

Design and Modification of a Wall to Reduce Heat Transfer from Engine Compartment to Passenger Cabin in Medium Bus

by

GAMACHISA MITIKU TADESSE



A thesis submitted to

Department of Mechanical Systems and Vehicle Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirement for the Degree of Master of Science in
Automotive Engineering

Adama Science and Technology University

Adama, Ethiopia

June, 2019

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APPROVAL OF BOARD OF EXAMINERS

We, the undersigned, members of the Board of Examiners of the final open defense by **GAMACHISA MITIKU** have read and evaluated his thesis entitled“ **Design and Modification of a Wall to Reduce Heat Transfer from Engine Compartment to Passenger Cabin in Medium Bus**” and examined the candidate. This is, therefore, to certify that the thesis has been accepted in partial fulfillment of the requirement of the Degree of masters of Science in Automotive Engineering.

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CANDIDATE'S DECLARATION

I hereby declare that the work which is being presented in the thesis entitled “**Design and Modification of a Wall to Reduce Heat Transfer from Engine Compartment to Passenger Cabin in Medium Bus**” in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Engineering is an authentic record of my own work carried out from October 2018 to June 2019 under the supervision of Dr. V.Chandraprabu, (Asst. Prof.) Department of Mechanical system and Vehicle Engineering, Adama Science and Technology University, Ethiopia.

The matter embodied in this thesis has not been submitted by me for the award of any other degree or diploma. All relevant resources of information used in this thesis have been duly acknowledged.

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This is to certify that the above statement made by the candidate is correct to the best of my knowledge and belief. This thesis has been submitted for examination with my approval.

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Dr. V.Chandraprabu, (Asst. Prof.)	_____	_____

ACKNOWLEDGEMENT

First of all, I would like to thank the almighty God, for giving me the knowledge and wisdom and providing me the resources to acquire education. I also want to thank my parents and siblings without their support and guidance I wouldn't have been able to achieve what I have so far in life. And also i want thank all my friends, teacher and all the society supporting me by any things. I would like to express my sincere gratitude for my research advisor; Dr. V.Chandraprabu, (Asst. Prof.) for his excellent guidance, moral support and constructive criticism, corrections and helps. Finally, I would like to thank, mechanical engineering teachers and laboratory assistants for their continuous support and comments.

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NOMENCLATURE

Symbols

Q	Heat flow
k	Thermal conductivity
h	Total thickness
λ_n	Modified thermal conductivity
C_p	Specific heat
T^i	Inner temperature
T^o	Outer temperature
R	Thermal resistance
$U \& k$	Heat transfer coefficient
α	Thermal diffusivity
w	Mass flow rate, kg/s
V	Velocity of stream, m/s
A	Flow area, m ²
ρ	Density of fluid, kg/m ³
p	Absolute pressure, Pa
Q	Discharge flow rate, m ³ /s
A	Orifice area, m ²
$p_1 - p_2$	Pressure drop
m	Mass flow
dA	Differential area
U	Constant
Λ	Thermal Conductivity
q	Heat flux
T	Ambient Temperature
D	Diameter in mm
L	Length in mm
W	Width in mm
A_s	Surface area in m ²
r	Radius
T_s	Average of surface temperature

ρ_a	Density of air
Q	Heat (kW)
H	Heat transfer coefficient $\text{W/m}^2.\text{K}$

Abbreviations

AC	Air conditioning
CFD	Computational fluid dynamics
D2	Carbon tool steel
EC	Engine compartment
EPS	Expanded polystyrene
FC	Forced convection
GHG	Greenhouse Gas
GPS	Graphite Expanded Polystyrene
HB	Brinell hardness
HRC	Hardness in Rockwell C Scale
LSF	Load-bearing light steel frame
MDF	Medium density fiberboard
NC	Natural convection
RANS	Reynolds Averaged Naviere Stokes
TC	Thermal conductivity

ABSTRACT

Most of the vehicles in modern times have a cooling system throughout the vehicle interior which is helpful in cooling the vehicle from the heat generated due to different reasons like the environment condition, sun light, body temperature, engine temperature and others. The vehicle engine temperature is one of the main causes arising from engine compartment by the heat transferring through the wall between the cabin and the engine compartment which didn't get enough attention in designing and production of most vehicles mainly buses. this walls requires a special design and analysis in order to reduce the heat transferring and maintain the cabin temperature while adjusting temperature inside compartment itself which was the main objective of this thesis work, instead of searching for the solution of the problem which is already been created it is better to reduce it from the cause In first place and that is reducing the heat transfer from the engine through any possible way. So this work has included the activities like selecting, designing insulation system, modeling, analyzing and experimenting. Instead of using regular insulation system, air flow heat rejection system has been made which reduces the heat transfer by 80.84 % while maintaining and reducing engine inside space temperature by 11.6 % when compared with the common type insulation systems. After experimentation, this makes the new system better than other competent insulations and finally the outcome has brought positive results which are maintained and comfortable cabin (passenger) space and adjusted engine compartment temperature.

Keywords: Heat transfer, Firewall, Insulation, Heat rejection, Forced convection

CHAPTER ONE

INTRODUCTION

1.1 Background

The surface public transport brings an alternative for future and offers many challenges for vehicle producers to create more reliable, more safe and more comfortable vehicles, automobiles, locomotives, coaches, railcars, tramways, suburban and underground units other vehicles etc. Internal aerodynamics and thermodynamics include ventilation and heating the driver's cabins or passenger compartment, which play important roles in the vehicle success on the market. Professional codes for computational fluid dynamics and thermodynamics as the robust tools for industrial application helping engineers to carry out lots of design ideas. One of that type is in this contribution describing, the solution of ventilation and maintaining heat of the cabin.

Heat transfer in all industrial applications consists of three types (modes) in each moment. The heat transfer modes are conduction, convection and radiation. In the case, when the thermal quality of the cabin interior is simulated, only conduction and convection play dominant roles. Radiation heat transfer is not included in heat balance in the cabin space because radiation is strongly dependent on the temperature and the temperature range in the cabin is assumed low with a small influence on the total heat distribution [1].

Internal combustion engines produce a lot of heat during their combustion cycle. This has a negative effect when it reaches various sensitive components such as sensors, batteries, vehicle inside compartment body and starter motors. The engine temperature has an effect on the passenger cabin. As a result, thermal insulation is necessary to prevent the heat from the exhaust and combustion reaching these components.

Thermal insulation is the reduction of heat transfer (the transfer of thermal energy between objects of differing temperature) between objects in thermal contact. Heat flow is an inevitable consequence of contact between objects of different temperature.

Thermal insulation provides a region of insulation in which thermal conduction is reduced and can be achieved with specially engineered methods or processes, as well as with suitable object shapes and materials where the separation of the engine compartment and the passenger cabin parts are located which are almost in contact with the driver and some of the passengers[2] sufficiently high surface temperatures on the inside of external

components during the winter, contributing to ensuring a feeling of comfort inside the room (comfort promoting thermal insulation); this allows for the same level of comfort to be achieved with lower room air temperatures and thus less energy consumption, which reduces undesired heat input and thus the overheating of rooms during the height hot season [3] .

Generally, the maximum temperature should not rise at any single point to be higher than 180°C, and the mean temperature rise across the whole sample must not be higher than 140°C and, most primary fires start in the engine compartment. The danger level is high with many types of fuels in a confined space together with the exhaust system's high temperatures. The most common location of the bus engine is at the back of the bus. Although there are also some buses with the engine located in the middle of the bus, lying centrally in the floor or standing on the side. To achieve a high level of passenger comfort with respect to temperature and noise, there must be thermal and acoustical insulation between the engine and passenger compartments. This insulation can simultaneously provide high fire resistance, provided all surfaces are covered. [13].

1.2 Statement of the problem

Most medium buses and old model buses are widely used type of buses for public transportation purpose but most of them are imported from the source country as a load carrying vehicles other than humans which will be later reshaped to public transporting vehicles which has lots of drawbacks. Local body and frame production companies are participating in removing the original body and replacing with a new locally made body. So in case of these medium buses the engine compartment position becomes exposed to the passengers and the driver which is directly below the front seats where the driver and the opposing side passenger's seats are located. They are highly exposed to the separation wall but most of these walls in between are not designed considering many thermal requirements and vehicle service year therefore; the heat generated below affects passengers comfort. This needs full attention since the heat coming from the engine combustion process, exhaust line, thermal reactions and other reasons will not be stopped with just a simple sheet metal which is common on different buses. In most area of country (Ethiopia), hot weather condition makes the case worse for the driver and passenger inconvenience which also causes skin diseases; driver may become sleepy causing different car accidents. Also some part of the engine like electric wires, hoses and others which are made of plastics will lose their property due to repeated hot temperature inside

the compartment which requires the insulation between the passenger room and the engine room to cool itself.

1.3 Objectives

1.3.1 General Objective

The general objective of the thesis is to design and modify, wall which reduces heat transfer from engine compartment to cabin and to analyze the engine thermal condition for midi bus vehicle.

1.3.2 Specific Objectives

The specific objectives are:

- Selecting materials with low thermal conductivity for insulation purpose in relation to the heat generated.
- Modeling and checking effect of material type using CATIA software then checking for the temperature difference and identifying critical positions using ANSYS software.
- Comparison of previous system and materials with the newly proposed solution.
- Experimenting the prototype and checking the temperature difference.
- Compare the inside engine compartment temperature results.

1.4 Significance of the thesis

The significance of the proposed thesis works,

- Maintaining the temperature of the vehicle cabin from the cause (source).
- Maintaining the temperature mainly near the engine compartment.
- Adjusting engine temperature.

1.5 Beneficiaries

- Automotive companies, spare part suppliers, owners of the vehicle, researcher generally the community using transportation means are the beneficiaries.

1.6 Delimitation (Scope)

The engine position is different from vehicle to vehicle, which is placed in the front end, back, close to front end below the cabin depending on the vehicle type, while this project can be used for many type of vehicle depending on their engine position, This MIDI BUS was selected due to the front location of engine compartment which is fully exposed to the cabin from above and partially from sides so this thesis were limited in designing and modifying the model showing heat transfer load distribution using CFD also experimenting the prototype using own test setup for front placed engine vehicle type.

1.7 Organization of the study

This study consists of five chapters. The first chapter is introduction about the research study. It includes the background of study, problem statement, objectives, and scopes, significance of the study and the structure of report. Chapter two focuses on the literature review based on the previous related works and experiments done. Chapter three describes materials and methodology of the procedures followed in experimentation, analysis and modeling which was done. The results and discussions are presented in chapter four. The generated data of heat transfer, temperature difference, heat transfer coefficient and modeling and analysis and experimentation results are presented in form of tables and graphics for quantitative and qualitative discussion respectively. The temperature distributions are also discussed to understand the best design characteristics. The correlation between the experiment result and simulation result is also discussed. The final chapter of this thesis is conclusion and recommendation.

CHAPTER TWO

LITERATURE REVIEW

2.1 Heat transfer to the cabin and importance of insulation

Tadeusz and Zbigniew (2010) have made a research that there are several important factors influencing the road accidents. Temperature inside the vehicle is ranked third after alcohol and seat belts. For this reason, maintaining thermal comfort in the passenger compartment is essential. Thermal comfort is provided by the air conditioning system, which consumes more energy. In the case of electrically powered vehicles, this results in a shorter range. Optimization of such systems is therefore required. The article proposed a set of equations describing the thermal conditions inside the vehicle, which are the result of appropriate energy balances for air, interior elements, the authors study focused on unsteady air-conditioning of the vehicle interior.

The measurement data was compared with the results obtained through numerical solutions of the proposed set of equations. When the turbulence is higher than 20 %, extensive blending occurs at the stream threshold between the air blown into the compartment and the interior air. This leads to an increase in the amount of air in the stream with an increase in the distance from the air inlet opening. In the low turbulent flow, with the degree of turbulence between 10 and 20 %, cold air flows in and causes the warm interior air, which absorbed heat from the surrounding objects, to go up by convection. As a result, the differences in temperature in the vertical cross-section are reduced considerably, which makes the heat transfer study an important issue. So **Bahrami (2009)** has indicated in thermodynamic the amount of heat transfer as a system undergoes a process from equilibrium state to another were considered. Thermodynamics gives no indication of how long the process takes. In heat transfer, the rate of heat transfer is a concern. Since the basic requirement for heat transfer is the presence of a temperature difference. Temperature difference is the driving force for heat transfer, the total amount of heat transfer during a time interval needs to be determined. Since steady heat conduction in Plane Walls, conduction is the transfer of energy from the more energetic resistance to heat flow with low particles of a substance to the adjacent less energetic ones as result of interactions between the particles.

The reviewed paper has defined that Insulators are materials primarily to provide thermal conductivity. The figure 1 shows the model of the system used in the

article. Thermal resistance network consider steady one dimensional heat flow through two plane walls in series which are exposed to convection on both sides, under steady state condition.

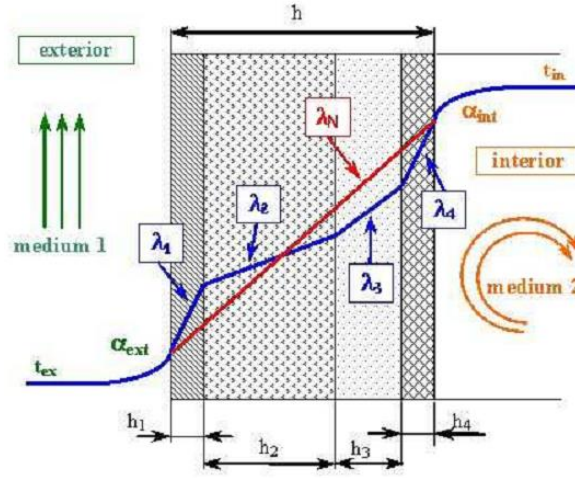


Figure 1: Heat transfer model [1]

$$\begin{matrix} \text{Rate of heat} & & \text{Rate of heat} & & \text{Rate of heat} & & \text{Rate of heat} \\ \text{Convection} & = & \text{conduction} & = & \text{conduction} & = & \text{Convection} \\ \text{Into the wall} & & \text{through the wall 1} & & \text{through the wall 2} & & \text{from the wall} \end{matrix} \quad \dots 2.1$$

Schuster (2007) has dealt with industrial application of numerical methods to the prediction of thermal situation in the rail vehicle interior. Basic principles of heat transfer were summarized to explain both theoretical background of simulations and engineering approach to solving temperature conditions in the vehicle interior. The main part of the contribution describes the solution of the locomotive driver's cabin heating and controlling the temperature levels. This contribution is a brief overview of both possibilities of engineering modeling of heat transfer modes and results in the simulation of the real locomotive cabin heating/ventilation system design. This shows that Fluent CFD code allows incorporating the numerical models for heat transfer by relatively simple way:

- The wall model
- Conduction model
- The convective heat transfer model

Conductive heat transfer which is caused by “diffusion” of heat due to temperature gradient. Heat flux q_s (W/m^2) through the wall of thickness h_i were written:

$$q = \frac{\lambda_i}{h_i} (t_{s2} - t_{s1}) \quad \dots 2.2$$

where t_{s1} and t_{s2} are temperatures of cold or hot wall surface and λ_i (W/m.K) is thermal conductivity of the solid walls material. The locomotive cabin consists of “multi-layer” walls. They are composed of the external sheet metal layer, of several parts of the cabin body as beams or supports and finally of the noise and the thermal insulation layers. In this case term can be modified:

$$q = \frac{\lambda_n}{h} (t_{s_k} - t_{s_l}) \quad \dots 2.3$$

Where h is the total thickness of the “multi-layer” wall. The total number of layers is $n = k - 1$. λ_n (W/m.K) is the modified thermal conductivity of the “multi-layer” wall, which is

$$\lambda_n = \frac{h}{\sum_{i=1}^n \frac{h_i}{\lambda_i}} \quad \text{Where number of layers is } i = 1, 2 \dots n. \quad \dots 2.4$$

This contribution dealt with the simulations of the thermal phenomenon which arise in the driver’s cabin during the heating or cooling regimes. The main part of the contribution described the possibilities of engineering heat transfer simulations. The Fluent CFD code models were created, verified and used for the convective and conductive heat transfer calculations. The Fluent heat transfer models were used for the simulation of the thermal conditions in the cabin inner space during the extremes heating for winter and cooling for summer. The obtained results of these simulations enabled to confirm the conditioned system with the tested power-output is able with reserve to keep comfortable conditions in the cabin. The average values of air temperatures near the driver’s body are more than 24°C for a heating regime and less than 18°C for a cooling regime) and they satisfy the standard requirements. **Dragan and Ferenc (2012)** discussed about the main methods for both the improvement of the microclimatic conditions and the reduction of air-conditioning energy consumption in any type of cabin in two ways:

Passive prevention of:

- Cabin heat gain, using principles of heat rejection and insulation;
- Heat transfer to the operator’s body by reducing the transferred solar radiation, as well as thermal radiation and heat transfer from the cabin interior surfaces.

In Active cooling, the operator's body uses the effective use of airflow from the air-conditioning system. The aims of the research were:

- To identify and evaluate the most important influences on heat transfer processes in the human-cabin-environment system.

- To explain the effects of the ventilation (air distribution) system design and settings on the operator's heat loss and thermal sensation in hot conditions.

The floors were made of a steel sheet supported by a steel frame. For the purposes of sound and heat insulation, the floor has an interior lining and, in some designs, an outer lining. The powertrain (transmission) of typical tractors is positioned beneath the cab. The powertrain of smaller tractors partly "protrudes" into the cab space, while larger tractors have a flat floor. In the interior, rubber or polymer flooring is used. Some cabs have a foam insulation material between the flooring and the floor. If there is additional heat and sound insulation beneath the floor, composite materials are used. The typical composite heat shield is made of reflective outer aluminum layer bonded on synthetic fiber core.

Analysis and optimization of the air conditioning system, experimentally verified numerical methods (CFD) with computer simulated person or virtual thermal manikin are an inevitable tool. Heat gain from the powertrain Heat transfer through the floor was modeled as the sum of the radiation from hot powertrain surfaces and the natural convection from the surrounding air, as in. It was assumed that the air and the outer surfaces of the powertrain have the temperature of 80°C (based on temperature of the transmission oil; data from tractor service manuals). In addition to the radiative properties and the temperature of surfaces, geometry also has some influence on the heat exchange. The geometry is described by the view factor. Because the distance between the powertrain and the floor is small, it is assumed here that the view factor is equal to unity. For a simple floor without a heat shield on the outer side and with rubber flooring only, the estimated value of the heat flux through the cab floor caused by powertrain is around 110W/m². If there is a heat shield beneath the floor and insulation between the flooring and the metal, the heat gain can be reduced to one third. Consequently, in the first case, the inner temperature of the floor will be above 50°C, while in the second case the temperature will be lower, around 34°C.

Tseng et al. (2014) has understood that automobile air-conditioning systems are not very efficient even for use in conjunction with internal combustion engines. This problem is more critical if they are to be used in electric vehicles because conventional systems consume a significantly more energy and therefore tremendously shortens the travel distance per charge of an electric vehicle. Besides the development of new generation air-conditioning system, one of the easier remedies is to reduce the heating effects of a cabin so that the requirement for air-conditioning can be minimized and thus its energy consumption by the system. this work aimed in identifying some of the key

factors that may help minimize the cooling load of an electrical vehicle. Experiments were made with 5:1 reduced scale experimental cabin model has been built based on the actual measurement of a Formosa Martiz. Since the distance between the light Source and the car model was controlled within 0.5 cm, the error of the similar solar flux was 9W/m^2 . Since the car cabin in this work is a reduced model by 5:1, the solar flux employed here is equivalent to a solar flux. The temperature variation at different locations associated to the car cabin model. Were compared correspond to the case whose glasses are tinted or not by window film installation, respectively. Based on the above values; the simulated solar flux q was roughly 278W/m^2

2.2 Heating rather than cooling

These articles show that the other world is searching for a solution to heat the passenger cabin not to cool them down since their environment condition requires heat so **Wirth et al. (2014)**, unlike combustion engines, electrified powertrains have much lower levels of waste heat that can be used to heat the passenger cabin. Additionally, the refrigerant compressor that is traditionally linked directly to the internal combustion engine needs to be powered electrically. Consequently, in fully electrically powered vehicles, independent heating and air-conditioning systems need to be implemented, and these have to be powered by the vehicle's battery itself. The operation of these systems has immediate effects on the driving range, thus resulting in conflicting objectives between driving range and thermal comfort, the heating and air conditioning of the passenger's cabin was accomplished through a temperature-conditioned air mass flow.

The methodological coupling of CFD based DoE with an automotive-specific criteria catalogue has led to an objective conclusion basis that allows clear recommendations for an insulation strategy to be developed. The strategy has been implemented for the optimization of the smart electric drive's thermal insulation. **OumSaad et al. (2015)** has found out that heating load calculation is based on peak or near-peak conditions, which normally occur early in the morning before sunrise, in addition, in cold countries, the ambient temperature variation during the winter months is not significant. The development of air conditioning system has been growing rapidly due to people needs of having comfortable condition. The cooling system is one of the important systems, because it is related to passenger comfort and important to study transient response of vehicle under real driving conditions. The procedure of this research

was simulation analysis using MATLAB program that uses mathematic equation, made the cabin model, etc. The heat occurred in the cabin consists of metabolic, direct, diffuse, reflected, ambient, exhaust, engine, ventilation, and AC loads has been calculated with three different time's condition which was performed using cooling load simulation tool of MATLAB/GUI. A negative cabin load was provided to the cabin in order to decrease its temperature to the comfort level. the cooling load calculation is presented on the basis of some parameters such as metabolic load, direct load, diffuse load, reflected load, ambient load, exhaust load, engine load, ventilation load, and AC load. The cooling load in the car cabin was calculated by the GUI program from MATLAB software.

2.3 Heat transfer reduction

Nicholas and Patrick (2014) have made a study on the use of latent heat thermal energy storage for thermally buffering vehicle systems. Vehicle systems with transient thermal profiles are classified according to operating temperatures in the range of 0–800°C. Thermal conditions of those applications are examined relative to their impact on thermal buffer requirements, and prior phase change thermal enhancement studies for these applications are discussed. In addition a comprehensive overview of phase change materials covering the relevant operating range were given, including selection criteria and a detailed list of over 700 candidate materials from a number of material classes. Promising material candidates are identified for each vehicle system based on system temperature, specific and volumetric latent heat, and thermal conductivity. Based on the results of previous thermal load leveling efforts, there is the potential for making significant improvements in both emissions reduction and overall energy efficiency by further exploration of PCM thermal buffering on vehicles. Recommendations were made for further material characterization, with focus on the need for improved data for metallic and solid-state phase change materials for high energy density applications. Here, **Mao et al. (2010)** made a full-size of machine-level; under-hood thermal analysis has been conducted using CFD. In particular, an iterative approach has been developed and was used to speed up the convergence of the numerical solution of the energy equation, where the large heat source term poses a computational challenge. The Reynolds Averaged Naviere Stokes (RANS) method was applied to simulate under-hood turbulent flow, while a net- radiation enclosure model was included in the energy equation. The numerical results are used to verify and validate the concept design and prototyping of off-highway

heavy-duty trucks, including parameters and components such as: the speed of cooling fans; core size of the heat exchangers; temperature; pressure drop; thickness of insulation and design options of the heat shield. Another objective is to bench mark a one-dimensional lumped in house tool for industrial cooling system concept design and analysis. The CFD method is a powerful tool that can identify possible design drawbacks, evaluate design options, improve the heat transfer efficiency and provide optimal engineering solutions and strategies for complicated heavy-duty thermal management.

The temperature at the engine exhaust manifold is more than 700°C and, thus, insulation at the inner wall of the pipes that link to the after-treatment equipment's is required. So they used different insulation materials and thicknesses are dictated by the design standard used. Also recommended that a heat shield be placed between the "hot box" (combustion chamber) and the control instruments close to the "hot box".so the result were If the temperature is above the temperature limit one may need to protect other neighboring components from the after-treatment equipment. An improved case which is based on an improved insulation layer, a heat shield is a good option for most industrial designs and component manufacture. In order to validate the thermal analysis results of the system, so several critical components and sub systems were selected for comparison of the surface temperatures in which some values have been measured in test bed while other data obtained from field test.

2.4 Engine compartment temperature

Binner et al. (2006) have investigated thermal Protection is of vital importance in the development of passenger cars. In the last years thermal issues have been increasingly investigated by numerical means. Computational fluid dynamics (CFD) is today an efficient way to optimize the aerodynamics of a new car or to investigate the under hood flow, especially the mass flux through the heat exchanger package. Additionally the computation of under hood flow field and air temperature distribution is today common practice. This allows, for example, an engineer to judge the performance of the engine fan and vents as well as look for hot air spots within the under hood. The computations of two standard steady state load cases – 240km/h and 40km/h hill climb are shown and some results were briefly discussed. The computation of component temperatures in the engine compartment requires a multimode heat transfer analysis considering conduction, convective heat transfer and thermal radiation. Due to very high temperatures next to the exhaust manifold, in particular, a thermal radiation analysis was found to be required

which were described an under hood thermal methodology based on coupling sophisticated and specialized tools.

The methodology used has computed a new steady-state load case, briefly described as idle without operating fan. Load case was defined as a first step in order to compute thermal soak. Experimental and numerical results are compared in order to validate the commercial CFD code Star CD™ for buoyancy driven flows and the thermal analysis with natural convection. The experimental Setup of A radiator and small fans are provided as an additional heat exchanger operating outside the car. The radiator was built up in a chamber next to the test rig with the car. The coolant flow was diverged from the main stream by flexible tubes and a variable pump. T-junctions are used to set up two parallel fluid streams over the car's radiator and the external one.

A 1D study revealed that only a small part of the coolant actually passed through the car's radiator for this experimental setup. Additionally the flow through this radiator was inverted. Nevertheless the setup is able to cool the engine and the whole engine compartment. In this section some results of the additional load case "idle without fan" are presented. The surface heat may reach 250°C in critical condition while first the 115°C coolant temperature configuration is investigated. Additionally the modification with closed grill and a coolant temperature of 105°C is computed and compared to the open one. **Hammarström et al. (2008)** have done one part of the project to identify designs and routines that create high risk for fire and means to mitigate these risks. Fires are typically a result of heat, vibrations, defective maintenance, and compromises in the design. Most fires start in the engine compartment and therefore actions to reduce fire risk were focused on this part of the vehicle and suggested an action like ,Insulating all hot surfaces in the engine compartment (thermal insulation), Equipping the engine compartment with better detection systems in combination with fire extinguishing systems.so this article tells us that insulation (I) is the ability of an element with fire on one side to maintain the temperature of the unexposed side below a certain level. Some times higher than 180°C and the mean temperature rise across the whole sample must not be higher than 140°C and, most primary fires start in the engine compartment. The danger level is high with many types of fuels in a confined space together with the exhaust system's high temperatures.

