EXPERIMENTAL AND NUMERICAL ANALYSIS OF DIESEL ENGINE EXHAUST HEAT RECOVERY USING TRIPLE TUBE HEAT EXCHANGER

by

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In this study, a 2000 mm long triple tube heat exchanger was designed and manufactured in this work with three intermediate tubes having annulus space of 26, 32, and 36 mm, respectively. Experimental investigations were carried out in the system to assess the percentage of energy savings. On the Diesel engine experiments were conducted by varying the load conditions at 25%, 50%, 75%, and 100%, respectively. Motor speed was varied at 900, 1200, and 1500 rpm, respectively, for each load condition. Also incorporated during the study were counter current, co-current with counter current, counter current with co-current, and co-current fluid flow patterns. It is found that, while increasing the load and speed, the heat transfer rate of the heat exchanger increased. It is also observed that, the fluid counter current flow pattern gave better performance compared to other flow pattern types. The effects of the operating parameters on the heat exchanger's performance are represented by the Nusselt number and effectiveness. The results of the experiments were also compared with the thermal energy storage performance of the double tube heat exchanger. It is found that, compared to double tube heat exchanger, 20% of fuel energy was saved by using triple tube heat exchanger as waste heat recovery system.

Key words: *heat exchanger, waste heat recovery, triple tube, counter current, double tube*

Introduction

Today, for many engineering applications, heat extracted from exhaust waste gas is used. Because of their capabilities and benefits, heavy-duty Diesel engines are widely installed and operated in industries for electrical energy production, transportation, *etc.*, but about 30% of the input energy is wasted by exhaust gas emissions and water cooling. It is observed that for quite some time the traditional heat exchangers are used to recover heat from waste gases. But, to achieve maximum heat recovery, they are expensive and require very high heat transfer area. Another issue is the deposition of carbonaceous matter on the surface of the heat exchanger due to hot gas discharge. Soot fouling is the major cause of the drastic decay of thermal efficiency in conventional shell and tube heat exchanger [1]. From some of the literature [2-4] it has been observed that typical Diesel engine converts about 38% of the input energy supplied into useful work and the remaining energy is wasted through exhaust gases for coolant and lubricating oil including frictional losses.

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Hatami *et al.* [5-7] and Vivekananden *et al.* [8] conducted a response surface methodology experiment to obtain an optimized fine type heat exchanger design to recover waste heat from a Diesel engine's exhaust gas. They reported that, while the height of the fins influences the drop in pressure, the number of fins affects the heat recovery. In another experiment, the new design of a heat exchanger to recover exergy from a Diesel engine exhaust, they proposed a design consisting of twenty vortex generators with optimum heat exchanger dimensions and angle of attack to achieve more exergy recovery as well as minimal pressure drop. They also compared Nusselt number results, friction factor, and recovered exergy from the new heat exchanger design with the simple double pipe heat exchangers. A vortex generator heat exchanger was examined to recover exergy from an OM314 Diesel engine exhaust gas. Twenty vortex generators with an attack angle of 30° were used to improve heat recovery and low back pressure in the exhaust gas with five engine loads and two exhaust gasses. Based on the results, the central composite design optimization study was conducted. Senthil Kumar *et al.* [9] used a spherical shape dimpled tube to numerically analyse the heat transfer rate of the triple concentrate tube heat exchanger. Govindhasamy *et al.* [10] designed a heat exchanger with twisted tape inserts in the corrugated tube and reported that this type of heat exchanger increased the heat transfer rate around 235.3%. In order to recover waste heat, Mavridou *et al.* [11] conducted experiments on two types of heat exchangers with finned surfaces attached to the exhaust gas side. They designed a classic heat exchanger for shells and tubes with a staggered cross-flow as well as a heat exchanger for cross-flow plate. Kumar *et al.* [12] developed a dynamic model for the analysis of a modular two-phase heat exchanger. Based on the reviews carried out on the design of the heat exchangers required for the exhaust gas heat recovery system, a new type heat exchanger must be developed to meet the requirements of high heat transfer rate and heat exchanger efficiency [10]. The current study focuses on filling this gap by introducing a newly developed triple tube concentric tube heat exchanger that is attached to a heavy duty diesel engine's exhaust gas side. Efforts have been made to determine the optimum rate of heat transfer from the waste gas and to improve the efficiency of the heat exchanger by adjusting the engine loads, speeds and intermediate exhaust gas diameter and flow patterns.

