ISSN 2229-5518

EFFECT OF PIN FINS IN HEAT RECOVERY SYSTEM FOR IC ENGINES

Rone T Ray Muthalaly, Roy J, Libin V Dhar, Sooraj.H.A, Rahul.H, Midhun V C

Abstract—The increasingly worldwide problem regarding the increasing needs of energy and its shortage. The IC engine which is the major consumer of energy has an efficiency of only 30% to 40% in converting heat into work output. About 30%heat is being carried away by cooling systems and 30% to the exhaust system. Efforts are made to catch this 30 % energy of exhaust gases. If this waste heat energy is tapped and converted into usable energy, the overall efficiency of an engine can be improved. A waste heat recovery system has the ability to convert some of this waste heat into electricity. So heat recovery from the exhaust is one of the simple methods to reduce energy wastage. For any heat recovery system heat exchangers are an integral part. Here a basic attempt is made to analyze the thermal and hydrodynamic effect of pin fins in a heat a heat recovery system using CFD tool ANSYS Fluent 14.5. The results show that the use of pin fins enhanced the heat transfer with little sacrifice in pressure drop. Through this analysis we can see a satisfactory improvement in waste heat recovery from exhaust gas from an IC engine.

Index Terms— CFD, Engine Exhaust heat recovery, Heat transfer enhancement, Heat exchanger, Pin fins

1 INTRODUCTION

Large multinational car companies like BMW, Ford, Renault and Honda have demonstrated their interest in exhaust heat recovery, developing systems that make use of TEGs[1]. Moreover, the global energy demand will increase by almost 35% by 2030 compared with the 2005 level or by up to 95% without the use of energy efficient technologies [2]. A waste heat recovery system has the potential to convert some of this waste heat into electricity and consequently reduce the fuel consumption of the car by reducing the load on the car alternator.

With reference to the work reviews of existing exhaust heat exchangers and propose several internal structures: an inclined plate, a parallel plate structure, separate plate with holes, a serial plate structure and novel plate structure [4]. About 6 different designs were carried and out of which pipe structure type remain the most efficient. Apart many pin fin arrangement has been changed to encapsulate the best suitable way to trap most of the heat [5]. The overall heat transfer, friction factor and the effect of the various design parameters on the heat transfer and friction factor for the heat exchanger equipped with square cross-sectional perforated pin fins were investigated experimentally and concluded that the fin height and th shape of the fin have prior importance in heat trapping.

Thermo electric generators (TEG) are used in converting the available heat energy into electric energy. TEG when incorporated in conventional heat exchanger (as in parallel plate type or plate fin type heat exchanger) can produce energy. Compared to other waste heat recovery technologies, the use of TEGs in a waste heat recovery system has many desirable attributes such as silence, small size, scalability and durability. The most common used TEG is Bismuth telluride module. It is ought to design the heat exchanger between the exhaust manifold and the silencer. The TEG is placed above the heat exchanger such a way that it occupies maximum surface area. Ideal heat exchangers recover as much heat as possible from an engine exhaust at the cost of an acceptable pressure drop, the design of fins also place an important role in the power generation. Apart from the other fins (like plate or embedded) pin fins have the advantage of consuming less space, economical, and less back pressure. Though production cost is often the primary limitation, several other selection aspects such as temperature ranges, pressure limits, thermal performance, pressure drop, fluid flow capacity, clean-ability, maintenance, materials, etc. are important apart from all the pin fin design like conical elliptical, concave and convex shaped designs, it is easier to design and manufacture the conventional cylindrical pin fin. Apart from all shapes of pin fins the elliptical type pin fin have best heat transfer capabilities [6]. The pin fin arrangement must be varied and analyzed such that it will not cause any pressure drop and temperature analysis must be also carried out. Various design of the arrangement of fins has been taken into consideration [2]. From the whole suitable design is taken that have the maximum heat holding capacity for longer time. CFD is a powerful tool that is capable of comparing several heat exchangers under the same boundary conditions.

