Shell and Tube Type Heat Exchanger for Waste Heat Recovery from Diesel Engine Exhaust-Case Study

Dipak S. Patil[1], Dr. R. R. Arakerimath[2], Dr. P. V. Walke[3], Dr. R. S. Shelke[3]

[1] *Research Scholar in Department of Mechanical Engineering,G. H. Raisoni College of Engineering, Nagpure, Maharashtra, India*

[2] *Department of Mechanical Engineering, G. H. Raisoni College of Engineering and Management, Pune, Maharashtra, India*

[3] *Department of Mechanical Engineering G. H. Raisoni College of Engineering, Nagpure, Maharashtra, India*

Abstract- Automotive engines reject a considerable amount of thermal energy to the surroundings. About 30-40 % energy is released through the exhaust. Significant improvement in fuel economy could be attained by recovering the exhaust heat. One of the most important issues is to develop an efficient heat exchange system which provides optimal recovery of heat from engine exhaust. Waste heat recovery from engine exhaust can be achieved using various types of heat exchangers like recuperators, shell & tube, spiral, plate type heat exchangers. Shell & tube heat exchangers have the ability to transfer large amounts of heat in relatively low cost, serviceable designs and reliability. In the present work, 1-2 shell & tube type heat exchanger has been designed, fabricated and tested for its performance. The waste heat has been recovered & utilized for heating water. To assess the performance of the heat exchanger, prototype has been designed, fabricated and tested for its performance. The design of the heat exchanger was done by carrying out simulation process using design program developed for shell & tube heat exchanger. The effect of waste heat recovery on engine performance has also been studied by conducting tests on engine with and without heat exchanger. The results show that the deviation between predicted and experimental values of outlet temperatures of water and gas are within ±8% and ±15% respectively. Also, effect of heat exchanger on engine output is within 3%.

Keywords – Waste heat recovery, 1-2 shell and tube type heat exchanger, Engine, overall heat transfer coefficient, Diesel exhaust.

I. INTRODUCTION

Energy is an important entity for the economic development of any country. The rapid industrial and economic growth in India and China where one third population of the world is present has increased the need for energy rapidly in the recent years. Considering the need for environmental protection and also in the context of uncertainty over future energy supplies, attention is concentrated on the utilization of sustainable energy sources and the energy conservation methodologies. High capacity diesel engines are one of the most widely used power generation units. Nearly two-third of input energy to the engine is wasted through exhaust gas and cooling water. On the other side, primary energy is spent for low temperature applications like hot water generators, generation of low pressure steam for process heating etc. It is imperative that serious and concrete efforts should be launched for conserving the energy through waste heat recovery techniques. Such a waste heat recovery would ultimately reduce the overall primary energy requirement and also the impact on global warming.

Waste heat is generated in a process by the way of fuel combustion or chemical reaction, and then released in the environment. IC engines also release large quantity of hot flue gases to the atmosphere. If some of this waste heat could be recovered, considerable amount of primary fuel could be saved. The energy loss through exhaust gases cannot be fully recovered. However, much of the heat could be recovered by selecting the appropriate type of heat exchanger and its efficient design [1]. Depending on the temperature level of exhaust stream and the proposed application, different heat exchange devices, heat pipes and combustion equipment's can be employed to facilitate the use of the recovered heat. The shell and tube heat exchanger is the most widely used type of industrial heat transfer equipment. Initially, only plain tubes were used in shell & tube heat exchangers. Heat exchangers are commonly used in a wide range of applications like heating, air-conditioning, phase change. Pandiyarajan et al. [1] have investigated shell and finned tube heat exchanger integrated with an IC engine setup to extract heat from the exhaust gas and a thermal energy storage tank used to store the excess energy available. The performance parameters pertaining to the heat exchanger and the storage tank such as amount of heat recovered, heat lost, charging rate, charging efficiency and percentage energy saved are evaluated. S. Mavridou et al. [2] have studied comparative design with conventional and state of the art heat transfer enhancements. Two different heat exchanger configurations with different types of heat transfer surfaces have been examined regarding their weight, volume, induced pressure drop and their effect on the vehicle (freight weight, etc.). The shell and tube configuration was

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examined first consisting initially of smooth, and then finned and dimpled circular tubes in order to increase the heat transfer area. The plate and fin configuration was also studied, where plain fins or metal foam were examined as a means of heat transfer.

Pertti Kauranen et al. [3] have designed exhaust gas heat recovery system and latent heat accumulator. The true benefits of the system for fuel economy and emission reduction should be evaluated by more exact measurements and calculations of the engine and exhaust gas heat balance as well as by measuring the total fuel economy and emissions of the vehicle including the additional heater and by considering the additional weight of the latent heat accumulator and exhaust gas recovery. Paola Bombarda et al. [4] have compared kalian and organic rankine cycle for heat recovery applications in the frame of medium temperatures (typical for Diesel engine discharge) and low power levels were investigated, so as to compare their thermodynamic performance. A preliminary optimization procedure for the most important parameters was performed for both cycles prior to cycle comparison. On the basis of the calculations performed and with the assumption considered, it emerges that the kalian cycle requires a very high maximum pressure in order to obtain high thermodynamic performance. M. Talbi et al. [5] used exhaust gases for combined heat and power applications and studied theoretical performance of four different configurations of a turbocharger diesel engine and absorption refrigeration unit combination when operating in a high ambient day temperature of 35oC. Dr. Madhukar S. Tandale et al. [6] have developed an analytical model for carrying out design simulations of the Pancake type heat exchanger. For the validation of the analytical model, pancake type heat exchanger was fabricated in the laboratory with two sets of four pancakes. Total of eight pancakes were used in the heat exchanger. Stainless steel (SS-316 seamless) was used for tubes and mild steel for shell. D. Del Col et al. [7] conducted experimental test on prototype of shell & tube type evaporator with R22. The data has been compared against a computational procedure for shell and tube type heat exchangers. Both the effect of inlet quality and superheat are well predicted by the procedure. K. C. Leong et al. [8] developed software for the thermal and hydraulic design of shell and tube heat exchangers with flow induced vibration checks has been developed in a Windows-based Delphi programming environment. Its user-friendly input format and excellent color graphics features make it an excellent tool for the teaching, learning and preliminary design of shell and tube heat exchangers. G.T. Polley et al. [9] have done work in fouling factors of heat exchanger. It also dominates the life time cost of pre-heat trains. Yusuf Ali Kara et al. [10] made computer-based design model for preliminary design of shell-and tube heat exchangers with single-phase fluid flow both on shell and tube side. The program covers segmentally baffled U-tube, and fixed tube sheet heat exchangers one-pass and two-pass for tube-side flow. The program determines the overall dimensions of the shell, the tube bundle, and optimum heat transfer surface area required to meet the specified heat transfer duty by calculating minimum or allowable shell-side pressure drop. The transfer of heat to and from process fluids is an essential part of most of the chemical Processes, I.C. engine etc. Therefore, Heat Exchangers (HEs) are used extensively and regularly in process and allied industries and are very important during design and operation. The most commonly used type of heat exchanger is the shell-and-tube heat exchanger [11].

