International Journal of Engineering & Technology
CENTRE OF PROFESSIONAL RESEARCH PUBLICATIONS
ISSN 2049-3444

International Journal of Engineering and Technology Volume 2 No. 6, June, 2012

Effect of Radiator Fins Blockage by Clay Soil on the Engine Cooling Temperature

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ABSTRACT

The paper studies the effect of clay soil blocking the heat transfer area of the radiator on the engine coolant temperature by developing a mathematical model. The results of the study indicated that the percentage area covered resulted in a proportional increase of the inlet and outlet temperatures of the coolant in the radiator. The mathematically model developed also predicted the experimental data very well. Regression analysis showed that for every 10% increase in the area of the radiator covered with clay soil resulted in the corresponding increase of about 21 ^{o}C of the outlet temperature of the radiator coolant. A t-test showed that no significant difference between the experiment and the mathematical model prediction. Thus, irrespective of the type of material that blocks the radiator surface area, the coolant temperature rises in proportion with the area covered by the radiator.

Keywords: *clay, radiator, fins, tube, engine cooling, temperature*

NOMENCLATURE

$A_{fr.t}$	coolant tube frontal area, m^2
F_l	fin length, m
A_f	fin heat transfer area, m^2
A_c	total heat transfer area on coolant side, m^2
$A_{p,c}$	total coolant pass area, m^2
Re	reynolds number, dimensionless
Nu	nusselt number, dimensionless
Pr	prandtl number, dimensionless
ρ	density of the fluid, kg/m^3
V	velocity of the fluid, m/s
L	length of tube, m
k	thermal conductivity of the fluid, W/m.k
C_p	specific heat capacity of the fluid, kJ/kg.K
h	heat transfer coefficient, J/m^2.K
α	thermal diffusivity of the fluid, m^2/s
μ	viscosity of the fluid, centipoise
υ	kinematic viscosity of the fluid, m^2/s
U	overall heat transfer coefficient
h	heat transfer coefficient, W/m^2 ^{o}C
n_o	total surface efficiency of an extended fin surface
R_f	fouling factor (W/m^2 ^{o}C)$^{-1}$
$A_{fr,r}$	radiator core frontal area m^2
A_a	total heat transfer area on air side m^2
A_c	total heat transfer area on coolant side m^2
h_a	heat transfer coefficient on air side ,W m^{-2} K^{-1}
h_c	heat transfer coefficient on coolant side, W m^{-2} K^{-1}
L_h	louver height, m
L_l	louver length, m
F_h	Fin height, m
D_h	hydraulic diameter, m

Y_l	coolant tube length, m
ε	The radiator thermal efficiency
C_a	heat capacity of air
C_c	heat capacity of coolant
T_{ci}	Coolant inlet temperature, ^oC
T_{co}	Coolant outlet temperature, ^oC
T_{ao}	Air outlet temperature, ^oC
T_{ai}	Air inlet temperature, ^oC

Subscript

a	air side
c	coolant side
w	coolant wall
h	height
ci	coolant inlet temperature
co	air outlet temperature
ai	air inlet temperature

1. INTRODUCTION

When internal combustion engine is operating at full throttle, the maximum temperature reached by the burning gas may be as high as 2700 ^oC [1]. The heat transfer that occurs within an engine is extremely important for its proper operation. About 30 - 40% of the total chemical energy that enters an engine in the form of fuel is converted to useful crankshaft work. Most of the materials used for engine components may not be able to withstand the high temperature reached during the combustion process and would quickly fail if they are not properly cooled. Appropriate temperature distribution is highly critical in the correct functioning of the engine and preservation of all the components.

Engine cooling system using radiator as its cooling medium is one method of dissipating heat from the engine. The most familiar example of an air-to-liquid heat exchanger is a car radiator. The coolant flowing in the engine picks up heat from the engine block and carries it to the radiator. From the radiator, the hot coolant flows into the tubes side of the radiator. The relatively cool air flowing over the outside of the tubes picks up the heat, reducing the temperature of the coolant. Due to poor heat conductivity of air, the heat transfer area between the surface of the radiator and the air must be maximized. This is done by using fins on the outside of the tubes. The fins improve the efficiency of radiator and are commonly found on most liquid-to-air radiators [2].

