HT2008
August 10-14, 2008, Jacksonville, Florida USA

HT2008-56469

Thermal Analysis of Radiator Core In Heavy Duty Automobile

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ABSTRACT

Background: Heat dissipation is one of the most critical considerations in engine design and with an efficient cooling system; performance of the engine can be dramatically improved. All internal combustion engines convert chemical energy into mechanical power. Around 70% of the energy is converted into heat and therefore, the primary job of the cooling system is to keep the engine from overheating by transferring this heat to the air. A radiator transfer's heat from the hot coolant to the air and an effective design of radiator will ultimately lead to enhanced engine performance by reducing the heating effect.

Methods and results: A mathematical expression for the rate of heat dissipation from the radiator core was derived and a modification in the design was proposed in the radiator core by changing the structure of the tubes from cylindrical to helical. The rate of heat dissipation for both designs was compared with similar boundary conditions by varying the magnitude of all design parameters in a specific range that have same magnitude of area of cross section, length of the radiator core and coefficient of thermal conductivity for the tube. Enhanced rate of heat dissipation for helical structure confirms the efficacy of the proposed design.

Keywords: Radiator core; heat transfer; forced convection; helical structure; heat exchanger.

INTRODUCTION

Automobile radiator is one of the types of cross flow heat exchangers that is an important part of vehicle engine. It is normally used as a cooling unit for the engine where mixture of water and glycol is the medium for heat transfer. The fluid (coolant) moves in a closed system from the radiator to the engine where it conducts heat away from the engine parts. The radiator is typically mounted behind the vehicles grille with outside air driven through the radiator by the vehicle's forward motion that is often supplemented by a fan. Thus the radiator transfers the heat from the fluid inside to the atmosphere thereby cooling the engine.

Vithayasai et. al. [1] analyzed the effect of electric field on the performance of an automobile radiator. In their work, electric field was supplied on the air side of the exchanger. By adjusting the voltage in a specific range it was observed that the unit with electric field pronounced better heat transfer rate especially at a low frontal velocity of air. Oliet et. al. [2] presented a set of parametric studies performed on automotive radiators by means of detailed rating and design of heat exchangers. Their analysis mainly focused on the influence of working conditions on both the fluids (air and coolant) by showing the impact of selected coolant fluid on the rate of heat transfer. They also demonstrated the influence of certain geometric parameters such as fin pitch and louver angle on radiator global performance and significant design conclusions were reported by analyzing the overall behavior of the radiator. Witry et al. [3] introduced aluminum roll bonding technique used in manufacturing wide range of heat exchangers that can help in augmenting heat transfer whilst reducing pressure drops. Computational fluid dynamics (CFD) results were obtained that showed tremendous levels of possible performance improvement on both sides of heat exchanger. Dittus & Boelter [4] discussed the fundamentals involved in the heat transfer from coolant in the simplest type of tubular structure. All the above mentioned techniques were basically aimed at augmenting rate of heat transfer from the coolant which improves the engine performance in tropical conditions.

In order to augment the rate of heat dissipation, the radiator core consisting of thin cylindrical tubes were changed to helical structure to expose more surface area to the atmosphere. The magnitudes of main parameters which affect the rate of heat transfer such as mass flow rate of the coolant in the tubes and relative velocity of the air with respect to the vehicle were varied in a specific range and variation of the rate of heat dissipation with these parameters was shown graphically. By analyzing these variations, we showed that the rate of heat dissipation could be improved to a great extent.

NOMENCLATURE

Q_x Rate of heat transfer for a length x .

Q_c Rate of heat transfer for the cylindrical structure.

U_x Change in internal energy for length x .

Q_h Rate of heat transfer for the helical structure.

V_c Velocity of the coolant.

K_m Coefficient of thermal conductivity of the tube.

V_a Relative velocity of the air.