It is essential for the designer to have a good working knowledge of the mechanical features of STHEs and how they influence thermal design. The principal components of an STHE are: shell, shell cover, tubes, channel, channel cover, tube-sheet, baffles, and nozzles. Other components include tie-rods and spacers; pass partition plates, impingement plate, longitudinal baffle, sealing strips, supports, and foundation [12]. Diesel exhaust is a complex mixture of gases and fine particles that are emitted by internal combustion engines using diesel oil as fuel. The gaseous component of diesel exhaust is similar to the combustion products of other fuels. Although the adverse effects of diesel emissions are due both to the gaseous and particulate components, the toxicity of diesel exhaust is often expressed in relation to its particulate component. Several agencies have classified diesel exhaust as a carcinogen. Complete and incomplete combustion of fuel in diesel engines results in a complex mixture of gases and particles composed of hundreds of organic and inorganic compounds. The physical and chemical characteristics of diesel exhaust are dependent on many factors such as the composition of the fuel, the characteristics of the engine and the conditions under which the diesel is burned. From light diesel vehicle, typical diesel exhaust gas composition by weight is: Carbon dioxide (CO_2) - 7.1%, Water Vapour (H_2O) - 2.6%, Oxygen (O_2) - 15.0%, Nitrogen (N_2) - 75.2%, Carbon monoxide (CO) - 0.03%, Hydrocarbons (C_3H_6) - 0.0007%, Nitrogen Oxide (Nitrogen dioxide NO_2) - 0.03%, Hydrogen (H_2) - 0.002% etc. [13].

In the present work, heat recovery system consisting of a 1-2 shell & tube heat exchanger has been designed, fabricated, and tested for waste heat recovery from diesel engine exhaust. Tap water was used as heat transfer fluid (HTF) on shell side to extract heat from exhaust gas. Thermal performance of heat recovery heat exchanger has been studied under various engine operating conditions.

II. PROBLEM FORMULATION

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The problem is to design heat exchange system for waste heat recovery from a diesel exhaust gas and use this heat to water heating. The recovered heat can be utilized for water distillation, control the temperature of vehicle cabinet, etc. The data regarding exhaust gas composition, temperature, pressure and mass flow rate of gas, was taken by conducting tests on diesel engine at various loads in the laboratory. Using this data, heat potential available in the exhaust gas is calculated using energy balance equation.

$$\dot{m}_w\, C_{pw}\left(T_{wo} - T_{wi}\right) = \dot{m}_g\, C_{pg}\left(T_{gi} - T_{go}\right) \quad (1)$$

Heat loss from wall of shell of the heat exchanger is within 5-10 %.

$$Q_w = 0.9\, Q_a \quad (2)$$

LMTD is calculated by using following equation. The correction factor was calculated for 1-2 shell & tube type of heat exchanger from chart.

$$\Delta Tm = ((T_{gi} - T_{wo}) - (T_{go} - T_{wi}))/(ln\,((T_{gi} - T_{wo}))/((T_{go} - T_{wi}))) \ x\ F \quad (3)$$

Total surface area of heat exchanger is calculated. Assume overall heat transfer coefficient (Uo ass)

$$A = Q/(U_o \,.\, \Delta T_m) \quad (4)$$

Total Length of tube

$$L = A_{/}\,\pi d_i \quad (5)$$

Mass flow rate of diesel exhaust is calculated from energy balance equation

$$m_g = \frac{\dot{m}_w\, c_{pw}\ (T_{wo} - T_{wi})}{c_{pg}\ (T_{gi} - T_{go})} \quad (6)$$

Inner diameter of tube

di = do – 2 t (7)

Further, gas side film heat transfer coefficient is calculated using following equations.

Area available for gas flow

$$A_{fl\cdot g} = \frac{\pi}{4}\, d_i^{\,2} . N_T \quad (8)$$

Velocity of diesel exhaust

$$V_a = \dot{m}_a/(\rho_a\, Afl_a) \quad (9)$$

Reynolds number of diesel exhaust

$$Re_g = \frac{\rho_g V_g\, d_i}{\mu_a} \quad (10)$$

Using following correlations for Nusselt number for diesel exhaust flow [14]

If Reynolds number (Reg) < 2300

For developing flow

$$Nu_g = 1.86\left[\left(\frac{D}{L}\right) Re_g\ X\ Pr_g\right]^{1/3} x \left[\frac{\mu_g}{\mu_s}\right]^{0.14} \quad (11)$$

For developed flow

$$Nu_g = 3.66 + \frac{0.0668\left(\frac{d_i}{L}\right) Re_g \,.\, Pr_g}{1 + 0.04\left(\frac{D}{L}\right) Re_g \,.\, Pr_g\ )^{2/3}} \quad (12)$$

If 2300 < Reg > 5 x 106,

The Gnielinski equation

$$Nu_g = \frac{\left(\frac{f}{8}\right)[(Re_g - 1000)Pr_g]}{1 + 12.7\left(\frac{f}{8}\right)^{1/2}[(Pr_g^{\frac{2}{3}}) - 1]}(1 + \frac{D}{L})^{2/3}$$

(13)

$$f = (0.79 \ln Re_g - 1.64)^{-2}$$

Heat transfer coefficient of gas side

$$h_g = \frac{Nu_g . K_g}{d_i} \tag{14}$$

The water side heat transfer coefficient is calculated using following equations.

