THERMODYNAMIC ANALYSIS OF COMBINED ORGANIC RANKINE CYCLE USED FOR POWER GENERATION

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This is to certify that report entitled **“THERMODYNAMIC ANALYSIS OF COMBINED ORGANIC RANKINE CYCLE USED FOR POWER GENERATION”** by **MANISH KUMAR** is in partial fulfilment for the award of Master of Technology (M.Tech) in **Thermal Engineering at Delhi Technological University**. This work was completed under my supervision and guidance. He has completed his work with utmost sincerity and diligence. The work embodied in this project has not been submitted for the award of any other degree to the best of my knowledge.

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ABSTRACT

In this study the Combined Organic Cycle focuses the performance of two fluids R123 and R245fa. Though earlier studies hold innumerable options, this work provides a much clearer statement on IHE in subcritical ORC system setting a new model taking pressure drop by loops and pinch point into thought. The common operating fluids R123 and R245fa have been chosen for subcritical cases. A steady temperature of 200ºC is used for warmth supply while maintaining a constant mass flow rate of it is 1 kg/sec. The Engineering Equation solver (EES) software package is utilized for analysis. A different approach to calculation of most heat exchange in Internal Heat Exchanger (IHE) is use when modelling a supercritical cycle, due to large dynamic heat component close to the critical point. Also, an alternative approach is required to compute the outlet temperature of the warmth source and identify the point of the pinch in supercritical cases. The results prove that IHE is useful in a subcritical case, but will improve system performance only partially for the low pressure stage in a supercritical scenario. In subcritical cycle it's discovered that power obtained is 27.33 kilowatt and cycle efficiency is 12.03% at an evaporator pressure of 1326 kPa and therefore the second law efficiency is 40.41% at an evaporator pressure of 1326 kPa using fluid R123. The cycle efficiency is coming out to be 11.27%, the second law efficiency is coming out to be 38.67% and net-work is 28.38 kW using fluid R245fa. Analysis is done on combined Organic Rankine cycle in which the exhaust coming out of heat exchanger of Brayton cycle is supplied to the Organic Rankine cycle. The analysis is done on both fluids R123 and R245fa.

In combined cycle 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.

Also, in this study various cases of Brayton cycle is analyzed such as simple Brayton cycle, with intercooling, with reheating and with both intercooler and re-heater. The study is done on fluids R123 and R245fa.

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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

SUBSCRIPTS:

O	Environment
1-22	State points of the Cycle
ad	acid dew point
bp	boiling point
C	cold stream
cond	condensation
Cr	Critical
evap	evaporation
H	hot stream
IHE	internal heat exchanger
In	flow into the system
Out	flow out of the system
P	Pump
pp	pinch point
r	working fluid
sub	subcooling
sup	supercooling
exp	expander

INTRODUCTION

Energy security, economic development and atmosphere protection are not very well balanced these days which has put further pressure on energy demand which is closely connected to the economic process. At the same time, the over-consumption of fossil fuels has led to a lot of surroundings issues like heating, ozone depletion and atmosphere pollution. Fossil fuels are still the dominant form of energy resources worldwide, accounting for 77 of the increasing energy demand 2007-2030 (IEA 2009). When combined with the quick development of business, a significant increase in energy shortages and blackouts has been noticed a lot frequently everywhere around the globe. Due to of those reasons, utilizing inferior waste heat for energy consumption. The most ordinarily investigated in inferior heat supply and waste heat utilizations these days are Organic Rankine Cycle (ORCs) and Kalina Cycle (binary fluids and fluid mixtures). When low grade heat is used, the conventional steam Rankine cycle does not offer acceptable performance as a result of its poor thermal efficiency and big volume flows, and so therefore known as Organic Rankine Cycle (ORCs) are projected. These cycle use organic substances as operating fluid system comprises a minimum of 4 major comparisons, a regenerator is additionally enclosed within the current calculation.

According to waste heat recovery plant by U.S. DOE, industrial processes within U.S. consumes about 35 quadrillion BTU (Quads) of energy annually (BCS INC 2008). This quantity is concerning 30 minutes of total energy consumed within the U.S. This report categorizes the waste heat supported the temperature of the waste material. Table 1 in addition shows of waste and work potential of each waste heat cluster (BCS INC, 2008). The waste heat and work potential depends on the reference temperature of 25ºC. This information indicated that low temperature heat supply leads to 60 minutes of total waste heat supply.

It's calculable that 290 trillion energy unit annually or 33rd of this low temperature heat will be recaptured into helpful works. Low temperature heat supply recovery presents the biggest chance to recover among alternative discarded heat.

1.1 Organic Rankine Cycle

Organic Rankine Cycle can generate power from natural and waste heat sources similar to geothermal, biomass, radiation and waste heat returning from industrial processes and fuel consumption, solar radiation etc. Therefore, it is considered an important and promising method of renewable energy production process. The ORC uses associate organic substances for operating fluid as implicit by the word 'Organic'.

In Organic Rankine Cycle power system, the selected fluid should increase the thermal efficiency considering the heat supply and long term performance temperature range. The temperature ranges from 100ºC to about 350ºC.

	Temperature Range		**Waste Heat (trillion BTU per year)**		**Work potential (trillion BTU per year)**
	ºF	ºC	77F[25ºC]	300F[150ºC]	77F[25ºC]
Low	<450	<230	903	37	287
Med	450-1200	230-650	466	130	216
High	>1200	>650	108	89	86
Total	-	-	1478	256	589

Table 1.1: Yearly national unrecovered heat

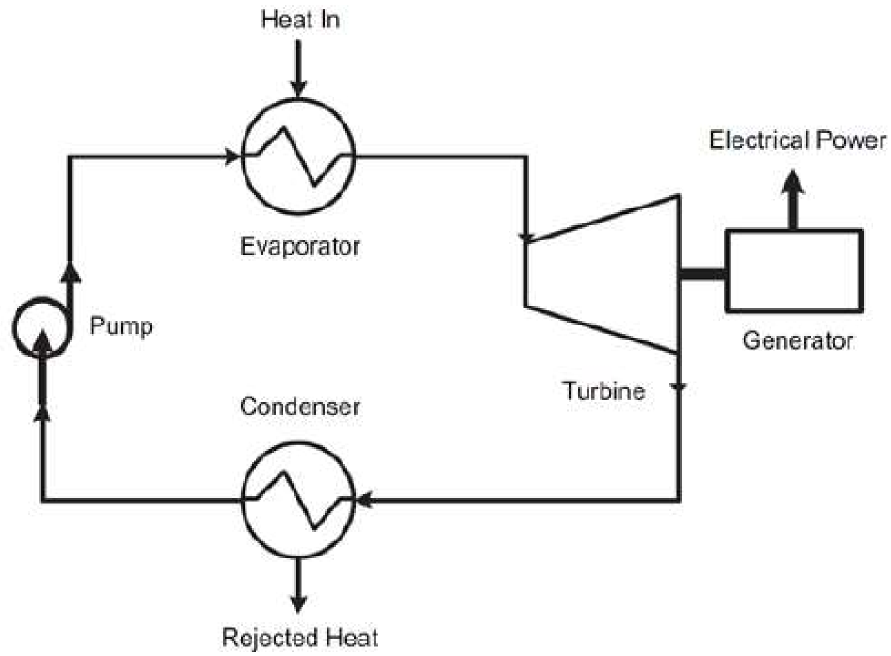

Fig 1.1 Basic Organic Rankine Cycle

Rankine cycle gives the highest efficiency once low temperature heat is regenerate into electricity. The waste heat coming from combustion engines and processes in industries: hot exhaust gases coming from blast furnaces in steel industry and from the kilns in ceramic industry, exhaust gases returning from the engines and turbines, hot liquids that are went to cool kilns and furnaces etc. The ORC uses organic fluid as operating fluid in place of water. Organic Rankine Cycle is employed for establishment of the many commercial plants and check plants. The refrigerants that are used most typically in ORC include R123, R600, R11 and R114 etc. These days the interest is concentrated on new refrigerants like R1234yf, R1234ze etc.

The net output, initial and second law efficiency are the indication of system performance besides the operating fluid properties. As far as maximum pressure is bothered, the ORC will be subcritical and supercritical. The cycles will be utilized with internal device or while not IHE. Sure discussions are continued with some report station that exergy, efficiency and power output stays unchanged whereas some stating thermal efficiency rises however output is stationary. It’s also expressed that inductions of IHE greatly reduces expansion ratio, evaporation ratio, evaporation pressure being below a certain limit.

1.2 Simple Brayton Cycle:

Named after the Boston engineer, George Brayton, the basic gas turbine cycle was initially proposed around 1870. The Brayton cycle is useful for turbines where both compression and expansion occurs in rotating machinery. The basic gas turbine was patented by John Barber in 1791. Gas turbine engines are popularly used in aircraft propulsion and electric power generation. They are also common at stationary power plants, where they are used to generate electricity as stand-alone units or with steam power plants for a combined output.

The Brayton cycle highlights the standard model for a gas turbine power cycle. A compressor, a combustor, and a turbine constituted the three primary components of a gas turbine. According to the principle, the compressor compresses the air, which is then mixed with fuel and burned in the combustor under a consistent pressure. This air can also be heated by a waste heat flow. The resultant high pressure and temperature gas is then allowed to expand through a turbine to perform work. The majority work produced in the turbine is used to support the compressor while the remaining is available for further applications. This is commonly used for stationary power generation plants (electric utilities) and mobile power generation engines (ships and aircraft). For power plant applications, the turbine output is used to provide shaft power to drive a generator. A jet engine aircraft is propelled by the reaction thrust of the exiting gas stream. The turbine provides enough power to run the compressor and deliver auxiliary power. The gas stream acquires further energy in the cycle than what is necessary to drive the compressor. The remaining energy is then used to propel the aircraft forward.

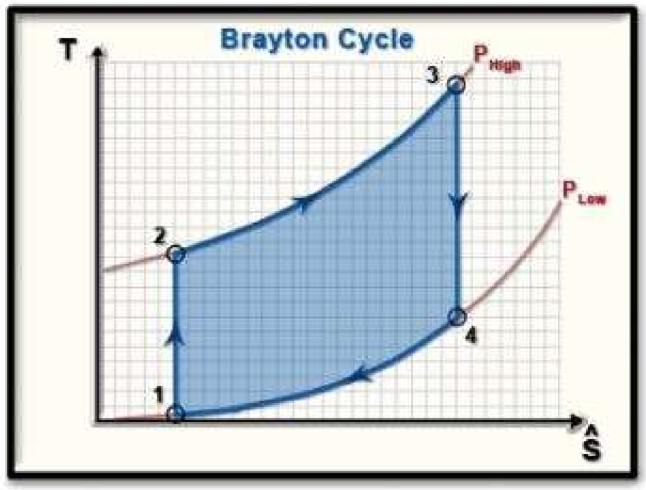

Fig. 1.2 T-S Diagram of Simple Brayton Cycle

1.3 Advantages:

Rankine Cycle gives the most effective efficiency and power output with an organic fluid as an operating fluid rather than water once waste heat at moderate inlet temperature is used as a heat source. This is mainly due to the fact that the specific vaporization heat of water is way more than organic fluids. Thus, the warmth supply fluid that's to be cooled is followed higher by the organic operating fluid. In the ORC method more electric power can be generated from a source of heat when exhaust gas is cooled down.

1.4 Working Fluids

The following operating fluids during this study: R123 and R245fa. Several researchers investigated these operating fluids and are representative operating fluids that are employed in ORC, Not solely HFC-134a is employed however additionally HFC-245fa and iso-pentane are employed in an Organic Rankine Cycle to collect low heat from the industrial plants.

Ethanol performs fine at medium temperature level i.e. 200-300ºC hence, it is also a common as an automotive ORC operating fluid. The evaporation temperature for organic substances is comparatively lower as compare to steam, this being the most reason and advantage of exploitation them as operating fluid in ORC. The low evaporation temperature makes it attainable for them to vaporize or superheat with just low or medium heat sources. Additionally, the expander life are often extended by using the organic substances which are classified as dry or isentropic fluids. Under the saturated or superheated tends to be dry. The particular performance needs to be verified by playing elaborate simulations for the model that employs heat exchanger. The major focus these days is kept on the subcritical cycle, however a number of the research on critical cycle using heat exchanger has also been done. The aim of the study is to identify the impact of internal heat exchanger on the sub-critical, as well as super-critical Organic Rankine Cycle. Net power output, thermal and exergy efficiency are the key performance analysis parameters observed in this study.

The study analyses the subcritical and supercritical model on an individual basis for locating the pinch point and scheming outlet temperature for hot stream, a nested iteration methodology is projected. It's found within the conclusion that the system performance is promoted by internal heat exchanger in subcritical case whereas in supercritical cases the internal heat exchanger's effects are passionate about a particular heater pressure.

1.5 Saturated Cycle Model:

With a compact system design, this is the most basic cycle that can achieve an acceptable thermal efficiency when recovering low temperature waste heat.

The T-S diagram of the cycle within the Fig 1.3. It shows that a continuous pressure is maintained on the operating fluid after isentropic pumping in the evaporator until it becomes a saturated vapor. This is then followed by isentropic expansion method.

Based on the type of operating fluid, the condition of operating fluid under the expansion method can be

a) A two phase mixture (wet fluids)

b) Superheated state (dry fluids)

c) Saturated vapor state (isentropic fluids)

After the expansion is complete, the fluid undergoes isobaric condensation in the condenser and reverts the fluid back to a saturated liquid form.

1.6 Trilateral Cycle Model:

It is an easy cycle as compared to traditional binary cycle. This cycle results in an improved competitive performance for low temperature sources, especially when a proper two phase expander is utilized. Additionally, the overall cost of operation is low when equivalent parts are used.

This cycle consists of 2 features, the first being that the supply temperature profile and operating fluid temperature profile are matched perfectly. Also, the overall operation is maintained at reasonable pressure. The schematic diagram is shown within the Figure 1.4. The pumping pressure that is on paper same as heater pressure is lower than the critical pressure of operating fluid, while the temperature for expander inlet is kept at a target saturated temperature.

The cycle involves a physical state of operational fluid preceding its isobaric heating to the bubble point. This is then followed by isentropic expansion of operating fluid to a 2 phase mixture and then back to a liquid state. The first difference between natural and standard cycle is that the operating fluid is in a liquid state when it leaves the heater and thus the expansion happens inside the two-phase range.

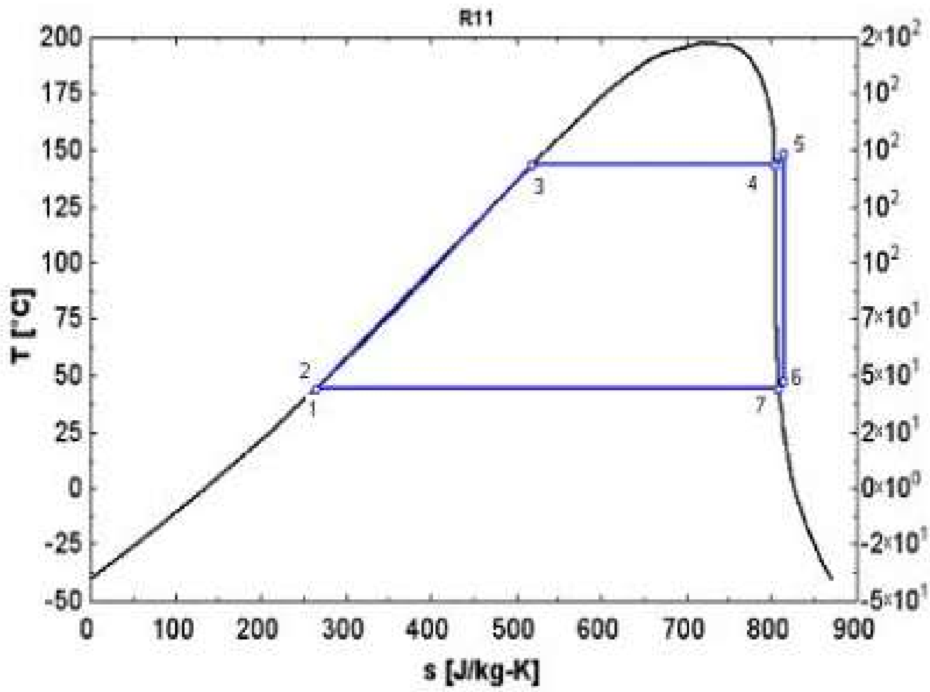

Fig 1.3 T-S diagram of Saturated Cycle

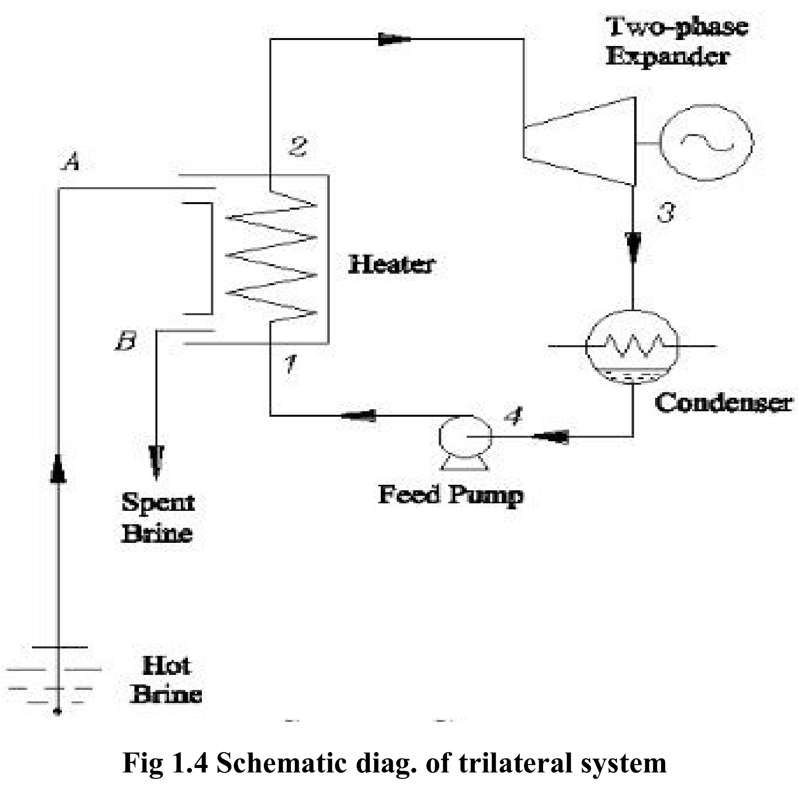

Fig 1.4 Schematic diag. of trilateral system

1.7 Superheated Cycle Model:

Unlike the saturated or trilateral cycle, the superheated cycle permits additional heat into the ORC system which changes the operating fluid to a superheated state. The superheated cycle, in some cases, achieves a much higher thermal efficiency compared to that of saturated cycle.

After the isentropic pumping method, the fluid it heated to a superheated state, with a temperature exceeding saturated vapor state at a similar pressure. Since the high inlet temperature results in a larger production, the temperature is maintained above the critical value at the expander inlet. The operating fluid is then passed through the expander with, followed by the isentropic growth and is then condensed to a saturated liquid under constant pressure.

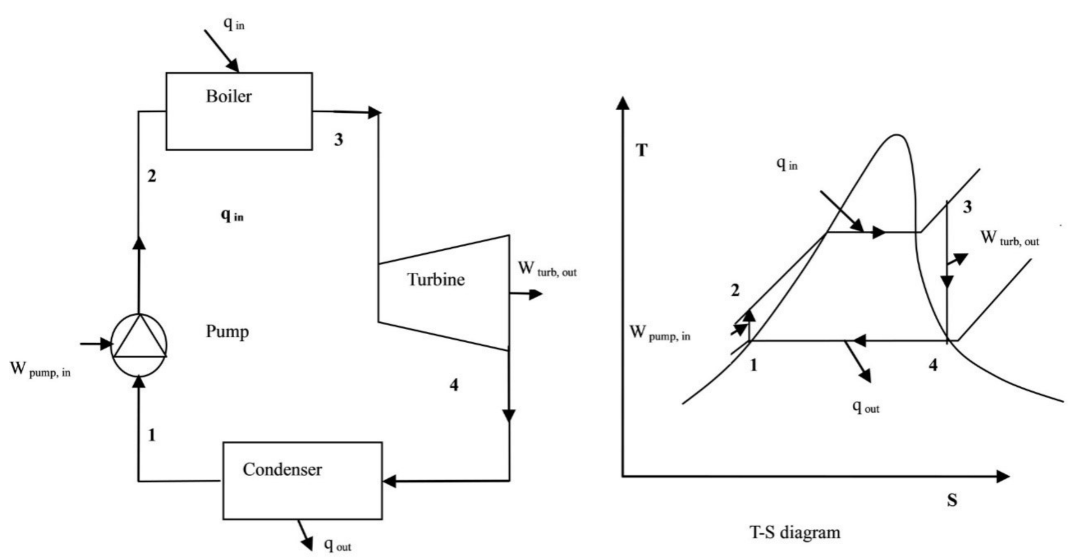

Fig 1.5 Schematic diag. and T-S plot of Superheated System

1.8 Subcritical Cycle Model:

The subcritical cycle have nearly operational condition as in superheated cycle, except that the pumping pressure is maintained above the pressure required in case of the superheated cycle while still being under the critical pressure.

The T-S diagram is shown within the Figure 1.6 that sounds like the superheated cycle. The processes involve isentropic pumping followed by isobaric heating, isentropic expansion and at last isobaric condensation.

The expander water temperature is kept beyond the critical temperature of operating fluid once the expansion process during this cycle.

