FORCED CONVECTIVE HEAT TRANSFER USING TURBULENCE PROMOTERS

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Abstract :
Internal Combustion Engines (IEC) is one of the most important prime movers used in automobile, manufacturing, aerospace sectors etc. It is a well-known fact that most of the chemical energy obtained by the combustion of fuels are carried out by the exhaust gases. It is known fact that exhaust gas from an IEC carries away about 30 – 40% of heat of combustion. The energy available in this stream of exhaust gases goes as a waste (if not utilized). This paper presents a concept of recovering waste heat energy of exhaust gas of diesel engine by placing a heat exchanger in the exhaust manifold so that energy from the exhaust gases can be used for preheating fuel. A shell and tube heat exchanger is fabricated for diesel engine to recover waste heat from exhaust. Maximum fuel temperature achieved for counter flow arrangements at 50% of full load at 1440 rpm. Effectiveness of heat exchanger is found to be 81%. Waste heat recovered at 50% full load condition is found to be 72%. Turbulence promoters are useful in enhancing the heat transfer and mass flow rates in the given section. As well, Nanofluids are known to effectively improve heat transfer under turbulent flows. Turbulent flow is maintained under varying conditions of 5 – 20 m/s inlet velocity and heat carry away fluid maintained at 6 lts/ min. Enhanced heat transfer rate is observed in DI/ Ag+ NP compared to Cold Water at 5^0c and DI/ Alum NP nanofluids at 28^0C employing annular turbulence promoters embedded with inner spleen sections laterally fabricated along the Shell of the Heat Exchanger.

Index Terms: IEC, Diesel Engine, Nanofluids, Nano Particles, DI/Ag+, DI/Alum.

I. INTRODUCTION

In among of main power in the transportation, construction, fishery and agriculture machinery, engine has played an important part and consumed more than 60% of fossil fuel, thus it was able to result in exhausting the fossil fuel. Recent propensity about using the energy sources aiming at reducing the fossil fuel consumption as well as pollution was considered as urgent task. Up to now, the major consumer of fossil fuel was the internal combustion engines (ICE), however only about 30-40% energy of combustion in the engine chamber was transformed into useful mechanical work. The rest heat source expelled to the environment or lost through exhaust gases and cooling water/oil were approximately 25–35%, hence it was necessary to utilize and recover the waste heat to increase the heat efficiency of internal combustion engines. The waste heat recovery and utilization not only saved energy but also reduced the toxic pollution. Engine manufacturers have implemented and improved the latest techniques to increase thermal efficiency by enhancing the fuel-air mixing, using turbo-charger, and variable valve timing or advance combustion chamber.

WH was heat generated by the fuel combustion or chemical reaction. In the time of engine run, four sources of WH such as exhaust gas, cooling oil/water/liquid, lube oil, and turbocharger were dissipated to the atmosphere from the engine. WH depended on not only the temperature of the waste heat gases, but also mass flow rate of exhaust gas of engines. Exhaust gas temperature of diesel engines after leaving the engine were as high as 450 - 600°C. Consequently, the higher the exhaust gas temperature was, the higher the heat value was, however, the temperature of exhaust gases were limited by the laws of thermodynamics. Total energy from diesel engines was shown in the Figure 1. WH recovery from diesel engines brought many big benefits not only high power engines, but also smaller engine. Benefits from WH recovery might be divided into direct or indirect benefits.

2. Problem Definition

The increasingly worldwide problem regarding rapid economy development and a relative shortage of energy, the internal combustion engine exhaust waste heat and environmental pollution has been more emphasized heavily recently. Out of the total heat supplied to the engine in the form of fuel, approximately,