High temperatures, hot surfaces and a variety of combustible materials make the engine compartment a high risk area. This means that the engine compartment design, including parts, must be of a high standard and quality to ensure a low fire risk. **Addendum 117: Regulation No. 118.** States that approval of a vehicle type with regard to the burning

behavior and/or the capability to repel fuel or lubricant of the components used in The interior compartment, the engine compartment and any separate heating Compartment. "Interior compartment" means any compartment intended for passengers, Drivers and/or crew. The interior compartment shall be bounded by the roof, floor, side walls, doors, outside glazing, the rear compartment, bulkhead, or the plane of the rear seat and Back support. While, "Engine compartment" means the compartment in which the engine is installed and in which a combustion heater may be installed.

2.5 Possible fire wall and under hood design

2.5.1 Fire resistance

Gerlich et al. (1996). Have analyzed Light steel-frame building systems are becoming more prevalent in commercial, industrial and residential construction and investigates the performance of load-bearing light steel-frame systems exposed to fire. Methods are presented for calculating the reduction of steel strength and stiffness at elevated temperatures, and for predicting the deflections resulting from temperature gradients and P-A effects. Heat transfer modeling by computer is used to predict steel framing temperatures for systems exposed to the standard ISO 834 time temperature curve and real fires. Three full-scale furnace tests were carried out to evaluate analytical predictions. A design procedure is proposed starting the thermal properties the temperature rise of a steel member as a result of heat flow is a function of the thermal conductivity and specific heat of the material. Therefore the thermal conductivity as a function of temperature is given by the following equation:

$$k = - 0.022T + 48, \text{ for } 0 < T < 900^{\circ}\text{C} \quad \dots 2.5$$

Where k is the thermal conductivity (W/mK)

The specific heat (C) for most steels increases gradually with temperature. A constant value of 600J/kgK is suggested for temperatures below 600°C, data for higher temperatures are not required. The study was carried out to gain understanding of the performance of load-bearing light steel frame (LSF) walls and to model their performance against the ISO 834 and real compartment fires. Structural testing at room temperature was carried out to check relevant structural design codes. The effect of temperature on the steel strength and stiffness was investigated. In order to predict stud bending moments, analytical methods are proposed for calculating the thermal deflection arising from a temperature gradient across the steel members and the superimposed

deflection due to P-A effects. Heat transfer modeling using TASEF was carried out to predict steel framing temperature. Three full-scale furnace tests were carried out to evaluate the proposed model against the ISO 834 time-temperature curve and a significantly hotter test fire.

2.5.2 Wall plus Air cooling system

Wang et al.(2013) have studied that the engineering vehicle radiator module is an important part of the cooling system using air as a medium. The heat transfer performance of a radiator module has a direct impact on the state of cooling system thermal equilibrium. Therefore, it is essential for the engine vehicle to make accurate calculation of the heat transfer performance. Both the air flow rate through the radiator module and air flow distribution in the engine compartment are important parameters affecting the heat transfer performance of the radiator modules. In the simulation, the characteristic of air flow through the radiator module in engine compartment is researched. The simulation results were validated with experimental data. The research results indicate that the hot air reflow in engine compartment is the main factor that leads to radiator module overheating. Since the air flow rate through the radiator module in the engine compartment is an important parameter affecting the heat transfer performance of the radiator modules. With the rapid development of the technology of computing power and numerical calculations, computational fluid dynamics (CFD) simulation analysis as a design research tools, and applications in the research of the engineering vehicles. Compared with the experimental study, the use of the CFD technology has high calculated speed, high precision, low cost and can be simulated liquid flow in complex structure. Boundary conditions and impact parameter can be changed analysis of various parameters for the impact of the research system will be made, then validate the results of the analysis by the test data, and reduce the number of test to save development cost and shorten the development cycle.

For experimenting the boundary conditions are that in the entrance is velocity inlet, and pressure outlet at the exit. The remaining four surface of virtual wind tunnel were set with the wall, and the wall is the standard wall, no-slip boundary wall. Under the corresponding working conditions, with the principles of the relative velocity, the entrance condition of virtual wind tunnel using different uniform air flow velocity as the speed of the inlet boundary. Analyses were focused on air flow distribution in power compartment. Then

based on the law of air flow distribution, the characteristic of air flow through the radiator module simulation data with test contrast, the results are in good agreement. It was proved that the computation method is feasible. There is air reflow phenomenon between the top of the power compartment and the top of the radiator module. Due to the presence of air reflow, resulting in part of hot air flowing through the radiator module to re-flowing into the radiator modules. This part of the recirculation of hot air causes the radiator heat exchanger performance degradation. In the design process improved the design to reduce reflux.

Tripanagnostopoulos et al. (1996) have made a study on Hybrid PV-Thermal systems which can be integrated on building facade and inclined roof, instead of plain PV modules, providing simultaneously electricity and heat. Heat extraction by air circulation is considered a simple PV cooling mode, by which PV electrical efficiency can be kept at a satisfactory level, building undesirable heating can be avoided during summer and space heating of building can be achieved during winter. Aiming to low cost improvements of building integrated air cooled hybrid PV-Thermal systems was tested experimental models consisting of pc-Si PV modules and air channel at PV rear surface, regarding system inclination, air channel depth, air channel opposite surface emissivity and heat extraction modes. A water cooled PV module, a PV module with back insulation and a PV module with both surfaces free to ambient were also tested in order to compare their performance with that of the air cooled PV systems. From the results it is estimated that simple heat extraction modes as the roughened opposite channel surface and the placing of thin metallic sheet inside air channel can be considered low cost system improvements. High air flow rate increases the thermal output of the air cooled hybrid PV systems, but the electrical input by the fan (to overcome the increased pressure drop) reduces the net electrical gain. The systems were thermally insulated and during tests air was circulating through the formed air channel as shown in *figure 2*. In all performed tests the system with the modifications (PV/S) was tested in comparison with the reference system (PV/R) for the same operation conditions (position mode, weather conditions), but with several changes in the air channel (air channel depth, addition of fins, tubes, flat sheet). Pyrometer were used to measure the incoming solar radiation on PV plane, air flow meter to measure air flow rate and thermocouples to measure temperature at several points on the devices (front and rear PV surface, input and output of air, opposite channel surface, ambient temperature, etc.)

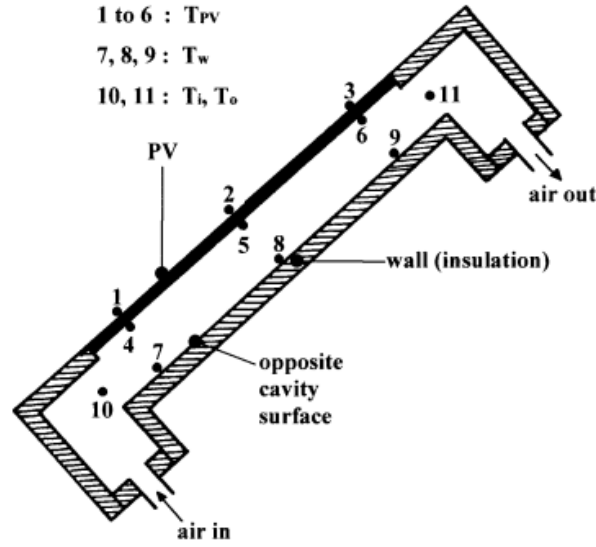


Figure 2: Air flow cooling [17].

High air flow rate increases the thermal output of the air cooled hybrid PV systems, but the electrical input by the fan (to overcome the increased pressure drop) reduces the net electrical gain. Wide air channel depth results to a lower pressure drop, reducing the needed electrical input by the fan, but decreasing the thermal output of the system.

Florent and Maroun (2017) have also studied wall model where there is thermal network between the cabin interior and the outside environment, the thermal balance at the outer wall node T_{wall}^{out} applied (by convention, heat flow is positive when heat goes from outside into the cabin). Similarly, thermal balance at the inner wall node T_{wall}^{in} was expressed. The model requires a full list of parameters. There are not presented in detail here due to conciseness considerations. Basically, the car geometry was described by the wall areas and orientations. In addition, thicknesses and thermal parameters are required. For the following results, Parameters from a mid-size car are applied in this paper. With an internal mass separated from the air cabin by a convection resistance have been analyzed. Integrating directly the internal mass capacitance to the air, (i.e. without convection resistance) increases by 9% and 28% respectively the heating needs and the cooling needs. In addition, if the inner radiative couplings are considered between the walls and the air (instead of between the walls and the internal mass), the cooling needs are increased by 15%. Of course, those two approaches do not represent the reality, but this analysis emphasizes the fact that the internal mass has to be considered properly.

2.6 Under hood thermal issues

Lam et al. (2012) have made a study on temperature variations inside a car under hood are largely controlled by the heat originating from the engine block and the exhaust manifold. Excessive temperatures in the under hood can lead to the faster deterioration of engine components and may affect the thermal comfort level inside the passenger cabin. On the paper the computational fluid dynamics investigations to assess the performance of a heat shield in lowering the peak temperature of the engine components and firewall in the under hood region of a typical passenger car were presented. The simulation used the finite volume method with the standard k- ϵ turbulence model and an isothermal model for the heat transfer calculations with Some Engine components properties shown in *table 1*. The results show that the heat shield managed to reduce the peak temperature of the engine components and firewall by insulating the intense heat from the engine block and exhaust and regulating the airflow inside the under hood region. Boundary conditions used

Table 1: Some Engine components properties [19]

Components	Material	Thickness	Temperature
Engine	Steel	0.5 cm	365 K
Exhaust manifolds	Steel	0.4 cm	673 K
Battery	Aluminum	0.1 cm	0 W/m ²
Oil filter	Aluminum	0.1 cm	0 W/m ²
Firewall	Steel	0.3 cm	Coupled
Car wall	Steel	0.2 cm	Coupled
Heat shield	Steel	1.5 mm	Coupled

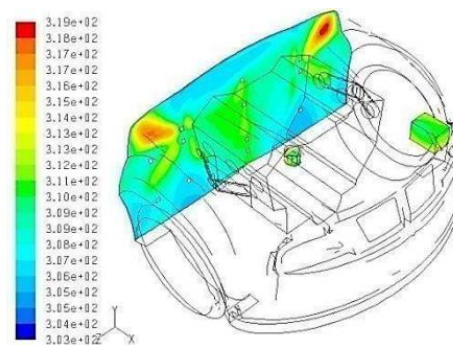


Figure 3: CFD under hood heat distribution [19]

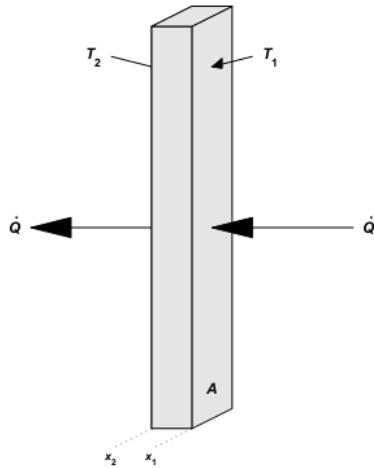
The commercial CFD code FLUENT was used for the simulations which was represented on *figure 3*. In brief, FLUENT applies a standard Finite Volume Method to solve the

governing continuity, momentum and energy equations. For the current problem, pressure-velocity coupling was achieved using the SIMPLE algorithm, available as standard in FLUENT (ANSYS Inc., 2010). In this study it was found that the proposed heat shield design can act as a good measure in improving the heat distribution and cooling performance on engine components, regardless of its positioning. A heat shield not only serves as an insulator, but also helps to regulate the air flow in the under hood. Simulation results show that the distribution of heated air in the under hood is more regulated as the heat from the heat source is channeled in a more appropriate manner, resulting in lower peak temperatures for various under hood components. The study showed the potential of the CFD method in optimizing the cooling processes in the under hood, as heat dissipation and air circulation within the under hood can be designed out. **björklund (2004)** has reviewed what Biot (1804) and Fourier (1822) came up with which is the equation describing the fundamentals for heat Transfer in a material. In the one-dimensional case the heat flow Q through a wall with area A is of figure 4.

$$Q = -\lambda A \frac{\partial y}{\partial x} \dots 2.6$$

Where λ is the thermal conductivity,

This equation is called Fourier's law of heat conduction. If we assume that there are no heat sources or sinks in the material, Q has to be equal through the whole wall. λ were assumed to be constant in the wall material and the wall face temperatures to be T_1 and T_2 .



$$Q = \lambda A \frac{T_1 - T_2}{x_1 - x_2} \dots 2.7$$

Figure 4: One-dimensional heat flow [20].

If multi-layer wall, as in the case with the walls in a truck cabin, Q must still be the same in all part of the wall. For a three layer wall that is

$$Q = \lambda_a A \frac{T_1 - T_2}{\Delta x_a} = \lambda_b A \frac{T_2 - T_3}{\Delta x_b} = \lambda_c A \frac{T_3 - T_4}{\Delta x_c} \dots 2.8$$

So solving for Q becomes

$$Q = A q \frac{T_1 - T_4}{\frac{\Delta x_a}{\lambda_a} + \frac{\Delta x_b}{\lambda_b} + \frac{\Delta x_c}{\lambda_c}} \dots 2.9$$

Here is found that the heat flow is proportional to the wall face temperatures. It will not introduce a non-linearity., the thermal conductivity in the layers and the thickness of the layers were found. The denominator was estimated in by measurements of the heat flow.

2.7 Experimental setup

khalid et al. (2011)presented paper reports of an experimental study of the aero thermal phenomena in the vehicle under-hood compartment as investigated by measuring temperature, convective heat flux, and radiative heat flux. Measurements are carried out on a passenger vehicle in wind tunnel S4 of Saint -Cyr-France. The under hood space is instrumented by 120 surface and air thermocouples and 20 flux meters. Measurements were performed for three thermal functioning conditions while the engine is in operation and the front wheels are positioned on the test facility with power-absorption-controlled rollers. In the thermal analysis, particular attention is given to measuring absorbed convective heat fluxes at component surfaces.

It is shown that, in some components, the outside air entering the engine compartment (for cooling certain components) can in fact heat other components. This problem arises from the under hood architecture, specifically the positioning of some components downstream of warmer components in the same air flow. Optimized thermal management suggests placing these components further upstream or isolating them from the hot stream by deflectors. Given style constraints, however, the use of air deflectors is more suitable than under hood architectural changes. Much of the present paper is devoted to heat flux analysis of the specific thermal behaviors in the under hood compartment (especially the absorption of convective heat fluxes) and to a description of a new control approach exploiting air deflectors to optimize under hood aero thermal management. The combustion temperatures for motor fuel are around 2200°C and the temperatures of exhaust gas at the cylinder head outlet are of the order of 1000°C. It is therefore necessary to cool the piston, valves, and cylinder walls in order to prevent them from melting. The difference between primary energy and combustion engine power is then arranged among exhaust energy, energy lost by convection (in the cooling system and under the hood) and energy lost by radiation. The problem of under hood aero thermal management essentially entails controlling these heat exchanges and their impact on component temperatures.

By considering the surface temperatures and the convective heat transfer coefficient measured by the two flux meters to be approximately the same, one can deduce from the

overall heat fluxes measured by the two flux meters the convective and radiative heat flux exchanged at the surface. Aero thermal experiments were performed in wind tunnel S4 of Saint-Cyr. Wind tunnel Experiments were carried out for three different thermal functioning points that simulate more or less severe rolling situations from the thermal point of view. The thermal soak phase follows the slowdown and simulates the vehicle stopping after a significant heat load. Here thermal inertia maintains temperatures higher even though the engine has stopped. In this case, cooling airflow under the hood is achieved only by fan rotation and/or free convection.

The present paper gives a physical analysis of particular under hood aero thermal behaviors, especially convective heat flux absorption, and presents a new optimization procedure based on this physical analysis that entails redistribution of cooling airflow in the vehicle under hood compartment. The results from It has been shown by some examples that outdoor air entering the under hood compartment to cool it by convection can in fact heat up some components too, while **Wischhusen et al. (2012)** described a modular system level modeling approach for the thermal behavior of an automotive cabin. The model is parameterized with geometric and physical data. At the end a set of 6 parameters is used to calibrate the model with two measurement data sets: one for a passive heat up and active pull down and one for a cold heat up. The procedure can be used as a recipe for developing own models of the same kind which were used in integrated thermal management studies. Automotive cabins, calibration, thermal simulation, air conditioning, are integrated thermal management. The assessment of the thermal behavior of an automotive cabin as a part of the whole vehicle becomes more and more important while the air conditioning system is not just responsible to cool and heat the passenger compartment but also has to condition other sensitive parts. Alternative, energy-saving vehicles concepts require innovative concepts to manage multiple heating and cooling loads. This has to be achieved by thoroughly optimizing many factors: e.g., energy consumption, component utilization as well as life-time reduction and passenger comfort. The following physical effects are modeled by the component models of the library: Heat transfer by convection, Heat transfer by conduction, Heat transfer by direct and diffuse irradiation, distinction between short wave and Long wave irradiance, Convective mass transfer, Condensation of moisture, Carbon dioxide emission and balance for re-circulation air controls a single air volume approach for a very popular middle class sedan car. The model consists of external wall partitions that are exposed to external boundary

conditions on the outside for the ceiling, the floor, the left side wall, the right side wall. For demonstration reasons air spaces have been integrated into the ceiling and side walls in order to simulate air temperatures here as well. It is usually not easy to determine properties of the multi-layered cabin walls. Another challenge was defined by the later calibration of the cabin model since the physical parameters of each layer are potentially uncertain. Therefore, it may be a better approach to calculate average properties for a compound of materials and calibrate three property parameters for a wall.

It is important to understand that the car were located in an artificial experimental setup and will not move during all driving cycles. Instead, the air velocity of the surrounding air is changed accordingly. During the passive heat up the vehicle is actually exposed to a small air flux to prevent overheating on some external surfaces. The exact wind speed is unfortunately not known but was assumed to be small. Another large impact on the car's heat balance is imposed by walls of the experimental facility which emit long wave radiation. so it has been found out that, Calibration Results the process of the Human Comfort model is required in order to determine important, unknown parameters that have a large impact on the thermal behavior of the cabin. Those are usually ,Internal and external heat transfer coefficient, Emission and absorption factors of internal and external surfaces.

2.8 Material property

2.8.1 Thermal conductivity

Thermal insulation materials which are specifically designed to reduce the heat flow by limiting heat conduction, convection, radiation or all three while performing one or more of the following functions:

- Conserving energy by reducing heat loss or gain
- Controlling surface temperatures for personnel protection and comfort
- Facilitating vapor flow and water condensation of a process
- Increasing operating efficiency of heating / ventilating / cooling, plumbing, steam, process and power systems found in commercial and industrial installations

In this book it was discussed that thermal Conductivity λ is a thermo-physical property that determines the ability of a material to transfer heat. The value of the thermal conductivity is characterized by the quantity of heat passing per unit of time per unit area at a

temperature drop of 1°C per unit length. It depends on the medium's phase, temperature, density, molecular bonding, humidity and pressure. The R value is a measure of thermal resistance used in the building and construction industry. The higher the number, the better the effectiveness of the building insulation, R value is the reciprocal of U value [23].

$$\text{Thermal resistance } R = \frac{d}{\lambda} \quad \dots 2.10$$

$$(\Delta T = Rq) \quad \dots 2.11$$

Often in heat transfer the concept of controlling resistance is used to determine how to either increase or decrease heat transfer. Heat transfer coefficients represent how much heat is able to transfer through a defined region of a heat transfer area. The inverse of these coefficients are the resistances of those areas. The heat transfer coefficient k or U value is defined as: presenting the thermal conductivity of different material as follows,

$$k = \frac{\lambda}{d} = \text{u value (k value)} \quad \dots 2.12$$

Different conceptual view-point for Fourier's law is the heat-transfer rate may be considered as a flow, and the combination of thermal conductivity, thickness of material, and area as a resistance to this flow. So it is stated that, the temperature is the potential, or driving, function for the heat flow, and the Fourier equation may be written [24];

$$\text{Heat flow} = \frac{\text{thermal potential difference}}{\text{Thermal resistance}} \quad \dots 2.13$$

Heat transfer analysis is the ratio of the thermal conductivity to the heat capacity is another important property termed the thermal diffusivity, which has units of m²/s:

$$\alpha = \frac{k}{\rho C_p} \quad \dots 2.14$$

It measures the ability of a material to conduct thermal energy relative to its ability to store thermal energy. Materials of large will respond quickly to changes in their thermal environment, whereas materials of small will respond more sluggishly, taking longer to reach a new equilibrium condition [25].

2.8.2 Selecting low thermal conductive and strong material

Guru et al. (2018) have studied that high chromium high Carbon tool steel (D2 steel) is widely used for making molds and dies as it possesses high hardness. The major concern in D2 is that the temperature of the mold increases with number of cycles of operation which

is due to the insufficient heat transfer from the mold cavity to the surface. An additional cooling system can be provided to overcome the aforementioned problem but it leads to higher production cost and it complicates the mold design. As the effect of Copper on steels increases thermal conductivity, in the present study Copper is added to D2 in the composition varied in steps of 0.5% up to 2.0%. Specimens were fabricated using gas casting technique and heat treatment (quenching and tempering) was done. As the mold requires high strength, hardness and high thermal conductivity, hardness (Vickers) and thermal conductivity tests were conducted. Hardness test was taken using Vickers hardness testing machine and it was converted to the equivalent HRC and tensile strength. The maximum hardness value of 57.5 HRC was obtained at 2.0% Cu. Similarly, thermal conductivity test was taken and it was observed that maximum thermal conductivity of 25.3W/mK was obtained at 2.0% of Cu. It was observed that the copper addition increases the thermal conductivity of the D2 steel proportionally.

Micro structural analyses were also done and the images were discussed. Steel was better in low thermal conductivity before the addition of copper. Steels are generally above 1100°C as the carbides require higher temperature to dissolve in the matrix and the homogenization can be achieved at the higher temperature. At the same time Hardness tests were conducted as per the ASTM E18 method at the load of 150kg using Micro Vickers Hardness testing machine and equivalent Rockwell Hardness values were taken.

Thermal conductivity test was the major mode of heat transfer in the mold is conduction. Hence the thermal conductivity of the mold should be considerably high enough for a mold so that the cooling and heating can be effectively done during the cycle of operation. The thermal conductivity is done as per the ASTM E 1225-13 method using Guarded-Comparative-Longitudinal Heat Flow Technique so, Results and Discussion The hardness test was conducted in a micro Vickers hardness testing machine as per the ASTM E 18 method and the equivalent Rockwell hardness C scale results are tabulated in the process parameters of heat treatment will be selected according to the required hardness and strength. Here the property of steel is required mainly while the copper is additional while the thermal conductivity was observed from *figure 26* shown below.

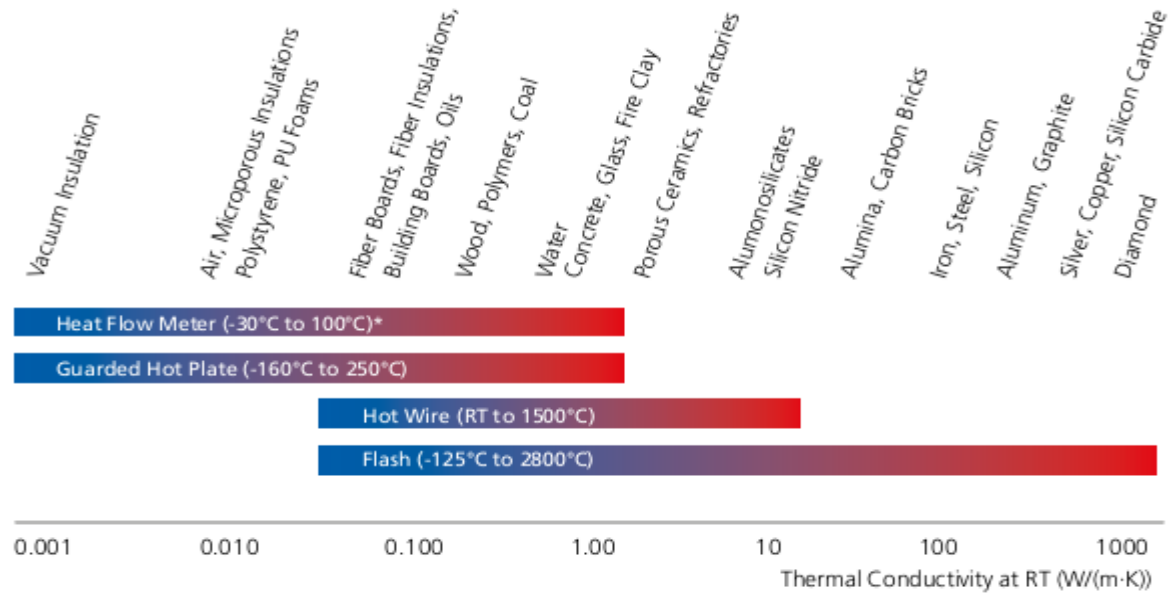


Figure 5: Thermal conductivity of different materials [26].

Again here in this paper **Zhang et al** used a set of experimental devices were designed and built to investigate the effects of average temperature of lamination, contact pressure, and lamination number on the effective axial thermal conductivity of silicon steel multi-stacked material in the loading and unloading processes under atmospheric environment. Specimens are stacked by oriented silicon steel sheets, the content of silicon is approximately 3.3 %, and thickness is 0.23mm. An average temperature is in the range of 200–800°C, and contact pressure is between 0.16 and 16.68MPa. Experimental results indicated that: effective axial thermal conductivity presented an obvious relationship with the average temperature and contact pressure of lamination but showed independence from the lamination number. Effective axial thermal conductivity is more sensitive to contact pressure in the loading process than in the unloading process. The correlation equations of effective axial thermal conductivity have an accuracy of 15 %.

While experimenting first, the contact pressure of specimens at the lowest value was set and the target temperature and heating furnace. Then, the moment that the heater reaches the target temperature was recorded and the system reaches a steady state. Thereafter, the load was increased to a high level and the moment that the heat transfer reaches another steady state was recorded. The second step was repeated to obtain experimental data under different loads. After the load reaches the maximum value, then unloading experiment was started and decreases the load gradually. Recording the temperature distribution of

specimens and heat conducting rod at each step when, the temperature change of every thermocouple was less than 0.2°C in 30 min. This criterion was essentially used to ensure the stability of temperature and the reliability of data, So that the result becomes the lamination number varies from 20 to 50, contact pressure is from 0.16MPa to 18MPa, and average temperature is from 200°C to 800°C . Each experiment is conducted with certain number of lamination, and contact pressure was increased from the lowest to the maximum value gradually and then decreased to the minimum value gradually. The hot- and cold-side cylindrical stainless-steel heat conducting rods are constructed through the same process and possess the same surface properties. The oriented silicon steel sheets are obtained from the same silicon steel. The effective axial thermal conductivity of multi-stacked silicon steel sheets is significantly affected by the average temperature of lamination. Effective axial thermal conductivity increases averagely by 0.15W/mK per 100°C before 700°C and then increases by 0.6W/mK per 100°C after 700°C . The effect of temperature is similar at the loading and unloading processes.

