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A numerical evaluation of a novel recovery fresh air heat pump concept for a generic electric bus

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ABSTRACT

Since the outbreak of the worldwide COVID-19 pandemic, public transportation networks have faced unprecedented challenges and have looked for practical solutions to address the rising safety concerns. It is deemed that in confined spaces, operating heating units (and cooling) in non-re-circulation mode (i.e., all-fresh air mode) could reduce the airborne transmission of this infectious disease, by reducing the density of the pathogen and exposure time. However, this will expectedly increase the energy demand and reduce the driving range of electric buses. To tackle both the airborne transmission and energy efficiency issues, in this paper a novel recovery heat pump concept, operating in all-fresh air mode, was proposed. The novelty of this concept lies in its potential to be applied to already manufactured/in-service heat pump units as it does not require any additional components or need for redesigning the heating systems. In this concept, the cabin exhaust air is directed to pass through the evaporator of the heat pump system to recover part of the waste heat from the cabin and to improve the efficiency of the system. In this paper, a 0D/1D coupled model of a generic single-deck cabin and a heat pump system was developed in the Simulink environment of MATLAB (R2020b) software. The model was run in two different modes, namely the all-fresh air (as a baseline and a recovery heat pump concepts), and the air re-circulation mode (as a conventional heat pump concept with a 50% re-circulation ratio). The performance of these concepts was investigated to evaluate how an all-fresh air policy could affect the performance of the system, as well as the energy-saving potential of the proposed recovery concept. The performance of the system was studied under different ambient temperatures of −5 °C, 0 °C, and 5 °C, and for low and moderate occupancy levels. Results show that implementing the all-fresh air policy in the recovery and baseline concepts significantly improved the ventilation rate per person by at least 102% and at most 125%, compared to the air-re-circulating heat pump. Moreover, adopting the recovery concept reduced the power demand by at least 8% and at most 11%, compared to the baseline all-fresh air heat pump, for the selected fan and blower flow rates. The presented results in this paper along with the applicability of this concept to in-service mobile heat pumps could make it a feasible, practical, and quick trade-off solution to help the bus operators to protect people and improve the energy efficiency of their service.

ARTICLE INFO

Keywords:
Heat pump
Waste heat recovery
Electric bus
COVID-19
Contagious diseases

1. Introduction

Since the emergence of COVID-19 and the pandemic it has created, mass transportation including buses have faced unprecedented challenges in providing people with safe, reliable, and functional services. The main challenge here is linked with the valid concerns of the spread of COVID-19 when a large number of people travel in a limited space [1,2]. After the early ambiguous period of facing the new threat, now scientists have a better understanding of its fatality and transmission potentials, especially when people are in close vicinity and/or closed spaces. In fact, COVID-19 and other respiratory pathogens are believed to be spread mainly through contact with contaminated surfaces and by infectious droplets ($d_p > 10 \mu m$), and plausibly aerosols ($d_p \leq 10 \mu m$). So mask-wearing and social distancing (between 1 and 2 m) are strongly advised to reduce the transmission risks. Even though the airborne transmission is still a matter of discussion, many authors have identified it as a plausible transmission mechanism in closed spaces [3-6]. Droplets and aerosols are generated by the infected person via breathing, talking, sneezing, and coughing. Droplets are heavier particles that deposit after some time due to their weight while aerosols can be dragged by the air...
Appropriate ventilation rate should be adopted to dilute the concentration of the pathogens and avoid air re-circulation or increase the fresh airflow in closed spaces such as public transport systems, meaning low fresh air rates into the passenger vehicles [20-22]. Several researchers have suggested that ventilation is essential to deal with airborne transmission in closed spaces and an appropriate ventilation rate should be adopted to dilute the concentration of the pathogens [7-9]. If the ventilation inside closed spaces such as public transport systems is poor, meaning low fresh air rates into the space and high air re-circulation, the pathogens can build up and travel through the area [10-14].

