Waste Heat Utilization for Power Generation Using Organic Rankine Cycle (ORC)

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***Abstract*: Energy in from of waste heat is liberated from industries in huge amount. The usage of waste heat for generation of electrical power is very important because of depleting fossil fuels and increasing environmental pollution. Organic Rankine cycle (ORC) provides suitable solution for transformation of low grade energy. This study aims on first law and second law analysis of basic ORC scheme with toluene as a working fluid and Nitrogen stream at temperature of 300^{0}C as waste heat source. The ORC is simulated in engineering equation solver (EES). The performance of the ORC is analyzed by varying turbine inlet pressure ranging from 1 MPa to 3.9 MPa. The first law efficiency is found to be 20.98% for the minimum pressure and 24.02 % for the maximum pressure. Similarly, second law efficiency for the minimum pressure is 30.05% and 35.15% for the maximum pressure. The ORC scheme is further analyzed by superheating toluene at turbine inlet pressure of 2 MPa. The degree of superheat ranges from 4^{0}C to 24^{0}C. The results reveal that the first and second law efficiencies for minimum value of degree of superheat, are 22.84% and 32.95% respectively, and first law and second law efficiency, for maximum value of degree of superheat, slightly drops to 22.75% and 32.77% respectively. It is evaluated that increasing turbine inlet pressure improves performance of the ORC, whereas, superheating slightly declines the performance.**

***Keywords*: First Law Analysis, Organic Rankine Cycle, Second Law Analysis.**

I. INTRODUCTION

The Organic Rankine cycle (ORC) operates in similar way as traditional Rankine cycle. The major difference found in between Organic Rankine cycle and traditional Rankine cycle is of rate of heat addition or heat source temperature and working fluid used. The Organic Rankine cycles incorporates organic working fluids e.g. hydrocarbons, Refrigerants and silicon oils. Traditional Rankine cycle operates with steam as working fluid. Organic Rankine cycle requires low grade energy for power output. The studies show that 50% or more amount of total heat liberated in industry is wasted as low grade heat [1].ORC technology has drawn attention of researches around the globe because of its suitability for usage of low grade energy for getting mechanical power. The low grade energy includes waste heat from industries and renewable energy sources. E.Bellos et al., [2] studied ORC operated by solar and waste heat. The temperature ranges from 150 °C to 300 °C. Among the working fluids analyzed, toluene shows highest power output as compared to n-pentane, MDM and cyclohexane. T.-C hung [3] examined ORC with p-Xylene,Toluene,R113,Benzene,and R123 as working fluids. P-Xylene exhibits maximum efficiency while Benzene exhibits the minimum. Mirzae et al.,[4] investigated ORC with waste heat source application. The results obtained from study reveal that Ethylbenzene, P-xylene,m-xylene have greater efficiency, net power output and lesser total cost, compared to other working fluids considered for analysis. Liu et al.,[5] noticed that the certain molecules with hydrogen bond, such as ammonia, ethanol, and water result in wet fluid condition due to higher enthalpy of vaporization, and such working fluids are observed to be inappropriate for ORC. It is further evaluated that first law efficiency for several working fluids is not strong function of the critical temperature. Lai et al., [6] analyzed linear siloxanes , Alkanes and aromates as working fluids for high temperature heat source operated ORCs. The results ensure cyclopentane as leading working fluid. Quoilin et al.,[7]studied back work ratio as function of evaporator temperature. It is found that toluene had lowest back work ratio as compared to R123yf, R134a, R245fa, and n-pentane. Bao and Zhao [8] determined that the molecular complexity, ratio of latent heat of vaporization and sensible heat and critical parameters can be used for choosing working fluids for ORC. The choosing process is also effected by type of heat source and its temperature level. Mago et al.,[9]analyzed basic and regenerative ORC with different working fluids. The optimal performance is represented by R113 which is the fluid with highest boiling among the fluids selected for analysis. Panesar [10] analyzed ORC using toluene and hexamethyldisiloxane as working fluids. The study exhibits that the blends offer 22%-24% enhancement in the net power produced as compared to pure working fluids. Several studies are carried out for analysis of ORC. The detailed thermodynamic analysis of ORC scheme operating on waste heat is rarely found. This study focuses on first law and second law analysis of ORC with waste heat source nitrogen stream having temperature of 300 ^{0}C and, toluene as working fluid. The physical properties, safety data and environmental properties for toluene are mentioned in Table. 1.