Luis and Alejandro (2017) have demonstrated the potential thermal, environmental and cost advantages of a thermal insulation, which consists on several air chambers between layers of an insulation material with low infrared emissivity. The multimodal heat transfer involved (conduction, convection and radiation) were modeled and numerically solved for different materials and designs. The first design proposed uses multiple EPS (expanded polystyrene) layers of 1 cm thickness separating air chambers. It achieves global thermal transmittances ranging from $0.439\text{W/m}^2\text{K}$, to $0.126\text{W/m}^2\text{K}$, for 4 –13 layers, respectively. In this way, material savings up to 55% were obtained with respect to the solid EPS insulation, leading to lower embodied impacts and cost. A second design, based on thin 0.3cm MDF panels coated with low-emissivity paint, gave thinner walls than the previous one, but higher embodied impacts and cost. A third design, based on EPS separation layers coated with very low emissivity aluminum foil, leads to significant improvements in all aspects considered. For a given total transmittance of $0.1\text{ W/m}^2\text{K}$, this design reaches savings of 77% in material, 66% in cost, 72% in embodied impacts, and 38% reduction in wall width with respect to solid EPS insulations.

Investigation was the performance of an insulation system based on multiple air gaps created between flat layers of an insulation material with low infrared emissivity. A model to calculate the multimodal heat transfer by conduction, convection and radiation has been developed, and the numerical method to calculate this kind of systems was performed in a

spreadsheet. Three designs have been studied and the obtained total thermal transmittance compared to solid EPS insulation. A first design, based on air gaps separated by several 1 cm regular EPS was proposed (EPSW). The model allowed calculating the optimum air-gap distance, which varies with the number of layers, and for EPSW ranges from 2.0cm to 2.9cm for arrays of 4 and 13 layers, respectively. This effect causes a noticeable reduction of the heat transferred by conduction-convection mechanism, leading to material savings around 50% and similar reductions in cost and embodied impacts. However, as a disadvantage, for EPSW the total width of the insulation wall can be enlarged up to 77%.

Drawback was, two other designs were studied: the use of thin layers of MDF coated with low emissivity paint (MDFP), and the use of Aluminum foil on EPS separation layers (EPSF). The MDFP design resulted in a reduction of insulation total width, but still having values much larger than solid EPS. For equal heat transmittance, MDFP has lower cost and embodied GHGs than solid EPS; however, the embodied energy is higher. On the other hand, the EPSF design leads to significant better performance in all aspects considered. For instance, for a given total transmittance of $0.1\text{W/m}^2\text{K}$, this design reaches savings of 77% in material, 66% in cost, 72% in embodied energy and impacts, and 38% reduction in wall width with respect to solid EPS insulations.

A contribution of this work was. First, it provides a practical tool for the thermal analysis of new insulation assemblies based on multiple air-gap chambers and reflective coatings, simplifying the calculation of the important role of convection-conduction and infrared radiation mechanisms in air chambers. Although this concept has been proposed and studied before, the complete thermal modeling developed here allows extending the application of the concept to multiple air chambers. Secondly this considers the advantages of the multiple gap concepts. Thirdly, they were able to assess the environmental embodied energy and GHG emissions of each alternative, and lay out the method to perform the assessment on future insulation proposals within the concept of mixing insulation materials, reflective coatings and air gaps. Also **Lee et al (2016)** have presented the results of a study and investigation on the feasibility of using a constant thermal conductivity for intumescent coating when calculating protected steel temperature in fire, based on analyzing a series of fire tests on intumescent coating protected steel sections with a range of section factors and intumescent coating thicknesses. Having a constant thermal conductivity enables simplified analytical equations to be developed for design purpose. The constant thermal conductivity is calculated as the temperature averaged value within

the temperature range of interest for fire resistance design. The constant thermal conductivity is then used to calculate the protected steel temperatures and the calculated steel temperatures are compared with the measured values from the fire tests.

The results of this comparison suggest that it is feasible to use a constant thermal conductivity for intumescent coating. The constant thermal conductivity value would be dependent on the intumescent coating thickness and the steel section factor. However, this issue was dealt with obtained intumescent coating thermal conductivity from the fire tests on steel plates, because a comparison between results based on steel plate fire tests and those based on steel section fire tests shows close agreement. The results also indicate that the constant thermal conductivity of intumescent coating converges to a fixed value, independent of the steel section factor, when the steel section factor is high, suggesting that the constant thermal conductivity method is particularly applicable to thin-walled sections with high section factors.

Two fire testing furnaces experiment was conducted in Tongji University, China, for a small furnace (internal dimensions: 1000mm× 1000mm× 1200mm and a large one (internal dimensions: 4500mm× 3000mm×2200mm).The interior surfaces of the furnaces were lined with ceramic fiber materials of thickness 200mm to efficiently transfer heat to the specimens. The small furnace was used for fire tests on all the steel plates, the hot rolled I-shaped sections and the hot rolled channel-shaped sections. The large furnace was used for fire testing all the welding H-shaped section specimens. The fire temperature–time relationship in both furnaces was according to the ISO 834 standard fire condition. Shows typically the gas temperatures recorded at different locations in the big furnace, indicating relatively uniform temperature distribution in the furnace.

The average gas temperatures of all the fire tests, confirming that all fire tests almost achieved the intended standard fire temperature-time curve. Shows typical steel temperatures measured at different locations, showing uniform temperature distribution the steel section, Similarly **Robert et al. (2009)** have studied the development of a hot plate method capable of using air as a standard reference material for the steady-state measurement of the thermal conductivity of very small test samples having thermal conductivity on the order of air were dealt. As with other approaches, care were taken to ensure that the heat flow through the test sample is essentially one-dimensional. However, unlike other approaches, no attempt is made to use heated guards to block the flow of heat from the hot plate to the surroundings. It is argued that since large correction factors must be applied to account for guard imperfections when sample dimensions are small, it may

be preferable to simply measure and correct for the heat that flows from the heater disc to directions other than into the sample.

Experimental measurements taken in a prototype apparatus, combined with extensive computational modeling of the heat transfer in the apparatus, showing that sufficiently accurate measurements can be obtained to allow determination of the thermal conductivity of low thermal conductivity materials. Suggestions were made for further improvements in the method based on results from regression analyses of the generated data. The computational fluid dynamics (CFD) code FLUENT (ANSYS, Inc.) were used to model the apparatus. It uses a finite volume method to discretize the continuity, momentum, and energy equations. Equations may be solved alone or in combination. Also a series of experiments were conducted to determine if measurements of sufficient precision could be made to allow accurate determination of thermal conductivity, and to demonstrate the feasibility of using air as a calibration standard. The dimensions of this apparatus, discussed in the Experimental Design section, differ from those used to model the “air gap” case in that the diameters of the large PMI foam discs were increased somewhat in order to take further advantage of the thermal insulation provided by the air gap. Also, the thickness of the PMI foam discs were increased by 50 % in order to allow affirmer attachment to the threaded rods that hold the fixture in place

2.9 Air inlet or nozzle design

The principle of conservation of mass and energy can be applied to the venturi and nozzle geometries to give through the following.

$$w = \rho V_1 A_1 = \rho V_2 A_2 = A_2 \sqrt{\frac{2\rho(P_1 - P_2)}{1 - \beta^4}} \quad \dots 2.15$$

Where,

w = mass flow rate, kg/s

V = velocity of stream, m/s

A = flow area, m²

ρ = density of fluid, kg/m³

p = absolute pressure, Pa

$\beta = (D_2/D_1)$ for venturi and sharp edge orifice and d/D for flow nozzle

Note: Subscript 1 refers to the entering conditions; subscript 2 refers to the throat conditions. Because the flow through the meter is not frictionless, a correction factor C is defined to account for friction losses. If the fluid is at a high temperature, an additional

correction factor F_a should be included to account for thermal expansion of the primary element. Because this amounts to less than 1% at 250°C, it can usually be omitted. The dimensions can be obtained as on *figure 6* after using Equation below [31].

C is the friction loss correction factor

$$w = CA_2 \sqrt{\frac{2\rho(P_1 - P_2)}{1 - \beta^4}} \dots 2.16$$

The orifice flow rate equations then becomes,

$$Q = KA_2 \sqrt{\frac{2(P_1 - P_2)}{\rho}} \dots 2.17$$

Where

Q = discharge flow rate, m^3/s

A_2 = orifice area, m^2

$p_1 - p_2$ = pressure drop as obtained by pressure taps, Pa

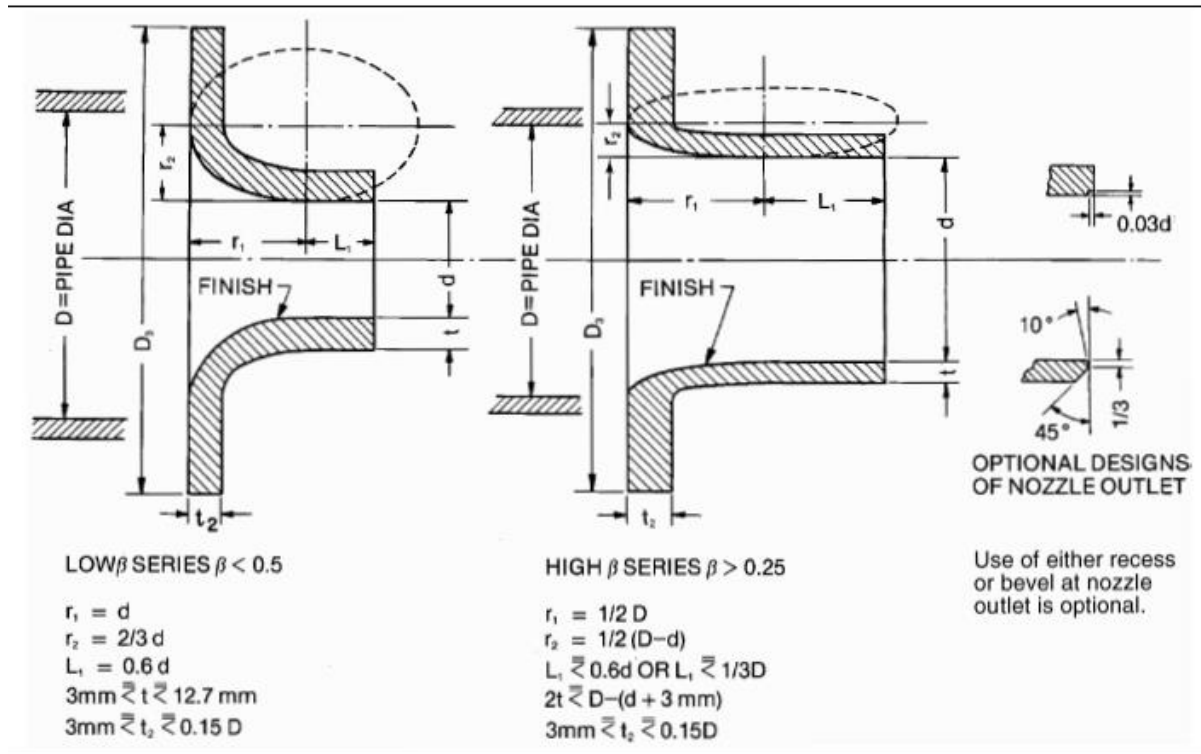


Figure 6: Standard nozzle design dimension [31].

The relation of Conservation of mass applied to fluid flow in a conduit requires that mass not be created or destroyed. Specifically, the mass flow rate into a section of pipe must equal the mass flow rate out of that section of pipe if no mass is accumulated or lost. This requires that;

$$m = \int \rho V dA = \text{constant} \quad \dots 2.18$$

Where m is mass flow rate across the area normal to flow, V is fluid velocity normal to differential area dA , and ρ is fluid density. Both ρ and V may vary over the cross section A of the conduit. When flow is effectively incompressible ($\rho = \text{constant}$) in a pipe or duct flow analysis, the average velocity is then

$$m = \rho VA \quad \dots 2.19$$

Or

$$Q = \frac{m}{\rho} = AV \quad \dots 2.20$$

Where, Q is volumetric flow rate [32].

2.9.1 Air filtration

Automotive air conditioning text book has some means of air inlet and cleaning the air inlet nozzles, since it is one important requirement as *figure 7* shows fibers in the filter preventing large particulates from entering the system and trap the really small ones, the filter is electrostatically charged. Due to this electrostatic charge, the filter attracts particles like a magnet attracts iron filings. Besides removing visible particles, the filter also removes pollen, spores and different types of dust [33].



Figure 7: Air inlet and filtering [33]

2.10 Summary of the literature

Much of the present papers are devoted to heat flux analysis of the specific thermal behaviors in the under hood compartment (especially the absorption of convective and conductive heat fluxes) and to a description of a new control approach to optimize under hood and cabin aero thermal management. Where the maximum temperature in under hood or vehicle compartment is 150°C to 250°C was taken as a reference for most of the studies Thermal Conductivity k is a thermo-physical property that determines the ability of a material to transfer heat. The value of the thermal conductivity is characterized by the quantity of heat passing per unit of time per unit mass at a unit temperature drop. It depends on the medium's phase, temperature, density, molecular bonding, humidity and pressure, the R thermal resistance value in which analytical values can be obtained.

Thermal insulation materials are specifically designed to reduce the heat flow by limiting heat conduction, convection, radiation or all three while performing one or more of the following functions were understood. Different steel alloys become the most preferable insulation material with a required thermal conductivity, strength, since steel possesses high hardness because some researchers were trying to add or make an alloy of steel like copper, and silicon in order to increase its thermal conductivity while keeping its strength which makes steel good material having both the strength and low thermal conductivity at the same time.

Aero thermal experiments were performed; and carried out for different thermal functioning points that simulate more or less severe rolling situations from the thermal point of view. The thermal soak phase follows the slowdown and simulates the vehicle stopping after a significant heat load. Here thermal inertia maintains temperatures higher even though the engine has stopped

Both the air flow rate through the radiator module and air flow distribution in the engine compartment is important factors. Some of investigations were the performance of an insulation system based on multiple air gaps created between flat layers of an insulation material. Different models to calculate the multimodal heat transfer by conduction, convection and radiation has been developed, while numerical method to calculate this kind of systems was performed in a spreadsheet also. The numerical results are used to verify and validate the concept design and prototyping of off-vehicles including parameters and components such as: the speed of cooling fans; core size of the heat exchangers;

temperature; pressure drop; thickness of insulation and design options of the heat shield. Experimental Design section, model the “air gap” Advantage of the thermal insulation is due to provided air gap. So from different articles review I have come to a decision that the heat transfer reduction can be reached through multiple air gap insulation system and validate the obtained result with the CFD values.

Some literature review shows that for thermal investigation, The CFD analyses are a powerful tool that can identify possible design drawbacks, evaluate design options, improve the rate of heat transfer and provide optimal engineering solutions and strategies for complicated thermal management which is going to be used in this thesis in order to show the load distributions over different surfaces and compare its results with the experiment.

Experiments have been made using methods like wind tunnel experimentation, artificial experiment setup. This makes artificial or own experimental setup an option which gives actual result. The materials and equipment's will be prepared and analysis can be made. Heat and blow of air can be achieved, and the results are obtained after comparing the models.

CHAPTER THREE

MATERIALS AND METHODS

In heat transfer analysis many methods are applicable in order to make sure of the results obtained are accurate different methods should be used which makes the use of different material in different conditions. The main focus here was the experimentation but to make sure of the results obtained modeling and analyzing was also additionally used.

3.1 Materials

Here in this thesis work materials were used for two (2) different cases

1. Materials used for experimentation
2. Materials used for modeling and analysis

3.1.1 Materials for experimentation

a. Steel sheet metal

Steel is an alloy of carbon and iron with the following material properties which makes it preferable.

- High tensile strength
- Low cost
- Used widely in automobiles and other application areas
- Corrosion resistance
- Strength
- Formability
- Ductility
- Low thermal conductivity when compared with other hard materials.

b. Air has a low thermal conductivity and used to extract the heat from the air gap created.

Air is used as insulation and heat removing medium since it has low thermal conductivity and light in weight which makes it preferable other fluids.

c. Measuring instruments which were used in this thesis work are;

- Thermocouple is used to measure the temperature on the surface and also room temperature
- Thermometer used to measure temperature inside the cabin

- Pressure gauge.

d. Working major laboratory equipment's

- Compressor, to create air flow through the air gap.
- Heater, to obtain required temperature under the surface.

e. Materials used for sample preparation

- Welding machine, Electrodes
- Grinding machine and
- Drilling machine

3.1.2 Materials used for modeling and analysis

The CFD method is a powerful tool that can identify possible design drawbacks, evaluate design options, improve the heat transfer efficiency and provide optimal engineering solutions and strategies for complicated thermal management.

- CATIA V5 R21 was used to construct the designed sample and model it.
- ANSYS used in order to analyze the effect of heat over the compartment surface for efficient insulation ability and conductivity of the materials.

3.2 Methods

The method of doing this project follows the next procedures from the start to end. The steps are dependent on each other since one comes after the other.

- Start of the project work
- Literature review
- Data collection
- Experimentation and testing
- Modeling and analysis
- Result and discussion
- Conclusion and recommendation
- Document preparation

3.2.1 Data collection

For the baseline model of public mid bus shape seen *figure 8* ISUZU NPR 400 AMT was selected, the dimensions of the engine compartment which was considered for heat transfer analysis, public transport mid bus was bench marked for the study. It is a model which is

popular in Ethiopia when selecting for this title. Engineering parameters of this model was kept as same for the new engine compartment design and was selected as the baseline for studying the heat transfer analysis.



(a)



(b)

Figure 8: a) Isuzu vehicle from company, b) Reshaped Isuzu vehicle

The necessary data's were collected for the design requirement in order to get enough information about the engine capacity, challenges and thermal concerns. Regarding the engine compartment and some relevant data's shown in *table 2* data's were collected in the following ways;

- Interviewing midi bus drivers ,
- Visiting automobile industries and dealing with concerned bodies in Bishoftu automotive industry and Hagbes Vehicle Company.
- Dealing with reshaping companies

- Interviewing different technicians
- And measurements.

Table 2: Specification of ISUZU NPR 400 AMT vehicle engine compartment

Required data of the Engine compartment	Details and Dimensions	Remark
width	600mm	Measured
Height	100mm	Measured
Length	1140mm	Measured
Inside height	85mm	Measured
Inside width	570mm	Measured
Inside length	1110mm	Measured
Max driving speed	120 km/hr.	Obtained from interviewing drivers
Maximum inside compartment temperature	250°C	
Vehicle model	ISUZU NPR 400 AMT	Reshaped to public bus
Engine type	Diesel Engine	

3.2.2 Experimentation

Method of preparation

The model were prepared according to the design in a reduced scale, with two insulation flat surface as housing for the EC and separation of the cabin surface and heat source.

Scaling

The size was reduced by scale of (original size: prototype size) 1:3 ratio. The original size was reduced three times. The area is reduced and the heat for experiment is equally considered with the actual value because, aim of experiment was to compare the results of different models.

The steps which were followed during the production of the prototype are indicated in the following figures.

Step 1. Cutting the metals into required dimension and numbers as shown in figure 9.



Figure 9: Metal cutting and preparation

Step 2. Assembly procedure

The assembly procedure took place as shown figures 10-11.



Figure 10: Assembly procedure step 1.



Figure 11: Assembly procedure step 2.



Figure 12: Assembly procedure step 3.



Figure 13: Final assembly.

Experimentation:

Artificial experimental setup were constructed as shown in figure 13 and 14 with compressor generating the required heat extraction procedure; the heat were be generated beneath the floor surface with the help of heater to a required temperature.



Figure 14: Experiment setup.

Calibration of measuring instruments

The instruments used for collecting data's must be calibrated before proceeding to the next work.

1. Thermocouple - Is a temperature measuring device which needs calibration
It is important to check and calibrate the measuring devices weather it working correctly or not, the device measurement has been compared with three different kind of temperature measuring device and the result becomes positive to proceed.

Experiment conditions



Figure 15 : (a) Single flat sheet metal (b) Wooden sandwich sheet metal.

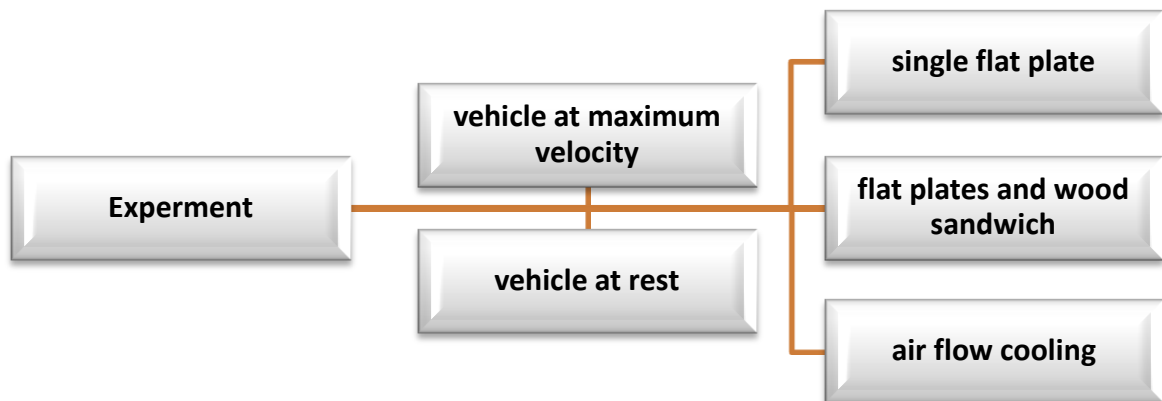


Figure 16 : Experiment condition for different materials

For three different insulation and heat transfer materials there will be two (2) different experiments.

1. Single flat plate(sheet metal)

Two different vehicle velocities

1. Whenever the engine is working but Vehicle at rest
2. Level road driving

2. Flat plate and wood sandwich insulation

Two different vehicle velocities

1. Whenever the engine is working but Vehicle at rest
2. Level road driving

3. Air flow cooling and insulation s system

Two different air or vehicle velocities

1. Whenever the engine is working with full throttle but Vehicle at rest (air velocity $V=0$)
2. Driving at maximum speed (air velocity $V=\max$)

1. **Single flat sheet metal or plate**

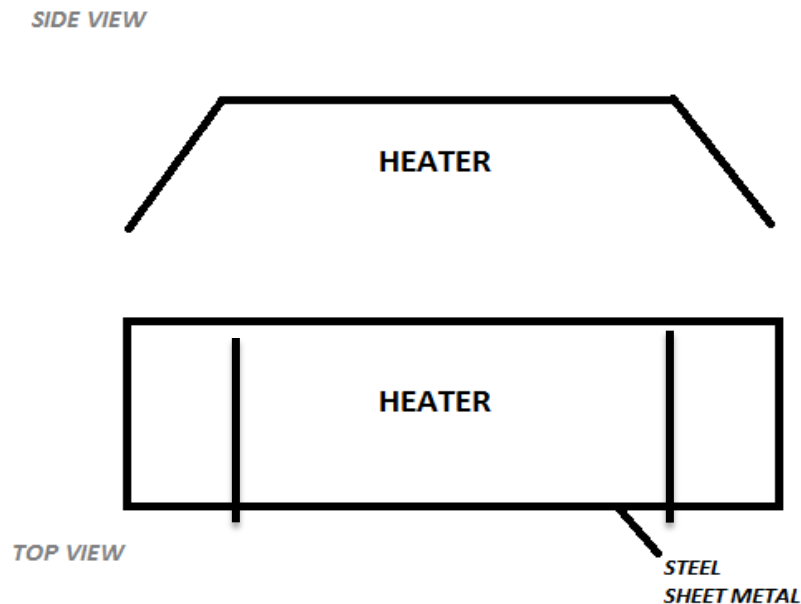


Figure 17: Single flat sheet metal or plate

1. Flat plate and wood sandwich insulation

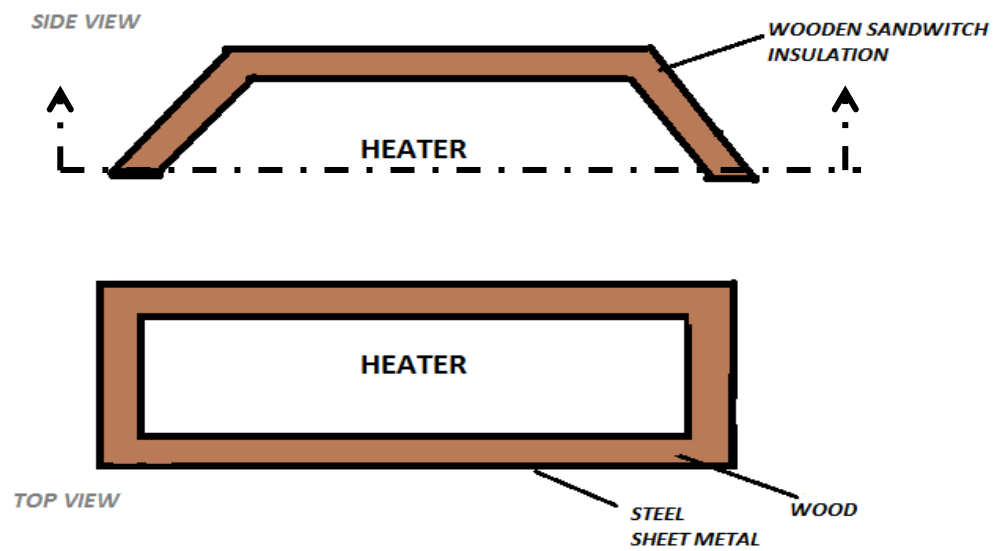


Figure 18: Flat plate and wood sandwich insulation

2. Flat plate with Air flow

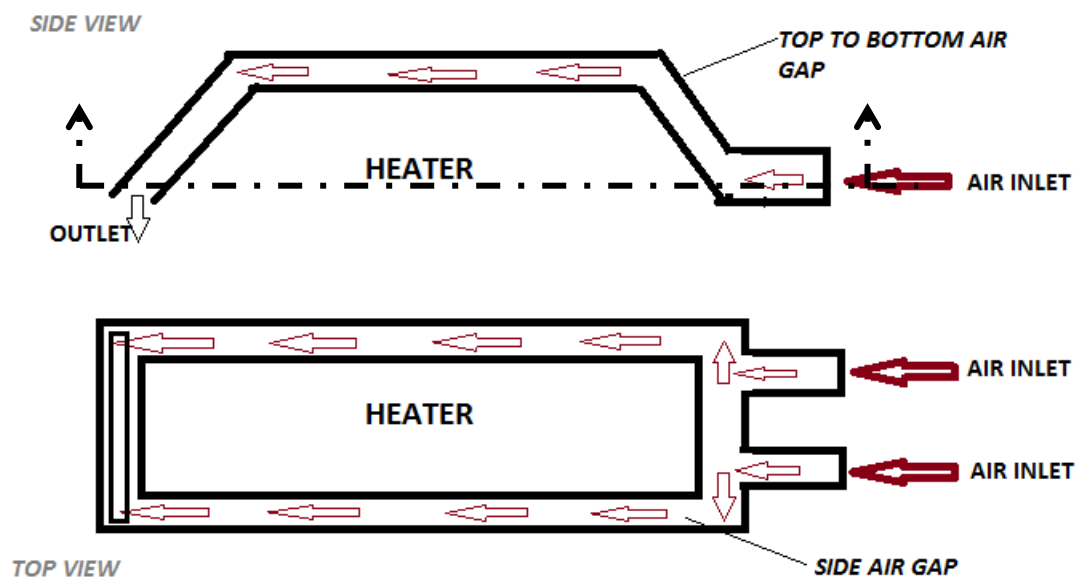


Figure 19: Flat plate with Air flow

Procedure

The previously in service heat reduction and the designed surface will be computed and the difference will be registered. For this procedure the vehicle will be assumed that it is in motion so the air cooling system were due to the vehicle moving speed.

