THE 1-D ANALYSIS OF COOL DOWN SIMULATION OF VEHICLE HVAC SYSTEM

by

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In this paper, a detailed combined 1-D model of HVAC systems of a vehicle were developed by using the LMS imagine LAB AMESIM software package. The numerical simulations were considered for soaking and cool down analysis under different environmental conditions. The thermal performance of different refrigerants as R-134a and R-1234yf were evaluated in terms of thermal performance and energy consumption. According to the soaking simulation results, the cabin air temperature values ranged from 49 °C to 57 °C in general. The maximum increase in cabin air temperature value was about 22 °C obtained for 1000 W/m² solar load. The total time until reaching the steady-state conditions for a target temperature value (23.5 °C) was different for all simulations. The total time was calculated as 910 seconds for 1000 W/m² solar load by using R-134a refrigerant loop. The results also showed that although the thermal performance of R-134a was slightly better, R-1234yf can be used due to its environmental properties with acceptable performance. The calculated COP values during cooldown analysis were ranged from 1.71 to 4.52 in general. The minimum value was obtained for the cases which had a maximum solar load and higher cabin interior temperature values. The calculated temperature data for soaking and cool down analysis were in good agreement with the reference data presented in this study. These numerical results are very important for reducing the thermal load of the vehicle cabin considering energy consumption of the HVAC system for different thermal conditions.

Key words*: 1-D simulation, refrigerants, cool down simulation, cooling, vehicle cabin*

Introduction

Passenger thermal comfort is an important design criterion for automotive manufacturers. Moreover, the thermal comfort of a driver in a vehicle cabin ensures the occupants' health and safety too. The HVAC system of a vehicle has to provide the quality and temperature of the air for all weather conditions. In summer conditions, according to the American Society of Heating Refrigerating and Air Conditioning Engineers standards, cabin interior air temperature should be in the range of 23-28 °C in general [1]. The HVAC system of the vehicle is used to keep the cabin interior air temperature at these desired levels. However, there are some physical tests such as heating, deicing and defogging, soaking and cool down tests*, etc.* that the HVAC system needs to be satisfied. The soaking tests represent a vehicle parked in the sun

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without running the HVAC system to get the variations of cabin interior air temperature and also the maximum temperature values. After the soaking test, the cool down test of an HVAC system has to be performed to get the desired cabin interior temperature with a reasonable time. In cooldown tests, the HVAC system is activated so that the air-flow from the vents is also started under various driving conditions. These soaking and cool down analyses of the HVAC system can be carried out both experimentally and numerically. Sen and Selokar [1] numerically investigated cool down of a sports utility vehicle (SUV) by using the CFD method. In their model, the vehicle parked under 1000 W/m^2 solar load for soaking analysis, the inside temperature of the vehicle was reached about 65 °C and some local areas had temperature values up to 70-80 °C, which were below the dash surface due to low air velocity. Cool down simulations were also carried out to evaluate the temperature and velocity distributions of the vehicle. Patidar performed the soaking and cooling analyses both experimentally and numerically. A 3-D SUV model with seven passengers was used in CFD simulations. In the soaking period of this study, 1000 W/m^2 solar load was imposed on the vehicle cabin, and the temperature inside the cabin increased from 45 °C (ambient temperature) to about 60 °C for two hours. The cool down simulations were also performed with air conditioning in the operating re-circulation mode. The temperature values were monitored at different locations, and the HVAC performance was also predicted, and hot spots were identified inside the cabin [2]. Mathur [3, 4], experimentally investigated the vehicle's cockpit module at cold-soaking conditions and at high ambient with solar loads to get the influence of energy stored in various components (*e.g.*, instrument panel, HVAC system, heat exchanger, wire harness*, etc.*). The author concluded that in order to design a better HVAC system, stored energy in these components should take into account and experiments showed that the instrument panel stores the maximum amount of energy at a given temperature. Sevilgen and Kilic [5], investigated cooling and heating periods of an automobile cabin with a manikin, including also solar radiation effects. During the experimental study, only four of the console vents were fully opened in the vehicle. In their numerical study, they developed a 3-D CFD model of the vehicle cabin with an only driver inside and used the velocity and temperature values obtained from their experimental study. They also examined the effect of using different boundary conditions defined on human body surfaces to evaluate the thermal comfort [5-9].

