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Investigation of Thermal Similarity and Performances of Solar Thermal Air Collector with Heat-Soaked Vehicle Cabin

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ABSTRACT

This article explores the potential of vehicle interiors, which can reach temperatures of up to 120°C due to solar exposure, as efficient solar collectors. Comparative studies have been conducted on the thermal behavior of heat-soaked vehicle cabins and dedicated solar collectors, highlighting their potential benefits in heating, cooling and ventilation. Thermal soak tests reveal a remarkable similarity in thermal performance between the two, with minor temperature variations of up to 5.3°C between the equivalent components. These temperature deviations are mainly due to differences in absorptivity characteristics, in which the black absorber plate exhibits superior absorption capabilities compared to the grey interior cabin surfaces. Integration of ambient air through the inlets on the cabin floor helps with initial cooling of the hot soaked cabin. The collector benefits from a preheated air mixture (comprising heat-soaked cabin air and ambient air), which improves its operating temperatures. This characteristic is attributed to the synergistic integration of the two systems. By decoupling these systems and aligning the properties of the absorber plate, the cabin and the collector could become thermally identical. The potential of the heated vehicle cabin as a power generation source is underscored, which presents opportunities to supplement the energy demand in remote locations. This study illuminates the untapped potential of vehicular solar heat capture, highlighting its feasibility, diverse applications, and significant implications for sustainable energy strategies and community development.

1. Introduction

Vehicles parked in direct sunlight will have extreme temperatures: 80 °C for air and 120 °C for the dashboard [1]. This temperature exceeds the level of thermal comfort upon entry and has been proven to be deadly to children and pets when it is forgotten or left intentionally. Medical professionals see that the interior of a vehicle functions as an oven, accelerating perspiration in confined children. This increased sweating leads to increased evaporation of bodily fluids, increasing the potential for hyperthermia, a severe medical condition involving elevated body temperature, and

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ultimately presents a substantial risk of death [2]. Another health problem for vehicle occupants is the alarming emission of volatile organic compounds (VOCs) as the cabin temperature increases sharply; VOCs are emitted exponentially from hot interior surfaces such as the dashboard, backboard, seat covers, and other accessories [2-4]. As a result, prolonged exposure to VOC will eventually lead to cancer development [5,6]. Furthermore, decreasing the cabin temperature by the air conditioning system (AC) to maintain a comfortable environment for vehicle occupants will increase fuel consumption and tailpipe emissions [7]. The heating, cooling and ventilation loads of buildings are the most consumed in total energy consumption. Thus, using the solar air collector concept as a passive load reduction approach can considerably save energy bills and reduce carbon dioxide emissions. In a hot climate, a solar chimney could bring the temperature of the school room close to the ambient air, reducing the cooling load of the room, ensuring energy savings and thermal comfort [8]. The application of solar air collectors has covered a wide range of areas, but vehicle cabins instead of solar air collectors for heating, cooling and ventilation have not been introduced. This study highlighted the thermal similarity of vehicle thermal cabin soak with solar thermal collector, highlighting its feasibility, diverse applications, and significant implications for sustainable energy strategies and community development.

2. Similarity between Vehicle Cabin and Solar Air Collector

2.1 Physical Similarity Analysis

Figure 1 shows the physical model of the solar air collector and the heat transfer process. Air enters the collector at the bottom opening and the hot air exits from the top of the collector.

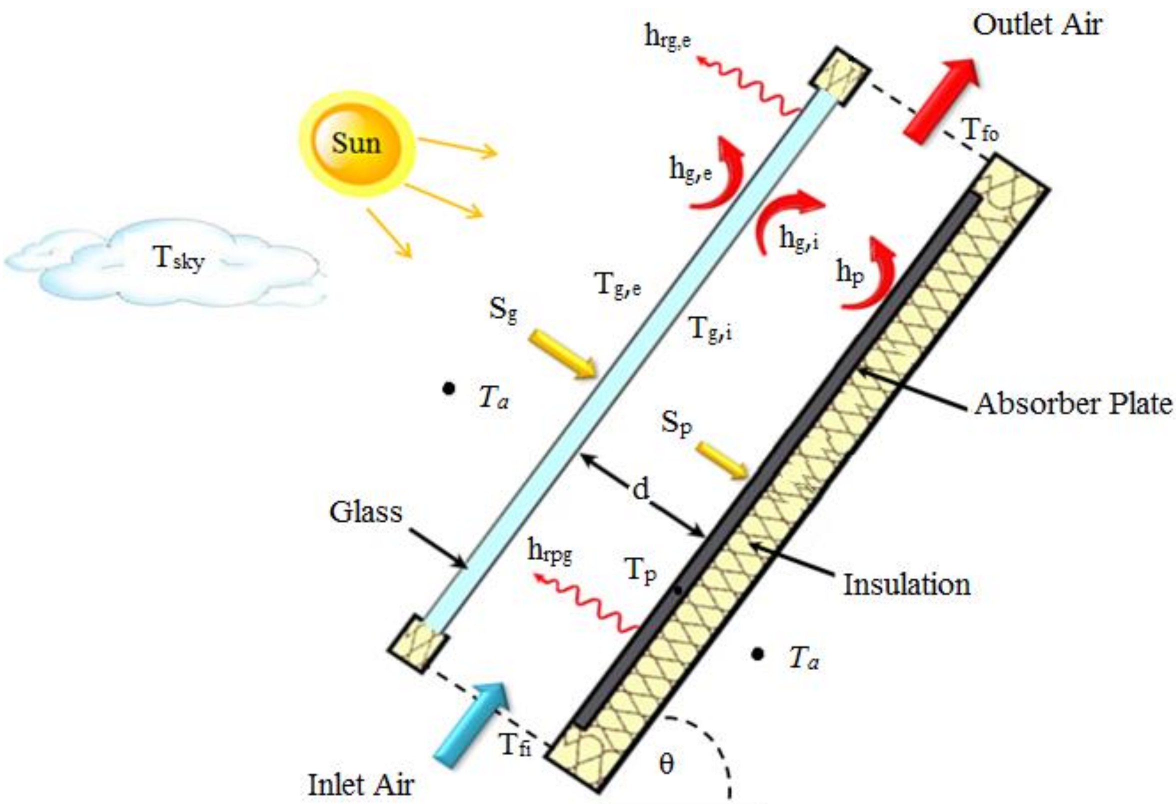

Fig. 1. Schematic diagram of the heat transfer in the physical model

The vehicle cabin is similar to the concept of a solar air collector. However, the cabin can be compared to an efficient solar collector and is the reason for the development of the greenhouse effect phenomenon [9]. The vehicle cabin has windows that act as a collector transparent cover for sun rays to pass through and raise interior temperatures. Furthermore, the interior surface temperature, such as the dashboard, steering wheel, and seats, can reach as high as the black absorber plate temperature of the solar air collector. Since the cabin has two openings, it can be ideally suited as a solar collector with a unique interior absorber.

