



(RESEARCH ARTICLE)



Advanced waste heat recovery system based on organic rankine cycle

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Abstract

In this research, Organic Rankine Cycle (ORC) is used to recover heat from exhaust gas of a four-stroke diesel engine. After retrofitting ORC to the engine, Brake power increased from 10473.91 kW to combined – cycle Brake power of 10736.00kW, thermal efficiency increased from 36.01% to combined – cycle thermal efficiency of 51.32% and Exhaust gas temperature decrease from 358°C to 120°C at the exit of the turbocharger. ORC with R12, R22, R134a and R290 as working fluids at saturation and superheated temperatures, pressures and condenser pressures at different ranges were used to compare refrigerants performance in converting low grade exhaust gas waste heat into useful work. This research presents theoretical analysis on four different refrigerants. Applying the above-mentioned refrigerants as working fluid superheated vapour temperature for R12 is 131.72°C, R134a is 129.37°C, R22 is 113.40°C and R290 is 116.95°C. ORC Power generated by turbine gives 94.98kW, 95.56kW. 130.32kW. 262.64kW respectively, ORC Thermal efficiency gives 36%, 29%, 37% and 38% for R12, R22, R134a, and R290 respectively. Combined – cycle power for each of the refrigerant gives 10568.89kW, 10604.23kW, 10569.47kW and 10736.00kW respectively, combined – cycle thermal efficiency for each refrigerant gives 51.14%, 51.18%, 51.14% and 51.32% for R12, R134a, R22 and R290 respectively. R290 offers optimal performance compared to other refrigerants used in this research. The retrofitting of the ORC has saved some supposedly waste exhaust heat energy and has increased both combined cycle power output and thermal efficiency of the engine cycle.

Keywords: Refrigerant; Superheated vapour; Thermal efficiency; Combined power; Brake power; Retrofitting

1 Introduction

Across the globe today, oil price has drastically increased more than its estimation by renown economist. On the other hand, the recent battle by government to reduce or eliminate the release of heat into the environment due to its harmful effect on the ecosystem and its surrounding. This has brought about thorough research by engineers with government assistance to reduce the amount of heat (exhaust gases) released by engines, furnace, and boiler with the aim of improving their efficiency. In engines, the exhaust gases are the “Waste heat” because they are no longer required for use by the engine and it has to be sent-out from the cylinder of an engine so that combustion can take place for more heat to be generated. Waste heat recovery reduces the amount of heat carried by the exhaust gases and thereby reduces global warming.

Waste heat recovery is a method of arresting (capturing) heat energy that is not in practical use again and transporting them to the point where another useful heat can be generated through another medium. This useful heat can be used to create additional heat to the system or used for other purposes such as electrical or mechanical generation. Waste heat can be released at any temperature; it can either be high, medium, or low temperature. But the higher the temperature released, the higher the quality of the waste heat, and the easier the optimization of the waste heat recovery process. It

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is important to utilize the minimum lowest temperature of waste heat in order to obtain higher power from it and increase engine performance and its efficiency (5).

In order to determine the kind of waste heat recovery system (WHRS) to be used, it is important to know the amount and grade of heat recoverable from the process. We have many types of WHRS which can be used in capturing and recovering waste heat.

These systems are heat exchangers, boilers and direct electrical conversion devices. Waste heat temperatures are grouped as follows:

- High temperature 1,200°F and above
- Medium temperature 450°F – 1,200°F
- Low temperature 450°F and below

Low temperature energy is the most difficult energy to be recovered despite being the largest amount of energy released, it constitutes about 60% of waste heat rejected making it very large and available energy source, the data above shows that despite the fact that low temperature waste heat carries less thermal energy. It has larger volume for energy recovery; waste heat can be recovered through the use of boiler for generating useful work (1).

Boiler may be defined as an enclosed vessel by which heat of combustion or heat from external sources is transferred to the liquid through the enclosed vessel thereby causing the liquid to reach its boiling point for the purpose of steam generation or other purposes.

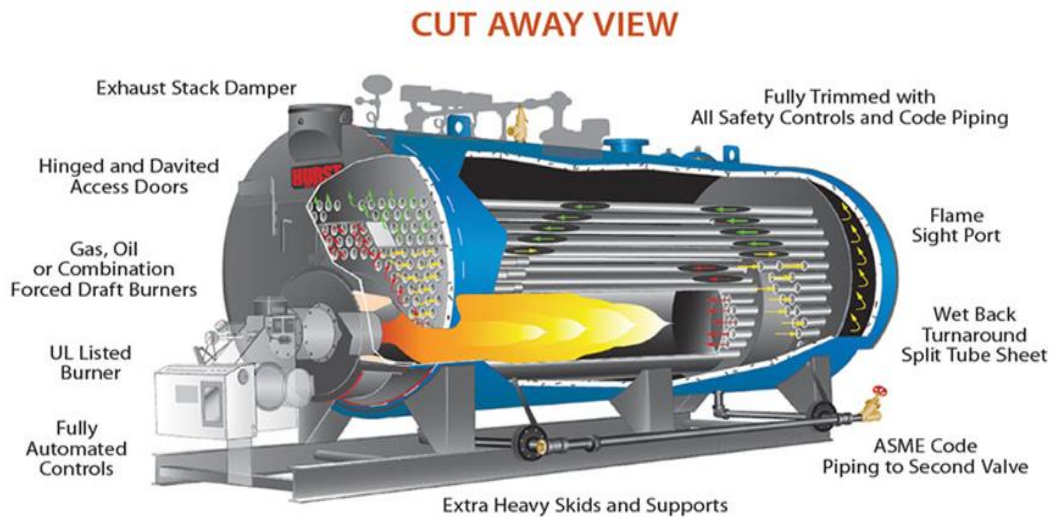


Figure 1 Schematic Diagram of Exhaust Gases Waste Heat Boiler

Waste heat boiler uses liquid tube boiler because of its ability to generate fast steam at high pressure, hot exhaust gases from plant, furnace, incinerator and gas turbine are channel in such a way that it passes over a tube containing liquid thereby transferring heat to the liquid through convection. The liquid absorbs the heat and raise its temperature up to boiling point and get vaporized in the tube thereby generating steam/vapour which is directed (channeled) to steam drum for which it is drawn out for heating purpose or processing steam.

Exhaust gases usually come in low or medium temperature range a more compatible boiler with low boiling point liquid is required in order to conserve space. This kind of boiler can be constructed using finned liquid tube in order to increase effective heat transfer area on the gas side. In the process of trying to recover waste heat, the quality of waste heat produce must be put into consideration in order to determine the type of WHRS to be used.

Due to the quality of waste heat produced from chilled cooling liquid to high temperature from waste gases. The higher the waste heat produced, the greater the heat recovered and greater lost effectiveness. Waste heat is used for different purposes such as preheating of air (charge) for combustion, preheating of boiler feed water and space heating. The current trend is to ensure lowest minimum waste heat to improve system efficiency thereby reducing the cost of its operation.

Boiler is used primarily for steam generation, it is used in all household (i.e all houses across the world) despite the heat source it is used to raise the temperature of liquid from one purpose to the other, domestically, boiler is for boiling water, and cooking etc. The most commonly used household boiler is called “kettle”. In domestic boiler, gas (cooking gas) fire wood, charcoal, electricity etc is used as heat source, while tripod stand is used for supporting the boiler as the case may be.

