Chapter 3: Organic Rankine Cycle Modifications

This chapter contains energy, exergy and economic analyses of two novel ejector-enhanced organic Rankine cycles and a comparison with basic cycle.

3.1. Cycle Description

Components of reference (basic) organic Rankine cycle are the evaporator (vapour generator), power-producing expansion device (e.g., turbine), condenser, pump and working fluid. The type of working fluid may be dry, isentropic or wet fluid. Here, R123, R601a and R1233zd(E) are considered as working fluids, which are dry fluids. Two novel ejector-enhanced ORCs (EEORC), i.e., EEORC-1 and EEORC-2, along with basic ORC are analyzed and optimized for the given heat source temperature. The superheated isopentane is then allowed to expand in a turbine after being heated in an evaporator to a saturated vapour state 3 from the pressurized isopentane. At state 4, working fluid leaves the turbine and condenses into the saturated liquid condition 1. The working fluid is heated using a counter-flow evaporator from the pump outlet (state 2) to the saturated vapour state 3, passing via the saturated liquid state 3l. The net work is produced from states 3 to 4 of the turbine expansion. The presence of an ejector enhances the turbine work output by lowering the turbine exit pressure.

Figure 3.1 and Figure 3.2 show the schematic layout and T-s diagram of EEORC-1, respectively. In EEORC-1, the liquid working fluid is pumped (process 1-2) and enters into the preheater section, then divides into primary and secondary streams at the inlet of the evaporator section (saturated liquid state). Primary and secondary streams of the working fluid flow through the ejector and turbine, respectively. The primary working fluid stream is bleeding from the saturated liquid state of the evaporator (exit of preheater) and enters the nozzle section of the ejector. The motive stream flows through the nozzle and gets high velocity at the nozzle exit, which entrains the secondary stream. The secondary working fluid stream flows through the evaporator and gets heated to the saturated vapor state by the heat source fluid (Therminol-VP1 is used here). Then this fluid expands in the turbine and generates power. The secondary stream from the exit of the turbine enters the constant cross-section of the nozzle and gets mixed with the motive stream. This mixed stream then flows through the diffuser of the nozzle to increase the condenser pressure by slowing down the velocity. The diffuser exit stream enters the condenser, where heat is removed by cooling fluid (water). Then this stream is pumped to the evaporator pressure where it gets preheated and the cycle continues.

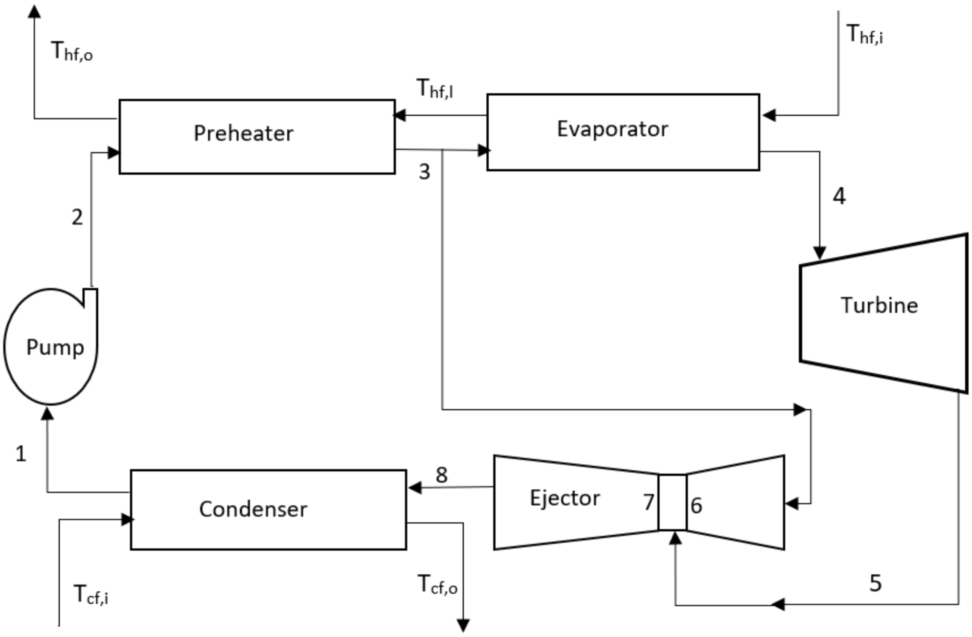

Figure 3.1. Schematic layout of proposed EEORC-1

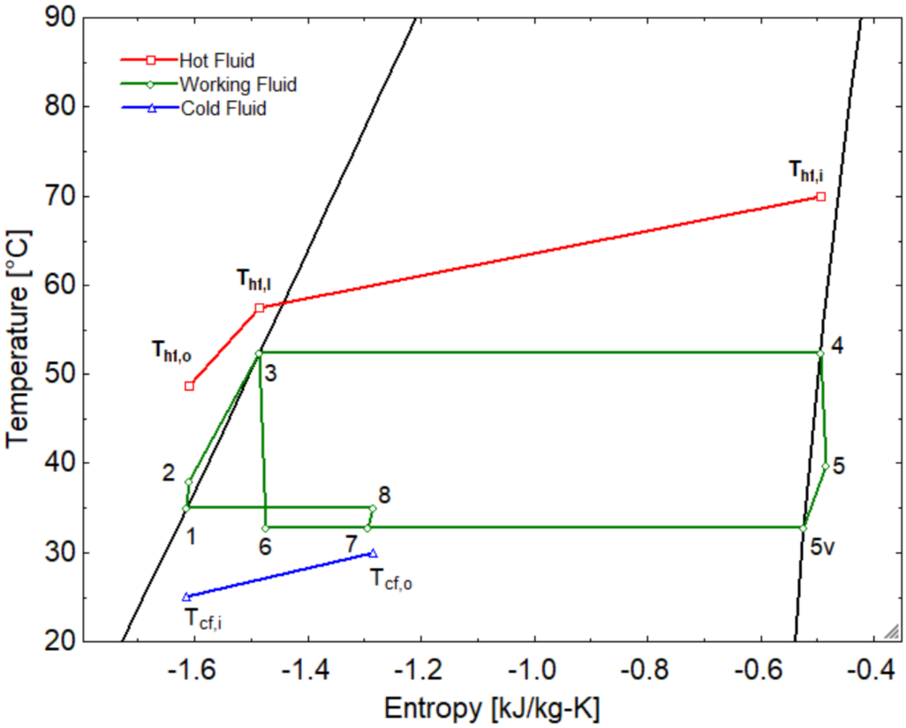

Figure 3.2. Temperature-entropy diagram of proposed EEORC-1

Figure 3.3 and Figure 3.4 show the schematic layout and T-s diagram of EEORC-2, respectively. The difference in EEORC-2 as compared to EEORC-1 is the presence of double pumps, double evaporators and the exit state of the motive stream. In EEORC-2, the working fluid at the condenser exit is divided into two streams. Pump-1 and pump-2 pump these two streams to comparatively high pressure and low pressure, respectively. The primary stream from pump-2 flows to the ejector through a low-pressure boiler (preheater + evaporator) and the secondary stream from pump-1 gets heated through a high-pressure boiler (preheater + evaporator) and expands in the turbine to generate power. Hot fluid (Therminol-VP1) enters the high-pressure boiler and divides into two streams. These streams are further used to heat the low-pressure boiler (preheater + evaporator) and high-pressure preheater and exit out to the surrounding. The low-pressure's primary stream is heated up to the saturated vapor state through the low-pressure boiler and then enters the ejector's nozzle section. The high-pressure's secondary stream is heated up to the saturated vapor state in the high-pressure boiler and then enters the turbine for power generation. The rest of the phenomena in the ejector are the same as those in the EEORC-1 ejector and then condense by cooling water, and the cycle repeats.