2 METHOD

In this work, the main motive is to conduct a single phase three dimensional analysis of the flow through a heat exchanger fitted with pin fins arrangement to determine the pressure drop and temperature distribution inside the pipe. This cannot be analysed with available analytical method so we need to depend on numerical methods like CFD. Here the CFD analysis is carried out in Fluent 14.5. The governing equation for this problem are discussed in the following section

K-EPSILON MODEL

The K-epsilon model is one of the most common turbulence models. It is a two equation model which gives a general description of turbulence by means of two transport equations (PDEs) [7]. The K-epsilon model has been shown to be useful for free-shear layer flows with relatively small pressure gradients. Similarly, for wall-bounded and internal flows, the model gives good results only in cases where mean pressure gradients are small; accuracy has been shown experimentally to be reduced for flows containing large adverse pressure gradients. One might infer then, that the K-epsilon model would be an inappropriate choice for problems such as inlets and compressors [8].

TRANSPORT EQUATIONS

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+P_k+P_b-\rho\varepsilon-Y_M+S_k$$

$$\frac{\partial}{\partial t}(\rho\epsilon)+\frac{\partial}{\partial x_j}(\rho\epsilon u_j)=\frac{\partial}{\partial x_j}[(\mu+\frac{\mu_t}{\sigma_\varepsilon})\frac{\partial\epsilon}{\partial x_j}]+\rho C_1 S_\epsilon-\rho C_2\frac{\epsilon^2}{k+\sqrt{v\epsilon}}+C_{1\varepsilon}\frac{\epsilon}{k}C_{3\epsilon}P_b+S_\epsilon$$

Where,

$$C_1=\max\left[0.43,\frac{\eta}{\eta+5}\right],\quad \eta=S\frac{k}{\epsilon},\quad S=\sqrt{2S_{ij}S_{ij}}$$

In these equations, $P_k$represents the generation of turbulance Kinetic energy due to the mean velocity gradients, calculated in same manner as standard k- epsilon model. P_b is the generation of turbulance kinetic energy due to Buoyancy calculated in same way as standard k –epsilon model.

Modeling Turbulent viscosity

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

Where

$$C_\mu=\frac{1}{A_0+A_s\frac{kU^*}{\epsilon}}$$

$$U^*\equiv\sqrt{S_{ij}S_{ij}+\Omega'_{ij}\Omega'_{ij}}\quad ;$$

$$\Omega'_{ij}=\Omega_{ij}-2\epsilon_{ijk}\omega_k\ ;$$

$$\Omega_{ij}=\bar{\Omega}_{ij}-\epsilon_{ijk}\omega_k$$

Where $\bar{\Omega}_{ij}$ is the mean rate of rotation tensor viewed in a rotating reference frame with the angular velocity ω_k. The model constants A_0 and A_s are given by :

$$A_0=4.04,\quad A_s=\sqrt{6}\cos\emptyset$$

$$\emptyset=\frac{1}{3}\cos^{-1}(\sqrt{6}W),\quad W=\frac{S_{ij}S_{jk}S_{ki}}{\tilde{S}^3},\quad S\tilde{S}=\sqrt{S_{ij}S_{ij}},$$

$$S_{ij}=\frac{1}{2}\left(\frac{\partial u_j}{\partial x_i}+\frac{\partial u_i}{\partial x_j}\right)$$

Model constants

$C_{1\epsilon}=1.44,\quad C_2=1.9,\quad \sigma_k=1.0,\quad \sigma_\epsilon=1.2$

Present analysis is based on the following assumptions:

1. Fluid flow is assumed to be in single phase.
2. The effect of gravity is not considered in the analysis.
3. The axial conduction at ends of the heat exchanger domain is neglected.
4. Heat transfer to the cold fluid flowing outside is taken in simplified manner by applying convection boundary condition
5. Inside surface of the pipe is assumed to be smooth.

