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ESTIMATION OF AUTOMOBILE COOLING LOADS FOR AIR CONDITIONING SYSTEM DESIGN

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ABSTRACT

Automobile air conditioning by vapour compression refrigeration system increases the amount of energy consumed. A vapour absorption refrigeration system which will utilize vehicle engine waste heat either from the exhaust gases or from the cooling system thereby reducing energy consumption is being proposed. Estimation of automobile cooling load was undertaken as part of an investigation to determine the suitability of a vapour absorption refrigeration system for automobile air conditioning system. Heat balance method was used to estimate the various thermal loads, radiation loads and cooling load of a car with four passengers in Lagos, Nigeria. The value of the cooling load obtained was 2.35kW and it was estimated under the hot weather condition of the sun to give a fairly accurate result for design purpose.

1. INTRODUCTION

The first automobile air conditioning system introduced by Packer in 1939 used a vapour compression refrigeration system (Tiwari and Parishwad, 2012). In the last 70 years, automobile air conditioning systems have undergone gradual and continual improvements in performance and efficiency as a result of improvements in the individual components (McLaughlin, 2005).

Alternative systems powered by heat which can be used in automobiles are solid adsorption cooling systems, thermo acoustic refrigeration, active magnetic regenerator system, thermo electric devices and vapour absorption refrigeration (VAR) systems (Zoontjens et al., 2005).

Air conditioning of a vehicle can be done by vapour compression refrigeration system (VCRS) or vapour absorption refrigeration system (VARS). Using a VARS will lead the automobile manufacturing industries into new technological ideas. For example, the compressor in a VCRS needs a pulley and a belt drive attached to the engine shaft before it can work, using a VARS eliminates that. A thorough cooling load calculation helps to minimize system over-sizing and the associated negative effects on thermal comfort and humidity control.

The aim of this study was to properly estimate the various heat load of the air conditioning system of a vehicle.

2. MATERIALS AND METHOD

2.1. Materials

A KIA PICANTO 2013 model with three cylinder engine was used for this study. The material properties used were in accordance with Fayazbakhsh and Bahrami (2013) and is shown in Table 1.

Table 1: Material properties

Property	Glass	Vehicle body
Conductivity (W/mk)	1.05	0.2
Density (kg/m^3)	2500	1500
Transmisivity	0.5	0
Absorptivity	0.3	0.4
Specific heat (J/kgK)	840	1000
Thickness (mm)	3	10

2.2. Weather Data

To effectively carry out this analysis weather data of the selected area of Lagos, Nigeria (Lagos State was choosing because the company that was used as a case study is situated at Lagos State, Kia picanto model was used because it was the model that was readily available and it carries only four persons), (Latitude: 6.5^o N, Longitude: 3.4 ^{o} E) were obtained. Dry bulb temperature (DBT) was obtained as 34^{o}C, while wet bulb temperature (WBT) was obtained as 26 ^{o}C. It is desired that the temperature be maintained at 24^{o}C DBT and relative humidity of 50%.

2.3. Heat Balance Method

Heat balance method (HBM) was used to analyze the various thermal loads and radiation loads of the vehicle (Fayazbakhsh and Bahrami, 2013). The total thermal load on the vehicle cabin is given by Equation (1). The recommended heat-gain values for passengers and design specifications for the vehicle model are presented in Tables 2 and 3 respectively.

$$Q_T = Q_P + Q_{Dir} + Q_{Ref} + Q_{Amb} + Q_{Exh} + Q_{Eng} \quad (1)$$

Where Q_T = Total load, Q_{Amb} = Ambient load,Q_{Dir} = Direct radiation load, Q_{Eng} = Engine load, Q_{Exh} = Exhaust load, Q_{REF} = Reflected load and Q_P= Occupant load.

2.3.1. Heat gain due to vehicle occupants

Heat gained due to vehicle occupants was determined for the driver and three passengers using Table 2. The sensible heat gain due to occupants (Q_{SCP}) and latent heat gain due to occupants (Q_{LCP}) were obtained from Equations (2) and (3) respectively.

$$Q_{SCP} = Q_{PS} \times Q_{AP} \quad (2)$$

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$$Q_{LCP} = Q_{AP} - Q_{SCP} \quad (3)$$