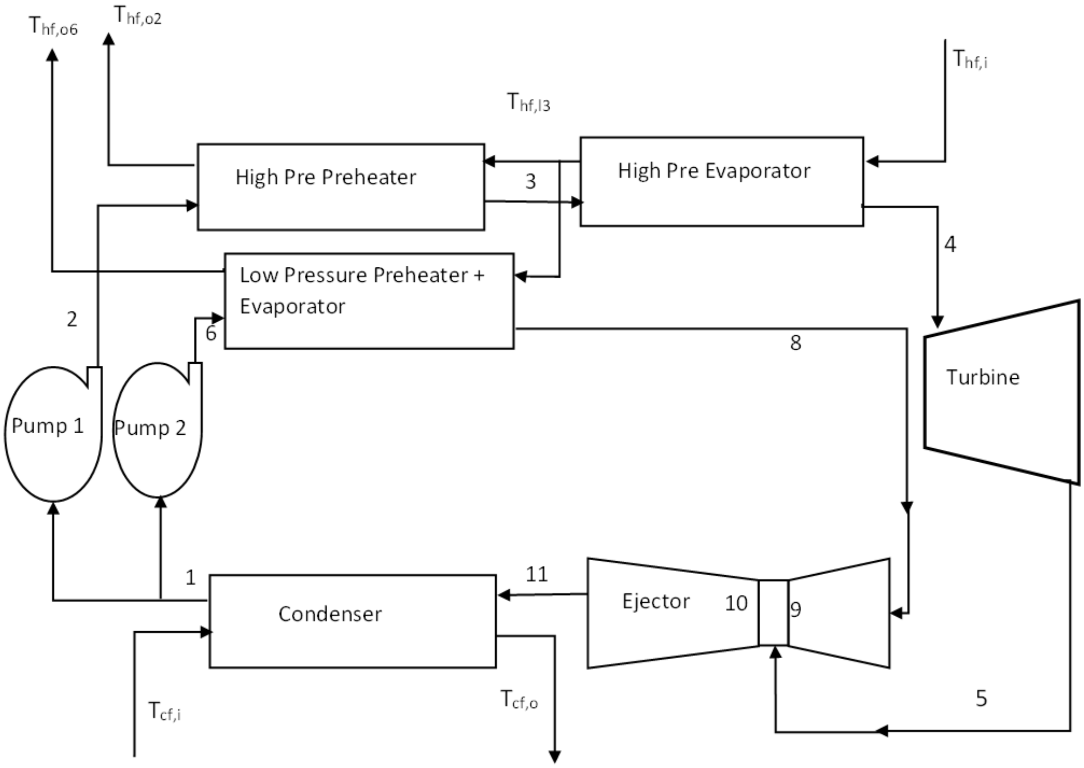

Figure 3.3. Schematic layout of proposed EEORC-2

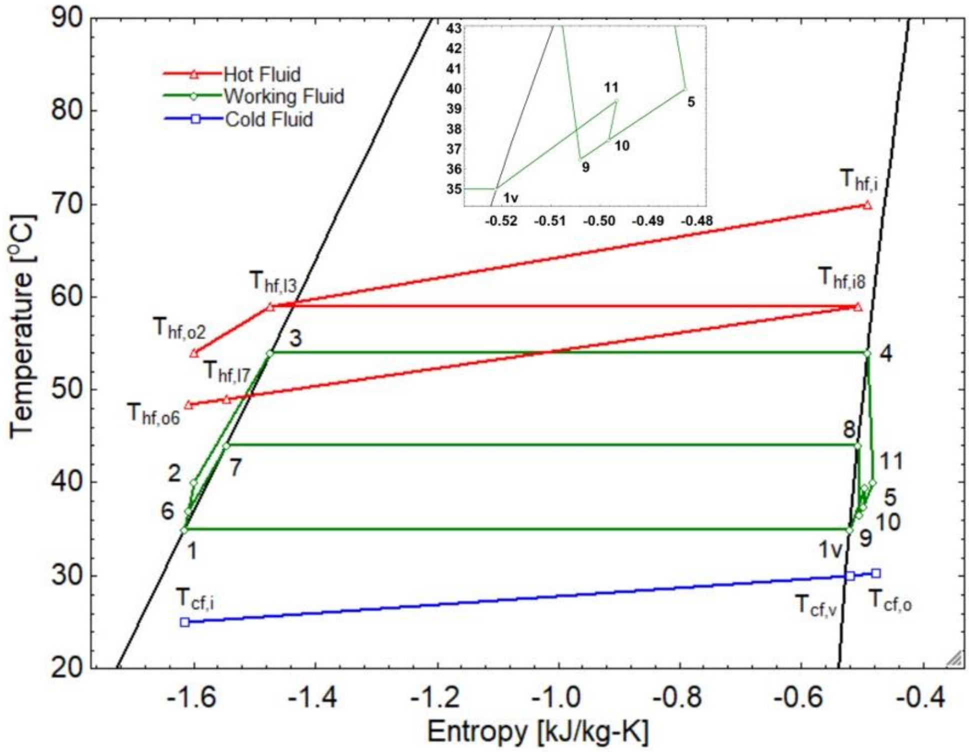

Figure 3.4. Temperature-entropy diagram of proposed EEORC-2

3.2. Modelling and Simulation

3.1.1. Mathematical Modelling

The required mathematical modelling for thermodynamic analysis and component design is taken from Chapter 2 (Eqs. 2.1 - 2.27). Other than the above, additional assumptions, equations and correlations are mentioned below.

(i) Nozzle and diffuser have given isentropic efficiencies.

(ii) Both the motive and the suction stream reach the same pressure at the inlet of the constant area mixing section of the ejector. There is no mixing between the two streams before the inlet of the constant area mixing section.

(iii) Similar to other components, ejector inlet and outlet velocities have been neglected.

(iv) The cost of the ejector has been ignored because the cost of the ejector is much lower than that of other equipment.

For the ejector, the nozzle exit velocity is given by:

$$c_{nz,out} = \sqrt{2000\left(h_{nz,in} - h_{nz,out}\right)} \tag{3.1}$$

Momentum and energy balances at constant area intermixing chamber of the ejector are given by, respectively:

$$\dot{m}_{wf,sec}c_{tur,out} + \dot{m}_{wf,pri}c_{nz,out} = \dot{m}_{wf}\frac{c_{d,in}}{\sqrt{\eta_m}} \tag{3.2}$$

$$\dot{m}_{wf,sec}\left[h_{tur,out} + \left(\frac{{c_{tur,out}}^2}{2000}\right)\right] + \dot{m}_{wf,pri}\left[h_{nz,out} + \left(\frac{{c_{nz,out}}^2}{2000}\right)\right] = \dot{m}_{wf}\left[h_{d,in} + \left(\frac{{c_{d,in}}^2}{2000}\right)\right] \tag{3.3}$$

The diffuser outlet of the ejector is calculated by the following equation:

$$h_{d,out} = h_{d,in} + \left(\frac{{c_{d,in}}^2}{2000}\right) \tag{3.4}$$

The pressure lift ratio of the ejector is calculated as:

$$PLR = \frac{P_{con}}{P_{sec,nz,in}} \tag{3.5}$$

The ejector irreversibility of the ejector is calculated as:

$$I_{eje} = T_{amb}\left(\dot{m}_{wf}s_{con,in} - \dot{m}_{wf,tur}s_{tur,out} - \dot{m}_{wf,eje}s_{eje,in}\right) \tag{3.6}$$

Now, the overall exergy balance is given by:

$$Ex_{in} = W_{net} + I_{pump} + I_{tur} + I_{evp} + I_{con} + I_{eje} \tag{3.7}$$

3.1.2. Proposed economic models

To market and commercialize the new advanced technology (enhanced cycle) developed in the field of the energy production sector, increased power generation, total cost consumed and selling price are the main important parameters. The main aim of technological advancement is to generate more power at the same previous resources (source temperature, flow rate, etc.) used and to provide more hours of power availability at the same or cheaper rates so that more population can get benefitted from more hours of power availability but at a cheaper rate. To get the power (electricity) at a cheaper rate along with industry benefits using proposed novel cycles, three economic models have been chosen and compared with reference to basic ORC. Here, three different economic models (M1, M2 and M3) have been discussed for the two cases. 1st case is when only old technology is there and for 2nd case is when new technology completely replaces the old technology. In order to provide cheap electricity at a larger amount, the government may subsidize the total cost consumed on new technology and research & development work [118].

Let's consider the simple linear power production and selling price function.

$$C_{ele} = a_{ii} - b_{ii}W_{net} \tag{3.8}$$

Where, constant $a, b > 0$ and C_{ele} is the selling price of electricity and W_{net} is the power production of old and new technology, respectively.

The subscript "ii" refers to the economic model M1, M2 and M3. The condition of economic models is described below.

Economic model 1 (M1) is based on when the unit selling price of electricity is the same (real word functioning criteria):

$$C_{ele}^{case1} = C_{ele}^{case2} \tag{3.9}$$

Economic model 2 (M2) is based on when the total hourly profit is the same:

$$\text{Profit}_{total}^{case1} = \text{Profit}_{total}^{case2} \tag{3.10}$$

Economic model 3 (M3) is based on when the unit profit is the same:

$$\text{Profit}_{unit}^{case1} = \text{Profit}_{unit}^{case2} \tag{3.11}$$

3.1.3. Simulation procedure

Based on the above mathematical models, a simulation code has been developed in Engineering Equation Solver for the proposed EEORC systems. The code can predict the performances of proposed systems for the given heat source, heat sink, ambient temperature and component efficiencies. PPTD is defined effectively when designing the evaporator and condenser. All physical and thermodynamic properties of considered fluids are calculated using an in-built subroutine in software. Both proposed systems have been optimized for maximum net work output. The heat exchanger area is calculated to determine the capital investment cost. Proposed economic modelling is executed on the software to analyse the best thermodynamic cycle suited to give more power but at cheaper rates to the consumer. Figure 3.5 represents the flow chart of the whole simulation process.