The shell side equivalent diameter (hydraulic diameter), for an equilateral triangular pitch arrangement

$$D_e = \frac{4[\frac{P_T^2 . \sqrt{3}}{4} - \frac{\pi d_o^2}{8}]}{\frac{\pi d_o}{2}} \tag{15}$$

For square pitch arrangement

$$D_e = \frac{4[\frac{Ps^2}{4} - \frac{\pi d_o^2}{4}]}{\pi d_o^2} \tag{16}$$

Clearance between adjacent tubes

$$C = P_T - d_o \tag{17}$$

The tube bundle cross flow area, at the center of the shell

$$A_s = (D_i . C . B)/P_T \tag{18}$$

Baffle spacing

$$B = \frac{1}{5} D_i \tag{19}$$

The shell side mass velocity

$$G_s = \frac{\dot{m}_w}{A} \tag{20}$$

Reynolds number of water

$$Re_w = \frac{G_s D_e}{A} \tag{21}$$

Using same correlations for Nusselt number for water side which are used for gas side

Heat transfer coefficient of water side

$$h_w = \frac{Nu_w . \kappa_w}{D_o} \tag{22}$$

Overall heat transfer coefficient without fouling

$$U_{o\ th} = \frac{1}{\frac{1}{h_o}\frac{d_o}{d_i} + \frac{d_o}{2\kappa_m} \ln\frac{d_o}{d_i} + \frac{1}{h_w}} \tag{23}$$

Overall heat transfer coefficient with fouling

$$U_{o\ th} = \frac{1}{\frac{1}{h_o}\frac{d_o}{d_i} + \frac{d_o}{2\kappa_m} \ln\frac{d_o}{d_i} + \frac{1}{h_w} + Rf_o + (\frac{d_o}{d_i} \times Rf_i)} \tag{24}$$

Gas side pressure drop without fouling

$$\Delta P = \left(\frac{4 f LV_g^2 \rho_g}{2 d_i}\right) \tag{25}$$

Gas side pressure drop without fouling for two pass tube

$$\Delta P = \left(\frac{4 Np f LV_g^2 \rho_g}{2 d_i}\right) + \left(\frac{4 Np V_g^2 \rho_g}{2}\right) \tag{26}$$

Experimental overall heat transfer coefficient, Uo

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exp.

$$Q_{avg} = Q_{exp} = \left(\frac{Q_w + Q_g}{2}\right) \quad (27)$$

$$Q_{exp} = U_{exp} . A . \Delta T_m \quad (28)$$

$$U_{o\ exp} = \left(\frac{Q_{exp}}{A . \Delta T_m}\right) \quad (29)$$

Deviation between theoretical and experimental overall heat transfer coefficient

$$\%\ Deviation = \frac{U_{o\ exp} - U_{o\ th}}{U_{o\ exp}} \ x\ 100 \quad (30)$$

III. EXPERIMENTAL INVESTIGATION

The important criterion in the design of waste heat recovery system is the proper selection of heat exchanger with optimum conditions. In the present work, the objective is to extract heat from diesel exhaust gas for heating tap water. In order to validate the design program, a small heat exchanger was fabricated and tested for its performance. The performance of diesel engine compared with and without heat exchanger:

3.1 Selection of Heat Exchanger

Various types of heat exchangers are available for recovery of waste heat. Selection of the heat exchangers for a particular waste heat recovery application depends on various factors such as heat potential available for recovery, possible use of recovered heat, space available for installing the unit and economics of heat recovery, etc.

3.2 Experimental Setup

The experimental setup consists of a four cylinder, four stroke, water cooled, Mahendra and mahendra (M&M) make diesel engine (bore 88.9 mm, stroke 101.6 mm, rated power 27.6 kW at 5000 rpm) coupled to an eddy-current dynamometer, integrated with a heat recovery heat exchanger (HRHE). Figure 1, 2 and 3 show the schematic diagram of the experimental setup, the photographic view of the tube bundle and shell and tube type heat exchanger respectively. The heat recovery system is a 1-2 shell and tube type heat exchanger, made up of mild steel with tube side fluid as exhaust gas and shell side fluid as tap water. Total tube area is 1.7 m2. The details of the shell and tubes in the HRHE are given in Table 1. The cross sectional view of the 1-2 shell and tube type heat exchanger is shown in Fig. 4. The HRHE was fitted into the exhaust pipe of the engine. The exhaust gas from the engine was allowed to flow either to the heat exchanger or to the atmosphere by using valves. Water was circulated through shell side of the heat exchanger using a pump.

