ASSESSING THE EFFECT OF DIRT ON THE PERFORMANCE OF AN ENGINE COOLING SYSTEM

BY

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CERTIFICATION

I hereby declare that this submission is my own work towards the Master of science in Mechanical Engineering degree and that, to the best of my knowledge, it contains no material previously published by another person nor material which has been accepted for the award of any other degree of the University, except where due acknowledgment has been made in the text.

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ABSTRACT

This thesis looked at the effect of sand blocking the heat transfer area of the radiator and its effect on the engine coolant through the conduct of experiments and a mathematical model developed. The results 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 pointed out that every 10% increase area of the radiator covered with silt soil resulted in an increase of about 1.7 ^{o}C of the outlet temperature of the radiator coolant. Similarly, using clay as a cover material, 10% of the area covered of the radiator resulted in an increase of about 2 ^{o}C of the outlet temperature of the radiator coolant.

Statistical analysis pointed to the fact that the result obtained for clay, silt and the mathematical model were not significantly different. Thus, irrespective of the type of material that blocks the radiator surface area, the coolant rises with proportion of the radiator covered.

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ABBREVIATIONS

AC	Air-conditioning
ANOVA	Analysis of Variance
EG	Ethylene Glycol
EGR	Exhaust Gas Recirculation
LMTD	Log Mean Temperature Difference
NO_x	Nitrogen Oxide
NTU	Number of Transfer Units
SAE	Society of Automobile Engineers
PD	Power Developed
VE	Volumetric Efficiency
BTDC	Before Bottom Dead Centre
DOHC	Double Overhead Camshaft
CR	Compression ratio
RPM	Revolution per Minute
IC	Internal Combustion
CS	Compression Stroke

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DEDICATION

To my wife Mrs. Linda Oduro and children: Nana Kwadwo Ansah Oduro, and Nana Kwadwo Boakye Oduro.

CHAPTER ONE INTRODUCTION

1.1 BACKGROUND

When internal combustion engine is operating at a full throttle, the maximum temperature reached by the burning gas may be as high as 2700 ^{o}C [1]. The expansion of the gas during the power stroke lowers the temperature considerably. However, the minimum normal operating temperature of the gas that occurs during the exhaust stroke may be at least 800 ^{o}C [2]. The entire engine absorbs some heat from the hot gas and the amount of heat absorbed is proportional to the temperature, the area exposed to the gas and the duration of the exposure. This heat will raise the temperature of the components of the engine. Sometimes, the temperature of the exhaust gas is above the melting point of engine materials such as aluminium.

The heat transfer that occurs within an engine is extremely important for its proper operation. About 35% of the total chemical energy that enters an engine in the form of fuel is converted to useful crankshaft work, and about 30% is carried away in a form of exhaust gas through the exhaust pipe. This leaves about one-third of the total energy, which is dissipated to the surroundings of the engine by some mode of heat transfer [1].

Materials used in the making of some components of the engine may not 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 its components and preservation of the lubricant. The combustion chamber wall, piston crown, the upper end of the cylinder and the region of the exhaust port are exposed to the hottest gas and reaches the highest temperature. The resulting thermal expansion of these parts distort them from their correct shape, causing gas leakage, loss of power, valve burning, gasket hardening and possibly even cracking of the cylinder head.

Furthermore, the electrical and electronic components are affected by high temperature, which may cause malfunctioning and poor performance [3] of the major components of a vehicle and engine stalling. Moreover, the oil film that should lubricate the piston and the cylinder walls may be burnt or carbonized, causing excessive wear and even seizure of the piston. Power is lost by the heating of the fresh gas entering the cylinder, which reduces the volume of air intake and, thereby, reducing the power output.

Lastly, an increase in temperature of the fresh gas increases the ability to detonate, thus reducing the compression ratio [3]. Some parts of the surface of the combustion chamber may be hot enough to cause pre-ignition, which may result in serious damage to the engine if it persists.

It is, however, important not to overcool the engine. Heat is necessary to assist in vaporizing the fuel inside the cylinder during the compression stroke (CS). When the engine is overcooled, some of the heat which may be utilized to expand the gas may be lost. Reduction in vaporization may cause gas produced by combustion to condense on the cylinder wall. This leads to dilution of the oil in the sump and the addition of harmful corrosive acids [4].

Removal of the oil film from the cylinder wall by the vaporized fuel leads to increase cylinder bore wear. Inadequate lubrication of the engine, due to the oil not being warm enough to flow freely, results in greater frictional losses. In general, the economy and life of the engine is reduced. Overcooling is normally prevented with thermostat inserted in the cooling system. The thermostat allows the engine to reach its normal operating temperature. It controls the flow of coolant through the radiator depending upon the engine water-jacket temperature. When the engine is cold, the thermostat prevents coolant from leaving the jacket until a higher coolant temperature is reached.

At this stage, the thermostat valve opens and coolant is allowed to pass from the water jacket into the radiator at a rate sufficient to maintain temperature within the normal range and, thereby, improves fuel consumption. This improves the quick warming up of the engine. When the engine reaches its normal operating temperature, the cooling system begins to function. The cooling system cools the engine rapidly when it is hot, but slowly when the engine is cold or warming. Hence, the thermostat improves fuel efficiency.

Two general types of cooling system are utilised in most types of vehicles found in Ghana. They are air cooling system, and a combination of air and liquid cooling system [5]. In air cooling engine, there are spaces between the cylinder arrangements in order to allow enough air to flow in between the cylinders. Most engines for airplanes, motorcycles, power lawn mowers, and chain saw are air cooled. A multi cylinder engine using the air cooled system requires a large or special fan as well as large fins or a cowling to direct the air to the hot sections of the engine, and such equipment becomes bulky if the system is to be effective.

This point is better appreciated when it is remembered that in terms of volume, 4,000 times more air than water must pass through an engine to dissipate an equal amount of heat. Large fans absorb engine power when driven by the engine and reduce its output. The absence of a sound muffling coolant jacket may cause noisy engine operation and the possibility of higher temperatures when compared with liquid cooled engines [6].

On the other hand, the water cooling system provides an improved cooling of the engine over the air cooled. Usually, it has an impeller water pump to speed up the circulation of the water in the engine and a heat exchanger unit, which is called radiator. The radiator used on cars act as a reservoir, which stores water for the engine cooling system.

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 and in some high efficiency liquid-to-liquid radiators [8].

The radiator is normally placed in front of the engine to enhance flow of air in the cooling of the engine. As a radiator it 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 [9].

In Ghana, most of the vehicles move on untarred roads that result in a lot of dust and mud blocking a large proportion of the radiator fins area. As a result, engines may overheat, which may significantly affect their performance.

The main focus of this thesis is to assess the effect of dirt and mud on the performance of radiators of vehicles.

1.2 PROBLEM STATEMENT

Over the years, vehicles that are used in the country are imported from Europe, North America and other Asian countries. The vehicles are manufactured without taking into consideration the nature of the roads in Ghana. When these cars are brought into the country, they encounter numerous problems such as overheating. As the cars move on the untarred roads, dust and mud get in contact with the radiator fins. This dust and mud block the fins and impede the flow of air through the radiator. As a result the temperature of the coolant in the engine rises and significantly affects the performance of the engine.

In Ghana, vehicles pass through a lot of dust and mud which eventually settle on the engine and the radiator. Vehicles in cities, towns and villages encounter these problems since most of the roads these vehicles ply on are untarred roads. These dirt and mud accumulate on the fins of radiator and other component of the cooling system. The accumulation narrows the heat dissipation air passage area.

There has been little or no research conducted on the effects of mud on the cooling system in Ghana. Little is known on the quantification of mud on the radiator fins and its effect on the engine performance.

1.3 OBJECTIVES

The main objective of this thesis is to look at the critical quantification of dirt on the radiator and its effect on the engine performance. The focus will be to look at the effect of mud on the radiator fins and its impact on temperature variation in the inlet and outlet of the radiator hoses. The study will looks at the way forward to minimize overheating.

1.3.1 SPECIFIC OBJECTIVES

The specific objectives of the study are:

a) To determine the effects of blockage of fins and fin louvers on the inlet and outlet temperature of the radiator.

b) To assess the effects of blockage of fins and fin louvers on the rate of cooling of an engine.

c) Build a mathematical model to describe the phenomenon of heat transfer in the radiator.

1.3.2 SCOPE

The study will be limited to a four-cylinder four-stroke engine, with a complete engine and radiator assembly. The device to be studied is a cross –flow finned tube radiator used for both petrol and diesel cars. The coolant flows in a vertical tubes and the area available for the transfer on the air side is greatly extended the use of louvered fins.

1.4 APPROACH

An experiment will be conducted by using a four cylinder four stroke engine with complete radiator assembly. Mathematical equations will be used to model the radiator. Heat balances can be calculated from the proposed measurements. Air velocity as well as inlet and outlet temperatures are sufficient to calculate the heat transferred per unit time.

Once the heat transfer coefficients for the water and air-side of the radiator are found, they can be combined to obtain the overall heat transfer coefficient, and subsequently calculate the theoretical heat lost. Two thermocouple probes will be used to read the input and the output temperatures of coolant in the radiator. Two different soil types will be used.

Silt and clay soil type will be chosen for the experiment because of their ability to stick to the radiator surface area. The radiator will first be 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 will be kept constant throughout the experiment. The effective heat transfer area of the radiator will be divided into ten (10) equal parts representing 10% of the effective heat transfer area. The effective heat transfer area will be partially covered with either clay or silt soil to the required percentage before running the engine. The engine will be run for about fifteen (15) minutes to attain stable conditions before readings will be taken.

CHAPTER TWO LITERATURE REVIEW

2.1 INTRODUCTION

When new engines are developed, they will be expected to operate under severe conditions and demanding load profiles Kem. [10], thus, increasing the demand of effective engine cooling systems. The automotive industry is continuously involved in a strong competitive market to obtain the best automobile design in multiple aspects. Some of the aspect includes performance, low fuel consumption, aesthetics, safety, cheaper and low maintenance.

According to Oliet [11], the air-cooled heat exchangers found in a vehicle radiator, Air-Conditioning (AC) condenser, evaporator and charge air cooler, has an important role in its weight and also in the design of its front-end module. In addition, it has a strong impact on the car aerodynamic behavior. Looking at these challenges, the design process is optimized to obtain the optimum design.

The high working temperatures of the internal combustion engine which are the cause of its high efficiency are at the same time a source of great practical challenge in construction and operation. Moreover, there are challenges in obtaining materials capable of continuously enduring the high temperatures to which the cylinder, piston and valves are subjected to, while at the same time retaining sufficient strength to withstand the high working loads [1]. The maximum temperature attained during combustion is approximately equal to the melting point of platinum, and the temperature even of the exhaust gas is above that of melting point of aluminium. Thus, it is essential that heat be abstracted from the combustion chamber at a sufficient rate to prevent a dangerous temperature being reached. Unfortunately, it is quite impossible to prevent the hot gas from flowing over the metal surfaces. Excessive cooling also prevents proper vaporisation of the fuel, leading to further waste. In addition, it may lead to dilution of the crankcase oil by un-vaporized fuel, which gets down past the piston. On the other hand, a high operating temperature change weight and reduces volumetric efficiency (VE) on account of the excessive heating of the incoming charge. Consequently, there is reduction of output power.

Over the years there have been increasing demands on the engine cooling system of a car. This has been caused by a steady increase in engine output; combined with a reduction in the size of the cooling inlets and an increase in auxiliary components [12]. A water-cooling system is employed in almost every new car. The water coolant passes through the engine extract heat around the combustion chamber and dissipates the heat in the radiator. The radiator transfer heat from the coolant to the air flowing through the fins of the radiator. The air flowing is driven by a combination of the forward motion of the car and from a fan enclosed in a shroud attached to the radiator.

The basic requirement of a radiator is to provide a sufficiently large cooling area for transmission of heat from the coolant to the air. The construction of the centre of the radiator or core varies, but in general the water passages terminate at a header tank at the top of the radiator, and a smaller collecting or lower tank at the bottom. In addition to an opening which enables the cooling system to be topped up, the header tank allows for expansion and contraction of the coolant within the system [13]. The trend towards low bonnet lines has led to the adoption by some vehicle manufacturers of a cross-flow radiator, the water tubes being placed cross-wise with a small collection tank at each side, and a separate header tank. The alternative to a cross –flow would be to have a large number of short vertical tubes, but this would increase the amount of soldering and hence the chance of leakage [14].

Normally, the automobile radiator exchanges heat between hot water and cool air. Due to its low air-side thermal performance, the louvered fins are attached to the unit at the air-side surface to increase the heat transfer area. This kind of fin positioning enormously increases the air-side heat transfer coefficient.

2.2. FUNCTION OF ENGINE COOLING SYSTEM

In all mechanical systems, conversion of energy from the primary source to useful work cannot be achieved with 100% effectiveness. In internal combustion (IC) engines, only a fraction of the energy generated from the combustion of fuel in the cylinders produces useful work. For a typical vehicle, considerable amount of the energy produced by fuel is dissipated approximately in three ways [15]. About 35 to 45 percent of heat energy goes to useful work which drives the vehicle. The exhaust gas takes about 30 to 40 percent of the energy, and the rest is wasted. Thus, there is an amount of 22 to 28 percent of heat produced by combustion required to be dissipated. Part of this heat is usable in areas such as warming the cabin in cold weather for passenger comfort; and maintaining the engine at an optimum temperature to achieve effective combustion and lubrication. The remainder is excess and must be removed. The method employed for removing the excess heat to surrounding air in typical vehicle is via a liquid-cooled indirect cooling system.

The principle of operation of the cooling system is to circulate a water-based coolant carrying the unwanted heat from the cylinders and engine head, and dissipating it by movement of air at the radiator. The engine could be represented as a big thermal capacitance where there is an incoming of energy from the combustion and the friction. The coolant is in contact with the mass through a wall and there is convective heat transfer between wall and coolant.

2.2.1. ENGINE

The combustion and friction in the engine produce heat that has to be removed to avoid excessive increase of the temperature. The engine is cooled using three cooling media, which are water, air and oil.

The engine cylinders are surrounded by a water-filled cavity, the water jacket or cooling jacket. An important engineering dimension is the water jacket depth, defined as the distance from the top plate plane to the lowest point in the water jacket. In earlier grey cast engine blocks, this dimension was as much as 95% of the length of the cylinder running surface. In modern cast iron engine block designs, the water jacket ends in the area swept by the lower piston ring. The water jacket is even shorter in depth than that of modern aluminium engine blocks. Figure 2.1 shows a schematic of a typical liquid-cooled indirect cooling system employed in vehicles. In the figure, the water jacket depth corresponds to about one third of the length of the cylinder running surface.

This is made possible because the thermal conductivity of aluminium is higher than that of cast iron materials. In addition, compression heights are shorter than that of the earlier grey cast iron engine block. A short water jacket reduces the coolant volume in the engine and, thus, the engines weight. Having small coolant volume and thermal capacitance shortens the engine's warm-up phase, with positive effects in terms of unburned hydrocarbon emissions and the response time for the catalytic converter.

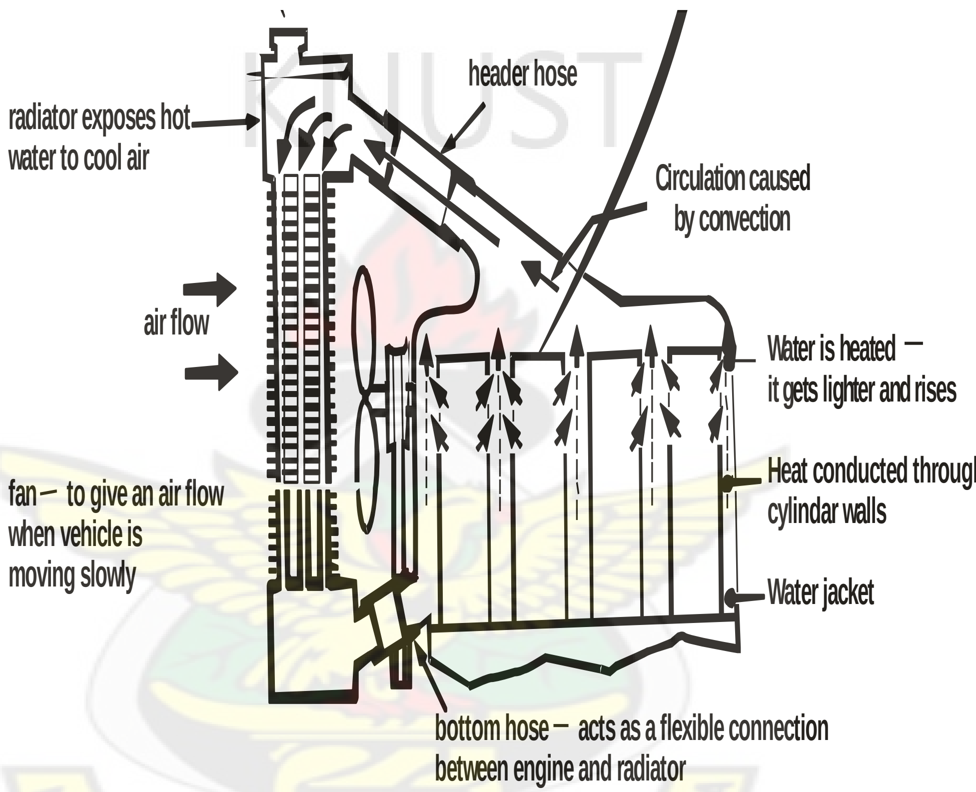


Figure 2.1: Liquid-cooled indirect cooling system

The air going through the radiator and the bypass of the radiator cools down the engine. There are some engines, which are cooled only by air instead of the water coolant. Today, only a very few manufacturers still use air-cooled cylinders in automotive engines [15]. Heat dissipation in air-cooled cylinders is dependent upon the thermal conductivity of the cylinder fins and on the cylinder materials, shape of the cooling fins and the way in which cooling air passes across the fins.

The oil that lubricates the engine also cools down the engine. In some cases the oil is refrigerated by the engine cooling system to improve the efficiency of the engine, or to maintain the oil below the maximum permissible temperature. This oil is cooled with engine oil cooler that uses either liquid coolant or air in the cooling module. The ones that use coolant are preferably located near the engine to enhance heat transfer and may have a round disc shape, disc stack, or flat tube design and are usually made of aluminium. Direct cooling with oil and air coolers is also common. They are constructed from very high-pressure soldered flat tube aluminium design and are arranged in the cooling module. In commercial vehicles, cooling is always performed with coolant, while the coolers are normally installed in an opening in the crankcase where they are exposed to the main flow of the coolant. The most widespread design is plate-type coolers which are made of stainless steel with inserts on the inside through which the oil flows.

More recently, the use of aluminium coolers with high capacities and comparable strengths but roughly half the weight has become possible. The oil used in transmission and the hydraulic oil used in power steering or other servo systems are also cooled in some cases.

2.2.2. ENGINE COOLING STRUCTURE AND OPERATION

The conventional internal combustion engine cooling system consist of a thermostat, water pump, radiator, and engine block water jacket (as shown in Figure 2.2,) and copper impregnated wax enclosed thermostatic valve, which regulates the coolant temperature by controlling flow of the coolant through the radiator.