The systems will be thermally insulated during tests air was circulating through the formed air channel.

- Using thermometer and thermocouple reading the temperature difference on the surface and space of the room.

3.2.3 Convective heat transfer coefficient analysis

The numerical analysis is made for the proposed air flow cooling system, Temperature variations inside a car under hood are largely controlled by the heat originating from the engine block and the exhaust manifold. This affects the thermal comfort level inside the passenger cabin.

Since the material to be used as insulation are already identified, the design and the ideal model is already known even if it requires detail analysis and experimenting ,now the design is all about understanding the final outcome which is understanding whether the result has brought the reduction in heat transferred .

Two types of heat transfer modes are to be applied starting from the engine compartment to the passenger cabin.

- When the vehicle is at rest - Conduction heat transfer and natural convection
- When the vehicle is in motion - Conduction and forced convection

The design considers the above two conditions and analyze important parameters to Prevent the cabin heat gain, using principles of heat rejection and insulation; then identify and evaluate the most important influences on heat transfer processes in the human-cabin-environment system, If there is a heat shield beneath the floor and insulation between the flooring and the metal, the heat gain can be reduced. So that air can be used as a heat shield.

Heat transfer

- Heat flux (q)
- Ambient Temperature(T)
- Heat transferred across the air gap

The design of an insulation or heat reduction system is governed by the insulated operating values, which the plant requires after insulation. The values may be:

- Radiation
- Thermal conditions – Heat loss/Heat gain
- Process temperature drop or rise
- Personnel protection temperature
- Optimal economic conditions
- Thermal conductivity of insulation material
- Ambient temperature
- Wind velocity[34]

Heat Transfer in Natural Convection

Aim: To determine the heat transfer coefficient for a losing heat by natural Convection. In contrast to forced convection, the natural convection phenomenon is due to the temperature difference between the surface and the fluid and is not created by any external agency.

The present experimental setup is designed and fabricated to study the natural convection phenomenon in terms of average heat transfer coefficient and its comparison with the value obtained using an appropriate correlation.

The apparatus consists of an electric heating element which is kept in the vertical tube, which in turn heats the surface. The heat is lost from the surface to surrounding air by natural convection. The temperature is measured by thermocouples. The analysis is to be made for the flap or top since the material throughout the whole body is the similar.

Specifications:

1. Diameter of the inlet tube = 20mm
2. Length of the inlet tube = 150mm
3. No. of thermocouple positions = 3 at a time
4. One thermocouple reads the temperature of air in the duct
5. Temperature indicator = 0 – 400°C

Procedure:

1. Put on the supply & adjust heater to obtain the required heat input.
2. Wait till the fairly steady state is reached, which is confirmed from temperature readings ($T_1 - T_3$ (three thermocouple positions over the top surface))
3. Note down surface temperatures at various points ($T_1 - T_3$) over single surface.
4. Note down ambient temperature

Observation data

These values are for the vehicle at rest (0 km/hr.) while the engine is still working which makes the engine temperature medium, and values are to be observed for 15 minutes. And temperature of is take as inner surface temperature so the other values will be taken in relation this value.

Note: there is no forced air flow in this condition only natural convention is available.

Known values

$$T_s = 240^\circ\text{C}$$

$$T_a = 30^\circ\text{C}$$

$$V = 192\text{V}$$

$$I = 5.96\text{A}$$

$$A_s = 2h(1+w) = 2 \times 20(600 + 1140) = 69.6\text{m}^2 \quad \dots 3.1$$



Figure 20: Input voltage and current measurement

Calculation

(1) Average heat transfer coefficient (h)

$$h = q / (A_s(T_s - T_a)) \text{ Where, } h = \text{convective heat transfer coefficient} \quad \dots 3.2$$

$$= 0.07829\text{W} / \text{m}^2 \cdot \text{K}$$

$$q = V \times I \text{ Watts} = 5.96 \times 192 = 1144.32 \text{ W [figure 20]....3.3}$$

$$A_s = \text{Surface area in } = 69.6 \text{ m}^2$$

$$T_s = \text{Average of surface temperature} = 240 \text{ }^\circ\text{C}$$

$$T_a = \text{Ambient temperature} = 30 \text{ }^\circ\text{C}$$

Heat Transfer in Forced Convection

Aim: is to determine heat transfer coefficient for in forced convection. If a heater is exposed to ambient room having air without any external movement there is a movement as a result of density gradient neat plate. This is natural convection as opposed to the forced convection, which is found in case of a fan blowing air over or under the plate. The quantity “h” is called convection heat transfer.

The apparatus consists of blower unit fitted with the test pipe. The test section surrounds the heater. Six thermocouples are placed in the air system at the entrance & exit, of the test section & thermocouples are embedded on the test section to measure the air temperature. Test pipe is connected to the delivery side of the blower along with the orifice to measure flow of air through the pipe. A temperature indicator is provided to measure temperatures of the wall in the test section. Airflow is measured with the help of orifice meter and the water manometer fitted on the panels.

Specification:

- | | |
|-------------------------------|------------------------|
| 1. Delivery pipe dia. (Di) | = 0.02m |
| 2. Length of test section (L) | = 1145m |
| 3. Compressor with motor | = 3HP |
| 4. Nozzle converging Diameter | = 0.02m |
| 5. Thermocouples | = K type, T type |
| 6. Temperature Indicator | = 0 – 400 °C |
| 7. Heater | = Nichrome wire heater |

Procedure:

1. Starting the supply.
2. Starting the compressor
3. Starting the heating of the test section
4. Taking the readings of all thermocouples at an interval of 30 sec until the Steady state is reached.
5. Also noting down the heater input (in terms of voltage and current)

Thermocouple positions:

T_s = Temperature of wall in the test section, average is taken

T_{inlet} = Air inlet temperature @ $T_s = 240^\circ\text{C}$

T_{outlet} = Air outlet temperature

Precautions taken during experiment:

1. Stopping blower or compressor in between the testing period is forbidden.
2. Operate selector switch of temperature indicator gently.
3. No taking data's that exceed the heater input temp to 240°C

This makes the engine temperature maximum, and values are to be observed for 15 minutes, and temperature of maximum heater temperature.

$h = 1$ since constant air pressure is assumed = 1bar, Vehicle at maximum speed (120km/hr.)

Calculation

1. Bulk mean temperature of air

$$T_{bm} = \frac{T_I + T_o}{2} = 56 + 34/2 = 45^\circ\text{C} \quad (318\text{K}) \quad \dots 3.4$$

Properties of air at T_{bm} (table 23)

$$\rho_a = 1.1614 \text{ kg/m}^3$$

$$C_p = 1.007 \text{ KJ/Kg-K}$$

2. Mean mass flow rate of air (m_a)

$$m = \rho * V * A = 2,683.67 \text{ kg/s} \quad \dots 3.5$$

Where:

ρ : Density of the flowing liquid or gas = 1.1614 kg/m^3

V : Flow speed = 33.2 m/s

A : Flow area, = 69.6 m^2

m : Mass flow rate, in kg/Heat of air (q)

$$q = m * C_p * (T_o - T_i) = 59,454.029 \text{ kg/s} \quad \dots 3.6$$

3. Heat transfer coefficient (h)

$$h = \frac{q}{AX(T_s - T_{bm})} = 4.3806 \text{ W/m}^2\text{-K} \quad \dots 3.7$$

Where $A = 2h(l+w)$ area of plate [35]

This shows that the heat transfer coefficient in natural convection is high when compared to the forced one since it took the air more time while passing. [35]

Radiation analysis

There is a radiation heat transfer between the two plates and the engine surface but since the result of the experiment was taken from the inside lower surface of the plate the radiation heat can be calculated regarding the analysis radiation heat becomes insignificant.

3.2.4 Modeling and analysis

Since the right material is selected determined modeling and analysis will precede using software's ANSYS, and CATIA.

- CATIA, The model with exact dimensions like areas ,thickness
With an air gap in between is to be used and constructed using this software.
- ANSYS fluent, parameter like thermal conductivity for both the air and steel sheet metal, type of material, exact dimension, and mode of thermal heat transfer will be fed in to it in order to obtain exact amount of heat distribution throughout the area and space of the cabin.

- wall model
- conduction and convection model

Conductive heat transfer is caused by “diffusion” of heat due to temperature gradient. Heat flux q

t_{s1} and t_{s2} are temperatures of cold or hot wall surface and

λ_i (W/m.K) is thermal conductivity

Comparison of the numerical and the modeling will be finally brought together for the better understanding.

3.2.5 CATIA 3D modeling

The 3D modeling over CATIA was constructed as shown in *figures 21 and 22* using over all measurement data's from table 2.

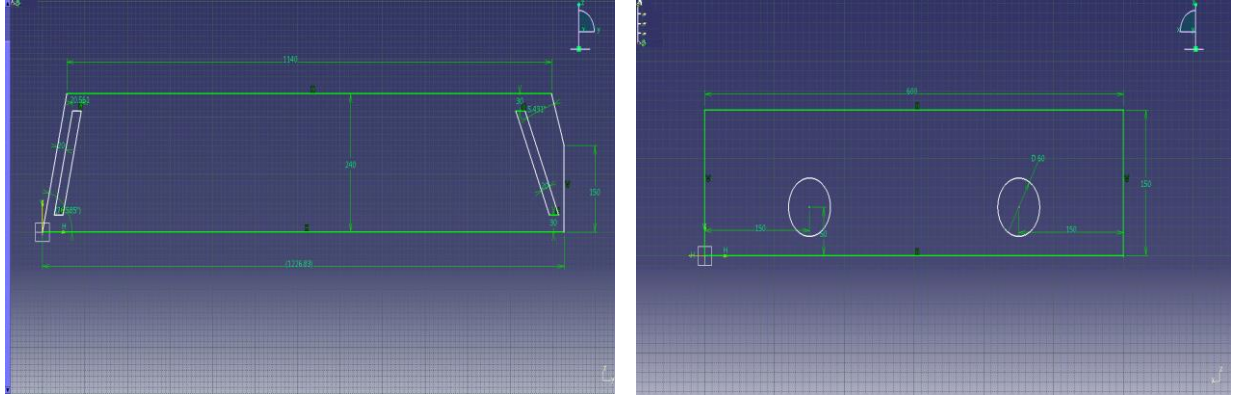
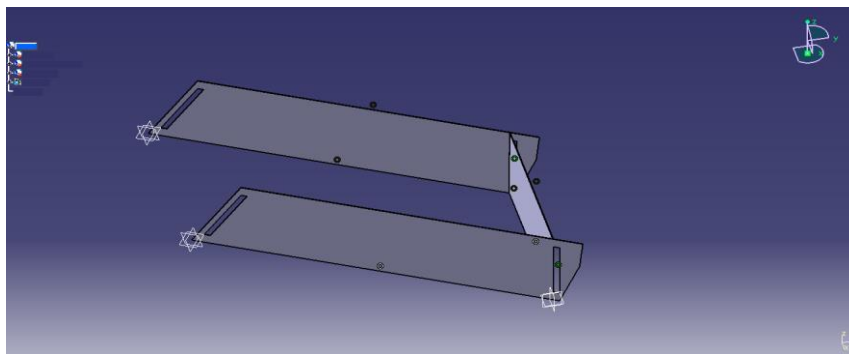
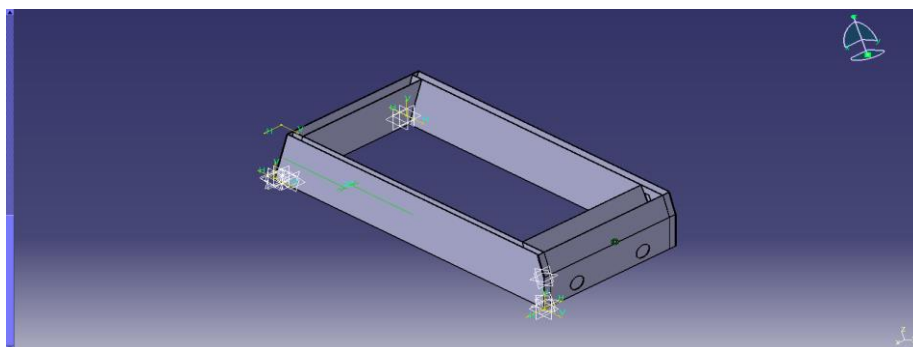


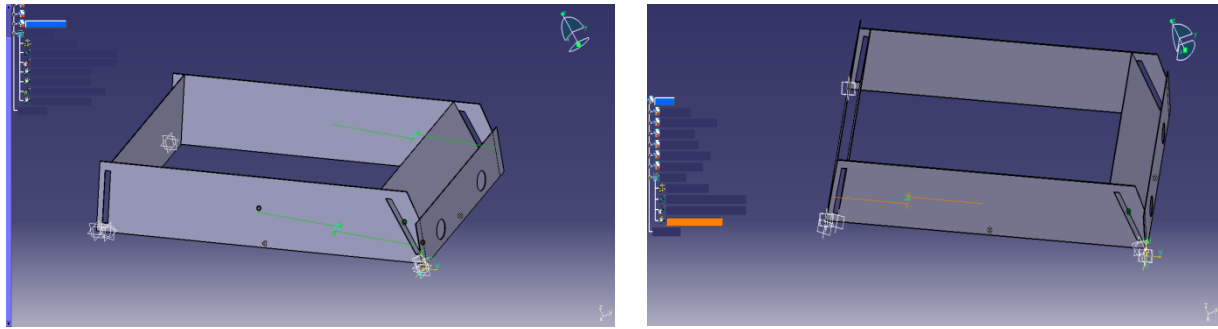
Figure 21: CATIA startup drawing outer front and inner side



(a)



b)



c)

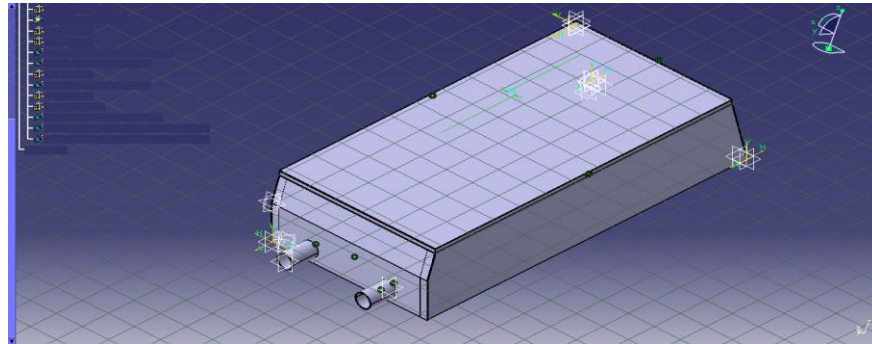


Figure 22: CATIA (a) Assembly first stage (b) Next stage assembly (c) Assembly finalization.

3.2.6 CFD (ANSYS) modeling and analysis

Since the proposed engine compartment has air flow gap throughout its surrounding, the modeling and analysis were made for selected area which is the top (flap) with air gap, Input values were taken from experimental results of 8th minute in which from the literature it has been found out that critical temperature on the engine compartment lower surface becomes 240°C from *table19*, so relating the values we can obtain other outputs.

Used parameters,

$V_{\max}$ air is 120km/hr. (33.3m/s),

Air pressure at the inlet 2 bar, at the out let 1bar,

Hot surface temperature 513.13K (240°C)

Inlet air T_{in} 34°C, 307.15K

Outlet air T_{out} 56°C, 329.15K

Time 8th min

Procedures

The selected top or flap of the air flow cooling was constructed as shown on the *figures 23-26* and analyzed.

Drawing the geometry,

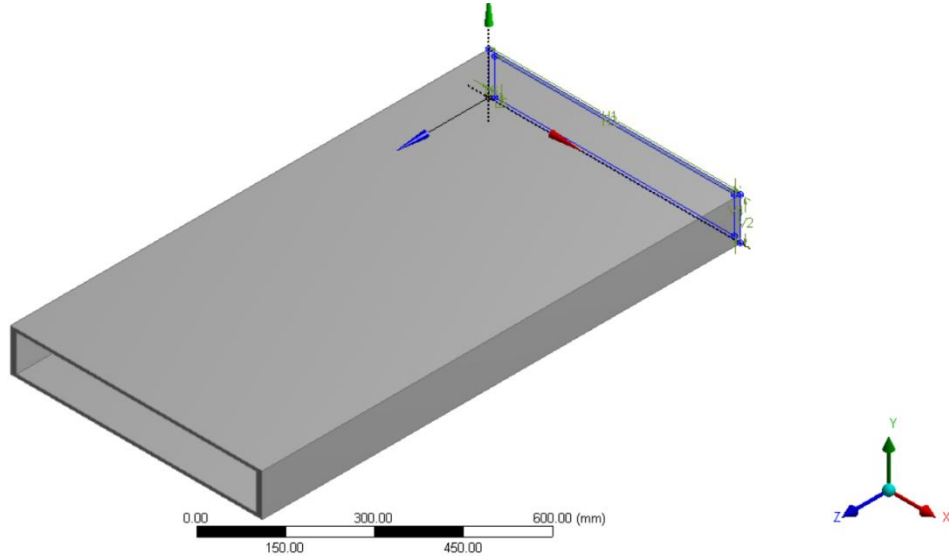


Figure 23: Constructed geometry

Meshing the geometry,

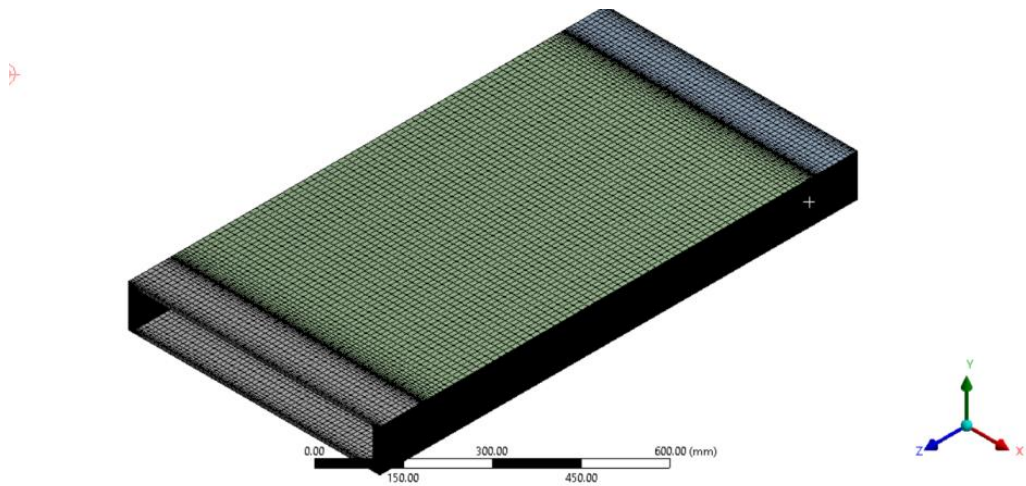


Figure 24: Meshing the geometry,

The meshing provides the accuracy of the modeling, in this procedure quadrilateral meshing was the type of meshing used in as shown in *figure24*.

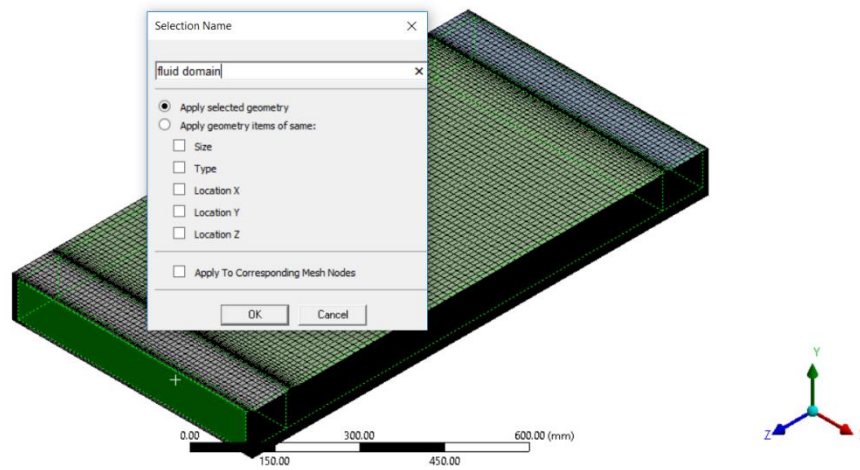


Figure 25: Naming fluid domain

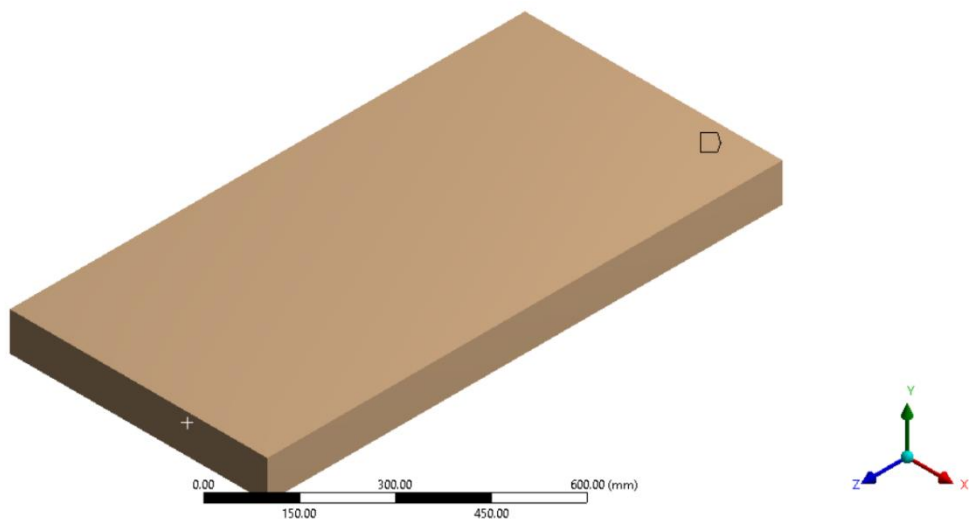


Figure 26: Fluid domain identification

3.3 Cost estimation

During the development the engineering approach or concept was taken into account. The necessary cost of manufacturing should be reduced; here the alternative design will reduce cost without sacrificing quality or performance requirement.

Based on the current cost of different materials and equipment's the following financial breakdown is done as shown on table 3.

Sheet metal dimensions

$$A_{\text{side}} = [(100 \times 1140) + (85 \times 1140)] \times 2 = 416,700 \text{ mm}^2$$

$$A_{\text{rear \& front}} = [(100 \times 600) + (85 \times 570)] \times 2 = 216,900 \text{ mm}^2$$

$$A_{\text{flap}} = (600 \times 1140) \times 2 = 1,368,000 \text{ mm}^2$$

$$A_{\text{total}} = 2001600 \text{ mm}^2$$

$$2,001,600 / 2,000,000 = 1.0008 \text{ mm}^2$$

Table 3: Budget (cost) of wooden sandwich insulation.

N o.	Item name	specification	Qty	Unit price	Total price(birr)	remark
1.	Steel sheet metal	(2000x1000)mm	1*1.0008	1400	1,401.12	
2.	Welding electrode	2.5mm	1 pack	150	150	
3.	Labor cost		5 days	200	1,000	
	Sum				2,551.12	
	Contingency (10%)				255.112	
	Total cost (air flow cooling system)				3,061.344	
4.	Wood (oak)	(2300x1200)mm	1*0.3635	600	218.1	
	Sum	Adding cost of wood to the sum			3,046.21	
	Contingency (10%)				304.62	
	Total cost (air flow cooling system)				3350.83	

The cost is surely reduced by 289.5 birr

Note, the single flat plate has very low performance which makes it incompetent regarding cost.

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Experimental result

The experiment was started at 30°C room temperature at the moment for each experiment, and the results were collected with 30 seconds of time variation.

Assumptions

- All the experiment was done with constant speed.
- Air pressure 1 bar and equal air velocity were used just for the case of experiment in the case of forced convection and constant for every procedure.

4.1.1 Result for model one (M1)

EC with flat plate (flat sheet metal)

Different Thermocouple positions as shown in *figure 27*.

1. Rear face of the compartment
2. Inside compartment
3. Front face of the compartment
- 4F. Side face location
- 4M. Side face middle location
- 4R. Side face rear location
- 5F. Top surface location
- 5M. Top surface middle location
- 5R. Top surface rear location

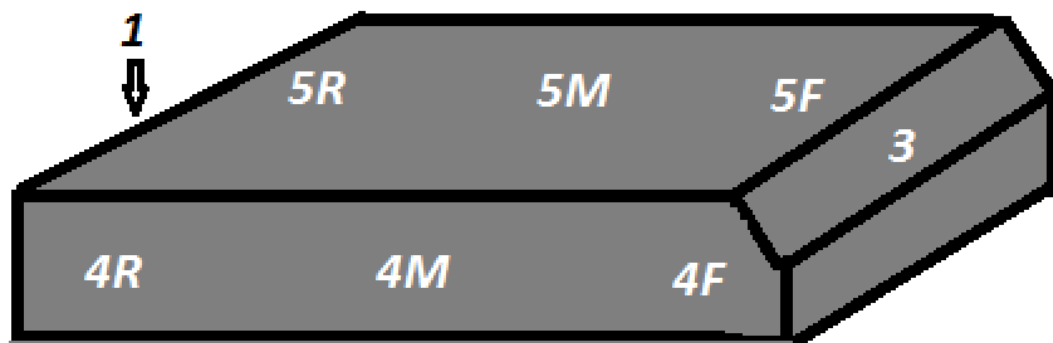


Figure 27: Thermocouple location for no air flow cooling

Condition, There are Two (2) different vehicle velocities and since there is no air flow cooling it is assumed that the vehicle is full throttling when at rest also, which makes the experimental result for both at rest and in maximum vehicle motion condition equal. So the result obtained can be used for comparison with other models both at rest and in motion conditions. Here the obtained results are expressed in *table 5*.

