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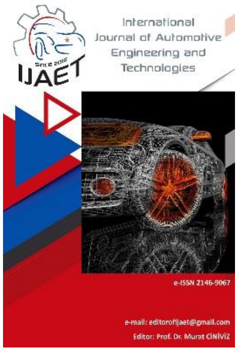

Original Research Article

Calculation of heat transfer from the bottom of a coach to the passenger compartment

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ABSTRACT

In the current study, amount of heat transfer to the passenger compartment through the floor, which is one of the cooling load components of the vehicle, under normal working conditions were examined. For detailed investigations, floor of the passenger compartment was sub-divided into 11 regions by considering the material properties and the thermal condition under the region. By referencing the test results and making some assumptions, mean temperatures of the regions under the floor were determined. And then the heat transfer from the bottom of the vehicle to the passenger compartment was calculated. At the end of study, mean temperatures, heat transfers and heat fluxes of the regions were examined and some improvement modifications advised for heat transfer and economy points of views.

Keywords: Heat transfer, coach, temperature distribution, thermal comfort, cooling load

1. Introduction

Thermal protection is increasingly important in the development process of passenger cars. Tightly packaged engine compartments and strongly increased engine power demand extensive testing and analysis [1].

Modern man's desire for mobility has been a major factor in technical developments. The bus (from Latin omnibus: "for all") has played a significant role in this development. Within the introduction of enclosed bodywork, passengers were given a certain level of comfort. Today, buses still account for a major portion of road transport. The most convincing argument to the passenger in favor of bus travel, however, is the comfort offered, and the interior climate plays a key role here [2].

An important factor affecting the comfort of bus passengers is the heat transfer from the engine compartment through the floor of the passenger compartment. Especially in the summer times, this heat negatively affects the comfort of the passengers.

All of the engineers know that, the temperature of the regions under a coach is higher than the passenger compartment and as a result of this;

some amount of heat is transferred to the passenger compartment. But in literature there is no information about the temperature levels of a coach under side. With this study, it is aimed to determine these temperatures and the heat transfer amount to the passenger compartment.

On the other hand some researchers conducted studies on under hood thermal investigations for automobiles, and trucks. Also some studies conducted to predict the cooling load of a bus.

Büyükalaca et al., (2011) used Radiant Time Series (RTS) method for the calculation of cooling load of a bus. In the study, they introduced Radiant Time Series (RTS) method briefly and gave the important points for the application of the method to a bus [3].

Fournier and Digges (2004) investigated 4 different models of automobiles under hood temperatures in their study with a test procedure similar to present study. In their study, 11 thermo-couples were installed to 4 different vehicles' under hood and measurements were done with 3 different test conditions, which are stationary, constant speed driving and uphill driving [4].

Binner et al. (2006), in their study investigated the under hood temperature distribution of a sport car under maximum speed and low speed uphill climbing test conditions [1].

Kulkarni et al. (2012) conducted a study on under hood flow management of heavy commercial vehicle to improve thermal performance. In the study under hood flow management for 25T truck has been carried out by flow analysis by CFD method using commercial software. As a result of this study they improved velocity and mass flow rate of the air passing through the radiator, and engine room of a truck. Since the airflow around the engine was improved, at the end of modifications, heat transfer to the driver cabin was decreased and heat rejection on radiator and exhaust manifold increased [5].

A computational study was conducted by Xiao et al. (2008), in order to characterize the heat transfers in a sedan vehicle underbody and the exhaust system [6].

Reddy et al., (2019) has conducted a study on analysis of air conditioning system used in buses. The aim of the study was to analyze the performance of a bus shell by considering identifying practical solutions in order to reduce the impact of air conditioning on bus, consumption and, therefore, on air pollution. The analysis was carried considering several parameters, including passenger capacity, local climatic conditions, and fuel consumptions. For the analysis, a bus with passenger capacity of 60 people was selected and then its heat load capacity was determined by considering different conditions like seasons and various loads [7].

