© 2014 IJEDR | Volume 2, Issue 1 | ISSN: 2321-9939

The Load Calculation of Automobile Air Conditioning System

[1]Jignesh K. Vaghela, [2]Dr. Ragesh G. Kapadia
[1]M.E. Student, [2]Principal
[1]Thermal Engineering, SVMIT, Bharuch-392001, Gujarat, INDIA
[2]Principal, SVMIT, Bharuch-392001, Gujarat, INDIA
[1]jignesh.vaghela725@gmail.com, [2]rag260475@gmail.com

***Abstract*— The Load Calculation of Automobile Air Conditioning System is presented. From the load calculation, cooling capacity can be calculated & thus tonne of refrigeration required is found out. The Heat Balance Method (HBM) is used for estimating the heating and cooling loads encountered in a vehicle cabin. Mathematical models of heat transfer phenomena are used to calculate the different load categories. Mathematical load calculation models are devised and collected from various sources for load estimation. Case study of Wagon R car is introduced under arbitrary driving conditions. Simplified geometry & typical material properties of a Wagon R car are considered as input parameters for studies.**

In the case study of Wagon R Car, the engine, exhaust, and reflected radiation loads may be neglected from consideration. On the other hand, the direct and diffuse radiation loads are important AC loads that tend to give rise to the cabin temperature. The cabin temperature decreases from a soak temperature of 80°*C* to the comfort temperature after almost 10 minutes represented by the pull-down time. After the pull-down time, a steady-state situation is achieved where the loads are balanced and a zero net load is maintained for the cabin for the rest of the period. This research aims to provide a basis for estimating thermal loads in vehicle cabins. Cooling capacity i.e. tonne of refrigeration required is found out from this load calculation. The result of this study can be used by HVAC engineers to design more efficient car AC systems.

***Index Terms*— Load Calculation, Automobile, Automobile Air Conditioning System.**

I. INTRODUCTION

For estimating the heating and cooling loads encountered in a vehicle cabin, the Heat Balance Method (HBM) is used. Mathematical models of heat transfer phenomena are used to calculate the different load categories. Mathematical load calculation models are devised and collected from various for load estimation. Lumped-body assumptions are made[10].

Nomenclatures	
A - Apparent Solar Irradiation at Air Mass = 0 (W/m^2)	**Greek Letters**
A_{Du} - DuBois Body Surface Area (m^2)	α - Absorptivity
B - Atmospheric Extinction Coefficient	β - Altitude Angle (°)
c - Specific Heat (J/kg K)	Δt - Time Step Size (s)
C - Diffuse Radiation Factor	ΔT - Temperature Change (K)
DTM - Deep Thermal Mass (J/K)	ϕ - Relative Humidity (%)
e - Enthalpy (J/kg)	λ - Surface Element Thickness (m)
h - Convective Heat Transfer Coefficient (W/m^2K)	θ - Angle between Surface Normal and Sun Position (°)
H - Human Body Height (m)	ρ - Density ($kg/m3$)
$\dot{I}$ - Radiation Heat Gain per Unit Area (W/m^2)	ρ_g - Ground Reflectivity
k - Conductive Heat Transfer Coefficient (W/mK)	Σ - Surface Tilt Angle from Horizon (°)
m - Mass (kg)	τ – Transmissivity
$\dot{m}$ - Mass Flow Rate (kg/s)	**Subscripts**
M - Metabolic Rate (W/m^2)	a - Cabin Air
P - Air Pressure (Pa)	AC - Air Conditioning
P_s - Water Saturation Pressure (Pa)	Amb - Ambient
R - Surface Overall Heat Transfer Resistance (m^2K/W)	$comf$ - Comfort Condition
RPM - Engine Revolutions per Minute (1/min)	Dif - Diffuse Radiation
S - Cabin Surface Element Area (m^2)	Dir - Direct Radiation
t_c - Pull-Down Constant (s)	Eng - Engine
t_p - Pull-Down Time (s)	Exh - Exhaust
T - Temperature (K)	i - Inside
T_0 - Initial Cabin Temperature (K)	Met - Metabolic
U - Surface Overall Heat Transfer Coefficient (W/m^2K)	new - New Time Step
V - Vehicle Speed (m/s)	o - Outside
W - Human Body Weight (kg)	old - Old Time Step
X - Humidity Ratio (kg water/kg dry air)	Rad - Radiation

Theory of load calculation procedure used in Heat Balance Method is represented. Case study of Wagon R car is introduced under arbitrary driving conditions. Simplified geometry & typical material properties of a Wagon R car are considered as input parameters for studies.

Variation in Thermal loads with respect to time is represented. Also variation in Net cabin load & cabin temperature with respect to time is plotted in graph.

II. LITERATURE REVIEW

Efficient design of car air conditioning (CAC) has been the centre of attention of automotive manufacturers and academic researchers during the last few decades. Reduction of fuel consumption and tailpipe emission are two crucial targets for the auto industry to improve the efficiency[10].

- **Welstand, J., Haskew, H.**

Air conditioning operation is proven to have a significant impact on the emissions and fuel economy; *e.g.* AC usage can increase NO_x emission from 15% to 100% . Welstand *et al.* studied the effects of AC on vehicle emissions and fuel consumption.[1].

- **Lambert, M. A., and Jones, B. J.**

The AC power consumption of mid-size cars is estimated to be higher than 12% of the total vehicle power during regular commuting [2].

- **Johnson, V.**

Further, AC loads are the most significant auxiliary loads present in conventional ICE vehicles today; its energy use even outweighs the energy loss to rolling resistance, aerodynamic drag, or driveline losses for a typical vehicle. The U.S. alone consumes about 7 billion gallons of fuel a year for AC systems of light-duty vehicles. The AC load of a 1200-*kg* sedan, under peak conditions, can amount to 6 *kW*, which can deplete the vehicle's battery pack quickly [3].