1.9 Supercritical Cycle Model:

In the critical Cycle, the expander inlet conditions i.e. pressure and temperature are unbroken above the critical point of operating fluid.

The main advantage of the cycle is that the hot supply temperature will be raised to the next level that can't be reached in alternative cycles.

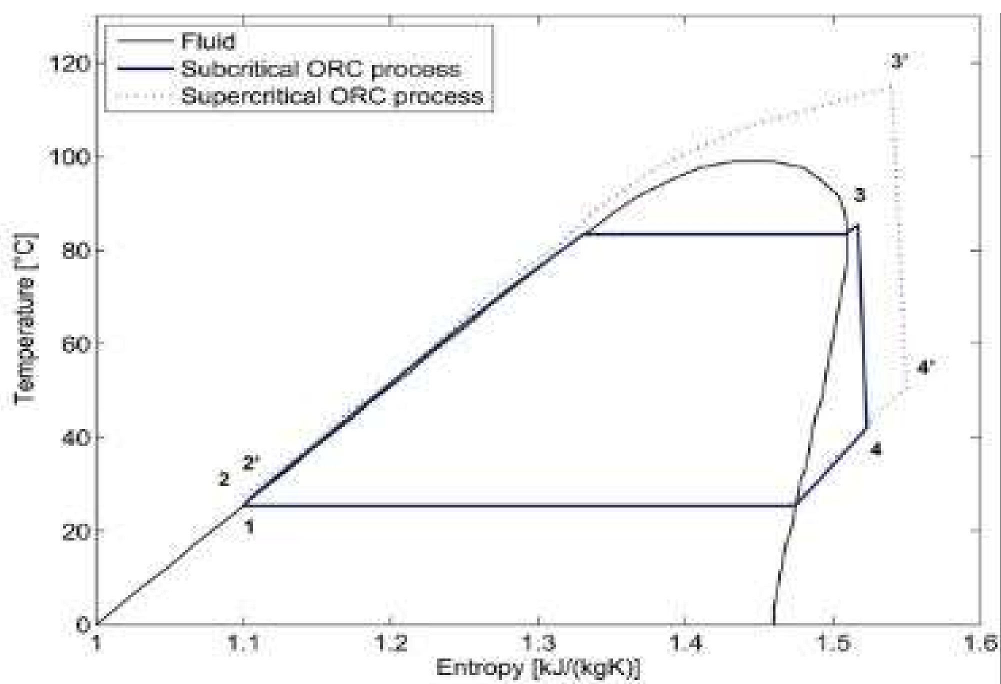

Fig 1.6 T-S diag. showing subcritical and supercritical

CHAPTER 2

LITERATURE REVIEW

Declamatory literature is available on Organic Rankine Cycle, thermodynamic modeling and simulation of thermal efficiency with various working fluids.

A brief of literature survey is given below:

2.1 Literature survey on Theoretical & Experimental study of ORC

Serrano et al. [1] carried out the theoretical investigation and has instructed the feasibleness of introducing a waste heat recovery system in 2 stage turbocharged HDD engine. The waste heat recovery (WHR) is earned by introducing a Rankine cycle that uses an organic substance or directly water as operating fluid looking on energetic performance concerns. They additionally instructed an alternate for rising the general thermal efficiency of internal-combustion engine consists of recovering the energy lost by suggests that of a waste heat recovery (WHR) system. These solutions are supported adapting one among the turbochargers by removing its rotary engine and making an attempt to recover the energy by Rankine cycle. Finally, the rotary engine of the Rankine cycle provides the recovered energy on to the compressor of this turbocharger.

J. et al. [3] suggested the strategy for customizing an organic Rankine cycle to a fancy heat supply for economical energy conversion, demonstrated on a Fischer Tropsch plant. Organic Rankine Cycle offer an alternate to ancient steam temperature unit cycle for the conversion of low grade heat supply into power, wherever typical steam power cycle is thought to be inefficient. A large process plant typically has multiple low temperatures waste heat supply obtainable for conversion to electricity by a low temperature cycle, leading to composite heat supply with a complex temperature-enthalpy profile. Organic fluids are sculptured employing a pure substance information. The pinch analysis technique of forming composite curve is applied to research the impact of every building block on the temperature-enthalpy profile of the ORC heat demand.

Wang et al. [4] carried out the performance analysis of double organic Rankine cycle for discontinuous temperature waste heat recovery. The optimum operation of many operating fluids are calculated by a procedure using MATLAB and REPROP. The influence of outlet temperature of warmth supply on internet power output, thermal efficiency, power consumption, mass rate of flow, expander outlet temperature, cycle irreversibility and exergy potency at a given pinch temperature difference has been used to get a thermodynamically understanding of the ORC performance.

Larjola [5] suggested that electricity from industrial waste heat exploitation high speed organic Rankine cycle. Within the conversion of cold heat into electricity the best efficiency is obtained in several cases by using AN Organic Rankine Cycle. The ORC method additionally be possible also in warm temperature applications. This paper deals with an ORC style, during which a high speed oil free turbo generator feed pump is employed.

Li et al. [6] carried out analysis of external heat loss from a little scale expander employed in organic Rankine cycle. With the lowering of the ORC, the engine shaft power isn't solely determined by the enthalpy drop by the expansion method however additionally the external heat loss from the expander. On paper and by experimentation support in evaluating tiny scale expander heat loss is rare. They give a quantitative study on the convection, radiation and conductivity heat transfer from a kw-scale expander. A mathematical model is constructed and valid.

Quoilin et al. [7] did performance and style optimization of a low value solar organic Rankine cycle for remote power generation. Recent interests in little scale solar thermal combined heat and installation has coincided with the demand growth for distributed electricity providers in space poorly served by centralized power stations. One potential technical approach to fulfill this demand is parabolic trough solar thermal collector including Organic Rankine Cycle engine. He additionally describes the planning of solar organic Rankine cycle being put in in Kingdom of Lesotho for rural electrification purpose. The system carries with it parabolic trough collectors, a tank, a tiny low scale ORC engine scroll expanders. A model of every part is developed that permits sizing the various parts of cycle and evaluates the performance of system.

Heberle et al [8] did energy primarily based fluid choice for a geothermic Organic Rankine Cycle for combined heat power generation. During this study the choice of combined heat and power generation was thought-about for geothermic resources at a temperature level below 450k. Series and parallel circuits of an organic Rankine cycle and extra heat generation was organized by second law analysis. The result shows that the second law efficiency of energy station will be considerably enlarged compared to power generation.

Liu et al. [9] carried out investigation of 2 stage Rankine cycle for power plant. A 2 stage Rankine cycle for power generation is given during this paper. It's product of water steam Rankine Cycle and Organic Rankine bottoming Cycle. By using an organic fluid with density over water. The performance of 2 stage Rankine cycle operative with those completely different operating fluids is evaluated. System efficiency may be increased by introducing a regenerator for a few of operating fluids.

Aleksandra & Borsukiewicz [10] suggested pumping within Rankine cycle and created calculation based mostly result for the pumping work on ORC. Analysis has been administrated for eighteen completely different organic fluids that may be used an operating fluids within the subcritical ORC system. A trial was created to search out correlations between numerous thermo-physical properties of operating fluids, specific work and power of cycle.

Clemente et al. [11] proposed the energy potency analysis of organic Rankine cycle with scroll expanders for co-generative applications. The model of scroll machine is applicable to calculate the performance of each a compressor and expander, as perform of pure mathematics of device and dealing fluid.

Garcia-Rodriguez et al. [12] proposed solar power-driven Rankine cycle for water production. He additionally centered that on the analysis of distributed solar power-driven generation systems for driving a reverse diffusion method process. Results signifies the desalinization system coupled to solar power-driven organic Rankine cycle exhibit lower specific consumption of solar power than solar distillation.

Sprousell and Depick [13] suggested the review of temperature unit Cycle for combustion engine exhaust waste heat recovery. This paper review the history of combustion engine exhaust waste heat recovery specializing in organic Rankine cycle since this thermodynamical cycle works well with medium grade energy of exhaust. Choice of cycle expander and dealing fluid are primary focus of the review. Results demonstrate a possible fuel economy improvement around ten.

Florian and Dieter [14] suggested exergy based mostly fluid choice for a geothermic Organic Rankine Cycle for combined heat and power generation.

Algieri and Morrne [15] suggested the comparative energetic analysis of hot temperature subcritical and transcritical Organic temperature unit cycle during a biomass application within the Sibari district. The current work aims to research the energetic performance of ORC's for tiny scale applications.

Hui-tao et al. [16] did the improvement of temperature exhaust gas waste heat fuelled organic Rankine cycle. Cycle parametric analysis was performed to look at the consequences of thermodynamical parameters on the cycle.

Kosmadakis et al. [17] suggested the constant theoretical study of a 2 stage solar Rankine cycle for Ro distillation. The current work issues the constant study of an autonomous 2 stage solar ORC for Ro desalinization. The most aim is to estimate the efficiency similarly on calculate annual energy out there for desalinization.

J.P. Roy and Ashok Mishra [18] presents an analysis of regenerative Organic Rankine Cycle (ORC), supported constant quantity optimization using R-123 and R-134a throughout superheating at a continuing pressure of 2.50 MPa underneath realistic conditions. The aim was to pick a much better operating fluid on the idea of obtained system efficiency, turbine work output, irreversibility rate and second law efficiency underneath applied fastened and variable heat supply temperature conditions, R-123 has been found a more robust operating fluid than R-134 for changing low grade heat to power. A computer program has been developed to parametrically optimize and compare the system and irreversibility ratio with will increase in rotary engine water temperature (TIT) underneath completely different heat supply temperature conditions to get the optimum operative conditions whereas using R-123 as the operating fluid throughout superheating at varied rotary engine water pressures for the use of the waste heat sources of temperatures higher than 150^0C. The calculated results reveal that an water pressure of 2.70 MPa offers the most system efficiency, rotary engine work output and second law efficiency with minimum irreversibility rate, irreversibility magnitude relation and system mass rate of flow up to a TIT within the vary of 165^0C-250^0C.

W.Li et al. [19] did research within the paper and sets some initial conditions, boundary conditions, and hypothesis for a mathematical model. 3 types of pure fluid and one mixture were hand-picked as operating fluids and their constant quantity changes were calculated below totally different evaporating temperatures. Once organic Rankine cycle (ORC) is supplied with internal device (IHE), its parameters varies whereas using totally different operating fluids.

Dai Y et al. [20] did his constant optimization and comparative study of Organic Rankine Cycle for low grade waste heat recovery in saturated cycle.

Karellas et al. [21] has given a subcritical cycle and stimulated it beneath constant operation conditions as superheated cycle, however, the pumping pressure is unbroken above the pressure used for superheated cycle however slightly less than the critical pressure for operating fluids.

V. Lemort et al. [22] presents the results of an experimental study administered on a model of an open-drive oil-free scroll expander integrated into an ORC operating with refrigerant HCFC-123. By exploiting the expander performance measurements, the eight parameters of a scroll expander semi-empirical model are then known. The model is ready to calculate variables of 1st importance like the mass rate, the delivered shaft power and also the discharge temperature, and secondary variables like the availability heating-up, the exhaust cooling-down, the close losses, the interior escape and also the mechanical losses. The utmost deviation between the predictions by the model and also the measurements is two hundredth for the mass rate, 5-hitter for the shaft power and 3K for the discharge temperature. The valid model of the expander is finally wont to quantify the various losses and to point how the planning of the expander can be altered to realize higher performances. This analysis detected that the internal leakages and, to a lesser extent, the availability pressure drop and also the mechanical losses are the most losses moving the performance of the expander.

A.Schuster et al. [23] **investigated** and find out that the use of organic operating fluids for the conclusion of the thus known as Organic Rankine Cycle (ORC) has been established to be a promising resolution for decentralized combined heat and power production (CHP). The method permits the employment of low temperature heat sources, giving an advantageous efficiency in tiny scale applications. This is often the explanation why the quantity of energy and biomass fired power plants supported this technology are enhanced among the last years. The favorable characteristics of ORC build them appropriate for being integrated in applications like solar desalinization with reverse diffusion system, waste heat recovery from biogas digestion plants or micro-CHP systems. During this paper, the state of the art of ORC applications are going to be given beside innovative systems that are simulated in an exceedingly method simulation setting using experimental data. The results of the simulations like efficiencies, water production rates or possible electricity cost are given and mentioned.

F.Herbele& D. Bruggemann [24] studied the choice of combined heat and power generation was thought of for energy resources at a temperature level below 450 K. Series and parallel circuits of an Organic Rankine Cycle (ORC) and a further heat generation were compared by second law analysis. Looking on operative parameters criteria for the selection of the operating fluid were known. The results show that because of a combined heat and power generation, the second law efficiency of an energy station will be considerably enhanced as compared to a power generation, the second law efficiency of an energy station will be considerably enhanced as compared to a power generation. The foremost economical conception may be a series circuit with an organic operating fluid that shows high critical temperatures like iso-pentane. For parallel circuits and for power generation, fluids like R227ea with low critical temperatures are to be most well-liked.

J.P. Roy, M.K. Mishra, A. Mishra [25] carried out parametric optimization and performance analysis of a waste heat recovery system supported Organic temperature unit Cycle, using R-12, R-123 and R-134a as operating fluids for power generation are studied. The cycles are compared with heat supply as waste heat of flue gas at 140^0C and 312 Kg/s/unit mass rate of flow at the exhaust of ID fans for 4×210 MW, NTPC Ltd. Kahalgaon, India. Optimization of rotary engine inlet pressure for max work and efficiencies of the system on the saturated vapor line and isobaric superheating at completely different pressures has been carried out for the chosen fluids. The results show that R-123 at corrected pressure evaluated among all the chosen fluids. The Carnot efficiency for R-123 at corrected pressure evaluated underneath similar conditions is near to the particular potency. It will generate 19.09 MW with a mass rate of flow of 341.16 Kg/s having a pinch point of 5^0C, 1st law efficiency of 25.30% and also the second law efficiency of 64.40%. Therefore choice of an Organic Rankine Cycle with R-123 as operating fluid seems to be a alternative system for utilizing inferior heat sources for power generation.

J.Li, G.Pei, Y.Z.Li, J.Ji [26] find out that with the reducing of the Organic Rankine Cycle (ORC), the engine shaft power isn't solely determined by the enthalpy drop by the expansion method however additionally the external heat loss from the expander. Theoretical and experimental support in evaluating tiny scale expander heat loss is rare. This paper presents a quantitative study on the convection, radiation and conductivity heat transfer from a kW-scale expander. A mathematical model is constructed and valid. The results show that the external radiative or convective heat loss constant was regarding 3.2 or 7.0 W/m2K once the ORC operated around 100^0C. Radiative and convective heat loss coefficients enhanced because the expander operation temperature enhanced. Conductive heat loss because of the affiliation between the expander and also the support accounted for an oversized proportion of the warmth loss. The fitting relationships between heat loss and mean temperature distinction were established. It's recommended that low conduction material be embodied within the support of expander. Mattress insulation for compact expander may be eliminated once the operation temperature is around 100^0C.

S. Quoilin et al. [27] carried out performance and style optimization of a low value solar organic Rankine cycle for remote power generation. Recent interests in little scale solar thermal combined heat and installation has coincided with the demand growth for distributed electricity providers in space poorly served by centralized power stations. One potential technical approach to fulfill this demand is parabolic trough solar thermal collector including Organic Rankine Cycle engine. He additionally describes the planning of solar organic Rankine cycle being put in in Kingdom of Lesotho for rural electrification purpose. The system carries with it parabolic trough collectors, a tank, a tiny low scale ORC engine scroll expanders. A model of every part is developed that permits sizing the various parts of cycle and evaluates the performance of system. Totally different operating fluids area unit compared, and 2 totally different growth machine configurations area unit simulated (single and double stage).

N.Yamada, M.N.A. Mohamad, T.T. Kien [28] HFO-1234yf could be a new refrigerant with a virtually zero warming potential (GWP) and gas depletion potential (ODP); it exhibits thermodynamical properties just like HFC-134a. The potential if HFO-1234yf as an operating fluid for organic Rankine cycle (ORC) is elucidated through a primary order simulation. A basic thermodynamical model of ORC with 5 forms of cycles trilateral, saturated, superheated, sub-critical, and critical wont to compare the thermal efficiency of HFO-124yf thereupon of different operating fluids. HFO-1234yf was found to supply a thermal efficiency that was like that of HFC-134a. This paper provides a helpful clearly shows the simplest attainable thermal efficiency among the 5 forms of cycle for numerous expander recess and condensation temperatures. the best thermal potency vary (8.8%- 11.4%) was obtained once the critical ORC was used at an expander inlet temperature of 170^0C and condensation temperature vary of 20-40^0C for the given pump and expander potency. it's concluded that HFO-1234yf could be a potential operating fluid for ORC applications, particularly for those with low to medium temperature heat sources.

F.A. Al-Sulaiman, I. Dincer, F. Hamdullahpur [29] This a part of the study presents the thermo economic improvement formulations of 3 new tri-generation systems using organic Rankine Cycle (ORC): SOFC-tri-generation, biomass-tri-generation, and solar-tri-generation systems. A thermo economic modeling is used using the precise exergy costing (SPECO) methodology whereas the improvement performed using the Powell's methodology to reduce the product value of tri-generation (combined cooling, heating, and power). The results facilitate in understanding the way to apply the thermodynamical modeling and thermo economic improvement to a tri-generation system.

F.A. Al-Sulaiman, I. Dincer, F. Hamadullahpur [30] In this half II of the study, 3 new tri-generation systems are examined. These systems are SOFC-tri-generation, biomass-tri-generation, and solar-tri-generation systems. This study reveals that the utmost tri-generation-exergy efficiencies are regarding thirty eighth for the SOFC-tri-generation system, twenty eighth for the biomass-tri-generation system and eighteen for the solar-tri-generation system. Moreover, the most price per exergy unit for the SOFC-tri-generation system is about 38$/GJ, for the biomass-tri-generation system is 26$/GJ and for the solar-tri-generation system is 24 $/GJ. This study reveals that the solar tri-generation system offers the simplest thermo economic performance among the 3 systems. This is often as a result of the solar-tri-generation system has the lowest price per exergy unit. Moreover, the solar tri-generation system has zero carbon dioxide emissions and it's supported a free renewable energy supply.

2.2 Objectives of Present Work:

- The performance of combined ORC cycle is analyzed by varying heater pressure and due to that effect in various parameters like combined cycle efficiency and combined net-work.
- The accomplishment of Internal Heat Exchanger in subcritical Organic Rankine Cycle methods are discovered.
- The result of several operating parameters like high pressure as well as low pressure turbine inlet temperature, heater(recuperator), pressure of cooler, pre-compressor, re-compressor pressure and temperature efficiencies on the outcomes of the cycle.
- A different methodology of calculative pinch point location has been implied.
- It provides helpful understandings for designing and processing of Internal Heat Exchanger to Organic Rankine Cycle.

CHAPTER-3

SYSTEM DESCRIPTION

3.1 Subcritical ORC

The root configuration of ORC system is shown below in Fig.3.1 which consists of a condenser, heater, pump, growth device and a cooling cycle. The heater and condenser are outlined as 3-stage models by Quolin and Wei dynasty. The thermodynamically processes of the subcritical ORC system correlating to the numbers are depicted in Fig 3.1 and also are demonstrated with the help of T-S diagram below in Fig 3.2. The curve that is bridging all the points 5, 9, 10 and 4 is known as saturation curve. The quantities namely, Ti, pi, hi and si indicated the temperature, pressure, specific enthalpy and specific entropy at state point i.

