Analysis of heat pump performance in battery electric buses

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Abstract:
Battery Electric Buses (BEB) driving range is one of the main challenges limiting their adoption at a massive scale. Heating and air conditioning of the bus cabin is an energy-intensive process that results in additional energy consumption from the sole onboard energy source. The limited capacity of the battery makes it crucial to quantify this load and study its impact on the bus range. Heat pumps (HP) are the main heating, ventilating and air conditioning technology deployed in electric buses due to their superior performance over electric resistance heaters. However, their performance witness major variations at different external temperatures and operating conditions such as air and refrigerant flow rates and compressor speed. This paper quantifies the thermal load in BEB and studies the HP performance at different weather and operating conditions. First, cabin thermal comfort conditions are set, and HP components technological constraints are presented. Then, a HP model is integrated into a BEB model to fulfill its thermal needs. Simulations are conducted at different external temperatures analyzing the HP performance. Moreover, this paper conducts a sensitivity analysis to study the effect of HP operating conditions on the Coefficient of Performance (COP) and energy consumption. The simulation shows that the compressor speed, the air recirculation rate and the mass flow rate of the air blown into the cabin have the most significant impact on the HP COP.

Keywords:
Battery Electric Buses, Heat Pumps, Coefficient of Performance, Thermal Load, Autonomy

1. Introduction

The rapid expansion of global transportation sets serious challenges in terms of CO2 emissions to meet the long- term 60% reduction target in greenhouse gas (GHG) emissions in 2050 according to the 2011 Transport White Paper[1]. The transportation sector GHG emissions are one-fourth of the total emissions[2]. Several countries are shifting their diesel bus fleets to more environmentally friendly technologies to meet the continuously tightened environmental legislation. [3]. Being a zero-emissions technology, BEB plays an important role in this transition [3].

EU countries are utilizing their efforts to enhance the poor air quality specifically in urban areas because it puts serious threats on the citizens' health. The rapid increase rate in carbon emissions puts a serious risk strengthening the greenhouse effect and causing global warming [3]. Their main concern is to decrease the tail-pipe emissions of internal combustion engines from which harmful pollutants such as nitrogen oxides and particulates are emitted. BEB, as a technology having zero tailpipe emissions, can enhance the overall air quality in the cities [3]. However, this technology still faces several due to the limited specific energy and power of the on-board batteries [4]. BEB charging needs and technology may vary to adapt to the city's schedule of the public transportation operation[3]. But, with any choice of charging technique, for a city to operate and schedule BEB in their transportation fleet, the city should consider how many kilometers can the BEB covers on one charge[4]. This factor is called the BEB range. BEB range is directly related to the energy consumption of the bus. An increase in the energy consumption of the bus will decrease its range and creates a disturbance in the schedule and routes planned for the bus to cover. One of the major contributors to the energy consumption of the BEB is the energy needed to heat up or cool down the cabin [4].

The thermal load needed by the bus to achieve thermal comfort varies at different outdoor conditions. This thermal load is supplied by a HVAC (heating, ventilating and air conditioning) unit that consumes energy directly from the bus battery [4,5]. The higher this thermal load, the greater the energy consumed by the HVAC unit. While the Bus is operating, at extreme outdoor conditions, the energy consumption of the HVAC unit is significantly increasing the total energy required by the bus which in turn reducing the range of the BEB [4]. In fully electric vehicles (FEV), heating of the cabin can reduce the overall driving range by 50% at extreme outdoor conditions[5,6]. That being said, BEB range is sensitive to the HVAC technologies used in a vehicle.

In BEB, Heat Pump (HP) technology is widely deployed for heating and cooling operations [5]. HP uses an electric compressor powered by the sole on-board power source [4]. despite their effectiveness as an HVAC technology used for both heating and cooling, HP shows major variations in their performance at the different outdoor condition and operating parameters[5]. At extreme winter outdoor conditions, the efficiency of the heat pump decreases significantly [5].

In the literature, few of the studies analyzed the performance of a heat pump in battery electric buses. Most of the articles studied the performance of the vapor compression cycle (VCC) that the heat pump technology is based on. Different control approaches are available due to the variety of control parameters and applications. The number of control parameters selected by each study is related to the purpose of the study itself and the type of vehicle used.[4] targeted to design and experiment a modified new model of a HP system to an electric bus with a dynamic on-road charging. [7] showed only the variation of the COP of the system with respect to the variation of the compressor speed and outdoor conditions. [8] varied the compressor speed and the air flow rate over the condenser for an electric vehicle. However, the HVAC unit in [8] consisted of 3 loops and one of the loops is the HP.[9] controlled the air mass flow rate at the inlet of the condenser and recirculation rate in an electric vehicle. It is important to note that, in a conventional car, the compressor is mechanically connected to the engine working with a constant speed. In BEB, the usage of a variable speed electric compressor allows the control of the speed of the compressor. [10] targeted in their application the efficiency of the Vapor Compression Cycle. They controlled the compressor speed and the expansion valve and monitored the superheat level and pressure difference between the condenser and the evaporator [10]. However, their system only included the vapor compression cycle and mainly they varied one stream without considering the HVAC unit interaction with the bus cabin. The HVAC system in a car has more variables to consider especially with the air stream playing an important factor in the system. The air flows through the condenser and the evaporator to either cool or heat the refrigerant. The air stream conditions highly affect the heat transfer on the condenser and the evaporator. [11] also had the same as [9] control parameters to control and optimize the vapor compression cycle. In their approach, the air inlets to the heat exchangers have known boundaries whereas, in the HVAC application, this is not the case.