Besides emission and fuel efficiency, passengers' comfort is another major factor that should be considered in the design of new vehicles. In fact, auto manufacturers pay a significant attention to driver and passenger comfort which is directly linked to AC system. This trend is evidenced by new features such as multi-zone climate control and heated/cooled seats that can be found even in recent compact vehicles. Based on these observations, it is of both environmental and economic interest to seek new methods to improve the efficiency and performance of AC systems of vehicle.

- **Fanger, P. O.**

A clear understanding of the heating and cooling loads, encountered by the passenger cabin, is a key prerequisite for an efficient design of any mobile AC system. The function of AC system is to compensate for the continuous changes of cabin loads in order to maintain the passenger comfort within a thermal 'comfort zone'. Fanger's model of thermal comfort has been extensively used in AC research and applications as the basis for comfort assessment [4].

- **Ingersoll, J.**

Based on Fanger's model, Ingersoll *et al.* developed a human thermal comfort calculation model specific to automobile passenger cabins. While performing thermal load calculations for a vehicle cabin, their model can be used to assess the corresponding status of thermal comfort [5].

- **ASHRAE Handbook of Fundamentals**

ASHRAE Handbook of Fundamentals provides two major thermal load calculation methodologies: Heat Balance Method (HBM) and Weighting Factor Method (WFM) [6].

- **Kamar, H.**

HBM is the most scientifically rigorous available method and can consider more details with less simplifying assumptions. An advantage of HBM is that several fundamental models can be incorporated in the thermal calculations. Although HBM is more accurate than WFM, it is easier to implement WFM for load calculation in a passenger vehicle. However, when more detailed information of the vehicle body and thermal loads is available, HBM is the preferred choice [7].

- **Zheng, Y., Mark, B., and Youmans, H.**

Zheng *et al.* devised a simple method to calculate vehicle's thermal loads. They calculated the different loads such as the radiation and ambient loads. A case study was performed and the results were validated using wind tunnel climate control tests. The different loads were separately calculated and summed up to give the total heat gain or loss from the cabin.

In this study, a simple method to calculate vehicle heat load is developed. The cooling load temperature differential (CLTD) method is used to calculate the heat gain of a sunlit roof and wall (door). This is done in one step by using ASHRAE data. The calculation presented here takes into account the geometrical configuration of the vehicle compartment including glazing surfaces (shading), windshield and roof angle, and vehicle orientation. Special attention is given to the calculation of direct and diffuse incidence solar radiation through the windshield and skylight glass. The vertical glass' solar heat gain is evaluated by using ASHRAE data. The U value method is used to calculate heat transfer between the outside and inside cabin. Heat gains from infiltration, occupant, and HVAC unit blower motors are considered in the cooling load calculation.

The method accuracy was validated using wind tunnel tests. The results showed the predicted cooling load is very close to the tested value, and the deviation between calculated and tested heat loads is smaller with fresh air mode than that with recirculation mode [8].

- **Ding, Y. and Zito, R.**

Ding and Zito also used a lumped model for the cabin and solved the corresponding transient heat transfer differential equation analytically. Their analytical solution can be used as a benchmark for basic problems like the cool-down test. This paper describes a first order differential equation that relates the cabin heat transfer coefficient, discharge panel temperature and discharge volumetric air flow to the average interior temperature. The solution to this equation leads to an overall understanding of automotive air conditioning designs and tests [9].

This research aims to provide a basis for estimating thermal loads in vehicle cabins. Although other load estimation models exist as mentioned above, the main emphasis of this study is to account for the dynamic changes to the AC loads that occur in real-world scenarios. The results of this study can be used by HVAC engineers to design more efficient mobile AC systems. A specifically calibrated model for any vehicle can be later implemented to provide accurate predictions of upcoming thermal load variations under various driving and environmental conditions. As such, the ultimate objective of this study is to create a platform for real-time control of the AC system, including AC compressor, heat exchanger fans, passenger(s) seat temperature and window glazing, to achieve superior fuel efficiency and passenger comfort.

III. Model Development & Load Calculation Procedure

We consider a lumped model of a typical vehicle cabin [10]. The net heat gain by the cabin can be classified under nine different categories. The total load as well as each of these loads can either be positive (heating up the cabin) or negative (cooling down the cabin) and may depend on various driving parameters. In the following, the models developed for each of these load categories are presented and discussed. Some of the correlations used in the present model are based on experiments performed on certain vehicles, which are used here for general validation of the model. New correlations can be readily plugged into the present model can be tailored to any new vehicle, after specifying those correlations for the case. The summation of all the load types will be the instantaneous cabin overall heat load gain. The mathematical formulation of the model can thus be summarized as

$$\dot{Q}_{Tot} = \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} + \dot{Q}_{AC} \quad (1)$$

All of the above $\dot{Q}$ values are thermal energies per unit time. $\dot{Q}_{Tot}$ is the net overall thermal load encountered by the cabin. $\dot{Q}_{Met}$ is the metabolic load. $\dot{Q}_{Dir}$, $\dot{Q}_{Dif}$, and $\dot{Q}_{Ref}$ are the direct, diffuse, and reflected radiation loads, respectively. $\dot{Q}_{Amb}$ is the ambient load. $\dot{Q}_{Exh}$ and $\dot{Q}_{Eng}$ are the exhaust and engine loads due to the high temperature of the exhaust gases and the engine. Finally, the term $\dot{Q}_{Ven}$ is the load generated due to ventilation, and $\dot{Q}_{AC}$ is the thermal load created by the AC cycle.

Figure 1 schematically shows the various thermal load categories encountered in a typical vehicle cabin. Some of the above loads pass across the vehicle body plates/parts, while others are independent of the surface elements of the cabin.