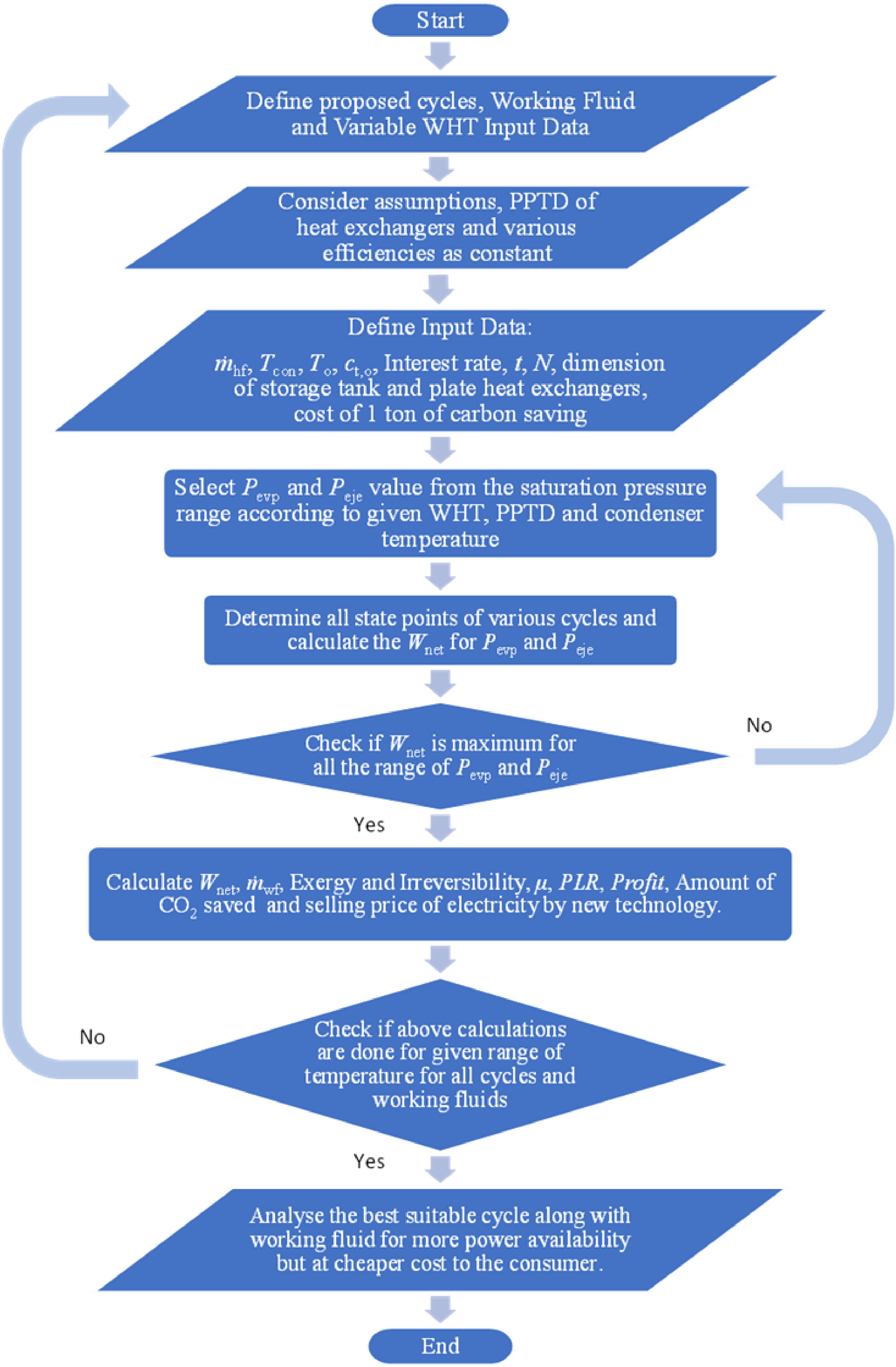

Figure 3.5. Flow chart of the simulation procedure for EEORC

3.1.4. Model validation:

Table 3.1. Comparison of result obtained from reference with simulated values

Parameter	Reference value	Calculated value	Error (%)
Pump work	2.50 W [38]	2.51 W	0.4
Turbine work	60.20 W [119]	63.58 W	5.61
Turbine work	64.48 W [38]	63.22 W	1.95
Thermal efficiency	2.59% [38]	2.59%	0
Ideal Rankine efficiency	7.93% [38]	7.87%	0.75
Evaporator area	0.60 m^2 [119]	0.62 m^2	3.3
Condenser area	0.80 m^2 [119]	0.85 m^2	6.3

As the proposed cycle is new, the basic ORC is validated only to check the accuracy. Under similar operating conditions to Farrokhi et al. [38] and Liu et al. [119], the simulation is performed and compared with the reference values, which are given in Table 3.1. Thermodynamic simulation achieves a good result with acceptable error for Farrokhi et al. [38]. The area of the evaporator and condenser also shows a good result with a maximum error of 6.4% despite the correlation used [119]. These results are obtained from current simulation analyses and reference experimental analyses with very small and accepted deviation caused by instrument calibration error and real-ideal phenomena differences.

3.3. Results and Discussion

Working fluid properties of novel modification of ORC are shown in Table 3.2.

Table 3.2. Selection of working fluids [120]

Working Fluid	Alternative name	Critical temperature [^oC]	Critical Pressure [MPa]	Normal boiling point [^oC]	Triple point temperature [K]	ASHRAE Safety group	ODP	GWP (100 years)
2-Methylbutane	R601a (Isopentane)	187.2	3.37	27.85	112.65	A3	0	5

2,2-Dichloro-1,1,1-Trifluoroethane	R123	183.7	3.66	27.78	166	B1	0.02	79
Trans-1-Chloro-3,3,3-trifluoropropene	R1233zd(E)	165.6	3.57	18.32	195.15	A1	0	4.5

Input parameters of novel studied EEORCs are given in Table 3.3. Cold source and condenser temperature is selected based on specific location ambient temperature. For this simulation, latitude 25.42° N and longitude 82.97° E Varanasi, India, are selected. The hot fluid mass flow rate is selected in such a way as to justify the work output. Other parameters are taken from the reference. The selection of working fluid is based on critical temperature, pressure and environmental effect. Desired critical temperature range is 150-200 °C for a wide temperature range of working. Working fluid should have lower ozone depletion potential and global warming potential and meet with the standard ASHARAE safety groups.

Table 3.3. Input parameters for Basic and EEORCs

Input Parameter	**Unit**	**Modified ORC**
Source inlet temperature	^oC	70
Cold source inlet temperature (Water)	^oC	25
Condenser Temperature	^oC	35
Ambient Temperature	^oC	30
Hot Fluid mass flow rate (Therminol VP1)	kg/s	100
$PPTD_{evp}$	^oC	5 [114]
$PPTD_{con}$	^oC	5 [114]
Pump Efficiency	-	0.7 [115]
Turbine Efficiency	-	0.85 [116]
Motive nozzle Efficiency	-	0.9 [121,122]
Mixing Efficiency	-	0.9 [121,122]
Diffuser Efficiency	-	0.85 [121,122]

Turbine exit velocity	m/s	0
Annual working hour	h	7000
Annual Interest rate	-	10%
Width and Length of PHE	mm, cm	5, 15
Diameter and height of tank	cm, cm	30, 70

Performance comparison is presented in Table 3.4 for studied mean input parameters (Table 3.3) for different power cycles and working fluids. For the comparative analysis of basic and advanced cycles for the different working fluids, the low-grade WHT of 70 $^{\circ}$C is chosen. All cycles are optimized for maximum net work output. Ejector-enhanced cycles exact more heat from a given heat source and yield more net work but lower thermal efficiency due to lower optimum evaporator pressure. Basic cycle and EEORC-2 show almost equal maximum net work output for all working fluids, but EEORC-1 shows different value with different working fluid and a maximum for R123 working fluid. Thermal and exergy efficiencies are cycle-dependent and thus show very less variation on working fluid. The mass flow rates of the working fluid and cold fluid, entrainment ratio and pressure lift ratio are obtained in order to maximize net work output. The total cost invested in EEORC-1 is the maximum due to the highest irreversibility and evaporator area. At 70 $^{\circ}$C source temperature, yearly profit is maximum for the basic cycle, but at a higher source temperature, either EEORC-1 or EEORC-2 may give a better result. The amount of CO_2 saving is maximum for EEORC-1 due to maximum irreversibility and heat input required and so converting of CO_2 saved amount in monetary value gives maximum total profit with CO_2.

