THEORETICAL AND EXPERIMENTAL ANALYSIS OF WASTE HEAT RECOVERY EFFECTIVENESS OF A DIESEL ENGINE

Abstract

The study utilized the exhaust gas from a diesel engine to preheat water in the constructed shell and tube heat exchanger.

The theoretical analysis of the heat exchanger was carried out using the Log Mean Temperature Difference (LMTD) method. The Volumetric flowrate of the water was manipulated using a valve and the resulting output temperature of water leaving the heat exchanger was recorded. Experimentation was carried out to determine the effects of volumetric flow rate on the output temperature and the effectiveness of the heat exchanger. After the test and data analysis, it was discovered that that at flow rate of 3.0 Liter per minute (LPM) the effectiveness of the heat exchanger was peak at 43.34%. The volumetric flow rate of water is inversely proportional to the output temperature of water and it was also established that the effectiveness of the heat exchanger depends on output temperature of and the mass flow rate of the water. Also it was proven that by preheating water before it enters the boiler of the Rankine cycle the efficiency of the cycle increases.

Keywords - *Heat Exchanger, Volumetric flow rate, Output Temperature, Effectiveness.*

1. Introduction

Present day innovative work endeavors relating to combustion engine design are to a great extent driven by the need to decelerate the global consumption of fossil fuels. The increased consumption of fossil fuels results in the increased emission of greenhouse gases which enhance global warming. Presently the demand for fossil fuels is increasing due to the increase in global population and industrialization [1]. Moreover, the amount of fossil fuels remaining on earth is decreasing and will soon run out due to increase in demand.

Fossil fuels are used for powering automobiles, as a fuel source in power plants that generate electricity and many other applications. All types of combustion engines, from heavy duty diesel engines to simple two stroke gasoline engines have one major similarity which is loss of energy from incomplete combustion of fuel and exhaust of gas to the atmosphere and this does not constitute to useful engine output. Heat is lost through heat transfer to the surroundings, the vehicles cooling system and majority of which is lost to the exhaust gas which is a by-product of the combustion reaction to produce mechanical power. Attempts have been made to modify the combustion engine by improving heat transfer, metallurgical enhancements and reducing exhaust temperature however the laws of thermodynamics place a lower limit on the exhaust temperature.

The purpose of this study is to theoretically and experimentally analysed the effectiveness of the waste heat recovers from an internal combustion engine via the exhaust gas. This wasted heat can be utilized to improve the performance of a Rankine cycle.

2. The Internal Combustion Engine

Internal combustion engines find a wide range of application in the productive sectors of the world as they are predominantly used in the manufacturing and the transport sectors for power generation, production and the transportation of goods and services [2].

Engines are devices that produce mechanical power by conversion of another form of energy (usually a fuel). Modern combustion engines consist of a piston fixed on top of a connecting rod which connects the piston to the crankshaft. The piston assembly is fitted inside a cylindrical combustion chamber where an air fuel mixture is sprayed through the intake valve (Figure 1) and it is then compressed by motion of the piston and ignited by either a spark plug or the excessive pressure depending on the fuel type i.e. Gasoline or diesel [3].

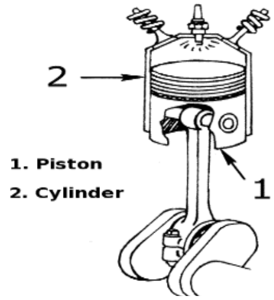
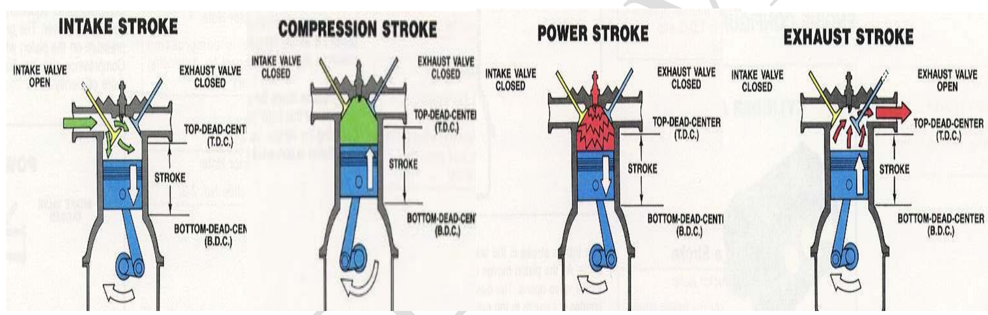


Figure 1: Piston in cylinder assembly [3]

The process of how motion is produced in an engine can be summarized as follows:

a) The piston is at the highest position in the cylinder. The piston moves downward creating a pressure difference while the intake valve opens and inserts an Air – Fuel mixture into the cylinder as shown in Figure 2 (a). This is known as the intake stroke.



(a) (b) (c) (d)

Figure 2: stroke of an Internal Combustion Engine [3]

b) The intake valve is now closed and the piston rises therefore compressing the air-fuel mixture. The piston rings act as seals to ensure there is no leakage and compression can occur until the required pressure is met as shown in Figure 2 (b). This is known as the compression stroke.

c) When the required pressure is obtained the pressurized air-fuel mixture is ignited by a spark plug which provides an exposed electrical spark of very high voltage or by compression where the fuel ignites with excessive pressure. This causes the mixture to rapidly expand and the large increase in volume forces the piston downward as shown in Figure 2 (c). This linear motion is converted to rotary motion via the crankshaft. This is referred to as the power stroke.

d) This piston is forced downward until it reaches maximum displacement. The exhaust valve is opened and the hot gas from the combustion process is forced upward by the piston and it leaves the combustion chamber through the opened exhaust valve as shown in Figure 2 (d).

2.1 The Path of the Exhaust Gas

Exhaust gas for each cylinder is ejected through the cylinder's individual exhaust valve which is located at the top of the cylinder. All of the gas is accumulated in a casted metal unit known as the exhaust manifold (as shown in Figure 3) where it is temporarily stored. From the manifold the hot gas travels through the exhaust pipe where it is discarded to the environment. The proper removal of this hot gas is vital for maximum performance of any internal combustion engine.

Figure 3: Exhaust manifold for a 4-cylinder engine [3]

2.2 The Rankine Cycle

The purpose of the Rankine cycle is to convert mechanical work (done in the turbine) into electricity. It involves the periodic evaporation and condensation of a chosen working fluid. Many impracticalities associated with the Carnot cycle are overcome in the Rankine cycle by changing certain components (example compressor is replaced by a boiler) of the cycle. The Carnot cycle has a greater efficiency than the Rankine cycle but a lower Work ratio than the Rankine cycle [4]. The basic equipment involved in the Rankine cycle is shown in Figure 4.

