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Computational Simulation of Energy Analysis for Industrial Application Using ORC with Eco-Friendly Refrigerant

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Abstract- The construction of the modern world has achieved too much by increasing the availability and widespread use of various models of energy. The ORC uses a liquid that is more suitable than in water at the point of use of heat with a low temperature. The ORC cycle, unlike the racing circuit, is attractive for local energy and small for electrical energy. In this experiment this was done using EES software to test the thermodynamically ORCs using different fluids. The influence of the working fluid on the ORC performance of the discharge power lift, overall performance and rank efficiency were presented. The R290 has a higher overall performance than that of turbine efficiency as much as 70% -88% when it is similar to the R600a and R134a. If ORC is going to work with greater turbine efficiency, the R290 report depends on whether we want to know more about overall performance or Rankine efficiency.

Keywords:- Power output, overall efficiency, Rankine efficiency, turbine efficiency, ORC, R290, R600a, R134a.

I. INTRODUCTION

Over the years, the interest in restoring low levels of heat has grown exponentially. Many researchers have come up with a variety of ways to generate electricity from low-energy sources in solar energy, home heating, biomass and wind power warm waste. Of all these, ORC is considered to be the most cost-effective due to its simple design and availability of equipment. The ORC uses a liquid that is more suitable than water at the point of use of heat with a low temperature. The ORC cycle, unlike the racing circuit, is particularly attractive for local energy and small for electrical energy [1].

The process of extracting energy from heat waste by various industrial processes is called heat dissipation process. In some applications, refurbishments and fire hydrants are used to regenerate and reheat in their systems. The operation of the solid waste heat back into the step cycle is not supported when the solid waste heat is low. The low-temperature electric heating system can easily make the environment easier to use ORC.

Apart from functions such as temperature; the major difference between ORC and the Rankine cycle is the water consumption used in each cycle. In Rankine gas environment water is the only working fluid that can be used while in ORC there are more than one hundred working fluids that can be used. The discovery of new fluids to work for ORC is ongoing.

The size of the ORC process depends on the temperature of the material at which the fluid operates. The negative temperature, the environment and the safety of each working fluid vary. Safety and environmental information for working fluids are generally not included. Selecting the appropriate fluid for ORC systems is critical for better performance and higher operating results [2].

Work fluid such as water does not pose a risk to the environment because it is toxic, non-flammable, has no global warming potential and no ozone depletion potential. Many organic working liquids are not environmentally friendly because they have ozone depletion potential and some cause a greenhouse effect that is harmful to the environment.

Some organic working liquids have symptoms of high toxicity and high flammability. The majority of ORC liquids relative to water have low NBP. Due to the low NBP, the ORC working fluid needs low heat but water to evaporate and recover little heat energy from heat sources. Water has a negative saturation of a steam line slope in the T-s diagram, while the organic working fluids have three different saturation steam line slopes, viz. infinite, positive and negative steam lines. Turbo-tensile has more advantages when operating fluid has either a positive or infinite steam line slope.

One of the advantages of the positive and infinite slope is that the working fluid in both leaves the turbocharger as overheated steam, thus eliminating the risk of corrosion. In addition, when the working fluid has a positive or infinite slope, the steam in the evaporator does not need to be overheated and therefore smaller and cheaper ORC components can be used [3].

II. LITERATURE REVIEW

Eryanto et al. (2020) [4] introduced a comparative analysis with ORC, Regenerative ORC (RORC) and RORC with internal heat exchange (IHE). The results indicate that RORC with IHE has the highest value for both energy (21.74%) and allergy efficiency (25.26%), while net power produced 5479 kW. This shows that the addition of OFOH and IHE can increase these factors, improve performance and reduce energy consumption from the cycle.

Köse, Özkano. fl. (2020) [5] evaluated the performance enhancement of SRC and ORC systems used as bottom systems in GT based triple composite systems for different temperature and pressure inlet turbines. In the first step, the SRC was optimized for different pressures (from 10 bar to 100 bar) and temperature (from saturated steam temperature to 480°C). In the second step, a parametric optimization is performed to evaluate the overall effect of acetone, R113, R141b, R152a, R245fa and R365mfc on the thermodynamic performance of ORC (Organic Rankine Cycle) with increased inflow pressure and turbine temperature. In parallel with the optimization, the ORC turbine inlet pressure was raised from 7.5 bar to a significant working fluid pressure of 2.5 bar.

Thanganadar et al. (2019) [6] aimed to explore the full potential of a commercial gas turbine cycle and analyzed the maximum efficiency and cost of electricity in five carbon dioxide cycles.

Sun and colleagues (2018) [7] developed a new connected top-bottom circuit for cascading utilization of effluent temperature, in which high-temperature energy was extracted using a CO2 circuit as a peak cycle when the inlet inflow temperature was above 500 ° C.

Mecheri and Moullec (2016) [8] concluded that CO2 hydroelectric power plants have theoretically offered about 6% LHV relative performance to improve efficiency (from about 45% to 48%) with existing materials at current operating conditions.