Temperature distribution of engine room of a 12-m coach in different driving conditions was examined by Mezarcıöz, (2018). For detailed investigations, engine room was subdivided into six regions by considering the mechanical component layout and thermal condition of the regions. For determination of the temperatures, a test vehicle, which was equipped with 14 thermocouples, tested under three different test conditions. These are constant high speed, uphill climbing and stationary test conditions. At the end of the study, the temperatures of each region under three different driving conditions were determined [8].

In this study, it is focused on the temperatures of the regions under the passenger compartment and amount of heat transfer to the passenger compartment by conduction and convection under regular working conditions. Also some improvements are advised to improve the heat isolation and cost reduction.

2. Material and method

In the present study, the amount of heat transfer from the bottom of a 12-m length coach to the passenger compartment under regular operational conditions was calculated by considering the components that can be assumed as a heat source under the vehicle like, engine, transmission, exhaust muffler, axles etc. The bottom of the passenger compartment sub-divided into 11 regions by considering the material properties (material, thickness and thermal conductivity) and predicted mean temperature under the regions. Then, all the regions examined in detail. Codes and names of the regions can be seen in Figure 1 and Table 1 respectively.

In order to calculate the heat transfer mentioned above, firstly the thermal properties of the heat transfer areas, in other words, the section properties of the passenger compartment must be specified.

The material properties of each section are given in the Tables from 2 to 6. Also section view of region FL1 can be seen as an example in Figure 2.

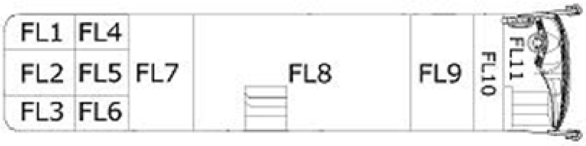


Figure 1. Codes of the regions

Table 1. The names and codes of the regions

Detail Region Name	Detail Region Code
Above the Radiator	FL1
Above the Engine	FL2
Above the Exhaust	FL3
Above the Rear Left Luggage	FL4
Above the Transmission	FL5
Above the Battery	FL6
Above the Rear Axle	FL7
Above the Luggage Compartment	FL8
Above the Front Axle	FL9
Above the Fuel Tank	FL10
Below the Driver Platform	FL11

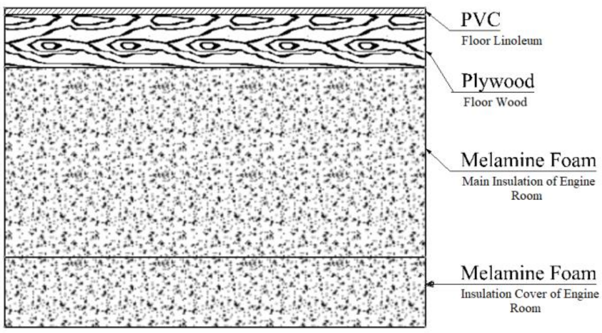


Figure 2. Section view of region FL1.

Table 2. Material properties table for the regions FL1, FL2, FL3 and FL6

No	Section Name	Material	Thickness (m)	Thermal Conductivity (W/mK)
1	Floor Linoleum	PVC	0.002	0.116
2	Floor Wood	Plywood	0.015	0.130
3	Main Insulation of Engine Room	Melamine Foam	0.054	0.040
4	Insulation Cover of Engine Room	Melamine Foam	0.020	0.040

Thermal conductivity of the materials are taken from Yılmaz (1999) [9].

Table 3. Material properties table for region FL4

Number	1	2
Section Name	Floor Linoleum	Floor Wood
Material	PVC	Plywood
Thickness (m)	0.002	0.016
Thermal Conductivity (W/mK)	0.112	0.130

Table 4. Material properties table for region FL5

Number	1	2	3
Section Name	Floor Linoleum	Floor Wood	Metal Sheet
Material	PVC	Plywood	Steel
Thickness (m)	0.002	0.015	0.002
Thermal Conductivity (W/mK)	0.116	0.130	53.0

Table 5. Material properties table for region FL7

Number	1	2	3
Section Name	Floor Linoleum	Floor Wood	Main Insulation of Engine Room
Material	PVC	Plywood	Melamine Foam
Thickness (m)	0.002	0.015	0.054
Thermal Conductivity (W/mK)	0.116	0.130	0.040

