

Article

Optimising Ventilation Strategies for Improved Driving Range and Comfort in Electric Vehicles

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Abstract: A car cabin's small volume makes it vulnerable to discomfort if temperature, humidity, and carbon dioxide levels are poorly regulated. In electric vehicles, the HVAC system draws energy from the car battery, reducing the driving range by several dozen kilometres under extreme conditions. A 1D simulation model calibrated for the Renault ZOE was used to evaluate the effects of ventilation parameters on thermal comfort, humidity, and power consumption. The results highlighted the interdependence of factors such as the recirculation ratio and blower flow rate, showing that energy-efficient settings depend on ambient conditions and other factors (such as occupancy, vehicle speed, infiltration). Adjustments can reduce heat pump energy use, but no single setting optimally balances power consumption and thermal comfort across all scenarios. The opti-CO₂ mode is proposed as a trade-off, offering energy savings while maintaining safety and comfort. This mode quickly achieves the cabin temperature target, limits carbon dioxide concentration at a safe level (1100 ppm), minimises fogging risks, and reduces heat pump power consumption. Compared to fresh air mode, the opti-CO₂ mode extends the driving range by 9 km in cold conditions and 26 km in hot conditions, highlighting its potential for improving energy efficiency and occupant comfort in electric vehicles.

Keywords: thermal comfort; 1D simulation; energy efficiency; heat pump; HVAC system



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1. Introduction

This study presents a comprehensive analysis of energy-efficient ventilation strategies for electric vehicle cabins, focusing on the interplay between thermal comfort, air quality, and power consumption. Our research introduces the novel opti-CO₂ mode, a smart HVAC (Heating, Ventilation, and Air-Conditioning) strategy that optimises energy use while maintaining a safe and comfortable cabin environment.

In a vehicle, the HVAC module aims to offer thermal comfort to the passengers, with a limited amount of energy. An overview of the airflows in a cabin passenger vehicle is displayed in Figure 1. Generally, the air inlet is located at the bottom of the windshield. The outside air is driven into the cabin with the help of the blower. This air flows around the occupants, which are a source of heat. The outlet of the cabin is at the rear back of the vehicle, through the decompression traps. In a vehicle, there is also the recirculation mode, either instead of the fresh air inlet, or as a complement. In that case, there can be a mix between fresh air and recirculation. In addition, there is the infiltration phenomenon, referring to the air entering the cabin by other means than the blower operation, through small gaps and bad sealing in the car body. The “infiltration” term refers to the air not treated thermally by the HVAC unit. Infiltration occurs in particular conditions, and is

triggered by high vehicle speed and low fresh air flow rate (low relative pressure in the cabin). The calibration process of the infiltration flow rate in an electric vehicle is detailed in a previous publication [1]. In the current article, the simulation model used to evaluate the HVAC system includes the infiltration model. This offers a new approach to existing numerical models from the literature [2], as they do not take into account the impact of this fresh air that penetrates the cabin.

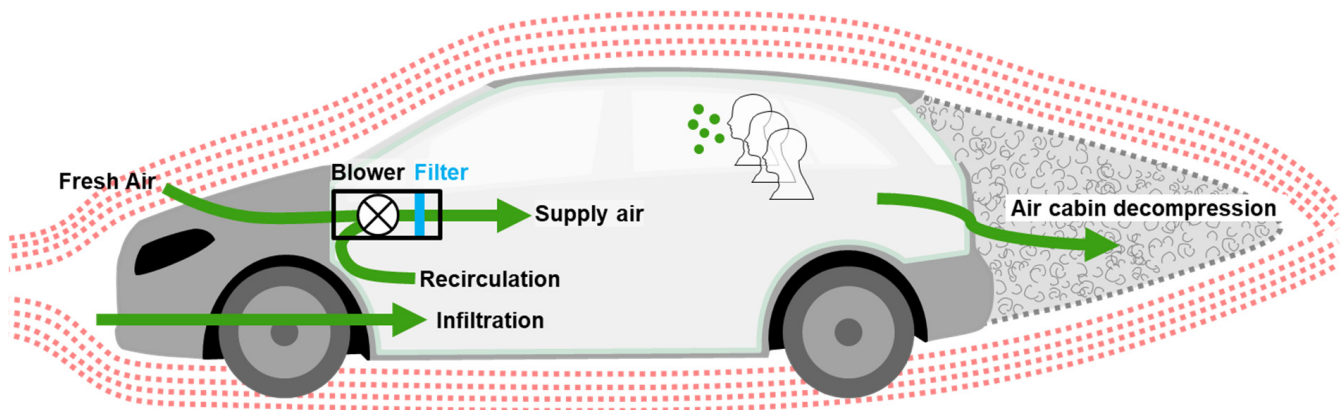


Figure 1. Overview of airflows in a vehicle cabin.

1.1. Thermal Comfort

The passenger compartment is a confined space, containing one or more people in only a few cubic metres. In addition, driving a vehicle requires a certain level of concentration. In order to avoid fatigue and a degradation of the driving skill, it is preferable to have a feeling of comfort, especially thermal comfort. One study relates driver fatigue to different temperature settings [3]. It is found as well that thermal stress negatively influences the driver's abilities.

The sensation of thermal comfort felt by a person is a mix of physical and mental conditions. It depends on several factors related to both the individual and the environment. The standard ASHRAE 55 (American Society of Heating, Refrigerating and Air Conditioning Engineers) defines a comfort zone based on six variables [4]: the metabolism of the occupants, the degree of insulation of the occupants' clothing, the relative humidity of the air, the temperature of the air, the temperature due to radiation from the walls, and air flow velocity. However, thermal comfort is a complex concept that is difficult to ensure and is not limited to the above physical parameters. It is also influenced by subjective factors, including psychological and socio-cultural criteria, as well as individual characteristics such as gender, age, constitution, and health [5]. From the literature [6–8], the main environmental factors influencing the perception of thermal comfort in car cabins are the following: temperature, relative humidity, air velocity, and solar radiation.

Assuming that the temperature is at an acceptable level, the comfort level of relative humidity for humans ranges from 30% to 70% (ideally between 40% and 60%) [6,9,10]. Below 30%, there are some effects on the human body: eye and throat irritation, and the drying of the mucous membranes and nasal passages [6,9]. Above 70%, the excess of water can cause a reduced evaporative cooling of the body through sweating, causing a suffocating sensation. An experimental study evaluates the effects of the car cabin environment on passenger comfort and fatigue [11], under different driving conditions. It is demonstrated that the rate of complaints related to fatigue increases with high humidity conditions (>60%). It is also showed that indoor humidity and local air velocity have a negative impact on eye dryness sensation and visual fatigue, which could also result in further distractions due to discomfort while driving.

Air velocity is also a parameter influencing human comfort. It influences heat exchange by convection and increases evaporation from the skin surface. It is recommended that it does not exceed 0.2 m/s [5]. However, in hot and humid summer environments, higher air movements can be tolerated (e.g., 1.2 to 2.5 m/s, corresponding to a light breeze).

A factor that can lead to discomfort is solar radiation, although it is not taken into account in thermal comfort indices [7]. This parameter does not only influence the cabin air temperature, but is also a local discomfort parameter (visual dazzling). Additionally, several personal factors such as metabolism and clothing insulation can affect human comfort levels [7]. The metabolism affects the heat generation rate inside the body, therefore requiring different energy balancing loads. Some typical heat generation values range from 60 to 115 W per square metre of skin area during driving [6]. Finally, the clothing of the occupants can also affect the body heat losses and comfort due to the insulation provided by the clothing [6,7]. The variability of clothing insulation between individuals further highlights the subjectivity of thermal comfort, as differences in clothing choices result in varying levels of heat loss and comfort, making it challenging to satisfy all occupants with a single thermal setting.

1.2. Fresh Air Sensation in a Car Cabin

In addition, an important aspect of comfort has not been taken into account so far: air exchange. The renewal of indoor air is essential. It allows the evacuation of pollutants and odours, but also the water vapour and carbon dioxide released by individuals. Indoor air should not only be free of harmful substances and at a good thermal level, but also be perceived as fresh, odourless, and non-irritating. Certain trace substances, which have no effect on health, can be malodorous, such as butyric acid produced by sweating.

A method for assessing the perception of fresh air in an indoor environment involves monitoring the concentration of carbon dioxide (CO₂). This odourless and colourless gas, which is a by-product in the exhalation of living organisms, serves as an effective indicator of human-emitted bio-effluents (i.e., odours) that are typically considered undesirable for comfort. Therefore, by monitoring CO₂ levels, the overall quality of the indoor environment can be evaluated.

The effects of the exposure to CO₂ have been studied to assess the human tolerance of staying in confined spaces (submarines, basements, caverns, etc.). The extent of the effects depends on the concentration in the atmosphere, the duration of exposure, and numerous physiological (age, vascular condition, etc.) and climatic factors (temperature and oxygen pressure). There are no significant variations in a person's biological parameters below 5000 ppm. This limit is used as the maximum occupational exposure limit in workplaces of many countries [12]. This limit is given in order to avoid health issues, and not for a comfort purpose.

The ASHRAE provides a CO₂ limit that aims to satisfy good air quality in closed areas such as car cabins. It is specified that the CO₂ concentration should not be higher than 700 ppm over the ambient concentration in the atmosphere [13]. This guideline is based on the use of indoor CO₂ concentration as a surrogate for levels of human bio-effluents. It should be noted that this guideline is not related to any health effect from the CO₂ itself, but only to an odour sensitivity caused by human bio-effluents that can cause discomfort. The typical ambient concentration level of CO₂ is approximately 400 ppm (0.04%). Hence, the CO₂ concentration that should not be exceeded in a car cabin is 1100 ppm. Above this limit, outside air must be introduced to decrease the CO₂ concentration.

Many studies show that, in a car cabin, the CO₂ concentration can accumulate quickly [14–16]. Depending on the ventilation mode and the number of occupants, the CO₂ concentration can be up to 4000 ppm after a few minutes in recirculation mode. Such a level

maintained over the long term can be harmful, particularly for road professionals (e.g., bus or taxi drivers) that spend a lot of time in their vehicle. Several studies show the effects of an exposition of CO₂ at such a concentration in closed volumes (rooms and buildings).

The first one links the concentration of indoor CO₂ concentration with the amount of sick symptoms in office buildings [17]. A sore throat, tight chest, and wheezing can be correlated to bad air renewal in office buildings (CO₂ above 800 to 1000 ppm). In another study, an author suggests that the prolonged exposure to low-levels of CO₂ can alter bone metabolism and blood calcium concentration in young adults (CO₂ above 3000 ppm) [18]. The most impactful study about CO₂ effect at a low concentration is probably the work of [19]. The test results are given in Figure 2. From 600 to 1000 ppm, statistically significant decrements occurred in five of nine categories of decision-making performance. At 2500 ppm, the decrease in performance occurs in seven categories. For instance, initiative, basic strategy, or information usage show very poor results (dysfunctional). Only the tests requiring a full focus are maintained at a good result compared to 600 ppm. This result is also confirmed in another study [20], but with a smaller group sample. They suggest that the CO₂ level may directly influence human performance.

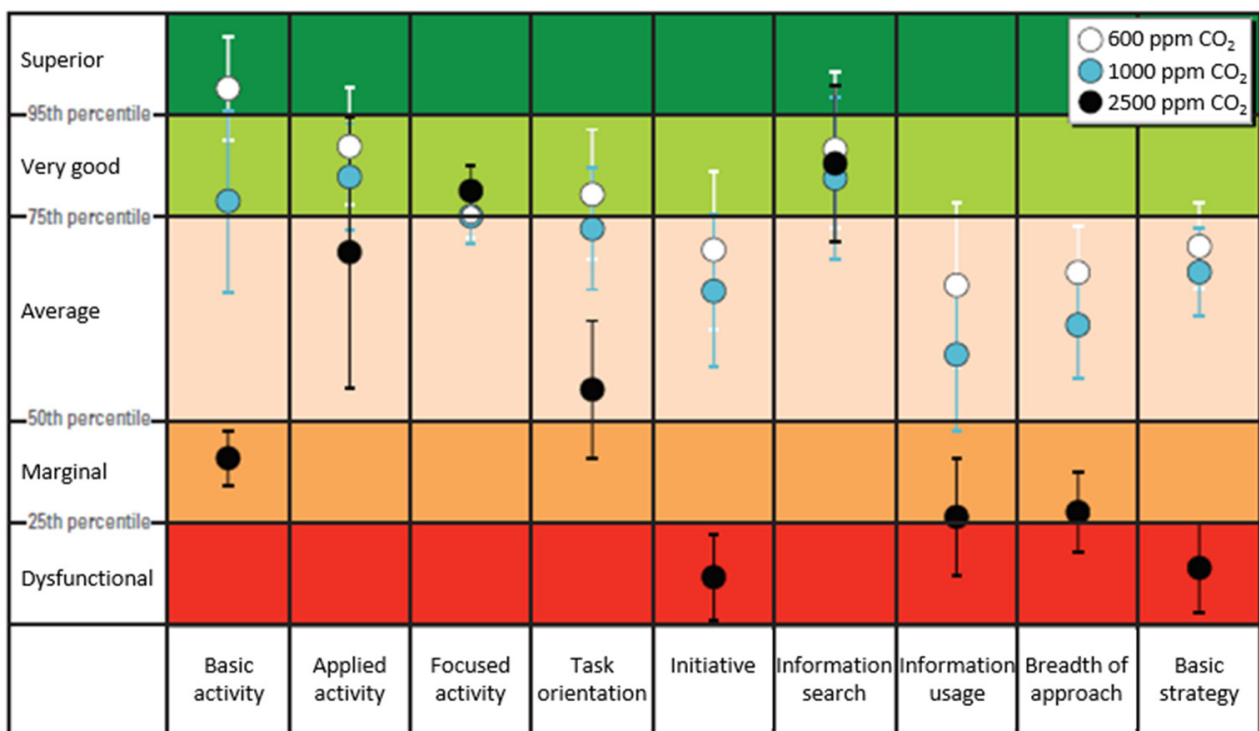


Figure 2. The effect of CO₂ on human decision-making performance, adapted from [19].