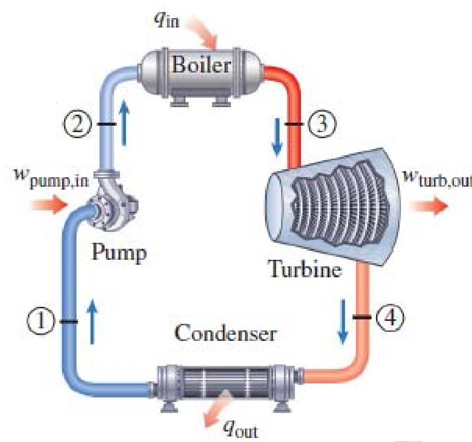


Figure 4: Plant layout for the Rankine cycle [4]

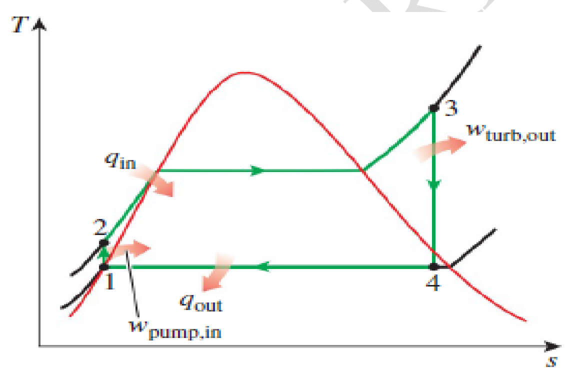


Figure 5: T-s diagram for the Rankine cycle with superheat [4]

Cengel and Boles (2015) stated that the working fluid in the Rankine cycle undergoes the following processes (shown in Figure 5):

1 – 2 Working fluid enters the pump at state 1 as a saturated liquid and is compressed to the operating pressure of the boiler.

2 - 3 Working fluid enters the boiler at state 2 as a saturated liquid and exits at state 3 as a superheated vapor.

3 - 4 The superheated vapor at state 3 enters the turbine at the boiler pressure where it expands and produces work which rotates a shaft that is coupled to an electric generator.

4 -1 The working fluid exits the turbine and enters the condenser where it changes phase and is directed towards the pump to continue the cycle.

2.3 Heat Exchanger

The basic function of a heat exchanger is to transfer heat from one fluid to another. In many heat exchangers, fluids transfer heat through a separating wall or boundary however some heat exchangers exchange heat via direct interaction. Examples of heat exchangers include evaporators, condensers, cooling towers and radiators. Heat exchangers can be classified according to flow arrangements, heat transfer mechanisms and construction features among other criteria [5].

2.3.1 Classification According to Flow Arrangement

Flow arrangement is related to the temperature levels, maximum velocity allowed, effectiveness, thermal stresses, maximum/ minimum pressure [5]. Flow can either be single pass or multi pass.

2.3.1.1 Multi-pass

A fluid is said to have made one full pass if it flows through the entire length of the Heat Exchanger. It makes several passes when the fluid flows through the length of the Heat Exchanger and then switches direction as shown in Figure 6. Examples include multi-pass crossflow heat exchangers, condensers [5].

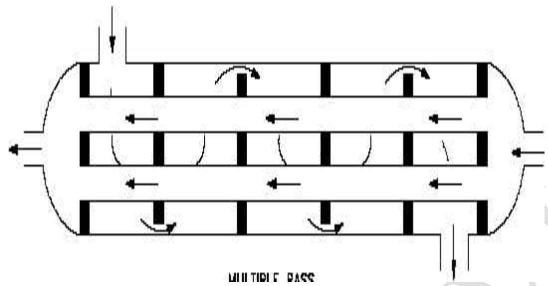


Figure 6: Multi-pass counter- flow Heat Exchanger [5]

2.3.1.2 Single-pass

A heat exchanger is considered a single pass unit if both fluids make one pass in the exchanger and exits as shown in Figure 7.

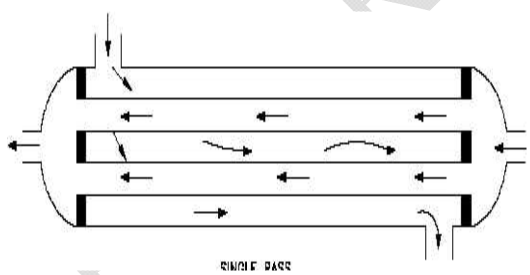


Figure 7: Single pass counter-flow Heat Exchanger [5]

2.3.2 Classification According to Heat Transfer Mechanism

The basic heat transfer mechanisms employed for transfer of thermal energy from the fluid on one side of the exchanger to the separating wall are conduction, convection and radiation.

2.3.3 Classification According to Construction Features

Heat exchangers are most commonly categorized by their construction features. Major construction types include: Tubular, plate type and regenerative exchangers.

2.3.3.1 Plate Type Heat Exchanger

Design includes numerous narrow plates which can be smooth or modified to increase surface area as shown in Figure 8. Typically, they cannot accommodate high temperature and pressures as well as large temperature differences. The plates are sealed around the edges to prevent leaks and heat loss (usually by gaskets) and they are held firmly together in a support which is clamped together by long bolts. Example include plate fin heat exchanger.

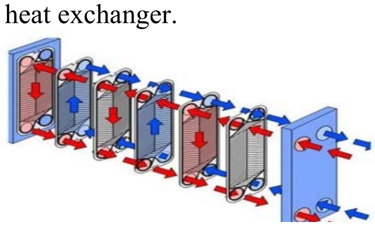


Figure 8: Plate type Heat exchanger [5]

2.3.3.2 Tubular Heat Exchanger

These usually consist of rectangular, circular, elliptical, round or flat tubes. This is a favorable design because the heat exchange surface area is manipulated easily by changing the tube diameter and length of tube [5]. They are functional at high temperatures and pressures however fouling can be major concern. Shell and tube heat exchangers are composed of tubes housed in a hollow cylinder where one fluid flows through the tubes and the other flows across and along the tubes inside the cylinder as shown in Figure 9.

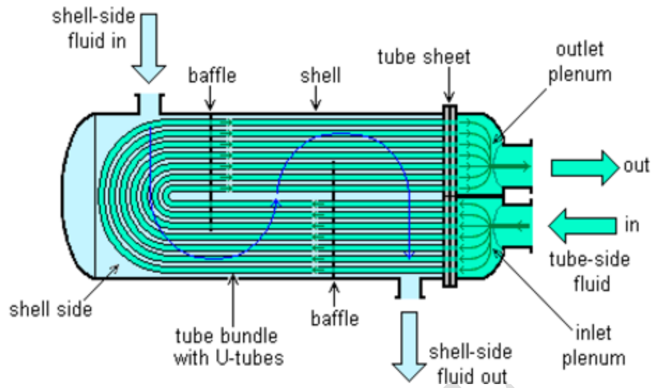


Figure 9: Shell and Tube Heat Exchanger [6]

The Tubular Exchanger Manufacturers Association (TEMA) has standardized a variety of front and rear head types as well as shell types as shown in Figure 10.