Table 6. Material properties table for region FL8, FL9, FL10, FL11

Number	1	2
Section Name	Floor Linoleum	Floor Wood
Material	PVC	Plywood
Thickness (m)	0.002	0.015
Thermal Conductivity (W/mK)	0.116	0.130

As can be seen from the tables, while in the regions, accommodating heat and sound sources like engine, transmission, radiator and exhaust muffler, named as engine room, double layer special isolation materials and 15 mm thickness plywood are employed, in the rear axle region single layer isolation material is employed. Also in the front axle and luggage room regions only 12 mm thickness plywood is used as a separator without any extra isolation.

It can be assumed that a coach is generally driven in high speed in highway conditions. Also, the vehicle is forced to climb uphill in some portion of this highway condition. Additionally, a bus can wait in idle in a small time period of life. So the life cycle of a coach can be assumed as 70% of highway condition, 20% of uphill condition and 10% of idle condition.

In order to determine the temperatures in these conditions a study was conducted by Mezarcıöz (2015). In the study a test vehicle equipped with 31 thermo-couples (Figure 3) tested under these test conditions [10].

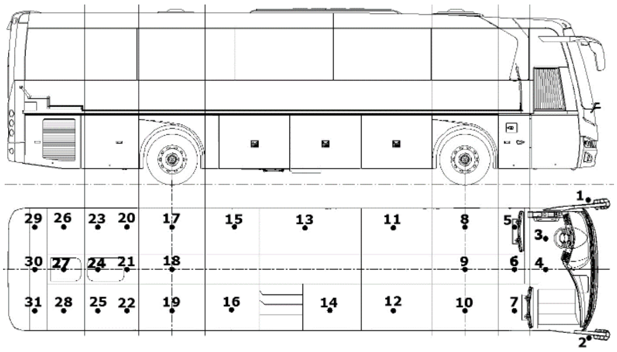


Figure 3. Thermo-couple installation plan

Here, the thermo-couples numbered 1 and 2 were mounted to the side mirrors of the vehicle and used to measure the outside temperature during the tests.

In high speed test condition, test vehicle was driven in a straight highway with a constant speed of 100 km/h in the last gear step of the transmission and the temperatures were recorded for 50 km. In ramp climbing test condition, measurements were taken while the test vehicle was forced to climb in a ramp with a slope of about 15% at 30 km/h. In stationary test condition, the test vehicle was parked in a place where it would be exposed to direct sunlight, and engine room temperature measurements were taken while idling. These measurements were taken under the conditions of 35°C of average outdoor temperature.

In all test conditions, a period of time was waited until the temperatures were stable. Data was recorded in high speed and stationary test conditions in 30 minutes and in ramp climbing conditions 10 minutes. Under all test conditions, the average outdoor temperature was recorded as 35°C.

Cabin temperature of the vehicle is assumed as 24 °C by considering the summer comfort conditions by SAE J1503 and Temsa (2007) [11, 12].

For the calculation of heat transfer coefficients the following procedure was used by considering the heat transfer section properties, inside and outside temperatures.

Total heat transfer coefficient was calculated by employing Equation 1.

$$\frac{1}{U} = \frac{1}{h_{in}} + \frac{s_1}{k_1} + \frac{s_2}{k_2} + ... + \frac{s_n}{k_n} + \frac{1}{h_{out}} \qquad (1)$$

Here inner surface heat transfer coefficient is taken from ASHREA standards, as 8 W/mK for free convection conditions [13].