2.2 Thermal Similarity Analysis

Fundamentally, the energy conversion behavior that occurs within the vehicle cabin is similar to that of a solar air collector. For instance, the heat transfer mechanisms that occur in the cabin glazing are similar to those in the solar air collector glass cover. As shown in Figure 2, some of the incident solar energy received in the cabin glazing surfaces is lost through convection and radiation towards the sky and the other to the cabin interior by convection and re-radiation in the thermal infrared wavelength range [7]. Thus, the interior cabin surfaces will gain the heat loses from the interior glazing surfaces and from absorbing the shortwaves of the incident solar radiation once they pass through the glazing of the cabin.

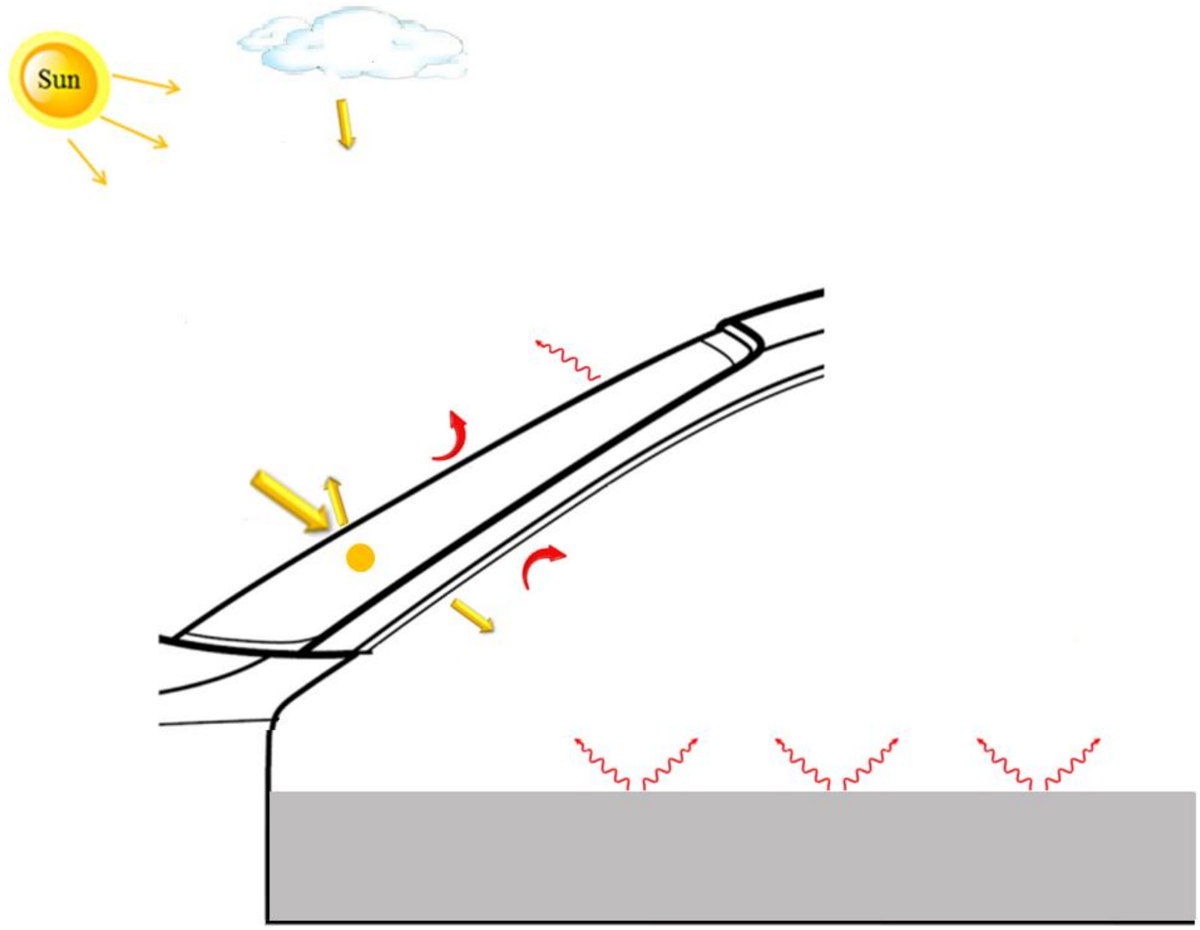

Fig. 2. Heat energy balance in vehicle cabin

2.2.1 Thermal modelling and governing equations of the vehicle cabin and solar collector

Figure 1 and Figure 2 show the heat transfer mechanisms in both systems during exposure to direct solar load. Thermal modelling of the cabin and collector was developed using the principles of heat and mass transfer and solar engineering calculations, which assumed a uniform temperature distribution. In principle, both models have three energy balance equations: one for the exterior surfaces (cabin's glazing, roof, collector cover, enclosure), another for the interior air, and the last for the interior surfaces (interior cabin components (e.g., seats, dashboard, steering wheel, floor, other accessories) and absorber plate. Since the vehicle cabin can be considered similarly to the solar collector, the following energy balance equations for the main components of both systems would be the following

(i) The energy balance equation for the air

Inside the cabin

Inside the vehicle cabin, the primary heat source of the cabin air thermal load is long-wave radiant energy, which is released from the interior surfaces and the roof after they absorb the shortwaves of the incident solar radiation that passes through the cabin glazing during direct exposure to the sun rays [10-12]. Thus, cabin air gradually becomes hotter by convection heat transfer from these masses [13-18].

The energy balance equation for the air inside the cabin can be written as follows:

$[Convective\ heat\ transfer\ from\ the\ cabin\ interior\ surfaces\ to\ the\ air]\ +$
$[convective\ heat\ transfer\ from\ the\ windshield\ to\ the\ air]\ =$
$[the\ heat\ gain\ by\ the\ cabin\ air]$.

Newton's law of cooling gives the following expression for the heat transfer by convection from the interior surfaces of the cabin and the surface of the interior windshield to the cabin air:

$$h_{s,i}\, A_{s,i}(T_{s,i} - T_c) + h_{w,i}\, A_w(T_{w,i} - T_c) = q"_{cabin} \quad (1)$$

The heat gain by the cabin air q''_{cabin} can be calculated using the following equation:

$$q"_{cabin} = m_c C_P (T_c - T_a) \quad (2)$$

In the collector

The energy balance equation for the air in the solar collector channel can be written as:

$[Convective\ heat\ transfer\ from\ the\ absorber\ plate\ to\ the\ air]\ +$
$[convective\ heat\ transfer\ from\ the\ glass\ to\ the\ air]\ =$
$[the\ heat\ gain\ by\ the\ air\ in\ the\ solar\ collector's\ channel]$.

