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Evaluation of the Fuel Consumption Rate for a Fully Loaded Engine and a Non-Air Conditioned Engine of an Automobile (A Case Study of Automobiles in Bonny Island)

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ABSTRACT

The evaluation of the fuel consumption rate for a fully loaded engine and a non-air conditioned engine of an automobile has been carried out using analytical method and the equations were implemented in MATLAB computing environment. In this research, air psychometrics and heat balance were applied to the vehicle cabin to model the various heating and cooling loads transferred to the cabin via radiation, convection and conduction, and the experiment was conducted using Peugeot 307 car. Automobile air conditioning systems was analyzed to compensate the continuous changes of the cabin thermal loads in order to maintain passenger thermal comfort. The key results obtained show that fuel consumption rate is dependent on thermal load and other auxiliary demands of a vehicle such as aerodynamics, rolling resistance, inertia and irradiation. It was found that the AC load which is about 80% of the engine total load resistance forms the most prevalent influencing constraint of fuel consumption rate in the car. It was also found that fuel consumption rate increased with vehicle speed to a maximum: 91(gpm) with AC and 89 (gpm) no AC test conditions. The result obtained also shows that at high vehicle speed between 70-90 km/hr., fuel consumption rate for both AC and no AC test conditions are the same, but at speed above 90km/hr., AC cars consume less amount of fuel.

Keywords: automobile, fuel consumption rate,

INTRODUCTION

As the population of the world increases far more than available energy resources to cater for the said population, there is an increasing need for the more efficient utilization of energy. One of the ways in which this can be achieved is through improved fuel economy. A number of factors affect the rate of fuel consumption in vehicles. One of which is the use of air conditioners. This use is pertinent to users in countries situated in the tropical region with temperatures naturally so high that driving becomes uncomfortable without the use of air conditioners. But although air conditioners provide satisfaction for vehicle users especially in tropical regions, this satisfaction comes at a cost –increased fuel consumption. This naturally leaves vehicle users in a dilemma involving an opportunity cost between increased satisfaction and added fuel consumption.

The upshot of this dilemma is a myth which on surface value can be justified. This common myth suggests that air conditioners when used in vehicles increases fuel consumption. This is partly true but when speed is factored in, it becomes true only for certain ranges of speed. At very high speeds, aerodynamic drag is factored in due to introduction of more openings in the vehicle through the windows. Our study is done with a vehicle fully air conditioned with windows completely up and again repeated with windows down and turned air conditioner off. Another point that debunks this argument is the power required to operate the air conditioner compressor at high speeds is lower than it is at low speeds as the engine has already achieved higher rpm at high speeds. This is experimentally proven by the American Society of Automotive Engineers. With the following already established, the framework for this study is set in place.

Statement of the problem

Efficient use of fuel not only saves cost but is becoming more and more relevant in today's world as the supply of fuel resources is not increasing at the same rate as the demand. Also the more fuels especially from fossil is burnt and consumed the more the emission of greenhouse gases. This increases overall world temperatures to levels that can be harmful for our environments.

In regions that are already too hot, there is a need to improve the ergonomic conditions of the people resident in those regions through the use of air conditioners. But the problem that follows air conditioner usage in vehicles is the problem of fuel consumption. If these consumption rates cannot be justified, then people will be reluctant to make use of them. But the problem associated with air conditioners and fuel consumption is although already obvious that they consume an additional amount of fuel, there are situations where their usage outweighs the far more fuel consumption that would have been incurred had they not be used. So, when should the small extra fuel consumption rate of air conditioners be tolerated in order to minimize the much larger fuel consumption rate due to trying to overcome aerodynamic drag? That is one of the problems.

Another problem is the climate change: Where fossil fuels are still used in vehicles, it is important to both man and the environment that they are efficiently used. If we are able to enlighten more people on how to maximize fuel economy through air conditioner usage not only do we solve the problem of cost, we also solve an environmental problem.

Aim and objectives

The aim of the study is to evaluate the fuel consumption rate for a fully loaded engine and non-air conditioned engine of an automobile. The key objectives of the study are

i. To study experimentally the fuel consumption rate of an air conditioned vehicle
ii. To measure using an experimental model, the amount of fuel consumed when an air conditioning system is switched on with vehicle moving at different speeds.
iii. To evaluate the differences in the total load on the engine when it is fully loaded at different speed and when the AC are switched off at same speed.
iv. To evaluate the differences in specific fuel consumption rate when the engine is fully loaded at different speed and when the AC are switched off at same speed.
v. To ascertain a mathematical relationship between fuel consumption rate of air conditioning system and change in speed.
vi. To compare fuel consumption rate of the vehicle at different speeds when the air conditioning system is switched on and off.
vii. Suggest the optimum speeds to turn on and off the air conditioning system in order to maximize fuel.

Scope of the Study

The scope of the study is to evaluatee the fuel consumption rate of fully- loaded engine and non-air-conditioned engine of an automobile.

Significance of the Study

The study will be beneficial to the following groups:

i. **Vehicle users:** This study will be beneficial to vehicle users as it will help in cost savings through maximization of fuel
ii. **Environment:** This study will implement a more efficient fuel usage that will be beneficial to the environment as emissions will be cut down reducing greenhouse effects and thus limiting climate change.
iii. **Vehicle Manufacturers:** This study will serve as a source of information for the vehicle manufacturers in the automobile sector.
iv. **Researcher:** This research work will be valuable to other researchers who may carry out the similar research work in related field for reference purposes.

Definition of Terms

Fuel: Any substance which gives heat energy on combustion. It is any material either solid, liquid or gas that can be made to react with other substances, releasing varying forms of energy as heat.

Fuel Consumption: The amount of fuel used or consumed by the vehicle per unit time.

Air Conditioning: This is the process of removing heat and moisture from the interior of an occupied space, improving the comfortability of the occupants. It involves controlling conditions of temperature, humidity, purity and air motion in an enclosed space independent of external conditions.

Aerodynamic Drag: This is a mechanical force that opposes the motion of an object through fluid.

Limitations of the Study

Every research study comes with a number of constraints and challenges. In the course of carrying out this project, the following challenges were encountered:

i. Lack of finance to fuel vehicle in order to carry out repeated testing so as to improve investigation accuracy.
ii. Absence of easily available access roads void of traffic congestion within the Port Harcourt metropolis to allow for especially high speeds driving to match our investigations.
iii. Use of only vehicle in the experiment which may not have captured a universal result applying to every vehicle with different engine ratings and sizes of air conditioning units.

Thus, this study is limited to fully-loaded engines and non-air conditioned engines without regards to the topography of the road and the effects of gear selections on fuel consumption rate of the engines.

LITERATURE REVIEW

Extent of past works

Recent literature shows that various tests have been performed on vehicles to determine the fuel consumption rate of various vehicle designs. Rugh (2010) stated that although no automotive industry consensus existed on vehicle A/C fuel use test procedure, a recommendation including operating vehicle over repeated drive cycles or at a constant speed until steady-state cabin air temperature is attained and running A/C off and A/C on tests in order to calculate cool down and steady-state A/C fuel use while gathering data for both cool-down and steady-state passenger compartment thermal conditions.

Shean (2013) in the Oak Ridge National Laboratory highlighted some important points on A/C effect on fuel economy. His findings show that A/C fuel consumption increased slightly with increasing vehicle speed, and that open windows in vehicle resulted in an increased fuel consumption due to increasing aerodynamic drag that further increased with an increase in vehicle speed. Shean further demonstrated that under certain conditions such as high vehicle speed leads to an increased aerodynamic drag when the windows are down. Consequently, fuel consumption rate is significantly increased due to high thermal load on the air conditioning unit and cabin area, and so it is more economical to wind up windows and turn on the vehicle's A/C.

Kiss (2013) modeled an air conditioning system using MATLAB/Simulink. A transient A/C simulation tool was incorporated into vehicle simulation models which provides a tool for developing more efficient A/C systems on thorough consideration of the transient A/C system performance. This also helped in modeling the A/C system load on engine in more sufficient detail. Similarly, Nielsen (2016) developed a 1D model using GT-SUITE software where he obtained the energy used by an A/C system. This was later verified by on-site vehicle experimental test with a Volvo S60 which matched closely the predictions based on his model. From his findings, he was able to propose energy saving measures that included proper utilization of the A/C.