Liu and colleagues (2018) [9] proposed a new conceptual system integrated with a CO2 cycle to recover the exhaust heat from the boiler exhaust at 350 ° C.

Saleh (2018) [10] introduced an optimized ORC video system with an allergy efficiency of 53.8% with R602 as the working fluid. Compared to the Kalina cycle, the Organic Rankine cycle and the triple power cycle could be more economical.

Colorado (2017) [11] performed an advanced exergy analysis on a single- stage absorption heat transformer operating with a lithium bromide water solution. The total irreversibility of the cycle was estimated in 1.046 kW in which avoidable part sharing 14.78% could be reduced by improving its design and configuration.

Mohammadi and colleagues (2019) [12] evaluated the real potential of enhancement for the recompression CO2 cycle performance by means of calculating the first and second splitting levels of exergy destruction. They revealed that for improvement priority of components obtained by the conventional exergy analysis was different from that achieved by the advanced exergy analysis.

Galindo and his colleagues (2016) [13] performed a high exergy analysis of an ORC-based system combined with an IC engine and suggested that the expander be the first improvement priority.

Nami and her colleagues (2017) [14] applied the exergy high-tech analysis to an ORC dual-fluid power plant with a geothermal heat source.

Liu, Xiangyang et al. (2020) [15] presented a system for converting the exhaust gas (EG) waste water and jacket cooling water (JCW) ME into electrical and cooling energy that is mainly required on the ship. The proposed waste heat recovery system has three sub-cycles, namely the Rankine vapor cycle (RC), the organic Rankine cycle (ORC) and the absorption refrigeration cycle (ARC), which perform well in high, medium and low usage-temperature heat sources, respectively.

Liu, Xiangyang et al. (2020) [16] Rankine steam and organic cycles are combined to convert the waste heat of the exhaust gas and cooling water of a marine engine into mechanical energy jacket. Some of the jacket cooling water is used as the working fluid of the Rankine steam cycle subsystem to efficiently use jacket cooling water heat and avoid increased ship weight due to the excess water. The performance of the proposed system for recovering 14-cylinder two-stroke marine engine waste heat was simulated and compared with the performance of the WHRSs based on a single steam Rankine cycle (SSRC) and a double-rank organic Rankine cycle (DPORC).

Kim, Jun-Seong et al. (2019 [17] presented the thermodynamic performance of waste heat recovery systems of a marine gas turbine was analyzed. These systems combine the steam Rankine cycle and the organic Rankine cycle to form a dual-loop cycle. Working fluids R32, R134a, R152a and R1234yf, that are low-global warming potential, were selected for the organic Rankine cycle.

Bălănescu et al. (2019) [18] presented the fuel savings and their cost are assessed. Two organic working fluid were considered, namely R134a and R123. The study shows that efficiency of the power plant increases by roughly 1.1 % when the ORC unit is added. Taking into account the current concerns regarding the fossil fuel depletion, the estimated fuel savings could be considered significant.

III. SYSTEM MODELING

1. Working Principle and Mathematical Models of SRC and ORC:

The Rankine cycle waste heat generation system consists of three subsystems: heat source (medium low temperature waste heat vapor) system, Rankine cycle system and cooling source (cooling water) system. The main tradition and process of the ORC system and the traditional SRC system is the same, but the difference is that the ORC system uses low-boiling organic working fluid instead of water with high pressure vapor.

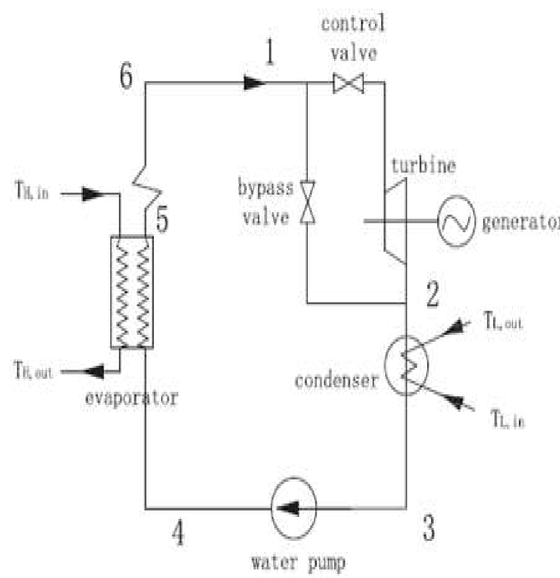

Fig 1. Schematic of ORC cycle.

Process 1 to 2 is an adiabatic expansion process of working fluid inthe turbine, the ideal reversible adiabatic expansion process is regarded as an isentropic process, but the turbine efficiency ηt needs to be considered in the actual calculation process, and then the output power of working fluid in the turbine is Wt:

$$W_t = m_f (h_1 - h_2) \square_t$$

Where, $h_1$and h_2 are the enthalpy (KJ/Kg) at inlet and outlet of the turbine respectively, m_f is mass flow rate (kg/s) of refrigerant entering the turbine.

