Thermo-Hydraulic performance of Internal finned tube Automobile Radiator

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ABSTRACT

Thermo-fluid behaviour of finned and finless tube was investigated experimentally and numerically using Fluent 14. One meter length aluminium tube of diameter 0.044 m subjected to constant heat flux on the outer surface was chosen both for numerical and experimental purpose. High velocity of air supplied by the blower and orifice meter was used to measure the flow rate. The thermocouples are used for measuring the inlet, outlet and wall temperatures. Finned tube consisting of longitudinal rectangular sectioned fins attached on the inner periphery of the tube. The numerical simulations were performed by solving 3-D conservation equations of mass, momentum and energy with two equation based k-ε turbulent model. It was found from the investigations that for the same heat flux and Reynolds number, the wall temperature was lowest, Nusselt number was 2.5 times for 8 numbers of internal fins finned tube compared to plain tube and convective resistance was minimum for 8 numbers of internal finned tubes.

Key words: CFD, Nusselt number, friction factor, heat transfer coefficiente, wall temperature, thermocouples

Nomenclature

D	diameter of tube (m)
f	friction factor
H	height of the fin (m)
h	heat transfer coefficient (W/m^{2}C)
L	length of the tube (m)
n	number of tube
q	heat flux
Re	Reynolds number'
T	temperature (K)
W	width of the fin (m)
x	axial distance (m)

Subscripts

avg	average
B	bulk
O	outlet
x	local
sx	axial distance along the wall

1. INTRODUCTION

The radiators are generally used in automobiles for the purpose to transfer the heat of the coolant which is circulated around the internal combustion engines to the atmosphere so that the temperature of the engine is always kept below safe operating temperature. The radiators are specially placed in front of the vehicles and in turn increase the frontal area and during the movement of the vehicle it produces maximum drag force. To overcome this drag force more engine power will be required and that will increase the fuel consumption and decrease the engine efficiency. So we can tell that the engine efficiency depends on the size of the radiators. By reducing the size of the radiator the engine efficiency increases and vice versa. The size of the radiator can be reduced by using different type's methods and heat transfer augmentation technique is one of them without sacrificing the efficiency. Attaching the fins on either side of the flow passage is one of the heat transfer augmentation technique and different researchers worked on this are described below.

Recently experimental study performed by different researchers shows that friction factor and Nusselt number is higher for rectangular and triangular fins compared to round crest fins [1]. Moore and Joshi [2] concluded from experimental investigation by introducing a small amount of tip clearance in pin fin increases the heat transfer and reduces the pressure drop. Moon and Lau [3] conducted experimental study on rectangular channel to study the steady heat transfer between the two blockages having holes in the turbulent for nine different staggered arrays of holes in the blockages. They found that the blockages enhanced the heat transfer, but the pressure drop was increased significantly. For an offset fin array it was seen that the friction factor decreases when the Reynold's number changes from 10000 to 20000 and at higher Reynolds number the friction factor is increased [4]. Experimental investigation conducted by Islam and Mozumder [5] for smooth tube and T-shaped internally fined tube revealed that the average heat transfer coefficient of finned tube was two times of smooth tube and the friction factor was five times than that of smooth tube. Li et al. [6] numerically analyzed a specially treated polymer circular tube which is attached with longitudinal fins for Reynolds number ranges from 2000 to 20000 and it was concluded that the convective heat transfer was enhanced for the oval shaped tube. Extensive numerical study for turbulent flow inside finned tube shows that after a certain height of fin, the Nusselt number increases significantly with fin height [7]. Three dimensional numerical investigations by Liting et al. [8] revealed for a wavy finned tube heat exchanger with delta winglet vortex generators that Nusselt number and friction factor both increased with angle of attack and friction factor always increased with wavy angle. Agra et al. [9] investigated numerically the hydrodynamic and thermal behaviour of two helically finned tube and two corrugated tubes with Reynolds number varies from 12,000 to 57,000 for the water flowing in the inner tube. They have found that the corrugated tubes had the higher heat transfer coefficient than the smooth tube but a lower coefficient than the helical finned tubes.