4.1.2 Result for model two (M2)

Flat plate and wood sandwich insulation

Condition, Similarly there were Two (2) different vehicle velocities and since there is no air flow cooling it is assumed that the vehicle is at full throttling when at rest also, which makes the experimental result for both at rest and in maximum vehicle motion condition equal. So the result obtained can be used for comparison with other models both at rest and in motion conditions. Here also the obtained values can be seen in *table 6*.

In this case the data's are obtained from the same position with that of single flat plate sheet metal since there is no air flow cooling in both as shown in *figure 27*.

4.1.3 Result for model three (M3)

Vehicle with air flow cooling system

Different Thermocouple locations from *figure 28*;

1. Rear face of the compartment
2. Inside compartment
3. Front face of the compartment
- 4F. Side face location
- 4M. Side face middle location
- 4R. Side face rear location
- 5F. Top surface location
- 5M. Top surface middle location
- 5R. Top surface rear location
6. Air outlet
7. Air inlet

The data's were taken at a single position except for top and side since the areas to be measured are small in dimension. But for the side and top position the values are taken from three different nodes as shown in figure below.

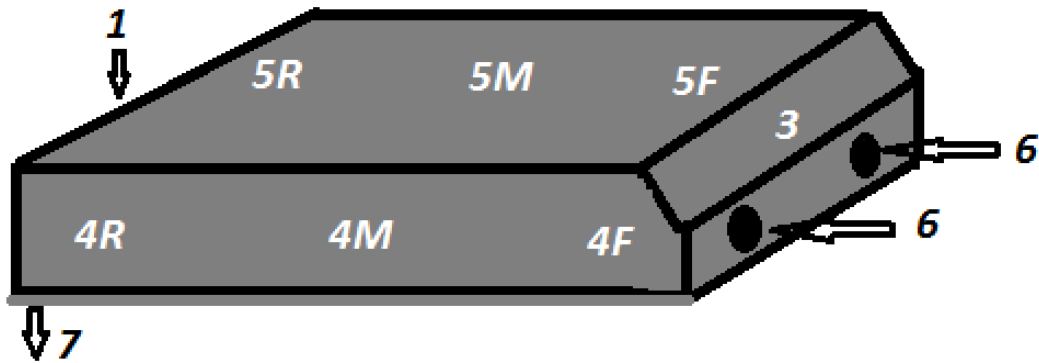


Figure 28: Thermocouple location for air flow cooling system

Condition, there were Two (2) different vehicle velocities and there is air flow cooling.

- Vehicle with air flow and maximum velocity (represented by M3 a)
- Vehicle at rest but engine is working with full throttle (represented by M3 b)

The obtained experimental values can be obtained from *tables 7 and 8 respectively*

4.2 Variation graph

The graph represents the variation of the experimental results of all the three (3) models at the similar position.

Representation

- M1 EC with flat plate (flat sheet metal)
- M2 Flat plate and wood sandwich insulation
- M3 a Vehicle with air flow and maximum velocity
- M3 b Vehicle at rest but engine is working with full throttle

It is important to know that the percentage values are calculated for the 8th min where the inside temperature is referenced at 240°C.

Comparison of two models with the newly proposed solution in case of forced convection

Rear face

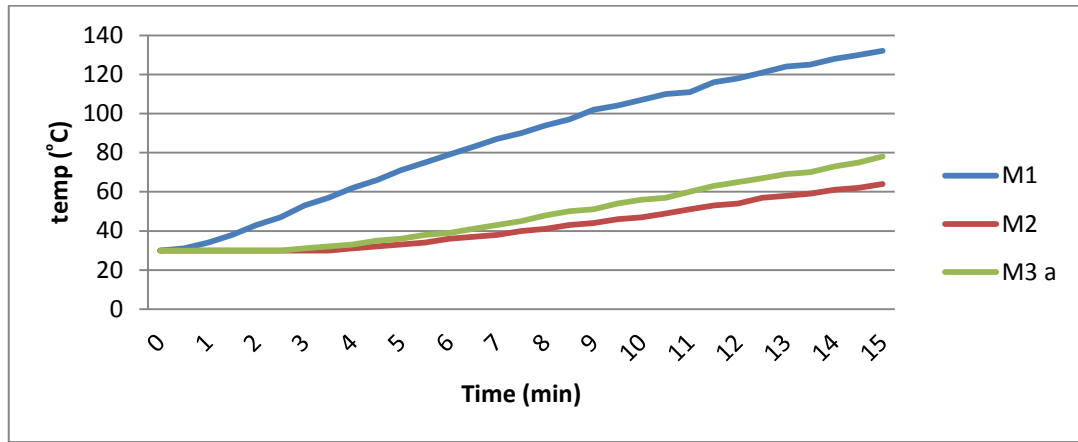


Figure 29: Rear face temperature-time graph

figure 29 indicated that when temperature at thermocouple position 1 (rear face) is compared with forced convection air flow cooling, the forced air flow cooling has better performance than that of single sheet metal and closer to sheet metal and wooden sandwich this makes it good option. From table 9a percentage heat reduction obtained are 60.9%, 82.2%, 80% respectively.

Front face

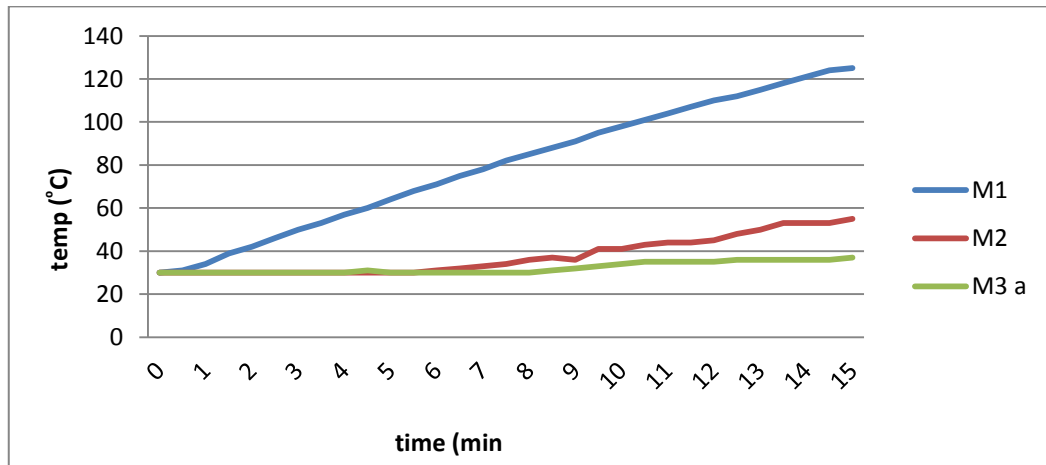


Figure 30: Front face temperature-time graph

The green line on figure 30 is the modified air flow temperature at thermocouple position 3 (front face) when compared with others, the forced air flow cooling has better performance than both of single sheet metal and sheet metal wooden sandwich. Percentage heat reduction obtained is 64.5%, 85%, 87.5% respectively.

Side surface

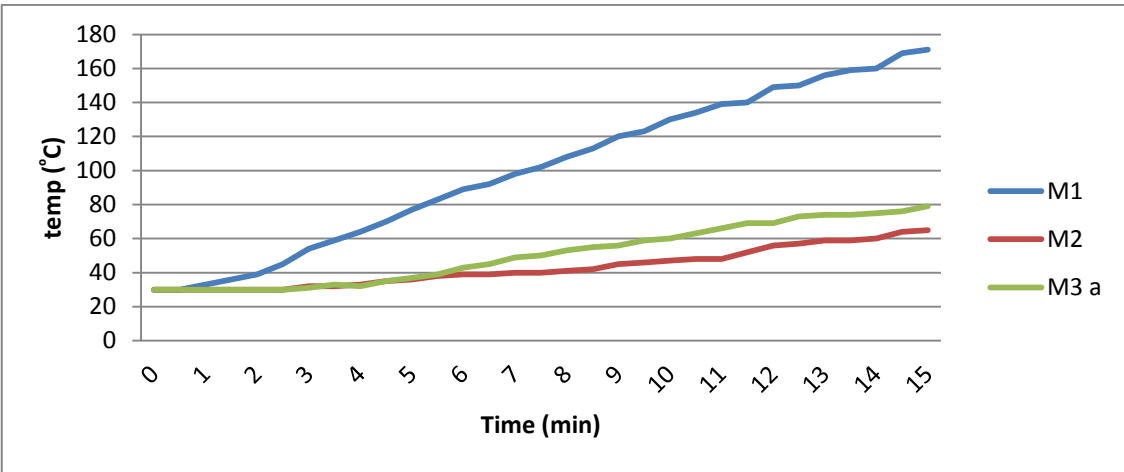


Figure 31: Side face location 4F temperature-time graph

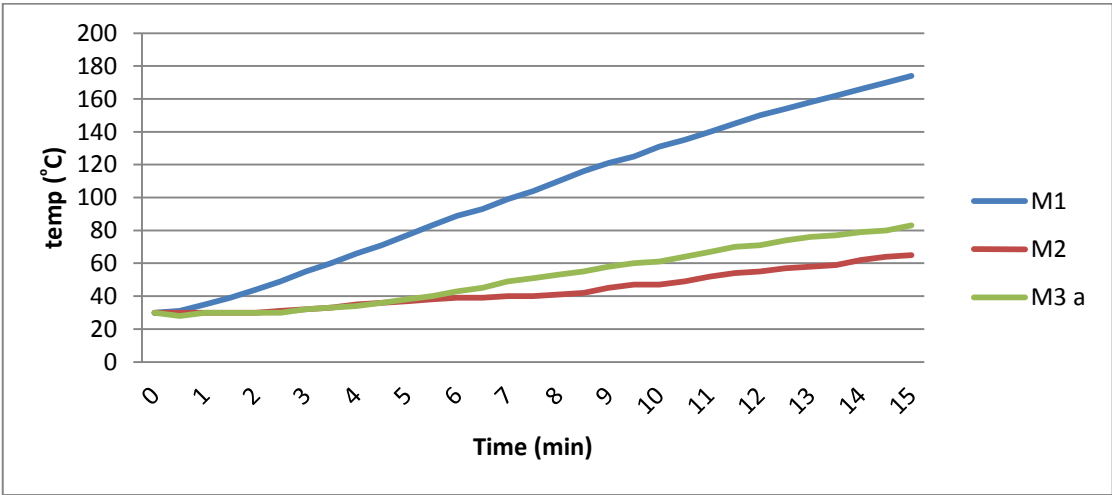


Figure 32: Side face location 4M temperature-time graph

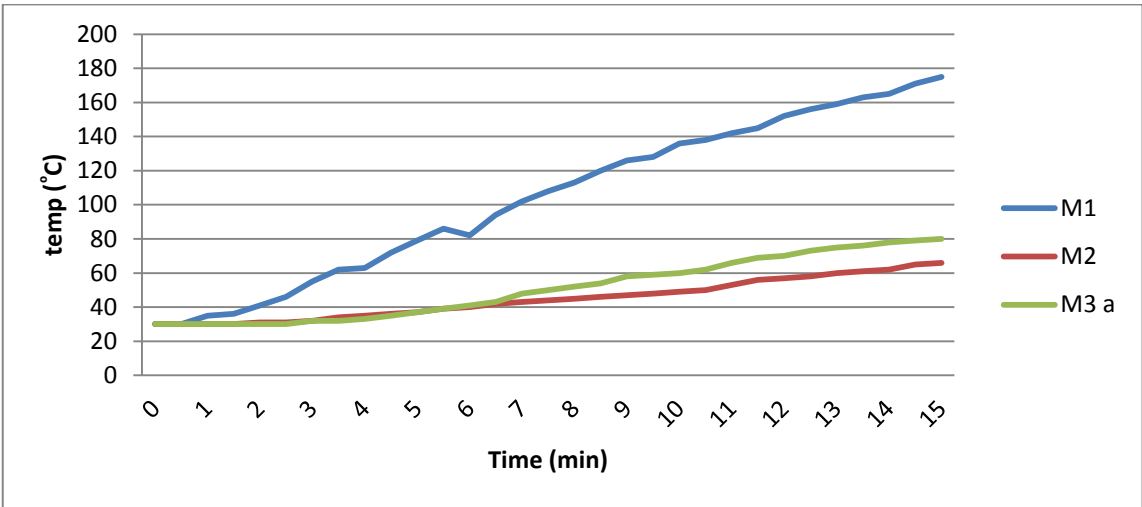


Figure 33: Side face location 4R temperature-time graph

Temperatures at thermocouple position side face were shown on the *figures31-33*.all the above three graphs show a result over the same face with closer values. When the others are compared with forced convection air flow cooling, the forced air flow cooling has better performance than that of single sheet metal and closer to sheet metal and wooden sandwich. Where, 4F, 4M, 4R are side face front position, middle and rear position as shown on *figures 27 and 28*.

Sheet metal and wooden sandwich has closer temperature reduction values both of them have better results than that of single sheet metal. From *tables (10a, b and 11)* respectively percentage heat reduction obtained are (55%, 82.9%, 77.9%), (54.1%, 82.9%, 77.9%), (52.9%, 81.2%, 78.3%) respectively.

Top surface

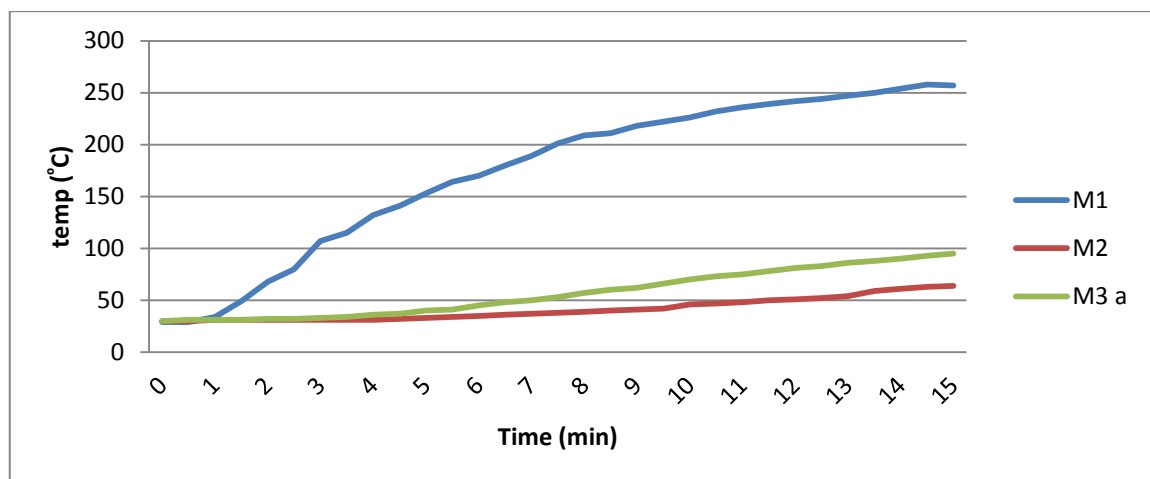


Figure 34: Top surface location 5F temperature-time graph

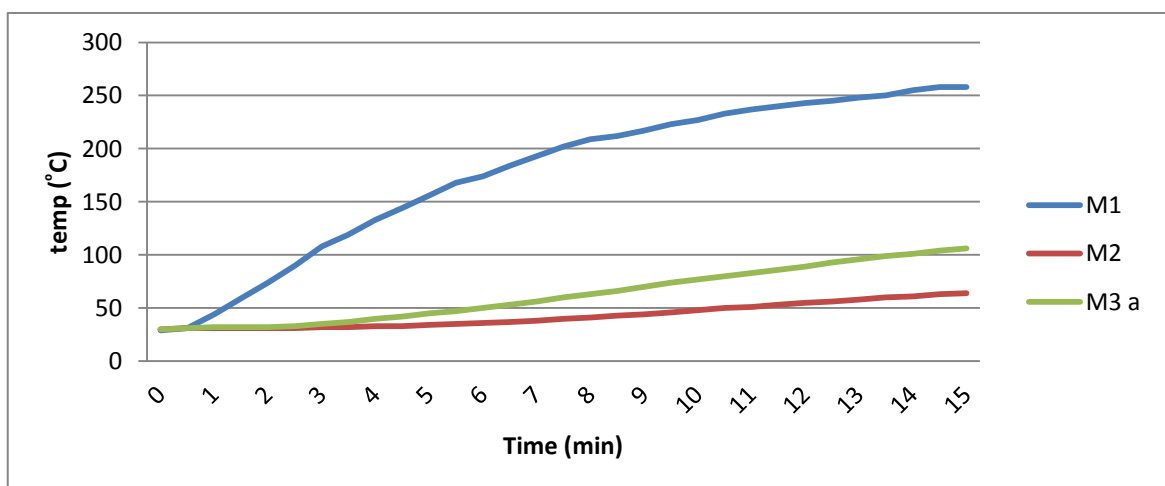


Figure 35: Top surface location 5M temperature-time graph

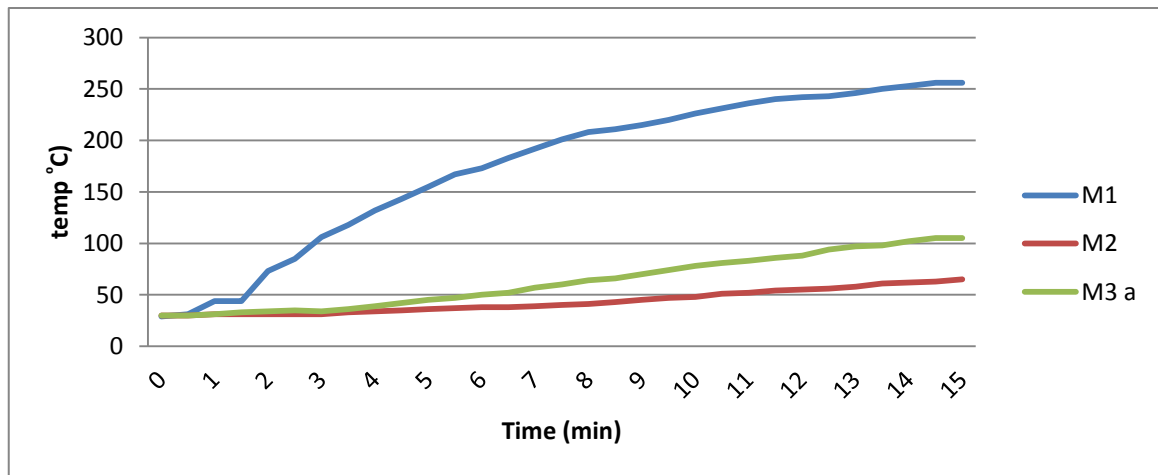


Figure 36: Top surface location 5R temperature-time graph

Here the obtained result on the above *figures34-36* for top position were also taken at three (3) locations (5F, 5M, 5R) as shown on *figures 27 and 28*.

The obtained results of forced convection air flow cooling were compared with others. The forced air flow cooling has better performance than that of single sheet metal and closer result were obtained with sheet metal and wooden. The obtained experimental results were shown on *tables12 and 13 a, b* where the 8th minute percentage temperature reduction becomes, (12.9%, 83.7%, and 76.2%), (12.9%, 82.1%, 73.7%), (13.4%, 82.1%, 73.3%) respectively.

Inside

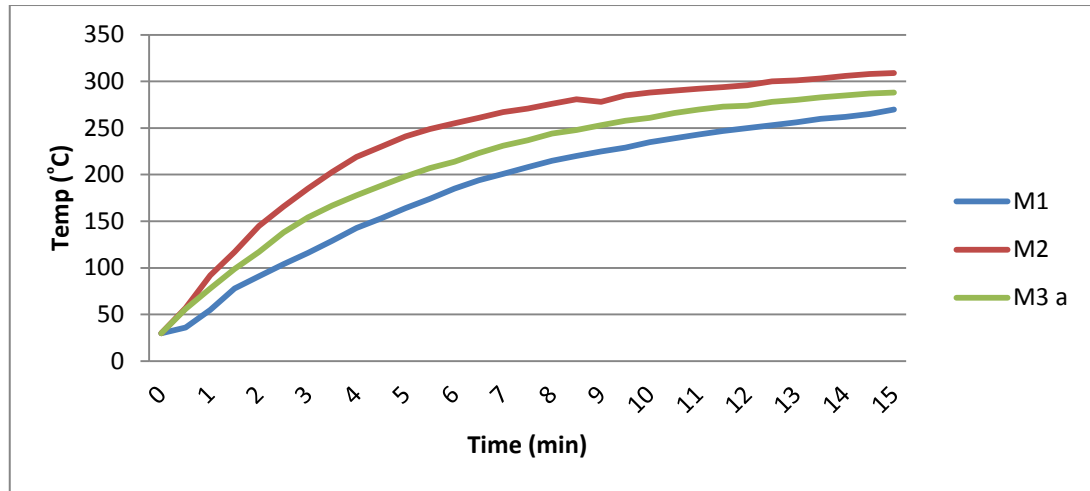


Figure 37: Inside compartment temperature-time graph

figure 37 shows the inside compartment temperature-time curve has better result than wooden sandwich insulation but the worst case of the single sheet metal plate shows better result than that of forced air flow convection system which is highly affecting the cabin temperature as shown in all the above graphs. The value of inside temperature here can be compared with the other after wall temperature differences depending on the time variation.

The temperatures at 8th min are 215°C, 276°C, and 244°C for single sheet metal, from table 19 wooden sandwiches and air flow cooling respectively.

Comparison of two models with the newly proposed solution in case of natural convection

Rear face

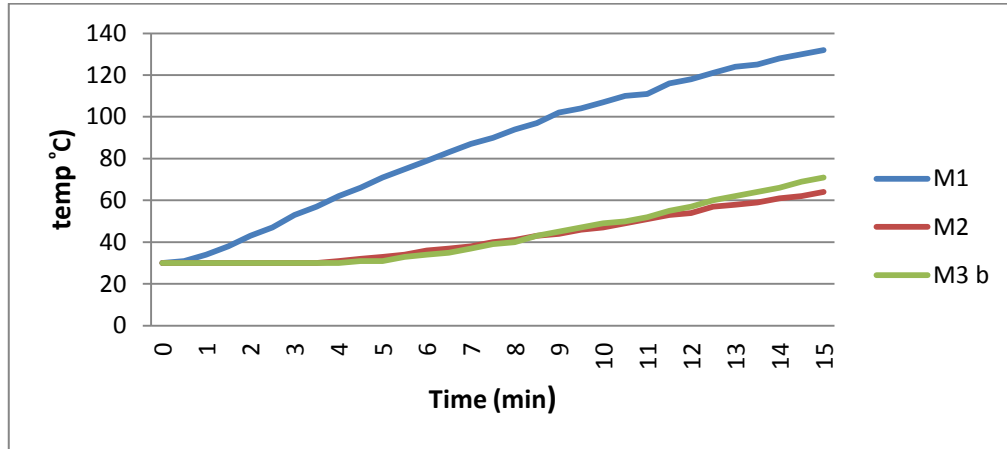


Figure 38: Rear face temperature-time graph

From the results obtained on *figure 38* the temperature at thermocouple position 1 (rear face) in *table 14a* shows the single sheet metal is still in worst condition but the result in case of wooden sandwich and air flow cooling has a good result. The 8th min percentage heat reduction obtained is 60.8%, 82.9%, 83.3% respectively.

Front face

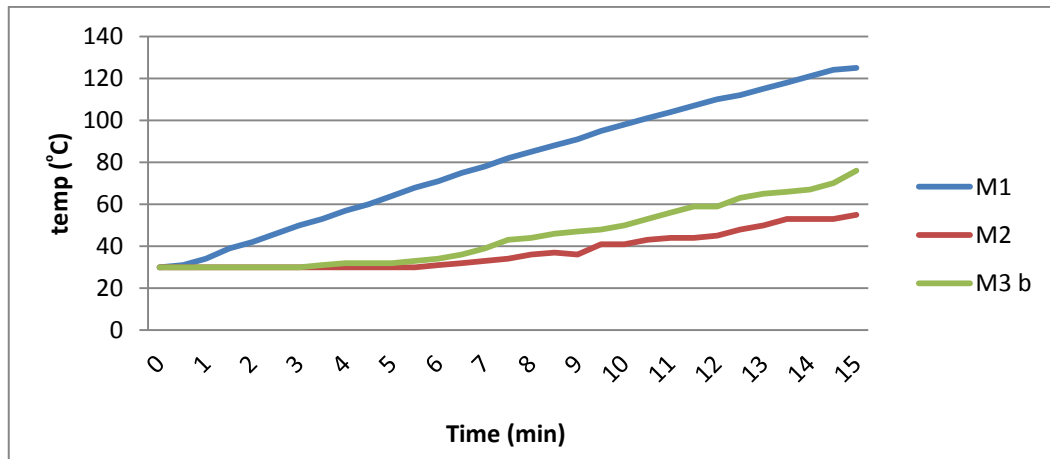


Figure 39: Front face temperature-time graph

On the above *figure 39* modified air flow (M3b) temperature at thermocouple position 3 (front face) when compared with others, the natural air flow cooling has good performance than single sheet metal but the results shows it has closer result with sheet metal wooden sandwich. As shown on *table (14b)* and, percentage heat reduction obtained is 64.5%, 85%, 81.6% respectively.