Process 2 to 3 is an isobaric exothermic process of working fluid in the condenser, the heat released is Qc:

$$Q_c = m_f (h_2 - h_3)$$

Where, h_2 and h_3 (KJ/Kg) are inlet and outlet conditions of condenser respectively and m_f mass flow rate of refrigerant (Kg/s) entering the condenser

Process 3 to 4 is an adiabatic compression process in the working fluid pump, the ideal reversible adiabatic compression process isconsidered as a isentropic process.

But the pump efficiency η_p needs to be considered in the actual calculation process, and the output power of the working fluid in the turbine is Wp:

$$Wp = m_f (h_4 - h_3)/\eta_p$$

Where, h_3 and h_4 are inlet and outlet conditions of pump respectively, η_p is isentropic efficiency of pump.

Process 4 to 1 is an isobaric endothermic process in theevaporator. The heat absorbed in the evaporator is Qe:

$$Qe = m_f (h_1 - h_4)$$

Where h_1 and h_4 are inlet and outlet conditions of evaporator respectively, m_f (kg/s) is mass flow rate of refrigerant entering the evaporator.

The power generation capacity of unit quality working fluid is P:

$$P = E/m_f = w_t \eta_g$$

Where P is system power generation capacity, kW; wt is the specificwork of working fluid, kJ/kg; η_g is the generator efficiency, %.

The thermal efficiency of the system is η_e.

$$\eta_e = W_{net}/E_{in}$$

Where, W_{net} is net workdone produced by turbine, E_{in} is heat input given to the evaporator.

Table 1. Data taken under consideration.

Circular name	Turbine inlet temperature (°C)	Turbine inlet pressure (Mpa)	Condensation temperature (°C)	η_p	η_t
SRC	150-350	0.2-1.4	45	70%	80%
ORC	100-180	1.5-3	37	85%	70%-88%

2. Properties of the Working Fluids Used for the Investigation:

2.1 R600a (Iso-Butane): R600a (Iso-Butane) is refrigerant grade Iso-Butane used as a replacement for R12 and R134a in a variety of high temperature refrigeration applications. R600a (Iso-Butane) is a hydrocarbon that is becoming increasingly popular due to its low Global Warming Potential (GWP).

2.2 R134a(Tetrafluoroethane (CF3CH2F)): R134a is also known as Tetrafluoroethane (CF3CH2F) from the family of HFC refrigerant. With the discovery of the damaging effect of CFCs and HCFCs refrigerants to the ozone layer, the HFC family of refrigerant has been widely used as their replacement.

2.3 R290 (Propane): R290 (Propane) is refrigerant grade propane, used as an alternative to R404A and R407 series refrigerants in new refrigeration and air conditioning systems. R290 (Propane) is a Hydrocarbon and although highly flammable it is an efficient refrigerant that has a low Global Warming Potential (GWP).

IV. RESULTS AND DISCUSSION

1. Simulation results of the SRC power generation system.

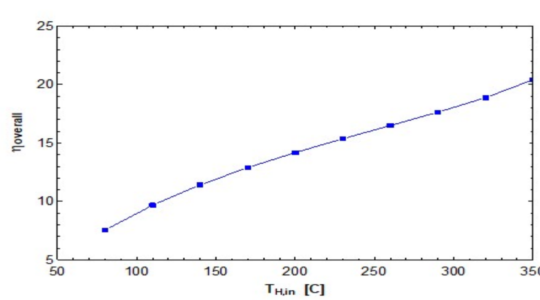

Fig 2. Variation of overall efficiency with heat source temperature.

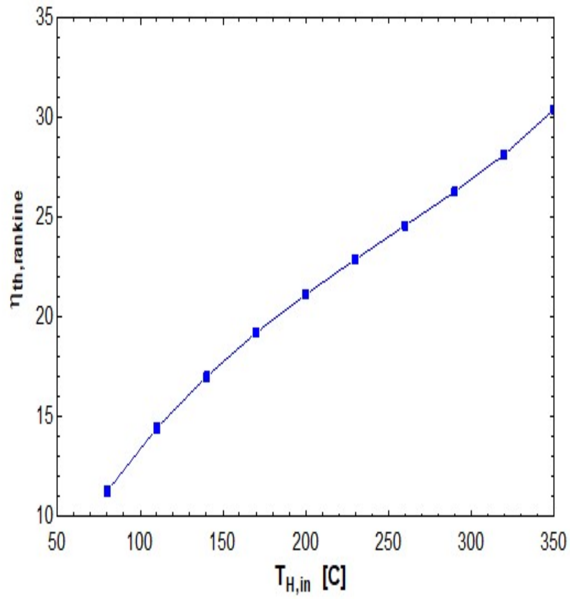

Fig 3. Variation of Rankine efficiency with heat source temperature.