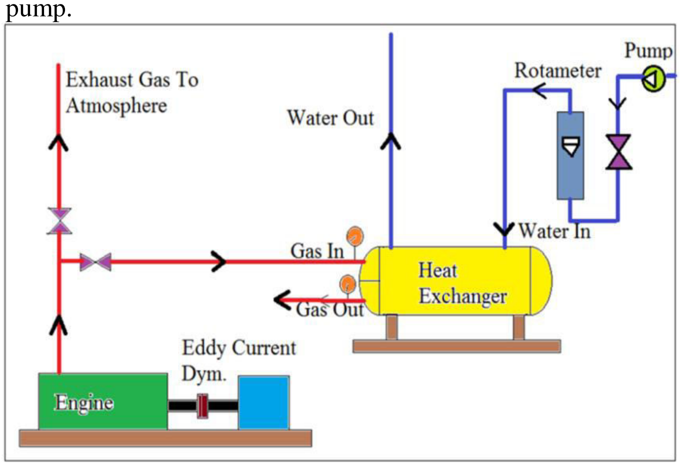

Figure 1. Schematic diagram of the experimental setup

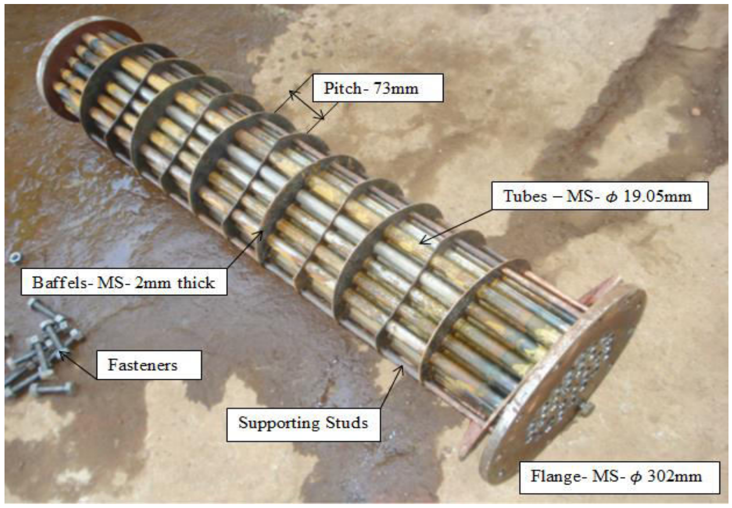

Figure 2. Photographic view of tube bundle

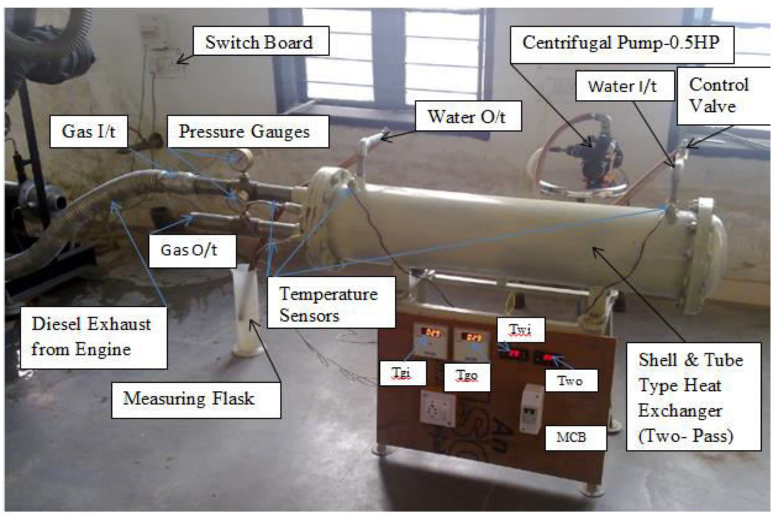

Figure 3. Photograph of the heat exchanger

3.3 Experimental Methodology

The experiments were conducted by operating the engine at various load/torque conditions. An eddy current dynamometer is used to vary the load on the engine. First the experiment is conducted at 50 N m torque. Initially the exhaust gas is not allowed to flow through the heat exchanger to avoid carbon deposition on the tube surface. After a short duration from the start of engine, the exhaust gas is allowed to pass through the tube side (two pass) of the heat exchanger while ensuring the water circulation through the shell side.

Computerized four stroke, four cylinder, turbocharge diesel engine is used for this experiments. The EPA software works with NI 6221 Card provided with the System only. This software communicates with various systems of test rig and collects the data, displays and saves data. The engine is variable speed engine; it can be varied speed of the engine with throttle valve. In the present work engine speed 1500, 1700 and 2000 rpm were used. Gas side pressure difference has been measured by pressure gauges, which are mounted on inlet and outlet exhaust gas line. Rotamerter is used to measure the mass flow rate of water. The temperature readings are continuously monitored in the inlet and outlet temperature of the HRHE. The above said measurements are used to evaluate the heat recovered. Several experiments are conducted to check the repeatability of the results. The experiments are conducted for 70, 90, and 110 N-m torque. Another experiment is also conducted to determine the performance of the diesel engine with and without heat exchanger. Validation of the design program with a small 1-2 shell and tube heat exchanger and tested for its performance. The results along with the evaluated parameters are analyzed and discussed in the following section.

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Table-1 Heat Exchanger Specification

Parameters	Specification
Shell material	Mild steel
Shell outer diameter	219 mm
Shell thickness	5 mm
Tube type	Plain type
Tube material	Mild Steel
Tube outer diameter	19.05 mm
Tube thickness	1.6 mm
Length of shell	1000 mm
No of tubes inside the shell	18
Type of tube layout	Triangular layout
Tube pitch	24 mm
No. of baffles	11
Baffle pitch	73mm
Thermal conductivity of M.S.	45 W/m K

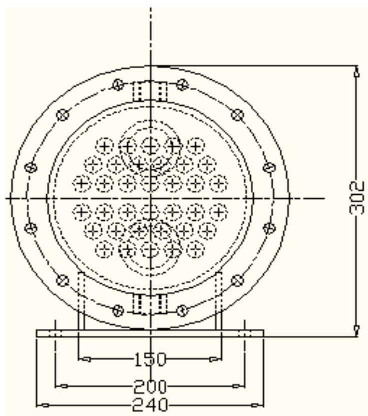

Figure 4. Cross sectional view of the heat exchanger
(All dimensions are in mm)

IV. RESUL AND DISCUTIONS

The results obtained from the experimental investigation for the engine operated at various load conditions are studied in detail and presented. The exhaust gas in an internal combustion engine carries about 30 to 40% of the heat of combustion. In the present work, attempts have been made to recover the maximum possible heat from the exhaust gas through a 1-2 shell and tube heat exchanger and to measure the performance of diesel engine with and without heat exchanger. Outlet temperature of water and gas are predicted by computer program, it is compared with outlet temperature of water and gas from heat exchanger.

4.1 Performance of Heat Recovery Heat Exchanger

Fabricated model of shell and tube type heat exchanger was tested for heat transfer performance by using diesel engine exhaust. Recover heat from exhaust was used for water heating. Test was conducted for various mass flow rates of water. Before conducting trials, separate trial of the engine was conducted for estimation of mass flow rate of exhaust gas. An engine was run at various speed i.e.1500, 1700, 2000 rpm with 70, 90, 110 N-m torque. There were seven mass flow rates of water were used i.e. 0.0316, 0.0370, 0.0452, 0.0500, 0.0550, 0.0625, 0.0694 kg/s. Nine condition were used for the seven mass flow rate of water i.e. total trial conditions were 63.