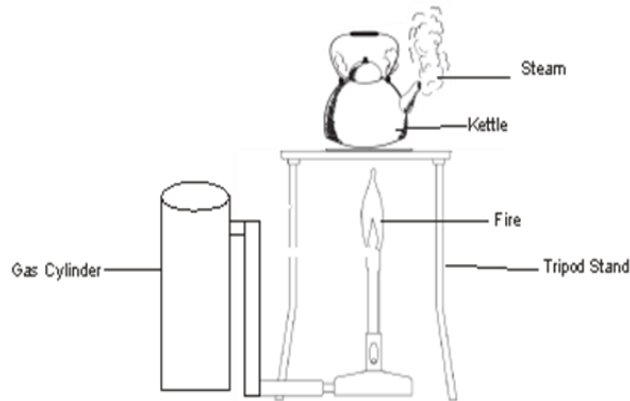


Figure 2 Schematic Diagram of a Domestic Boiler

Domestic boiler uses the following component: tripod stand, heat source (i.e cooking gas, electricity, fire wood etc.) and kettle. Heat is generated through a heating medium and boiler is filled with liquid mostly water. Tripod stand is placed on top of the heat source with the boiler on top of the tripod stand. This help in rising the temperature of the working fluid (liquid) to it saturated temperature (boiling point). When the liquid is boiled, steam is sent out in vapor state.

Boiler efficiency is determined by how much steam is generated by a particular amount of waste heat that is supplied to the boiler, steam quality is measured by the amount of liquid present in the steam that is produced. Apart from increasing system efficiency, WHRS help in reducing environmental pollution which is of great concerned to the government because of it negative effect on parts and animals.

Pollution is defined as the release or injection of the harmful energy or matters into the environment with the ability to cause damage to the ecosystem. Pollutant refers to those substances or form of energy in which when released to the environment (directly or indirectly) causes adverse change on biosphere.

Pollution is classified according to it receiving agent, namely:

- Air pollution
- Water pollution
- Land pollution

Air pollution is defined as the presence of contaminant in the atmosphere in such quality and duration capable of causing injury to human health, animal or plant life. Air pollution is mostly caused by the release of harmful gases like carbon monoxide, nitrogen dioxide and sulphur dioxide into the air by cars, marine engines, factories and companies.

The following are major causes of air pollution:

- Burning of fossil fuels
- Emission from vehicles, marine engines, etc
- Emission from factories

- Mining operations
- Indoor air pollution

Air pollution causes the following effects namely:

- Respiratory and health problems
- Global warming
- Acid rain
- Effect on wild life
- Depletion of ozone layer

Global Warming: This is caused as the result of increase in greenhouse gases such as carbon dioxide, carbon monoxide, methane and other gases present in abundance in the atmosphere. Global warming increases the atmospheric temperature and pressure thereby making survival difficult for living species

Acid Rain: During fossil fuels burning, harmful gases such as nitrogen oxide and sulphur oxide are release into the atmosphere. When rain falls, the water mixed with those dangerous gases and becomes acidic and then fall on the ground in the form of acid rain and pollutes the water bodies. This result in causing damage to humans, animals and plants.

Due to the negative effect of heat release from fuel combustion to the environment from marine engines it is necessary to find a lasting solution to reduce the amount of heat release from marine engine to the environment. To this end, effective use of WHRS as stated in this research work will help in achieving waste heat reduction from marine engines thereby reduces or preventing ozone layer depletion and also reduces the amount of fuel burnt thus reduces amount of greenhouse gases generated by marine engines (9).

Solutions to Prevent Global Warming

From research it has been proven that 80% of heat release to the atmosphere is caused by the burning of fossil fuels from cars, marine engines, factories and companies' engines as the result of releasing unused gases (exhaust gas) into the atmosphere. In order to control this, there must be a means to make use of unused gases or collect unused gases for useful purposes. The unused gases are referred to as "waste heat". Waste heat can be used in generating steam for practical usage, steam is used for various purposes such as mechanical work, generation of electricity and space heating. In order generate steam, advance waste heat recovery system is used to utilize the lowest possible waste heat for useful purpose.

In May 2005 the International Maritime Organization (IMO) set certain rules in preventing heat released to the environment caused by marine operation, and in 2011 IMO. Set ground breaking mandatory technical and operational energy efficiency measures to reduce the amount of heat and greenhouse gases such SO_x, NO_x and CO₂ emission from ships to marine environment, this is contained in annex vi of convention and regulations to prevent or minimize pollution.

1.1 Statement of the Problem

Continuous release of heat from engine into the atmosphere causes ozone layer depletion which is of more concerned to government and other establishments. Apart from environmental degradation cause by marine engines, high cost of fuel has also created serious problem to the economy of the Nation. In order to improve engine efficiency with low fuel consumption due to high cost of fuel, Advance waste Heat Recovery System based on Organic Rankine Cycle will provide solution to the problem mentioned above.

1.2 Solution to the Problem

This research work will bring to bear the effective ways of reducing the amount of heat released by marine engines into the atmosphere by utilizing the exhaust gases released by ship engines to heat waste heat boiler (WHB) to produce useful work thereby preventing environmental degradation and also reduce the amount of fuel consume by the engine.

1.3 Aim and Objectives

The aim of this research is to carry out design analysis of an organic Rankine cycle waste heat recovery system to improve the performance of a marine engine.

Objectives

The Specific objectives of this research are to:

- Identify the marine engine to be retrofitted and determine its performance parameters
- Determine the performance parameters (power and thermal Efficiency) of the Organic Rankine Cycle using various refrigerants.
- Compare performances of ORC of the various Refrigerants to determine the one with the optimal performance.
- Carry out analysis of the effect of superheating the vapour of the ORC.
- Determine the combined cycle (CC) power and thermal efficiency and comparing with the single marine engine cycle (MEC) power and efficiency

1.4 Scope of the study

Advanced waste heat recovery system based on Organic Rankine Cycle involves the study of exhaust gases released by engine and the boiling point of the refrigerants coupled with their physical properties. The exhaust gas is channeled to the boiler which in turn heat the refrigerant inside the boiler, steam is generated in the process and it is used in generating electricity or turning of proportion system for ship movement. This project is delimited to waste heat recovery system using refrigerant.

1.5 Significance of the study

The following benefits will be derived from this research work:

- Improving marine engine efficiency
- Reducing the amount of heat rejected to the environment

It will reduce amount of fuel consumed by marine engine with respect to the combined – cycle power generated.

2 Material and methods

This research focused on advanced waste heat recovery system based on Organic Rankine Cycle. An operational slow speed engine will be used as a case study for this work.

2.1 Thermodynamic Analysis Methods

Organic Rankine Cycle: (ORC) is a thermodynamic cycle that work the same way with rankine cycle but only differed in the working fluid they used. Rankine cycle uses water which has high boiling point and ORC uses high molecular liquid (refrigerant) with boiling point lower than that of RC in order to utilize the lowest possible waste heat available in the system to produce useful energy which couldn't be utilized by RC.