On the other hand, in electric vehicles with limited energy storage capacity, heating and air conditioning of the cabin could be the main auxiliary power consumer, especially in extreme ambient conditions, that could reduce the driving range by more than 50% [15-17]. In recent years, there has been growing interest in the application of Heat Pumps (HPs) in the automotive industry due to the electrification of powertrain and the loss of waste heat from internal combustion engines. In this regard, HPs’ higher Coefficient of Performance (COP) and lower energy consumption, compared to electric heaters, make them a feasible alternative for heating in electric buses. To minimize heating loads when using HPs, it is common practice to use re-circulated air with a proper mixing ratio with fresh air. Zhang et al. [18] found that for a passenger electric car with the maximum return-air condition, the heating demand reduced by 46.4-62.1% (when the ambient temperature was –5 °C and –20 °C, respectively) compared to the all-fresh air condition. Pan et al. [19] evaluated that in an electric vehicle, utilizing re-circulated air could reduce heating energy in an HP system by 33-57% and increase the driving range by 11-30%.

To date, several numerical and experimental studies have investigated the Heating, Ventilation, and Air Conditioning (HVAC) of passenger vehicles [20-22]; however, there is much less research conducted specifically on buses [23-25]. Numerical studies have focused on the cabin’s thermal conditions, as well as the performance of the HVAC systems, mainly through detailed CFD [21,26] or lumped-parameter models [22,27-31]. Mezrhab and Bouzidi [22] developed a numerical nodal model based on the finite difference method to evaluate the cabin’s thermal conditions of a passenger car in summertime. They described the cabin as a network of solid and fluid nodes representing the compartment body and the air inside it, respectively. They investigated the impacts of solar radiation and the radiative features of the vehicle on the temperature inside the cabin. Marcos et al. [27] proposed a simplified thermal model to calculate the thermal load for the air conditioning of a passenger car without including any models for the HVAC system. Torregrosa-Jaimé et al. [28] investigated the cabin of a minibus using a lumped-parameter transient thermal model. Recently O’Boyle et al. [32] suggested a lumped-parameter model estimating the thermal load of a single-deck bus, without modeling the HVAC unit, to help with the creation of control strategies to minimize energy consumption. In the same vein, several researchers have looked into the coupled models of the cabin and HVAC systems. Schaut and Sawodny [29] used a lumped-parameter cabin sub-model with a resistance heater (heating mode)/evaporator (cooling mode) as the HVAC unit, with a blower to control the heating/cooling power and airflow rates. Afrasiabian et al. [30] introduced a dynamic model by coupling an AC system with a cabin sub-model. They used their model to predict the effects of the occupancy level on the dynamic performance of the AC system and the real-time thermal conditions of the cabin. Later Afrasiabian et al. [31] adopted a coupled model of a multi-unit AC system and the cabin of a generic bus to evaluate different control strategies on the power demand and discomfort level inside the cabin.

Moreover, even though the concept of utilizing waste heat in electric vehicles is not new, most studies and research have focused on recovering the waste heat from battery and electric motor. It is usually achieved by coupling the cooling circuits of the battery pack and electric motor with the heat pump system by employing an additional heat exchanger [25,33]. Yet, very limited studies have proposed to recover
the heat from cabin exhaust air by employing specific heat exchangers known as Heat (or Energy) Recovery Ventilator (HRV) in electric vehicles. A search of the literature revealed that Denkenberger et al. [34] studied the prototype of a microchannel heat exchanger to be employed in vehicles. On top of that, Bellocchi et al. [35] studied the employment of an air-to-air heat exchanger in which the fresh outside air could exchange both latent and sensible heat with air coming from the cabin through a separating membrane. However, in both papers [34,35], the authors suggested recovering heat from the cabin exhaust air by using a specialized heat exchanger.

In this study, we introduce a recovery heat pump concept that would address both the passengers’ safety and energy demand concerns in a single-deck electric bus. The importance and originality of this study is that it suggests recovering waste heat from the cabin exhaust air without employing any additional heat exchangers. This could significantly improve the performance of the heat pump when an all-fresh-air policy is in place (COVID-mode). The novel idea consists of forcing the warm air in the cabin to pass through the evaporator (on the cold side) and consequently increasing the evaporator temperature and the available heat to be absorbed by the refrigerant. To investigate the performance enhancement of the new recovery system, we developed a model by coupling a heat pump and a cabin of a generic bus sub-models. Three heating concepts were modeled, namely: a Conventional Heat Pump (CHP), a Baseline Heat Pump (BHP), and a Recovery Heat Pump (RHP).

