# Numerical And Experimental Analysis Of Heat Transfer Through Various Types Of Fin Profiles By Forced Convection

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## Abstract

In all engine two types of cooling systems are utilized, liquid-cooling & air-cooling, among the two types of engine cooling system, liquid-cooling is widely used due to its capacity to reject large amount of heat and air-cooling is preferred for small capacity engine in which the cooling system is much simpler in design, lighter in weight and low in cost. In air-cooled engine annular fins with different fin profiles are used for heat transfer enhancement, therefore it is important for an air-cooled engine to utilize fin profiles effectively to obtain heat transfer enhancement. The current work purpose is to suggest the optimum fin profile for heat transfer enhancement. A feasibility study is conducted on actual engine, conducting the experiment on actual engine is tedious and costly so, similar type of test rig is prepared. Fins are subject to constant heat source and cooled by airflow. Experiments were conducted for rectangular and triangular fin profiles for several air velocities ranging from 0 to 11 m/s. The temperature distribution in the fin was measured for several locations simultaneously to calculate the amount of heat being transferred. Experiment is repeated for both fins, Similarly CFD simulation is carried out for both fins. Finally experimental and CFD simulated result are compared. From the comparison it proves that annular fins with rectangular fin profiles are more suitable for heat transfer enhancement as compared to triangular fin profiles.



## 1. Introduction

Since from long time, there has been a progress demands for high efficiency and high power output engines. This implies that, it is necessary to study in details of engine systems and subsystems of which cooling system is also important component. Among the two types of cooling system, liquid-cooling and air-cooling, liquid-cooling is widely used for large capacity and air-cooled engine are used for small capacity. Air cooled engines are release the heat to atmosphere by forced convection. The rate of heat transfer depends up on the air velocity and geometry of the fins profiles. The extended fins profiles well known for enhancing the heat transfer in engines. There have been number of on air cooled engine fins.

Magarajan [1] observed that with increase in fin pitch heat transfer increases up certain extent. Heat release from the cylinder is analyzed at a wind velocity of 0 km/h. The trial conducted for 600 s, heat release by ethylene glycol through cylinder to fins is about 28.5 W for 10 mm pitch and 33.90 W for 20 mm pitch. Kumbhar [2] studied the rectangular fins with triangular perforations on it. It is observed that perforation size increases heat transfer increases up to certain extent & after that it start decreases. Vijay [3] optimizes the cooling system of scooter engine by modification in cowl profile and cooling fan blade angle. Observed that there is significant reduction in temperature of oil and fin surface with 3.1% reduction in fan power, while maintaining the same flow rate. Mohd Faizal [4] observed the effect of moist cooling. on engine performance. Engine is cooled by moist air instead of dry air due to moisture heat carrying capacity of air increases. Pulkit [5] in his study experienced that heat lost at same vehicle speed increases with decrease in atmospheric temperature. Also, with temperature remaining constant, the heat transfer coefficient increases with velocity leading to increased heat transfer. An excess heat loss from the engine surface is undesirable as it results in decreased efficiency and excess fuel consumption. Alan [6] Conducted simulation on annular fin stack with specification of OD-100, L-40, P-14. Perform the mesh independency study, only by refining the size of mesh solution accuracy not accurate. Variables in the models are Boundary surface size, Cell Growth rate, Prism layer thickness, Prism Layer growth rate, Primary prism layer cell size, Cell density. Observed that effect of different mesh size 15, 10, 5 mm on temperature is 0.5 K.