None of the studies conducted a sensitivity analysis on all the possible control parameters of the HP. In addition, none of the studies analyzed the performance of a heat pump coupled to a BEB. Nevertheless, the studies were analyzing the Refrigeration cycle or the HP alone without the integration of the model into a BEB thermal model. Hence, this paper presents a model of a heat pump that is integrated into a BEB model to satisfy the cabin thermal needs. Moreover, it presents sensitivity analysis on all possible HP control parameters that were discussed in the literature review, in order to assess their impact on the Coefficient of Performance (COP) of the HVAC unit.

The paper structure is as follow: Section 2 introduces the HP operation and main components and defines the thermal comfort needs in BEB. Section 3 introduces the HP model developed in this study and explains the integration of the HP model in the BEB model. In section 4, sensitivity analysis is presented, and the results are discussed and analyzed. Finally, section 5 features the main findings of this paper.

2. HVAC unit in a BEB

2.1. Heat Pump Configuration

Heat pumps transfer thermal energy into (heating) or out of (cooling) an enclosed space such as a bus cabin. HP is based on the vapor compression cycle. The vapor compression cycle (VCC) is the most dominant in air – conditioning or heating applications. HP includes 4 main components: Compressor, condenser heat exchanger, evaporator heat exchanger, and an expansion valve. A 3 – way valve directs the flow to switch between shifting the HP function from cooling the cabin to heating it and vice versa. The purpose of the HP in heating is to heat the air blown into the cabin which is achieved by the VCC. The HP main components connections are described in figure 1.

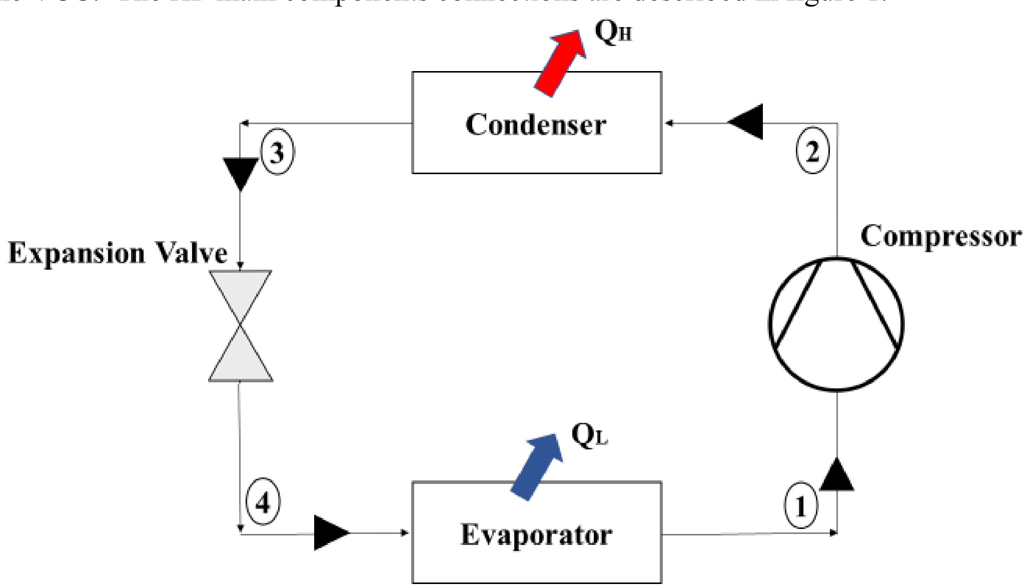

Figure 1 Vapor Compression Cycle configuration.

Referring to figure 1, the VCC cycle is split into a low- and high-pressure mediums. The compressor compresses the refrigerant from state one from low pressure and temperature to high pressure and temperature at state two. The refrigerant then witnesses a phase change from vapor to a liquid state in the condenser through heat exchange with air thereby the air cools the refrigerant and in return, the air gets heated. The expansion valve decreases the pressure of the refrigerant to the low pressure set at constant enthalpy. Finally, through the evaporator, the refrigerant returns to its vapor state after exchanging heat with the air and cooling the latter.

The air is directed through fans in the heat exchangers. Mainly, in heating mode, the condenser is the inner heat exchanger in the bus and the evaporator is the outer one. It is important to note that this paper conducts a sensitivity analysis on the performance of the HP in the heating mode only.

Each component is selected to meet the targeted heating capacity. The chosen compressor is a variable speed scroll compressor with a displacement of 121 cm^3 and has a speed range between 500 rpm to 7000 rpm. The condensers' selected geometry has a heating capacity of 40 KW in order to meet the thermal need of the bus cabin at the worst-case scenario (-10 degree Celsius). R-1234yf refrigerant was selected for this heat pump application since it has a zero Ozone Depletion Potential. Moreover, compared to the most common R – 134a refrigerant's Global Warming Potential (GWP) of 1430, R1234yf has a GWP of 4. The components specifications are presented in Table 1.