For the calculation of h_{out} values equation 2 was employed.

$$h_{out} = \frac{Nu \times k}{L_k} \qquad (2)$$

Nusselt Number can be calculated by using Equation 3 [9].

$$Nu = \sqrt{\left(Nu_{L,Lam}\right)^2 + \left(Nu_{L,Tur}\right)^2} \qquad (3)$$

Local Nusselt Numbers can be calculated by using Equations 4 and 5 [9].

$$Nu_{L,Lam} = 0{,}664 x Re^{1/2} x Pr^{1/3} \qquad (4)$$

$$Nu_{L,Tur} = \frac{0{,}037 \times \mathrm{Re}^{0,8} \times \mathrm{Pr}}{1 + 2{,}443 \mathrm{Re}^{-0,1} \times (\mathrm{Pr}^{2/3} - 1)} \qquad (5)$$

By using the dry air properties in average temperature, Reynolds and Prandtl numbers can be calculated by employing Equations 6 and 7 [9].

$$\mathrm{Re} = \frac{\upsilon \times L_k}{\nu} \qquad (6)$$

$$\mathrm{Pr} == \frac{\nu}{\frac{k}{(c_p \times \rho)}} \qquad (7)$$

For the dry air properties in average temperature, the following equations were used [9].

$$k = 0{,}02418 \times \left(\frac{273 + T_{ort}}{273} \right)^{0,85} \qquad (8)$$

$$c_p = 1{,}005 + 0{,}006 \times \left(\frac{T_{ort}}{100} \right)^{1,73} \qquad (9)$$

$$\rho = 348{,}1 \times \left(\frac{P}{273 + T_{ort}} \right) \qquad (10)$$

$$\eta = 1{,}724 \times 10^{-5} \times \left(\frac{273 + T_{ort}}{273}\right)^{0{,}77} \quad (11)$$

$$\nu = \frac{\eta}{\rho} \quad (12)$$

At the end of these preparations, the amount of heat transfer from each of the regions to the passenger compartment was calculated by using Equation 13.

$$Q = U \cdot F \cdot \Delta T \quad (13)$$

3. Results and Discussions

As a result of test results and assumptions, the average temperatures of the regions under the vehicle floor were determined as shown in Table 7.

Table 7. Average temperatures of the regions under the vehicle floor [10]

Detail Region Name	Highway (%70) Temp (°C)	Uphill (%20) Temp (°C)	Idle (%10) Temp (°C)	Weighted Average Temperature (°C)
Above the Radiator	71.2	79.5	54.3	71.2
Above the Engine	72.0	80.4	56.3	72.1
Above the Exhaust	74.8	90.5	66.7	77.1
Above the Rear Left Luggage	55.1	57.3	57.0	55.7
Above the Transmission	47.5	61.5	59.2	51.5
Above the battery	54.7	57.8	56.6	55.5
Above the Rear Axle	41.8	38.4	45.3	41.5
Above the Luggage Com.	32.4	31.5	31.7	32.2
Above the Front Axle	36.3	34.0	36.6	35.9
Above the Fuel Tank	36.5	34.3	36.0	36.0
Below the Driver Platform	35.4	33.8	34.8	35.0

Heat transfer from each of the regions to the passenger compartment was calculated by means of the calculation tables, which one of a sample is shown in Table 8.

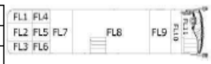
As can be seen from the sample calculation table (Table 8) Reynolds Number is over 500.000, so it can be said that the flow under the passenger compartment is turbulent.

The procedure explained above to calculate the amount of heat transfer was followed for each of the 11 regions. Then, the results were shown in Figure 4 and tabulated in Table 9.

Table 8: Sample calculation table of convective and conductive heat transfer

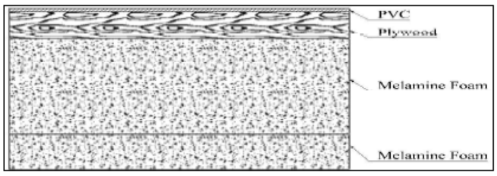
CALCULATION TABLE OF HEAT GAIN BY CONDUCTION AND CONVECTION

Main Region Name	Vehicle Floor
Main Region Code	FL
Detail Region Name	Above the Radiator
Detail Region Code	FL1
Heat Transfer Surface Area (m2)	0,964



Definition	Notation	Value	Unit
Inside Temperature	T_{in}	24	°C
Inside Conventional Heat Transfer Coefficient	h_{in}	8	W/m^2K
Outside Temperature	T_{out}	71,2	°C
Mean Temperature	T_{avg}	47,6	°C
Temperature Difference	ΔT	47,2	°C
Characteristic Length	L_k	0,7	m
Outside Air Velocity	u	50	km/h
Outside Air Velocity	u	13,9	m/s