There are different fin profiles available but among all the profiles, which is the suitable for heat transfer enhancement that is analyzed in current work. The present work details the feasibility study carried out to improve air-cooled systems by changing the fins profiles. The optimal cooling rate is required for efficient operation. If the cooling rate is decreases, results in overheating leading to seizure of the engine. At the same time, an increase in cooling rate affects the starting of the engine and reduces the efficiency

## 2. Methodology

### 2.1. Experimental set-up

Since conducting an experiment on actual engine is tedious and costly process. For simplification test rig is prepared in CAD software for simulation, which save time and cost. The method provides better control on measured parameter and maintains accuracy. The experimental setup consists of annular fins with different fin profiles. Annular fin liner with 13 fins fitted in PVC duct, its one end fitted with blower and other is opened. The blower is driven by electric motor with belt-pulley arrangement. A 7.5 HP electric motor used to generate air velocity up to 12m/s shown in Fig 1. The fins base is heated at constant heat flux by gas heater. The thermocouples are used to measure the temperature of fin surface and air. Anemometer used to measure the air velocity.

Simulation is carried out at different air velocities on cylinder liner with two different fins profiles specifically, rectangular and triangular. Test setup remains unchanged only replacement of rectangular and triangular fins profiles to be done at the time of simulation. Fig. 2 show the schematic diagram of experimental set-up.

**Figure** 1: Experimental set-up

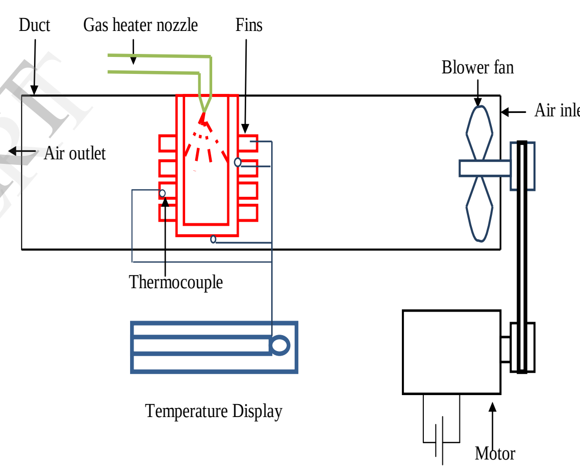



**Figure** 2: Schematic diagram of experimental set-up

### 2.2. Location of thermocouples

K- Type thermocouples utilized for measurement of surface temperature and temperature of air flowing through duct. To locate the thermocouple hole of 1.5 mm diameter and 3 mm depth created on surface of the fins. The probe of the thermocouple with adoiltoit liquid penetrate in the created hole, and kept it for 24 hr for perfect bonding between thermocouples and surface of the fins. On the fins surface only 9 thermocouples are located to avoid the flow abstractions. All the thermocouples are at an angle $90^0$ apart shown in the Fig. 3

**Table** 1: Position of thermocouple.

| Thermocouple | Location |
|---|---|
| $T_0$ | $0^0$ Liner surface temperature |
| $T_1$ | $0^0$ Fin tip temperature |
| $T_2$ | $0^0$ Fin base temperature |
| $T_3$ | $90^0$ Fin tip temperature |
| $T_4$ | $180^0$ Liner surface temperature |
| $T_5$ | $180^0$ Fin tip temperature |
| $T_6$ | $180^0$ Fin base temperature |
| $T_7$ | $270^0$ Fin tip temperature |
| $T_8$ | Liner bottom temperature |
| $T_{In}$ | Air inlet temperature |
| $T_{Out}$ | Air outlet temperature |

**Figure** 3: Location of thermocouples at different location on fins surface

## 2.3. Modeling and design

The test rig was modeled in CATIA and simulated in CFD software Fig. 4. The annular cylinder liner with rectangular fin profiles shown in Fig. 5. By keeping other parameter constant shape of the fin profiles is changed from rectangular to triangular and modeled is simulated.

