ISSN 2250-3153

Computational Fluid Dynamics Study of IC Engine - Energy Recovery System

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Abstract- It is confirmed that in internal combustion engines (ICEs), more than 30–40% of fuel energy wastes from the exhaust and just 12–25% of the fuel energy converts to useful work. On the other hand, statistics show that producing amounts of the internal combustion engines growth very fast and the concern of increasing the harmful greenhouse gases (GHG) will be appeared. So, researchers are motivated to search alternative solutions by using conservation ways, recover the heat from the waste sources in engines which not only reduces the demand of fossil fuels, but also reduce the GHG and help in energy saving. On the other hand, because exhaust gas heat recovery using heat exchanger may make a pressure drop and effects on the engine performance hence its design is very important and crucial. To select an appropriate heat exchanger design limitations for each heat exchanger type firstly should be considered. 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, cleanability, maintenance, materials, etc. are important. One of the most effective methods to increase heat transfer is using the fins which are widely used by the researchers across the heat recovery designers. Some of the special HEXs designs to recover the exhaust heat are introduced. This paper aims to model the heat transfer through exhaust gases to a cold fluid with modeling the fins with suitable viscous model to calculate the heat recovery amount. Two cases of previous experimental work are selected and numerical results are compared with experimental outcomes. Also, effect of fins size and engine load and speed on heat recovery is examined graphically.

Index Terms- CFD, Heat Exchangers, Heat Recovery, Numerical Simulation

I. Introduction

Recently, Ghazikhani et al. estimated in an experimental work that brake specific fuel consumption (BSFC) of the diesel engine could be improved approximately 10% in different load and speeds of an OM314 diesel engine by using the recovered exergy from a simple double pipe heat exchanger in exhaust. Pandiyarajan et al. designed a finned-tube heat exchanger and they used a thermal energy storage using cylindrical phase change material (PCM) capsules and found that nearly 10–15% of fuel power is stored as heat in the combined storage system in different loads. In another experimental work, Lee and Bae designed a little heat exchanger with fins in the exhaust by design of experiment (DOE) technique. They reported that fins should be in the exhaust gases passage for more heat transfer and designed 18 cases in different fins numbers and thicknesses and found the most effective cases. Zhang et al. modeled a finned tube evaporator heat exchanger for an ORC cycle. They concluded that waste-heat recovery efficiency is between 60% and 70% for most of the engine's operating region also they mentioned that heat transfer area for a finned tube evaporator should be selected carefully based on the engine's most typical operating region. Recently, Hossain and Bari used a new HEX for heat recovery in a diesel engine experimentally and numerically. They applied SST k–ω for their modeling and they optimized the working fluid pressure and the orientation of heat exchangers and found the additional power increased from 16% to 23.7%.

II. METHODOLOGY

Two different cases of heat exchanger which previously are used by researchers for exhaust waste heat recovery are selected which are shown in Fig. 1.

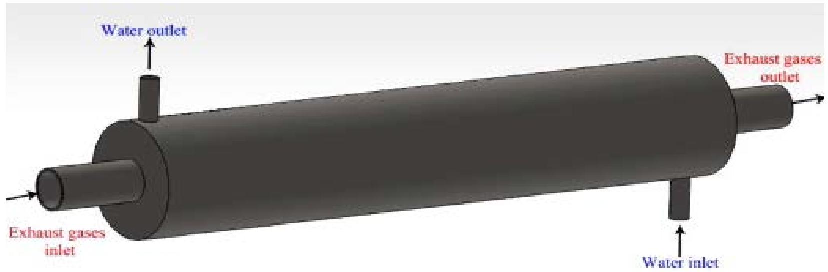

Fig. 1 Double pipe Heat exchanger

Case 1 is a simple double pipe heat exchanger with water coolant which is installed in the exhaust of an OM314 diesel engine by Ghazikhani et al. The length of this HEX is 70 cm with 12 cm inlet and 14 cm outlet diameters. In this case, water mass flow rates are in 10–100 g/s and exhaust gases are in 30–60 g/s ranges in different engine's operating condition. Also, temperature range for water is 10–25 and for exhaust gases is 100–220 degree of centigrade in different engine loads and speeds. Case 2 is an optimized finned heat exchanger with special design which a mixture of 50% water and 50% ethylene glycol circulate around the tube as coolant used by Lee and Bae to recover the heat from a gasoline engine.