A load estimation model proposed by Bahrami (2013) used heat balance method (HBM) to estimate and investigate various loads include cooling loads in vehicle cabin. Using a lumped-body approach, the model was implemented by a computer code applicable to arbitrary driving conditions. Another model known as CARSIM, was developed by Mohd (2012) where a semi-empirical model was used for simulating thermal and energy performance of an automotive air conditioning system in passenger vehicles. The model consisted of two sections, namely the empirical evaporator correlations and the dynamic load simulation. This model when compared with road test data shows a reasonable prediction of the performance of the automotive air conditioning system with accuracy.

Arturo (2018), also developed the GT-SUITE model which he used as a simulation tool. The air conditioner was modelled according to its various components such as the compressor, evaporator, thermal expansion valve and condenser. The vehicle's cabin, an additional sub-system that influenced energy consumption was also considered. The simulated model showed good agreement with the test data for important parameters such as the compressor power consumption and the air temperature after the evaporator with only a 6.25 % percent difference between the test data and the simulated model. In the way, a vehicle cabin was co-simulated using TAITherm and GT-SUITE software and then the results were compared with results from simulations from STAR-CCM+ model. The cabin model was used to simulate a pull-down, where the air inside the cabin was first heated up due to solar radiation and then afterwards cooled down using the A/C system.

Shete (2015) conducted the influence of automotive air conditioning load on fuel economy of internal combustion engine vehicles, he suggested that the factors affecting air conditioning load on an engine include, climatic conditions such as temperature, pressure and relative humidity, cabin conditions, compressor speed, the difference between climatic and cabin conditions as well as overall efficiency of the air conditioning system. Shete also found that the overall efficiency of the air conditioning system depends on the type of condenser, evaporator, compressor and refrigerant used as well as vehicle speed. He further stated that for maximum efficiency, electrically driven compressors were to be used with tube in type evaporators. Also, evaporative condensers were to be used as well as refrigerants having higher critical temperature, pressure, thermal conductivity and latent heat of vaporization as it will result in lesser load on the engine.

Jungwoo (2012) used what he considered a new approach to analyzing the effect of each component of the air conditioning system on the fuel consumption, which he developed through a systematic sequence of processes. The driving torque of the alternator was obtained through a single-component experiment. The effect of the electric load applied to the alternator on the driving torque was then measured. Linear recursive equations from these data were used to derive the driving torques of the blower, the cooling fan and the clutch. The driving torque of the compressor was verified by installing a refrigerant line and a blower on the compressor. And then finally, the contribution of each A/C system component to the fuel consumption was calculated. The fuel consumption increased linearly with increasing engine load at a constant speed. A linear recursive equation of the fuel consumption versus the engine torque was derived from the experimental data. The estimated fuel consumption and the measured data showed an error of less than 3.2%. This study found that among the A/C system components, the most important component affecting the fuel consumption was the compressor which caused a fuel consumption increase of 77–89% with the blower, 6–12%; the cooling fan, 4–10%; and the clutch, 0.7–2%.

Research Gap and Expected Outcome

The works conducted in recent literature is not exhaustive on the subject matter. Vehicle information and design parameters are not conspicuously delineated especially the manufacturer of these vehicles. Also, lack of data to compare simulation results is a drawback as identified by (Arturo,2018). The absence of a conventionally accepted method of determining fuel consumption rate of A/C system has also increased the difficulty of the research in this area (Rugh, 2010). Time was a constraint; as certain works did not have the luxury of sufficient time for a more in depth study. Also, not all researchers had access to sufficient tools and equipment to conduct more detailed study. Generally, it was nearly impossible to model simulations that showed zero deviations from real life scenarios. But in this research work, On-Board Diagnostic (OBD) software(Poegout planet 2000) is used to obtain data and implemented it in MATLAB simulation to evaluate the overall fuel consumption rate for both AC and Non-AC Test conditions.

MATERIALS AND METHODS

The amount of fuel consumed depends on the engine, the type of fuel used, and the efficiency with which the output of the engine is transmitted to the wheels. This fuel energy is used to overcome

- Rolling resistance primarily due to flexing of the tires
- Aerodynamic drag as the vehicle motion is resisted by air, and
- Inertia and hill-climbing forces that resist vehicle acceleration, as well as engine and drive line losses

Fuel efficiency is a historical goal of automotive engineering, and its associated terminologies are discussed below:

Fuel Economy

It is a measure of how far a vehicle will travel with a gallon of fuel, expressed in miles per gallon (mpg). This is a popular measure used for a long time by consumers. It is used also by vehicle manufacturers and regulators, mostly to communicate with the public. As a metric, fuel economy actually measures distance traveled per unit of fuel

Fuel Consumption

This is usually the inverse of fuel economy. It is the amount of fuel consumed in driving a given distance. It is measured in the United States in gallons per 100 miles (gpm), and in liters per 100 kilometers (lpkm) in Europe and elsewhere throughout the world. Fuel consumption is a fundamental engineering measure that is directly related to fuel consumed per 100 miles and is useful because it can be employed as a direct measure of volumetric fuel savings. Fuel consumption is also the appropriate metric for determining the yearly fuel savings if one goes from a vehicle with a given fuel consumption to one with a lower fuel consumption. Because fuel economy and fuel consumption are reciprocal, each of the two metrics can be computed in a straight forward manner if the other is known.

In mathematical terms, if fuel economy is X and fuel consumption is Y, then their relationship is expressed by XY = 1 which is not linear relationship (National Research Council [NRC], 2011). It added that in a conventional vehicle propelled by an internal combustion engine, either spark ignition (SI) or compression (CI), most of the energy in the fuel goes to the exhaust and to the coolant (radiator), with about a quarter of the energy doing mechanical work to propel the vehicle. This is partially due to the fact that both engine types have thermodynamic limitations, but it is also because in a given drive schedule the engine has to provide power over a range of speeds and loads; it rarely operates at its most efficient point. This is illustrated in figure 3.1 below. And a typical thermal load of the cabin of an automobile vehicle is illustrated in figure 3.2 showing the various possible factors of heat load on an air-conditioning unit.