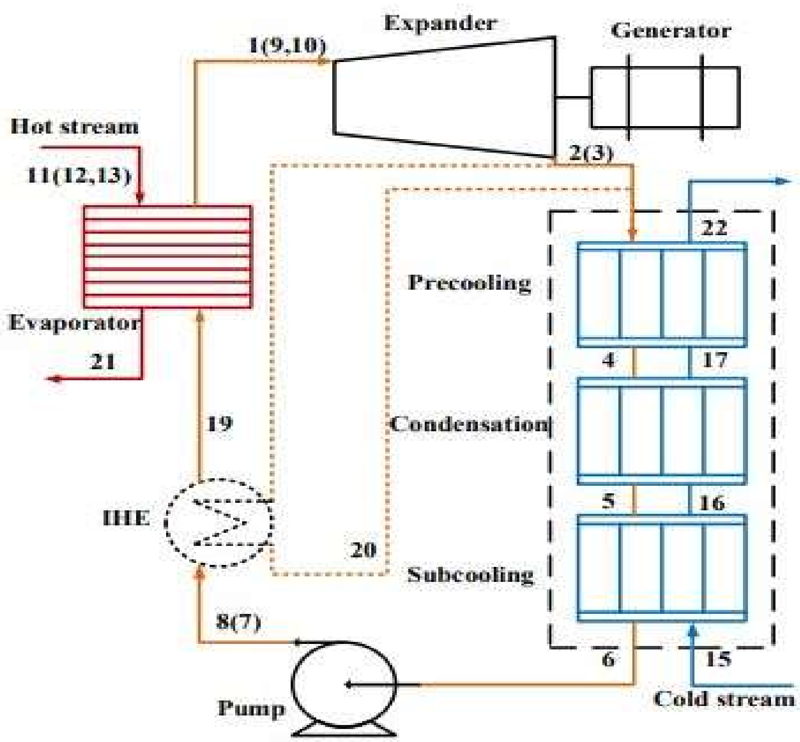

Fig 3.1 Schematical diag. of Subcritical ORC system with IHE

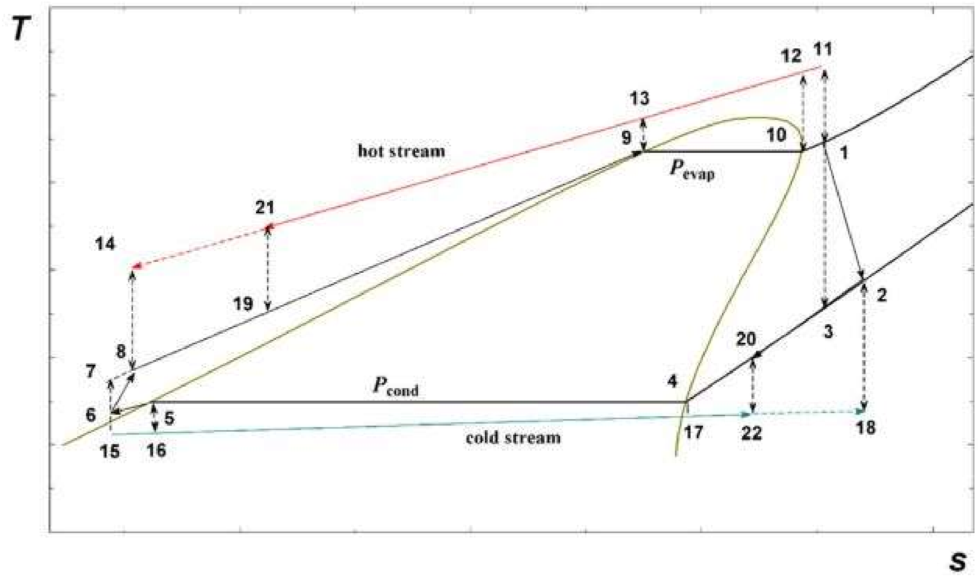

Fig 3.2 T-S diag. of the Subcritical ORC System

At point 1, the working fluid have temperature T1 = Thex + Tsup and the pressure being Pi, is seen as the superheated vapor and from here it enters the expander and subjected to an expansion until it reaches to point 2 where it is having pressure P2 and temperature T2. During the growth, the expander delivers the work Wt. It is now at point 2, where the operating fluid goes into the condenser. During this time, it releases the specific heat Qcond to the cold steam and returns to the sub-cooled state point 6 at constant pressure. At the state point 6 in cycle, the operating fluid is that the sub-cooled liquid with temperature T6 = Tcond – Tsub and the correlating vapor pressure is P6 = P2, both the temperature being the lowest ones within the Organic Rankine Cycle. Then the fluid undergoes adiabatic compression and reaches the temperature which is the highest temperature within the Organic Rankine Cycle at state point 1. Processes 1-2 and 6-8 are the real case studies whereas the processes 1-3 and 6-7 are ideal ones in which s1 = s3, s6 = s7.

When IHE is projected, the liquid operating fluid before getting into the evaporator exchanges heat with the gas operating fluid coming out of expander within the IHE. The liquid operating fluid gets heated from T8 to T19 whereas the gas operating fluid cooled from T2 to T20.

As it is extremely well known that if we know any 2 parameters out of the 4 parameters Ti, Pi, hi, si of the state point then the other 2 parameters can be determined. And for doing these thermodynamically calculations, Engineering Equation Solver Ver. 8 are employed. This software system is straightforward and versatile to make use of identical equation of state as utilized in REFPROP-NIST. This is a flexible software and an awfully useful tool.

3.2 Supercritical ORC

In a critical Organic Rankine Cycle, method that happens in heater is a complex 2 phase process, merely implying that a 3 stage model can't be used. However the other facilities are same with nearly no variations among them. The schematic diagram is analogous to the diagram shown in Fig 3.3 with the sole distinction that a heater is replaced by a 1 stage model. Fig 3.3 shows the supercritical system with IHE and thermodynamically processes over T-S diagram are shown below in Fig 3.4.

Due to complexness of 2 phase changes, there occurs an issue in locating the position of pinch point. During a subcritical cycle, by default the pinch point is assumed in the middle of point 9 and 13 and this assumption in most cases is reasonable. In other way, the temperature difference between T11 and T1 is too high. However, during a critical cycle, shifts of operating fluid evaporation progress and constant pressure lines of the heat supply are similar; therefore it becomes tough to spot the pinch point position qualitatively.

Some of the preceding studies showed their work by taking the outlet temperature of the heat source or by assuming the site of pinch point, or by setting and defining effectiveness of internal heat exchanger at a constant value of €. This simplified the overall calculation whereas it'd conjointly led to the result that the temperature difference of the pinch point would be less or quite the value that is already set, that isn't in conformity with the important condition in heat exchanger. The given minimum heat transfer temperature difference ΔTpp is used for calculative the effectiveness of Internal heat exchanger i.e. €, but the derivation of reversing process isn't real. Here, an iteration technique is recommended through that the pinch point position within the critical cycle will be located. Once ΔTe is set, the operating fluid's evaporation zone is split equally to N number of parts within the direction of entropy.

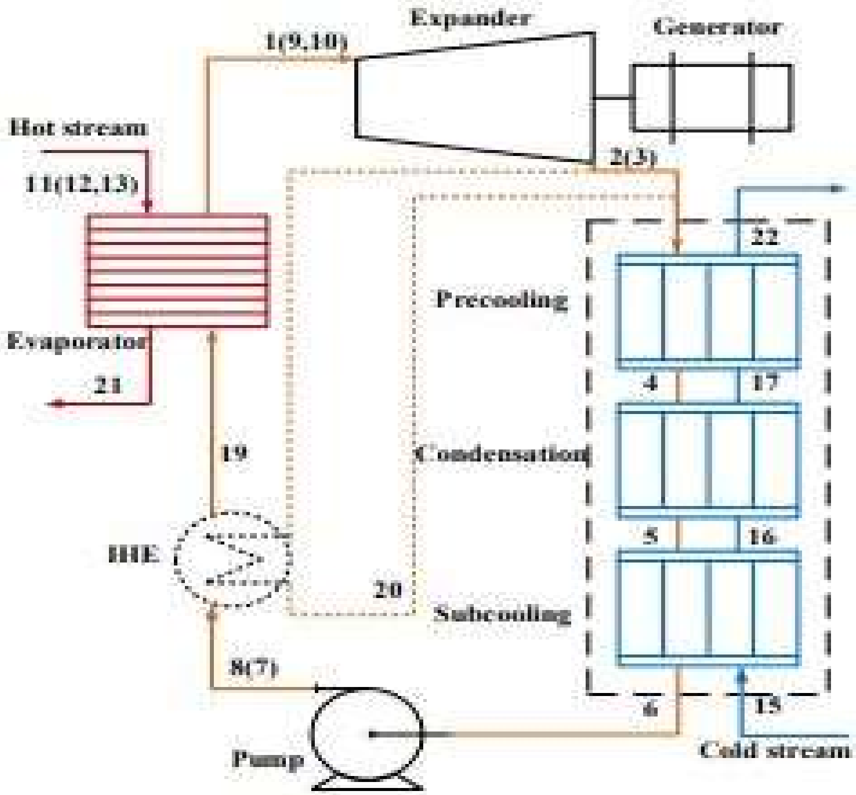

Fig 3.3 Schematic diag. of Supercritical ORC System with IHE

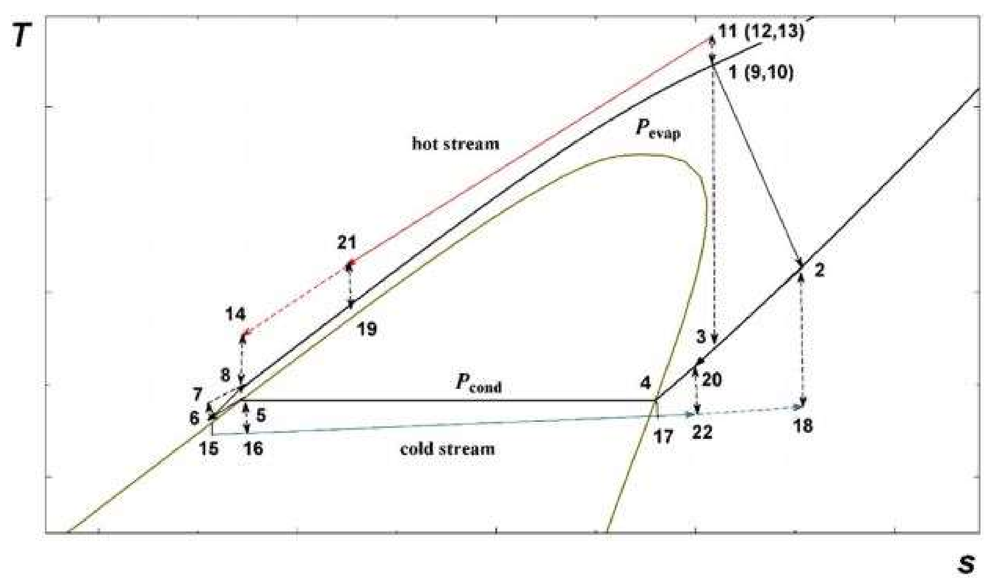

Fig 3.4 T-S diag. of Supercritical ORC

The lowering of the outlet temperature of warmth supply is employed to seek out the minimum gap between operating fluid evaporation isobar and heat source isobar. Once this equals ΔT_{pp}, the outlet temperature will be determined and the position of pinch point is N. Fig. 3.5 demonstrates this methodology and Fig. 3.6 is the flow diagram that is written as per this methodology.

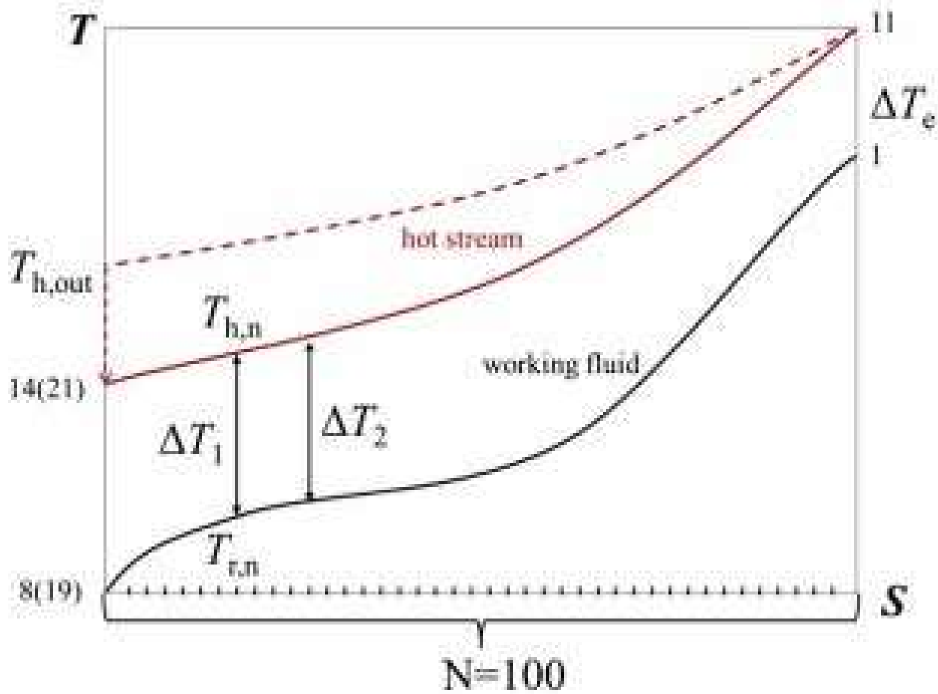

Fig 3.5 Schematic diag. of Iteration Method for Supercritical ORC Pinch point locating

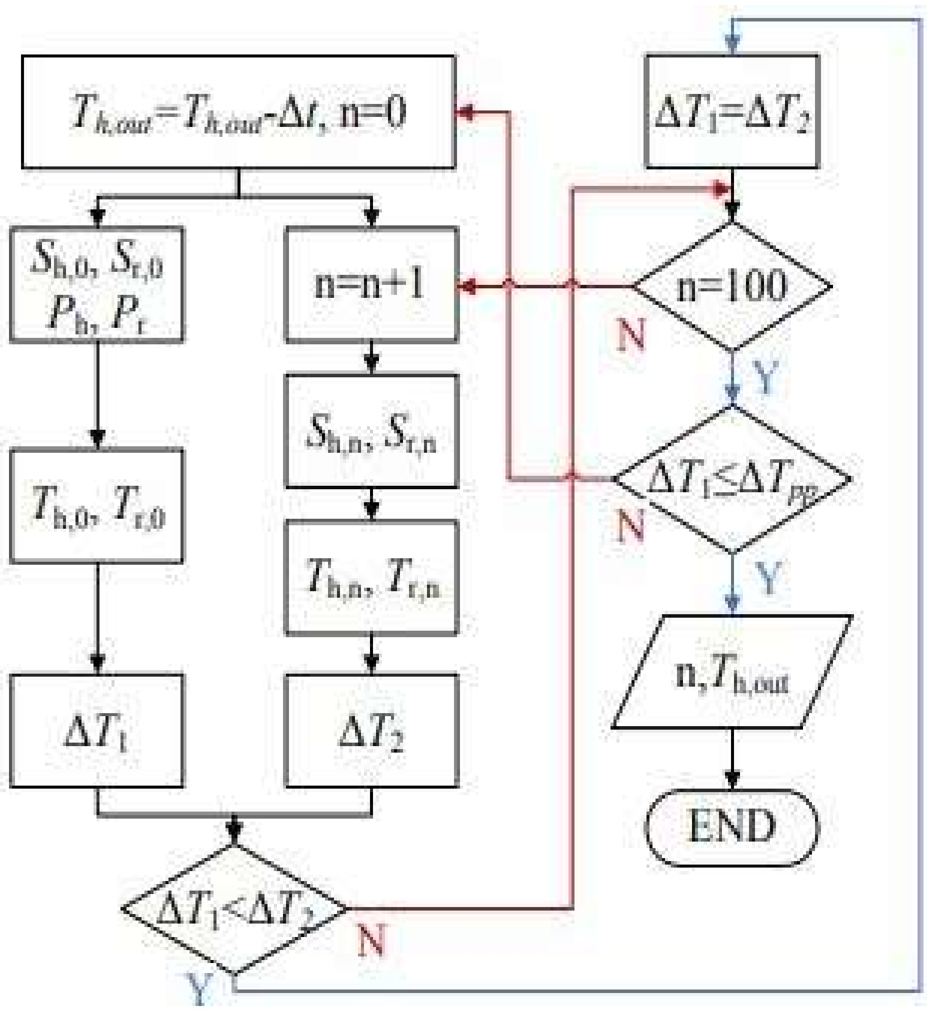

Fig 3.6 Flow chart of the Pinch point and Hot stream Outlet temperature Calculation program

3.3 Simple Brayton Cycle:

The simple Brayton cycle can be shown in the Fig.3.7. High temperature gas is supplied to the turbine where the gas expands and gives the shaft work. Then it is send to cooler where it releases the heat to the sink and after that the gas was compressed to highest pressure and temperature in compressor and then it is send to the combustion chamber where heat is supplied.

The cycle efficiency can be increased by doing the compression in two stages such as using intercooling or by doing the expansion process in two stages i.e. reheating. So it is beneficial using intercooling and reheating.

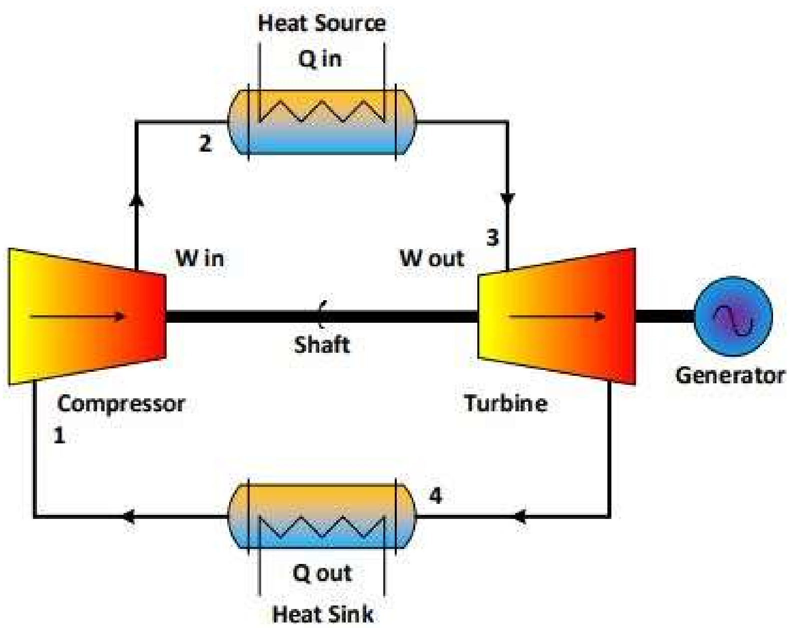

Fig. 3.7 Schematic Diagram of Simple Brayton cycle

Processes:

1-2: Isentropic compression of the compressor

2-3: Constant pressure heat inclusion in the combustion chamber

3-4: Isentropic Expansion of the turbine

4-1: Constant pressure heat rejection of the cooler.

For thermodynamic analysis of Brayton cycle the following assumptions are taken:

- The turbine's isentropic efficiency is taken to be 90% and pump's isentropic efficiency is taken to be 89%.
- The cycle is working at steady state conditions.
- Dropping of pressure in the heat exchangers can neglected.
- Minimum temperature of cycle is set as 25ºC.

3.4 Brayton Cycle with Intercooling:

In this cycle multi compression is done in intercooler. The low-pressure stream is firstly made to enter a heat exchanger (pre-cooler) and was allowed to cool thereafter. The cooled flow was then made to go along the pre-compressor, from where it is made to compress to a halfway pressure. Then, the fluid enters the intercooler and is cooled down there again before entering the main compressor. Fig.3.8 shows the layout of Brayton cycle with Intercooling.

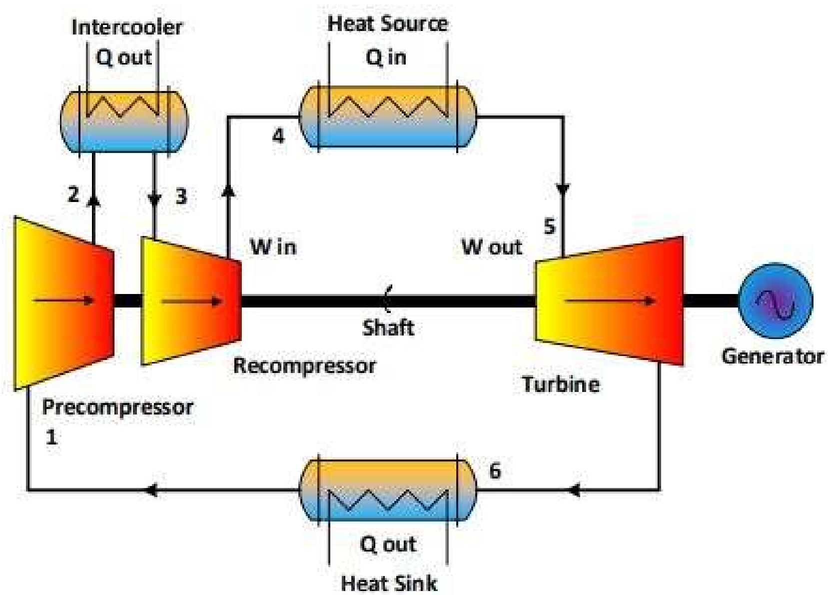

Fig. 3.8 Brayton cycle with intercooling

The inlet temperature in the compressors does not need to be equal, but since only 1 cold sink is need to be used, the temperature should be equal. Hence, the compressor inlet temperature is alike in this above study.

3.5 Brayton cycle with Reheating

The third cycle is with reheating. Fig. 3.9 shows the layout of Brayton cycle with reheating. It is similar to simple Brayton cycle i.e. after compression process the gas goes to combustion chamber and then to the turbine for expansion, the only difference is that the expansion process is done in multiple stages i.e. first the gas will expand in High pressure turbine and after reheating it expands in low pressure turbine.

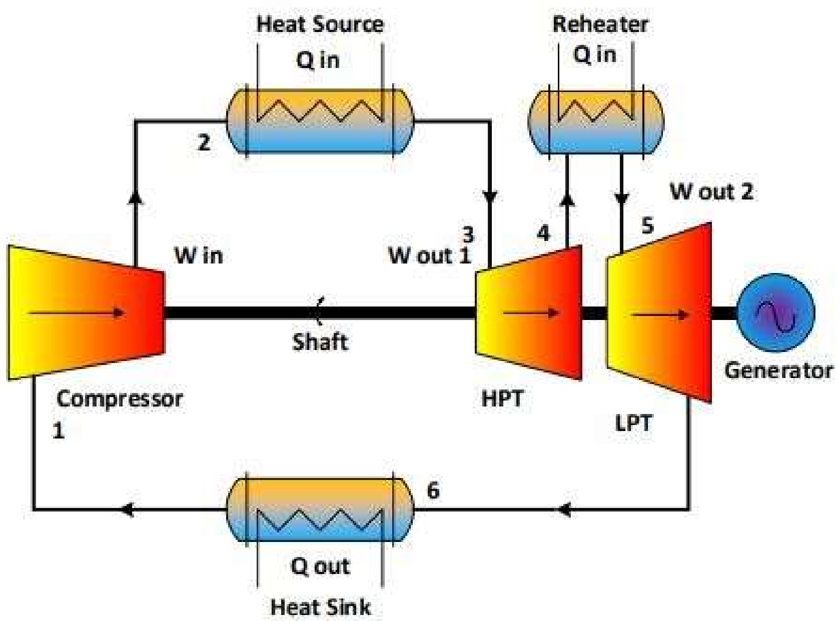

Fig. 3.9 Layout of Brayton cycle with Reheating

The reheating resulted in the improvement of the cycle efficiency by raising the Carnot temperature of the cycle. The fluid at high pressure leaves the compressor and passes into the gas heater or heat exchanger as the cold stream (2). The stream at high temperature and pressure pass on to the HP turbine (3) and faces energy loss in the HP turbine door (4). Low- pressure stream has gained energy in the re-heater and made to exit through the LP turbine (5). It is possible to initiate more than one reheat stage where two reheat stages being used. In the case of 3 stages of reheat, there occurs an inclusion of another turbine body into the working system. At the end, the heat is rejected in the pre-cooler (6), where the working fluid is allowed to cool to the temperature according to the temperature of the compressor inlet.

