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Thermodynamic models for combined cycle power plants used in organic Rankine & Brayton Cycles

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Abstract

As global energy demand continues to grow, combined cycle power plants are becoming more relevant**.** In combined cycles**,** the power output loss in gas turbines for ambient air temperatures over the standard 15˚C**,** the gas turbine power output is increased up to 30% and also the exhaust flow is increased. However, the exhaust temperature will be reduced due to the cooling. The reduction on the gas temperature partially compensates the increase in the exhaust flow and the final effect is an increase in the steam turbine power output of 2 – 5 %. Therefore in a combined cycle , the gas turbine power increased from 20 – 30% and steam turbine power increased to 2 – 5 % In this paper thermal models were developed for Rankine cycle using organic fluids, Brayton gas turbine cycle and its performance improvements using different methods using organic fluids and combined cycle and performance were carried out and it was found that the cycle efficiency is increasing on increasing of heater pressure for both R123 and R245fa and efficiency is coming more for R123 as compared to R245fa while the net-work is also increasing on increasing of pressure ratio but more network increases in R245fa as compared to R123..
Keywords: Thermodynamic Modelling, Subcritical ORC, combined cycle power plants, Organic Rankine & Brayton Cycles

1. Introduction

A combined-cycle power plant uses both a gas and a steam turbine together to produce up to 50 percent more electricity from the same fuel than a traditional simple-cycle plant. The waste heat from the gas turbine is routed to the nearby steam turbine, which generates extra power. In the combined cycle gas turbine (CCGT) plant, a gas turbine generates electricity and the waste heat is used to make steam to generate additional electricity via a steam turbine; this last step enhances the efficiency of electricity generation. Normally in the combined cycle power plants (CCPPs) a gas turbine generator generates electricity while the waste heat from the gas turbine is used to make steam to generate additional electricity via a steam turbine. The output heat of the gas turbine flue gas is utilized to generate steam by passing it through a heat recovery steam generator (HRSG), so it can be used as input heat to the steam turbine power plant. This combination of two power generation cycles enhances the efficiency of the plant. While the electrical efficiency of a simple cycle plant power plant without waste heat utilization typically ranges between 25% and 40%, a CCPP can achieve electrical efficiencies of 60% and more. Supplementary firing further enhances the overall efficiency. The high fuel utilization factor of the plant contributes to low lifecycle costs. The lot of work on thermodynamic analysis were carried out on the combined power plants using organic fluids as explained by various investigators including Sanjay Vijayaraghavan & Goswami et.al [1] developed new thermodynamic cycle for the simultaneous production of power and cooling from low temperature heat sources. The proposed cycle combines the Rankine and absorption refrigeration cycles, providing power and cooling as useful outputs. Initial studies were performed with an ammonia-water mixture as the working fluid in the cycle. This work extends the application of the cycle to working fluids consisting of organic fluid mixtures. Organic working fluids have been used successfully in geothermal power plants, as working fluids in Rankine cycles. An advantage of using organic working fluids is that the industry has experience with building turbines for these fluids. A

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commercially available optimization program has been used to maximize the thermodynamic performance of the cycle. And also discussed the advantages and disadvantages of using organic fluid mixtures as opposed to an ammonia-water mixture and also found that the thermodynamic efficiencies achievable with organic fluid mixtures, under optimum conditions, are lower than those obtained with ammonia-water mixtures, and the refrigeration temperatures achievable using organic fluid mixtures are higher than those using ammonia-water mixtures. Mishra R.S. and Dharminder Sahu [2] carried out energy and exergy analysis of Gas turbine-Organic Rankine combined cycle with solar reheating of organic fluid is done and results are compared with simple combined cycle, combined cycle with regeneration, combined cycle with solar reheating of organic fluid. The performance of the system is compared using different organic fluids (i.e. R134a, R245fa, Acetone, and R1234yf) at different organic Rankine cycle and it is found that the maximum pressure and maximum temperature , the R1234yf shows maximum increase in efficiency by regeneration about 70%. Acetone shows maximum organic Rankine cycle efficiency of 25.96%. Exergetic efficiency of combined cycle with regeneration and reheating 64%, 72%, 64%and 82% for R134a, R245fa, R1234yf and Acetone respectively. Acetone is recommended for practical applications due to its highest energy and exergetic efficiency among all selected organic fluids but some important problem related to flammability and explosion risk have to be considered while managing it. After Acetone, R245fa can be considered as better option in solar reheated combined cycle plant with regeneration. Al-Sulaiman Fahad A, Hamdullahpur Feridun, Dincer Ibrahim"[3] carried out detailed exergy analysis of selected thermal power systems driven by parabolic trough solar collectors (PTSCs) is presented. The power is produced using either a steam Rankine cycle (SRC) or a combined cycle, in which the SRC is the topping cycle and an organic Rankine cycle (ORC) is the bottoming cycle and examined using seven organic fluids (i.e. R134a, R152a, R290, R407c, R600, R600a, and ammonia) . and found that the R134a combined cycle demonstrates the best exergetic performance with a maximum exergetic efficiency of 26% followed by the R152a combined cycle with an exergetic efficiency of 25%. Alternatively, the R600a combined cycle has the lowest exergetic efficiency, 20–21%. In this paper thermal models were developed for Rankine cycle using organic fluids, Brayton gas turbine cycle and its performance improvements using different methods using organic fluids and combined cycle and performance were carried out.

2. Thermodynamic Modelling

This chapter deals with the set of equations that are used for the different steps of analysis to be performed. They are presented in order of the showed system performance.

2.1 Energy Analysis

The Ist law of thermodynamics is the basis for the energy analysis. The final results includes the net output and the thermal efficiency. As already mentioned in the assumptions, their value is dependent only on evaporation pressure i.e. Pevap = P1 = P8. Following are the set of equations for different components.

2.1.1 Subcritical ORC

The equation for heater/heat exchanger:

$$Qhex= \dot{m}hf\ (h_{11} – h_{14}) = \dot{m}cf(h_1 – h_8) \quad (1)$$

The equation of the expander

$$\eta_{T,i} = (h_2 - h_1) / (h_3 - h_1) \quad (2)$$
$$W_T = \eta T,m*(h_1 – h_2) \quad (3)$$

Where ηT,mis the mechanical efficiency of expander.
The equation of condenser:

$$Q_{cond} = \dot{m}cf\ (h_{18} – h_{15}) = \dot{m}r(h_2 – h_6) \quad (4)$$

The equation of fluid pump:

$$W_{pump} = (h_8 – h_6) \quad (5)$$

Equation for the net system output:

$$W_{sys} = \dot{m}hf(h_1 – h_2) - \dot{m}cf(h_8 – h_6) \quad (6)$$

Equation for finding the cycle thermal efficiency:

$$\eta_{th} = W_{sys}/Q_{hex} \quad (7)$$

Taking into account the effect of IHE, the equation (1) and (4) are changed to

$$Q_{hex} = \dot{m}_{hf*(}h_{11} – h_{21}) = \dot{m}r*(h_1 – h_{19}) \quad (8)$$
$$Q_{cond} = \dot{m}_{cf*}(h_{22} – h_{15}) = \dot{m}r*(h_{20} – h_6) \quad (9)$$

For the Internal Heat Exchanger:

$$(h_{20} – h_2) = (h_8 – h_{19}) \quad (10)$$

2.1.2 Subcritical ORC Using R245fa

The equation for heater/heat exchanger:

$$Q_{hex} = \dot{m}_{hf}(h_{11} – h_{14}) = \dot{m}cf(h_1 – h_8) \quad (11)$$

The equation of the expander

$$\eta_{T,i} = (h_2 - h_1) / (h_3 - h_1) \quad (12)$$

$W_T = \eta_{T,m*}(h_1 - h_2)$ (13)
Where ηT,mis the mechanical efficiency of expander.
The equation of condenser:

$Q_{cond} = \dot{m}_{cf}(h_{18} - h_{15}) = \dot{m}_r(h_2 - h_6)$ (14)

The equation of fluid pump:

$W_{pump} = (h_8 - h_6)$ (15)

Equation for the net system output:

$W_{sys} = \dot{m}_{hf}(h_1 - h_2) - \dot{m}_{cf}(h_8 - h_6)$ (16)

Equation for finding the cycle thermal efficiency:

$\eta_{th} = (W_{sys}/Q_{hex})$ (17)

Taking into account the effect of IHE, the equation (11) and (14) are changed to

$Q_{hex} = \dot{m}_{hf*}(h_{11} - h_{21}) = \dot{m}r*(h_1 - h_{19})$ (18)
$Q_{cond} = \dot{m}_{cf*}(h_{22} - h_{15}) = \dot{m}_{r*}(h_{20} - h_6)$ (19)

For the Internal Heat Exchanger:

$(h_{20} - h_2) = (h_8 - h_{19})$ (20)

2.1.3 Simple Brayton Cycle

The energy equation of simple Brayton cycle are as follows:
For the compressor:

$W_C = (h_2 - h_1)$ (21)

Where WC is the work of compressor.