4.2 Graphs of Overall Heat Transfer Coefficient vs. Mass Flow Rate of Water

Figure 5a shows that theoretical and measured overall heat transfer coefficient depends on engine exhaust temperature. The exhaust temperature decreased from 170oC to152oC, both OHTC decreased. Last two reading shows exhaust temperature increased from 152oC to 158oC, the OHTC increased. Outlet temperature of diesel engine exhaust depends on engine speed and load/torque. It was observed that experimental OHTC more than theoretical OHTC. Deviation between OHTC was less at 0.055 kg/s mass flow rate of water and maximum at 0.0316 kg/s mass flow rate of water. Figure 5b shows that the temperature of exhaust was not maintained at same condition of the engine like 1500 rpm and 90 N-m because the throttle of the engine changed its position due to engine vibration. Because of this reason exhaust temperature was fluctuated therefore OHTC deviation varied with exhaust temperature. Maximum OHTC deviation obtained at low mass flow rate of water. Minimum deviation observed at 0.055kg/s mass flow rate of water.

Figure 5c indicated that the exhaust temperature is constant at first three and last two mass flow rate of water. It was observed that the OHTC deviation is same at respective mass flow rate of water. Overall heat transfer coefficient of the heat exchanger was same for the first three measurements because at that condition exhaust temperatures of the engine were same. The OHTC deviation varied with temperature of engine exhaust. Minimum deviation was about 17 % and maximum deviation was about 25 %, as shown in figure 5d. Figure 5e shows that theoretical OHTC is in between 35 - 40 W/m2 K and experimental OHTC is in between 42 - 50 W/m2 K for all mass flow rate of water. At 0.0550 kg/s mass flow of water, exhaust temperature of engine was very low as compare to anther mass flow rate of water at 1700 rpm and 110 N-m torque. It was noticed that OHTC deviation was low i.e. 18 %.

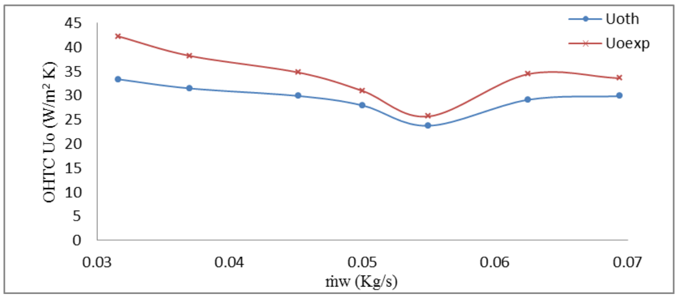

Figure 5a. Overall heat transfer coefficient vs. Mass flow rate of water at 1500rpm and 70 N-m torque

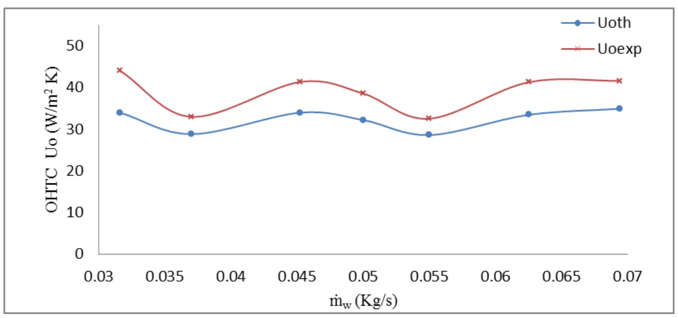

Figure 5b. Overall heat transfer coefficient vs. Mass flow rate of water at 1500rpm and 90 N-m torque

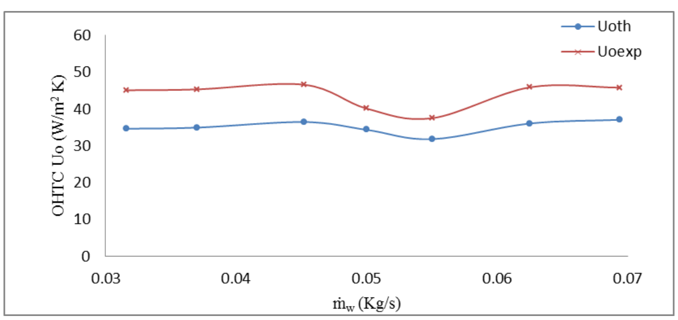

Figure 5c. Overall heat transfer coefficient vs. Mass flow rate of water at 1500rpm and 110 N-m torque

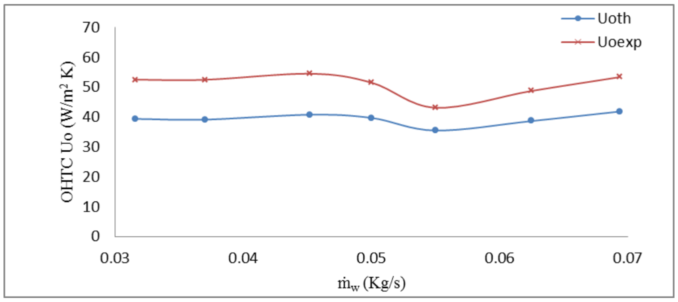

Figure 5d. Overall heat transfer coefficient vs. Mass flow rate of water at 1700rpm and 70 N-m torque

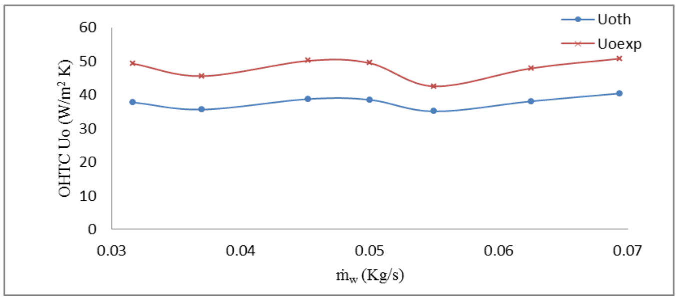

Figure 5e. Overall heat transfer coefficient vs. Mass flow rate of water at 1700rpm and 90 N-m torque

It was shown in figure 5f. Exhaust temperature increased with increasing speed and torque of the engine. Measured OHTC 66.66 W/m2 K and theoretical OHTC 49.24 W/m2 K were obtained at 0.0694 Kg/s mass flow rate of water. Maximum deviation was 27 % at 0.0370 kg/s as shown in figure 5g. Overall heat transfer coefficient deviation for measured and theoretical was near about same for all mass flow rate of water.