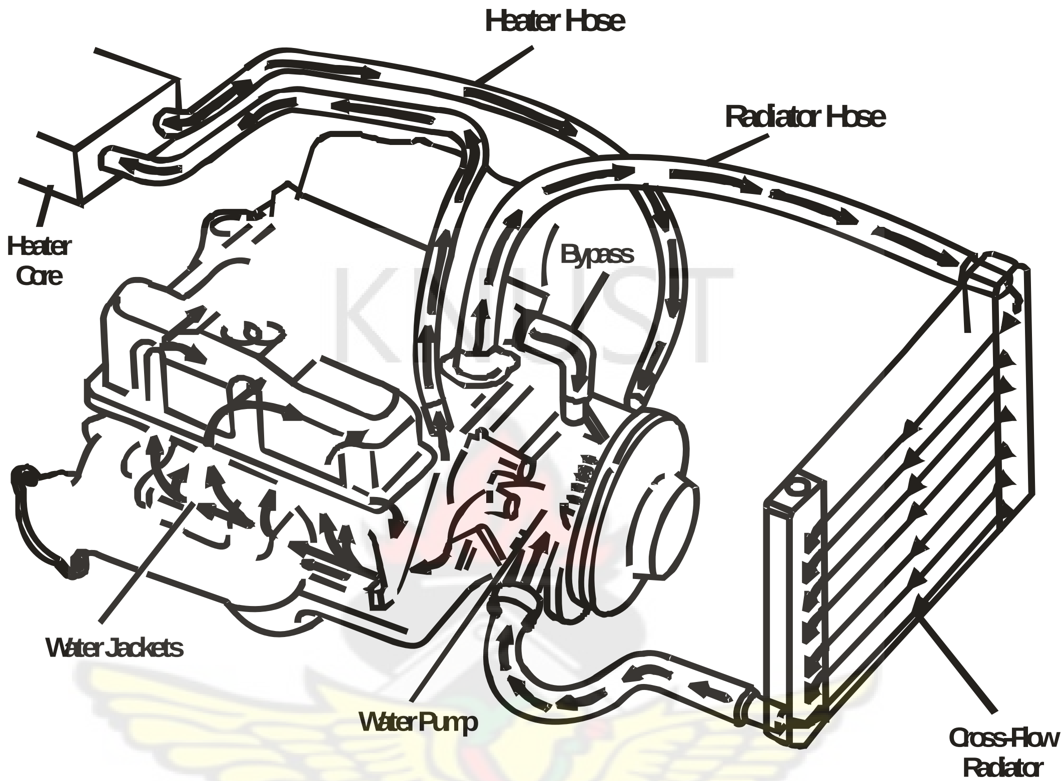


Figure 2.2: Coolant Path and Component of Automobile Engine Cooling System

A belt driven centrifugal pump traditionally circulates coolant through the system based on the engine speed. The engine is normally designed to maintain the engine's component temperatures within prescribed operating ranges during both steady-state and transient driving. By cooling the flow of coolant, the thermostat valve controls the temperature in the engine. Some of the coolant heated in the engine block may be used to heat the cabin air, and the water introduce back into the circuit in the pump or somewhere else. Spark ignition and the diesel engines have the same engine cooling system, but with minor modifications [3] to suit their operation. In the diesel engines, the air is usually compressed and cooled before it is introduced into the cylinder. This yields some structural differences such as air-to-air intercooler layer in the radiator, and has significant effect on the air temperature of the radiator. The exhaust gas is used to run a turbine that has the same shaft as the compressor, and some of this exhaust gas is introduced into the combustion air to reduce the nitrogen oxides (NOx) emissions. This is done through an exhaust gas recirculation (EGR) valve.

2.2.3 LOCALIZED COOLING

Localized cooling occurs when jets of coolant are directed to local hot spots such as valve seats and spark-plug areas. This may be accomplished by means of a suitably drilled gallery pipe fed directly from the water pump. In addition, nozzles may be pressed into the head casting at each of the coolant passages between the cylinder block and head jackets. Figure 2.3 shows a section through the exhaust valve.

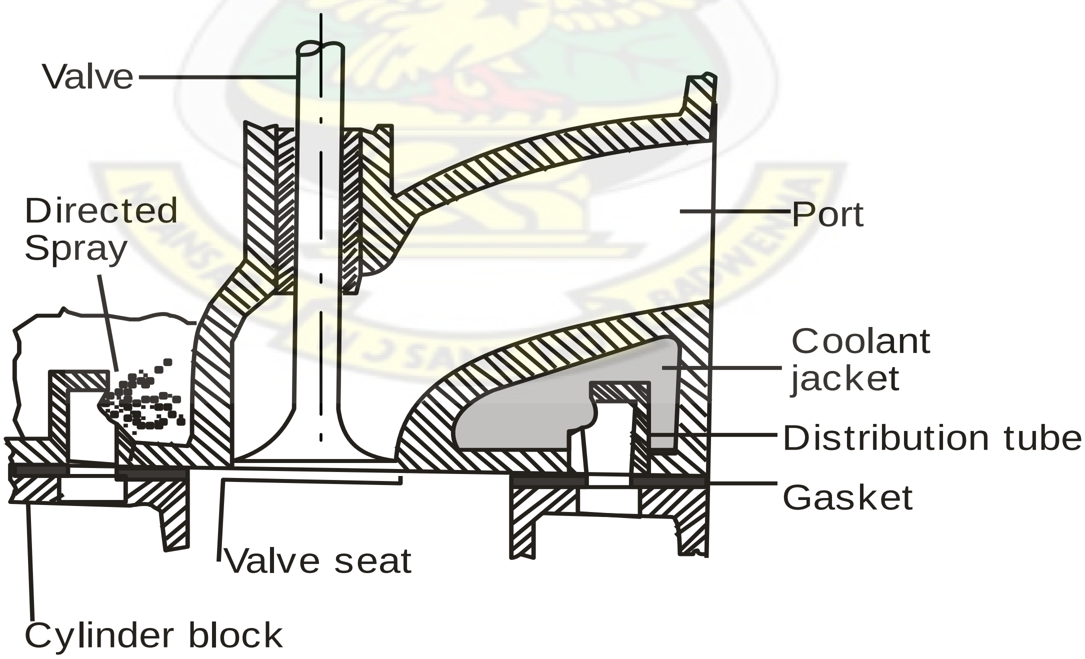


Figure 2.3: Direct cooling in engine cylinder head

In the Figure, the nozzles take the form of thimbles provided with suitably positioned one or two ports to direct local streams of coolant to the desired spots. A pressed locating snug ensures correct alignment of the nozzles. A six-cylinder head may have as many as ten of these jets directed towards the exhaust valve pockets.

2.3 BASIC STRUCTURE OF AUTOMOTIVE RADIATORS

Radiators used for vehicle engine cooling are cross-flow type heat exchangers, as airflow passes perpendicular to the coolant through the heat exchangers. The airflow is generally induced by the combination of moving vehicles and the cooling fans.

2.3.1. AIR PASSAGES

Radiators with extended surfaces consisting of multi-louvered fins are commonly employed in modern vehicles, as shown in Figure 2.4. The extended surfaces significantly enhance the heat transfer not only by providing additional heat dissipation surface area. Furthermore, it is enormously increased by reducing the thickness of the boundary layer by inducing a series of flat-plate leading edges and interrupting the growth of the boundary layer along the fin surface [16]. Hosoda et al. [17] compared a plate fin heat exchanger with one with parallel-louvers exchanger, and found that the air-side heat transfer performance of the louvered fin structure was 60% higher than that of the plain plate fin equivalent.

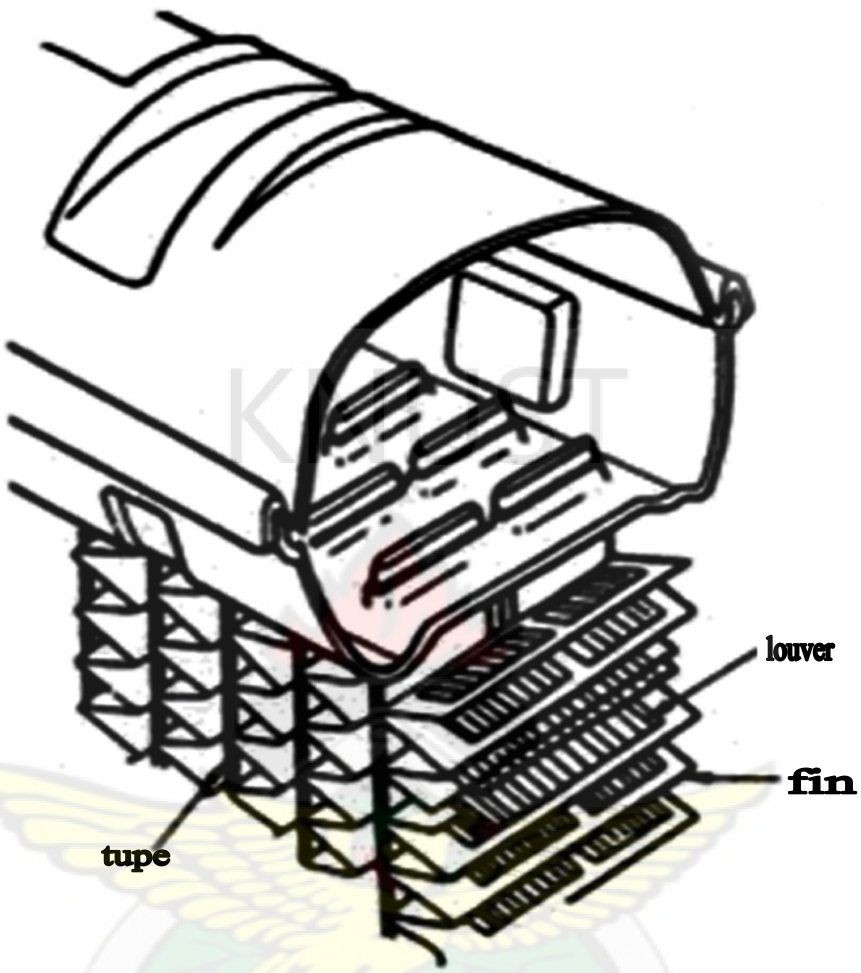


Figure 2.4: Tube- and- corrugated louvered fin radiator

2.3.2. COOLANT PASSAGES

Orientation of the coolant tubes may be horizontal or vertical. Shown in the Figure 2.5 is a radiator with coolant tubes oriented to the vertical. According to SAE HS-40 [19], the horizontal flow type radiator is more commonly used in passenger vehicles, than the vertical ones. The coolant tubes are generally based on flat tube design. The advantages includes higher heat transfer area per unit of flow area and the wakes of the tubes cause less reduction of heat transfer in downstream regions. In addition, the small projected areas of the tubes do not create profile drag and result in higher fin efficiency.

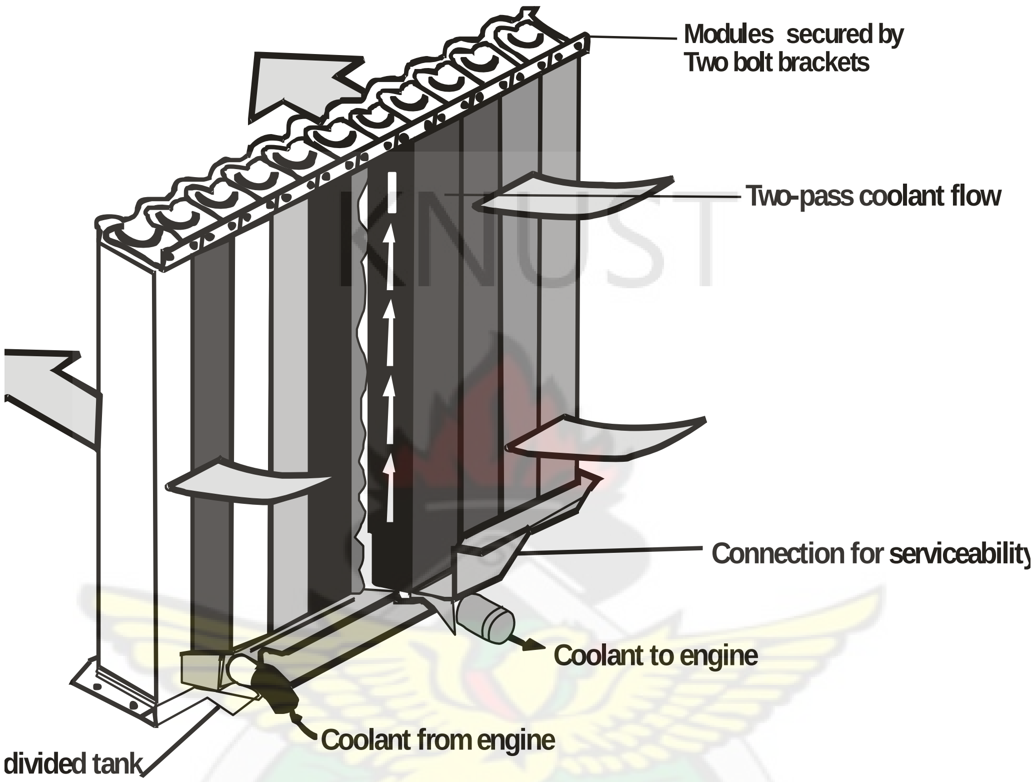


Figure 2.5: Radiator direction of coolant flow

2.3.3. COOLANT

For the purposes of increasing the boiling point of the coolant and preventing corrosion in the cooling systems, coolants are generally a mixture of water, anti-freeze and possibly various corrosion inhibitors. It is noted that the use of ethylene glycol (EG) mixture which is a widely used anti-freeze, generally reduces the heat transfer performance compared with pure water.

2.3.4. THE MOST COMMON RADIATOR CONFIGURATION

This analytical model was developed specifically for the most common radiator configuration, which consists of air-cooled radiator, cross-flow radiator, corrugated louvered fins and flat coolant tubes with horizontal flow structure. The traditional vehicle radiator was made by non-ferrous metal materials, with copper fins and brass pipes. In Europe, they have been suppressed in cars since 1975 and in commercial vehicles since 1988 by further developed aluminium alloys offering a weight reduction of up to 30%, high pressure resistance, and a higher corrosion resistance [20].

A radiator may consist of two layers of a heat exchanger. On the front is the layer of air conditioning heat exchanger, and the second layer is the engine cooling heat exchanger. The structures of these two heat exchangers are different. The air conditioning heat exchanger has a two- phase refrigerant and structure of folded pipes. Due to the decrease in the volume of refrigerant as it passes through the heat exchanger, the number of folding per unit length decrease from the refrigerant inlet to the outlet.

The engine cooling radiator is a simpler structure formed by two vertical main leaders that are connected through a group of smaller horizontal pipes as shown in Figure 2.6. The horizontal pipes can be round or oval. In between the horizontal pipes are fins to increase the heat exchanging surface. These fin structures may have different shapes and fins have stamped mini- fins called louvered fins.

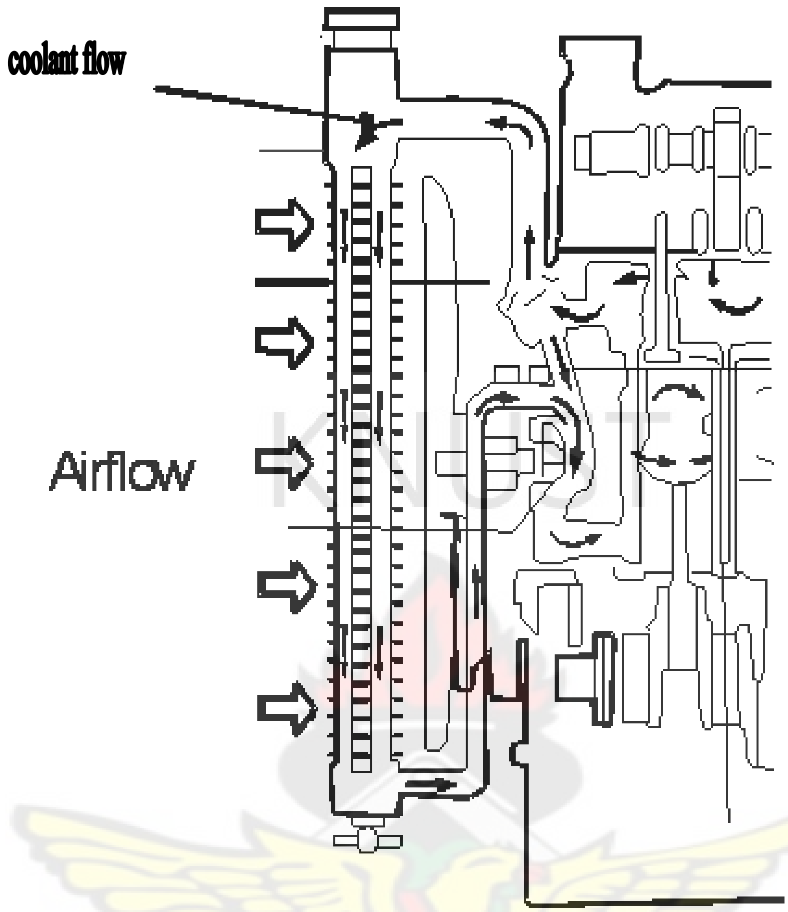


Figure 2.6: Engine cooling radiator structure

2.3.5. RADIATOR CONSTRUCTION

Copper radiators have now evolved into two types of water-tube construction, the early honeycomb or cellular construction as shown in Figure 2.7 and 2.8 are not commonly used in cars. The early types have certain advantages in appearance, ease of repair and smaller proportionate immobilization by a single obstruction. However, they suffer from the important heat transfer defect that the ratio of metal–air to water–metal dissipating surface is practically smaller.

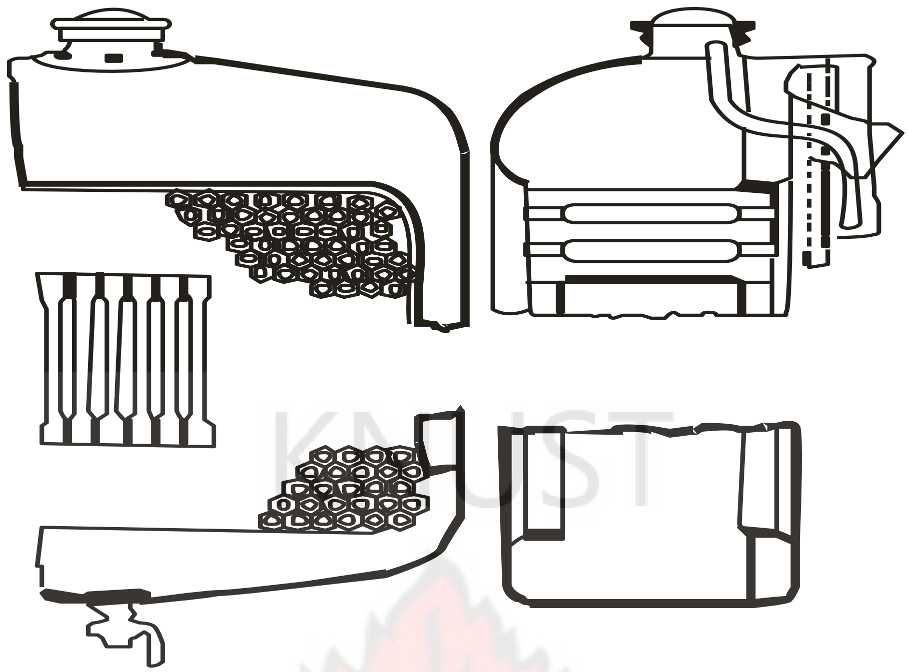


Figure 2.7 Honeycomb radiator

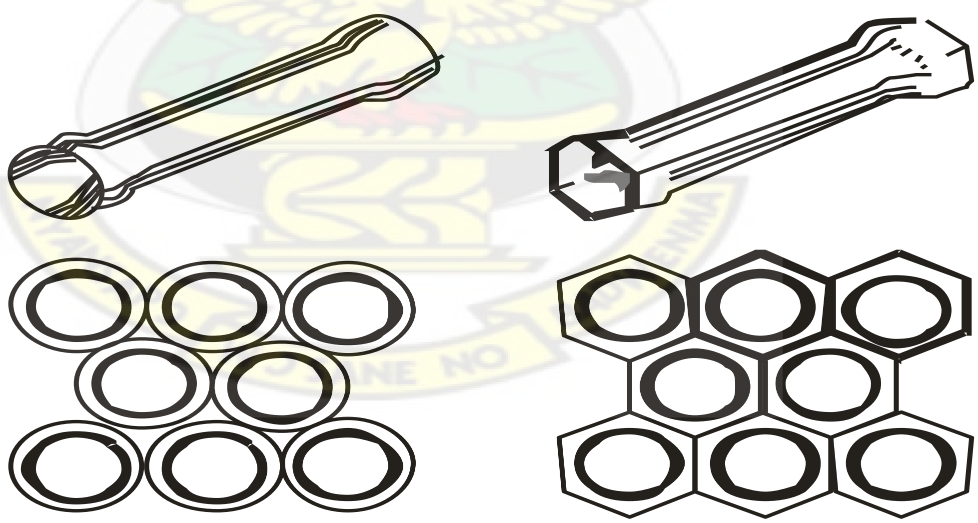


Figure 2.8: Honeycomb tube

In modern tubular constructions, appearance and protection are provided by the ornamental grille which is now universal in passenger vehicles. A radiator block may have a simple design, efficient and lighter in weight as much possible. Figure 2.9 shows the vertical water tube type which is found on heavier types of commercial vehicles. The construction consists essentially of light cast alloy, top and bottom tanks bolted to cast side pillars. The cooling element consists of a nest of copper tubes soldered into brass upper and lower tube plates, which are bolted to the top and bottom tanks. This construction is relatively inexpensive and readily for accessibility and for repair. The tubes take various forms of cross section and circular section, are usually provided with external gills to give an additional surface to aid transfer of heat from metal to air.