REGIONAL SECTION OF HEAT TRANSFER



S.N.	Section Name	Material	Thickness (m)	Coefficient of Thermal Conductivity (W/mK)
1	Floor Lineleum	PVC	0,002	0,116
2	Floor Wood	Plywood	0,015	0,130
3	Main Insulation of Engine Room	Melamine Foam	0,054	0,040
4	Insulation Cover of Engine Room	Melamine Foam	0,020	0,040

PROPERTIES OF DRY AIR AT MEAN TEMPERATURE

Thermal Conductivity	k	0,028	W/mK
Specific Heat	c_p	1.006,661	J/kgK
Density	ρ	1,086	kg/m^3
Dynamic Viscosity	μ	1,95E-05	kg/m/s
Kinematic Viscosity	ν	1,80E-05	m^2/s
Prandtl Number	Pr	0,71	-
Reynolds Number	Re	541.030,96	-
Local Laminar Nusselt Number	$Nu_{L,Lam}$	411,12	-
Local Turbulant Nusselt Number	$Nu_{,L,Tur}$	1.168,41	-
Nusselt Number	Nu	1.238,63	-
Outdoor Conventional Heat Transfer Coefficient	h_{out}	57,05	W/m^2K
Overall Heat Transfer Coefficient	U	0,47	W/m^2K
Total Heat Transfer (Convection + Conduction)	Q	21,41	W

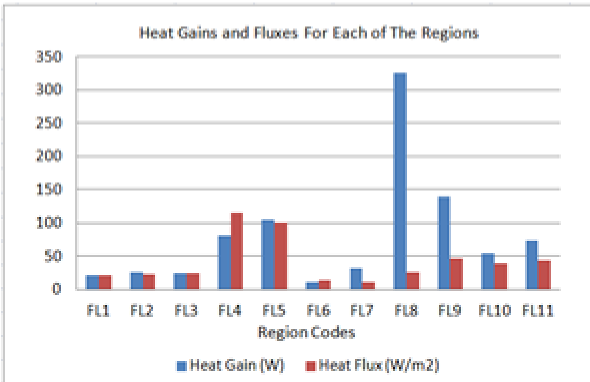


Figure 4. Heat gains and fluxes for each of the regions

Table 10 shows the relationship between climatic zones, physiological needs and the demands made on air conditioning performance based on a tour coach with a length of up to 12 m [2].

As can be seen from Table 10, total cooling load of a coach operating in warm temperature climatic zone is 32 kW, also the test vehicle has a 32-kW A/C unit. Here it can be understood that, the cooling load originated by the bottom of the passenger compartment is only 3% of the total cooling load of the vehicle. Here it can be seen that, the heat transfer from the bottom of the coach has a little percentage in total cooling load. The reason for this small rate is the efficiency of the isolation material (double layer and made of melamine foam) employed in the engine room. Although the main usage aim of this isolation is to reduce the noise coming from the engine room, it is also seen that current isolation materials are very efficient from the heat isolation point of view.

Table 9. Amount of heat transfer from vehicle floor region

Region Name	Region Code	Heat Gain (Watt)	Heat Flux (W/m^2)
Above the Radiator	FL1	21.4	22.2
Above the Engine	FL2	26.0	22.6
Above the Exhaust	FL3	24.1	25.0
Above the Rear Left Luggage	FL4	80.9	114.9
Above the Transmission	FL5	105.5	100.0
Above the Battery	FL6	11.6	14.8
Above the Rear Axle	FL7	32.2	10.8
Above the Luggage Compartment	FL8	325.4	25.9
Above the Front Axle	FL9	140.1	46.9
Above the Fuel Tank	FL10	53.9	39.9
Below the Driver Platform	FL11	73.3	43.4
Vehicle Floor	FL	894.4	32.9

Table 10. A/C power requirements in relation to the climatic region and the physiological perceptions of the passengers [2].