Experimental set-up and procedure

A triple tube heat exchanger (TTHE) is used in this current study to extract waste heat from the engine exhaust gas released by the Diesel engine so that the engine performance output can be studied under different load and speed conditions. The thermal energy storage device is used for the collection and storage of heat energy from exhaust gas. The three-tube heat exchanger configuration has been designed to enhance the convective heat transfer rate between hot exhaust gas and cold water flowing around the flue gas sides. Figure 1 showed the schematic view of the experimental set-up which consists mainly of a 2-cylinder, four-stroke and water-cooled Diesel engine with a rated power capacity of 7.36 kW and a test rig consisting of separate circuits for measuring fuel-flow rate, air-flow rate and water-flow.

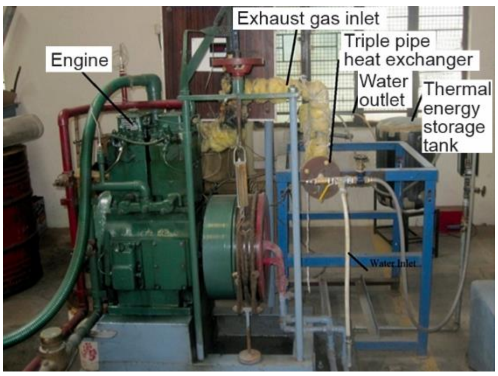

Figure 1. Experimental set-up pictorial view

The triple tube heat changer was manufactured with internal, intermediate and outer tubes with 13 mm, 45 mm, and 70 mm diameters, respectively, 3 mm wall thickness, whereas the heat exchanger length is 2000 mm. The engine exhaust gas pipe outlet is attached to the intermediate heat exchanger tube where a flow control valve is used to control the exhaust gas flow rate. With the help of the water circulation pump, cold water from the water tank is circulated between the heat exchanger's inner and outer tubes. To measure and control the quantity of water circulating, rotameter and flow control valves are used. To receive the hot water from the heat exchanger, a well-insulated thermal storage reservoir has been set up. Temperature indicator instrument is used to measure temperatures with the help of K-type thermocouples that are fixed at various test rig locations. Digital manometers are used to measure the pressure drop in the exhaust gas pipe that is connected to the flow circuits of water and exhaust gas. The experiments for the current study were conducted for various load conditions ranging from 25% to 100% in the 25% steps as well as speed conditions with different flow types.

Numerical analysis

The governing equations describing mass equation, momentum equation, and energy equations regarding the thermo fluid characteristics of the triple concentrate tube heat exchanger with a set of nonlinear particle differential equations are given below.

– Mass equation

$$\frac{\partial \rho}{\partial t} + div(\rho u) = 0 \tag{1}$$

– continuity equation

$$\frac{\partial (u)}{\partial x} + \frac{\partial (v)}{\partial y} + \frac{\partial (w)}{\partial z} = 0 \tag{2}$$

– X momentum equation

$$\frac{\partial (\rho uu)}{\partial x} + \frac{\partial (\rho vu)}{\partial y} + \frac{\partial (\rho wu)}{\partial z} = \frac{\partial \sigma xx}{\partial x} + \frac{\partial \tau yx}{\partial y} + \frac{\partial \tau zx}{\partial z} \tag{3}$$