Table. 1. Physical, Environmental and Safety data for toluene [2], [16]

M (Kg/kmol)	92.14
$P_{critical}$ (kPa)	4106
$T_{critical}$ (K)	591.8
ρ_0 (Kg/m^3)	862.4
T_{bp} (K)	384
Cp_0 (kJ/KgK)	1.701
K_0(W/mK)	0.1338
GWP	n.a
ODP	0
ASHRAE 34	B3
Working fluid type	dry

II. MATERIALS AND METHODS

A. System Description

The ORC selected for analysis consists of turbine, condenser, pump and evaporator, as shown in fig. 1. Considering fig.1, Process 1-2 involves pressurization of working fluid in pump, process 3-4 incorporates expansion of working fluid in turbine. Whereas, process 2-3 involves constant pressure heat addition in evaporator and process 4-1 includes heat rejection at constant pressure. The hot nitrogen stream is used as a heat source. The ORC scheme is equipped with water cooled condenser. The ORC scheme is assumed to be operating with constant pressure heat addition and rejection. There is also no loss of pressure along the pipes of the system. The components are considered to be operating in steady state condition. Kinetic and potential energy of working fluid are neglected. Kinetic, potential and chemical energy of working fluid are also ignored.

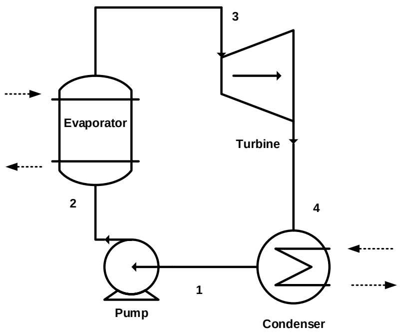

Fig. 1: Schematic of ORC

Table. 2. Constants considered for thermodynamic analysis [9], [14]

Heat source Nitrogen stream temperature	300^0C
Heat source Nitrogen stream pressure	100 kPa
Isentropic efficiency of pump	85%
Hot Nitrogen stream mass flow rate	2 kg/s
Isentropic efficiency of turbine	80%

Table.3 Thermo-physical properties of ORC at each state point

State point	T K	P MPa	h Kj/Kg	s kJ/kgK	Ψ kJ/kg	E kW
1	318.3	0.01	-123.2	-0.351	1.054	0.3557
2	319.3	2.6	-119.6	-0.3493	4.158	1.403
3	555.2	2.6	590.1	1.19	255.1	86.08
4	419.8	0.01	420.3	1.295	54.19	18.28
Hot nitrogen stream inlet	573	0.1	598	7.525	84.51	169
Hot nitrogen stream outlet	460	0.1	478.3	7.292	34.12	68.24
Condenser Water inlet	298	0.1	104.2	0.3648	0	0
Condenser Water outlet	308	0.1	146	0.5092	0.6865	3.009

B. Mathematical Model

The mathematical model used for evaluating performance of ORCs is presented in this section. The constant parameters considered during thermodynamic analysis are mentioned in Table no. 2. The performance of ORCs is determined by applying first law and second law of thermodynamics on each component. Mass balance, law of conservation of energy, and exergy balance for any open system are mentioned below. These mathematical relations are simplified by considering open system with steady state condition and negligible variation in *potential and kinetic energy*. These mathematical relation are represented respectively, by:

$$\sum \dot{m}_{in} = \sum \dot{m}_{ex} \tag{1}$$
$$Q - \dot{W} = \sum \dot{m}_{ex}\, h_{ex} - \sum m_{in} h_{in} \tag{2}$$
$$\dot{X}_{heat} - \dot{W} = \sum \dot{m}_{ex}\, \Psi_{ex} - \sum m_{in}\, \Psi_{in} + \dot{I} \tag{3}$$

The exergy transfer by heat transfer $\dot{X}_{heat}$ is calculated by:

$$\dot{X}_{heat} = \sum \left(1 - \frac{T_0}{T}\right) \times Q \tag{4}$$

The specific flow exergy of fluid stream at a state point is evaluated by:

$$\Psi = h - h_0 - T_0 \times (s - s_0) \tag{5}$$

The exergy rate associated with a fluid stream at a state point is given by

$$\dot{X} = \dot{m}\Psi = \dot{m} \times [h - h_0 - T_0 \times (s - s_0)] \tag{6}$$