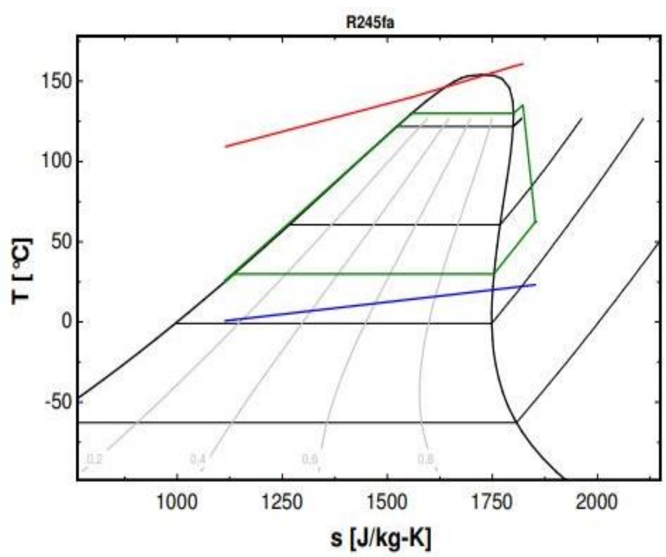

Figure 1: T-S diag. of Subcritical ORC system using R245fa

h_2 and h_1 is the enthalpy at the exit and inlet of the compressor respectively.
For the Combustion Chamber:

$Q_{in} = (h_3 - h_2)$ (22)

Where Qin is the heat input in the combustion chamber.
And h_3 and h_2 is the enthalpy at the exit and inlet of the combustion chamber respectively.
For the Turbine:

$W_T = (h_3 - h_4)$ (23)

Where WT is the work of the turbine.
And h3 and h4 is the enthalpy at the inlet and the exit of the turbine respectively.
For the cooler/ heat exchanger:

$Q_{out} = (h_4 - h_1)$ (26)

Where Q_{out} is the heat output in the cooler or heat exchanger.
And h_4 and h_1 is the enthalpy at the inlet and exit of the cooler respectively.
Net-work of the Brayton cycle:

$W_{NET} = (W_T - W_C)$ (27)

Isentropic Efficiency of the compressor:

$\eta_C = (h_{2,s}-h_1)/(h_2-h_1)$ (28)

Where $h_{2,s}$ is the isentropic enthalpy of the compressor.
Isentropic Efficiency of the turbine:

$\eta_T = (h_3-h_4)/(h_3-h_{4,s})$ (29)

Where h4,s is isentropic enthalpy of the turbine.
Cycle Efficiency of the Brayton cycle:

$\eta_{cycle} = (W_{NET}/Q_{in})$ (30)

2.1.4 Combined Cycle

$W_{net,combined} = W_{net,Brayton} + W_{net,Rankine}$ (31)
$E_{tacombined} = W_{net,combined} / (Q_{in,hex,Rankine} + Q_{in,combustor})$ (32)

2.2 Exergy Analysis

The drawback of energy analysis is that the conversion of energy of evaporator and condenser could not be done. Exergy analysis is an energy conversion coefficient that focuses on quantity as well as quality of energy, and this parameter more profoundly tells the essence of losses and energy conversion as compared to energy analysis. The exergy destruction in each component can also be calculated by this analysis.
The exergy analysis can be done by calculating the specific

flow exergy at exit and entry of sources and also at each state point in cycle. The following expression gives the quantity:

$$e = h - h_0 + T_0(s_0 - s) \tag{33}$$

Where the subscript 0 represents the environmental conditions which are considered to be 15ºC and atmospheric pressure. Following are the equations for different components:
Equation for Evaporator:

$$E_{d,evap} = \dot{m}_{hf*}(e_{11} - e_{14}) + \dot{m}_{r*}(e_8 - e_1) \tag{34}$$

Equation for expander:

$$E_{d,T}=\dot{m}_{r*}(e_1 - e_2) - W_T \tag{35}$$

Equation for condenser:

$$E_{d,cond} =\dot{m}_{r*}(e_2 - e_6) + \dot{m}_{cf}*(e_{15} - e_{18}) \tag{36}$$

Equation for working pump:

$$E_{d,P} = W_P + \dot{m}_{r*}(e_6 - e_8) \tag{37}$$

Expression for system total exergy destruction rate:

$$E_{d,Tot} = E_{d,evap} + E_{d,P} + E_{d,cond} + E_{d,T} \tag{38}$$

The overall exergy efficiency is calculated as:

$$\eta_{ex} = 1 - \{E_d,T_{ot} / \dot{m}_{hf} *(e_{11} - e_{14})\} \tag{39}$$

If internal heat exchanger is considered, then Eqn. 34 and 36 is changed to

$$E_{d,evap} = \dot{m}_{r*}(e_{19} - e_1) - \dot{m}_{hf*} (e_{21} - e_{11}) \tag{40}$$
$$E_{d,cond} = \dot{m}_{r*}(e_{20} - e_6) - \dot{m}_{cf*}(e_{22} - e_{15}) \tag{41}$$

Expression for internal heat exchanger:

$$E_{d,IHE} = \dot{m}_{r*}(e_8 - e_{19}) - \dot{m}_{r*}(e_{20} - e_2) \tag{42}$$

The equations 25 and 26 are modified as:

$$E_{d,Tot} = (E_{d,evap}+E_{d,P}+E_{d,cond}+E_{d,IHE} + E_{d,T} \tag{43}$$
$$\eta_{ex}= 1 - \{E_{d,Tot} / \dot{m}_{hf*}(e_{11} - e_{21})\} \tag{44}$$

3. Result and Discussion

3.1 Subcritical ORC

3.1.1 Effect of cycle efficiency

Fig.2 shows the effect of cycle efficiency by varying the heater pressure. It can be seen from the figure that by increasing the heater pressure the cycle efficiency is also increasing but there is a limit on increasing of heater pressure i.e. critical pressure of the fluid. In this study analysis is done on two fluids R123 and R245fa for subcritical case respectively. It can be seen from the fig. 2 that R123 is having more efficiency than R245fa at a same particular heater pressure. It can be seen from the fig. 2 the efficiency of the Rankine cycle at heater pressure 1326 kPa is 12.04% for fluid R123 and efficiency of Rankine cycle at heater pressure 1326 kPa is 9.08% for fluid R245fa.
So, from the results it can be seen R123 is having more efficiency than R245fa but the amount of chlorine content in R123 makes it unusable as it depletes the ozone layer and is not an ecofriendly fluid.

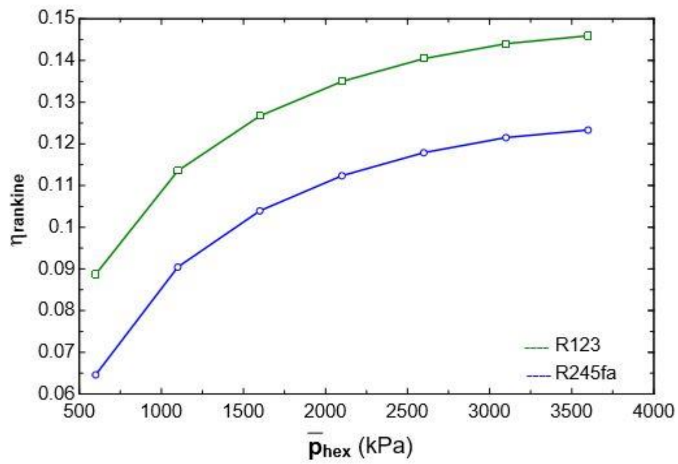

Figure 2: Variation of cycle efficiency with variation of heater pressure (kPa)

3.1.2 Effect on exergy flow rate

Fig. 3 shows the change of exergy flow rate of hot fluid by varying the heat pressure. The exergy flow rate of hot fluid is 67.75 kW at heater pressure of 1326 kPa for the fluid R123 and for fluid R245fa the exergy flow rate is 73.39 kW at heater pressure of 2127 kPa.

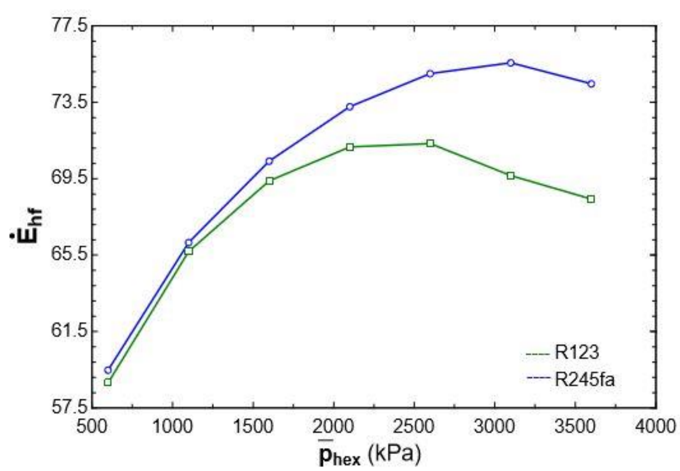

Figure 3: Variation of Exergy flow rate of hot fluid with variation of heater pressure

3.1.3 Effect on heater temperature

Fig.4 shows the change in heater temperature by varying the heater pressure. It can be seen from the Fig. 4 the heater temperature is 125 ºC at heater pressure 1326 kPa for fluid R123 and for fluid R245fa the heater temperature is 95 ºC at heater pressure 1326 kPa.

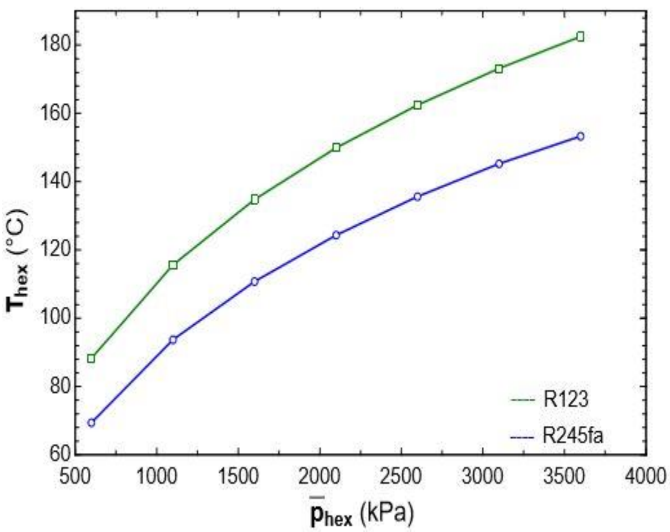

Figure 4: Variation of heater temperature with Variation of heater pressure

3.1.4 Effect of net-work

Fig.5 shows the graph between heater pressure and net-work output. It can be seen from the Fig.5 that the net-work output is 27.33 kW at heater pressure 1326 kPa for fluid R123 and 28.38 kW at heater pressure 2126 kPa for fluid R245fa. From the figure it can be seen that Net-work output is coming more for R123 than R245fa.