However, the soaking and cool down analyses can be performed by using 1-D models. The 1-D model simulations can give us an average temperature inside the cabin, but CFD simulations can give us both average temperature values inside the cabin and the temperate distributions at the local surface of the cabin interior. Moreover, CFD models can be used for the estimation of local discomfort zones of the human body and the calculation of heat transfer interactions between the human body and its environment. On the other hand, temperature and velocity distributions at the local surface of cabin or human models, hot or cold spots*, etc.,* cannot be monitored by using the 1-D models. There are also co-simulation between CFD and 1-D models, which combine the advantages of both approaches. Shah generated both CFD and 1-D models to get cool down simulation of a vehicle and determined the influence of solar irradiation, selected cabin wall materials, surface areas, refrigerant charge of HVAC system, heat exchanger dimensions by using 1-D model but the temperature distribution inside the vehicle was monitored by using CFD method [10]. In another study, air temperature value at the evaporator outlet was calculated from the 1-D model and used as an input value for the CFD model of a cool down analysis [11, 12]. Zhai [13] built a 1-D model that include a group of correlations for air conditioning system components and decided a combination of the condenser's fin density, compressor type, and evaporator air quantity. Austin and Botte [14] modeled air conditioning and engine cooling systems of the vehicle by using Flowmaster 1-D software package. Patil *et al.* [15] developed a 1-D model for a parked car subjected to solar heat loads to investigate the thermal assessment of the automobile cabin. The developed model can predict the thermal response of a car cabin and its internal aggregates, such as seats, dashboards, roof*, etc.,* for hot day solar loading conditions. There were also 1-D combined models include refrigerant loop and vehicle cabin used to simulate soaking and cool down analysis and to predict the HVAC system performance of a vehicle. The 1-D combined models provide cool-down cycles with desired vehicle speeds or driving cycles such as the New European Driving Cycle (NEDC) and allow to get the effects of material types, insulation thickness, vehicle paint*, etc.*, on the HVAC system performance [16-18]. The comparison of refrigerant thermal performance can also be achieved by using 1-D models. The R-134a is still widely used refrigerant in vehicle air conditioning systems, but properties in view of ozone depletion potential and global warming potential has not been efficient according to today's legal regulations. The R-1234yf has similar thermodynamic properties and no need too many modifications on the conventional HVAC systems to replace R-134a. The R-1234yf is also compatible with legal regulations [19, 20].

The effects of compressor speed, compressor discharge temperature*, etc.*, on the performances of these two refrigerants were investigated by carried out numerous experimental and numerical studies. The researchers concluded that R-1234yf is a suitable alternative refrigerant to the vehicle HVAC systems due to its environmental properties, acceptable performance*, etc.* In addition, R-1234yf can be substituted in the conventional vehicle HVAC systems with minor modifications [21-24].

In this study, a detailed combined 1-D model of the HVAC system and vehicle cabin was developed by using LMS imagine LAB AMESIM software. The 1-D model was used for soaking and cool down analyses of a vehicle cabin under NEDC driving cycle conditions. The effect of both environmental and interior parameters such as solar load, cabin interior temperature, and using different refrigerants on the HVAC system performance was investigated numerically. The results of the 1-D model and the reference data for cooling periods were compared and discussed.

Materials and methods

All of the 1-D models were generated by using LMS imagine LAB AMESIM that a simulation software can model and analysis of 1-D systems and also quickly estimate the effects of the variable parameters [25]. The screen shot of the software interface was given in fig. 1.