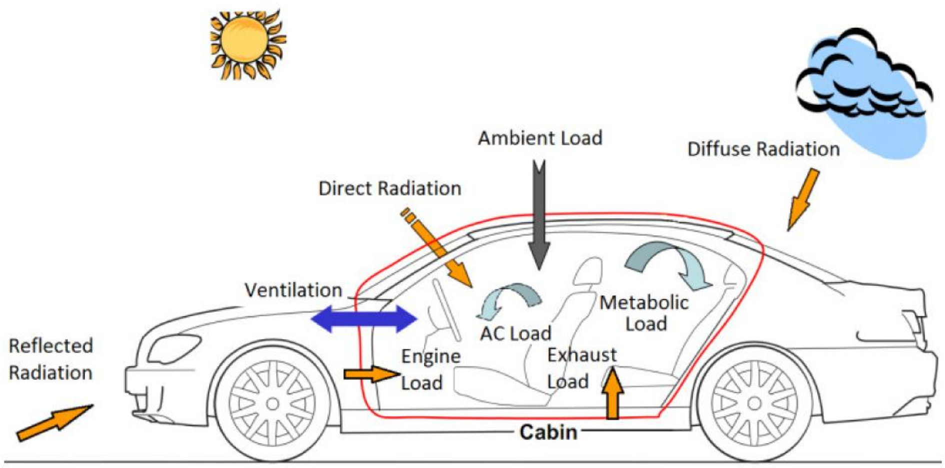

Figure 1. Schematic representation of thermal loads in a typical vehicle cabin [10].

Each thermal load is calculated assuming a quasi-steady-state condition. Load calculations are performed at time steps during the period of interest, and after every time step, all the load components are algebraically summed up and the new cabin air temperature and surface element temperatures are calculated as

$$\Delta T_i = \frac{\dot{Q}_{Tot}}{m_a c_a + DTM} \Delta t$$

$$\Delta T_s = \frac{\dot{Q}_s}{m_s c_s} \Delta t \quad (2)$$

Where T_i and T_s are the change in the cabin and surface element temperatures at the current time step. DTM is the sum of all the deep thermal masses i.e. the overall thermal inertia of all objects other than air present inside the cabin. These objects include the seat structures, the dash, the dash components, etc. which are combined with the cabin air in the lumped model. t is the time step, m_a is the cabin air mass and ca is the air specific heat. m_s and c_s are the mass and specific heat of each of the surface elements and $\dot{Q}_s = \dot{Q}_{s,Rad} + \dot{Q}_{s,Amb}$ is the net heat gain by a surface element consisting of the heat gain by radiation, $\dot{Q}_{s,Rad}$, and the heat gain from ambient, $\dot{Q}_{s,Amb}$.

- **Metabolic Load**

The metabolic activities inside human body constantly create heat and humidity (*i.e.* perspiration). This heat passes through the body tissues and is finally released to the cabin air. This amount is considered as a heat gain by the cabin air and is called the metabolic load. The metabolic load can be calculated by

$$\dot{Q}_{Met} = \sum_{Passengers} MA_{Du} \quad (3)$$

Where M is the passenger metabolic heat production rate. It is found from the tabulated values in ISO 8996 [11] based on various criteria such as occupation and activity levels. For a driver and a sitting passenger, the values can be estimated as 85 W/m^2 and 55 W/m^2, respectively. The Dubois area A_{Du}, which is an estimation of the body surface area as a function of height and weight, is calculated by [11]

$$A_{Du} = 0.202 W^{0.425} H^{0.725} \quad (4)$$

where W and H are the passenger weight and height respectively.

- **Radiation Load**

The heat gain due to solar radiation is a significant part of the cooling loads encountered in vehicles. According to ASHRAE [6], solar radiation heat load can be categorized into direct, diffuse, and reflected radiation loads. Direct radiation is that part of the incident solar radiation which directly strikes a surface of the vehicle body, which is calculated from

$$\dot{Q}_{Dir} = \sum_{Surfaces} S\tau \dot{i}_{Dir} \cos\theta \quad (5)$$

Where I_{Dir} is the direct radiation heat gain per unit area and θ is the angle between the surface normal and the position of sun in the sky. τ is the surface element transmissivity and S is the surface area, respectively. Before local sunrise and after local sunset, simply no radiation loads are considered. The direct radiation heat gain per unit area is found by

$$\dot{i}_{Dir} = \frac{A}{\exp\left(\frac{B}{\sin\beta}\right)} \quad (6)$$

where A and B are constants tabulated in ASHRAE Handbook of Fundamentals [6] for different months. β is the altitude angle that is calculated based on position and time. Diffuse radiation is the part of solar radiation which results from indirect radiation of daylight on the surface. During a cloudy day, most of the solar radiation is received from this diffuse radiation. The diffuse radiation heat gain is found by

$$\dot{Q}_{Dif} = \sum_{Surfaces} S\tau \dot{i}_{Dif} \quad (7)$$

Similarly, $\dot{i}_{Dif}$ is the diffuse radiation heat gain per unit area which is calculated from

$$\dot{i}_{Dif} = C\dot{i}_{Dir} \frac{1+\cos\Sigma}{2} \quad (8)$$

where Σ is the surface tilt angle measured from the horizontal surface and the values for C are tabulated in [6]. Reflected radiation refers to the part of radiation heat gain that is reflected from the ground and strikes the body surfaces of the vehicle. The reflected radiation is calculated by

$$\dot{Q}_{Ref} = \sum_{Surfaces} S\tau \dot{i}_{Ref} \quad (9)$$

$\dot{i}_{Ref}$ is the reflected radiation heat gain per unit area, is calculated from

$$\dot{i}_{Ref} = \left(\dot{i}_{Dir} + \dot{i}_{Dif}\right) \rho_g \frac{1-\cos\Sigma}{2} \quad (10)$$

where ρ_g is the ground reflectivity coefficient. Based on the absorptivity of each particular surface element, a percentage of the incident radiation load can be absorbed by that surface, hence increasing its temperature. The net absorbed heat of each surface element due to radiation can thus be written as

$$\dot{Q}_{s,Rad} = S\alpha\left(\dot{i}_{Dir}\cos\theta + \dot{i}_{Dif} + \dot{i}_{Ref}\right) \quad (11)$$

Where α is the surface absorptivity.