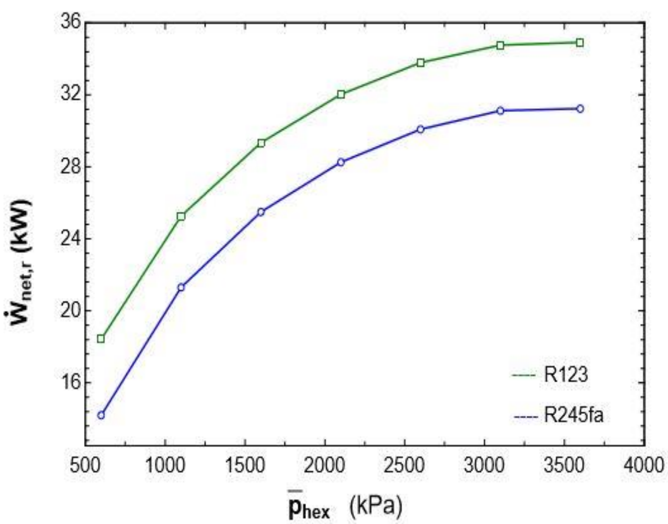

Figure 5: heater pressure Vs Net-work output

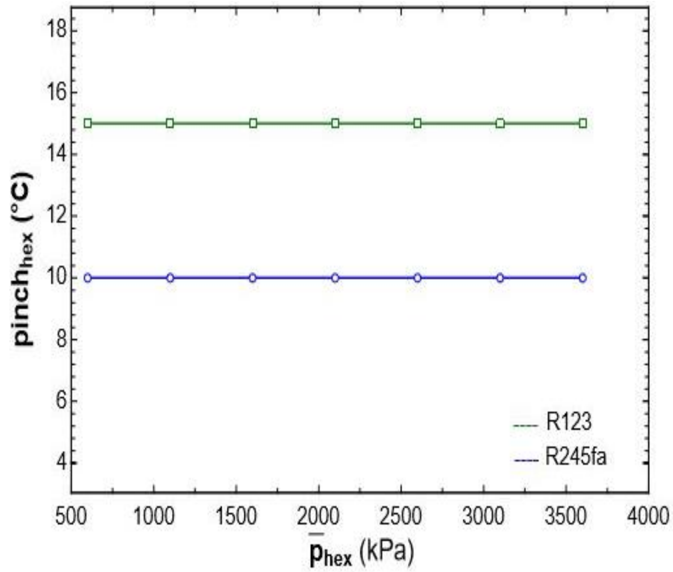

Figure 6: heater pressure Vs pinch point

3.1.5 Effect of pump work

Fig.7 shows the change in pump work by varying the heater pressure. It can be seen from the Fig. 7 that R123 fluid is taking less work to run a pump while R245fa fluid is taking more work to run a pump. The pump work is 1.047 kW at heater pressure 1326 kPa for R123 fluid and 1.832 kW for fluid R245fa at heater pressure 2127 kPa.

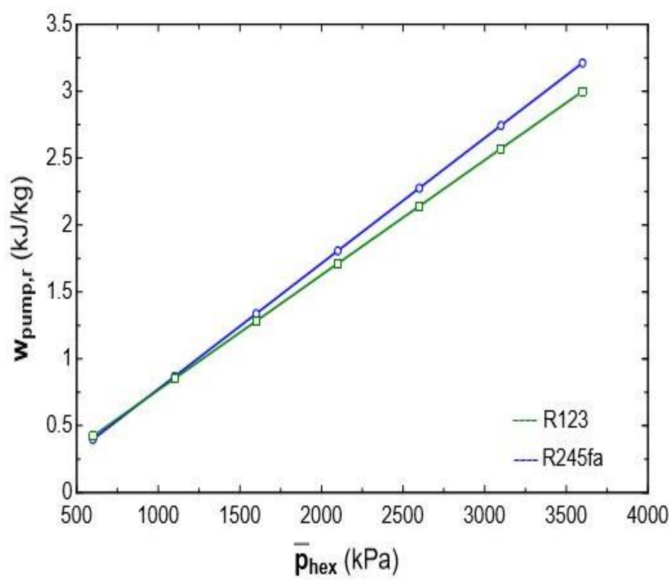

Figure 7: Variation of pump work with Variation of heater pressure

3.1.6 Effect of turbine work

Fig.8 shows the variation between heater pressure and turbine work. It can be seen from the Fig. 8 that fluid R123 is having more turbine work than fluid R245fa at same pressure. For fluid R123 the turbine work is 28.38 kW at heater pressure 1326 kPa. For fluid R245fa the turbine work is 24.56 kW at heater pressure 1326 kPa.

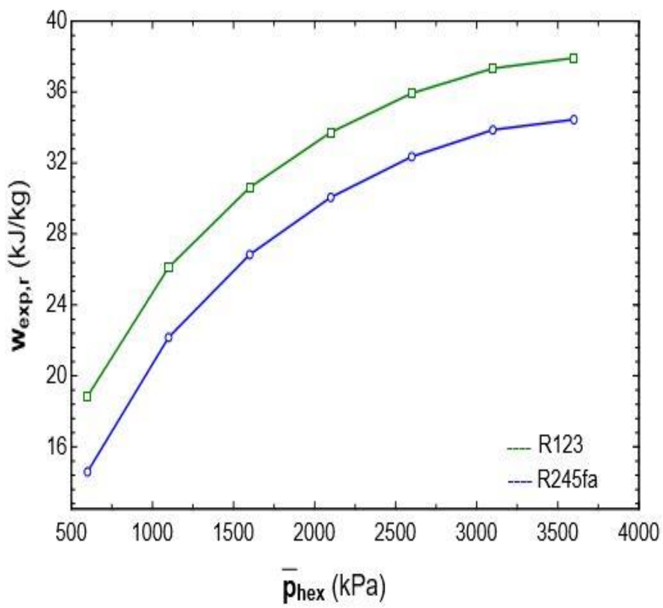

Figure 8: Variation of turbine work with Variation of heater pressure

3.1.7 Effect of second law efficiency

Fig.9 shows the graph between second law efficiency and heater pressure. It can be seen from the Fig. 9 that the second law efficiency is 40.41% at heater pressure 1326 kPa for fluid R123 and 38.67% at heater pressure 2127 kPa for fluid R245fa

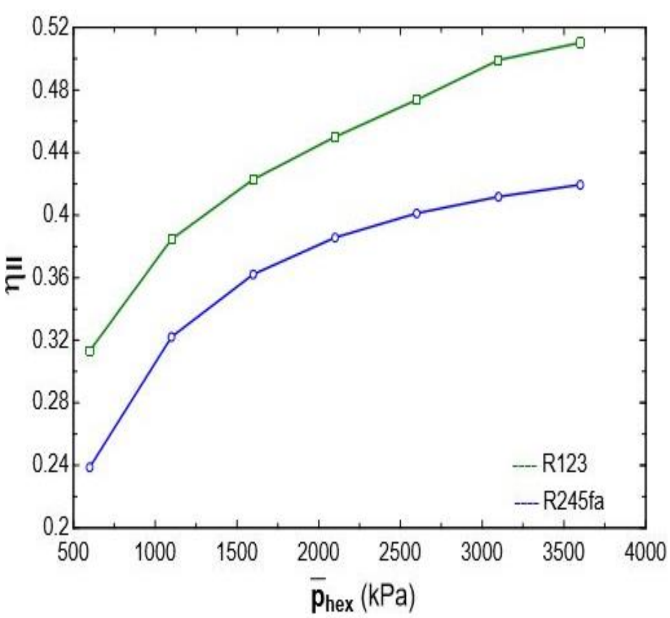

Figure 9: Variation of second law efficiency Variation of heater pressure

3.2 Simple Brayton Cycle

3.2.1 Effect of compressor inlet pressure

Fig.10 shows the effect of pressure at compressor inlet on the efficiency of Brayton cycle. The analysis is done on two fluid i.e. R123 and R245fa. It can be seen from the Fig.10 that for a given value of P1, increasing the turbine inlet temperature (T3) results in increase in efficiency of cycle. This is because ($\eta_{th} = 1 - T4/T3$) as the T3 increases, the denominator increases and the fraction decreases which will result in increasing the efficiency. It can be seen from the Fig. 10 the cycle efficiency is 3.601 % at 94 kPa compressor inlet pressure for fluid R123 and 11.41% at 95 kPa compressor inlet pressure for fluid R245fa.

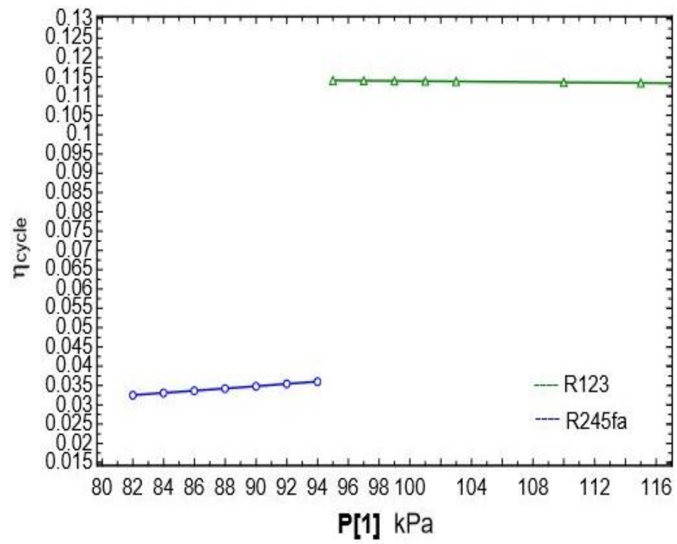

Figure 10: Variation of cycle efficiency with variation of compressor inlet pressure

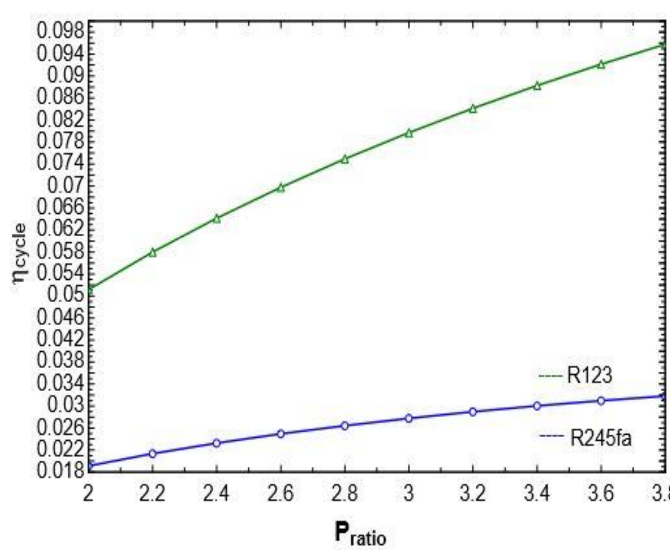

Figure 11: Variation of cycle efficiency with variation of Pressure ratio

3.2.2 Effect of pressure ratio

Fig.11 shows the variation between pressure ratio and cycle efficiency. It can be seen from the Fig.11 that the cycle efficiency is 2.896 % at pressure ratio 3.2 for fluid R245fa and 8.4 % for fluid R123 at 3.2 pressure ratio.

3.2.3 Effect of minimum operation Temperature

Fig.12 shows the graph between minimum operation temperature and cycle efficiency is 3.548 % at T1 = 25 ºC for fluid 245fa and 11.39 % for fluid R123.