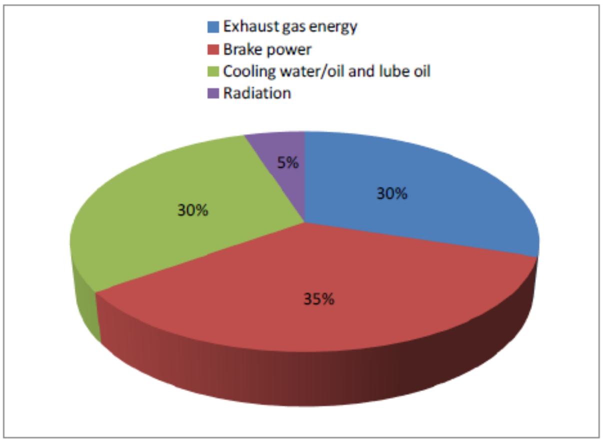


Figure 1. Total energy in diesel engines

Table 1: Various Engine and There Output (K.T.Wojciechowski)

Engine Type	Power Output (kW)	Waste Heat
Small air cooled diesel engine	35	30-40 % of energy waste loss from IC engines
Water air cooled engine	35-150	
Earth moving machineries	520-720	
Marine applications	150-220	
Trucks and road engines	220	

30 to 40% is converted into useful mechanical work; the remaining heat is expelled to the environment through exhaust gases and engine cooling systems, resulting in to entropy rise and serious environmental pollution, so it is required to utilized waste heat into useful work. The recovery and utilization of waste heat not only conserves fuel (fossil fuel) but also reduces the amount of waste heat and greenhouse gases damped to environment. The study shows the availability and possibility of waste heat from internal combustion engine, also describe loss of exhaust gas energy of an internal combustion engine. Possible methods to recover the waste heat from internal combustion engine and performance and emissions of the internal combustion engine. Waste heat recovery system is the best way to recover waste heat and saving the fuel. Modern research and development efforts relating to combustion engines and vehicle design are largely driven by the pressing need to reduce the global consumption of fossil energy carriers and the resulting emissions of the greenhouse gas carbon dioxide. The limited supply of fossil fuels is one of the most important factors underpinning these efforts. Oil and natural gas are currently the most important energy carriers used in transportation, with oil accounting for 93% of the energy used in this sector in 2013. At current rates of production, the world's proven oil reserves will expire in approximately 52 years, while the prognosis for natural gas is 54 years. However, an even more important factor is that as production rates start to decline, the limited supply of fossil fuels will become increasingly problematic. Global oil production is expected to peak before 2030 and may do so before 2020. A second, possibly even more important, factor underpinning the desire to develop energy efficient vehicles relates to emissions of greenhouse gases. The combustion of fossil fuels generates CO_2 emissions, which absorb re-radiated heat from the earth's surface and thereby contribute to global warming. This anthropogenic greenhouse effect alters natural marine and terrestrial carbon cycles, reducing the environment's capacity for CO_2 storage. In the year 2005, the transport sector was responsible for slightly more than 23% of the world's CO_2 emissions, with road transport accounting for 17%. The largest share of the globe's CO_2 emissions (45%) originated from fossil fuels burned for energy generation. Overall CO_2 emissions have increased by 80% since 1970 (and those from the transportation sector have increased by more than 100%), contributing to an average atmospheric temperature increase of around 0.8 °C Cover the same period. While this may sounds small in absolute terms, the long term effects of this trend are predicted to be devastating for life on earth. Diesel emissions and control are still very much in the forefront. Interest in the diesel powertrain for LD applications is continuing, and may be increasing as a result of tightening vehicular CO_2 regulations. The combination of criteria pollutant and efficiency mandates will push diesel technologies in both sectors. The non-road market is implementing technologies to meet new 2011-12 emissions tightening, and technologies are moving into development for the 2014 step. Large locomotive and marine engines are also coming under emissions pressure (but will not specifically be covered here). As a result, a lot of research has been devoted to increasing combustion engine efficiency by reducing these losses. This can be done in various ways, including reducing losses due to mechanical friction, optimizing the engine to increase the efficiency of the combustion process, and creating superior gas exchange paths

Hydrocarbon and CarbonMonoxide Control Diesel oxidation catalysts (DOC) have been applied to engines for more than 20 years, yet we are still improving them and learning fundamentals. They serve two primary purposes to oxidize hydrocarbons (HC) and CO that is innate in the exhaust or added to provide fuel for regenerating a DPF, and to generate NO_2, which is used to oxidize soot on a continuous basis or for improving the low temperature performance of SCR catalysts. On the latter point, Spurk, investigated NO_2 coming from a catalyzed DPF for use in a downstream SCR system. Surprisingly, they found the NO_2 coming out of the DOC and going into the DPF is not as important as the HCs coming from the DOC. Essentially, the HCs going into the DPF can interfere with the NO_2 formation in the DPF. The Pt/Pd ratio is much more important to NO_2 formation than precious metal loading on the DPF.

