International Journal of Engineering Technology and Applied Science
(ISSN: 2395 3853), Vol. 8 Issue 2 February 2022

Thermal Analysis of Organic Rankine Cycle Based on Different Fluid Type

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Abstract- The ORC works with a high molecular mass organic working fluid with the characteristic of having a phase change of liquid to vapour occurring at a temperature which is lower than the phase change of water to steam for a given pressure. The recovery of low grade heat can be achieved using organic fluids. These low grade heat sources can be from biomass energy, solar energy, geothermal energy and industrial waste. This work presents an analysis of organic Rankine cycle (ORC) using R143a, R600a, R134a and RC318 as working fluids under constant external conditions. An EES program has been developed to parametrically compare the first and second law efficiencies and turbine size factor with increase in turbine entry temperature, heat source temperature, turbine efficiency and pinch point temperature difference.

Keywords- ORC, EES, simulation model, Rankine cycle, First law efficiency, Second law efficiency, turbine size factor.

I. INTRODUCTION

Over the past years, the interest in recovering low grade heat has grown rapidly. Many researchers have come up with several ways of generating electrical power from low temperature heat sources available in solar energy, domestic boilers, biomass and industrial waste heat. Among all these the ORC is considered to be the most suitable due to its simple design and availability of components.

The ORCs use organic working fluids which are more suitable than water in the context of using heat source with low temperatures. The ORC cycle unlike conventional steam cycles is an attractive yardstick for local and small scale power generation.

Frank W. Of ledt patented the naphtha engine in 1883 which has the same application as the ORC. The naphtha was used in place of water as working fluid so as to replace the steam engine on boat [5]. During fractional distillation of crude petroleum oil, distinct liquid hydrocarbon naphtha is produced. Since the heat of vaporization for naphtha is lower compared to water, it was seen that if a certain amount of heat is added to the naphtha it produces more vapor and therefore, more work output could

be realized from the engine if water is used. There was a high risk of explosion when steam boats started using naphtha engine, for this reason the coast guides made it mandatory for operators tohave licenses which later resulted in the population growth of the naphtha engine [6]. The discovery by Frank W Ofledt was a substitute for using steam engines. Figure 2.1 shows an article about naphtha engine (1890) while figure 2.2 shows a simple design of naphtha engine.

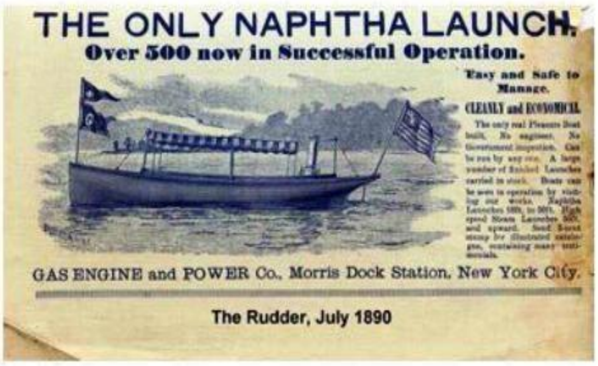

Figure 1: An article on naphtha engine.

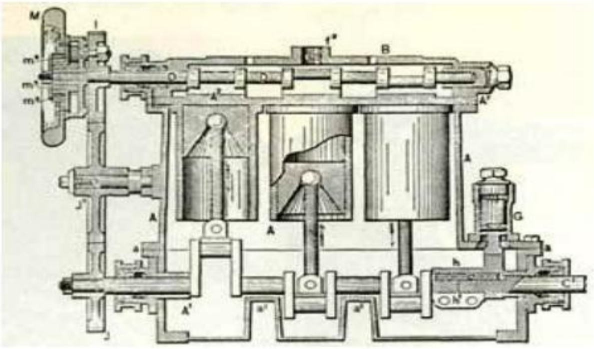

Figure 2: Sample design of naphtha engine.

The first prototype of the ORC system was first developed by Harry Zvi in the early 1960s [9]. This prototype was mainly used to recover low grade heat which is similar to the solar energy used to convert low temperature sources to electrical power. A turbine capable of working and operating at a comparatively low temperature was also developed by Harry Zvi. This invention was later privatized in 1965 by an Israeli company [10].