Table 1 Summary of the specifications of the HP components

Name	Type	Geometry	Parameters
Compressor	Variable Speed Scroll Compressor	Displacement:	121 cm^3
		Speed Range:	500 – 7000 rpm
Condenser	MPET Cross Flow Heat Exchanger	Heating Capacity:	20 KW
		Heat transfer Area (Airside):	0.2067 m^2
Evaporator	MPET Cross Flow Heat Exchanger	Heat transfer Area (Airside):	0.0598 m^2
Expansion Valve	Electronic Expansion Valve	Orifice Valve	

2.2. Thermal Comfort in BEB

The bus cabin has thermal needs that the HP should meet. Thermal comfort put restrictions on several parameters including cabin temperature and humidity, blown airspeed, blown air temperature, and convergence time to reach steady state. Table 2 summarizes the thermal comfort parameters that are considered in this study.

Table 2 Thermal comfort conditions [12,13]

Parameters	**Thermal Comfort Conditions**
Cabin Temperature	$19℃ - 23℃$
Cabin Relative Humidity (RH)	30% - 50%
Convergence Time (-28.8 ℃ to 20℃)	70 minutes
Maximum Blown Air Temperature	49℃
Maximum Blown Airspeed on any Passenger	0.508 m/s
Minimum Flow Rate per Passenger	0.42 m^3/s

The cabin temperature and RH are respectively the temperature and humidity inside the cabin that should be maintained to achieve thermal comfort. American Public Transportation Association (APTA) restricts the heating unit to be able to heat the cabin from -28.8 °C to 20°C in 70 minutes. However, in this paper, a convergence time of 10 minutes was targeted to heat the cabin from 0°C or 10°C to at least 19°C. Also, a convergence time of 20 minutes is taken to heat the cabin from -10°C to at least 19°C. Before blowing air to the cabin, the blown temperature should be greater than the set cabin temperature and not exceeding the maximum limit to avoid any discomfort for the passengers. Moreover, the maximum blown air speed must not be exceeded to avoid any discomfort for the passenger.

Recirculation influences the CO_2 concentration inside the cabin and the formation of fogging on windows. The maximum recirculation rate specified by APTA is 90%. However, [14] conducted a study to examine the relation of fogging formation with the external temperature, external humidity and the speed of a vehicle. For the sensitivity analysis conducted by this study, a maximum of 80% of the recirculation rate was taken [14].

3. Model

3.1. Heat Pump Model

The heat pump is modeled on Dymola software. Dymola has multi-engineering capabilities that allow the integration of different engineering domains [15]. The components found in Dymola represent physical components that are described by their corresponding differential and algebraic equations. The graphical connections between the models represent their physical coupling which makes the model intuitively organized the same way as the physical system is composed. All the components used were selected from a built-in library in Dymola called ThermalSystems and TSMobileAC. In order to meet the thermal need of the cabin at -10°C, two identical heat pumps were installed in the bus.

The condenser and evaporators selected are built-in the TSMobileAC library heat exchangers. An expansion valve with a variable effective flow area is selected from the ThermalSystems library. All the system components parameters calculations are based on specific equations for each parameter provided by Dymola. For example, in the condenser heat exchanger (HX), mass and energy balances are applied to calculate the enthalpies at inlets and outlets of the HX ports. Similarly, mass and energy balances are used to calculate the enthalpies at the inlet and exit of the compressor. The COP of the system is calculated using the heat flow rate supplied by the condenser to the cabin and the work of the compressor using equation (1), (2), and (3).

$$COP = \frac{Q_H}{W_{shaft}} \tag{1}$$

$$W_{shaft} = \dot{m}_{ref}\left(h_{discharge} - h_{suction}\right) \tag{2}$$

$$Q_H = \dot{m}_{ref}\left(h_{in}^{evap} - h_{out}^{evap}\right) \tag{3}$$

The superheat level was calculated using equation (4) where it is used in section 4.4.

$$SH = T_{evaporator\ exit} - T_{sat} \tag{4}$$

Where T_{sat} is the temperature at which the refrigerant is a saturated vapor at the evaporator pressure.

$$\eta_{isentropic} = \frac{h_{discharge\ isentropic} - h_{suction}}{h_{discarge} - h_{suction}} \quad (5)$$

3.2. HVAC – Bus Model

The BEB model is developed by the current co-author. The bus is 12-meter long with a maximum passengers capacity of 50 passengers. The bus model incorporates all the heat losses and gains in the cabin. It considers the radiation, convection, and conduction heat transfers with the environment, in addition to transmitted and absorbed solar flux.

The bus heat exchange with the environment takes place through its walls. The wall model is an array of walls covering the top, bottom, front and back walls of the bus body. The walls can be opaque representing the metal body or can be glazing on behalf of the glass body. The walls have an outer side facing the environment and an inner side facing the cabin interior. On the outer level, the model considers the convection and radiation between the wall outer side and the exterior. Through the wall, the model includes conduction from the wall outer side to the wall inner side. Through the glazing part of the wall, the model incorporates the transmitted heat flow and the absorbed heat flow. On the inner level, heat transfer through convection and radiation between the wall inner side and the cabin interior is taken into consideration. Adding to that, The internal mass exchange heat through convection and radiation. The model considers the passengers' thermal loss and the passenger's water vapor generation. One limitation of the BEB model is that it doesn't take into consideration the losses associated with the opening and closing of the bus at different stop stations.