3. Turbulence Promoters

Heat transfer enhancement techniques improve the heat transfer coefficient which brings about a reduction in the heat transfer area or the increase of the heat transfer capacity. Heat transfer enhancement techniques are classified in two main groups: active and passive. Active methods require external power, for instance, fluid suction or injection, electrostatic fields, surface fluid vibrations, etc. Passive methods consist in the modification of the heat transfer surface of the system. Examples of these are the heat transfer inserts that generate turbulence and produce eddies and vortices. The main feature of such devices is that they reduce the laminar boundary layer next to the walls which is the major resistance to heat transfer inside a tube. This technology is considered a key heat transfer enhancement due to their high performance, low cost, ease of construction, simple installation and removal for cleaning. They can be made on almost any material of construction such as aluminum, copper, carbon steel and stainless steel, among others.

Performance Comparison Methods : One of the most widely used performance parameters used to measure the improvement of heat transfer is the Thermal Enhancement Factor (η). The term assumes that the pumping power between the bare tube and the tube with inserts are the same.This is expressed as

$$(V) \cdot (\Delta P)\, p = [V \cdot \Delta P]\, s \qquad (1)$$

$$\left(f \cdot Re^3\right)_p = \left(f \cdot \mathrm{Re}^3\right)_s$$

Or

$$\left(Re_s\right) = \mathrm{Re}_p \left(\frac{f_p}{f_s}\right)^{\frac{1}{3}}$$

$$\eta = \frac{h_p}{h_s}\bigg|_{pp} = \frac{\left(\frac{Nu_p}{Nu_s}\right)}{\left(\frac{fp}{fs}\right)^{\frac{1}{3}}}$$

The relationship between the friction factor and the Reynolds number is given by:

The Thermal Improvement Factor (η) is defined as the ratio between the heat transfer coefficient, hp, of the tube with the insert and that of the bare tube hs. It is important to mention that all turbulence promoters reduce their η as Re increases, except for the triple helical tape insert [10] where the factor increases as Re does.

Selection Guide

The selection of the suitable turbulence promoter is important to bear in mind the following aspects:

- The values of η must be larger than unity. Above this value, more heat can be recovered for the same geometry and pumping power.
- ii. The Reynolds number range where the insert is to be used must be of the same range. Otherwise there is no guarantee that the rate of heat transfer increment is maintained.
- iii. Operating pressures must not exceed the limits established by the materials of construction.
- iv. It is important to stick to these selection guides in order to maximize the benefit of the application.

4. **Exhaust Gas Temperature (EGT):**WHR system requires waste recovery equipment to recover heat from the streams and transform it into a useful form for utilization. This is done using energy conversion devices. Over the past two decades, much research has been directed towards this. S N Srinivasaet. al.,(2012) have attempted to explore the various possibilities of waste heat energy recovery methods in conventional commercial two wheeler and four wheelers. In this context, a new concept of hybrid engine has also been discussed. The heat energy contained in the exhaust gases are recovered in three different methodologies.

- Firstly, the waste heat energy is utilized to burn an additional amount of fuel.
- The second stage, a thermoelectric generator producing electrical energy by utilizing the heat of exhaust gases.
- The third stage energy recovery is done by coupling a compressor and an alternator.

Kruiswyk (2008) developed components, technologies, and methods to recover energy lost in the exhaust processes of an internal combustion engine and utilize that energy to improve engine thermal efficiency by 10%. Saidu et al. (2012) have studied the different technologies to recover the heat wasted from the exhaust gas of IC engines and concluded that there is a huge potential for extractingthe waste heat from the exhaust gas of IC engines. ShekhNisar Hossain Rubaiyat(2010) conducted experiments to measure the exhaust waste heat available from a 60 kW automobile engine and a computer simulation was carried out to improve the design of the heat exchanger. Two heat exchangers were used: one to generate saturated and the other to generate super heated vapor. It is found that with the exhaust heat available from the diesel engine at least 18% additional power can be achieved.