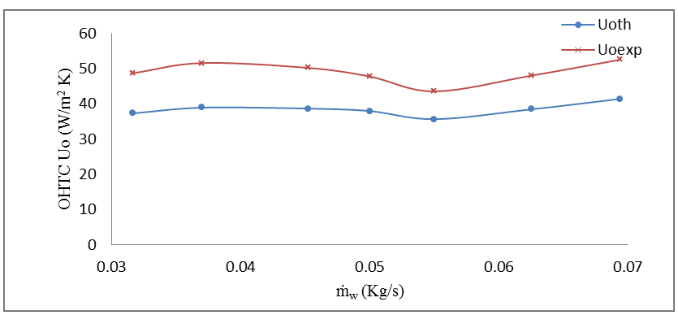

Figure 5f. Overall heat transfer coefficient vs. Mass flow rate of water at 1700rpm and 110 N-m torque

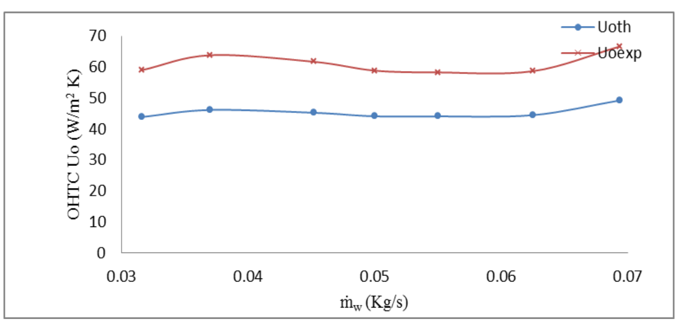

Figure 5g. Overall heat transfer coefficient vs. Mass flow rate of water at 2000rpm and 70 N-m torque

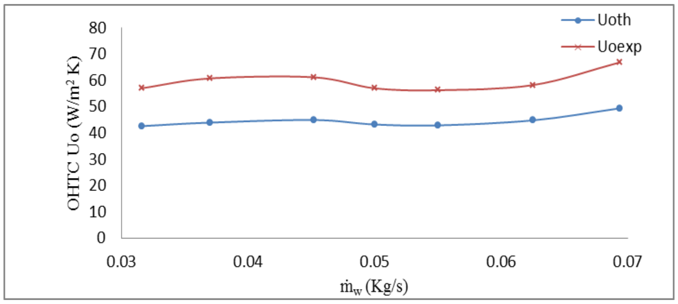

Figure 5h. Overall heat transfer coefficient vs. Mass flow rate of water at 2000rpm and 90 N-m torque

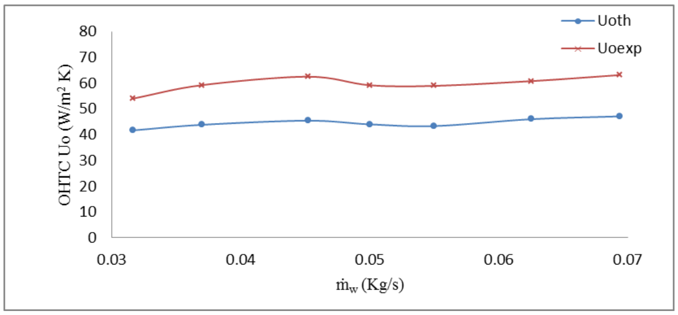

Figure 5i. Overall heat transfer coefficient vs. Mass flow rate of water at 2000 rpm and 110 N-m torque

It can be shown in figure 5h. Maximum temperature of exhaust was obtained for this case is 233oC and experimental OHTC was 62 W/m2 k. The trend of both OHTC is same as shown in figure 5i.

b Graphs of Outlet Temperatures of Gas & Water vs. Mass Flow Rate of Water

Measured and predicted outlet temperature of water and exhaust decreased with increasing mass flow rate of water. Temperature ranges of the fluids were increased by increasing mass flow rate of water. Measured outlet temperature of water and exhaust is more than predicted temperatures of water and exhaust. Experimental and predicted outlet temperature of water is much closer than outlet exhaust temperature. Outlet water temperature deviation is zero percentage at 0.055 kg/s mass flow rate of water and maximum deviation is 4 %. Maximum outlet exhaust deviation is 15%, as shown in figure 6a.

Figure 6b shows that measured exhaust temperature is less than predicted exhaust temperature at 0.0316 mass flow rate of water so exhaust deviation was negative. About zero percent water temperature deviation was obtained at 0.055 kg/s mass flow rate of water. Outlet exhaust temperature decreased with decreasing mass flow rate of water but at 0.0694 kg/s it was increased. It was occurred due to increasing exhaust temperatures from the engine (inlet exhaust temperature for the heat exchanger). Figure 6c shows that maximum outlet water deviation is about 5 % at 0.0316 mass flow rate of water. When mass flow rate is increased, all measured and predicted temperature were decreased but at last two mass flow rate of water it was increased due to increasing in exhaust temperature from engine. Outlet temperature of exhaust and predicted out let temperature of water were same at 0.0316 kg/s mass flow rate of water and outlet water temperature deviation is maximum at the same mass flow rate of water as shown in figure 6d.

Figure 6e shows that negative deviation for exhaust temperature because measured value of exhaust is less than predicated value. Positive deviation was obtained for outlet temperature of water. Predicted outlet water temperatures were same at 0.0625 and 0.0694 kg/s mass flow rate of water. All outlet exhaust temperatures deviation were negative at all mass flow rate of water except 0.055 kg/s mass flow rate of water as shown in figure 6f.

Figure 6g indicates that all exhaust temperature deviations were negative and all deviations of water were positive. Temperature difference between measured and predicted temperature of water and exhaust increased with increasing mass flow rate of water. Water and exhaust temperature deviation increased along with increasing exhaust temperature as shown in figure 6h. Measured and predicted outlet water temperatures were same at 0.0316 kg/s mass flow rate of water. Figure 6i shows that measured and predicted water temperature decreased with increasing mass flow rate of water. Same thing was not obtained for exhaust temperature. Water and exhaust temperature deviation were 7 and -14 % at 0.0370 kg/s mass flow rate of water.