Based on the above mentioned studies a comparison of fluid flow and heat transfer characteristics between smooth tube and internally finned tube for a turbulent flow conditions has been chosen for the present investigation. The experiment is performed for developing flow condition and is subsequently validated numerically using **Fluent 14.**

2. EXPERIMENTAL SET-UP

The schematic diagram of experimental set-up has been shown in the Fig. 1. The test section consists of an aluminium tube of 1 m length with two types of geometry. One is smooth tube having 0.05 m outside diameter and 0.044 m inside diameter and the other is finned tube of same dimension that of smooth tube having four internal longitudinal rectangular-section fins spaced at equal distances attached at the internal circumference of the tube by means of casting methods. All the four fins are of equal size of 0.9 m length, 0.01 m depth and 0.003 m width. Radial holes were drilled on the outer surface of the tube along axial flow direction for the fitment of thermocouples. Air is supplied into the test pipe by a blower as shown in Fig. 1 and the air flow rate is regulated by means of regulator valve fitted at the end of the test pipe. A manometer was used to measure pressure drop across the orifice from where the air flow rate inside the tube was measured. Heating coils were wound around the aluminium pipe to supply constant heat fluxes when it was connected to a power supply with help of a magnetic contactor and temperature controller. Heat fluxes are controlled by means of electric supply, current flow is regulated by voltage regulator (Variac). Ammeter and voltmeter are used to measure the current and voltage respectively for calculation of heat flux. To avoid the heat losses to the atmosphere three layers of insulations are provided which consists of 7 mm thickness of asbestos rope, 25 mm thickness of glass wool and a PVC pipe of 3 mm thickness.

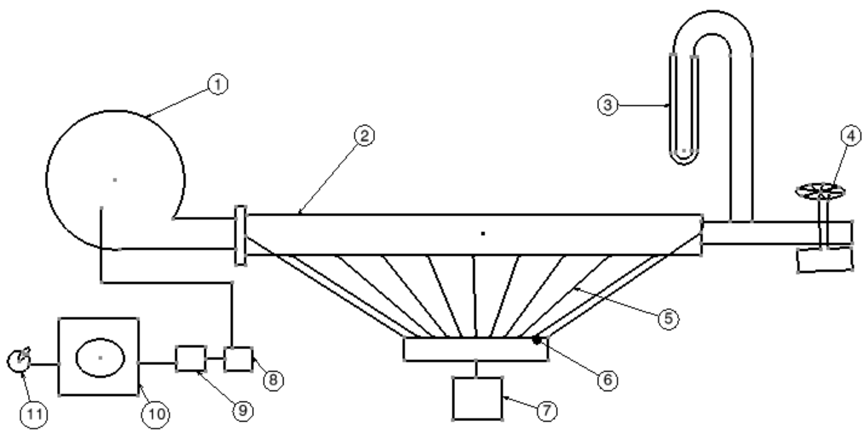

Fig.1: Schematic diagram of the test rig

1. Blower 2. Test section 3. Manometer, 4. Flow regulator, 5. Thermocouple 6. Selector switch 7. Temperature indicator 8. Voltmeter, 9. Ammeter, 10. Voltage regulator, 11. On/Off switch

The test rig is supplied modulated current by regulating it through the voltage regulator. The voltage and current are set at a particular position by watching the voltmeter and ammeter respectively. There are 9 numbers of thermocouples spaced at equal distances on the outer surface of the test pipe to measure the temperatures axially at different nodes. One thermocouple was attached at inlet and three thermocouples were attached at the outlet to measure the inlet and outlet temperatures respectively. At the outlet three thermocouples are placed radially at different distance and the average temperature of three thermocouples readings is the measure of outlet temperature. A snap view of the experimental set-up has been shown in Fig. 2.

Fig.2: Snap view of experimental set up

Fig.3: Internal fins on inner periphery of the tube

3. MATHEMATICAL FORMULATION

The numerical simulations are performed for an internally finned tube of diameter, D and length, L as shown in the Figure 3. Air at room temperature is allowed to enter into the tube from one end and the other end is exposed to atmosphere. Rectangular fins are attached at the inner periphery of the tube and runs in axial direction. The investigation started with four rectangular fins and subsequently fin numbers changed by keeping the mass flow rate of air was kept constant. The flow field in the domain would be computed by using incompressible, 2D axi-symmetric, Navier-Stokes equations with two-equation based k-ε turbulence model along with the energy equations. The fluid used is air, at temperatures taken between 300 to 500 K.