Tube layout can be characterized by the included angle between tubes as shown in Figure 11. A tube layout of 30° produces the greatest tube density and hence it is the most commonly used [6]. The number of tubes that can be placed within a shell depends on tube layout, tube outside diameter, pitch size, number of passes and shell diameter as shown in Table 1s.

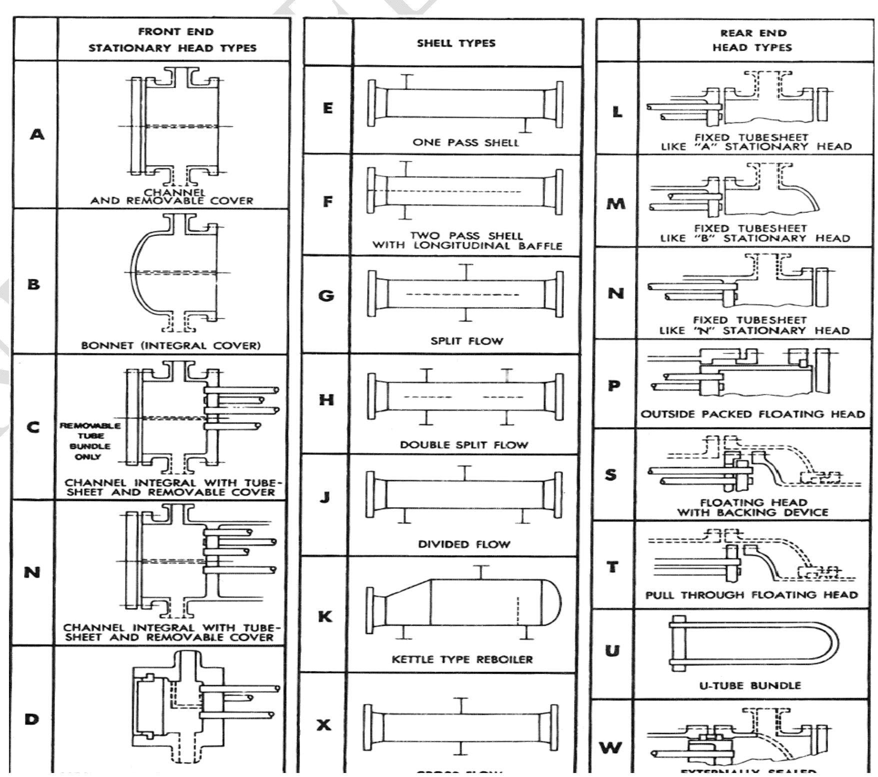


Figure 10: TEMA Standard Shell Types [6]

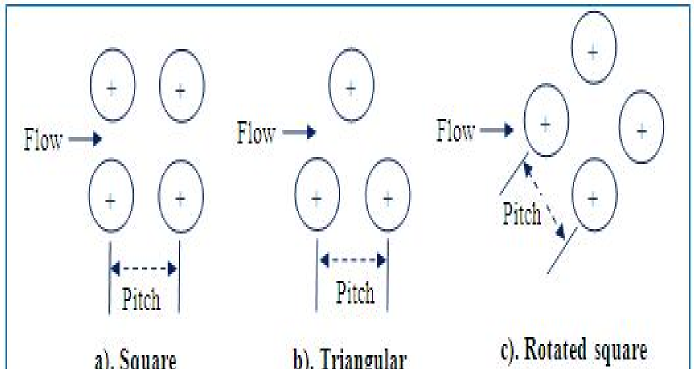


Figure 11: Tube Layout showing angle between tubes [7]

Table 1: Number of Tubes that can be placed within a Shell [6]

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
3/4-in. O.D. Tubes on 1-in. Triangular Pitch					
8	37	30	24	24	
10	61	52	40	36	
12	92	82	76	74	70
13 1/4	109	106	86	82	74
15 1/4	151	138	122	118	110
17 1/4	203	196	178	172	166
19 1/4	262	250	226	216	210
21 1/4	316	302	278	272	260
23 1/4	384	376	352	342	328
25	470	452	422	394	382
27	559	534	488	474	464
29	630	604	556	538	508
31	745	728	678	666	640
33	856	830	774	760	732
35	970	938	882	864	848
37	1074	1044	1012	986	870
39	1206	1176	1128	1100	1078
1-in. O.D. Tubes on 1 1/4-in. Triangular Pitch					
8	21	16	16	14	
10	32	32	26	24	
12	55	52	48	46	44
13 1/4	68	66	58	54	50
15 1/4	91	86	80	74	72
17 1/4	131	118	106	104	94
19 1/4	163	152	140	136	128
21 1/4	199	188	170	164	160
23 1/4	241	232	212	212	202
25	294	282	256	252	242
27	349	334	302	296	286
29	397	376	338	334	316
31	472	454	430	424	400
33	538	522	486	470	454
35	608	592	562	546	532
37	674	664	632	614	598
39	766	736	700	688	672
3/4-in. O.D. Tubes on 1-in. Square Pitch					

3.0 Theoretical Analysis of the Heat Exchanger

The method selected to design (size) the Heat exchanger was the LMTD method because it was the simplest method to vary design parameters to obtain the optimum design.

3.1 Determination of Heat Duty (Q)

Determine Heat duty, which is the amount of energy to be transferred to achieve desired temperature change. This is given by the equation:

$$Q = \dot{m}_w \, c_w \, (T_{C,o} - T_{C,i}) \qquad (1)$$

Where $\dot{m}_w$ = Mass flow rate of water [kg/s]

c_w = Specific heat capacity of Water [kJ/kg-K]

$T_{C,o}$ = Outlet Temperature of Cold Fluid [K]

$T_{C,i}$ = Inlet Temperature of Cold Fluid [K]

The specific heat capacity of water (c_w) is taken to be 4.187 kJ/kg-K [4].

The mass flow rate of water ($\dot{m}_w$) was predetermined to be 0.0333 kg/s

Therefore, heat duty (Q):

$$Q = \dot{m}_w \, c_w \left(T_{C,o} - T_{C,i}\right)$$
$$= (0.0333)(4.187)(80 - 29)$$
$$= 7.11 \; kW$$

3.1.2 Determination of Exhaust Gas Outlet Temperature ($T_{H,O}$)

The theoretical temperature of the exhaust gas exiting the Shell ($T_{H,O}$) can be determined by performing an energy balance on the Heat exchanger. Governing equation is as follows:

$$Q = \dot{m}_w \, c_w \left(T_{C,O} - T_{C,i}\right) = \dot{m}_g \, c_g (T_{H,i} - T_{H,O}) \qquad (2)$$

Where $\dot{m}_w$ = Mass flow rate of water [kg/s]

c_g = Specific heat capacity of Exhaust Gas [kJ/kg-K]

$T_{H,o}$ = Temperature of Exhaust gas at outlet [K]

$T_{H,i}$ = Temperature of Exhaust gas at inlet [K]

The specific heat capacity of the exhaust was approximated to be 1.1 kJ/kg-K [4].