K_c Coefficient of thermal conductivity of coolant.
T_a Ambient temperature.
K_a Coefficient of thermal conductivity of air.
T_i Inlet temperature of the coolant.
h_c Convective heat transfer coefficient of coolant.
T_x Temperature of the coolant at length x .
$\overline{h_a}$ Average Convective heat transfer coefficient of air.
L Length of the radiator core.
R Outer radius of the tubular section.
A Area of cross section of radiator core.
R_c Outer radius of the tubular section with cylindrical structure.
R_h Outer radius of the tubular section with helical structure.
R_H Radius of the helix.
ρ_c Density of coolant.
r Inner radius of the tubular section.
ρ_a Density of air.
r_c Inner radius of the tubular section with cylindrical structure.
q Rate of flow of coolant.
r_h Inner radius of the tubular section with helical structure. .
μ_a Coefficient of viscosity of air.
C_{Vc} Specific heat of coolant at constant volume.
μ_c Coefficient of viscosity of coolant.
C_{Pc} Specific heat of coolant at constant pressure.
n Number of tubes.
C_{Pa} Specific heat of air at constant pressure.
n_h Number of tubes with helical structure.
g Gap between two tubes.
n_c Number of tubes with cylindrical structure.
g_r Gap between two circular rings of the helical structure.
Re_c Reynolds number of the coolant flowing.
Pr_c Prandtl number of the coolant flowing.
Re_a Reynolds number of the air flowing.
Pr_a Prandtl number of the air flowing.
$\mathrm{N_R}$ Number of rows of tubes in the radiator core.
m Mass flow rate of the coolant.

MATHEMATICAL MODELING

A small segment of the tubular section was considered as a computational domain for thermal analysis. The physical problem describes a flow of the coolant which dissipates heat across the cylindrical shell through conduction and convection to the air flowing across the tube transversely. The expression for the rate of heat dissipation was derived using Fourier's law, Newton's law of cooling and the First law of thermodynamics .The domain was considered to be axisymmetric. Fluid flow was assumed to be incompressible and turbulent. Heat transfer was considered to be in a steady state and the changes in the specific heat capacity, coefficient of viscosity and coefficient of thermal conductivity of air coolant and the material of the tube due to temperature variations were assumed to be negligible.

Using the assumptions mentioned above, a general expression for the rate of heat transfer for a cylindrical tube of length x was developed. $Q_x = \dfrac{(T_x - T_a)}{R_e}$ $, U_x = \rho_c q C_{Vc} (T_x - T_i)$. **[7]**

Where thermal resistance $R_e = \dfrac{1}{2\pi x}\left(\dfrac{1}{h_c r} + \dfrac{1}{h_a R} + \dfrac{\ln\left(R/r\right)}{K_m}\right)$

Since the flow was assumed to be incompressible, change in specific volume of the coolant is negligible. $dQ_x = -dU_x$ **- (1)**

The following equation attributed to Dittus and Boelter [4], was used assuming $\mathrm{Re}_c \le 2100$, $Nu_c = 0.023\,\mathrm{Re}_c^{\,0.8}\,\mathrm{Pr}_c^{\,a}$

$Nu_c \Rightarrow$ Nusselt number of the coolant flowing

Where, a = 0.4, when the fluid is heated
a = 0.3, when the fluid is cooled

$$h_c = \frac{K_c}{2r}\left(Nu_c\right), \quad Nu_c = \left(0.023\left(\frac{2\rho_c V_c r}{\mu_c}\right)^{0.8}\left(\frac{\mu_c C_{Pc}}{K_c}\right)^{0.3}\right)$$

$\because\ V_c = \dfrac{q}{\pi r^2 n}$ From the above equation we get,

$$h_c = \frac{\lambda_c}{r^{1.8} n^{0.8}}, \quad \lambda_c = \frac{K_c}{2}\left(0.023\left(\frac{2\rho_c q}{\pi\mu_c}\right)^{0.8}\left(\frac{\mu_c C_{Pc}}{K_c}\right)^{0.3}\right)$$

In this situation, the cross flow of one fluid (air) over the bank of tubes takes place, while the second fluid (coolant) at a different temperature passes through the tubes. The bank of tubes are either staggered or aligned in the direction of fluid velocity as shown in **Fig. 1 [5]**. Flow conditions within the bank are dominated by boundary layer separation effects and by the wake interactions, which in turn influence convection heat transfer. The tubes of the first few rows act as a turbulence grid, which increases the heat transfer coefficient for the tubes in the following rows, however, heat transfer conditions stabilize, such that little change occurs in the convection coefficient further.

