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A Comparative Thermodynamic Analysis of the Rankine Cycle Using Various Organic Working Fluids

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Abstract

A comparative thermodynamic analysis of the Organic Rankine Cycle (ORC) using various working fluids was done using models developed in this work with the ASPEN Plus software to determine the thermal and exergy efficiencies of the ORC when running on six selected promising working fluids. The process was designed for power generation using a low heat source of about 40°C. Two layouts were considered; one was a typical ORC while the other was an ORC with a recuperator for heat energy savings. The process was run with two temperature conditions of 30 degrees superheat and 30°C at the turbine inlet. Six working fluids; three wet (R-152a, R-134a, and R-32)) and three dry (R-600, R-600a and R-245fa) were selected based on their physical, chemical, environmental and economic criteria. The results showed that no single fluid perfectly met all requirements but in a trade-off, their overall performances at the turbine inlet temperature of 30°C were preferred and R-32 emerged as the best ranked followed by R-600a and then R-134a; next was R-152a then R-600 and lastly R-245fa.

Keywords

Working fluids, Thermodynamic analysis, ORC, Thermal efficiency Exergy

1. Introduction

Energy conservation in the world is becoming very critical leading to the quest for sustainability and adequate measures for energy conversion and management. One of the main global concerns presently is the cost of energy for operation which can be reduced by minimizing energy losses due to irreversibilities thereby increasing efficiencies in the process units. One way to enhance energy efficiency is to determine where and to what degree energy is being wasted along a process route using thermodynamic analysis [1]

The global energy landscape is changing rapidly and shifting away from the use of fossil fuels to other green and clean energy sources in order to protect the environment and foster human and animal health. One of these improvements towards green energy is the use of organic fluids in the Rankine Cycle for power generation using low heat sources [2].

The Rankine cycle used in power plants with steam as the working fluid is one of the simplest and earliest forms of vapour power cycles. Substituting a hydrocarbon or a hydrocarbon-based fluid for water gave rise to the Rankine cycle power plant being termed "organic" with naphtha being the first organic fluid to be so deployed [3]. Naphtha was used commercially instead of steam in the earliest Organic Rankine Cycle (ORC) due to its lower heat of vaporization than that of water. A given amount of heat input produced more vapor from naphtha for the power cycle resulting in greater power output [3]. Since naphtha is sourced from fossil fuels (coal and petroleum oil), its use in the ORC is fast becoming undesirable and hence outdated.

The efficiency of the Rankine cycle is the ratio between the net work produced to the heat input at the boiler. This efficiency can be increased by lowering the condenser pressure, superheating the vapor to higher temperatures or increasing the boiler pressure. Most importantly however, the chosen working fluid for the Rankine cycle greatly affects the efficiency, running cost and depreciation rate of the components [4]. In order, therefore, to operate at optimum conditions there is the need for a comparative thermodynamic analysis of the Rankine cycle using various promising working fluids as addressed in this work.

Basically, the working fluid in a heat engine is a gas or liquid, that primarily converts thermal energy into mechanical energy in a cyclic process of vaporization and condensation while in a heat pump (refrigerator, air conditioner, cooling plant, etc.), where the working fluid is sometimes referred to as a refrigerant, coolant, or working gas, it transports thermal energy from a colder to a hotter region with the consumption of mechanical or electrical power. These fluids will all be referred to as *working fluids* in a generic sense in this work.

Four generations of working fluids (refrigerants) have been identified over the years according to the selection criteria that define them namely; whatever worked (1830-1930), safety and durability (1931-1990), stratospheric ozone protection (1990-2010) and global warming mitigation (2011-Date) [5]. A working fluid in any heat engine or heat pump system should possess certain chemical, physical and thermodynamic properties that would make it both economical and safe to use under prescribed conditions. The criteria for selecting working fluids hence include favourable thermodynamic and physical properties, thermal and chemical stability, environmental safety, availability, cost effectiveness, zero-toxicity and compatibility with the materials of construction and lubricant used.

A thermodynamic analysis is the assessment of the thermodynamic imperfections present in a system or process for the purpose of providing various means of avoiding or reducing their effects. There are various methods of thermodynamic analysis but in this work, the second law analysis and the exergy analysis were employed. Regarding the second law analysis, although the *ideal* Rankine cycle is internally reversible, the actual cycle is not because the turbine and pump do not operate isentropically also, there are finite heat transfers at the evaporator and condenser. Thermodynamic analysis through the second law reveals these irreversibilities and where they occur so that they could be minimized [6].

Exergy analysis on the other hand, is used to determine the amount of useful work (exergy) that is destroyed as a result of process irreversibilities when a system is brought into thermodynamic equilibrium with a reference environment [7]. Exergy is the total amount of useful work the system can perform [8, 9]. By exergy analysis, we determine the parts of the process where exergy is being destroyed so as to mitigate such losses.

This work is aimed at developing and employing a model for the determination and hence comparison of the thermal and exergy efficiencies that would be achieved in a Rankine cycle using different working fluids heated with a low heat source (< 40^{o}C).

2. Methods

Two layouts were considered in this work; the operational principle of the first layout (Layout-1) is the same as a conventional ORC whose simplified schematic diagram is shown in Figure 1, while a recuperator was included to obtain the second layout (Layout-2) shown in Figure 2.