The heat pump supplies the cabin with the heat flow rate Q_H. In the heat pump, the evaporator is fed with the temperature and the humidity of the external environment. The air at the inlet of the condenser takes into consideration the mix of air in case of recirculation. Recirculation is one of the parameters that this paper target to study its effect on the heat pump. The configuration of the HVAC unit and cabin is presented in figure 2. The properties of the mixed air were estimated using equations (6) and (7).

$$T_{mix} = (1 - recirc) * T_{ext} + recirc * T_{cabin} \quad (6)$$

$$\omega_{mix} = (1 - recirc) * \omega_{ext} + recirc * \omega_{cabin} \quad (7)$$

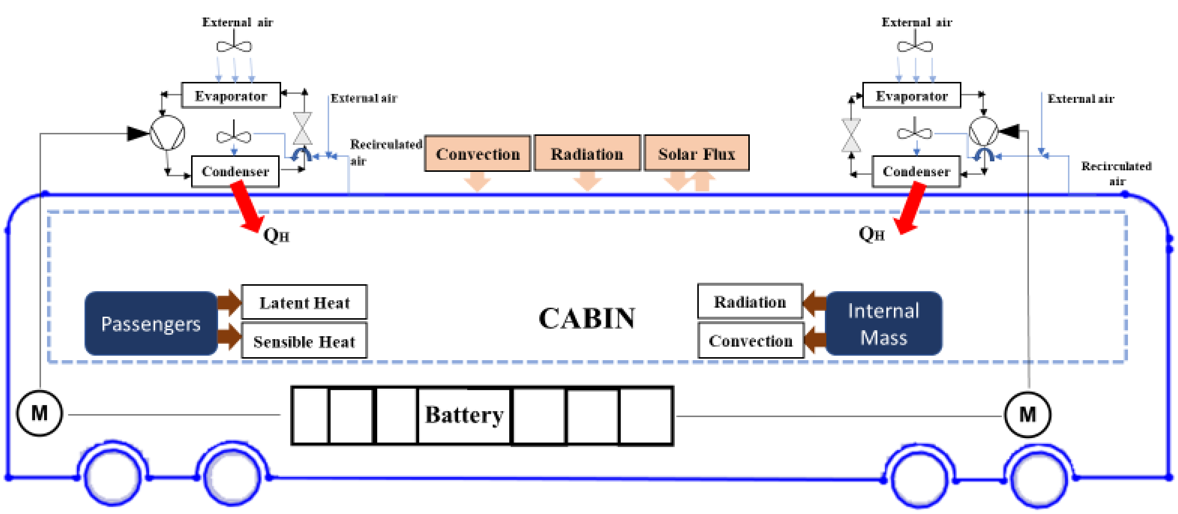

Figure 2 HVAC - cabin configuration.

4. Results

The five main parameters collected from literature papers that are interesting to study their variation effect on the heat pump performance of BEB are: the compressor speed, the mass flow rate of the air entering the condenser, the mass flow rate of the air entering the evaporator, the recirculation rate, and the flow effective area of the expansion valve. To analyze the variation of one parameter, the other parameters were held constant. The same sensitivity analysis is conducted at -10°C, 0°C and 10°C. The temperatures selected represent typical winter temperatures in France taking the -10°C as the extreme outdoor temperature scenario[16]. Table 2 summarizes all the sensitivity analysis parameters with the conditions considered.

The ranges of all parameters at different external temperature are selected based on the thermal comfort requirements in a bus discussed in section 2.2. For all the ranges selected the cabin temperature must reach a temperature between 19 °C and 23 °C in the convergence time specified.

Table 3 Sensitivity analysis parameters ranges

Sensitivity analysis	**Parameter varied**	**Range**		
		-10°C	**0**°C	**10**°C
Test 1	Compressor Speed	[3900 – 4260] (rpm)	[2400 – 3000] (rpm)	[600 – 1200] (rpm)
Test 2	Air Mass Flow Rate at the Inlet of the Condenser	[0.45 – 0.75] (kg/s)	[0.45 – 0.75] (kg/s)	[0.2 – 0.45] (kg/s)
Test 3	Recirculation Rate	[30% - 70%]	[30% - 60%]	[0% - 30%]
Test 4	Air Mass Flow Rate at the Inlet of the Evaporator	[0.4 – 0.8] (kg/s)	[0.4 – 0.8] (kg/s)	X
Test 5	Expansion Valve Opening	[2.1– 3] (mm^2)	[1.4– 1.96] (mm^2)	[0.4– 1.1] (mm^2)

4.1 Effect of the Compressor Speed on the COP of the HP

In figure *Figure 3*, the effect of the variation of the compressor speed on the COP of the HP at the external temperatures -10°C, 0°C, 10°C is presented. The COP decreases by 40%, 25% and 15% with the increase in the compressor speed by 600 rpm at -10°C, 0°C, 10°C respectively. There are two reasons behind the decrease in the COP. First, as the compressor speed increases, the isentropic efficiency of the compressor decreases. The isentropic efficiency is calculated using equation (5). The compressor speed increase allows higher pressure ratios in the system due to the increasing condensation and decreasing evaporation pressures. Higher pressure ratios decrease the isentropic efficiency of the compressor in the speed range considered[17]. For example, at 0°C, the isentropic efficiency decreased by 18% with the increase in the compressor speed by 20%. The second reason behind the decrease in the COP is the increase in the frictional losses at higher compressor speeds. The increase in the frictional power of the compressor decreases its mechanical efficiency. For example, at 0°C, the frictional power of the compressor increases by 10%.