In the CHP, the air is mixed with 50% of fresh air and re-circulated inside the cabin to reduce the energy consumption of the HP unit. While in both the BHP and RHP, the all-fresh-air policy was implemented to eliminate air re-circulation inside the cabin and lower the pathogen’s airborne transmission risks. In the BHP, the evaporator was solely exposed to the outside cold air while in the RHP, the warm cabin air was mixed with the outside cold air and then flowed through the evaporator before being vented outside. In this paper, the system operation, performance, power demand, and the fresh air provided per person in the CHP, BHP, and RHP concepts were investigated. The findings make an important contribution to the field of public transport during pandemics, when air re-circulation should be avoided, by demonstrating the RHP’s potential both to improve the ventilation rate and reduce the energy consumption compared to the CHP and BHP concepts, respectively.

2. Methodology

Three different models were developed for the CHP, RHP, and BHP concepts in the Simulink and MATLAB environment (2020b). The Thermal, Moist Air, and Two-Phase physical domains from Simscape toolbox were used to build up two sub-models for each case, namely a heating unit and the cabin. The former is based on a vapor compression cycle for a heat pump and the latter is formed by a network of thermal elements to describe the principle heat loss/gain/storage within the cabin or through the bus body. These two sub-models are coupled through the evaporators and condensers, while they also exchange heat with the ambient environment.

2.1. HP model

The heating unit sub-model is composed of the main components required for a vapor compression cycle namely; an evaporator (external heat exchanger), accumulator (ACC), Compressor (Comp), condenser (internal heat exchanger), Electric Expansion Valve (EEV), fans, blowers, and sensors, as shown in Fig. 1. In this study, R134a flows through these components as the refrigerant exchanging heat with moist air through the condenser and evaporator. The refrigerant absorbs heat at the evaporator after the outside cold air is drawn through it by pulling fans. The superheated vapor passes through an accumulator installed at the exit of the evaporator to make sure solely the superheated vapor goes to the compressor and to prevent any droplets from deteriorating the compressor’s operation. The low-pressure and cold superheated refrigerant becomes high-pressure/temperature after compression. At the condenser, heat is exchanged between the refrigerant and the fresh (or mixed in the CHP) air moved by the blowers. The subcooled vapor passes through the expansion valve to enter the evaporator again as a low-pressure/temperature two-phase refrigerant and to complete the cycle. Fig. 1 shows that in the CHP case, fresh air from outside is mixed with the re-circulated air and is blown into the cabin while the evaporator is exposed to 100% outside cold air. This is a common practice to reduce the waste heat through vents as well as energy consumption. As Fig. 1 illustrates, in the BHP model, both the evaporator and condenser are fed with 100% fresh air in order to deal with pandemic policies. This will expectedly increase the energy demand from the heating unit but is particularly crucial when the risk of contagious diseases is high, and health concerns are the first priority and requirement to operate any public transportation system. While both the BHP and RHP benefit from all-fresh-air into the cabin, the proposed RHP is distinctive by recovering waste heat at the evaporator, as seen in Fig. 1. This architecture is noteworthy as it does not need major modifications such as redesigning the system or adding new components, thus can be applied to the systems that are already in service with minor modifications.

In all three cases, the heat pump components are identical and only the system architectures are different. The compressor is the key component that determines the capacity of the system by regulating the refrigerant mass flow rate according to Eq. 1:

$$m_c = \eta_v \rho_{v, \text{sys}} \nu V_{\text{comp}}$$

Here, $\eta_v$ stands for the volumetric efficiency, $\rho_{v, \text{sys}}$ (kgm$^{-3}$) is the refrigerant density at the compressor’s suction line, $\nu$ is the frequency (Hz), and $V_{\text{comp}}$ is the compressor’s displacement volume (m$^3$ rev$^{-1}$). In this study, a variable speed compressor was modeled using a controlled mass flow rate source where the vapor is compressed through an isentropic

![Fig. 1. Schematic of the CHP (Left), BHP (Center), and RHP (Right) concepts. The dotted-line boxes show the heat pump units.](image-url)
have the following parameters; \( B_m = 60 \, (\text{W m}^{-2}) \), \( B_h = 1.7 \, (\text{m}) \), and \( B_k = 70 \, (\text{kg}) \). In Eq.3, the main loads entail the heat transfer with ambient through the convective and conductive mechanisms \( \dot{Q}_{amb} \), heat generated by the people on board \( \dot{Q}_{met} \), and heat vented outside \( \dot{Q}_{vent} \). In this equation, \( \dot{Q}_{HP} \) stands for the heat delivered from condenser into the cabin, and the effects of thermal masses to store the heat are represented by \( \dot{Q}_{store} \). In this study and for the nominal operational conditions of the HP systems, the solar radiation was assumed negligible in the winter time. The distinctive thermal conductivity coefficients of the body and windows are assumed uniform, as well as the internal and external convective heat transfer coefficients, as listed in Table 1.