2.1 Process Route Description

2.1.1 Layout-1

The steps in the components which constitute the process route for Layout-1 (without a recuperator) are as follows:

Step (1-2): *Pumping* - the working fluid as a subcooled liquid is pumped to the evaporator from the condenser.

Step (2-3): *Evaporation and superheating* - the working fluid enters the evaporator to absorb thermal energy from a heat source and undergoes phase change from subcooled liquid to saturated vapor and then to superheated vapour.

Step (3-4): *Expansion* - the superheated vapor enters the turbine where it expands to produce useful work. The working fluid leaves the turbine still as superheated vapor but with much lower enthalpy.

Step (4-1): *Condensation* - working fluid at low pressure enters the condenser where it rejects heat to the cold reservoir and undergoes phase change to subcooled liquid.

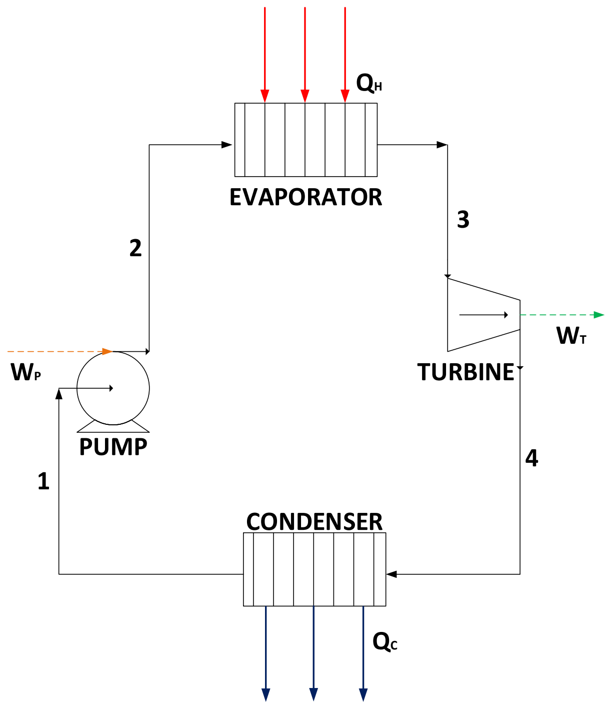

Figure 1. Layout-1 (Rankine Cycle).

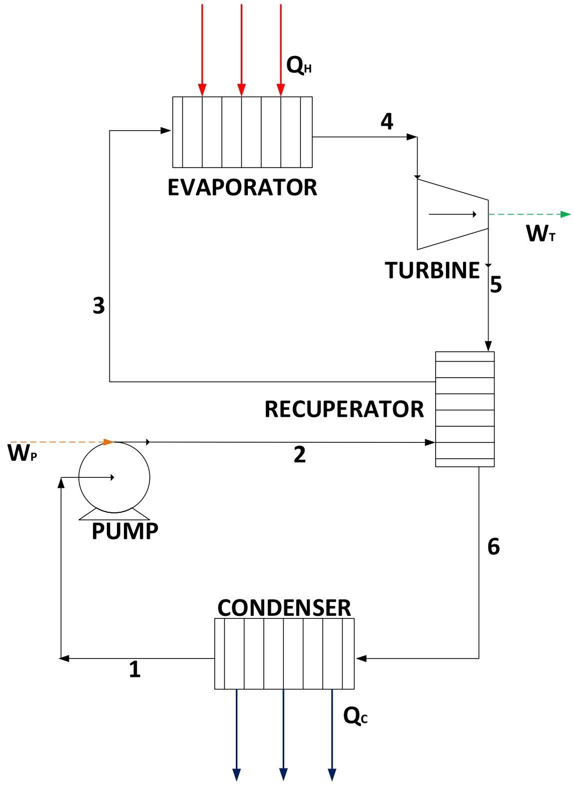

Figure 2. Layout-2 (Rankine Cycle with a Recuperator).

2.1.2 Layout-2

The steps comprising the process route for Layout-2 (with a recuperator) are as follows:

Step (1-2): *Pumping* - the subcooled liquid working fluid is pumped to the recuperator from the condenser.

Step (2-3): *Preheating* - heat is transferred in the counter-current heat exchanger (recuperator) from the super-heated vapor leaving the turbine to the subcooled liquid leaving the pump.

Step (3-4): *Evaporation* - the preheated working fluid enters the evaporator to absorb more thermal energy from a heat source where it undergoes phase change from subcooled liquid to saturated vapor and then to superheated vapor.

Step (4-5): *Expansion* - the superheated vapor enters the turbine where it expands to produce work. The working fluid leaves the turbine still as superheated vapor.

Step (5-6): *Precooling* - the stream from the turbine loses some heat in the recuperator to the subcooled liquid from the pump. At the exit of the recuperator, the working fluid is a saturated vapor.

Step (6-1): *Condensation* - at the condenser, the saturated vapor loses heat to the cold heat sink and undergoes phase change to subcooled liquid.

2.2 Process Modeling

The developed process models used to carry out the thermal and exergy analyses of the processes for both layouts are presented in this section. The general assumptions for the processes are:

- Steady state conditions.
- No pressure drop or heat loss in the evaporator, recuperator, condenser and pipes.
- Isentropic efficiency of both the pump and the turbine is 80%.