Side surface

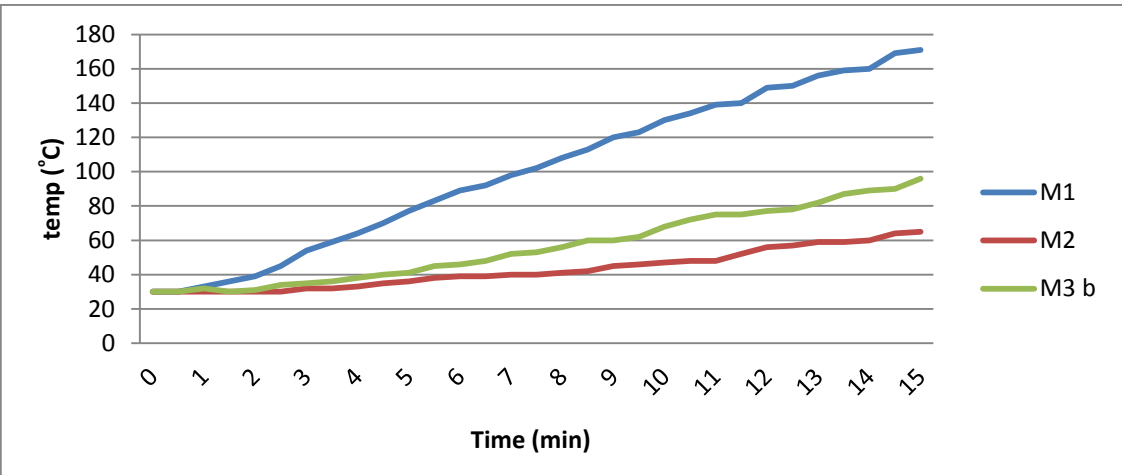


Figure 40: Side face location (1)4F temperature-time graph

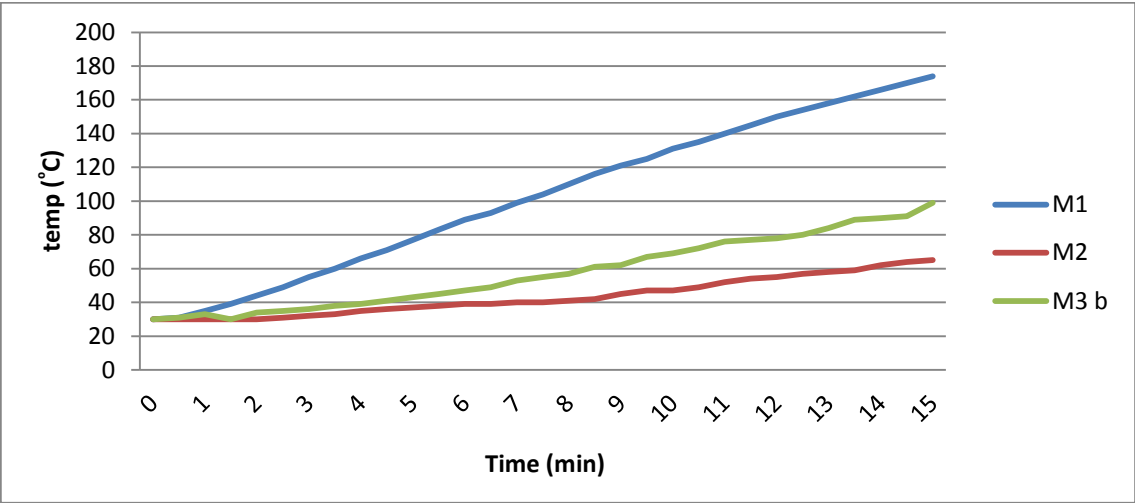


Figure 41: Side face location (2)4M temperature-time graph

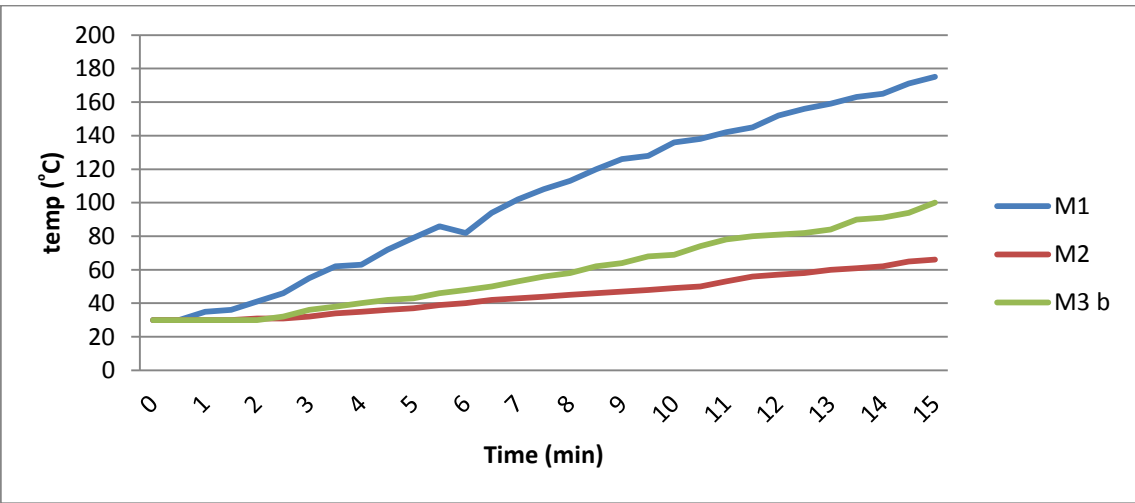


Figure 42: Side face location (3)4R temperature-time graph

In all the three (3) cases on the above *figures 40-42* (side face location 1, 2, 3) 4F, 4M, 4R) the only difference is that the reading were taken for single surface at different node but almost closer variation is there. When the others are compared with natural convection air flow cooling, it has better performance than that of single sheet metal and closer to sheet metal and wooden sandwich. Where, 4F, 4M, 4R are side face front position, middle and rear position as shown on *figures 41 and 42*, also the temperature increases toward the out let as per the result.

Respectively percentage heat reductions obtained from the graphs are (55%, 82.9%, and 76.6%) (54.2%, 82.9% and 76.2%) (52.9%, 81.2%, and 75.8%) respectively.

Top surface

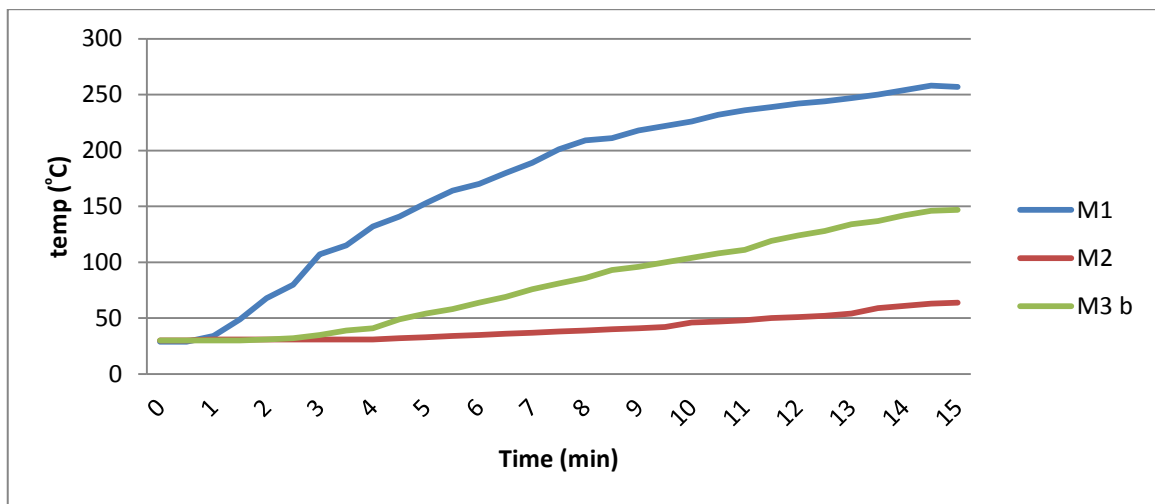


Figure 43: Top surface location (1) 5F temperature-time graph

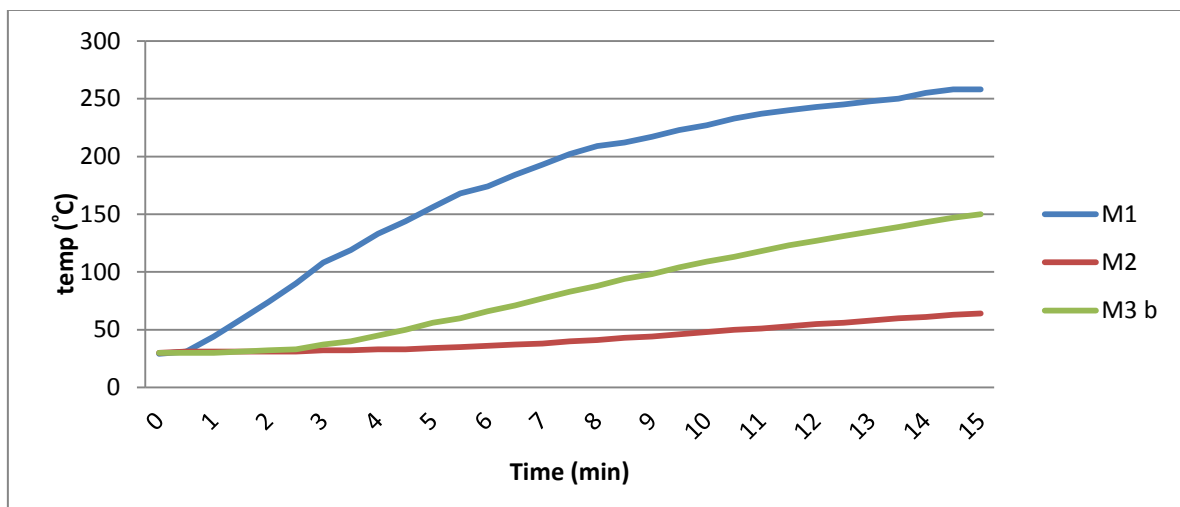


Figure 44: Top surface location (2) 5M temperature-time graph

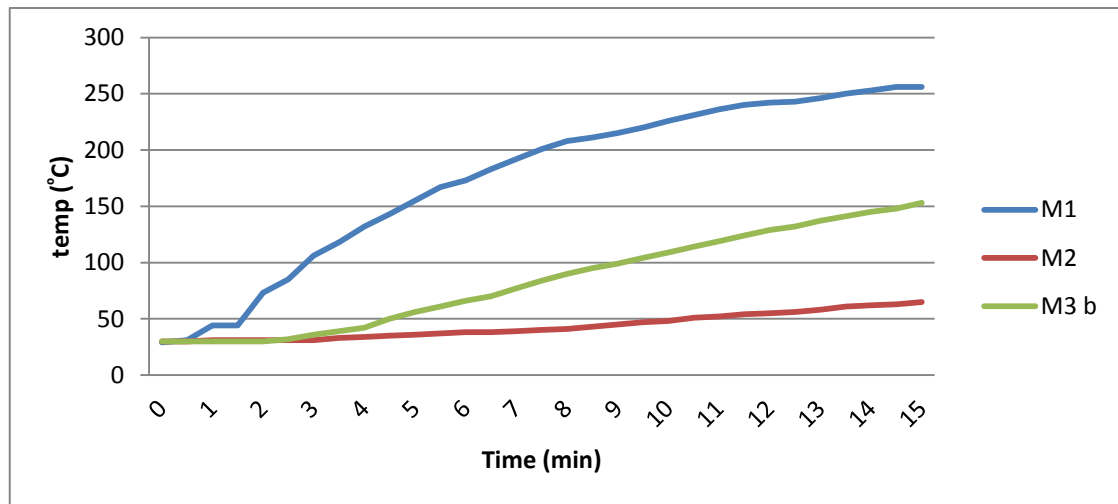


Figure 45: Top surface location (3) 5R temperature-time graph

The results obtained in these positions in figures 43-45 are the temperature for air flow cooling has better result than that of single sheet metal but the wooden sandwich has better result regardless of its high inside compartment temperature shown *table 20*.

In all the three (3) cases top surface location 5F, 5M, 5R as shown on *figures 27 and 28*, the only difference is that the reading were taken for single surface at different node but almost closer variation is there. The temperature increases toward the out let as per the result.

The obtained results of forced convection air flow cooling were compared with others. The forced air flow cooling has better performance than that of single sheet metal and closer result were obtained with sheet metal and wooden. The obtained experimental results were shown on *tables 17 and 18 a, b* where the 8th minute percentage temperature reduction becomes(12.9%,83.7%,64.1%)(12.9%,82.9%,63.3%)(13.4%,82.9%,62.5%)respectively.

Inside

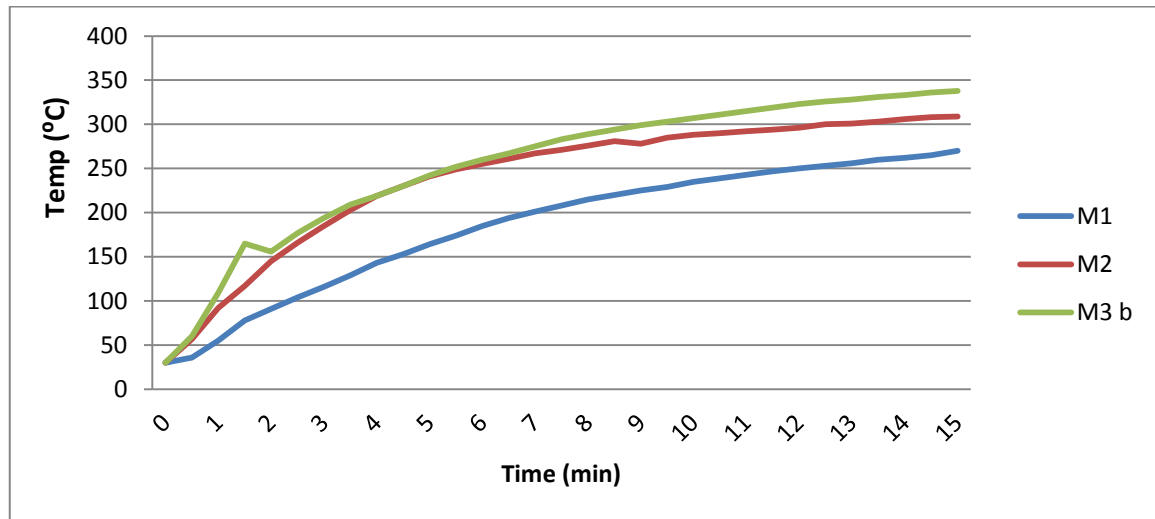


Figure 46: Position 2 inside compartment temperature-time graph

The figure 46 shows that the natural convective has closer result to that of wooden sandwich sheet metal. But the single sheet metal has lower heat than both since it is affecting the cabin environment highly since the heat transfer is greater than others.

4.3 Temperature change graph

4.3.1 Forced convection heat transfer temperature change (ΔT)

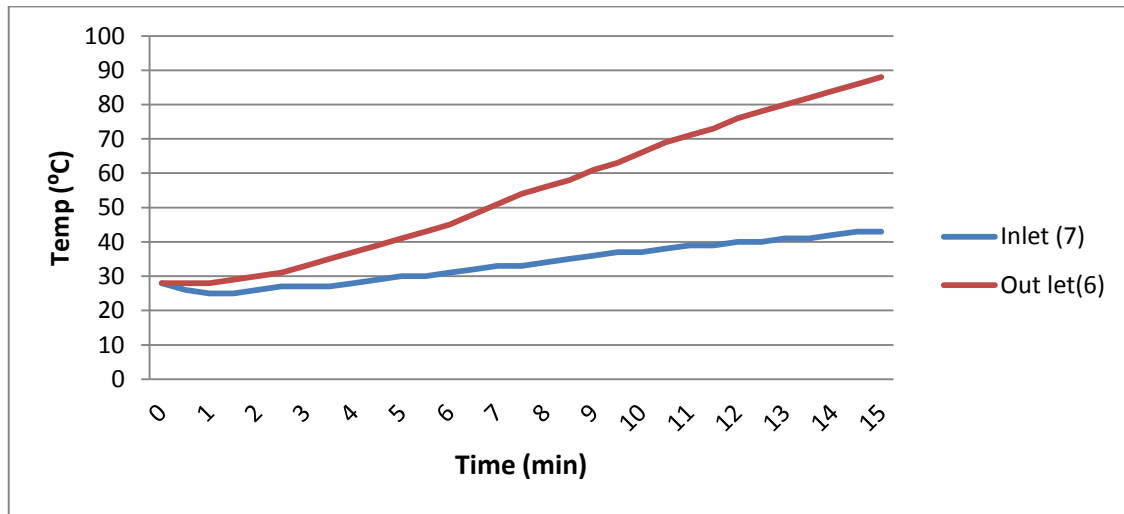


Figure 47: Inlet and outlet temperature-time graph in forced convection

This above *figure 47* result shows the air temperature at the outlet has greater value and also increasing than the temperature at inlet since it is carrying the heat generated inside the air gap. The numerical values are expressed in *table 21*.

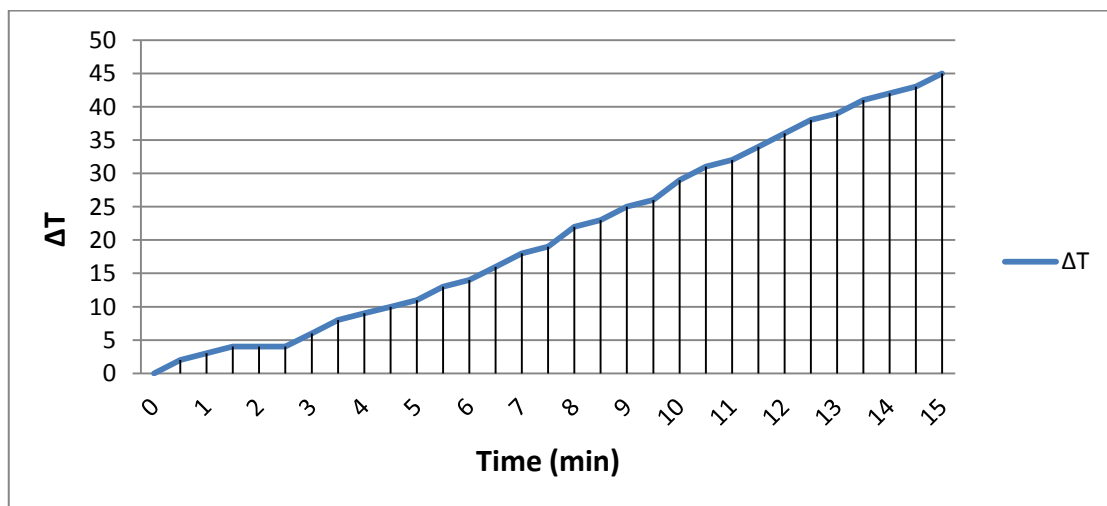


Figure 48: ΔT from inlet to out let with time graph in forced convection

Temperature rise or change of air on the *figures 47 and 48* is achieved due to hot inner surface of the compartment, this helps to disperse the hot environment before it reaches the outer surface. so the air temperature at the inlet rises when living through the out let. The graph shows continuous rise of temperature change with time as the engine temperature rises.

4.3.2 Natural convection heat transfer temperature change (ΔT)

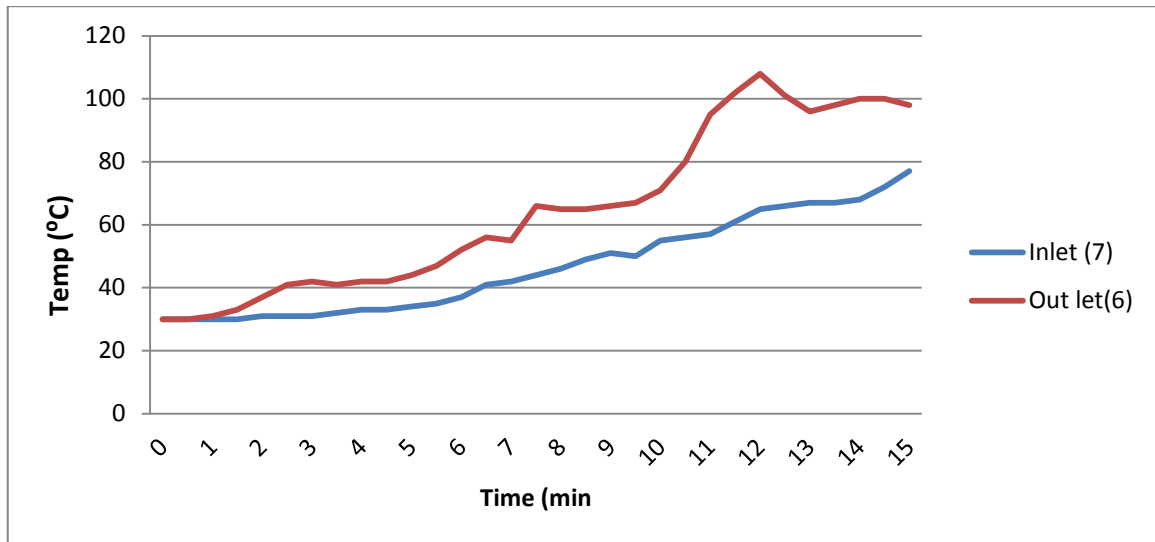


Figure 49: Inlet and outlet temperature-time graph in natural convection

This result on the *figure 49* shows the air temperature at the outlet has greater value and also increasing than the temperature at inlet as expressed in *table 22*.

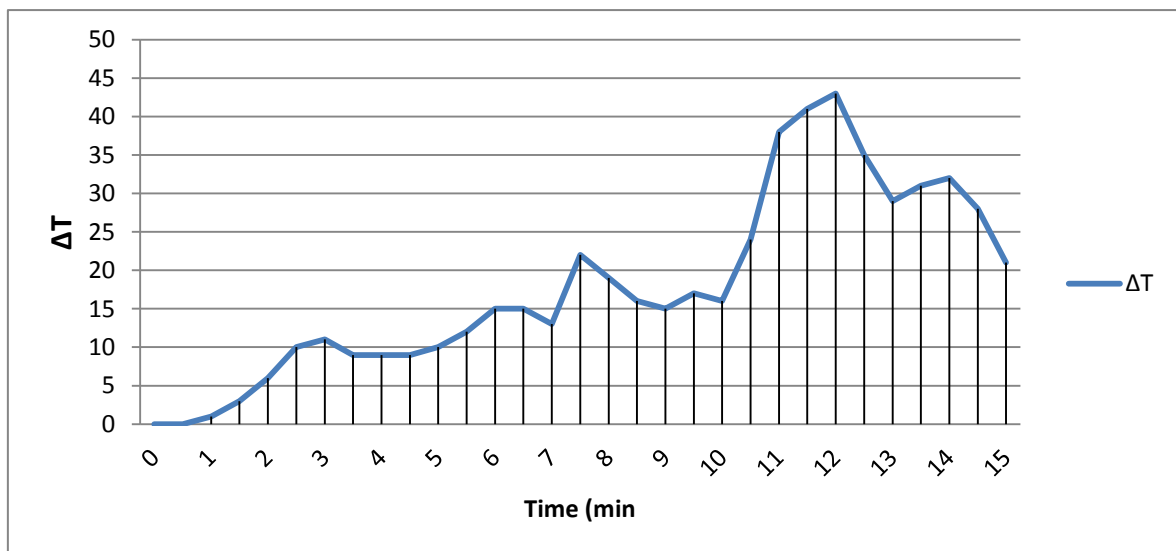


Figure 50: ΔT from inlet to outlet with time graph in natural convection

The outlet temperature shown on the *figures 49 and 50* has fluctuation because of variation of air in natural convection. Temperature rise or change of air is achieved due to hot inner surface of the compartment, this helps to disperse the hot air in the air gap space before it reaches the outer surface. This results the air temperature at the inlet rises when living through the outlet.

4.4 CFD load distribution result

4.4.1 Temperature distribution

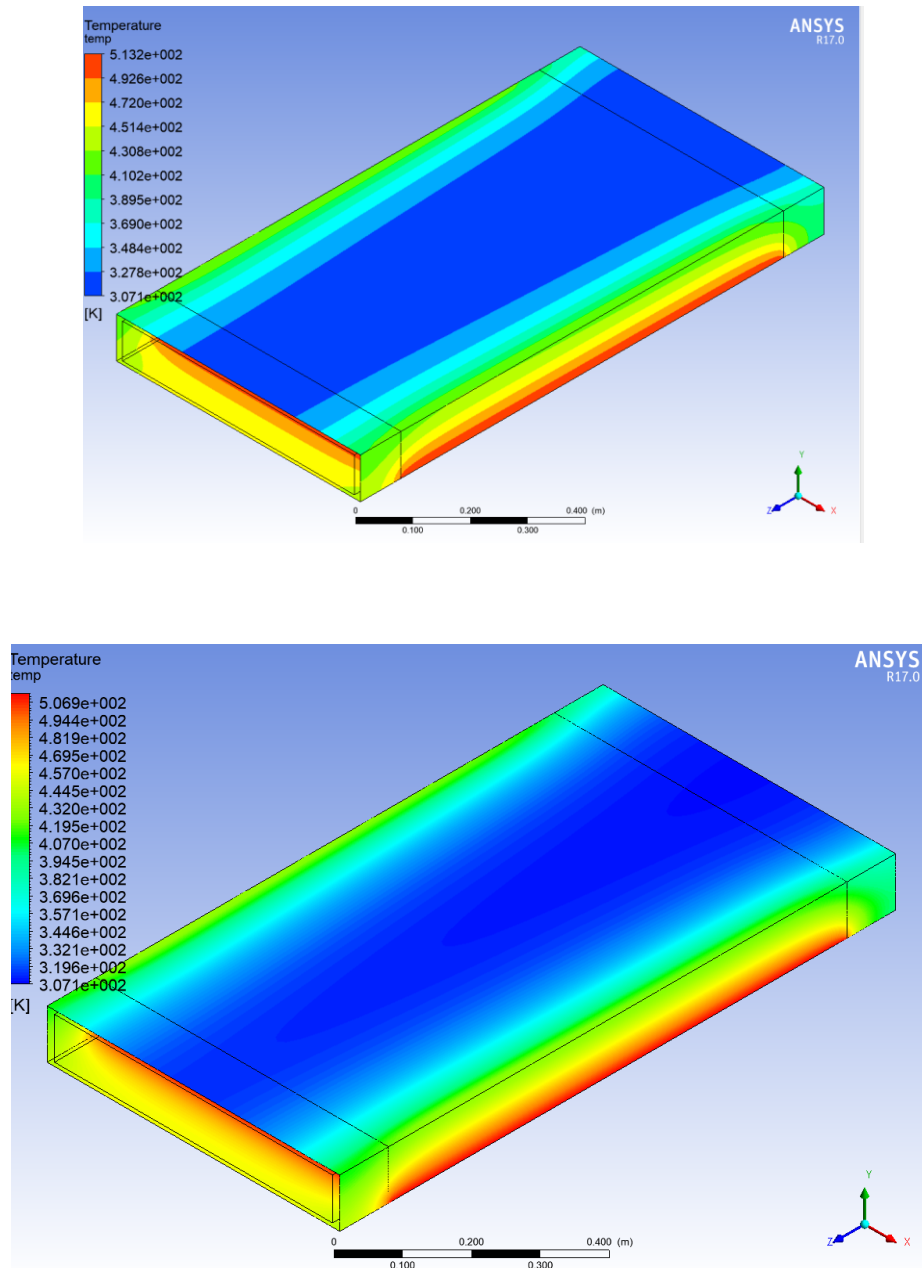


Figure 51: Temperature at the top

Referencing 240°C on *figure 50 and 51* found at 8th minute of forced convection inside temperature input was given. The result shows that the temperature at the location of the heater is 235.55°C(508.7K) is where 240°C is the input value which is displayed and the top lower value is but the reduced temperature at top position was 58.8°C(332K) which was obtained after the CFD analysis.