Fig.2 Schematic of their geometries

Both these studies investigated the amount of waste heat recovery in different engine loads and speed. This study aims to model both these HEXs numerically using commercial CFD code, FLUENT software which a schematic of their geometries are shown in Fig. 2.

The mesh generation software Gambit, combined with the FLUENT software which depends on the finite volume methods as the described by Patankar, helped to define the boundary layers and zone types. The mesh was then exported to FLUENT. Full information about each case is described in the results and discussion section. Every simulation case takes approximately 9 h to get converged solutions.

The numerical simulation was performed with a three dimensional steady-state turbulent flow system. To solve the problem, governing equations for the flow and conjugate heat transfer were modified according to the conditions of the simulation setup. As the problem was assumed to be steady, the time dependent parameters were dropped from the equations.

In this paper, three viscous models are examined. Renormalization-group (RNG) k–ε model, Shear-stress transport (SST) k–ω model and Reynolds stress model (RSM). These three models were selected because previous studies which are reviewed by previous authors, introduced these models as efficient models. For RNG k–ε model thermal effect is considered in the enhanced wall treatment panel.

III. RESULTS AND DISCUSSION

As mentioned before, two cases shown in Fig. 2 are modeled numerically in different load and speeds, furthermore four different geometries are considered for case 2 to show the effect of fin number and sizes. As an approximation, the properties of air can be used for diesel exhaust gas calculations which the error associated with neglecting the combustion products is usually no more than about 2%. Due to high temperature in exhaust, temperature dependent properties are considered for exhaust gases which Fig. 3 shows those variations.

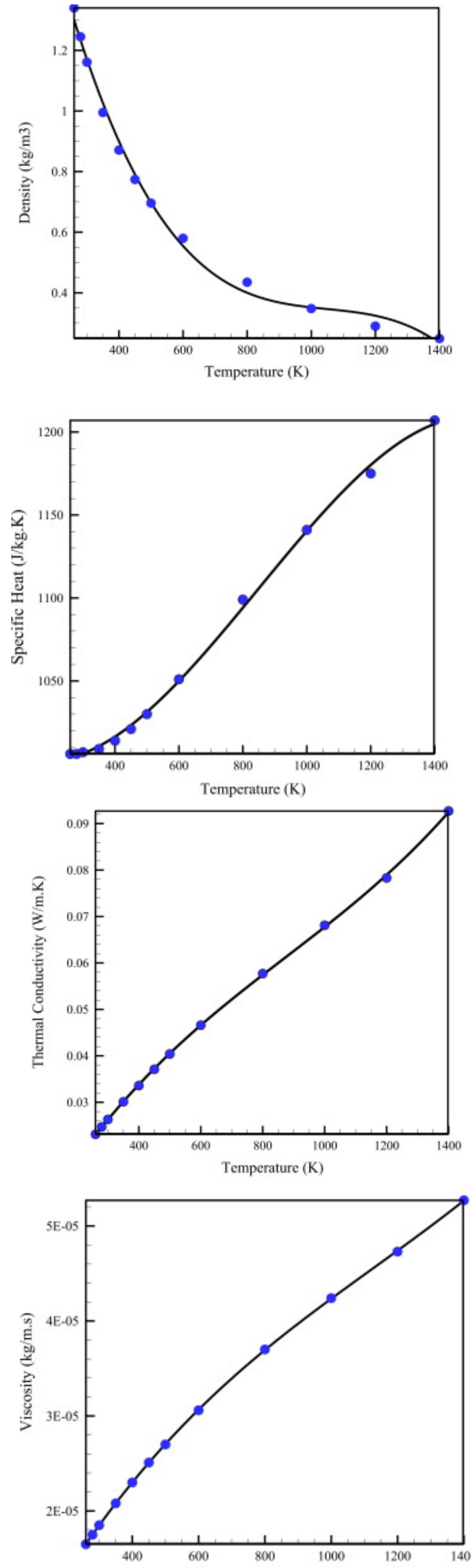

Fig.3 Temperature vs Temperature dependent properties graphs

For each property, a fourth order polynomial is plotted which its equation and coefficients are shown in Table 1.