Therefore:

$$T_{H,O} = T_{H,i} - \frac{Q}{\dot{m}_g \, c_g}$$
$$= 180 - \frac{7.11}{(0.09)(1.1)}$$
$$= 108.2 \text{ °C}$$

3.1.3 Determination of LMTD

The Log Mean Temperature Difference (LMTD), is the logarithmic average of the temperature difference between the hot and cold feeds at the end of a Heat Exchanger. For counter-flow heat exchanger the LMTD can be calculate using the equation:

$$T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \qquad (3)$$

Where, $\Delta T_1 = 180 - 29 = 151\text{°C}$

$\Delta T_2 = 108 - 80 = 28 \text{ °C}$

$$T_{LM} = \frac{151 - 28}{\ln \frac{151}{28}}$$

$$= 73\text{°C}$$

3.1.4 Determination of Heat Transfer Coefficients

To determine the Convective Heat Transfer coefficient of the Tube Fluid:

From the Copper Tube Manual shown in Tab
le 2, the standard dimensions of $\frac{3}{8}$" tubes were used to perform the theoretical analysis.

Table 2: Standard Dimensions of Commercial Copper Tube [8]

Nominal or standard size, inches	Nominal dimensions, inches		
	Outside diameter	Inside diameter	Wall thickness
⅜	.500	.450	.025
½	.625	.569	.028
¾	.875	.811	.032
1	1.125	1.055	.035
1¼	1.375	1.291	.042
1½	1.625	1.527	.049
2	2.125	2.009	.058
2½	2.625	2.495	.065
3	3.125	2.981	.072
3½	3.625	3.459	.083

a) The flowrate of the water is known as well as the cross sectional area of the tube, therefore the velocity of the water (v) in the tube can be computed using the equation below.

$$Velocity\ of\ fluid, v = \frac{V_w}{A_i} \tag{4}$$

Where V_w = Volumetric Flow Rate of water [m^3/s]
A_i = Cross-Sectional Area of inner tube [m^2]

The flowrate was predetermined to be 2 LPM which is approximately 0.0000333 m^3/s

Therefore,

$$Velocity\ of\ fluid, v = \frac{V_w}{A_i} \tag{5}$$

$$= \frac{0.0000333}{\frac{\pi}{4}(0.01143)^2}$$

$$= 0.3245\ m/s$$

b) Compute Reynold's Number which is a dimensionless number used to determine the type of flow. The Reynold's Number can be calculated using the equation below.

$$Re = \frac{\rho\ v\ d_i}{\mu}$$

Where ρ = Density of water [kg/m^3]
v = Velocity of Water in tube [m/s]
d_i = Inner diameter of tube [m]
μ = Dynamic Viscosity of water [Pa s]

The viscosity of water is taken to be 0.00089 Pa s [9].
The density of water is taken to be 1000 kg/m^3 [4].

Therefore:

$$Re = \frac{\rho\ v\ d_i}{\mu} \tag{6}$$

$$= \frac{(1000)(0.3245)(0.01143)}{0.00089}$$

$$= 4167.455$$

$Re > 4000$ hence flow is turbulent

c) Compute the Prandtl Number, which is a dimensionless number approximating the ratio of momentum diffusivity to thermal diffusivity. The Prandtl Number can be computed using the equation below.

$$\mathrm{Pr} = \frac{c_w\,\mu}{k} \tag{7}$$

Where c_w = Specific heat capacity of Water [kJ/kg-K]
μ = Dynamic Viscosity of water [Pa s]
k = Thermal Conductivity of water [W/mK]

The thermal conductivity of water is taken to be 0.608 W/mK [9]

Therefore:

$$\mathrm{Pr} = \frac{c_w\,\mu}{k} \tag{8}$$

$$= \frac{(4187)(0.00089)}{0.608}$$

$$= 6.129$$

d) Calculate Nusselt Number which is the ratio of convective to conductive heat transfer across (normal to) the boundary. The Nusselt Number for a Liquid being heated can be calculated using the equation (9).

$$Nu = (0.023)\,Re^{0.8}\,Pr^{0.4} \tag{9}$$

Where Re = Reynold's Number
Pr = Prandlt Number

$$Nu = (0.023)\,Re^{0.8}\,Pr^{0.4}$$

$$= (0.023)(4167.455)^{0.8}(6.129)^{0.4}$$

$$= 37.375$$

e) Determine Convective Heat Transfer Coefficient of the Tube Fluid (h_i) using the equation below.

$$Nu = \frac{h_i\,d_i}{k} \tag{10}$$

Where Nu = Nusselt Number
d_i = Inner diameter of tube [m]
k = Thermal Conductivity of water [W/mK]
h_i =Convective heat transfer coefficient of fluid in tube [W/m^{2}K]

Therefore,

$$h_i = \frac{Nu\,k}{d_i} \tag{11}$$

$$= \frac{(37.375)(0.608)}{0.01143}$$

$$= 1988.01\ \mathrm{W\,m^{-2}\,K^{-1}}$$

3.1.5 Determination of Overall Heat Transfer Coefficient

The Overall Heat Transfer Coefficient for an un-finned tubular heat exchanger can be determined using the equation below.

$$U_o = \frac{1}{\frac{r_o}{r_i}\frac{1}{h_i} + \frac{r_o}{r_i}F_i + \frac{r_o \ln\frac{r_o}{r_i}}{k} + \frac{1}{h_o}} \tag{12}$$

Where r_o = Outer radius of tube [m]

r_i = Inner radius of tube [m]

h_i = Heat Transfer Coefficient of fluid inside tube [W/m^{2}K]

F_i = Fouling factor of fluid inside tube [W/m^{2}K]

h_o= Heat transfer coefficient of fluid outside tube [W/m^{2}K]

The Convective heat transfer coefficient of the exhaust gas (h_o) is approximated to be 200 W/m^{2}K [6]

The Fouling factor for water (F_i) was taken as 0.0001 m^2/ WK [6]

The thermal conductivity of the Copper tube (*k*) is taken as 401 W/mK [9]

