DESIGN AND FABRICATION OF HEAT EXCHANGER - RADIATOR

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‘Saya/Kami* akui bahawa telah membaca
karya ini dan pada pandangan saya/kami* karya ini
adalah memadai dari segi skop dan kualiti untuk tujuan penganugerahan
Ijazah Sarjana Muda Kejuruteraan Mekanikal (Termal Bendalir)’

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α_a = Ratio of total heat transfer area of one side of the exchanger to its volume, m^2/m^3

A = Total heat transfer area, m^2

A_0 = Free flow area of one side of exchanger, m^2

A_{fr} = Frontal area of one side of the exchanger, m^2

B = Air flow length in the exchanger, m

C_a = Stream heat capacity rate for air

c_p = Specific heat loss of coolant

C_w = Stream heat capacity rate for water

$C*$ = Heat capacity rate ratio

C_μ = Specific heat at constant dynamic viscosity

$C_{p,a}$ = Specific heat of air

$\in$ = Heat exchanger effectiveness

f = Friction factor

g = Gravity

G_a = Core mass velocity

j_a = Multiplied factor of heat exchanger

L = Length of each tube

L_p = Louver pitch

m_c = Coolant mass flow rate

$N_{u,w}$ = Nusselt's number

Q = Heat Loss

R_t = Ratio of the convection heat transfer characteristic to the convection mass transfer characteristic for the simultaneous convection heat and mass transfer for tube.

Re = Reynold’s number

r_h = Fin radius

T_i-T_0 = Temperature Drop

U_a = Air side area

V = Total volume of the exchanger, m^3

W_a = Specific thermal resistance of air

ρ = Water density

μ = Viscosity

μ_t = Viscosity in tube

CHAPTER I

INTRODUCTION

1.1 Background

Radiators and convectors are types of heat exchanger designed to transfer thermal energy from one medium to another for the purpose of cooling and heating. The majority of radiators are constructed to function in automobiles, buildings, and electronics. In automobiles with liquid-cooled internal combustion engine, a radiator is connected to channels running through the engine and cylindrical head through which a liquid is pumped. This liquid is typically a half-and-half mixture of water and ethylene glycol or propylene glycol known as antifreeze. The radiator transfer heat from the liquid inside to the air outside, thereby cooling the engine. Radiators are generally mounted in a position where they will receive airflow from the forward movement of vehicle such as behind the grill. Where engines are rear-or-mid-mounted, it is usually necessary to mount the radiator behind the front grill, so as to achieve sufficient airflow even through this requires long coolant pipes.

Automobile radiator is one type of cross flow heat exchanger which is an important part of vehicle engine. Normally, it is used as a cooling unit of the engine and the water is heat transfer medium. The heat exchanger configuration is louvered fin and flat tube. In some cases, the unit is also modified to be a heat recovery device or a heat exchanger in industrial processes because of its low price compared to other heat exchanger types and it is available in the market.

Automobile radiators are constructed of a pair of header tanks linked by a honeycomb core. This core usually made of stacked layers of metal sheet, pressed to form channels and soldered or brazed together. For many radiators were made from brass or cooper cores soldered to brass headers. Modern radiators save money and weight by using plastic headers and may use aluminum cores. This construction is less easily repaired than traditional materials. Vintage cars may also have used radiator cores made from coiled tube, a less-efficient but simpler construction. Other factors influence the temperature of the engine including radiator size and type of radiator fan. The size of radiator is chosen such that it can keep the engine at the design temperature under the most extreme conditions a vehicle is likely to encounter.

Airflow speed through a radiator is a major influence on the heat it loses. Vehicle speed affects this in rough proportion to the engine effort thus giving crude self-regulatory feedback. Where an additional cooling fan is driven by the engine, this also tracks engine speed similarly. Engine-driven fans are often regulated by a viscous-drive clutch from the drive belt which slips and reduces the fan speed at low temperatures. This improves fuel efficiency by not wasting power on driving the fan unnecessarily. On modern vehicles, further regulation of cooling rate is provided by either variable speed or cycling radiator fans. Electric fans are controlled by a thermostatic switch or the engine control unit. Electric fans also have the advantage of giving good airflow and cooling at low engine revs or when stationary such as in slow-moving traffic.