According to these findings, it is clear that the driving of a vehicle can be degraded if the CO₂ level is too high. The ASHRAE guideline of 1100 ppm appears as a good target, since the degradation of human performance is limited at this level. Several studies have used this guideline inside a vehicle cabin [14,21,22]. This level is not considered a health risk, but a surrogate for human comfort. It could be possible to increase slightly the target value if the duration of exposure is limited.

1.3. HVAC Energy Management in Electric Vehicles

Thermal comfort is an essential service in cars. The efficiency of electric motors results in low thermal losses. The waste heat is not sufficient to heat the cabin. Another issue is the high-energy demands of electric actuators. Thermistors can only convert electrical energy into thermal energy. Even though they are efficient (almost a ratio of one), the fast

demands for thermal comfort require several kilowatts. A study [23] demonstrates that the energy required to cool the cabin of electric vehicles can reduce the driving range by 35% depending on the weather conditions and battery capacity.

Other alternatives, such as the reversible heat pump, exist to reduce the energy consumption of this service. This solution is developed for the Renault ZOE car (Figure 3). An important advantage of this system is that it can provide more thermal power than the consumed electrical power. This is possible because of the ability to transfer thermal energy from the environment. The ZOE is Renault's first production electric car to use a reversible heat pump [24]. The Renault ZOE is equipped with a "direct/direct" or "air/air" type of heat pump, as shown in Figure 3. In heating mode, the system extracts heat from the external environment and transfers it into the vehicle's interior, even in low temperatures. The refrigerant begins as a low-pressure gas, which is compressed to a high-pressure, high-temperature state. It then passes through the interior condenser, where heat is transferred to the cabin air. The refrigerant, now cooler but still under high pressure, moves through an expansion valve, causing a drop in pressure and temperature, turning it into a cold liquid. This cold refrigerant absorbs heat from the outside air in the exterior heat exchanger, evaporating back into a gas, and the cycle repeats to maintain cabin temperature.

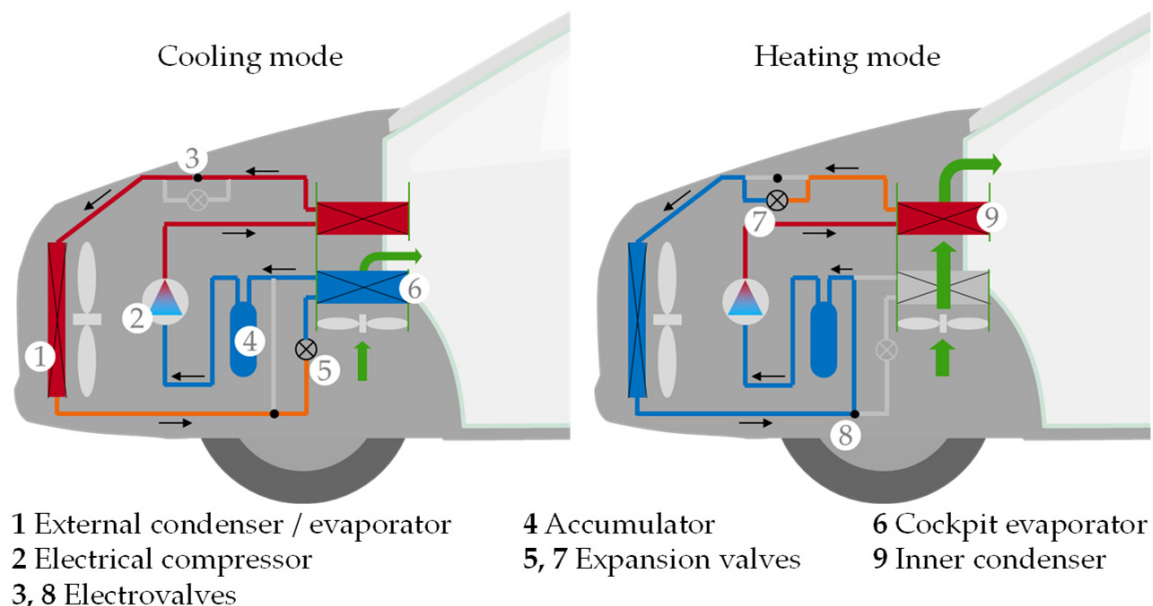


Figure 3. Renault ZOE reversible heat pump, adapted from [24].

2. The Building of a 1D-Simulation Tool

A 1D-simulation multi-physic model is developed to evaluate the HVAC system in a car cabin (Figure 4). It is based on numerous individual elements connected to describe the complete system. The GT-Suite v2024 software provide predictive correlations and literature models. The holistic approach focuses at a macro scale level. In that regard, the model must be calibrated and validated with experiments. The input model data is fulfilled to match the characteristics of a Renault ZOE (eco2 phase 1). It is an electric car (65 kW), with a cabin volume of 2.5 m³. In the scope of this project, different test campaigns (around two dozen) are performed. The results are used to calibrate each sub-component (cabin, heat exchanger, compressor. . .) and for the global validation of the model.

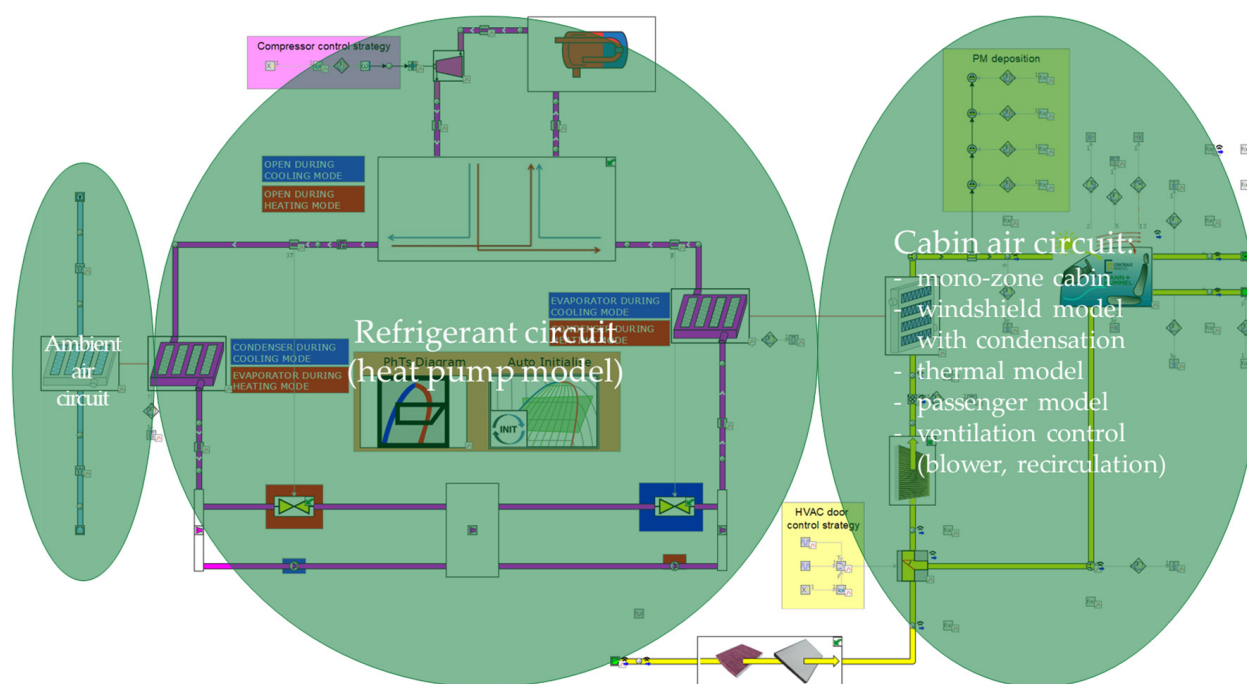


Figure 4. Overview of the complete simulation model in GT-Suite.

2.1. Heat Pump Model and Cabin Air Circuit

The heat pump circuit of the model (i.e., the left part of Figure 4) contains several components: a compressor, an accumulator, two heat exchangers, a four-way switch, two expansion valves, and some piping elements, while the refrigerant working fluid is R134a. This part of the model is described in a previous article [25].

The cabin air circuit, depicted on the right side of Figure 4, includes the following components: the recirculation flap, the blower, the evaporator or condenser (depending on the scenario), filter elements, the cabin model, and various piping elements. Air from the external environment enters the system downstream of the recirculation flap, which functions as a three-way valve, regulating the amount of air sourced from either the outside environment or recirculated from the cabin. The blower drives airflow through the pipes, targeting a user-defined volumetric flow rate in its branch. Subsequently, the air is either heated or cooled in the heat exchanger connected to the heat pump circuit before entering the cabin. Air exits the cabin via either the recirculation path or the outside environment. Additionally, an infiltration inlet is connected to the cabin element to represent air entering directly from outside the vehicle. This unfiltered and unconditioned air can significantly impact the thermal balance and air quality within the cabin. A previous study describes the modelling and calibration of this infiltration flow rate [1].

The cabin is modelled as a single-zone volume using the generic GT-Suite module to compute the thermal balance within a medium-sized vehicle cabin. Modifications have been made to include additional humidity calculations, condensation formation on glass surfaces, and air infiltration. The mono-zone model assumes complete homogeneity within the cabin, which may not fully capture localised differences near air vents for example. This approach, as opposed to using multiple volumes or computational fluid dynamics, is favoured due to its fast computational time and minimal input requirements. It is suitable for the current study, which has its focus primarily on evaluating global energy consumption and air quality trends rather than localised variations within the cabin. GT-Suite provides a template that is calibrated for medium-sized vehicles, such as the Renault ZOE.

There is a thermal interaction between the air circulating within the cabin, the internal cabin elements, and external environmental factors such as sunlight and wind. The modelling of the cabin involves numerous processes, as illustrated in Figure 5, and they are also described in a previous study [25].

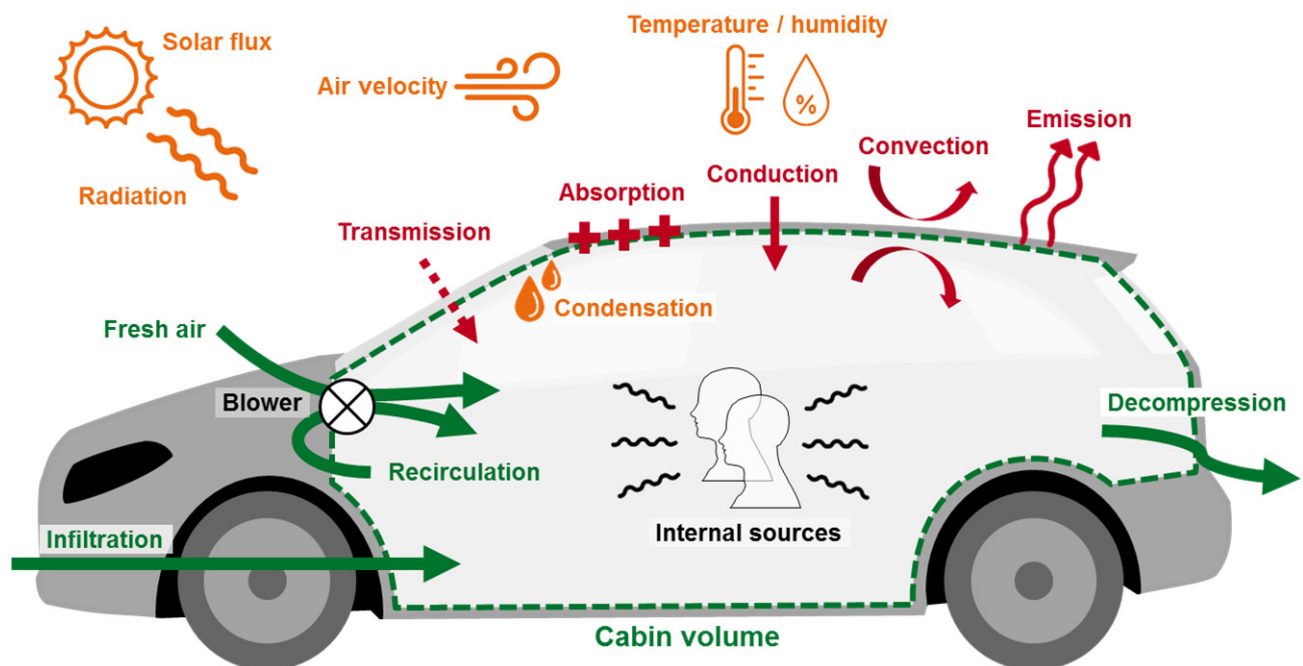


Figure 5. Modelling the energy balance of a car cabin in GT-Suite [25].