The radiator is normally placed in front of the engine to enhance flow of air in the cooling of the engine. The radiator consists of two tanks and a core or matrix from which heat is radiated from the coolant to the air. The total heat dissipated from the engine through the coolant to the air depends on the total fin area on the radiator surface [3]. The louvered fin surfaces are very common for engine cooling, air-conditioning apparatus, aircraft, and air cooler.

The popularity of the louver surface may attribute to its superior heat transfer performance by continuous renewing the boundary layer of the air flow. For automotive application, such as radiators, condensers, and evaporators, the louver fins were generally brazed (or soldered, mechanically expanded) to a flat, extruded tube, with a cross section of several independent passages, and formed into a serpentine or a parallel flow geometry.

There are many researchers that have looked at the air-side performance of louver fins exchanger geometry such as Webb and Jung [4] Davenport [5], Achaichia and Cowell [6], Rugh et al. [7], Tanaka et al. [8], Chang and Wang [9] who presents considerable amount of air-side data for louver fin geometry. Based on the test results of 91 samples, Chang and Wang [9] and Chang et al. [10] developed a general heat transfer and friction correlation for air side louver fins.

Lin et al. [11] presented an interesting study of specific dissipation (SD) sensibility to radiator boundary conditions (air and coolant inlet temperatures and mass flows). Their conclusions were assessed by numerical and experimental work. Juger and Crook [12] reported an experimental testing on two radiators of the same flow area but with the tubes in vertical or horizontal position, therefore studying the influence of tube length vs. number of parallel tubes. They carried out this analysis for three different coolant fluids. Gollin and Bjork [13] experimentally compared the performance of five commercial radiators working with water and five aqueous glycol mixtures. Chen et al. [14] experimentally analysed a sample radiator and developed regression equations of heat dissipation rate, coolant pressure drop and air pressure drop in function of the boundary conditions. Ganga Charyulu et al. [15] presented a numerical analysis (based on ε- NTU method) of a radiator in a diesel engine, centring the attention on the influence of fin and tube materials and the boundary conditions on both fluids.

Interesting as these results may be no study has been conducted on the effect of clay soil or dirt blocking part of the heat transfer surface area the radiator. The vehicle population in Ghana stands at about 1, 500, 000 cars (Driver and Vehicle Licensing Authority, [16] with an annual importation of about 100 000 vehicles. Most of these cars end up developing engine related problems a year or two later. This may be due to the nature of roads in the country. About 60% of roads in Ghana are untarred [17]. It is envisaged that clay particles from untarred roads end up blocking parts of the surface of the radiator. This study, thus, takes a critical look at the effect of clay soil blocking parts of the surface area of the radiator and its corresponding effect on the performance of the engine temperature.

2. EXPERIMENTAL SETUP

The engine used for the experiments was four-cylinder four stroke engine. The engine was water cooled engine with a tank capacity of 5.5 litres. It was petrol engine which has a compression ratio of 9.8 to 1 and operates on the Otto-cycle.

Table .Shows the technical information/specification of the engine used for the experiment.

Baseline specification of the engine used	
Year	1996
Manufacturer	Nissan
Model range	Primera
Engine Capacity	1597cc
Engine Type	GA 16
Number of cylinders	4/DOHC
Engine Firing Order	1-3-4-2
Compression Ratio	9.8:1
Cooling System capacity	5.5 litres of water
Thermostat opening	76.5c
Radiator Pressure	0.78-0.98 bar
Ignition timing basic BTDC	Engine/rpm 10+2/625
Idling Speed	750+50 rpm

The radiator of the engine was a General Motors' serpentin-fin cross-flow type of size 65 mm × 35 mm in length and breadth respectively. The numbers of tubes were one row of 66 tubes with a thickness of 2 mm. The fins were copper made with a thickness of 0.5 mm, a height of 16 mm and spaced 3 mm apart.

The inlet and the outlet temperatures of water, and the external surface of the radiator were measured by a set of K-type thermocouples being calibrated with ±0.1 ^{0}C accuracy. The thermocouple was manufactured by Cola-Parma Instrument Company with model number 8110-10. A mercury-in bulb thermometer was used to measure the ambient temperature.