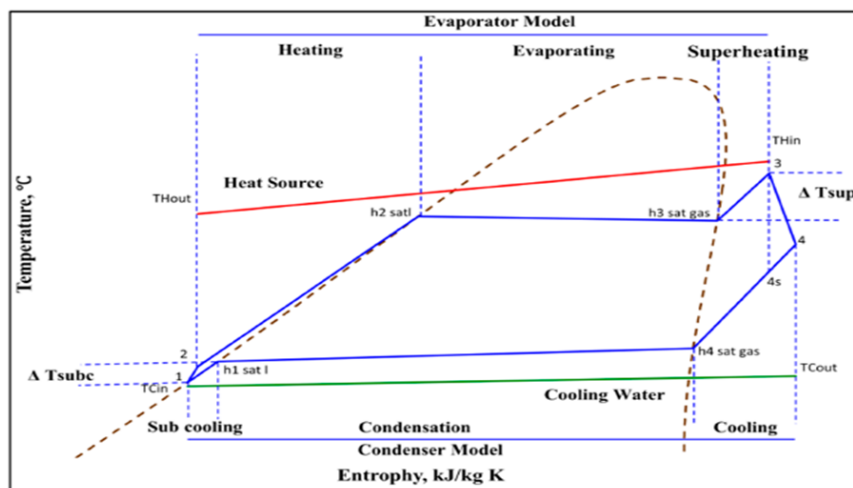


Figure 3 Thermodynamic T – S Diagram for ORC with Superheater

- In process 2 – 3: heat is added to the boiler isobarically.
- In process 3 – 4: Expansion of steam occurs in the turbine adiabatically.
- In process 4 – 1: heat is release in the condenser isobarically
- In process 1 – 2: pumping of refrigerant occurs adiabatically.

The T – S diagram as demonstrated in figure 3 shows thermal cycle of ORC system. The analysis of heat transfer process (thermodynamic process) serves as the main option to find out the cycle performance which has been constructed. Thermal processes used here consist of second law of thermodynamic, mass conservation and energy conservation. This is been analyzed by applying all of them (conservations) to the cycle components which is made up of pump, boiler, turbine and condenser

Four working fluids are used to test run the cycle, at stage 1, the refrigerants are made to flow through the pump at different interval, and they were compressed isentropically to designed boiler pressure. At stage 2, pump work occurred without heat transfer. Energy conservation relating to pump is expressed as follows:

$$W_{pump} = (h_2 - h_1) \tag{1}$$

Where:

h_1 = enthalpy of the pump at entrance

h_2 = enthalpy of the pump at exit

And h_2 and h_1 indicate exit and entrance enthalpy of the pump whose values depend solely on temperature and pressure, while \dot{m} represent mass flow rate of the whole system.

The compressed fluid flowed throughout boiler from stage 2 to stage 3, heat transfer occurred in the boiler and the working fluid get heated. The boiler heat transfer process is calculated using the mathematical expression as shown below.

$$Q_{in} = (h_3 - h_2) \tag{2a}$$

$$Q_{out} = (h_4 - h_1) \tag{2b}$$

Where Q_{in} = heat supplied to the system

Q_{out} = heat released at the condenser

h_3 = enthalpy at exit of evapourator.

h_4 = enthalpy at condenser

It can be further calculated using

$$\Delta T_{boiler} = \frac{\Delta T_{max} - \Delta T_{min}}{\ln \frac{\Delta T_{max}}{\Delta T_{min}}}$$

$$= \frac{(TH_{out} - T_2) - (TH_{in} - T_3)}{\ln \frac{TH_{out} - T_2}{TH_{in} - T_3}}$$

$$U_{in} = \frac{Q_{in}}{\Delta T_{boiler}}$$

U_{in} = heat transfer co – efficiency

ΔT_{boiler} = logarithm mean temperature difference

Pinch point is the difference in the temperature of the working fluid i.e the hot and cold fluid.

$$Pinch = TH_{in} - T_3 \tag{3}$$

TH_{in} = Hot temperature at inlet

T_3 = Temperature at evaporator

For electricity produced, and energy consumed by pump. Back work ratio can be used to represent the fraction between them.

$$BWR = \frac{W_p}{W_t} \quad (4)$$

BWR = Back work ratio

W_p = Work done on Pump

W_t = Work done Turbine

Superheating temperature (ΔT_{sup}) at the discharge side of the boiler (stage 3) has high temperature and pressure, the high vapour flow through turbine which got expanded and produces power output and released to the condenser at (stage 4). the turbine power output is expressed as follow:

$$W_t = (h_3 - h_4) \quad (5)$$

For ideal isentropic expansion from stage 3 – to – stage 4s, the actual expansion is from 3 to 4. Therefore, the turbine isentropic efficiency ($\eta_{is,exp}$) this is expressed as follows:

$$W_{net} = W_t - W_p \quad (6a)$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad (6b)$$

Where W_{net} = Network Output

η_{th} = Thermal Efficiency, (3).

In the above equation, enthalpy values may be substituted from steam table, mollier charts in other to get efficiency value. In ORC, work is done in the turbine and pumps and heat addition and rejection occurs in boiler and condenser respectfully.

In the process where h is not found in the vapour table, below formula was used for interpolation.

$$y = y_1 + (x - x_1) \frac{(y_2 - y_1)}{(x_2 - x_1)} \quad (7)$$

y = linear interpolation work

x = independent variable

x_1, y_1 = values of the function at one point

x_2, y_2 = values of the function at another point.

2.2 Determination of Waste Heat Recovery

In exhaust manifold, the amount of heat that is contained in exhaust gas is a function of both the temperature and mass flow rate of the exhaust gas.

$$\dot{Q} = \dot{m} \times C_p \times \Delta T \quad (8)$$

Where;

\dot{Q} = heat loss (KJ/min)

\dot{m} = exhaust gas mass flow rate (Kg/min)

C_p = specific heat of exhaust gas (Kg/KJ.K)

ΔT = change in temperature (K)

The waste heat is always necessary to be higher than the waste sink temperature. The temperature difference between the waste heat and that of the waste sink is a major determinant of waste heat quality for possible utilization. This difference influenced the rate at which heat is transferred across the waste heat recovery system and also help in obtaining maximum theoretical efficiency of the system for converting heat source energy to another form of energy.

Calculating Heat Loss through Diesel Engine

$$V_r = \frac{V_c + V_s}{V_c} \quad (9)$$

V_r = compression ratio

V_c = clearance volume

V_s = swept volume

$$V_c = \frac{V_s}{V_r - 1}$$

$$s.f.c = \frac{\dot{m}_f}{power}$$

$$\dot{m}_f = s.f.c \times power \quad (10)$$

\dot{m}_f = mass flow rate of fuel

s.f.c = specific fuel capacity

$$\dot{V} = V_s \times N \quad (11)$$

N = speed

\dot{V} = volume rate

$$\eta_v = \frac{\dot{m}_a}{\rho_a \times n \times v_s}$$

$$\dot{m}_a = \eta_v \times \rho_a \times n \times v_s \quad (12)$$

η_v = volumetric efficiency

\dot{m}_a = mass flow rate of air

Mass flow rate of exhaust gas (\dot{m}_E) can be calculated using the equation below

$$\dot{m}_E = \dot{m}_f + \dot{m}_a \quad (13)$$

Heat loss in exhaust

$$Q_E = \dot{m}_E \times c_p \times \Delta T \quad (14)$$

ΔT = change in temperature

c_p = Specific heat capacity

The mass flow rates of exhaust gas rely on the engine size and speed. Hence the larger the engine size and speed, the higher the amount of exhaust gas emitted. Clearance Volume: this is the volume of the cylinder that remains when the piston is at top dead center. Swept volume: this is the piston area multiply by the stroke.