### 2.3. Model verification

To verify the numerical model a series of simulations was carried out for the BHP case where \( T_{amb} = -5 \, ^{\circ}\text{C} \), and \( N = 3\) (low occupancy level). The results were compared with a refrigeration toolset called CoolPack toolkit (V1.5). This toolkit includes EESCoolTools (as an Engineering Equation Solver tool in the refrigeration area), Refrigeration Utility, and Dynamic analysis tools that are suitable for designing, sizing, and analyzing refrigeration systems. Here the instantaneous subcooling and superheating degrees were obtained from the Simulink dynamic model, and imported as input into the CoolPack toolkit. As illustrated in Fig. 3 with the above-mentioned assumptions, the current model can accurately predict the instantaneous power requirement of the HP system to run with the specifications listed in Table 1, as well as the amount of heat delivered into the cabin. Moreover, the predictions of the refrigerant mass flow rate of such a system are in good agreement in both models. For the required power, delivered heat, and the refrigerant mass flow rate, predicted by Simulink and CoolPack toolkit, the maximum error is approximately 2%.

### 3. Results and discussion

In order to evaluate the performance of the CHP, BHP, and RHP concepts, it is essential to establish common comparison criteria with the same operational and functional conditions. Due to the dynamic nature of this problem, in which both the HPs’ performance and the cabin conditions affect each other in a reciprocal way, it is important to identify the operating point of the individual components based on the whole HP systems’ operation. The compressor speed, the fan and blower flow rates, and the opening of the EEV all should be adjusted to minimize the power demand of the HP systems and to deliver the required heat into the cabin while maintaining the same thermodynamic parameters, such as the superheating and subcooling degrees, within the same ranges. A system operation map was created through conducting a series of numerical simulations, for each case, to find the optimum value of the compressor RPM (Revolutions Per Minute) and the EEV opening. Section 3.1 describes the assumptions and constraints under which the optimum figures were obtained. Likewise and as described in Section 3.2, a set of simulations were carried out to show how different modes of fans and blowers affect the power demand and recovery potential of the RHP concept, compared to the BHP and CHP cases.

#### 3.1. Operating point of the compressor and EEV

For given blower and fan airflow rates, the system performance parameters including the COP and the heating capacity vary by changing the compressor speed and EEV opening. This is mainly due to the changes in the refrigerant mass flow rate, superheating, and subcooling degrees. Fig. 4 shows a system operation map created for the BHP case with \( T_{amb} = 0 \, ^{\circ}\text{C} \) and low occupancy level. For all the presented results hereafter, a similar contour map was created to determine the optimum operating point for the compressor RPM and the EEV opening where the required power of the HP was minimum, and the following constrains

### Table 1

| Model’s parameters.                                      | External Heat Exchanger (Evaporator) | Internal Heat Exchanger (Condenser) |
|---------------------------------------------------------|-------------------------------------|------------------------------------|
| Tubes Length                                            | 1.5 (m)                              | 1.15 (m)                           |
| Number of tubes                                         | 60 (2 passes)                        | 16 (2 passes)                      |
| Total fin surface area                                  | 40 (6 m²)                            | 8.2 (m²)                           |
| Tube bank grid arrangement                              | Staggered                            | Tube bank grid arrangement         |
| \( R_H \)                                               | 25 (%)                               | 67.5 (m²)                          |
| \( T_{r} \)                                             | 7 (°C)                               | 111 (c m² K⁻¹)                     |
| \( k_{con, body} \) [32]                                | 0.35 (W m⁻¹ K⁻¹)                     | 480 (W)                            |
| \( k_{con, amb} \)                                      | 0.75 (W m⁻¹ K⁻¹)                     | 580 (W)                            |
| \( \eta_{con, ext} \)                                   | 25 (W m⁻² K⁻¹)                       | 0.65                               |
| \( \eta_{amb, ext} \)                                   | 5 (W m⁻² K⁻¹)                        | 0.9                               |
| \( \delta_{amb} \)                                      | 0.05 (m)                             | 1 (m)                              |
| \( \delta_{con} \)                                      | 0.005 (m)                            | 1 (m)                              |
were satisfied:

- The subcooling degree at the exit of the condenser should be $5^\circ C < T_{\text{sub}} < 10^\circ C$.
- The refrigerant should leave the evaporator at a minimum superheating degree ($T_{\text{sup}} < 5^\circ C$).
- Required warm-up time is fixed and equals $30^\prime \pm 30^\prime$.