The mathematical relation for First law efficiency and Second law efficiency is represented by equation (7) and (8), respectively.

$$\eta_I = \frac{\dot{W}_{turbine} - \dot{W}pump}{Q_{evaporator}} \tag{7}$$

$$\eta_{II} = \frac{Exergy\ utilized}{Exergy\ avaiable} = \frac{\dot{W}_{turbine} - \dot{W}pump}{(\dot{X}_{in\ wf} evaporator + \dot{X}_{in_{gas}} evaporator) - (\dot{X}_{in\ wf} pump + \dot{W}pump)} \tag{8}$$

IV. RESULTS AND DISCUSSION

The ORC is analyzed using first law and second law of thermodynamics. The turbine inlet pressure and degree of superheat for ORC scheme are increased to investigate their effect on First law efficiency, System Total lost work rate, Net Power Output, and Second law efficiency. The Performance parameters of ORC are calculated using the data mentioned in table no.3. The evaluated performance parameters are presented in table no. 4. These performance parameters are evaluated by keeping Evaporator and Condenser pressure fixed at 2.6 MPa and 10 kPa, respectively.

Table. 4. Performance Parameters of ORC

Description	Data	
Turbine work	Kw	57.29
Pump Work	Kw	1.218
Evaporator duty	Kw	239.4
Condenser duty	Kw	183.4
Net power output	Kw	56.07
Mass flow rate (Toluene)	Kg/s	0.3374
System Total lost work rate (ORC)	kW	41.7
First Law efficiency	%	23.42
Second Law efficiency	%	33.93

A. *Impact of inlet pressure of turbine on Performance of ORC*

The ORC scheme is thermodynamically analyzed by changing turbine inlet pressure to find out its effect on First Law efficiency, Net power output, second Law efficiency, and Total lost work rate. The pressure is enhanced from 1 MPa to 3.9 MPa. The Pressure 3.9 MPa is slightly below the critical pressure for toluene.