This energy balance can be mathematically formulated as follows:

$$h_p\, A_p(T_p - T_f) + h_{\mathrm{g}}\, A_{\mathrm{g}}(T_{\mathrm{g,i}} - T_f) = q"_{\mathrm{coll}} \tag{3}$$

The heat gain by air can be calculated using the following equation:

$$q"_{\mathrm{coll}} = m_{coll} C_f (T_{fo} - T_{fi}) \tag{4}$$

The mean air temperature (T_f) of the air in the gap can be calculated using a simple linear equation based on experimental observations [19]. It is given as in the following equation:

$$T_f = \gamma T_{fo} + (1 - \gamma) T_{fi} \tag{5}$$

(ii) The energy balance equation for the glazing

The heat transfer calculation for the radiation exchange between the inside surfaces of the cabin will be the most difficult due to the variety of materials used and their irregular shapes. The following assumptions are made for simplifying the analysis

(a) Both systems are under steady-state conditions, one-dimensional heat transfer mode and laminar flow.
(b) Glazing thermal mass and conduction through glass are neglected due to small thickens.
(c) The interior of the cabin is considered as one part.

Figure 1 and Figure 2 illustrate the heat balance at the glazing surfaces, which involved convection and radiation from the glazing to the interior and to the outside air of both systems.

Cabin glazing heat balance

Like the glass cover in the solar air collector, the cabin glazing is a semitransparent medium where solar irradiation can be partially reflected, absorbed and transmitted, Figure 2. On the interior/exterior surface of both systems, it is considered heat exchange between the surrounding and the surface by radiation and convection. The energy balance equations for the cabin windows are all similar; thus, in order to avoid repeating the similar equations, only the energy balance equation for the windshield will be exhibited in this paper.

$$\begin{aligned}&[Incident\ solar\ radiation] + \\ &[radiative\ heat\ transfer\ from\ the\ cabin\ interior\ surfaces\ to\ the\ windshield] = \\ &[convective\ heat\ loss\ from\ the\ windshield\ to\ the\ cabin\ air] + \\ &[convective\ heat\ loss\ from\ the\ windshield\ to\ the\ ambient\ air] + \\ &[radiative\ heat\ transfer\ from\ the\ exterior\ windshield\ surface\ to\ the\ sky].\end{aligned}$$

Mathematically, it can be formulated as follows.

$$\begin{aligned}&S_{\mathrm{w}} A_{\mathrm{w}} + h_{rsw,i}\, A_{s,i}(T_{s,i} - T_{\mathrm{w},i}) = \\ &h_{\mathrm{w},i}\, A_{\mathrm{w}}(T_{\mathrm{w},i} - T_c) + h_{\mathrm{w},e} A_{\mathrm{w}}(T_{\mathrm{w},e} - T_a) + h_{rw,e}\, A_{\mathrm{w}}(T_{\mathrm{w},e} - T_{sky})\end{aligned} \tag{6}$$

Glass Cover Heat Balance

The energy balance equations for the glass cover can be written as:

$[Incident\ solar\ radiation]\ +$
$[radiative\ heat\ transfer\ exchanged\ between\ the\ absorber\ plate\ and\ glass\ cover]\ =$
$[convective\ heat\ loss\ from\ the\ glass\ cover\ to\ the\ air\ in\ the\ flow\ channel]\ +$
$[convective\ heat\ loss\ from\ the\ glass\ cover\ to\ the\ ambient\ air]\ +$
$[radiative\ heat\ transfer\ from\ the\ exterior\ glass\ cover\ surface\ to\ the\ sky]$.

Mathematically, it can be formulated as follows.

$$S_{\mathrm{g}}A_{\mathrm{g}} + h_{rpg}\,A_p(T_p - T_{\mathrm{g},i}) = h_{\mathrm{g},i}\,A_{\mathrm{g}}(T_{\mathrm{g},i} - T_f) + h_{\mathrm{g},e}A_{\mathrm{g}}(T_{\mathrm{g},e} - T_a) + h_{r\mathrm{g},e}\,A_{\mathrm{g}}(T_{\mathrm{g},e} - T_{sky}) \quad (7)$$

(iii) The equation for the energy balance of the interior surfaces

Interior cabin surface heat balance

Interior cabin surfaces, such as the dashboard and seats, have irregular shapes that complicate heat transfer calculations. It has led many authors, particularly Khayyam *et al.,* [20], Fayazbakhsh and Bahrami [21], and Wang and Xiang [22], to assume that these surfaces were flat to simplify their thermal cabin models. Furthermore, the intricate nature of radiative heat transfer among interior cabin surfaces, as highlighted by Marcos *et al.,* [23], has been omitted from consideration due to its potential to significantly complicate the computational process. Furthermore, the authors assert that heat transfer originating from the door panels can be ignored, as its impact is deemed negligible compared to heat transfer originating from the windows.

The energy balance equation for the interior surfaces after the above simplifications can be written as:

[Incident solar radiation] = [convective heat transfer to cabin air]

$$S_{s,i}A_{s,i} = h_{s,i}\,A_{\mathrm{s,i}}(T_{\mathrm{s,i}} - T_c) \quad (8)$$

Absorber plate heat balance

The energy balance equation for the absorber plate can be written as:

[Incident solar radiation] = [convective heat loss to air in the flow channel] + [radiative heat loss to the glass cover] + [conduction to the ambient]. Mathematically, it can be formulated as follows:

$$s_P A_P = h_p A_p(T_p - T_f) + h_{rp\mathrm{g}}A_p(T_p - T_{\mathrm{g,i}}) + \frac{k_{ins}A_p(T_p - T_a)}{\Delta w_{ins}} \quad (9)$$

(iv) Heat transfer coefficients

The energy balance equations are composed of convective and radiative heat transfer coefficients such as h_{rpg}, h_g, and h_p. These coefficients can be obtained from the empirical correlation for the Nusselt number. From the nature of the heat transfer and the type of flow, the Reynolds number (Re) or the Grashof number (Gr) must be calculated, and then the appropriate Nu correlations can be selected. For example, these correlations are given by previous studies [23-35].

$$N_u = \frac{hL}{k} \rightarrow h = \frac{N_u k}{L} \tag{10}$$

where L is the characteristic length, h is the convection heat transfer coefficient, and k is the thermal conductivity of the fluid.

(v) Solar radiation heat flux

The incident solar radiation heat flux absorbed by the transparent surface (cabin glazing and glass cover) is given by:

$$S = \alpha I_t \tag{11}$$

The incident solar radiation heat flux absorbed by the solid surface (cabin interior surfaces and absorber plate) is given by the following:

$$S = \tau_g \alpha I_t \tag{12}$$

Total incident solar irradiance can be estimated from different models for the calculation of solar radiation, such as in previous studies [36-39].