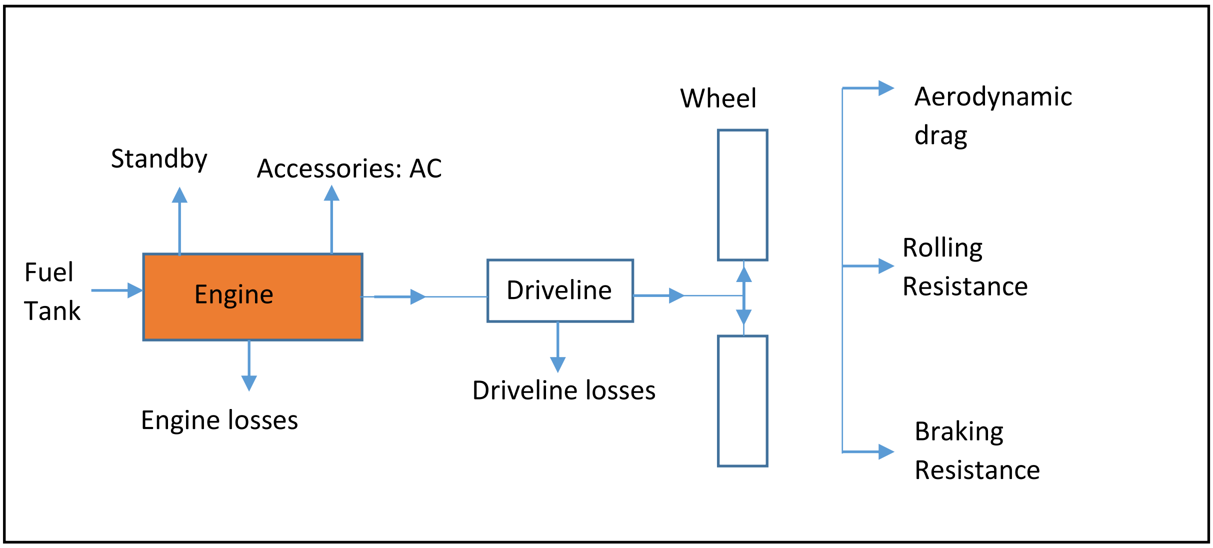
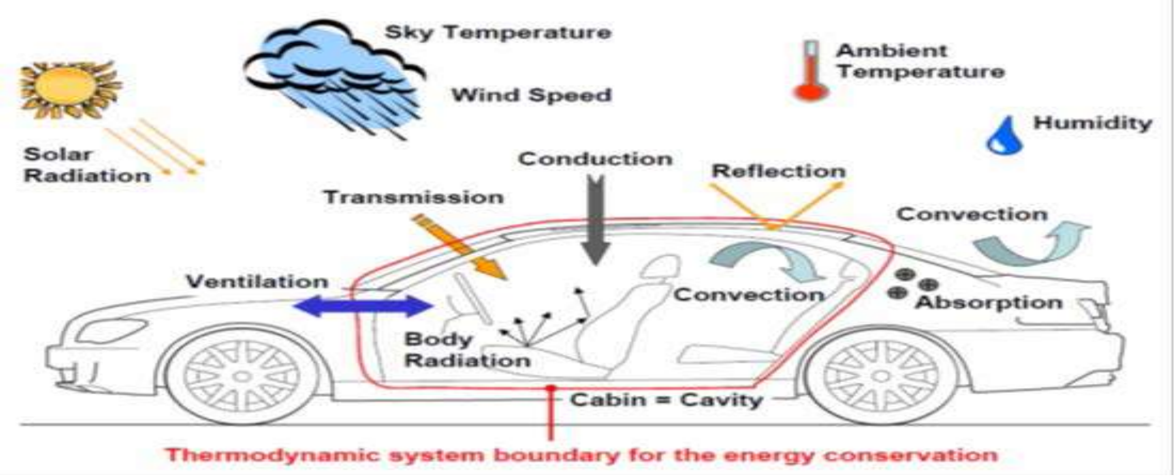


Figure 3.1 Schematic of Power Flow in an Automobile



Thermodynamic system boundary for the energy conservation

Vehicle Load Analysis

The total vehicle load neglecting road curvature according to Ross (1997) is given as:

$$P_{load} = P_{tires} + P_{air} + P_{inertia} + P_{ac} \qquad (3.1)$$

where P_{tires}, P_{air}, $P_{inertia}$ and P_{ac} (W) are the Power required to overcome rolling resistance, air resistance, accelerate the car and to drive the accessories, respectively.

Power is the time-derivative of energy expressed as

$$P = \frac{dE}{dt} \qquad (3.2)$$

One liter of gasoline fuel contains approximately 8.787kWh according to (Werne, 2013). He established that the factor relating power and fuel consumption can be expressed as

$$fuel\ consumption = \frac{Energy}{Distance} = \frac{Power}{Speed} \qquad (3.3)$$

Air Resistance

Ross (1997) shows that the power required to overcome aerodynamic resistance) is given as

$$P_{air} = \frac{\rho C_D A V^3}{2} \qquad (3.4)$$

Rolling Resistance

The number of cycles of deformation is a linear function of the distance traveled, and the energy used to overcome rolling resistance is approximately proportional to the distance traveled Transportation Research Board (TRB) (2006). As given below:

$$P_{tires} = C_R.m.g.v \tag{3.5}$$

where C_R(-), m(kg), g (ms^{-2}), V (m/s) are the coefficient of rolling resistance specific to the tire, mass of the car, acceleration due to gravity and the vehicle speed, respectively.

Inertia, Speed and Acceleration

$$P_{inertia} = \delta.m.a.v \tag{3.6}$$

where δ (-), a(m/s^2) are the correction factor for the rotational and linear inertia of the car and the acceleration, respectively.

Accessories/Air conditioning Load

Farington and Rugh (2000) found that the air conditioning system is the single largest auxiliary load on a vehicle. The cabin psychometrics showing the comfort condition of a vehicle occupant is demonstrated in figure 3.3 below as a result of thermal effect of figure 3.2.

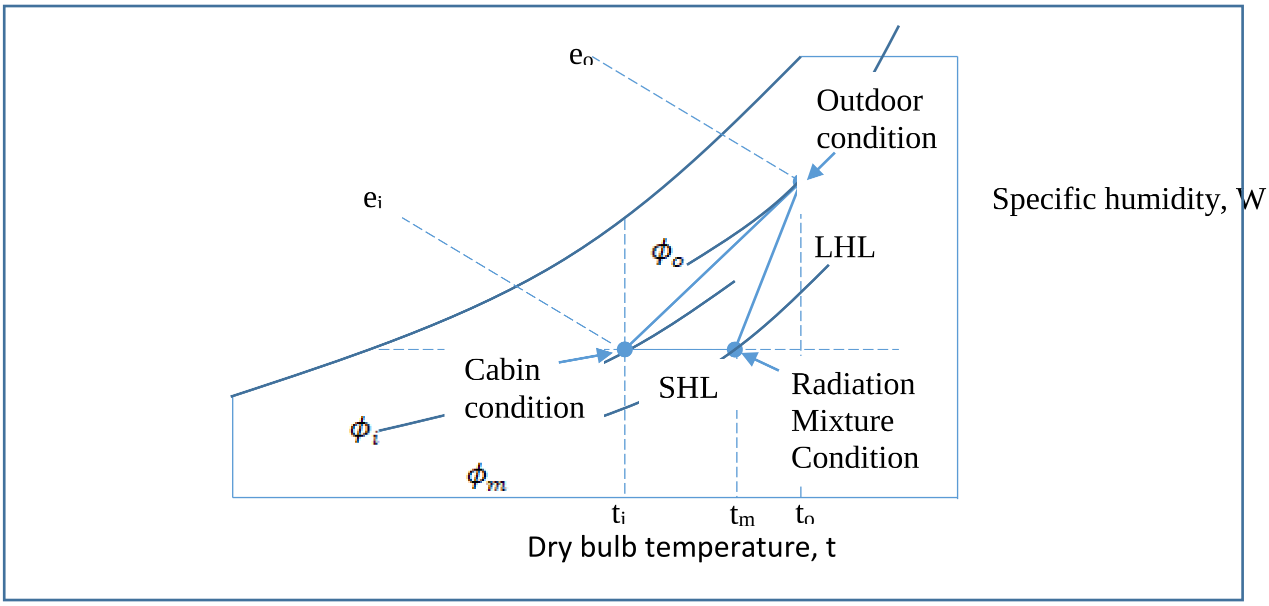


Figure 3.3 Cabin Psychometric Heat Load

Metabolic Load

The heat released to the cabin air due to metabolic activities inside the human body contributes to the heat gain by the cabin air. This is given as the sum of the individual heads present in the vehicle according to the equation below:

$$\dot{Q}_m = \sum_{i=1}^{n} M.A \tag{3.7}$$

$$A = 0.202\,W^{0.425}H^{0.725} \tag{3.8}$$

where n is the number of persons present in the vehicle. $i = 1,2, \dots, n$

M is the individual body mass, and A is the body surface area as a function of passenger height (H) and weight (W) according to ISO standard (2004).

Radiation Load

Heat gain due to radiation is a significant part of the cooling load encountered in vehicles according to ASHRAE (1988). Solar radiation heat load is classified into 3 categories: direct, diffuse and reflected radiation loads.

Direct Radiation Load

The sum of direct radiation over the surfaces is given as:

$$\dot{Q}_{dir} = \sum_s A_s.\tau\,.I_{dir}.cos\theta \tag{3.9}$$

$$I_{dir} = \frac{A}{e^{\left(\frac{B}{sin\beta}\right)}} \tag{3.10}$$

where A and B are constants tabulated in ASHRAE (1988) for different months. β is the altitude angle.