3.6 Brayton cycle with Intercooling and Reheating:

In this given cycle, there is multistage compression and expansion. The stream, which exits from pre-cooler, enters into the pre-compressor (1) see Fig.3.10. After the stage of compression, the fluid passes on to an intercooler to dissipate heat (2). After the intercooler, stream enters to the main compressor (re-compressor) where its pressure and temperature increased (3).

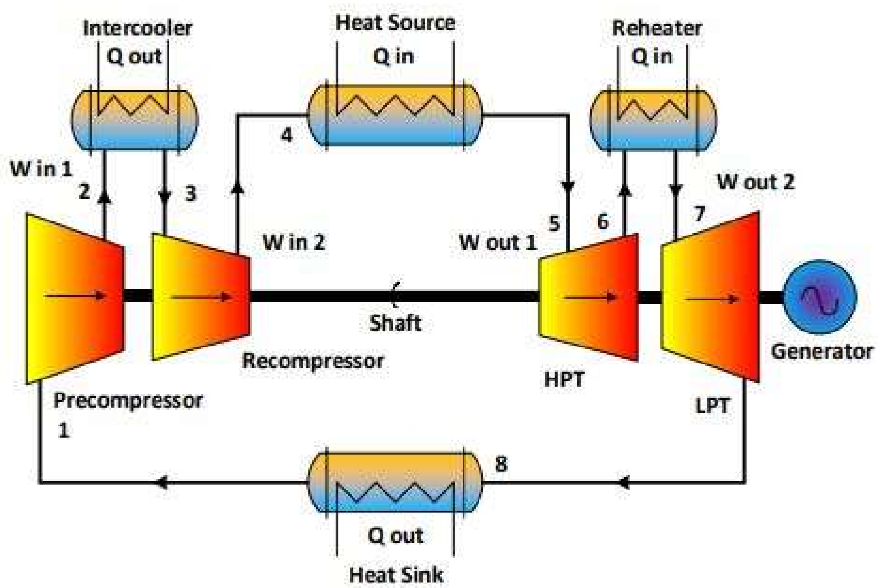

Fig. 3.10 Layout of Brayton cycle with Intercooler and Re-heater

After this procedure, stream absorbs the heat from the heat exchanger (gas heater) (4 to 5) and made to move to HP turbine. The outlet fluid from HP turbine is hot in the re-heater at constant pressure (6). The low-pressure hot fluid (7) moves the LP turbine and allowed to expand there (8).After the generation of electricity, the fluid was made to go through heat sink where the remaining heat is dissipated and thereafter enters to a pre-compressor and the complete process starts all over again.

3.7 Combined cycle

The combined cycle can be shown in the Fig.3.11. The heat rejected by the heat exchange or cooler of gas turbine is supplied to the heat exchanger or heater of the steam turbine cycle. Rest all the processes are same. First the stream is compressed in the compressor where its temperature and pressure is increased then it is send to the combustion chamber where it absorbs heat from the source and the hot temperature stream is directed to the turbine for further expansion. After this turbine process heat is rejected in the heat exchanger or cooler and the stream goes back to the compressor.

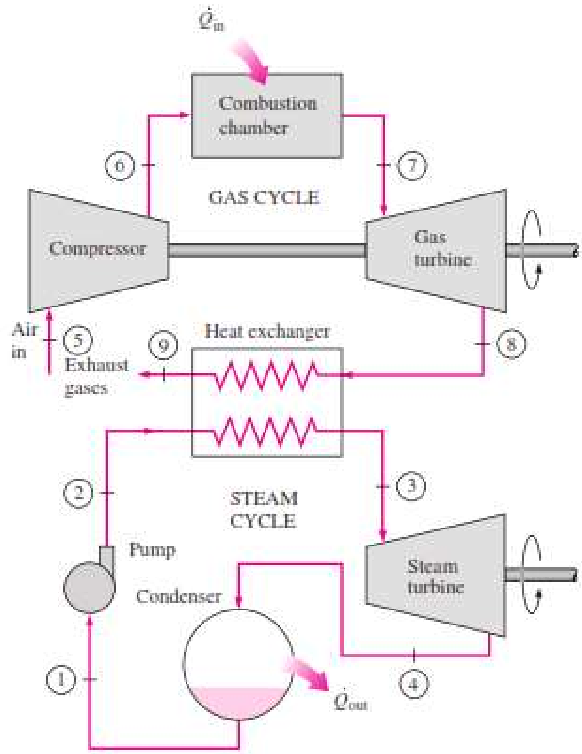

Fig. 3.11 Layout of the combined cycle

The heat which is rejected by the heat exchanger or cooler of the gas turbine is send to the heater of the Rankine cycle and hat is allowed to expand in the turbine from where it gives the work which is required. After that the stream goes to the condenser where it rejects heat to the sink and condenses and becomes saturated liquid and this saturated liquid goes to the pump of the Rankine cycle which icreases pressure the fluid and cycle goes on.

Here in this study the fluid which circulates in the cycle is R123 and R245fa.

3.8 Global Model

The system safety, economic potency of ORC system and environment protection is greatly influenced by the selection of operating fluids. The wide used fluids includes the applicable fluids with sensible thermodynamics, economic features, environmental protection and safety issues like fluids having low liquid specific heat, viscous medium, toxicity, flammability, gas depletion potential, issues regarding global warming, and price.

Keeping in mind the above things, the operating fluid R123 and R245fa are chosen for subcritical cases. The basic environmental and thermodynamically properties of the mentioned fluids are shown in Table 3.1.

Working Fluid	T_{cr} (ºC)	P_{cr} (MPa)	T_{fp} (ºC)	T_{bp} (ºC)	Safety	ODP	GWP
R123	183.7	3.668	-107.2	27379	B1	0.012	120
R245fa	154.1	3.651	-106.7	14.9	A1	0	1030

Table 3.1 Basic thermodynamic and environmental properties of R123 and R245fa

The following assumptions are created within the analysis so as to modify the computing and complicacy that are neither the key points not they have an effect on the accuracy of results coming back within the model advised.

- The entire system is considered to be in equilibrium state.
- The specific heat of the new stream is replace with the one amongst hot air.
- The loss of heat and small leakages through the pipeline and elements are neglected.

Some unaltered parameters concerned within the calculations are shown in Table 3.2.

Parameter	Sign	Unit	Value
Environment temperature	T_0	⁰C	15
Low Grade Heat flow temperature	$T_{h,in}$	⁰C	200
Cooling water temperature	$T_{c,in}$	⁰C	15
Condensation Temperature	T_{cond}	⁰C	30
Low Grade Heat Mass Flow Rate	$q_{v,h}$	Kg/s	1
Superheat Degree	T_{sup}	⁰C	3
Degree of Subcooling	T_{sub}	⁰C	1
Pinch Point	ΔT_{pp}	⁰C	10
Temperature difference at hot stream inlet of Evaporator	ΔT_e	⁰C	15
Adiabatic efficiency of Expander	$\eta_{T,i}$	1	0.65
Mechanical Efficiency of Expander	$\eta_{T,m}$	1	0.9
Working fluid pump Adiabatic efficiency	η_{is}	1	0.8
Working fluid pump Mechanical efficiency	η_{Pm}	1	0.9
Total circulation loop Flow resistance	dP_{hf}	kPa	600
Total resistance at condenser cooling water side	dP_{cf}	kPa	300

Table 3.2 Assumptions of Streams and Components

CHAPTER-4

THERMODYNAMIC MODELLING

This chapter deals with the set of equations in which various measures are used for the analysis to be performed. They are introduced to show the performance of the system.

4.1 Energy Analysis

The Ist law of thermodynamics is the basis for the energy analysis. The final results includes the net-work output and the efficiency of the cycle. Their value is dependent only on heater pressure $P_{hex} = P_1 = P_8$ as per assumption mentioned before. Following are the set of equations for different components.

4.1.1 Subcritical ORC

The equation for heater/heat exchanger:

$$Q_{hex} = \dot{m}_{hf}(h_{11} - h_{14}) = \dot{m}_{cf}(h_1 - h_8) \quad (1)$$

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 $\eta_{T,m}$is 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:

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

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

$Q_{hex} = \dot{m}_{hf}*(h_{11} – h_{21}) = \dot{m}_r*(h_1 – h_{19})$ (8)

$Q_{cond} = \dot{m}_{cf}*(h_{22} – h_{15}) = \dot{m}_r*(h_{20} – h_6)$ (9)

For the Internal Heat Exchanger:

$h_{20} – h_2 = h_8 – h_{19}$ (10)

4.1.2 Subcritical ORC Using R245fa

The equation for heater/heat exchanger:

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

The equation of the expander

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

$W_T = \eta_{T,m}*(h_1 – h_2)$ (13)

Where $\eta_{T,m}$is 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)

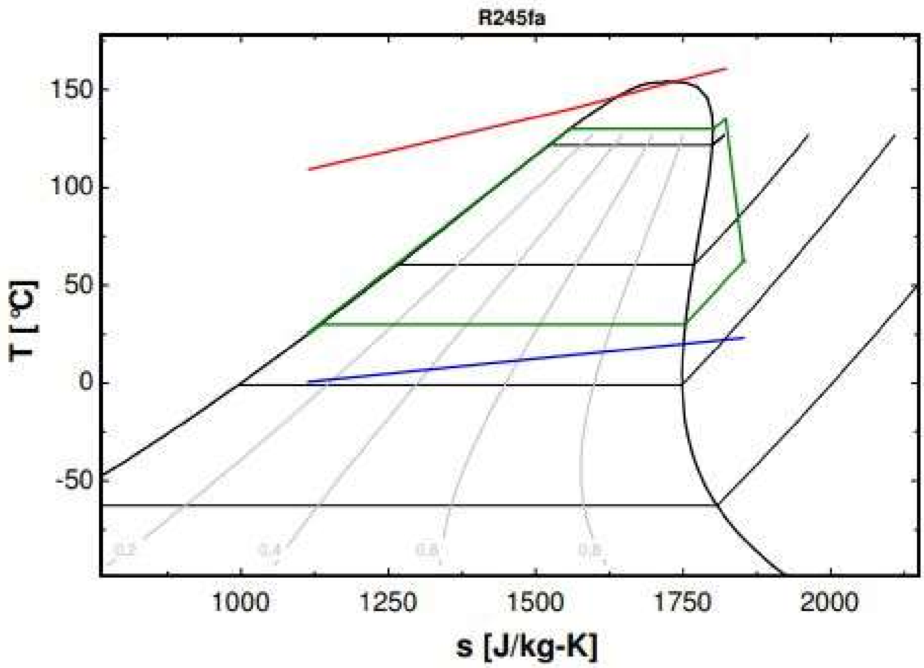

Fig. 4.1 T-S diag. of Subcritical ORC system using R245fa

4.1.3 Simple Brayton Cycle

The energy equation of simple Brayton cycle are as follows:

For the compressor:

$W_C = h_2 - h_1$ (21)

W_c, h_2 and h_1 stands for compressor work, exit enthalpy and inlet enthalpy of the compressor respectively.

For the Combustion Chamber:

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

Q_{in}, h_3 and h_2 stands for heat input, exit enthalpy and inlet enthalpy of the combustion chamber respectively.

For the Turbine:

$W_T = h_3 - h_4$ (23)

W_T, h_3 and h_4 stands for the turbine work, inlet enthalpy and exit enthalpy of the turbine respectively.

For the cooler/ heat exchanger:

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

Q_{out}, h_4 and h_1 stands for heat output, inlet enthalpy and exit enthalpy in the cooler or heat exchanger.

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 h4,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 $h_{4,s}$ is isentropic enthalpy of the turbine.

Cycle Efficiency of the Brayton cycle:

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

4.1.4 Combined Cycle

$W_{net,combined} = W_{net,Brayton} + W_{net,Rankine}$ (31)

$eta_{combined} = W_{net,combined} / (Q_{in,hex,Rankine} + Q_{in,combustor})$ (32)

4.2 Exergy Analysis

The downside of energy examination is that the transformation of energy of heater and condenser were not effective. Exergy examination is an energy transformation coefficient that focuses on quality of energy, and this parameter deeply tells the principle of losses and energy transformation as compared to energy examination. The exergy destruction in each component can also be calculated by this analysis.

The exergy examination is done by computing specific flow exergy for 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 as 25ºC and atmospheric temperature. Following are the equations for various constituents:

Equation for Heater:

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

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:

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

The overall exergy efficiency is calculated as:

$$\eta_{ex} = 1 - \{E_{d,Tot} / \dot{m}_{hf} *(e_{11} - e_{14})\} \tag{39}$$

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

$$E_{d,hex} = \dot{m}_r*(e_{19} – e_1) - \dot{m}_{hf}* (e_{21} – e_{11}) \tag{40}$$

$$E_{d,cond} = \dot{m}_r*(e_{20} – e_6) - \dot{m}_{cf}*(e_{22} – e_{15}) \tag{41}$$

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:

$$E_{d,Tot} = E_{d,hex} + E_{d,P} + E_{d,cond} + E_{d,IHE} + E_{d,T} \tag{43}$$

$$\eta_{ex}= 1 – \{E_{d,Tot} / \dot{m}_{hf}*(e_{11} – e_{21})\} \tag{44}$$

CHAPTER- 5

RESULTS AND DISCUSSIONS

5.1 Subcritical ORC

5.1.1 Effect of cycle efficiency

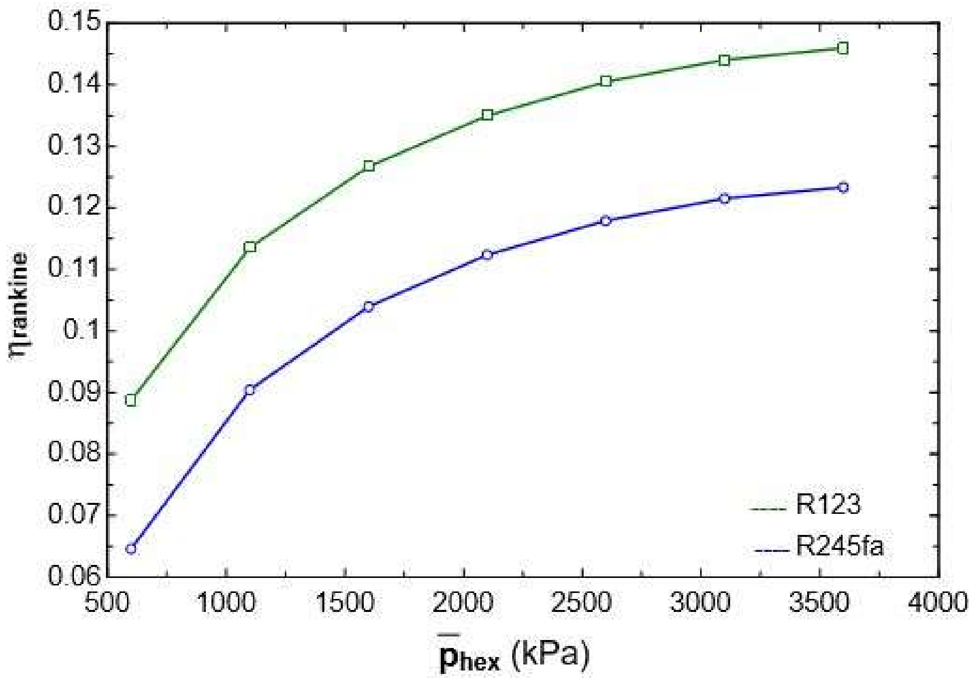

Fig. 5.1 heater pressure Vs cycle efficiency

Figure 5.1 shows the outcome of cycle efficiency by varying the heater pressure. As per the figure 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. 5.1 that R123 is having more efficiency than R245fa at a same particular heater pressure. From fig. 5.1 the Rankine cycle efficiency at heater pressure 1326 kPa is 12.04% for fluid R123 Rankine cycle efficiency at heater pressure 1326 kPa is 9.08% for fluid R245fa.

So, from the result it can be seen R123 is having more efficiency than R245fa but the amount of chlorine content in R123 makes it unstable as it depletes the ozone layer and is not an ecofriendly fluid.

5.1.2 Effect on exergy flow rate

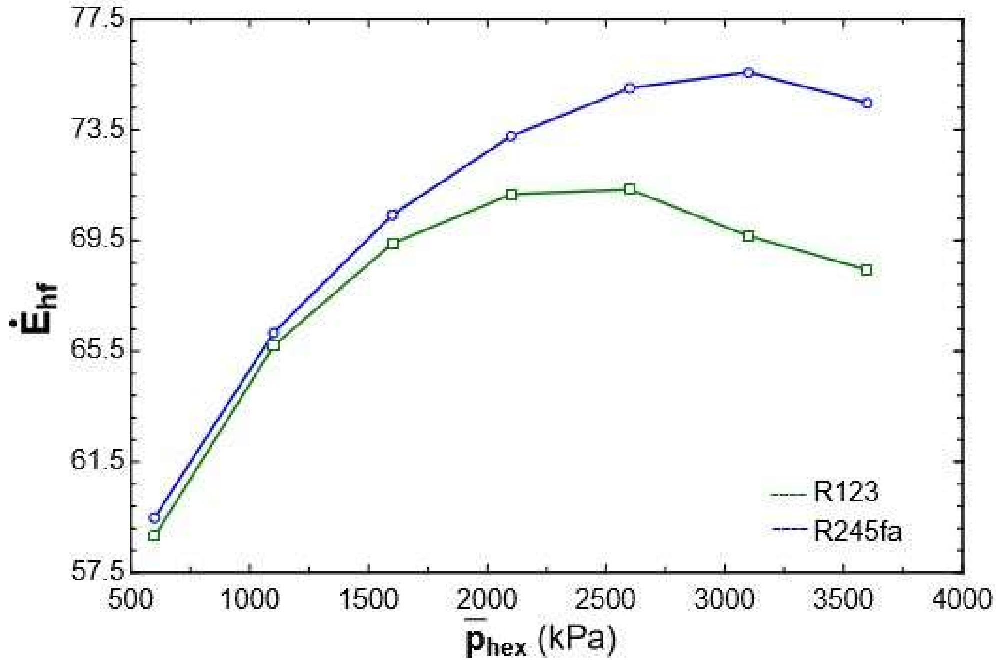

Fig 5.2 heater pressure Vs Exergy flow rate of hot fluid

Fig 5.2 shows the change of exergy flow rate for hot fluid by varying heater pressure. As per the figure the exergy rate flow of hot fluid is 67.75 kW at heater pressure of 1326kPa for the fluid R123 and for fluid R245fa the exergy flow rate is 73.39 kW at heater pressure of 2127 kPa.