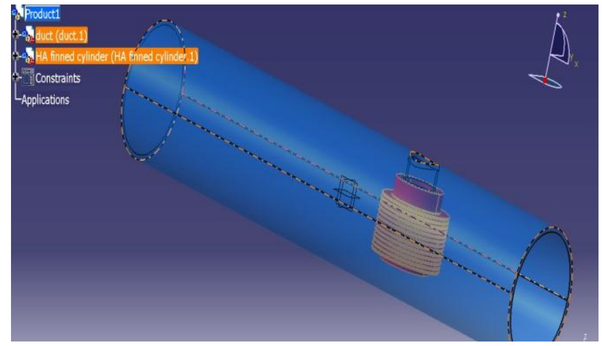
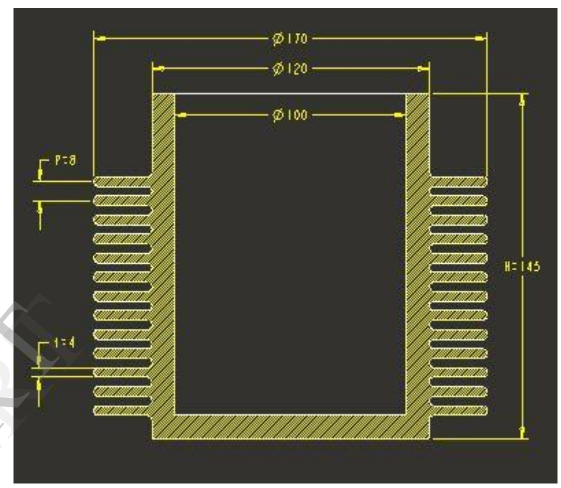
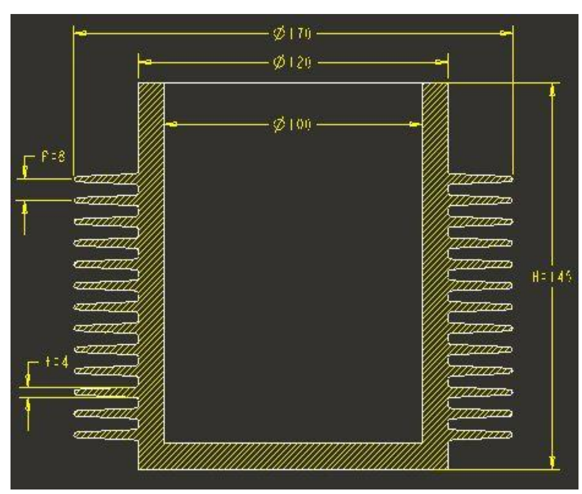



**Figure** 4: CAD model




(a) Rectangular profile




(b) Triangular profiles

**Figure** 5: Annular cylinder liner with different fins profiles

## 2.4. Computational modeling

The computational domain consists of a PVC duct and rectangular fins, PVC large dimensions containing the finned cylinder at its centre. It was focused on the fins and appropriate boundary conditions were applied at the domain ends to maintain continuity. The domain was made longer at outlet of the cylinder to allow for wake formation.

## 2.5. Meshing

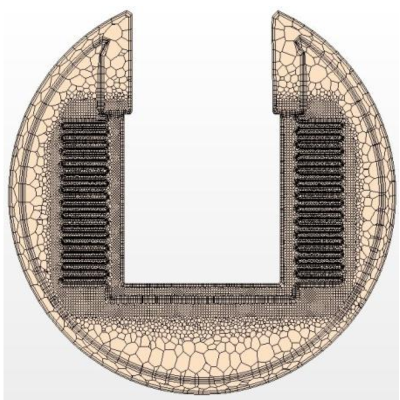


**Figure** 6: Prism layer with controlled volume mesh

From meshing model, surface remeshing, tetrahedral and polyhedral mesh type is selected for surface and volume mesh generation respectively with extruder at its outlet. Latter on specify the base size of the element and number of prism layer with absolute or relative prism layer thickness. After this initialize the meshing operation. Fig. 6 section of volume mesh

## 2.6. Mathematical model

The 3-dimensional heat flow through cylinder liner and fins were simulated by solving the appropriate governing equations viz. conservation of mass, momentum and energy using CFD code which work by finite volume method.