- **Ambient Load**

The ambient load is the contribution of the thermal load transferred to the cabin air due to temperature difference between the ambient and cabin air. Exterior convection, conduction through body panels, and interior convection are involved in the total heat transfer between the ambient and the cabin. Equation (12) shows the general form of the ambient load model.

$$\dot{Q}_{Amb} = \sum_{Surfaces} SU\left(T_s - T_i\right) \quad (12)$$

where U is the overall heat transfer coefficient of the surface element. T_s and T_i are the average surface temperature and average cabin temperature, respectively. U has different components consisting of the inside convection, conduction through the surface, and outside convection. It can be written in the form

$$U = \frac{1}{R} \quad \text{where} \quad R = \frac{1}{h_o} + \frac{\lambda}{k} + \frac{1}{h_i} \qquad (13)$$

where R is the net thermal resistance for a unit surface area. h_o and h_i are the outside and inside convection coefficients, k is the surface thermal conductivity, and λ is the thickness of the surface element. The thermal conductivity and thickness of the vehicle surface can be measured rather easily. The convection coefficients h_o and h_i depend on the orientation of the surface and the air velocity. Here, the following estimation is used to estimate the convection heat transfer coefficients as a function of vehicle speed [12]

$$h = 0.6 + 6.64\sqrt{V} \qquad (14)$$

where h is the convection heat transfer coefficient in W/m^2K and V is the vehicle speed in m/s. Despite its simplicity, this correlation is applicable in all practical automotive instances [12]. The cabin air is assumed stationary and the ambient air velocity is considered equal to the vehicle velocity.

Numerical simulations can also be used to provide the model with convection coefficients that have higher accuracy and take into account the different orientation and position of every surface component. Similar to the radiation load above, a portion of the ambient load across the body surface is absorbed by the body plate material. The heat gain or loss of each surface element is the difference between the heat gained from the ambient by the surface, and the heat released to the cabin by the surface. Thus we can write the net absorbed heat as

$$\begin{aligned} \dot{Q}_{s,Amb} &= SU(T_o - T_s) - SU(T_s - T_i) \\ &= SU(T_o - 2T_s + T_i) \end{aligned} \qquad (15)$$

where T_o, T_i, and T_s are the ambient, cabin, and surface average temperatures, respectively.

- **Exhaust Load**

Conventional and hybrid electric vehicles have an Internal Combustion Engine (ICE) that creates exhaust gases. The Exhaust Gas Temperature (EGT) can reach as high as 1000°C [13]. Because of the high temperature of the exhaust gas, some of its heat can be transferred to the cabin through the cabin floor. Considering S_{Exh} as the area of the bottom surface in contact with the exhaust pipe, the exhaust heat load entering the cabin can be written as

$$\dot{Q}_{Exh} = S_{Exh} U (T_{Exh} - T_i) \qquad (16)$$

where U is the overall heat transfer coefficient of the surface element in contact with the exhaust pipe and it should be calculated by Eq. (13) assuming no external convection since the exhaust temperature is measured at the outer side of the bottom surface. S_{Exh} is the surface area exposed to the exhaust pipe temperature and T_{Exh} is the exhaust gas temperature. The temperature of the exhaust gases in Celsius degrees is estimated by [14]

$$T_{Exh} = 0.138RPM - 17 \qquad (17)$$

where RPM is the engine speed in revolutions per minute.

- **Engine Load**

Similar to the exhaust load above, the high temperature engine of a conventional or hybrid car can also contribute to the thermal gain of the cabin. Equation (18) shows the formulation used for calculating the engine thermal load.

$$\dot{Q}_{Eng} = S_{Eng} U (T_{Eng} - T_i) \qquad (18)$$

where U is the overall heat transfer coefficient of the surface element in contact with the engine and S_{Eng} is the surface area exposed to the engine temperature. The overall heat transfer coefficient can be calculated by Eq. (13) assuming no external convection, since the engine temperature is measured at the outside of the front surface. T_{Eng} is the engine temperature and is estimated in Celsius degrees by [14]

$$T_{Eng} = -2\times10^{-6} RPM^2 + 0.0355RPM + 77.5 \qquad (19)$$

- **Ventilation Load**

Fresh air is allowed to enter the vehicle cabin to maintain the air quality for passengers. As the passengers breathe, the amount of CO_2 concentration linearly increases over time. Thus, a minimum flow of fresh air should be supplied into the cabin to maintain the passengers comfort. Arndt and Sauer [15] reported the minimum fresh air requirements for different numbers of passengers in a typical vehicle. For instance, a minimum of 13% fresh air is needed for a single passenger.

On the other hand, Fletcher and Saunders [16] reported the air leakage from different vehicle types. They showed that for typical vehicles, leakage occurs as a function of the pressure difference between the cabin and the surroundings as well as the vehicle velocity. For a small sedan car at a pressure difference of 10 Pa, a leakage of 0.02 m^3/s was reported [16]. Because of the air conditioning and ventilation, the cabin pressure is normally slightly higher than the ambient. Thus, the ventilation load has to take the leakage air flow rate into account. Meanwhile, in the steady-state operation, the built-up pressure is assumed to remain constant. Hence, ambient air is assumed to enter the cabin at the ambient temperature and relative humidity, and the same flow rate is assumed to leave the cabin at the cabin temperature and relative humidity.