1.2 Objectives

There are several objectives that will be cover through finishing this research. The objectives are:

i. Built two complete experimental set-ups that are the radiator test on 1.5cc Wira engine and wind tunnel test to identify performance characteristic of various automotive radiator design under simultaneous actual heat dissipation loading and under various climatic conditions.
ii. To analyze heat transfer performance characteristics of automotive compact heat exchanger units under real conditions using experimental technique.
iii. To generate a bank of performance data required in the development of new heat exchanger design for both radiator and condenser system.
iv. Design and fabricate the annular shaped radiator.

1.3 Scope

The scope of this research will cover about the new design of heat exchanger for automotive vehicle. This new design will be fabricated and tested using experimental methods. It is the detailed geometrical arrangement of the fins and tube that determines the heat rejection performance. This research will provide the necessary data and knowledge leading towards the development of new heat exchanger design which will be applicable to the national automotive industry and worldwide heat exchanger industry.

1.4 Problem Statement

Nowadays radiator designed for automotive vehicle is commonly in square pattern. Besides that, it requires much space including the fan system. If the design can be improved, it will become more efficient for vehicle cooling system usage in terms of space and performance. From this research the radiator that is commonly used in automotive industry will be changed into circular pattern and test this new design to make sure it is reliable for automotive industry production. Hence, this project will clarify the effect of shape between the ordinary square shaped radiators compared to the circular shaped radiator.

CHAPTER II

LITERATURE REVIEW

2.1 Introduction

There are several previous researchers that were studying about heat exchanger. They were covered about the design, performance, characteristics, and also numerical analysis. C. Oliet et. al (2007) had studied on Parametric Studies on Automotive Radiators, D. G. Charyulu et. al (1999) had studied on Performance Evaluation of A Radiator in A Diesel Engine-A Case Study, S. C. Kim et. al (2007) had studied on Performance Evaluation of a CO_2 Heat Pump System for Fuel Cell Vehicles Considering The Heat Exchanger Arrangements, Q. Yu et. al (2006) had studied on Carbon-Foam Finned tubes in Air-Water Heat Exchangers, while H. J. Kim et. al. (2008) had studied on A Numerical Analysis for The Cooling Module Related to Automobile Air Conditioning System. These studies are focusing on the design, performance, characteristics, and numerical analysis of the heat exchanger in automotive industrial.

2.2 Heat Exchanger

According to C. Oliet et. al. (2007) the air-cooled heat exchangers found in a vehicle radiator (radiator, AC condenser and evaporator, charge air cooler, etc.) have an important role in its weight and also in the design of its front-end module, which also has a strong impact on the car aerodynamic behavior. Looking at these challenges, an optimization process is mandatory to obtain the best design compromise between performance, size/shape, and weight. This optimization objective demands advance design tools that can indicate not only the better solution but also the fundamental reason of the performance improvement.

H. J. Kim et. al. (2008) states that the purpose of a vehicle cooling system is to ensure that the engine is maintained at its most efficient practical operating temperature. The cooling airflow in today’s vehicle engine cooling systems is generated by a ram effect resulting from the vehicle’s motion and suction produced by fan operation. This airflow passes through the grill, condenser, radiator, cooling fan, and other components, removing the rejected heat to the surrounding environment. The flow rate and the temperature of the air in front of the radiator have a strong influence on the heat dissipation capacity of the radiator and on the performance of the cooling system. Therefore, the airflow is an essential factor affecting the engine cooling system performance and has always been of primary concern in the engine cooling system design.

Air-water heat exchangers are commonly employed in engine cooling, high-power electronics cooling, and heat recovery units for power generation systems said Q. Yu et. al (2006). The resistance to convective heat transfer on the air side of the heat exchanger dominates in the design of these heat exchangers. Large numbers of impermeable metal fins are used to provide additional surface area on the air side of the heat exchanger to lower the total convective thermal resistance however the associated increase in surface area also results in large pressure drops, which must be overcome by higher fluid power input on the air side. Hence, the classical problem in heat exchanger design emerges: an optimum balance must be found between the thermal resistance (convective heat transfer) and the hydrodynamic resistance (pressure drop).

D. G. Charyulu et. al. (1999) found that the performance evaluation of a radiator mounted on a turbo-charged diesel engine has been made with and without fouling factor. Heat transfer estimation indicates that the radiator is over designed. The characteristics of the radiator have been analyzed for different tube rows with varying air mass velocities to enable the design engineer to select the size depending upon the requirement and application.