Therefore:

$$U_o = \frac{1}{\frac{r_o}{r_i}\frac{1}{h_i} + \frac{r_o}{r_i}F_i + \frac{r_o \ln\frac{r_o}{r_i}}{k} + \frac{1}{h_o}} \quad (13)$$

$$= \frac{1}{\frac{0.00635}{0.005715}\frac{1}{1988.01} + \frac{0.00635}{0.005715}(0.0001) + \frac{0.00635 \ln\frac{0.00635}{0.005715}}{401} + \frac{1}{200}}$$

$$= 176.31 \text{ W/m}^2\text{K}$$

3.1.6 Determination of Surface Area Required

Calculate the outside surface area required for heat addition. This can be done using Fourier's equation as shown below (assuming Friction factor of unity)

$$Q = U_o A_o \Delta T_{LM} \quad (14)$$

Where U_o = Overall Heat Transfer Coefficient [W/m^{2}K]

ΔT_{LM} = Log Mean Temperature Difference [℃]

A_o= Outer Area of tube [m^2]

Q = Heat duty [W]

Therefore:

$$A_o = \frac{Q}{U_o \Delta T_{LM}}$$

$$= \frac{7110}{(176.31)(73)}$$

= 0.5524 m^2

3.1.7 Determination of Tube Length

Equation to determine the length of tube required

$$L = \frac{A_o}{d_o \pi} \quad (15)$$

Where L= Length of tube required [m]

A_o= Outer Area of tube [m^2]

d_o= Outer diameter of tube [m]

Therefore:

$$L = \frac{A_o}{d_o \pi}$$

$$= \frac{0.5524}{\pi (0.0127)}$$

$$= 13.85\ m$$

Using Standard Value *L*= 14m

Number of tube passes (revolutions) = $\frac{Length\ of\ tube}{Circumference\ of\ one\ revolution}$

$= \frac{14}{0.4\pi}$

= 11.14 ≡ 12 tube passes (revolutions)

Number of shell passes = 1

Shell diameter should be selected in such a manner to give a close fit to the tubes. The clearance between the Inner Shell wall and the Tubes depends on the length of tube, number of passes and the tube diameter. Shells are usually made of Industrial Pipes of standard diameters [10].

Since the diameter of the tube coils equals 0.4m, the Shell diameter was taken to be 0.5m leaving a clearance of 0.05m between the tube and the inner wall of shell.

3.1.8 Determination of Pressure Drop in Tube

The pressure drop is the difference in pressure when the fluid enters the tube to when the fluid exits the tube. The pressure drop in a counter flow heat exchanger with circular tubes is given by the equation below:

$$\Delta P = f\ \frac{L}{d_i}\ \frac{\rho\, v^2}{2} \tag{16}$$

Where ΔP = the pressure drop [kPa]

f = Darcy Friction factor

L = Length of tube [m]

d_i = Inner diameter of tube [m]

ρ = density of tube fluid [kg/m^3]

v = velocity of fluid in tube [m/s]

Surface roughness of Copper, Ɛ = 0.0000015m [9]

Relative roughness = $\frac{\varepsilon}{d_i}$ = 1.3 ×10 $^{-4}$

From Moody chart shown in Figure 12, the Darcy friction factor was found to be approximately 0.045

Therefore:

$$\Delta P = f\ \frac{L}{d_i}\ \frac{\rho\, v^2}{2} \tag{17}$$

$$= 0.045\ \times \frac{14}{0.01143} \times \frac{(1000)(0.3245)^2}{2}$$

= 2902 Pa

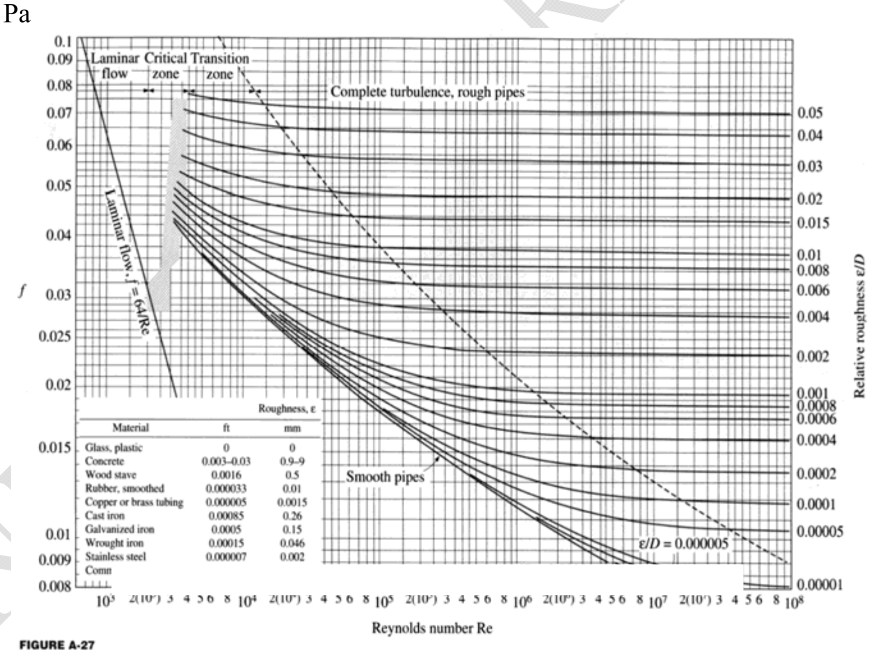


Figure 12: Moody Diagram used to obtain Friction Factor [9]

3.2 Determination of (Theoretical) Effectiveness of Heat Exchanger

The effectiveness of a heat exchanger is the ratio of the actual heat transfer rate to the maximum possible heat transfer rate.

The Heat capacity of both fluids are determined using the Specific Heat capacity and the mass flow rate.

$$C_c = \dot{m}_w\, c_w \tag{18}$$

Where $\dot{m}_w$ = Mass flow rate of water [kg/s]

c_w = Specific heat capacity of Water [kJ/kg-K]

Therefore

$$C_h = \dot{m}_g\, c_g \qquad (19)$$
$$= 0.09 \times 1.1$$
$$= 0.099 \text{ KJ kg}^{-1}\text{ K}^{-1}$$
$$C_H = \dot{m}_g\, c_g$$

Where $\dot{m}_g$ = Mass flow rate of gas [kg s^{-1}]

c_g = Specific heat capacity of Exhaust gas [kJ/kg-K]

Therefore,

$$C_c = \dot{m}_w\, c_w$$
$$= 0.0333 \times 4.187$$
$$= 0.1394 \text{ kJ/kg-K}$$

Since $C_c > C_H$ the effectiveness of the Heat Exchanger can be computed using equation (20):

$$\varepsilon = \frac{C_c\,(T_{C,O} - T_{C,i})}{C_H(T_{H,i} - T_{C,i})} \qquad (20)$$

Where C_w = Heat capacity of Water [kJ/kg-K]

C_H = Heat capacity of Exhaust gas [kJ/kg-K]

$T_{C,O}$ = Temperature of water at outlet [K]