Heat is lost in an engine through the following medium:

- Energy converted to useful work.
- Energy transfer to coolant.
- Energy lost with the exhaust gas (4).

2.3 Selection of Working Fluid

In choosing working fluid for ORC certain conditions has to be considered to ensure smooth running of the cycle with less equipment. The following has to be put into consideration:

- It must be cheap and easily available.
- Need to be non – toxic, non – corrosive and chemically stable.
- The working fluid should have smaller specific heat so that sensible heat supplied could be neglected.
- Saturated vapor line should be steep enough so that state after expansion has high dryness fraction.
- Working fluid should have high density so that the size of the plant becomes smaller.
- Working fluid should show significant decrease in volume upon condensation.
- It should have it critical temperature within metallurgical limits.
- The freezing point of the working fluid must be below atmospheric pressure so that there will be no chance of freezing in condenser (8).

Below are the selected refrigerants chosen for this work as shown in Table 1

Table 1 Refrigerants and their properties

Type	ASHRAE Number	IUPAC Chemical Name	Molecular Formula	ODP	GWP	Boiling Point (°C)	Critical Temperature (°C)	Critical Pressure (Kpa)
CFC	R - 12	Dichlorodifluoromethane	CCl_2F_2	1	10,200	-29.80	111.97	4,136
HFC	R-143a	1,1,1-Trifluoroethane	$\text{C}_2\text{H}_3\text{F}_3$	0	4,800	-47.6	72.89	3,776
HCFC	R-22	Chlorodifluoromethane	CHClF_2	0.05	1,760	-40.70	96.14	4,990
HC	R-290	Propane	C_3H_8 or $\text{CH}_3\text{CH}_2\text{CH}_3$	< 0	3	-42.1±.2	96.7	4,248

Table 2 Refrigerants group and atomic molecules

Prefix	Meaning	Atoms in the Molecule
CFC	Chlorofluorocarbon	Cl, F, C
HCFC	Hydrochlorofluorocarbon	H, Cl, F, C
HFC	Hydrofluorocarbon	H, F, C
H	Hydrocarbon	H, C

2.4 Data Source

The data that is used in this research work are as follows:

Table 3 Pressure and temperature table

Pressure(bar)	Temperature(°C)	Evaporation pressure(bar)
5	65	0.6
6	70	0.7
7	75	0.8
8	80	0.9
9	85	1
10	90	1.1

Table 4a 18 Cylinder MAN 51/60DF ENGINE Data

Parameter	Value
Number of cylinders	18
Cylinder bore	510mm
Piston stroke	600mm
Displacement per Cylinder	122.5lt
Compression ratio	13.3
Engine speed	500rpm
Engine power	17550Kw

Table 4b Calculated data for Diesel Engine

Calculated Conditions for Four Stroke Diesel Engine	
Ambient Temperature T_0 (°C)	15
Ambient Pressure (bar)	1.01
Outlet Temperature @ Turbocharger T_c °C	244
Temperature at the beginning of combustion	1059
Pressure at the beginning of combustion P_2 (bar)	172.9
Temperature at the completed expansion T_3 (°C)	822
Pressure at the end of expansion P_3	10.9
Exhaust gas temperature at the turbocharger. T_e	12

2.4.1 Combined – cycle ORC

the method that is used in advanced waste heat recovery system based on organic rankine cycle depend on the characteristics of the working fluid (refrigerants), waste heat temperature, pressure and the working fluid performance. Engine waste heat characteristics, boiler efficiency and turbine performance are obtained from previous experience by means of asking question and collecting of data from appropriate authorities, to confirm the accuracy and also breach the gap between theory and practical aspect, Rankine cycle is used to carry out the overall efficiency of ORC. Formula for ORC as stated in theoretical principle is applied.

In order to improve ORC for greater system efficiency, superheater need to be Inco-operated in the system Organic Rankine cycle with preheater further modification of Rankine Cycle with superheater:

$$Q + W = dh \tag{15}$$

In boiler

$$Q_1 + W = h_1 - h_4$$

Since there is no work done in the boiler, $W = 0$

Hence;

$$Q_1 = h_1 - h_{f4} \tag{16}$$

Turbine, since there is no heat at adiabatical state,

$$Q_{12} + W_{12} = h_2 - h_1$$

Then $Q = 0$

$$W_{12} = h_1 - h_2 \tag{17a}$$

Or work output

$$-W_{12} = h_2 - h_1 \tag{17b}$$

At Condenser

$$Q_2 + W = h_2 - h_{f3}$$

Since there is no work done in condenser, $W = 0$

$$Q_2 = h_2 - h_{f3} \tag{18a}$$

Heat rejected in condense

$$-Q_{23} = h_{f3} - h_2 \tag{18b}$$

At Pump

$$Q_3 + W_3 = h_{f4} - h_{f3}$$

Isentropic, ($S = S_4$) ($Q_{34} = 0$)

$$W_3 = h_{f3} - h_{f4} \tag{19}$$

Network input for the system

$$\varepsilon W = W_{12} + W_{34}$$

$$\text{i.e } \varepsilon W = (h_2 - h_1) + (h_4 - h_3) \tag{20a}$$

or network output

$$-\mathcal{E}W = (h_1 - h_2) - (h_4 - h_3)$$

If the feed pump is neglected

$$\text{Network output } -\mathcal{E}W = (h_1 - h_2) \quad (20b)$$

Heat supplied in the boiler, from equation

$$Q_{451} = h_1 - h_4$$

$$\text{Organic Rankine Cycle Efficiency } (\eta_{ORC}) = \frac{\text{network output}}{\text{heat supplied in the boiler}}$$

$$\text{i.e, } \eta_{ORC} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

$$\text{or } \eta_{ORC} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_3) - (h_4 - h_3)}$$

$$\text{When the feed pump is neglected, the equation becomes } \eta_{ORC} = \frac{(h_1 - h_2)}{(h_1 - h_3)} \quad (21)$$

When feed pump term is to be included, it is necessary to evaluate the quantity, W_{34}

$$\text{pump work} = W_{34} = h_4 - h_3$$

For incompressible (i.e, $V = \text{constant}$) increase in enthalpy for isentropic, compression is given by

$$(h_4 - h_3) = V(P_4 - P_3)$$

For Reversible adiabatic process

$$dQ = dh - Vdp = 0$$

Therefore

$$dh - Vdp = 0$$

$$\text{i.e } \int_3^4 dh = \int_3^4 Vdp$$

for liquid, since V is approximately constant, we have

$$h_4 - h_3 = V \int_3^4 dp = V (P_4 - P_3), (7).$$

Therefore;

$$\text{pump work input} = (h_4 - h_3) = V(P_4 - P_3) \quad (22)$$

Equation for engine parameter

$$I.P = \frac{nP_{mi}LANK \times 10}{6} \text{kW} \quad (23)$$

Where

I.P = Indicated Power

n = Number of cylinders

P_{mi} = Indicated mean effective pressure

L = Length of stroke (m)