The thermal balance of the system is calculated assuming a homogeneous temperature in the cabin (mono-zone model). For the three heat transfer modes (convection, conduction, and radiation), the thermal flux φ (W) is calculated by multiplying the flux density J (W/m²) and the surface area A_s (m²) according to the following Equation (1):

$$\varphi = J A_s \quad (1)$$

The convective heat transfer coefficient h is estimated using the literature correlations [26,27] or calibration data if available. The wall temperature is taken on the side of the convection (e.g., outside wall temperature for convection with exterior air). In the cabin model, the conduction phenomenon is the transfer of energy within solid materials (i.e., the cabin walls). Each material layer is characterised by its thermal conductivity coefficient. Doors, and the roof and floor are discretized in three layers made of polyamide, polyurethane, and stainless steel. The materials' mass and geometry (thickness and surface area) are adjusted to match the characteristics of a Renault ZOE. A standard vehicle glass material is selected for the windshield, side windows, and rear window. Details on the material properties are given in Table 1. The solar view factor coefficient describes the portion of solar flux reaching each part of the cabin. Finally, the radiative heat transfer depicts the electromagnetic radiations of the system. A grey-body modelling is assumed (with absorptivity), described by the Stefan–Boltzmann's equation.

Table 1. Cabin model decomposition and material characteristics (* thermal conductivities are temperature dependent).

	Material	Total Mass	Total Surface Area	Thickness	Thermal Conductivity at 300 K *	Solar View Factor
	-	kg	m ²	mm	W/m/K	1 = Sun Overhead
Door inside	Polyamide 66	20	3.5	27	0.33	0.1
Door insulation	Polyurethane			126	0.025	
Door outside	Stainless steel			27	14.9	
Floor inside	Polyamide 66	10	3.5	24	0.33	-
Floor insulation	Polyurethane			112	0.025	
Floor outside	Stainless steel			24	14.9	
Roof inside	Polyamide 66	30	2.0	7.5	0.33	1.0
Roof insulation	Polyurethane			35	0.025	
Roof outside	Stainless steel			7.5	14.9	
Front windshield	Glass	20	1.0	5	1.4	0.7
Side windows	Glass	30	1.0	5	1.4	0.1
Rear window	Glass	10	0.7	5	1.4	0.5
Interior materials (dash, seats...)	Lumped material	30	6.0	-	5.0	-

To simplify the model, wind effects are not considered, meaning the air velocity on the windshield is assumed equal to the vehicle speed. For the windshield, the external convection coefficient is calculated using the Nusselt number Equation (2) [25].

$$Nu = \frac{h \cdot L_c}{\lambda} \quad (2)$$

The Nusselt number Nu can be calculated using correlations that depend on the flow and the surface considered. It involves the calculation of other non-dimensional numbers (Reynolds Re , Prandtl Pr). In the case of forced convection (as on the windshield), the correlations are the Equations (3) and (4), in which the flow is considered turbulent if $Re > 5 \cdot 10^{-5}$ [28]:

$$\text{Laminar flow : } Nu = 0.664 \cdot Re^{\frac{1}{2}} \cdot Pr^{\frac{1}{3}} \quad (3)$$

$$\text{Turbulent flow : } Nu = \left(0.037 \cdot Re^{\frac{4}{5}} - 871 \right) \cdot Pr^{\frac{1}{3}} \quad (4)$$

This enhanced convection heat transfer model provides a more accurate prediction of the windshield surface temperature, which is crucial for assessing the condensation risk and determining the overall cabin temperature.

The human body is a non-negligible source of heat, CO₂, and humidity. The exhalation flow rate from the occupants (driver and passengers) is simulated at the following rates:

- CO₂ mass flow rate = 32.6 g/h
- H₂O mass flow rate = 30 g/h under cold conditions (below 10 °C), 50 to 65 g/h under standard conditions (20 °C and 50% of relative humidity), and 100 g/h under tropical conditions (above 30 °C and 65% of humidity)

These values represent averages for one human adult engaged in normal seated activities (e.g., office work, reading, driving...) [12,29–31]. The values remain constant

for the duration of the simulation and do not vary with changes in cabin temperature or humidity.

Finally, occupants are a source of heat. A standard 75 W/person value is kept in the model as it is in accordance with the several literature references for a cabin passenger model [32–34]. This value is used in the calculations of the global cabin heat transfer balance as a heat source term. An additional source 100 W is taken into account in order to represent the heat coming from the electrical motor and the dashboard auxiliaries, as suggested by GT-Suite. This value is an estimated average based on typical thermal loads observed in electric vehicles during normal operation, including heat dissipation from the power electronics and cabin lighting. While the precise heat generation may vary depending on vehicle operational conditions, the chosen value provides a reasonable approximation for modelling purposes.

2.2. Model Validation on a Renault ZOE in a Cold Environment (Heating Mode)

In order to validate the model, the results of an experimental test and a simulation are compared. The test is a heating process in a $-19\text{ }^{\circ}\text{C}$ environment during 40 min. This duration allows the complete stabilisation of the cabin temperature. The Renault ZOE is in its original configuration. Tests are performed in a climatic chamber with a large fan in front of vehicle to represent the wind resistance. No solar flux is considered (indoor test). One operator is in the driver seat for the duration of the experiment. There is uncertainty on the heat released from this occupant. As the simulation model is adapted to match the experimental setup, three cases are considered with 0.66, 1, and 1.33 occupants in the cabin. The sensible heat released from occupants is respectively 50 W, 75 W, and 100 W. It allows an error margin for this unknown parameter. The Renault ZOE HVAC system is placed in automatic mode. The resulting blower flow rate and the recirculation ratio are recorded and injected in the simulation. These curves are presented in Figure 6 along with the vehicle speed scenario and the ambient temperature. Given the zero vehicle velocity and the relatively high fresh air flow rate, it is assumed that there are no infiltrations in the cabin. This assumption is advocated in a previous study [1].

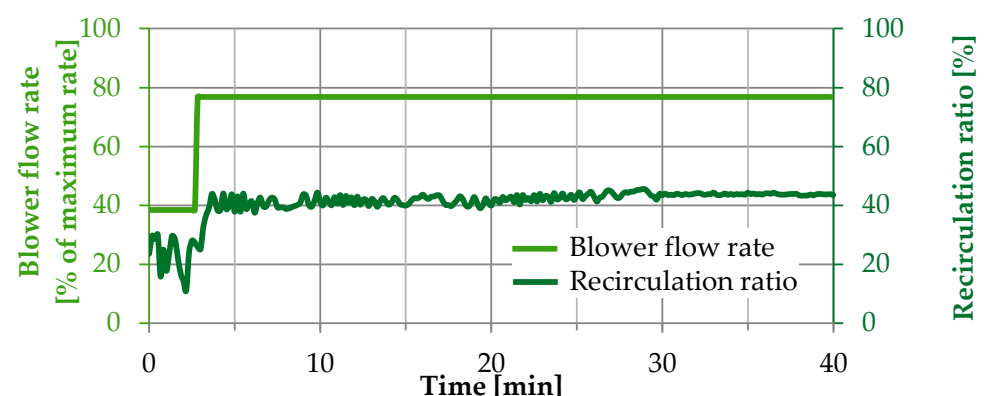


Figure 6. Input data for the simulation (supply air flow rate and recirculation ratio).

In the experiment, the cabin temperature is recorded at two locations: the front passenger seat at the feet and the front passenger seat at the head. It should be noted that the ventilation grid system is arranged to blow only through the bottom grids (near the front passengers' feet). All of the other ventilation grids (central, windshield, and sides) are closed. In the model, the mono-zone assumption cannot differentiate the two locations of temperature recordings. However, the head temperature is selected as the main data to represent the spatial average cabin temperature. Indeed, the feet temperature is closer to a supply air temperature than an estimation of the average cabin temperature. The initial

cabin temperature in the simulation is set at the test front head value (-11.8°C). Finally, in the model, the heat pump compressor strategy (PID regulator) is tuned to match the experiment results. The simulation results are compared with the test results in Figure 7.

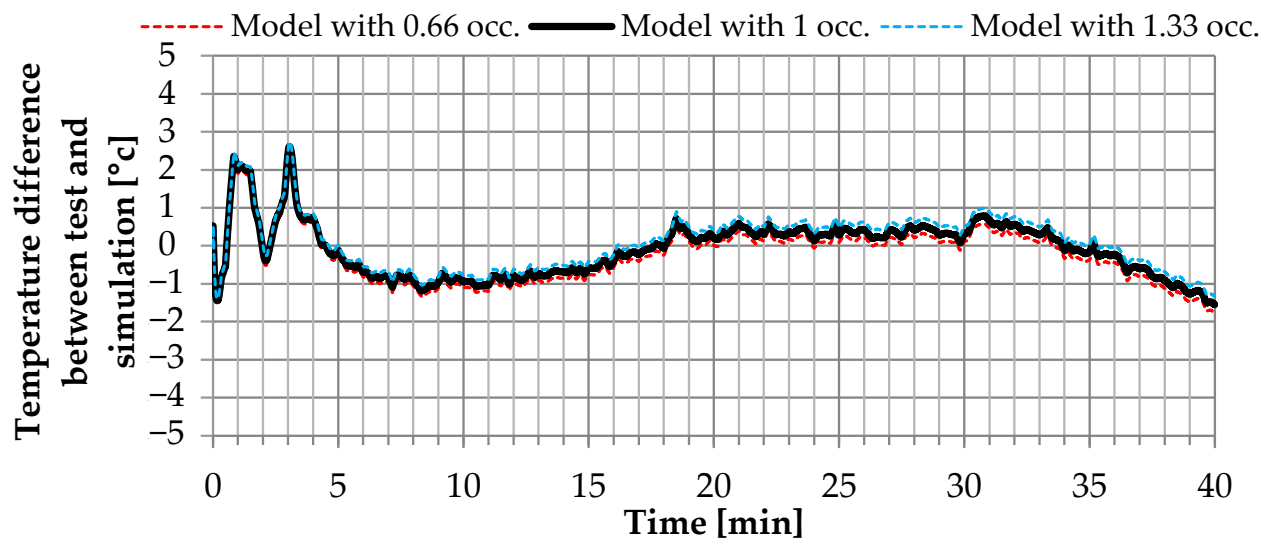


Figure 7. A comparison between the experiment and simulation of a cabin temperature rise in a cold climate.

Three simulation curves are displayed depending on the number of occupants. From 0.66 to 1.33, the curve differences are minor ($+0.5^{\circ}\text{C}$ at best): the thermal output from the test operator has a negligible effect. The main differences from test to simulation are at the beginning of the scenario in the initial temperature rise. The shift of the recirculation ratio at 3 min is visible on the simulation curve. In the test, the shift is only visible on the front feet sensor. Afterward, the maximum temperature differences are -1.3°C at 8 min, $+0.7^{\circ}\text{C}$ at 30 min, and -1.7°C at the end of the scenario (zero speed) when the cabin temperature reaches 22°C . Overall, the correlation is acceptable. While the mono-zone modelling is not designed for perfect curve matching, it accurately predicts the cabin heat-up process under cold ambient conditions. The cabin cool down in hot environment is not experimented. It cannot be directly validated, but only indirectly through the validation of the heat pump model in cold conditions.

3. Evaluation of the Energy Consumption and Thermal Comfort with Numerical Simulations

One key challenge for e-mobility is to increase the driving range. The heating and cooling of the cabin drains energy from the battery, which cannot be used for the powertrain of the vehicle. The objective is to investigate the power consumption of the HVAC system in order to achieve a thermal comfort in the cabin. The idea is to find solutions to minimise this energy consumption, and thus increase the car driving range. Different scenarios and configurations are implemented in the simulation tool, with particular interest on an opti- CO_2 mode. Modifications and improvements on the heat pump system (compressor, heat exchangers. . .) are not considered. This study focuses on the ventilation system.