5.1.3 Effect on heater temperature

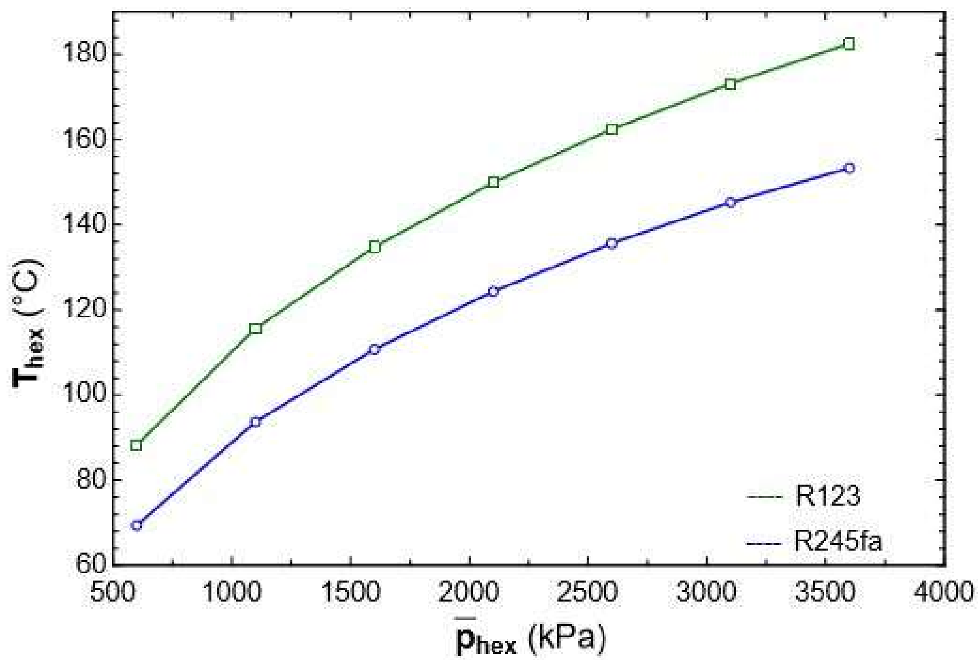

Fig 5.3 heater pressure Vs heater temperature

5.1.4 Effect of net-work

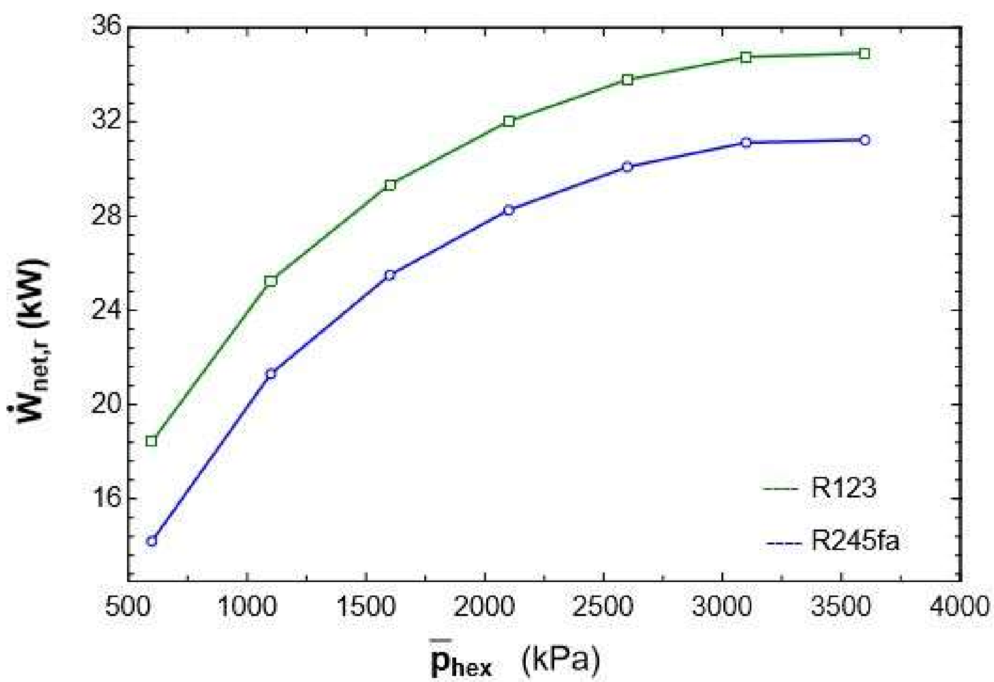

Fig 5.4 heater pressure Vs Net-work output

5.1.5 Effect of pinch point

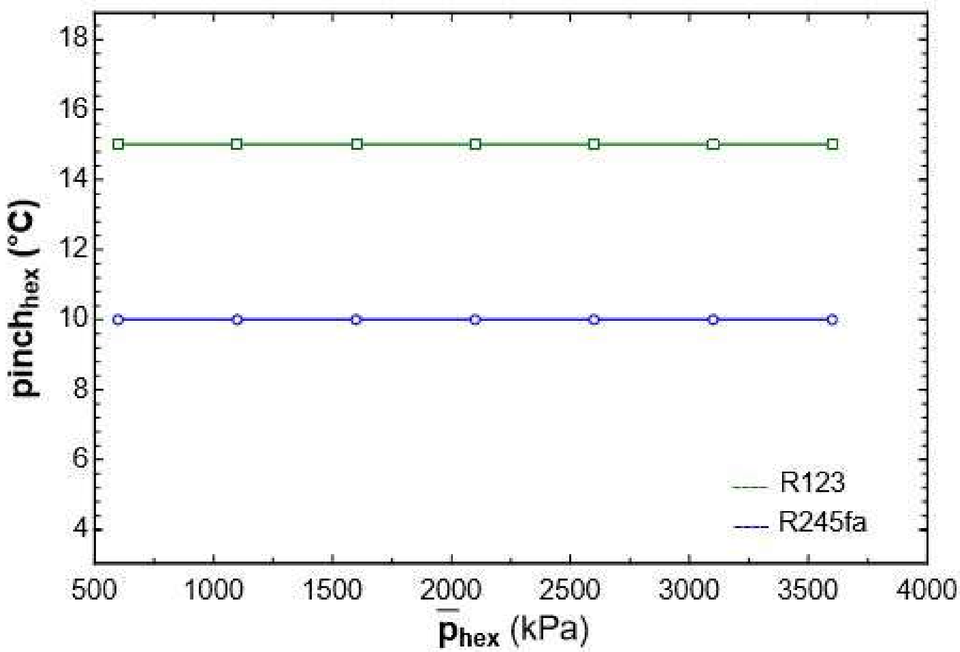

Fig 5.5 heater pressure Vs pinch point

5.1.6 Effect of pump work

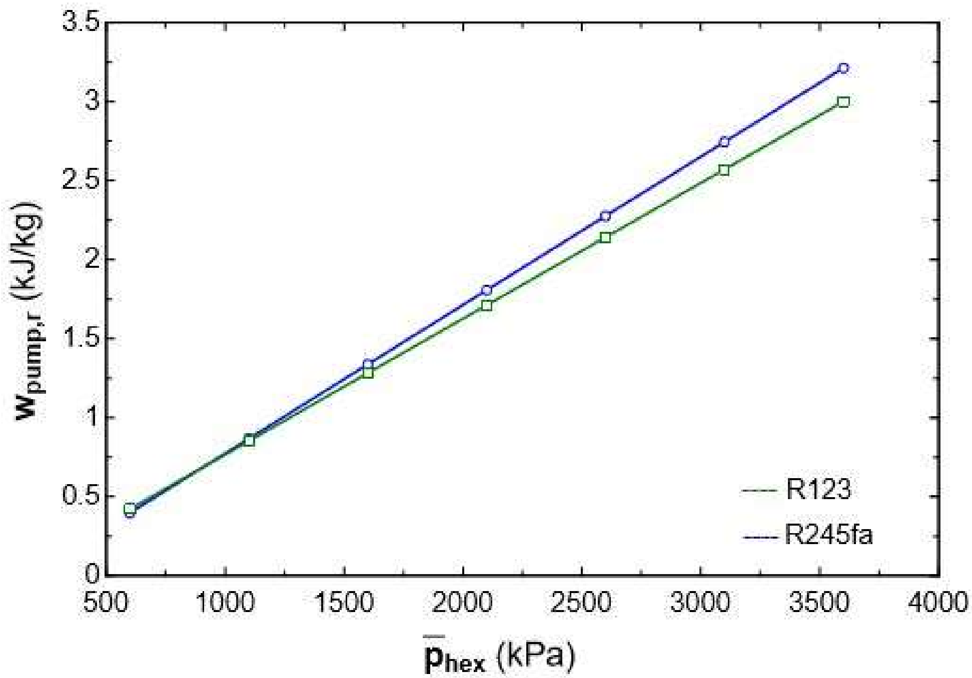

Fig 5.6 heater pressure Vs pump work

5.1.7 Effect of turbine work

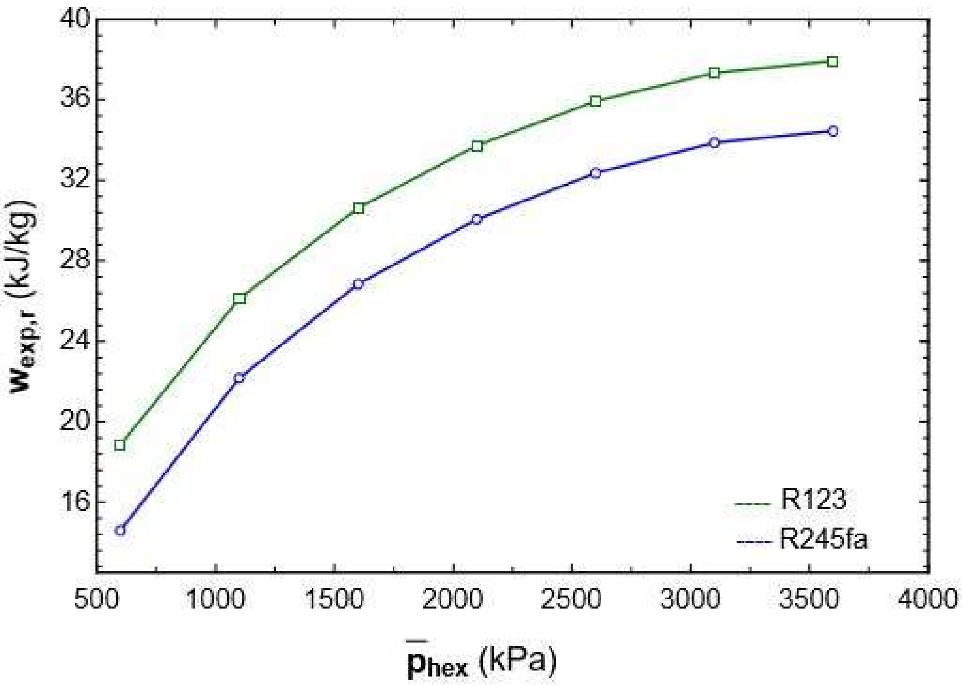

Fig 5.7 heater pressure Vs turbine work

5.1.8 Effect of second law efficiency

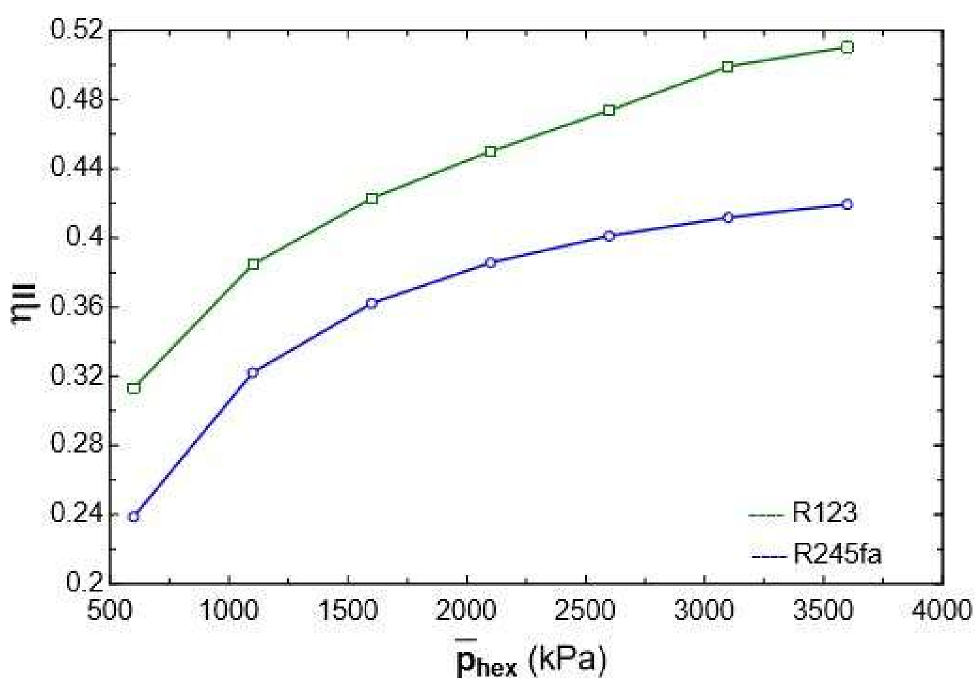

Fig 5.8 heater pressure Vs second law efficiency

Fig 5.3 shows the change on heater temperature by varying the heater pressure. From Figure 5.3 the heater temperature is 125ºC at heater pressure 1326 Kpa.

Fig 5.4 shows the graph between heater pressure and net-work output. As per figure 5.4 the total 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 the total work output is more for R123 as compared to that of R245fa.

Fig 5.6 shows the variation in pump work by varying the heater pressure. As per Fig. 5.6 the 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 5.7 indicates the graph between heater pressure and turbine work. From the Fig 5.7 the 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.

Fig 5.8 shows the graph between second law efficiency and heater pressure. As per figure 5.8 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.

5.2 Simple Brayton Cycle

5.2.1 Effect of compressor inlet pressure

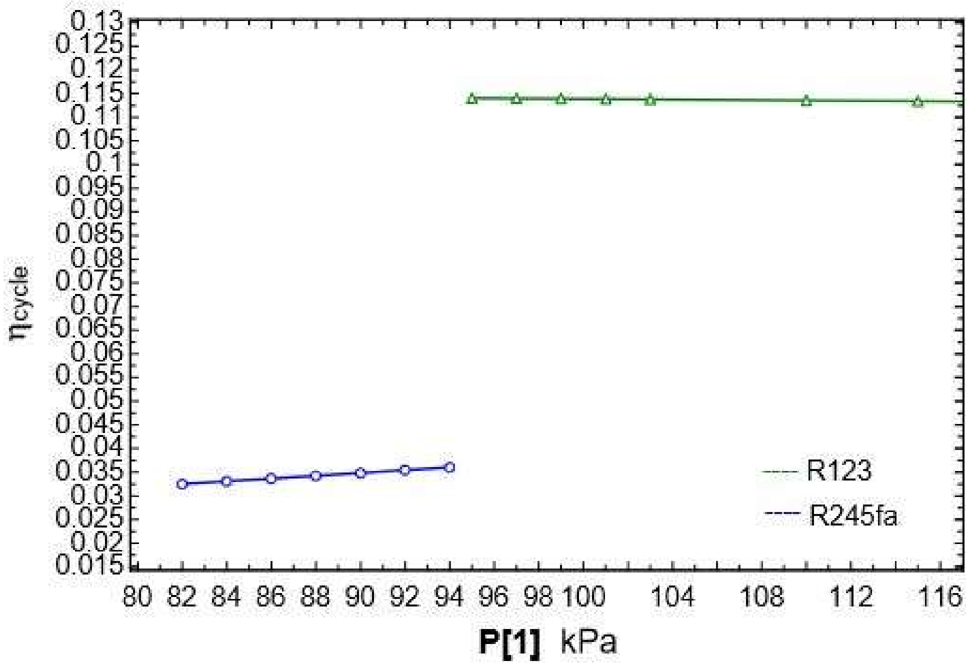

Fig 5.9 Compressor inlet pressure Vs cycle efficiency

5.2.2 Effect of pressure ratio

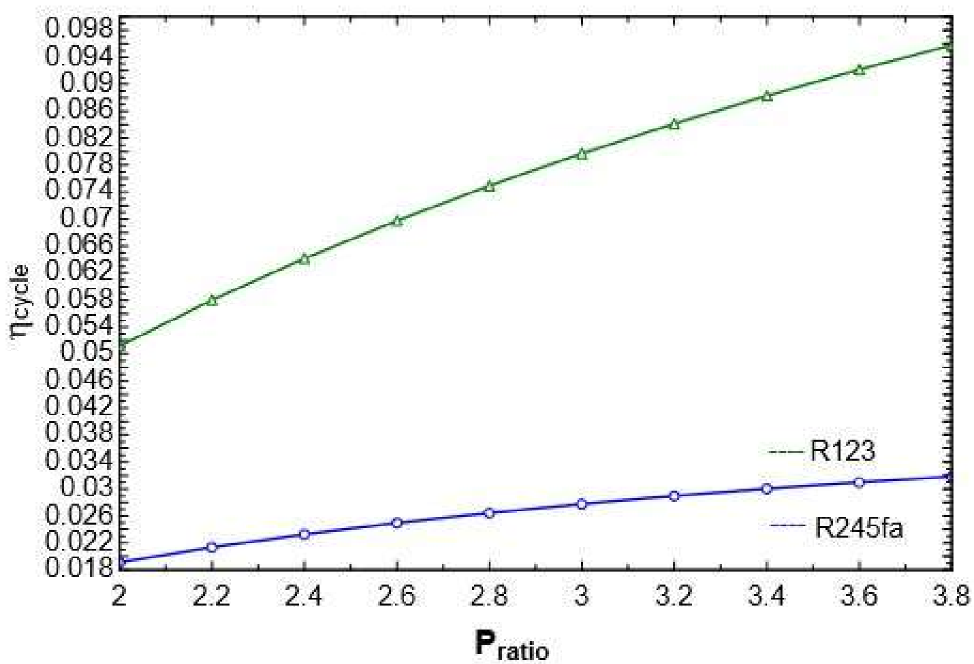

Fig 5.10 Pressure ratio Vs cycle efficiency

5.2.3 Effect of minimum operation Temperature

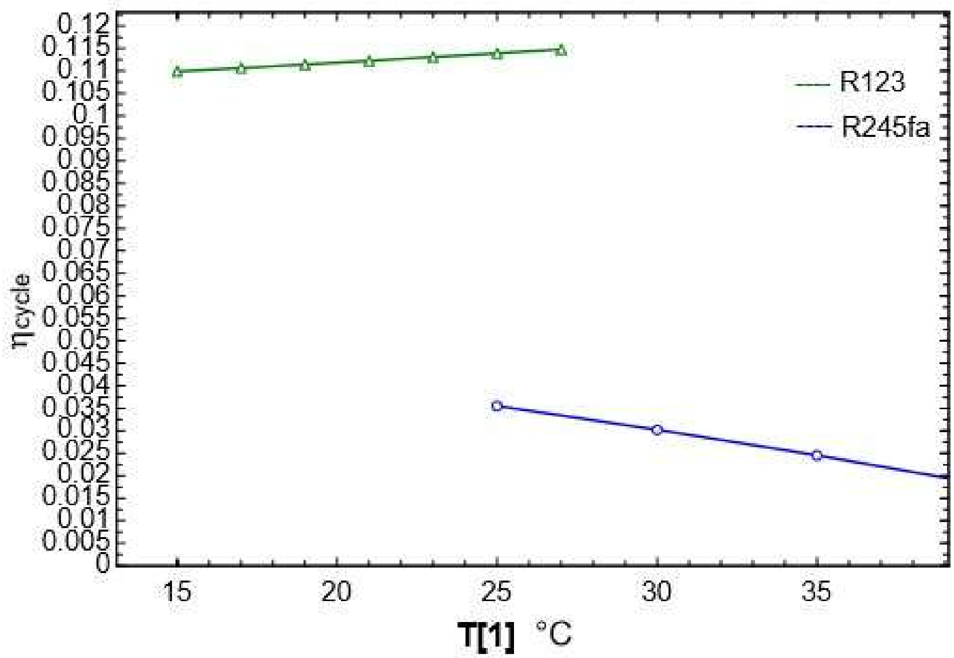

Fig 5.11 Minimum Operation temperature Vs cycle efficiency

5.2.4 Output of pressure ratio on the compressor and turbine work

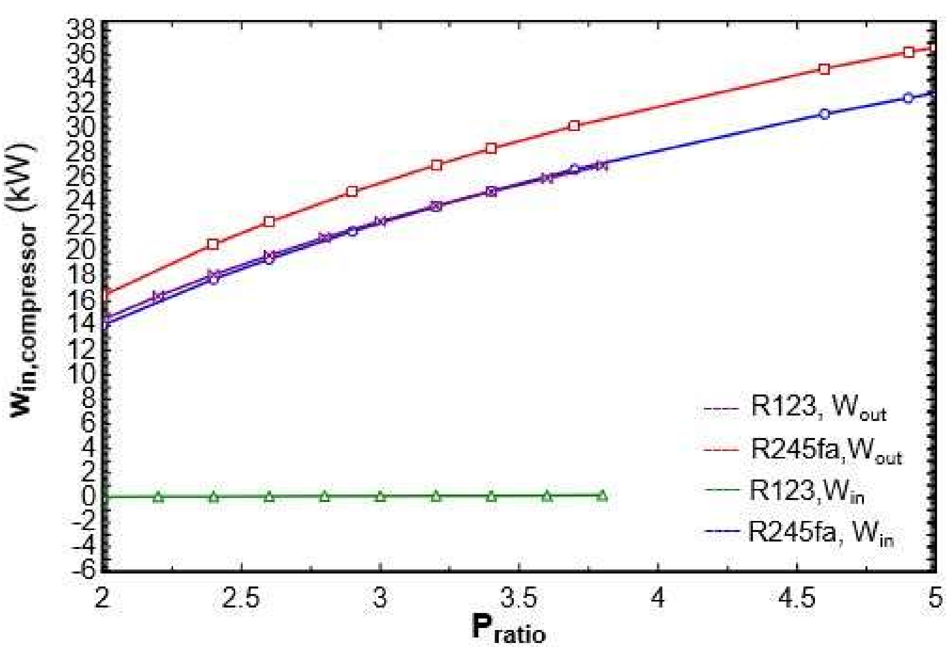

Fig 5.12 Pressure ratio Vs compressor and turbine work

Fig 5.9 indicates the outcome of pressure at inlet of compressor on the efficiency in Brayton cycle. The examination done on two fluids are R123 and R245fa. It can be seen from the Fig 5.9 that there is increase in the cycle efficiency due to the increase of inlet temperature (T3) at certain value of P1.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. From Figure 5.9 the efficiency of cycle is 3.601% at 94 kPa compressor inlet pressure for fluid R123 and 11.41% at 95kPa compressor inlet pressure for fluid R245fa.

Fig. 5.10 shows the variation between pressure ratio and cycle efficiency. It can be seen from the Fig. 5.10 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.

Fig 5.11 shows the graph between minimum operation temperature and cycle efficiency is 3.548% at T1 = 25ºC for fluid 245fa and 11.39% for fluid R123.

Fig 5.12 shows the variation between pressure ratio and compressor & turbine work. It can be seen from the Fig. 5.12 that for fluid R123 the work input and work output is 0.1691 kW and 23.67 kW respectively are 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.