– Y momentum equation

$$\frac{\partial (\rho uv)}{\partial x} + \frac{\partial (\rho vv)}{\partial y} + \frac{\partial (\rho wv)}{\partial z} = \frac{\partial \tau xy}{\partial x} + \frac{\partial \sigma yy}{\partial y} + \frac{\partial \tau zy}{\partial z} \tag{4}$$

– Z momentum equation

$$\frac{\partial (\rho uw)}{\partial x} + \frac{\partial (\rho vw)}{\partial y} + \frac{\partial (\rho ww)}{\partial z} = \frac{\partial \tau xz}{\partial x} + \frac{\partial \sigma yz}{\partial y} + \frac{\partial \tau zz}{\partial z} \tag{5}$$

– energy equation

$$\frac{\rho c_p}{k}\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) + \frac{\mu}{k}\left[2\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial y}\right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial u}{\partial x}\right)^2\right] \tag{6}$$

The momentum and energy equations are solved by second order up wind scheme. The transport equations of K-ε model is given:

$$\sum_{\mathrm{f}}^{Nfaces} u_{\mathrm{f}} \varphi_{\mathrm{f}} A_{\mathrm{f}} = \sum_{\mathrm{f}}^{Nfaces} \Gamma_{\varphi} (div\varphi)_n A_{\mathrm{f}+S_{\varphi}} \tag{7}$$

The technique of CFD was used to find the complete system solution for which detailed procedure was elaborated. The initial analysis was conducted for various mesh sizes so that the correct number of elements can be identified which enumerate accurate results in a short time. Adaptive-grid refinement technique with 1533370 nodes was used to refine the grid density. Based on the experimental results, the boundary conditions are used to analyse the triple concentrate tube heat exchanger in which hot and two cold fluids are set at inlet temperatures and velocities. The pressure at the heat exchanger outlet is set to zero after describing the material properties of both hot and cold fluids. Semi implicit pressure method linked equation technique was used to solve the triple concentrate tube heat exchanger's computational domain. In this work, the influence of heat transfer in the 26 mm, 32 mm and 36 mm diameter intermediate tubes was analysed across a range of parameters.

Results and discussions

The experiments were conducted on the THE to estimate the optimum rate of heat transfer from the waste gas and to enhance the efficiency of the heat exchanger by adjusting the engine loads, speeds, intermediate diameters, and flow patterns of the exhaust gas. The optimization procedure was carried out in this current study by fixing one design variable while maintaining the other design variable as a constant and feasible solution can be found.

Comparisons of effectiveness and Nusselt number between DTHE and TTHE

Variations in efficiency and calculated Nusselt numbers for double tube heat exchanger (DTHE) as well as TTHE are shown in fig. 2. The TTHE thermal performance was compared with the DTHE performance for similar conditions. From the figure it can be seen that, with the increase in engine load, the Nusselt number and efficiency of both heat exchangers gradually increased. This is due to the gradual increase in exhaust gas temperature as the engine load gradually increases. So when compared to lower loads, both parameters show higher values. The results also revealed that, compared to the DTHE, the efficiency of the TTHE is higher at all engine loads. At the same operating conditions, the efficiency of the TTHE increased 20.40% more compared to that of the DTHE. This is due to more contact area of the exhaust gas counter current flow pattern and thus higher heat transfer rate in the heat exchanger for triple tubes. Similarly, it is observed that, in the same operating conditions, the Nusselt number of the TTHE also showed 33.33% higher than that of the DTHE. Adding the third tube to the TTHE has helped to increase the surface area because of which higher heat transfer rate acceleration compared to

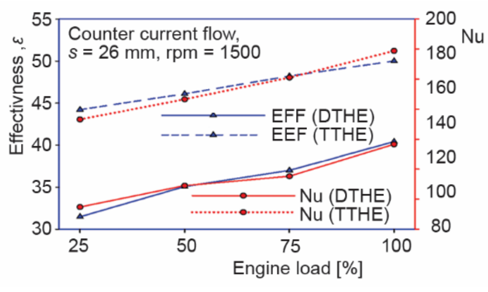

Figure 2. Comparisons between DTHE and THE of effectiveness and Nusselt number

DTHE acceleration. More exhaust gas circulation time within the TTHE was another reason for higher heat transfer rate and hence higher heat exchanger Nusselt number was achieved.