The sky temperature can be obtained according to the following equation [40]:

$$T_{sky} = 0.0552 T_a^{1.5} \tag{13}$$

(vi) Mass flow rate

Therefore, in that case, the mass flow rate equation becomes applicable. However, the buoyancy-driven natural draught flow rate through the two openings can be roughly estimated using the mass air flow equation [41,42]:

Cabin Mass Flow Rate Equation

$$m_{cabin} = C_d \rho A_o \sqrt{\frac{2gH(T_c - T_a)}{(1 + A_r^2)T_a}} \tag{14}$$

Collector mass flow rate equation

$$m_{coll} = C_d \rho A_o \sqrt{\frac{2gL \sin\theta (T_f - T_c)}{(1 + A_r^2)T_c}} \tag{15}$$

The compilation of equations involved in the modeling are listed in Table 1.

Table 1
Summary of equations used for car cabin and solar air collector

Equation	Vehicle cabin	Solar collector
Energy balance equation for the air	$h_{s,i}\,A_{s,i}(T_{s,i}-T_c)+h_{w,i}\,A_w(T_{w,i}-T_c) = q"_{cabin}$	$h_p\,A_p(T_p-T_f)+h_g\,A_g(T_{g,i}-T_f) = q"_{coll}$
Heat gain	$q"_{cabin} = m_c C_P(T_c-T_a)$	$q"_{coll} = m_{coll} C_f(T_{fo}-T_{fi})$
Energy balance equation for the glazing	$S_w A_w + h_{rsw,i}\,A_{s,i}(T_{s,i}-T_{w,i}) = h_{w,i}\,A_w(T_{w,i}-T_c) + h_{w,e}A_w(T_{w,e}-T_a) + h_{rw,e}\,A_w(T_{w,e}-T_{sky})$	$S_g A_g + h_{rpg}\,A_p(T_p-T_{g,i}) = h_{g,i}\,A_g(T_{g,i}-T_f) + h_{g,e}A_g(T_{g,e}-T_a) + h_{rg,e}\,A_g(T_{g,e}-T_{sky})$
Energy balance equation for the interior surfaces	$S_{s,i}A_{s,i} = h_{s,i}\,A_{s,i}(T_{s,i}-T_c)$	$s_P A_P = h_p A_p(T_p-T_f)+h_{rpg}A_p(T_p-T_{g,i}) + \frac{k_{ins}A_p(T_p-T_a)}{\Delta w_{ins}}$
Heat transfer coefficients	$N_u = \frac{hL}{k}$	$N_u = \frac{hL}{k}$
Solar radiation heat flux	$S = \tau\alpha I_t$	$S = \tau\alpha I_t$
Mass flow rate	$m_{cabin} = C_d\rho A_o\sqrt{\frac{2gH(T_c-T_a)}{(1+A_r^2)T_a}}$	$m_{coll} = C_d\rho A_o\sqrt{\frac{2gL\sin\theta\,(T_f-T_c)}{(1+A_r^2)T_c}}$

3. Usages of Accumulated Heat inside Hot-Soaked Vehicle

3.1 Energy Source for Power Generation

Recently, automakers and researchers have been working to exploit techniques that recover heat losses from the vehicle engine that are dissipated by the cooling circuit or carried out by the exhaust gases through the tailpipe to improve the thermal efficiency of the engine, in turn fuel economy. Also, to reduce the heat gain by the interior of the vehicle cabin for improving thermal comfort and fuel economy. According to Seebeck, engine heat losses and cabin heat gain can be used as a source of energy for the thermoelectric power generator system to supply electricity to cooling devices to reduce the cabin temperature of the vehicle soaked in the sun or to charge batteries [43,44].

Gowtham *et al.,* [44] recovering the heat from the exhaust of the vehicle engine by using a thermoelectric generator to cool down a hot cabin temperature of the vehicle, as shown in Figure 3. The authors claimed that at least five units of their prototypes will be able to equalise the interior cabin temperatures with the ambient in 20 min. Sunawar *et al.,* [43] used the accumulated heat inside the hot soaked vehicle cabin as a source of energy for the thermoelectric generator system to supply electricity to the cooling devices to reduce the temperature of the vehicle cabin soaked in the sun and charge batteries. The authors stated that for this approach to be feasible, more areas need to be considered. According to their calculation, if 1 m^2 of vehicle roof is used, the thermoelectric generator could generate up to 280W of electric power.

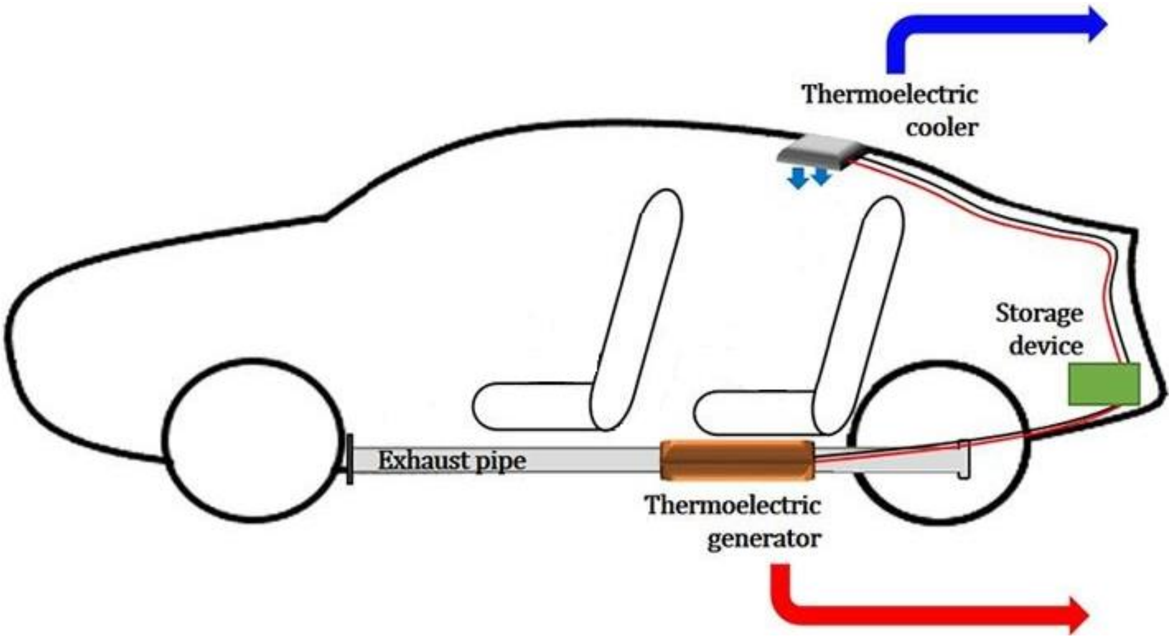

Fig. 3. Thermoelectric power generator installed on exhaust pipes [45]

4. Materials and Methods

Since using a hot soaked vehicle cabin as a solar air collector has not been reported in the existing literature, the objective of this paper is to perform a comparison study to show that the thermal behaviour of a soaked vehicle cabin under the direct sun is similar to the performance of the solar air collector for promoting space heating, cooling, and ventilation.