II. METHODOLOGY

1. System Modeling

To aid in analysis of engineering problem it is necessary to realize the Physical model in a mathematical model. To do this, we first write state point equations of thermodynamic properties and then develop a polynomial for thermodynamic properties with the help of software or, directly taken from the reference.

Therefore this chapter involves the description of physical model, mass, and energy balance, assumptions, state point equations and thermodynamic properties. We will calculate and compare thermal efficiency, generating capacity, etc. of power systems, namely AC-ORC under the same heat source conditions.

2. System Description Working Principle And Mathematical Model Of Orc

The ORC investigated in this paper is shown in Fig. 3. It consists of four different processes: process1-2 (expansion through turbine), process 2-3 (heat rejection in condenser), process 3-4 (pressurized in pump), and process 4-1 (heat addition in evaporator).

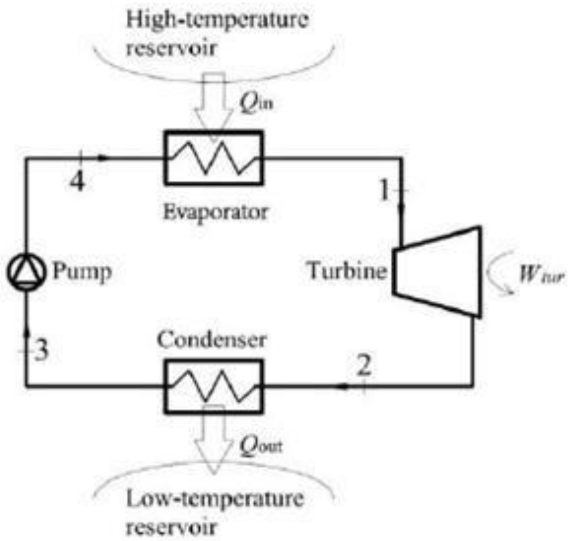

Figure 3: Schematic diagram of the organic Rankine cycle.

The analysis is based on the following assumptions:

1. The system is operating under steady-state condition,
2. No undesired pressure drop and heat loss occur in the system,
3. Working fluid at the evaporator and condenser exits is saturated, and
4. Isentropic efficiencies for the turbine and pump.

III. RESULT ANALYSIS

1. Effect of Heat Source Temperature

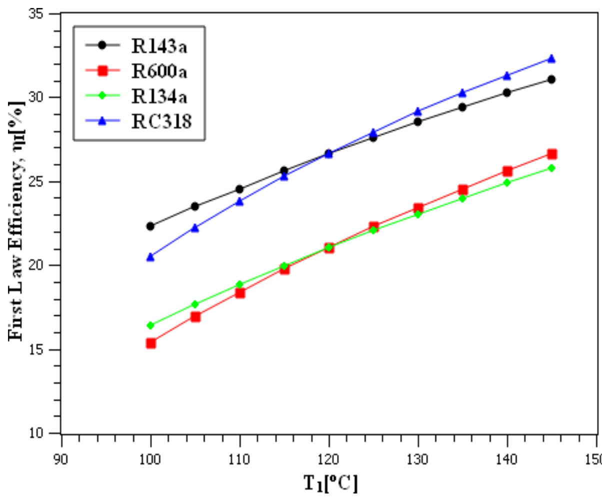

Figure 4: Variation of first law efficiency vs heat source temperature.

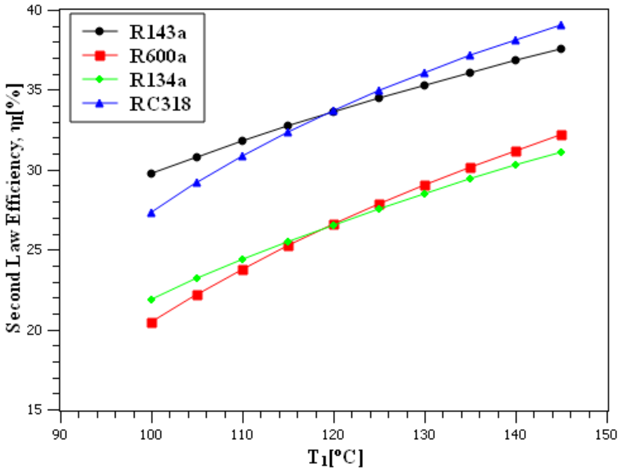

Figure 5: Variation of second law efficiency vs heat source temperature.