Table 3.4. Comparison of different cycles and working fluids at WHT = 70 $^{\circ}$C

Parameter	Basic ORC (R601a)	Basic ORC (R123)	Basic ORC (R1233zd(E))	EEORC 1 (R601a)	EEORC 1 (R123)	EEORC 1 (R1233zd(E))	EEORC 2 (R601a)	EEORC 2 (R123)	EEORC 2 (R1233zd(E))
Net work output [kW]	103.6	103.8	103.6	118.3	122.2	114.4	107.4	107.8	107.2

Thermal efficiency (%)	3.76	3.80	3.77	2.07	2.13	2.00	2.72	2.73	2.73
Exergy efficiency (%)	39.69	40.04	39.76	30.08	31.03	29.05	32.28	32.45	32.40
Working fluid mass flow rate [kg/s]	7.66	15.47	14.00	88.91	198.73	168.54	11.09	22.55	20.11
Coolant mass flow rate [kg/s]	123.3	123.9	124.7	267.3	268.1	268.5	178.6	180.5	179.1
Entrainment ratio	NA	NA	NA	0.086	0.077	0.086	1.880	1.850	1.690
Pressure lift ratio	NA	NA	NA	1.169	1.205	1.168	1.021	1.023	1.026
Total irreversibility [kW]	157.4	155.5	156.9	275.1	271.5	279.4	225.3	224.3	223.6
Evaporator area [m^2]	29.13	30.14	30.56	346	506.9	657.6	66.16	90.83	87.66
Condenser area [m^2]	25.31	24.72	24.78	36.32	36.73	36.98	78.98	79.66	79.93
Total cost invested [Thousand $]	183.12	183.08	184.68	308.26	340.21	366.67	213.14	219.34	220.17
Total cost consumed [$/h]	3.26	3.26	3.29	5.48	6.05	6.52	3.79	3.90	3.92

Total cost earned [$/h]	13.67	13.70	13.67	15.62	16.13	15.1	14.18	14.22	14.15
Profit [$/h]	10.42	10.45	10.39	10.14	10.08	8.59	10.38	10.32	10.23
Profit [Thousand $/year]	72.92	73.14	72.71	70.96	70.53	60.03	72.69	72.26	71.61
CO_2 Saving [ton/year]	6760	6710	6750	1400	1410	1410	9710	9700	9640
Profit with CO_2 [$/h]	20.08	20.03	20.02	30.15	30.16	28.66	24.25	24.18	24.00

Figure 3.6 represents the effect of waste heat inlet temperature variation on net work output. Net work output depends upon the difference of enthalpy difference of the turbine work and pump work for the same mass flow rate. Net work output is maximum for EEORC-1 due to the use of an ejector in a particular configuration system. In this configuration PLR and working fluid mass flow rate gives maximum value. In EEORC-1, the turbine's exit pressure will be lower and the secondary fluid mass flow rate will be higher. These parameters cumulatively maximize the net work output of EEORC-1. Net work output increases with increasing the waste heat inlet temperature because the amount of heat added to the evaporator is high and so higher the working fluid enthalpy at the turbine inlet (evaporator exit). Higher turbine inlet enthalpy is the main dominating factor to increase in net work output. EEORC-1 gives maximum net work output for the temperature range of 60-180 °C. At the lower temperature range of 60-150 °C, R123 and at the higher temperature range of 155-180 °C, R1233zd(E) has the highest net work output among the studied working fluids for EEORC-1. At 151-154 °C, all working fluids of EEORC-1 show almost equal net work output.

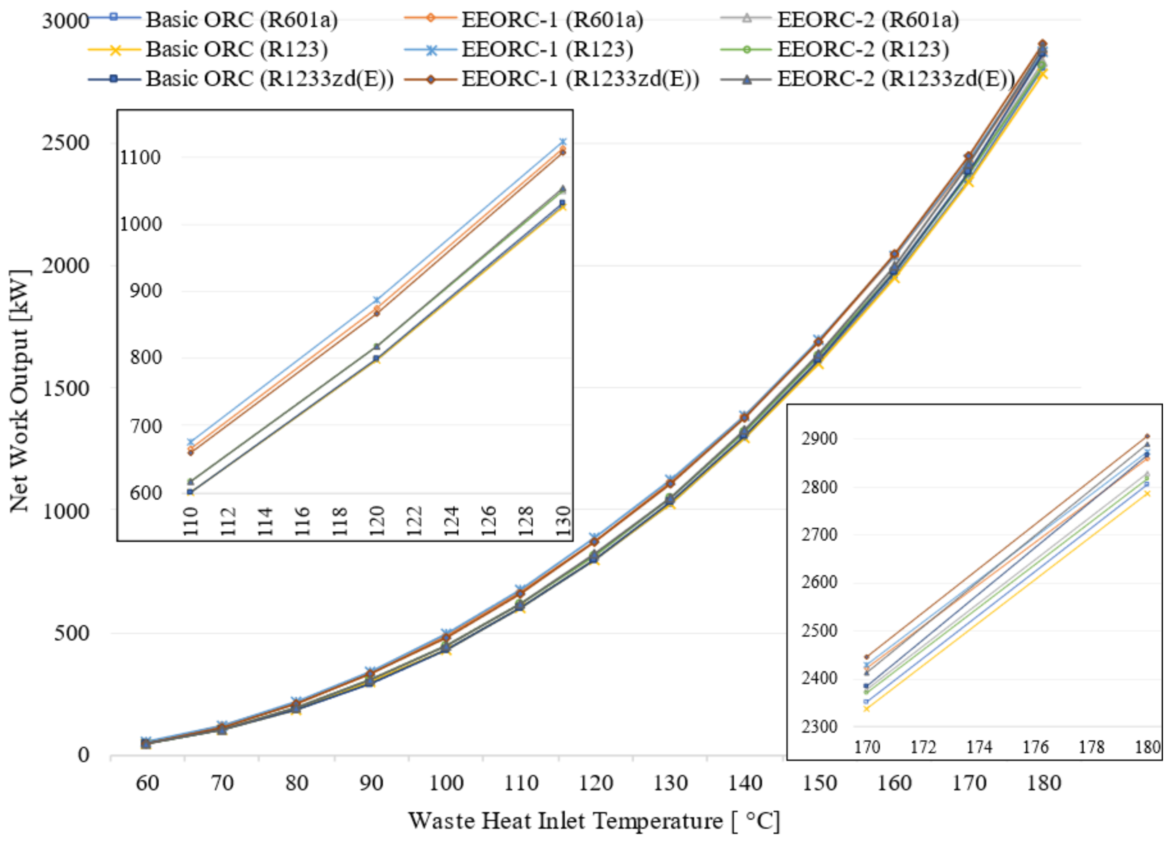

Figure 3.6. Effect of waste heat inlet temperature variation on net work output

Figure 3.7 represents the effect of the variation of waste heat inlet temperature on the ejector exit pressure. Figure 3.7 shows that ejector exit pressure first decreases to the minimum level and then increases. The value of ejector exit pressure depends on the optimized ejector exit pressure condition to maximize the net work of each ejector enhanced cycle at each temperature point to exist the cycles. The lower value of the ejector exit pressure helps to increase the net work output. In Figure 3.7, EEORC-1 has a lesser value of ejector exit pressure in comparison to EEORC-2 for the same working fluid in the temperature range of 60-180°C. Among all working fluids, R123-based EEORC-1 has a minimum ejector exit pressure value at each source temperature. The ejector pressure value cannot go below at higher temperatures because of thermodynamic constraints derived from the simulation method used to maximize the net work output. Thermodynamic constraints are primary and secondary mass flow rate values and temperature difference between hot source fluid exit and pump exit.