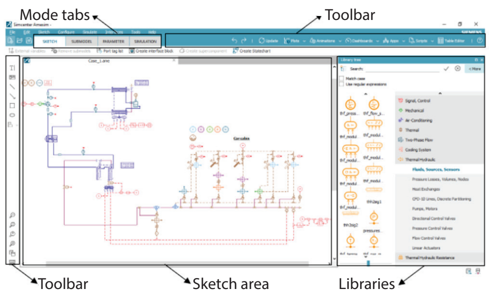


Figure 1. The screen shot of the LMS imagine LAB AMESIM software interface

The developed 1-D model consists of two sub models, which were vehicle cabin and refrigerant loop models. By using these sub models, effects of geometric details such as surface areas, inclination angle of the surfaces, the thickness of the cabin exterior walls, material properties of vehicle components, thermal and optical properties of these materials, solar radiation, ambient temperature and humidity, the color of the vehicle, and the vehicle speed were included to the 1-D model simulations. On the other hand, for the thermal assessment of the vehicle cabin in different situations, the HVAC system performance has to be evaluated with the vehicle cabin model. For this purpose, the complete 1-D model of a vehicle cabin is shown in fig. 2.

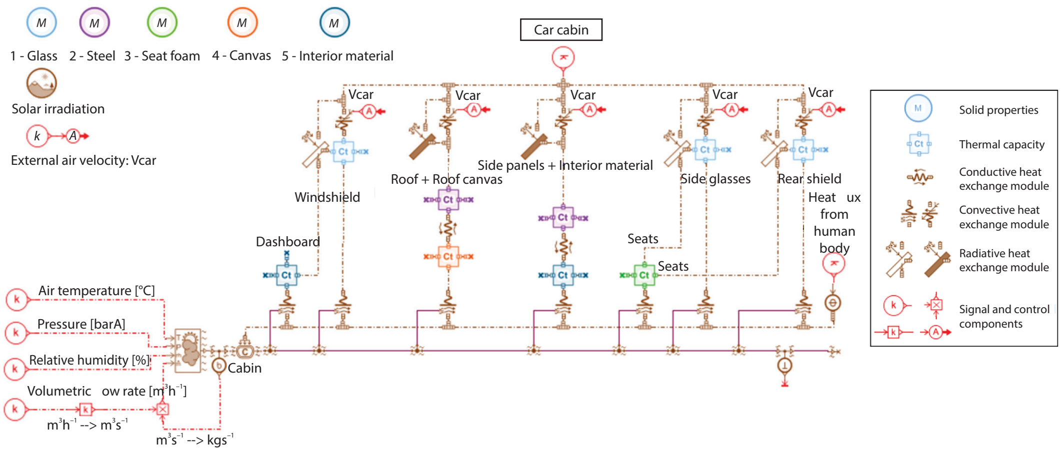


Figure 2. The developed 1-D vehicle cabin model used in soaking and cool down simulations

In the vehicle cabin model, different materials such as glass, steel, seat foam, canvas were used for the exterior surfaces. The relative humidity of the air, cabin interior, and external air temperature values can also be defined by using this model for the initial conditions of 1-D mathematical calculations. During the soaking and cool down simulations, we assumed that the vehicle cabin was subjected to constant solar loads varied from 600-1000 W/m^2 to get the effects of solar radiation on the cabin interior and surface temperature values. Windshield, rear shield, and side glasses were assumed as transparent surfaces. On the other hand, dashboard, roof, seats, side panels and interior materials were assumed as opaque surfaces. The color of the vehicle used in this study was selected as taupe, and the absorption coefficient of the color was assumed as 0.9 [26].

For the soaking analysis, the vehicle cabin model was only used under different environmental conditions. However, the cabin model worked together with the refrigerant loop in cool down simulations. At the beginning of the soaking period, the initial temperature of the vehicle cabin interior temperature was selected as a value of 35 °C. The cooldown simulation had started after soaking analysis to obtain the initial cabin interior temperature value of the vehicle cabin. The refrigerant loop was modeled by using conventional air conditioning system components which were compressor, condenser, expansion valve and evaporator. And the refrigerant loop used in this study is shown in fig. 3. These components were also calibrated individually by using the reference data [18].