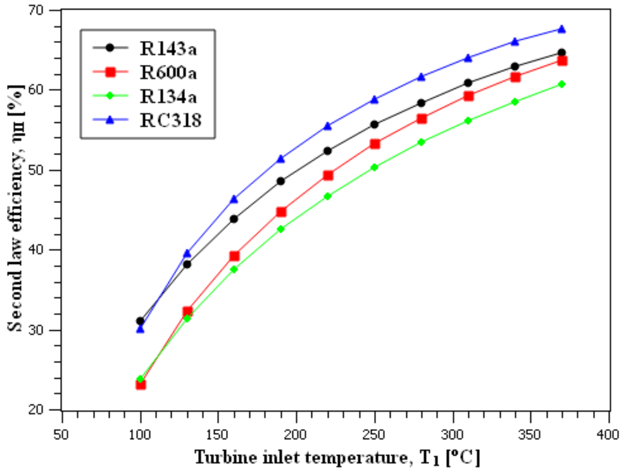

Figure 8: Variation Of Second Law Efficiency Vs Turbine Inlet Temperature.

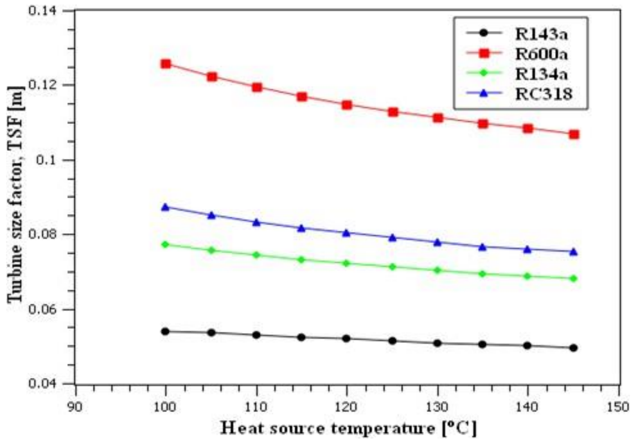

Figure 6: Variation of turbine size factor vs heat source temperature.

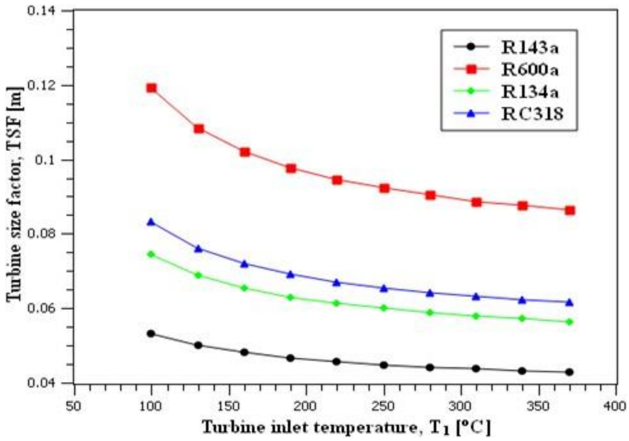

Figure 9: Variation Of Turbine Size Factor Vs Turbine Inlet Temperature.

2. Effect of Turbine Inlet Temperature

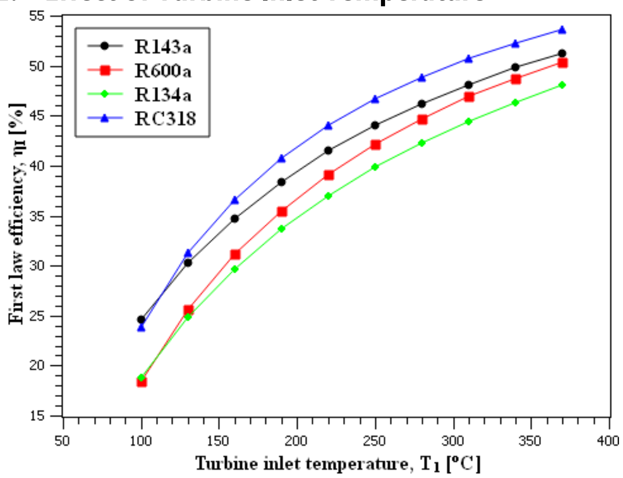

Figure 7: Variation Of First Law Efficiency Vs Turbine Inlet Temperature.