Table 1. Temperature-dependent properties of exhaust gases				
Exhaust gas properties	A	B	C	D
ρ(kg/m3)	2.50e+00	−5.95e−03	5.57e−06	−1.77e−09
Cp(J/kg K)	1.02e+03	−1.51e−01	4.54e−04	−1.78e−07
μ(kg/m s)	1.32e−06	6.74e−08	−3.74e−11	1.11e−14
k(W/m K)	−3.18e−03	1.18e−04	−7.70e−08	2.93e−11

Solid phases which contain tubes and fins are considered to be carbon steel which its thermal properties and cold fluid properties are shown in Table 2.

Table 2. Thermal properties of coolants and solid phases.				
	ρ(kg/m3)	Cp(J/kg K)	μ(kg/m s)	k(W/m K)
Water	998.2	4182	0.001003	0.6
Ethylene glycol	1111.4	2415	1.61e−02	0.252
Water–ethylene glycol (50–50)	1050.44	3499	0.8e−03 (in 80 °C)	0.4108
Carbon steel	7858	486	–	52

First, to model the case 1 with water coolant, three mesh numbers are constructed to show the mesh independency. Table 3 shows these mesh number values while volume 1 represent to gases pass, volume 2 is solid phase (walls or fins) and volume 3 is for coolant area.

Mesh number	Volume1/cooper	Volume2/cooper	Volume3/tetrahedral	Sum of meshes
Mesh number 1	6<comma>182	17<comma>700	22<comma>780	46<comma>662
Mesh number 2	8<comma>480	25<comma>200	31<comma>593	65<comma>273
Mesh number 3	16<comma>530	41<comma>400	51<comma>442	109<comma>32

As seen in Fig. 4 which is plotted for temperature of central line and velocity of exhaust outlet for T¼60 N.m and 1600 rpm, solution is approximately independent to mesh numbers.

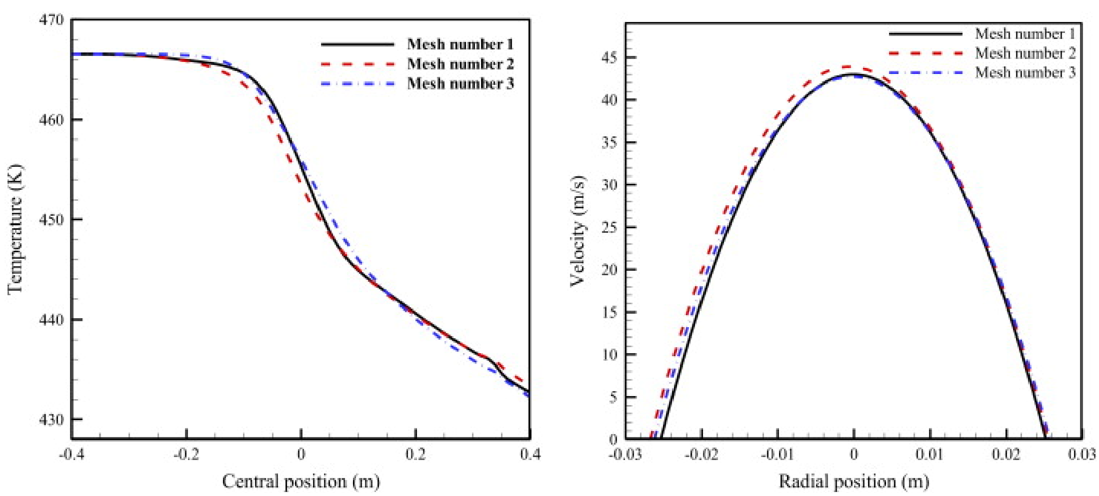

Fig.4 Graphs for different mesh numbers at different positions

To find the best viscous model among those of described in the previous section, problem is solved in the condition of different engine loads when engine speed is 1600 rpm. Outcomes for exhaust and water outlet temperatures are depicted in Fig. 5_ which confirms that RNG k–ε and SST k–ω has an acceptable accuracy compared to experimental outcomes.