3 GEOMETRY DETAILS

The figure3 shows the cut section of the heat exchanger with pin fins fitted inside the pipe. The pin fin arrangements are made in three different planes and are equally spaced. In each plane there are 15 pin fins and a total of 45 pin fins in total. Further geometry details are described in the table 3.2 and table 3.3.

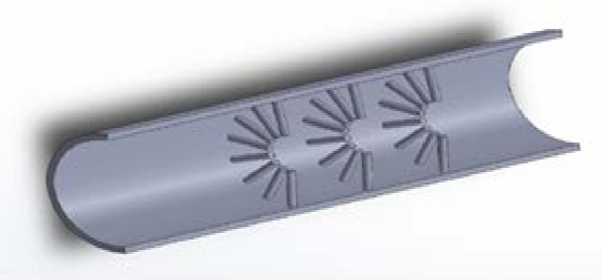


Fig.3 sectional view of heat exchanger

International Journal of Scientific & Engineering Research, Volume 7, Issue 4, April-2016
ISSN 2229-5518

Table3 1 nomenclature of the properties

Symbol	Quantity	unit
T_1	Inlet fluid temperature	K
T_2	Outlet fluid temperature	K
ρ	Density	kg/m^3
k	Conductivity	W/m K
C_p	Specific heat capacity	J/kg K
μ	Dynamic viscosity	kg/m s
p	Pressure drop	P_a
V_1	Velocity at inlet	m/s

Table3.2 cylinder configuration

Outside diameter(mm)	Thickness (mm)	Length(mm)
50	5	180

Table 3.3 fin configuration

Length (mm)	Diameter(mm)	Number in a row	Pitch(mm)
15	25	15	30

4 Mesh Details

Meshing of the computational domain was done by using ANSYS ICEM CFD in ANSYS Workbench 14.5. The number of elements in the domain are 1024749 and number of nodes are 236601 with an average orthogonal quality of 0.8627. Grid indepentance is obtained at this level.
The fluid and solid domain are discretized by sizing method [9]. In the fluid domain region closer to the inside pipe wall is finely meshed by providing inflation which is necessary to comprise the effect of thermal as well as hydrodynamic boundary layer. Figure4.1

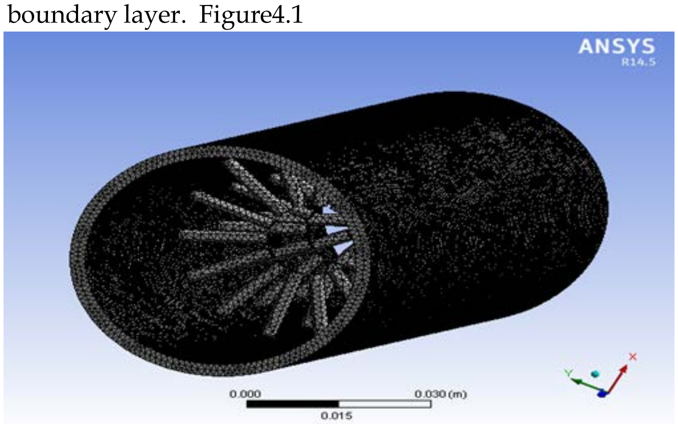


Fig4.1 Meshing in solid domain 1

5 Solution Methodology

The governing equations were solved using the FLUENT from ANSYS 14.5. The second order upwind scheme was used for the convective term discretization of momentum, energy, turbulent kinetic energy and turbulent dissipation whereas least square cell based scheme for gradients and standard for pressure equation quantities at cell faces are computed using a multidimensional linear reconstruction approach. SIMPLE algorithm is being used here for pressure and velocity coupling. The solution was assumed to be converged when the difference was limited to the third decimal point for solution of velocity terms and the sixth decimal point for energy and continuity solutions.