The following equation attributed to Grimison [5], was used assuming $2000 \le \mathrm{Re}_a \le 40{,}000$ and $\mathrm{N_R} < 10$ by considering pragmatic application of this research.

$\overline{Nu_a} \Rightarrow$ Average Nusselt number of the air flowing.

$\overline{Nu_a} = 1.13 C_1 \mathrm{C}_2 \mathrm{Re}_a^w \mathrm{Pr}_\mathrm{a}^{0.33}$, Hence we obtain

$$\overline{h_a} = \frac{K_a}{2R}\left(\overline{Nu_a}\right), \quad \overline{Nu_a} = \left(1.13\mathrm{C}_1\mathrm{C}_2\left(\frac{2\rho_a V_{\max} R}{\mu_a}\right)^w \left(\frac{\mu_a C_{Pa}}{K_a}\right)^{0.333}\right)$$

$V_{\max} \Rightarrow$ Maximum velocity of the air occurring.

For Aligned $w \approx 0.613$; $\mathrm{C}_1 \approx 0.278$. [5].

For Staggered $w \approx 0.562$; $\mathrm{C}_1 \approx 0.472$ [5].

$\mathrm{C}_2 \Rightarrow$ Correction factor which can be obtained from **Table. 1**

Therefore h_c and $\overline{h_a}$ are derived from the above empirical relations of forced convection. If all the tubes are arranged in aligned fashion, $V_{\max} = \dfrac{(2R+g)V_a}{g}$ [5]. We get,

$$\overline{h_{a1}} = \frac{\lambda_a}{R^{(1-2w)}}, \quad \lambda_a = \frac{K_a}{2}\left(1.13\mathrm{C}_1\mathrm{C}_2\left(\frac{2\rho_a\left(g/R + 2\right)V_a}{g\mu_a}\right)^w \left(\frac{\mu_a C_{Pa}}{K_a}\right)^{0.333}\right)$$

Now if the tubes are arranged in staggered fashion. We get,

$$\overline{h_{a2}} = \frac{\lambda_a}{R^{(1-w)}}, \quad \lambda_a = \frac{K_a}{2}\left(1.13\mathrm{C}_1\mathrm{C}_2\left(\frac{2\rho_a\left(\dfrac{g}{R}+2\right)V_a}{(\sqrt{5}\left(\dfrac{g}{R}+2\right)-2)\mu_a}\right)^w \left(\frac{\mu_a C_{Pa}}{K_a}\right)^{0.333}\right)$$

$\because\ V_{\max} = \dfrac{(2R+g)V_a}{((2R+g)\sqrt{5}\ -2R)}$ [5]. Where $\overline{h_{a1}}$ and $\overline{h_{a2}}$ are the average convective heat transfer coefficients of air for the aligned and staggered arrangements of the tubes in the radiator core respectively. In our paper staggered arrangement of the tubes was considered and the following equations for the heat transfer were derived by assuming $\left(\dfrac{g}{R}+2\right) \approx 2$.

From **(1)** we get, $\quad dT_x = -dx\dfrac{(T_i - T_a)}{B\rho_c q C_{Vc}},$

$$B = \frac{1}{2\pi}\left(\frac{(rn)^{0.8}}{\lambda_c} + \frac{1}{\lambda_a R^w} + \frac{\ln\left(R/r\right)}{K_m}\right) \because\ B\rho_c q C_{Vc} \gg x$$

Now the expression for the number of tubes is obtained as

$$n_c = \left[\frac{A}{(2R_c+g)^2}\right] \text{ and } n_h = \left[\frac{A}{(2R_H + 2R_h + g)^2}\right]$$

Where [y] denotes the integral part of y $\forall$ $y \in \mathrm{IR}$. Hence the rate of heat transfer for length L of the radiator core from the coolant for cylindrical structure and helical structure is obtained as

$$Q_c = n_c\frac{L(T_i - T_a)}{B_c}, \quad B_c = \frac{1}{2\pi}\left(\frac{(r_c n_c)^{0.8}}{\lambda_c} + \frac{1}{\lambda_a R_c^{\ w}} + \frac{\ln\left(R_c/r_c\right)}{K_m}\right)$$

$$Q_h = n_h\frac{2\pi R_H(L-g_r)(T_i - T_a)}{(g_r + 2R_h)B_h}, \quad B_h = \frac{1}{2\pi}\left(\frac{(r_h n_h)^{0.8}}{\lambda_c} + \frac{1}{\lambda_a R_h^{\ w}} + \frac{\ln\left(R_h/r_h\right)}{K_m}\right)$$