To create the system operation map, different compressor RPM and EEV opening were inserted into the model as input and both the system power demand and warm-up time were predicted. In this map, the horizontal and vertical axis show the normalized EEV opening and compressor RPM, respectively. For the simulated range of the EEV opening and RPM, the colored contour illustrates the normalized power demand, and their respective warm-up time (minutes) is shown with dashed contour lines. Due to the considered constraints, the accepted warm-up time is within the red dash lines. The warm-up time is the needed time for the cabin to reach the set-point temperature and the abovementioned normalized parameters are defined as:

$$RPM_n = \frac{RPM}{RPM_{\text{max}}}$$

$$A_{\text{EEV},n} = \frac{A_{\text{EEV}}}{A_{\text{EEV},\text{max}}}$$

$$W_n = \frac{W}{W_{\text{max}}}$$

Here the average power demand takes into account both the compressor, fans, and blowers power over the total running time of the bus ($t_{\text{tot}}$), and reads as:

$$W_{\text{av}} = \int (W_{\text{comp}} + W_f + W_b) \, dt / t_{\text{tot}}$$

As Fig. 4 illustrates, for the BHP concept at low occupancy level and when $T_{\text{amb}} = 0^\circ C$, the optimum values of the compressor RPM and the EEV opening are determined based on the required power and the warm-up time which are those within the dotted ellipse.

### 3.2. Blower/Fan airflow rate

In practice, waste heat from the cabin could be controlled by varying both the total airflow rate and the mixing ratio of fresh and re-circulated air into the cabin. The lower this ratio, the lower the energy waste due to ventilation. However, this reduces the air quality inside the cabin and over time increases the CO2 concentration. In the RHP and BHP concepts, where the all-fresh air policy is applied, and in the absence of air re-circulation, the concerns over the air quality inside the cabin are reduced. Therefore, the airflow rates into the cabin could be reduced to a bare minimum to decrease waste heat. However, the airflow rates through the condenser and evaporator should be high enough to provide the required subcooling and superheating degrees on the refrigerant side. On top of that, for the RHP architecture the ratio between the blower and fan flow rates is expected to be an important parameter as it should affect the potential of the evaporator to recover the heat from the warm air before being vented.

In this study, three different modes for the blowers that are 900, 207, and 223.
1350, and 1800 (m$^3$·hr$^{-1}$) along with a fan flow rate of 3600 (m$^3$·hr$^{-1}$) were considered. Three different combinations of fan and blower speeds were studied, for two levels of low ($N = 3$) and medium occupancy ($N = 16$). To reduce the power demand of heating, the set point temperature was determined based on a sliding scale. Recently, Afrasiabian et al. [38] studied the effects of implementing sliding set-point temperature on the comfort level inside the cabin and the energy consumption, in different ambient temperatures. They showed that when $T_{amb} = -5 \degree$C, the power demand would reduce by about 61% for $T_{set} = T_{amb} = 12 \degree$C comparing to the case where $T_{set} - T_{amb} = 22 \degree$C. They also showed that reduced set-point temperatures would still provide people with acceptable comfort conditions as long as their normal outdoor clothing level was not reduced when people onboard. Therefore, in the current study three ambient temperatures, $-5 \degree$C, $0 \degree$C, and $+5 \degree$C, were investigated and the set-point temperature was set as $T_{set} = T_{amb} + 12 \degree$C. In all cases, the initial temperature of the cabin is assumed $T_i = T_{amb} + 5 \degree$C.

Fig. 5 shows the normalized power demand for three different combinations of the blower and fan airflow rates when $T_{amb} = 0 \degree$C. As expected, the CHP case is the most efficient system in terms of energy consumption, and the BHP is the most energy-intensive system. As this figure demonstrates, the RHP case could successfully reduce the energy consumption when an all-fresh air policy is implemented in a bus. As anticipated for all the cases, and due to the lower metabolic heat gain, the required power for the low occupancy level is higher compared to the medium level. For $T_{amb} = 0 \degree$C using a BHP system increases power demand by at least 20% and at most 51%. While adopting a RHP demands 15–30% more power compared to lower blower flow rate.