The COP decrease rate varies among the different external temperatures. As the external temperature gets higher, the temperature difference between the refrigerant and air streams at the inlets of the condenser decreases. This leads to a less increase in the heating capacity at higher temperatures. The compressor work contribution to decreasing the COP becomes more significant.

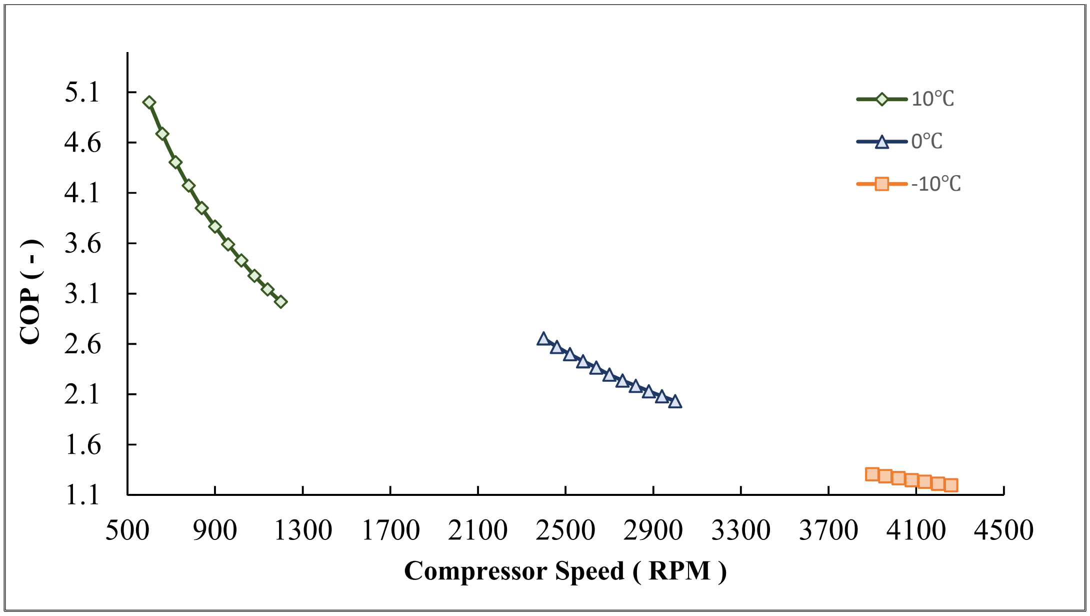

Figure 3 Compressor speed versus COP at -10℃, 0℃ and 10℃.

4.2 Effect of the mass flow rate of the air at the inlet of the condenser

The variation of the air mass flow rate at the condenser inlet versus the COP at the external temperatures -10℃, 0℃, 10℃ is presented in figure 4. Increasing the air mass flow rate at the condenser inlet improves the COP. The COP increases by 8.2% as the air mass flow rate increases by 47%. The increase in the air mass flow rate increases the convective heat transfer coefficient. The overall heat transfer coefficient increases also due to the proportionality between the two coefficients [18]. The increase in the overall heat transfer coefficient increases the heating capacity of the condenser.

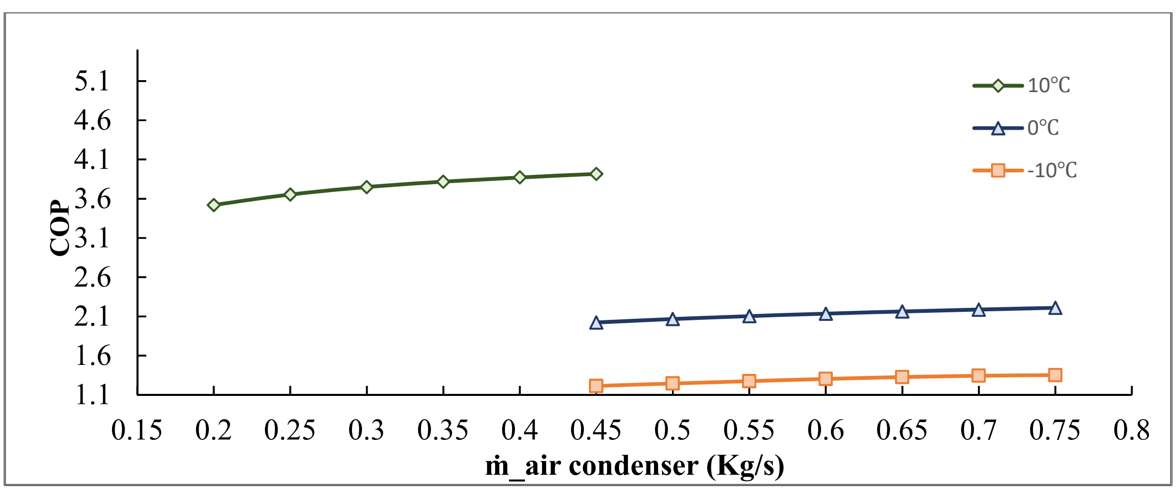

Figure 4 Mass flow rate of air at the inlet of the condenser versus COP at -10℃, 0℃ and 10℃.