Diffuse Radiation Heat Load

Summing over the surfaces, the diffuse radiation is given as:

$$\dot{Q}_{dif} = \sum_s A_s.\tau.I_{dif} \tag{3.11}$$

$$I_{dif} = C.I_{dir}\frac{1+\cos\theta}{2} \tag{3.12}$$

where I_{dif} is the diffuse radiation heat gain per unit area, and θ is the surface tilt angle measured from the horizontal surface and the values for C are tabulated in the ASHRAE handbook of fundamentals.

Reflected Radiation Heat Load

Reflected radiation heat load over the surfaces is given as:

$$\dot{Q}_{Ref} = \sum_S A_s . \tau . I_{Ref} \tag{3.13}$$

$$I_{Ref} = \rho_g \left(I_{dir} + I_{dif}\right)\left(\frac{1-\cos\theta}{2}\right) \tag{3.14}$$

where ρ_g is the ground reflectivity coefficient.

Net absorbed heat of each surface element due to radiation can be written according to ISO Standard (2004) as

$$\dot{Q}_{s,Rad} = A_s \alpha \left(I_{dir} \cos\theta + I_{dif} + I_{Ref}\right) \tag{3.15}$$

Ambient Load

This is the contribution of the thermal load transferred to the cabin air due to temperature difference between the ambient and the cabin air. According to Ingersoll *et al*. (1992), the ambient heat load is given as

$$\dot{Q}_{amb} = \sum_s A_s U(T_s - T_i) \tag{3.15}$$

$$U = \frac{1}{R_{th}} \tag{3.16}$$

$$R_{th} = \frac{1}{h_o} + \frac{\lambda}{\kappa} + \frac{1}{h_i} \tag{3.17}$$

$$h = 0.6 + 6.64\sqrt{v} \tag{3.18}$$

where U is the overall heat transfer coefficient of the surface elements based on the inside convection, conduction through the surfaces and outside convection, and T_s and T_i are the average surface temperature and average cabin temperature, respectively. R_{th} is the net thermal resistance of a unit surface area, h_o and h_i are the outside and inside convection coefficients, κ is the surface thermal conductivity, v the vehicle speed, and λ the thickness of the surface element. Ingersoll *et al*. (1992) emphasized that these correlations are applicable in all practical automobile.

Net ambient heat absorbed is given as:

$$\dot{Q}_{s,amb} = A_s U(T_o - 2T_s + T_i) \tag{3.19}$$

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

Engine Load

The engine thermal load is given as:

$$\dot{Q}_{Eng} = A_{Eng} . U\left(T_{Eng} - T_i\right) \tag{3.20}$$

$$T_{Eng} = -2x\,10^{-6} N^2 + 0.0355N + 77.5 \tag{3.21}$$

where N(rpm) is the engine speed

Ventilation Load

Arndt and Saver (2004) reported the minimum fresh air that should be supplied into the cabin to maintain passengers comfort in terms of air quality. They established that the ventilation heat gain consists of the sensible and latent heat loads as shown in figure 3.3 above. This is expressed as given below

$$\dot{Q}_{ven} = \dot{m}_{ven}(e_o - e_i) \tag{3.22}$$

$$e = 1006T + (2.501x\,10^6 + 1770T)w \tag{3.23}$$

where $\dot{m}_{ven}$ is the ventilation mass flow rate, and $e_o,\ e_i\ and\ T$ are the ambient and cabin enthalpies and the air temperature for the ambient and cabin conditions, respectively.

The humidity ratio in gram of water per gram of dry air is given as

$$w = 0.62198\,\frac{\phi P_s}{100P - \phi P_s} \tag{3.24}$$

where ϕ is the relative humidity, P and P_s are the air pressure and water saturation pressure, respectively.

Air Conditioning Load

In hot ambient conditions, AC cooling load is negative and it is required for maintaining comfort conditions according to (Vaghela & Kapadia, 2014).

$$\dot{Q}_{AC} = \left(\dot{Q}_{dir} + \dot{Q}_{dif} + \dot{Q}_{Ref} + \dot{Q}_{amb} + \dot{Q}_{Eng} + \dot{Q}_{ven}\right) - \frac{\left[(m_a C_a + DTM)(T_o - T_{comf})\right]}{t_c}$$

(3.25)

$$t_c = \frac{t_p}{\ln(T_o - T_{comf})} \tag{3.26}$$

where t_c is the pull-down constant

Tractive Force and Tractive Energy

These concepts are useful for understanding the role of vehicle mass, rolling resistance, and aerodynamic drag. The instantaneous tractive force required to propel a vehicle is given by the National Research Council (NRC) (2011), taking into account that tractive force is evaluated at the wheels of the vehicle, as:

$$F_{TR} = R + D + \left[m + 4\left(\frac{I_w}{r_w^2}\right)\right] a \tag{3.27}$$

where $R = r_o mg$ the rolling resistance, and $D = C_D A \frac{V^2}{2} \rho$ aerodynamic drag. C_D is the drag coefficient, m is the vehicle mass, V the velocity, $a = \frac{dV}{dt}$ is the acceleration. A is the frontal area, r_o is the tire rolling resistance coefficient, g is the acceleration due to gravity, I_w is the polar moment of inertia of the four wheels, r_w is the wheel effective rolling radius, and ρ is the density of air.

The tractive energy required to travel an incremental distance ds is $F_{TR} V dt$, and its integral over all portions of the driving schedule is the total tractive energy provided $F_{TR} > 0$.

According to Sovran and Blaser (as cited in NRC, 2011), the tractive energy of a vehicle per unit distance is given by the relation

$$\frac{E_{TR}}{S} = m\left[\alpha r_o + \beta\left(\frac{C_D A}{m}\right) + \gamma\left(1 + \frac{4 I_w}{m r_w^2}\right)\right] \tag{3.28}$$

where S is the total distance travelled in the driving test schedule and $\alpha, \beta \ and \ \gamma$ are constants.

Fuel consumption rate according to NRC (2011) is given by the expression

$$g^* = \frac{\frac{E_{TR}}{\eta_{dr}^*} + E_{acc}}{\rho_f \left(\frac{\eta_b^*}{\eta_{b,max}}\right) \eta_{b,max}} + g_{braking}^* + g_{idling}^* \tag{3.29}$$

This is better expressed by Sovran (1983) as

$$g^* = \frac{bsfc}{\rho_f} BP \tag{3.30}$$

$$G^* = \left(\frac{264.172}{60x0.0000621371}\right)\frac{g^*}{V} \tag{3.31}$$

where g^*, $g_{braking}^*$, g_{idling}^* (m^3/h) are the volumetric fuel consumption rate over the driving schedule, braking and idling, respectively. E_{TR} is the tractive energy, ρ_f(kg/m^3) is the fuel density, η_{dr}^* is the average drive train efficiency, $\eta_{b,max}$ is the maximum engine brake thermal efficiency, V(m/s) is the vehicle speed, G^*(gpm) is the fuel consumption rate, and E_{acc} is the energy required to power the accessories (Pumps, and compressor).

Equation (3.27) shows clearly that fuel consumption rate can be significantly reduced by reducing E_{TR} and E_{acc} or increasing $\frac{\eta_b^*}{\eta_{b,max}}$, respectively.

Effect of Drive Train

The indicated power (kW) of the engine is given as

$$IP = \frac{\overline{P}_{mi} LA \frac{n}{60}}{1000} \qquad (3.32)$$

For 4 stroke n = N/2

where N(rpm) is the engine speed, $\overline{P}_{mi}$ (N/m^2) is the indicated mean effective pressure, A (m^2) is the cylinder area

Brake power is the power delivered to the output shaft of the engine, and it is the difference between the engine indicated power IP and friction power P_f and engine pumping loss P_{pl}. Engine auxiliaries include fuel, oil, and water pump, and compressor for air conditioning load.

$$BP = IP - \left(P_f + P_{pl}\right) \qquad (3.33)$$

Where P_f is the engine friction loss, and P_{pl} is the pumping loss.