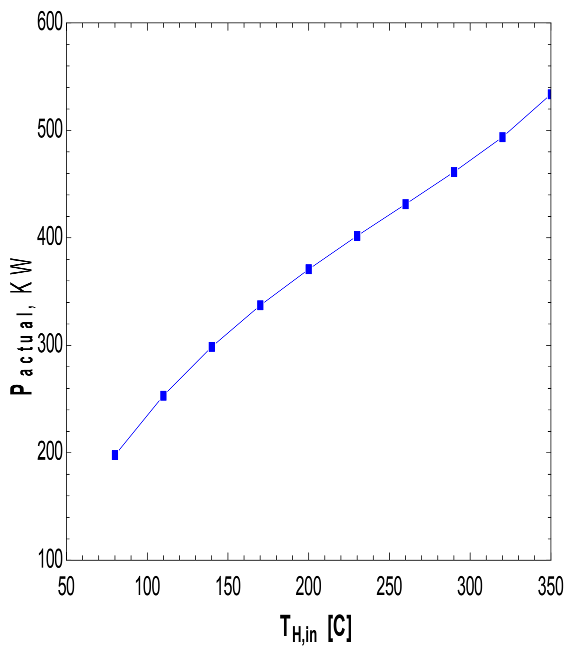

Fig 4. Variation of power output with heat source temperature.

Fig. 4.1-4.3 shows that with increase in heat source temperature overall efficiency, rankine efficiency and power output increases.

Also it is noted that increase of heat source temperatures leads to bigger heat transfer temperature differences in the evaporator, the irreversible loss increases, and the exergy efficiency declines.

2. Simulation Results of the ORC Power Generation System:

In this study different working fluid in the ORCs based on their thermodynamic, environmental and safety properties were used.

2.1 Variation of Heat Source:

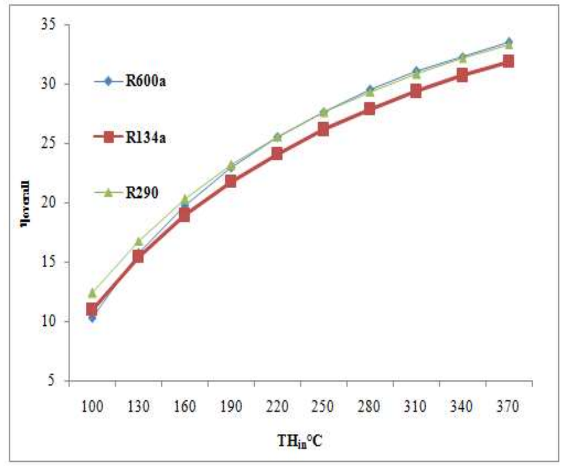

Fig 5. Variation of overall efficiency with heat source with different working fluids.

Fig. 4.2.11, shows the variation of overall efficiency with the various heat source temperatures for 3 different working fluids. It is obvious that as the heat source temperature increases, overall efficiency increases for all 3 different working fluids and maximum value was achieved with R290 while minimum value with R134a.

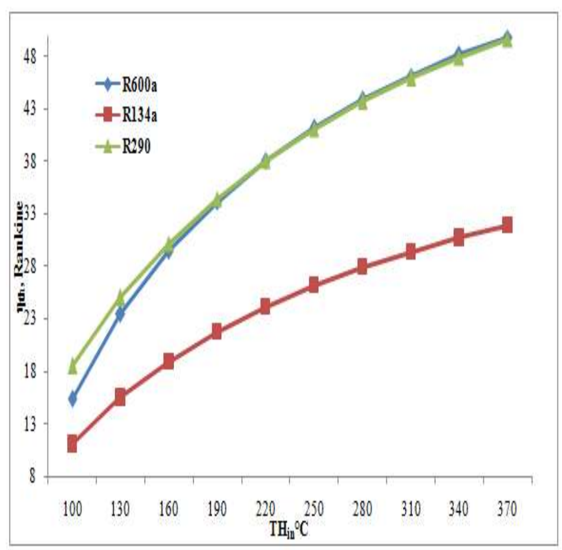

Fig 6. Variation of thermal efficiency of rankine cycle with heat source with different working fluids.

In fig. 4.2.12, shows the variation of thermal efficiency of rankine cycle with the various heat source temperatures for 3 different working fluids.

It shows the same trend as the heat source temperature increases, overall efficiency increases for all 3 different working fluids, maximum it was achieved with R290 and minimum it was achieved with R134a.

The higher thermal efficiency of R 290 is because of the higher latent heat of evaporation which indicates lesser mass requirement of refrigerant.The lower liquid density of R290 reflects the lower requirement of refrigerant mass resulting in lower friction and better heat transfer coefficients in evaporator and condenser.