The engine was fitted with two thermocouple probes to read the inlet and outlet temperature of the water in and out of the radiator. The radiator was first cleaned to get rid of all particles trapped in the fins and on the engine. The rotational speeds of the engine and the fan speed were held constant throughout the experiment. The effective heat transfer area of the radiator was divided into ten (10) equal parts representing 10% of the effective heat transfer area. The effective heat transfer area was then partially covered with clay soil to the required percentage before running the engine. The engine was run for about fifteen (15) minutes to attain stable conditions before readings were taken.

For each set-up, three readings were taken at intervals of two minutes. After each experiment, the radiator was removed and washed to get rid of all the sand particles and then dried in open air. Each experiment was repeated three times according to the random sample technique adopted.

2.1 Mathematical Model

In order to validate the result, a mathematical model was developed using Effectiveness – Number of Heat Transfer Units (ε-NTU) method and algorithm was developed using a Matlab to model the radiator. The model was developed to predict the outlet temperature of the coolant from the radiator given the dimensions of the radiator. The heat transfer process in the radiator was modelled as a forced convective heat transfer operation. The following equations forms Lin [17] were modified to calculate the heat transfer rate in the radiator. The parameters for calculating the heat transfer area are given in Table 2 below. The relevant equations are given from equation 1 to 25.

Table 2: Baseline parameters for the radiator heat transfer area

Core height	B_H
Core width	B_W
Core thickness	B_T
Number of coolant tubes in core depth dimension	N_r
Number of coolant tubes in one row	N_{ct}
Number of profiles	N_p
Number of fins in one meter	$N_{f/m}$
Louvre pitch	L_p
Louvre length	L_l
Fin thickness	F_t
Fin height	F_h
Fin pitch	F_p
Fin end radius	R_f
Angle of fin	α_f
Coolant tube length	Y_l
Coolant tube cross section length	Y_{cl}
Coolant tube cross section width	Y_{cw}

Coolant tube thickness	Y_t
Coolant tube pitch	Y_p
Coolant tube end radius	R_t

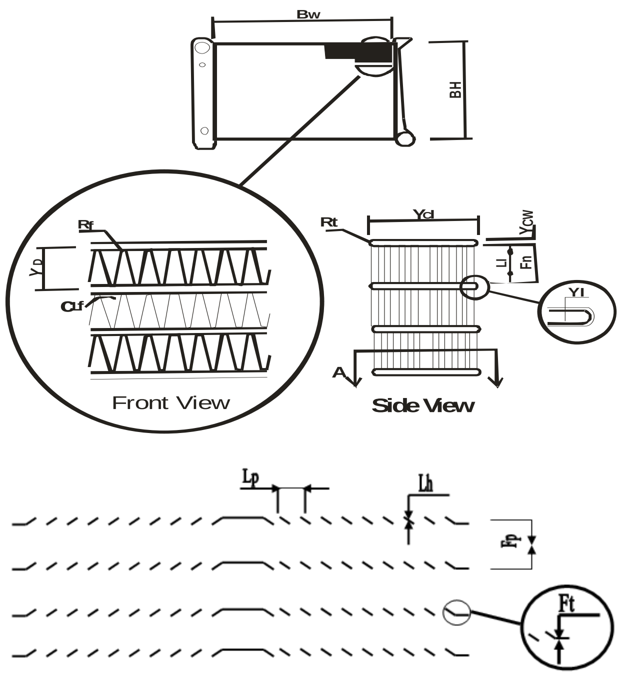

Fig.1.Definitions of the radiator parameters

The fin length, F_l is given by;

$$F_l = \pi R_f + \frac{F_h - 2R_f}{\cos\alpha} \qquad (1)$$

The radiator core frontal area, A_{fr} is given by;

$$A_{fr} = B_H B_w \qquad (2)$$

Coolant tube frontal area, $A_{fr,t}$ is given by;

$$A_{fr.t} = Y_{cw} Y_l N_{ct} \qquad (3)$$

Fin heat transfer area, A_f, is given by;

$$A_f = 2. B_T F_T N_f Y_l N_p \qquad (4)$$

The total heat transfer area on air side, A_a is given by;

$$A_a = A_f + 2N_{ct} Y_l N_f [(Y_{cl} - 2R_t) + (2\pi R_t)] \qquad (5)$$

The total heat transfer area on coolant side, A_c is given by;

$$A_C = [2\pi(R_t - Y_t) + 2(Y_{cl} - 2R_t)] Y_l N_{ct} N_r \qquad (6)$$

The total coolant pass area, $A_{p,c}$ is given by;

$$A_{p,c} = [\pi(R_t - Y_t)^2 + (Y_{cw} - 2Y_t)(Y_{cl} - 2R_t)] N_{ct} N_f \qquad (7)$$