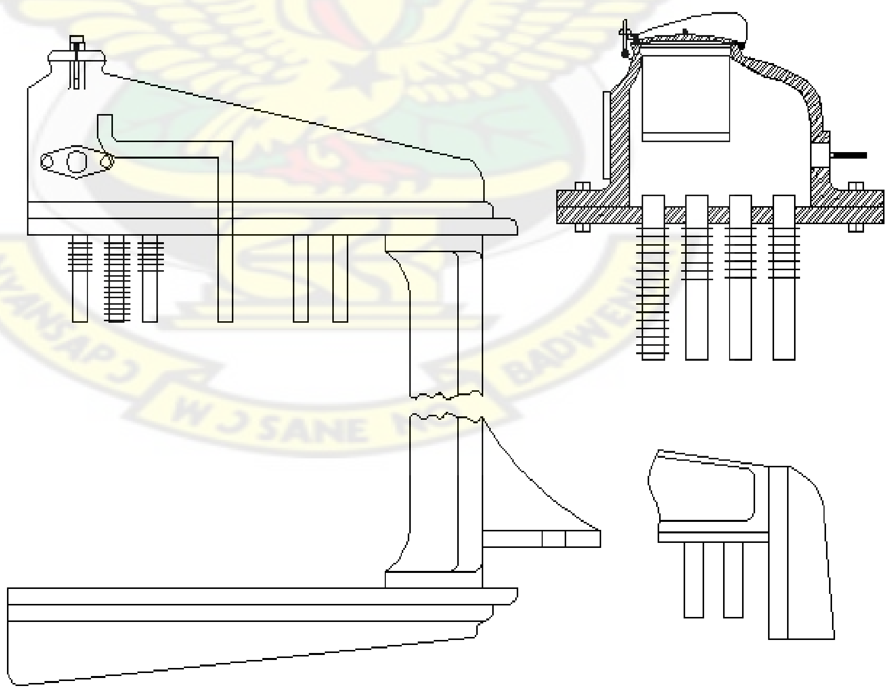
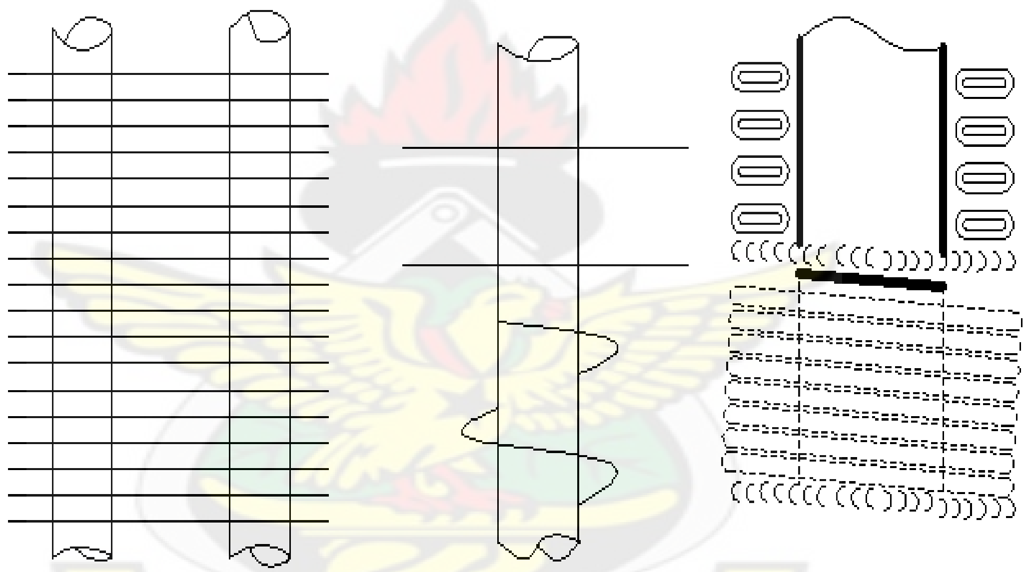


Figure 2.9: Radiator constructions for heavy commercial vehicles.

Various cross sections of tubes are illustrated in Figure 2.10. In (*a*) of the figure, the construction consists of a nest of six or eight tubes, which run from back to front of the block and end in the common gill plates. Different forms of plate gill applied to individual tubes are shown in (*b*) of the figure. In (c), the efficient and widely used still tube which employs a continuous coil of copper wire wound spirally around, and sweated to, the plain tube.



(a) (b) (c)

Figure 2.10: Radiator tubes

Figure 2.11 shows a popular current form of flattened tube and plate gill type, which can be assembled up to the size of block required. The flattened tubes give a more favorable ratio of heat transfer area to water weight than the circular tube. Usually, they are fabricated from a very thin strip of metal, which is folded to form longitudinal joint. The ratio of air surface to water surface can be varied very conveniently by the appropriate choice of radiator tubes, and the number of gill plates through which the tubes are threaded.

The dimples on air passage help to ensure the desirable turbulent motion of the air. Figure 2.11 illustrates the design of the film-tube type in which the core is built up of units formed from embossing a thin of copper strip. Normally, the width of the strip is equal to the required thickness of block and the thickness ranges from 50.5 mm (2”) to 76.0 mm (3”).

The block is built by first forming airway units of the full height of the block. This is done by folding the embossed strip around serpentine dimpled strip to provide the additional air surface. Alignment dimples are provided on the embossed strip to aid the setting up of the serpentine for soldering.

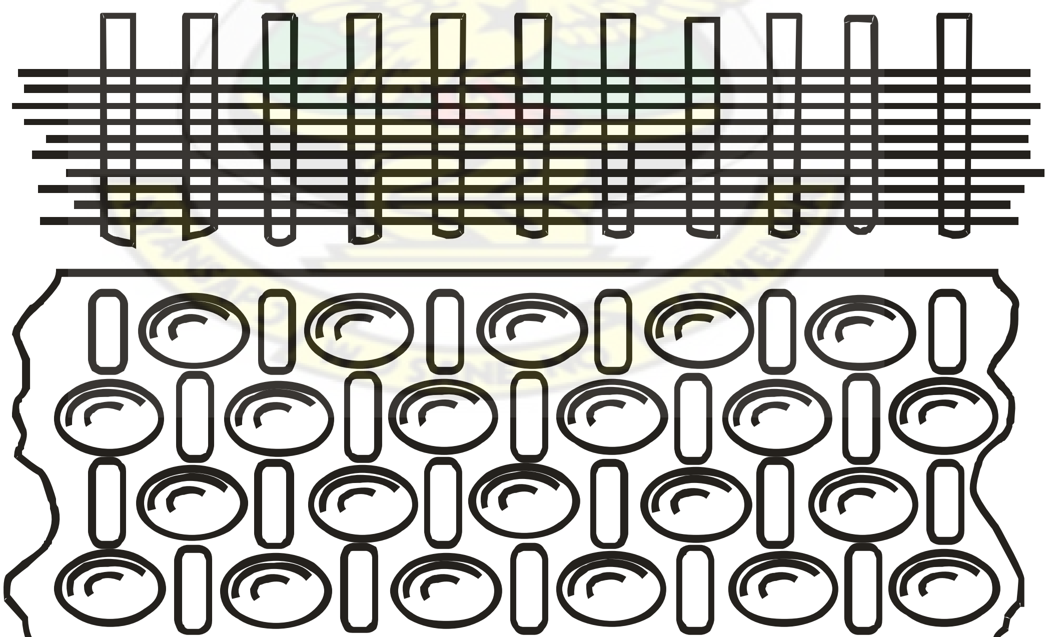


Figure 2.11: Flattened tubes, with plate gills

These airway units are then assembled in a jig in the required number and joined together at the corrugated edges, leaving straight vertical waterways of a lateral width equal to the full depth of the edge corrugations. In addition, the embossing of the strip provides strengthening flutes and spacing supports on the vertical centre line. The actual size drawing of Figure 2.12 should make the construction clearer. The ratio of air area to water area in the example illustrated is about 5: 1. Aluminum radiators have been available for many years but have not been widely used. The aluminum alloy radiators are mostly brazed in construction and are involved in a high temperature during the manufacturing process.

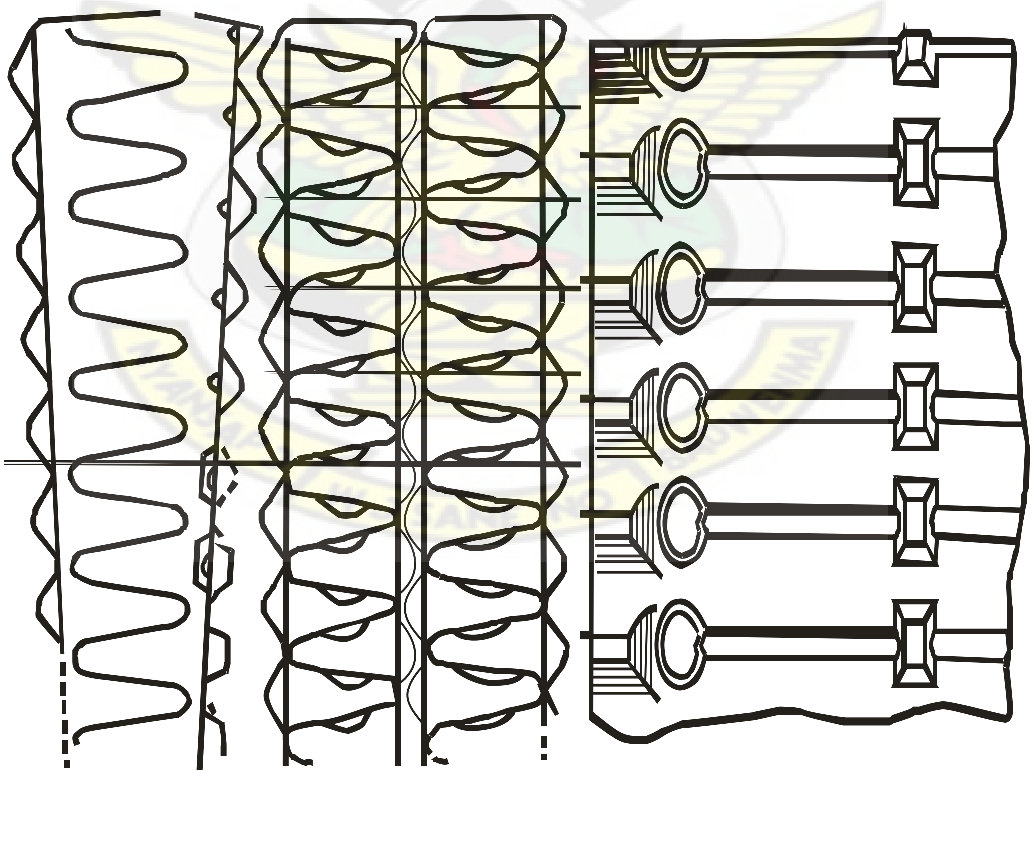


Figure 2.12: Film tube constructions (actual size)

2.4. CHARACTERIZING THE PERFORMANCE OF A HEAT EXCHANGER

Automotive radiators fall in the category of compact heat exchangers due to the restricted space in the vehicle available to the heat exchanger. From the literature review, it can be concluded that an effective automotive radiator may have a high heat transfer rate per surface area exposed to the air stream, a high heat transfer rate per frontal area and a high heat transfer rate per unit volume of the radiator [22]. Various radiator designs have been documented in literature that meets those objectives to a greater extent.

Differences between the designs that are included in this category are mostly marked by variations in the airflow channel, water flow channel and fin designs. Although different variations were reported to have benefited at certain flow rates and applications, most variations were introduced to obtain a single goal to reduce the formation of boundary layers on the heat transfer surface exposed to the air steam.

With the growth of a boundary layer on a surface, the convection heat transfer coefficient reduces in the flow direction of the passing fluid when the fluid has a laminar flow profile. The coefficient remains fairly constant and generally has a higher value in a turbulent flow profile [23]. Increasing heat transfer coefficients and the advantages gained when increasing the heat transfer area are constantly being investigated in order to obtain higher heat transfer rates. Two common radiator fins geometries were identified in published literature due to their high rate heat transfer rates per frontal area, surface area and volume. The two types of radiator fins geometries are offset strip and louver fin.

The offset strip fin geometry consists of a number of fins in such a manner that the forming boundary layer is disturbed. Webb [24] describes the offset strip fin method as a surface enhancement technique that dissipates the boundary layer in the wake region behind the strip. The efficiency with which this can be done is dependent on the length and thickness of the strip, the fin pitch as shown in Figure 2.13, as well as the Reynolds number of the flow over the surface

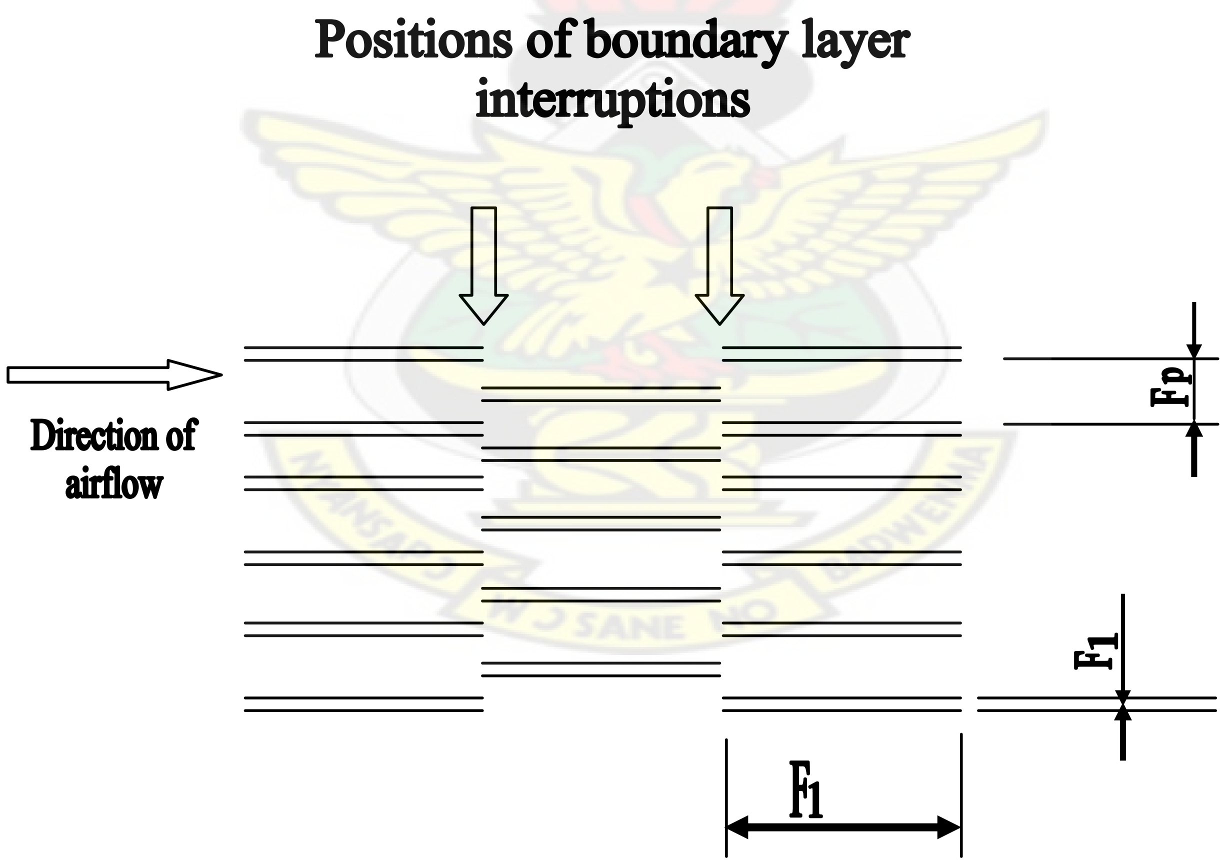


Figure 2.13: Schematic of offset strip fin geometry

In the manufacturing process of the louver fin geometry, the louvers are punched into the fin material causing a flow disturbance in the flow channel as shown in Figure 2.14.

In this case, the flow is disturbed by the angles geometry as well as the air stream entering the channel from the neighbouring channel. Similar to the offset strip fin, this enhancement technique is also Reynolds number dependent for a specific fin and louver pitch.