$T_{C,i}$ = Temperature of water at inlet [K]

$T_{H,i}$ = Temperature of Exhaust gas at inlet [K]

Therefore,

$$\varepsilon = \frac{0.1394(80-29)}{0.099(180-29)}$$
$$= 0.4756 = 47.56\%$$

4.0 Methodology

4.1 Experimentation

Necessary parameters will be measured prior to the water entering the Heat Exchanger and when the water is leaving the Heat Exchanger. The path taken by the water can be seen schematically in Figure 13 and the waste heat recovery assembly is shown in Figure 14. In order to achieve different volumetric flowrates of water and record the output temperature of the water, the following steps should be employed:

1) First the engine will be loaded to full load capacity and the exhaust gas will pass through the shell of the heat exchanger until a sufficient duration of time has elapsed.
2) Use the valve to regulate the flow of water such that the desired flowrate is seen on the Variable area rotameter flowmeter.
3) Pressure can be read directly off of the pressure gauge which is placed between the valve and flowmeter
4) In order to achieve different flow rates, the valve should be varied until the next desired flowrate is seen on the Variable area rotameter flowmeter
5) Temperature of water leaving the heat exchanger and Temperature of the exhaust gas leaving the shell would be measured using thermocouple thermometers.

The volumetric flow rate of water ($\dot{v}_w$) was varied in increments of 0.5 Liters per Minute (LPM) since that was the increments present on the scale of the Rotameter. The output temperature of the water was recorded using a thermocouple thermometer for the respective volumetric flow rate and the results are tabulated in Table 3.

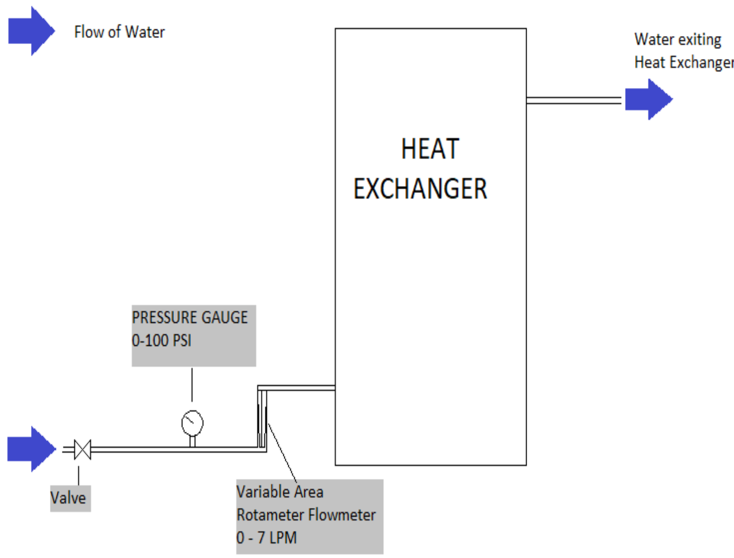


Figure 13: schematic of the path of water flow

Figure 14: Experimental set up - Waste Heat Recovery from Diesel Engine

Table 3: Outlet Water Temperature from Heat Exchanger for selected volumetric flowrate

Volumetric Flow rate of water, $\dot{v}_w$ / LPM	**Outlet Water Temperature from Heat Exchanger, ($T_{C,O}$) / °C**
1.5	74
2.0	72
2.5	66
3.0	60
3.5	52
4.0	48
4.5	43

5.0	41
5.5	37
6.0	33
6.5	30

From the output temperatures at the specific flowrates shown in Table 3, the heat capacity of the water at that flow rate can be calculated and hence the effectiveness of the Heat Exchanger at that specific flow rate was determined using equation (20). The results obtained are tabulated in Table 4.

Table 4: Effectiveness of the Heat Exchanger at the respective volumetric flowrate

$\dot{v}_w$ /LPM	$\dot{m}_w$/kgs^{-1}	$T_{C,O}$/°C	$T_{C,I}$/°C	$T_{H,O}$/°C	$T_{H,i}$/°C	C_c/kJK^{-1}	C_h/kJK^{-1}	Ɛ /%
1.5	0.0250	74	29	97	180	0.105	0.099	31.61
2.0	0.0333	72	29	96	180	0.1394	0.099	40.01
2.5	0.0417	66	29	96	180	0.175	0.099	43.31
3.0	0.050	60	29	97	180	0.209	0.099	43.34
3.5	0.0583	52	29	96	180	0.244	0.099	37.54
4.0	0.067	48	29	96	180	0.281	0.099	35.71
4.5	0.075	43	29	95	180	0.314	0.099	29.41
5.0	0.083	41	29	95	180	0.348	0.099	27.93
5.5	0.0917	37	29	93	180	0.384	0.099	20.55
6.0	0.100	33	29	93	180	0.4187	0.099	11.20
6.5	0.1083	30	29	93	180	0.453	0.099	2.95

Using the various flowrates, the heat duty (Q) required for sensible heat addition (Raising the temperature of the fluid up to the Saturation temperature (T_s)) was determined for the case where the Preheater was not used using equation (21). The results are tabulated in Table 5.

$$Q = \dot{m}_w c_w \left(T_S - T_{C,i}\right) \quad (21)$$

Table 5: Heat Duty required for Sensible Heat addition if the Preheater was not used

$\dot{v}_w$ / **LPM**	$\dot{m}_w$/ **kgs^{-1}**	$T_{C,I}$ / °C	T_S / °C	ΔT / °C	Q /kW
1.5	0.0250	29	100	71	7.43
2.0	0.0333	29	100	71	9.90
2.5	0.0417	29	100	71	12.40
3.0	0.050	29	100	71	14.86
3.5	0.0583	29	100	71	17.33
4.0	0.067	29	100	71	19.92
4.5	0.075	29	100	71	22.30
5.0	0.083	29	100	71	24.67
5.5	0.0917	29	100	71	27.26
6.0	0.100	29	100	71	29.73
6.5	0.1083	29	100	71	32.20

Using the output temperatures at different flowrates, the heat duty (Q) required for Sensible heat addition was calculated for the case where the Water preheater was used. The results are tabulated in Table 6.

Table 6: Heat Duty required for Sensible Heat addition when the Preheater is used

$\dot{v}_w$ / **LPM**	**$\dot{m}_w$/ kgs^{-1}**	$T_{C,I}$ / °C	T_S / °C	ΔT / °C	Q /kW
1.5	0.0250	74	100	26	2.72
2.0	0.0333	72	100	28	3.90
2.5	0.0417	66	100	34	5.94
3.0	0.050	60	100	40	8.374
3.5	0.0583	52	100	48	11.72
4.0	0.067	48	100	52	14.59
4.5	0.075	43	100	57	17.90
5.0	0.083	41	100	59	20.05
5.5	0.0917	37	100	63	24.19
6.0	0.100	33	100	67	28.05
6.5	0.1083	30	100	70	31.74

The following was considered for determining which fluid will flow through the tube and which will flow through the shell:

- The higher fouling fluid flows through the tube.
- The higher pressure fluid flows through the tube.
- The more corrosive fluid must flow through the tube otherwise both the shell and tube would be corroded.
- The stream with the lower heat transfer coefficient flows through the shell side.