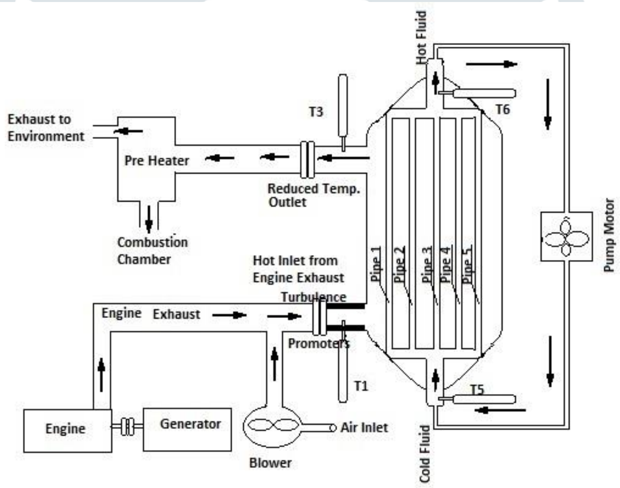
5. Heat Transfer through Heat Exchanger:

a. Counter Flow Heat Exchanger

The engine tests were conducted on single cylinder four-stroke diesel engine test rig, water cooled with a compression ratio of 16.5:1, running at a speed of 1500rpm with 5 HP rated power output. It was directly coupled to a rope brake dynamometer that permitted engine motoring fully or partially. Following is the methodology adopted.

- A simple concentric tube heat exchanger is used in the exhaust line as a preliminary design as shown in fig 1. Here the fuel pipe line is initially passed through the exhaust manifold and then connected to the fuel injector. T1 and T2 are exhaust gas temperatures before and after pre-heating. t1 and t2 are fueltemperatures at inlet and exit of heat exchanger. Engine is made to run at different loads and temperature of both fuel and exhaust gas is recorded. Here, exhaust gas flow rate is left uncontrolled. Injection timing is observed to be at TDC.

- It is observed in above tests that, at 6 kg load, exhaust temperature increased beyond 600°C and fuel temperature crossed 130°C and made difficult to control the engine running. Therefore, it is decided to control the exhaust gas flow rate and modify the heat exchanger type from simple concentric type to shell and tube type. Fig 2 below shows modified experimental setup. Here heat exchanger is not fixed directly in exhaust gas line. Two valves are used and both are regulated in combination to control exhaust gas flow rate so that preheat temperature is properly controlled. Initially parallel flow arrangement is used where both exhaust gas and fuel flow in same directions. Here the fuel injection timing is kept at 23°C BTDC which is changed from earlier timing i.e. exactly at TDC, to know the effect of injection timing on preheat temperature.



b. Heat Exchanger Design:

In this experiment, a shell and tube heat exchanger design is used. Shell is a tube connected to exhaust pipe of same diameter. So it is important here to design the tube length and number of turns for the tube. It is done as follows.

We know that,

heat transfer through heat exchanger is given by

$$Q = U\,A\,(LMTD) \tag{6}$$

Where,

Q = Heat exchanged, W
U = Overall heat transfer coefficient W/m2K
A = Area (m2)
LMTD = Logarithmic Mean Temperature Difference

LMTD for counter flow is given below,

$$LMTD = \{[(T_1-t_2) - (T_2-t_1)]/ \ln((T_1-t_2)/ (T_2-t_1))\} \tag{7}$$

Then area A required for heat transfer can be calculated by

$$A = \pi\,d\,L. \tag{8}$$

Where,

d = outer diameter of tube, m
L = Length of the tube.