Therefore we obtain, $\dfrac{Q_h}{Q_c} = \dfrac{2\pi R_H(L-g_r)B_c n_h}{(g_r + 2R_h)LB_h n_c}$

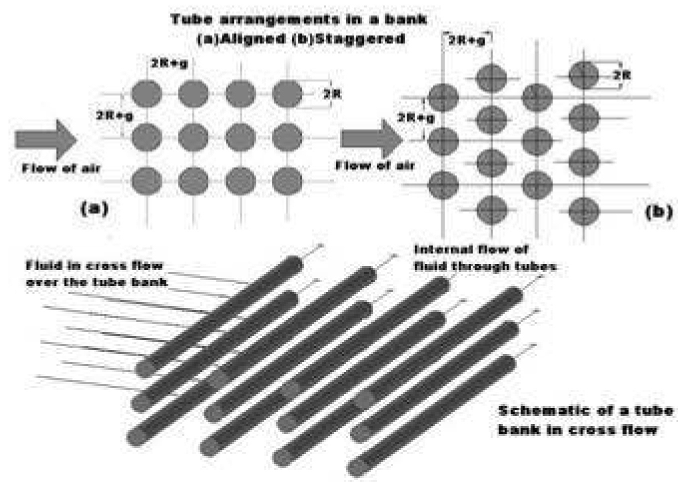

Fig. 1

RESULTS AND DISCUSSION

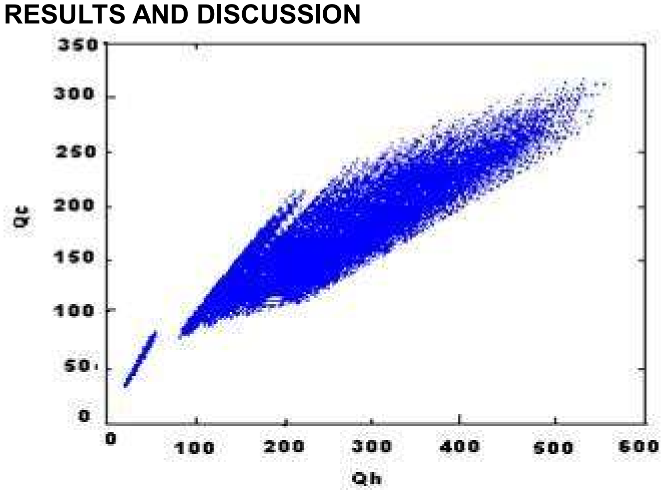

Fig. 2

Plot of Qc & Qh

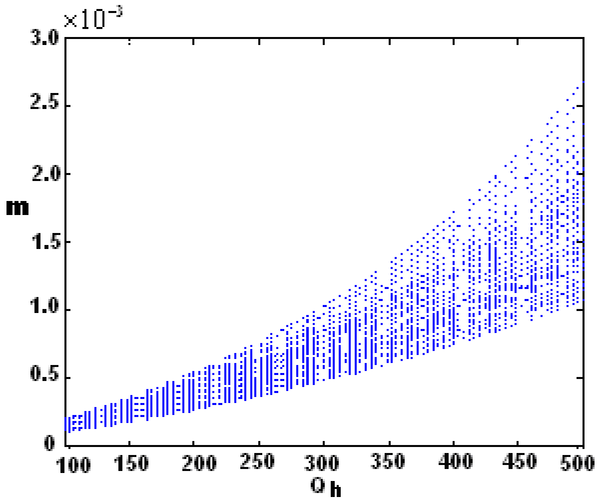

Fig. 3
Plot of m & Qh

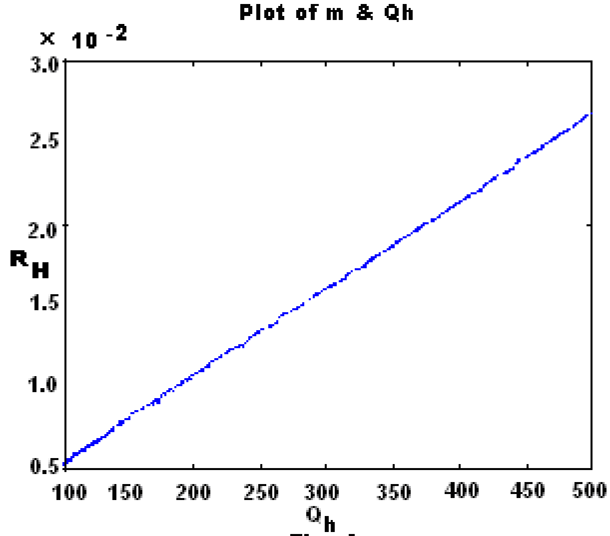

Fig. 4
Plot of RH & Qh

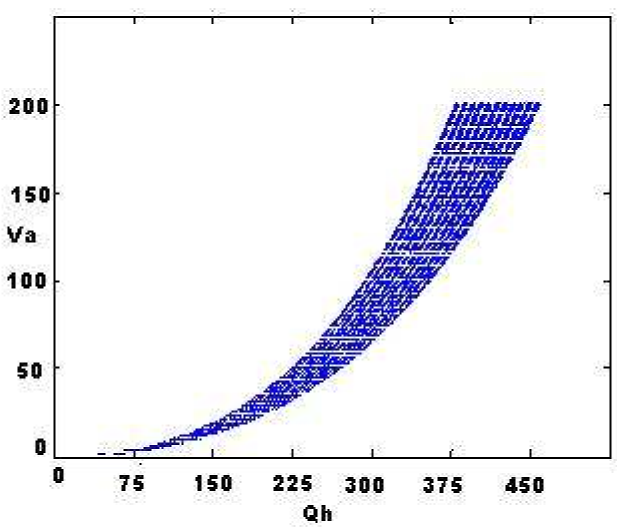

Fig. 5
Plot of Va & Qh

The main design parameters for the helical structure are r_h, R_h, R_H, g and g_r. Fig. 2 shows the graph plotted between Q_h and Q_c by varying all the design parameters that affect the rate of heat dissipation in a specific range. It is easy to see that $Q_h/Q_c > 1$, $\forall\ Q_h > 250$ from the figure. The graph plotted in Fig. 2 shows the comparison between rates of heat dissipation of both designs and reflects the essential novelty in this work. To design a helical structure any value of $Q_h > 250$ was selected depending up on the ambient conditions of the vehicle, a high Q_h value was chosen considering hot tropical conditions and corresponding values of other design parameters were known accordingly. More importantly centrifugal forces induce a secondary flow consisting a pair of longitudinal vortices that increase the magnitude of the convective heat transfer coefficient in the helical structure **[6].