The following dimensionless groups for convective heat transfer were used;

- **Reynolds number**

$$Re = \frac{\rho V L}{\mu} \qquad (8)$$

- **Nusselt number**

$$Nu = \frac{hL}{k} \qquad (9)$$

- **Prandtl number**

$$Pr = \frac{v}{\alpha} = \frac{\mu C_p}{k} \qquad (10)$$

2.1.1 Overall Heat Transfer Coefficient

The overall heat transfer resistance for radiators can be considered to be due to;

Mathematically, it can be defined as;

$$\frac{1}{UA} = \frac{1}{(n_o hA)_a} + R_{fa} + \frac{\Delta x}{(kA)_w} + \frac{1}{(n_o hA)_c} \qquad (11)$$

When all these assumption are taken into effect the corrected equation becomes;

$$\frac{1}{UA_{fr,r}} = \frac{1}{h_a A_a} + \frac{1}{h_c A_c} \qquad (12)$$

The air side heat transfer coefficient was taken from Davenport [18] and is given by;

$$h_a = \left[0.249 Re_{Ll}^{-0.42} L_h^{0.33} \left(\frac{L_h}{F_h}\right)^{1.1} F_h^{0.26}\right] \frac{\rho V_a C_{p,a}}{P_r^{2/3}} \qquad (13)$$

The coolant side Nusselt number was taken from Davenport [18] is given by;

$$Nu_c = 3.66 + \frac{0.0668\left(\frac{D_{h,c}}{Y_1}\right) RePr}{1 + 0.04\left[\left(\frac{D_{h,c}}{Y_1}\right) RePr\right]^{2/3}} \qquad (14)$$

2.1.2 The Ɛ-NTU Method

The heat transfer rate in the radiator is given by;

$$Q = \varepsilon C_{min}(T_{ci} - T_{ai}) \qquad (15)$$

Where C_{min} = minimum heat capacity rate

$$\varepsilon = \frac{Q}{Q_{max}} \qquad (16)$$

ISSN: 2049-3444

The actual heat transfer balance equation at steady state which is defined in terms of energy lost on coolant side and energy gained on the air side is given by;

$$Q = C_c(T_{ci} - T_{co}) = C_a(T_{ao} - T_{ai}) \quad (17)$$

The heat capacity ratio is defined as the product of the mass flow rate and the specific heat of the fluid;

For air: $C_a = m_a x C_{p,a} = A_a \rho_a V_a C_{p,a}$ (19)

For coolant: $C_c = m_c x C_{p,c} = A_c \rho_c V_c C_{p,c}$ (20)

The heat capacity ratio is defined as the ratio of the smaller to the larger capacity rate for the two fluid streams and is expressed as;

$$C_r = \frac{C_{min}}{C_{max}} \quad (21)$$

Where C_{min} is the smaller of C_a and C_c. According to SAE JI393 [19], the minimum capacity rate C_{min} is always on the air side. Hence

$$C_{min} = C_a \text{ and } C_{max} = C_c. \quad (22)$$

It follows therefore that the heat transfer rate is given by;

$$Q = \varepsilon C_a(T_{ci} - T_{ai}) \quad (23)$$

The number of heat transfer units (NTU) is the ratio of overall conductance UA to the smaller capacity rate $C_{min;}$

$$NTU = \frac{UA_{fr,r}}{C_{min}} = \frac{1}{C_{min}} \int_A U.dA_{fr,r} \quad (24)$$

The radiator effectiveness is defined as a function of both the NTU and the C_r by Kays and London [20] and is given by;

$$\varepsilon = 1 - exp\left\{\frac{NTU^{0.22}}{C_r}\left[exp(-C_r.NTU^{0.78}) - 1\right]\right\} \quad (25)$$

Based on the equations outlined above, the model was developed using ***MATLAB*** and run with the algorithm in Fig.3.