Another researcher, S. C. Kim et. al. (2007) found that a novel CO_2 heat pump system was provided for use in fuel cell vehicles, when considering the heat exchanger arrangements. This cycle which had an inverter-controlled, electricity-driven compressor was applied to the automotive heat pump system for both cooling and heating. The cooling and heating loops consisted a semi-hermetic compressor, supercritical pressure micro channel heat exchangers (a gas cooler and the cabin heater), a micro channel evaporator, an internal heat exchanger, an expansion valve and an accumulator. The performance characteristics of the CO_2 heat pump system for fuel cell vehicles were analyzed by experiments.

2.3 Mathematical Formulation and Numerical Model

In the following, the mathematical formulation is briefly described providing the corresponding discretised equations of the air, the fins, the tubes, and the coolant. The local empirical information (heat transfer coefficients, friction factors, pressure drop through singularities) and the thermo-physical properties that have been used on the air and coolant side modeling are also presented in detail.

2.3.1 Governing Equations

To solve the complicated physical phenomena corresponding to the complex geometry of the vehicle and cooling system model, there are necessary requirements. A simple coordinate system for stable solving problem is needed. Additionally, a special method defining the relation of the computational domain to the simple coordinate system and the complex geometry is also necessary. Generally the testing of the engine cooling module is completed when the coolant inlet temperature of the radiator varies little over the time of test. Therefore, the steady state continuity, momentum equation is used to solve the transient equation:

Continuity equation

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \tag{1}$$

x, y, z momentum equation

$$\frac{\partial(\rho u_k u_j)}{\partial x_k} = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_j}{\partial x_i}\right) + s_j \tag{2}$$

The stress tensor term split so that a portion of the normal stress appears in the diffusion term, and the rest is contained in S_j

$$s_j = f_j + \frac{\partial}{\partial x_j}\left[\mu \frac{\partial u_i}{\partial x_i}\right] - \frac{2}{3}\frac{\partial}{\partial x_j}\left(\mu \frac{\partial u_k}{\partial x_k}\right) + s_{source} \tag{3}$$

Energy temperature equation

$$\frac{\partial\left(\rho u_k T\right)}{\partial x_k} = \frac{\partial}{\partial x_k}\left(\frac{k}{C_p}\frac{\partial T}{\partial x_j}\right) + s_r \tag{4}$$

The Governing equation used in the design of heat exchangers is expressed in the general form: $q = UA\Delta T_m$, where U is the overall heat transfer coefficient based on the fin side total heat transfer surface area A, and ΔT_m is the mean temperature difference between the two fluids on the tube and fin sides. The Governing equation also can be written as:

$$q = \frac{\Delta T_m}{R_t} \tag{5}$$

2.3.2 Turbulence Model and Wall Function

When the flow is turbulent, the solution variables may be divided into a time average value and its instantaneous component. For most engineering purposes, it is unnecessary to resolve the details of the turbulent fluctuations. Therefore, the standard two equation *k*-*ε* model was applied:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$$

$$\frac{\partial\left(\rho u_k k\right)}{\partial x_k} = \frac{\partial}{\partial x_j}\left(\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial x_j}\right) + G - \rho\varepsilon \qquad (6)$$

$$\frac{\partial\left(\rho u_k \varepsilon\right)}{\partial x_k} = \frac{\partial}{\partial x_j}\left(\frac{\mu_\varepsilon}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x_j}\right) + \left(C_{1\varepsilon} G - C_{2\varepsilon}\rho\varepsilon\right)\left(\frac{\varepsilon}{k}\right) \qquad (7)$$

Where; $C_\mu = 0.09; \quad \sigma_k = 1.00; \quad \sigma_\varepsilon = 1.30; \quad C_{1\varepsilon} = 1.44; \quad C_{2\varepsilon} = 1.92$

A basic wall function equation for the velocity parallel to the wall was applied. The “logarithm law of the wall” equation is given as

$$u^+ = \frac{1}{k}\ln\left(Ey^+\right) \qquad (8)$$

k is Von Karman’s constant (0.4187) and *E* is an integration constant that depends on the roughness of the wall. For smooth wall with constant shear stress, *E* has a value of 9.0.