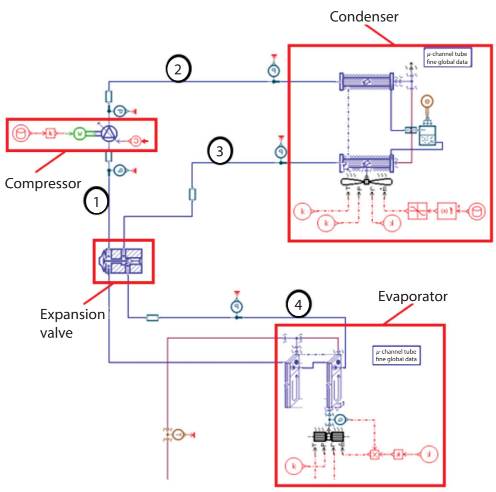


Figure 3. The Refrigerant loop 1-D model performed in this study

For getting precise numerical results, important parameters are compressor speed and displacement in the refrigerant loop. The compressor is generally a belt-driven component of the HVAC system and coupled to the internal combustion engine. Thus, the compressor speed is directly related to the engine speed for these vehicles [27-29]. The engine speed and NEDC driving conditions used in the refrigerant loop model are demonstrated in figs. 4(a) and 4(b), respectively. The NEDC driving conditions were included during all simulations. This driving cycle consists of four consecutive times urban driving cycle (UDC) and extra urban driving cycle (EUDC) cycles that amount to 780 seconds and 400 seconds, respectively. The UDC represents the typical driving conditions of busy European cities and is characterized by low engine load, low exhaust gas temperature. On the other hand, the EUDC represents more aggressive, high speed driving modes [30]. In addition, the compressor displacement can be controlled by the average temperature inside the cabin, and the minimum and maximum displacement values selected in this study were 20 cm^3 and 140 cm^3, respectively. Another important component of the HVAC system is the thermal expansion valve, which was modeled considering four-quadrant diagrams obtained from the reference study [18]. The diagram shows the relationship between the valve lift, temperature, and pressure at the evaporator outlet, and the mass-flow rate of refrigerant. Once the calibration was achieved, the components were assembled in the refrigerant loop. The vehicle speed was also considered and for getting comparative results of using different refrigerants in this model, R-134a and R-1234yf refrigerants were chosen in cool down simulations.

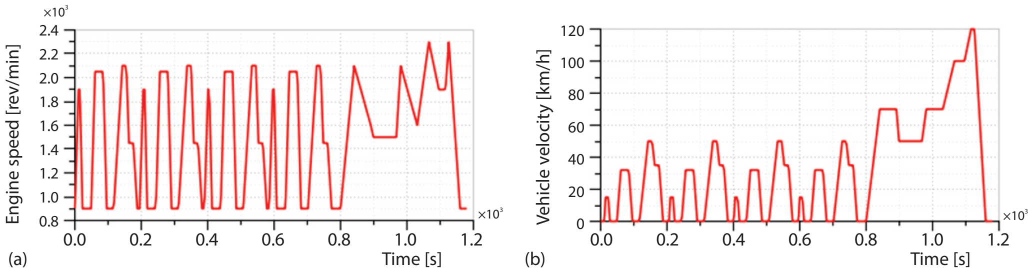


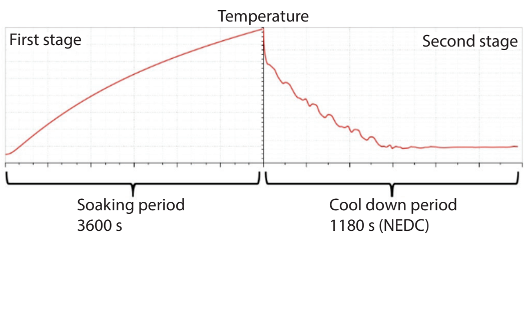
Figure 4. (a) The engine speed and (b) the NEDC driving conditions used in this study