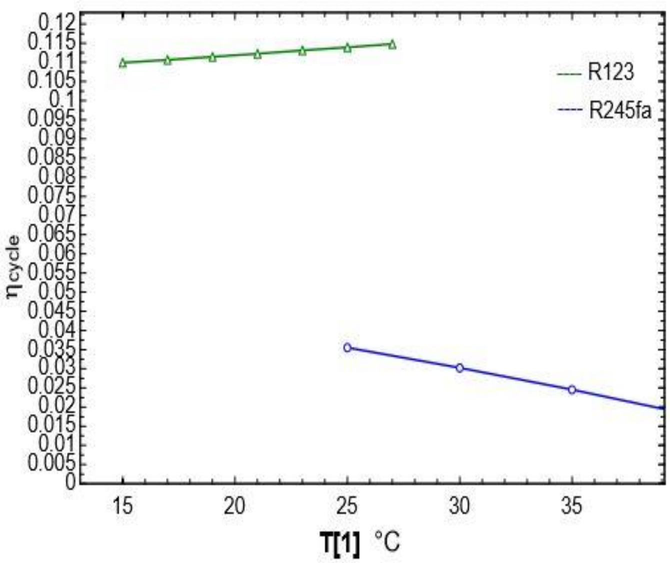

Figure 12: Variation of cycle efficiency with Variation of Minimum Operation temperature

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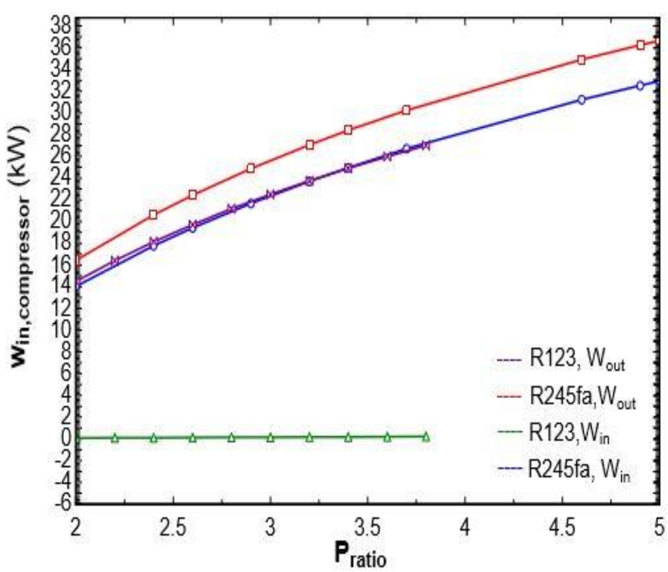

Figure 13: Variation of compressor and turbine works with Variation of Pressure ratio

3.2.4 Effect of pressure ratio on the compressor and turbine work

Fig.13 shows the variation between pressure ratio and compressor & turbine work. It can be seen from the Fig.13 that for fluid R123 the work input and work output is 0.1691 kW and 23.67 kW respectively at pressure ratio 3.2 and for fluid R245fa the work input and work output is 23.71 kW and 27.01 kW respectively at pressure ratio 3.2

3.3 Brayton cycle with Intercooling

The performance evaluation of Brayton Cycle with Intercooling is given below.

3.3.1 Effect of cooler pressure

Fig.14 show the variation of cooler pressure and cycle efficiency. It can be seen from the Fig. 14 the cycle efficiency is 11.39 % at 114 kPa cooler pressure for fluid R123 and 3.812 % at 114 kPa cooler pressure for fluid R245fa.

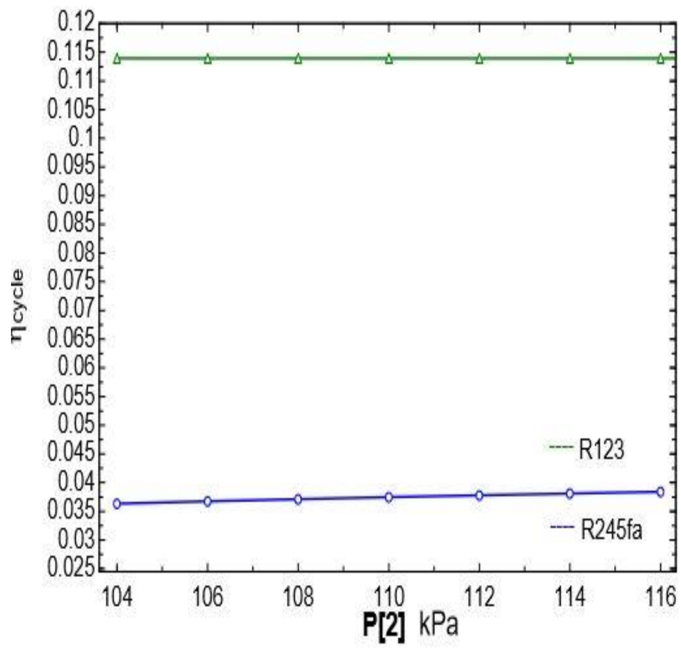

Figure 14: Variation of cycle efficiency with Variation of Cooler pressure

3.3.2 Effect of High pressure Inlet turbine temperature

Fig.15 shows the variation between turbine inlet temperature and efficiency of cycle. It can be seen from the Fig. 15 that the cycle efficiency is 11.47 % at 150 ºC for fluid R123 and 2.892 % at 150 ºC for fluid R245fa.

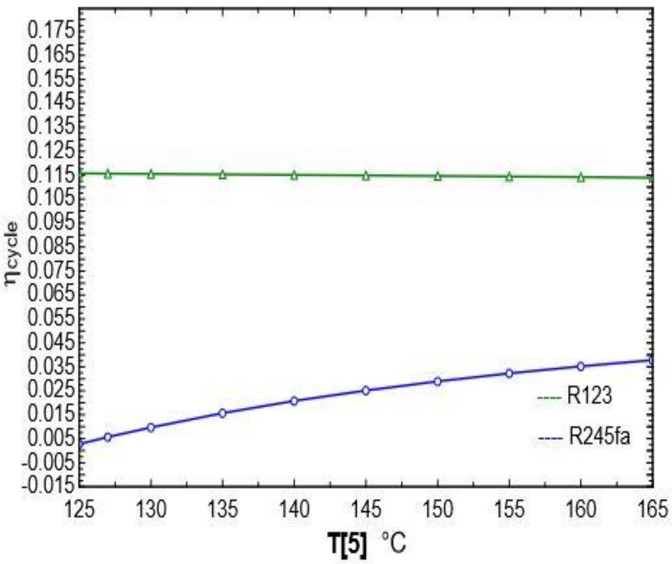

Figure 15: Variation of cycle efficiency with High pressure inlet turbine temperature

3.3.3 Effect of cooler pressure on cycle work

Fig.16 show the variation of cooler pressure and net-work. It can be seen from the Fig. 3.15 the net-work is 11.37 kW at 114 kPa cooler pressure for fluid R123 and 4.136 kW at 114 kPa cooler pressure for fluid R245fa.

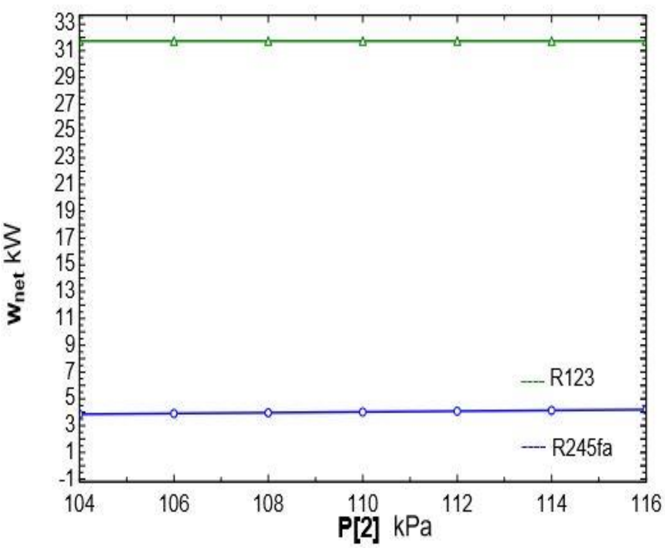

Figure 16: Variation of Cooler pressure Vs Variation of Net-work

3.3.4 The effect of pressure ratio

Fig 3.16 show the variation of pressure ratio and cycle efficiency. It can be seen from the Fig. 17 that the cycle efficiency is 8.412 % at pressure ratio 3.2 for fluid R123 and 3.054 % at pressure ratio 3.2 for fluid R245fa.

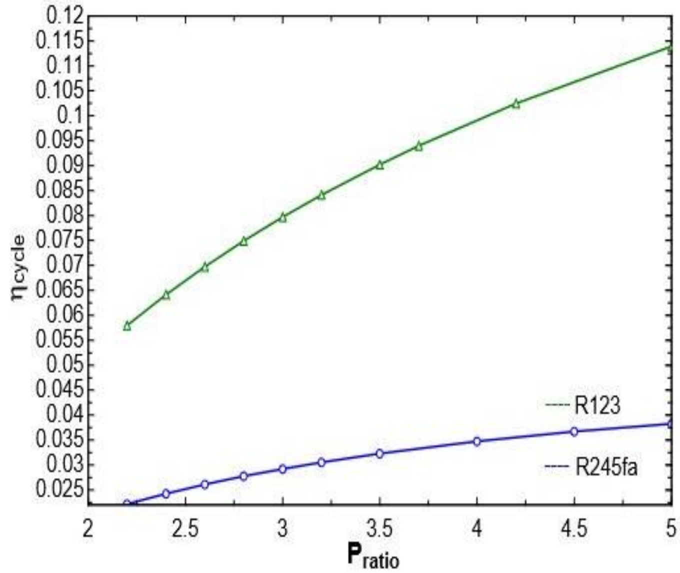

Figure 17: Pressure ratio Vs cycle efficiency

3.3.5 Effect of pressure ratio on compressor and turbine work

Fig.18 shows the variation between pressure ratio and compressor & turbine work. It can be seen from the Fig.18 that for fluid R123 the work input and work output is 0.1691 kW and 23.67 kW respectively at pressure ratio 3.2 and for fluid R245fa the work input and work output is 23.42 kW and 27.07 kW respectively at pressure ratio 3.2.