The processes were analyzed to determine and compare the thermal and exergy efficiencies of the cycles when operated with six selected promising working fluids shown in Table 1.

2.2.1 Model for thermal efficiency

The thermal efficiency model for Layout-1 is just as that of Layout-2 but excluding the pre-heating and the pre-cooling steps

The thermal efficiency model for Layout-2 was developed as follows:

Pumping (step 1-2)

$$W_P = \dot{m} \times (h_1 - h_2) \tag{1}$$

Pre-heating (step 2-3)

$$Q_{R1} = \dot{m} \times (h_3 - h_2) \tag{2}$$

Evaporation (step 3-4)

$$Q_E = \dot{m} \times (h_4 - h_3) \tag{3}$$

Expansion (step 4-5)

$$W_T = \dot{m} \times (h_5 - h_4) \tag{4}$$

Pre-Cooling (step 5-6)

$$Q_{R2} = \dot{m} \times (h_5 - h_6) \tag{5}$$

Condensation (step 6-1)

$$Q_C = \dot{m} \times (h_6 - h_1) \tag{6}$$

The thermal efficiency of the cycle is given by

$$\eta_{thermal} = \frac{W_T - W_P}{Q_E} \tag{7}$$

From the first law of thermodynamics, heat lost by the hot fluid is equal to heat gained by the cold fluid at the recuperator; therefore,

$$Q_{R1} = Q_{R2} \tag{8}$$

Hence,

$$(h_3 - h_2) = (h_5 - h_6) \tag{9}$$

But in real systems, due to irreversibilities, the value of T_3 cannot be equal to T_5 (except the recuperator is operating at 100% efficiency which is impossible). The efficiency of the recuperator is therefore:

$$\eta_{recuperator} = \frac{T_5 - T_6}{T_3 - T_2} \tag{10}$$

Equation (7) gives the thermal efficiency for both layouts (Figures 1 and 2)

2.2.2 Model for exergy efficiency

Since exergy is the potential for a system to cause a change as it achieves equilibrium with its surroundings, we define exergy here as the maximum useful work that can be extracted from the working fluid during the process in the equipment under consideration [10].

Exergy efficiency can also be expressed in terms of total exergy destruction and exergy supplied to the system. Exergy destruction rate of the Rankine cycle process steps for Layout-2 is given below while that of Layout-1 simply excludes the recuperator.

Pump

$$E_{DP} = \dot{m} \times T_0 \times [S_2 - S_1] \tag{11}$$

Evaporator

$$E_{DE} = \dot{m} \times \left[(S_4 - S_2) - \left(\frac{h_4 - h_2}{T_H}\right)\right] \tag{12}$$

Turbine

$$E_{DT} = \dot{m} \times T_0 (S_5 - S_4) \tag{13}$$

Condenser

$$E_{DC} = \dot{m} \times T_0 \times \left[(S_1 - S_6) - \left(\frac{h_1 - h_6}{T_L}\right)\right] \tag{14}$$

Recuperator

$$E_{DE} = \dot{m} \times T_0 \times [[(S_3 - S_2) - \frac{h_3 - h_2}{T_{R1}}] + [(S_5 - S_6) - \frac{h_5 - h_6}{T_{R2}}]]$$

$$E_{DR} = \dot{m} \times T_0 \times \left[\left[(S_3 - S_2) - \frac{h_3 - h_2}{T_{R1}}\right] + \left[(S_5 - S_6) - \frac{h_5 - h_6}{T_{R2}}\right]\right] \tag{15}$$

Hence,

$$E_{Dsystem} = E_{DP} + E_{DE} + E_{DC} + E_{DR} \tag{16}$$

For Layout-1 (without a recuperator),

$$E_{DR} = 0 \tag{17}$$

Also,

$$Exergy\ destryed = Exergy\ of\ Feed\ Stream - Exergy\ of\ Product\ Stream \tag{18}$$

Hence,

$$Exergy\ efficiency = \frac{Exergy\ of\ Product\ Stre\,am}{Exergy\ of\ Feed\ Stream} \times 100 \tag{19}$$

2.3 Selection of Working Fluids

The working fluids used in this work were required to be in their superheated state at about room temperature to save cost of evaporation and are particularly tailored for solar heat application. The six selected promising working fluids that met the necessary safety and economic criteria earlier mentioned are shown in Table 1.

Table 1. Selected Working Fluids and their Relevant Properties

Number	Name	Chemical Notation	Boiling point at 101.325Kpa, (^oC)	Freezing Point at 101.325Kpa, (^oC)	Critical Temp. (^oC)	Critical Pressure (Kpa)
R-32	Difluoromethane	CH_2F_2	-51.651	-136.81	78.105	5782.0
R-152a	1,1-difluoroethane	CH_3CHF_2	-24.023	-118.59	113.26	4516.8
R-600	Butane	$CH_3CH_2CH_2CH_3$	-0.49	-138.27	151.98	3796.0
R-600a	2-methylpropane (isobutane)	$HC(CH_3)_3$	-11.75	-159..42	134.66	3629.0
R-134a	1,1,1,2-tetrafluoroethane	CF_3CH_2F	-26.3	-26.074	-101.3	101.06
R-245fa	1,1,1,3,3-pentafluoropropane	$CF_3CH_2CHF_2$	15.3	-103	154.01	3651