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The overall heat transfer coefficient (OHTC) calculated for various mass flow rates of water. Over all heat transfer coefficient for different readings of the trial are plotted on a graph of overall heat transfer coefficient vs. mass flow rate of water which numbered as fig 5a-5i. It was observed that when speed and load of diesel engine increased the deviation between the overall heat transfer coefficients from the experimental results and analytical model were increased. A minimum deviation obtained at 1500 rpm speed, 70 N-m torque and 0.0550 kg/s mass flow rate of water was 7.5% and a maximum deviation obtained at 2000 rpm 110 N-m torque and 0.0452 kg/s. The experimental results shows that the deviation between the overall heat transfer coefficient from the experimental results and values obtained from analytical model are within 25%.It was seen that the deviation between measured and theoretical overall heat transfer coefficient is less for 0.0550kg/s mass flow rate of water for each case. Overall heat transfers coefficient increases when engine speed and torque were increased.

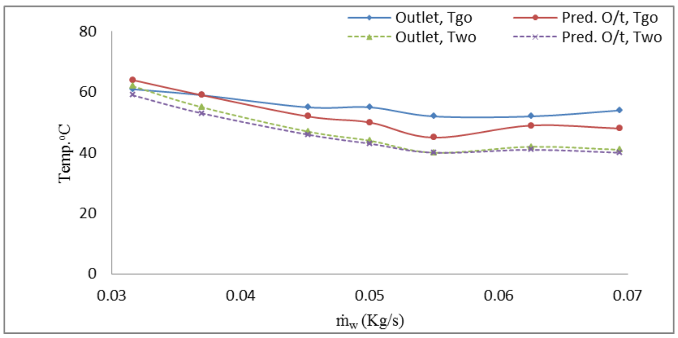

Figure 6a. Outlet temperatures of gas & water vs. mass flow rate of water at 1500rpm and 70 N-m torque

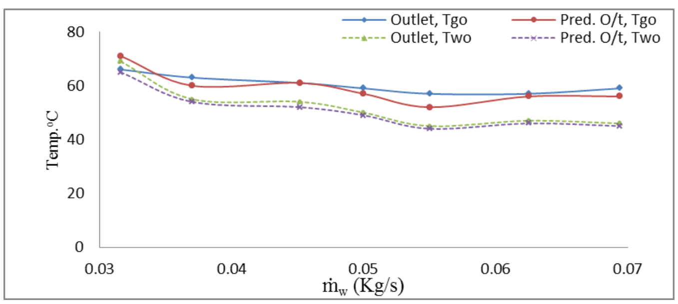

Figure 6b. Outlet temperatures of gas & water vs. mass flow rate of water at 1500rpm and 90 N-m torque

Outlet temperature of water decreases with increasing mass flow rate of water. According to experimental result, maximum outlet temperature of water is 91oC and minimum is 40o C. Outlet temperature of water can be predicted by □-NTU method of heat exchanger. The deviations of predicted outlet temperatures of water were increased with increasing speed and load of an engine. Heat losses affect predicted outlet temperatures of water. The deviation between the predicted temperatures and temperatures from experimental result are within ± 8%.

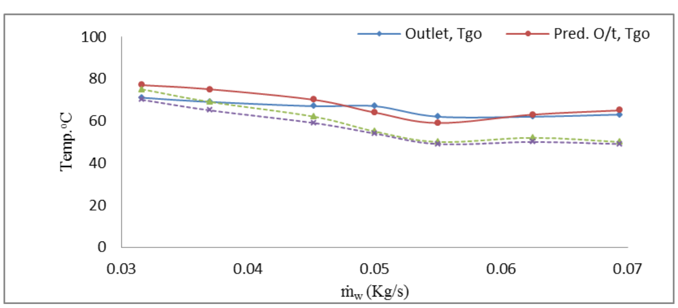

Figure 6c. Outlet temperatures of gas & water vs. mass flow rate of water at 1500rpm and 110 N-m torque

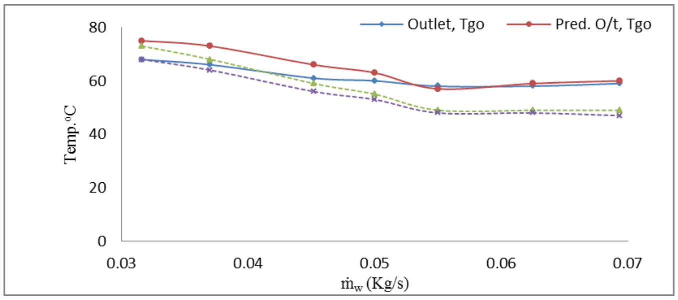

Figure 6d. Outlet temperatures of gas & water vs. mass flow rate of water at 1700rpm and 70 N-m torque

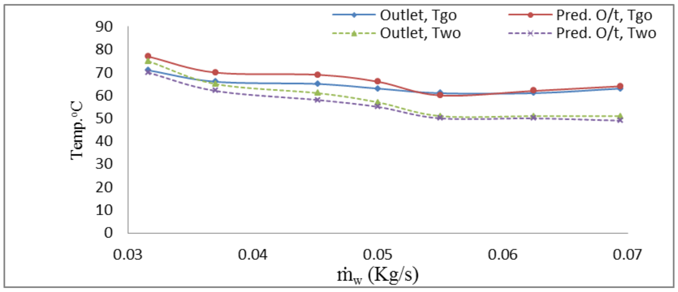

Figure 6e. Outlet temperatures of gas & water vs. mass flow rate of water at 1700rpm and 90 N-m torque

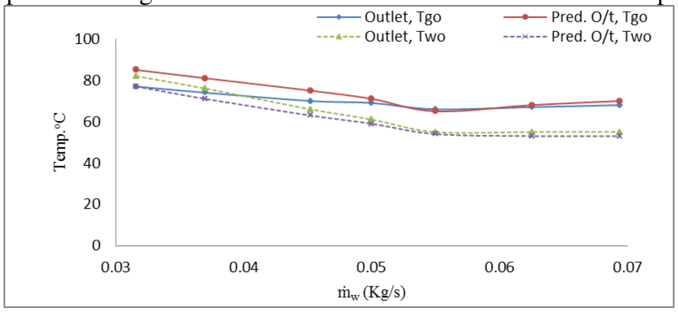

Figure 6f. Outlet temperatures of gas & water vs. mass flow rate of water at 1700rpm and 110 N-m torque

The deviation between the predicted outlet temperature of gas and measured outlet temperature of gas are within ±15%. Values of outlet gas and water temperature for different readings of the trial are plotted on a graph of outlet water temperature vs. mass flow rate of water as graph 6a - 6i Predicted outlet temperatures of water are less than measured outlet temperature of water.