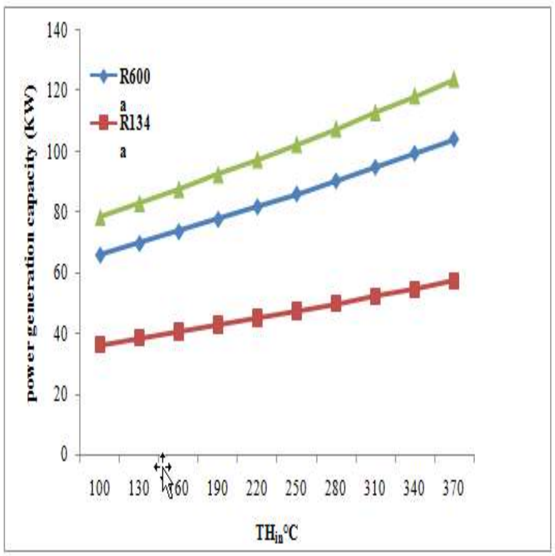

Fig 7. Variation of power generation with heat source with different working fluids.

In fig. 4.2.13, shows the variation of power output with the various heat source temperatures for 3 different working fluids. It is shows that as the heat source temperature increases, power output increases for all 3 different working fluids R290 has maximum and minimum it was achieved with R134a.Refrigerant viscosity is the major source of irreversibility and influences condensation and boiling heat transfer coefficients.

R290 has lower viscosity and higher thermal conductivity which improves the performance of condenser and evaporator. The higher specific heat of R290 gives lower discharge temperature.

2.2 Variation of Turbine Efficiency:

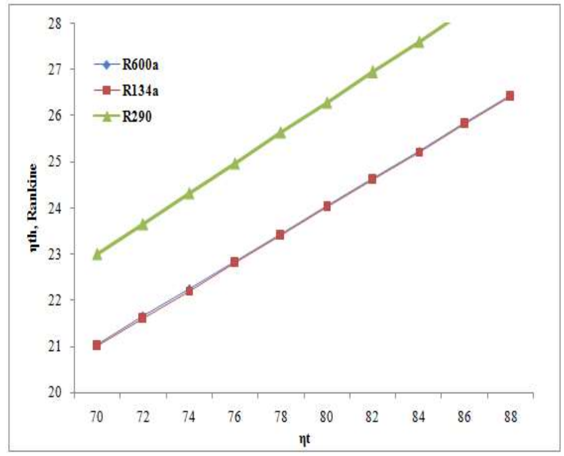

Fig 8. Variation of thermal efficiency with turbine efficiency with different working fluids.

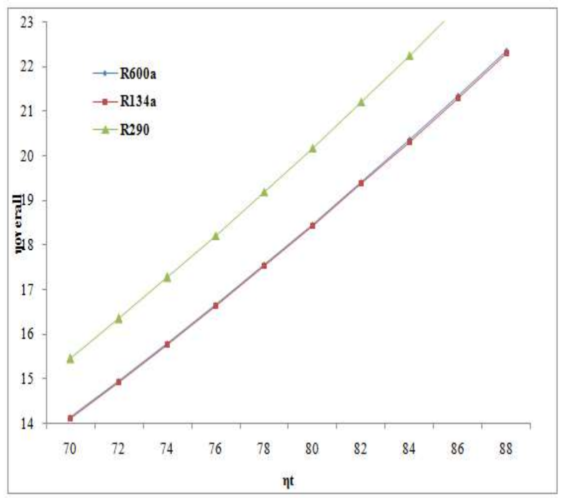

Fig 9. Variation of overall efficiency with turbine efficiency with different working fluids.

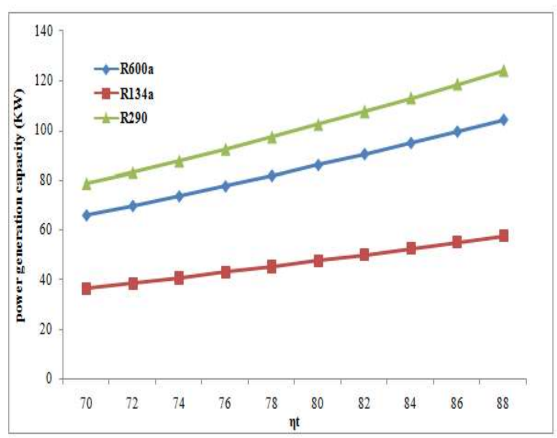

Fig 10. Variation of power generation capacity with turbine efficiency with different working fluids.

As shown by figure 4.2.21 and figure 4.2.22, R290 has larger overall efficiency over turbine efficiency range 70%-88% while it is low in case of R134a. If the ORC will be operated with larger turbine efficiency, R290 is suggested if center of our interest is to increase overall efficiency or thermal efficiency.R290 gives lower discharge temperature which is important factor improving the life of compressor.

Through the simulation and calculation of power generation,we know that the power generation of the system increases with the increase in turbine efficiency.

As shown by figure 4.2.23, R290 has larger power generation capacity over the turbine efficiency range 70%-88% while it is low in case of R134a. If the ORC will be operated for larger turbine efficiency R290 is suggested.