According to psychrometric calculations, ventilation heat gain consists of both sensible and latent loads. To account for both these terms, assuming a known flow rate of fresh air entering the cabin, the amount of thermal heat gain can be calculated from

$$\dot{Q}_{Ven} = \dot{m}_{Ven}\left(e_o - e_i\right) \quad (20)$$

Where m_{ven} is the ventilation mass flow rate and e_o and e_i are the ambient and cabin enthalpies, respectively. Enthalpies are calculated from [17]

$$e = 1006T + \left(2.501 \times 10^6 + 1770T\right)X \quad (21)$$

Where T is air temperature and X is humidity ratio in gram of water per gram of dry air. Humidity ratio is calculated as a function of relative humidity by

$$X = 0.62198\frac{\phi P_s}{100P - \phi P_s} \quad (22)$$

Where ϕ is relative humidity, P is air pressure, and P_s is the water saturation pressure at temperature T.

- **AC Load**

The duty of the air conditioning system is to compensate for other thermal loads so that the cabin temperature remains within the acceptable comfort range. In cold weather conditions, positive AC load (heating) is required for the cabin. Inversely, in warm conditions, negative AC load (cooling) is needed for maintaining the comfort conditions. The actual load created by the AC system depends on the system parameters and working conditions. In this work, it is assumed that an AC (or heat pump) cycle is providing the thermal load calculated by

$$\dot{Q}_{AC} = -\begin{pmatrix} \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \\ \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} \end{pmatrix} - \left(m_a c_a + DTM\right)\left(T_i - T_{comf}\right)/t_c \quad (23)$$

T_{comf} is the target comfort temperature as described and widely used by ASHARE standards [6]. This is the target bulk cabin temperature which is assumed comfortable at the conditions under consideration. t_c is a pull-down constant which determines the overall pull-down time. Pull-down time is defined as the time required for the cabin temperature to reach the comfort temperature within 1 K. Using Eq. (23) for the air conditioning load, the pull-down constant can be calculated from

$$t_c = \frac{t_p}{\ln\left|T_0 - T_{comf}\right|} \quad (24)$$

where T_0 is the initial cabin temperature.

Of course, the actual AC load depends on the system sizing and design. For a given system, the load may change depending on the compressor and fan speed as well. The actual power consumption of the cycle should be estimated by considering a suitable Coefficient of Performance (COP) for the vapour compression cycle. Equation (23) is used in this study as a guideline for analyzing the performance of an AC system in a typical vehicle. It shows that analyzing different scenarios with the AC cycle can help efficient sizing and control of the air conditioning cycle [10].

IV. CASE STUDY

Figure 2 : Wagon R car[18]

PROBLEM :

Wagon R car has the following specification values [10] :

MONTH :	JULY
TIME :	13:00 TO 16:00
LOCATION :	Bharuch , GUJARAT
LATITUDE :	21.7º N , 72.9ºE
DRIVER HEIGHT :	1.7 m
DRIVER WEIGHT :	70 kg
PASSENGER HEIGHT :	1.6 m
PASSENGER WEIGHT :	55 kg
VENTILATION FLOW :	0.1 m^3/sec
GROUND REFLECTIVITY :	0.02
AMBIENT TEMPERATURE :	35°C
INITIAL CABIN TEMPERATURE :	80°C
AMBIENT RELATIVE HUMIDITY :	70%
CABIN RELATIVE HUMIDITY :	50%
COMFORT TEMPERATURE :	23°C
PULL DOWN TIME :	10 min (600 sec)
DEEP THERMAL MASS :	5600 J/K

A. Perform the load analysis & Calculate the Total Thermal Load.

B. Estimate the cooling capacity of AC system.

C. Also plot the graph of (1) Thermal load v/s Time.

(2) Net cabin load v/s Time v/s Cabin temperature.

ANSWER :

CABIN GEOMETRY & MATERIAL PROPERTIES :