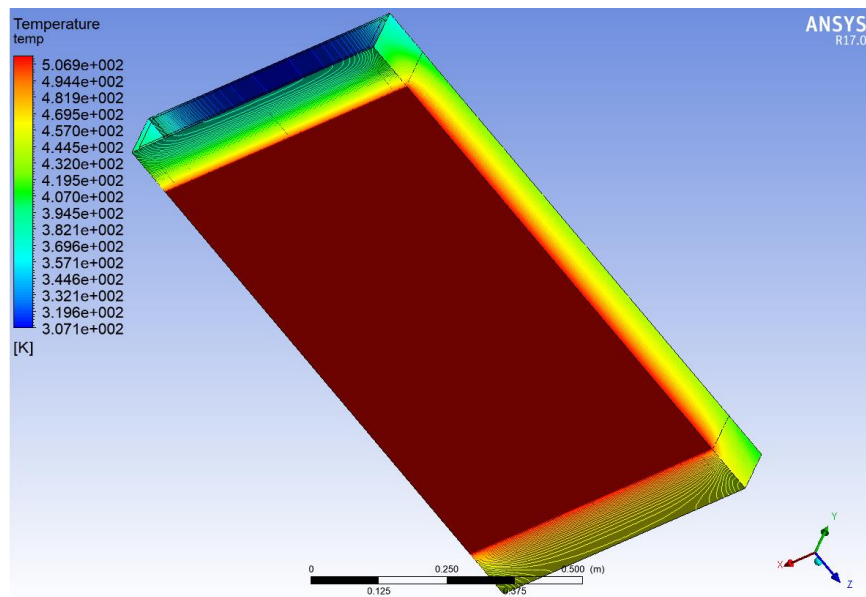


Figure 52: Temperature distribution at the bottom

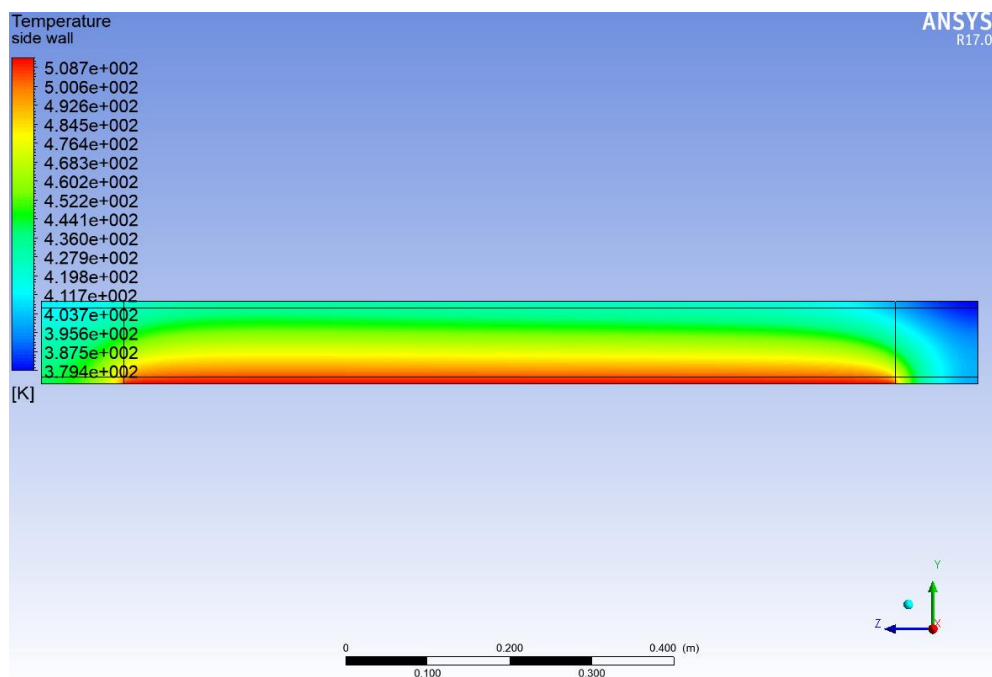


Figure 53: Temperature distribution at the side wall

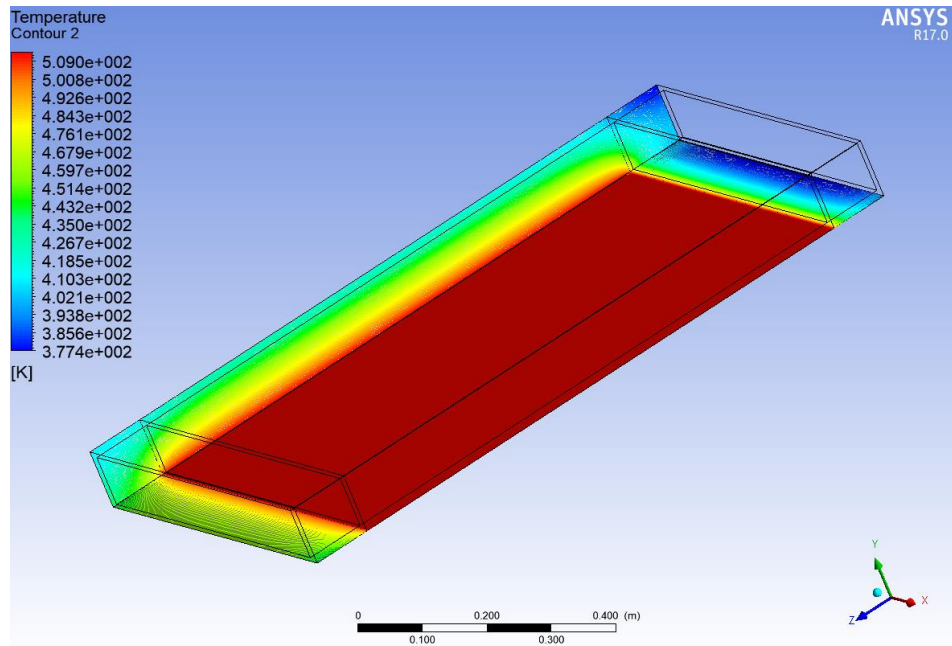


Figure 54: Temperature variation on the lower part of the flap

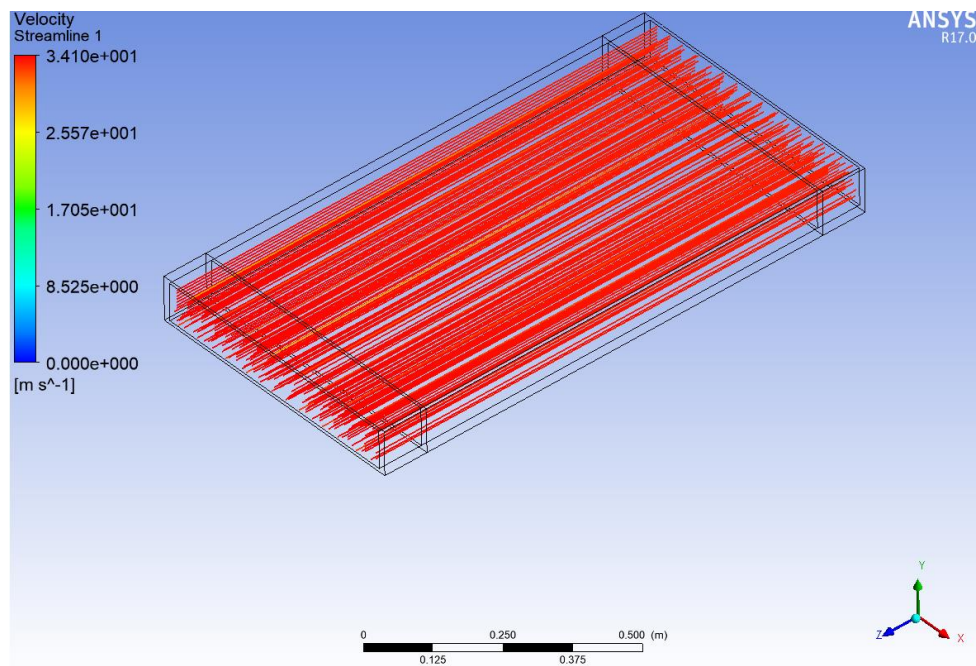


Figure 55: Velocity stream line

The flow pattern from inlet to out let is shown with the red colored stream lines on *figure 55*. And the variation of the velocity of air flowing is small which makes the stream lines uniform.

4.5 Discussion

Experimental values shows that the heat transfer in all the three methods were recorded and comparison was made, the result of the experiment shows that the heat transfer can be reduced if insulation is used. The computation here was that the wooden sandwich insulation between metals have reduced the heat transfer, while increasing the heat inside the engine compartment because the heat created beneath the floor don't have anywhere to go so it multiplies the inside temperature as shown on *figure 37,46*.

Figure 40 shows that the vehicle is in full throttle but whenever the engine is idle the result for the air flow cooling becomes less and less .On the other side, the experimental result shows that the sandwiched wood itself will lose its property through aging and heat which makes the burning surface it also have additional weight than that of air flow cooling.

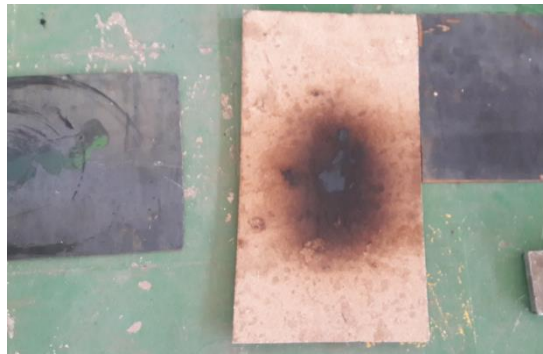


Figure 56: Burned wood (oak)

The inside heat causes the engine sensitive parts like hoses, electric wires and others to lose their property.

For wooden temperature let's take the inside temperature at 15min for single sheet metal, wooden sandwich with metal, air flow cooling, from figure 37, table 19 becomes 270°C, 309°C, 288°C while outside top middle from *figure 35, 17 a* shows 258°C, 64°C, 106°C note that the values taken are for the same type heater, at the same time the outside temperature of single sheet metal is maximum.

Since the single sheet metal is fully exposed to cabin environment and it uses atmospheric air as cooling medium but the forced air flow cooling system has a better performance in adjusting the engine temperature itself without affecting the cabin environment.

For this location when the convective heat transfer becomes natural which is not? Real since the vehicle is considered to be on full throttle, which is not possible but in this experiment it is assumed that the engine is working at full throttling even if the vehicle is at rest. Still the result of the single flat sheet metal here has lesser result since it is sharing the heat with the cabin environment, but the result shows that the natural convective has closer result to that of wooden sandwich sheet metal.

In air flow cooling as shown on *figures 42 and 44* the temperature increases as it is flowing through hot path and cooling. So the temperature at the inlet increases when living the air gap the curve shows the temperature change with time.

The values from CFD by selecting only the top or flap of the engine compartment since the materials are the same throughout the whole body; the flow line for air is available in every position as shown on *figure 11* so knowing the temperature distribution though out the whole surface becomes unnecessary therefore the flap is directly exposed to the heat and has wider area which is the reason for its selection in order to show the heat distribution.

Result comparison

Here the obtained results of the CFD and the experiment were compared for the temperature difference over the top surface for, 240°C inside temperature this shows the results are found in the closer range which is makes the overall results are valid.

Table 4: Result comparison

position	Measured Experimental value	CFD	Remark
Inside temperature	240 ⁰ C (513.15K)	235.55 (508.7)K	Table 19 input
Top position Temp	57-63 ⁰ C (330-336.15K) (table 12,13a,b)	33.8-58.8(307-332K) The obtained result from ANCYS	

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

- This study has revealed a new means of heat transfer reduction to passenger space just for the purpose of passenger comfort. The heat arising from engine compartment due to the burning fuel and driving force generated by the engine was different from vehicle to vehicle. The development by this study has reduced the heat transferring and heat multiplication inside the engine compartment. This heat affects some of engine components gradually so the principle of heat rejection was used, the air flowing through the air gap which do not have any contact with the engine body.
- Successful experimental analysis was done starting from preparing reduced scale prototype for the purpose of experiment .The data collection depending on the values from the thermocouple scale reading three different models have been analyzed. The results have been expressed through charts and tables. From the information gathered in literature review the maximum temperature inside engine compartment 240°C was selected for the purpose of CFD (ANSYS) analysis. The natural convection was done with assumptions same as forced convection even if there is no forced heat rejection the heater creates the same heat for others.
- The proposed air flow cooling system reduces heat transfer in 69.86 % better when compared to sheet metal which is worse for the passenger compartment. This shows that heat transfer of wooden sandwich metal has reduced 80.84 % while it is heavier since it has additional material for insulation purpose the sandwiched wood will lose its composition after some service time.
- Here in maintaining the temperature inside the compartment, the air flow cooling system has better result than others. Taking 240°C inside temperature the air flow cooling system maintains the heat by 11.6 % when compared to metal wooden sandwich method. While the experimental value of inside temperature of single flat plate sheet metal shows less value since it completely shares the heat with cabin environment which will affect the passengers comfort negatively. This makes the air flow cooling selected for its two side advantage.

- The CFD analysis shows the temperature distribution of air for the selected critical inside temperature. The heat distribution is clearly shown on the result of the analysis. The main heat transfer mode analyzed was forced convection, since natural convection was not common incidence since there is no one stepping on the throttle continuously. When the vehicle is at rest and engine is running, the engine will work on idle which have less temperature. While the experiment was done for the worst situation.

5.2 Recommendation

- Every vehicle assembling and body and frame companies can use this means of heat reduction for the purpose of passenger comfort and engine wellness as well.
- Any other students researchers should do more design if any luck is here on this issue of heat transfer.
- The department of automotive engineering should work with thermal engineering in order to solve this issue with better understanding and performance.
- Testing and experimental setup measuring should be prepared for feature working, which was a challenge during the work.
- CFD is a powerful tool this university should work on creating skilled students by adding this software as a single course.
- Additionally there are common insulating coating being used in modern vehicles in order to increase the performance of the proposed system it is possible to add additional insulations as example, aluminum layered expanded polystyrene.

5.3 Feature Work

Here most of the data's were taken from literature, while some were measured, the analysis was made for the compartment only the other parts like, (nozzle, if possible additional fan) can be designed.

If any change is necessary it is better to work on it, as a future work any one can show the overall heat transfer, the effects on engine and any other issue.

It was difficult to do this thesis since no body has done it which makes it difficult for me but now it is already proposed in this thesis so as this work is first work many more things and modifications can be done while the references obtained are related issues.

The vehicle selected here was medium buses, but by changing the shape size, vehicle type, orientation, and location as per engine specification and position using the same procedure or air flow heat rejection the heat transfer can be reduced as well as engine temperature can be adjusted.

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APPENDIX I

Experimental data's

Table 5: EC with flat plate (flat sheet metal) temperature-time values at different locations.

Time (min)	Temperature(°C)								
Thermocouple position	1	2	3	5 f	5m	5 r	4F	4M	4R
0	30	30	30	29	29	29	30	30	30
0:30	3	36	31	29	31	31	30	31	30
1	34	55	34	34	44	44	33	35	35
1:3	38	78	39	49	59	44	36	39	36
2	43	91	4	68	74	73	39	44	41
2:30	7	104	46	80	90	85	45	49	46
3	3	116	50	107	108	106	54	55	55
3:30	57	129	53	115	119	118	59	60	62
4	62	143	57	132	133	132	64	66	63
4:30	66	153	60	141	144	143	70	71	72
5	71	164	64	153	156	155	77	77	79
5:30	75	174	68	164	168	167	83	83	86
6	79	185	71	170	174	173	89	89	82
6:30	83	194	75	180	184	183	92	93	94
7	87	201	78	189	193	192	98	99	102
7:30	90	208	82	201	202	201	102	104	108
8	94	215	85	209	209	208	108	110	113
8:30	97	220	88	211	212	211	113	116	120
9	102	225	91	218	217	215	120	121	126
9:30	104	229	95	222	223	220	123	125	128
10	107	235	98	226	227	226	130	131	136
10:30	110	239	101	232	233	231	134	135	138
11	111	243	104	236	237	236	139	140	142
11:30	116	247	107	239	240	240	140	145	145
12	118	250	110	242	243	242	149	150	152
12:30	121	253	112	244	245	243	150	154	156
13	124	256	115	247	248	246	156	158	159
13:30	125	260	118	250	250	250	159	162	163
14	128	262	121	254	255	253	160	166	165
14:30	130	265	124	258	258	256	169	170	171
15	132	270	125	257	258	256	171	174	175

Table 6: Flat plate and wood sandwich insulation temperature-time values at different positions.

Time (min)	Temperature(°C)								
Thermocouple position	1	2	3	5 f	5m	5 r	4F	4M	4R
0	30	30	30	30	30	30	30	30	30
0:30	30	57	30	30	31	30	30	30	30
1	30	92	30	31	31	31	30	30	30
1:30	30	117	30	31	31	31	30	30	30
2	30	145	30	31	31	31	30	30	31
2:30	30	166	30	31	31	31	30	31	31
3	30	185	30	31	32	31	32	32	32
3:30	30	203	30	31	32	33	32	33	34
4	31	219	30	31	33	34	33	35	35
4:30	32	230	30	32	33	35	35	36	36
5	33	241	30	33	34	36	36	37	37
5:30	34	249	30	34	35	37	38	38	39
6	36	255	31	35	36	38	39	39	40
6:30	37	261	32	36	37	38	39	39	42
7	38	267	33	37	38	39	40	40	43
7:30	40	271	34	38	40	40	40	40	44
8	41	276	36	39	41	41	41	41	45
8:30	43	281	37	40	43	43	42	42	46
9	44	278	36	41	44	45	45	45	47
9:30	46	285	41	42	46	47	46	47	48
10	47	288	41	46	48	48	47	47	49
10:30	49	290	43	47	50	51	48	49	50
11	51	292	44	48	51	52	48	52	53
11:30	53	294	44	50	53	54	52	54	56
12	54	296	45	51	55	55	56	55	57
12:30	57	300	48	52	56	56	57	57	58
13	58	301	50	54	58	58	59	58	60
13:30	59	303	53	59	60	61	59	59	61
14	61	306	53	61	61	62	60	62	62
14:30	62	308	53	63	63	63	64	64	65
15	64	309	55	64	64	65	65	65	66

Table 7: Vehicle with air flow and maximum velocity (forced convection heat reduction)

Time min)	Temperature(°C)										
T pt.	1	2	3	5 f	5m	5 r	4F	4M	4R	6	7
0	30	30	30	30	30	30	30	30	30	28	28
0:30	30	56	30	31	31	30	30	28	30	28	26
1	30	78	30	31	32	31	30	30	30	28	25
1:30	30	99	30	31	32	33	30	30	30	29	25
2	30	117	30	32	32	34	30	30	30	30	26
2:30	30	138	30	32	33	35	30	30	30	31	27
3	31	154	30	33	35	34	31	32	32	33	27
3:30	32	167	30	34	37	36	33	33	32	35	27
4	33	178	30	36	40	39	32	34	33	37	28
4:30	35	188	31	37	42	42	35	36	35	39	29
5	36	198	30	40	45	45	37	38	37	41	30
5:30	38	207	30	41	47	47	39	40	39	43	30
6	39	214	30	45	50	50	43	43	41	45	31
6:30	41	223	30	48	53	52	45	45	43	48	32
7	43	231	30	50	56	57	49	49	48	51	33
7:30	45	237	30	53	60	60	50	51	50	54	33
8	48	244	30	57	63	64	53	53	52	56	34
8:30	50	248	31	60	66	66	55	55	54	58	35
9	51	253	32	62	70	70	56	58	58	61	36
9:30	54	258	33	66	74	74	59	60	59	63	37
10	56	261	34	70	77	78	60	61	60	66	37
10:30	57	266	35	73	80	81	63	64	62	69	38
11	60	270	35	75	83	83	66	67	66	71	39
11:30	63	273	35	78	86	86	69	70	69	73	39
12	65	274	35	81	89	88	69	71	70	76	40
12:30	67	278	36	83	93	94	73	74	73	78	40
13	69	280	36	86	96	97	74	76	75	80	41
13:30	70	283	36	88	99	98	74	77	76	82	41
14	73	285	36	90	101	102	75	79	78	84	42
14:30	75	287	36	93	104	105	76	80	79	86	43
15	78	288	37	95	106	105	79	83	80	88	43

Table 8: Vehicle at rest but whenever the engine is working with full throttle

Time min)	Temperature(°C)										
T pt.	1	2	3	5 f	5m	5 r	4F	4M	4R	6	7
0		30	30	30	30	30	30	30	30	30	30
0:30		60	30	30	30	30	30	31	30	30	30
1		109	30	30	30	30	32	33	30	31	30
1:30		165	30	30	31	30	30	30	30	33	30
2		156	30	31	32	30	31	34	30	37	31
2:30		177	30	32	33	32	34	35	32	41	31
3		194	30	35	37	36	35	36	36	42	31
3:30		209	31	39	40	39	36	38	38	41	32
4		219	32	41	45	42	38	39	40	42	33
4:30		230	32	49	50	50	40	41	42	42	33
5		242	32	54	56	56	41	43	43	44	34
5:30		252	33	58	60	61	45	45	46	47	35
6		260	34	64	66	66	46	47	48	52	37
6:30		267	36	69	71	70	48	49	50	56	41
7		275	39	76	77	77	52	53	53	55	42
7:30		283	43	81	83	84	53	55	56	66	44
8		289	44	86	88	90	56	57	58	65	46
8:30		294	46	93	94	95	60	61	62	65	49
9		299	47	96	98	99	60	62	64	66	51
9:30		303	48	100	104	104	62	67	68	67	50
10		307	50	104	109	109	68	69	69	71	55
10:30		311	53	108	113	114	72	72	74	80	56
11		315	56	111	118	119	75	76	78	95	57
11:30		319	59	119	123	124	75	77	80	102	61
12		323	59	124	127	129	77	78	81	108	65
12:30		326	63	128	131	132	78	80	82	101	66
13		328	65	134	135	137	82	84	84	96	67
13:30		331	66	137	139	141	87	89	90	98	67
14		333	67	142	143	145	89	90	91	100	68
14:30		336	70	146	147	148	90	91	94	100	72
15		338	76	147	150	153	96	99	100	98	77

APPENDIX II

Comparison of results

Comparison of two models with the newly proposed solution in case of forced convection

Table 9: a, T at position 1 (rear face) v/s FC, b: T at position 3 (front face) v/s FC

Temperature(°C)			
Time (min)	Model		
	M1	M2	M3 a
0	30	30	30
0:30	31	30	30
1	34	30	30
1:30	38	30	30
2	43	30	30
2:30	47	30	30
3	53	30	31
3:30	57	30	32
4	62	31	33
4:30	66	32	35
5	71	33	36
5:30	75	34	38
6	79	36	39
6:30	83	37	41
7	87	38	43
7:30	90	40	45
8	94	41	48
8:30	97	43	50
9	102	44	51
9:30	104	46	54
10	107	47	56
10:30	110	49	57
11	111	51	60
11:30	116	53	63
12	118	54	65
12:30	121	57	67
13	124	58	69
13:30	125	59	70
14	128	61	73
14:30	130	62	75
15	132	64	78

(a)

Temperature(°C)			
Time (min)	Model		
	M1	M2	M3 a
0	30	30	30
0:30	31	30	30
1	34	30	30
1:30	39	30	30
2	42	30	30
2:30	46	30	30
3	50	30	30
3:30	53	30	30
4	57	30	30
4:30	60	30	31
5	64	30	30
5:30	68	30	30
6	71	31	30
6:30	75	32	30
7	78	33	30
7:30	82	34	30
8	85	36	30
8:30	88	37	31
9	91	36	32
9:30	95	41	33
10	98	41	34
10:30	101	43	35
11	104	44	35
11:30	107	44	35
12	110	45	35
12:30	112	48	36
13	115	50	36
13:30	118	53	36
14	121	53	36
14:30	124	53	36
15	125	55	37

(b)

Table 10: (a) T at position (side face location 1)4F v/s FC. (b), T at position 4(side face location 2) 4M v/s FC

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 a
0	30	30	30
0:30	30	30	30
1	33	30	30
1:30	36	30	30
2	39	30	30
2:30	45	30	30
3	54	32	31
3:30	59	32	33
4	64	33	32
4:30	70	35	35
5	77	36	37
5:30	83	38	39
6	89	39	43
6:30	92	39	45
7	98	40	49
7:30	102	40	50
8	108	41	53
8:30	113	42	55
9	120	45	56
9:30	123	46	59
10	130	47	60
10:30	134	48	63
11	139	48	66
11:30	140	52	69
12	149	56	69
12:30	150	57	73
13	156	59	74
13:30	159	59	74
14	160	60	75
14:30	169	64	76
15	171	65	79

(a)

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 a
0	30	30	30
0:30	31	30	28
1	35	30	30
1:30	39	30	30
2	44	30	30
2:30	49	31	30
3	55	32	32
3:30	60	33	33
4	66	35	34
4:30	71	36	36
5	77	37	38
5:30	83	38	40
6	89	39	43
6:30	93	39	45
7	99	40	49
7:30	104	40	51
8	110	41	53
8:30	116	42	55
9	121	45	58
9:30	125	47	60
10	131	47	61
10:30	135	49	64
11	140	52	67
11:30	145	54	70
12	150	55	71
12:30	154	57	74
13	158	58	76
13:30	162	59	77
14	166	62	79
14:30	170	64	80
15	174	65	83

(b)

Table 11: T at position 4 (side face location 3) 4R v/s FC,**Table 12:** T at position 5 (Top surface location 1) 5F v/s FC,

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 a
0	30	30	30
0:30	30	30	30
1	35	30	30
1:30	36	30	30
2	41	31	30
2:30	46	31	30
3	55	32	32
3:30	62	34	32
4	63	35	33
4:30	72	36	35
5	79	37	37
5:30	86	39	39
6	82	40	41
6:30	94	42	43
7	102	43	48
7:30	108	44	50
8	113	45	52
8:30	120	46	54
9	126	47	58
9:30	128	48	59
10	136	49	60
10:30	138	50	62
11	142	53	66
11:30	145	56	69
12	152	57	70
12:30	156	58	73
13	159	60	75
13:30	163	61	76
14	165	62	78
14:30	171	65	79
15	175	66	80

(T-9)

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 a
0	29	30	30
0:30	29	30	31
1	34	31	31
1:30	49	31	31
2	68	31	32
2:30	80	31	32
3	107	31	33
3:30	115	31	34
4	132	31	36
4:30	141	32	37
5	153	33	40
5:30	164	34	41
6	170	35	45
6:30	180	36	48
7	189	37	50
7:30	201	38	53
8	209	39	57
8:30	211	40	60
9	218	41	62
9:30	222	42	66
10	226	46	70
10:30	232	47	73
11	236	48	75
11:30	239	50	78
12	242	51	81
12:30	244	52	83
13	247	54	86
13:30	250	59	88
14	254	61	90
14:30	258	63	93
15	257	64	95

(T-10)

Table 13: (a) Tat position 5 (top surface location 2) 5M v/s FC (b) Tat position 5 (top surface location 3) 5R v/s FC,

Temperature(°C)			
Time (min)	Model		
	M1	M2	M3 a
0	29	30	30
0:30	31	31	31
1	44	31	32
1:30	59	31	32
2	74	31	32
2:30	90	31	33
3	108	32	35
3:30	119	32	37
4	133	33	40
4:30	144	33	42
5	156	34	45
5:30	168	35	47
6	174	36	50
6:30	184	37	53
7	193	38	56
7:30	202	40	60
8	209	41	63
8:30	212	43	66
9	217	44	70
9:30	223	46	74
10	227	48	77
10:30	233	50	80
11	237	51	83
11:30	240	53	86
12	243	55	89
12:30	245	56	93
13	248	58	96
13:30	250	60	99
14	255	61	101
14:30	258	63	104
15	258	64	106

(a)

Temperature(°C)			
Time (min)	Model		
	M1	M2	M3 a
0	29	30	30
0:30	31	30	30
1	44	31	31
1:30	44	31	33
2	73	31	34
2:30	85	31	35
3	106	31	34
3:30	118	33	36
4	132	34	39
4:30	143	35	42
5	155	36	45
5:30	167	37	47
6	173	38	50
6:30	183	38	52
7	192	39	57
7:30	201	40	60
8	208	41	64
8:30	211	43	66
9	215	45	70
9:30	220	47	74
10	226	48	78
10:30	231	51	81
11	236	52	83
11:30	240	54	86
12	242	55	88
12:30	243	56	94
13	246	58	97
13:30	250	61	98
14	253	62	102
14:30	256	63	105
15	256	65	105

(b)

Comparison of two models with the newly proposed solution in case of natural convection

Table 14: (a) T at position 1 (rear face) and v/s NC. (b), T at position 3 (front face) and v/s NC.