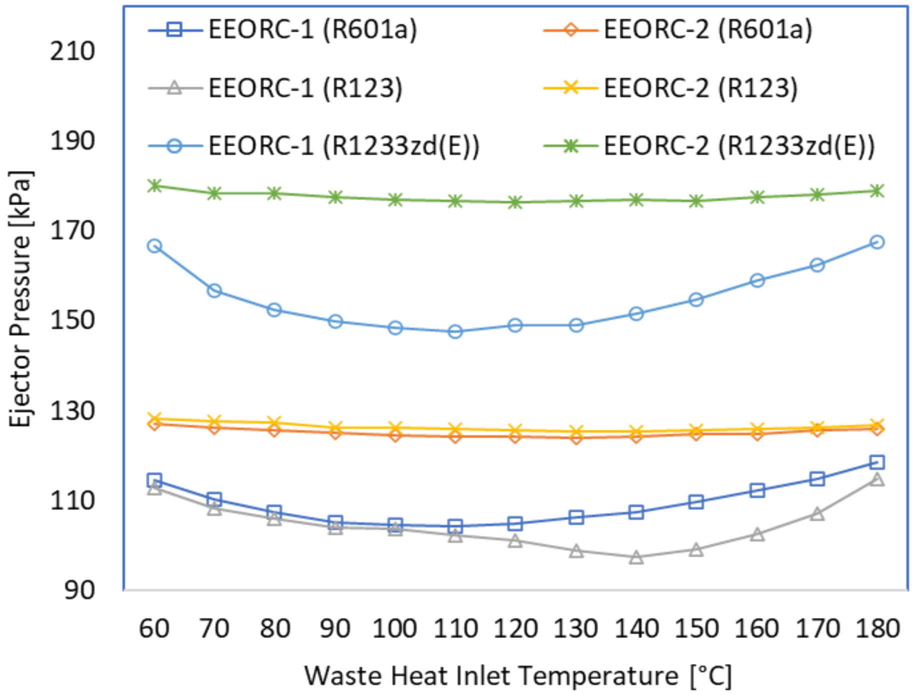

Figure 3.7. Effect on ejector pressure with the variation of waste heat inlet temperature

The Y-axis of Figure 3.8 represents the percentage increase of net work output on ejector-enhanced cycles (both EEORC-1 and EEORC-2) when compared with the basic ORC of the same working fluids. In Figure 3.8, the effect is shown for the percentage increase of net work output on the variation of waste heat inlet temperature. EEORC-1 has the greater and EEORC-2 has the lesser increase in net work output with downward concavity and linearly decreasing respectively with temperature increment. This is because the organic working fluid property is generally suitable for low-range temperature application and thus, irrespective of the cycles, at lower temperature range, organic working fluid shows promising results. On increasing heat source temperature, net work output value ratio of ejector-enhanced cycles (both EEORC-1 and EEORC-2) and the basic cycle will be decreasing because there is a limitation up to which turbine exit pressure can be lowered and so no significant increase in net work output of ejector-enhanced cycles.

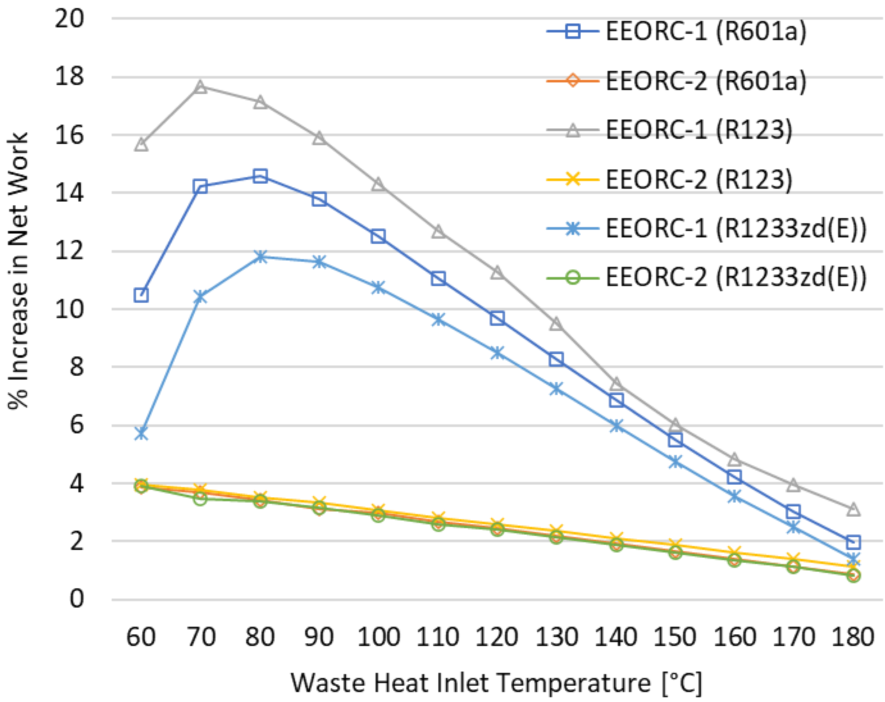

Figure 3.8. Improvement of net work output with waste heat inlet temperature

Figure 3.9 represents the effect on thermal efficiency with waste heat inlet temperature. For all temperature ranges, the thermal efficiency is maximum for basic ORC and least for EEORC-1. The basic cycle has maximum thermal efficiency than the studied ejector-enhanced cycles because less heat input is required at the same heat source temperature. Despite the lowest value of net work output of basic ORC, the cumulative effect of net work output and heat input maximizes the thermal efficiency of basic ORC. The amount of heat input plays a major role in the net work output. R123-based cycles have higher thermal efficiency because of increased net work output and so the basic ORC with R123 has the highest thermal efficiency for all temperature ranges. Due to the highest cycle temperature of EEORC-1, among others, it requires the highest heat input and so minimizes the cycle thermal efficiency.

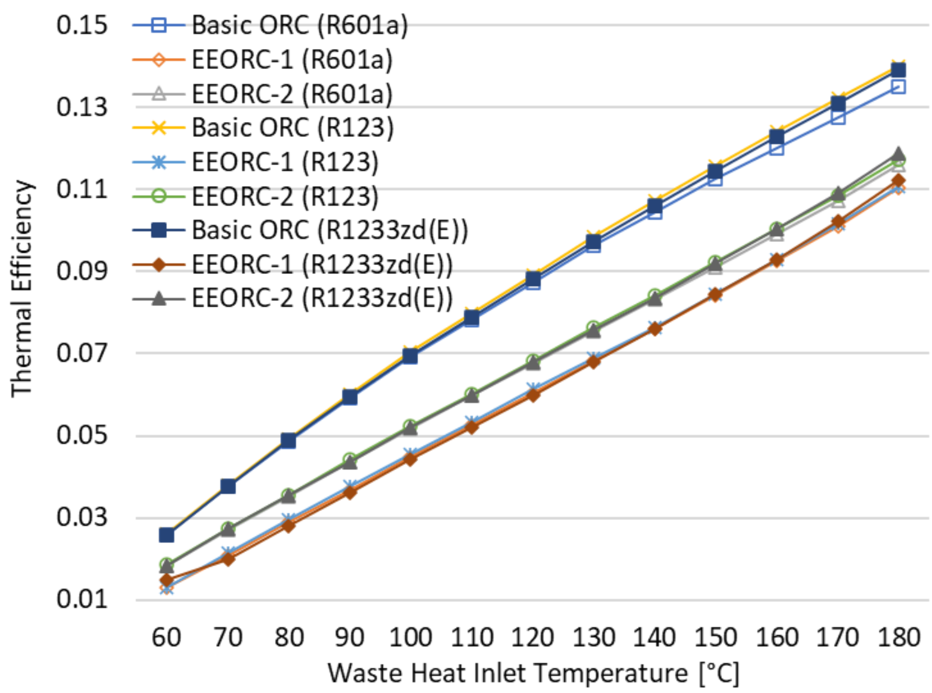

Figure 3.9. Effect on thermal efficiency with waste heat inlet temperature

Figure 3.10(a) and Figure 3.10(b) represent the effect on the entrainment ratio and pressure lift ratio with the variation of WHT, respectively. For all heat source temperature ranges, EEORC-1 has a lower entrainment ratio and higher-pressure lift ratio. The simulated value of the entrainment ratio and PLR is obtained from the maximum net work output condition. The R123-based cycle has the lowest entrainment ratio and highest PLR in its domain at a higher temperature range (greater than 140 °C) because of the fluid property. The lowest entrainment ratio and highest PLR are desirable for maximum net work output condition because of maximum enthalpy drop in the turbine due to more lowering of the turbine exit pressure.