The COP is the ratio between the rate of heat absorbed at the evaporator (cooling capacity) to the amount of power input at the compressor and it can be calculated:

$$COP = \frac{\text{Heat absorbed at the evaporator}}{\text{Compressor power}} \tag{1}$$

All the simulations were performed by using the same vehicle cabin model for soaking analysis and the vehicle cabin volume was designed as 2.78 m^3. The air conditioning system was operated in the re-circulation mode. The soaking simulation time was selected as 3600 seconds. The cooldown simulations were performed for 1180 seconds due to NEDC driving conditions. Once the soaking simulations were achieved, the results of the soaking simulations were used as the initial conditions of the cooldown simulations. The stages of the combined simulation and the combined of the vehicle cabin and refrigerant loop models can be shown in figs. 5 and 6, respectively.

Figure 5. The stages of combined simulation presented in this study



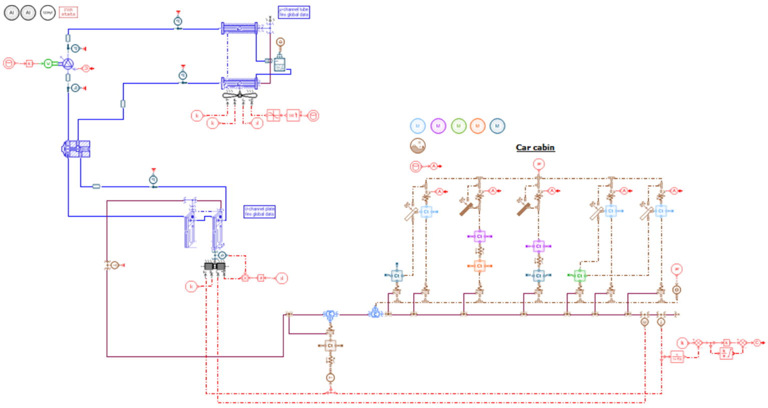


Figure 6. The combined 1-D model include a vehicle cabin and refrigerant loop used in this study

Results and discussions

The cabin air temperature values calculated for the soaking analysis is shown in fig. 7(a). These values were varied from 35-55 °C in general, and these results were compared to the temperature data obtained from a soaking analysis under similar conditions in reference data. The absolute difference in mean cabin interior temperature value between simulated and reference data were below 1 °C in general, fig. 7 (b). From these results, we can easily say that the calculated temperature data were in good agreement with the reference data.

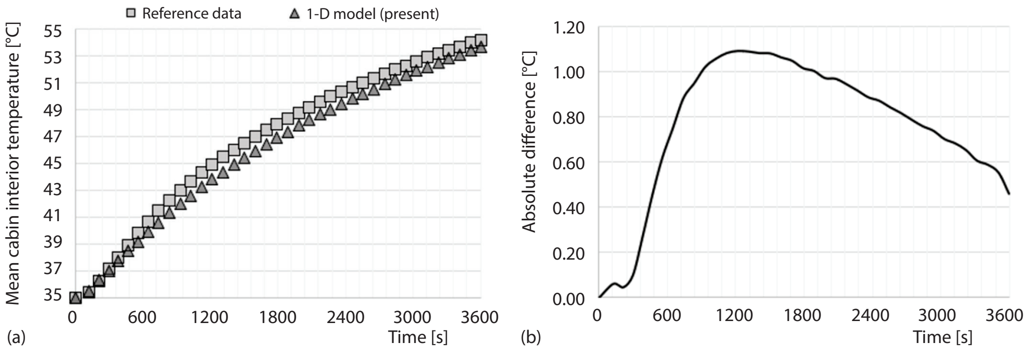


Figure 7. (a) The comparison and (b) absolute difference of the predicted mean cabin air temperature values of the present and reference study

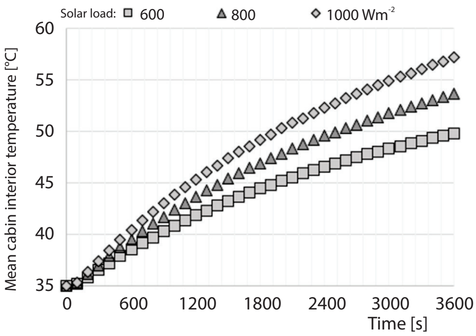


Figure 8. The average temperature inside the vehicle cabin during the soaking period