6 Material Properties and Boundary Conditions

Properties for the pipe and fin material along with the exhaust gas properties are shown in appendix. The Inlet side of the fluid domain is provide with velocity inlet condition with velocity of 0.92 m/s [9] with fluid temperature as 573 K while the outlet is provided with pressure outlet condition. The pipe surface is provide with convection wall temperature with convective heat transfer coefficient 15 W/m²K and fluid temperature be 300 K. this boundary condition will give an approximate effect of heat absorption done by cold fluid flowing along the pipe surface similar to the case of a double pipe heat exchanger. The end walls are providing with adiabatic wall boundary condition by assuming that the axial conduction at end of the pipe are neglected.

7 Result and Discussion

Various analyses are done on the above configuration and we obtain the following CFD models and are listed below. As the fluid moves away from the cylinder the heat get

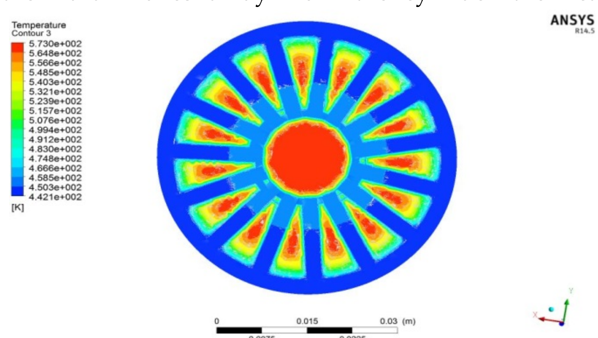


Fig7.1 Temperature distribution of plane 1

ISSN 2229-5518
absorbed by the fins at different positions placed in three rows with 15 fins. Temperature at three different planes are listed in

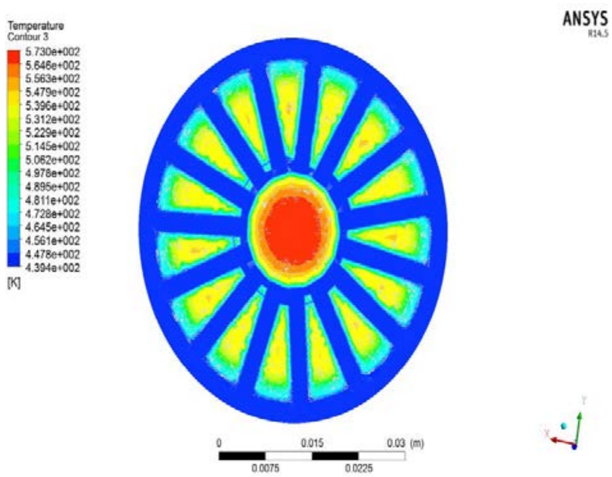


Fig 7.2 Temperature distribution at plane 2

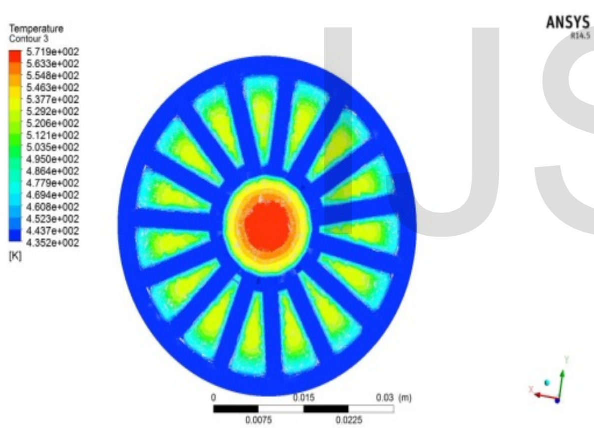


Fig 7.3 Temperature distribution at plane 3

The temperature contour of the entire cylinder is shown in figures below. The temperature intensity at different position show the absorption of heat .