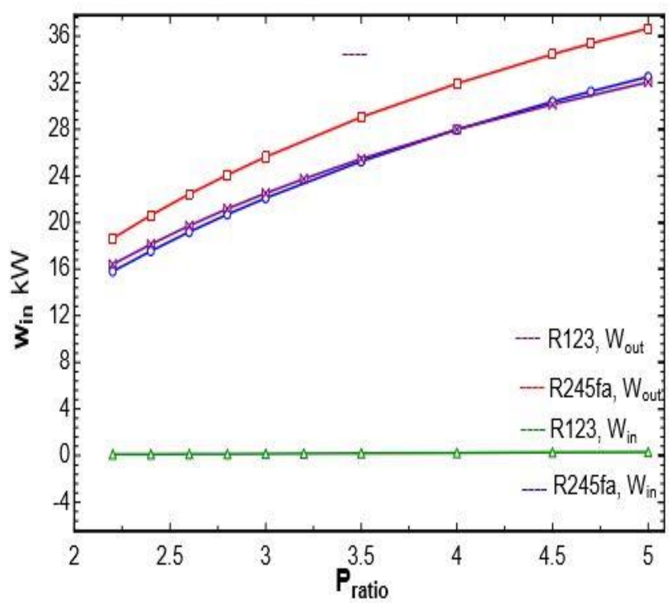

Figure 18: Variation of Pressure ratio Vs Variation of compressor and turbine work

3.3.6 The effect of Minimum cycle temperature

Fig.19 shows the graph between minimum operation temperature and cycle efficiency is 3.827 % at T1 = 25 ºC for fluid 245fa and 11.39 % for fluid R123.

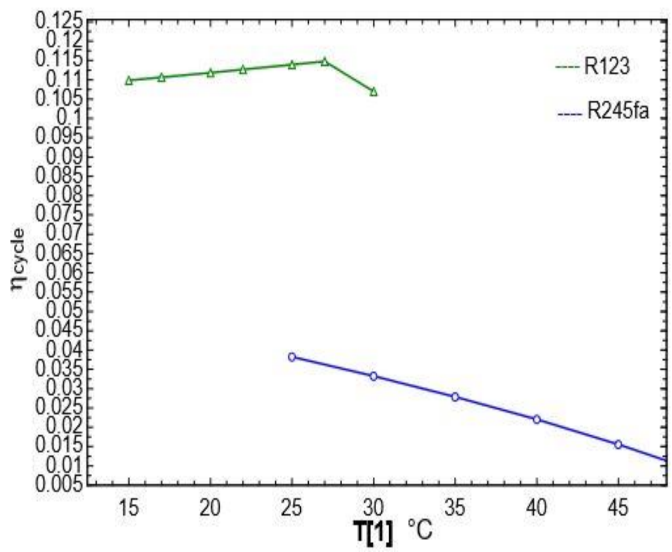

Figure 19: Variation of Minimum cycle temperature Vs Variation of cycle efficiency

3.4 Brayton with Reheating

3.4.1 The effect of Re-heater pressure

Fig. 20 show the variation of re-heater pressure and cycle efficiency. It can be seen from the Fig. 20 that the cycle efficiency is 10.69 % at 160 kPa re-heater pressure for fluid R123 and 3.372 % at 160 kPa re-heater pressure for fluid R245fa.

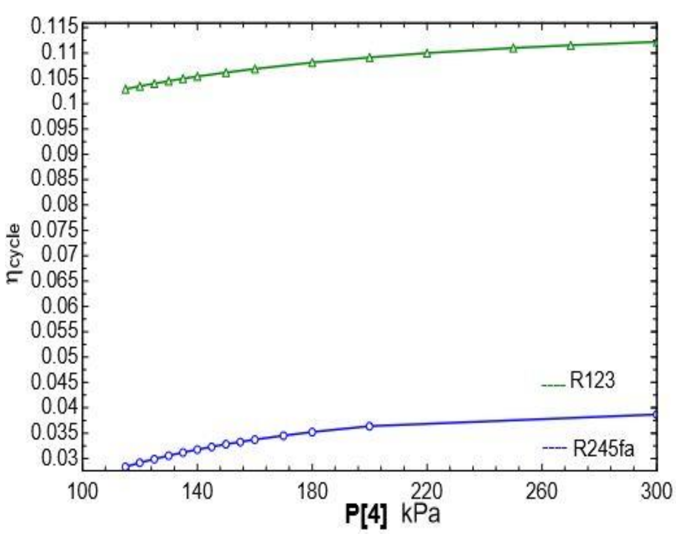

Figure 20: Variation of Re-heater pressure Vs Variation of cycle efficiency

3.4.2 The effect of high pressure turbine inlet temperature

Fig.21 show the variation of turbine inlet pressure and cycle efficiency. It can be seen from the Fig. 3.20 that the cycle efficiency is 10.31 % at 160 kPa turbine inlet pressure for fluid R123 and 2.562 % at 160 kPa turbine inlet pressure for fluid R245fa.

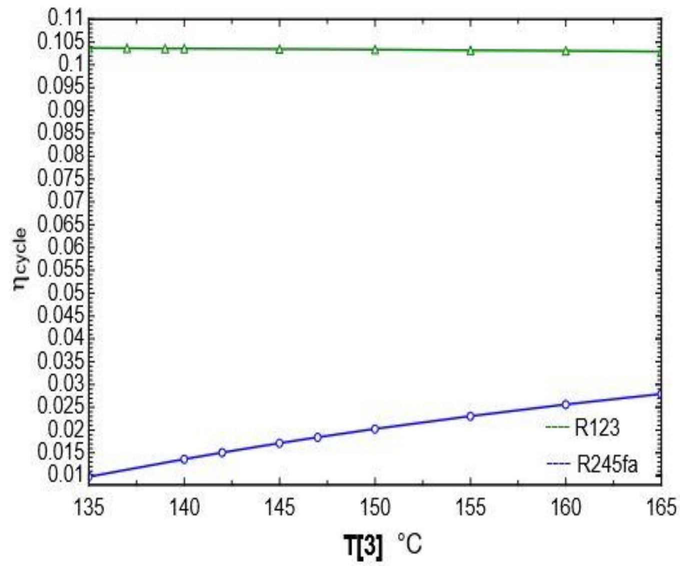

Figure 21: Variation of High pressure inlet turbine temperature Vs Variation of cycle efficiency

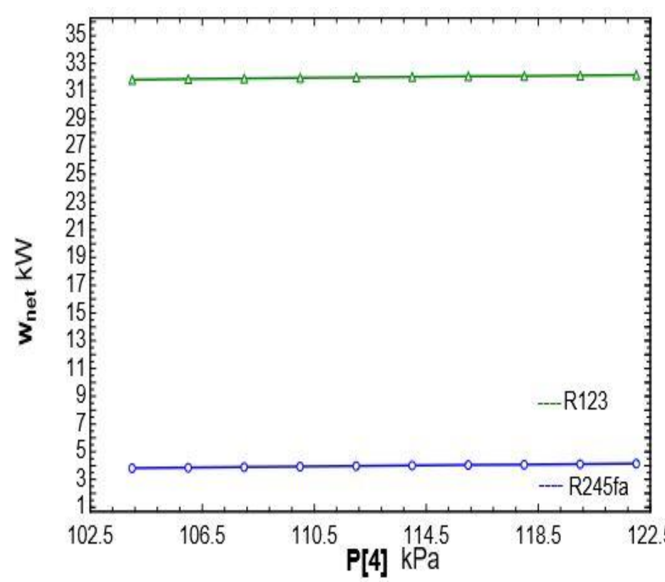

Figure 22: Variation of Re-heater pressure Vs Variation of Net-work

3.4.3 The effect of Re-heater pressure

Fig.22 show the variation of re-heater pressure and net-work. It can be seen from the Fig. 22 that the net-work is 32.13 kW at 120 kPa re-heater pressure for fluid R123 and 2.915 kW at 120 kPa re-heater pressure for fluid R245fa.

3.4.4 The effect of pressure ratio on cycle efficiency

Fig.23 show the variation of pressure ratio and cycle efficiency. It can be seen from the Fig. 23 the cycle efficiency is 7.4 % at pressure ratio 3 for fluid R123 and 2.4 % at pressure ratio 3 for fluid R245fa.

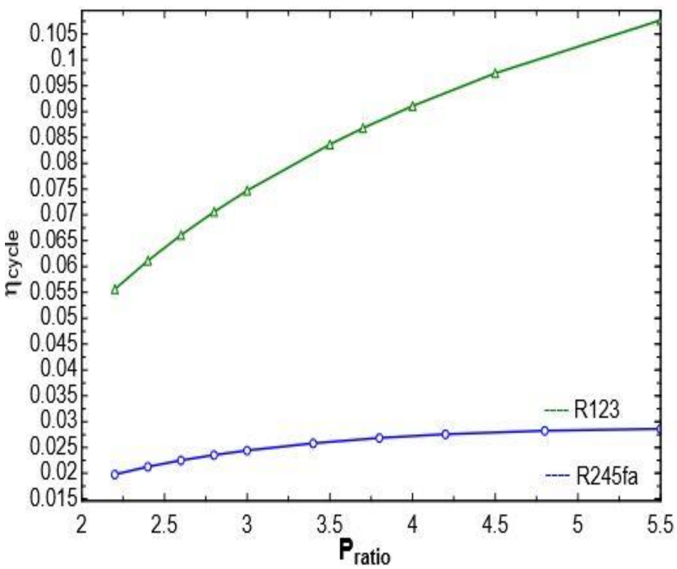

Figure 23: Variation of Pressure ratio Vs Variation of cycle efficiency

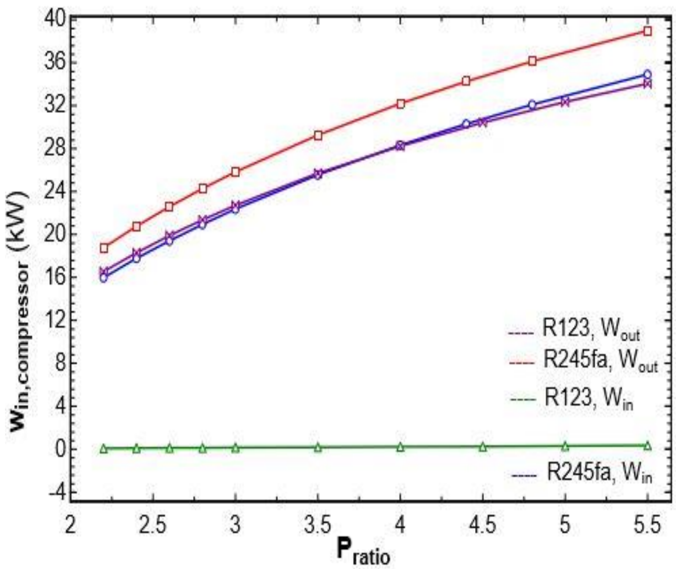

Figure 24: Variation of Pressure ratio Vs Variation of compressor and turbine works

3.4.5 The effect of pressure ratio on compressor and turbine work

Fig.24 shows the variation between pressure ratio and compressor & turbine work. It can be seen from the Fig. 24 that for fluid R123 the work input and work output is 15.38 kW and 22.72 kW respectively at pressure ratio 3.2 and for fluid R245fa the work input and work output is 22.38 kW and 25.83 kW respectively at pressure ratio 3.2.