The comparison of the average temperature inside the vehicle cabin for 600, 800, and 1000 W/m^2 solar loads are shown in fig. 8. At the end of the soaking period, the temperature values ranged from 49-57 °C. The maximum increase in cabin air temperature value was about 22 °C obtained for 1000 W/m^2 solar load. Another important result is that the significant increase was obtained ar the end of the twenty minutes of the soaking period.

The calculated cabin air temperature values during cool down simulations were compared to

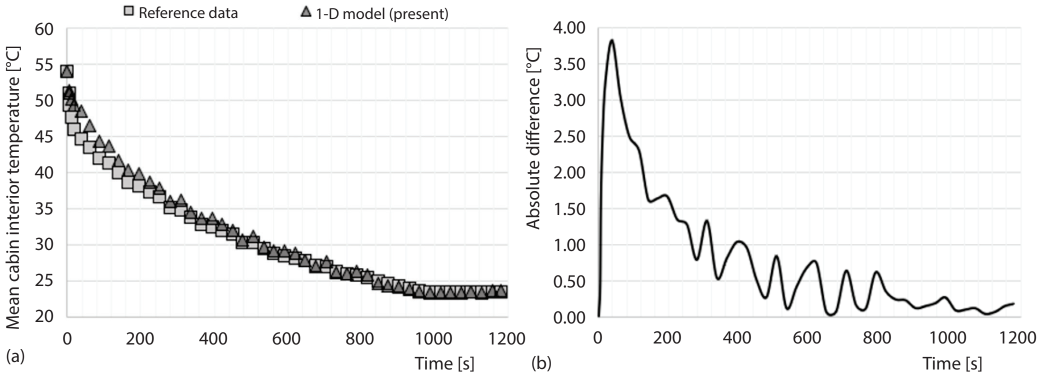


Figure 9. (a) The comparison and (b) absolute difference of the predicted cooldown simulation results of the present and reference study

the reference data, which were obtained for 800 W/m^2 solar load under the same conditions, fig. 9(a). The absolute difference in mean cabin air temperature values between simulated and reference data were in 0.5-3 °C in general, fig. 9(b).

The calculated cabin interior temperature values by using different solar loads and refrigerants are shown in figs. 10-12. All calculated temperature data had the same trend and the mean cabin air temperature values decrease *vs.* time. Nevertheless, the total time until reaching the steady-state conditions with a target temperature value (23.5 °C) was different for all simulations. The total time was calculated as 640, 730, and 910 seconds for 600, 800, and 1000 W/m^2 solar loads, respectively by using R-134a refrigerant loop. Whereas, this value was ranged from 720-1075 seconds for R-1234yf refrigerant loop. From these results, the cooldown analysis by using R-134a refrigerant loop had slightly better thermal performance compared to the results obtained for R-1234yf refrigerant loop.

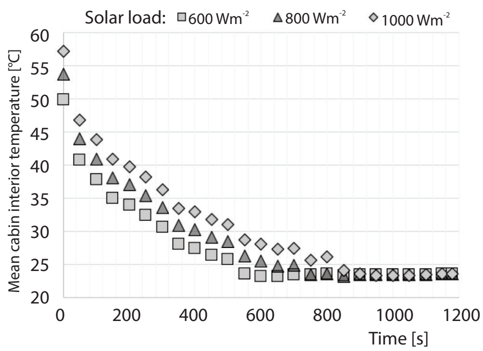


Figure 10. Predicted mean cabin air temperature values by using R-134a refrigerant