4.3 Effect of the mass flow rate of the air at the inlet of the evaporator

The increase in the air mass flow rate at the inlet of the evaporator increases the COP of the HP. The COP increase is less than 4% with the increase in the air mass flow rate by 50% at the external temperatures -10°C, 0°C. As mentioned in the introduction of section 4, all the simulations were conducted to reach the thermal cabin comfort in the specified convergence rate. For this reason, the variation of the air mass flow rate at 10°C was not presented since the range will not be significant to show. The air mass flow rate at the evaporator inlet increases the cooling capacity of the evaporator and the heating capacity of the condenser[19]. However, the increase in the heating capacity is higher than the increase in the cooling capacity. Adding to that, a negligible variation in the compressor work is observed.

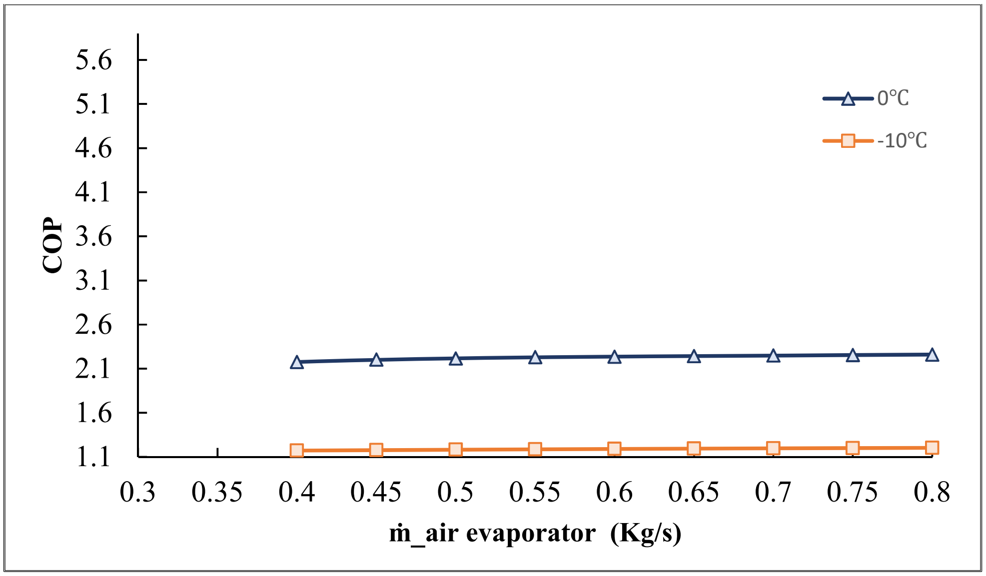

Figure 5 Mass flow rate of air at the inlet of the evaporator versus COP at -10°C *and 0*°C.

4.4 Effect of the effective flow area variation in the expansion valve on the COP

Analyzing the variation of the expansion valve effective flow area is challenging. To start with, superheating is the energy added to saturated vapor, resulting in an increase in its temperature and it is calculated using equation (4). Smaller expansion valve opening offers higher superheat level which can be safe for the operation of the compressor and increases the efficiency of the evaporator. Increasing the superheat to very high levels decreases the evaporator efficiency. However, decreasing the superheat to very low levels reaching zero degrees damages the compressor since liquid particles may still be present at the exit of the evaporator and decrease the efficiency of the evaporator [20]. There is an optimal point for the expansion valve opening to operate on in which before and after the COP of the system decreases.

This decrease in the COP, referring to figure 6, is around 2.5% with the variation of the expansion valve opening. The expansion valve is mainly used to regulate the superheat level in order to operate the evaporator at better efficiency and protect the compressor.