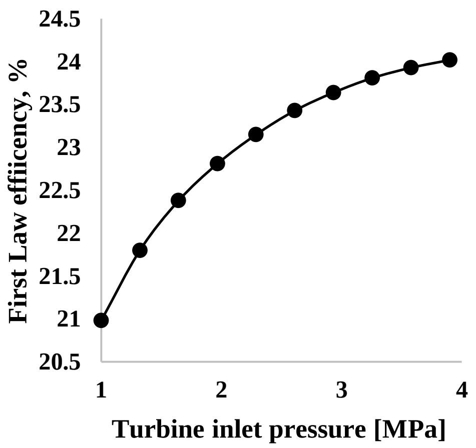

Fig. 2: Impact of Turbine inlet pressure on First Law efficiency

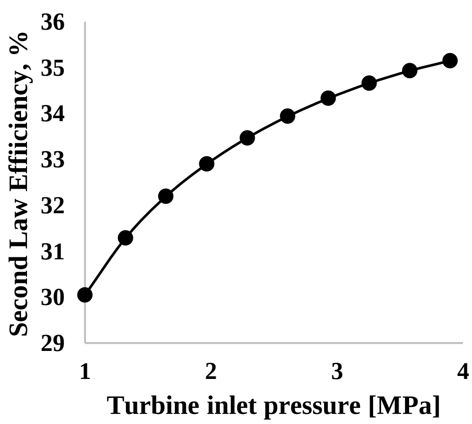

Fig. 3: Impact of Turbine inlet pressure on Second Law efficiency

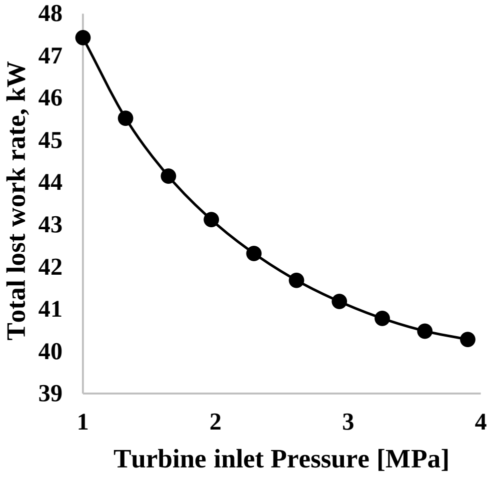

Fig. 4: Impact of Turbine inlet pressure on Total lost work rate

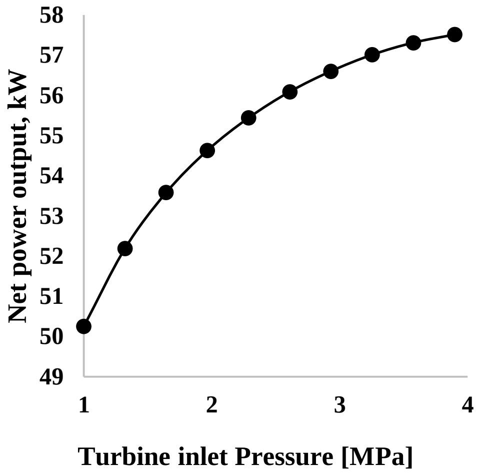

Fig.5: Impact of Turbine inlet pressure on Net Power output

Fig. 2 Depicts change in First Law efficiency with respect to turbine inlet pressure. It is detected that the First Law efficiency escalates by rise in turbine inlet pressure. The First Law efficiency at minimum pressure of 1 MPa is 20.98% and 24.02% at maximum pressure of 3.9 MPa. It is can be observed that percentage increase in efficiency from minimum to maximum pressure is 14.5 %. The rise in efficiency by increasing pressure is due to increased average temperature of working fluid at which heat is added to evaporator. Fig. 3 Depicts change in Second Law efficiency with respect to turbine inlet pressure. It is noticed that the efficiency rises by rising turbine pressure. The efficiency at minimum pressure of 1 MPa is 30.05% and 35.15% at maximum pressure of 3.9 MPa. It is found that percentage increase in the efficiency from minimum to maximum pressure is 17%. The rise in efficiency by increasing pressure is due to increased flow exergy of working fluid at turbine inlet. The relation between turbine inlet pressure and Total lost work rate can be checked from fig. 4, The Total lost work rate decreases by increasing turbine inlet pressure. The Total lost work rate at minimum pressure of 1 MPa is 47.43 kW and 40.28 kW at maximum pressure of 3.9 MPa. It is can be detected that percentage drop in Total lost work rate from minimum to maximum pressure (-18%). The drop is Total lost work rate is due to significant drop in lost work rate of evaporator and condenser (Heat exchangers). Fig.5 depicts relation between Net power output and turbine inlet pressure. The rise in turbine inlet pressure enhances net power output because of increased enthalpy at turbine inlet and significant drop in enthalpy along the expansion of working inside turbine resulting greater net power output. The Net power output at lowest pressure of 1 MPa is found to be 50.25 kW and 57.51 kW for highest pressure of 3.9 MPa. The percentage rise in net power output from lowest to highest pressure is 14.5%.

B. Impact of superheat degree on performance of ORC

The ORC scheme is thermodynamically analyzed by increasing superheat degree from 4^0C to 24^0C to determine its effect on First Law efficiency, Total lost work rate Net Power output and Second Law efficiency, and. The fig. 6 to 9 depict the effect of variation in superheat degree on the performance parameters. The superheating is carried out keeping evaporator pressure constant at 2 MPa.