To determine Number of Turns required, the formula used is,

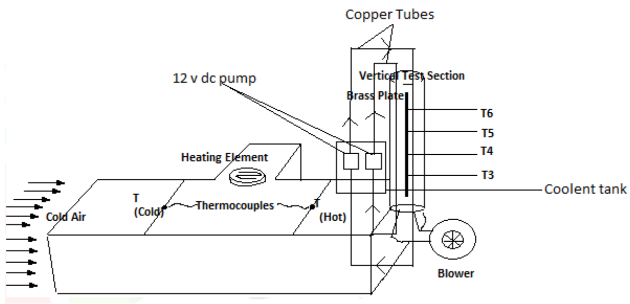
$$N = \frac{L}{\sqrt{(2\pi r)^2 + (3r)^2}} \tag{9}$$

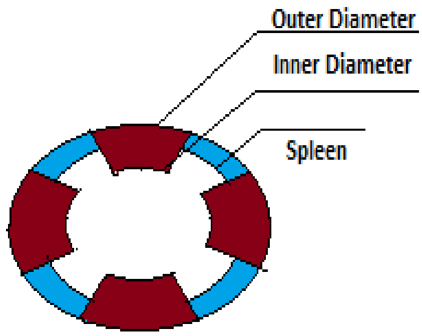
Where,

N = Number of turns L= Length of the tube. r = Radius of the shell.

6. Experimental Setup and Test Equipment[1]:

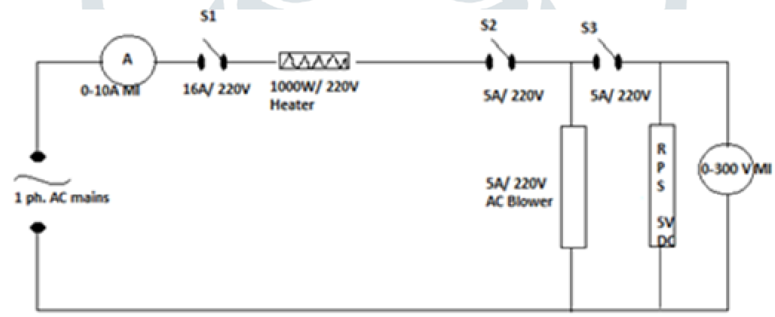
a. Vertical Test Section: An aluminium (AL7075 T6) test section of 60 mm diameter outer diameter and 50 mm inner diameter with internal spleens of 5 mm x 5 mm x 460 mm is fixed vertically to the ground. A heated coil (1000W/ 220V, 1Φ AC) is placed in the duct and heated air is blowed into the vertical test section using 1Φ - 220V/ 0.25 hp AC blower motor. Two thermocouples (k-types) are placed in the duct i.e. one before and other after the heated plate. Thermostat setting is set at 185^0celsius. Brass plate (5 x 5 x 375 mm) is fitted in the test section without contacting the inner walls. Four thermocouples are equally placed along the length of the brass plate equidistantly. All thermocouples are of K-type with visible temperature range between -32^0celsius to $+675^0$celsius.

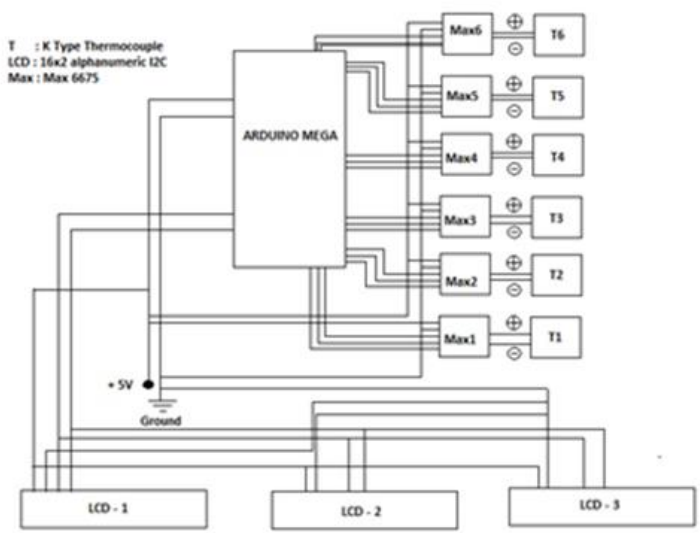




Two 12v DC submersible pumps are used to pump the coolants across the spleen test section. These 2 motors are placed in coolant tank, The test section represents a exhaust chamber of a C.I engine with one flue channel without inner lining of a cylindrical pipe, vertical inner spleen pipe is considered in one stage and a helical grooved cylindrical section is considered in the other. The Helical grooved is coupled with shell and tube cross flow type heat exchanger.