4.1 Experimental Setup

A car and solar air collector in Figure 4 were used to perform outdoor thermal hot soaking tests to experimentally compare the thermal performances of both systems. The car was modified to work as a solar air collector. It has two inlets on the cabin floor with the same area as the outlet on the cabin roof. All car windows and doors remained closed throughout the test. The solar air collector with adjustable arm mounted on the roof of a vehicle with an air gap of 0.1 m, a width of 0.77 m and a collector length of 1.12 m. Both systems were orientated facing south in an open space under direct sun without shadow interference during the experimental measurements.

Datataker DT80 has been used to measure collector temperatures (glass cover, absorber plate, and outlet air) and cabin temperatures in the car (windshield, dashboard, and outlet air), in addition to weather data such as ambient temperatures and global horizontal solar irradiance. The data acquisition system recorded the data every 60 seconds step interval throughout the test, and the sky conditions throughout the experiment were observed.

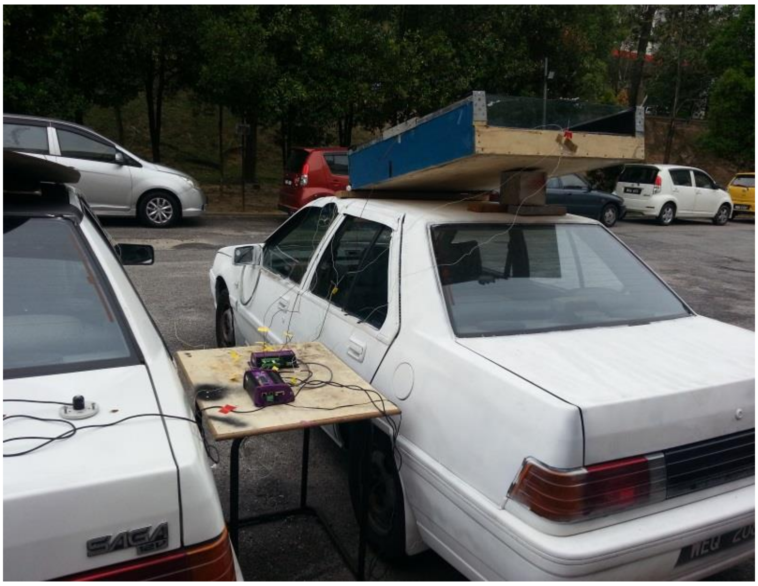

Fig. 4. Test car at the experimental field case

4.2 Experimental Soaking Procedure

The experimental soak procedure adopted in this paper was developed by the authors in previous work for the thermal hot soak test in the outdoors [46]. The soak tests were planned to be conducted on two different days (May 26, and June 6, 2017) to ensure different climatic conditions. The summary of date, soak period, instrumentation, orientation, sky condition and average climatic parameters are detailed in Table 2.

Table 2
Summary of the configurations tested during the soaking trials

Configuration	Soaking Period	Inlet	Orientation	Sky condition	Average ambient temperature °C	Average solar irradiance W/m^2
(I)	11:46-15:46	Recirculation (ON) and two openings on the cabin floor;	South-East	Partly cloudy	32.6	725
(II)	10:06-15:18	Recirculation (ON) and two openings on the cabin floor.	South-East	Mostly sunny	34.5	767

The temperatures of important locations in the car and solar chimney were selected for monitoring and recording air and surface temperatures as shown in Figure 5 by the black circle and summarized in Table 3.

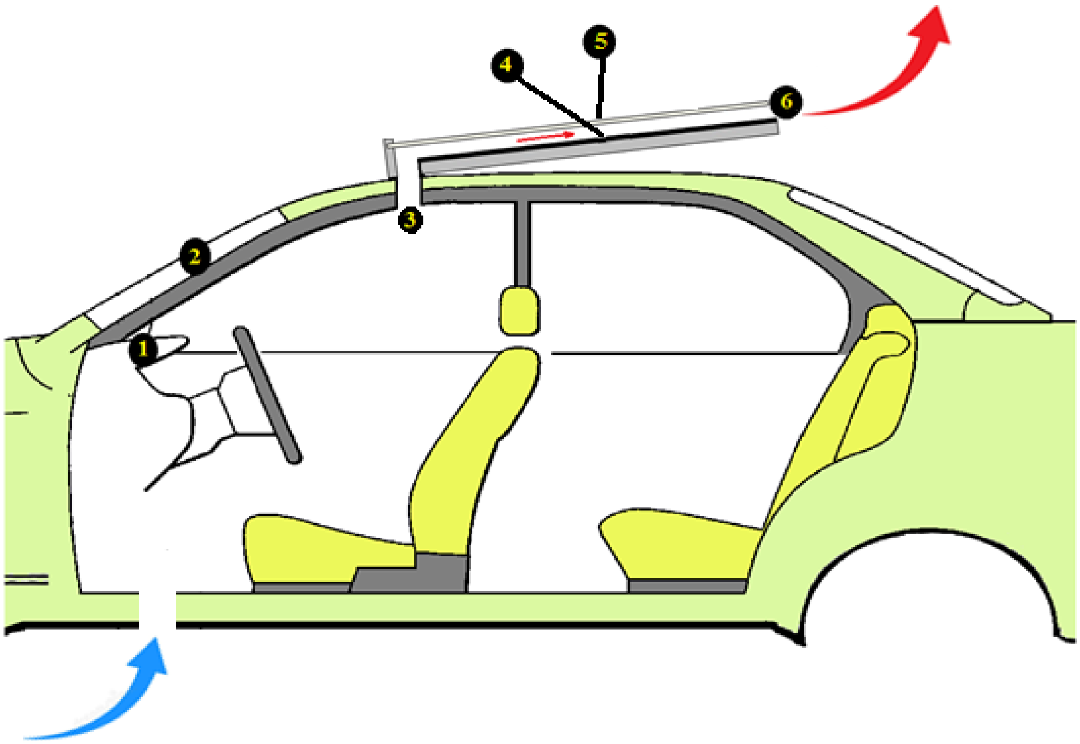

Fig. 5. Location of thermocouples in the cabin and solar collector

Table 3
Summary of the location of the car cabin and solar air collector for temperature measurements

No.	Location	Dimensions (cm)
1	Dashboard	In the centre
2	Windshield	60W & 33L
3	Cabin Air Outlet	In the centre
4	Absorber	In the centre
5	Glass cover	In the centre
6	Collector outlet air	In the centre

5. Results and Discussion

The experimental results of the hot thermal soaking test for both cases are listed in Table 4 and shown in Figure 6 to Figure 11. Table 4 summarised the maximum and average recorded temperatures for each location with the corresponding time, under average environmental conditions.