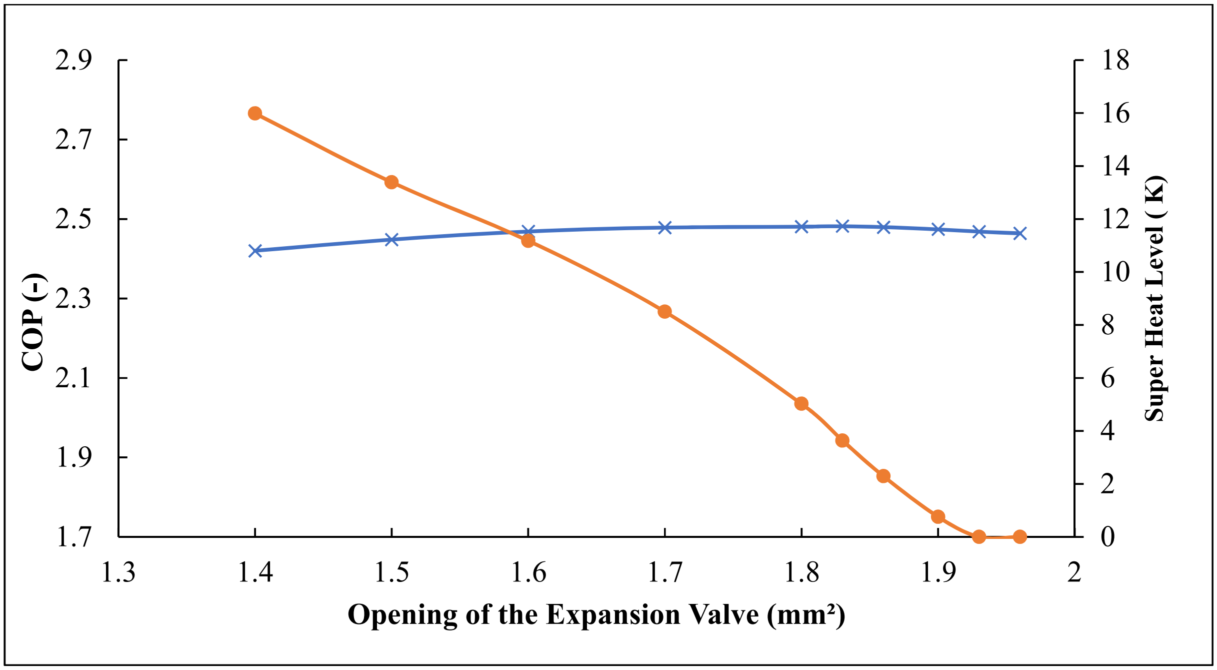

Figure 6 Opening of the expansion valve versus COP and superheat level at 0°C.

4.5 Effect of the recirculation rate on the COP

It is critical to study the direct effect of the recirculation rate on the COP of the system. Analyzing the recirculation rate variation while leaving all the other parameters constant is not intuitive.

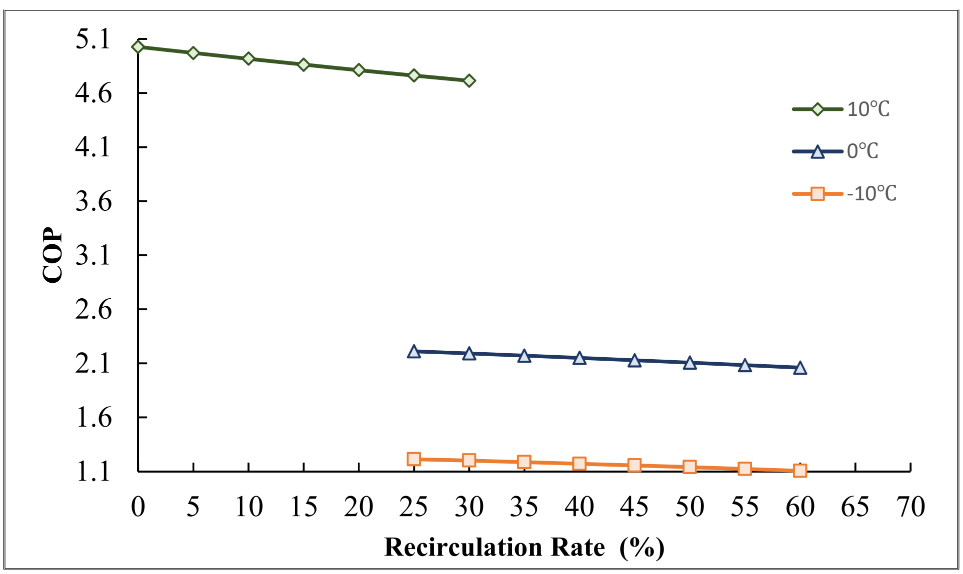

Figure 7 Recirculation variation versus COP at -10°C, *0*°C, 10°C.

Increasing the recirculation rate decreases the COP by referring to figure 7. The difference in temperature between the air and the refrigerant stream at the inlet of the condenser decreases which allows less heat transfer. However, as observed in figure 8, by increasing the recirculation rate, the targeted cabin temperature is reached in a period less than the specified convergence time so in order to meet the cabin temperature convergence time constraint, the compressor speed is regulated in a way the studied recirculation rates all converge the cabin temperature at the same time[21].