b. Electronic Sketch & Layout:





C. Procedure

1. Hot air is pumped into the test section under buoyant conditions.
2. The sensor data is collected until the steady state is achieved.
3. Dimensionless numbers are computed under varying blower input velocity conditions i.e. 2m/s, 6 m/s, 8 m/s,…., 20 m/s.
4. Then the 12v DC pumps are turned on. The coolant is passed in the test section in such a way that it is opposite to the flow of air. So, it resembles a cross flow heat exchanger.
5. Thermocouple data is collected under varying conditions.

6. For conditions under forced convection i.e. high Reynolds's numbers (turbulent flow); prandtl – Rayleigh numbers are computed.
7. Conventional wisdom predicts that with inlet velocity of more than 6.2 m/s is turbulent and forced convection values can be achieved. For velocity ranging between 3.7 m/s- - 5.6 m/s laminar flow is observed.
8. The experimentation is done in two ways by using two different conditions, the first one is using air and cold water and the second one is steam and cold water. To test the maximum convective heat transfer coefficient in the above two conditions. The cold water is at 5^0c.

7. Results & Discussion

Sl. No	Hot Fluid – Steam			Cold Fluid – Water @ 5^0c				Hot Fluid – Air			Cold Fluid – Water @ 5^0c			
	T_{amb}	T_{ch}	T_3	T_4	T_5	T_6	$T_{avg.}$	T_{amb}	T_{ch}	T_3	T_4	T_5	T_6	$T_{avg.}$
@ Inlet Velocity of 6 m/s														
1.	74	58	50	42.25	40.25	38.5	42.75	53	54	47.75	49	58.5	49	51.06
2.	73	54.25	51	41.75	40.75	38.25	42.93	52.5	52.25	49.25	49	54.75	47	50
3.	72.75	54.25	51	42	40.75	38.25	43	51.5	46.5	48.25	46	53.25	45	48.12
Sl. No	Hot Fluid – Steam			Cold Fluid – Water @ 5^0c				Hot Fluid – Air			Cold Fluid – Water @ 5^0c			
	T_{amb}	T_{ch}	T_3	T_4	T_5	T_6	$T_{avg.}$	T_{amb}	T_{ch}	T_3	T_4	T_5	T_6	$T_{avg.}$
@ Inlet Velocity of 6 m/s														
4.	74	58	50	42.25	40.25	38.5	42.75	53	54	47.75	49	58.5	49	51.06
5.	73	54.25	51	41.75	40.75	38.25	42.93	52.5	52.25	49.25	49	54.75	47	50
6.	72.75	54.25	51	42	40.75	38.25	43	51.5	46.5	48.25	46	53.25	45	48.12
7.	71.25	54	52.5	41.5	41	37.75	43.18	50.5	46	47.75	45.75	52.25	44.25	47.5
8.	71	53	53	41.25	40.5	38.25	43.25	50.25	45	48	45.5	51.75	44.25	47.37
9.	69.5	51.25	53	40.5	40.75	37.75	43	49.25	45.5	47.75	45.25	50.75	43.5	46.81
10.	70.5	52	53	40	40.75	38	42.93	47.75	45.5	48.25	45	50.5	43	46.68

Cross Flow Heat Exchanger with Counter Flow Cold water at 5^0c fluid mass flow rate of 6 lts/ min. Hot Air at varying speed