Table 4. Comparison between experimental and SST k–ω model results

T(N m)	T_{out} exhaust gases (K)				T_{out} water (K)			
Experimental	Minimum	Maximum	Average	Experimental	Maximum	Minimum	Average	Experimental
20	361.95	362.65	375.7	370.93	290.65	289.12	287.2	288.06
60	388.2	392.25	405.1	399.02	291.15	290.14	287.5	288.67
100	414.65	419.90	436.6	428.52	293.15	292.87	289.6	291.05

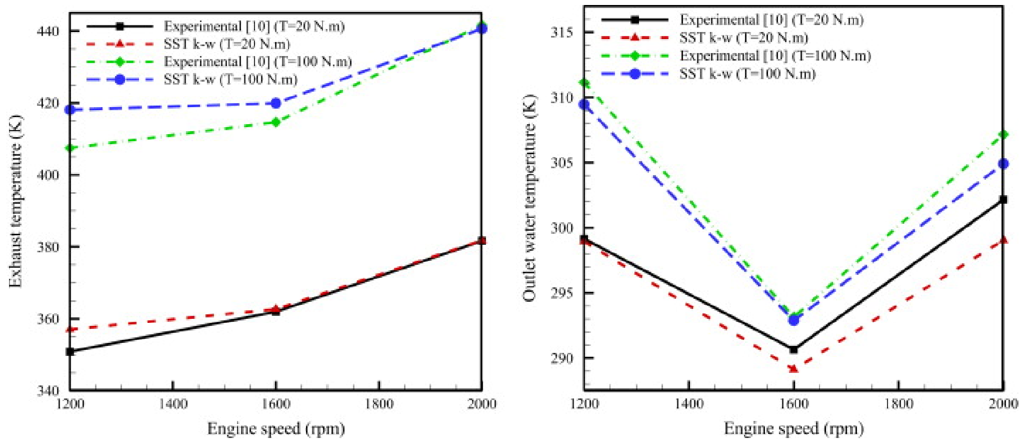

Fig.5 Graph for Engine speed vs Exhaust temp and water outlet temp

These figure confirm that RSM is not a suitable viscous model for these kind of the problems and although SST k–ω values is more close to experimental values, but its convergence is more difficult compared to RNG k–ε during the solution process. Contours of the Fig. 6 shows the temperatures in three engine loads in 1600 rpm. It is obvious that by increasing the engine torque, water outlet temperature increased due to higher exhaust temperature.