3.1. Simulation Settings

Three scenarios are considered with different ambient conditions (Table 2). These scenarios represent typical conditions of cold, medium, and hot environments. In the COLD case, the HVAC system runs in heating mode while in the other cases it is in cooling mode. In the MEDIUM scenario, although the outside and targeted temperatures are the

same (19 °C), the system is in cooling mode due to the heat input from passengers and solar radiation. The cabin temperature target is 19 °C in COLD and MEDIUM scenarios, and 25 °C in a HOT scenario. The outside CO₂ concentration is fixed at 400 ppm. It is assumed that the initial state of the cabin is the same as the outside conditions.

Table 2. Three ambient settings for simulation.

		COLD	MEDIUM	HOT
Ambient pressure	bar		1.01325	
Ambient temperature	°C	−15	19	30
Relative humidity	%	85%	40%	70%
Mixing ratio	g/kg _{dry air}	1.0	5.4	18.8
Ambient CO ₂ concentration	ppm (%)		400 (0.04%)	
Solar flux	W/m ²	250	500	750
Water emission per person	g/h	30	65	100
Sensible heat emission per person	W		75	
CO ₂ emission per person	L/h (g/h)		18.8 (32.6)	
Number of occupants	-	2 (unless specified otherwise)		
Road cycle	-	WLTC (duration = 1800 s)		
Blower flow rate	m ³ /h		200	
Fresh air ratio	%	Fixed (0–50–80–100%) or variable (opti-CO ₂ mode)		

The blower is controlled to supply a fixed 200 m³/h total air flux going through the 2.5 m³ cabin volume. It corresponds to a medium blower position (four out of eight on the Renault ZOE), and can be considered as common usage. The simulation is run on WLTC (Worldwide harmonised Light-duty vehicles Test Cycles), which has a total duration of 1800 s (30 min). It is sufficient for a temperature stabilisation inside the cabin.

The purpose of the simulation is to evaluate the possible energy gains with this system by varying the HVAC recirculation flap configuration. For each ambient scenario, five settings of the recirculation flap are evaluated. In four configurations, the ratio is fixed at 0%, 50%, 80%, or 100% of fresh air. For instance, with a position set at 80%, the proportion of outside air and recirculated air going through the cabin is respectively 80% and 20%. In the fifth setting, the recirculation flap is set in opti-CO₂ mode. The fresh air ratio is controlled in order to manage a cabin CO₂ concentration of 1100 ppm. Simulation results show that with the opti-CO₂ mode, the steady-state amount of fresh air required to stabilise the CO₂ level at 1100 ppm depends on the number of occupants in the vehicle (Table 3). For example, with four occupants, the air exchange rate is satisfactory with 79% of outside air, while it can be decreased down to 21% with one occupant. These values are only indicators. They depend on the blower flow rate, the outside CO₂ concentration, and the breathing rate of occupants. In addition, these stabilised values are obtained in case of no infiltration. In case of infiltration, there is an additional fresh air flow rate entering the cabin. In the United Kingdom, since 2002, the yearly average occupancy in a 4-wheeled car is 1.6 occupants per vehicle [35]. This figure drops to 1.2 occupants regarding business and commuting trips. For the current study, two occupants are considered in the vehicle.

Table 3. The amount of fresh air required to stabilise the CO₂ concentration at 1100 ppm, with a blower flow rate of 200 m³/h and assuming no infiltration.

Number of Occupants	Stabilised Vol. Ratio of Fresh Air in Opti-CO ₂ Mode
1	21%
1.2	24%
1.6	31%
2	39%
3	58%
4	79%
5	98%

There are 15 cases implemented in the simulation model: five recirculation flap settings and three ambient scenarios. Several variables are monitored to evaluate the comfort, like humidity (relative and absolute), temperature, or CO₂ concentration. Energy savings are assessed through the power consumption of the compressor in the refrigerant fluid circuit, and at the heat exchanger on the cabin side.

3.2. Thermal Comfort Analysis

At first, the analysis starts with the air fluxes in the system. The results regarding the airflow repartition of the five configurations are displayed in Figure 8. Only the COLD scenario is displayed in this figure as an example. Similar results are obtained with the MEDIUM and HOT scenarios. The opti-CO₂ configuration manages the fresh air flow rate just above 50 m³/h in the regulation stage. This flow rate is not steady because of the infiltration, which depends on the vehicle speed. The infiltration flow rate varies around 10 m³/h at the beginning of the cycle, and around 20 m³/h on the second half of the cycle. This inlet of outside air inside the cabin allows the opti-CO₂ control to decrease the fresh air inlet by opening the recirculation flap. On the 100% fresh air configuration there is no infiltration, as well as in the 80% fresh air case. In these cases, the amount of fresh air delivered by the blower is enough to create a positive pressure to prevent infiltration. It should be noted that the air entering the cabin by infiltration has the same properties as the exterior air: the same temperature, humidity level, and CO₂ concentration.

Regarding the CO₂ aspect in the cabin, the results of the five configurations of the COLD scenario are displayed in Figure 9. From 50% to 100% of fresh air, the CO₂ level inside the cabin stays below 1000 ppm. The amount of outside air driven into the cabin is sufficient to keep a safe environment for the two occupants. In opti-CO₂ mode, the control of the recirculation flap leads to an expected concentration of 1100 ppm. During the first 134 s of the cycle, the flap is completely opened to let the CO₂ concentration rise to the limit value. Then, the amount of fresh air is stabilised around 30%. This value is lower than the 39% of Table 3 because of the infiltration flow rate. The opti-CO₂ PID settings allow a straight stabilisation of the CO₂ level within the cabin. Finally, in the 0% fresh air configuration, the CO₂ concentration rises up steadily to high levels. After 10 min, the concentration fluctuates between 3000 and 3500 ppm until the end of the cycle. The fluctuations are the results of the infiltration flow rate. This is not a sustainable level regarding the comfort and safety of the occupants.

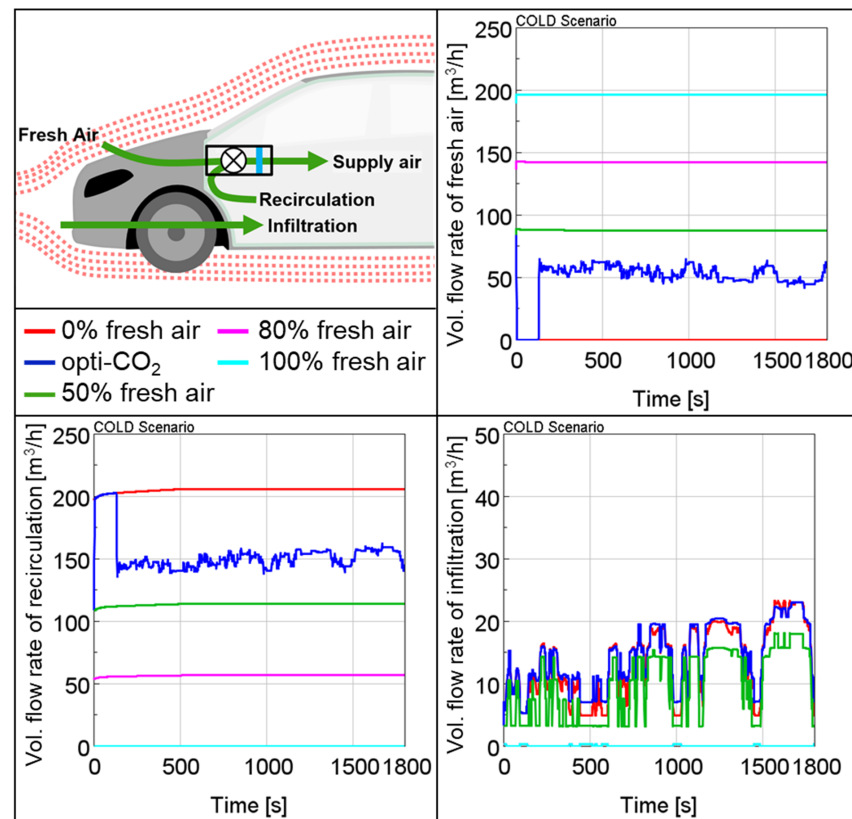


Figure 8. Airflow repartition for the five recirculation flap configurations of the COLD scenario.

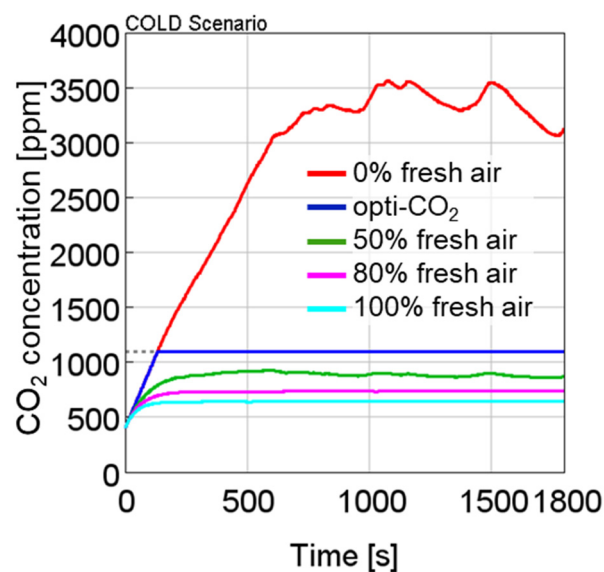


Figure 9. Cabin CO_2 concentration for the five configurations of the COLD scenario.

Another aspect of the passengers' comfort is the temperature and humidity levels in the cabin. The temperature curves of the COLD scenario are displayed in Figure 10 along with the curves of the opti- CO_2 configuration for the three scenarios. The heating of the cabin is faster in recirculation mode despite the infiltration, which brings cold outside air into the cabin. At 100% fresh air, the system takes 820 s to reach the 19°C target, while it takes 500 s in the 0% fresh air case. Contrary to the CO_2 aspect, the occupants benefit from a full recirculation mode. This is also verified for the MEDIUM and HOT cases, but at a lower amplitude since it is globally faster to reach the target temperature in these scenarios. As an example, the right-side chart of Figure 10 presents the opti- CO_2 configuration of the

three scenarios. While it takes 550 s to reach the temperature target in the COLD scenario, it is reduced down to 40 s in MEDIUM and HOT cases. Then, the thermal comfort of the occupants is reached quicker.

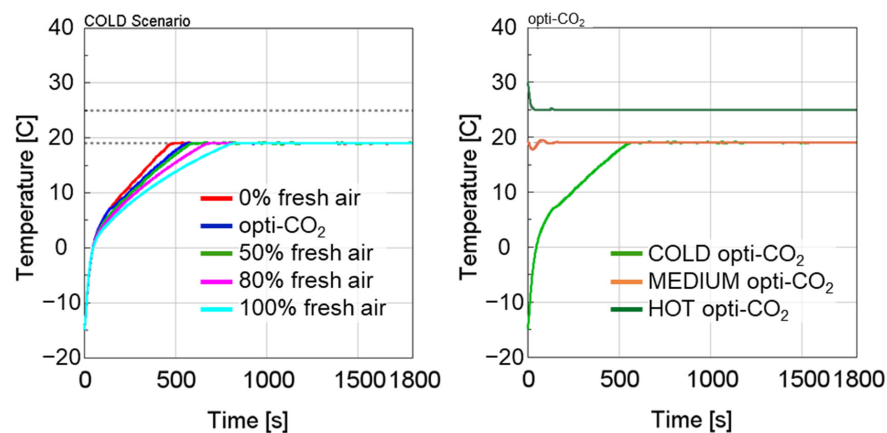


Figure 10. Temperature curves of the COLD scenario (left) and for the opti-CO₂ configuration (right).

Regarding the humidity aspect, there is no specific control made in order to keep the level in the comfort area (between 30% and 65% of the relative humidity and below an absolute humidity of 12 g_{water}/m³). The cabin humidity levels presented in Figure 11 are only a consequence of the thermal management. Globally, there are only few differences comparing the five recirculation flap configurations.

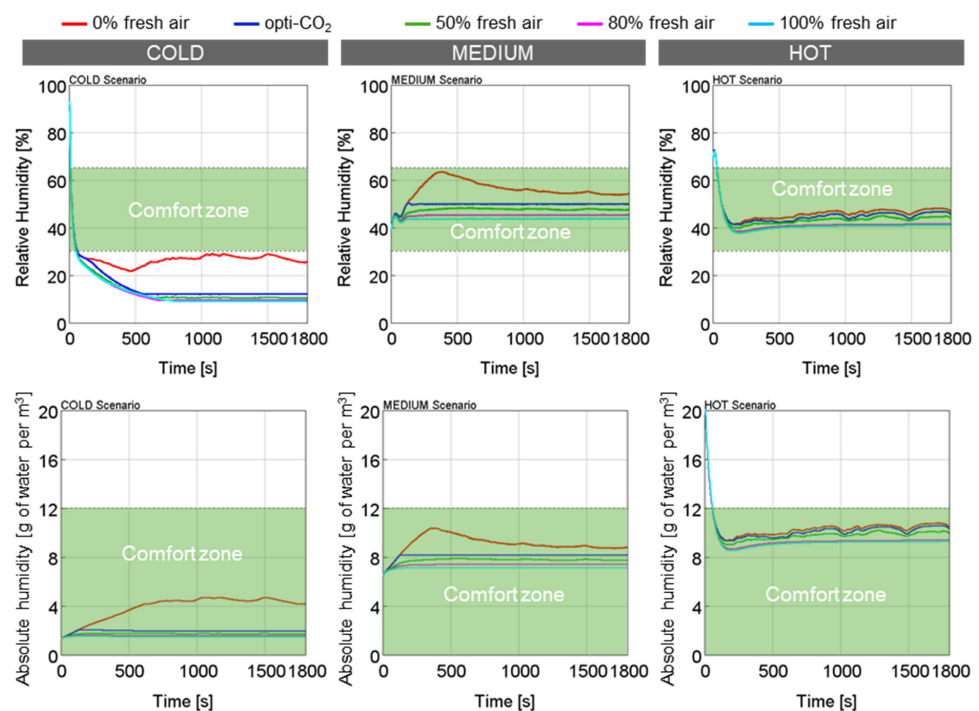


Figure 11. Relative humidity (top row) and absolute humidity (bottom row) on the three scenarios.