2.3 Variation of Pinch Point Temeperature Difference:

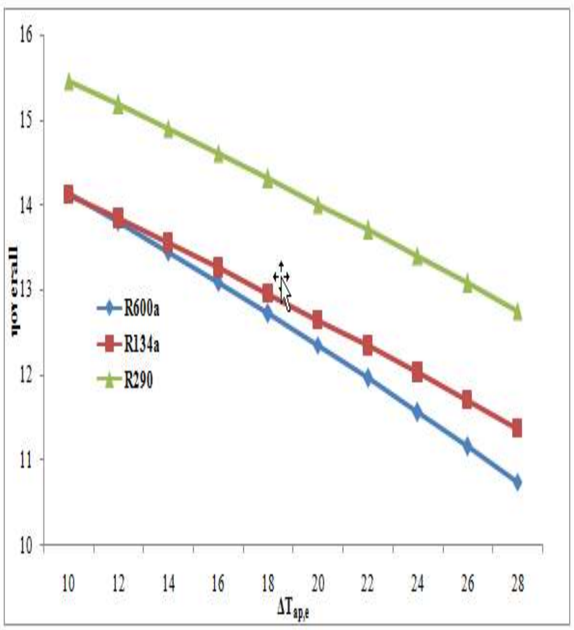

Fig 11. Variation of overall efficiency with pinch point temperature difference with different working fluids.

As shown by figure 4.2.31 and figure 4.2.32, R290 has larger overall efficiency over the pinch point temperature difference range of 10-25 °C while it is low in case of R600.

If the ORC will be operated for a larger pinch point temperature difference R290 is being suggested.

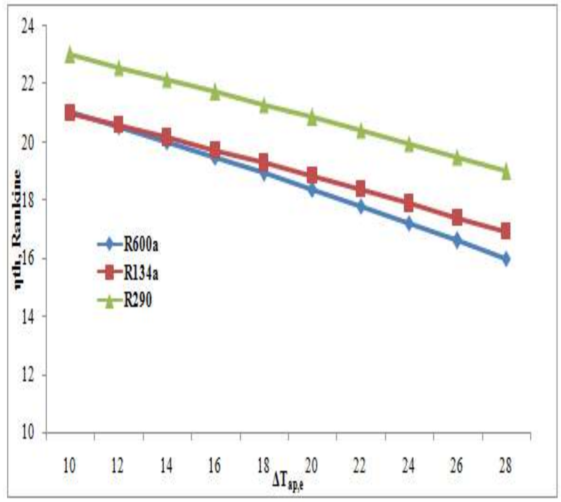

Fig 12. Variation of Rankine efficiency with pinch point temperature difference with different working fluids.

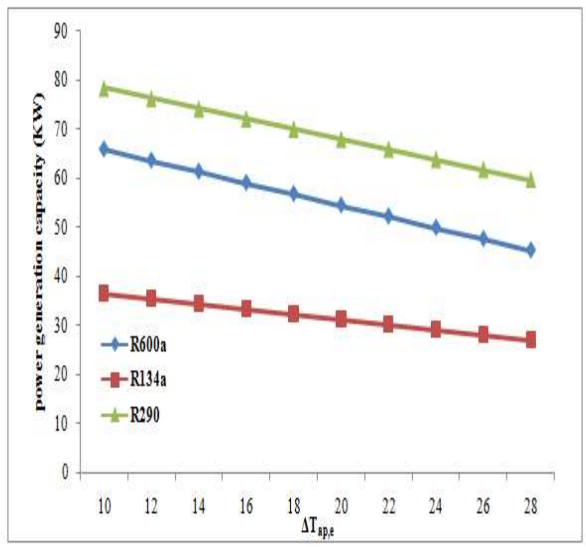

Fig 13. Variation of power generation capacity with pinch point temperature difference with different working fluids.

Through the simulation and calculation of power generation,we know that the power generation of the system decreases with the increase in pinch point temperature difference.

As shown by figure 4.2.33, R290 has larger power generation capacity over the pinch point temperature difference range of 10-25 °C while it is minimum in case of R134a. If the ORC will be operated for a larger pinch point temperature difference, R290 is suggested if largerpower generation capacity is our concern. The lower liquid density of R290 reflects the lower requirement of refrigerant mass resulting in lower friction and better heat transfer coefficients in evaporator and condenser.

3. Simulation Comparision of SRC and ORC Systems:

In this section simulation comparison of SRC and ORC has been done with varying heat source temperature, pinch point temperature difference and turbine efficiency.