Exhaust Gases		Hx Fluid		V in m/s	Re	Pr	Nu	h	$cp_{exhaust}$	mass $\rho * A * V$	$Q_{exhaust}$	cp_{fluid}	U	Q_{fluid}
Inlet	Out.	In	Out											
47.7	49.0	5	49	14	15573	0.720	279588	382336	1.005	0.024686	1.09163	1.01	632	13.793
49.2	49.0	5	49	14	15573	0.720	279588	382336	1.005	0.024686	1.09163	1.01	632	8.7598
48.2	46.0	5	46	14	16000	0.723	511928	690846	1.005	0.024686	1.0172	1.01	632	16.375
47.7	45.7	5	46	14	16000	0.723	511928	690846	1.005	0.024686	1.011	1.01	632	15.518
48.0	45.5	5	45	14	16000	0.723	684541	923787	1.005	0.024686	1.0048	1.01	632	17.042
47.7	45.2	5	45	14	16000	0.723	603187	814000	1.005	0.024686	0.99859	1.01	632	16.984
48.2	45.0	5	45	14	16000	0.723	858153	1158078	1.005	0.024686	0.99239	1.01	632	19.184
47.2	45.0	5	45	17	19429	0.723	1032774	1393728	1.005	0.024686	0.9923	1.01	632	15.352
49.7	45.2	5	45	17	19429	0.723	2651217	3577818	1.005	0.024686	0.9985	1.01	632	22.870
50.5	45.2	5	46	20	19429	0.723	3200930	4319654	1.005	0.024307	0.9832	1.01	632	22.153
50.5	46.2	5	46	17	19429	0.723	1962954	2649007	1.005	0.024307	1.0077	1.01	632	23.118
49.7	45.7	5	45	17	19429	0.723	1242272	1676446	1.005	0.024686	1.011	1.01	632	22.276
50.2	45.5	5	46	17	19429	0.723	998000	1346801	1.008	0.024307	0.9923	1.01	632	21.494
51.7	46.2	5	46	20	22857	0.723	479466	1346801	1.008	0.024307	1.0107	1.01	632	24.547
51.5	46.7	5	46	20	22857	0.723	503236	647040	1.008	0.024307	1.0229	1.01	632	23.958

52.0	46.7	5	47	20	22857	0.723	558624	679117	1.008	0.024307	1.0229	1.01	632	24.672
52.7	48.0	5	48	20	22857	0.723	591156	679117	1.008	0.024307	1.0536	1.01	632	22.830
53.0	48.5	5	48	20	22857	0.723	716302	797766	1.008	0.024307	1.0658	1.01	632	23.674
53.5	48.5	5	50	20	22857	0.723	716302	966651	1.008	0.024307	1.0658	1.01	632	19.266
copper tube outer dia = 6 mm; inner dia = 4 mm. length = 36 inches; no. of tubes = 2														

The results obtained during preliminary test and main tests are presented here. Along with this, an estimation of amount of waste heat recovered and the effectiveness is also done for both parallel and counter flow arrangements.

a. Results for Shell and Tube Heat Exchanger

Figure 5 shows temperature variation of preheated fuel with parallel and counter flow arrangement. The preheated fuel temperature at maximum load for parallel flow found to be 73°C and for counter flow is 90°C respectively. From the graph it is clear that counter flow heat exchanger is more effective than parallel.