In the COLD scenario, the relative humidity is lower than the comfort area. The reason is that the absolute humidity in the outside environment is very low (due to the low temperature). In the process, only a small amount of water is added into the system (from the occupants). Then, by heating up the air inside the cabin, the relative humidity is understandably low. In the MEDIUM scenario, the outside temperature is the same as the cabin target. Then, humidity variations in the cabin are the result of the occupant moisture.

The effect is more important as the recirculation percentage increases. Nevertheless, both the relative and absolute humidity are kept within acceptable comfort limits. Finally, in the HOT scenario, the outside air is loaded with water, such as in a tropical environment. The absolute humidity is near $20 \text{ g}_{\text{water}}/\text{m}^3$. The heat exchanger (evaporator) at the cabin side decreases the air temperature in order to achieve the 25°C target. In that process, the water vapour in the air condenses. The amount of condensate retrieved at the heat exchanger outlet varies from 1.2 L (100% fresh air) to 2.1 L (0% fresh air). At the heat exchanger outlet, the air is saturated in water (100% relative humidity). As the air is reheated in the cabin, it allows for a decreasing of the relative humidity down to comfortable levels, between 40% and 50%, for this scenario.

Alongside the comfort, an important impact of the humidity level in the cabin is the fog formation on the glass surfaces. This is a safety aspect for the driver. In all 15 cases, there is only one in which fog appears on the front windshield: 0% fresh air in the COLD environment. Indeed, in HOT and MEDIUM scenarios, the temperature of the glass surfaces is high enough to avoid fogging. In the COLD scenario, the glass temperature is low because of the outside environment. It creates a cold cluster favourable to the formation of condensate. In the 0% fresh air configuration, the absolute humidity in the cabin ($4 \text{ g}_{\text{water}}/\text{m}^3$) is twice as much as in the other four configurations of the same scenario ($2 \text{ g}_{\text{water}}/\text{m}^3$). This difference is sufficient to lead to the formation of fog on the all-glass surfaces on the cabin side (Figure 12). The rear windshield is the least impacted surface with a peak of condensate estimated at 0.2 g at the beginning of the cycle. On the side windows, the peak is obtained after one third of the cycle and is much more important (1.1 g). The glass surface that is the most impacted by fog formation is also the most important for the safety aspect: the front windshield. On this surface, the condensation is not well managed since it increases during the whole cycle, reaching a peak of 1.1 g at the end. This quantity is relatively high. Such a level during the major part of the cycle would certainly prevent a person from driving the car. For this specific case, solutions can be to dry the surface manually, or to increase the amount of fresh air driven by the blower. However, according to the charts of Figure 11, it is difficult to control the cabin humidity only with the recirculation ratio. Another solution could be the implementation of a humidity absorber in the cabin.

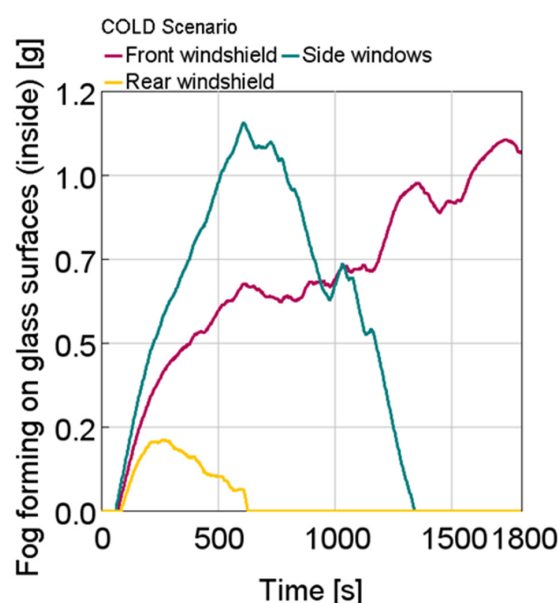


Figure 12. Condensation formation on glass surfaces (cabin side) in the 0% fresh air configuration of the COLD scenario.

Overall, cabin comfort is achieved in most cases. The thermal conditions are satisfactory across all scenarios; however, the target comfort temperature is not reached promptly in the COLD scenario. The recirculation mode should be avoided due to the CO₂ constraint, although other recirculation configurations appear to be safe. Humidity within the cabin is not regulated. In the HOT and MEDIUM scenarios, this is not problematic, as both absolute and relative humidity levels fall within the comfort range. In the COLD scenario, relative humidity inside the cabin is below the comfort threshold. Nonetheless, increasing humidity within the cabin is not advisable due to the risk of condensation forming on glass surfaces, which remain very cold due to external temperatures. This issue is particularly evident in the 0% fresh air configuration.

3.3. Power Balance Analysis

The second part of the analysis is from a power consumption point of view. In this subsection, the analysis is made over all the cycle durations. The power values presented hereafter are averaged during the whole WLTC.

As a first example, the HOT scenario is taken, with the 50% fresh air case. According to the simulation, on WLTC, the average power required to cool down the cabin is 2.58 kW. This is the power at the heat exchanger on the cabin side (evaporator), taken from the air and absorbed by the refrigerant fluid. In the refrigerant loop, the compressor consumes a 1.92 kW power. The whole compressor power is not transmitted to the refrigerant fluid. The mechanical friction of the shaft is taken into account. In addition, an isentropic efficiency is applied to represent the efficiency of the compressor at a given operating point. In this case, the work transmitted to the fluid is 1.00 kW (52% of the consumed power). Finally, the energy balance is completed at the heat exchanger on the outside air side (condenser), where a 3.58 kW work is removed from the refrigerant fluid to the outside air. This power is the sum of the power at cabin side and the fluid work in the compressor.

From an energy-saving point of view, the aim is to reduce the compressor power since it is directly taken from the car battery. Although there are possibilities to decrease the power consumption by improving the compressor features, these are not evaluated in the scope of this study. Similarly, the features of the heat exchangers are not evaluated. The objective is to reduce the need for cooling at the evaporator (cabin side). As a result, the compressor power would decrease as well. At a given cabin temperature target, the solution is to decrease the air exchange rate. Without the infiltration consideration, this is achieved by increasing the recirculation ratio. Then, the simulation tool can give an estimation of the amplitude of the gain. The five cases of the HOT scenario are displayed in Table 4.

Table 4. HOT scenario power balance.

Recirculation Flap Setting	100% Fresh Air	80% Fresh Air	50% Fresh Air	Opti-CO ₂	0% Fresh Air
Compressor power [W]	2810	2280	1920	1740	1360
Blower power [W]	77	74	67	66	66
Convective heat transfer at cabin side [W]	3650	3070	2580	2340	1740
Convective heat transfer at outside air side [W]	5140	4220	3580	3290	2420
Energy Efficiency Ratio (EER _{HOT/MEDIUM})	1.30	1.35	1.34	1.34	1.28

The compressor power decreases by 1.45 kW from full fresh air mode to full recirculation mode. This gain is explained by a large decrease in the convective heat transfer at the cabin side. It should be acknowledged that the gain at the heat exchanger at the cabin side (1.91 kW) is not fully translated into compressor gain (1.45 kW). The energy efficiency ratio ($EER_{HOT/MEDIUM}$) remains stable. The EER and COP (Coefficient of Performance) of the heat pump system are defined as the ratio of useful heat (supplied or removed) against the consumed work. They are calculated according to the Formulas (5) or (6) [36].

$$COP_{COLD} = \frac{\text{Power at condensor}}{\text{Power of compressor}} \quad (5)$$

$$EER_{HOT/MEDIUM} = \frac{\text{Power at evaporator}}{\text{Power of compressor}} \quad (6)$$

The blower power consumption also decreases as there is more recirculation, but the gain is negligible compared to the compressor gain. Indeed, an isentropic efficiency coefficient (0.5) is implemented to calculate the blower power and the losses that are being added to the fluid (thus effecting the outlet temperature). In fresh air mode, the air at the inlet of the blower is at ambient temperature. In recirculation mode, the air at the inlet of the blower quickly reaches the target temperature of the cabin. As a result, the average power of the blower during the whole scenario is lower in recirculation mode. Nevertheless, the difference remains negligible compared to the compressor power.

In the MEDIUM scenario, the analysis is similar than the HOT scenario, but with lower amplitude in the results: from full fresh air to full recirculation, the compressor power decreases from 670 W to 550 W. The results are displayed in Table 5. The $EER_{HOT/MEDIUM}$ is higher than in the HOT scenario, and remains stable over the five configurations of the recirculation ratio.

Table 5. MEDIUM scenario power balance.

Recirculation Flap Setting	100% Fresh Air	80% Fresh Air	50% Fresh Air	Opti-CO ₂	0% Fresh Air
Compressor power [W]	670	620	570	550	550
Blower power [W]	78	74	66	64	62
Convective heat transfer at cabin side [W]	1160	1080	990	930	900
Convective heat transfer at outside air side [W]	1430	1320	1220	1150	1130
Coefficient of performance ($EER_{HOT/MEDIUM}$)	1.74	1.74	1.74	1.73	1.62

In the COLD scenario, the heat pump system is in heating mode (contrary to MEDIUM and HOT scenarios). The results of the COLD scenario are displayed in Table 6. In this scenario, the compressor is used at its maximum speed for a large part of the road cycle. In fresh air mode, the average power consumed by the heat pump is 2.97 kW. In recirculation mode, this value is decreased to 2.31 kW, which represents a gain of 660 W. This gain is relatively important, although it is much lower than in the HOT ambient scenario (1.45 kW). Regarding the COP_{COLD} , the values are in the range of 0.9–1.0 depending on the recirculation ratio. It means that the power at the condenser is almost equivalent to the power of the compressor.

Table 6. COLD scenario power balance.

Recirculation Flap Setting	100% Fresh Air	80% Fresh Air	50% Fresh Air	Opti-CO ₂	0% Fresh Air
Compressor power [W]	2970	2750	2650	2550	2310
Blower power [W]	90	85	76	74	68
Convective heat transfer at cabin side [W]	3020	2720	2530	2360	2070
Convective heat transfer at outside air side [W]	1640	1490	1350	1250	1050
Coefficient of performance (COP _{COLD})	1.02	0.99	0.95	0.93	0.90

The consequences on the range of an electric vehicle can be coarsely estimated with Equation (7) [37].

$$Range_{loss} = Range_{max} - \frac{Range_{max}}{1 + \frac{Spec_{range} * Power}{Speed}} \quad (7)$$

The maximum range of the Renault ZOE is rounded at 300 km, the specified range at 6.6 km/kWh, and the average speed on WLTC is 47 km/h. With this calculation, it is possible to estimate the impact on the driving range depending on the recirculation flap configuration. The power consumption of the compressor and the resulting impact on the driving range are presented in Figure 13 for the three ambient scenarios. The “range gain” chart is calculated by subtracting the $Range_{loss}$ of the given configuration to the common 100% fresh air case. This configuration is used as the worst-case reference. The range gain in the other fresh air configurations represents the improvement from this common 100% fresh air configuration.

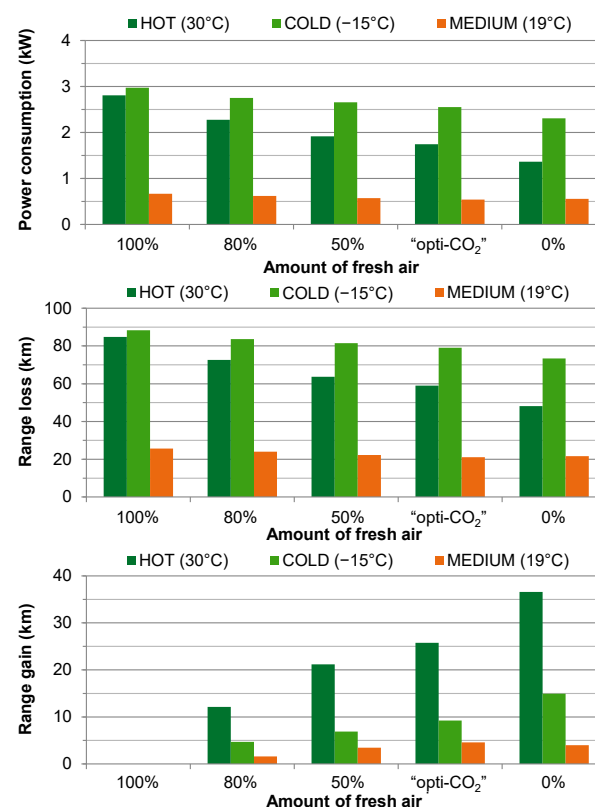


Figure 13. The power consumption needed for thermal comfort on three ambient scenarios and the impact on the driving range of an electrical vehicle.