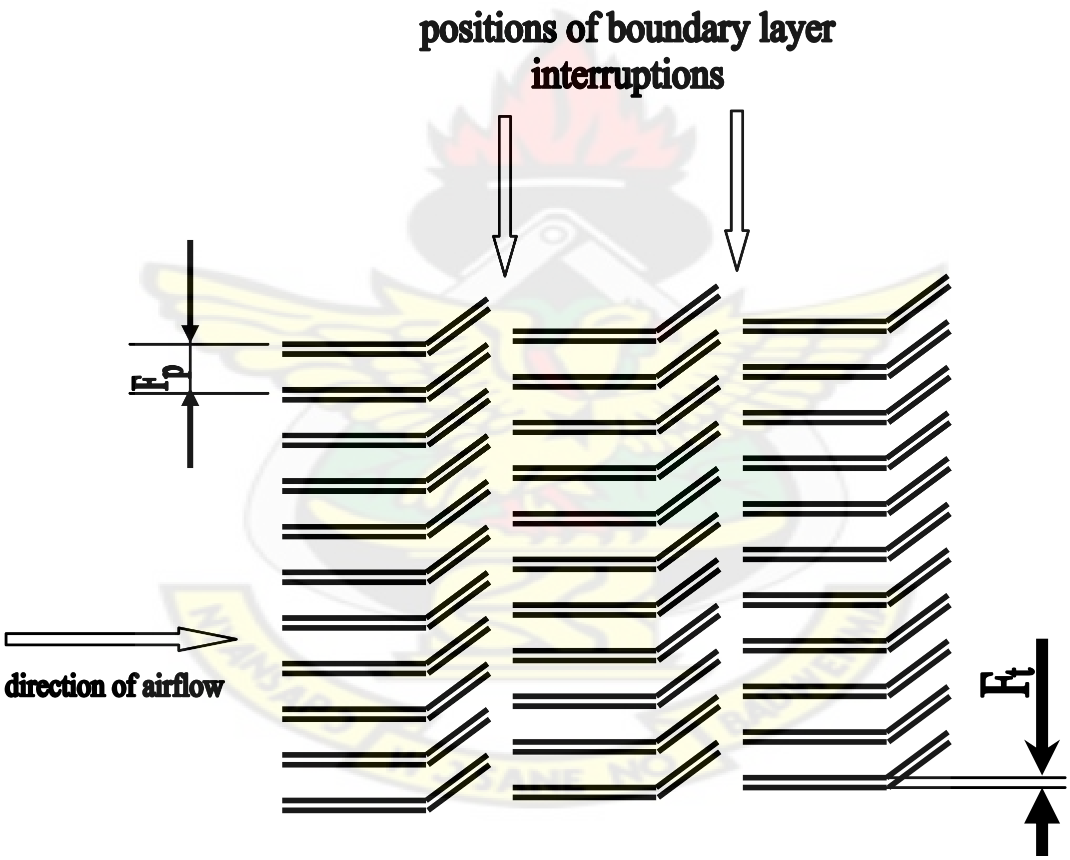


Figure 2.14: Schematic of louver fin geometry

The greater case has to do with the manufacture and assembling of the offset strip fin, as the fins must have an offset relative to each other within a certain tolerance. On the other hand, the louvers on the louver fins are punched into the fin material using high speed manufacturing processes and a large tolerance on the end product is permissible resulting in a less expensive radiator surface, hence its popularity in the commercial market [25]. Further to this advantage, it is found in literature that the louver fin geometry has superior heat transfer properties compared to the offset strip fin [26-27].

The two mentioned radiator fin geometries affect the pressure drop over a radiator which is an important parameter that needs to be considered [28]. An increase in pressure drop reduces the flow rate that in turn reduces the heat transfer coefficient. The trade-off between increasing the heat transfer and the resulting increasing in pressure drop is therefore, an important parameter that needs to be balanced at all times [29]. As the forming boundary layer which has a huge impact on the heat transfer rate is Reynolds number dependent, it can be expected that a perfect balance will only be obtained for a specific core at a certain flow rate.

The water tube design in radiator is a contributing factor to the pressure drop characteristics over the air side of a radiator. Change, Wang, Shyu and Hu [30] investigated louver fin heat with flat and round tubes. Comparing these radiators using augmentation techniques such as the volume goodness factor and the area goodness factor, it was found that the flat tube geometry offered significant advantages when compared to the round tube radiator configuration. Webb and Jung [31] as well as Zang [32] ascribed the advantages of the flat tube as the airflow being normal to the louvers, reduction in wake area behind the tube which has a positive effect on the heat transfer on downstream fins, and lower form drag associated with the design.

The advantages found in the literature with regard to the performance and manufacturing cost of the louver fin compared to the offset strip fin geometry, the louver fin was selected as the surface geometry to be used for the cars.

2.5. INFLUENCE OF VEHICLE CONFIGURATION ON THE COOLING SYSTEM

The vehicle itself has a huge effect on the performance of the cooling system. Three areas were identified where the vehicle has a direct impact on the cooling system characterisation model as the dependence of the waterflow rate on the engine speed, dependent of the airflow rate on the vehicle speed and heat generated by the engine to be dissipated through the cooling system.

The water circuit in a vehicle is isolated from any external influence on the system as a known mass of water is used in a closed circuit to transfer the heat from the engine to the radiator. The pump performance, in terms of the mass flow rate, will only change if additional restrictions are placed in the water system which will usually be associated with water system component change.

On the other hand, the airflow through the radiator is not only dependent on component that is directly related to the cooling system, and it is therefore essential to consider the factors influencing the airflow through the system. It is widely accepted in published literature that flow rate through an automotive radiator is generally the most important parameter that will dictate the heat transfer rate of the radiator [33-35].

Furthermore, it is known that the flow rate is dependent on the flow part through engine compartment and only the radiator. In an optimal flow circuit, the air passing through the radiator needs to be channelled through the component in such a manner that a minimum back pressure is induced behind the radiator, to maximum flow rate. Changes to component or configuration not directly rated to the cooling system will, thus, also have an influence on the flow rate, as it will either increase or decrease the restriction in the flow path. For this reason, it is important to be aware of factors that will increase or decrease airflow through the radiator core. General finding in literature on case studies performed in this field give some indications on the effect that certain components have on the flow stream.

Quantifying the influence of individual components on the airflow rate through a radiator will be dependent on the specific vehicle evaluated. Nevertheless, commons trends are found in the literature. El-Bourrini [36] investigated the effects of the front-end components on the airflow through the engine compartment. By measuring the airflow rate at various positions on the radiator frontal surface, and calculating the mean airflow through the radiator, a relationship between the vehicle speed and the air speed through the radiator was obtained. It was concluded that the total assembly of all font-end components reduced the possible airflow by 84% in the case of his test vehicle. Only 48% of the reduction was caused by components involved in the cooling system of the vehicle while the remainder was due to other vehicle components.

In a similar experiment, Bahnsen [37] evaluated a vehicle for which the drag coefficient could be reduced by approximately 7% through covering the under body of the vehicle. A large reduction in fuel consumption was noticed with this configuration. An alarming restriction in the airflow through the radiator core was however noted, and the need to re-evaluate the cooling system was listed as a field for further investigation.

2.6. AIR SIDE HEAT TRANSFER COEFFICIENT.

Airflow over a louvered fin array is complex, as illustrated in Figure 2.15. Rather than acting as surface roughness or turbulent generators, the louvers are used to deflect airflow from their incident direction and consequently the flow becomes aligned with the plane of the louvers. They enhance the heat transfer by providing multiple leading edges, associated with high heat transfer coefficients.

Due to the complexity of airflow over louvered fins, it is difficult to determine the heat transfer coefficient from conventional convection heat transfer equations, as the coefficient is a function of both fin geometry and flow conditions. Therefore, there have been several detailed investigations of the flow phenomena in louvered fins over the last two decades, in order to have better understanding of the mechanisms of louvered heat transfer. The heat transfer characteristics of the louvers are closely related to the flow structure.

2.6.1. BRIEF DESCRIPTION OF FLOW STRUCTURE IN LOUVER FINS

Beauvais [39] was the first to discover that louvers act to realign the airflow in a direction parallel to their own planes after using flow visualization on large-scale models. Following Beauvais, Beard and Smith [40] also conducted hot-wire measurements for local air velocity over louvers and confirmed the validity to use large-scale models for evaluation of these surfaces.

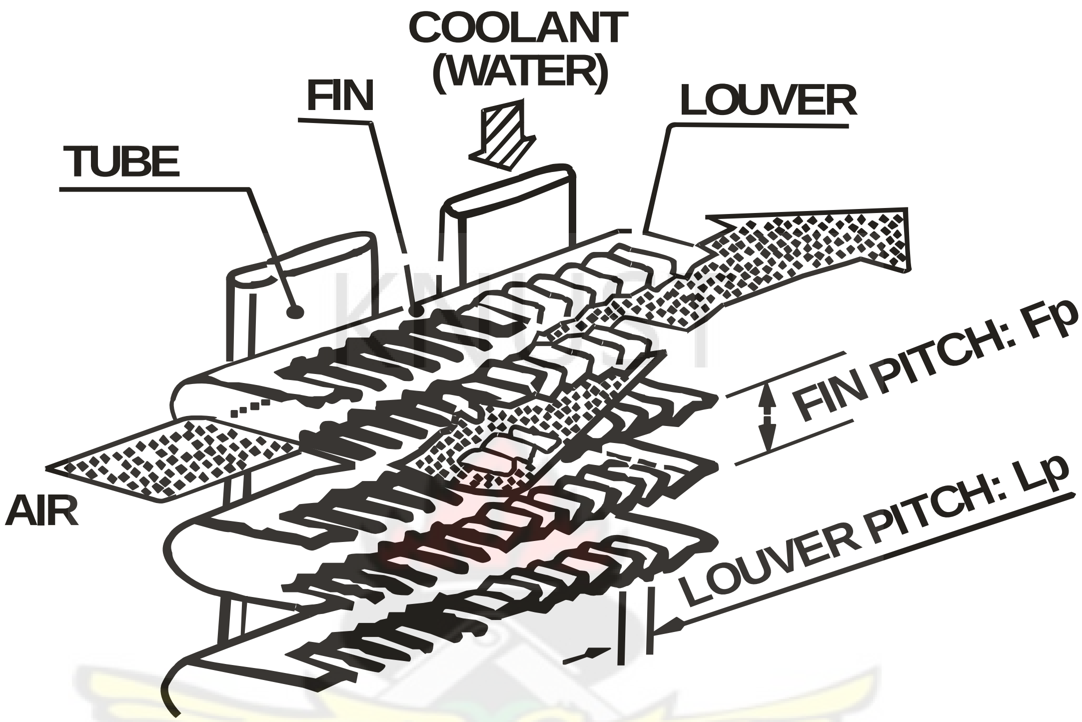


Figure 2.15: Multi-louvered fin geometry

They observed the flow-directing properties of the louvers, indicating the air traveling through the louver array not at all times parallel to the duct axis. Davenport [40] performed a detailed investigation on corrugated fin louvered radiator using smoke trace measurements, demonstrating that the flow structure within the louvered array was a function of Reynolds number. Figure 2.16 illustrates a section through a louver array in which two possible extreme flow directions are indicated.

Davenport found that at low Reynolds numbers the flow did not pass through the louvers but traveled axially through the fins and behaved like duct flow. Davenport explained that the developing boundary layers on the louvers became sufficiently thick to block off the gaps between louvers.

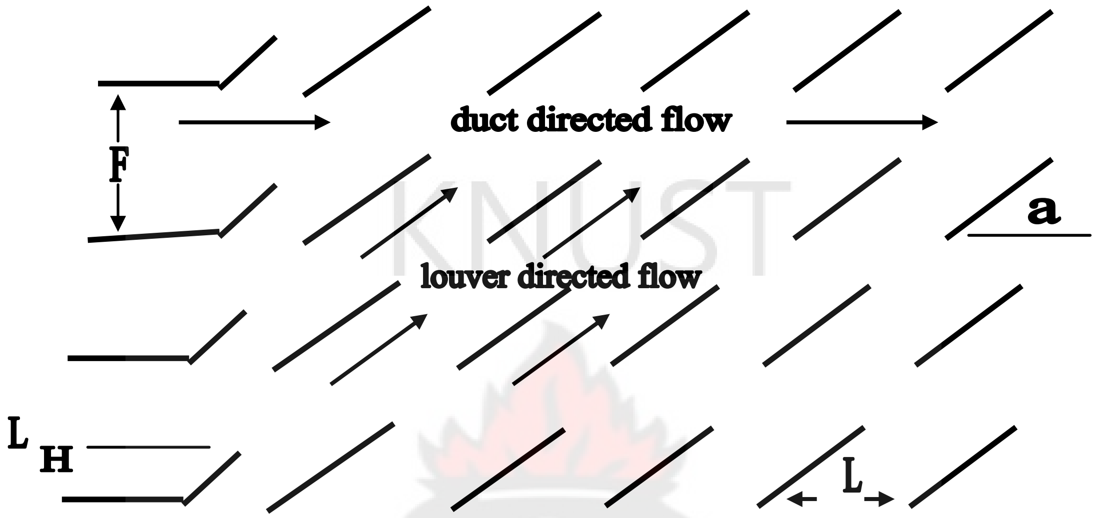
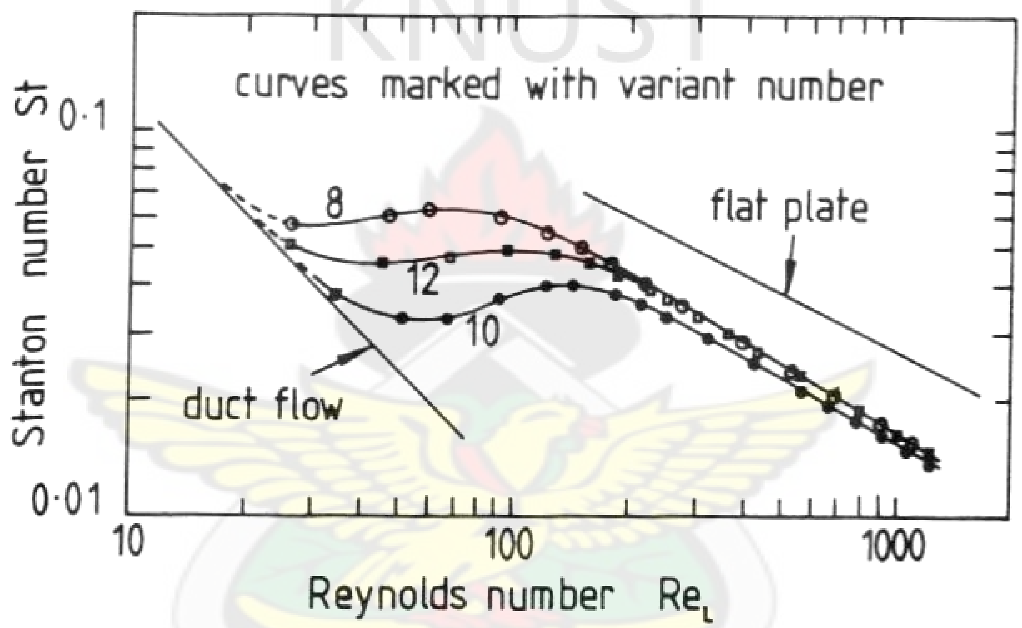


Figure 2.16: Section through louvered fin indicating possible flow directions

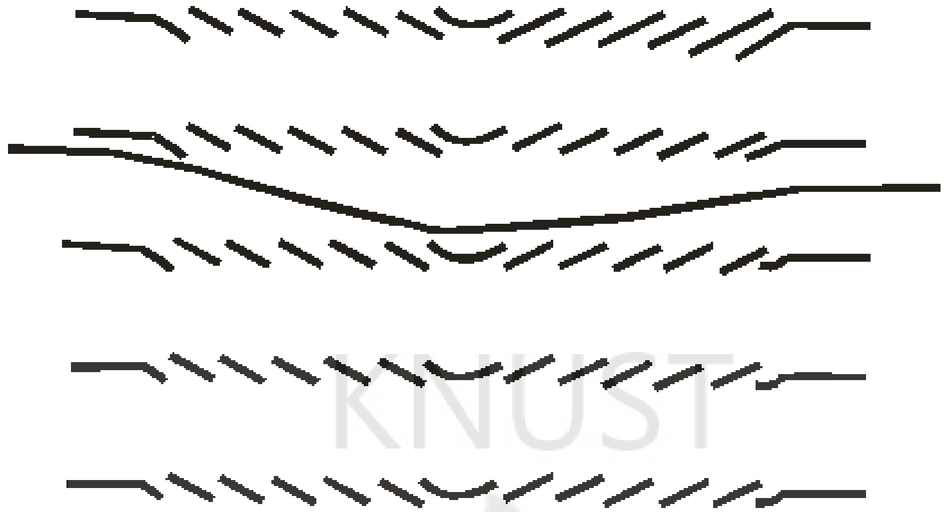
This gradually changed to an almost complete alignment with the louvers as Reynolds number was increased. At high Reynolds numbers, the flow was directed by the louvers flowing nearly parallel to them, behaving like flat-plate flow. In addition, Achaichia and Cowell [41] in a study of louvered fin radiator of the plate fin geometry, described a “flattening” behaviour of the Stanton number curves as shown in Figure 2.17. They explained that the behaviour was caused by the transition from duct flow to flat-plate flow, agreeing with the investigation by Davenport [40].

Another numerical study by Achaichia and Cowell [41] showed that the mean flow directions in the louvered fin array varied with Reynolds number, louver angle and the ratio of fin pitch to louver pitch [38] used a dye streamline method to visualize flow streamline in a 10 times scaled louvered fin model. Figure 2.18 shows the results of two louver arrays that have the same louver pitch but different fin pitches. It was shown that relative dimensions of the fin pitch and louver pitch significantly affected the airflow path.

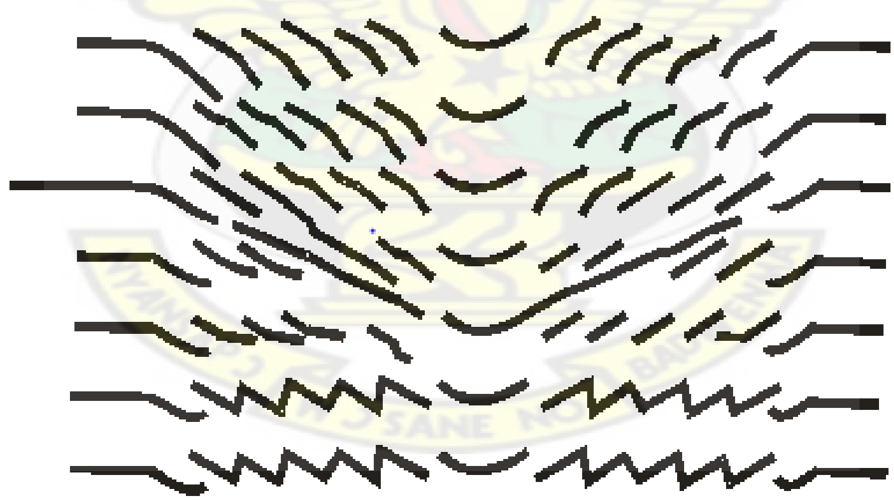


Variant	Fin Pitch (mm)	Louver Pitch (mm)	Louver Angle (deg)
8	2.11	0.81	29
10	3.33	0.81	29
12	2.16	0.81	20

Figure 2.17: Stanton number curves demonstrating transition from duct to flat-plate flow [41]



Large fin pitch, Fp =10mm



Small fin pitch, Fp = 5mm

Figure 2.18: Observed streamlines in louver fin array [38]

2.6.2 AIR-SIDE REYNOLDS NUMBER

The flow path over louvers is indicated to be dependent on Reynolds numbers for a given louvered fin array. However, it seems that the characteristic length is rather arbitrary for louvered fin surfaces. Therefore, it is important to understand which geometric parameters are important for heat transfer. Aoki et al. [42] state that the heat transfer performance of multi-louvered fins is influenced by fin geometry factors, such as fin pitch, louver pitch and louver angle. Davenport [16] after testing 32 samples of multi-louvered fin surfaces suggested that fin pitch and hydraulic diameter made no contribution to the correlation of Colburn's modulus j factor. Davenport concluded that, although the hydraulic diameter is relevant to heat transfer in plain fins, using the louver-pitch-based Reynolds number is more appropriate to describe the heat transfer on louvered fin surfaces. It can also help to reduce the number of variables in the heat transfer correlation. Most of the later research have been consistent with this finding and have used louver-pitch-based Reynolds number as a basis.

2.6.3. TYPES OF LOUVER FINS

The publication of heat transfer performance of corrugated louvered fin geometry which is the basic structure of modern radiators, has been very limited mainly for commercial reasons. Apart from the corrugated geometry, there are some other types of radiator geometry available [40]. These includes the flat tube and louvered plate fin, corrugated louvered fin with rectangular channel, corrugated louvered fin with splitter plate and rectangular channel and corrugated louvered fin with splitter plate and triangular channel [43].

One of the important studies on extended heat transfer surface was conducted by Kays and London [45] who published the first reliable and comprehensive test data on louvered surfaces for compact radiators.