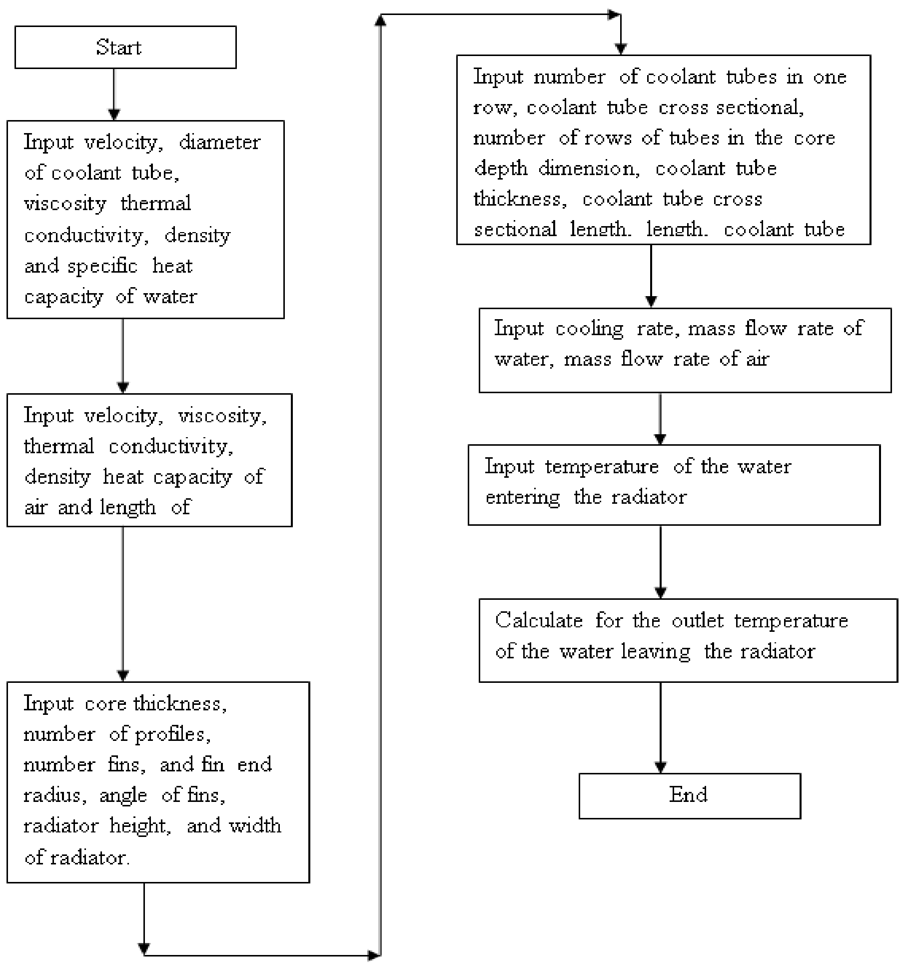

Fig.2. The Algorithm for running the model.

3. RESULTS AND DISCUSSION

3.1 Results Obtained Using Clay as the Cover Material

As shown in Fig. 3, the temperature of coolant from the radiator increased with an increase in the area of the radiator covered. As the area covered increased, there was also a corresponding increase in the temperature of the coolant coming from the engine.

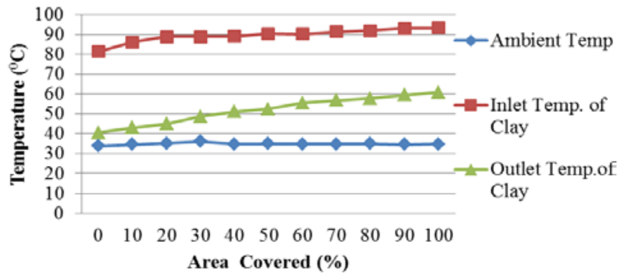

Fig. 3. Graph of Temperature verse Area covered using clay to cover the radiator

The highest outlet coolant temperature was achieved when 100% of the radiator was covered. During the experiment it was noticed that the engine stopped running after a short time when the radiator was completely covered. This was due to the inability of the coolant to take away enough heat from the walls of the engine thereby causing overheating and subsequently forcing the engine to stop running in order to prevent substantial damage to the engine. This is in line with literature because according to Pulkrabek [1] if the heat within the engine is not continuously removed to the required operating temperature there will be substantial damage to most of the engine parts. To avoid that, the engine could possibly stop operating.