Conservation of mass:

$$\nabla.\left(\rho.\vec{V}\right) = 0 \qquad \ldots (1)$$

X-momentum:

$$\nabla.\left(\rho u\,\vec{V}\right) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} \qquad \ldots.. (2.1)$$

Y-momentum:

$$\nabla.\left(\rho u\,\vec{V}\right) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + \rho g \quad \ldots (2.2)$$

Z-momentum:

$$\nabla.\left(\rho u\,\vec{V}\right) = -\frac{\partial p}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial z}{\partial z} + \rho g \qquad \ldots (2.3)$$

## 2.7. Boundary condition

Steady state conjugate heat transfer analyses carry out for finned cylinder at different air velocity varied from 0 to 12 m/s, and air temperature at ambient condition.

In simulation the three regions are created, among this region two are solid and one is fluid. In region cylinder liner and duct is solid region, air flowing through duct around the cylinder liner that is fluid region. The entire regions are separated by interfaces. The function of interface is to exchange the thermal properties.

In cylinder liner region the fins are subjected to constant heat source. The cylinder liner surface temperature maintained at $180^0$C, material properties of casted liner GG25 specified.

In duct region, duct of PVC material and open to atmosphere. Due to atmospheric condition its boundary are subjected to natural convection rate of 5 $W/m^2K$.

In fluid region, all the boundaries are subjected to adiabatic conditions. Only the inlet and outlet conditions of the entering fluid are specified as-

- Inlet face was given fixed mass flow condition. The mass flow rate of air and temperature of the entering air was chosen depending upon the case being solved.

- Outlet face was given fixed pressure condition of 1.01325 Pa representing the atmospheric pressure.

Field function set for heat transfer coefficient and nusselt number. For heat transfer coefficient and nusselt number reference temperature is specified as average temperature of fluid from inlet to outlet. For nusselt number reference length is specified as the hydraulic diameter, 0.032 m for rectangular and 0.033 m for triangular fin profiles.

## 2.8. Contour plots

### 2.8.1. Velocity distribution across fins

As the flow passing through duct at 12 m/s, before striking the fin surface flow velocity become constant but at the hanging point of fins, the gap between fins and duct is very small which accelerate the flow. At the back side of the fins there is very less velocity as indicated by blue color. Near the fins side the velocity is high due to less space in the region. The position of the thermocouples are given in the Fig. 7.

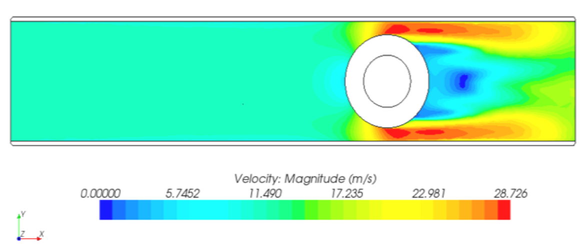
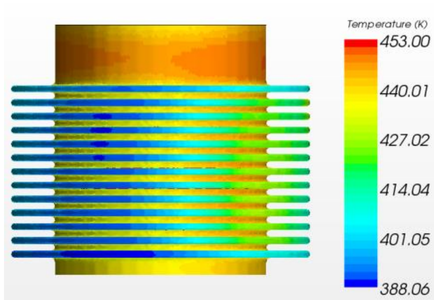
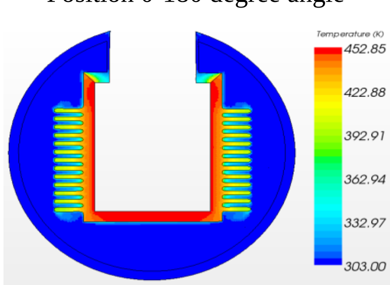



