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Experimental and Parametric Study of Extended Fins In The Optimization of Internal Combustion Engine Cooling Using CFD

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Abstract - In this experiment the single cylinder air cooled engines was assumed to be a set of annular fins mounted on a cylinder. Numerical simulations were carried out to determine the heat transfer characteristics of different fin parameters namely, number of fins, fin thickness at varying air velocities. A cylinder with a single fin mounted on it was tested experimentally. The numerical simulation of the same setup was done using CFD. The results validated with close accuracy with the experimental results. Cylinders with fins of 4 mm and 6 mm thickness were simulated for 1, 3, 4 & 6 fin configurations.

Keywords-Numerical simulation,,Annular fins.

I. INTRODUCTION

As the fossil fuel reserves are depleting day by day, the spiraling fuel price is pushing the technology towards it limit to provide engines which are highly efficient and produces high specific power. Air cooled engines are gradually phased out and are being replaced by water cooled engines which are far more efficient in dissipating heat, but in cases of two wheelers and certain other applications, air cooled engines are the only viable option due to space constraints.

The heat which is generated during combustion in an internal combustion engine should be maintained at the highest level possible to increase its thermal efficiency, but in order to prevent the thermal damage to the engine components and the lubricants some amount of heat must be removed from the system. If the cooling is not sufficient greater thermal expansion of the engine might occur and scuffing might occur between the pistons. Parameter analyzed in this paper includes fin number, fin thickness and wind velocities. The meshing for the computational problem was done primarily using the software ICEM CFD and the solver used was FLUENT.Hexa-meshing was used for all geometries. The numerical study is extended to study the effect of fin pitch, normal and tapered fins, effect of holes and slits in fins etc.

II. EXPERIMENTAL SET UP

The experimental setup consisted of cast iron as cylinder material and aluminum alloy 6101 as the fin material. The fin was machined to a thickness of 6mm and to a length of 50mm from the root of the fin. The cast iron cylinder was machined to an outer diameter of 80mm and inner diameter of 60mm.The height of the cylinder is 120mm. SAE40 oil was used as the heated fluid element. Initially the oil was heated to 200°C and poured into the cylinder and the heat transfer and the cooling rate of cylinder and fin is monitored at regular intervals of time. Temperatures at different parts of the fin are measured with the help of J-type thermocouples which are attached securely to prevent the interference due to atmospheric conditions. A multi input digital temperature display is used to note the temperatures.

Figure 1 Model of the experimental set up along with the point of attachment of the thermocouple

The volume of the hollow portion of the experimental cylinder is 300cm^3 . A heater with electrical power of 400 W and J type thermocouple were inserted into the cylinder to identify the temperature. The heat storage liquid was heated to approximately 200°C and is poured into the cylinder and the cap is placed above the cylinder. The temperature of the heat storage liquid was decreased to 120°C by cooling at room temperature. The heat release from the cylinder was obtained by multiplying the heat capacity of the heat storage liquid by the difference between 120°C and the temperature after every 30 seconds at an air velocity of 0 km/hr. The experiments were carried out at an ambient temperature of 33°C . The ambient temperature has a significant effect on the heat release from the cylinder. The experimental setup is shown in the image below.

Figure.2 Experimental setup

COMPUTATIONAL MODELING

Figure3 and 4 shows the meshed cylinder and fins which was used for CFD analysis and figure 5 is showing cylinder combined with fin. The dimensions of cylinder and fins are similar to the model in literature. The outer diameter of the cylinder is $\varnothing 180\text{ mm}$, inner diameter $\varnothing 80\text{ mm}$ and the thickness of 6 mm. The outer and inner diameters of the cylinder are $\varnothing 80\text{ mm}$ and $\varnothing 60\text{ mm}$. The cylinder is of a length of 120 mm ¹¹. The hollow cylinder, fins along with the outer air domain is created separately in Pro-E wildfire 4.0 and is then assembled. The output assembly design is created in Parasolid format file for grid generation in ANSYS-ICEM CFD meshing tool. Fig.6 showing the assembled geometry with air box

Figure 3 Meshed fin figure.4 cylinder figure.5 fin with Cylinder

Figure 6 Assembled Geometry with air box

GRID GENERATION

The 3-D model is then discretized in ICEM CFD meshing tool. In order to capture both the thermal and velocity boundary layers the entire model is discretized using hexahedral mesh elements which are accurate and involve less computational effort. Fine control on the hexahedral mesh near the wall surface allows capturing the boundary layer gradient accurately. The entire geometry is divided into four domains FLUID_OIL, FLUID_AIR_SURROUNDING, SOLID_FINS and SOLID_CYLINDER. The discretized model is checked to have a minimum angle of 27° and min determinant quality of 65 %. Once the meshes are checked for free of errors and minimum required quality it is exported to ANSYS FLUENT pre-processor.