Brake power (W) may also be obtained from the relation

$$BP = \frac{\overline{P}_{mb} LA \frac{n}{60}}{1000} = \frac{2\pi NT}{1000} \qquad (3.34)$$

where $\overline{P}_{mb}$ (N/m^2), L (m), and T (Nm) are the break mean effective pressure, cylinder length, cylinder area, engine speed, and engine torque, respectively.

Mechanical efficiency

$$\eta_m = \frac{BP}{IP} \qquad (3.35)$$

The brake thermal efficiency is given as:

$$\eta_b = \frac{3600}{bsfc(LHV)} \qquad (3.36)$$

The specific fuel consumption rate on brake power basis is expressed as:

$$bsfc = \frac{3600\dot{m}_f}{BP} \qquad (3.37)$$

where $bsfc$ (kg/kWh) and $\dot{m}_f$ (kg/hr.) are the break specific fuel consumption rate and the mass flow rate of fuel, and the fuel lower heating value LHV (kJ/kg), respectively.

RESULTS AND DISCUSSION

Input Data and Vehicle Test Parameters

The Input parameters: material properties the vehicle characteristics and the experimental data necessary for the computation and simulation of the various mathematical equations presented in chapter 3 are shown in Table 4.1, 4.2, and 4.3, respectively.

Table 4.1 Material Properties

Property	Unit	Symbol	Glass	Vehicle Body
Thermal conductivity	W/mK	κ	1.05	0.2
Density	kg/m^3	ρ	2500	1500
Transmissivity	-	ϵ	0.5	0
Absorptivity	-	α	0.3	0.4
Specific heat capacity	kJ/kgK	c	840	1000
Surface Element Thickness	Mm	λ	3	10

Table 4.2 Vehicle Test Parameters

Parameter Description	Symbol	Unit	Measurement value
Car type	Peugeot 307	-	-
Type of fuel	Gasoline		
Vehicle mass	m	kg	177.93
Transmission type	Automatic	-	-
Driver height	H_1	m	1.7
Driver body mass	M_1	kg	60
Vehicle frontal area	A	m^2	1.82
Passenger height	H_2	m	1.6
Passenger body mass	M_2	kg	80
Ventilation flow	$\dot{m}_{ven}$	kg/s	0.1
Ground reflectivity	ρ_g	-	0.02
Ambient temperature	T_o	°C	27
Initial cabin temperature	T_i	°C	
Ambient relative humidity	ϕ_o	%	78
Cabin relative humidity	ϕ_i	%	40
Comfort temperature	T_{comf}	°C	23
Deep thermal mass	DTM	J/K	5600
Drag coefficient	C_D	-	0.3

Table 4.3 Experimental Data

Speed (km/hr.)	**0**	20	30	40	50	60	70	80	90	100	110
Speed (rpm) no AC	700	1200	2000	2880	2560	2280	2720	3080	3520	3880	3360
Speed (rpm) with AC	840	1280	2160	2800	2440	2320	2720	3080	3560	3080	3320

Table 4.4 Engine Output Data

IP(kW)	BP(kW)	BP AC(kW)	$\dot{m}_f$(kg/kWh)	G*(m^3/s)	G*(gpm)	G* AC(gpm)
1456.7	1055.6	1266.7	2.4753	0.0034394	0	0
2255.9	1809.6	1930.2	4.2434	0.0058961	75.201	80.214
3800.1	3015.9	3257.2	7.0724	0.0098268	83.556	90.241
5004	4342.9	4222.3	10.184	0.014151	90.241	87.734
4374.3	3860.4	3679.4	9.0526	0.012578	64.171	61.163
4117.3	3438.2	3498.5	8.0625	0.011203	47.627	48.463
4840	4101.7	4101.7	9.6184	0.013364	48.701	48.701
5480.5	4644.5	4644.5	10.891	0.015133	48.254	48.254
6323.8	5308	5368.4	12.447	0.017295	49.02	49.577
5697.7	5850.9	4644.5	13.72	0.019064	48.63	38.603
5918.5	5066.8	5006.4	11.882	0.016509	38.284	37.828

Result from Simulation

A MATLAB program is developed to implement the various equations and the key results associated with vehicle fuel consumption rate are presented in this section. The variation of the engine fuel consumption rate against the vehicle load is shown in figure 4.1 for both the AC and no AC test conditions. It shows that the compressor load arising from Radiation, thermal mass, metabolic, ventilation, engine and ambient heat loads forms 88% of the engine loads requirements.

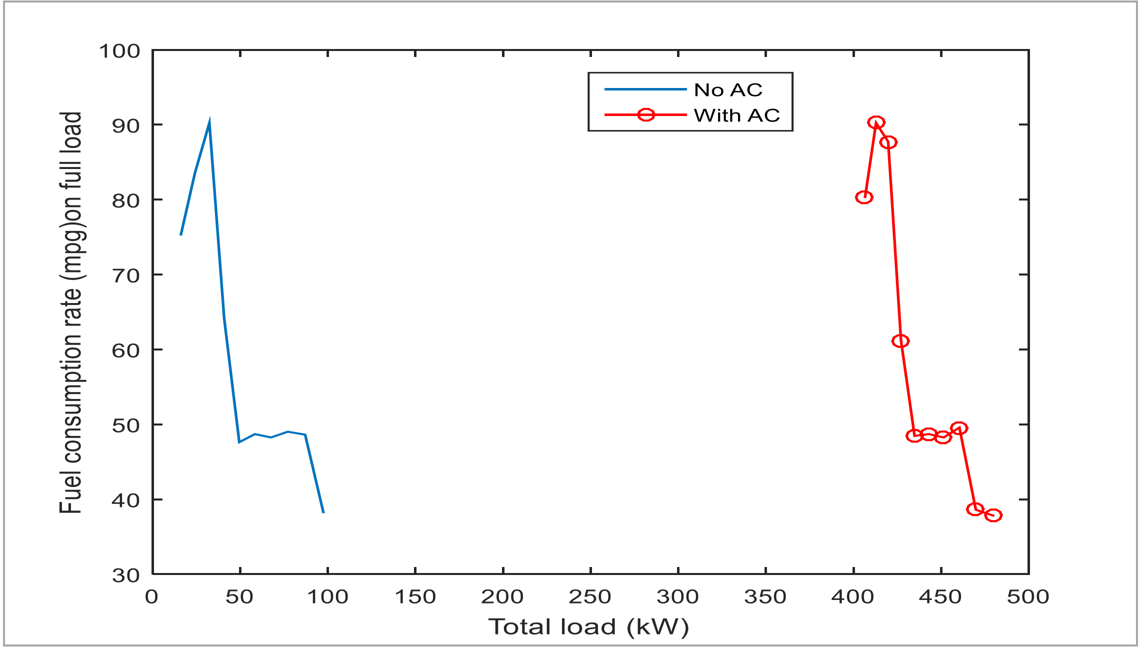


Figure 4.1 Fuel Consumption against Total Engine Load

Fuel consumption rate initially varies proportionately with brake power to a maximum and thereafter, fuel consumption rate is minimized for both AC and no AC test conditions as shown in figure 4.2. This shows that fuel consumption rate can be minimized on full load to about 40gpm (AC) and 37gpm (No AC), and at brake power of 4600kW fuel consumption rate for both test conditions is found to be 49 gpm regardless of speed and constraint factors.