Eventually, for the three scenarios, the best optimisation between cabin comfort and energy savings is achieved by setting the HVAC flap position with the opti-CO₂ mode (between 22% and 37% of fresh air depending on the infiltration intensity on WLTC with two occupants). Compared with the common case “100% fresh air” the power gains of the compressor in COLD, MEDIUM, and HOT scenarios are, respectively, 14%, 19%, and 38%. These gains are very important because the overall power consumption of the heat pump is a major drawback for the driving range of the car. In the 100% fresh air configuration, the range loss is estimated at 88, 26, and 85 km in the COLD, MEDIUM, and HOT scenarios, respectively. In comparison to the maximum driving range (300 km), such numbers are important. In the opti-CO₂ mode, the gains on the driving range are estimated at 9, 5, and 26 km compared to the 100% fresh air case. It would be possible to increase the gains by decreasing the amount of fresh air, but at the expense of the CO₂ safety aspect, the other bio-effluents emitted by the human body and the infiltration phenomenon. A solution can be the implementation of an absorbent system in the cabin to regulate the CO₂ concentration.

3.4. Opti-CO₂ Strategy Investigations

The idea of the opti-CO₂ mode is to find a trade-off between a moderate energy consumption and safe air inside the cabin. In this mode, the recirculation flap is regulated between its two extreme positions in order to achieve a target CO₂ concentration in the vehicle. In the previous simulation results, this target was set at 1100 ppm. The regulation depends on the number of occupants. Table 3 gives an estimation of the stabilised recirculation ratio needed to achieve the 1100 ppm target for different numbers of occupants. Then, the results of Figure 13 can be used to estimate (approximately) the consequences on the energy consumption. This is an approximation because there is also variation due to the heat and humidity released by the occupants. For example, with four occupants in the vehicle, the recirculation ratio is stabilised at 21%, and the results of the 80% fresh air case can be used as an approximation. The main difference would be during the transient stage, when the recirculation flap is completely opened in the opti-CO₂ mode. The range gain with four occupants would be 5, 2, and 12 km for the three scenarios (COLD, MEDIUM, and HOT) compared to the full fresh air cases. This stresses out the influence of the number of occupants on the results: the opti-CO₂ mode is very relevant with a low number of occupants, but cannot provide a major benefit when the cabin contains its maximum number of passengers.

Another way to retrieve a driving range gain is to increase the target CO₂ concentration in the cabin. Indeed, there is no standard imposing a strict limit of CO₂ concentration in a vehicle. The main concerns are from a health and safety point of view. The literature review shows that there is no direct health issue below 5000 ppm. The concerns are more about the decrease in performance regarding some cognitive tests. The exposure time is not really discussed in the literature. Most labour codes across the globe give guidelines for long-term exposure (several hours) and are never below 5000 ppm. For short-term exposure, less than an hour, the limit is increased (up to 30,000 ppm). In 2012, the average daily journey of a European driver is from 5 to 8 km [38], and drops below 2 km for 40% of French users [39]. Although these figures could have changed over the years (hopefully they are on the rise), it represents very short travels (a few minutes). In that scenario, it could be acceptable to increase the opti-CO₂ limit up to 2000 or 3000 ppm given the limited time of exposure. The simulation tool is used to evaluate what would be the impact on energy savings.

Table 7 is an extension of previous Table 3, with the ratio of fresh air in three opti-CO₂ modes (target at 1100 ppm, 2000 ppm, and 3000 ppm). The ratios are obtained from the simulation results in the stabilised stage, and assuming no infiltration. As expected, the

increase in the CO₂ target leads to a lower amount of fresh air required. The fresh air ratio is decreased by at least 50% when the target goes from 1100 ppm to 2000 ppm, and by at least 70% going to 3000 ppm.

Table 7. Stabilisation of the CO₂ concentration, with a blower flow rate of 200 m³/h and assuming no infiltration.

Number of Occupants	Stabilised Volumetric Ratio of Fresh Air in Opti-CO ₂ Mode		
	Target = 1100 ppm	Target = 2000 ppm	Target = 3000 ppm
1	21%	10%	5%
1.2	24%	12%	7%
1.6	31%	15%	8%
2	39%	17%	12%
3	58%	25%	17%
4	79%	35%	21%
5	98%	43%	26%

A thorough example is run with the simulation tool: a case with two occupants and the CO₂ target at 2000 ppm. The results of power consumption and the range gain estimations are added to the graph of Figure 14. Increasing the opti-CO₂ target from 1100 ppm to 2000 ppm gives an additional range gain in the three ambient scenarios (COLD, MEDIUM, and HOT): 8, 4, and 1 kilometres compared to the opti-CO₂ at 1100 ppm. These benefits must be weighed against the potential health risks caused by the higher CO₂ exposure. For instance, distinctions can be made between road professionals (such as cab and lorry drivers), commuters, and occasional drivers. Each application involves different cabin sizes, numbers of occupants, or durations of exposure. Additionally, a more sophisticated criterion could be implemented. In the simulation, the CO₂ limit is continuously fixed over time. A new control strategy could set a higher limit during steady-state operations, interrupted by regular periods of full fresh air intake. This strategy could be optimised by monitoring outside air conditions, ensuring that fresh air periods coincide with times when the outside air quality is high.

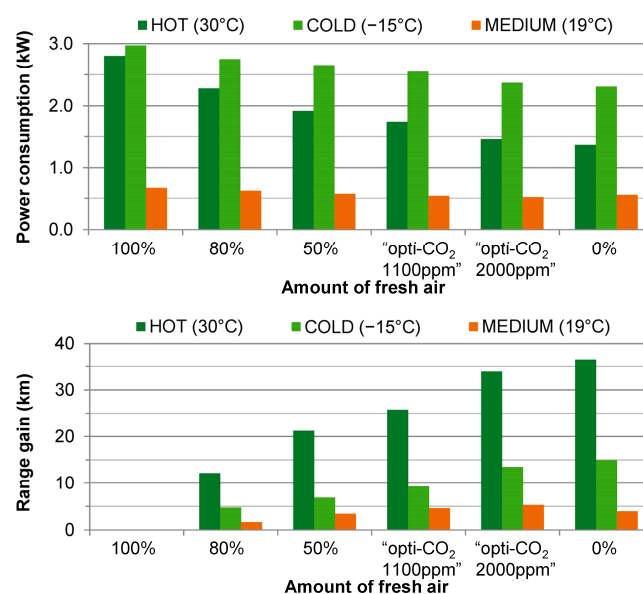


Figure 14. The power consumption (of compressor heat pump) needed for thermal comfort and the impact on the driving range of an electrical vehicle.

3.5. Influence Studies

The previous results were all done with a set of baseline parameters. Apart from the recirculation flap position, all of the parameters were fixed to reflect a common usage of a vehicle. In order to satisfy a rigorous analysis of the energy savings, it is important to study the influence of different elements. To simplify the process, the parameters are evaluated separately, meaning that only one parameter at a time is changed from the baseline configuration.

3.5.1. Influence of the Infiltration Model

Three configurations are compared, as follows: the models from the Renault ZOE and another thermal vehicle (Peugeot 208) detailed in a previous publication [1], and a theoretical model without any infiltration (“no infiltration”). It should be acknowledged that the only change is the infiltration model. For example, for the Peugeot 208 configuration, the potential heat released by the gasoline engine is not considered. All of the vehicle characteristics remain those of the Renault ZOE.

In all of the configurations, the blower flow rate is set at $200 \text{ m}^3/\text{h}$, which corresponds to a medium blower position: approximately position 4 (out of 8) for the Renault ZOE, and 3 (out of 6) for the Peugeot 208. In recirculation mode, this specific point of operation leads to a roughly twice more infiltration flow rate with the Renault ZOE compared to the Peugeot 208. Figure 15 displays the volumetric flow rate of infiltration for both vehicles, calculated by the simulation model. A total 154 simulations are run for each map: 11 recirculation flap positions and 14 vehicle speeds. The infiltration flow rate is taken in a steady-state stage, after a few seconds.

On WLTC, the blower drives 100 m^3 of air into the cabin during the whole cycle (30 min at $200 \text{ m}^3/\text{h}$), regardless of the configuration. Depending on the recirculation ratio, this volume is divided in two portions: recirculated air and fresh air. The infiltrated air is an additional inlet of air into the cabin, which depends on the recirculation ratio. The infiltration volume is at its maximum in the full recirculation mode: 7.7 m^3 with the Renault ZOE, and 3.3 m^3 with the Peugeot 208. For the full fresh air cases, and for the 80% fresh air cases, the infiltration flow rate is insignificant for both vehicles: null or less than 0.1 m^3 . In these configurations, the flow rate of fresh air driven by the blower prevents the infiltration phenomenon. For the intermediate case (50% fresh air), the infiltration volume of the Renault ZOE and the Peugeot 208 are, respectively, 4.8 m^3 and 2.3 m^3 . These volumes are based on the MEDIUM scenario results. While they may fluctuate slightly in COLD and HOT scenarios due to the changes in air density, the variations are minimal (less than 0.1 m^3).

The consequences on the compressor power consumption are displayed in Figure 16, for the three infiltration model configurations: Renault ZOE, Peugeot 208, and “no infiltration”. In the 100% and 80% fresh air cases, the power consumption over WLTC is the same since there are very few differences in the infiltration flow rate. In the 50% and 0% fresh air cases, there is a difference, but only for the HOT and COLD scenarios. In the MEDIUM cases, the infiltration does not appear to have an impact on the heat pump power consumption. Indeed, the temperature of infiltrated air is the same as the target temperature (19°C). In that scenario, and strictly from an energy point of view, the infiltration is helpful in order to maintain the temperature, as the heat pump is in cooling mode, in order to oppose the heat released by the occupants and the solar flux. However, this benefit is marginal. At best, it leads to an only 0.02 kW power decrease in the 0% fresh air case, compared to the “no infiltration” configuration. This translates into a 0.5 km driving range gain. The MEDIUM scenario is the only one in which infiltration is helpful.

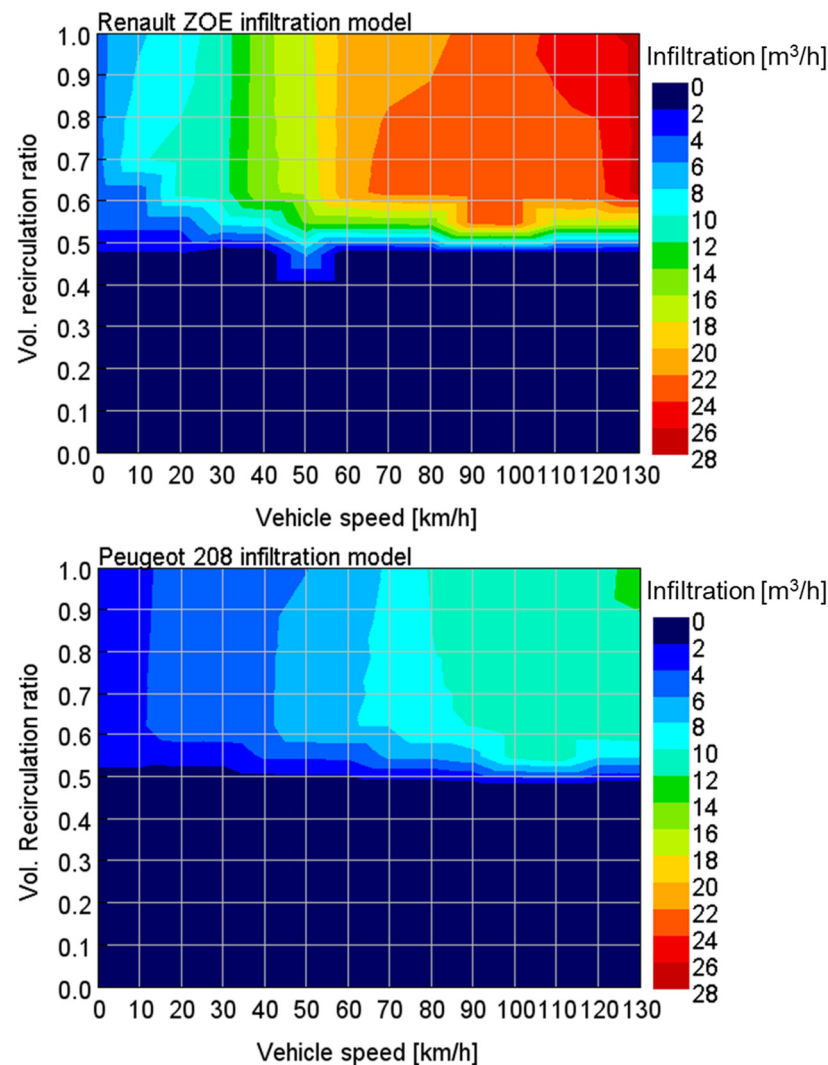


Figure 15. Infiltration volumetric flow rate for the Renault ZOE and the Peugeot 208, with a blower flow rate of $200 \text{ m}^3/\text{h}$.