It could be attributed to the higher airflow rates being vented outside thus increasing waste heat from the cabin. However, different concepts will have different sensitivity. For example, for low occupancy level and when $V_b = 1800$ (m$^3$·hr$^{-1}$) the CHP, RHP, and BHP require 6, 14, 22% more power compared to lower blower flow rate ($V_b = 900$ (m$^3$·hr$^{-1}$)), respectively. Furthermore, Fig. 5 demonstrates that the RHP could operate as a moderate option between the CHP and BHP, where its potential to reduce the power demand improves as the $V_b/V_f$ ratio increases. The Potential Energy Saving (PES) by the RHP system is defined as:

$$PES(\%) = \frac{W_{n,BHP} - W_{n,RHP}}{W_{n,CHP}} \times 100 \quad (8)$$

As implied by Fig. 5, by increasing $V_b/V_f$ from 0.25 to 0.5 the PES parameter increased from 4 to 21% at medium occupancy level. These figures are respectively 5 and 18%, at the low occupancy level. Hereafter and for the rest of this study, the operating blower and fan rates are considered 1350 and 3600(m$^3$·hr$^{-1}$). At the selected flow rates, for $T_{amb} = 0 \degree$C using the BHP concept increases power demand by 36% (and 32%) for moderate (and low) occupancy levels. While adopting the RHP system demands just 24% (and 20%) more power, when compared with the CHP.

3.3. Cycle performance analysis

Fig. 6 illustrates the evolving thermodynamic states of the refrigerant in the CHP (left), BHP (middle), and RHP (right) systems, when $N = 3$, $T_{amb} = 5 \degree$C, and $t_{R-amb} = 30 \pm 30 \degree$C. Here, $t_{R-amb}$ is the “Warm-up time” that it takes for the cabin air to reach the set-point value from the initial temperature, which is fixed for all the studied systems and conditions. After reaching the set-point, the main purpose of the HP is to maintain the cabin temperature within an acceptable range by alternating between ON and OFF modes, which is called “Boosting mode”. The “Initial cycle” in this diagram corresponds to the thermodynamic condition at the start of the warm-up mode, while the “Final cycle” corresponds to the maximum pressure reached by the refrigerant during the Boosting mode. The thermodynamic cycles evolve over the operating time from the initial to the final values within the warm-up and boosting modes, which are coloured in blue and red, respectively. As evident in the boosting mode, an RHP system operates with higher evaporator temperatures compared to a BHP, resulting in a lower power requirement. It is due to the fact that in an RHP concept, warmer air (mixture of the cabin and ambient air) flows through the evaporator, therefore more heat is available for the refrigerant to absorb in higher temperature and pressure. As a result of the higher temperature of refrigerant its vapour density at the suction line and its mass flow rate through the compressor increase by 5% compared to the BHP system, as shown in Fig. 7. Furthermore, as Fig. 7 demonstrates and for the sake of a valid comparison, for all these three cases the superheating and subcooling degrees were kept at the same values (approximately $\pm 1 \degree$C).

The instantaneous cabin temperature and relative humidity are shown in Fig. 8. As it is demonstrated, in all cases the cabin condition could be reached and maintained within the acceptable range successfully. However, the temperature and relative humidity in the BHP concept show a tendency to the right, indicating relatively longer operating time compared to the CHP and RHP systems.

Hereafter, in order to compare different cases and evaluate different concepts, the following non-dimensional and normalized parameters are used:

$$Q_{av} = \frac{Q_{av}}{Q_{av,\ max}} \quad (9)$$

$$t_{Op,\ av} = \frac{t_{Op}}{t_{Op,\ max}}, \quad m_{\ av} = \frac{m_{n, av}}{m_{n, av,\ max}}, \quad m_{n, av} = \int \dot{m}_n dt/t_{Op}, \quad (10)$$

Fig. 9 depicts the normalized power demand over six hours and under three different ambient conditions. The required power depends on a number of parameters including the heating capacity, COP, operation time, compressor and fan speed, ventilation strategies, and the thermal comfort policy. As this figure indicates, adopting a sliding set-point temperature would be beneficial in terms of the power demand for all cases, and also the system could operate with less divergence from its optimum and nominal conditions. Moreover as evident in Fig. 9, for all cases when the number of on-board people increases the required...
power is reduced by at least 19% and at most 22%, as the heat generated by the people would contribute more to the heat balance inside the cabin. Employing a BHP system increases the averaged power demand by at least 32% and at most 39%, compared to the CHP case. As this figure demonstrates, the HP architecture plays a substantial role in its performance, and adopting the RHP instead of the BHP suggests remarkable PES of at least 10% for $N = 3$ and $T_{amb} = 5^\circ C$, and at most 15% for $N = 3$ and $T_{amb} = -5^\circ C$.