3.4.6 The effect of minimum cycle temperature

Fig.25 shows the graph between minimum operation temperature and cycle efficiency is 2.837 % at T1 = 25 ºC for fluid 245fa and 10.29 % for fluid R123.

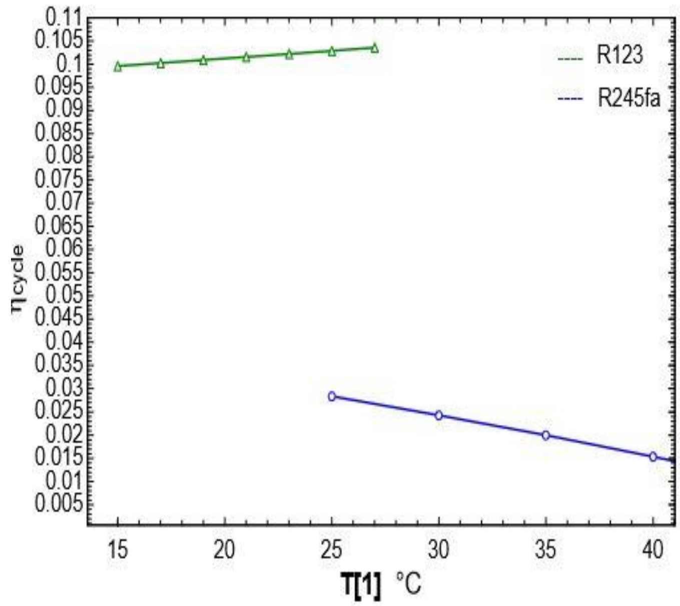

Figure 25: Variation of Minimum cycle temperature Vs Variation of cycle efficiency

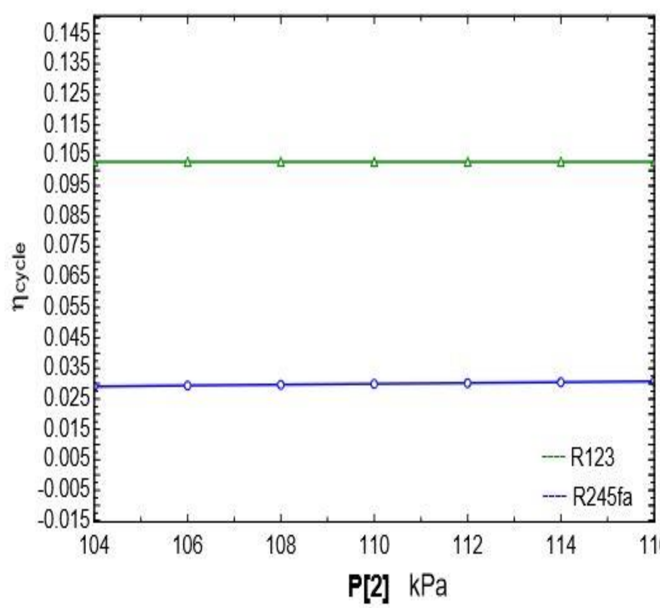

Figure 26: Variation of Cooler pressure Vs Variation of cycle efficiency

3.5 Brayton cycle with Intercooling and Reheating

3.5.1 The effect of cooler pressure

Fig.26 show the variation of cooler pressure and cycle efficiency. It can be seen from the Fig.26 that the cycle efficiency is 10.29 % at 114 kPa cooler pressure for fluid R123 and 3.051 % at 114 kPa cooler pressure for fluid R245fa.

3.5.2 The effect of high pressure Inlet turbine temperature

Fig.27 shows the graph between HP inlet turbine temperature and cycle efficiency is 2.31 % at T5 = 150 ºC for fluid 245fa and 10.33 % for fluid R123.

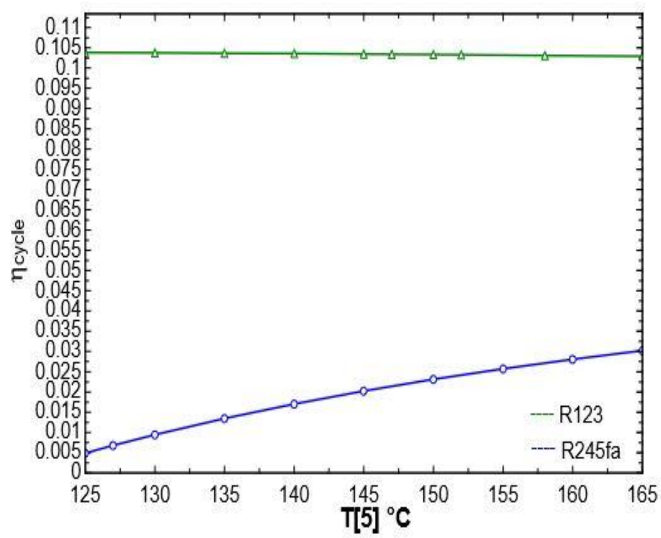

Figure 27: Variation of High pressure Inlet turbine temperature Vs cycle efficiency

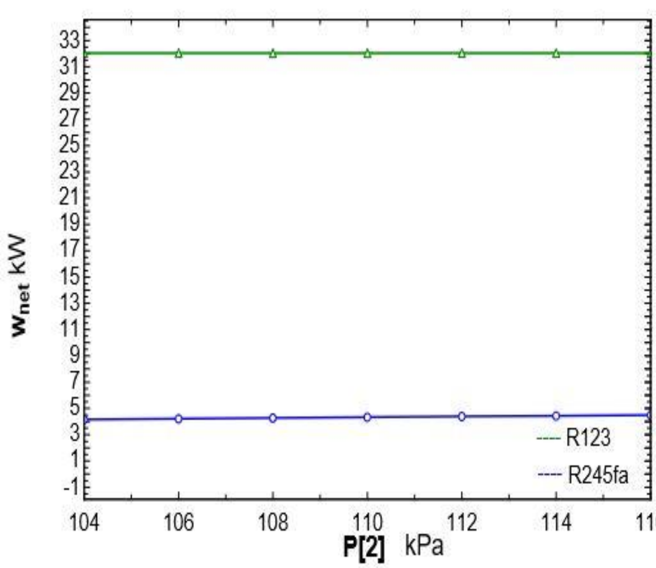

Figure 28: Variation of Heater pressure Vs Variation of Net-work

3.5.3 The effect of heater pressure on total cycle work

Fig.28 show the variation of heater pressure and Net-work. It can be seen from the Fig. 28 that the Net-work is 32.04 kW at 114 kPa heater pressure for fluid R123 and 4.439 kW at 114 kPa heater pressure for fluid R245fa.

3.5.4 Effect of pressure ratio on cycle efficiency

Fig.29 show the variation of pressure ratio and cycle efficiency. It can be seen from the Fig. 29 the cycle efficiency is 7.472 % at 3 pressure ratio for fluid R123 and 2.61 % at 3 pressure ratio for fluid R245fa.

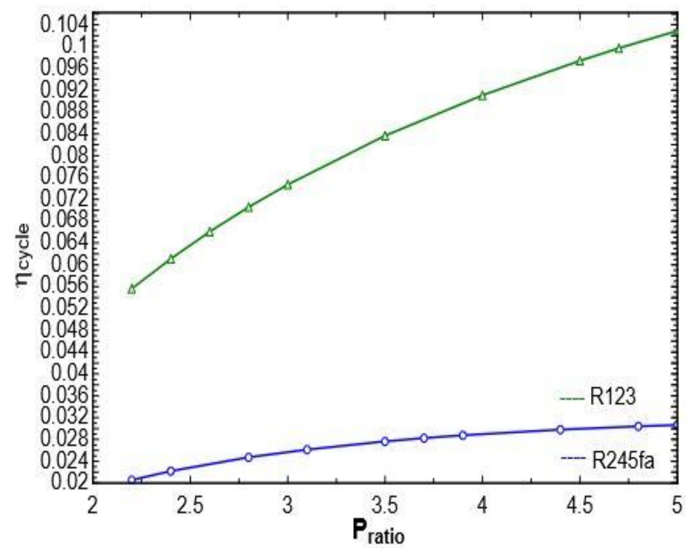

Figure 29: Variation of Pressure ratio Vs Variation of cycle efficiency

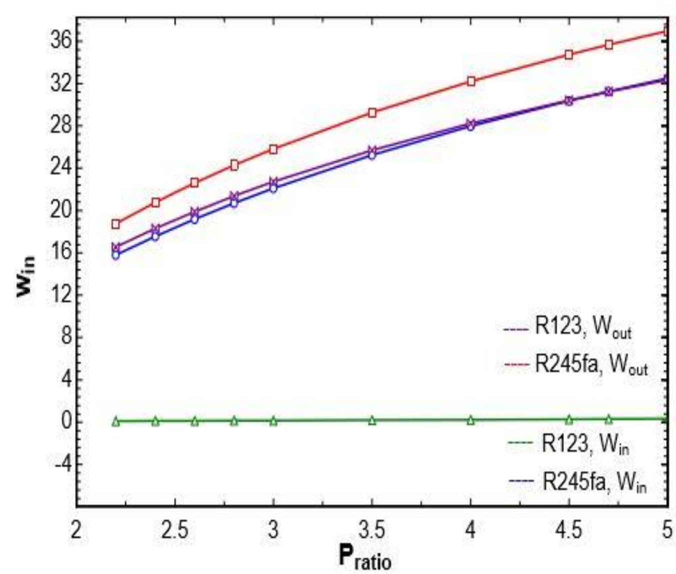

Figure 30: Variation of compressor and turbine work with variation of Pressure ratio

3.5.5 The effect of pressure ratio on compressor and turbine work

Fig.30 shows the variation between pressure ratio and compressor & turbine work. It can be seen from the Fig. 30 that for fluid R123 the work input and work output is 15.38 kW and 22.72 kW respectively at pressure ratio 3 and for fluid R245fa the work input and work output is 22.1 kW and 25.83 kW respectively at pressure ratio 3.