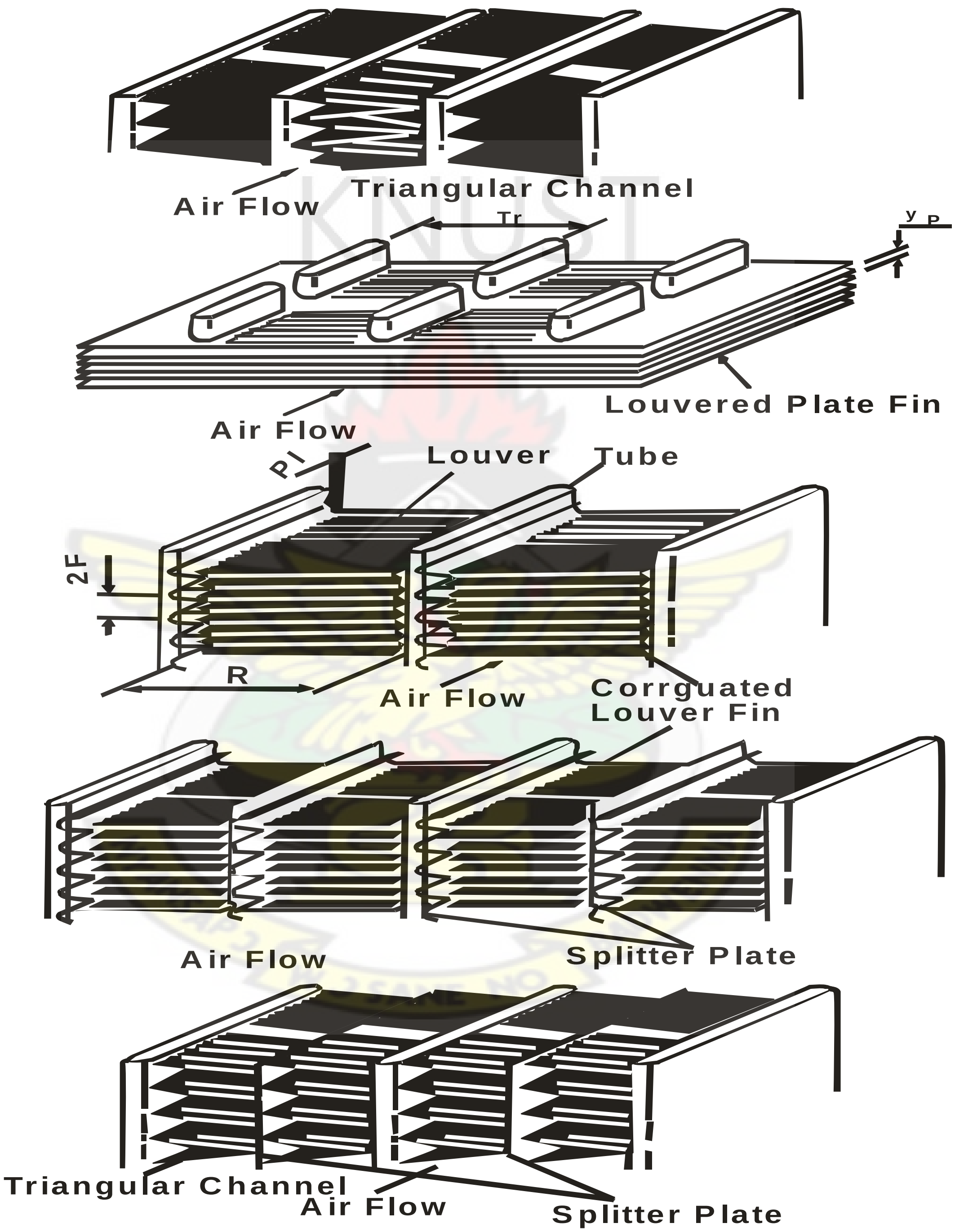


Figure 2.19: Types of louver fin heat Exchanger

The work provided valuable fundamental knowledge in this area. However, those louver designs are very different from the multi-louvered surface, which are widely used nowadays for automotive radiator cores, and the data are of little relevance when applied to modern radiators. In addition to Kays and London [45], there have been several research publications in regard to the heat transfer performance of louvered fin surfaces [46]. Types of louver fin include corrugated louver with triangular channel, corrugated louver with rectangular channel, corrugated louver with plate and rectangular channel, corrugated louver with splitter plate and triangular channel and plate-and-tube louver fin geometry (as shown above in Figure 2.19) .

2.6.4. COOLANT SIDE HEAT TRANSFER COEFFICIENT.

It is emphasized that the dominant thermal resistance is air-side convection and that the modeling is relatively insensitive to water side heat transfer. The coolant-side heat transfer coefficient is addressed in this section and can be evaluated by applying appropriate well-established heat transfer equations for flow inside tubes. The solutions to internal flow problem may be obtained from analytical derivations and empirical correlations. Conventional heat transfer equations for flow in tubes are often based on the consideration of circular tubes, and the coolant tubes in typical radiators are flat-oval shape.

The fluid flow condition in tubes can be laminar, turbulent or transition, as characterized by the fluid-flow Reynolds number based on the tube diameter, or on the hydraulic mean diameter for non-circular tube cross-sections. In laminar flow, the fluid particles move in a direction parallel to the tube walls and there is no velocity component normal to the axis of the tube. In contrast, by definition, turbulent flow is time-dependent, which is due to relatively small-scale vorticity in the flow. Its particles fluctuate in an irregular way around a steady time independent velocity. Although the flow has an average velocity parallel to the axis of the tube, it has instantaneous velocities both parallel and perpendicular to the axis.

There has been a large amount of fundamental work in understanding flow characteristics and heat transfer in tubes, it has been demonstrated that laminar flow becomes unstable as the velocity of flow increases in a given tube. The transition from laminar to turbulent flow occurs at a value of Reynolds number near 2300. The transition to turbulent flow generally takes place in the range of Reynolds number from about 2300 to 10000, and a fully turbulent flow mostly occurs at the Reynolds number above 10000. It has also been found that the transition of the flow would be greatly affected by the tube inlet configuration and surface roughness. In this analysis, the definitions for the various flow regimes in relation to the Reynolds numbers are listed below in Table 2.1.

Table 2.1: Flow classification

Laminar	2300 > ReDh
Transition	10000 ≥ ReDh ≥ 2300
Turbulent	ReDh > 10000

Source: Kays and London [45]

CHAPTER THREE EXPERIMENTAL SETUP

3.1 MATERIALS

This chapter gives a complete description of the experiments conducted on a four cylinder four stroke engine and the equations used for the model.

3.1.1. ENGINE

The engine used for the experiments was the four-cylinder four stroke engine shown in Figure 3.1. The engine is a water cooled engine with a tank capacity of 5.5 litres. It is a petrol engine which has a compression ratio of 9.8 to 1 and operates on the Otto-cycle. The other specifications of the engine are given in Table 3.1.

Figure 3.1: Engine for setup experimental

Table 3.1: Engine Specification

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

3.1.2. RADIATOR

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.

3.1.3. THERMOMETERS

Two different types of thermometers were used for the experiment. Two thermocouple probes were used to read the input and the output temperatures coolant in the radiator. The thermocouple was manufactured by Cola-Parma Instrument Company with model number 8110-10 and an accuracy of 0.10 ^{0}C. A mercury-in bulb thermometer was used to measure the ambient temperature.

3.1.4. COVERING MATERIAL

Two different soil types were used. Silt and clay soil were chosen for the experiment because of their ability to stick to the radiator surface area.

3.2 EXPERIMENTAL SETUP

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 either clay or silt 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.

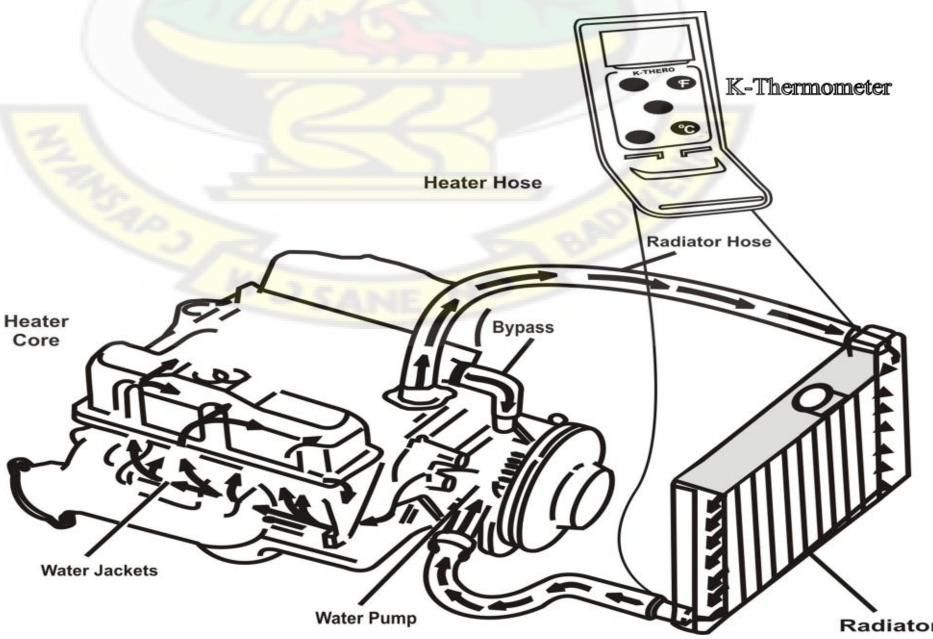


Figure 3.2: Schematic of the experimental set-up

3.3 RADIATOR HEAT TRANSFER MODEL

There are two common models available in literature for analysing the heat transfer process in a radiator. The two methods are;

- The log Mean Temperature Difference (LMTD) and
- Effectiveness – Number of Heat Transfer Units (ε -NTU) method

In modelling the radiator, the ε -NTU was used because, the focus of the thesis is on the outlet and inlet temperature of the water after the effective heat transfer area is then partially covered with either clay or silt soil to the required percentage before running the engine. When the inlet or outlet temperatures are to be estimated, the ε -NTU method comes in handy. The model was developed to predict the outlet temperature of the coolant from the radiator given the dimensions of the radiator, the flow rates of the fluids and the cooling load. In order to solve the analytical model, the following assumptions were made;

1. Constant coolant flow rate and fluid temperatures at both the inlet and outlet temperatures, that the system operated at steady state
2. There were no phase changes in the coolant
3. Heat conduction through the walls of the coolant tube was negligible
4. Heat loss by coolant was only transferred to the cooling air, thus no other heat transfer mode such as radiation was considered
5. Coolant fluid flow was in a fully developed condition in each tube
6. All dimensions were uniform throughout the radiator and the heat transfer of surface area was consistent and distributed uniformly
7. Pure water was used as the coolant

8. The thermal conductivity of the radiator material was considered to be constant
9. There were no heat sources and sinks within the radiator
10. There was no fluid stratification, losses and flow misdistribution.

The heat transfer process in the radiator was modelled as a forced convective heat transfer operation. The following equations for surface areas from Lin [47], were used. The parameters for calculating the heat transfer area are given in Table 3.2 below. The relevant equations are given from equation 3.1 to 3.7

Table 3.2: 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

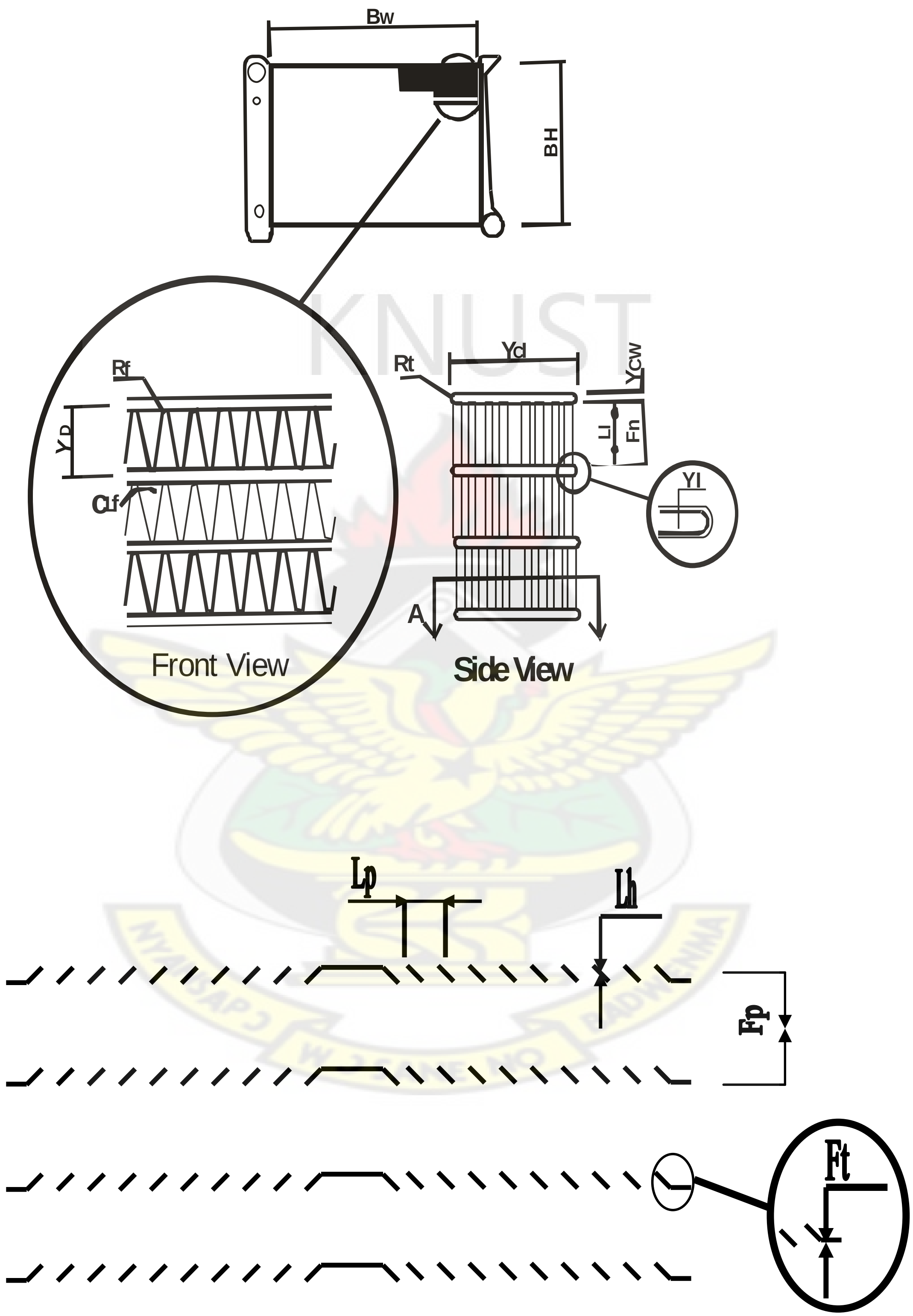


Figure 3.3 Definitions of the radiator parameters

The fin length, F_l is given by;

$$F_l = \pi \mathrm{R_f} + \frac{\mathrm{F_h} - 2\mathrm{R_f}}{\cos\alpha} \quad \text{.........} 3.1$$

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

$$A_{fr} = B_H B_w \quad \text{.........} 3.2$$

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

$$A_{fr.t} = Y_{cw} Y_l N_{ct} \quad \text{.........} 3.3$$

Fin heat transfer area, A_f, is given by;

$$A_f = 2. B_T F_T N_f Y_l N_p \quad \text{.........} 3.4$$

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

$$A_a = A_f + 2N_{ct} Y_l N_f [(Y_{cl} - 2R_t) + (2\pi R_t)] \quad \text{.........} 3.5$$

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

$$A_C = [2\pi(R_t - Y_t) + 2(Y_{cl} - 2R_t)] Y_l N_{ct} N_r \quad \text{.........} 3.6$$

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

$$A_{p,c} = [\pi(R_t - Y_t)^2 + (Y_{cw} - 2Y_t)(Y_{cl} - 2R_t)] N_{ct} N_f \quad \text{.........} 3.7$$

The following dimensionless groups for convective heat transfer were used;

- Reynolds number – which may de defined as ratio of flow momentum rate (inertia force) to viscous force for a particular geometry. It is given by;

  $$Re = \frac{\rho VL}{\mu} \qquad \ldots\ldots 3.8$$

- Nusselt number – which is defined as the ratio of the convective heat transfer coefficient (h) to the pure molecular thermal conductance (k/L), thus,

  $$Nu = \frac{hL}{k} \qquad \ldots\ldots 3.9$$

- Prandtl number – defined as the ratio of momentum heat diffusivity to thermal diffusivity of the fluid. It is solely a fluid property modulus

  $$Pr = \frac{v}{\alpha} = \frac{\mu C_p}{k} \qquad \ldots\ldots 3.10$$

Where 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

3.4 OVERALL HEAT TRANSFER COEFFICIENT

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

- Wall conductance
- Fouling on the air side
- Air side convection
- Fouling on the coolant side
- Coolant-side convection

Mathematically, it can be defined as;

$$\frac{1}{UA} = \frac{1}{(n_o hA)_a} + R_{fa} + \frac{\Delta x}{(kA)_w} + \frac{1}{(n_o hA)_c} \qquad \text{...3.11}$$

Where U = the overall heat transfer coefficient

h = heat transfer coefficient

n_o = total surface efficiency of an extended fin surface

R_f = fouling factor

Subscript a, c and w refer to air side, coolant side and coolant wall respectively.

For purposes of simplicity, the fouling factor on both the air side and the coolant side was assumed to be negligible. Also, the thermal resistance due to the coolant wall was assumed to be negligible as compared with the other terms because the coolant tube is most often made with either copper or aluminium both of which have large thermal conductivities whilst the thickness of the tube is usually small. The total surface efficiency of the fin was also assumed to be unity for purpose of simplicity.

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

$$\frac{1}{UA_{fr,r}} = \frac{1}{h_a A_a} + \frac{1}{h_c A_c} \qquad \text{......} 3.12$$

Where $A_{fr,r}$ = radiator core frontal area

A_a = total heat transfer area on air side

A_c = total heat transfer area on coolant side

h_a = heat transfer coefficient on air side

h_c = heat transfer coefficient on coolant side

The air side heat transfer coefficient was taken from Davenport [16] and is given by;

$$h_a = \left[0.249 Re_{Ll}^{-0.42} L_h^{0.33} \left(\frac{L_h}{F_h}\right)^{1.1} F_h^{0.26}\right] \frac{\rho V_a C_{p,a}}{P_r^{2/3}} \qquad \text{......} 3.13$$

Where L_h = louver height

L_l = louver length

F_h = Fin height

Other parameters retain their meaning as already defined above.

The coolant side Nusselt number was taken from Holman [9] is given by;

$$Nu_c = 3.66 + \frac{0.0668\left(\frac{D_{h,c}}{Y_1}\right) RePr}{1+0.04\left[\left(\frac{D_{h,c}}{Y_1}\right) RePr\right]^{2/3}} \qquad \text{......} 3.14$$

This equation was employed because of the laminar nature of the fluid flow in our radiator. Other correlations may be applicable depending on the nature of fluid flow – transition or turbulent.