ASSUMPTIONS

Some of the following assumptions are made for the analysis of the fins.

1. The heat flow through the fin is considered as in steady state, so that the temperature of the fin does not vary with time.
2. The contact thermal resistance between the cylinder the fin is not considered.
3. The temperature of the fin does not vary along the thickness of the fin.
4. The thermal conductivity of the fin material is uniform and constant.
5. The radiation heat transfer of the fin is neglected.
6. Uniform ambient temperature of 306 K is considered

GOVERNING EQUATIONS

The 3-dimensional heat flow through the cylinder and fins were simulated by solving the appropriate governing equations viz. conservation of mass, momentum and energy using ANSYS FLUENT code which work by finite volume approach^{12, 13}. Turbulence is taken care by Shear Stress Transport (SST) $k-\omega$ model.

$$\text{Conservation of mass: } \nabla \cdot (\rho \vec{V}) = 0 \text{ ----- (4)}$$

$$\text{X-momentum: } \nabla \cdot (\rho u \vec{V}) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} \text{ ----- (5.1)}$$

$$\text{Y-momentum: } \nabla \cdot (\rho v \vec{V}) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + \rho g \text{ ---- (5.2)}$$

$$\text{Z-momentum: } \nabla \cdot (\rho w \vec{V}) = -\frac{\partial p}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + \rho g \text{ --- (5.3)}$$

$$\text{Energy: } \nabla \cdot (\rho e \vec{V}) = -p \nabla \cdot \vec{V} + \nabla \cdot (k \nabla T) + q + \phi \text{ ----- (6)}$$

SST omega turbulence equation:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\Gamma_k \frac{\partial k}{\partial x_j} \right] + G_k - Y_k + S_k \text{ ----- (7)}$$

BOUNDARY CONDITION SETUP

Both the fluids SAE 40 oil and air are assumed to be incompressible fluids. Ambient temperature and pressure are assumed as 306 K and 101325 Pa respectively. The values of the boundary conditions like operating temperature, velocity of air are taken from the experimental work. Other boundary conditions like density, specific heat, thermal conductivity and other material properties are considered as constants throughout the analysis. The mesh is imported to ANSYS-FLUENT and then the domains are initialized.

The boundary conditions and the interface cylinder, fins, oil and air are set in the solver. The top and bottom of the cylinder surface are assumed to be adiabatic as it is insulated as per the experiment. The oil domain is initialized at a temperature of 423 K as the initial temperature of the domain as per the experiment. The heat transfer takes place due to natural convection and conduction up to 393 K so that the fins and cylinder can be initialized with some higher temperature value than ambient temperature. After the temperature reaches 393K air at inlet velocity of 0 km/hr. is passed over the cylinder and fins. The heat release from ethylene glycol from 393K after a time period of 10 minutes is calculated.

VALIDATION: Experimental results validated by Fluent

Figure7 Contour of Oil domain temp. at 20 sec

Contours of Static Temperature (k) (Time=2.0000e+01)
ANSYS FLUENT 12.0 (3d, pbns, mgke, transient) Apr 12, 2012

Figure8 Contour of Oil domain temp at 40sec

Contours of Static Temperature (k) (Time=4.0000e+01)
ANSYS FLUENT 12.0 (3d, pbns, mgke, transient) Apr 12, 2012

Figure9 Contour of Oil domain temp at 60sec

Contours of Static Temperature (k) (Time=6.0000e+01)
ANSYS FLUENT 12.0 (3d, pbns, mgke, transient) Apr 12, 2012

Figure10 Contour of Oil domain temp at 80sec

Figure10 Contour of Oil domain temp at 80sec

Figure13 Temperature difference at 12mm from root of fin

Figure11 Contour of oil domain temp at 100sec

Figure14 Temperature difference at 24mm from root of fin

Figure12 Temperature difference at fin root

From figures 12, 13&14, the CFD values obtained correspond to the value obtained by the experiments with a maximum error of $\pm 10\%$.