3.5.6 The effect of Minimum cycle temperature

Fig.31 shows the graph between minimum operation temperature and cycle efficiency is 3.064 % at T1 = 25 ºC for fluid 245fa and 10.29 % for fluid R123.

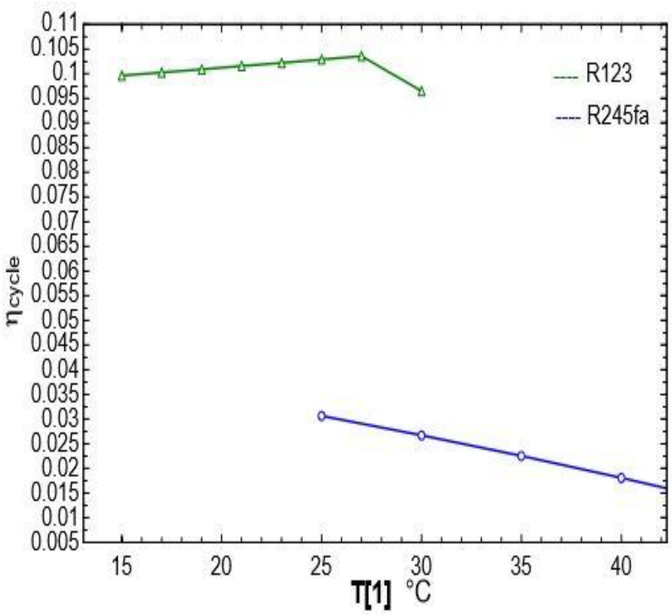

Figure 31: Variation of Minimum cycle temperature Vs Variation of cycle efficiency

3.6 Combined cycle

The performance evaluation of combined cycle is given below

3.6.1 The effect of heater pressure on combined cycle efficiency

Fig.32 shows the variation between heater pressure and combined cycle efficiency. It can be seen from the Fig. 32 that the combined efficiency, Rankine efficiency, Brayton efficiency is 12.43 %, 11.79 %, 10.72 % respectively at heater pressure 2600 kPa for fluid R245fa and 12.43 %, 14.04 %, 10.99 % respectively for fluid R123.

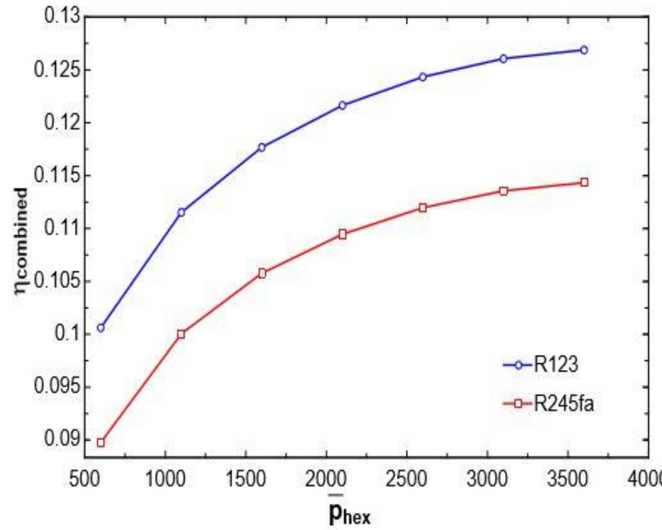

Figure 32: Variation of combined cycle efficiency with variation of heater pressure

3.6.2 The effect of heater pressure on combined net-work

Fig.33 shows the change in combined cycle net-work output by varying the heater pressure. It can be seen from the Fig. 33 that the combined cycle work is 63.21 kW at heater pressure 2600 kPa for fluid R123 and for fluid R245fa the combined cycle work is 64.16 kW at heater pressure 2600 kPa.

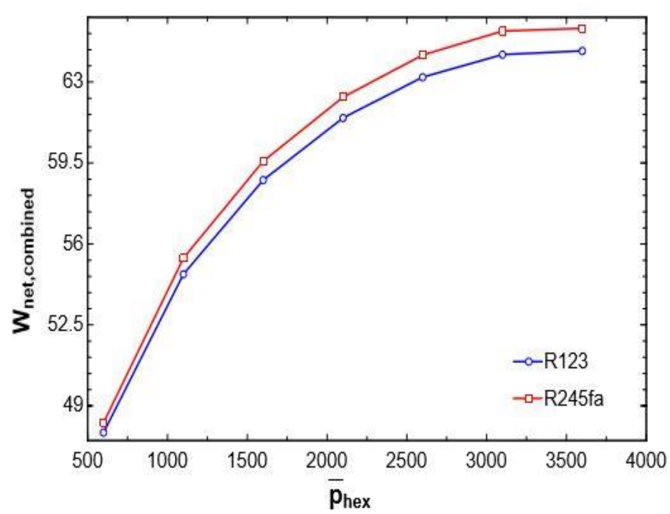

Figure 33: Variation of combined net-work with variation of Heater pressure

3.6.3 Effect of heater pressure on Turbine work of both Brayton and Rankine

Fig.34 shows the change in turbine work of Brayton and Rankine cycle by varying the heater pressure. It can be seen from the Fig. 34 that the turbine work for Brayton cycle is 35.93 kW and 30.08 kW for Rankine cycle at heater pressure

2600 kPa for fluid R123 and for fluid R245fa the turbine work for Brayton is 34.8 kW and for Rankine is 32.36 kW at heater pressure 2600 kPa.

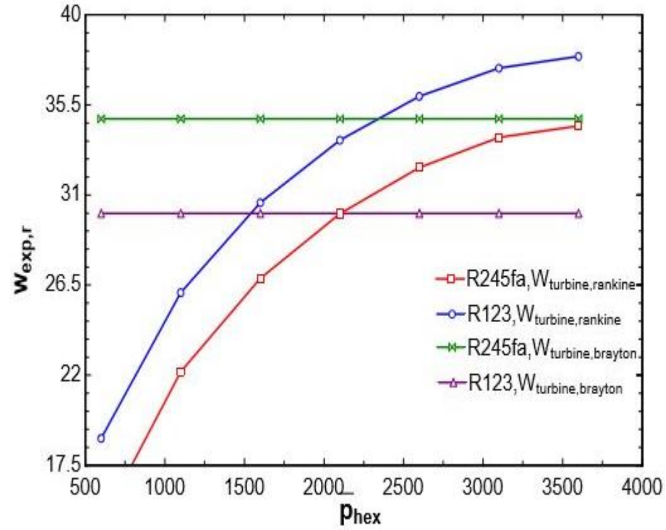

Figure 34: Variation of Turbine work with variation of heater pressure

3.6.4 Effect of heater pressure on pump and compressor work

Fig.35 shows the change in pump and compressor work of Brayton and Rankine cycle by varying the heater pressure. It can be seen from the Fig. 35 that the pump and compressor work for Brayton cycle is 2.142 kW & 0.6571 kW at heater pressure 2600 kPa for fluid R123 and for fluid R245fa the pump and compressor work for Rankine cycle is 2.276 kW & 0.7183 at heater pressure 2600 kPa.

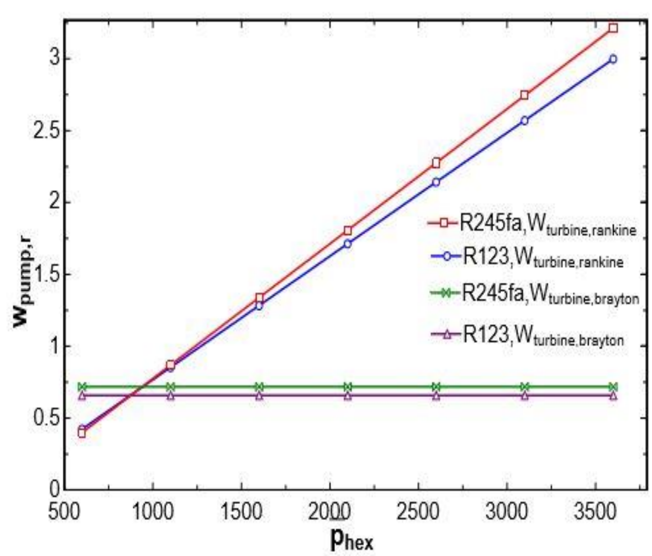

Figure 35: Variation of pump and compressor works with Variation of Heater pressure

3.6.5 The effect on heater temperature on combined efficiency

Fig.36 shows the change in combined cycle net efficiency by varying the heater temperature. It can be seen from the Fig. 36 that the combined cycle efficiency is 11.91 % at heater temperature 140 ºC for fluid R123 and for fluid R245fa the combined cycle efficiency is 11.27% at heater temperature 140 ºC.

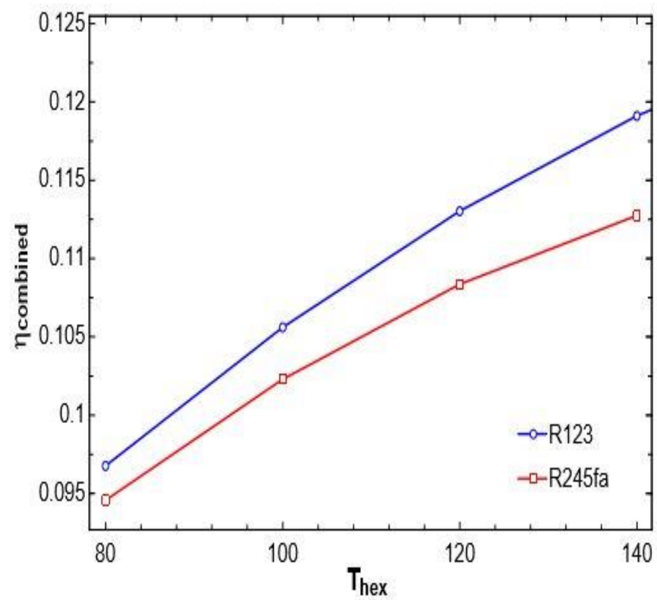

Figure 36: Variation of combined cycle efficiency with variation of Heater temperature

3.6.6 Effect of heater pressure on combined net-work

Fig.37 shows the change in combined cycle net-work by varying the heater temperature. It can be seen from the Fig. 37 that the combined cycle net-work is 59.74 kW at heater temperature 140 ºC for fluid R123 and for fluid R245fa the combined cycle net-work is 64.71 kW at heater temperature 140 ºC.

3.6.7 Effect of pressure ratio on combined cycle efficiency

Fig.38 shows the change in combined cycle net efficiency by varying the pressure ratio. It can be seen from the Fig.38 that the combined cycle efficiency is 13.43 % at pressure ratio 10 for fluid R123 and for fluid R245fa the combined cycle efficiency is 13.08 % at pressure ratio 10.