Where D_h = hydraulic diameter

Y_l = coolant tube length

3.4.1 THE Ɛ -NTU METHOD

The heat transfer rate in the radiator is given by;

$$Q = \varepsilon C_{min}(T_{ci} - T_{ai}) \quad \text{.......} 3.15$$

Where C_{min} = minimum heat capacity rate

The radiator thermal efficiency (ε) is defined as the ratio of the actual transfer rate from the hot fluid (coolant) to the cold fluid (air) in a given radiator to the maximum possible heat transfer rate. It is expressed as;

$$\varepsilon = \frac{Q}{Q_{max}} \quad \text{.......} 3.16$$

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;

$$Q = C_c(T_{ci} - T_{co}) = C_a(T_{ao} - T_{ai}) \quad \text{.......} 3.17$$

Where C_a = heat capacity of air

C_c = heat capacity of coolant

T_{ci} = Coolant inlet temperature

T_{co} = Coolant outlet temperature

T_{ao} = Air outlet temperature

T_{ai} = Air inlet temperature

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

For air: $C_a = m_a x C_{p,a} = A_a \rho_a V_a C_{p,a}$………..3.19

For coolant: $C_c = m_c x C_{p,c} = A_c \rho_c V_c C_{p,c}$ ………................................3.20

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;

$$C_r = \frac{C_{min}}{C_{max}} \quad \text{...3.21}$$

Where C_{min} is the smaller of C_a and C_c. According to SAE JI393 (1996), the minimum capacity rate C_{min} is always on the air side. Hence

$C_{min} = C_a$ and $C_{max} = C_c$. 3.22

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

$$Q = \varepsilon C_a (T_{ci} - T_{ai}) \quad \text{..3.23}$$

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

$$NTU = \frac{UA_{fr,r}}{C_{min}} = \frac{1}{C_{min}} \int_A U . dA_{fr,r} \quad \text{..3.24}$$

The radiator effectiveness is defined as a function of both the NTU and the C_r by Kays and London (1998) and is given by;

$$\varepsilon = 1 - exp\left\{\frac{NTU^{0.22}}{C_r}\left[exp(-C_r . NTU^{0.78}) - 1\right]\right\} \quad \text{.......................................3.25}$$

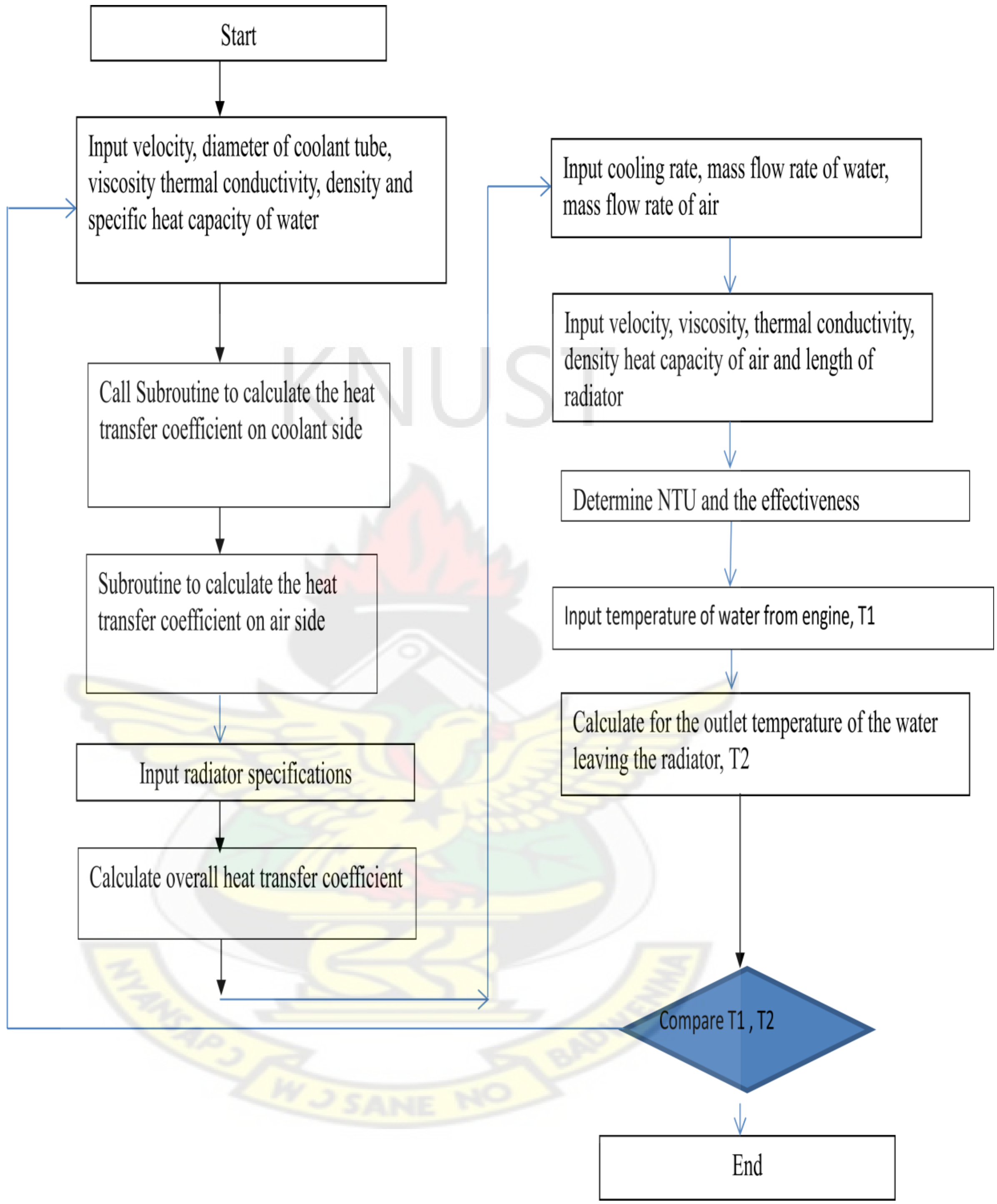


Figure 3.4: The Algorithm for running the model.

Based on the equations outlined above, the model was developed using MATLAB and run with the algorithm in figure 3.4

CHAPTER FOUR RESULTS AND DISCUSSION

This chapter discuses results obtained from the running of the experiments and compares it with results of the model developed.

4.1 RESULTS OBTAINED USING CLAY AS THE COVER MATERIAL

As shown in Figure 4.1 below, the temperature of coolant from the radiator increased with an increase in the area of the radiator covered increased. As the area covered increased, there was also a corresponding increase in the temperature of the coolant coming from the engine.

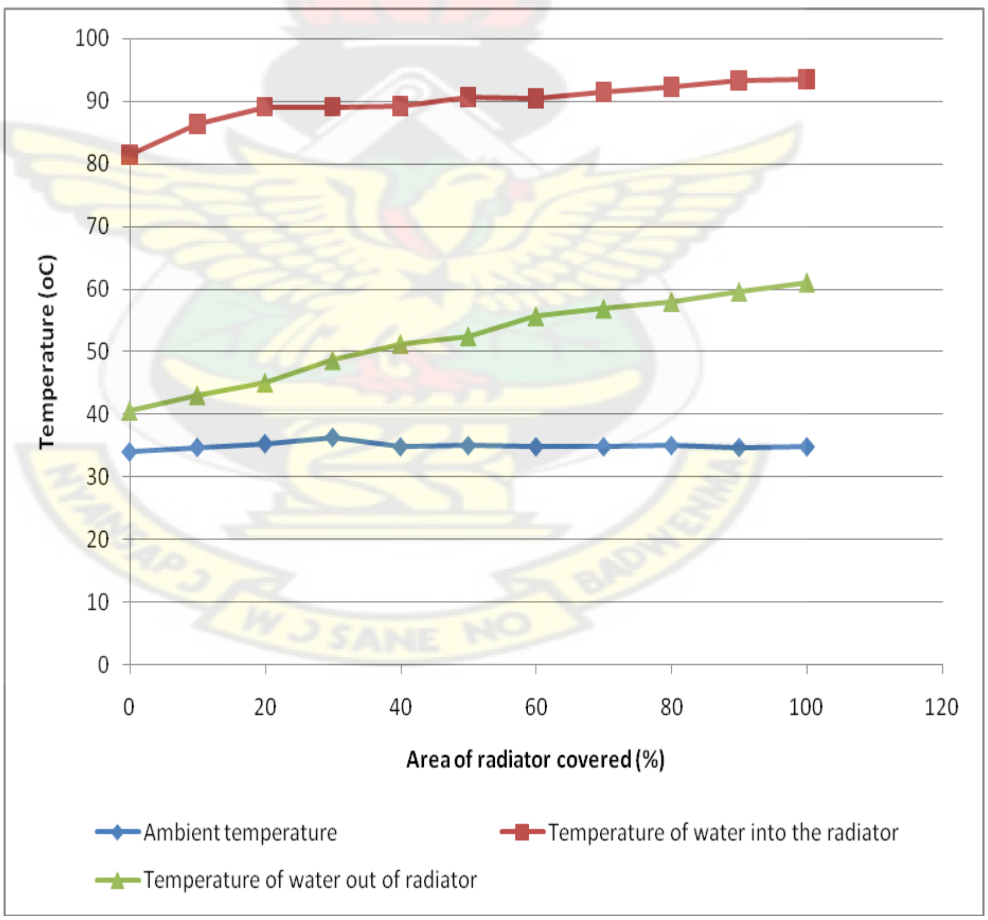


Figure 4.1: Graph of Temperature verse Area covered using clay to cover the radiator

The highest temperature obtained for the coolant into the radiator was achieved when the 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 from taking 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.

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

4.2 RESULTS OBTAINED USING SILT AS THE COVERING MATERIAL

As shown in Figure 4.2 below, the results obtained using silt as the covering material was identical to that of clay as covering material. It can be observed that as the percentage area covered increased, there was a corresponding increase in the outlet and inlet temperatures of the coolant fluid. When 80% of the radiator surface was covered, it was observed that the engine idling was not stable and the engine vibrated excessively. Also, the water within the expansion tank began to overflow just as it was observed when clay was used as the covering material. At 100% covering, the engine could not stay on and after a short time of running the engine stopped running. The highest inlet temperature of the coolant recorded was 91^{o}C and this was when100 % of the radiator surface was covered.

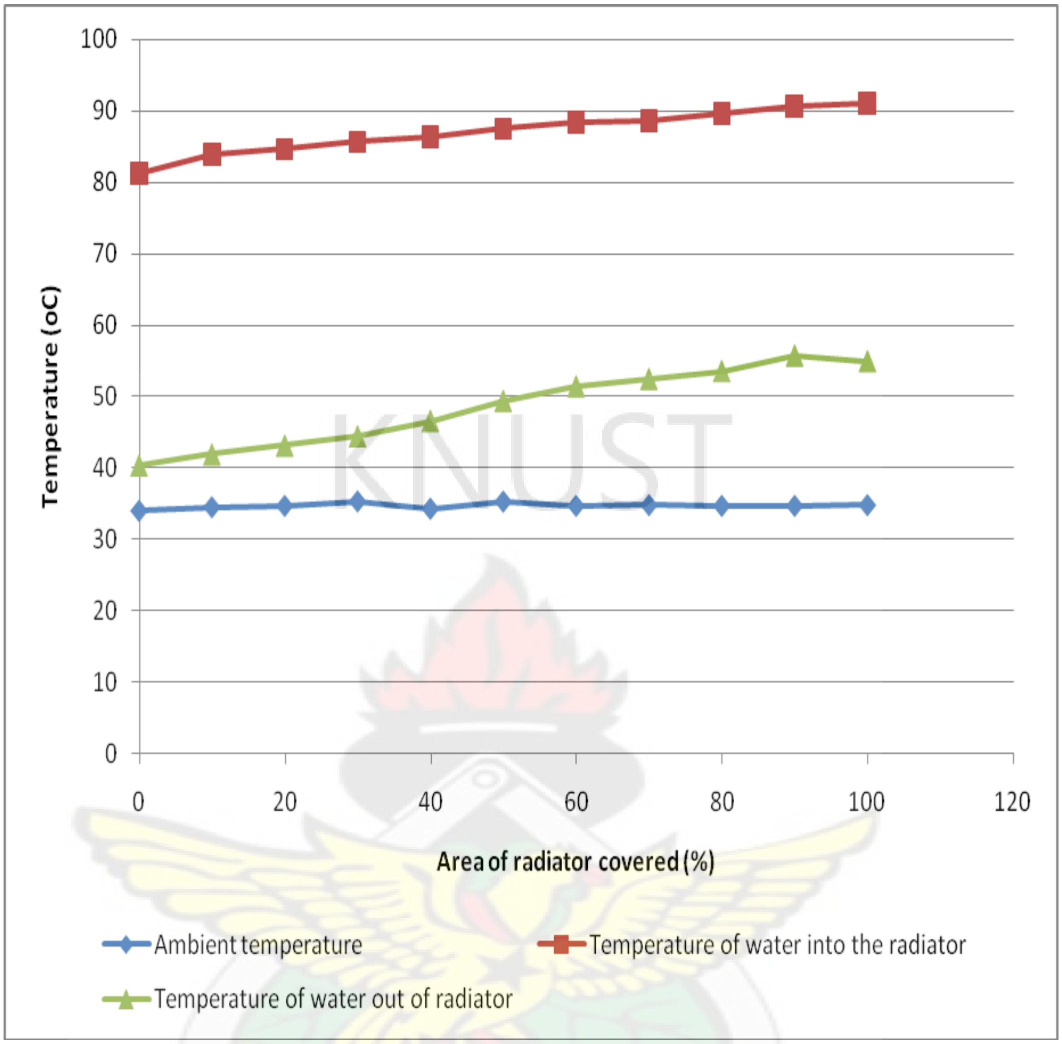


Figure 4.2: Graph of results obtained using silt to cover the radiator

4.3 DISCUSSIONS

When results obtained using clay was compared to that obtained using silt as shown in Figures 4.3 and 4.4, it was observed that the coolant temperature into the radiator was higher when clay was used as the cover material than when silt was used. This could be explained by the fact that clay stuck better on the radiator surface than silt. Because silt is loose, pocket of air space was observed on the covered radiator surface. This resulted in some amount of air passing through the air spaces to aid in the cooling of the coolant to a lower temperature than when clay was used. Since less heat was removed from the coolant, the resultant temperature of the coolant obtained from the radiator was also found to be higher for clay than silt as shown in Figure 4.4 below.

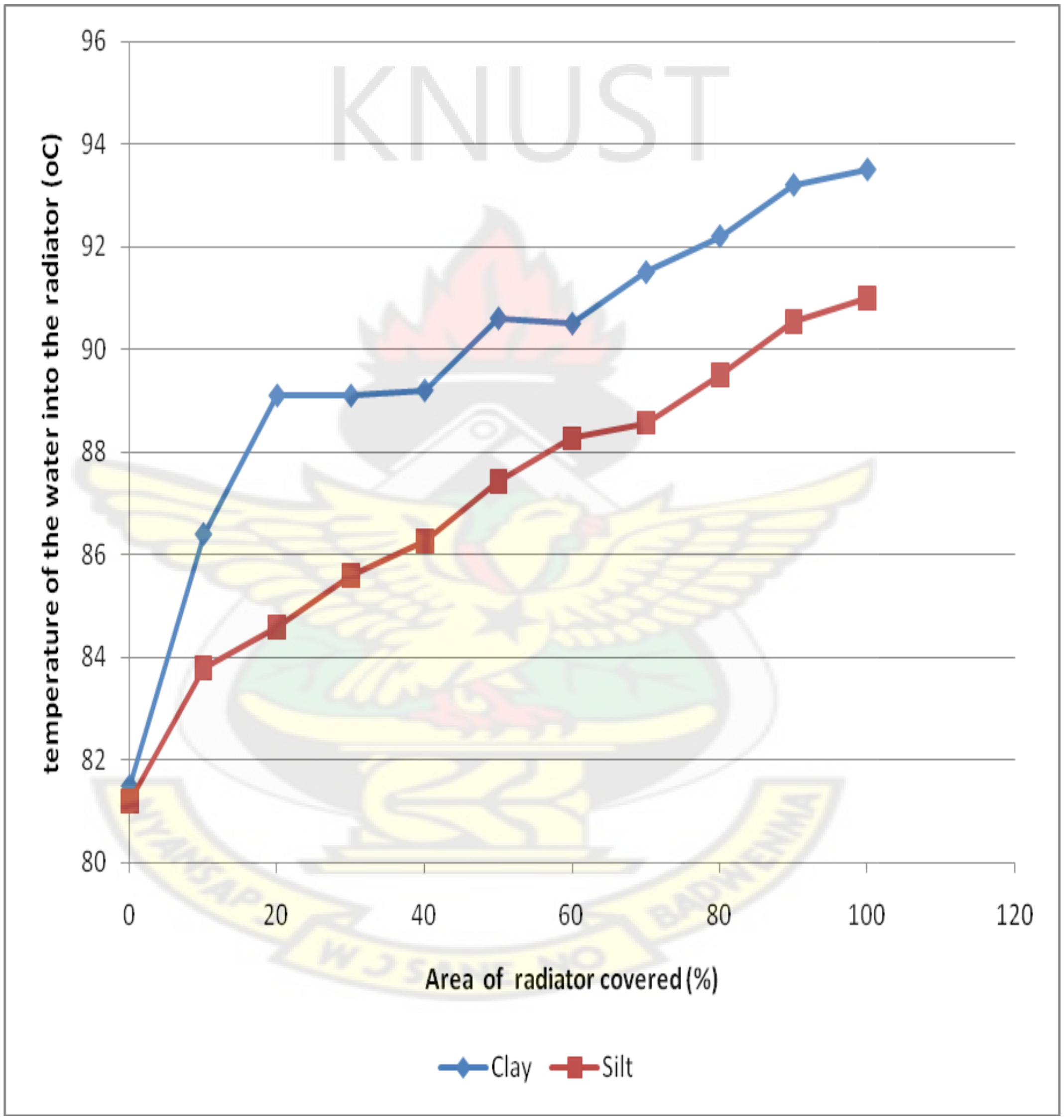


Figure 4.3: Graph of inlet temperature of water into the radiator against the percentage area covered

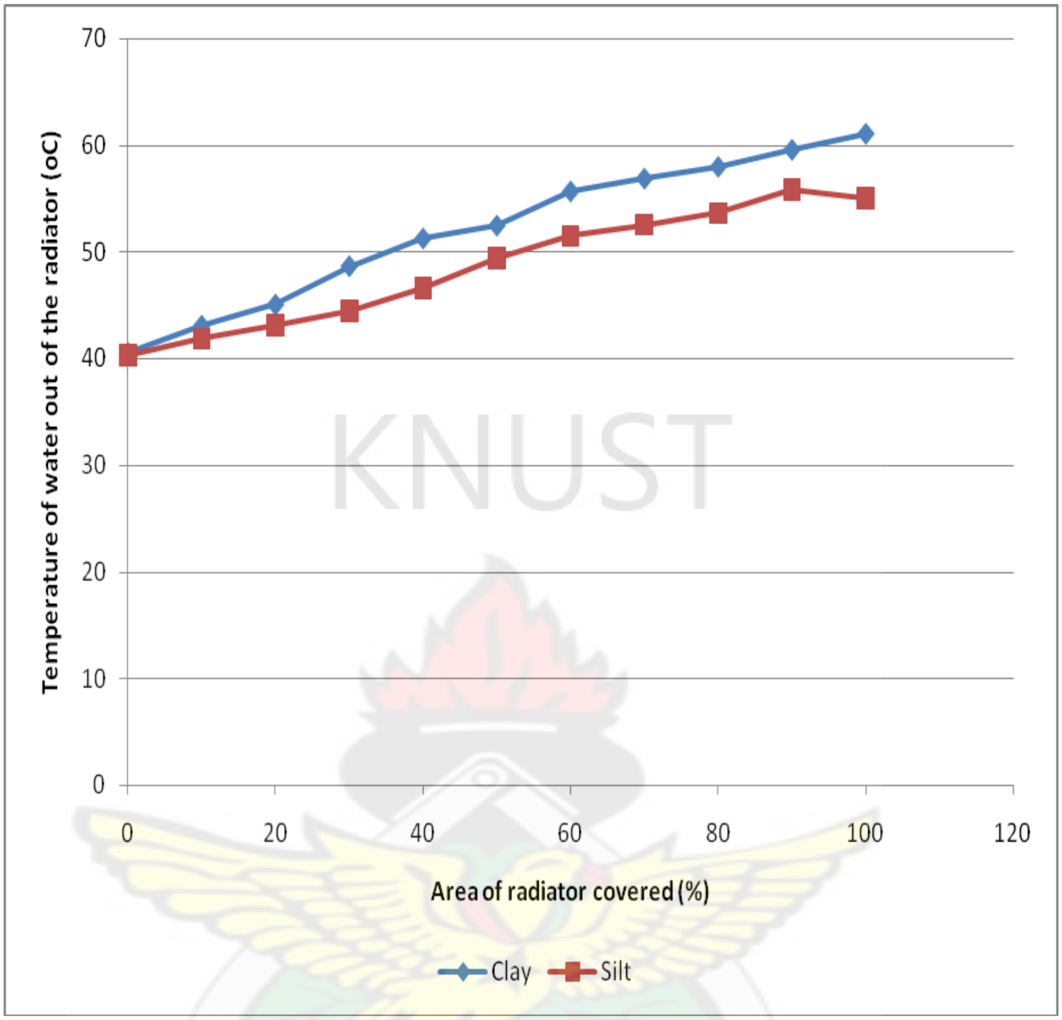


Figure 4.4: Graph of outlet temperature of water into the radiator against the percentage area covered

Also, when the result of the model was compared to the results obtained for the experiment as shown in Figure 4.5, it was observed that 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 when clay was the covering material better than silt. This is perhaps due to the fact that clay was a better coverage material than silt as already stated above. 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. In all cases this trend of increasing outlet temperature of the coolant from the radiator was observed. As shown in the figure 4.5 below, 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.