Effect of flow patterns

Now experiments have been conducted with four different flow arrangements of the two fluids inside the TTHE specifically with the counter-current, co-current with counter current, counter-current with co-current, and co-current to identify the optimum flow pattern for further enhancement of efficiency and Nusselt number. These flow patterns were, respectively, marked as A to D and the results are represented in the following graphs. Figure 3 shows the relationship between Nusselt numbers for all engine load conditions corresponding to the different flow pattern of the TTHE. It is observed that, at all engine loads at 1500 rpm, the Nusselt number for flow pattern (A) is higher than all other flow patterns. The Nusselt number corresponding flow pattern (A) is also evaluated at full load increased by about 23.4% when the engine was operated at a load of 25%. This may be due to the surface heat transfer associated with intermediate exhaust gas exhaust tube temperature side is more than the same with cold water side resulting in higher heat transfer rate.

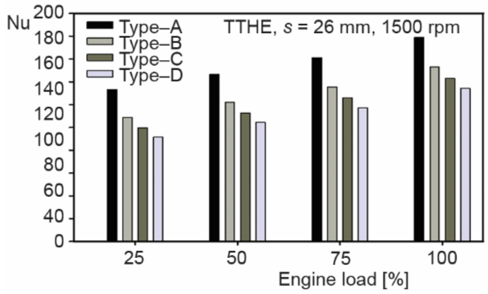

Figure 3. Variation of Nusselt numbers *vs.* different flow pattern of TTHE

Figure 4 shows the efficacy of the TTHE for different flow patterns in relation to different engine loads. It can be seen that, for all flow pattern arrangements, the effectiveness of the heat exchanger increases with the increase in engine load. Again, it is observed that, compared to all other flow patterns at all engine loads, the flow pattern (A) has achieved the highest possible efficiency value. It is also evaluated that, when compared with the 25% engine load under similar conditions, the efficiency of the heat exchanger having the flow pattern (A) is increased by 10.10% at full load. This is due to the average heat transfer rate that is higher at full load for counter-current flow pattern.

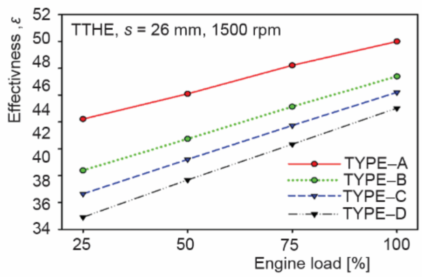

Figure 4. Variation of Effectiveness *vs.* different flow pattern of TTHE

Effectiveness and Nusselt number with respect to engine speed

The experiments were conducted to determine the optimum speed for different loading conditions using a TTHE attached to a heavy duty Diesel engine so that its efficiency and Nusselt number can be assessed for optimum speed. The engine speed was initially set at 900 rpm and the load conditions varied from 25% to full load in 25% steps. For further analysis of the study, experimental data were recorded. The same operating procedure was repeated and data collected by setting engine speed at 1200 rpm and 1500 rpm, by varying engine loads, respectively. The efficiency was presented in fig. 5 at different speeds and engine loads. The figure shows that, for all load conditions, the efficiency is almost directly proportional to the engine speed. From the figure, the maximum heat exchanger efficiency at an en-