Fig. 10 demonstrates the normalized averaged heat flow which is...
delivered to the cabin by the blowers and through the condenser. As this figure shows, the delivered heat into the cabin increases remarkably by at least 39% (and at most 49%) for the RHP, and at least 43% (and at most 58%) for the BHP concepts. In both cases, the waste heat is considerably higher than in a CHP system. It is because more heat is needed to be provided by the RHP and BHP to make up the heating demand and keep the cabin’s condition within the acceptable range. As Fig. 9 shows, the required power for a BHP and RHP concept to run with low occupancy level, respectively drop by 5 and 2% in \( T_{amb} = 5 \) °C, compared with when \( T_{amb} = -5 \) °C. Likewise and as depicted by Fig. 10, for the same occupancy level the normalized averaged delivered heat into the cabin by the BHP and RHP system is 9 and 6% lower for \( T_{amb} = 5 \) °C in comparison with when \( T_{amb} = -5 \) °C. This difference in the percentage of the additional power requirement and the delivered heat could be due to the lower COP of both concepts in higher ambient temperatures, as shown in Fig. 11, where the COP of the system is defined as:

\[
COP = \frac{W}{Q}
\]  

(COP of both the BHP and RHP systems is 4% higher for \( N = 3 \) and \( T_{amb} = -5 \) °C, in comparison with \( T_{amb} = 5 \) °C condition. Moreover, in spite of the fact that the power demand in the RHP concept is 8–11% (at least—at most) lower than BHP, its delivered heat into the cabin is just 3–6% lower. As can be seen in Fig. 11, the RHP improves the COP remarkably comparing to the BHP and CHP. This could be associated with its higher evaporator temperature, as illustrated in Fig. 6. The COP of the RHP system increases by at least 15% for \( T_{amb} = 5 \) °C in low occupancy level \( (N = 3) \) and at most 19% for \( T_{amb} = -5 \) °C with medium occupancy level \( (N = 16) \), compared to the CHP. These figures, are lower in the BHP system and falls between 9 and 14%.

Fig. 12 demonstrates the operation time of each system in different ambient temperatures. It is clear the BHP operates for longer periods of time in order to meet the cabin thermal requirements and this would also contribute to its higher power demand. The lower occupancy level also increases the operation time as the metabolic heat generation contributes less to the total heating balance inside the cabin, thus the HP works more and longer. The operation time of the RHP is 1–5% higher than the CHP, while these figures in the case of the BHP are 8–17%.

Regarding the ventilation rates, the BHP and RHP stand out and suggest significant improvements compared to the air-re-circulated system, the CHP. This is implied by Fig. 13 where fresh airflow rates into the cabin in terms of LPS parameter (Litres per Person per Second) are depicted. As it is shown the LPS for RHP is at least 102% and at most 108% higher than the CHP case. Likewise, a BHP offers at least 111% and at most 125% higher LPS compared with the CHP case. Again, the higher LPS of the BHP could be associated with the higher operation time, compared to the RHP system. It is believed that in the absence of air re-circulation inside the cabin in both the RHP and BHP concepts, this excess amount of fresh air would improve the cabin air quality and provide the passengers with a safer environment regarding the spread of contagious diseases such as the COVID-19 virus.

Comparing Figs. 9-12, we see that the BHP and RHP units work not only more efficiently but also more intensely to be able to keep the cabin temperature around the determined set-point. In fact, in colder ambient conditions, the evaporator should function at lower temperatures/pressures to be able to absorb enough heat from the ambient (or the mixture of ambient and cabin air in the RHP case). Therefore, the refrigerant vapour density at the suction side of the compressor drops as well as the refrigerant mass flow rate that results in lower heating capacities at lower ambient temperatures. In this situation, the compressor speed should increase to make up the heating demand and accordingly the EEV opening needs to be adjusted to provide enough superheating degree, which are shown in Figs. 14 and 15 respectively.