[11]

2.4 Simulation

The thermodynamic analysis of the cycle was performed using a process simulator; ASPEN Plus to obtain data. The isentropic efficiency of the pump and turbine was set to 80% and the flowrate of the working fluid was set to 3,600 kg/hr. The property equation of state model used for each component was:

- Working fluid: SRK
- Water: STEAMNBS
- Carbon dioxide: PENG-ROB

For each of the six selected working fluids, simulation was done at 30 degrees superheat and at 30^oC turbine inlet temperature for Layuot-1 and for Layout-2 making a total of twenty-four simulations

The purpose of simulating the process at two different turbine inlet temperatures (30 degrees superheat and 30^oC) is to obtain data to be used to calculate the direction (lower or higher turbine inlet temperature) in which the working fluids give better thermodynamic efficiencies. Another reason is to account for the performance of the working fluid when the plant is not working at optimum condition (due to fouling, mechanical faults, etc.) and heat transfer rate is significantly reduced but sufficient to superheat the working fluids.

3. Results and Discussion

This section presents the results of the thermal and exergy analyses of the selected working fluids and the comparison of their performances.

3.1 Thermal Analysis

Thermal efficiencies of all working fluids generally increased with increase in turbine inlet temperature. This can be seen by comparing Figures 3 and 4. It is also clear that the inclusion of a recuperator in Layout-2 (Figures 3 and 4) yielded higher thermal efficiencies. This is obviously due to the preheating and precooling effect which resulted in some energy savings. The overall highest thermal efficiency of 5.6148% was obtained using R-32 with a recuperator at a turbine inlet temperature of 30^{0}C. This is also obviously because it has the largest superheat of 81 degrees (at turbine inlet temperature of 30^oC) compared to other fluids used here.

Reference [12] conducted a similar research using R-134a and R-152a as working fluids. For turbine inlet temperature range of 360K to 400K, R-134a without a recuperator had thermal efficiencies ranging from 8.5% to 9.5% while R-152a ranged from 9.5% to 10.5%. Considering Layout-1 without a recuperator in this work, the turbine inlet temperature of 273K and 303K for R-134a yielded a thermal efficiency of 4.62% and 4.72% respectively while for R-152a, at a turbine inlet temperature of 283K and 303K the thermal efficiencies of 4.75% and 4.87% were obtained. Since the thermal efficiencies of R-134a and R-152a, being wet fluids, increase with increase in turbine inlet temperature, the results can be said to closely correlate.

Exergy Analysis

The exergy analysis of each unit and that of the overall process (cycle) were calculated and are shown in Figures 5 and 6. For each working fluid, the exergy at 30 degrees superheat and at turbine inlet temperature of 30^{0}C were calculated for process layouts 1 and 2. Comparing all four conditions for each working fluid it was observed that Layout-2 (with a recuperator) had higher exergy efficiencies with the exception of R-134a. The overall highest exergy efficiency of 96.4667% was obtained for R-134a without a recuperator and at a turbine inlet temperature of 0^{o}C

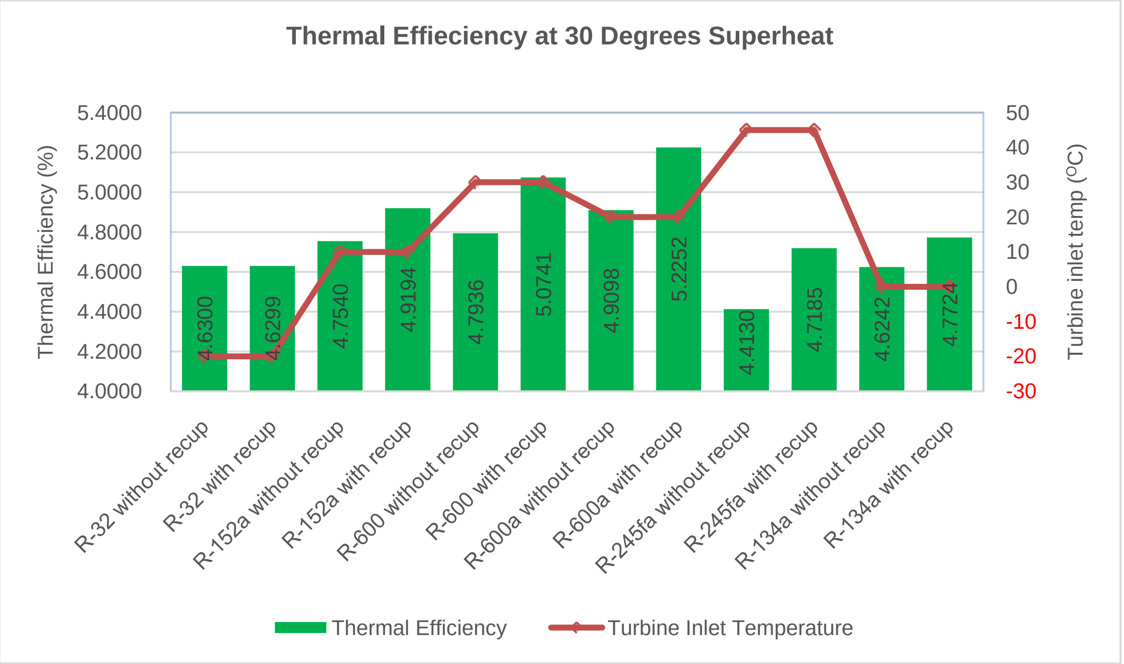

Figure 3. Thermal Efficiency at 30 Degrees Superheat.