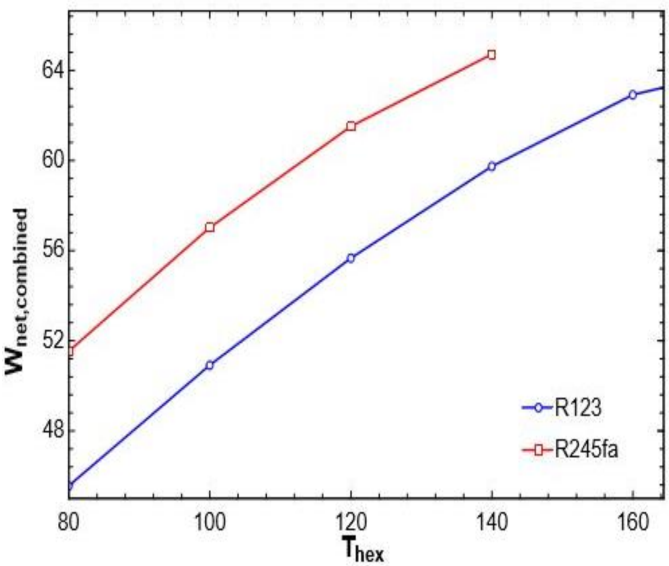

Figure 37: Variation of combined net-work with variation of Heater temperature

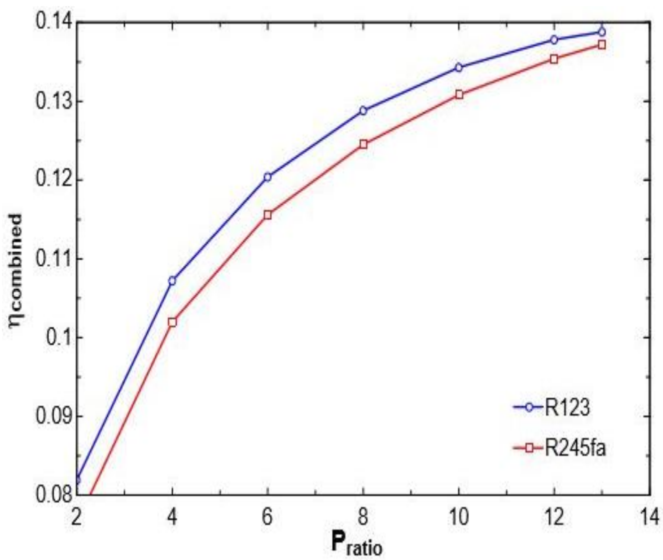

Figure 38: Variation of combined cycle efficiency with Variation of Pressure ratio

3.6.8 Effect of pressure ratio on combined net-work

Fig39 shows the change in combined cycle net-work by varying the pressure ratio. It can be seen from the Fig. 39 that the combined cycle net-work is 64.58 kW at pressure ratio 10 for fluid R123 and for fluid R245fa the combined cycle net-work is 72.96 kW at pressure ratio 10.

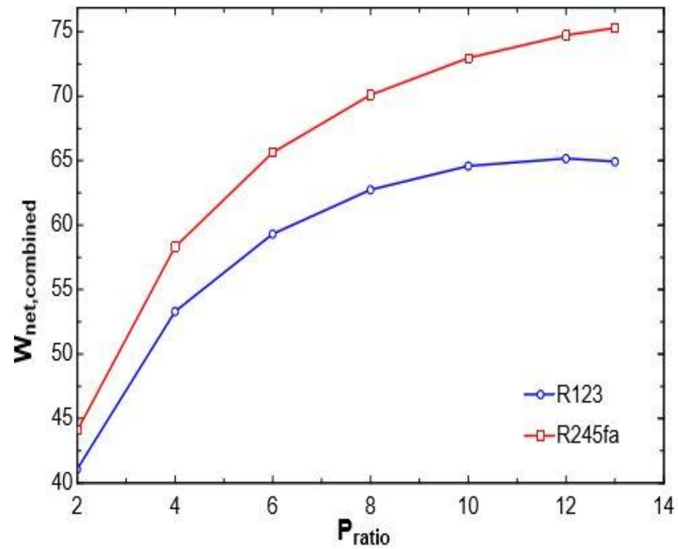

Figure 39: Variation of combined net-work with variation of Pressure ratio

3.6.9 Effect of compressor inlet pressure on combined efficiency

Fig.40 shows the change in combined cycle net efficiency by varying the compressor inlet pressure. It can be seen from the Fig. 40 that the combined cycle efficiency is 11.26 % at compressor inlet pressure 300 kPa for fluid R123 and for fluid R245fa the combined cycle efficiency is 10.8 % at compressor inlet pressure 300 kPa.

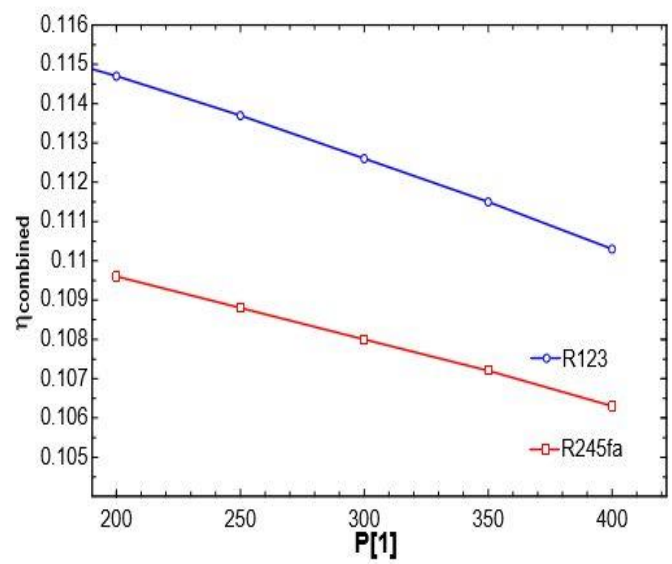

Figure 40: Variation of combined cycle efficiency with variation of Compressor inlet pressure

3.6.10 Effect of compressor inlet pressure on combined net-work

Fig.41 shows the change in combined cycle net-work by varying the compressor inlet pressure. It can be seen from the Fig. 41 that the combined cycle net-work is 55.03 kW at compressor inlet pressure 300 kPa for fluid R123 and for fluid R245fa the combined cycle net-work is 60.95 kW at compressor inlet pressure 300 kPa.

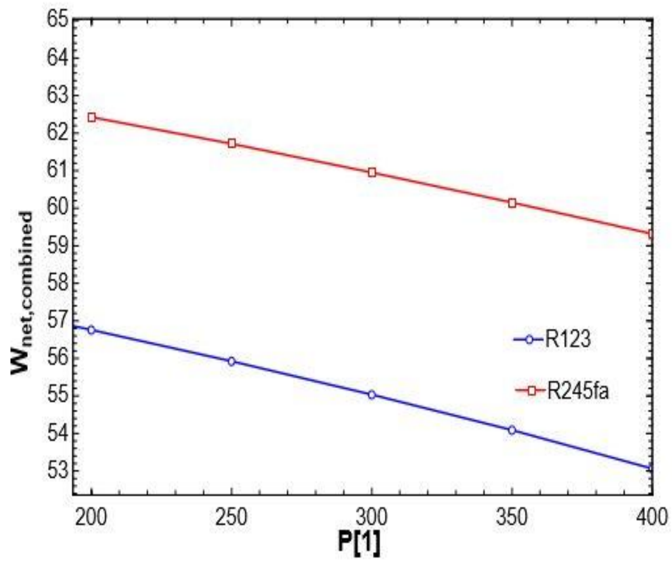

Figure 41: Variation of combined net-work with variation of Compressor inlet pressure

4. Conclusion

In this study, the simulation of subcritical organic Rankine cycle system with internal heat exchanger using R123 and R245fa is done which illustrates the impacts of IHE on ORC systems. Also, simulation is done on Brayton cycle and combined cycle. In combined cycle simulation is done using R123 and R245fa. The mass flow rate taken into account for analysis is 1 kg/s and the warmth source temperature is taken as 200ºC. The temperature difference of 10ºC is set as the minimum heat transfer temperature difference of heat exchanger. The simulation results show that there will be an optimum gas heater pressure for power cycles at certain cycle working conditions. The heater pressure will increase with increasing heat source temperature. The efficiency of the expansion machine will have more crucial influence on the cycle thermal efficiency than the pump efficiency does.

For combined cycle, the cycle efficiency is increasing on increasing of heater pressure for both R123 and R245fa. But efficiency is coming more for R123 as compared to R245fa at particular heater pressure.For combined cycle, the combined net-work of cycle is increasing on increasing of heater pressure for both R123 and R245fa but work output is more for R245fa. In combined cycle, cycle efficiency is increasing on increasing of pressure ratio but there is more increase seen in R123. Also, net-work is also increasing on increasing of pressure ratio but here more increase is seen in R245fa.In combined cycle, on increasing compressor inlet pressure the cycle efficiency and the net-work both are decreasing for R123 and R245fa.

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[2] R.S. Mishra & Dharmendra Sahu [2016]Thermodynamic analysis and comparison of various organic fluids for ORC in Gas turbine-Organic Rankine combined cycle plant with solar reheating and regeneration of ORC fluids, International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395 -0056 , Vol: 03, Issue: 08 ,2016 www.irjet.net p-ISSN: 2395-0072.

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Nomenclature

dP_{cf}	Total resistance at condenser cooling water side, kP_a
ε	Heat exchanger effectiveness
Q_{net}	Net heat transfer rate, W
dP_{hf}	Flow resistance in the working fluid circulation loop, kP_a
e	specific flow energy, kJ/kg
E_d	energy destruction rate, K_w
h	specific enthalpy, kJ/kg
m	mass flow rate, kg/s
P	pressure, KP_a
s	specific entropy, kJ/(kg K)
T	Temperature,
v	specific volume, m^3/kg
W	power output, kJ/s
w	specific power, kJ/kg
η_{ex}	energy efficiency
η_P	overall efficiency of the working fluid pump
$\eta_{P,i}$	isentropic compression efficiency of the working fluid pump
$\eta_{P,m}$	mechanical efficiency of the working fluid pump
η_{th}	thermal efficiency
$\eta_{T,i}$	internal efficiency of the expander
$\eta_{T,m}$	mechanical efficiency of the expander