Figure 6 and Figure 7 show that the variations of the dashboard temperature during the parking conditions are relatively close and similar to the pattern of the absorber plate, which indicated that both surfaces were highly impacted by the values of the solar irradiance. These results are similar to those reported by other researchers [47]. In case (I), the highest recorded temperature was 85.5ºC at 13:02 for the absorber surface and for the dashboard surface, it was 71.2ºC at 13:35. The average temperature difference was 11ºC. In case (II), the highest recorded temperature was 92.3ºC at 12:53 for the absorber surface and for the dashboard surface was 71.3ºC at 13:21. The average temperature difference was 18ºC.

Table 4
Summary of maximum recorded and average temperatures at target locations under average environmental conditions during thermal soak tests

Temperature Location ºC	Case	
	I	II
Max. Absorber	85.5	92.3
Time	13:02	12:53
Max. Dashboard	71.2	71.3
Time	13:35	13:21
Average temperature difference	11	18
Max. Glass cover	56.2	57
Time	13:12	12:53
Max. Windshield	54.4	53
Time	13:36	12:08
Average temperature difference	2.4	3.6
Max. Outlet Collector Air	62	59
Time	14:05	13:53
Max. Air in the exhaust cabin	51	53.5
Time	14:04	14:36
Average temperature difference	4.8	5.3

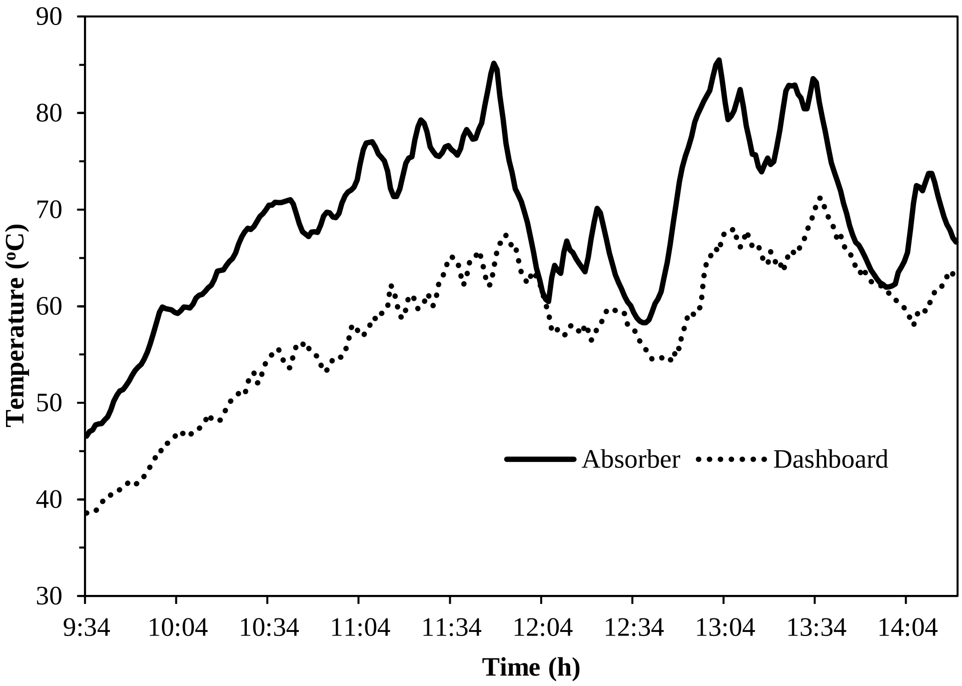

Fig. 6. Comparison of absorber and dashboard temperatures during thermal soaking test, case I

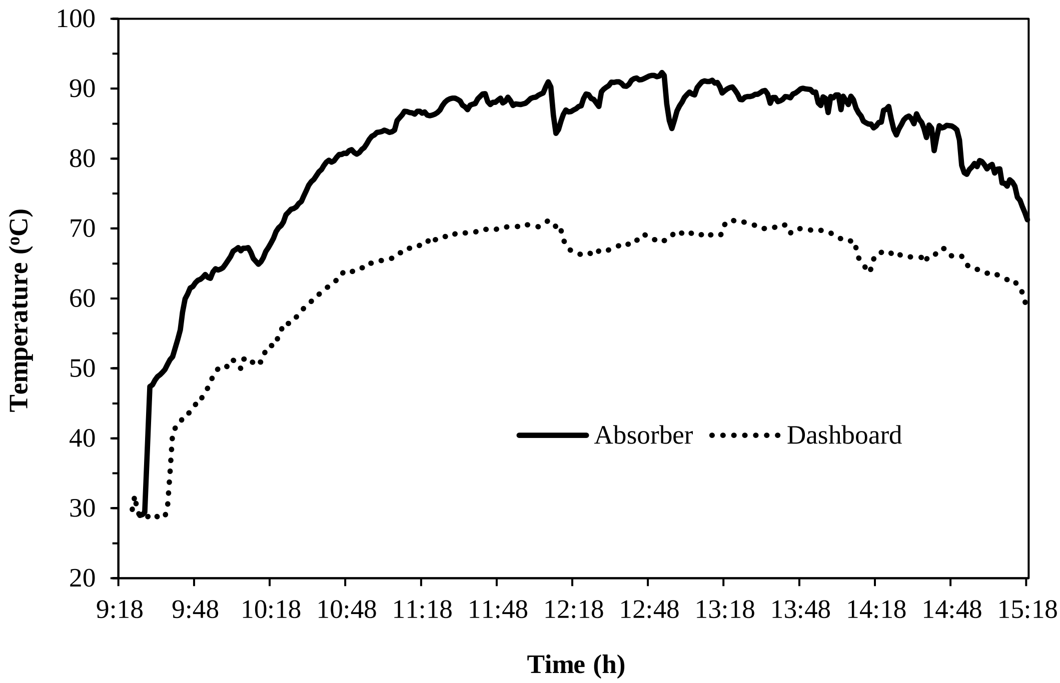

Fig. 7. Comparison of absorber and dashboard temperatures during thermal soaking test, Case II

The figures illustrate that the absorber plate in case (II) recorded higher temperatures than in case (I) due to the higher solar intensity on the day of the experiment. The absorber plate in both cases absorbed the most solar radiation due to the surface colour. The absorber had a black surface which captured more radiation than the dashboard surface (grey). If the vehicle cabin has black interior surfaces with the same absorber plate material, both will be thermally identical.