**Figure** 7: Velocity and position of thermocouple distribution across fins

### 2.8.2. Temperature distribution rectangular fin liner

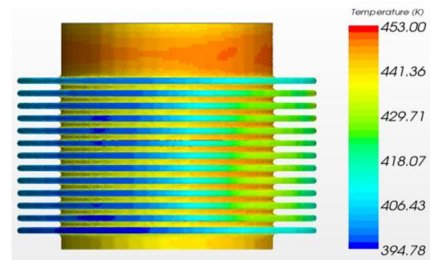



Position 0-180 degree angle




Position 90-270 degree angle

(A) Velocity = 11.38

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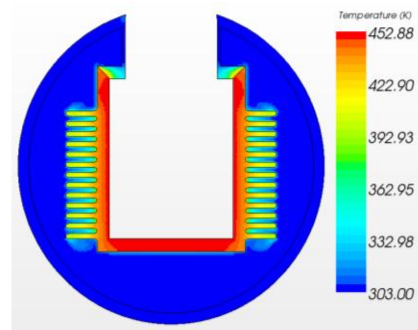
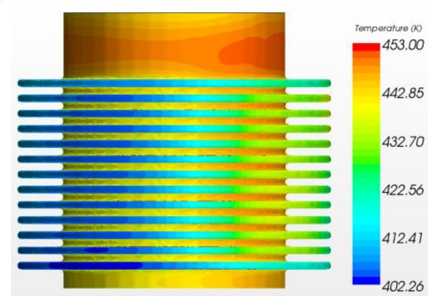
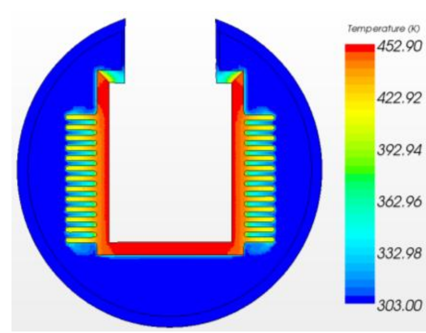



Position 90-270 degree angle

(B) Velocity = 8.76 m/s




Position 0-180 degree angle

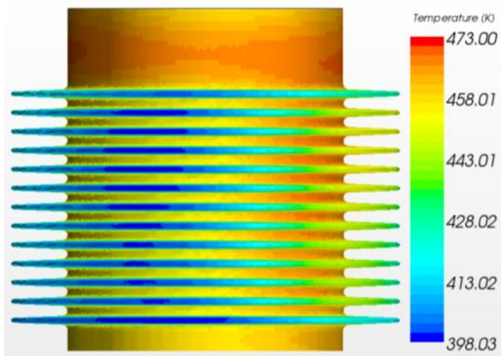



Position 90-270 degree angle

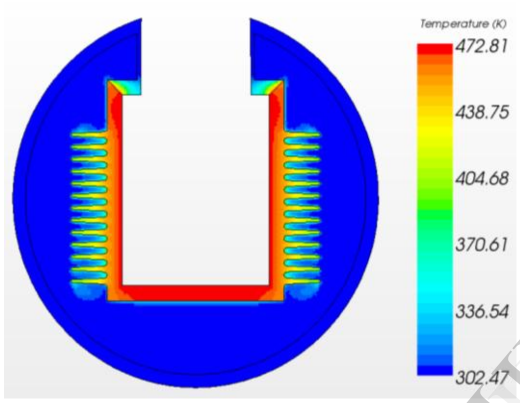
(C) Velocity = 6.2 m/s

**Figure** 8 (A-C): Temperature distribution in liner in rectangular fin liner at different location and at different air velocity