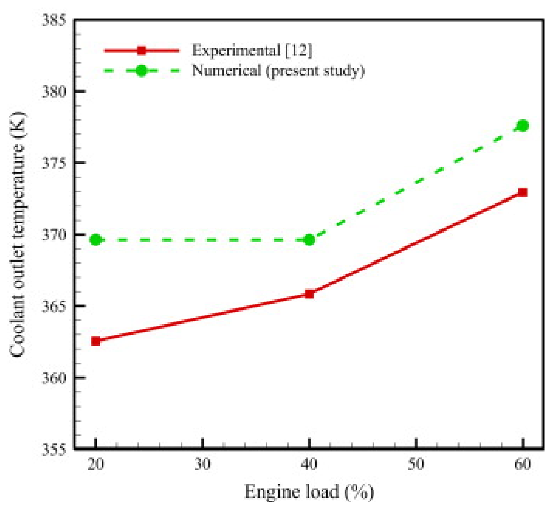

Fig.6 Temperature vs engine load graph

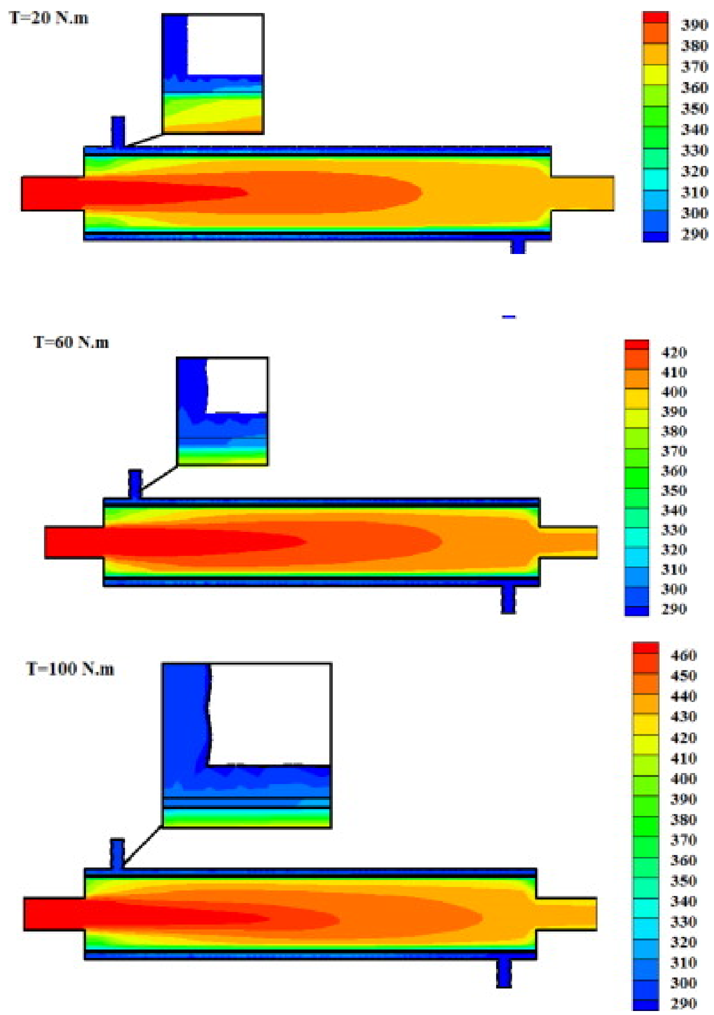

Fig.7 Temperature distribution contours for different torque values

In the zoomed areas of these contours, temperature distributions in the solid phase (pipe walls) and water are completely evident which is due to convection and conduction modeling. Maybe, one of the reasons of difference between outlet temperatures of experimental and numerical modeling is the location of thermocouples in the experiment. Because thermocouples measure just temperature of the one point, but in the numerical the average temperature is calculated while in each face, a maximum and minimum temperature occurs and thermocouples can sense each of them.

Table 4.Comparison between experimental and SST k–ω model results								
T(Nm)	Toutexhaust gases (K)				Toutwater (K)			
Experimental	Minimum	Maximum	Average	Experimental	Maximum	Minimum	Average	Experimental
20	361.95	362.65	375.71	370.93	290.65	289.12	287.21	288.06
60	388.2	392.25	405.19	399.02	291.15	290.14	287.53	288.67
100	414.65	419.90	436.60	428.52	293.15	292.87	289.69	291.05

Table 4 shows this matter and by choosing the nearest value point to experiment Fig. 7 is plotted which better results with experiments are observed.

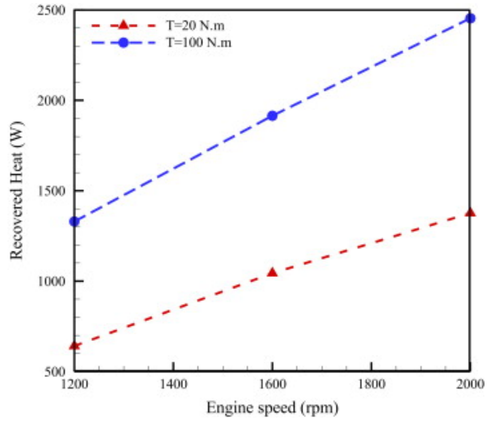

Fig.8 Recovered heat vs Engine speed Graph

Fig. 8 demonstrates the heat recovered amount in different engine load and speeds which its maximum is approximately 1400 W and occurs in 100 N m and 2000 rpm.

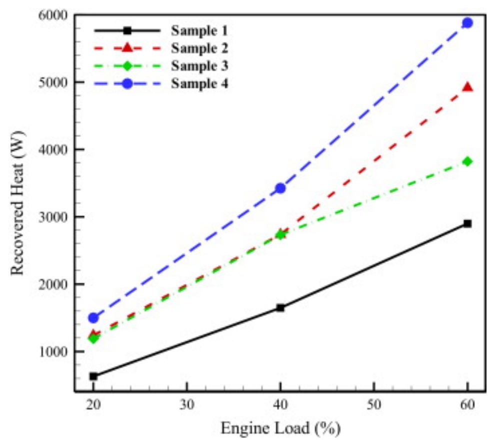

Fig.9 Recovered heat vs Engine load for different samples

In the second modeling (case 2), four samples from 18 experiment samples designed in is modeled. Samples 1, 8, 10 and 18 the data is from the published research and selected due to have the maximum and minimum efficiencies.