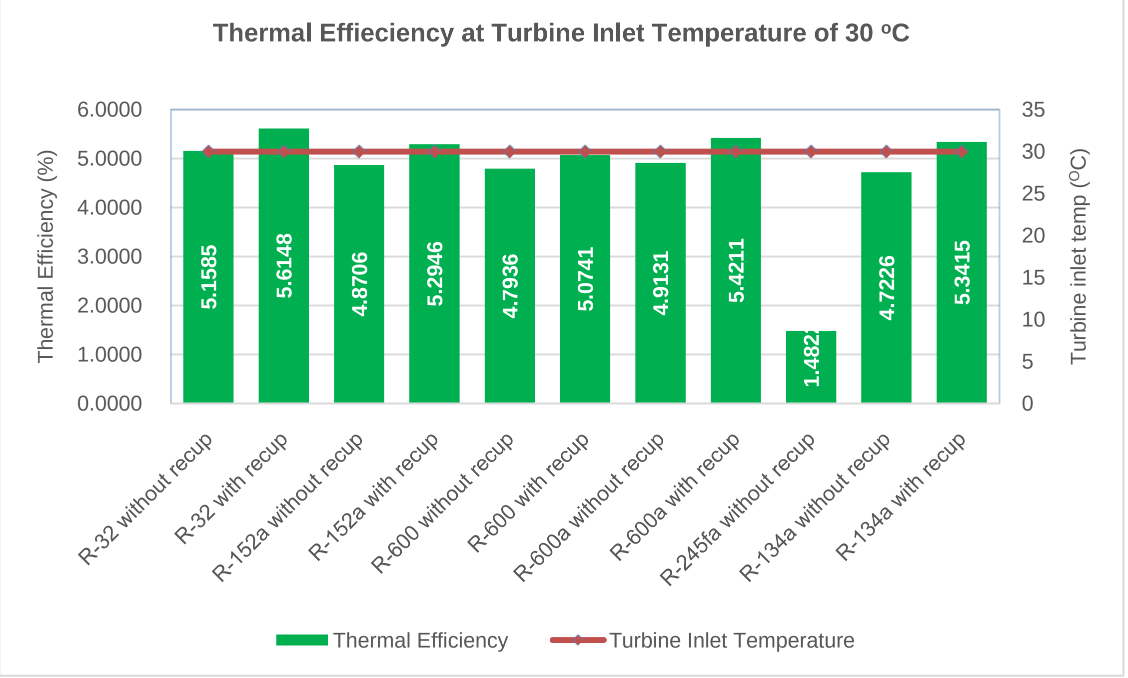

Figure 4. Thermal Efficiency at Turbine Inlet Temperature of 30^{o}C.

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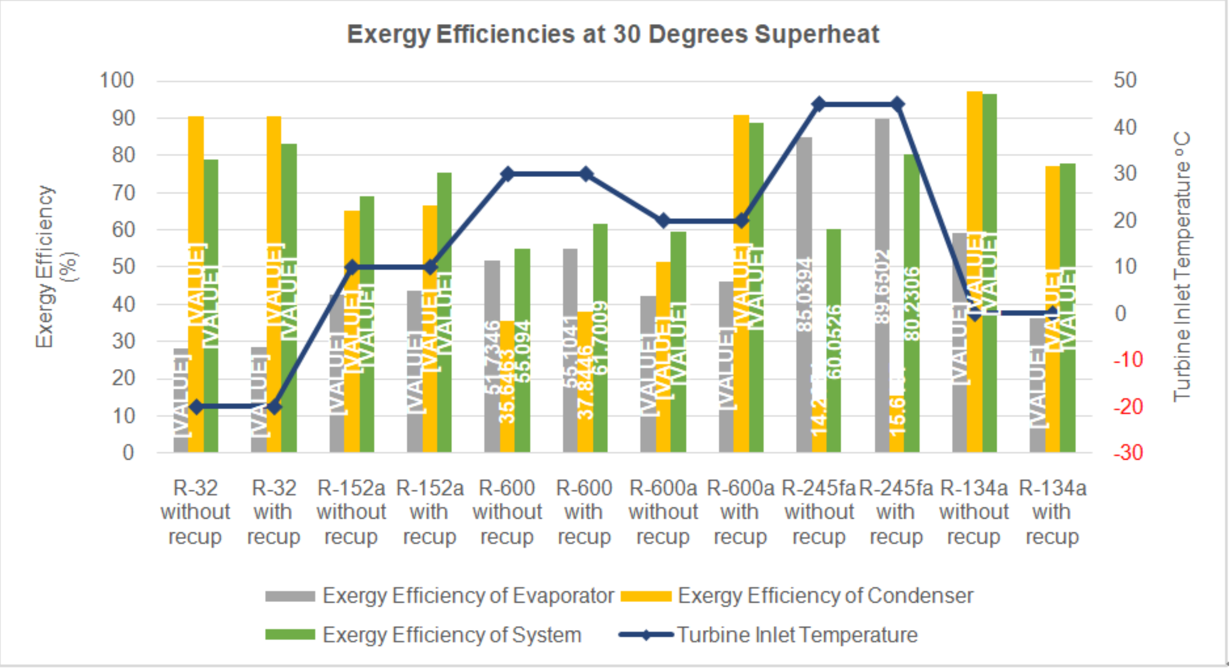


Figure 5. Exergy Efficiencies at 30 Degrees Superheat.