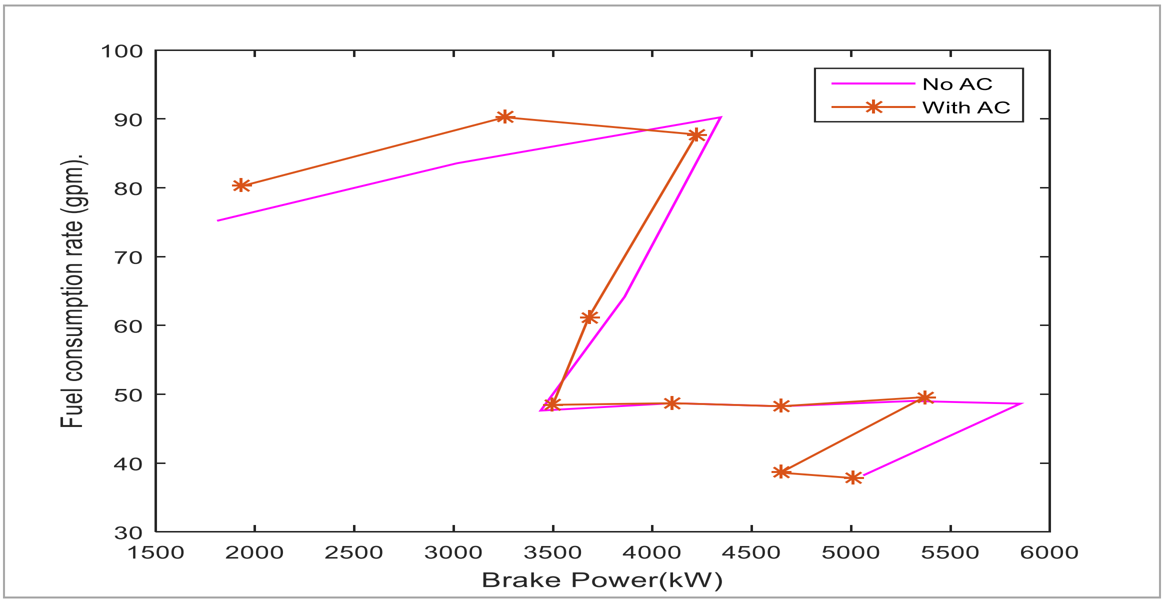


Figure 4.2 Variation of Fuel Consumption Rate Against Engine Brake Power

As shown in figure 4.3 below, Aerodynamic drag increases for both AC and non-AC conditions at initial speed of the engine. But as the speed increases, the aerodynamic drag for the both conditions becomes the same to some points and at higher speed, aerodynamic drag for AC condition decreases leading to less fuel consumption.

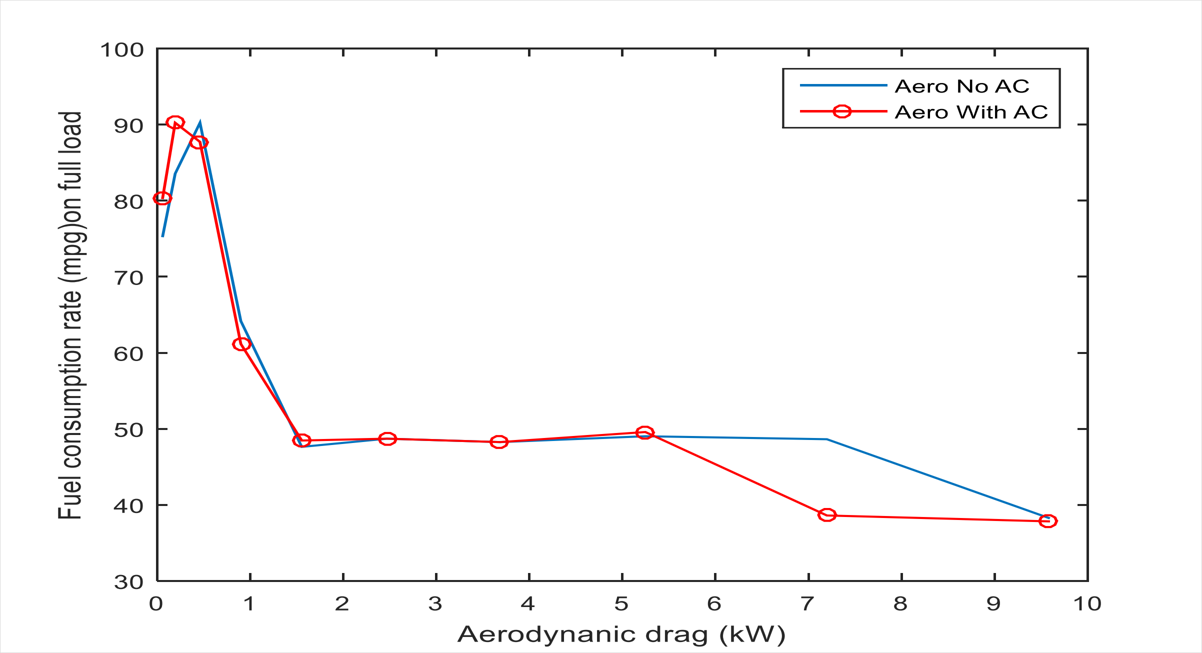


Figure 4.3 Variation of Fuel Consumption Rate Against Aerodynamic Drag

Fuel economy as the reciprocal of fuel consumption rate is shown in figure 4.4. It shows that fuel economy can be maximized by seeking alternative ways of lowering the fuel consumption rate in automobiles.

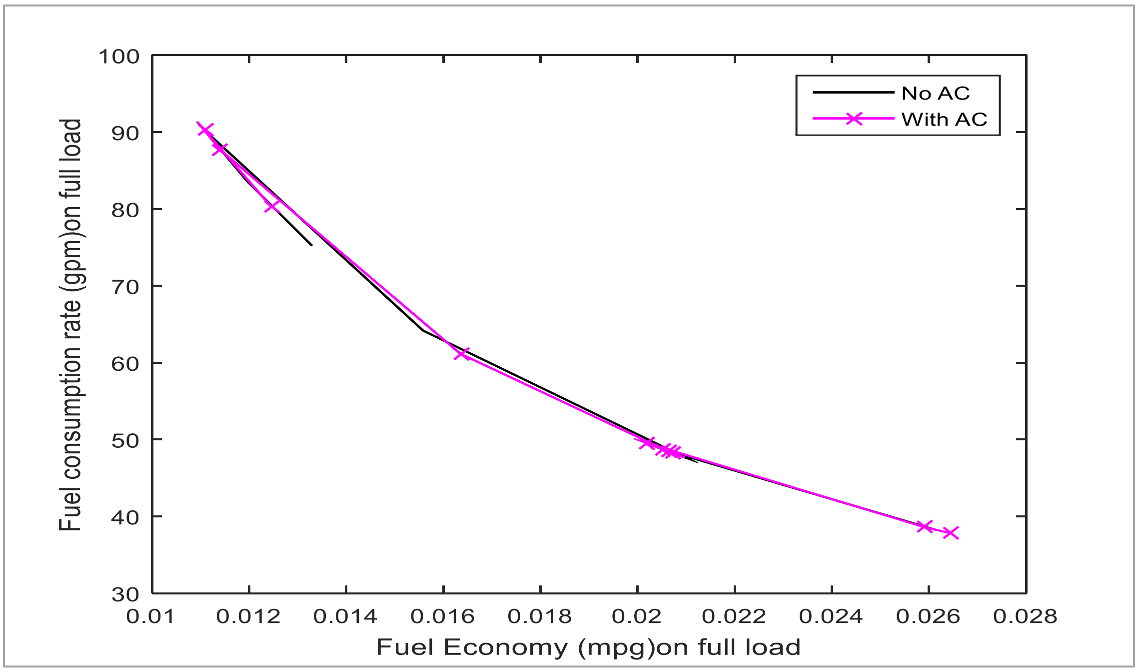


Figure 4.4 Fuel Consumption Rate and Fuel Economy Behavior

Fuel consumption increases with vehicle speed to a maximum: 91 (gpm) with AC and 89(gpm) no AC conditions as shown in figure 4.5. It also shows that at high vehicle speed above 70 km/hr. up to90km/hr., fuel consumption rate for both AC and no AC test conditions are the same. This means that AC cars can consume same amount of fuel as that of no AC cars on full load at high speeds, but at higher speed above 100km/hr., fuel consumption for non- AC condition becomes high than the AC conditions. Thus, it is advisable to turn on AC at the speed above 100km/hr.