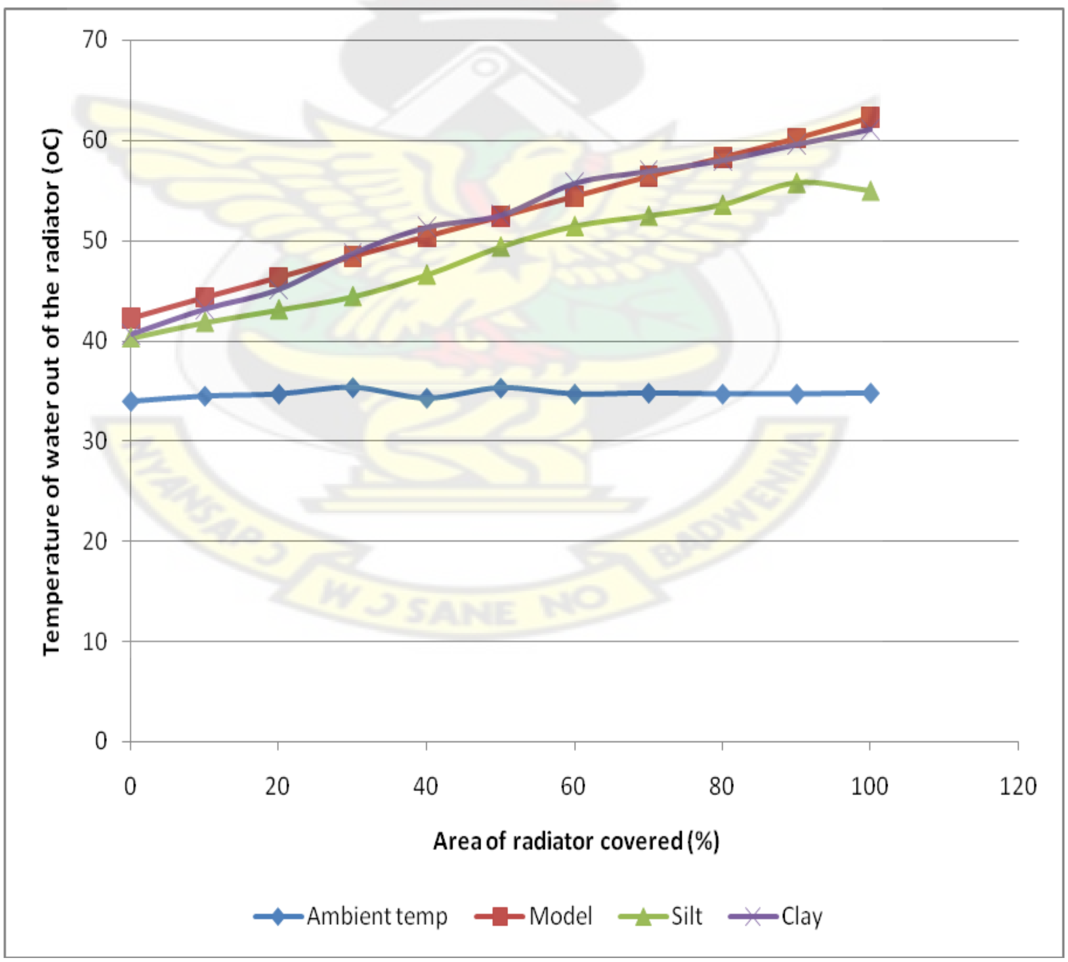


Figure 4.5: Comparison of model results of temperature of water out of the radiator with results obtained with silt and clay

Comparing the temperature of the coolant into the radiator for three scenarios – the model, clay covering and silt covering, it can be observed in Figure 4.6 that the results obtained were similar. The model was able to predict the heat transfer process quite well. In all the three cases the temperature of the coolant into the radiator increased as the area of the radiator covered also increased. This is due to the fact that when the surface area of the radiator is reduced by covering, less heat is taken out of the coolant, thus increasing the outlet temperature of the coolant from the radiator as observed above. Since the coolant is not well cooled, it picks up heat from the engine which increases its temperature higher than expected before entering the radiator.

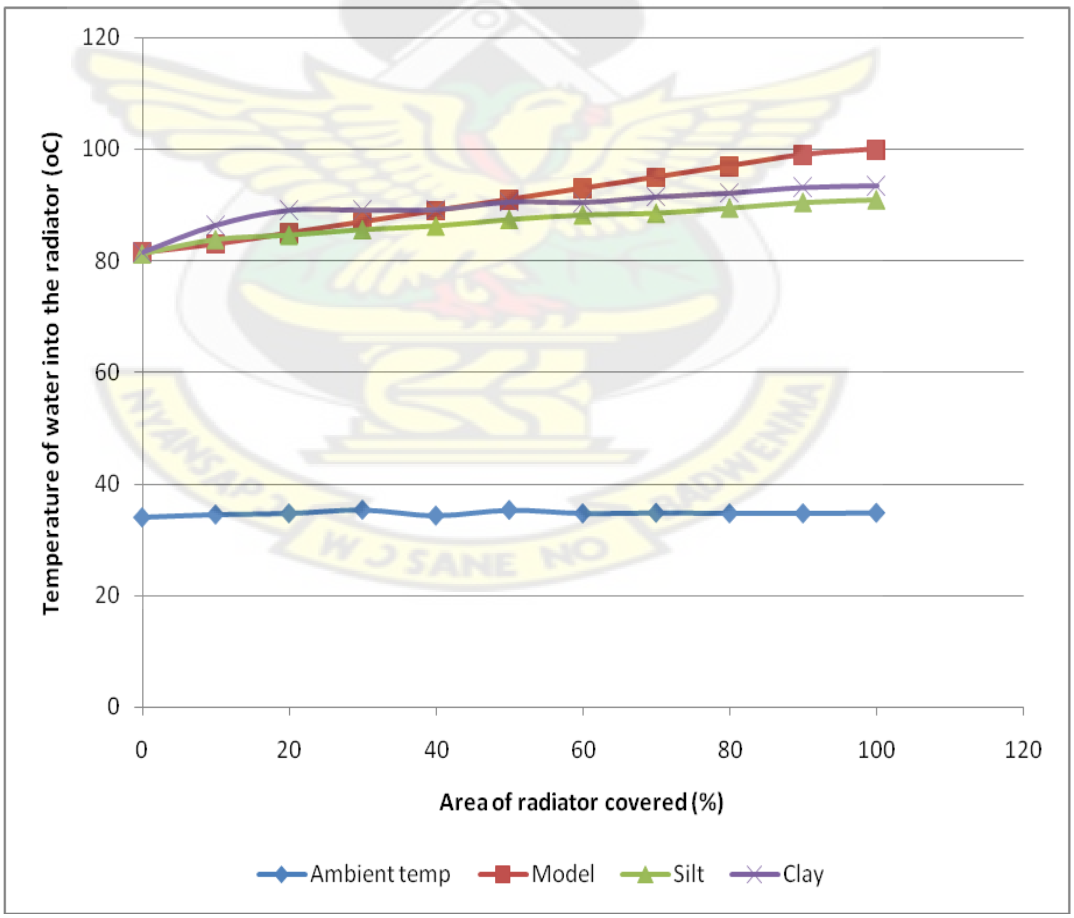


Figure 4.6: Comparison of results of inlet temperature of water into the radiator for the model with that of silt and clay

Presenting the result of Figure 4.7 and Figure 4.8 shows the graph of time against the outlet temperature for clay and silt. It was generally observed from the graph that, as the percentage of area covered increases there was a corresponding increased in the outlet temperature.

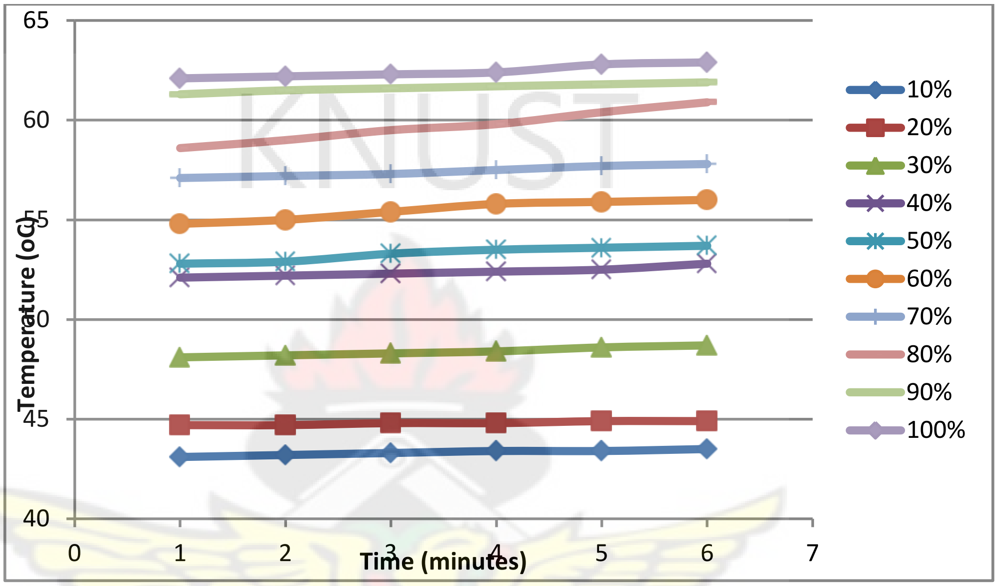


Figure 4.7: Graph of outlet Temperature of clay against Time

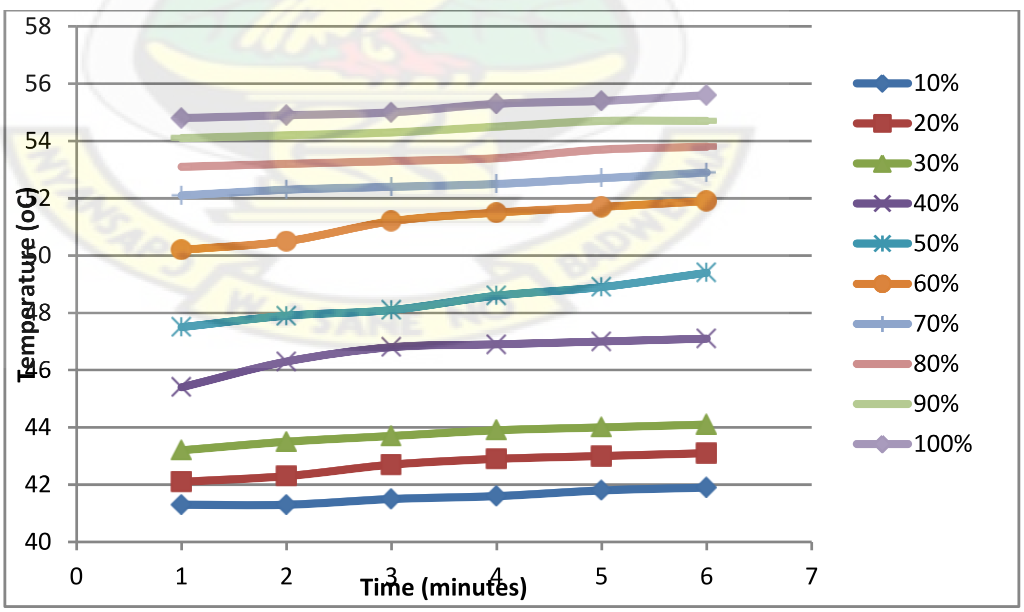


Figure4.8: Graph of outlet Temperature of clay against Time

Presented in Figures 4.9, 4.10, and 4.11, are the graphical regression relationships between the radiator coolant outlet temperatures and the proportion of the radiator surface area covered with clay, silt 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 4.1 shows the regression models used to predict the outlet temperatures of radiator coolant from the percentage of area covered of the radiator. The three models were developed using silt and clay to cover the radiator. The Matlab model was a simulated model to evaluate the validity of the silt and clay models. All the three models predicted well and were statistically significant at a 1% probability level. The coefficient of determination (R^2) of the three 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 silt model, for every 10% increase or decrease of the area covered of the radiator resulted in an average increase or decrease of about 1.7 degree Celsius of the outlet temperature of the radiator coolant. In the case of the clay model, a 10% increase or decrease of the area covered of the radiator resulted in about 2.1 degree Celsius of the outlet temperature of the radiator coolant (Table 4.1). In the Matlab model, a change in 10% of the area covered of the radiator resulted in a change of about 2 degree Celsius of the outlet temperature of the radiator coolant.

4.4 REGRESSION AND ANOVA ANALYSIS

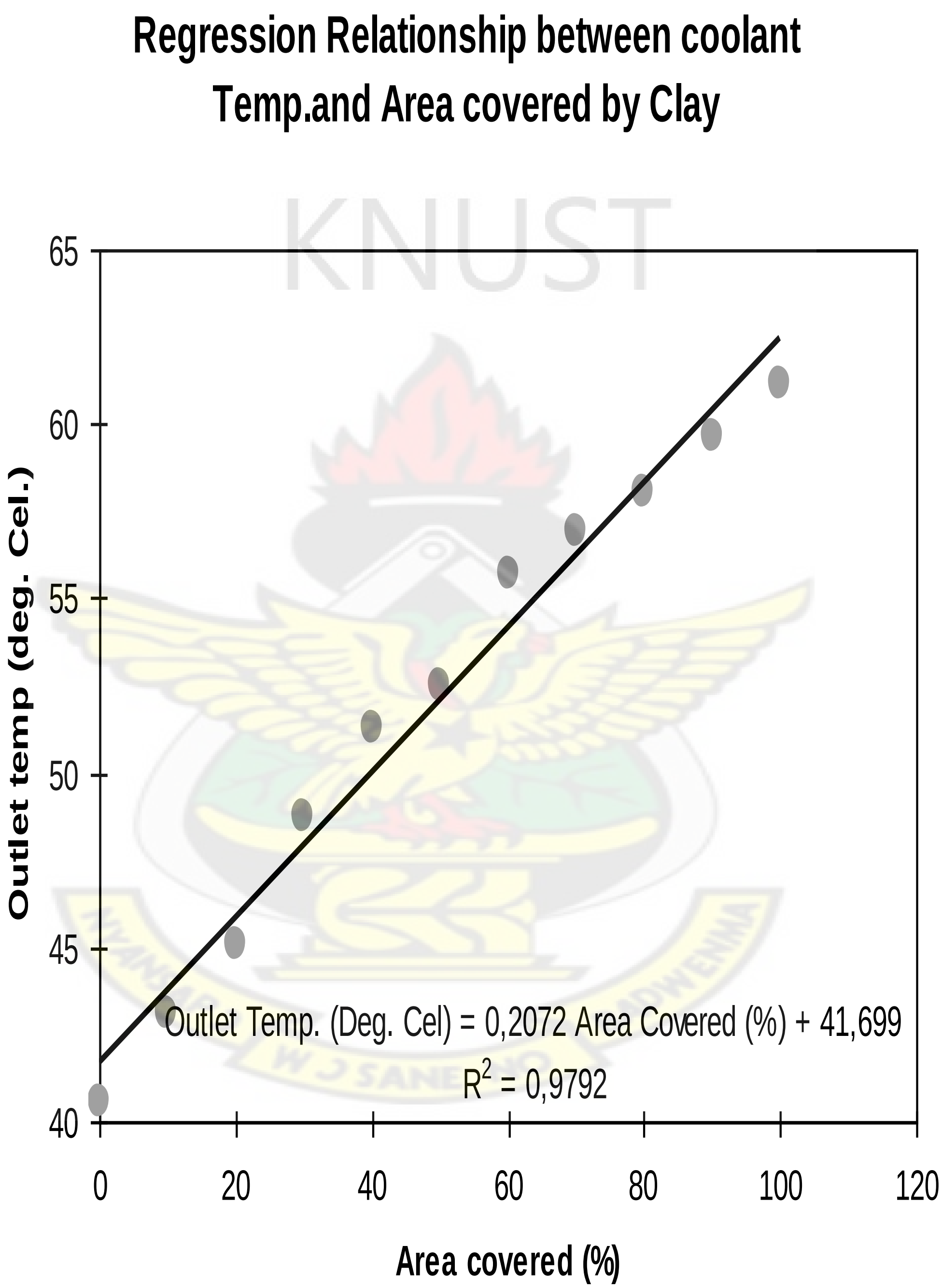


Figure 4.9: Regression Relationship between Radiator outlet Temperature and the Percentage of Radiator Surface Area Covered with Clay