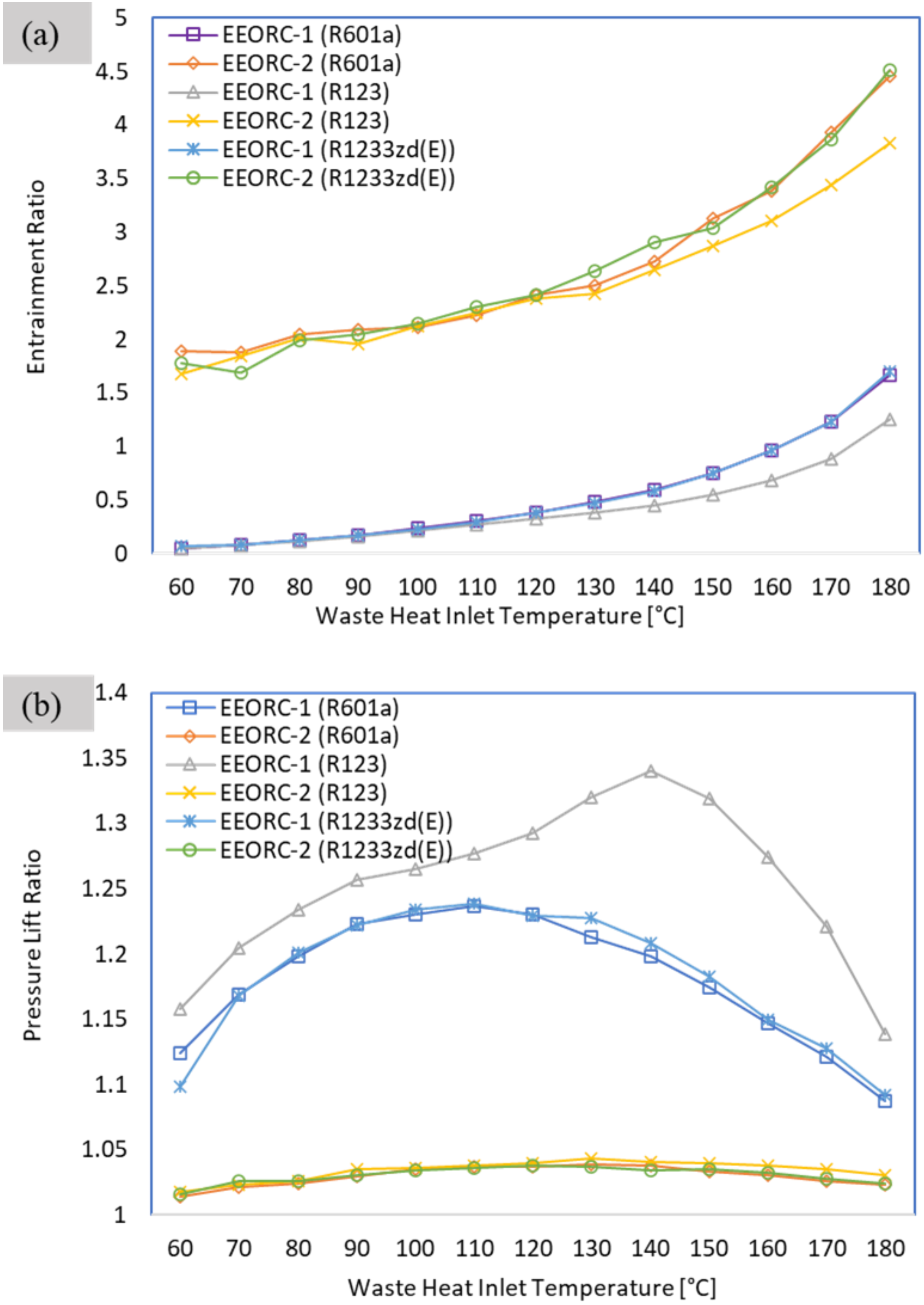

Figure 3.10. (a) Effect on entrainment ratio by variation of WHT, (b) Effect of PLR by the variation of WHT for different cycles

Figure 3.11 represents the effect on irreversibility at 70 °C for different components of various cycles and working fluids. This temperature is chosen for analysis purposes keeping in mind that the maximum % increase in net work output is at 70-85 °C temperature range and a comparative study is shown in Table 3.4 at 70 °C. Both ejector-enhanced cycles have higher irreversibility than basic due to higher evaporator pressure, ejector pressor and other components along with the ejector, which ultimately results in higher component irreversibility for maximum net work output. The presence of an extra component (ejector) will surely contribute to an increase in total irreversibility for EEORC. At a higher temperature range, each cycle's corresponding component irreversibility value will be lowered. Irreversibility of the evaporator and condenser becomes higher for another ejector-enhanced cycle due to handling more heat and thus the heat exchanger size increases. EEORC-1 with R1233zd(E) has the highest irreversibility and basic ORC with R123 has the lowest irreversibility at the heat source temperature of 70 °C.

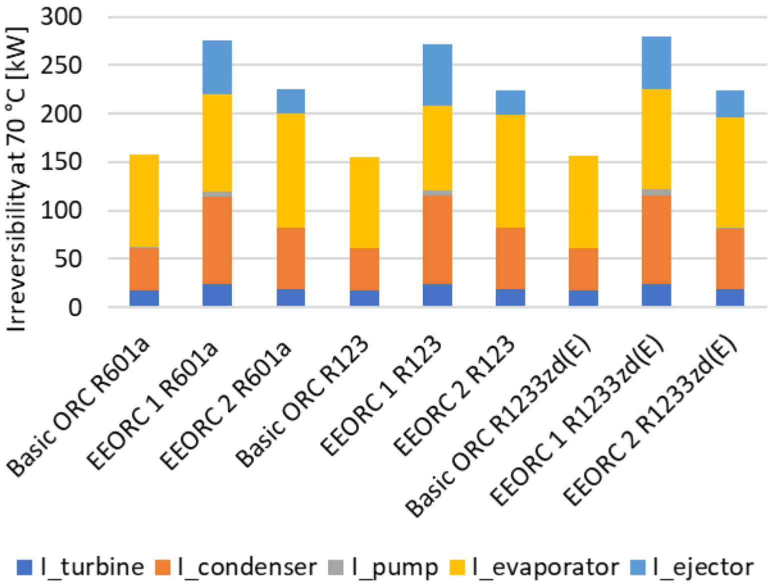

Figure 3.11. Irreversibility value of different cycle components at 70 °C

Figure 3.12 shows the variation of exergy efficiency along with waste heat inlet temperature. Exergy efficiency increases rapidly at the heat source temperature range of 60-90 °C because of the organic working fluid property. Basic ORC has higher exergy efficiency than studied ejector-enhanced cycles due to a large amount of input heat required (deduced by parametric optimization) at the same heat source temperature and thus exergy input will also increase. With the increase in heat source temperature net work output and exergy inlet will also increase and cumulatively decides how much exergy efficiency would increase. The exergy efficiency of basic and EEORC is significant at low WHT and further merges at higher WHT because of more and further less percentage increase in exergy. Basic ORC has the highest exergy efficiency and is not dependent upon working fluid used at a lower temperature range.

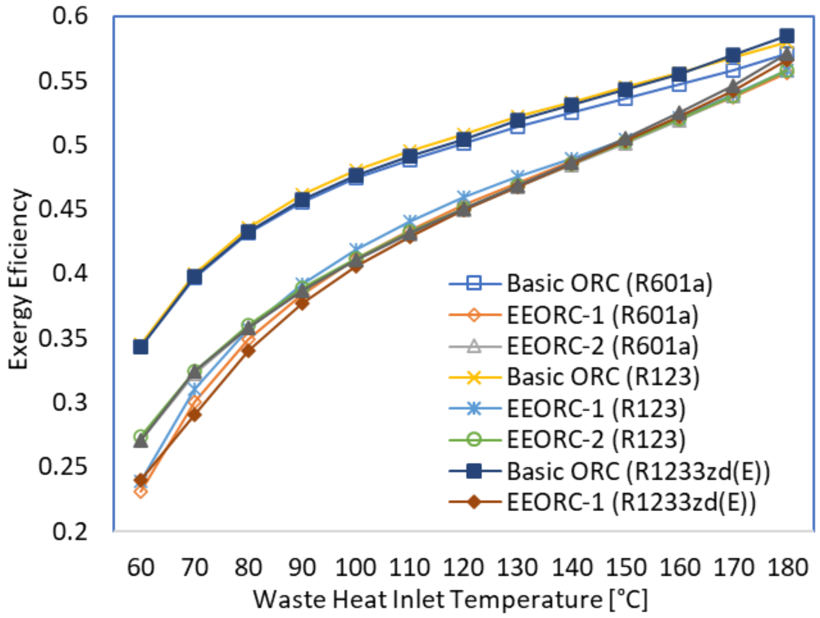

Figure 3.12. Effect on exergy efficiency with WHT for different cycles

Figure 3.13 represents the variation in size parameter with WHT. Size parameter of R123 working fluid-based cycles is the largest because of the cumulative effect of fluid property, lesser turbine isentropic enthalpy difference and higher turbine exit volume flow rate and fluid flowing through the turbine to produce maximum net work output from the shaft. From Figure 3.13, it can be concluded that the turbine size parameter is more dependent on the types of working fluid used rather than the types of cycles. The size parameter for all cases increases with increasing the WHT. The turbine size parameter directly affects the turbine cost and hence total cost consumed is the maximum for R123-based thermodynamic cycles.