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	30	30	30
0:30	31	30	30
1	34	30	30
1:30	38	30	30
2	43	30	30
2:30	47	30	30
3	53	30	30
3:30	57	30	30
4	62	31	30
4:30	66	32	31
5	71	33	31
5:30	75	34	33
6	79	36	34
6:30	83	37	35
7	87	38	37
7:30	90	40	39
8	94	41	40
8:30	97	43	43
9	102	44	45
9:30	104	46	47
10	107	47	49
10:30	110	49	50
11	111	51	52
11:30	116	53	55
12	118	54	57
12:30	121	57	60
13	124	58	62
13:30	125	59	64
14	128	61	66
14:30	130	62	69
15	132	64	71

(a)

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	30	30	30
0:30	31	30	30
1	34	30	30
1:30	39	30	30
2	42	30	30
2:30	46	30	30
3	50	30	30
3:30	53	30	31
4	57	30	32
4:30	60	30	32
5	64	30	32
5:30	68	30	33
6	71	31	34
6:30	75	32	36
7	78	33	39
7:30	82	34	43
8	85	36	44
8:30	88	37	46
9	91	36	47
9:30	95	41	48
10	98	41	50
10:30	101	43	53
11	104	44	56
11:30	107	44	59
12	110	45	59
12:30	112	48	63
13	115	50	65
13:30	118	53	66
14	121	53	67
14:30	124	53	70
15	125	55	76

(b)

Table 15: (a) T at position 4(side face location 1) 4 F v/s NC (b) T at position 4(side face location 2)4M v/s NC

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	30	30	30
0:30	30	30	30
1	33	30	32
1:30	36	30	30
2	39	30	31
2:30	45	30	34
3	54	32	35
3:30	59	32	36
4	64	33	38
4:30	70	35	40
5	77	36	41
5:30	83	38	45
6	89	39	46
6:30	92	39	48
7	98	40	52
7:30	102	40	53
8	108	41	56
8:30	113	42	60
9	120	45	60
9:30	123	46	62
10	130	47	68
10:30	134	48	72
11	139	48	75
11:30	140	52	75
12	149	56	77
12:30	150	57	78
13	156	59	82
13:30	159	59	87
14	160	60	89
14:30	169	64	90
15	171	65	96

(a)

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	30	30	30
0:30	31	30	31
1	35	30	33
1:30	39	30	30
2	44	30	34
2:30	49	31	35
3	55	32	36
3:30	60	33	38
4	66	35	39
4:30	71	36	41
5	77	37	43
5:30	83	38	45
6	89	39	47
6:30	93	39	49
7	99	40	53
7:30	104	40	55
8	110	41	57
8:30	116	42	61
9	121	45	62
9:30	125	47	67
10	131	47	69
10:30	135	49	72
11	140	52	76
11:30	145	54	77
12	150	55	78
12:30	154	57	80
13	158	58	84
13:30	162	59	89
14	166	62	90
14:30	170	64	91
15	174	65	99

(b)

Table 16: T at position 4(side face location 3)4R v/s NC**Table 17:** T at position 5 (top surface location 1) 5F v/s NC

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	30	30	30
0:30	30	30	30
1	35	30	30
1:30	36	30	30
2	41	31	30
2:30	46	31	32
3	55	32	36
3:30	62	34	38
4	63	35	40
4:30	72	36	42
5	79	37	43
5:30	86	39	46
6	82	40	48
6:30	94	42	50
7	102	43	53
7:30	108	44	56
8	113	45	58
8:30	120	46	62
9	126	47	64
9:30	128	48	68
10	136	49	69
10:30	138	50	74
11	142	53	78
11:30	145	56	80
12	152	57	81
12:30	156	58	82
13	159	60	84
13:30	163	61	90
14	165	62	91
14:30	171	65	94
15	175	66	100

(1)

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	29	30	30
0:30	29	30	30
1	34	31	30
1:30	49	31	30
2	68	31	31
2:30	80	31	32
3	107	31	35
3:30	115	31	39
4	132	31	41
4:30	141	32	49
5	153	33	54
5:30	164	34	58
6	170	35	64
6:30	180	36	69
7	189	37	76
7:30	201	38	81
8	209	39	86
8:30	211	40	93
9	218	41	96
9:30	222	42	100
10	226	46	104
10:30	232	47	108
11	236	48	111
11:30	239	50	119
12	242	51	124
12:30	244	52	128
13	247	54	134
13:30	250	59	137
14	254	61	142
14:30	258	63	146
15	257	64	147

(2)

Table 18:(a) T at position 5 (top surface location 2) 5M v/s NC, (b) T at position 5 (top surface location 3) 5R v/s NC

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	29	30	30
0:30	31	31	30
1	44	31	30
1:30	59	31	31
2	74	31	32
2:30	90	31	33
3	108	32	37
3:30	119	32	40
4	133	33	45
4:30	144	33	50
5	156	34	56
5:30	168	35	60
6	174	36	66
6:30	184	37	71
7	193	38	77
7:30	202	40	83
8	209	41	88
8:30	212	43	94
9	217	44	98
9:30	223	46	104
10	227	48	109
10:30	233	50	113
11	237	51	118
11:30	240	53	123
12	243	55	127
12:30	245	56	131
13	248	58	135
13:30	250	60	139
14	255	61	143
14:30	258	63	147
15	258	64	150

(a)

Temperature(⁰ C)			
Time (min)	Model		
	M1	M2	M3 b
0	29	30	30
0:30	31	30	30
1	44	31	30
1:30	44	31	30
2	73	31	30
2:30	85	31	32
3	106	31	36
3:30	118	33	39
4	132	34	42
4:30	143	35	50
5	155	36	56
5:30	167	37	61
6	173	38	66
6:30	183	38	70
7	192	39	77
7:30	201	40	84
8	208	41	90
8:30	211	43	95
9	215	45	99
9:30	220	47	104
10	226	48	109
10:30	231	51	114
11	236	52	119
11:30	240	54	124
12	242	55	129
12:30	243	56	132
13	246	58	137
13:30	250	61	141
14	253	62	145
14:30	256	63	148
15	256	65	153

(b)

Table 19: T at position inside compartment v/s forced convection.**Table 20:** T at position inside compartment v/s natural convection

Temperature($^{\circ}\text{C}$)			
Time (min)	Model		
	M1	M2	M3 a
0	30	30	30
0:30	36	57	56
1	55	92	78
1:30	78	117	99
2	91	145	117
2:30	104	166	138
3	116	185	154
3:30	129	203	167
4	143	219	178
4:30	153	230	188
5	164	241	198
5:30	174	249	207
6	185	255	214
6:30	194	261	223
7	201	267	231
7:30	208	271	237
8	215	276	244
8:30	220	281	248
9	225	278	253
9:30	229	285	258
10	235	288	261
10:30	239	290	266
11	243	292	270
11:30	247	294	273
12	250	296	274
12:30	253	300	278
13	256	301	280
13:30	260	303	283
14	262	306	285
14:30	265	308	287
15	270	309	288

(6)

Temperature($^{\circ}\text{C}$)			
Time (min)	Model		
	M1	M2	M3 b
0	30	30	30
0:30	36	57	60
1	55	92	109
1:30	78	117	165
2	91	145	156
2:30	104	166	177
3	116	185	194
3:30	129	203	209
4	143	219	219
4:30	153	230	230
5	164	241	242
5:30	174	249	252
6	185	255	260
6:30	194	261	267
7	201	267	275
7:30	208	271	283
8	215	276	289
8:30	220	281	294
9	225	278	299
9:30	229	285	303
10	235	288	307
10:30	239	290	311
11	243	292	315
11:30	247	294	319
12	250	296	323
12:30	253	300	326
13	256	301	328
13:30	260	303	331
14	262	306	333
14:30	265	308	336
15	270	309	338

(7)

Table 21: Air temperature change from inlet to out let with time in forced convection.**Table 22:** Air temperature change from inlet to out let with time natural convection.

Thermocouple Position	Temp(°C)		
	Inlet (7)	Out let(6)	ΔT
0	28	28	0
0:30	26	28	2
1	25	28	3
1:30	25	29	4
2	26	30	4
2:30	27	31	4
3	27	33	6
3:30	27	35	8
4	28	37	9
4:30	29	39	10
5	30	41	11
5:30	30	43	13
6	31	45	14
6:30	32	48	16
7	33	51	18
7:30	33	54	19
8	34	56	22
8:30	35	58	23
9	36	61	25
9:30	37	63	26
10	37	66	29
10:30	38	69	31
11	39	71	32
11:30	39	73	34
12	40	76	36
12:30	40	78	38
13	41	80	39
13:30	41	82	41
14	42	84	42
14:30	43	86	43
15	43	88	45

(8)

Thermocouple Position	Temp(°C)		
	Inlet (7)	Out let(6)	ΔT
0	30	30	0
0:30	30	30	0
1	30	31	1
1:30	30	33	3
2	31	37	6
2:30	31	41	10
3	31	42	11
3:30	32	41	9
4	33	42	9
4:30	33	42	9
5	34	44	10
5:30	35	47	12
6	37	52	15
6:30	41	56	15
7	42	55	13
7:30	44	66	22
8	46	65	19
8:30	49	65	16
9	51	66	15
9:30	50	67	17
10	55	71	16
10:30	56	80	24
11	57	95	38
11:30	61	102	41
12	65	108	43
12:30	66	101	35
13	67	96	29
13:30	67	98	31
14	68	100	32
14:30	72	100	28
15	77	98	21

(9)

APPENDIX III

Properties

Table 23 : Properties of air in relation to temperature

T (K)	ρ (kg/m ³)	c_p (kJ/kg · K)	$\mu \cdot 10^7$ (N · s/m ²)	$\nu \cdot 10^6$ (m ² /s)	$k \cdot 10^3$ (W/m · K)	$\alpha \cdot 10^6$ (m ² /s)	Pr
Air, $M = 28.97$ kg/kmol							
100	3.5562	1.032	71.1	2.00	9.34	2.54	0.786
150	2.3364	1.012	103.4	4.426	13.8	5.84	0.758
200	1.7458	1.007	132.5	7.590	18.1	10.3	0.737
250	1.3947	1.006	159.6	11.44	22.3	15.9	0.720
300	1.1614	1.007	184.6	15.89	26.3	22.5	0.707
350	0.9950	1.009	208.2	20.92	30.0	29.9	0.700
400	0.8711	1.014	230.1	26.41	33.8	38.3	0.690
450	0.7740	1.021	250.7	32.39	37.3	47.2	0.686
500	0.6964	1.030	270.1	38.79	40.7	56.7	0.684
550	0.6329	1.040	288.4	45.57	43.9	66.7	0.683
600	0.5804	1.051	305.8	52.69	46.9	76.9	0.685
650	0.5356	1.063	322.5	60.21	49.7	87.3	0.690
700	0.4975	1.075	338.8	68.10	52.4	98.0	0.695
750	0.4643	1.087	354.6	76.37	54.9	109	0.702
800	0.4354	1.099	369.8	84.93	57.3	120	0.709
850	0.4097	1.110	384.3	93.80	59.6	131	0.716
900	0.3868	1.121	398.1	102.9	62.0	143	0.720
950	0.3666	1.131	411.3	112.2	64.3	155	0.723
1000	0.3482	1.141	424.4	121.9	66.7	168	0.726
1100	0.3166	1.159	449.0	141.8	71.5	195	0.728
1200	0.2902	1.175	473.0	162.9	76.3	224	0.728
1300	0.2679	1.189	496.0	185.1	82	257	0.719
1400	0.2488	1.207	530	213	91	303	0.703
1500	0.2322	1.230	557	240	100	350	0.685
1600	0.2177	1.248	584	268	106	390	0.688
1700	0.2049	1.267	611	298	113	435	0.685
1800	0.1935	1.286	637	329	120	482	0.683
1900	0.1833	1.307	663	362	128	534	0.677
2000	0.1741	1.337	689	396	137	589	0.672
2100	0.1658	1.372	715	431	147	646	0.667
2200	0.1582	1.417	740	468	160	714	0.655
2300	0.1513	1.478	766	506	175	783	0.647
2400	0.1448	1.558	792	547	196	869	0.630
2500	0.1389	1.665	818	589	222	960	0.613
3000	0.1135	2.726	955	841	486	1570	0.536

Adopted from Incropera

Table 24: Material properties of different materials

<i>Other Materials (Continued)</i>				
Description/ Composition	Temperature (K)	Density, ρ (kg/m ³)	Thermal Conductivity, k (W/m · K)	Specific Heat, c_p (J/kg · K)
Ice	273	920	1.88	2040
	253	—	2.03	1945
Leather (sole)	300	998	0.159	—
Paper	300	930	0.180	1340
Paraffin	300	900	0.240	2890
Rock				
Granite, Barre	300	2630	2.79	775
Limestone, Salem	300	2320	2.15	810
Marble, Halston	300	2680	2.80	830
Quartzite, Sioux	300	2640	5.38	1105
Sandstone, Berea	300	2150	2.90	745
Rubber, vulcanized				
Soft	300	1100	0.13	2010
Hard	300	1190	0.16	—
Sand	300	1515	0.27	800
Soil	300	2050	0.52	1840
Snow	273	110	0.049	—
		500	0.190	—
Teflon	300	2200	0.35	—
	400		0.45	—
Tissue, human				
Skin	300	—	0.37	—
Fat layer (adipose)	300	—	0.2	—
Muscle	300	—	0.5	—
Wood, cross grain				
Balsa	300	140	0.055	—
Cypress	300	465	0.097	—
Fir	300	415	0.11	2720
Oak	300	545	0.17	2385
Yellow pine	300	640	0.15	2805
White pine	300	435	0.11	—
Wood, radial				
Oak	300	545	0.19	2385
Fir	300	420	0.14	2720

Adopted from Incropera

Table 25:Insulation material properties

<i>Insulating Materials and Systems</i>			
Description/Composition	Typical Properties at 300 K		
	Density, ρ (kg/m ³)	Thermal Conductivity, k (W/m · K)	Specific Heat, c_p (J/kg · K)
Blanket and Batt			
Glass fiber, paper faced	16	0.046	—
	28	0.038	—
	40	0.035	—
Glass fiber, coated; duct liner	32	0.038	835
Board and Slab			
Cellular glass	145	0.058	1000
Glass fiber, organic bonded	105	0.036	795
Polystyrene, expanded			
Extruded (R-12)	55	0.027	1210
Molded beads	16	0.040	1210
Mineral fiberboard; roofing material	265	0.049	—
Wood, shredded/cemented	350	0.087	1590
Cork	120	0.039	1800
Loose Fill			
Cork, granulated	160	0.045	—
Diatomaceous silica, coarse	350	0.069	—
Powder	400	0.091	—
Diatomaceous silica, fine powder	200	0.052	—
	275	0.061	—
Glass fiber, poured or blown	16	0.043	835
Vermiculite, flakes	80	0.068	835
	160	0.063	1000
Formed/Foamed-in-Place			
Mineral wool granules with asbestos/inorganic binders, sprayed	190	0.046	—
Polyvinyl acetate cork mastic; sprayed or troweled	—	0.100	—
Urethane, two-part mixture; rigid foam	70	0.026	1045
Reflective			
Aluminum foil separating fluffy glass mats; 10–12 layers, evacuated; for cryogenic applications (150 K)	40	0.00016	—
Aluminum foil and glass paper laminate; 75–150 layers; evacuated; for cryogenic application (150 K)	120	0.000017	—
Typical silica powder, evacuated	160	0.0017	—

Adopted from Incropera

Table 26: Properties of various materials at varying temp

Composition	Melting Point (K)	Properties at Various Temperatures (K)													
		Properties at 300 K				k (W/m · K)/ c_p (J/kg · K)									
		ρ (kg/m ³)	c_p (J/kg · K)	k (W/m · K)	$\alpha \cdot 10^6$ (m ² /s)	100	200	400	600	800	1000	1200	1500	2000	2500
Stainless steels															
AISI 302		8055	480	15.1	3.91			17.3	20.0	22.8	25.4				
								512	559	585	606				
AISI 304	1670	7900	477	14.9	3.95	9.2	12.6	16.6	19.8	22.6	25.4	28.0	31.7		
						272	402	515	557	582	611	640	682		
AISI 316		8238	468	13.4	3.48			15.2	18.3	21.3	24.2				
								504	550	576	602				
AISI 347		7978	480	14.2	3.71			15.8	18.9	21.9	24.7				
								513	559	585	606				
Lead	601	11340	129	35.3	24.1	39.7	36.7	34.0	31.4						
						118	125	132	142						
Magnesium	923	1740	1024	156	87.6	169	159	153	149	146					
						649	934	1074	1170	1267					
Molybdenum	2894	10240	251	138	53.7	179	143	134	126	118	112	105	98	90	86
						141	224	261	275	285	295	308	330	380	459
Nickel															
Pure	1728	8900	444	90.7	23.0	164	107	80.2	65.6	67.6	71.8	76.2	82.6		
						232	383	485	592	530	562	594	616		
Nichrome (80% Ni, 20% Cr)	1672	8400	420	12	3.4			14	16	21					
								480	525	545					
Inconel X-750 (73% Ni, 15% Cr, 6.7% Fe)	1665	8510	439	11.7	3.1	8.7	10.3	13.5	17.0	20.5	24.0	27.6	33.0		
						—	372	473	510	546	626	—	—		
Niobium	2741	8570	265	53.7	23.6	55.2	52.6	55.2	58.2	61.3	64.4	67.5	72.1	79.1	
						188	249	274	283	292	301	310	324	347	
Palladium	1827	12020	244	71.8	24.5	76.5	71.6	73.6	79.7	86.9	94.2	102	110		
						168	227	251	261	271	281	291	307		
Platinum															
Pure	2045	21450	133	71.6	25.1	77.5	72.6	71.8	73.2	75.6	78.7	82.6	89.5	99.4	
						100	125	136	141	146	152	157	165	179	
Alloy 60Pt-40Rh (60% Pt, 40% Rh)	1800	16630	162	47	17.4			52	59	65	69	73	76		
								—	—	—	—	—	—		
Rhenium	3453	21100	136	47.9	16.7	58.9	51.0	46.1	44.2	44.1	44.6	45.7	47.8	51.9	
						97	127	139	145	151	156	162	171	186	
Rhodium	2236	12450	243	150	49.6	186	154	146	136	127	121	116	110	112	
						147	220	253	274	293	311	327	349	376	
Silicon	1685	2330	712	148	89.2	884	264	98.9	61.9	42.2	31.2	25.7	22.7		
						259	556	790	867	913	946	967	992		
Silver	1235	10500	235	429	174	444	430	425	412	396	379	361			
						187	225	239	250	262	277	292			
Tantalum	3269	16600	140	57.5	24.7	59.2	57.5	57.8	58.6	59.4	60.2	61.0	62.2	64.1	65.6
						110	133	144	146	149	152	155	160	172	189
Thorium	2023	11700	118	54.0	39.1	59.8	54.6	54.5	55.8	56.9	56.9	58.7			
						99	112	124	134	145	156	167			
Tin	505	7310	227	66.6	40.1	85.2	73.3	62.2							

Adopted from Incropera

Composition	Melting Point (K)	Properties at Various Temperatures (K)													
		Properties at 300 K				k (W/m · K) / c_p (J/kg · K)									
		ρ (kg/m ³)	c_p (J/kg · K)	k (W/m · K)	$\alpha \cdot 10^6$ (m ² /s)	100	200	400	600	800	1000	1200	1500	2000	2500
Aluminum															
Pure	933	2702	903	237	97.1	302	237	240	231	218					
						482	798	949	1033	1146					
Alloy 2024-T6 (4.5% Cu, 1.5% Mg, 0.6% Mn)	775	2770	875	177	73.0	65	163	186	186						
Alloy 195, Cast (4.5% Cu)		2790	883	168	68.2	473	787	925	1042						
Beryllium	1550	1850	1825	200	59.2			174	185						
						990	301	161	126	106	90.8	78.7			
						203	1114	2191	2604	2823	3018	3227	3519		
Bismuth	545	9780	122	7.86	6.59	16.5	9.69	7.04							
						112	120	127							
Boron	2573	2500	1107	27.0	9.76	190	55.5	16.8	10.6	9.60	9.85				
						128	600	1463	1892	2160	2338				
Cadmium	594	8650	231	96.8	48.4	203	99.3	94.7							
						198	222	242							
Chromium	2118	7160	449	93.7	29.1	159	111	90.9	80.7	71.3	65.4	61.9	57.2	49.4	
						192	384	484	542	581	616	682	779	937	
Cobalt	1769	8862	421	99.2	26.6	167	122	85.4	67.4	58.2	52.1	49.3	42.5		
						236	379	450	503	550	628	733	674		
Copper															
Pure	1358	8933	385	401	117	482	413	393	379	366	352	339			
						252	356	397	417	433	451	480			
Commercial bronze (90% Cu, 10% Al)	1293	8800	420	52	14		42	52	59						
Phosphor gear bronze (89% Cu, 11% Sn)	1104	8780	355	54	17		785	460	545						
							41	65	74						
Cartridge brass (70% Cu, 30% Zn)	1188	8530	380	110	33.9	75	95	137	149						
							360	395	425						
Constantan (55% Cu, 45% Ni)	1493	8920	384	23	6.71	17	19								
						237	362								
Germanium	1211	5360	322	59.9	34.7	232	96.8	43.2	27.3	19.8	17.4	17.4			
						190	290	337	348	357	375	395			

Cross Section	$\frac{b}{a}$	$Nu_D = \frac{hD_h}{k}$		
		(Uniform q_s'')	(Uniform T_s)	Re_{D_h}
	—	4.36	3.66	64
	1.0	3.61	2.98	57
	1.43	3.73	3.08	59
	2.0	4.12	3.39	62
	3.0	4.79	3.96	69
	4.0	5.33	4.44	73
	8.0	6.49	5.60	82
	∞	8.23	7.54	96
	∞	5.39	4.86	96
	—	3.11	2.49	53

Adopted from Incropera

Table 27: Units and selected decimal multiples and submultiples

1	2	3
Quantity	Units and selected decimal multiples and submultiples	Unit symbol
Customary temperature	Degree Celsius	°C *
Temperature interval	Degree Celsius	°C *
Heat	Joule Kilojoule	J kJ
Heat flow	Watt Kilowatt	W KW
Intensity of heat flow rate and radiation	Watt per square meter	W/m ²
Thermal conductivity	Watt per meter degree Celsius	W/m °C *
Coefficient of heat transfer, thermal conductance of material of given thickness and of air spaces	Watt per square meter degree Celsius	W/m ² °C *
Thermal resistivity	meter degree Celsius per watt	m deg °C/W *
Specific heat and thermal capacity	kilojoule per kilogram degree Celsius kilojoule per cubic meter degree Celsius	kJ / kg °C * kJ/m ³ °C *
Calorific value	kilojoule per kilogram kilojoule per cubic meter	kJ/kg kJ/m ³
Refrigeration	Watt	W
Gas velocity	meter per second	m/s
Gas delivery rate	Cubic meter per second	m ³ /s

Table 28: Natural Convection Heat Transfer Coefficients

I. General relationships		$Nu = c(Gr Pr)^n$
		$h = c \frac{k}{L} \left(\frac{L^3 \rho^2 \beta g \Delta T}{\mu^2} \right)^n \left(\frac{\mu_c \rho}{k} \right)_f^n$
Characteristic length L		
Vertical plates or pipes		$L = \text{height}$
Horizontal plates		$L = \text{length}$
Horizontal pipes		$L = \text{diameter}$
Spheres		$L = 0.5 \times \text{diameter}$
Rectangular block, with horizontal length L_h and vertical length L_v		$1/L = (1/L_h) + (1/L_v)$
II. Planes and pipes		
Horizontal or vertical planes, pipes, rectangular blocks, and spheres (excluding horizontal plates facing downward for heating and upward for cooling)		
(a) Laminar range, when $Gr Pr$ is between 10^4 and 10^8		$Nu = 0.56(Gr Pr)^{0.25}$
(b) Turbulent range, when $Gr Pr$ is between 10^8 and 10^{12}		$Nu = 0.13(Gr Pr)^{0.33}$
III. Wires		
For horizontal or vertical wires, use $L = \text{diameter}$, for $Gr Pr$ between 10^{-7} and 1		$Nu = (Gr Pr)^{0.1}$
IV. With air		
$Gr Pr = 1.6 \times 10^6 L^3 \Delta T$ (at 21°C , L in m, ΔT in K)		
(a) Horizontal cylinders		
Small cylinder, laminar range		$h = 1.32(\Delta T/L)^{0.25}$
Large cylinder, turbulent range		$h = 1.24(\Delta T)^{0.33}$
(b) Vertical plates		
Small plates, laminar range		$h = 1.42(\Delta T/L)^{0.25}$
Large plates, turbulent range		$h = 1.31(\Delta T)^{0.33}$
(c) Horizontal plates, facing upward when heated or downward when cooled		
Small plates, laminar range		$h = 1.32(\Delta T/L)^{0.25}$
Large plates, turbulent range		$h = 1.52(\Delta T)^{0.33}$
(d) Horizontal plates, facing downward when heated or upward when cooled		
Small plates		$h = 0.59(\Delta T/L)^{0.25}$

*Adapted from ASHRAE 1997***Table 29:** Ideal-gas specific heats of air

Temperature, K	c_p kJ/kg·K	c_v kJ/kg·K	k
<i>Air</i>			
250	1.003	0.716	1.401
300	1.005	0.718	1.400
350	1.008	0.721	1.398
400	1.013	0.726	1.395
450	1.020	0.733	1.391
500	1.029	0.742	1.387
550	1.040	0.753	1.381
600	1.051	0.764	1.376
650	1.063	0.776	1.370
700	1.075	0.788	1.364
750	1.087	0.800	1.359
800	1.099	0.812	1.354
900	1.121	0.834	1.344
1000	1.142	0.855	1.336