3. Effect of Pinch Point Temperature Difference

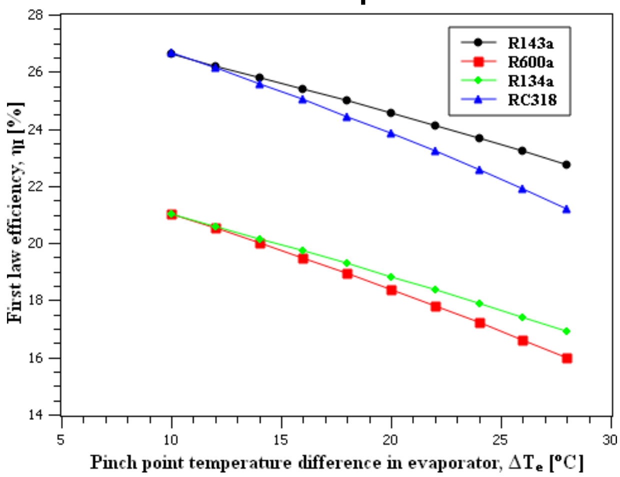

Figure 10: Variation of first law efficiency vs pinchpoint temperature difference.

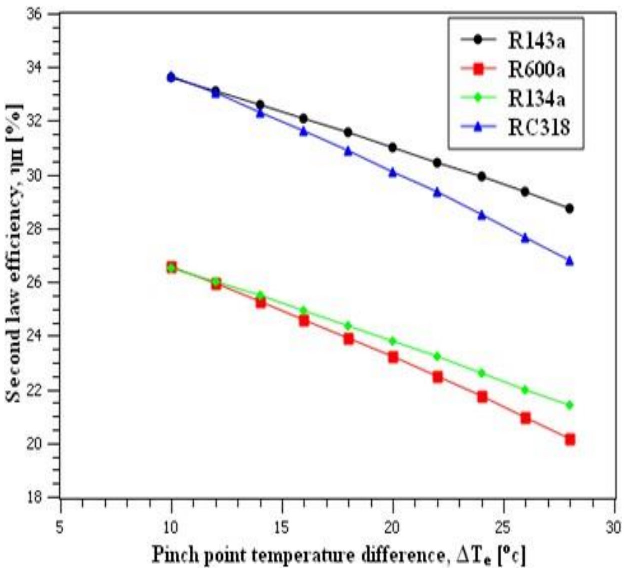

Figure 11: Variation of second law efficiency vs pinch point temperature difference.

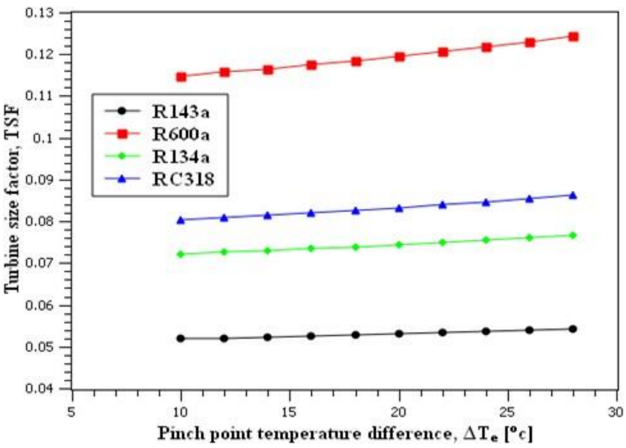

Figure 12: Variation of turbine size factor vs pinch point temperature difference.