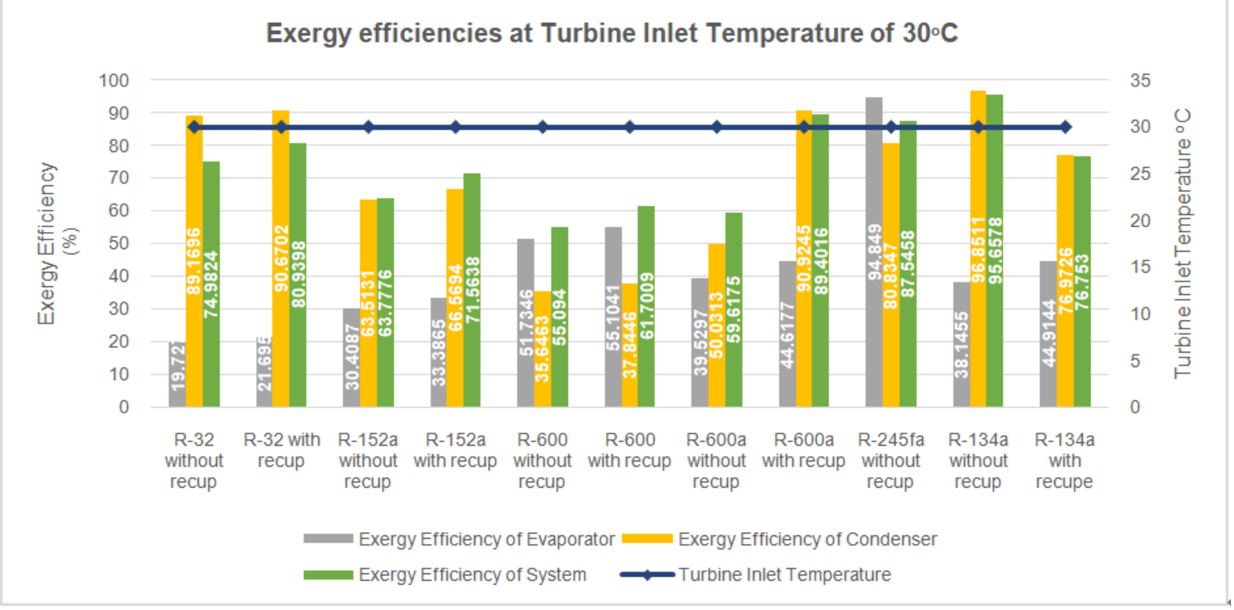


Figure 6. Exergy Efficiencies at Turbine Inlet Temperature of 30°C.

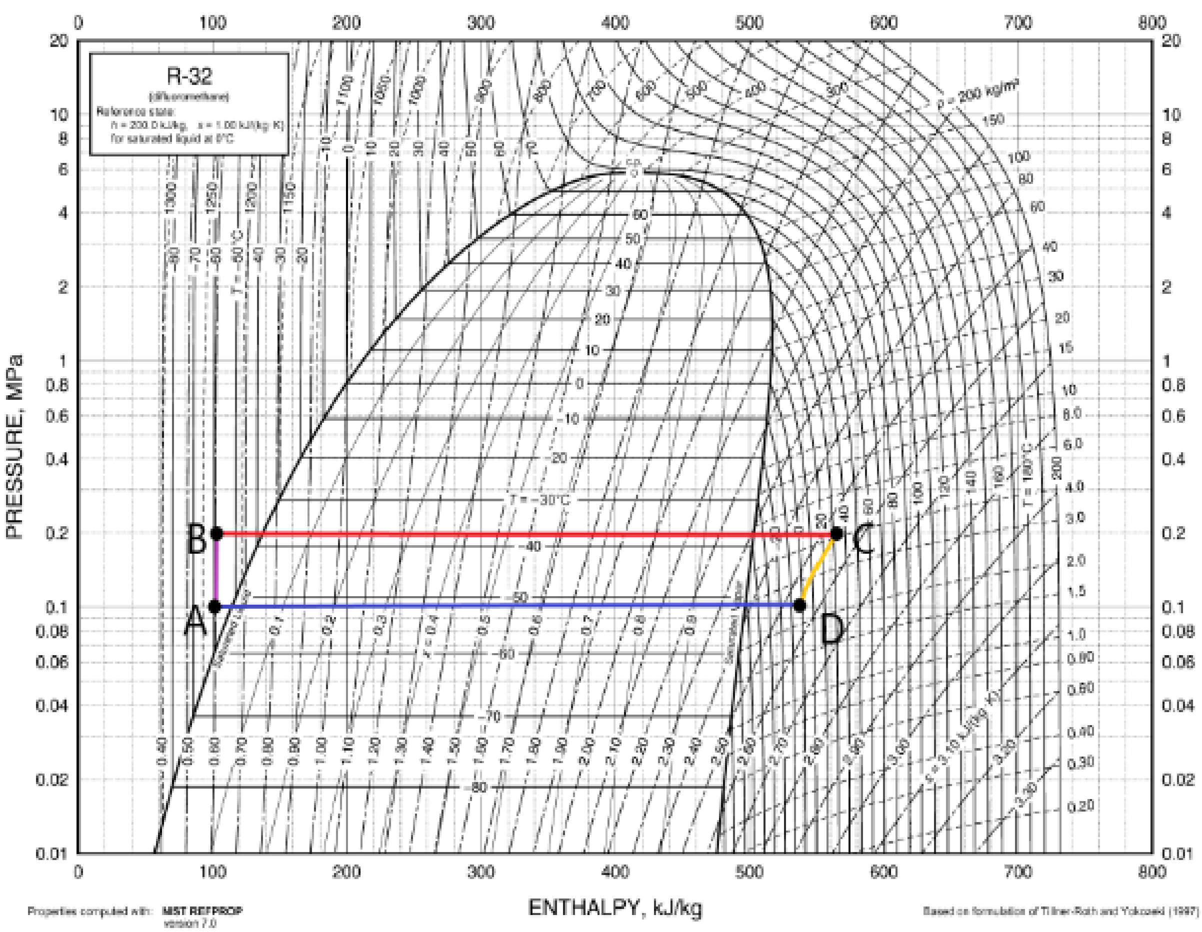
Exergy efficiency for processes at turbine inlet temperature of 30°C (higher turbine inlet temperature) was lower when compared to processes at 30 degrees superheat (lower turbine inlet temperature). Processes with and without a recuperator had almost similar exergy efficiencies at the pump and at the turbine for all working fluids. For the wet fluids (R-152a, R-134a, and R-32), the exergy efficiency at the evaporator was lower than at the condenser. For the dry fluids (R-600, R-600a and R-245fa), however, the reverse was the case as the exergy efficiency at the evaporator was higher than that at the condenser.

