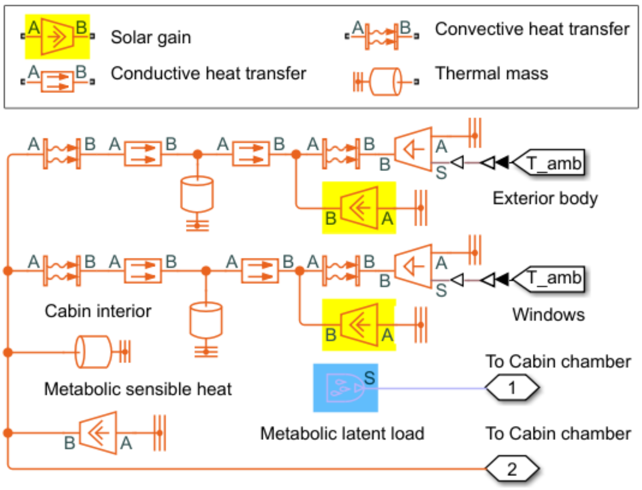

Fig. 2: The sensible and latent thermal network of the cabin

The passenger cabin is modeled as a constant volume chamber with allocated vent to the outside environment. Both sensible and latent thermal loads are connected to the cabin which is coupled with the HP sub-model via the condenser. As Fig. 2 shows the sensible load includes the solar radiation ($\dot{Q}_{sol}$), passenger heat generation ($\dot{Q}_{met}$), and heat transfer through convection/conduction between inside the cabin and environment ($\dot{Q}_{amb}$). The latent load is taken into account as the air humidification by the passengers. The energy equation for the cabin reads as:

$$m_a c_{p,a} \frac{dT_{cab}}{dt} = -\dot{Q}_m + \dot{Q}_{sol} + \dot{Q}_{amb} + \dot{Q}_{met} + \dot{Q}_{vent} + \dot{Q}_{HP} \tag{4}$$

Here $\dot{Q}_{HP}$ stands for the heat delivered to the cabin by the HP, $\dot{Q}_{vent}$ is the thermal loads due to the ventilation, and $\dot{Q}_m$ represents the heat stored/released due to thermal masses. Here $\dot{Q}_{sol}$ is the sum of the solar loads absorbed by the interior ($\dot{Q}_{sol,int}$) and exterior ($\dot{Q}_{sol,ext}$) surfaces, governed by the following equations:

$$\dot{Q}_{sol} = \dot{Q}_{sol,ext} + \dot{Q}_{sol,int} \tag{5}$$
$$\dot{Q}_{sol,ext} = \sum_{i=top,side,win} I'_{av}\, \tau_i\, S_i$$
$$\dot{Q}_{sol,int} = I'_{av}\, \alpha_{win}\, \tau_{int}\, S_{win}$$

where I'_{av} is the average irradiance that each surface area (S) receives, α_{win} is the transmitivity coefficient of windows, and τ_i is the absorptivity of the i^{th} surface of the bus. Moreover, the metabolic sensible and latent loads are defined as [18, 19]:

$$\dot{Q}_{met,l} = N\ h_l \times 1.44 \cdot 10^{-5} (kg.s^{-1}) \tag{6}$$
$$\dot{Q}_{met,s} = N \times 70 (Watt)$$

where h_l stands for the latent heat of condensation of water vapor and N is the number of people on-board. For the current study, the operating conditions and adopted parameters of the model are listed in Table 1.