As Figs. 14 and 15 indicate, in all the ambient temperatures, the compressor of the BHP system should operate with the highest speed.
with at least 37% and at most 52% higher RPM, and widest EEV opening which is at least 38% and at most 46% more than the CHP. In the RHP case, these figures are considerably lower than the BHP but still the system runs with 25–38% (at least – at most) higher speed and 31–36% wider EEV opening, compared to the CHP case. It is mainly due to the fact that in each ambient temperature and for their respective set-points, the condenser conditions in the BHP and RHP are similar even though the evaporator temperatures are different. In the BHP concept, the evaporator is exposed to cold airflow from outside; however, in the RHP a mixture of the cabin warm and the outside cold air passes through the evaporator that lets the refrigerant evaporates at higher temperatures/presures, as illustrated in Fig. 6. This would contribute to lower required work from the compressor in the RHP case than in the BHP case. Moreover and according to Eq. (1), the mass flow rate is a function of both the compressor RPM and the refrigerant density at the suction line. In the BHP case, in order to make up the heating demand, the unit operates at increased RPM to boost the volumetric flow rate. This necessitates opening the EEV more, compared to the RHP and CHP cases, as shown in Fig. 15. However, the RHP operates at higher evaporator temperature, resulting in higher density of the superheated refrigerant at the compressor inlet. In spite of lower compressor RPM and closer EEV in the RHP case, the higher refrigerant density at the suction line results in higher mass flow rates, compared to the BHP case. Fig. 16 shows that the normalized average refrigerant mass flow rate, over the operating time, in the RHP concept is 32% – 41% (at least - most) and 4%-6% higher than in the CHP and BHP concepts, respectively. In the CHP, the compressor speed and the refrigerant mass flow rates are the lowest as less heating capacity is required, among all the studied concepts, as shown in Fig. 10. It is worth noting that BHP delivered the most heat into the cabin despite having a lower mass flow rate than the RHP concept, which might be owing to its longer operation time, as seen in Fig. 10 and Fig. 15.

### 4. Conclusion

In this study, we investigated how using a novel recovery concept (RHP) could increase the efficiency of a heat pump in a generic single-deck bus cabin, when an all-fresh air policy was implemented. The proposed RHP concept here is based on recovering waste heat from the cabin exhaust air without using any additional heat exchanger. In this concept, the fresh air is delivered into the cabin and passes through the evaporator to improve both the cabin air quality and the system performance. The main motivation for avoiding air re-circulation in such a system is due to valid health concerns about its plausible negative effects on the spread of infectious diseases like the COVID-19 virus in public transportation systems. Three different coupled models of a generic bus cabin, and a heat pump unit were created where the latter one could be a baseline, recovery, or a conventional system. Two different occupancy levels were studied for the ambient temperature ranging from -5 to 5 °C, while a sliding cabin set-point temperature was employed. Accordingly, the results showed that:

- The RHP system increased the delivered fresh air into the cabin by at least 102% (111% in BHP) and at most 108% (125% in BHP), compared to the CHP system. This excess amount of fresh air along with no air-re-circulation policy could provide a safer environment for passengers and is expected to mitigate the risks of contagious diseases spreading.
- Adopting a 100% fresh air policy for the cabin would increase the waste heat as well. Thus, in the RHP system at least 39% (43% in the BHP) and at most 49% (58% in the BHP) more heat was delivered into the cabin to maintain its temperature around the set-point, compared to the CHP system.
- Compared to the CHP concept, the COP increased by at least 15% and at most 19 % in the RHP concept. These figures fell between 9 and 14% for the BHP concept.

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Fig. 14. Normalized compressor’s RPM of RHP, BHP, and CHP concepts in different operational conditions.

Fig. 15. Normalized EEV opening in RHP, BHP, and CHP concepts.

Fig. 16. Normalized refrigerant mass flow rates of RHP, BHP, and CHP concepts.
Employing the RHP concept could reduce the demanded power for heating the bus cabin by at least 8% and at most 11%, compared to the case where a BHP system is adopted. These figures mean that the RHP concept could suggest a Potential Energy Saving (PES) of at least 10% and at most 15%.

The operation times of the RHP and BHP concepts were 1–5% (at least –at most) and 8–17% higher than the CHP, respectively.

CRedit authorship contribution statement

Ehsan Afrasiabian: Conceptualization, Methodology, Writing – original draft. Roy Douglas: Conceptualization, Supervision. Marco Geron: Supervision, Writing – review & editing. Gareth Cunningham: Resources, Writing – review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgement

This work was supported by Innovate UK and Bamford Bus Company Limited trading as Wrightbus.

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