** Fig. 3 shows the variation of mass flow rate of coolant with the rate of heat dissipation and a specific value of q can be determined since $m = \rho_c q$. The plots of mass flow rate and relative velocity of air with the rate of heat dissipation in Fig.3 and Fig. 5 are well with in the acceptable limits to ensure the pragmatic application of this research. Fig. 4 shows the variation of R_H with Q_h linearly and unique value of R_H is determined from the graph plotted. Several options were available for enhancing heat transfer associated with internal flows. In this paper by coiling a tube, heat transfer was enhanced without inducing turbulence or additional heat transfer surface area.

Table. 1

N_R	1	2	3	4	5
C_2 for Aligned	0.64	0.80	0.87	0.90	0.92
C_2 for Staggered	0.68	0.75	0.83	0.89	0.92

Conclusions and future work

We presented a modification in the design of radiator core so as to function with augmented rate of dissipation that subsequently improves engine performance in tropical conditions. The shape of the cylindrical tubes was changed to helical in the radiator core to expose more surface area to the atmosphere. Such an approach of changing the design to increase the rate of heat dissipation does not seem to have reported in the literature. Fig. 5 shows the variation of Q_h with V_a. The future work of this effort includes developing a technique which augments the relative velocity of air V_a which improves the performance of the engine even at the low speeds of the vehicle.

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