Where Q_{PS} is the percentage sensible heat and Q_{AP} is the adjusted male/female passenger load.
Total heat gain due to occupants is given as:

$$Q_p = Q_{SCP} + Q_{LCP} \quad (4)$$

Total Sensible Heat is given as:

$$Q_{SC} = Q_{SCP} + Q_{Dir} + Q_{Ref} + Q_{Amb} + Q_{Exh} + Q_{Eng} \quad (5)$$

Sensible heat factor is given as:

$$SHF = \frac{Q_{SC}}{Q_T} \quad (6)$$

Table 2: Recommended heat-gain values for passengers and operators (ASHRAE, 2001)

Application	Adult male (w/person)	Adjusted male/female	% Sensible heat
Inter-city and commuter car, seated	132	117	61
Inter-city and commuter car, standing	139	132	56
Subway and Light Rail Vehicle, Seated	139	132	56
Subway and Light Rail Vehicle, standing	161	147	50
Operators / drivers	132	132	56

Table 3: Kia Picanto 2013 model design specifications

S/N	Parameters	Symbol	Specification
1	Cabin surface area	S_C	3.8m^2
2	Surface area exposed to exhaust pipe	S_{EXH}	0.8774m^2
3	Surface area of cabin exposed to the engine	S_{ENG}	1.18m^2
4	Surface area of roof	S_R	0.963m^2
5	Side window area	S_{SW}	1.44m^2
6	Minimum speed of the vehicle	V	5.56m^2
7	Petrol flow rate	V_P	0.12kg/s
8	Windscreen area	S_W	0.72m^2

2.3.2. Thermal and radiation loads

i. Ambient load

The ambient load which is the thermal load due to temperature difference between the ambient and cabin air is given by Equation (7).

$$Q_{Amb} = \sum_{passengers} S_c \times U_1(T_s - T_c) \quad (7)$$

Where S_c = Cabin surface area, T_s = Temperature of surface and T_c = Temperature of cabin

ii Exhaust load

Exhaust load is the load due to the high temperature of exhaust gases. The exhaust load was calculated using Equation (8).

$$Q_{Exh} = S_{Exh} \times U_2(T_{Exh} - T_c) \quad (8)$$

S_{Exh}= Surface area of exhaust, T_{Exh} = Temperature of exhaust

iii. Engine load

This is the load due to the high temperature of the engine. It was calculated from Equation (9).

$$Q_{Eng} = S_{Eng} \times U_3\left(T_{Eng} - T_c\right) \quad (9)$$

Where:

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$$U_1 = \frac{1}{R_1} \tag{10}$$

$$R_1 = \frac{1}{h_o} + \frac{t}{K} + \frac{1}{h_i} \tag{11}$$

R_1 = Net thermal resistance for a unit surface area,
h_o = Outside convection coefficient.
h_i = inside convection coefficient
K = Surface thermal conductivity.
$t =$ Surface element thickness.
According to Fayazbakhsh and Bahrami (2013), the outside convection coefficient is given as:

$$h_o = 0.6 + 6.64\sqrt{V} \tag{12}$$

where V= is the minimum vehicle speed

$$T_{Exh} = 0.138RPM - 17 \tag{13}$$

$$T_{Eng} = -2\times10^{-6}\,RPM + 0.0355\,RPM + 77.5 \;= 1500 \tag{14}$$

According to Ding and Zito (2001), h_i = 8.3 W/m^{2}K

$U_2 = \frac{1}{R_2}$, $R_2 = \frac{t}{K} + \frac{1}{h_i}$, $U_3 = \frac{1}{R_3}$, $R_3 = \frac{t}{K} + \frac{1}{h_i}$

iv Radiation load

Radiation loads are obtained for both the glass surfaces and other surfaces of the car. Equations 15 - 28 are the commonly used models for air conditioning calculations as suggested by ASHRAE (2001).

Direct radiation load

$$Q_{Dir} = \sum\nolimits_{Surface} S\ x\ \tau\ x\ I_{DN}\cos\theta \tag{15}$$

$$I_{DN} = \frac{A}{exp\left(\frac{B}{\sin\beta}\right)} \tag{16}$$

A is the apparent solar irradiation taken as 1230W/m^2 for months of January to December and 1080W/m^2 for mid-summer while B is the atmospheric extinction coefficient which has a values of 0.142 for the months of January to December. The declination, D is given as:

$$D = 23.47sin\frac{360\ (284+N)}{365} \tag{17}$$

Where day of the year starting from January, N= 21 for January 21, 2015 at Lagos, Nigeria
Local Solar Time, LST = 14.00 (2.00pm)