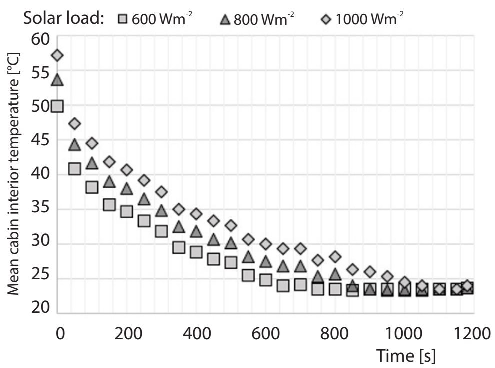


Figure 11. Predicted mean cabin air temperature values by using R-1234yf refrigerant

The mean cabin air temperature values obtained at the end of the driving cycle stages are listed in tab. 1. By using the stage of NEDC driving conditions, the mean cabin air temperature values had little differences considering different solar loads and refrigerants.

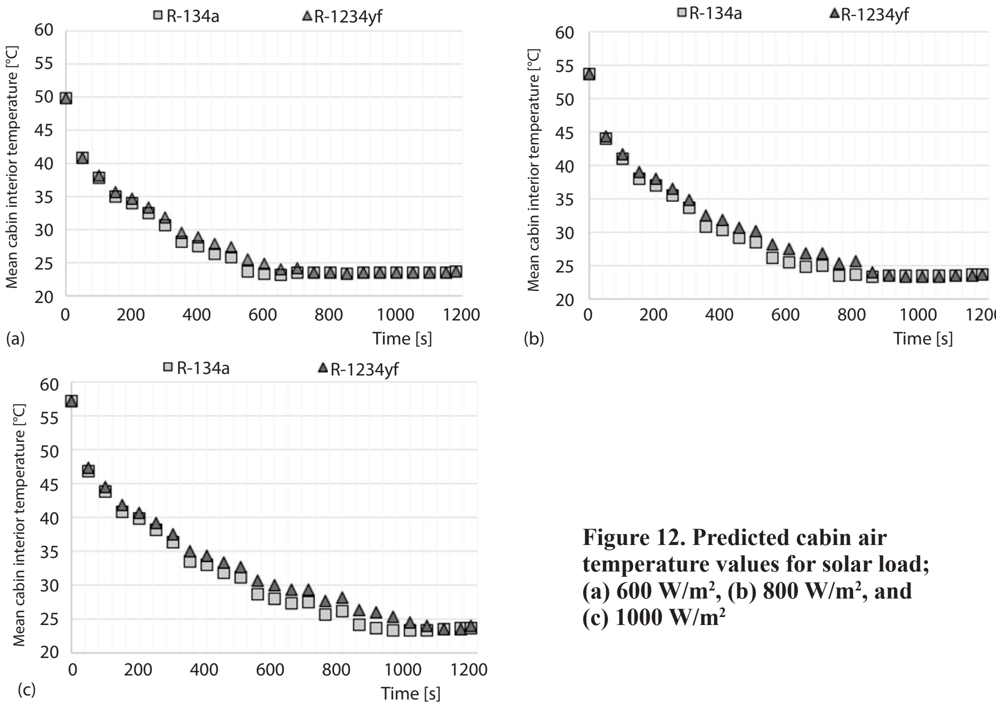


Figure 12. Predicted cabin air temperature values for solar load; (a) 600 W/m², (b) 800 W/m², and (c) 1000 W/m²

Table 1. The average temperature values inside the cabin at the NEDC stages

Refrigerant	Solar load [Wm⁻²]	*T* [°C]					
		NEDC					
		0 second	UDC1	UDC2	UDC3	UDC4	EUDC
R-134a	600	49.84	34.15	28.12	23.90	23.58	23.60
	800	53.63	37.17	30.93	26.57	23.69	23.63
	1000	57.15	39.99	33.57	29.12	26.02	23.65
R-1234yf	600	49.84	35.13	29.66	25.82	23.55	23.61
	800	53.63	38.10	32.39	28.43	25.65	23.64
	1000	57.15	40.87	34.97	30.91	28.09	23.97