RESULTS AND DISCUSSIONS

Temperature contours for 4mm thickness fins

Figure15a) 20km/hr Velocity

Figure15b)40km/hr Velocity

Figure16c) 60km/hr. Velocity

Figure15c)60km/hr Velocity

Figure16d) 80km/hr. Velocity

Figure15d)80km/hr Velocity

Figure17a) 20km/hr. Velocity

Figure16a) 20km/hr. Velocity

Figure17b) 40km/hr. Velocity

Figure16b) 40km/hr. Velocity

Figure17c) 60km/hr. Velocity

Figure17d) 80 km/hr. Velocity

Figure18 Temperature difference vs wind velocity for different no. of fins

Figure.19a)Temp. contour for 4mm fin(80km/hr)

Figure.19b)Velocity contour for 4mm fin(80km/hr)

Figure20 Temperature vs Lateral Distance from center of fin for 4mm .

Temperature contours for 6mm fin thickness

Figure21a) 20km/hr Velocity

Figure21b) 40km/hr Velocity

Figure21c) 60km/hr Velocity

Figure21d) 80km/hr Velocity

Figure22d) 80km/hr Velocity

Figure22a) 20km/hr Velocity

Figure23a) 20km/hr Velocity

Figure22b) 40km/hr Velocity

Figure23b) 40km/hr Velocity

Figure22c) 60km/hr Velocity

Figure23c) 60km/hr Velocity

Figure23d) 80km/hr Velocity

Figure24 Temperature difference vs wind velocity for different no. of fin

Figure.25a)Temp. contour for 6mm fin(80km/hr)

ig.25b)Velocity contour for 6mm fin(80km/hr)

Figure26 Temperature vs Lateral Distance from center of fin for 6mm .

Figures 15(a-d),16(a-d),17(a-d) shows the temperature contours for, three fin, four fin & six fins configurations respectively for 4mm fin thickness. The velocity profile (19(b)) and the heat contours of the fins (19 (a)) are also shown in diagrams. It can be noted from fig.18 that the temperature difference between the root and tip of the fin is maximum for 6 fins configuration be seen from the graphs and the temperature contours. It can be inferred from fig.20&fig.19a that the temperature difference from the fin root to the tip for natural convection is very low, but for forced convection particularly for the outside side of the cylinder the temperature difference is very high. This is due to greater velocity of the impinging air at the outside fin surface. Conversely the temperature difference is comparatively lower for the inside side of the cylinder due to the reduction in the effectiveness of the wind velocity due to the blockage by the cylinder, this can be seen clearly from the velocity contour shown in the fig.19(b).

Figures 21(a-d),22(a-d),23(a-d) shows the temperature contours for three fin, four fin & six fins configurations respectively for 6mm fin thickness. The figure 24 shows that the temperature difference between the fin tip and the root is maximum for the single fin configuration. The maximum difference obtained for the 6mm fin thickness is almost 15 degree Celsius less than the 4mm thickness; this can be due to the reduction in the effective surface area available for heat convection with respect to the area available for heat conduction. The temperature difference along the lateral direction of the fin is generally high compared to the 4mm thickness configuration. This can also be explained by the above said reason of the area available for heat convection.

Figure.27 Heat transfer vs. air velocity for 6mm and 4mm thickness 1 no. of fin

Figure.28 Heat transfer vs. air velocity for 6mm and 4mm thickness 3 no. of fin

Figure.29 Heat transfer vs. air velocity for 6mm and 4mm thickness 4 no. of fin.

Figure.30 Heat transfer vs. air velocity for 6mm and 4mm thickness 6 no. of fin

For figure 27-30, it shows the variation of the heat transfer with respect to velocity. The heat transfer was calculated directly from the fluent software. At zero velocity it is seen that the heat transfer from the 4mm

and 6mm fins are the same. When the velocity is increased it can be seen that the heat transfer is increased with due to forced convection and also due to the swirl generated between two fins which induces turbulences and hence higher heat transfer. For a larger fin thickness, the corresponding fin spacing is comparatively small. As a consequence, the generated swirled flow may mingle with the main flow and result in a higher heat transfer performance.

CONCLUSION

1. The difference of heat transfer between 4mm fins and 6mm fins are negligible same at zero velocity. So for stationary engines, fins of 4mm can be used with little difference in heat transfer, but helps in reducing the cost.
2. The heat transfer from 6mm fins is found to be the higher at high velocities. For high speed vehicles thicker fins provide better efficiency.
3. When fin thickness was increased, the reduced gap between the fins resulted in swirls being created which helped in increasing the heat transfer.
4. Large number of fins with less thickness can be preferred in high speed vehicles than thick fins with less numbers as it helps inducing greater turbulence and hence higher heat transfer.
5. The heat transfer from the outside portion of the fin is found to be less. This is our future scope of our study, as we try to increase its convection by providing slits and holes.

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