At 80% coverage of the radiator, it was observed that the engine idling was not stable, vibrating excessively. The water within the expansion tank began to overflow and the coolant tube was also observed to be very hot.

The graphs shown in Fig. 4, 5, 6 compare the results obtained using clay soil and the mathematical model.

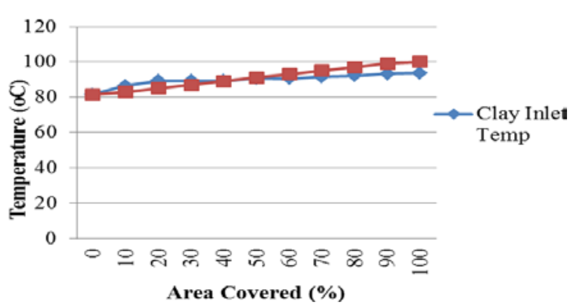

Fig.4. Graph of inlet temperature of water into the radiator against the percentage area covered.

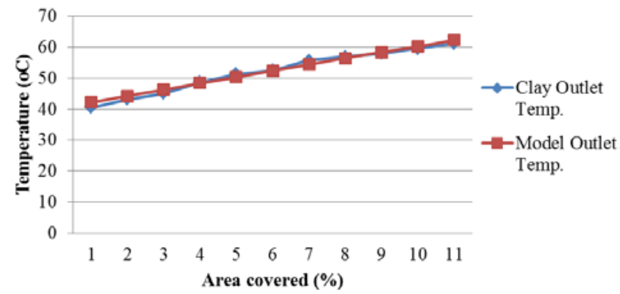

Fig.5. Graph of outlet temperature of water into the radiator against the percentage area covered.

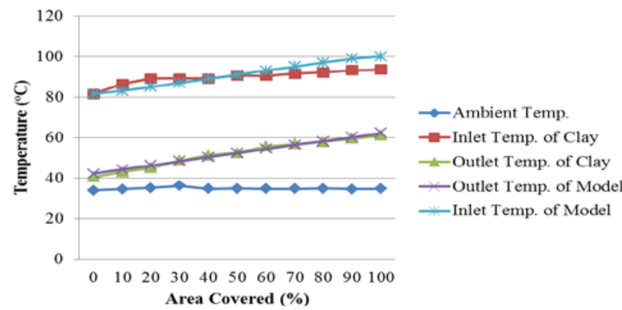

Fig.6. Comparison of model results of temperature of water out and out of the radiator with results obtained with clay.

In the graph (Fig. 4) compared the inlet temperature of the model with clay soil as a covering material whiles Fig.5 compared the outlet temperature of the radiator when covered with clay with the model. Also, Fig.6 compared both the inlet temperature and the outlet temperature of clay soil and the model. It was obvious that in the entire three scenarios, the model predicted the heat transfer phenomenon quite well. In spite of the many assumptions, it was observed that the temperature of the coolant from the radiator was comparable to results obtained for the experiments. The model predicted closely, the results obtained with the outlet temperature as shown in Fig.5. This is perhaps due to the fact that clay was a better coverage material and that it gives a better sticking on the radiator fins. It was generally observed that as the area of the radiator covered was increased, the temperature of the coolant from the radiator also increased considerably. This can be explained by the fact that the effective heat transfer area was reduced, thereby limiting the quantity of air admitted through the radiator for purposes of cooling the coolant. Achaichia and Conwell [6] agree to this as they also used numerical calculations to model a flow through a simplified two-dimensional louver array. They concluded that, the flow efficiency is reducing particularly and the lover angle decreased. In all cases this trend of increasing outlet temperature as the area covered also increased was observed. As shown in Fig.6 the temperature of the ambient air was found to be relatively constant. This therefore did not significantly affect the rate of heat transfer because the air mass was almost constant just as the thermal properties of the air.