### 2.8.3. Temperature distribution in triangular fin liner

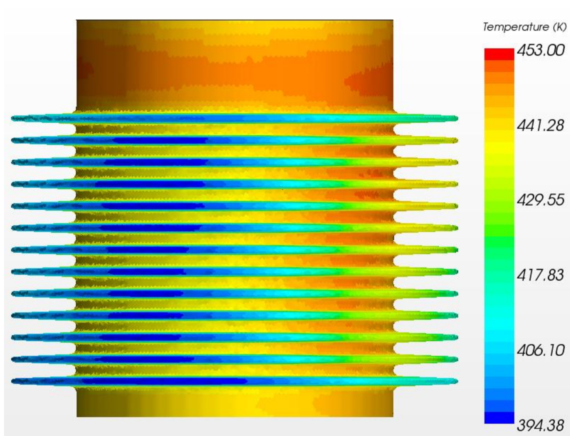



Position 0-180 degree angle

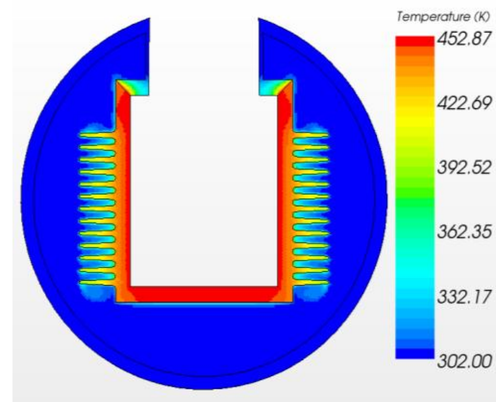



Position 90-270 degree angle

(a) Velocity = 11.4 m/s

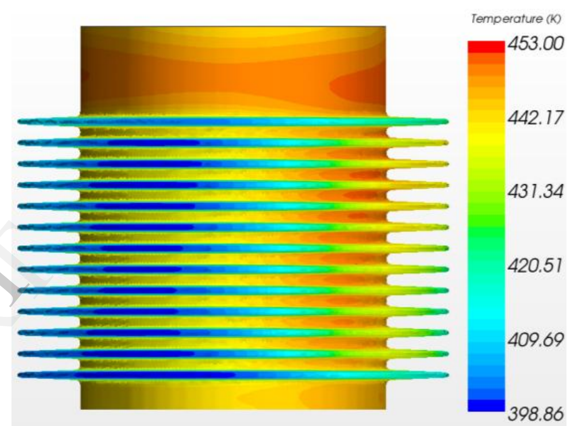



Position 0-180 degree angle

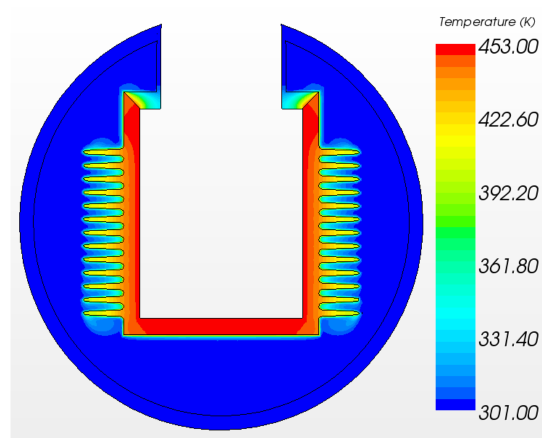



Position 90-270 degree angle

(b) Velocity = 8.74 m/s




Position 0-180 degree angle




Position 90-270 degree angle

(c) Velocity = 5.82 m/s

**Figure** 9 (a-c): Temperature distribution in triangular fin liner at different location and at different air velocity

In steady state simulation temperature distribution all over the fin surface at different air velocity and at different position at rectangular and triangular fins are given in Fig. 8 (A-C) and Fig 9 (a-c).

## 3. Result and Discussion

The convection heat transfer from fin surface to atmosphere by forced air is given as-

$$Q = hA(T_{avg} - T_{air}) \qquad \ldots (5)$$

Where,
Q: Heat flux from fin surface, h: Fin surface heat transfer coefficient, A: Surface area of fin, $T_{avg}$: Average fin surface temperature, $T_{air}$: Bulk temperature of air.