$$\text{Hour angle, } h = 15(\text{LST} - 12) \tag{18}$$

Tilt angle, ε = 0 (horizontal surface) and 90^o (vertical surface), Wall azimuth, ε= 0^o (southern surface), ε= 90^o (western surface) and ε= 180^o (northern surface)
The altitude angle, is given as:

$$\beta = \sin^{-1}(\cos(L)cos\ (h)\cos(d) + \sin(L)\sin(d)) \tag{19}$$

L = latitude, h = hour angle, d = angle of declination, $\tau =$ roof element transmissivity
Since surface element transmissivity of roof, wall and floor is zero, the direct radiation through those elements becomes zero. The incident angle is given as:

$$\theta = cos^{-1}(\sin(\beta)\cos(\varepsilon) + \cos(\beta)\cos(\alpha)\sin(\varepsilon)) \tag{20}$$

The azimuth angle is given as:

$$\alpha = [\pi - (\gamma + \varepsilon)]\ \times F \tag{21}$$

The solar azimuth is given as:

$$\gamma = \sin^{-1}\left(\frac{\cos(d)\times\sin(h)}{\cos(\beta)}\right) \quad (22)$$

F = 1 for afternoon, where $\pi = 180°$

***Diffuse radiation load* (Q_{DIF})**

$$Q_{Dif} = \sum_{Surface} S_{SW} \times \tau \times I_D \quad (23)$$

$$I_D = C \times I_D \times Fws \ (W/m^2) \quad (24)$$

Where I_D = diffuse radiation and C, a constant = 0.058 for January.
The view factor is given as:

$$\text{Fws} = \frac{1+cos\varepsilon}{2} \quad (25)$$

Ɛ = tilt angle and its value will be 0 for horizontal surfaces and 90° for vertical surfaces.

Reflected radiation loads

$$Q_{Ref} = \sum_{Surface} S_{SW} \times \tau \times I_{REF} \quad (26)$$

$$I_{Ref} = (I_{DIR} + I_D) \times P_G \times \text{FwG} \quad (27)$$

Ground reflectivity factor, $P_G = 0.2$
Angle factor is given as:

$$(\text{FwG}) = \frac{1-cos\varepsilon}{2} \quad (28)$$

FwG = 0, for horizontal surfaces and FwG = 0.5, for vertical surfaces.

3. RESULTS AND DISCUSSION

The values obtained for different types of load which make the total cooling load of the vehicle are presented in Table 4. The total heat load, Q_T was obtained as 2352.21W and the sensible heat factor, an important parameter for determining cooling coil capacity was 0.92. The various heat loads were gotten using various parameters which were gotten under the hot weather condition of the sun to get a fairly accurate heat load estimation so as to improve on the design and working conditions of the vehicle air conditioning system.

Table 4: Total cooling load

Q_{SCP}	288.O3
Q_{LCP}	194.97
Q_{AMB}	163.78
Q_{EXH}	705 83
Q_{ENG}	852.04
Q_{DIR}	115.2
Q_{DIF}	7.11
Q_{REF}	25.25
Q_T	2352.21W

4. CONCLUSION

The total heat load of the vehicle was estimated to be 2352.21W and as such the power to efficiently design and run the air conditioning system of the vehicle should be higher than the estimated heat load of the vehicle. This will help in proper and efficient design of the car air conditioning system.

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5. CONFLICT OF INTEREST

There is no conflict of interest associated with this work.

REFERENCES

ASHRAE (2001). Handbook of Fundamentals, American Society of Heating, Refrigerating and Air Conditioning, Atlanta, GA.

Ding, Y. and Zito, R. (2001). *Cabin heat transfer and air conditioning capacity* (No. 2001-01-0284). SAE Technical Paper.

Fayazbakhsh, M. A. and Bahrami, M. (2013). *Comprehensive modeling of vehicle air conditioning loads using heat balance method* (No. 2013-01-1507). SAE Technical Paper.

McLaughlin, S. (2005). An Alternative Refrigeration System for Automotive Applications. M.S. Thesis, Mechanical Engineering Department, Mississippi State University.

Tiwari, H. and Parishwad, G. V. (2012). Adsorption refrigeration system for cabin cooling of trucks. *International journal of emerging technology and advanced engineering*, *2*(10), pp. 337-342.

Zoontjens, L., Howard, C., Zander, A. and Cazzolato, B. (2005). Feasibility Study of an Automotive Thermo acoustics Refrigerator. In: Proceedings of ACOUSTICS 9-11 November 2005, Busselton, Western Australia.