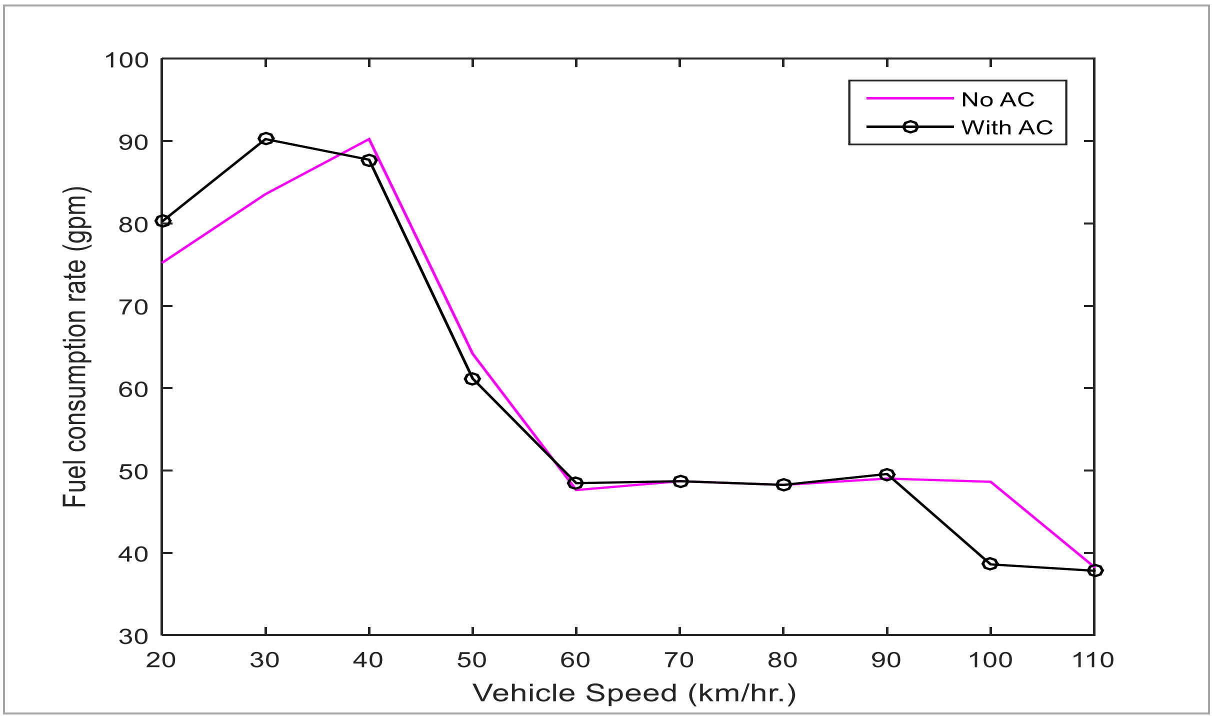


Figure 4.5 Variation of Fuel Consumption Rate Against Vehicle Speed

Engine output data is shown in table 4.4. it shows the fuel consumption rate G* (gpm) for both AC and no AC test conditions.

The fuel energy utilization and the losses the engine will have to incur is expressed in percentage as shown in figure 4.6.

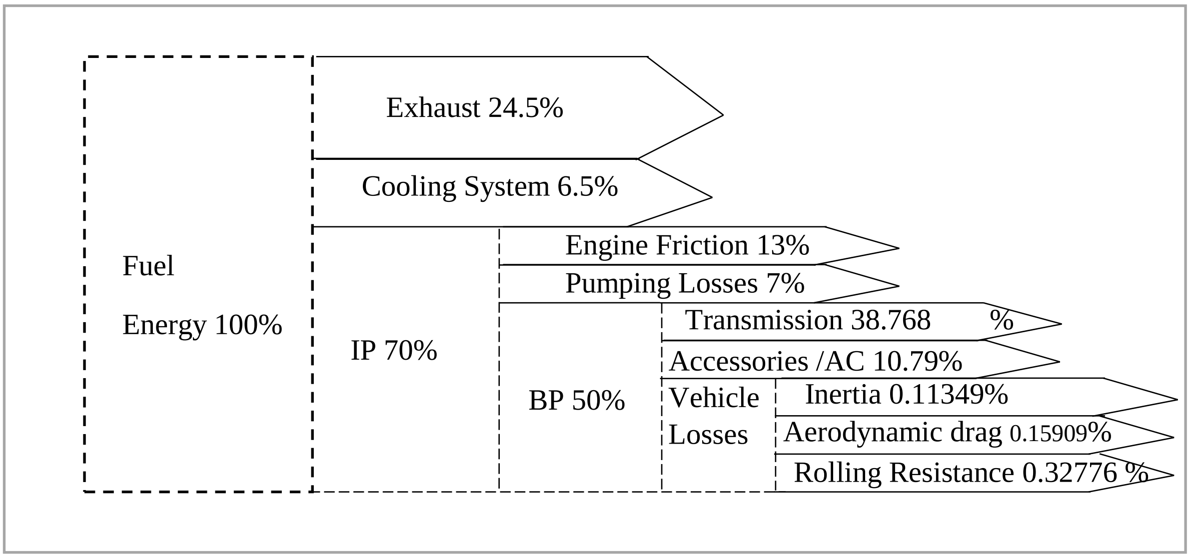


Figure 4.6 Fuel-energy use in automobiles for the present work. Adopted from the National Research Council (1992) and Shete (2015)

As shown in figure 4.7, the compressor load increases at a lower engine speed with AC condition and more fuel is consumed. But as the speed increases, the compressor load becomes constant and gradually decreases. The fuel consumption rate increases at the initial but decreases as the speed increases.

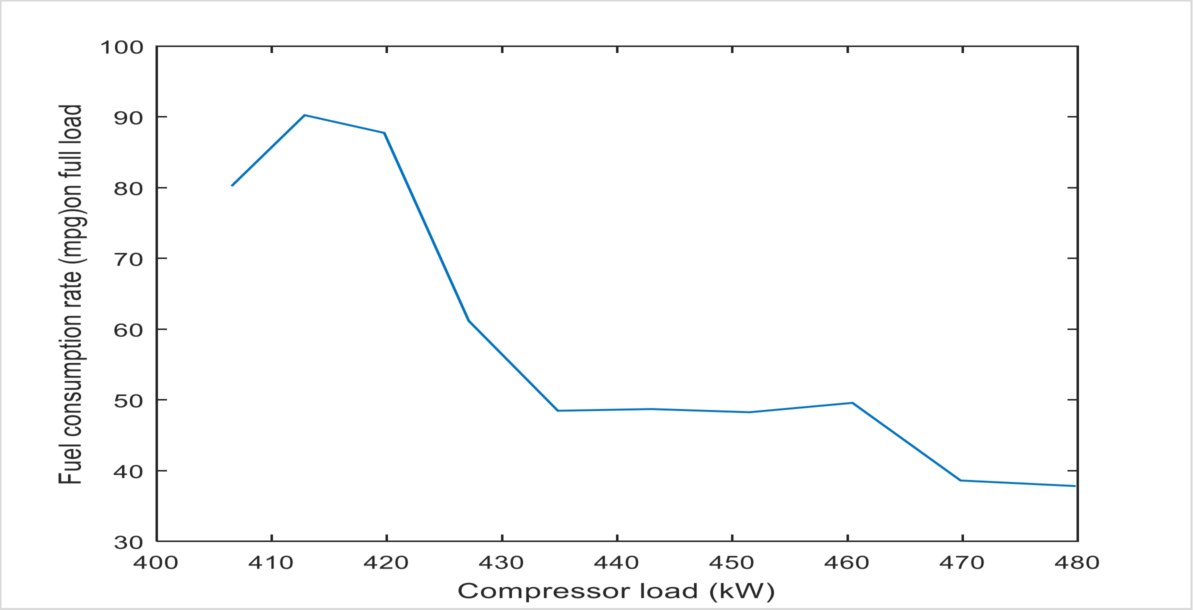


Figure 4.7 Fuel consumption rate against compressor load

RECOMMENDATION

Thermal load on the air conditioning unit can be reduced by using advanced window glazing and ventilation, seat based climate control, recirculated air and by the application of alternative cabin cooling techniques since these techniques can significantly reduce AC load and hence minimize fuel consumption rate.