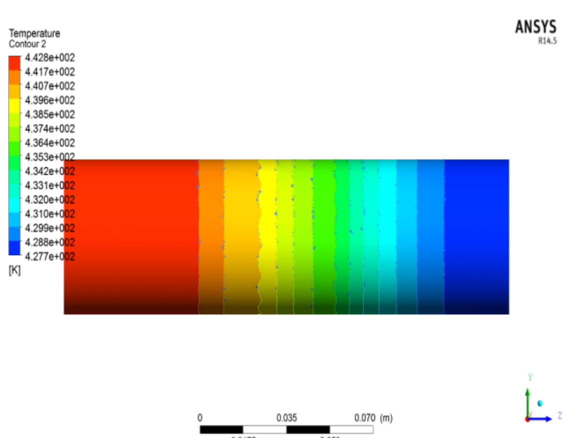
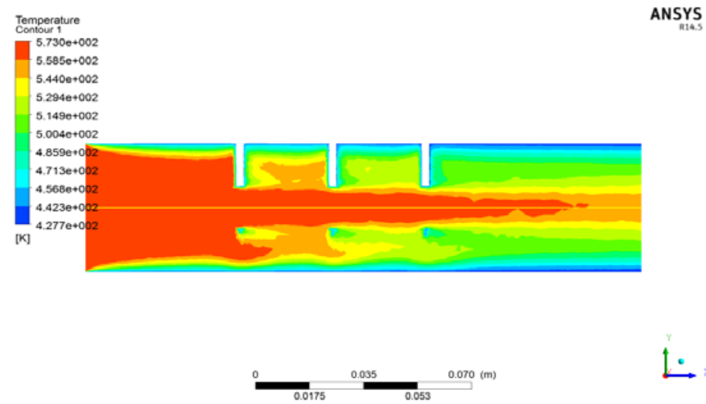


Fig 7.4 Surface Temperature contour

Fig7.5 Temperature contour along mid plane



Here it is analysed that the temperature at plane1 and plane 2 have maximum heat absorption and the pin fins arranged such that it absorbs the heat that is required for the thermo electric generator[10]. The pressure distribution along the centre line is shown in the fig 7.5. The high pressure exhaust gas coming out of the exhaust manifold need to be passed through the fins without causing much pressure drop (resulting intense vibrations and sound).The graph conveys that the current configuration will not cause much change in back pressure and can be employed.

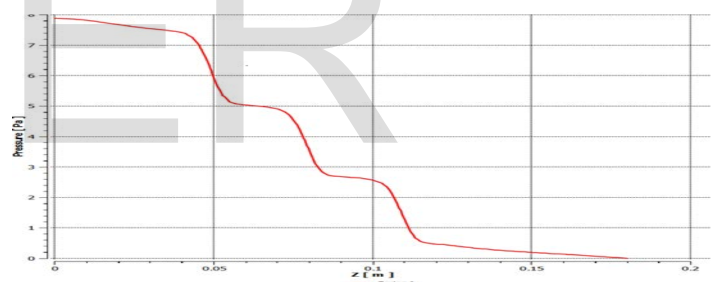


Fig 7.6 Pressure distribution along centre line

The graph conveys the relationship between average fluid temperature and the cylinder axial length. From the graph it is evident that the average fluid temperature goes on decreasing with increase in axial length.