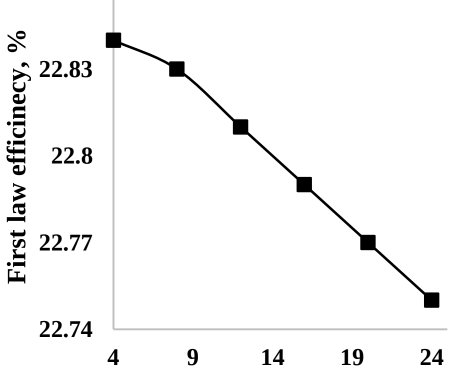

Fig 06. Impact of Degree of superheat on first law efficiency

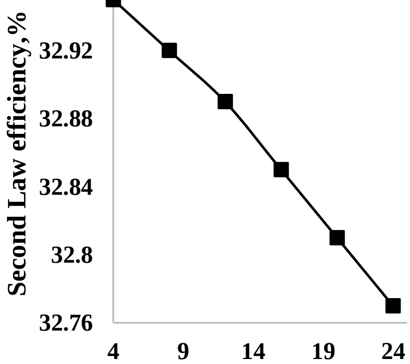

Fig 07. Impact of Degree of superheat on second law efficiency

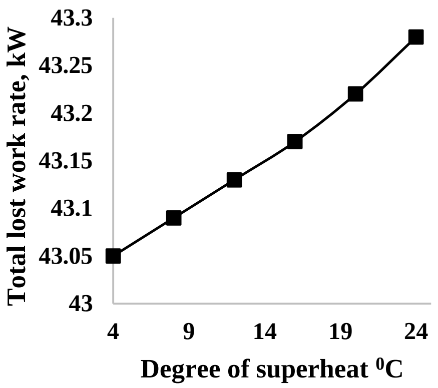

Fig. 08 Impact of Degree of superheat on Total lost work rate

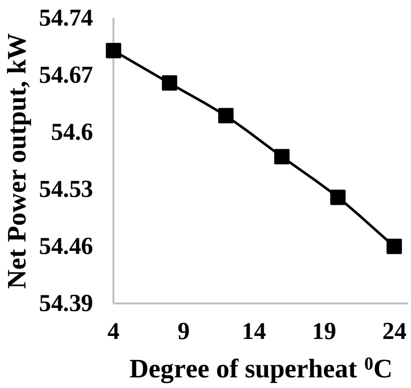

Fig. 09 Impact of Degree of superheat on Net power output

Fig. 6 shows the variation in first law efficiency with respect to superheat degree. It is noticed that superheating results slight drop in the efficiency. The efficiency is found to be 22.84% at lowest superheat degree (4^0C) and 22.75% at the highest degree of superheat (24^0C). The percentage drop in efficiency is - 0.3% .The effect of change in superheat degree on second law efficiency can be observed from fig. 7. It is noticed that superheating results slight drop in the efficiency. The efficiency is found to be 32.95% at lowest superheat degree (4^0C) and 32.77% at the highest degree of superheat (24^0C). The percentage drop in efficiency is (- 0.55 %). Fig. 8 represents the relation between superheat degree and Total lost work rate. The Total lost work rate is found to be 43.05 kW at lowest superheat degree (4^0C) and 43.28 kW at the highest degree of superheat (24^0C). The percentage rise in Total lost work rate is (0.5 %). The alteration of Net power output with respect to super heat degree is depicted in fig.9, The Net power output is found to be 54.7 kW at lowest superheat degree (4^0C) and 54.46 kW at the highest degree of superheat (24^0C). The percentage drop in Total lost work rate is (-0.44 %). Typically It is evaluated that superheating results slight drop in performance of ORC which is due to small range of superheat degree i.e. (4^0C to 24^0C). The negative impact of superheating on performance parameters is due abrupt drop in working fluid mass flow rate and corresponding gradual rise in enthalpy drop along turbine. Furthermore, superheating causes convergence of isobaric lines with temperature therefore superheating is not advantageous for dry working fluids (e.g. Toluene).

C. Lost work rate in components of ORC

The lost work rate is very crucial performance parameter. The lost work rate in evaporator, condenser, turbine and pump of ORC scheme is determined. The percentage lost work rate in components of ORC is also determined.