5.3 Brayton cycle with Intercooling

5.3.1 Effect of cooler pressure

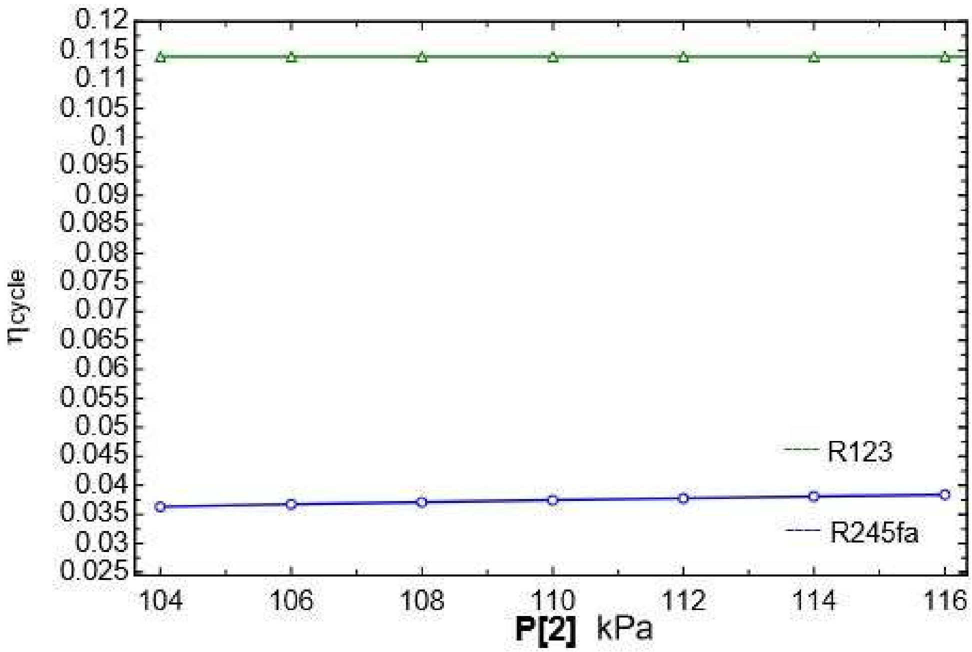

Fig 5.13 Cooler pressure Vs cycle efficiency

5.3.2 Effect of High pressure Inlet turbine temperature

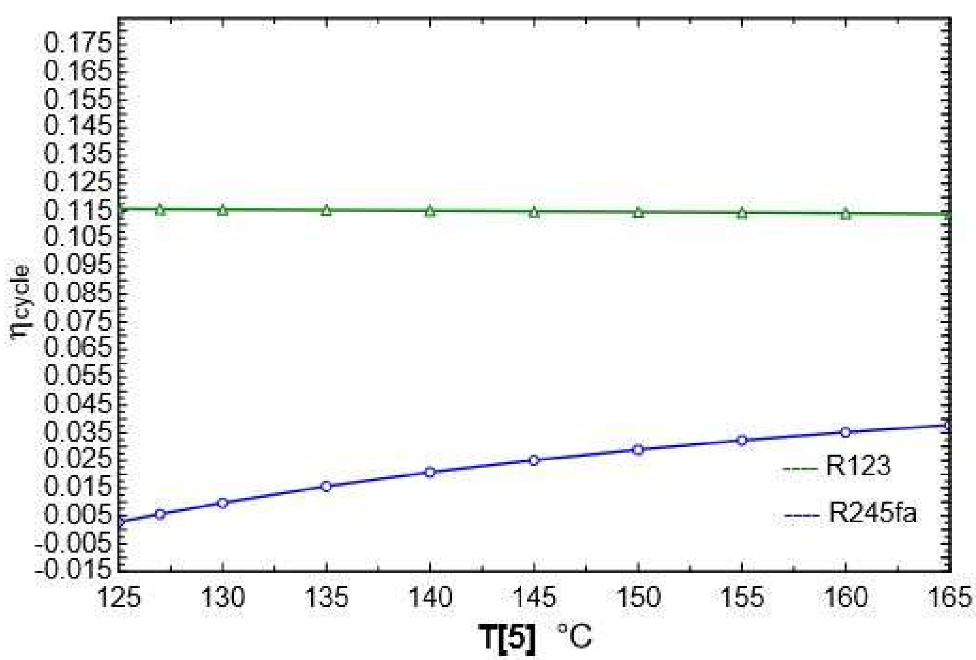

Fig 5.14 High pressure inlet turbine temperature Vs cycle efficiency

5.3.3 Outcome of cooler pressure on cycle work

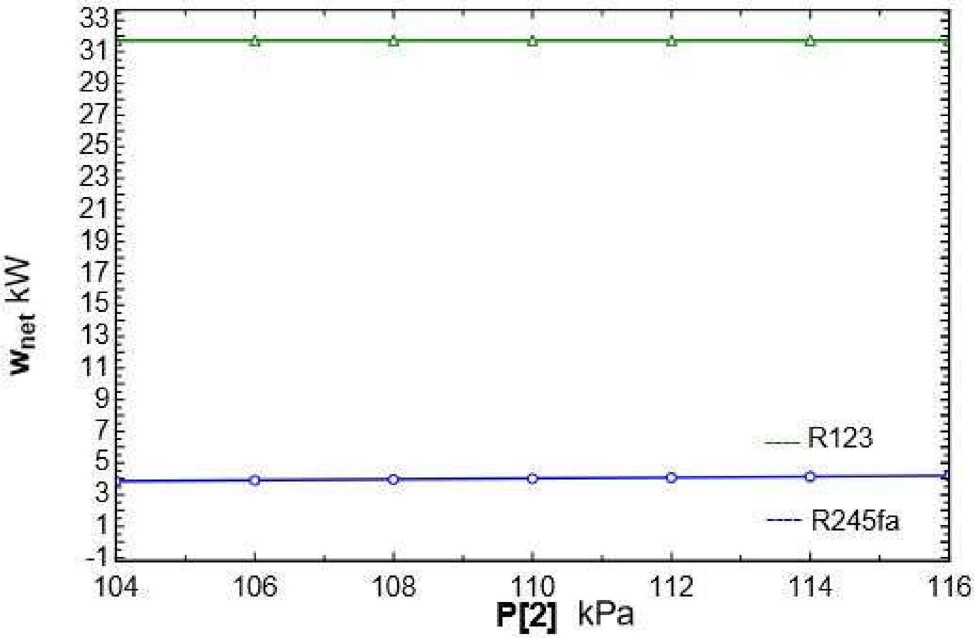

Fig 5.15 Cooler pressure Vs Net-work

5.3.4 The outcome of pressure ratio

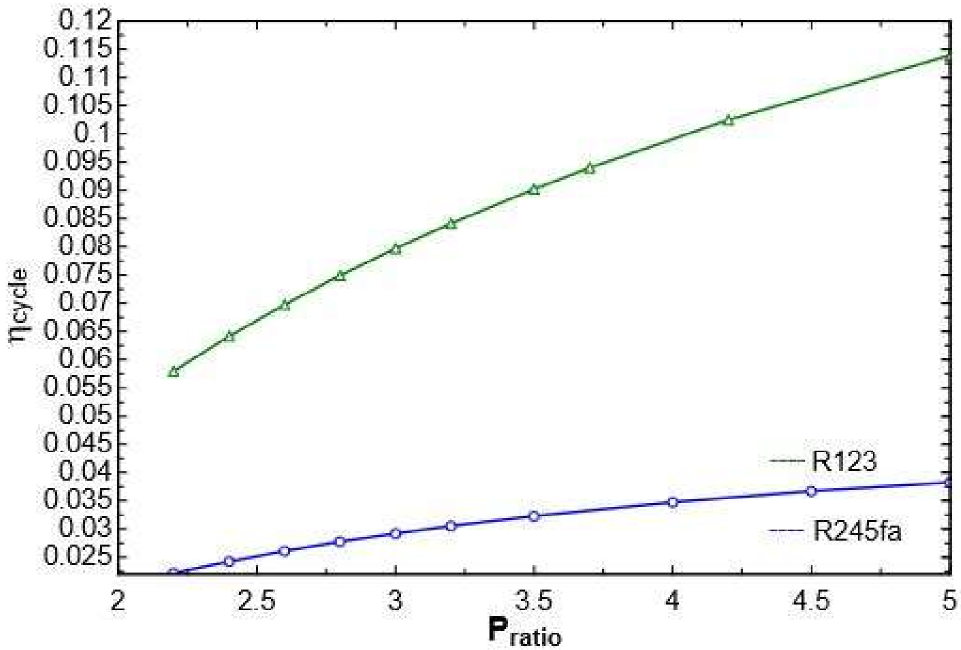

Fig 5.16 Pressure ratio Vs cycle efficiency

5.3.5 Outcome of pressure ratio on compressor and turbine work

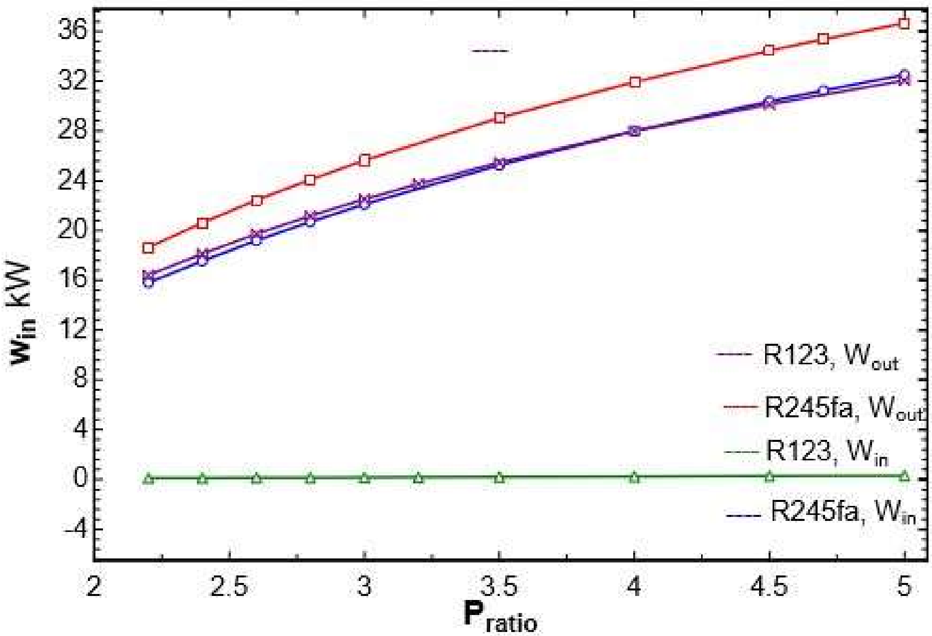

Fig 5.17 Pressure ratio V/s compressor and turbine work

5.3.6 Outcome of Minimum cycle temperature

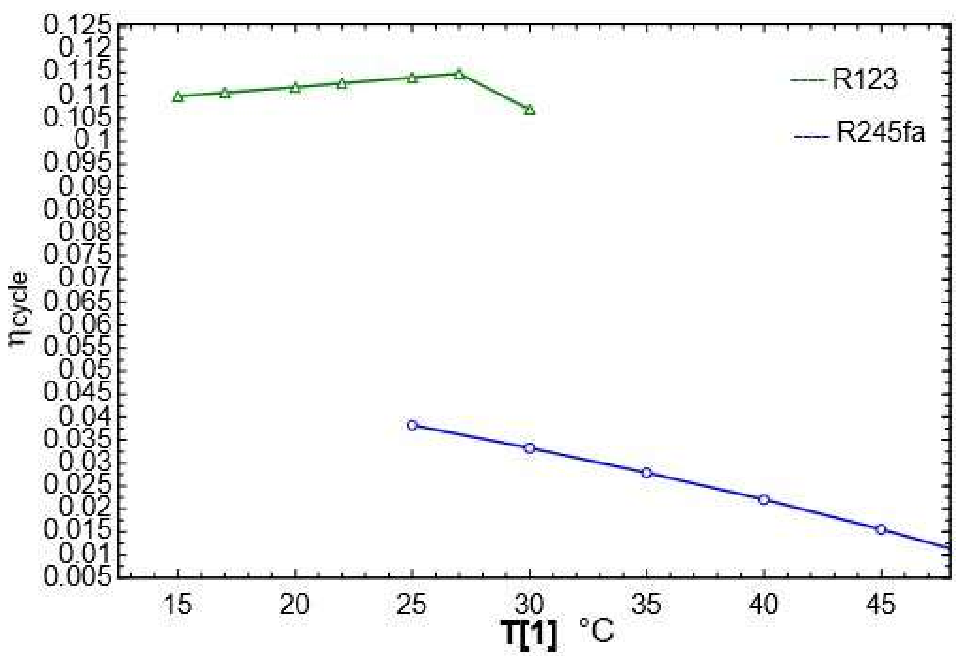

Fig 5.18 Minimum cycle temperature V/s cycle efficiency

Fig 5.13 show the variation of cooler pressure and cycle efficiency From the Figure 5.13 the efficiency of cycle is 11.39% at 114 kPa cooler pressure for fluid R123 and 3.812% at 114 kPa cooler pressure for fluid R245fa.

Fig 5.14 shows the variation between turbine inlet temperature and efficiency of cycle. From the Figure 5.14 that the efficiency of cycle is 11.47% at 150ºC for fluid R123 and 2.892% at 150ºC for fluid R245fa.

Fig 5.15 shows the variation of cooler pressure and net-work. From the Fig. 5.15 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.

Fig. 5.16 show the variation of pressure ratio and cycle efficiency. From the Figure 5.16 the efficiency of cycle is 8.412% at pressure ratio 3.2 for fluid R123 and 3.054% at pressure ratio 3.2 for fluid R245fa.

Fig 5.17 shows the variation between pressure ratio and turbine and compressor work. It can be seen from the Fig. 5.17 that for fluid R123 the work input and work output is 0.1691 kW and 23.67 kW and 27.09 kW respectively at pressure ratio 3.2.

Fig. 5.18 shows the graph between minimum operation temperature and cycle efficiency is 3.827% at T1 = 25ºC for fluid R245fa and 11.39% for fluid R123.