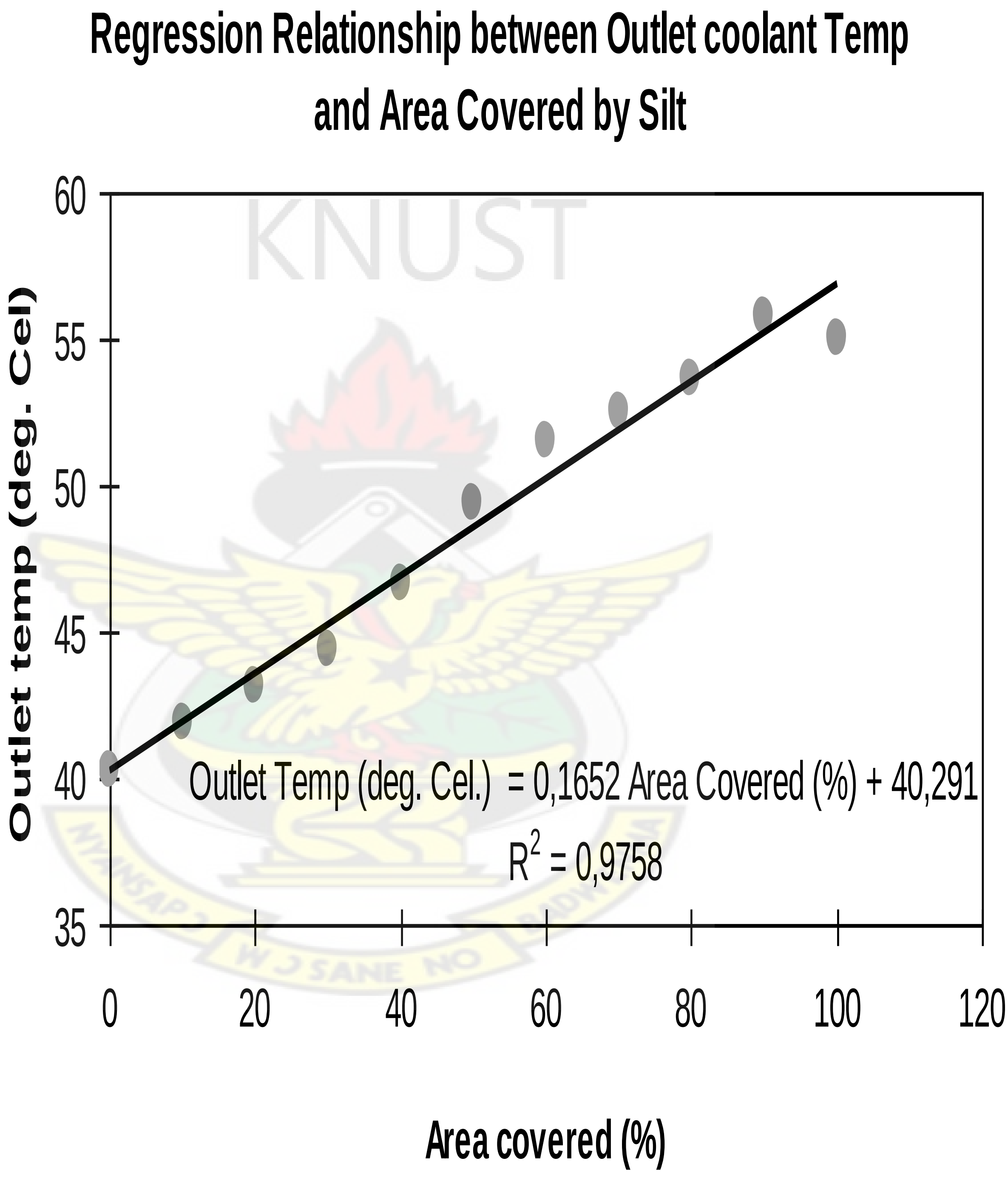


Figure 4.10: Regression Relationship between Radiator outlet Temperature and the Percentage of Radiator Surface Area Covered with Silt

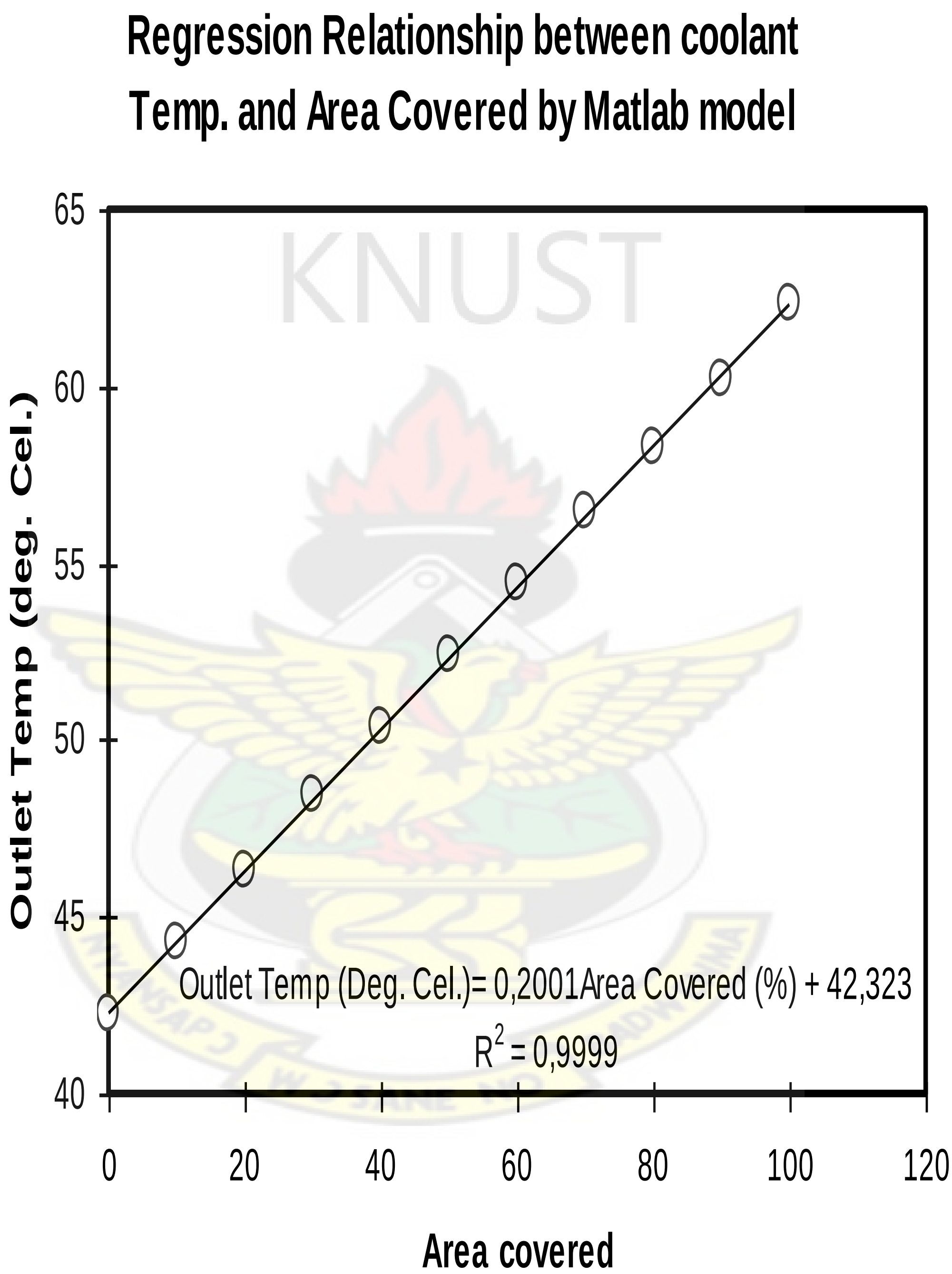


Figure 4.11: Regression Relationship between Radiator outlet Temperature and the Percentage of Radiator Surface Area Covered using Matlab Simulation

Table 4.1: 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**
Silt	T=0.1652A+40.29	0.9858	78.558	<0.0001
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

Analysis of Variance (ANOVA) was conducted to find out if there were statistically significant differences among the three models (Table 4.2). As shown in Table 4.2 there were no statistically significant differences among the three models. This indicates that the Matlab model was consistent with clay and silt models. It can also be said when the same amount of clay or silt covers the same area of a radiator the same temperature rise of the radiator coolant will result.

Table 4.2: 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	1425	0.255
Within Groups	1230,285	33	37,283		
Total	1336,531	35			

CHAPTER FIVE CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

Experiments were successfully conducted on the radiator of a four cylinder diesel at base load. Two different soil types, namely silt and clay, were used as covering material to cover the heat transfer area of the radiator to determine the heat transfer process during the running of the engine. It was observed that the inlet temperature of the coolant in the radiator increased as the percentage area of the radiator covered increased. It was also observed that the outlet temperature of the coolant from the radiator increased monotonically with increases in the percentage area of the radiator covered. In both cases, at 80% coverage of the heat transfer area of the radiator the engine vibrated excessively and the idling was not stable. The engine stopped running after a short while. At 100% coverage the engine stopped running immediately after starting. This phenomenon was expected because as the effective transfer area of the radiator is reduced, less heat is taken out and thereby affecting the temperature of the coolant and hence the performance of the engine.

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.

Clay was found as a better covering material as compared to silt. This is evident in the outlet temperature of the coolant from the radiator. 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 silt, 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.

5.2 RECOMMENDATIONS

The following recommendations have been arrived at to enable further research work to be carried out;

- Short engine test was conducted; however it is recommended that any future research should consider a medium to long term engine test where the engine will be run continuously for a day or two. This will help to establish the long term effect on the engine if part of the radiator is covered or blocked with sand.
- Routine maintenance should be carried out regularly to reduce the effect of blockage on the radiator surface area.
- This thesis did not look at the effect of fin angle on the heat transfer process. It is, therefore, recommended that any future work can also look at the effect of fin angle on the heat transfer process. As has been shown in the mathematical model above, the heat transfer process is dependent on the fin angle. The effect of the variation of fin angle, coupled with the

covering of heat transfer of the radiator on the engine performance, will help advice manufacturers of the best fin angle in a typical African setting,

- In running the model it was assumed that water was the coolant fluid which is the case in most cars used; however other coolant fluids with better heat transfer characteristics can also be considered, and
- It is also recommended that a finite differencing approach should be employed in any future research to solve the heat transfer problem. Finite differencing method can be employed to solve the mathematical model. This will provide a very good comparison between the straight ϵ -NTU method and the finite difference method.

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APPENDICES

Table A1: Experimental results obtained using clay as the covering material

Percentage area covered	Ambient temperature (°C)	Inlet temperature (°C)	Outlet temperature (°C)
0	34	81.5	40.6
10%	35	86.3	43.3
		86.9	43.4
		87	43.5
10%	35	86	42.4
		86.2	43.7
		86.6	43.9
10%	34	87	43.7
		87.2	43.9
		87.9	44
20%	36	88.5	45.2
		89.5	45.5
		91.3	46.7
	35	88.1	44.5
		88.3	44.7
		88.5	44.9
	35	88.6	44.7
		88.9	44.8
		89.9	44.9
30%	36	88.6	48.1
		88.9	48.2
		89.9	48.7
	37	89.5	49
		90.2	49.2
		90.7	52.4
	36	89.1	48.9
		89.3	49.2
		90.6	50.4
40%	35	89	51.1
		90.1	52
		90.2	53.2
	35	89.7	51.7
		89.9	52
		90.3	53.7
	34.5	89	51.1
		89.4	51.4
		89.9	52.7

50%	35	90.4	52
		90.5	53.6
		90.7	53.7
	35	90.8	52.3
		90.7	53.8
		90.9	53.8
	35	90.7	53.3
		90.8	53.9
		90.9	53.9
60%	35	91.1	55.8
		91.3	55.9
		92.1	56
	34.5	90.2	55.9
		91.5	55.9
		92.3	56.1
	35	90.3	55.3
		90.6	55.4
		91.4	55.7
70%	35	92.1	57.3
		92.5	57.6
		92.6	57.8
	34.5	90.3	56.4
		90.6	56.9
		90.7	57.4
	35	92.2	57.1
		92.6	57.4
		92.7	57.7
80%	35	92.1	58.6
		92.3	59.9
		92.5	61.2
	35	92	57.6
		92.4	58.9
		92.6	60.2
	35	92.4	57.7
		93.1	58.7
		93.2	61
90%	35	93.1	60.1
		93.2	62.3
		93.4	62.9
	34.5	93.1	59.2
		93.3	59.3
		93.4	60.1
	34.5	93.3	59.5

		93.6	60.4
		93.7	62
100%	35	93.4	59.5
		93.7	61.2
		93.9	62.4
	35	94	62.1
		94.1	62.4
		94.3	63.1
	34.5	93.2	61.7
		93.5	62.9
		93.7	63

Table A2: Experimental results obtained using silt as the covering material.

Space of area covered	Ambient Temperature (^oC)	Inlet temperature (^oC)	Outlet temperature (^oC)
0	34	81.2	40.3
10%	34	83.4	41.3
		84.9	41.6
		85.4	43.1
10%	34.5	83.8	41.9
		85.5	42.5
		86	42.8
10%	35	84.2	42.4
		84.7	42.9
		85.3	43.1
20%	34	84.2	42.2
		84.7	43.4
		85	43.5
	35	84.5	42.7
		84.6	42.9
		85.2	43.4
	35	83.4	43.1
		84.6	43.2
		85.1	43.6
30%	35	85.2	43.7
		85.3	43.9
		85.9	44.1
	35	85.7	44.3
		86.1	45.3
		86.3	45.9
	36	85.9	45.3

		86.4	46.2
		87.1	46.1
40%	35	85.8	46.4
		86.7	46.9
		86.5	47.2
	34	86.1	46.7
		86.5	46.9
		87	47.1
	34	86.9	46.7
		87.2	47.2
		87.3	47.4
50%	35	87.1	48.2
		87.4	48.9
		87.9	49.3
	36	87.3	49.4
		88	50.1
		88.1	50.2
	35	87.9	50.6
		88.1	50.8
		88.2	50.7
60%	34	88.4	50.9
		88.9	51.2
		89.1	51.9
	35	88.5	51.7
		88.8	52.3
		89.2	52.4
	35	87.9	51.8
		88.7	52.6
		89.1	52.7
70%	35	88.6	52.3
		89.4	52.8
		89.9	53.1
	34.5	88.4	52.5
		88.9	52.9
		89.2	53.3
	35	88.7	52.7
		88.9	52.8
		89.3	53.3
80%	35	88.9	52.5
		89.7	53.4
		89.9	54.5
	34	89.4	53.6
		90.1	54.8

		90.2	54.9
	35	90.2	54.7
		90.4	54.7
		90.3	55.2
90%	35	90.3	55.3
		90.7	54.4
		90.9	56
	35	90.2	56.1
		91.3	56.3
		91.6	55.9
	34	91.1	56
		90.9	55.6
		91.9	56.1
100%	35	90.9	56.4
		91.2	57.1
		91.3	57.3
	35	90.8	51.2
		91.3	58.3
		91.5	57.9
	34.5	91.3	57.3
		92.1	58.4
		92.5	58.9

Table A3: Results obtained using clay to cover the radiator

Area covered (%)	Ambient temp. (^oC)	Inlet temp (^oC)	Outlet temp. (^oC)
0	34	81.5	40.6
10	34.7	86.4±0.19	43.13±0.16
20	35.3	89.1 ±0.34	45.1±0.22
30	36.3	89.1±0.25	48.7±0.44
40	34.8	89.2±0.16	51.3±0.34
50	35	90.6±0.57	52.5± 0.24
60	34.8	90.5±0.25	55.7±0.09
70	34.8	91.5±0.33	56.9±0.15
80	35	92.2±0.14	58±0.44
90	34.7	93.2±0.07	59.6±0.46
100	34.8	93.5±0.12	61.1±0.38

Table A4: Results obtained using silt to cover the radiator

Area covered (%)	Ambient temp. (^oC)	Inlet temp. (^oC)	Outlet temp. (^oC)
0	34	81.2	40.3
10	34.5	83.8±0.29	41.86±0.22
20	34.7	84.59±0.18	43.11±0.15
30	35.33	85.6±0.19	44.43±0.33
40	34.3	86.27±0.17	46.6±0.10
50	35.3	87.43±0.13	49.4±0.3
60	34.7	88.27±0.14	51.47±0.21
70	34.8	88.57±0.15	52.5±0.11
80	34.7	89.5±0.16	53.6±0.3
90	34.7	90.53±0.19	55.8±0.2
100	34.8	91±0.18	55±0.77

Table A5: Results obtained using the model developed

Area covered (%)	Outlet temp. (^oC)	Inlet temp. (^oC)
0	42.2	81.5
10	44.3	83
20	46.3	85
30	48.4	87
40	50.4	89
50	52.4	91
60	54.4	93
70	56.4	95
80	58.3	97
90	60.2	99
100	62.3	100

APPENDIX B

MATLAB CODES

```
% DETERMINATION OF PARAMETERS
disp ('PARAMETERS FOR WATER SIDE')
disp (' ')
%determination of water side heat tranfer coefficient
% Determination of Water side Re
Re_w = 0; % Re is the Reynolds number
disp ('Water side Re')
D_w= input ('diameter =') ;
Rho_w = input ('Density of Water =');
V_w = input ('Velosity =');
Mu_w = input ('Viscosity = ');

Re_w = (Rho_w * V_w * D_w)/Mu_w

% Determination of the Prantl number Pr for Water
%disp ('Water side pr')
Pr_w= 0;

Cp_w =input ('Specific Heat Capcity of water =');
k_w =input ('Thermal conductivity of water =');

Pr_w = (Mu_w * Cp_w)/k_w
%Pr_w = input('prantl number =')
% Determination of Heat tranfer Coeffient of water

y1 = input ('Coolant tube length =');

Nuc= 3.66 + ((0.0668*(D_w/y1)* Re_w* Pr_w)/(1+ 0.04 *((D_w/y1)*Re_w *
Pr_w)^ (2/3)))

HA_w= Nuc*k_w/D_w

% *******Evaluate Ac*******

disp ('PARAMETERS FOR AIR SIDE')
disp (' ')
%determination of air side heat transfer coefficient
Re_a = 0; % Re is the air side heat tranfer coeffiecient

disp ('air side Re')
L_a= input ('lenght =');
```

```
Rho_a = input ('Density of air =');
V_a = input ('Velosity =');
Mu_a = input ('Viscosity = ');

Re_a = (Rho_a * V_a * L_a)/Mu_a

% Determination of the Prantl number Pr for Air
disp ('air side pr')
Pr_a= 0;

Cp_a =input ('Specific Heat Capcity of air =');
k_a =input ('Thermal conductivity of air =');
Lh= input ('Louvre Hieght =');
Ll = input('Louvre Length = ');
Fh = input ('Fin Height=');

Pr_a = (Mu_a * Cp_a)/k_a

% Determination of Heat tranfer Coeffient of air

HA_a = (0.249*Re_a^(-0.42)*Lh^(0.33)*(Lh/Fh)^(1.1)*Fh^(0.26))* ((Rho_a *
V_a* Cp_a)/(Pr_a)^(2/3))


disp ('PARAMETERS FOR OVERALL HEAT TRANSFER COEF')
disp (' ')
CAlph_f= 0;
Bt= input ('Core Thickness  = ');
Np= input ('Number of profiles = ');
Nf =input ('Number of fins = ');
Rf = input ('Fin end radius =');
Alp_f= input ('Angle of fin =');
Nct= input ('Number of coolant tubes in 1 row = ');
Ycc= input ('Coolant tube cross section length = ');
Rt = input ('Coolant tube End raduis =');
pi =22/7;
CAlph_f= cos (Alp_f);
cos_alp_f =angledim (CAlph_f,'radians','degrees');
F1 = pi * Rf +(( Fh -2* Rf )/(cos_alp_f))
Af= 2 *Bt * F1 *Nf* y1 *Np
AA=(Af +( 2* Nct * y1) * Nf *(Ycc-2 * Rt)+ 2* pi * Rt)

Bh = input ('Radiator Height = ');
Bw = input ('Width of radiator =');
No = input ('Air side efficiency =');

Afr= Bh * Bw*0.1
```

```
Nr = input ('number of rows of tubes in the core depth dimension =');
Yt = input ('coolant tube thickness =');
Ycl = input ('coolant tube cross sectional length =');

Ac = ((2*22/7*(Rt-Yt)+2*(Ycl-2*Rt))*(y1*Nct*Nr))

U = (No * HA_a *AA  * HA_w* Ac)/(Afr* (HA_w* Ac+No* HA_a*AA))
Qc = input('cooling rate = ');
mw = input('mass flow rate of water =');
ma = input('mass flow rate of air =');
Ca_w = mw*Cp_w
Ca = ma*Cp_a
Cr = Ca/Ca_w
NTU = U*Afr/Ca
e = 1-exp(((NTU^0.22)/Cr)*(exp(-Cr*(NTU^0.78))-1))
T2 = input('temperature of water entering the radiator =');
T1 = T2 -Qc/(e*Ca_w)
```