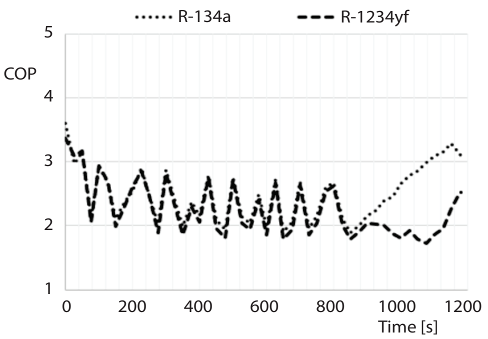


Figure 13. The calculated COP values of the air conditioning system for 1000 W/m² solar load

The calculated COP values during cool down analysis by using eq. (1) was ranged from 1.71-4.52 in general. The minimum value was obtained for the cases which had a maximum solar load and higher cabin interior temperature values. The calculated COP values of developed air conditioning system for 1000 W/m² solar load conditions is shown in fig. 13. It can be clearly shown that the results of the system used R-134a had better thermal performance than R-1234yf one. Another important result is that the significant increase in COP value was calculated after the desired temperature was reached because the selected compressor operates at a lower displacement at that temperature level.

Although R-134a is still widely used refrigerant in vehicle air conditioning systems, it will be no longer used because of today's legal regulations. The R-1234yf is an alternative refrigerant due to similarity of thermodynamic properties, to replace R-134a, some minor modifications to the conventional HVAC system are adequate, compatible with legal regulations and more environmentally friendly*, etc.,* [17-20, 26, 27].

For practical applications, it was also shown that the thermal performance of cooling process of HVAC system can be evaluated and improved by using 1-D models. For instance, the effects of the parameters such as materials, ambient temperature, solar loads, driving conditions, refrigerants, *etc*., were quickly estimated by using the developed 1-D vehicle cabin and refrigerant loop models. Thus, 1-D model of vehicle thermal management system allows performing transient simulations for the estimation of the dynamic behavior of the system.

Conclusions

In this study, a detailed combined 1-D model of the HVAC system and vehicle cabin was developed by using LMS Imagine LAB AMESIM software. We also employed the thermal performance of two different refrigerants used in the automotive industry.

According to the soaking simulation results, the temperature values ranged from 49-57°C, in general. The maximum increase in cabin air temperature value was about 22 °C obtained for 1000 W/m^2 solar load. Another important result is that the significant increase in mean cabin air temperature was obtained at the end of the twenty minutes of soaking period.

All calculated temperature data for cool down simulations had the same trend and the mean cabin air temperature values decrease *vs.* time. Highly transient conditions were obtained for ten minutes of the cooling period in general. But, the total time until reaching the steady-state conditions for a target temperature value (23.5 °C) was different for all simulations. The total time was calculated as 640, 730, and 910 seconds for 600, 800, and 1000 W/m^2 solar loads, respectively by using R-134a refrigerant loop. Whereas, this value was ranged from 720-1075 seconds for R-1234yf refrigerant loop. From the results of cool down simulations, R-134a refrigerant loop had slightly better thermal performance compared to the results obtained for R-1234yf refrigerant loop. The results also showed that although the thermal performance of R-134a was slightly better, R-1234yf can be used due to its environmental properties with acceptable performance.

The calculated COP values during cooldown analysis were ranged from 1.71-4.52 in general. The minimum value was obtained for the cases which had a maximum solar load and higher cabin interior temperature values. Another important result is that the significant increase in COP value was calculated after the desired temperature was reached because the selected compressor operates at a lower displacement at that temperature level.

Considering the parameters such as material type, solar loads, the thermal properties of outer and inner surfaces of the vehicle cabin, HVAC components, NEDC driving conditions, compressor speed, and displacement*, etc.* there were many factors that affected the numerical results. The calculated temperature data for soaking and cool down analysis were in good agreement with the reference data presented in this study. These numerical results are very important for reducing the thermal load of the vehicle cabin considering energy consumption of the HVAC system for different thermal conditions.

The developed combined 1-D model will be used to investigate the effects of using different heat exchanger types on the thermal performance of the system and can be also used to get the prediction of thermal comfort parameters in view of energy consumption for different environmental conditions.

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