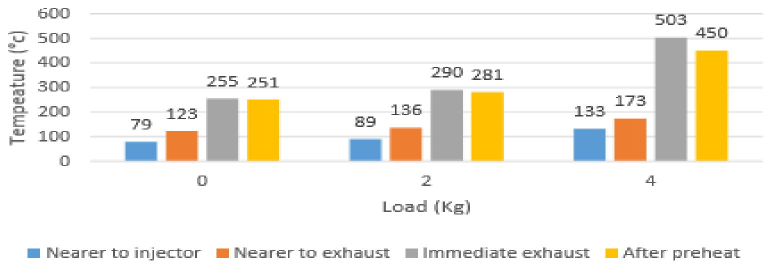


Figure 4. Graph of temperature distribution

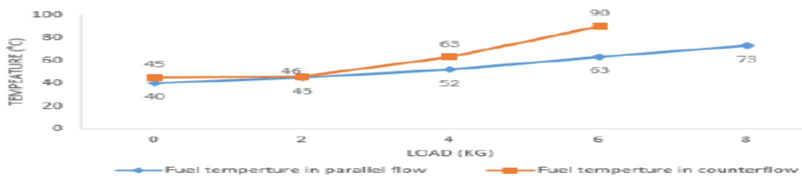


Figure 5. Preheated fuel temperature vs Load

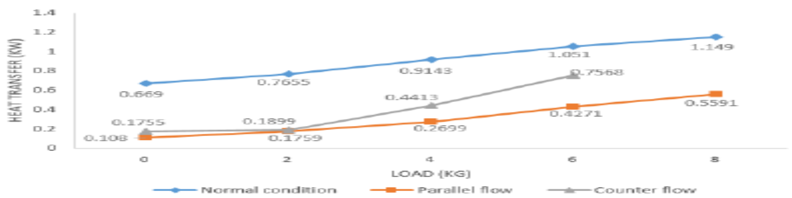


Figure 6. Graph of heat transfer vs load

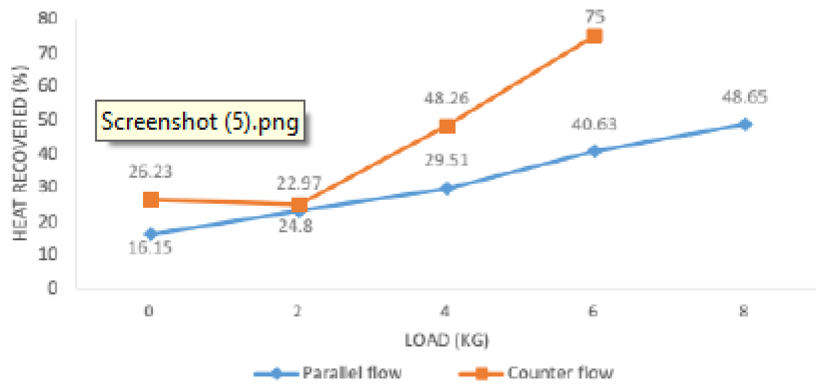


Figure 7. Percentage Heat Recovered vs load

b. Waste Heat Recovered

Since the objective is to determine waste heat recovery from the engine. Waste heat recovered is the ratio of heat absorbed by fuel (Qf) in the heat exchanger to the heat available in the exhaust gas (Qe) at the given load conditions. In figure 6, normal condition indicates heat available in the exhaust gas (Qe) and red lines indicates heat absorbed by the fuel for both parallel and counter flow arrangements. It is observed that at 4kg and 6 kg load heat absorbed by fuel is higher for counter flow heat exchanger. Figure 7 below shows heat recovered in percentages for both parallel and counter flow arrangements. It is found that waste heatrecovered for counter flow is more than that for parallel flow at same loading. This is mainly due to higher fuel temperature achieved in counter flow heat exchanger for the same engine loading than parallel flow type. 75% heat is recovered in counter flow as against 40.63% in parallel flow type for 6 kg load at 1500 rpm.

c. Effectiveness

Effectiveness of heat exchanger is calculated by considering full load conditions. In this case maximum load applied is 6 kg which is 50% of full load. It is calculated by using effectiveness-NTU graph [9] which is based on capacity ratio and temperatures of both hot and cold fluids. It is found that effectiveness for parallel flow is 75% and for counter flow 81%.

Conclusion:

In this work it is found that use of heat exchanger is a useful and simple method to utilize the waste heat energy available in the exhaust gas of diesel engine. Here shell and tube heat exchanger design is used with both counter and parallel flow arrangements. Following conclusion can be drawn from the experimental results:

- Heat Transfer is enhanced with internal spleen pipe arrangements by 35% – 40%.
- Inner walls of Heat Exchanger (Shell) is made of internal spleen geometry owing to its experimental validation of enhancing Heat Transfer, mentioned earlier.
- Fuel preheat temperature depends on flow rate of exhaust gases
- In shell and tube heat exchanger, fuel preheat temperature is from 45°C to 90°C under counter flow conditions for the loads ranging from 0kg to 8kg at 1440rpm.
- Effectiveness of heat exchanger is calculated by considering full load conditions. It is found that effectiveness for counter flow 81%.
- Waste heat recovered at 50% full load condition is found to be 72% under counter flow arrangements.

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