In the HOT and COLD scenarios, the infiltration is a disadvantage. It introduces air that is at extreme ambient temperature into a cabin that is at comfort level. The heat pump must provide more work to maintain this comfort level. The 0% fresh air case can highlight this over-consumption due to infiltration. From the Renault ZOE to the Peugeot 208, the power consumption decreases by 0.07 kW in the HOT scenario, and 0.11 kW in the COLD one. In the “no infiltration” configuration, these figures are more important, respectively, at 0.13 kW and 0.20 kW. The potential gains on the driving range can be extracted from Figure 16. In the HOT scenario, the benefit of the recirculation mode compared to the fresh air mode can increase from 37 to 41 km by removing the infiltration phenomenon. In the COLD scenario, the benefit increases from 15 to 20 km.

These 4 and 5 km bonuses of a driving range can be seen as significant for an electric vehicle. From an energy perspective, the removal of infiltration is globally efficient. The marginal benefits for a specific point of operation (MEDIUM cases) are overcome by any other scenario. However, the infiltration phenomenon can be difficult to eliminate. The fresh air mode can be used to create positive pressure in the cabin, and reduce the infiltration flow rate. However, with the objective to reduce the power consumption of the heat pump, this solution cannot be selected. The infiltration occurs most likely through gaps and bad sealing between elements of the car body. A different design or a new assembly method

could be a solution. The literature survey does not highlight any attempt of a structural solution to prevent infiltration.

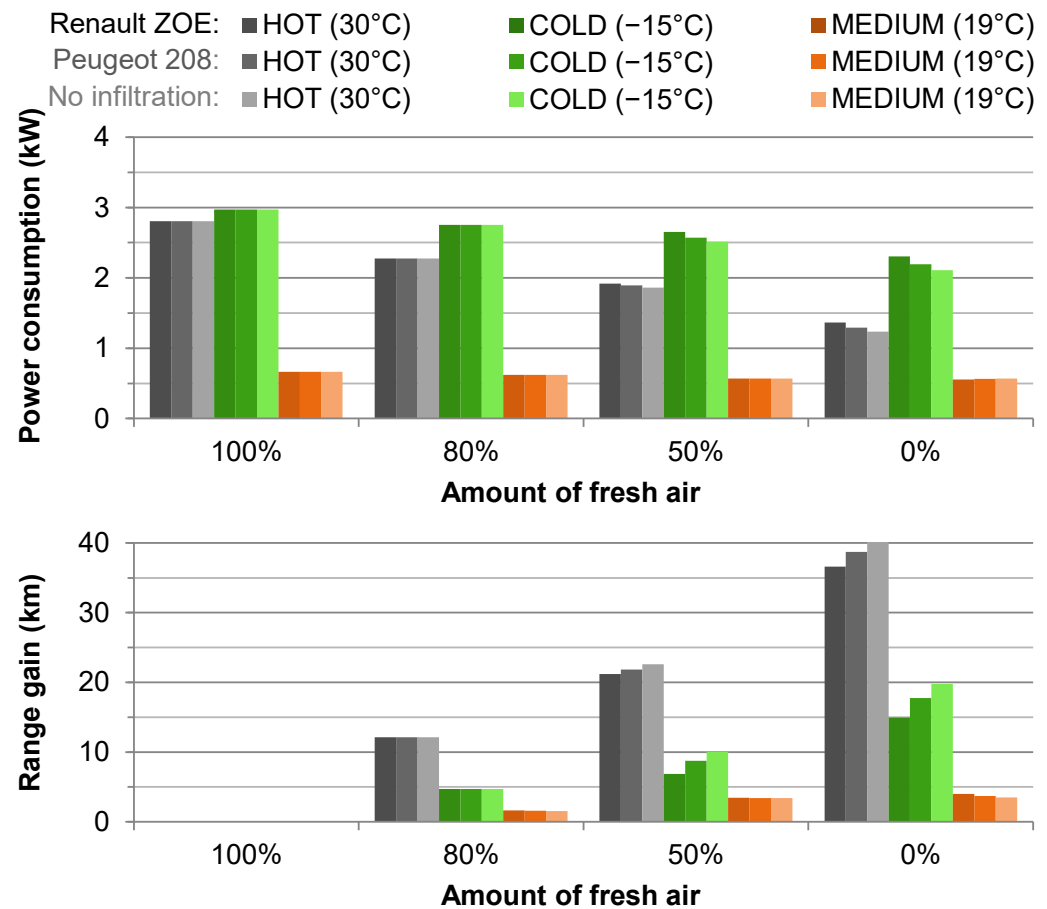


Figure 16. The power consumption of the compressor and driving range gain on three ambient scenarios, depending on the infiltration model.

3.5.2. Influence of the Ventilation Settings

The last study investigates the influence of the blower flow rate. The baseline parameters of the simulation are used, with the Renault ZOE infiltration model, on WLTC. In this subsection, the focus is on the blower action. Previously, the flow rate was fixed at 200 m³/h. This medium blower position for the Renault ZOE is considered as a common usage of the vehicle.

Infiltration Considerations

At first, the influence on infiltration is evaluated. Except in recirculation mode, an increasing blower flow rate can help to reduce infiltration. The simulation model is used to create a new map of the infiltration volume (Figure 17). A total 99 simulations are run, including 11 recirculation flap positions and nine blower flow rates (from 5 to 400 m³/h). The infiltration volume is computed on WLTC. Each simulation lasts 1800 s. The average computational time is 33 s per case.

Two zones can be extracted from the map: up and above 50% of recirculation. In the upper zone, there is more recirculated air than fresh air. A lower blower flow rate helps to reduce the infiltration flow rate. It is the opposite in the lower zone, where the infiltrations are limited by higher flow rates. In order to reduce, or even eliminate, the infiltration, it is best to keep the recirculation ratio below 45%, and the blower flow rate above 100 m³/h. If the recirculation ratio is set higher than 45%, it is best to reduce the blower flow rate.

The 100 m³/h setting appears as a good compromise regardless of the recirculation flap position. This flow rate corresponds to the blower position 1 (out of 8) on the Renault ZOE.

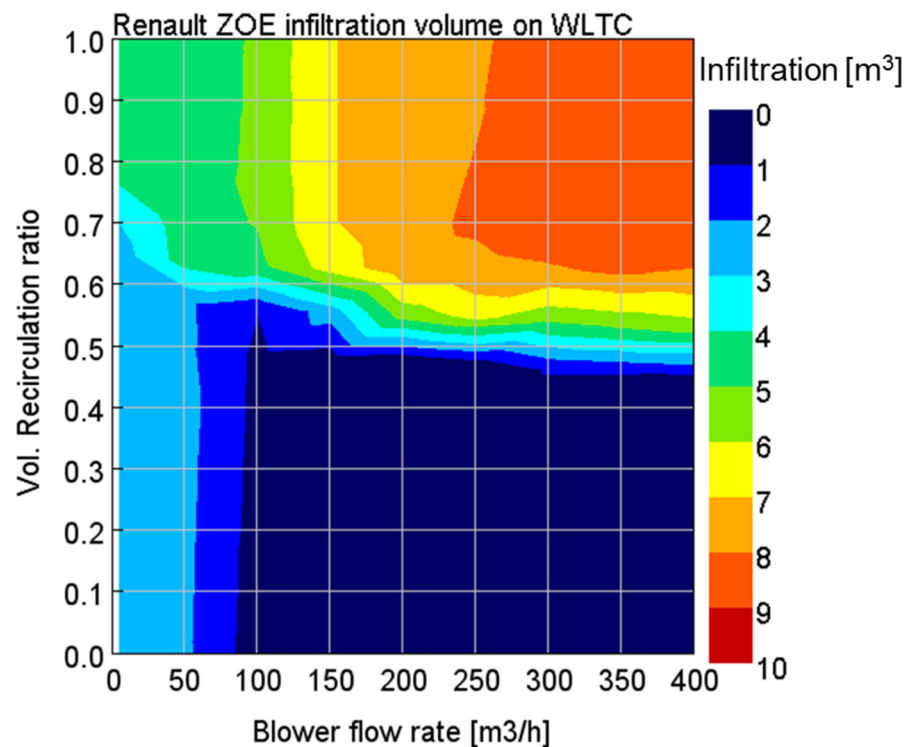


Figure 17. The volume of infiltration air on WLTC, for different ventilation configurations.

Opti-CO₂ Considerations

Alongside infiltration, the blower flow rate has a clear influence on the opti-CO₂ mode. The blower flow rate is separated into this fresh air flow rate and the recirculation. Then, as the blower flow rate increases, the portion of recirculated air increases as well. A few steady-state simulations are performed to quantify this investigation. The blower flow rate is fixed, from 5 to 400 m³/h. The number of occupants varies from 1 to 5, using the same values as in previous Tables 3 and 7. In order to simplify the analysis, no infiltration is considered for these simulations. The recirculation ratio is taken in the regulation stage, once the CO₂ concentration in the cabin reaches the 1000 ppm target. Figure 18 displays the recirculation ratio against the blower flow rate for different numbers of occupants.

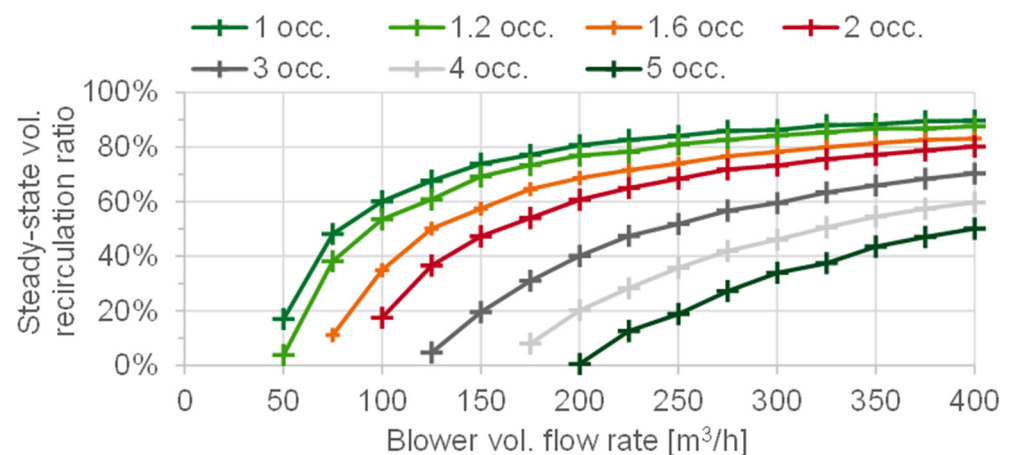


Figure 18. The steady-state recirculation ratio to maintain the cabin CO₂ concentration at 1100 ppm.

At higher blower flow rates, the recirculation ratio in the opti-CO₂ mode increases. Below a certain amount of blower flow rate, the CO₂ target in the cabin cannot be maintained. Even in full fresh air mode, the amount of fresh air is insufficient to keep the cabin CO₂ concentration at 1100 ppm or less. Since the convergence is not possible, these points are not displayed on the graph. The minimum amount of blower flow rate is 40 m³/h for one occupant, 90 m³/h for two occupants, and 200 m³/h for five occupants. Below these values, and assuming no infiltration, the CO₂ concentration rises above 1100 ppm, creating an unsafe environment for the occupants. The infiltration would provide some welcomed fresh air into the cabin, but it is not safe to rely on it since it depends on the vehicle speed.

For the baseline configuration at 200 m³/h and two occupants, the recirculation ratio of the opti-CO₂ mode in regulation stage is 61%, assuming no infiltration. Naturally, as the vehicle speed varies during WLTC, it creates more or less infiltration. This can provide more fresh air into the cabin, and thus reduces the CO₂ concentration. Then, the recirculation ratio in opti-CO₂ mode during WLTC varies slightly above 61% for higher velocities (up to 72% exactly).

Above the minimal flow rate to achieve the 1100 ppm target, the recirculation ratio increases quickly with the blower flow rate. Next, the curve reaches a plateau, meaning that a large increase in the blower flow rate would not lead to a significant change in the recirculation ratio. The case with one occupant can be taken as an example:

- As the blower flow rate varies from 50 to 200 m³/h, the recirculation ratio increases from 17% to 81% (+64%).
- As the blower flow rates increases from 200 to 350 m³/h, the recirculation ratio increases from 81% to 88% (+7%).