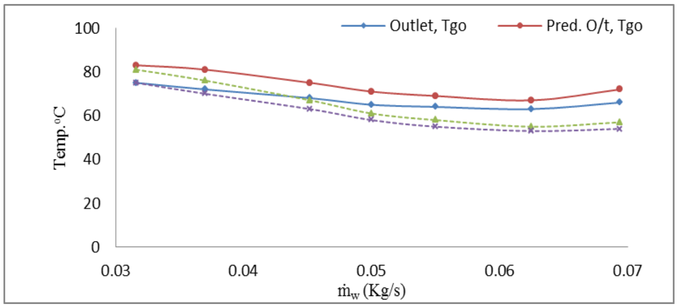

Figure 6g. Outlet temperatures of gas & water vs. mass flow rate of water at 2000rpm and 70 N-m torque

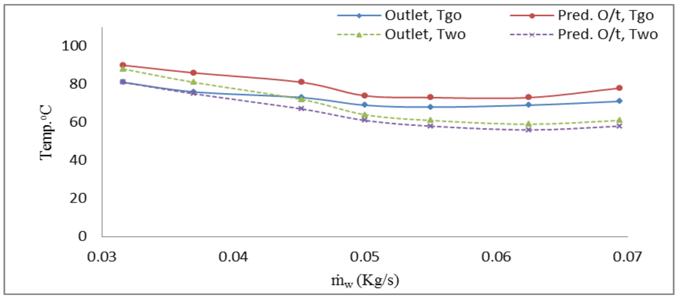

Figure 6h. Outlet temperatures of gas & water vs. mass flow rate of water at 2000rpm and 90 N-m torque

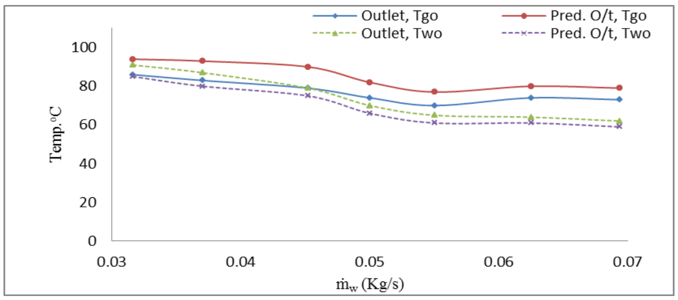

Figure 6i. Outlet temperatures of gas & water vs. mass flow rate of water at 2000 rpm and 110 N-m torque

4.3 Performance of the Diesel Engine

Performance of the diesel engine was measured with and without heat exchanger at 1500 rpm engine speed and 90 , 110 Nm torque and compared to each other. It was observed that brake power and indicated power slightly decrease. The heat balance sheet result shows that heat losses through diesel exhaust decrease for certain level and heat losses through cooling water remain same.

V. CONCLUSION

Computer program was developed to carry out design simulations for optimum design of the heat exchanger for waste heat recovery from engine exhaust. In order to validate the design program, a small 1-2 shell and tube heat exchanger was fabricated and tested for its performance. The exhaust from engine was utilized to heat the water.

Deviation in overall heat transfer coefficients obtained by using correlations and using experimental results. It was observed that when speed and load of diesel engine increased, the deviation between the overall heat transfer coefficients from the experimental results and analytical model were increased. The deviation between the predicted outlet temperatures of water and temperatures from experimental result are within ±8% and deviation for outlet gas temperature are within±15%.

The estimated values of heat transfer coefficients and friction factors are subject to uncertainties depending on values of Re. These uncertainties in heat transfer coefficients and friction factors mainly led to deviation in estimated and experimental values of the outlet temperatures. In order to see the effect of waste heat recovery on engine performance, tests were conducted under almost same operating conditions with and without heat recovery. The results obtained shows that the effect on engine performance is very small (3% change in output of the engine) which is mainly due to back pressure developed by use of heat exchanger in the exhaust line of the engine. However, the effect is negligible. The heat balance sheet result shows that heat losses through diesel exhaust decreases in the same proportion as that of heat recovery from exhaust, which is obvious. Nearly 10% of total heat (that would otherwise be gone as waste) is recovered with this system.

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NOMENCTURE

A	Cross section area, m2	Nu	Nusselt number
A/F	Air-fuel ratio	P	Tube pitch, m
Afl.g	Area available for gas Flow, m2	Pr	Prandlt number
Afl.w	Area available for water Flow, m2	PT	Tube pattern
B	Baffle spacing, m	Q	Heat load, W
C	Clearance between adjacent tubes, m	Re	Renold number
Cp	Sp heat at constant pressure, J/Kg oC	Rf	Fouling Factor
D	Diameter of shell, m	T	Temperature, oC,
De	Equivalent diameter of shell, m	$\dot{m}$	Mass flow rate, Kg/s
D	Diameter of tube	ΔP	Pressure drop, N/m2
F	Correction factor/ coeff. of friction	ΔT	Temperature difference, oC
F	Darcy friction factor	TL	Total length of tube, m
G	Mass velocity, Kg/sm2	T	Torque, N M
H	Heat transfer coefficient, W/m2 0K	T	Thickness of tube, m
L	Length of tube, m	Uo	Overall heat transfer coeff., W/m2K
NT	Number of tubes	V	Velocity of gas, m/s
Sub Scripts			
air	Air	e	Equivalent
atm	Atmospheric	exp	Experimental
avg	Average	f	Fuel
ass	Assume	g	Gas
c	Cold	h	Hot
cal	Calculated	i	Inner, Inside, Inlet
Greek Symbols			
μ	Viscosity, N s/m2	ρ	Density, kg/m3
k	Thermal conductivity, W/m oK		Effectiveness

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