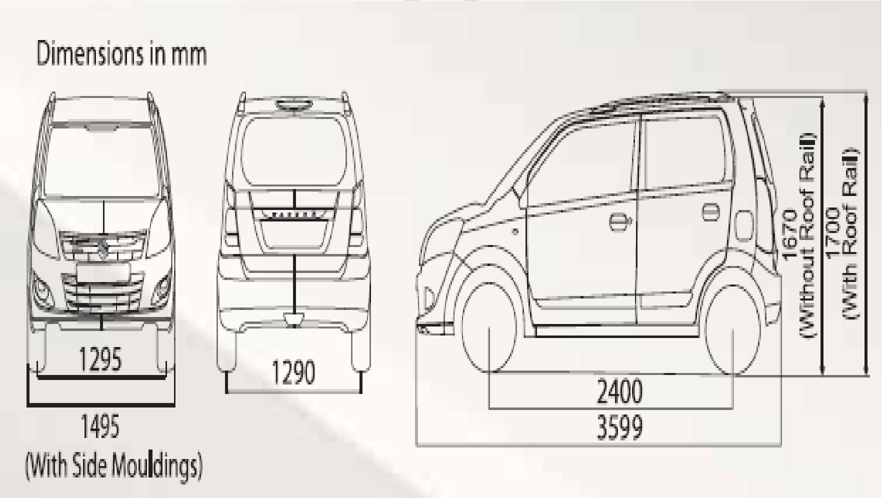

Figure : 3 : Front view, Rear view & Side view of Wagon R car**[18]**

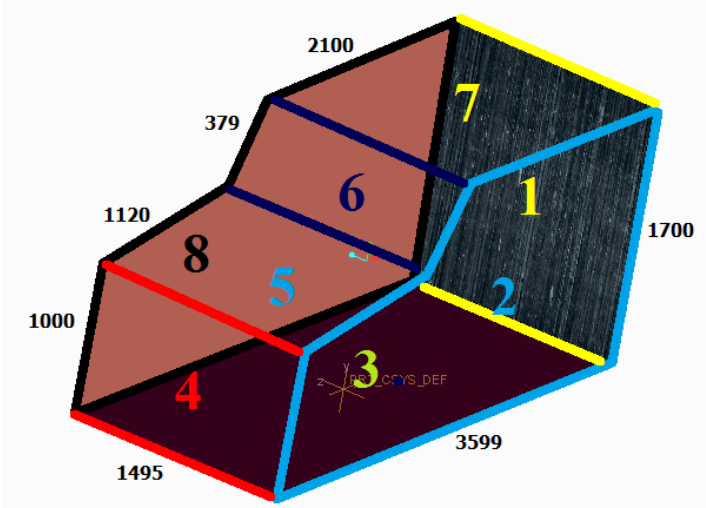

Figure 4 : Cabin Geometry

Property	Glass	Vehicle Body
Conductivity (W/mK)	1.05	0.2
Density (kg/m^3)	2500	1500
Transmissivity	0.5	0
Absorptivity	0.3	0.4
Specific Heat (J/kgK)	840	1000
Thickness (mm)	3	10

Table 1 : Material properties **[10]**

First lets defining the **Pull Down Time** :

Pull-down time is defined as the time required for the cabin temperature to reach the comfort temperature within 1 K.
Pull down time = 10 min (600 sec) .... (given)
Case A : During the pull down time (within first 10 minits)

(1) Metabolic Load :

DuBois area A_{Du} is :
$A_{Du} = 0.202W^{0.425}H^{0.725}$

For Driver :
W = 70 kg
H = 1.7 m

$A_{Du} = 0.202W^{0.425}H^{0.725}$
$= 0.202(70)^{0.425}(1.7)^{0.725}$
$= 1.80\ m^2$

$Q_{M1} = M * A_{Du}$
$= 85*1.80$ (for Driver , M = 85 W/m^2)
$= 153$ W

For Passenger :
W = 55 kg
H = 1.6 m

$A_{Du} = 0.202W^{0.425}H^{0.725}$
$= 0.202(55)^{0.425}(1.7)^{0.725}$
$= 1.55\ m^2$

$Q_{M2} = M * A_{Du}$
$= 55*1.55$ (for Passenger , $M = 55\ W/m^2$)
$= 85.7\ W$

$Q_M = Q_{M1} + Q_{M2}$
$= 153+85.7$
$=238.7\ W$
$\approx 240\ W$

(2) Radiation Load

(a) Direct Radiation load :

$$\dot{Q}_{Dir} = \sum_{Surfaces} S\tau \dot{I}_{Dir} \cos\theta$$

$=1000\ W$

(b) Diffuse Radiation load :

$$\dot{Q}_{Dif} = \sum_{Surfaces} S\tau \dot{I}_{Dif}$$

$= 190\ W$

(c) Reflected radiation load :

$$\dot{Q}_{Ref} = \sum_{Surfaces} S\tau \dot{I}_{Ref}$$

$= 0\ W$

(3) Ambient Load

$$\dot{Q}_{Amb} = \sum_{Surfaces} SU\left(T_s - T_i\right)$$

$= -200\ W$

(4) **Exhaust Load**

$$\dot{Q}_{Exh} = S_{Exh} U\left(T_{Exh} - T_i\right)$$

$= 0\ W$

(5) Engine Load

$$\dot{Q}_{Eng} = S_{Eng} U\left(T_{Eng} - T_i\right)$$

$= 0\ W$

(6) Ventilation Load

$$\dot{Q}_{Ven} = \dot{m}_{Ven}\left(e_o - e_i\right)$$

$= -1000\ W$

(7) AC Load

$$\dot{Q}_{AC} = -\begin{pmatrix} \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \\ \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} \end{pmatrix}$$
$$-\left(m_a c_a + DTM\right)\left(T_i - T_{comf}\right)/t_c$$

$= -2800\ W$

TOTAL LOAD :

$$\dot{Q}_{Tot} = \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} + \dot{Q}_{AC}$$

$= 240 + 1000 + 190 + 0 - 200 + 0 + 0 - 1000 - 2800$

= -2570 W (Since cooling is done in summer , this load is negative.)

∴ $\dot{Q}_{Tot}$ = **2570 W (Neglecting negative sign) ....(A)**

COOLING CAPACITY (During pull down time)

Cooling capacity $=\dfrac{\dot{Q}_{Tot}}{3.517*1000}$

$=\dfrac{2750}{3.517*1000}$

Cooling capacity = 0.78 TR ....(B)

Case B : After the pull down time (after 10 minits)

(1) Metabolic Load :