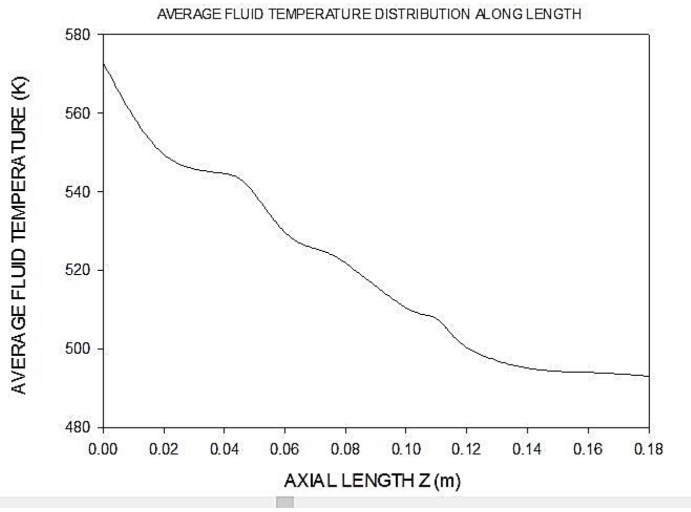


Fig 7.7 Temperature vs. axial length plot

8 CONCLUSION

As a result of this study, the following conclusions can be summarized

1. 3-Dimensional steady state analyses on a pin fin heat exchanger is been conducted.
2. The hydrodynamic and thermal characteristics in the heat exchanger are analyzed.
3. From this work we are able to obtain an idea about the pressure distribution and the temperature distribution inside a pipe fitted with pin fins
4. Heat transfer enhancement is evident from the result.
5. It is evident from the temperature contour, that using pin fins in heat exchanger, we can conclude
6. CFD models with a solid domain, liquid domain and fluid- solid interfaces were developed for the three row pin fin arrangement to stimulate the temperature and pressure field.
7. Further experiments and CFD simulations are in progress to conduct a system level optimization.
8. A completely passive and solid state exhaust heat recovery system can be developed using both TEGs and pin fins.
9. From this work, use of pin fin for heat transfer enhancement can be justified.

APPENDIX

Table for exhaust gas properties [9]

Exhaust gas properties	$A+B\times T+C\times T^2+D\times T^3$			
	A	B	C	D
ρ (kg/m³)	2.504012288761e + 00	-5.958486188418e - 03	5.578942358587e - 06	- 1.772600918994e - 09
C_p (J/kg K)	1.015580935928e + 03	-1.512248401853e - 01	4.544870294058e - 04	- 1.785063817137e - 07
μ(kg/m s)	1.325186910351e - 06	6.740061370040e - 08	-3.749043579926e - 11	1.110074961972e - 14
k(W/m K)	-3.182421851331e - 03	1.185847825677e - 04	-7.706004236629e - 08	2.939653967062e - 11

Table for properties of Aluminum (pipe material)

ρ (kg/m^3)	2719
C_P (J/kg K)	871
k (W/ m K)	202.4

REFERENCES

[1] Ramesh KUMAR, Ankit Sonthalia, and Rahul Goel, "Experimental Study On Waste Heat Recovery From An Internal Combustion Engine Using Thermoelectric Technology "

[2] Mohak Gupta, "Review on Heat Recovery Unit with Thermoelectric Generators"

[3] Shengqiang Bai, Hongliang Lu, Ting Wu, "Numerical and experimental analysis for exhaust heat exchangers"

[4] Tongchai Wang, Wailing Luan, Wei Wang, Shang Thu" Waste heat recovery through plate heat exchanger based thermo electric generator"

[5] Amol B. Dhumne, Hemant S. Farkade, "Heat Transfer Analysis of Cylindrical Perforated Fins in Staggered Arrangement"

[6] O˘guz Uz, Cengiz Camci, " Elliptical pin fins as an alternative to circular pin fins for gas turbine blade cooling applications"

[7] Bardina, J.E., Huang, P.G., Coakley, T.J. (1997), "Turbulence Modeling Validation, Testing, and Development ", NASA Technical Memorandum 110446.

[8] Jones, W. P., and Launder, B. E. (1972), " The Prediction of Laminarization with a Two-Equation Model of Turbulence" , International Journal of Heat and Mass Transfer, vol. 15, 1972, pp. 301-314.

[9] M. Hatami n, D.D.Ganji,M.Gorji-Bandpy, "Numerical study of finned type heat exchangers for ICEs exhaust waste heat recovery"

[10] A. Akbarzadeha, M. Mochizukib, R. Singhb, "A review of car waste heat recovery systems utilising thermoelectric generators and heat pipes"