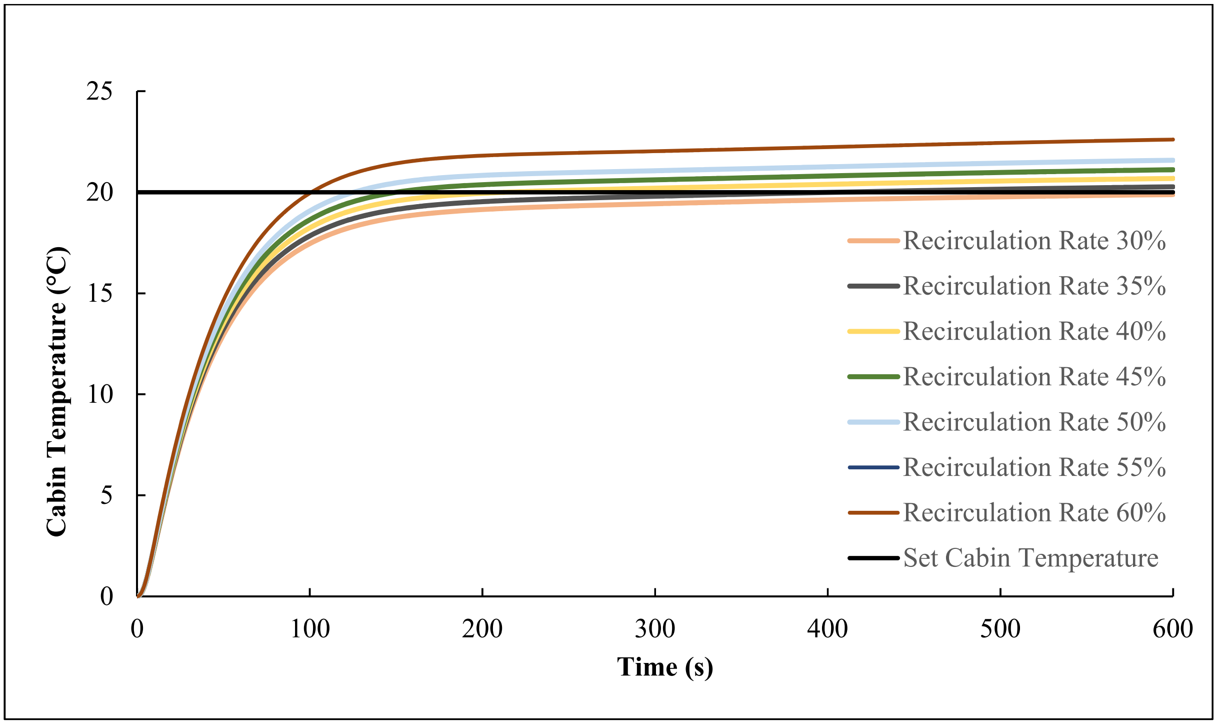

Figure 8 Convergence time versus the cabin temperature at different recirculation rates at 0℃.

In this case, higher recirculation rates will allow the compressor to operate at lower speeds. The decrease in the compressor speed will increase the COP of the system as discussed in section 4.1. So, the increase in recirculation rate has an indirect effect in increasing the COP by allowing the HP compressor to operate at lower speeds. An increase in the recirculation rate from 30% to 60% led to a 13.68% increase in the COP as presented in figure 9.

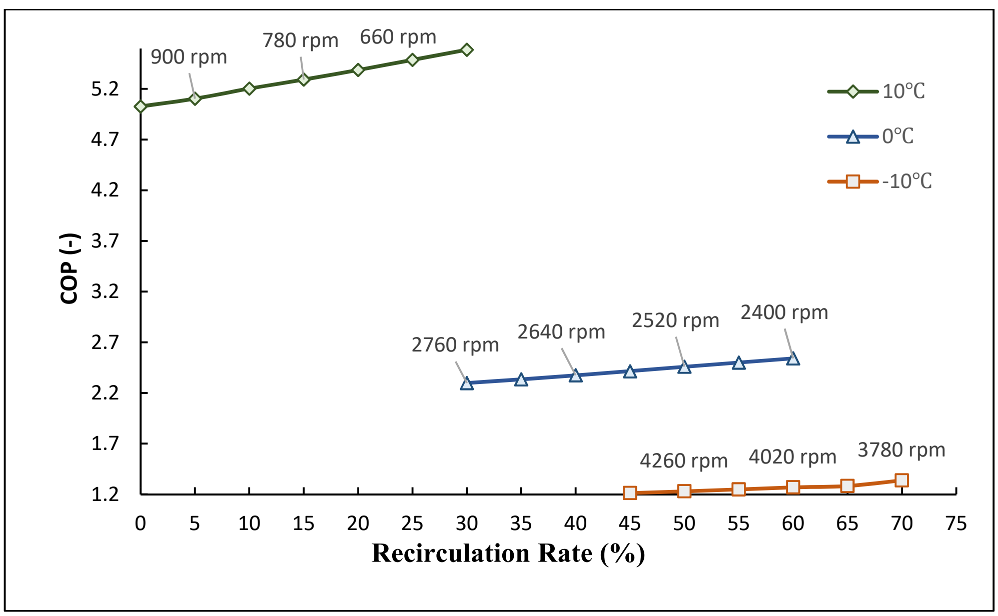

Figure 9 Recirculation rate versus COP at a specific compressor speed to meet the cabin comfort time convergence at -10℃, 0℃ and 10℃.

5. Conclusion

This paper studies the heat pump performance with respect to the variation of compressor speed, air recirculation rate, the air flow rate at the condenser inlet, the air flow rate at the evaporator inlet, and the expansion valve opening. The focus was to study the possibility of improving the COP by varying the main parameters in the HVAC unit in a BEB. The results assure that there is a room for improvement in the performance of the HP in BEB. As analyzed in section 4, the variation of the compressor speed by 600 rpm, the recirculation rate 35%, and the air mass flow rate at the condenser by 40% have the most significant effect on the variation of the COP of the HP with a percentage of 25%,10%, and 8.2% respectively. However, the variation in the air mass flow rate at the evaporator inlet and the opening of the expansion valve effect the COP in less by 4%. In BEB, electric compressors, having the greatest impact on COP, can be controlled unlike in diesel buses where the compressor is mechanically connected to the compressor operating at a constant speed. Thus, the three parameters with the highest effect on the COP can be used in control theory to optimize the performance of the HP decreasing the energy needs to heat up the BEB cabin and thus reducing the impact on the BEB range.

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