$Q_M \approx 240$ W

(2) Radiation Load

(a) Direct Radiation load :

$$\dot{Q}_{Dir} = \sum_{Surfaces} S\tau \dot{I}_{Dir} \cos\theta$$

=1000 W

(b) Diffuse Radiation load :

$$\dot{Q}_{Dif} = \sum_{Surfaces} S\tau \dot{I}_{Dif}$$

= 190 W

(c) Reflected radiation load :

$$\dot{Q}_{Ref} = \sum_{Surfaces} S\tau \dot{I}_{Ref}$$

= 0 W

(3) Ambient Load

$$\dot{Q}_{Amb} = \sum_{Surfaces} SU\left(T_s - T_i\right)$$

= 200 W

(4) Exhaust Load

$$\dot{Q}_{Exh} = S_{Exh}U\left(T_{Exh} - T_i\right)$$

= 0 W

(5) Engine Load

$$\dot{Q}_{Eng} = S_{Eng}U\left(T_{Eng} - T_i\right)$$

= 0 W

(6) Ventilation Load

$$\dot{Q}_{Ven} = \dot{m}_{Ven}\left(e_o - e_i\right)$$

= 600 W

(7) AC Load

$$\dot{Q}_{AC} = -\begin{pmatrix} \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \\ \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} \end{pmatrix}$$

$$-\left(m_a c_a + DTM\right)\left(T_i - T_{comf}\right)/t_c$$

≈ -2230 W

TOTAL LOAD :

$$\dot{Q}_{Tot} = \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} +$$

$$\dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} +$$

$$\dot{Q}_{AC}$$

= 240 + 1000 + 190 + 0+ 200 + 0 + 0 + 600 - 2230

= 0 W

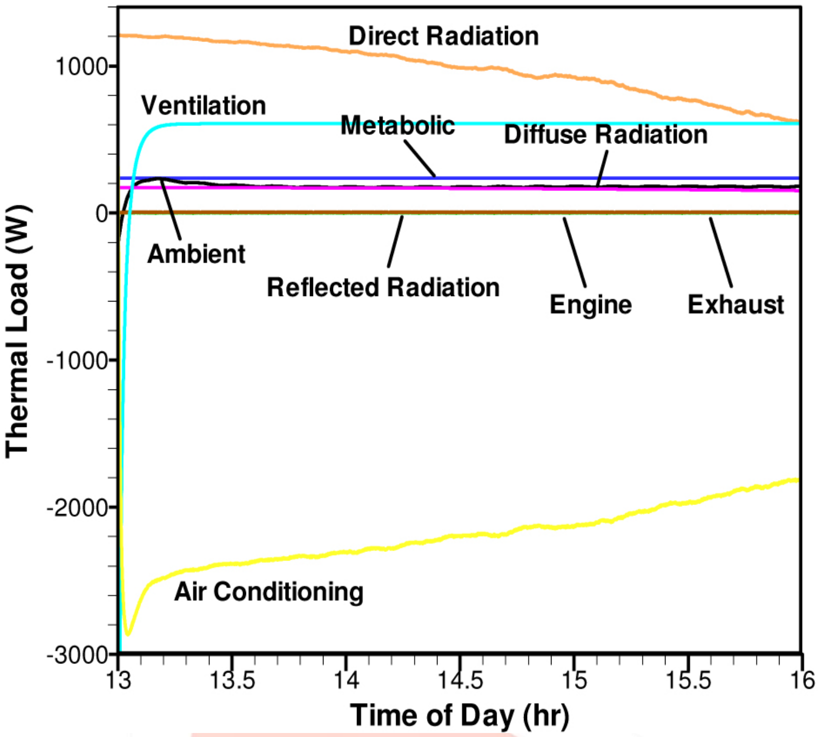

Figure 5 : Thermal load v/s Time...(C)

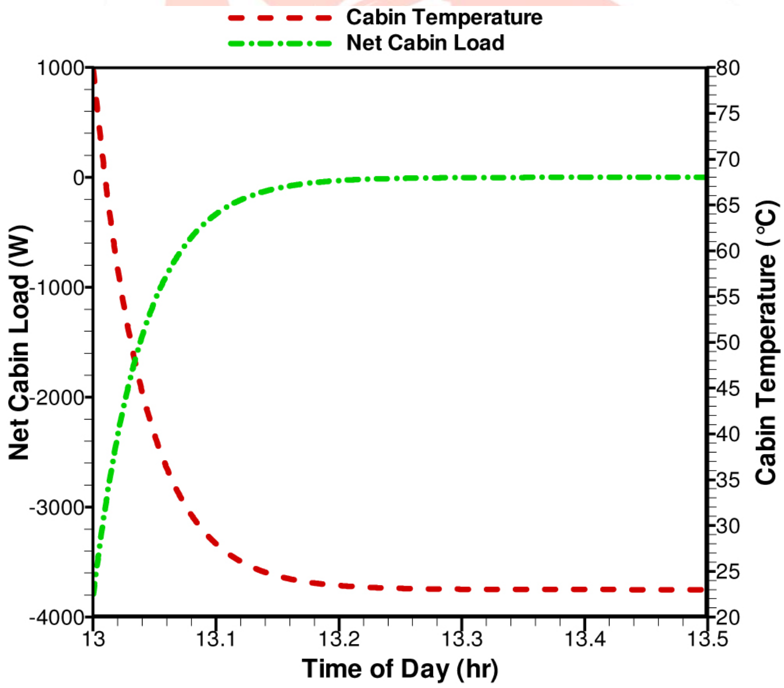

Figure 6 : Net cabin load v/s Time v/s Cabin temperature.........(C)

V. RESULTS & DISCUSSION

Figure 5 shows the contribution of each load category in the net thermal load gained by the cabin. It can be seen that the engine, exhaust, and reflected radiation loads are negligible. We can conclude that when seeking guidelines for reducing cabin heat gains in this driving condition, the engine, exhaust, and reflected radiation loads may be neglected from consideration.

The direct and diffuse radiation loads, on the other hand, are important AC loads that tend to give rise to the cabin temperature. It is observed that the direct radiation load decreases due to the decrease in the sun elevation angle after midday.

Metabolic load is another positive load which is constant due to no change in the number of passengers. Ventilation and ambient loads are functions of the temperature difference between cabin and ambient. During the first 5 minutes, the cabin temperature is higher than the ambient. It results in the negative starting values of these loads. After the cabin temperature reaches the steady condition, the warmer ambient imposes almost constant positive ventilation and ambient loads.