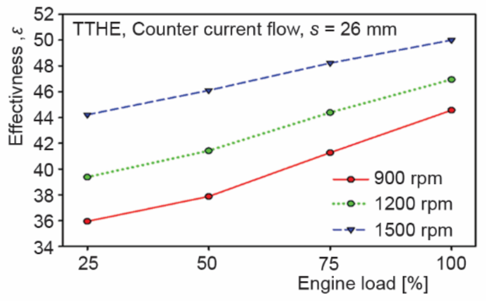

Figure 5. Effectiveness *vs.* engine load for different engine speed

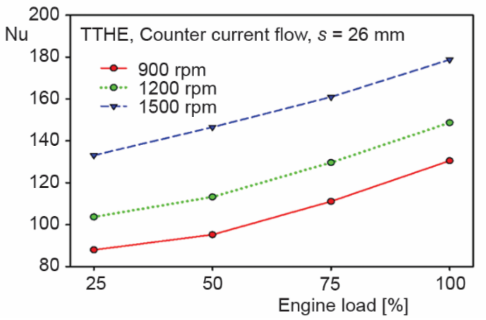

Figure 6. Nusselt number *vs.* engine load for different engine speed

gine speed of 1500 rpm is observed. It is also observed that when the engine speed was increased from 900 rpm to 1500 rpm at full load condition, the efficiency of the TTHE was improved by 12%. This clearly indicates that at higher engine speed, heat recovery will be more viable. As the engine load increases, the temperature of the exhaust gas also increases as a result of which the engine releases higher heat.

Figure 6. represents the the Nusselt number assessed at different speeds and conditions of load. It is seen from the figure that, with all three engine speeds as well as load conditions, the Nusselt number forms an approximately linear relationship. It is deduced from the figure that the heat exchanger's maximum Nusselt number at an engine speed of 1500 rpm was obtained. It is also observed that, when the engine speed varies from 900 rpm to 1500 rpm at full load condition, the Nusselt number of TTHE has been improved by 29.7%. This is due to an increase in exhaust gas velocity from inlet to outlet as an increase in engine load resulting in an increase in turbulence and the coefficient of convective heat transfer. This causes less heat exchanger pressure drop and higher intake system pressure density. It is therefore observed that, with the increase in engine load, less heat loss percentage is observed.

Percentage energy saved

The following equation indicates the percentage of fuel power saved when the TTHE is inserted when it is replaced by the DTHE requiring either fuel or electric power:

$$E_s = \frac{Q_C}{m_f CV} \tag{8}$$

From the results it can be concluded that the Diesel engine using a TTHE with an intermediate tube diameter of 26 mm provides a higher Nusselt number and efficiency when operated with full load at a speed of 1500 rpm as well as with counter current flow pattern. The results show considerable energy can be saved in the fuel and that it varies from 8% to 10% as the load increased from 25% to full load. Due to the higher specific fuel consumption at part load condition, the percentage of energy saved is lower at lower loads along with higher heat loss during the charging process for longer duration. Senthil Kumar *et al.* [13] analysed the waste heat recovery system in one of the literatures by designing the DTHE with helical fin and DTHE with porous tube. These heat exchangers have extracted 1-8% of energy. The newly designed TTHE shows 20% more efficient while comparing them.

Conclusion

In this current study, the performance of the TTHE was analysed both experimentally and numerically to identify the optimum design configuration for which a higher rate of waste heat recovery, Nusselt number and heat exchanger efficiency was achieved. The experiments were conducted during the analysis by varying the engine load, speed, fluid and intermediate tube diameter flow patterns. The TTHE experimental results were also compared numerical analysis with an error of ±10% using the CFD model. The results revealed that when the engine is operated at full load, the heat exchanger's efficiency, and Nusselt number increased. It is also observed that, using the counter current flow pattern, the higher values of the Nusselt number and heat exchanger efficiency were achieved. It was found during the investigations that the efficiency of the TTHE is increased at a lower value of the intermediate tube diameter at a fixed mass-flow rate. The TTHE therefore contributes higher heat exchanger efficiency and Nusselt number in addition to energy savings of 20% compared to the DTHE.

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