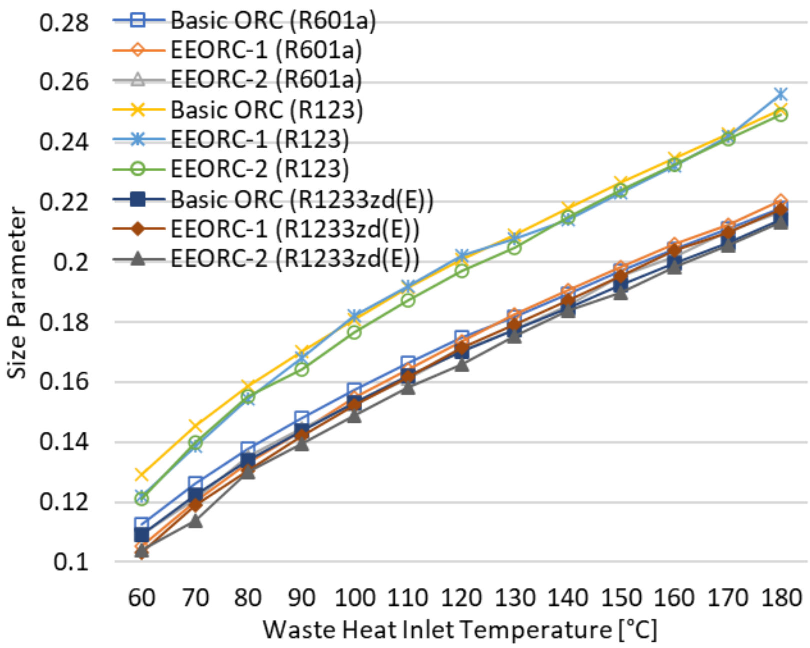

Figure 3.13. Effect on size parameter of turbine with WHT for different cycles

Figure 3.14 represents the variation in the total heat exchanger area (combined evaporator area and condenser area) with WHT. The total heat exchanger area increases with WHT. The significant deviation in the heat exchanger area at a higher heat source temperature range is due to the handling of more heat interaction in the evaporator and condenser to achieve the minimum log mean temperature difference value to maximize the net work output. This higher heat exchanger area value may bring down by appropriate judging the selling price of net work output value and capital investment in the heat exchanger. A larger area heat exchanger costs higher capital investment. EEORC-1 has a larger area because this cycle produces maximum power and so a larger heat input value requires at optimum pressure and lowest log mean temperature difference condition. Basic ORC has a lower area value because of less power production and simple thermodynamic heat exchanger design. The total heat exchanger area for EEORC-1 and EEORC-2 is dominated by the evaporator and condenser areas, respectively.

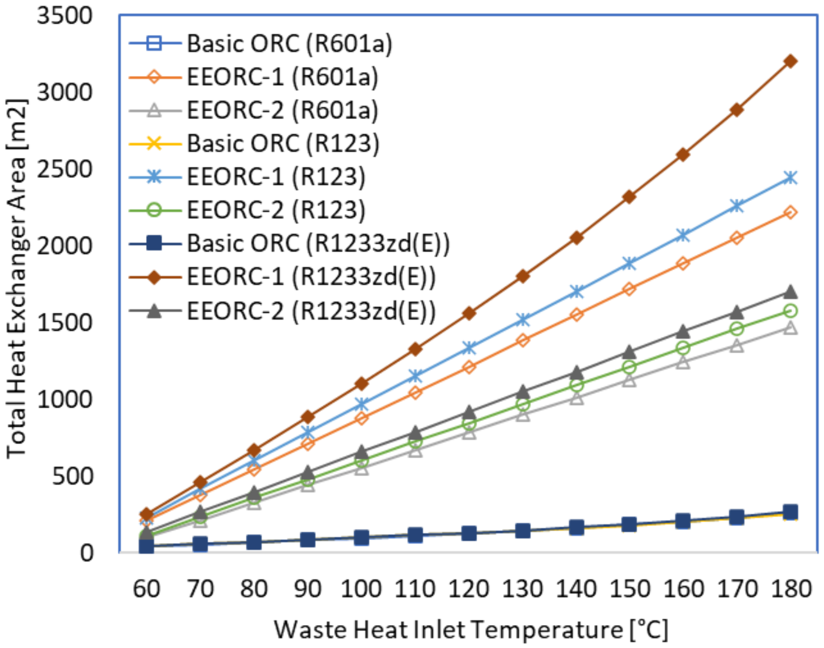

Figure 3.14. Effect on total evaporator and condenser areas with WHT for different cycles

Figure 3.15 represents the total cost consumed by different cycles for individual cycle components at 70 °C. The total cost is the sum of the capital investment, running cost and maintenance cost calculated (Table 3.4) per hour basis. As expected, the basic cycle has the lowest and EEORC-1 has the largest total cost rate due to the cost associated with the heat exchanger area and turbine size. Pump cost is significant in EEORC-1 due to more handling of working fluid mass. EEORC-1 with R1233zd(E) has the highest value due to maximum evaporator pressure and area, so the evaporator cost rate increases significantly. The available correlation [103,104] used to determine the turbine cost is for general-purpose international standards and the author feels that there is a need for some modified turbine cost correlation specially designed for waste heat recovery application for better prediction of turbine cost which will be locally manufactured.

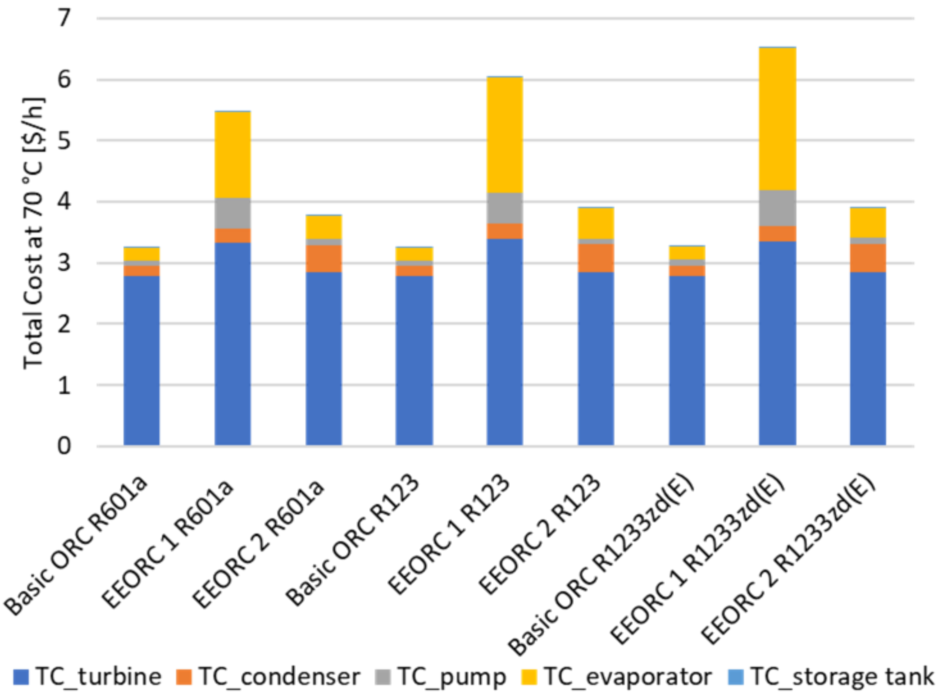

Figure 3.15. Value comparison on components cost rate of different cycles at 70 °C

Figure 3.16(a) represents the per-hour profit and per-unit selling price of three new different proposed economic models with the variation of WHT. Figure 3.16(b) represents the total annual profit for each power cycle and working fluid combination. In the figure, results show when old technology is completely replaced by new technology, then which economic model should follow so that both consumer and producer get the benefit? With the new technology, power plant production capacity will increase at the same WHT, which results in more availability (generation) of electric power. In developing countries, there is a scarcity of power due to the economy, technology and population. So, for fast development, such technology is required which can produce power at a cheaper rate with the support of the government so that more population can get benefitted from more hours of power availability but at a cheaper rate. In the current period, electricity is being sold to a consumer (country citizen) and per unit electricity price may be increased or decreased depending upon the government/industry price standard. Advancement in technology plays no role in per-unit electricity price fluctuation. Figure 3.16(a) represents the result for economic model-1 (M1) when per unit electricity selling price (SP) is the same, economic model-2 (M2) when total profit (TP) is the same, economic model-3 (M3) when unit profit (UP) is same and compare these results with the old technology having an existing economic model. In economic model

M1, the total profit rate is enhanced and maximum among other economic models at higher WHT (120-180 °C), by keeping the selling price constant with respect to the model followed by old technology. In this model, manufacturers enjoy gaining more profit, but the consumer will only get more hours of daily power availability at the same cost. In economic model M2, the total profit rate is constant but the selling price decreases with respect to the old technology economic model, so this model is favorable for the consumer as they get more hours of power availability at a cheaper rate and manufacturer's total profit remains the same. This model can be implemented by government power plants for the welfare of its citizen, especially those situated in rural areas. In economic model M3, the total profit rate is increased and the selling price decreases at a higher WHT range (95-180 °C). This model is not suitable for the lower WHT range, i.e., 60-95 °C, because of its higher value of selling price than the old technology. Figure 3.16(b) Variation on annual Net Profit with WHT for different cycles and working fluids when only existing (old) economic model is considered, among which EEORC-1 gives better results than EEORC-2. This graph serves the purpose of showing the variation in annual profit for different cycles and fluids for the old existing economic model. EEORC-1 gives maximum profit at the WHT range of 85-110 °C with a maximum of 7.32 % increment in profit by EEORC-1 (R123) at 90 °C.