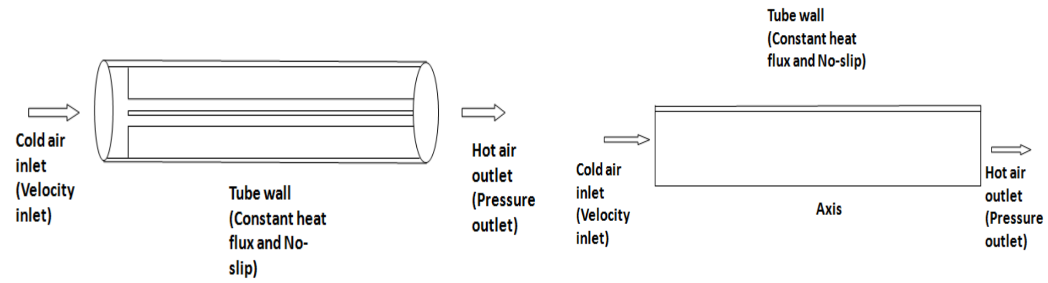

Fig. 4: Schematic diagram of computational domain and the boundary condition applied to it (a) finned tube; (b) un-finned tube

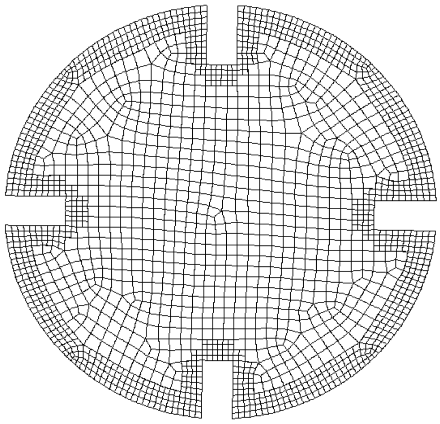

Fig.5: Cross sectional view of the internal finned tube showing the meshes inside the tube and around the fins

Two cylindrical aluminium pipes exactly of the same size and shape are used both for numerical and experiment investigation as shown in the Fig.4. For smooth tube two dimensional axi-symmetric models has been chosen. Fluent 14 is used to determine the flow field and temperature field in the finned tube domain would be computed by using 3-D (two dimensional axi-symmetric model for smooth tube), incompressible Navier-Stokes equations with a two equation based *k*- ε turbulence model along with the energy equation. Air is treated to be incompressible at room temperature is used in the simulation at injection velocity (which is below 10 m/s).

Fig. 4a and 4b shows the boundary conditions respectively for finned tube and smooth tube. As the density is taken as a function of temperature according to ideal gas equation therefore SIMPLE algorithm with PRESTO (Pressure Staggered Option) scheme has been used for better convergence. Under relaxation factors (0.3 for pressure, 0.7 for momentum and 0.8 for *k* and ε) were used for the convergence of all the variables. Convergence of the discretized equations were said to have been achieved when the whole field residual for all the variables fell below 10^{-3} except energy equation and for energy equation residual was set 10^{-6}.

Governing Equations

Conservation of mass

$$\nabla.(\rho \vec{v}) = 0 \tag{1}$$

Conservation of momentum

$$\nabla.(\rho \vec{v}\vec{v}) = -\nabla p + \nabla.\left(\mu^{eff} \nabla \vec{v}\right) \tag{2}$$

Conservation of energy

$$\frac{D(\rho T)}{Dt} = \frac{\partial}{\partial x_i}\left[\left(\frac{\mu}{\Pr} + \frac{\mu_t}{\Pr_t}\right)\frac{\partial T}{\partial x_i}\right] \tag{3}$$

Turbulence kinetic energy

$$\nabla.(\rho \vec{v} k) = \nabla.\left(\frac{\mu_t}{\sigma_k}\nabla k\right) + G_k - \rho\varepsilon \tag{4}$$

Turbulent rate of dissipation

$$\nabla.(\rho \vec{v} \varepsilon) = \nabla.\left(\frac{\mu_t}{\sigma_\varepsilon}\nabla \varepsilon\right) + \frac{\varepsilon}{k}\left(C_{1\varepsilon}G_k - C_{2\varepsilon}\rho\varepsilon\right) \tag{5}$$

Where G_k=Generation of turbulent kinetic energy due to mean velocity gradient and turbulent or eddy viscosity

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

$$C_{1\varepsilon} = 1.44; C_{2\varepsilon} = 1.92; C_\mu = 0.09; \sigma_k = 1.0; \sigma_\varepsilon = 1.3, \Pr_t = 1$$

The effect of fin number on wall temperature can be seen from the Figure 6. The present computations have been performed on rectangular sectioned longitudinal finned tube of length 1 m and diameter 70 mm. As the fin number was increased from 0 to 8 the wall temperature is dropped to the lowest value and after that when we increase the fin number further (i.e. n=12, 16 and 20) the wall temperature is raised again.