Hence the tube fluid was selected to be water and the shell fluid was the exhaust gas. Tube material was selected to be Copper due to its relatively high Thermal Conductivity (k=401Wm^{-1}K^{-1}). Materials with a lower thermal conductivity will require a larger length of tube in order to achieve the desired heat exchange area.

4.2 Results and Discussion

Figure 15 shows a line graph which was used to help visualize the results obtained in Table 3.

From the graph it is seen that a linear relationship exists between the Volumetric flowrate of water and the Output temperature of the water. It is observed that as the volumetric flowrate increases the water output temperature decreases (inversely proportional relationship). This can be explained by the slower the flowrate, the slower the fluid velocity ($Velocity\ of\ fluid, v = \frac{\dot{v}_w}{A_i}$ as $\dot{v}_w$ decreases the velocity of the fluid decreases) hence the water takes a longer time to pass through the tube of fixed length and this allows for greater time for heat transfer to the water.

The maximum flowrate tested was 6.5 LPM and this yielded the lowest output temperature of water which was 30℃. The lowest flowrate tested was 1.5 LPM and this yielded the highest output temperature of the water which was 74℃. However, at this low flow rate the output stream of water was non-uniform due to air pockets being trapped between regions of water. It was concluded that to obtain a uniform stream of water at the output, the volumetric flow rate of water should not be too low**.** Theoretically by reducing the volumetric flow rate of water ($\dot{v}_w$), the mass flow rate ($\dot{m}_w$) will decrease and the heat duty (Q) will decrease therefore less heat will be required to achieve the desired temperature change. This phenomenon is validated by the results of the experiment as it is seen that when the volumetric flow rate of water is decreased the output temperature of water is increased.

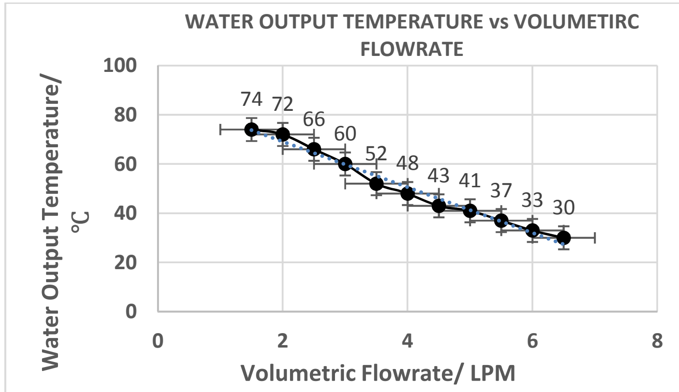


Figure 15: Line graph of Volumetric Flow Rate of water vs Output temperature of water

The flow rate however could not be reduced too much as there are many consequences involved such as;

1) The velocity of the fluid will become too low. From the equation

$$Velocity\ of\ fluid, v = \frac{\dot{v}_w}{A_i}$$

It is seen that the volumetric flow rate of water is directly proportional to the velocity of the water. If the velocity becomes too low the fictional forces within the pipe will become too large to overcome hence the water will not be able to reach the exit of the tube.

2) The gravitational force will be too large to overcome and hence the flow will begin to move downward instead of upward.
3) The exit stream of water may not be a continuous stream as air pockets may develop within the tube.

Figure 16 shows a line graph which was used to help visualize the results obtained in Table 4.

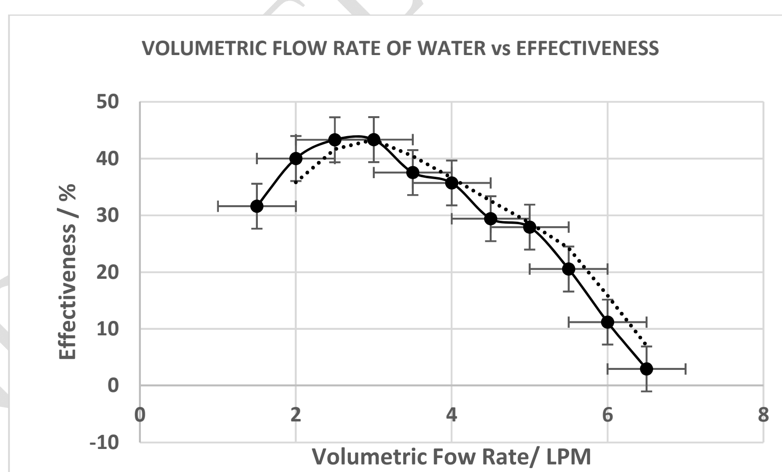


Figure 16: Line graph of Volumetric Flow Rate of water vs Effectiveness of Heat Exchanger

From Table 4 it is seen that the Heat capacity of water (C_c) changed accordingly to the change of mass flow rate of water ($\dot{m}_w$) which depended on the volumetric flow rate of water ($\dot{v}_w$). As the mass flow rate of water increased the heat capacity increased (directly proportional relationship). Heat Capacity (C) can be defined as the heat required to raise the temperature of a substance by one degree or one Kelvin. Exchanger heat transfer effectiveness is the ratio of the actual heat transfer rate in a heat exchanger to the thermodynamically limited maximum possible heat transfer rate.

From the graph it is seen that the volumetric flowrate that yielded the highest effectiveness was 3.0 LPM. From the equation, $C_c = \dot{m}_w\, c_w$, it is seen that the Heat Capacity of the water (C_c) is directly proportional to the mass flow rate of the water (ṁw). Hence as the mass flow rate of water (ṁw) increases, more heat is required to raise the temperature of the water by one degree or one Kelvin. Therefore regarding the constructed Heat Exchanger, the effectiveness (ε) depended on both the Heat capacity of the water (C_c) and the output temperature of the water ($T_{C,O}$) since the Inlet temperature of the Exhaust gas ($T_{H,i}$) and the inlet temperature of the water ($T_{C,i}$) remained constant throughout the experiment. From the equation, $= \frac{C_c\,(T_{C,O} - T_{C,i})}{C_H(T_{H,i} - T_{C,i})}$, the maximum effectiveness (43.34%) was obtained when C_c= 0.209 kJK^{-1} and $T_{C,O}$ = 60 °C.