Figure 8 shows the measurement results of case (I). The highest recorded temperature was 56.2ºC at 13:12 for the glass cover surface and for the windshield surface was 54.4ºC at 13:36. The average temperature difference was 2.4ºC. Figure 9 shows the measurement results of case (II). The highest recorded temperature was 57ºC at 12:53 for the glass cover surface and for the windshield surface was 53ºC at 12:08. The average temperature difference was 3.6ºC.

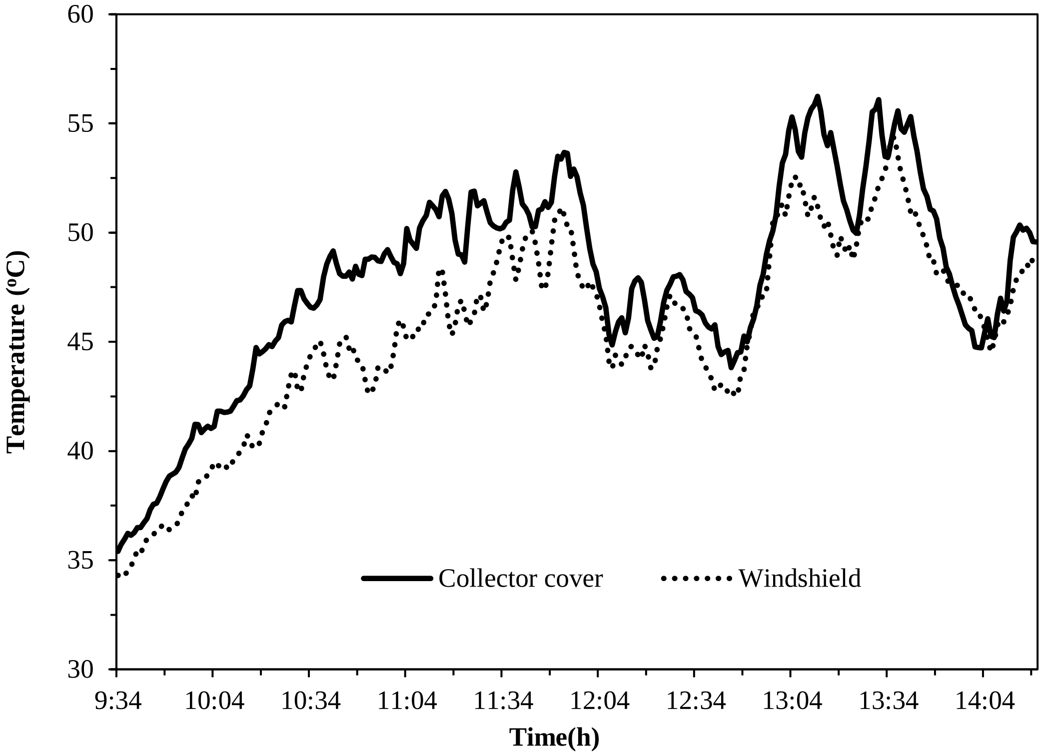

Fig. 8. Comparison of collector cover and windshield temperatures during thermal soaking test, case I

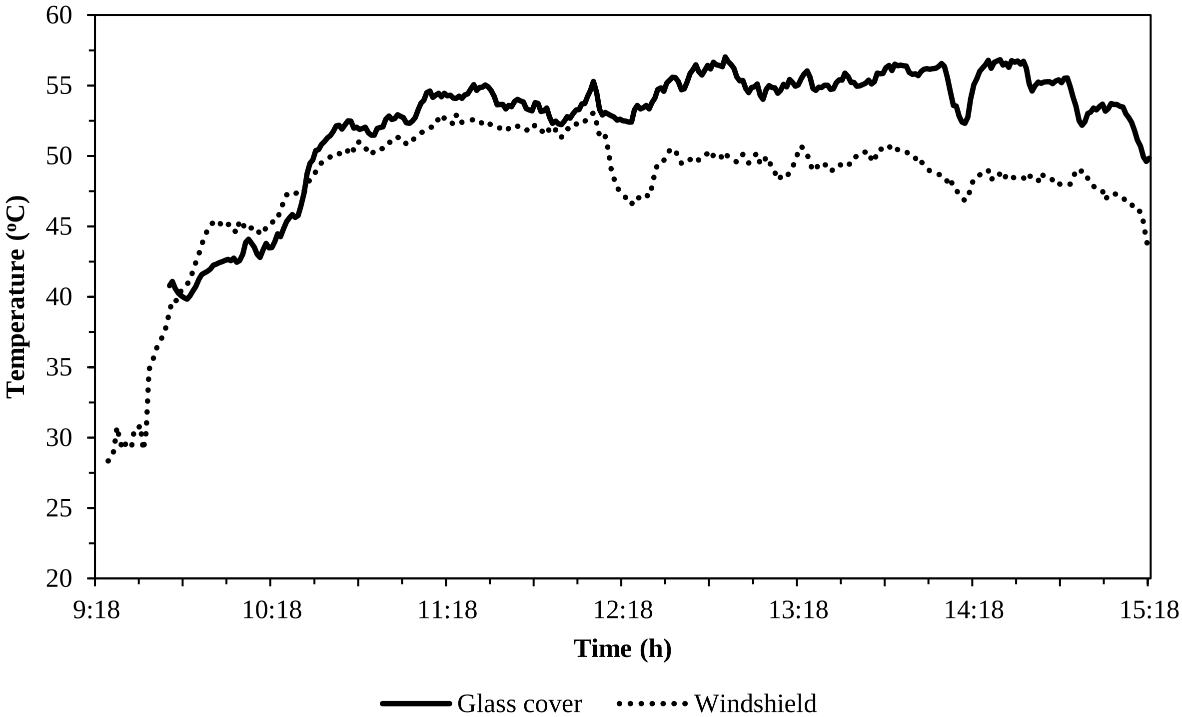

Fig. 9. Comparison of the collector cover and windshield temperatures during thermal soaking test, Case II

Both figures show that the variations in windshield temperature during parking conditions are very close and similar to the pattern of the glass cover, indicating that both surfaces were thermally equivalent. This is because both surfaces were made of glass and had almost identical thermal properties.

Figure 10 and Figure 11 show that the variations of the outlet solar collector and the cabin air temperatures are relatively close and have similar patterns due to the physical and thermal similarity between the car cabin and the solar air collector. In case (I), the highest recorded temperature for the air temperature of the outlet solar collector was 62 °C at 14:05, while for the air of the outlet cabin was 51 °C at 14:04. The average temperature difference was 4.8 ºC. In case (II), the highest recorded temperature for the outlet solar collector outlet was 59 ºC at 13:53, while for the cabin outlet air it was 53.5 ºC at 14:36. The average temperature difference was 5.3 ºC.