A = Area of piston (m^2)

K = $\frac{1}{2}$ for 4 – stroke engine and 1 for 2 – stroke engine

$$B.P = \frac{2\pi NT_q}{60 \times 1000} \text{KW} \quad (24)$$

Where

B.P = Brake Power

N = Speed in rpm

T_q = Torque (Nm)

$$\text{Mechanical Efficiency} = \frac{B.P}{I.P} \quad (25)$$

Equation for exhaust gas

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{r-1}{r}}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{r-1}{r}} \quad (26)$$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{r-1}{r}}$$

$$T_4 = \frac{T_3}{(r_p)^{\frac{r-1}{r}}} \quad (27)$$

T = Temperature

$$\text{CC Power} = P_E + P_{ORC} \quad (28)$$

$$\text{CC Thermal Efficiency} = \frac{P_E + P_{ORC}}{Q_{E \text{ Input}}} \quad (29)$$

P_E = Power Output of single Engine

P_{ORC} = Power Output at ORC

$Q_{E \text{ input}}$ = Heat input to diesel (2)

3 Results and discussion

3.1 Determining the Performance Parameters of the ORC using various Refrigerants

Four refrigerants were selected and properly examined comparatively to determine the effect of superheated temperature. Meanwhile, ORC operations were kept similar in all cases with increased in temperature, pressure and exhaust gas temperature the characteristics of all the fluid selected are summaries in table 1. under different conditions thermal efficiencies were found to increase from 1 – 2%. Below are tables showing calculated result for four refrigerants selected.

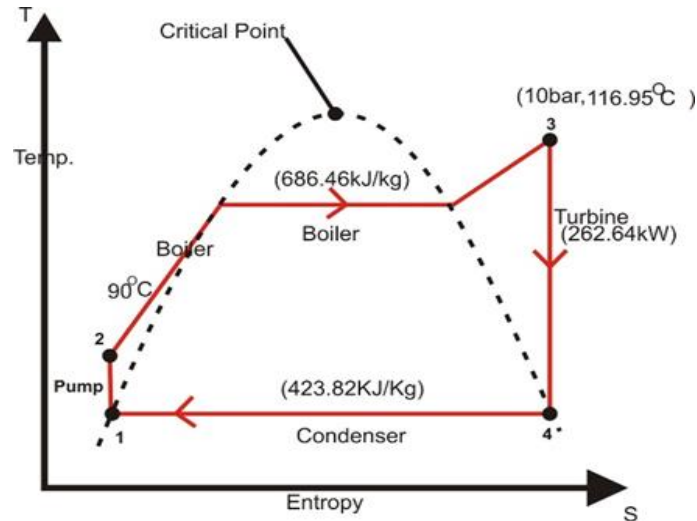


Figure 4 T – S Diagram of Organic Rankine Cycle showing designed point for R290

- In Stage 2 – 3: heat is added to the boiler isobarically.
- In Stage 3 – 4: Expansion of steam occurs in the turbine adiabatically.
- In Stage 4 – 1: heat is release in the condenser isobarically
- In Stage 1 – 2: pumping of refrigerant occurs adiabatically (9)

The following stages shows the working principle of Organic Rankine Cycle and its components, this is used in the calculation process for all refrigerants examined in this research work. Equation 2a, 2b, 6a and 6b were used to calculate for Q_{in} , Q_{out} , W_{net} and η_{ORC} respectively in Table 5, 6, 7 and 8.

Table 5 Result for R12

Pressure (bar)	T ₁ (°C)	T ₂ (°C)	Q _{in} (KJ/Kg)	Q _{out} (KJ/Kg)	Network Output	Thermal Efficiency
5	65	80.65	240.5	170.85	69.65	28
6	70	92.06	244.54	169.54	75	30
7	75	102.72	248.56	168.45	80.11	32
8	80	112.81	252.49	167.3	85.19	33
9	85	122.45	256.43	166.23	90.2	35
10	90	131.72	260.4	165.42	94.98	36

Table 5 shows computational data for R12, this was obtained using it thermodynamic properties,

The data in the table were achieved using vapour table as a bed rock for thermal properties of the refrigerants.

P = Initial Pressure at the evaporator
 T₁ = Initial temperature at the evaporator
 T₂ = superheated temperature leaving superheater
 Q_{in} = Heat supplied to the Evaporator
 Q_{out} = Heat released by the condenser
 W_{net} = Network output
 η_{ORC} = Thermal Efficiency of ORC system

Table 6 Result for R134a

Pressure (bar)	T ₁ (°C)	T ₂ (°C)	Q _{in} (KJ/Kg)	Q _{out} (KJ/Kg)	Network Output	Thermal Efficiency
5	65	80.7	317.64	223.95	93.68	29
6	70	91.55	323.28	222	101.28	31
7	75	101.69	329.97	220.25	109.72	33
8	80	111.31	334.57	218.65	115.92	34
9	85	120.51	340.27	217.16	123.11	36
10	90	129.37	346.14	215.82	130.32	37

R134a results were obtained following the same steps and formula as that of R12 but only differed in terms of thermal properties which shows the uniqueness of each of the refrigerants.

Table 7 Result for R22

Pressure (bar)	T ₁ (°C)	T ₂ (°C)	Q _{in} (KJ/Kg)	Q _{out} (KJ/Kg)	Network Output	Thermal Efficiency
5	65	-	-	-	-	-
6	70	-	-	-	-	-
7	75	-	-	-	-	-
8	80	-	-	-	-	-
9	85	104.59	324.31	233.95	90.36	27
10	90	113.4	328.28	232.71	95.56	29

Initial temperature and pressure are the same for all the refrigerants selected for this work, the critical temperature and pressure were observed which causes difference in each of their results.

Table 8 Result for R290

Pressure (bar)	T ₁ (°C)	T ₂ (°C)	Q _{in} (KJ/Kg)	Q _{out} (KJ/Kg)	Network Output	Thermal Efficiency
5	65	66.74	622.19	437.31	184.88	29
6	70	77.93	633.68	433.6	200.08	31
7	75	88.41	647.48	431.15	216.334	33
8	80	98.33	659.87	428.34	231.526	35
9	85	107.82	672.45	425.7	246.75	36
10	90	116.95	686.46	423.82	262.641	38

Superheated temperature increased alongside with pressure and temperature used.

3.2 Comparing Performances of ORC of the Various Refrigerants

From the solution algorithm, the program calculated different pressure and network output. This help in determination of optimal efficiency and operating condition of each of ORC for certain given data. Figure 5 through 10 shows result of the calculated heat supplied, heat rejected, network output, thermal efficiency against evaporation temperature, superheated temperature and evaporation pressure. Figure 5 shows network output for the four refrigerants chose in this work.