2.3.3 Relations for Surface and Core Geometry

Certain geometrical relations are necessary in the application of the basic heat transfer and flow friction data to the design problem. The following geometrical factors are required as a design result for each of the two sides of the complete exchanger core:

A total heat transfer area, m^2;
A_0 free flow area of one side of exchanger, m^2;

A_{fr} frontal area of one side of the exchanger, m^2;

B air flow length in the exchanger, m;

V total volume of the exchanger, m^3;

α_a ratio of total heat transfer area of one side of the exchanger to its volume, m^2/m^3.

The equations below give the relations between surface and core factors for one side of the exchanger.

$$r_h = L(A_0 / A) \qquad (9)$$

$$A_0 = \left(\sigma A_{fr}\right) = (Ar_h / L) = \left\{A\sigma/(L\alpha)\right\} \qquad (10)$$

$$A_{fr} = LH \qquad (11)$$

$$V = LBH \qquad (12)$$

$$\alpha = A/V = \left\{A/(LA_{fr})\right\} = (\sigma / r_h) \qquad (13)$$

$$\sigma = (A_0 / A_{fr}) = \left\{Ar_h /(LA_{fr})\right\} = (Ar_h)/V \qquad (14)$$

Air side calculations:

Stream heat capacity rate, C_a

$$C_a = W_a \times C_{p,a} \qquad (15)$$

Heat transfer coefficient, h_a

$$h_a = j_a G_a C_{p,a} /(P_{r,a})^{2/3} \qquad (16)$$

$$j_a = 0.174/(R_{e,a})^{0.383} \qquad (17)$$

Where, $G_a = core\ mass\ velocity = W / A_0 = W / A_{fr}\sigma_a$ (18)

$R_{e,a} = \mathrm{Re}\,ynolds\ number = G_a \times D_{h,a} / \mu_a$ (19)

Temperature effectiveness of fins (fin efficiency)

$$\eta_f = Tanh(ml)/(ml) \qquad (20)$$

Where, $m = \left[2 \times h_a /(k_f \times \delta)\right]^{1/2}$ (21)

Total surface temperature effectiveness

$$\eta_0 = 1.0 - (1.0 - \eta_f) \times A_f / A \qquad (22)$$

Pressure drop (neglecting the expansion and contraction loss coefficient)

$$\Delta P_a = P_i G_a^2 /(2P_i \rho_i) \times \left[(1-\sigma_a^2) + 2\left\{ \left(\rho_i / \rho_0 \right) - 1 \right\} + (fB\rho_i / r_h)(1/\rho)_m - (1-\sigma_a^2)(\rho_i / \rho_0) \right] \quad (23)$$

Where, $f = 0.3778 / R_{e,a}^{0.3565}$ (24)

$$(1/\rho)_m = \left\{ (1/\rho_i) + (1/\rho_0) \right\} / 2 \quad (25)$$

Water side calculations:

Stream heat capacity rate, C_w

$$C_w = W_w \times C_{p,w} \quad (26)$$

Heat transfer coefficient

$$h_w = N_{u,w} \times k_w / D_{h,w} \quad (27)$$

Where Nusselt number,

$$N_{u,w} = 0.023 \times \left(R_{e,w} \right)^{0.8} \times \left(P_{r,w} \right)^{0.3} \quad (28)$$

Reynold’s number,

$$R_{e,w} = G_w \times D_{h,w} / \mu_w \quad (29)$$

Pressure drop

$$DP_w = G_w^2 \times f_w \times H / \left[2 \times \rho \times \left(D_{h,w} / 4 \right) \right] \quad (30)$$

Where, $f_w = 0.079 \times \left(R_{e,w} \right)^{-0.25}$ (31)

Heat capacity rate ratio, $C*$

$$C* = C_{\min} / C_{\max} = C_a / C_w = C_c / C_h \quad (32)$$

Heat exchanger effectiveness, $\in$

$$\in = C_h \times (T_{h1} / T_{h2}) / \left[C_{\min} \times (T_{h1} - T_{c1}) \right] \quad (33)$$

Or

$$\in = C_c \times (T_{c2} / T_{c1}) / \left[C_{\min} \times (T_{h1} - T_{c1}) \right] \quad (34)$$

Overall heat transfer coefficient, based on air side area (U_a):

Neglecting the very small wall resistance,

$$1/U_a = 1/(\eta_0 h_a) + 1/\left[\left(\alpha_w / \alpha_a \right) h_w \right] \quad (35)$$