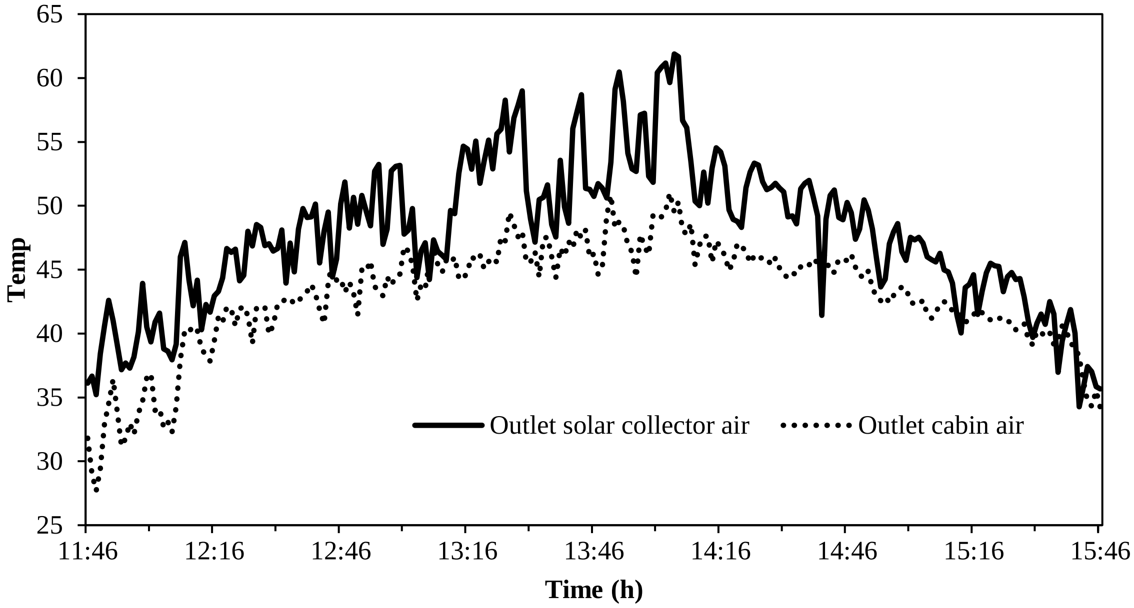

Fig. 10. Comparison of solar collector outlet and cabin air temperatures during thermal soaking test, case I

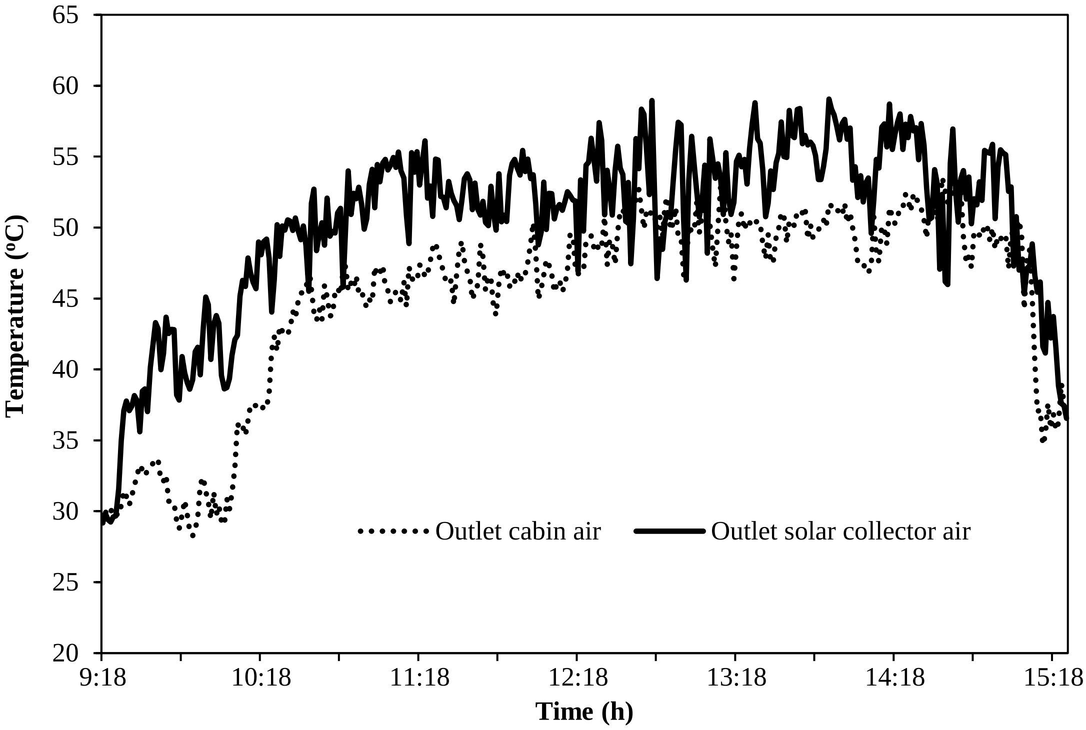

Fig. 11. Comparison of the solar collector outlet and cabin air temperatures during the thermal soaking test, Case II

From Table 4, it can be seen that case (II) has higher temperature increase than case (I) and reached the maximum value before case (I), as case (II) recorded a higher solar intensity than case (I). It was also observed that the outlet collector air recorded a higher temperature than the glass cover in both cases, although the glass cover has a large thermal capacity compared with the air. It was not surprising because the rise in air temperature can be attributed to the convective heat transfer from the black absorber plate (with the highest temperature) and the glass cover itself to the air in the flow channel. In addition, we attribute this observation to the heat transfer mechanism carried out by re-radiation in the thermal infrared wavelength range and convection due to wind, which takes place on the outer surface of the glass cover.

According to the table and figures above, in contrast to the measurement of temperatures of different locations on the solar collector, the cabin temperature of the car of different locations in both cases was very close with an average temperature difference of about 0.4 ^{o}C. Since the cabin is larger than the collector, it has many internal components with different thermal capacities, such as seats, floors, dashboards, etc. These components could function to regulate the temperature of the interior of the cabin.

6. Conclusion

The thermal soaking tests conducted in this study revealed a remarkable similarity in thermal performance between vehicle cabins, which can reach temperatures of up to 120 °C due to solar exposure and dedicated solar collectors. Despite the minor temperature variations that ranged from 2.4 to 18 °C, the equivalence of the components was evident, with interior surfaces, such as the dashboard, effectively functioning as absorber plates, and the windshield serving a role similar to a glass cover. The study suggests that if the two systems were decoupled and the vehicle cabin had black interior surfaces, it is plausible to achieve thermal identicality. This potential alignment in thermal properties highlights the feasibility of utilizing vehicle cabins as efficient alternatives to traditional solar air collectors for heating, cooling, and ventilation purposes. This approach not only presents an innovative energy source but also provides an energy-efficient solution for heating, cooling, and ventilation. The findings of this study underscore the untapped potential of vehicular solar heat capture, emphasising its diverse applications and significant implications for sustainable energy strategies.

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