Presented in Figures 7.and 8 are the graphical regression relationships between the radiator coolant outlet temperatures and the proportion of the radiator surface area covered with clay and Matlab simulation respectively. As expected it can be seen that as the percentage of the radiator surface covered increases the outlet temperature of the radiator coolant increases. Table 3, shows the regression models used to predict the outlet temperatures of radiator coolant from the percentage of area covered of the radiator. The model was developed using clay soil to cover the radiator. The Matlab model was a simulated model to evaluate the validity of the clay model. The two models predicted well and were statistically significant at a 1% probability level. The coefficient of determination (R^2) of the two models ranged from 0.98 to 0.99. At least 99% of the variation in the outlet coolant temperatures was explained by the percentage of the area covered of the radiator. For the clay model, for every 10% increase or decrease of the area covered of the radiator resulted in an average increase or decrease of about 21 degree Celsius of the outlet temperature of the radiator coolant as shown in (Table 3.). In the Matlab model, a change in 10% of the area covered of the radiator resulted in a change of about 20 degree Celsius of the outlet temperature of the radiator coolant.

3.2 Regression and ANOVA Analysis

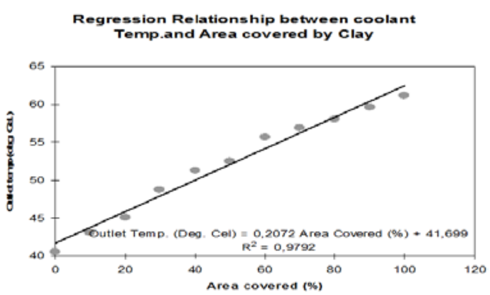

Fig.7: Regression Relationship between Radiator outlet Temperature and the Percentage of Radiator Surface Area Covered with Clay

Table 3: Models to predict outlet temperature (°C) of coolant from the Area covered of the radiator (%).

Model	Regression equation	R^2	T – Test (Two-tailed)	P-value
Clay	T=0.2072A+41.699	0.9792	70.039	<0.0001
Mat lab	T=0.20A+42.323	0.999	905.980	<0.0001

T= radiator outlet Temperature, A= Percentage of the radiator surface area covered by silt or clay

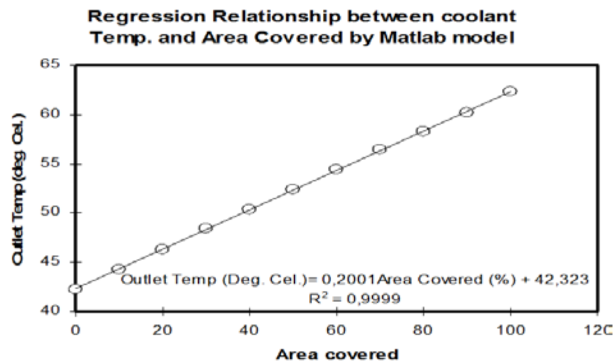

Fig.8: Regression Relationship between Radiator outlet Temperature and the Percentage of Radiator Surface Area Covered using MATLAB Simulation

Analysis of Variance (ANOVA) was conducted to find out if there were statistically significant differences among the two models (Table 4.). As shown in Table 4, there were no statistically significant differences among the three models. This indicates that the Matlab model was consistent with clay models. It can also be said when the same amount of clay covers the same area of a radiator the same temperature rise of the radiator coolant will result.

Table 4: Analysis of Variance (ANOVA) for comparison outlet temperatures based on the silt, clay and Matlab models

	Sum of squares	*Df*	*Mean Square*	*F-Value*	*p-value*
Between Groups	106,246	2	53,123	1.425	0.255
Within Groups	1230,285	33	37,283		
Total	1336,531	35			

4. CONCLUSIONS

The experiments were successfully conducted on the radiator of a four cylinder four stroke engine. The main soil type used was clay as a covering material for the radiator surface area. It was observed that both the inlet and outlet temperature of the coolant in the radiator increased as the percentage area of the radiator covered increased.

A mathematical model was developed to predict the heat transfer phenomenon. The model developed predicted the heat transfer so well. It was observed that the inlet temperature of the coolant in the radiator increased as the percentage area of the radiator covered increased. The outlet temperature of the coolant from the radiator also increased as the percentage of the area of the radiator increased. It was also observed that the heat transfer model developed agreed with the experimental results.

Statistical analyses of the experimental results obtained were carried out to determine how significant the differences were. The analysis of variance results pointed to the fact that the results obtained for clay and the mathematical model were not significantly different at 95% confidence interval. This point to the fact that, irrespective of the covering material, the effect of temperature rises will effectively be the same. This work did not look at fin angle and it is recommended that any future work can also look at the effect of fin angle on the heat transfer process.

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