TABLE 1: MODEL'S PARAMETERS

$m_{b,ext+win}$	8500	$kg,$	T_{amb}	5	°C,
$m_{b,int}$	800		T_{init}	5	
$Volum_{cab}$	75	$m^3,$	$\dot{V}_{comp}$	17.1	$m^3.hr^{-1}$
I'_{av}	100	$W.m^{-2},$	w_f & w_{bl}	350	W
$k_{conv,ext}$	30	$W.m^{-2}K^{-1},$	$(at\ 1200\ m^3.hr^{-1})$		
$k_{conv,int}$	5		η_{is}	0.75	
RH_{init}	25	%,	η_v	0.9	

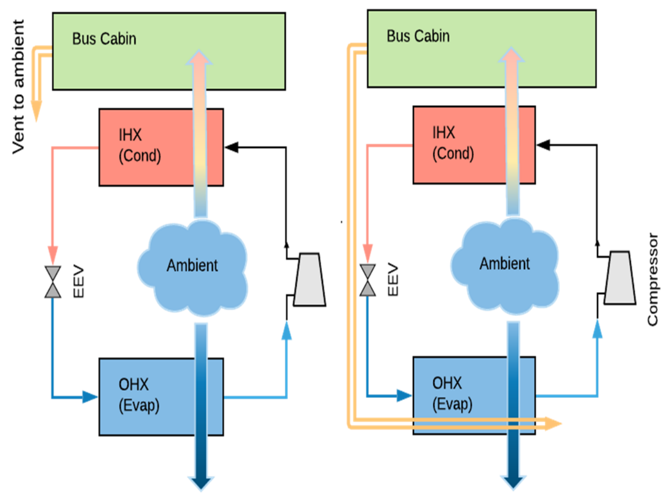

Fig. 3: Schematic of the BHP (Left) and RHP (Right)

III. RESULTS AND DISCUSSION

In order to make all-fresh-air heat pumps with no circulation inside the cabin more efficient, the ventilation rate should be kept at a minimum to reduce the heat waste from the vent and/or extract as much heat as possible in RHP mode. In this study, we investigated three different airflow rates of the blower and fan, namely: 600, 900, and 1200 ($m^3.hr^{-1}$). Two different levels of occupancy were considered, 5 and 25 people on-board. We narrowed down the combinations of the airflow rates into six modes where $V_{Amb} \geq V_{Cab}$. The set-point temperature is 17 $°C$ and the compressor, fan, and blower are switched ON when the temperature of the cabin drops below 16 $°C$ and they stop operating when it reaches 18 $°C$. Moreover, the expansion valve was regulated according to the following logic:

- The subcooling degree at the exit of the condenser should be 10 $\pm$ 0.5 $°C$.
- The refrigerant should leave the evaporator at a minimum superheating degree.
- Since the HP should operate in a wide range of conditions, it is sometimes difficult to meet both the above criteria. In this case, the expansion valve should be adjusted to meet the second criterion (minimum superheating).

Fig. 4 demonstrates the needed time for the cabin to reach from the initial temperature ($T_{init} = 5\ °C$) to 15 $°C$. As this figure indicates, the higher the fresh airflow rate into the cabin, the higher the warm-up time in BHP, especially for N=5 with a low level of metabolic heat generation. In fact, for higher V_{Cab} more heat will be wasted from the cabin without being recovered that explains the jump at the warm-up time for 900×900 compared to 1200×600 and 900×600 ($m^3.hr^{-1}$) modes. Employing RHP decreases both the warm-up time and sensitivity of the HP to the airflow rates. As this figure depicts, the number of people on-board has an important effect on the warm-up time which is more significant in BHP. Indeed by using RHP the warm-up time reduces at most (& at least) about 57% (& 9%) compared with BHP, when N=5 and the airflow rates are 1200×1200 (& 1200×600) ($m^3.hr^{-1}$). As it is evident, the advantage of RHP over BHP is narrowed for lower V_{Cab} where less heat is wasted as well as for the higher level of occupancy (N=25) where more heat is generated by the people inside the cabin. Furthermore in BHP, reducing the air flow rate through the evaporator raises the warm-up time, especially for N=5 where HP needs to extract the heat from 100% cold fresh air, which is lower for lower V_{Amb}.