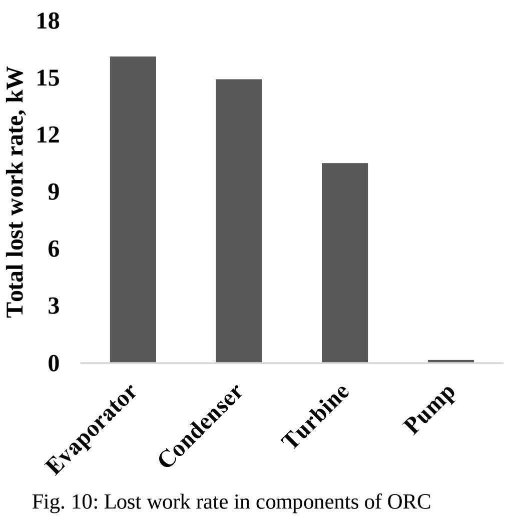

Fig. 10: Lost work rate in components of ORC

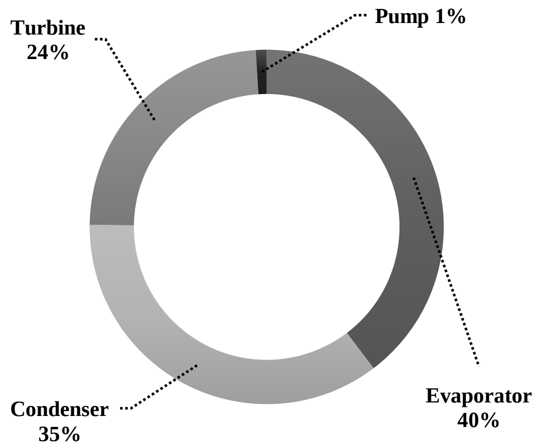

Fig. 11: Percentage lost work rate in components of ORC

Fig. 10 shows lost work rate in components of ORC. The lost work rate is determined at Evaporator pressure (2.6 MPa) and (10 kPa) condenser pressure. The maximum lost work rate is observed in evaporator 16 kW followed by condenser 14 kW , turbine 10.51 kW and pump 0.17 KW. It is observed that heat exchangers (evaporator and condenser) show greater lost work rate as compared to turbomachinery (Turbine in pump). The greater lost work rate in heat exchangers is due mechanism of exergy transfer by heat transfer resulting significant loss of work potential. Furthermore, it is evaluated that maximum percentage lost work rate is detected in evaporator (40%) and smallest in pump (1%) , as shown in fig.11.

V. CONCLUSION

The thermodynamics based analysis of ORC scheme operating on waste heat source is carried out to determine its performance. The performance variables taken into account are first law efficiency, Total lost work rate , Net power output, and second law efficiency. Moreover, the performance of ORC is evaluated by increasing turbine inlet pressure from 1 MPa to 3.9 MPa and varying super heat degree from 4^0C to 24^0C.

- It is concluded that increasing turbine inlet pressure (1MPa to 3.9 MPa) results rise in First law efficiency, and net power output , Second law efficiency out by 14.5%,14.5% , 17%, Whereas percentage drop in Total lost work rate is equal to -18%.
- It is further concluded that superheating ORC with toluene as a working fluid results negative impact on performance variables of ORC.
- It can be concluded that maximum percentage lost work rate is shown by evaporator (40%) and minimum by pump (1%). Turbine shows less percentage lost work rate (24%) as compared to condenser (35%) and evaporator (40%).

VI. RECOMMENDATIONS

Waste heat utilization is important because of depleting fossil fuel and rising environmental issues. In this study, waste heat from an industry is utilized for power generation only but along with power generation, it can also be utilized for process heating and cooling. Several heat driven refrigeration cycles e.g. vapour absorption refrigeration units can be operated with the available waste heat. Moreover, this work is limited to usage of toluene as a working fluid for ORC. The blends of several working fluids can also be used to examine the performance of ORC.

NOMENCLATURE

Symbols			
T_{bp}	Boiling point temperature, K	Ψ	Specific flow exergy, KJ/Kg
P	Pressure, kPa	ODP	Ozone depletion potential
$P_{critical}$	Critical pressure, kPa	GWP	Global warming potential
Ρ	Density, kg/m^3	M	Molar mass ,Kg/Kmol
$\dot{W}$	Power, kW	K	Thermal conductivity ,W/mK
Q	Heat transfer, kW	Cp	Specific heat at constant pressure, KJ/KgK
$\dot{m}$	Mass flow rate, Kg/s	***Subscripts***	
$\dot{X}$	Flow exergy, kW	0	Ambient
$\dot{X}_{heat}$	Exergy transfer by heat transfer ,kW	in	Inlet
h	Specific enthalpy, Kj/Kg	ex	Exit
$\dot{I}$	Total lost work rate , kW	wf	Working fluid
s	Specific entropy, KJ/KgK	gas	Waste heat nitrogen stream

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