Upon increasing the flowrate beyond 3.0 LPM the effectiveness of the Heat exchanger began decreasing because the mass flowrate of water continued increasing and the heat being supplied by the exhaust gas remained constant therefore the output temperature of the water would be lower and the effectiveness of the heat exchanger would reduce. The highest flowrate tested was 6.5 LPM and this produced both the lowest output temperature of water and the lowest effectiveness. Factors affecting the effectiveness of the Heat exchanger are:

1) Output temperature of water ($T_{C,O}$)
2) Flow arrangement of the streams i.e. co-flow and counter-flow
3) Volumetric flow rate ($\dot{v}_w$)/ Mass flowrate ($\dot{m}_w$) which affected the Heat Capacity of the water (C_c)
4) Geometry and type of the heat exchanger

The Heat duty required for sensible heat addition of preheated water and non-preheated water is shown in Table 5 and Table 6. A line graph of Heat duty (Q) for preheated water and non-preheated water vs volumetric flow rate of water was plot on the same axes for comparative purposes. The line graph is shown in Figure 17.

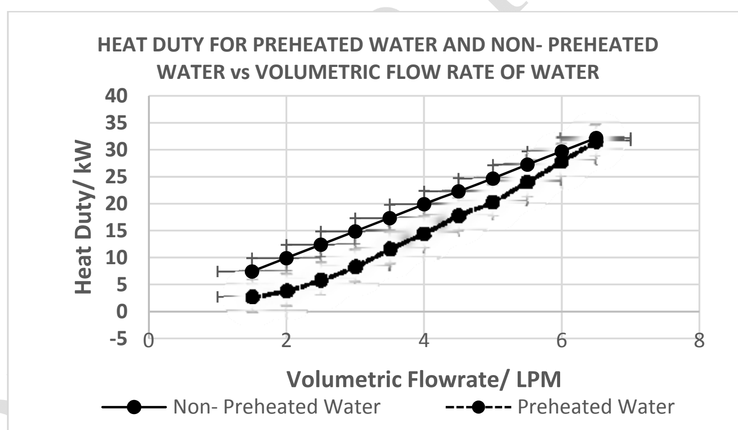


Figure 17: Line graph comparing the Heat duty for sensible heat addition for preheated water and Non- preheated water

In the boiler of the Rankine cycle the water undergoes three heating processes, i.e.

1) Sensible heating of the water i.e. raising the temperature of the water from the inlet temperature to the saturation temperature at the specific pressure.
2) Latent heat addition so that phase change can occur (temperature remains constant at the saturation temperature at this stage and phase change occurs)
3) Heat the vapor to a higher temperature than saturation temperature (superheat)

By preheating the water before entering the Boiler in the Rankine cycle the sensible heat addition needed to raise the temperature to the saturation temperature at that pressure would be reduced as seen in Figure 17. From the graph it is seen that the preheated water required less heat addition to raise the temperature to the saturation temperature. By preheating the water, the fuel demand in the boiler would reduce and hence the efficiency of the Rankine cycle would increase. Further emphasis can be provided using equations as follows:

Without pre-heater: Heat duty, $Q = \dot{m}_w\, c_w(\, T_s - T_C)$

With pre-heater: Heat duty, $Q_p = \dot{m}_w\, c_w(\, T_s - T_{preheat})$

Where Q = Heat duty without preheater

Q_p = Heat duty with Preheater

c_w= Specific heat capacity of water

$\dot{m}_w$= Mass flow rate of water

T_s= Saturation temperature at that specific pressure

T_C= Temperature of the cold water entering boiler

$T_{preheat}$ = Temperature of the water exiting the Preheater and entering the boiler

Given that $\dot{m}_w$, c_w and T_s are the same in both situations then

$\dot{m}_w\, c_w(\, T_s - T_C) > \dot{m}_w\, c_w(\, T_s - T_{preheat})$

Which implies $Q > Q_p$

The mathematical expression above shows that the Heat duty without a water preheater is greater than the heat duty with a water preheater. This implies that without a water preheater more fuel will be required from the boiler to increase the temperature of the working fluid to its saturation temperature at that pressure. The efficiency of the Rankine cycle is determined by the equation given below.

$$\eta = \frac{W_{net}}{Q_{input}}$$

Where W_{net} = Net-work in the cycle

Q_{input} = Heat input

From the equation it is seen that the efficiency of the cycle is inversely proportional to the Heat input (Q_{input}) of the cycle hence as the heat input decreases the efficiency increases. The heat input (Q_{input}) can provide a direct measure of the amount of fuel required by the Boiler and by decreasing Q_{input} the fuel demand of the Boiler decreases. This implies that the water preheater will Increase the efficiency of the Rankine cycle and reduce the operating cost required to run the cycle.

The difference in the actual values obtained from testing and the theoretical values can be explained by the Heat loss of the exhaust gas as it passes through the shell. Even though a thin layer of insulating material was used around the exterior of the shell, it was insufficient and therefore there was still a lot of heat loss to the environment. Other factors such as environmental conditions (weather, wind speed, humidity) and misfire in the engine could also contribute to this temperature difference, however the major cause was insufficient insulating material which led to a lot of heat loss to the environment.

The theoretical value of the effectiveness of the Heat exchanger calculated in section 5.2 was 47.56%, however the actual value obtained upon testing the Heat exchanger was 40.01% as seen in Table 4. The actual value differed from the calculated theoretical value of the Heat Exchanger since the output temperature of the water did not reach 80℃ due to major heat loss to the environment. The output temperature obtained was 72 ℃ which yielded an effectiveness of 47.56% at a volumetric flowrate of 2 LPM.

The outcomes achieved in this project validates that waste heat in exhaust gas from a combustion engine can be utilized to make the Rankine cycle more efficient by preheating the water prior to it entering the boiler.

Conclusion

The Heat Exchanger was sized using the LMTD method whereby the heat duty was found followed by the Log Mean Temperature Difference (LMTD), then the heat transfer coefficients and finally the length of copper tube required to obtain the desired temperature was found. The Heat exchanger was then manufactured and the experimentation phase of the project was conducted. The Volumetric flowrate of the water was manipulated using a valve and the resulting output temperature of water leaving the heat exchanger was recorded. This information was used to determine the effect the volumetric flow rate of water has on the output temperature of the water as well as the effectiveness of the heat exchanger. After testing and analyzing the data it was concluded that the volumetric flow rate of water is inversely proportional to the output temperature of water and it was also established that the effectiveness of the Heat Exchanger depended on output temperature of the water and the mass flow rate of the water and it was found that at 3.0 LPM the effectiveness of the heat exchanger was highest (43.34%). Also it was proven that by preheating water before it enters the boiler of the Rankine cycle the efficiency of the cycle increases.

Ultimately, all the objectives of the project were achieved and the findings from this study validated that the heat from Exhaust gas from a combustion engine can be used to preheat water prior to it entering the Boiler in the Rankine cycle.

Ethical: NA

Consent: NA

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