Table.5 Geometries of samples

Sample No.	Collant Flow direction	Fin Length (mm)	Fin Thickness (mm)	No. of Fins	Inner wall Thickness (mm)	Length of heat exchanger (mm)
Sample 1(1in[12])	Clockwise	10	1.5	10	2	18
Sample 2(8in[12])	Clockwise	20	2	20	3	18
Sample 3(10 in[12])	Counter clockwise	10	1.5	20	4	20
Sample 4(18 in[12])	Counter clockwise	20	2.5	15	2	20

Table 5 presents the complete geometries of these four samples. A mixture of 50% water and 50% ethylene glycol is considered for coolant which in samples 1 and 2 circulate clockwise while in samples 3 and 4 circulates counter clockwise. Type and number of generated meshes for these samples are presented through Table 6.

Table 6. Mesh numbers and construction for four samples of case 2 in Table 4				
Samples	Volume 1/cooper	Volume 2/tetrahedral	Volume 3/tetrahedral	Sum of meshes
Sample 1	84<comma>042	98<comma>639	95<comma>576	278<comma>257
Sample 2	76<comma>824	194<comma>326	169<comma>232	440<comma>382
Sample 3	112<comma>740	219<comma>038	154<comma>329	486<comma>107
Sample 4	82<comma>840	158<comma>682	180<comma>849	422<comma>371

Contours of Fig. 10 which are plotted for 20% load and 4500 rpm shows the effect of temperature distribution in the heat exchanger and zoomed areas clearly show the conduction heat transfer through the fins.

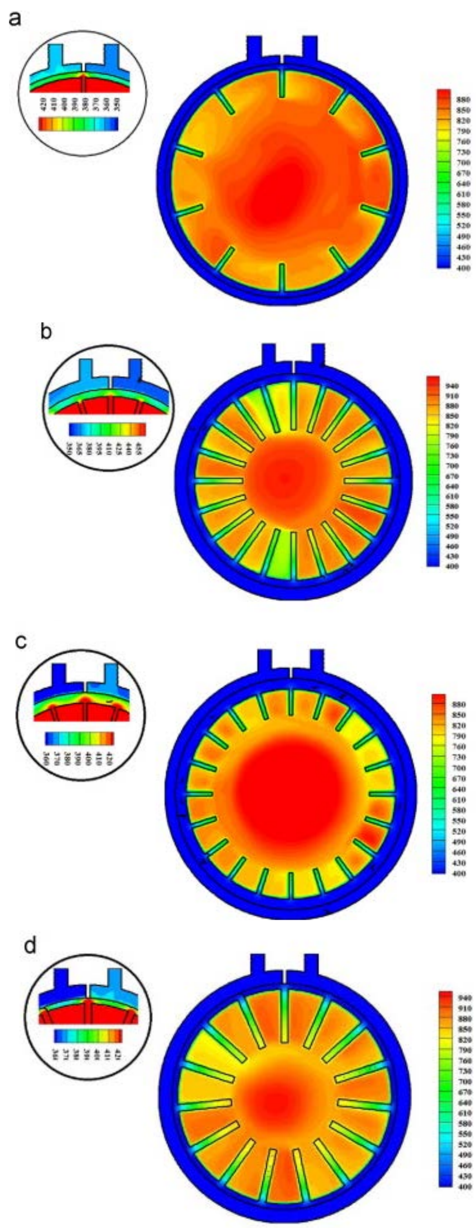

Fig.10 temperature distribution in heat exchanger

Comparisons of outlet cold fluid temperature for all samples with experimental results are shown in Fig. 11 (left) and for sample 4 (right) in different engine loads.