Fig. 5 shows the percentage of the operation time over six hours for both BHP and RHP and different airflow rates. As expected the HP should work longer for the lower occupancy level. Here again, recovering the heat from the cabin's vent in RHP reduces the operation time remarkably. In N=25, RHP reduces the operation time by at most 39% and at least 15% that would reduce the maintenance costs and the power requirement for RHP, as demonstrated in Fig. 6. This figure shows the average power demand over six hours that reads as

$$w_{av} = \int_t \left(w_{comp} + w_f + w_{bl}\right) dt\, /\Delta t \tag{7}$$

where w_f and w_{bl} stand for the power requirements of the fan and blower, respectively. As this figure shows, the lower the cabin's airflow rate, the lower the power demand. RHP reduces the average power by at most 36% and at least 8% for N=25. Likewise, RHP decreases the power requirement for N=5 by at most 17% and at least 6%.

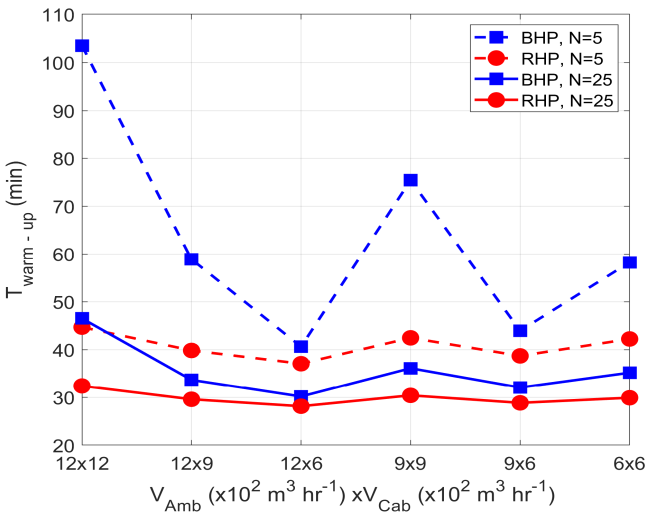

Fig. 4: Warm-up time for BHP (in blue) and RHP (in red)

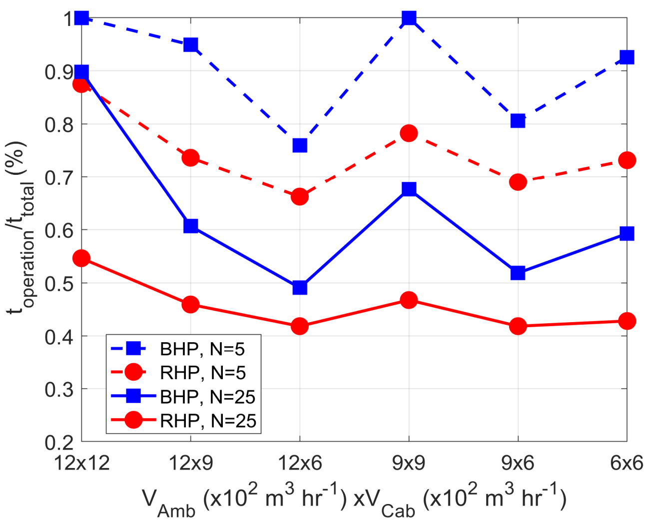

Fig. 5: Operation time ratio (over six hours) of the BHP and RHP

Power demand and safety-wise, three modes of $V_{Amb\times Cab}(m^3.hr^{-1} \times m^3.hr^{-1})$ are chosen here as potential optimum modes, namely: A) 1200×1200 & B) 1200×900 for N=25, and C) 600×600 for N=5. The respective fresh air rates per person are calculated based on the operation time and listed in Table 2. Moreover, Fig. 7 demonstrates the dynamic temperature and Relative Humidity (RH) inside the cabin over six hours under the respective selected modes. It is worth noting that, as the cabin is fed with all-fresh-air with no air circulation, there would be no humidity accumulation inside the cabin. As this figure implies in both N=25 and 5, RHP shows superior performance compared to BHP in terms of the warm-up and operation times, as previously discussed; however, BHP always provides better ventilation in terms of $[l.s^{-1}]$ per person.