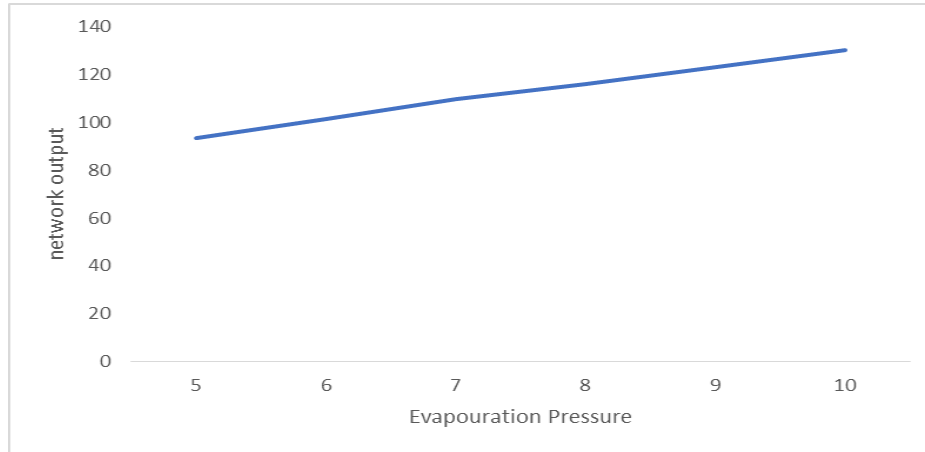


Figure 5 Network Output Vs Evaporation Pressure

Network output increases as the pressure of the cycle increases. In the case of R-22, at low pressure there was no network output because the pressure was at critical stage and could not be accounted for in vapour table. Under the same condition, R12 shows good performance, R134a better performance and R290 demonstrated best performance in terms of network output. Figure 4.3 shows thermal efficiency against evaporation inlet pressure. The least efficiency in both R290 and R134a turn out to be the highest efficiency for R-22 because of its boiling point. These two refrigerants (R134a and R290) show better thermal efficiency in this work.

During computation of data in this work, it was observed that all thermal efficiencies increase as the pressure increases as show in Figure 6.

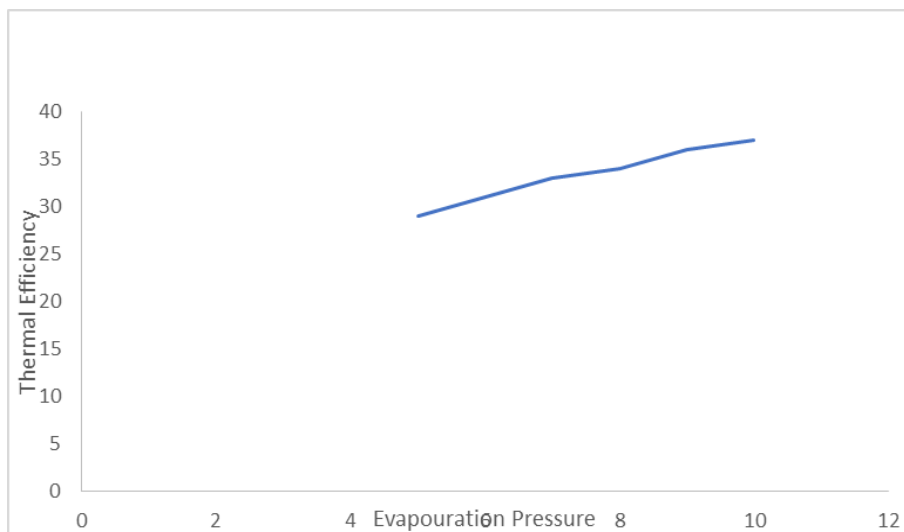


Figure 6 Thermal Efficiency vs Evaporation Pressure

From the calculation, it is shown that increase in temperature lead to more heat supplied in the cycle as it is the case with all the refrigerant used in this work, though, some refrigerants show high heat supplied than others. The highest

heat supplied in R – 12, R134a, R – 22 and R290 are 260.4, 364.14, 328.28 and 686.46 respectively as shown in table 5 through 8 and also figure 7, increased in temperature is cause by increase in load which help in reducing fuel consumption rate in an engine. From 65°C to 90°C as shown in table 5 through 8, the calculation shows higher fuel saving at high engine load.

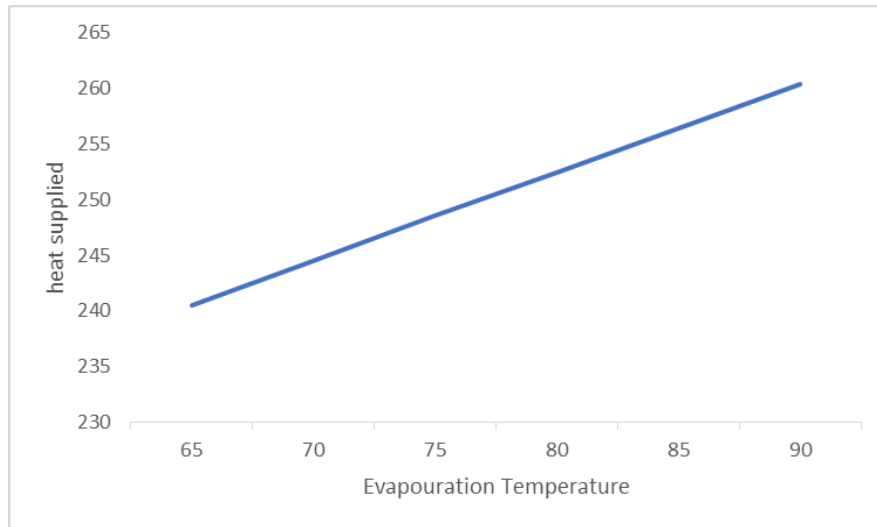


Figure 7 Evaporation heat Vs Evaporation Temperature

It is also observed that the higher the temperature, the smaller the amount of waste heat released as shown in the case of R290 that released smaller heat compared to heat that it received, heat received and heat rejected by each of the refrigerants is as follows: R – 12 received 260.4 and 165.42 is rejected, R134a received 346.14 and 215.82 is rejected, R – 22 received 328.28 and 232.71 is rejected, R290 received 686.46 and 423.82 is rejected as demonstrated in figure 8. therefore, increase in temperature reduces the amount of heat sent – out by the cycle to the environment, this help in reducing ozone layer depletion.

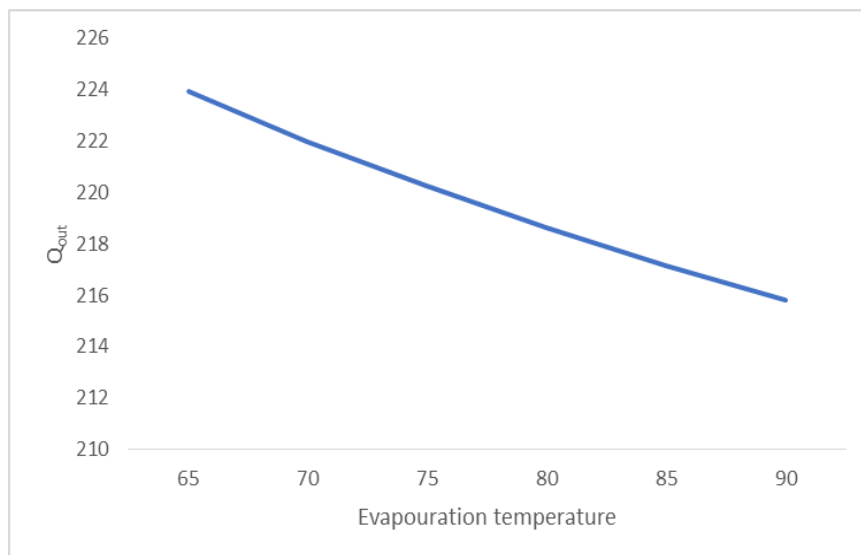


Figure 8 Q_{out} Vs Evaporation Temperature

This further shows that more work is done in the cycle as demonstrated in figure 9 when there is increased in temperature, large amount of heat is used in the cycle and more work is done thereby leading to reduction in fuel consumption and greater efficiency.