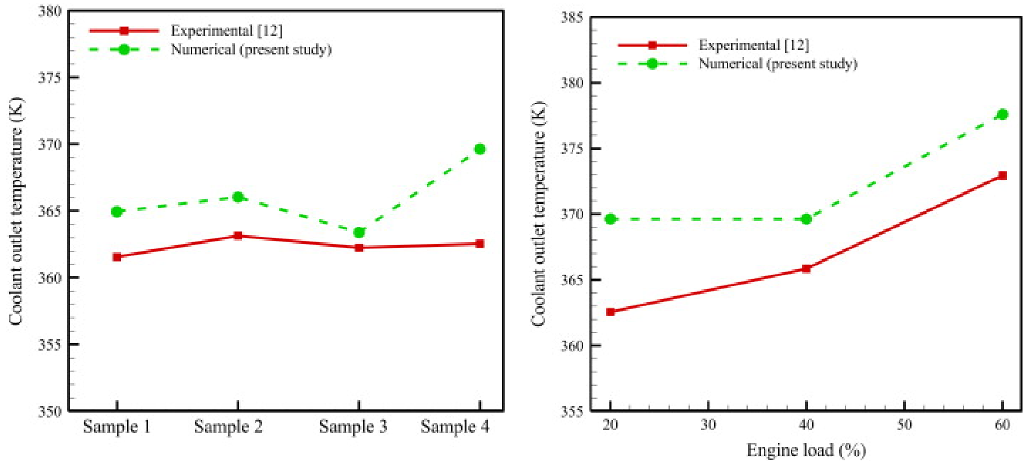

Fig.11 Comparison of experimental and numerical values for load and coolant temperature

As revealed Lee and Bae, it seems that samples 1 and 3 have the smallest effectiveness and samples 2 and 4 have the largest cooling effectiveness due to water outlet temperature. Because sample 4 has the maximum cooling performance, its temperature contours in different loads are shown in Fig. 12 in front and side view of the central plane.

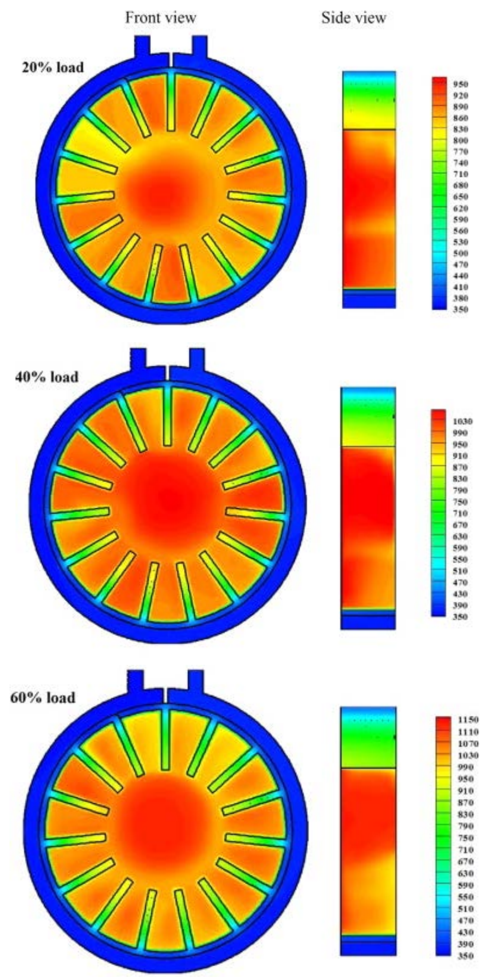

Fig.12 temperature contours for different loads

Finally by calculating the recovered heat amount, Fig. 13 is depicted for all four samples in different engine loads.

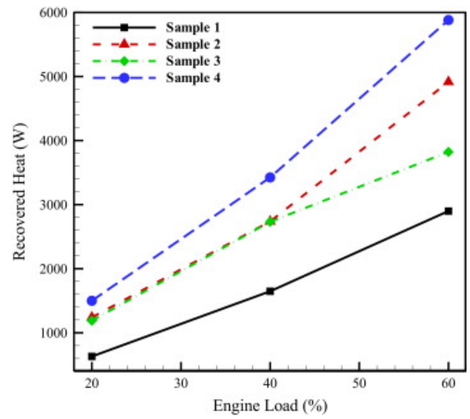

Fig.13 Recovered heat vs Engine load graph

As seen, maximum heat recovery is approximately 5900 W which occurs when engine speed is 4500 rpm and 100% load in sample 4 due to higher fin length, thickness and numbers.

IV. CONCLUSION

In this paper IC engines exhaust waste heat is recovered by using the double pipe heat exchange numerically. Heat recovery from exhaust gases using double pipe heat exchanger were modeled successfully and the transferred heat to cold fluid is calculated as the recovered heat. Results show that SST k–ω and RNG k–ε are suitable viscous models, but RSM has not good results compared to experimental outcomes. Also, graphs and contours reveal that recovered heat can be improved by increasing the fin numbers and length where maximum heat recovery occurs in high engine load and speeds.

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