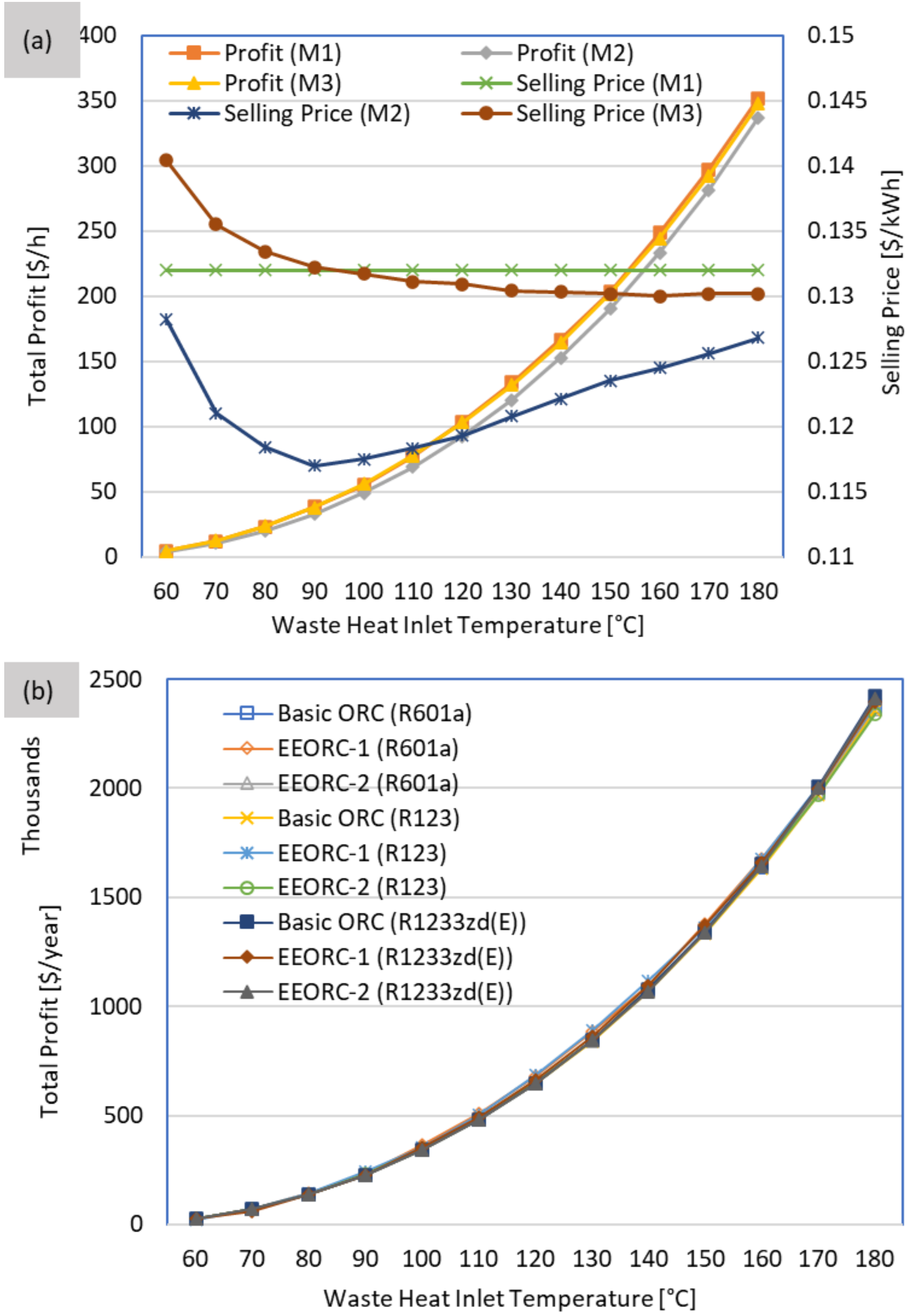

Figure 3.16. (a) Variation on total profit and selling price of EEORC-1 (R123) with WHT, (b) Variation on net annual profit with WHT for different cycles and working fluids

Figure 3.17 represents the amount of yearly CO_2 saving with the variation of WHT for different cycles. When power is produced from conventional coal-fired plants and then from boiler exhaust, a large amount of CO_2 emits, but when the same amount of power can be extracted from the waste heat source and at zero CO_2 emission, this is called CO_2 saving. There are many national and international agencies that encourage industries to emit as low as possible CO_2 and they support and encourage in terms of the monetary value of carbon credit gained. The amount of CO_2 saved converted into monetary value further adds profit to the industry. EEORC-1 saves the maximum amount of CO_2 and then EEORC-2 and basic ORC at all WHT ranges and thus add extra yearly profit.

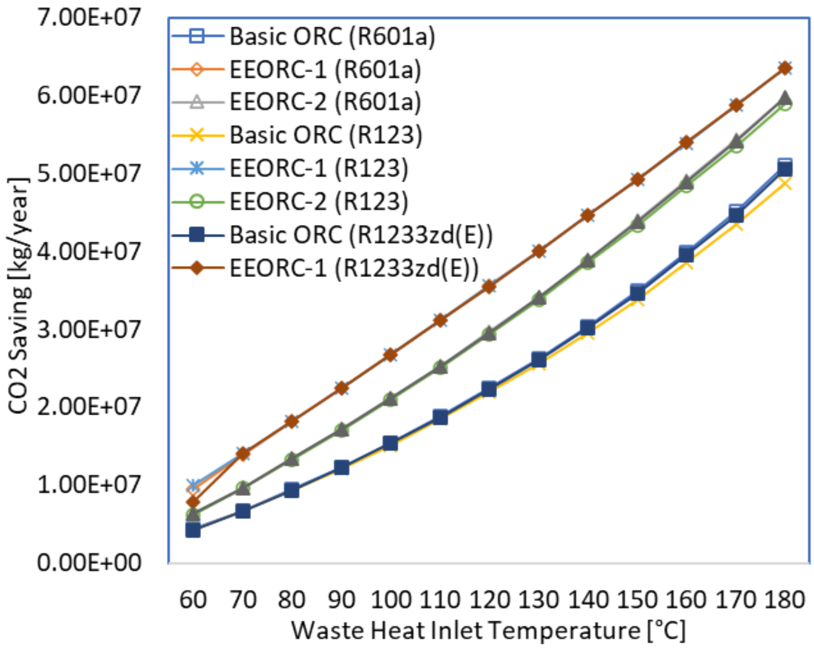

Figure 3.17. Variation of CO_2 saved with WHT for different cycles and working fluids

3.4. Important Findings

Two new ORC modification (EEORCs) are proposed and compared their 4E performances with the basic ORC for maximum net work output condition. Three economic models are chosen and used for the profit comparison of proposed cycles with the basic cycle for more power availability at a cheaper rate. By analysing the simulation results, the following conclusions are made.

(i) EEORC-1 with R123 gives maximum value for net work (122.2 kW), whereas basic ORC with R123 gives maximum thermal efficiency (3.8%), exergy efficiency (40.0%) and yearly profit (73143 \$/year) at 70°C WHT.

(ii) For the studied heat source temperature range and working fluids, EEORC-1 yields maximum net work output with R123 for a lower temperature range (60-150 °C) and with R1233zd(E) for a higher temperature range (155-180°C).

(iii) Thermal and exergy efficiencies increase with WHT and are maximum for basic ORC with all studied working fluids for the whole studied range of WHT.

(iv) The heat exchanger size of EEORC is higher, whereas the size parameter is fluid-dependent (higher for all cycles with R123) for all WHT ranges and thus, the turbine cost increases significantly with cycle advancement.

(v) If economic model M1 is considered, then basic ORC with R123 will provide the highest yearly profit. Economic model M2 is the best when the industry gets subsidized by the government for new technology installation and in the reverse industry has to provide power at a lower selling price. The industry enjoys a 7.32% increase in profit with M2 model (compared with M1) when EEORC-1 (R123) is running at 90 °C WHT (some economic models are not feasible at 70 °C).

(vi) The EEORC-1 cycle saves the maximum amount of CO_2 emission, which will significantly reduce global warming. Also, CO_2 emission savings can be monetized in terms of carbon credit given by different international and governmental agencies, which can further benefit the power plant industry.

Overall, it can be concluded from this study that EEORC-1 is the best thermodynamic cycle among studied cycles as it produces maximum net power output with R123 and saves the maximum amount of CO_2 emission. The economic model M2 is the most suitable (among M1, M2 and M3), where both the industry and consumer get benefitted.