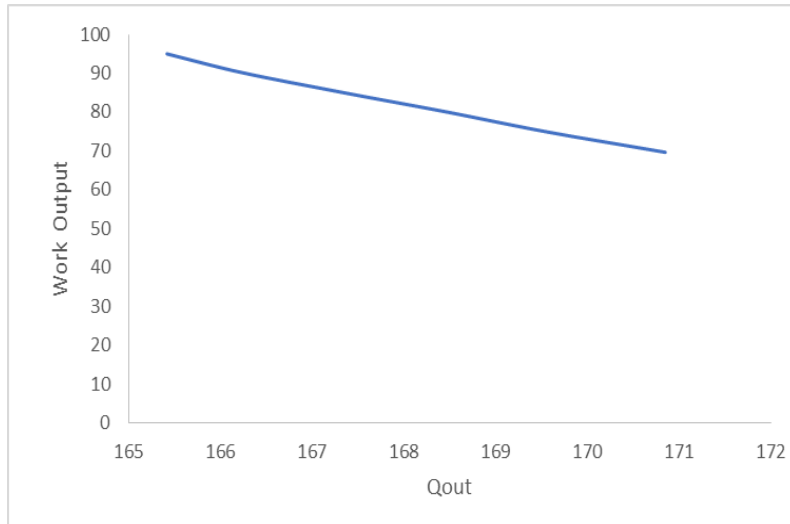


Figure 9 Work Output Vs Qout

Figure 10 shows changes in temperature and superheated temperature, heat transfer co-efficient increases in superheated vapour degree. The superheated temperature as shown in figure 10 inclined gradually throughout the experiment when there is increase in temperature.

This condition was accepted due to the superheated vapour and normal temperature always being in the same position. This result in irreversibility of the cycle. Meanwhile, increased in superheated vapour led to increase in heat transfer co-efficient for working fluids despite difference in their critical temperature and pressure. For working fluid R – 22 it shows stagnated increased in superheated vapour while others show rapid increased in superheated vapour as the temperature increases. This increased in superheated temperature, which is due to high heat being Captured, thus, when superheated temperature is in highest grade, the irreversibility becomes high, which means less wasted heat released to the environment by the cycle.

3.3 Effect of Superheating the Vapour of the ORC

The effect of the ORC exhaust pressure on the efficiency of the cycle is shown for selected refrigerants in Figure 10. Result is given out in superheated cycles in all cases. As demonstrated, increase in exhaust.

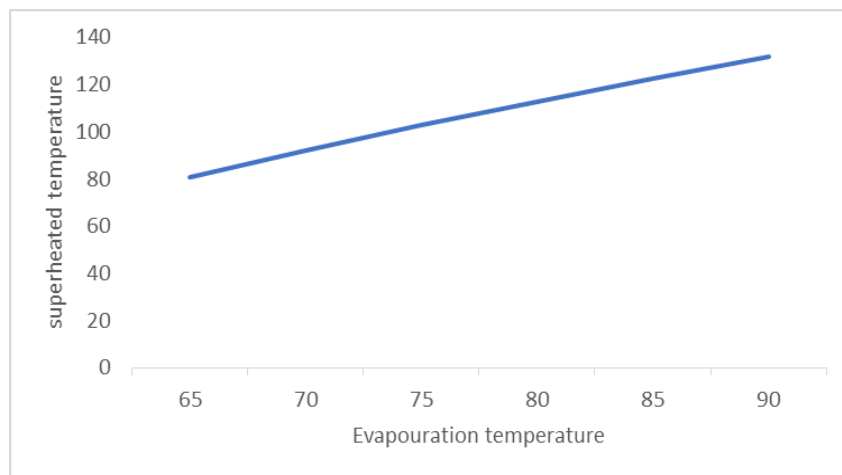


Figure 10 Superheated Temperature Vs Evaporation Temperature

Pressure always led to better cycle efficiencies for all refrigerants chosen in this work. It can be seen that the effect of the exhaust pressure is more dramatic at low pressure, heat released is high as shown in table 5 through 8, each of the refrigerant at 0.6bar exhaust gas as the following heat released R – 12 = 170.85 °C, R134a - 223.95 °C, R290 - 437.31 °C these happened to be the highest throughout the cycle for each of them as compared to (6).

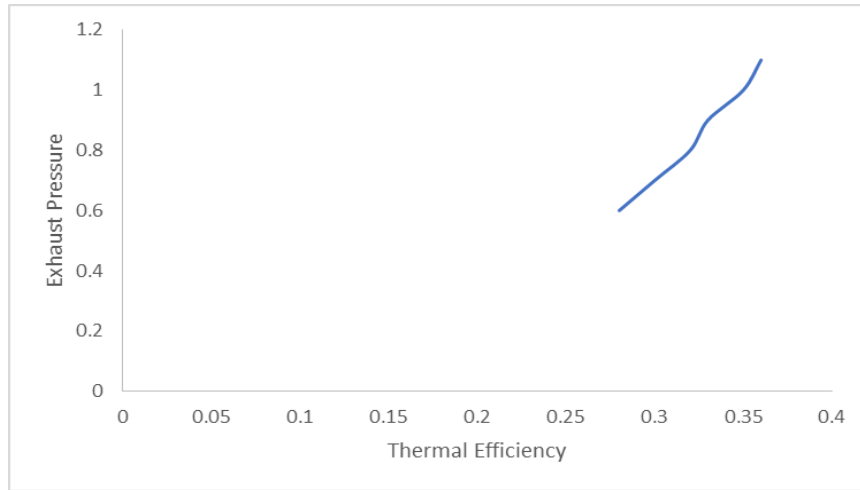


Figure 11 Exhaust Pressure Vs Thermal Efficiency

The most favourable condition of exhaust pressure for a refrigerant is therefore control by its critical pressure and refrigerant with higher critical pressures has greater ability for efficiency improvement. meanwhile, R134a, R-22, R290 and R-12 may further entrench their performance at elevated pressures. To evade the risk of leakages and increased costs for special safety equipment, exhaust pressure should be set at minimum pressure. Finally, Among the selected refrigerants, the ORC cycle of R290 is selected as the most correct refrigerants due to high thermodynamic performance.