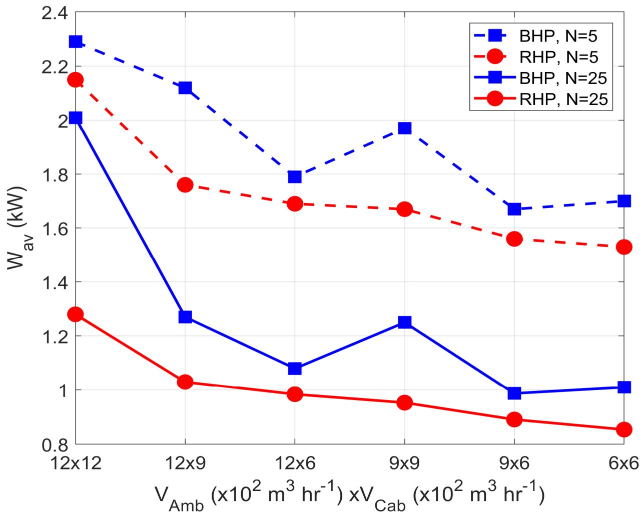

Fig. 6: Average power requirement of BHP and RHP

TABLE 2: AIRFLOW RATE $[l.s^{-1}]$ PER PERSON, FOR N=25 (A AND B) AND N=5 (C)

Mode	A	B	C
BHP	12.0	6.1	30.9
RHP	7.3	4.6	24.4

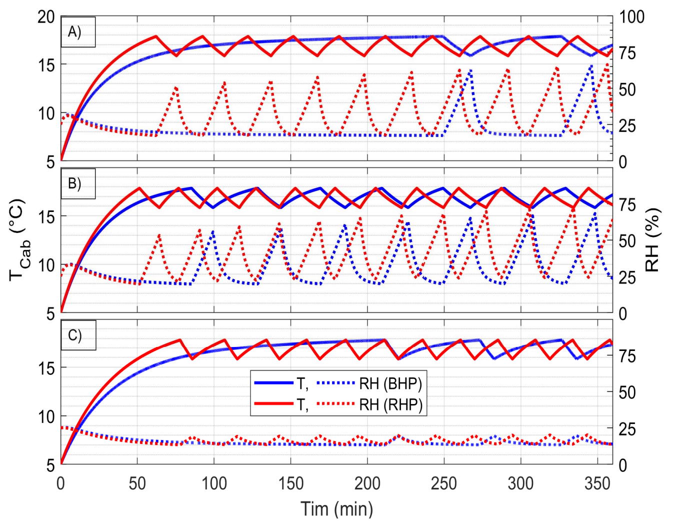

Fig. 7: The real-time cabin's temperature and relative humidity (dotted lines) for [N, $V_{Amb\times Cab}(m^3.hr^{-1})$]: A) [25, 1200$X$1200], B) [25, 900$X$900], C) [5, 600$X$600]

IV. CONCLUSION

Due to the health concerns over the negative effects of air circulation inside a bus on the spread of contagious diseases, such as COVID-19 virus, we investigated how employing a recovery system (RHP) can improve the performance of a heat pump with no air circulation inside a bus cabin. In this study, we developed a dynamic model of one heat pump coupled with the cabin with two configurations, BHP and RHP. Two different occupancy levels were studied for 5 and 25 people on-board. As results showed and for the whole range of the implemented airflow rates:

- For both cases and at a fixed V_{Amb}, the lower V_{Cab} the lower the warm-up time, power demand, and operation time. BHP is more sensitive to V_{Cab} compared with RHP.
- Employing RHP can reduce the warm-up time by at most 57% (&30%) and at least 9% (&7%) for N=5 (& N=25).
- Using RHP can reduce the operation time by at most 23% (&39%) and at least 13% (&15%) for N=5 (& N=25) which results in lowering the maintenance cost and extending the life of the compressor, fan, and blower.
- Employing RHP can reduce the power requirement by at most 17% (&36%) and at least 6% (&8%) for N=5 (& N=25).

As the results showed and in case of no air circulation policy in buses, the use of RHP is quite promising compared to BHP since it provides more available energy to be absorbed by the refrigerant in the evaporator and reduces the power demand, warm-up and operation times remarkably.

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