The AC load formula of Eq. (23) reaches a peak absolute value of about $3000W$ at the beginning of the scenario. After this time, the pull-down of the cabin temperature finishes and the AC load reaches a balance with the rest of the loads. Then, the absolute AC load value gradually decreases since no more pull-down is required and the contribution of the direct radiation load as a positive heat gain is decreasing as well.

Figure 6 shows the variation of cabin air temperature with time. The net heat gain in the cabin has been plotted as well. Negative heat values mean heat loss from the cabin, while positive values mean heat gains by the cabin. Figure 6 shows that the cabin temperature decreases from a soak temperature of $80°C$ to the comfort temperature after almost 10 minutes represented by the pull-down time.

After the pull-down, a steady-state situation is achieved where the loads are balanced and a zero net load is maintained for the cabin for the rest of the period.

VI. Conclusions

Automobile air conditioning systems is designed to compensate the continuous changes of the cabin thermal loads in order to maintain passenger thermal comfort. In this research, the Heat Balance Method (HBM) is applied to a vehicle cabin to model the various heating and cooling loads transferred to the cabin via radiation, convection, or conduction.

Mathematical models of heat transfer phenomena are used to calculate the different load categories. Mathematical load calculation models are devised and collected from various for load estimation. Lumped-body assumptions are made.

To perform case study on Wagon R car, specific material properties and the simplified geometry is considered. Some load categories such as the engine, exhaust, and reflected radiation are often negligible, while others such as the ambient or ventilation load can play important roles in the variation of cabin temperature.

This research aims to provide a basis for estimating thermal loads in vehicle cabins. From the load calculation, cooling capacity can be calculated & thus tonne of refrigeration required is found out. The result of this study can be used by HVAC engineers to design more efficient car AC systems. By efficient AC system, Reduction of fuel consumption and tailpipe emission are possible.

Acknowledgments

The author would like to thank SAE Technical paper [10] and SVMIT, Bharuch for their valuable input during the course of this investigation.

References

[1] Welstand, J., Haskew, H., Gunst, R., and Bevilacqua, O., “Evaluation of the Effects of Air Conditioning Operation and Associated Environmental Conditions on Vehicle Emissions and Fuel Economy,” SAE Technical Paper 2003-01-2247, 2003, doi:10.4271/2003-01 2247.

[2] Lambert, M. A., and Jones, B. J., “Automotive Adsorption Air Conditioner Powered by Exhaust Heat. Part 1: Conceptual and Embodiment Design,” Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 220(7):959-972,2006,doi: 10.1243/09544070JAUTO221

[3] Johnson, V., “Fuel Used for Vehicle Air Conditioning: A State-by-State Thermal Comfort-Based Approach,” SAE Technical Paper 2002-01-1957, 2002, doi: 10.4271/2002-01-1957.

[4] Fanger, P. O., “Thermal Comfort: Analysis and Applications in Environmental Engineering,” McGraw-Hill, New York, 1972.

[5] Ingersoll, J., Kalman, T., Maxwell, L., and Niemiec, R., “Automobile Passenger Compartment Thermal Comfort Model - Part II: Human Thermal Comfort Calculation,” SAE Technical Paper 920266, 1992, doi: 10.4271/920266.

[6] ASHRAE Handbook of Fundamental, American Society of Heating, Refrigerating, and Air Conditioning, Atlanta, GA, 1988.

[7] Kamar, H., Senawi, M., and Kamsah, N., “Computerized Simulation of Automotive Air-Conditioning System: Development of Mathematical Model and Its Validation,” International Journal of Computer Science Issues, 9(2):23-34, 2012

[8] Zheng, Y., Mark, B., and Youmans, H., “A Simple Method to Calculate Vehicle Heat Load,” SAE Technical Paper 2011-01-0127, 2011, doi:10.4271/2011-01-0127.

[9] Ding, Y. and Zito, R., “Cabin Heat Transfer and Air Conditioning Capacity,” SAE Technical Paper 2001-01-0284, 2001, doi:10.4271/2001-01-0284.

[10] Fayazbakhsh, M. and Bahrami, M. “ Comprehensive Modeling of Vehicle Air Conditioning Loads Using Heat Balance Method”, SAE Technical paper, 2013-01-1507, doi :10.4271/2013-01-1507.

[11] ISO Standard, “Ergonomics of the Thermal Environment- Determination of Metabolic Heat Production,” ISO Standard 8996, Rev. 2004.

[12] Ingersoll, J., Kalman, T., Maxwell, L., and Niemiec, R., “Automobile Passenger Compartment Thermal Comfort Model - Part I: Compartment Cool-Down/Warm-Up Calculation,” SAE Technical Paper 920265, 1992, doi: 10.4271/920265.

[13] Talbi, M., and Agnew, B., “Energy Recovery from Diesel Engine Exhaust Gases for Performance Enhancement and Air Conditioning,” Applied Thermal Engineering, 1(5): 5-10, 2002.

[14] Khayyam, H., Kouzani, A. Z., and Hu, E. J., “Reducing Energy Consumption of Vehicle Air Conditioning System by an Energy Management System,” Presented in IEEE The 4th International Green Energy Conference, China, 2009.

[15] Arndt, M., and Sauer, M., "Spectroscopic Carbon Dioxide Sensor for Automotive Applications," Sensors, Proceedings of IEEE 2004, 1:252-255, 2004.
[16] Fletcher, B., and Saunders, C., "Air Change Rates in Stationary and Moving Motor Vehicles," Journal of Hazardous Materials, 38:243-256, 1994.
[17] Singh, A. K., Singh, H., Singh, S. P., and Sawhney, R. L., "Numerical Calculation of Psychrometric Properties on a Calculator," Building and Environment, 37(4):415-419, 2002, doi:10.1016/S0360-1323(01)00032-4.
[18] www.marutiwagonr.com