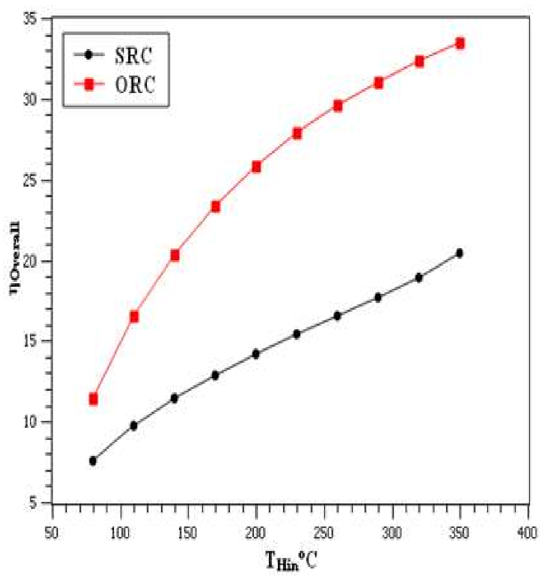

Fig 14. Overall efficiency with heat source of SRC and ORC.

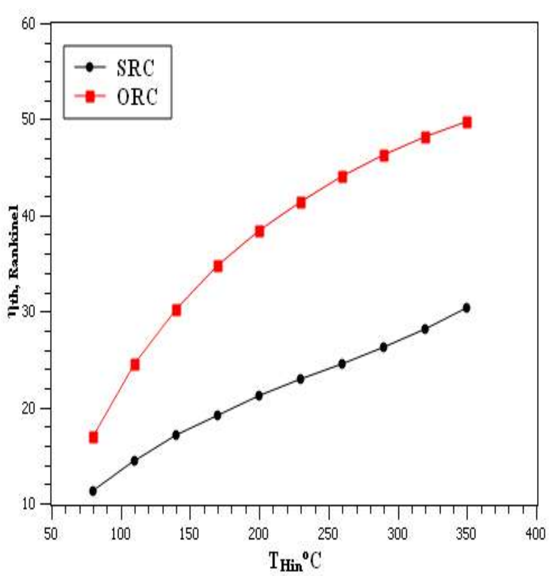

Fig 15. Rankine efficiency with heat source of SRC and ORC.

From fig. 4.31-4.32, it is observed that when the heat source temperature is around 100–370 ºC, overall efficiency and Rankine efficiency of ORC system is greater than that of the SRC systems.

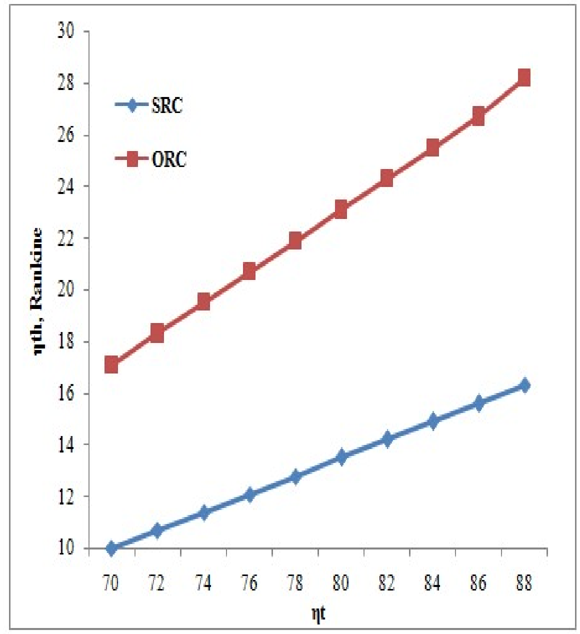

Fig 16. Overall efficiency with turbine efficiency of SRC and ORC.

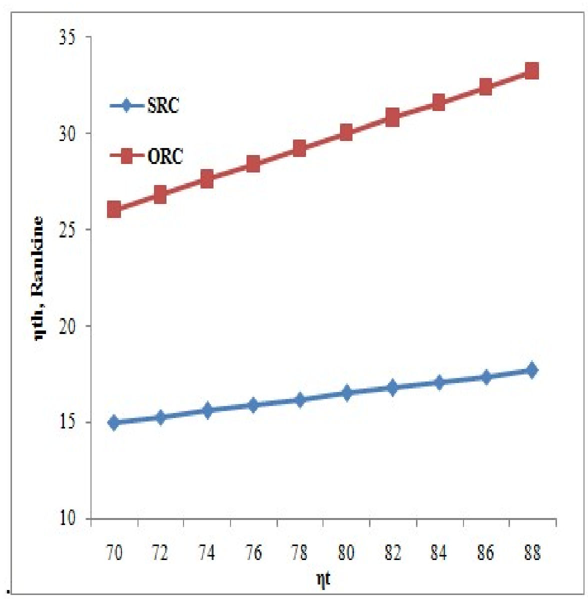

Fig 17. Rankine efficiency with turbine efficiency of SRC and ORC.

From fig. 4.33-4.34, it is observed that when the turbine efficiency is varied around 70–80%, overall efficiency and Rankine efficiency of ORC system is greater than that of the SRC systems.