Finally, from an energy saving point of view, assuming the opti-CO₂ is selected for the vehicle, it is best to increase the blower flow rate. It leads to an increase in the recirculation flow rate, and thus to a decrease in the power consumption of the heat pump. In order to quantify the gains, simulations are performed at two different blower flow rates: 100 and 300 m³/h. The opti-CO₂ mode is selected, and all parameters, apart from the blower flow rate, are in their baseline configuration (with infiltration). Results at 100 and 300 m³/h are analysed along the baseline simulation at 200 m³/h. The average power consumption of the heat pump on WLTC is given in Figure 19.

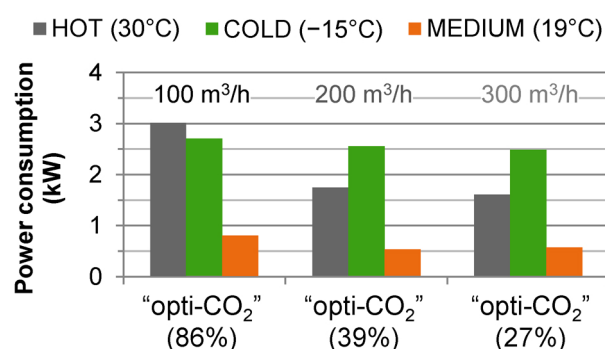


Figure 19. The power consumption of the heat pump on WLTC in opti-CO₂ mode for three blower flow rates.

The fresh air ratio is stabilised approximately at 83% in the 100 m³/h configuration, 39% at 200 m³/h, and 27% at 300 m³/h (these ratios vary slightly depending on the infiltration flow rate during WLTC). There is a clear advantage in increasing the blower flow rate from 100 to 200 m³/h: the power consumption is reduced by 1.3 kW in the HOT scenario, and 0.2 kW in the COLD and MEDIUM scenarios. This can lead to, respectively, 30 and 4 km of a driving range benefit. The gains of going from 200 to 300 m³/h are not

as impressive. The power consumption is reduced by less than 0.15 kW for all ambient conditions. According to these results, it appears that the blower flow rate should be relatively high in order to enhance the opti-CO₂ mode. However, above a certain level that depends on the number of occupants, the benefits are progressively less impactful. In practice, the presence sensors, already in use in most cars (for the seatbelt warning), can be used to know the number of occupants. With this information, the blower flow rate of the opti-CO₂ mode can be set at a proper value.

Global Considerations

A blower flow rate can be selected to minimise the infiltration flow rate, and thus minimise its impact on energy consumption. However, the blower action is not limited to the infiltration. It has a major role in the ventilation system for the heating or cooling of the cabin. Two new sets of simulations are performed at two different blower flow rates: 100 and 300 m³/h. The 12 cases (three ambient scenarios and four recirculation configurations, excluding the “opti-CO₂” configuration presented in the previous subsection) are run for each blower flow rate. All of the other parameters are in their baseline configuration (WLTC, two occupants, and Renault ZOE infiltration model). The results are compared with the original configuration at 200 m³/h, and displayed in Figure 20. The “opti-CO₂” cases from the previous subsection could be inserted between the “100%” and “80%” cases for the 100 m³/h configuration, and between the “50%” and “0%” for the 200 and 300 m³/h configurations.

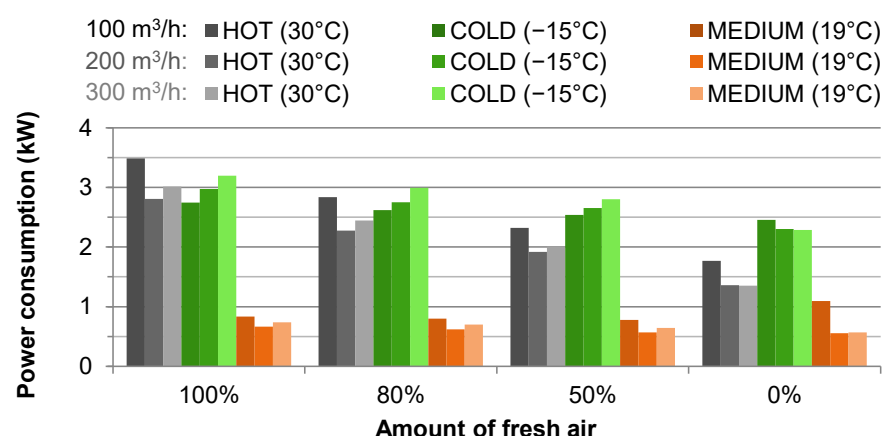


Figure 20. The power consumption of the heat pump on WLTC for different blower flow rates.

At first sight, there is no clear view of which blower configuration would be best for energy savings:

- The 100 m³/h configuration is best for three cases: COLD scenarios with 50% to 100% of fresh air.
- The 200 m³/h configuration is best for seven cases: HOT scenarios with 50% to 100% of fresh air, and all of the MEDIUM cases.
- The 300 m³/h configuration is best for two cases: HOT and COLD scenarios with 0% of fresh air (the three opti-CO₂ cases could be added).

In recirculation mode, which is the best configuration for energy savings, the 200 and 300 m³/h configurations have very similar results (less than 0.02 kW of difference). In this mode, the blower does not drive any external flow rate into the cabin. By increasing the blower flow rate, the air velocity and flow rate through the heat exchanger are increased as well. Then, the convective heat rate of this exchanger is also increased. This leads to a quicker temperature rise inside the cabin. However, increasing the blower flow rate also

has a consequence on the infiltration. As explained in a previous subsection, in recirculation mode, a higher blower flow rate leads to more infiltrations. This inlet of outside air into the cabin is a drawback for the power balance. According to the simulation results, it is still a better option to increase the blower flow rate. The gains could be higher by eliminating the infiltration issue.

For the other recirculation configurations (50% to 100% of fresh air), the analysis of each ambient scenario must be made separately. In heating mode (COLD scenario), the power consumption increases with higher blower flow rates. From 100 to 300 m³/h, the power increases from 2.8 to 3.2 kW in the full fresh air case. In order to save energy, it is best to have a lower blower flow rate. Indeed, it leads to introduce a lower amount of fresh air into the cabin. During WLTC, the total volume of fresh air introduced in the 100% fresh air case is, respectively, 50, 100, and 150 m³. Regarding infiltrations, the 200 and 300 m³/h configurations are sufficient to prevent any infiltration in the 80% and 100% fresh air cases. The 100 m³/h configuration allows a little flow rate of infiltration into the cabin, of the order of 0.8 m³ for the whole duration of WLTC in the 100% fresh air case, and 1.2 m³ in the 80% fresh air case. Then, once again, the benefits of the 100 m³/h configuration could have been higher by eliminating the infiltration issue. However, this gain would have been limited as follows: the volume of fresh air introduced by infiltration is negligible compared to the volume of fresh air introduced by the blower.

In cooling mode (HOT and MEDIUM scenarios), the configuration at 200 m³/h is slightly better than at 300 m³/h. This is similar to the COLD scenario. However, the configuration at 100 m³/h leads to the highest power consumption. It seems to underperform in comparison with the COLD scenario. The example of the HOT scenario at 100% fresh air is taken for the analysis (the following considerations can be translated for the other cases). The compressor power is displayed in Figure 21.

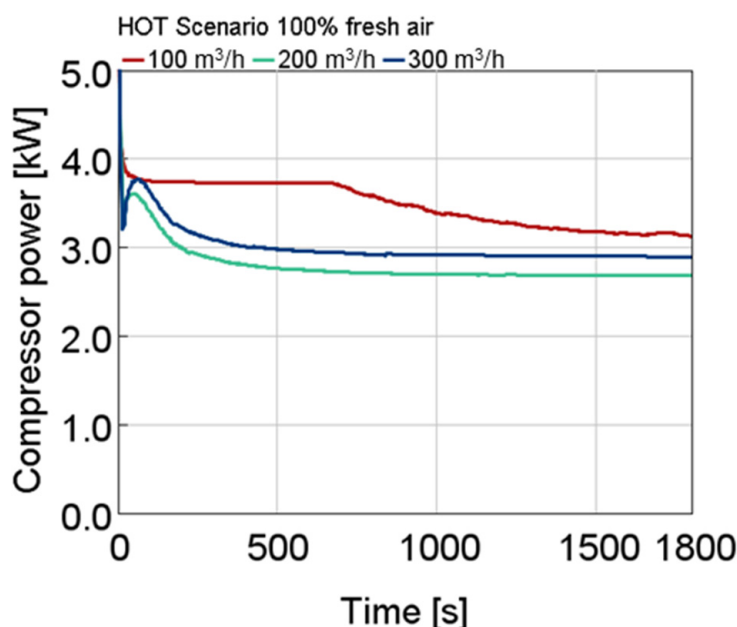


Figure 21. A comparison of the three blower flow rates with the HOT scenario—100% fresh air case.

In the 100 m³/h configuration, the compressor operates at full power during 670 s. It is the time needed to reach the cabin temperature target of 25 °C. Although the amount of fresh air introduced in the cabin is the lowest at 100 m³/h, the heat pump is not helped by the solar flux, the heat released by the occupants, or through the firewall, and the infiltration phenomenon. Except for the infiltration, these factors were beneficial in the COLD scenario.

In the HOT scenario, the compressor at full speed is not able to reach the cabin temperature target quickly. At 200 or 300 m³/h, the convective exchange at the heat exchanger on cabin side is more important than at 100 m³/h. The target temperature is reached briefly after the start (90 s) for both configurations at 200 and 300 m³/h. Then, the heat pump compressor reaches its cruising speed in a few moments. The overall power consumption is the lowest at 200 m³/h.

In summary, the blower flow rate is very important in the thermal balance of the cabin. Its joint action with the recirculation flap dictates the amount of fresh air that must be thermally treated. Furthermore, depending on the vehicle speed, it has an influence on the amount of infiltrated air coming into the cabin. Finally, in opti-CO₂ mode, it effects directly the recirculation ratio during the regulation stage.

4. Conclusions

For a given vehicle and an ambient scenario, the primary elements the occupant can control to manage energy consumption are the ventilation settings: the recirculation ratio and blower flow rate. The simulation results quantified the impact of these elements. The evaluation of key factors revealed the interdependency of each parameter, making it challenging to isolate the effect of a single element. Depending on ambient conditions and other parameters (such as the number of occupants and vehicle speed), the blower flow rate and recirculation ratio can be adjusted to reduce the energy consumption of the heat pump. However, it is not possible to identify a single setting that minimises the power consumption while maximising the thermal comfort in all scenarios. From an energy perspective, regardless of the recirculation ratio, operating the blower at an intermediate setting seems to provide a good trade-off. Additionally, the comfort of the occupants, particularly regarding air velocity inside the cabin and blower noise, should be considered when selecting the blower setting. For the recirculation ratio, it is best to be as close as possible to the full recirculation mode. The main limitation is the CO₂ concentration in the cabin and the associated health risks for the occupants.

In this context, the opti-CO₂ mode appears to be a good trade-off, reducing the power consumption while maintaining a safe and comfortable environment for the occupants. In this smart mode, the recirculation flap is managed to ensure a moderate air renewal in the vehicle. In a common case, the recirculation ratio is regulated at an intermediate position, avoiding the energy-consuming fresh air mode and the health-compromising full recirculation mode. The opti-CO₂ mode provides proper thermal comfort: the cabin temperature target is reached quickly, the CO₂ concentration is maintained at a safe level, and the risk of fogging is low. Additionally, the power consumption of the heat pump is limited. Compared to the common fresh air mode, the driving range gains are estimated at 5, 9, and 26 km in the MEDIUM, COLD, and HOT scenarios, respectively.

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Nomenclature

A_s	Surface area (m^2)
h	Convective heat transfer coefficient ($W/m^2/K$)
J	Flux density (W/m^2)
L_c	Characteristic length of the windshield (m)
Nu	Nusselt number (-)
Pr	Prandtl number (-)
Re	Reynolds number (-)
$Power$	Power consumption affecting the driving range loss (kW)
$Range_{loss}$	Loss of driving range of the electric vehicle (km)
$Range_{max}$	Maximum driving range of the electric vehicle (km)
$Spec_{range}$	Specified range of the electric vehicle (km/kWh)
$Speed$	Average vehicle speed (km/h)
φ	Thermal flux (W)
λ	Thermal conductivity ($W/m/K$)
ASHRAE	American Society of Heating, Refrigerating and Air Conditioning Engineers
CO_2	Carbon dioxide
COP	Coefficient of Performance
EER	Energy Efficiency Ratio
H_2O	Water
HVAC	Heating, ventilation and air-conditioning
PID	Proportional Integral Derivative
WLTC	Worldwide Harmonised Light Vehicle Test Cycle

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