5.4 Brayton with Reheating

5.4.1 The effect of Re-heater pressure

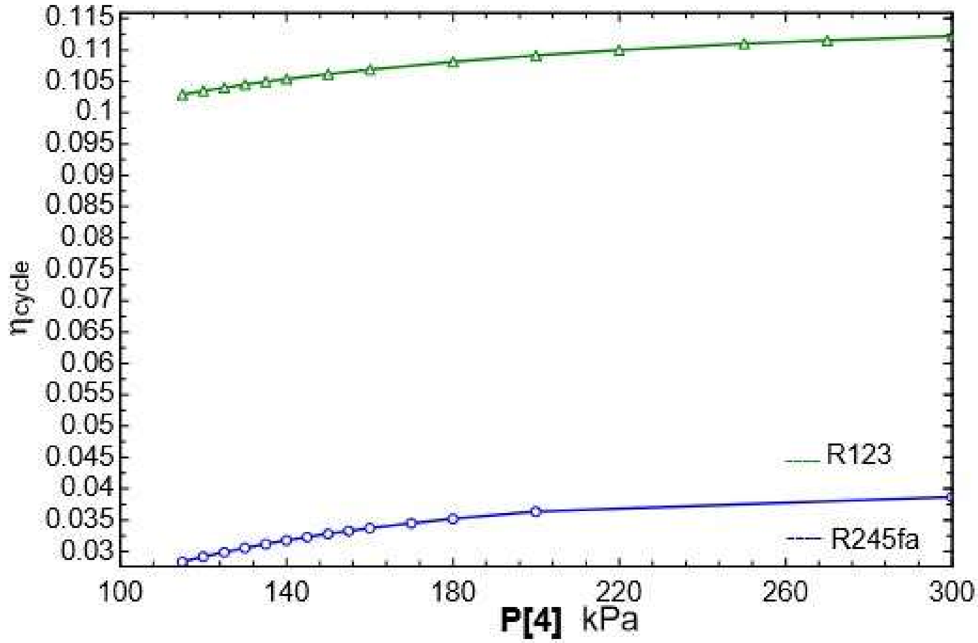

Fig 5.19 Re-heater pressure Vs cycle efficiency

5.4.2 The outcome of high pressure turbine inlet temperature

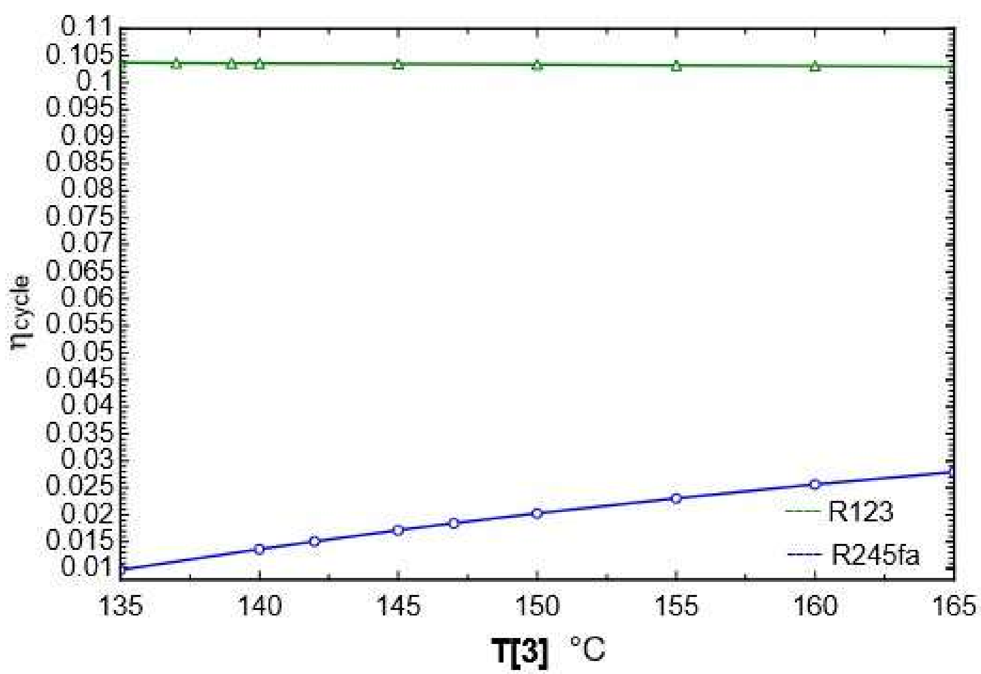

Fig 5.20 High pressure inlet turbine temperature V/s cycle efficiency

5.4.3 Outcome of Re-heater pressure

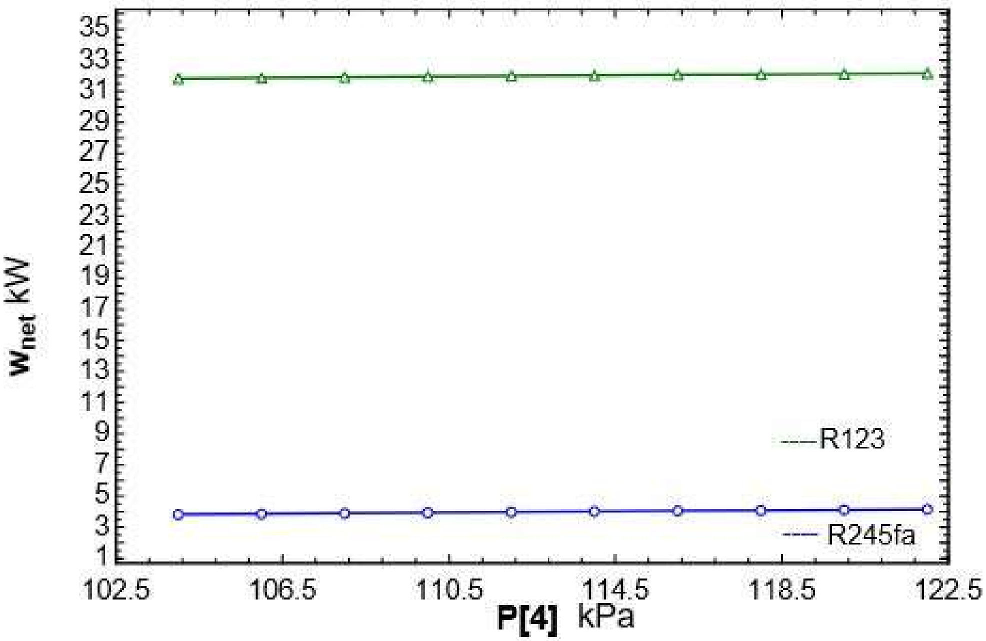

Fig 5.21 Re-heater pressure V/s Net-work

5.4.4 Outcome of pressure ratio on efficiency of cycle

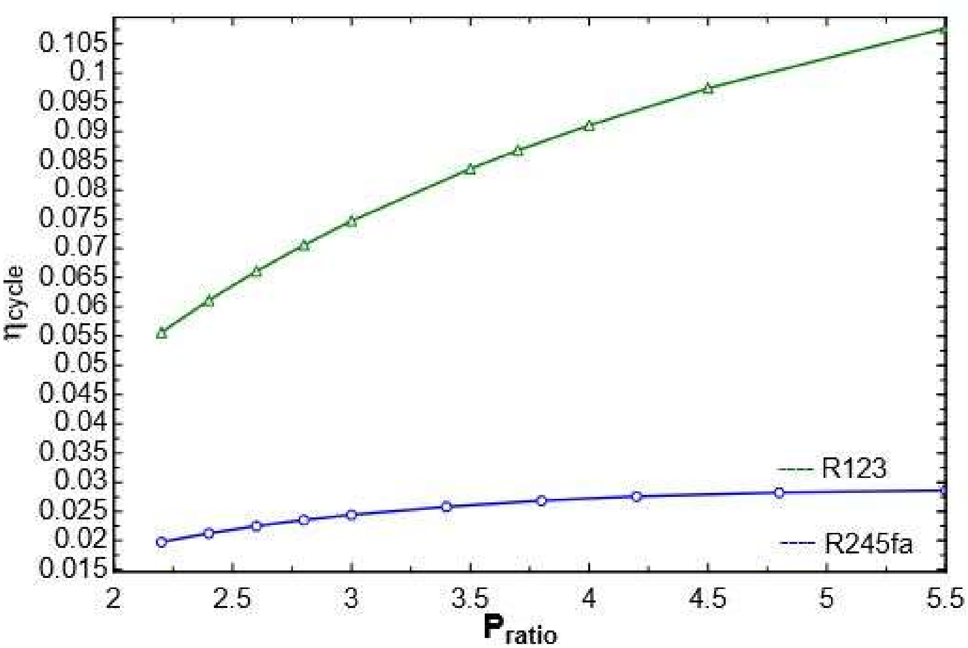

Fig 5.22 Pressure ratio V/s efficiency of cycle

5.4.5 Outcome of pressure ratio on compressor and turbine work

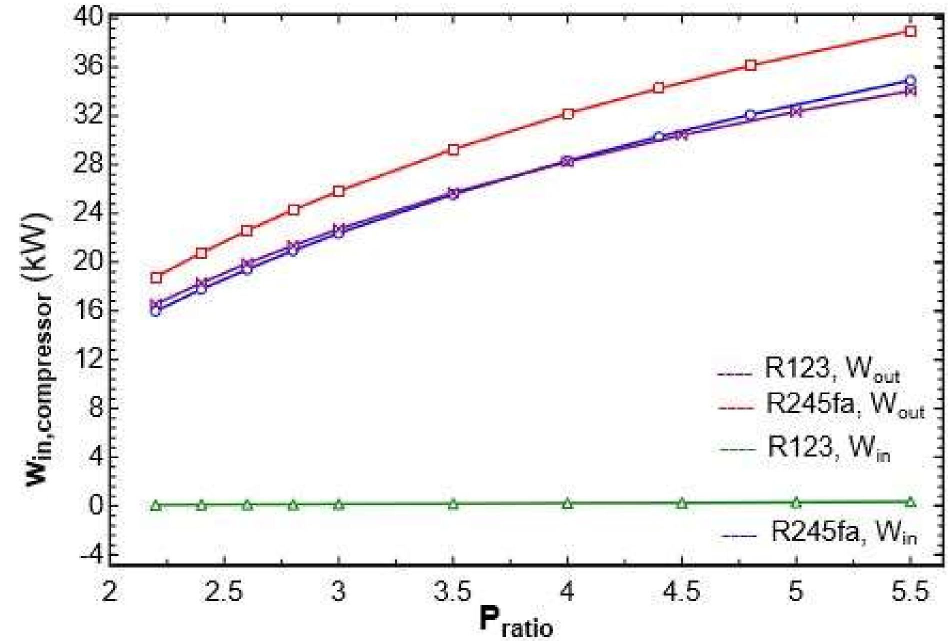

Fig 5.23 Pressure ratio V/s turbine and compressor work

5.4.6 Outcome of minimum cycle temperature

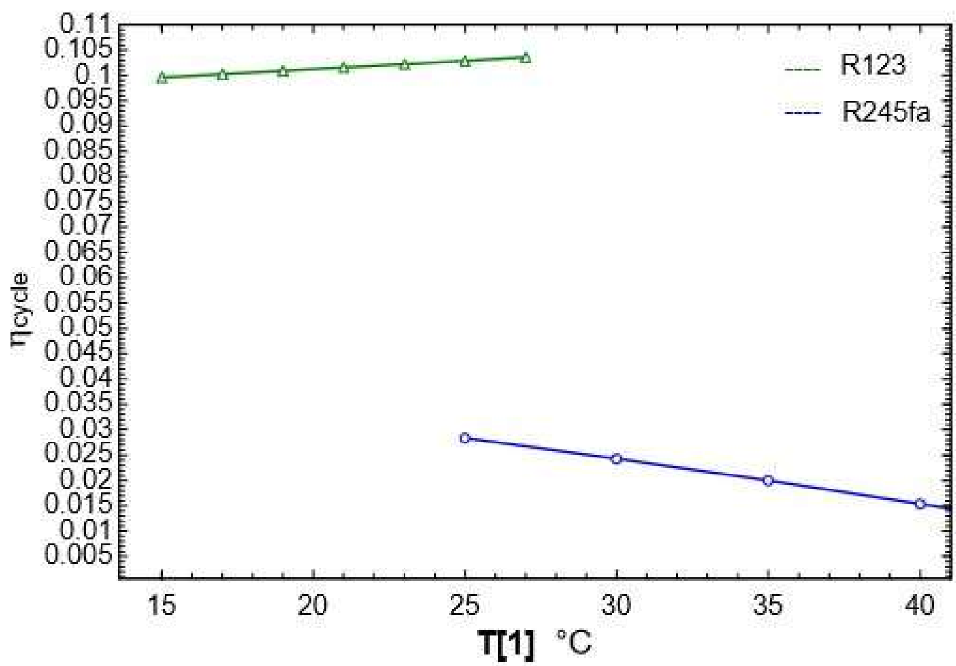

Fig 5.24 Minimum cycle temperature V/s cycle efficiency

Fig 5.19 show the variation of re-heater pressure and cycle efficiency. As per the Fig 5.19 the efficiency of cycle is 10.69% at 160 kPa re-heater pressure for fluid R123 and 3.372% at 160 kPa re-heater pressure for fluid R245fa.

Fig 5.20 indicates the changes of turbine inlet pressure and cycle efficiency. As per the Figure 5.20 that the efficiency of cycle is 10.31% at 160 kPa turbine inlet pressure for fluid R123 and 2.562% at 160 kPa turbine inlet pressure for fluid R245fa.

Fig 5.21 show the variation of re-heater pressure and net-work. As per the Figure 5.21 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.

Fig 5.22 show the variation of pressure ratio and cycle efficiency. As per the Figure 5.22 the efficiency of cycle is 7.4% at pressure ratio 3 for fluid R123 and 2.4% at pressure ratio 3 for fluid R245fa.

Fig 5.23 show the variation of pressure ratio and compressor & turbine work. It can be seen from the Fig 5.23 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 respectively at pressure ratio 3.2.

Fig 5.24 shows the graph between minimum operation temperature and cycle efficiency is 2.837% at T1=25ºC for fluid R245fa and 10.29% for fluid R123.

5.5 Brayton cycle with Intercooling and Reheating

5.5.1 Outcome of cooler pressure

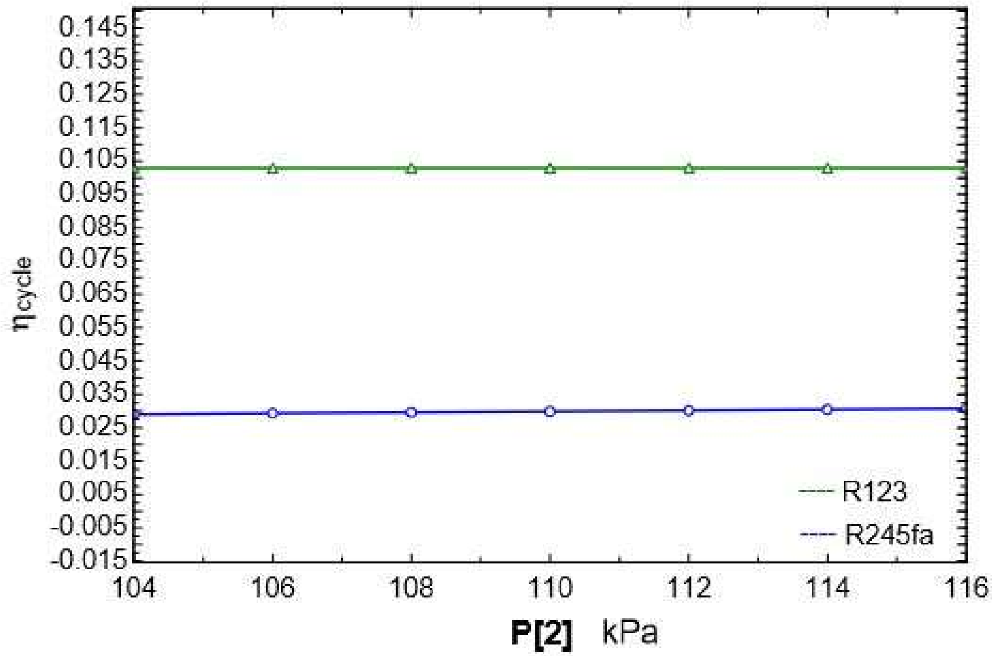

Fig 5.25 Cooler pressure V/s cycle efficiency

5.5.2 Outcome of high pressure Inlet temperature of turbine

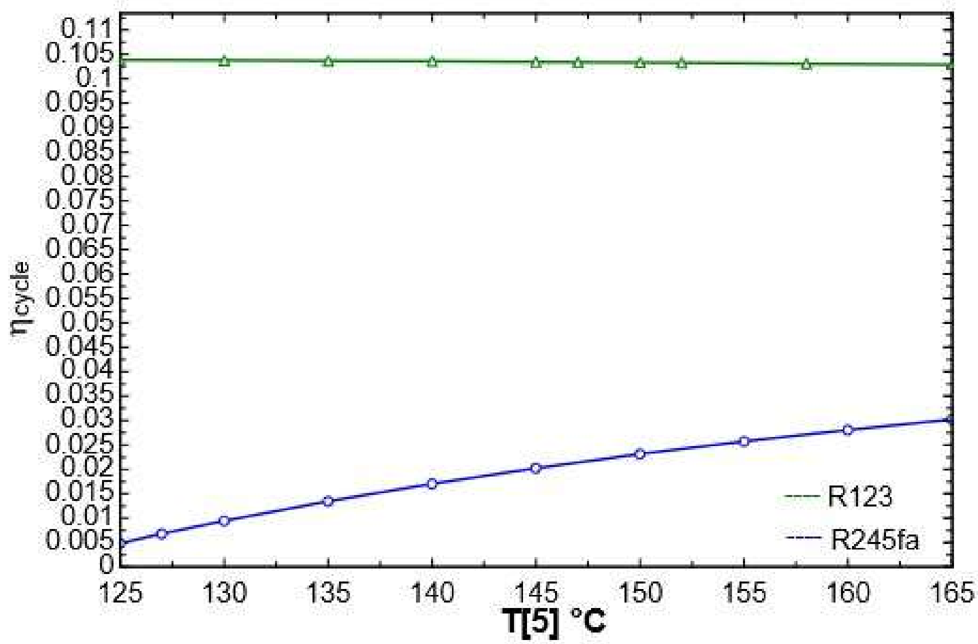

Fig 5.26 High pressure Inlet temperature of turbine V/s cycle efficiency

5.5.3 Outcome of heater pressure on total cycle work

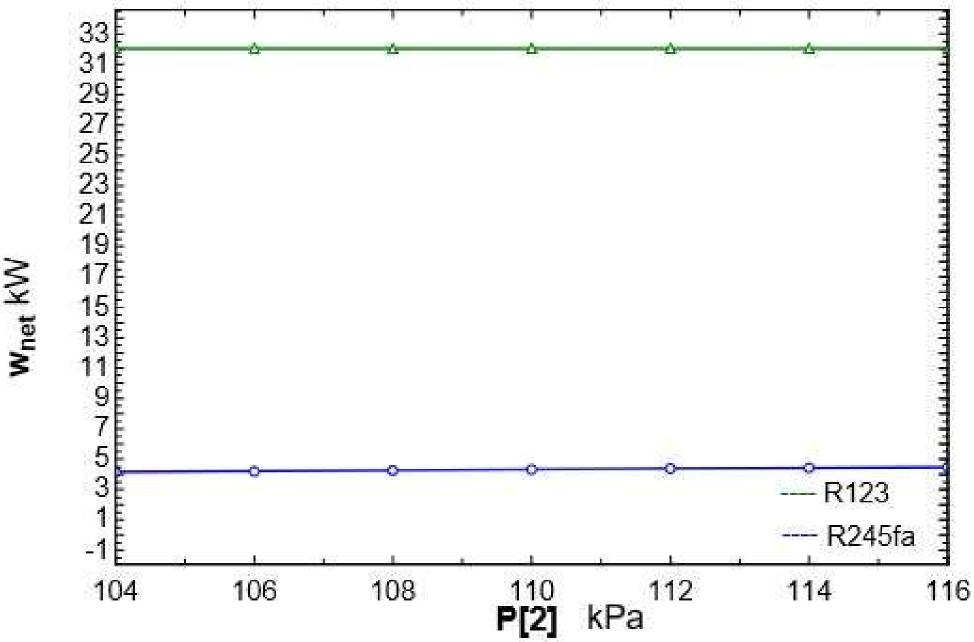

Fig 5.27 Heater pressure Vs Net-work

5.5.4 Outcome of pressure ratio on cycle efficiency

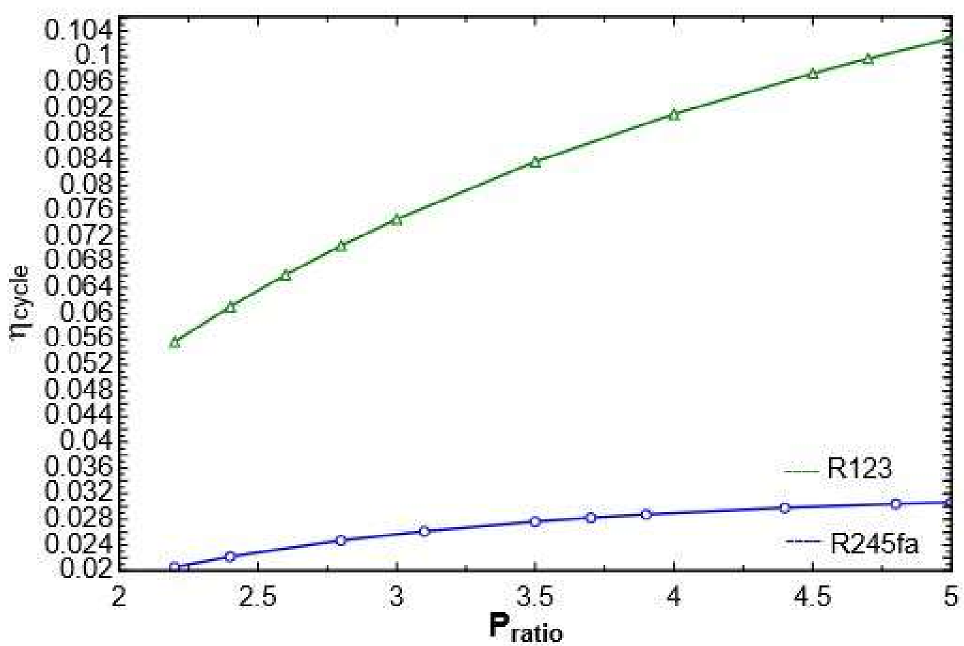

Fig 5.28 Pressure ratio V/s efficiency of cycle

5.5.5 Outcome of pressure ratio on turbine and compressor work

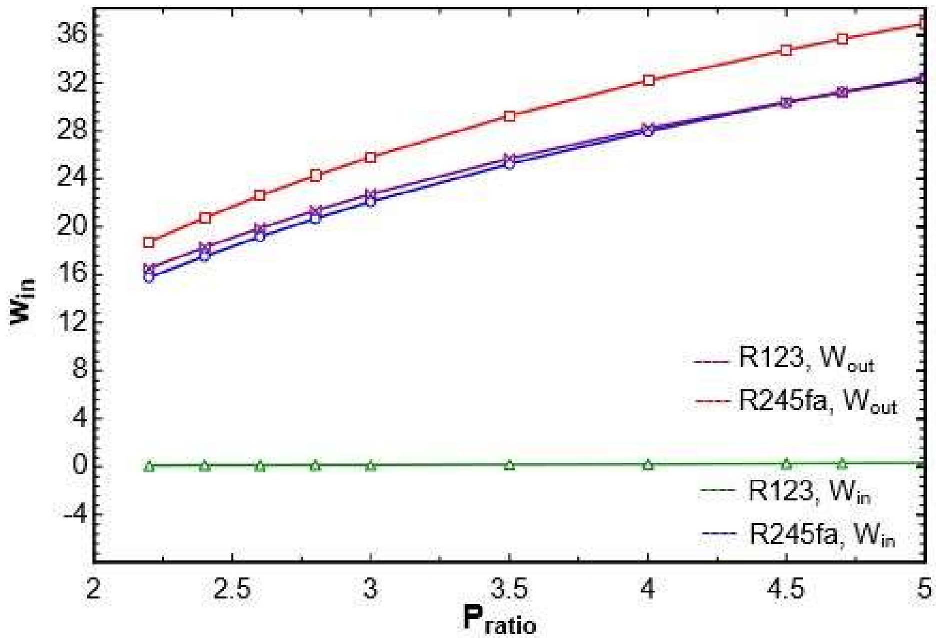

Fig 5.29 Pressure ratio V/s compressor and turbine work

5.5.6 Outcome of Minimum cycle temperature

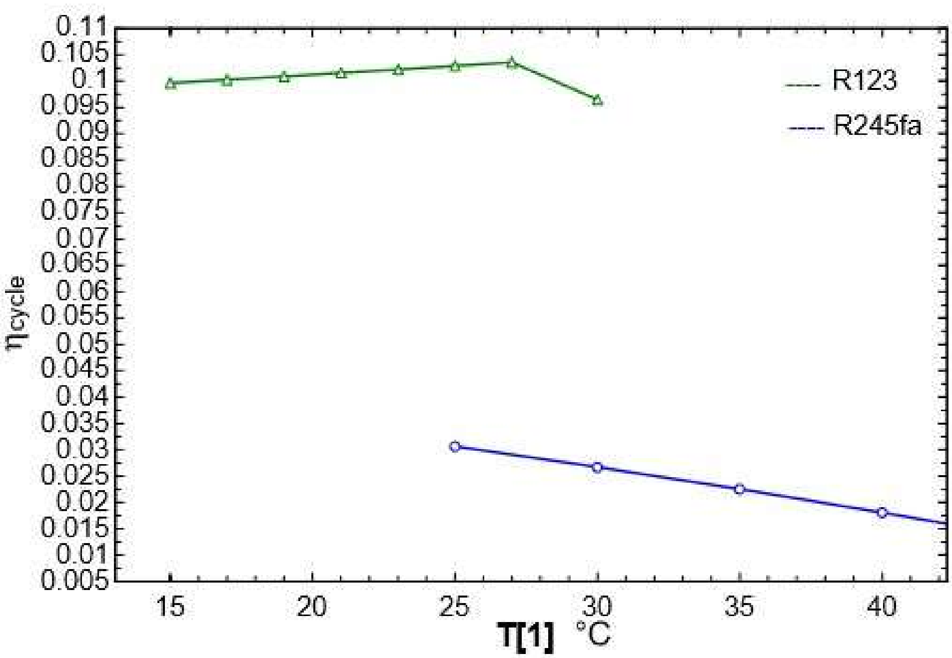

Fig 5.30 Minimum cycle temperature Vs cycle efficiency

Fig 5.25 show the variation of cooler pressure and cycle efficiency. As per the Figure 5.25 the efficiency of cycle is 10.29% at 114 kPa cooler pressure for fluid R123 and 3.051% at 114 kPa cooler pressure for fluid R245fa.

Fig 5.26 show the graph between HP inlet turbine temperature and cycle efficiency is 2.31% at T5 =150ºC for fluid R245fa and 10.33% for fluid R123.

Fig 5.27 show the variation of heater pressure and Net-work. As per the Figure 5.27 the total 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.

Fig 5.28 show the variation of pressure ratio and cycle efficiency. As per the Fig 5.28 the efficiency of cycle is 7.742% at 3 pressure ratio for fluid R123 and 2.61% at 3 pressure ratio for fluid R245fa.

Fig 5.29 show the variation of pressure ratio and compressor & turbine work. It can be seen from the Fig 5.29 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 3.2.

Fig 5.30 shows the graph between minimum operation temperature and cycle efficiency is 3.064% at T1=25ºC for fluid R245fa and 10.29% for fluid R123.