From [12] work, the exergy efficiency of R-152a obtained for turbine inlet temperature ranging from 360K to 400K was between 40% and 46%. Here also, exergy efficiency for R-152a was 75% and 63% at turbine inlet temperatures of 283K and 303K respectively with an inclusion of a recuperator.

R-245fa with the highest boiling point temperature of 15.3°C showed a very different behavior. Though its thermal efficiency at 30 degrees superheat was the overall lowest without a recuperator (4.4%), it gave a reasonable value of 4.7 with a recuperator as seen in Figure 3. At the simulation conditions of 30°C turbine inlet temperature however, it had a ridiculously low thermal efficiency of 1.4% without a recuperator and the system failed completely when run with a recuperator (Figure 4).

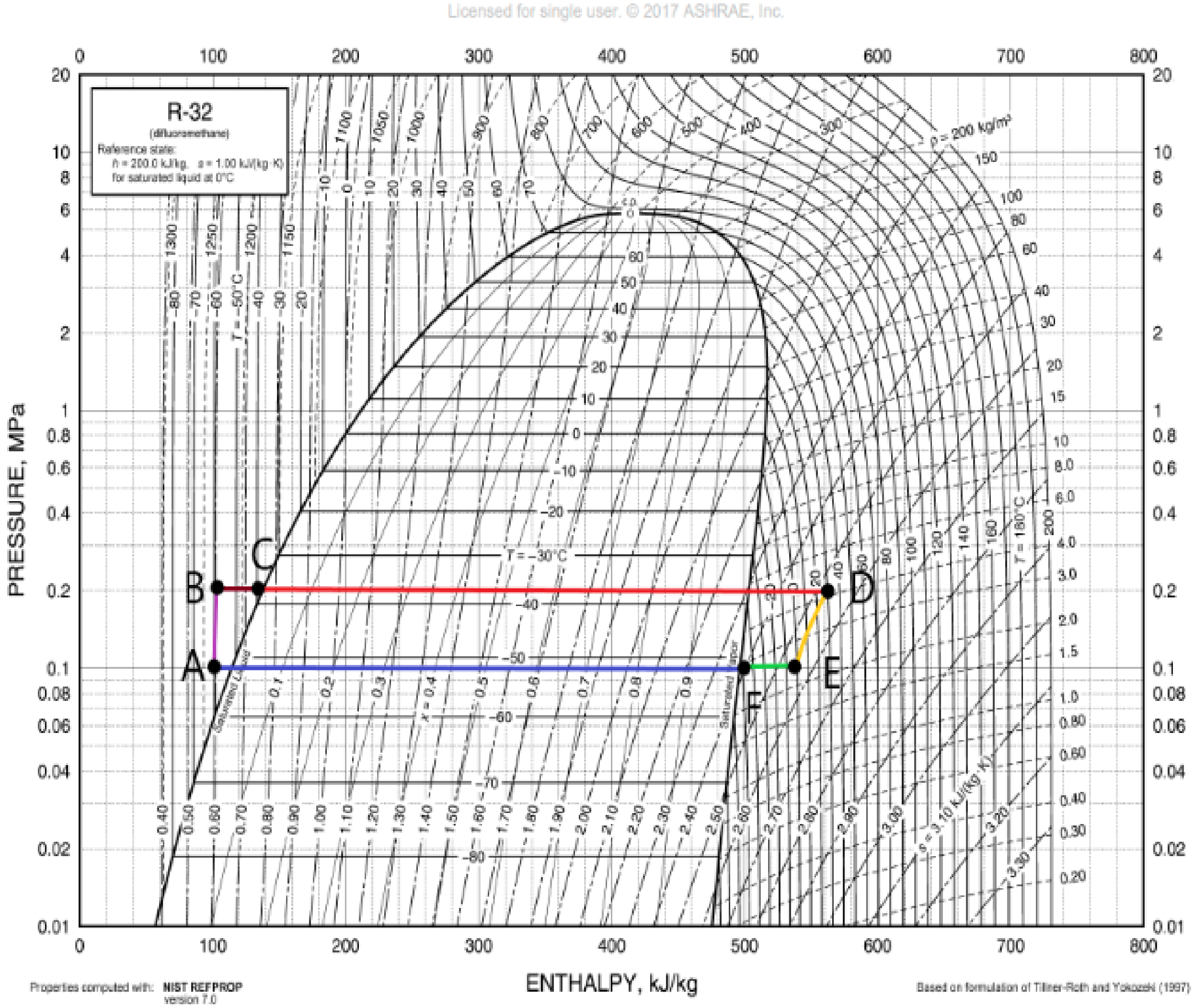
For the exergy efficiency, it gave extremely low values at the condenser at 30 degrees superheat both with a recuperator (14.2%) and without one (15.6%) as shown in Figure 5. At 30°C turbine inlet temperature (Figure 6), it failed to run successfully with a recuperator but gave a reasonable value of 87.5% for the cycle exergy efficiency without a recuperator; hence Figures 4 and 6 do not include any values for R-245a with a recuperator. By closely examining the error messages of the failed simulations, the process failure appears to be mainly due to the fluid crossing its saturated vapor curve into the two-phase region during the expansion step in the turbine and yielding liquid droplets which often cause damages in the turbine. This problem could be mitigated by blending this fluid with other compatible fluids to suit this specific application [13].