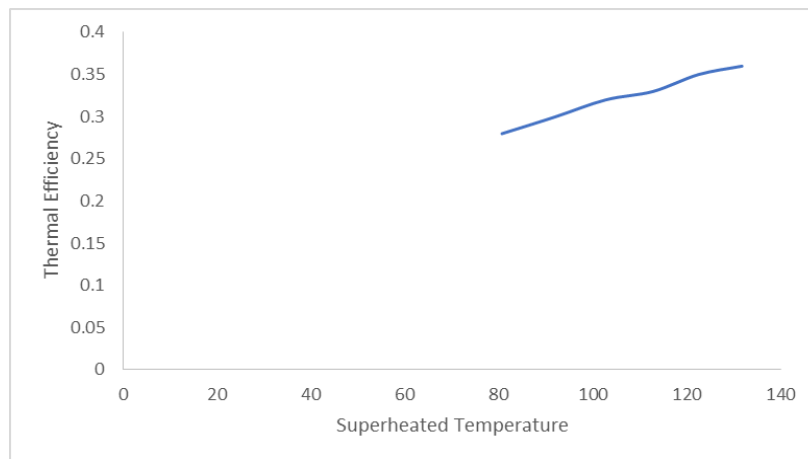


Figure 12 Thermal Efficiency Vs Superheated Temperature

When there is liquid at the final stages of turbine expansion its causes turbine blade corrosion and wear. Due to the damage cause by steam, simple Rankine cycle is inappropriate and superheating of the fluid vapor above the saturation temperature is always needed thereby required huge safety equipment. Effect of superheating temperature is shown on the efficiency improvement of ORC as demonstrated in the case of R12, R22, R134a and R290 as it is in Figure 12. All refrigerants used are at their optimum pressure mentioned before and superheating temperature is increased. As can be seen, an increase in superheating temperature always leads to an overall efficiency improvement.

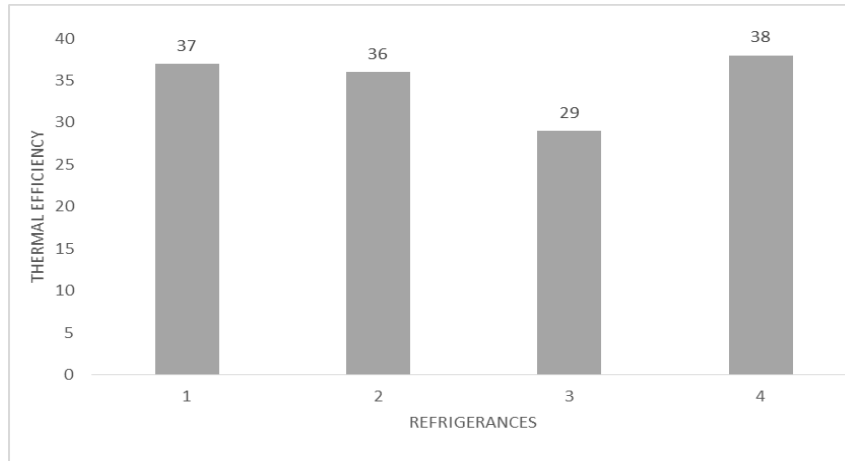


Figure 13 Column Chart showing Highest Range of Thermal Efficiency of each Refrigerant.

From figure 12, R-22 has the lowest thermal efficiency followed by R – 12 which has 36, R134a has 37% and propane R290 top chart with 38% this is because of the difference that appeared in their critical temperature and pressure which affect their boiling point.

3.4 Result of Combined – Cycle Power and Thermal Efficiency

The effect of combined – cycle (CC) brings about improvement in power Output and thermal efficiency of marine engine. CC power is achieved by adding the power of the engine which is 10473.91kW to the power developed by ORC, the single engine thermal efficiency is 39.4% while the CC thermal efficiency is obtained by dividing CC power with the amount of heat input to the diesel engine. All these was obtained by retrofitting ORC to the engine which gives the CC power of each of the refrigerant to be 10568.89kW, 10604.23kW, 10569.47kW and 10736.00kW and CC thermal efficiency of 51.14%, 51.18%, 51.14% and 51.32% for R12, R134a, R22 and R290 as shown in table 8, CC power and CC thermal efficiency. This shows improvement in the diesel engine used. It also helps in reducing amount of fuel consumed by the engine.

Table 9 Retrofitting ORC to single marine engine

Refrigerants	P_E (kW)	P_{ORC} (kW)	CC Power(kW)	CC Thermal Efficiency (%)
R12	10473.91	94.98	10568.89	51.14
R134a	10473.91	130.32	10,604.23	51.18
R22	10473.91	95.56	10,569.47	51.14
R290	10473.91	262.64	10,736.00	51.32

4 Conclusion

In this research work the (advanced waste heat recovery system based on ORC); four Organic compounds (refrigerants) were examined. The refrigerants were assumed to operate under the same pressure, temperature and evaporation gas with their different thermodynamic properties. Environmental and safety criteria were observed as well. The study was undertaken for 4 working fluids under ORC conditions and the effect of key operation conditions such as Evaporation pressure at inlet, superheating temperature, evaporation temperature, Evaporation pressure at outlet, network output, heat received and heat rejected were examined. After proper examination of all the working fluid (refrigerants) through appropriate selection of these operation conditions, propane (R290) was selected as the most appropriate working fluid among the four refrigerants examined due to its high thermodynamic performance, its low environmental impact and its low flammability-toxicity characteristics. Analysis has shown that R290 provide highest work output and thermal efficiency than others.

Single Marine Diesel Engine used in this research work shows tremendous improvement in power output and efficiency after retrofitting each of the four refrigerants (R12, R22, R134a, and R290) to the engine thereby making it a Combined Cycle Engine. R290 yield a greater improvement when it was retrofitted to the single marine diesel engine than other

refrigerants used in this research. This is believed that it will lead to proper reduction of fuel consumption, reduce amount of heat released by exhaust gas to the environment, improve engine efficiency and increase in turbine life with low safety equipment. Given the high work output of 262.641 kJ/kg at temperature of 90 °C and pressure of 10 bar, these improvements are translated into enormous cost savings during operation. The methodology employed in the present study may also serve as a guideline for future optimization of other energy systems.

ORC working fluid are sometimes in dry gas phase, to this end, excessive superheating is not needed because of its negative effect on maximum working pressure, temperature and overall performance of ORC. Waste heat for ORC can also be used in evaporator or super heater. In order to reduce the cost of material and safety measures, superheater should not be more than 20bar. Superheating led to increase in temperature vapour and performance of the cycle by improving the power output, thermal efficiency and then reducing amount of heat rejected to the environment thereby reducing ozone layer depletion.

Diesel engine with Four stroke version Turbocharger has temperature of 822°C, Exhaust gas temperature 358°C, brake power of 10473.91kW and thermal efficiency of 39.41%. After retrofitting with ORC, the combined – cycle brake power increased to 10736.00kW and thermal efficiency of 51.32% was achieved. This shows improvement in the engine compared to when ORC was not retrofitted.

Contribution to Knowledge

This research has demonstrated a method of recovering supposedly wasted exhaust heat of a diesel engine using ORC to improved diesel engine performance thereby achieving greater work output and higher thermal efficiency. It also gives inside on easy way of comparing ORC in order to determine the one with optimal performance, further improvise mechanical way of controlling global warming through waste heat utilization. It Provide inside on effect of superheating ORC vapour and how to identified best engine for retrofitting ORC.

Recommendation

The followings are recommended for further research work:

- Identify and retrofit two stroke marine engine and determine its performance parameters
- Determine the amount of the exhaust heat flow of the marine engine for two stroke engines
- Determine the organic Rankine cycle power and thermal efficiency of other refrigerants order than what is in this research work.
- Determine the combined cycle (CC) power and thermal efficiency of a two-stroke marine engine retrofitted with an ORC and compare it with two stroke marine diesel engine cycle (MEC) power and efficiency.

Compliance with ethical standards

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