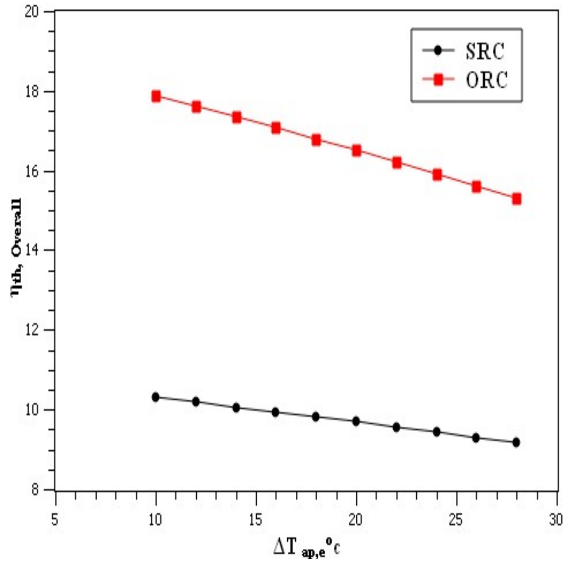

Fig 18. Overall efficiency with pinch point temperature difference of SRC and ORC.

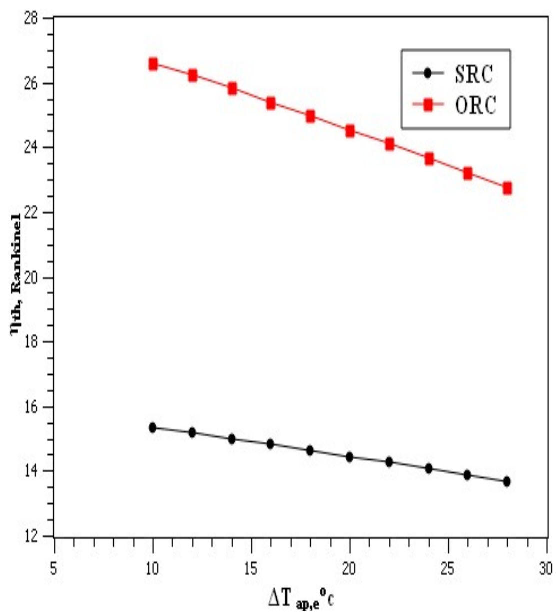

Fig 19. Rankine efficiency with pinch point temperature difference of SRC and ORC.

From fig. 4.35-4.36, it is observed that when the pinch point temperature difference is varied around 10–30ºC, overall efficiency and thermal efficiency of ORC system is greater than that of the SRC systems.

V. CONCLUSION

In this work simulations were carried out using the EES software package to analyze thermodynamically the SRC and that of ORCs utilizing different working fluids. The influence of the working fluid on the ORC performance in terms of net power output, overall efficiency and Rankine efficiency has been presented.

It shows that with increase in heat source temperature overall efficiency, rankine efficiency, turbine work and power output increase in case of SRC system. R290 has larger overall efficiency and thermal efficiency over turbine efficiency range 70%-88% while it is similar in case of R600a and R134a. If the ORC will be operated with larger turbine efficiency, R290 is suggested for increase in overall efficiency or thermal efficiency. Through the simulation and calculation of power generation, we know that the power generation of the system increases with the increase in turbine efficiency.

As shown by figure 4.2.23, R290 has larger power generation capacity over the turbine efficiency range 70%-88% while it is approximately same in case of R134a and R600a. If the ORC will be operated for larger turbine efficiency, R290 is suggested .

R290 has larger overall efficiency over the pinch point temperature difference range of 10-25 °C. If the ORC will be operated for a larger pinch point temperature difference, R290 is suggested. Through the simulation and calculation of power generation, we know that the power generation of the system decreases with the increase in pinch point temperature difference.

R290 has larger power generation capacity over the pinch point temperature difference range of 10-25 °C while it is approximately same for in case of R134a and R600a. If the ORC will be operated for a larger pinch point temperature difference, hence R290 is suggested.

It is observed that when the heat source temperature is around 100–370 ºC, overall efficiency and Rankine efficiency of ORC system is greater than that of the SRC systems. It is observed that when the turbine efficiency is varied around 70–80%, overall efficiency and Rankine efficiency of ORC system is greater than that of the SRC systems.

It is observed that when the pinch point temperature difference is varied around 10–28ºC, overall efficiency and Rankine efficiency of ORC system is greater than that of the SRC systems. When the waste heat is free, thermal efficiency is only important as far as it is capable of reducing overall system cost per kW of net power produced. Higher refrigerant operating temperatures can easily result in higher equipment cost.

R290 gives lower discharge temperature which is important factor improving the life of compressor. Refrigerant viscosity is the major source of irreversibility and influences condensation and boiling heat transfer coefficients. R290 has lower viscosity and higher thermal conductivity which improves the performance of condenser and evaporator. The higher specific heat of R290 gives lower discharge temperature. The higher thermal efficiency can be because of the reason that the higher latent heat of evaporation indicates lower refrigerant mass requirement. The lower liquid density of R290 reflects the lower requirement of refrigerant mass resulting in lower friction and better heat transfer coefficients in evaporator and condenser.

VI. ACKNOWLEDGEMENT

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