However, Air conditioning systems contributes extra CO_2 emissions by means of heat rejection into the environment thereby increasing the noise level of the environment via temperature and solar irradiation. Therefore, researchers in this area should give proper attention to air conditioning unit designs and also seek alternative ways of minimizing general engine exhaust in order to enhance a friendly environment.

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APPENDIX A

MATLAB PROGRAM

```
%% MATLAB PROGRAM TO DETERMINE THE FUEL CONSUMPTION RATE OF A full load MOVING VEHICLE
% for
% (1) NO AC
% (2) WITH AC
'======================================================================';
clear, clc
format shortG
%%
% Data
LHV=12.06;         % LHV kWh/kg Gasoline (liquid Petrol)=12.06*3600kJ/kg
Da=0.4064;         % WHEEL DIAMETER (m)=16(inch)
J=(pi*Da^4)/32;    %POLAR MOMENT OF INERTIA
ro=0.012;          % tire rolling resistance m
m=177.93;          % asuumed mass of car kg
g=10;              %acceleration due to gravity
R=ro*m*g;          % Wheel rolling resistance
rw=Da/2;           %E ffective rolling radius
CD=3;A=1.82;den=1.23;  %den=density of air
VV=[0 20 30 40 50 60 70 80 90 100 110]; %Measured Vehicle speed m/s
V=VV*(1000/3600);
D=(CD*A*den.*V.^2)/2; % aerodynamic drag
a=8.31;            % Car acceleration m/s^2
CD=0.3;            %Drag coefficient%
CR=1.2;            %Coefficient of rolling resisitance
sig=0.5;           %Assumed
%Air Resistance  (kW)
    Pair=(den*CD*A.*V.^3/2)/1000;
% Rolling Resistance (kW)
    Ptire=CR*m*g.*V/1000;
% Inertia (kW)
    Pinertia=sig*m*a*V/1000;
%% Accessories
% A Metabolic load
    M1=60;H1=1.7;%(m)& (kg)
    M2=80;H2=1.6;
    A1=0.202*(M1^0.425)*(H1^0.725);
    A2=0.202*(M2^0.425)*(H2^0.725);
    Qm=(M1*A1 + M2*A2)/1000;% (kW)
% B Radiation Load
% Net Radiation load
    theta=45;
    theta2=theta;
    AA= 0.5;  %Assumed constant
    B=0.2;    %assumed constant
    Idir=AA/(exp(B/sind(theta2)));
    C=0.5;    %Assumed
    Idif=C*Idir*((1+cosd(theta))/2);
    deng=0.02;
    Iref=deng*(Idir+Idif)*((1-cosd(theta))/2);
    As=4;% Assumed surface area of car
    absopt=0.4;% absorptivity
    Qrad_net=As*absopt*(Idir*cosd(theta) + Iref + Iref)/1000;
%C  Abient load
    h=0.6+6.648*sqrt(V);
    lamda=0.01;%m
```

```
    kappa=0.2;
    Rth=1./h + lamda/kappa + 1./h;
    U=1./Rth;% Overall heat transfer coefficient
    Ts=30+237;
    Ti=23+273;
    To=273+27;%linspace(21+273,30+273,10);
    Qamb=As*U.*(To-(2*Ts)-Ti)/1000;
%D Engine load
Qeng=0;
%E Ventilation load
P=1;%bar
Ps=0.02808;%bar
Phil=0.78;% relative humidity
mven=0.1;%ventilation mass flow rate (kg/s)
    w=(0.62198*Phil*Ps)/((100*P)-(Phil*Ps));
    eo=1006*To + (2.501e6 +1770*To)*w;
    ei=1006*Ti + (2.501e6 +1770*Ti)*w;
    Qven=mven*(eo-ei); %kW
%Total air conditioning load
Q_AC=abs(Qrad_net+Qamb+Qeng+Qven+Qm);

%%Engine Parameters
N= [700 1200 2000 2880 2560 2280 2720 3080 3520 3880
3360];%linspace(3000,5800,10);           % ENGINE SPEED RPM
Nac= [840 1280 2160 2800 2440 2320 2720 3080 3560 3080 3320];
T=240;                % TORQUE Nm
BP=2*pi*N*T/1000;     % BRAKE POWER kW
BPac=2*pi*Nac*T/1000; % BRAKE POWER kW
EFFM=0.6;             %Assumed mechanical efficiency
FP=(0.13*BP);         % friction loss be pf
PL=0.05*BP;           %Pumping loss
IP=BPac+FP+PL;        % INDICATED POWER
FTR=R+D+(m+4*(J/rw^2))*a; %tractive force
%S=100;                  %Total distance traveled during (m) the test
alpha=0.061;          %alphs
beta= 0.00314;        %Beta
gama=0.74;            %Assumed
ETR_S=m*(alpha*ro+ beta*(CD*A/m)+ gama*(1+4*(J/(m*rw^2)))); %Tractive
energy/distance
nb=0.70;             % Assumed brake thermal efficiency
bsfc=(nb.*LHV)./3600;
mf=BP.*bsfc;            %Mass flow rate of fuel
mfac=BPac.*bsfc;        %kg/hr.
denf=719.7;             %density of fuel kg/m^3
gg =(bsfc.*BP)/denf;   %fuel consumption rate m^3/h
ggac=(bsfc.*BPac)/denf;%With AC fuel consumption rate m^3/h
% s=linspace(100,200,10); %distance traveled
G=(264.172*gg)./(60*0.0000621371*V); %fuel consumption rate (gpm)
Gac=(264.172*ggac)./(60*0.0000621371*V);
GG=1./G;%Fuel economy (mpg)
GGac=1./Gac; % With AC
%% TOTAL LOAD
QT = Q_AC + Pair + Ptire + Pinertia;%With AC
QT1 = Pair + Ptire + Pinertia; %No AC
%% plot
    figure (1),
    plot(VV,G,'b'),hold on
    plot(VV,Gac,'r')
    xlabel('Speed (km/hr.)')
    ylabel('Fuel consumption rate (gpm)')
    legend('No AC','With AC')
```

```
    figure (2),plot(BP,G,'m'),hold on
    plot(BPac,Gac,'r')
    xlabel('Brake Power(kW)')
    ylabel('Fuel consumption rate (gpm).')
    legend('No AC','With AC')
    figure (3), plot(N,QT1), hold on
    plot (Nac, Q_AC,'r')
    xlabel('Speed RPM')
    ylabel('Heat Load (kW)')
    legend('No AC,','With AC')
    figure(4),plot(GG,G), hold on
    plot(GGac,Gac)
    xlabel('Fuel Economy (mpg)on full load')
    ylabel('Fuel consumption rate (gpm)on full load')
    legend('No AC','With AC')
    figure (5)
    plot(QT1,GG),hold on
    plot(QT,(GGac))
    ylabel('Fuel consumption rate (mpg)on full load')
    xlabel('Total load load (kW)')
    legend('No AC','With AC')
%     figure(5)
%     plot(V,BP./V)
% Percentage values
Ptransm=BP-(Pair+Ptire+Pinertia+Q_AC);
Pair2=((Pair./BP)*100)';
Ptire2=((Ptire./BP)*100)';
Pinertia2=((Pinertia./BP)*100)';
Pt=((Ptransm./BP)*100)';
Qac=abs((Q_AC./BP)*100)';


%%
%Results

disp('       =============================No
AC================================================')
disp('        IP(kW)        BP(kW)     BPac (kW)  mf(kg/kWh)    g(m^3/h)
G(gpm)  Gac(gpm)')
disp([IP' BP' BPac'  mf' gg'  G'  Gac'])
disp('
===========================================================================
========')
disp('Transmission(%)     AC load(%)     Aero(%)   Rolling Resis(%)
Inertia (%)')
disp([Pt/2 Qac/2 Pair2/2 Ptire2/2  Pinertia2/2])
```