The cycle on the phase diagram for the highest ranked fluid among the six considered here, which is R-32, is shown in Figure 7 for Layout-1 and Figure 8 for Layout-2 indicating the phase changes undergone by the fluid as it traverses the entire cycle.



Purple line (AB) = gas compression at the pump,
Red line (BC) = heat addition at the evaporator,
Yellow line (CD) = gas expansion at the turbine,
Blue line (DA) = heat rejection at the condenser

Figure 7. Layout-1 for R-32 on its phase diagram. (Based on the formulation of [14], [15]).



Purple line (AB) = gas compression at the pump
Whine line (BC) = Heat addition at the Recuperator
Red line (CD) = heat addition at the evaporator
Yellow line (DE) = gas expansion at the turbine
Green line (EF) = heat absorption at the recuperator
Blue line (FA) = heat rejection at the condenser

Figure 8. Layout-2 for R-32 on its phase diagram. (Based on the formulation of [14], [15]).

4. Conclusion

From the above results and analyses, the following conclusion can be made:

A recuperator almost always increases the thermal efficiency and the exergy efficiency of working fluids.

The exergy efficiency of the cycle decreases as the turbine inlet temperature increases except for R600a while the thermal efficiency increases as the turbine inlet temperature increases.

From the results of this research, it was seen that there is no ideal working fluid that can meet all the necessary criteria for an ORC. In a trade-off among the thermodynamic properties, environmental effects, safety requirements and cost of the selected working fluids used in this work, the best ranked working fluid out of the chosen six was R-32a followed by R-600a, then R-134a; next was R-152a then R-600 and lastly R-245fa.

Nomenclature

W_P work done by the pump (consumed work)
W_T work done by the turbine (produced work)
Q_E heat added to the working fluid at the evaporator
Q_C heat lost by the working fluid at the condenser
Q_{R1} the heat added to the working fluid at the recuperator
Q_{R2} the heat lost by the working fluid at the recuperator
$\dot{m}$ the mass flowrate
h_1 the enthalpy at the pump inlet
h_2 enthalpy at pump outlet
h_3 enthalpy at recuperator outlet to the evaporator
h_4 vapour enthalpy out of the evaporator
h_5 vapour enthalpy at turbine outlet
h_6 enthalpy at condenser inlet
E_{DP} exergy destruction rate in the pump
E_{DE} exergy destruction rate in the evaporator
E_{DT} exergy destruction rate in the turbine
E_{DC} exergy destruction rate in the condenser
E_{DR} exergy destruction rate in the recuperator
$E_{D\ system}$ exergy destruction rate in the system
S_1 entropy at pump inlet
S_2 entropy at pump outlet
S_3 entropy at recuperator outlet to the evaporator
S_4 entropy at evaporator outlet
S_5 entropy at expander outlet
S_6 entropy at condenser inlet
T_0 ambient temperature in kelvin
$T_H = \frac{T_{IN} - T_{OUT}}{2}$ at the evaporator
$T_L = \frac{T_{IN} - T_{OUT}}{2}$ at the condenser
$T_{R1} = \frac{T_{IN} - T_{OUT}}{2}$ at the recuperator (from pump to evaporator)
$T_{R2} = \frac{T_{IN} - T_{OUT}}{2}$ at the recuperator (from turbine to condenser

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