### 3.1. Determination of heat transfer coefficient

As the given case is the forced convection, the flow is turbulent, constant wall temperature and flow through pipe. To find out the convective heat transfer Churchill and Bernstein equation is used,

$$Nu = 0.3 + \frac{0.62 Re_D^{1/2} Pr^{1/3}}{\left[1+\left(\frac{0.4}{Pr}\right)^{\frac{2}{3}}\right]^{1/4}} \left[1 + \left(\frac{Re_D}{282000}\right)^{\frac{5}{8}}\right]^{4/5} \qquad \ldots (6)$$

Where, n is 0.3 for cooling and 0.4 for heating application.

$$Re = \frac{VD}{v} \qquad \ldots (7)$$

As the flowing fluid is air the prandtl number = 0.701 for present case.

$$Nu = \frac{hD}{k} \qquad \ldots (8)$$

From the above expression the convective heat transfer coefficient is calculated..

### 3.2. Velocity measurement

In experimentation the velocity is measure at different grid point and average velocity is considered for analytical calculation and simulation purpose. From the experimental result. It is observed that velocity in rectangular fin profile is slightly higher than triangular fin profile. From Fig. 10 show that in both profiles there is slight variation in velocity.

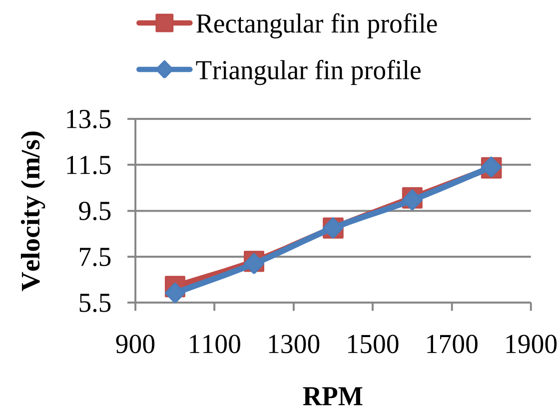



**Figure** 10: Experimental velocity measurement

### 3.3. Temperature variation

Temperatures are measured at various locations in the liner at different velocity. There is continuous variation in temperature due to variation of velocity. As the velocity increases temperature decreases in both triangular fins and rectangular fins. The trend followed by simulated and experimental result is same as shown in Fig. 11.

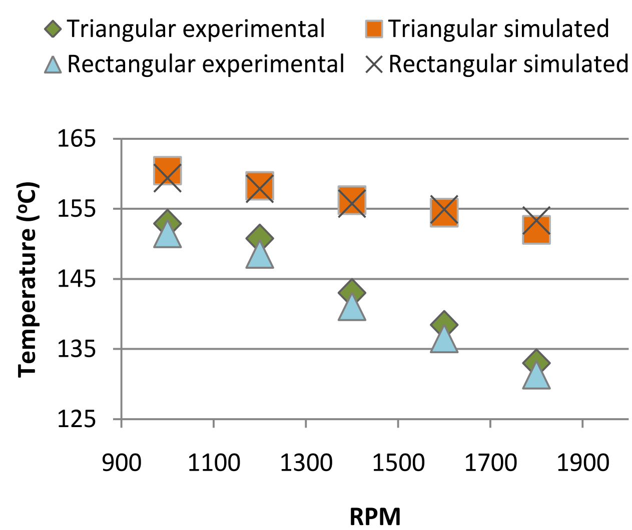



**Figure** 11: Experimental and simulated temperature variation

In experimentation temperature of triangular fin profile is higher than rectangular fin profile. Similarly in simulation also triangular fin profile temperature is higher than rectangular fin profile.

### 3.4. Heat transfer coefficient variation

In experimentation the heat transfer coefficient is calculated from Churchill and Bernstein equation. In simulation the heats transfer coefficient direct output.