4. Effect of Turbine Efficiency

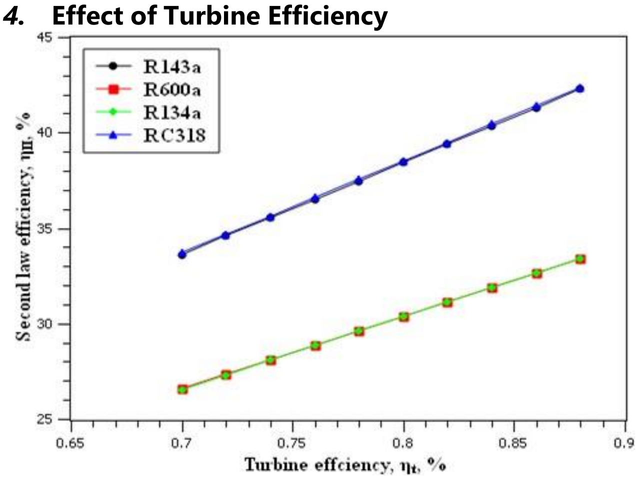

Figure 13: Variation of first law efficiency vs turbine efficiency.

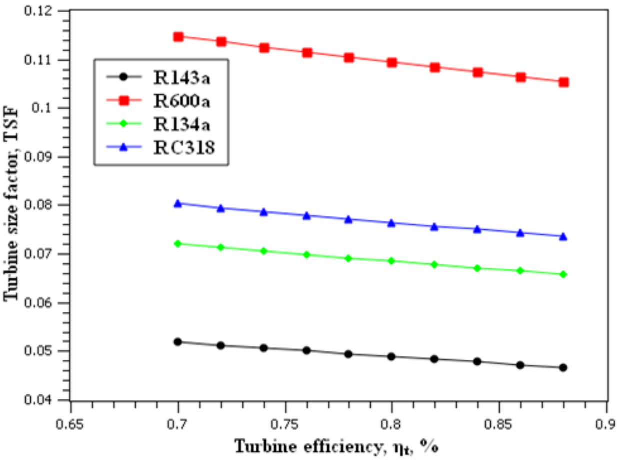

Figure 14: Variation of second law efficiency vs turbine efficiency.

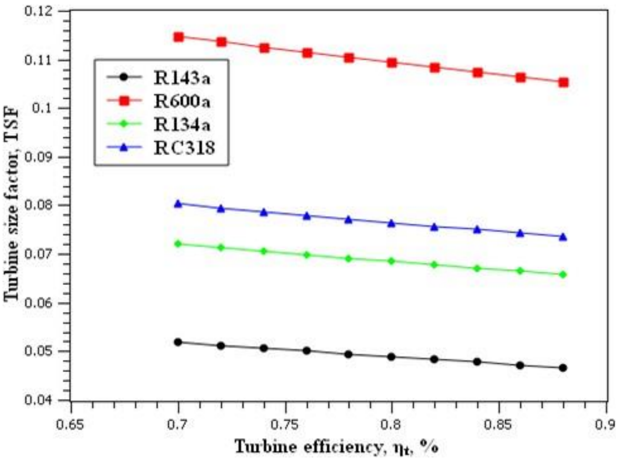

Figure 15: Variation of turbine size factor vs turbine efficiency.

IV. CONCLUSION

Based on the present analysis, it is found that

1. It is found that with different working fluids, the influences of the heat source temperature on the first law and second efficiency are similar. It shows that the first law and second law thermal efficiency increases monotonically with the increase in heat source temperature. With the heat source temperature rising from 100 to 145 ºC, the first law and second law efficiencies approximately increases for all working fluids. RC318 obtains the highest thermal efficiency, followed by R143a, R600a and R134a which shows relatively poor performance.

   It is seen that the turbine size factor always decreases as the heat source increases. For the conditions under consideration, small size factors are obtained for R143a at high TETs. R600a equires the largest size parameter due to the ver low evaporation pressure. Overall, R143a has the lowest turbine size parameter at all the TETs.

2. It is found that with different working fluids, the influences of the turbine inlet temperature on the first law and second law efficiency are similar. It shows that the first law and second law thermal efficiency increases monotonically with the increase in turbine inlet temperature. With the TET rising from 100 to 370ºC, the first law and and second law efficiencies approximately increases for R143a, R600a, R134a and for RC318. RC318 obtains the highest thermal efficiency, followed by R143a, R600a and R134a which shows relatively poor performance.
3. It is seen that the turbine size factor always decreases as the TET increases. For the conditions under consideration, small size factors are obtained for R143a at high TETs. R600a requires the largest size parameter due to the very low evaporation pressure. Overall, R143a has the lowest turbine size parameter at all the TETs.

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