Climate Zone	Desire for...	Cooling Capacity	Heating Capacity	Fresh air Flap
Tropical rain and dry climates	... Perceptibly cool and cold air movement, partial dehumidification, often only recirculated air	44 kW	-	No
Warm temperate climate, continental climate, dry period in summer	... Heating depending on region, perceptible ventilation, recirculated or fresh air portion	39 kW	32 kW	Yes
Warm temperate climate, cold winter, rain	... in part higher heating output, low air flow, perceptible cooling in summer	32 kW	38 kW	Yes

From Table 9, it is seen that the maximum heat flux (Amount of heat transfer through per meter square area) is in the region above the rear left luggage compartment. Since there is no heat and sound source inside that compartment, extra isolation was not applied to this section. However when the mean temperatures of the regions were investigated it is seen that the average temperature of this region is 55.7 °C, and here is a high temperature region, because of the high amount of heat transfer from engine, radiator and transmission compartments. So it is advised an application of 20 mm thick, single layer isolation to this compartment to prevent the heat transfer to the passenger compartment.

Similarly, 2 mm steel plate placed under the transmission inspection lid, increases the amount of heat transfer to the passenger compartment. Because, this plate works as a heat storage. As a result of this study it is advised to change the inspection lid structure into a different type having isolation material, instead of steel plate.

Since the heat transfer area is higher than the other regions, it can be thought that heat transfer from luggage room region is very high. However, when heat flux values given Table 9 and Figure 4 were investigated, it was seen that this value is very low compared to the other regions. So, since there is no heat and sound source in the luggage room, there is no need to employ an extra isolation material to the luggage compartment region, as is applied currently. These results proved that the current application (Plywood without any isolation material) is suitable from isolation and economy points of views.

The heat transfer value above the front axle region is a little high compared to the other regions. So it is advised a single layer, 20 mm thick isolation material to this region, this will positively affect both the heat and sound isolation. This isolation will work especially for the sound created by the tires.

In the examinations, it is seen that there is no isolation material application to the region under the driver platform. However, the heat flux value of this region is over the mean value of the whole bottom. Application of isolation to this region can supply an extra comfort to the driver in the very cold winter times by preventing the heat transfer from the passenger compartment to out of vehicle.

Over the battery case, double layer isolation was employed and as a result of this application, heat flux value of this region is determined very low. Since there is no noise source in this compartment, by considering the amount of heat transfer through this region, as advised for the rear left luggage, application of 20 mm thick, single layer isolation will provide a little amount of economic advantage.

In the rear axle region, although the heat transfer amount is very low, by considering the noise created by the differential and the tires, application of double layer isolation must be kept.

4. Conclusion

As a conclusion, the amount of convective and conductive heat transfer from the bottom of the passenger cabin and engine room of a 12-m coach was calculated and examined in detail. At the end of the study; extra isolation materials are added to some regions to improve the passenger comfort by reducing the rate of heat transfer. In some of the regions, some portion of applied ineffective isolation materials are canceled without any lack of comfort and provide the manufacturer a cost saving. When evaluated from the heat and sound isolation point of views, it can be said that current engine room isolation material and application is suitable.

Nomenclature

U: Total Heat Transfer Coefficient (W/m^2K)
h_{in}: Inner convection heat transfer coefficient (W/m^2K)
s: Thickness of the material (m)
k: Thermal conductivity (W/mK)
h_{out}:Outer convection heat transfer coefficient (W/m^2K)
Nu: Nusselt Number
k: Thermal conductivity of dry air at average temperature.(W/mK)
L_k: Characteristic Length (m)
$Nu_{L,Lam}$:Local Laminar Nusselt Number
$Nu_{,L,Tur:}$Local Turbulent Nusselt Number
F: Heat transfer surface area (m^2)
ΔT: Temperature difference (°C)
Re: Reynolds Number
Pr: Prandtl Number
υ: Kinematic viscosity ($m^2.s^{-1}$)
C_p: Specific heat (kJ/kgK)
ρ: Specific gravity (kg/m^3)
T_{ort}: Average temperature (°C)
P: Pressure (Bar)
η: Dynamic viscosity (kg/m/s)

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