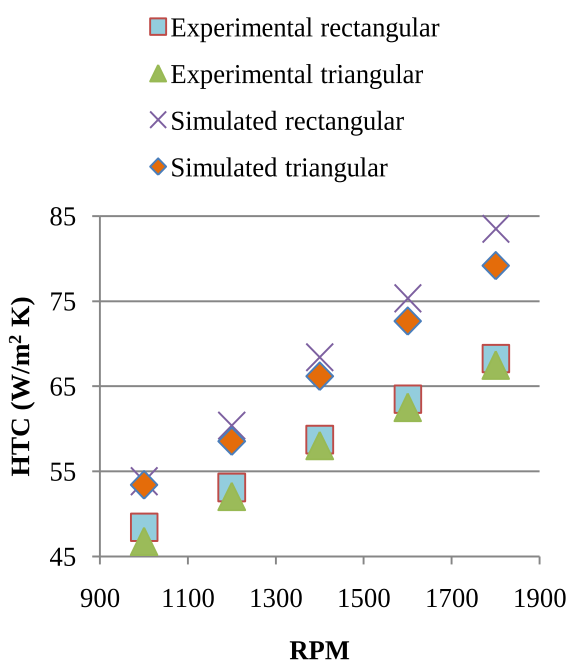



**Figure** 12: Experimental and simulated HTC variation

In experimental result observed that, heat transfer coefficient of rectangular fin profile is higher than triangular fin profile. In experimental result of both fin profile have 2% difference is observed. Similarly, in case of simulation also observed that heat transfer coefficient of rectangular fin profile is higher than that of triangular fin profile. In simulation result of both fin profile have 4% difference observed.

In experimental and simulated both profiles it is observed that heat transfer coefficient increases with increasing velocity. But in comparison with experimental and simulated results of both profile observed that there is 12 to 18% difference as shown in Fig. 12.

### 3.5. Heat transfer

In experimentation the heat transfer coefficient is calculated. In simulation the heat transfer coefficient direct output.

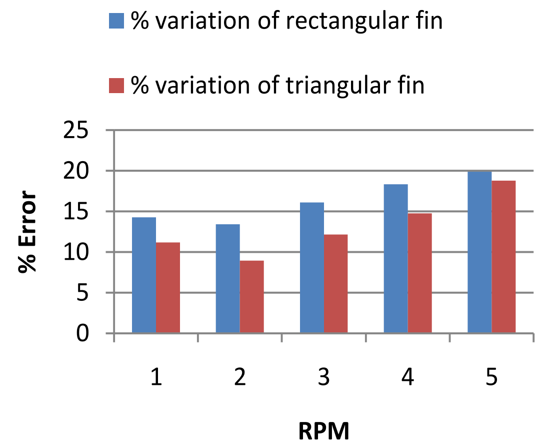



**Figure** 13: Experimental and simulated heat transfer variation

In experimental result observed that, heat transfer of rectangular fin profile is higher than triangular fin profile. In experimental result of both fin profile 10 to 12% difference is observed. Similarly, in case of simulation also observed that heat transfer of rectangular fin profile is higher than that of triangular fin profile. In simulation result of both fin profile 8 to 10% difference observed.

In experimental and simulated both profiles it is observed that heat transfer increases with increasing velocity. But in comparison with experimental and simulated results of both profile observed that there 14 to 19% variation of heat transfer Fig 13.

## 4 Conclusion

- The air flow velocity in abstraction with rectangular fin profile is higher than triangular fin profiles.
- Surface temperature of triangular fin profile is higher than rectangular fin profile at different air velocity.
- Heat transfer coefficient increase with increases with increases in velocity in both profiles. In comparison of both profile rectangular fin profile have higher heat transfer coefficient than triangular fin profile.
- Amount of heat transfer increases with decreasing the surface temperature or increasing air velocity in both profiles. In comparison of both profile rectangular fin profile transfer large amount of heat than triangular fin profile.

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