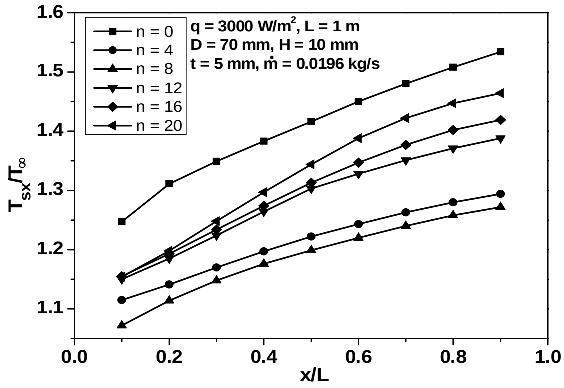

Fig.6: Effect of fin numbers on wall temperature

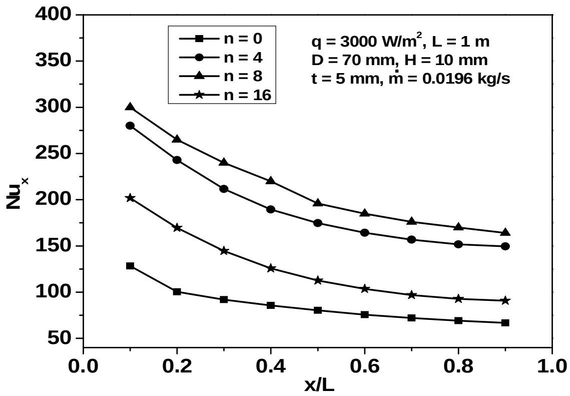

Fig.7: Effect of fin numbers on Nusselt number

As the number of fin is increased in an internal finned tube, the transfer of heat to the air is increased up to a certain number of fins but after that obstruction is created by the fins within the tube and the heat transfer rate is adversely affected and the wall temperature is increased.

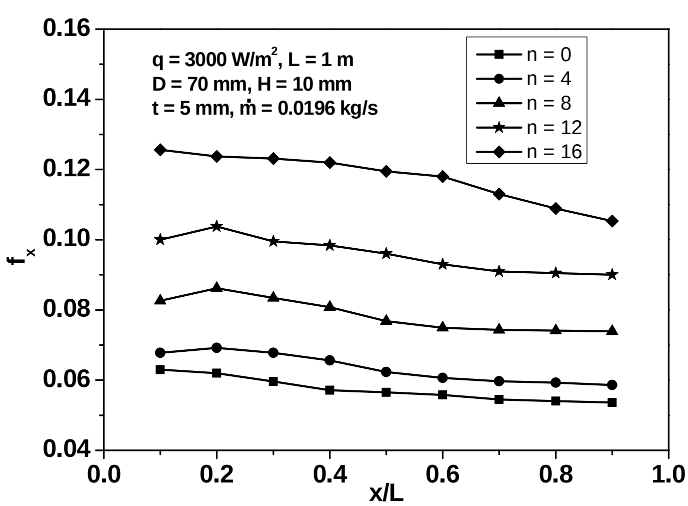

Fig.8: Effect of fin number on friction factor

This is also visualised from Figure 7 that the Nusselt number is highest for 8 numbers of fins and the Nusselt number is increased approximately 2.5 times for 8 numbers fins compared to plain tube. So it can be concluded that the convective resistance is minimum when the fin number is 8 and after that it increases significantly with the fin numbers. However the pressure drop is increased along with the fin numbers and due to that there was a sharp rise in the friction factor that can be seen from Figure 8.

4. CONCLUSIONS

Numerical and experimental investigations have been performed to study the thermo-fluid behaviour of finned tube and plain tube. Experiment was performed for a constant Reynolds and wall heat flux. From the analysis the following conclusions can be drawn:

The wall temperature of internally finned tube was found to have a lowest value for 8 numbers of fins which could have been reduced by 16% compared to a plain tube. The Nusselt number is highest for 8 numbers of fins and the Nusselt number is increased approximately 2.5 times for 8 numbers fins compared to plain tube. The pressure drop is increased along with the fin numbers and due to that there was a sharp rise in the friction factor.

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