5.6 Combined cycle

5.6.1 Outcome of heater pressure on combined efficiency of cycle

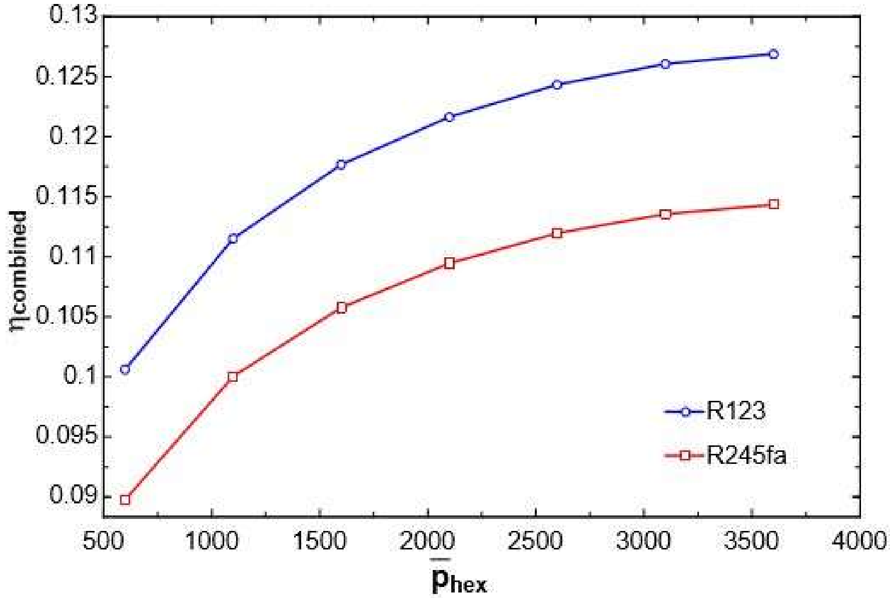

Fig 5.31 Heater pressure V/s combined cycle efficiency

5.6.2 The effect of heater pressure on combined net-work

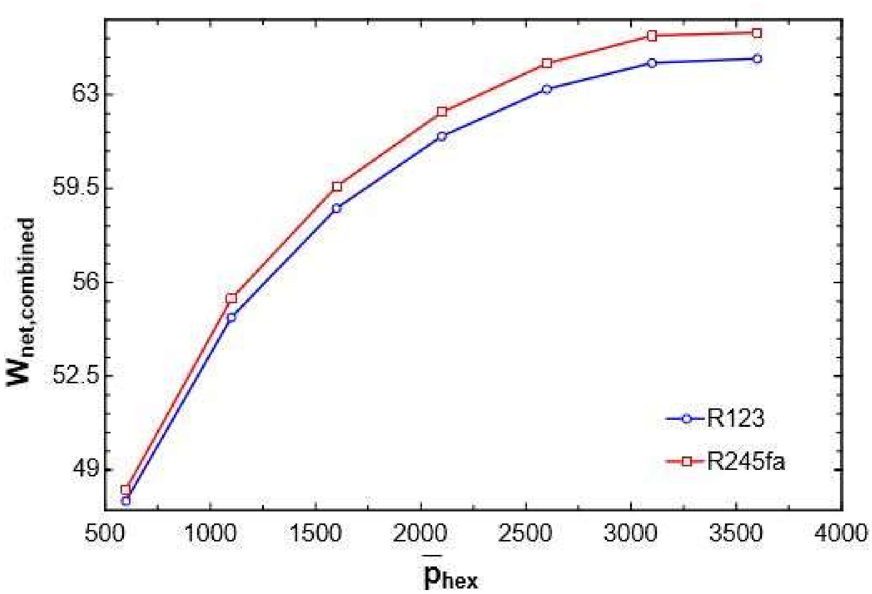

Fig 5.32 Heater pressure V/s combined net-work

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

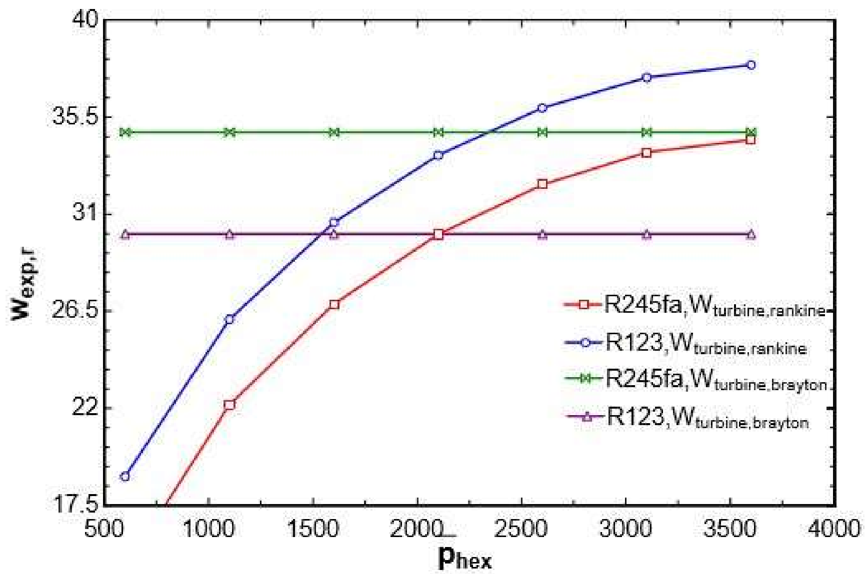

Fig 5.33 Heater pressure Vs Turbine work

5.6.4 Effect of heater pressure on pump and compressor work

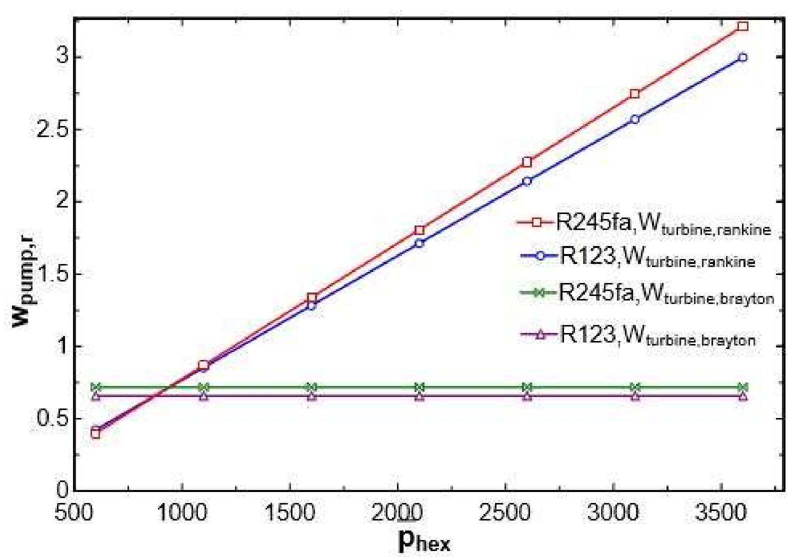

Fig 5.34 Heater pressure Vs pump and compressor work

5.6.5 The effect on heater temperature on combined efficiency

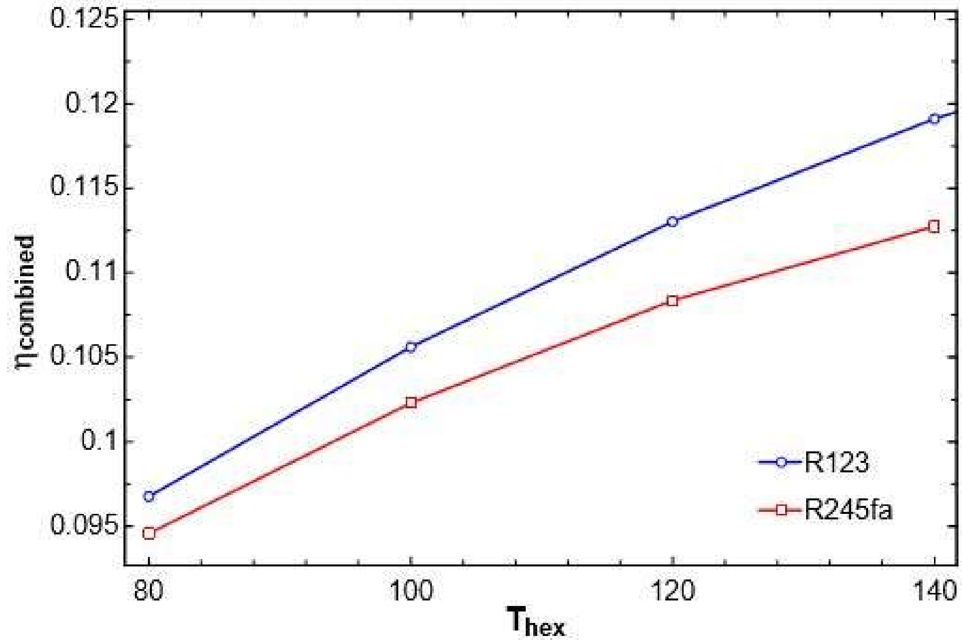

Fig 5.35 Heater temperature Vs combined cycle efficiency

5.6.6 Outcome of heater pressure on combined net-work

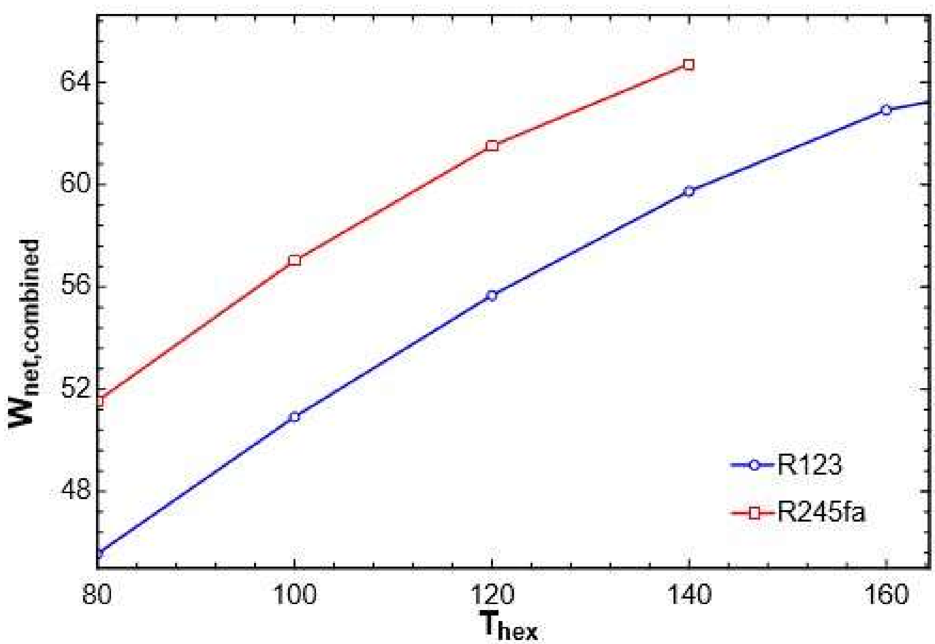

Fig 5.36 Heater temperature V/s combined net-work

5.6.7 Outcome of pressure ratio on combined cycle efficiency

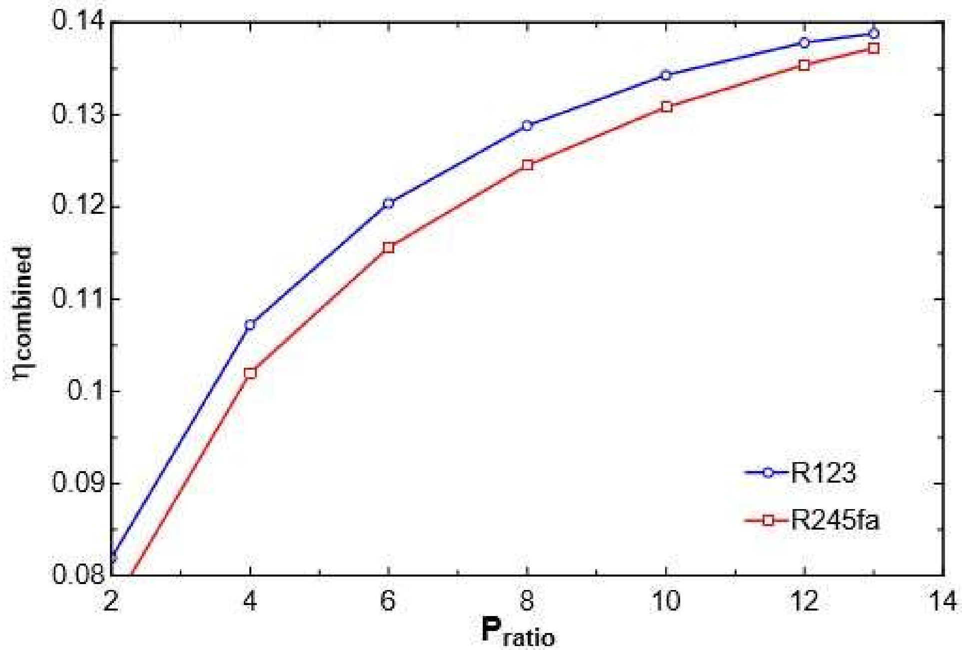

Fig 5.37 Combined cycle efficiency V/s Pressure ratio

5.6.8 Outcome of pressure ratio on combined net work

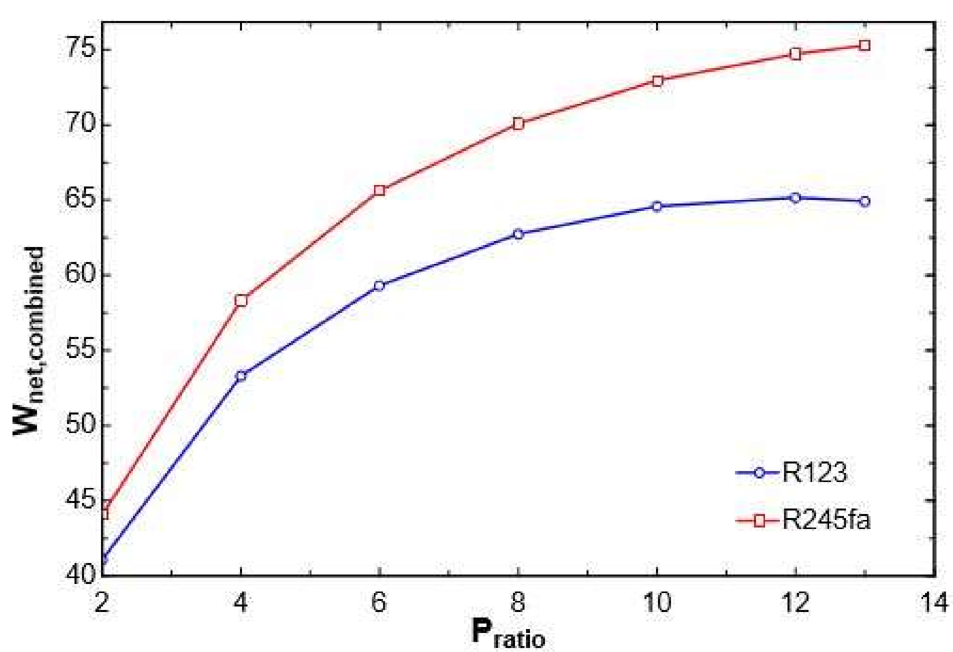

Fig 5.38 Combined net-work V/s Pressure ratio

5.6.9 Effect of compressor inlet pressure on combined efficiency

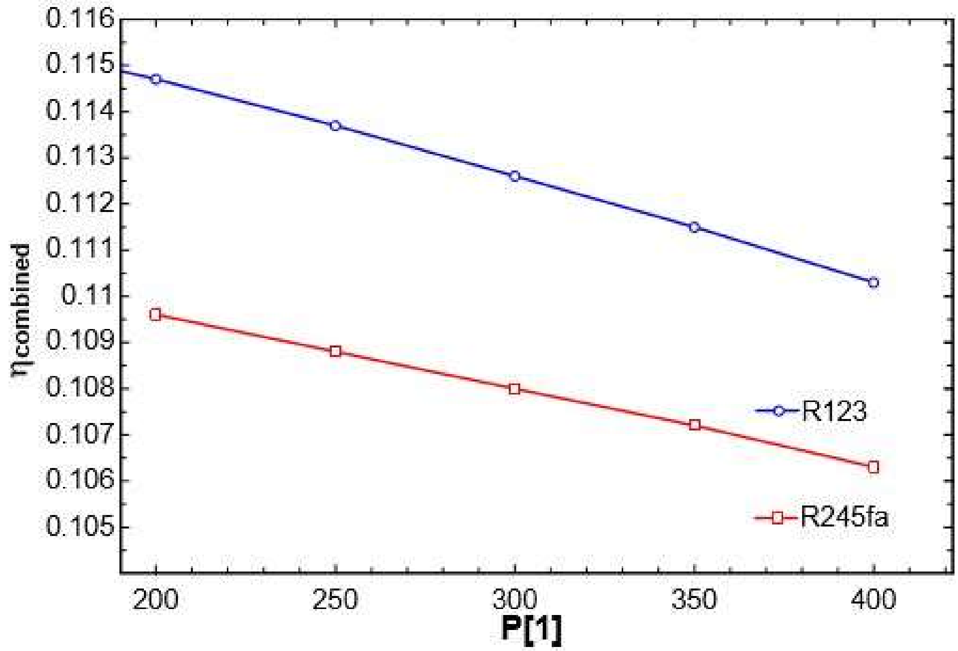

Fig 5.39 Compressor inlet pressure Vs combined cycle efficiency

5.6.10 Effect of compressor inlet pressure on combined net-work

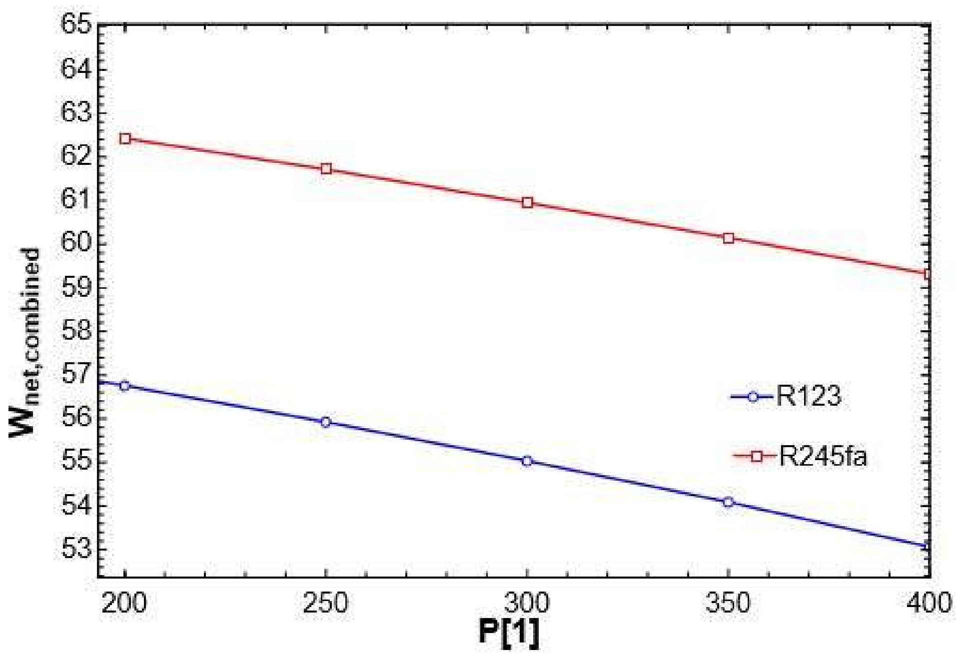

Fig 5.40 Compressor inlet pressure Vs combined net-work

Fig 5.31 shows the variation between heater pressure and combined cycle efficiency. As per the Fig 5.31 the efficiency of combined cycle, Rankine efficiency, Brayton efficiency is 12.43%, 14.04%, 10.72% respectively at heater pressure 2600 kPa for fluid R245fa and 12.43%, 14.04%, 10.99% respectively for fluid R123.

Fig 5.32 shows the change in total work output of combined cycle by varying the heater pressure. It can be seen from the Fig 5.32 that the combine 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.

Fig 5.33 shows the variation in turbine work of Rankine and Brayton cycle by varying the heater pressure. It can be seen from the Fig 5.33 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.

Fig 5.34 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 5.33 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 kW at heater pressure 2600 kPa.

Fig 5.35 shows the change in combined cycle net efficiency by varying the heater temperature. As per the Fig 5.35 the efficiency of combined cycle is 11.91% at heater temperature 140ºC for fluid R123 and for fluid R245fa the combined cycle efficiency is 11.27% at heater temperature 140ºC.

Fig 5.36 shows the change in combined cycle net-work by varying the heater temperature. As per the Fig 5.36 the combined cycle total work is 59.74 kW at heater temperature 140ºC for fluid R123 and for fluid R245fa the combined cycle efficiency is 64.71 kW at heater temperature 140ºC.

Fig 5.37 shows the change in combined cycle net efficiency by varying the pressure ratio. As per the Fig 5.37 the efficiency of combined cycle is 13.43% at pressure ratio 10 for fluid R123 and for R245fa the efficiency of combined cycle is 13.08% at pressure ratio 10.

Fig 5.38 shows the change in combined cycle net-work by varying the pressure ratio. As per the Fig 5.38 the combined cycle total work is 64.58 kW at pressure ratio 10 for fluid R123 and for R245fa the combined cycle net-work is 72.96 kW at pressure ratio 10.

Fig 5.39 shows the change in combined cycle net efficiency by varying the compressor inlet pressure. As per the Fig 5.39 that efficiency of combined cycle 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.

Fig 5.40 shows the change in combined cycle net-work by varying the compressor inlet pressure. As per the Fig 5.40 the combined cycle total 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.

CHAPTER 6

CONCLUSION

In this thesis, the simulation of subcritical Organic Rankine Cycle system with internal heat exchanger using R123 and R245fa is done which shows the influences of internal heat exchanger on Organic Rankine Cycle systems. Also, simulation is done on Brayton cycle and combined cycle. In combined cycle simulation is done using R12 and R245fa. The mass flow rate taken into account for analysis is 1kg/s and the warmth source temperature is taken as 200ºC.The temperature difference of 10ºC is set as the pinch point of heat exchanger.

The simulation outcomes illustrates that there will be an optimal heater pressure for Brayton cycles at various operating conditions.

- On increasing temperature of heat source it is found that there is increase in heater pressure.
- The turbine efficiency will have more vital effect on the thermal efficiency of cycle as compared to that of the pump efficiency.
- 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.

6.1 Future Scope

- An ample study of various upcoming working fluids comparison can be done in both supercritical and subcritical and systems that are equipped with Internal Heat Exchanger.
- More economical factors like heat exchanger and UA values and area can also be investigated.

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