CHAPTER-1

INTRODUCTION

Energy security, economic development and environment protection are not well balanced today and the energy demand is still closely connected to the economic growth. At the same time, the increasing consumption of fossil fuels has led to more and more environment problems such as global warming, ozone depletion and atmospheric pollution. Fossil fuels still play the dominant role in energy resources worldwide, accounting for 77% of the increasing energy demand 2007-2030 (IEA, 2009). Furthermore, along with the fast Development of industry, energy shortages and blackouts have appeared more and more frequently all over the world. Due to all these reasons, utilizing low-grade waste heat for energy production has attracted more and more attention for its potential in reducing the fossil fuel consumption.

The most commonly investigated cycles in low-grade heat source and waste heat utilizations today are Organic Rankine Cycles (ORCs) and Kalinga cycle (binary fluids and fluid mixtures). When utilizing low grade waste heat, the traditional steam Rankine cycle does not give satisfactory performance due to its low thermal efficiency and large volume flows; and thus so called Organic Rankine cycle(ORCs) have been proposed. These cycle use organic substances as working fluid system consist of at least five major component : evaporator, turbine, pump, condenser and working fluid. To make an equal comparison with the CO_2 process later, a regenerator is also included in the current calculation.

According to waste heat recovery report by U.S. DOE, industrial processes in the United States consumes approximately 35 quadrillion BTU (Quads) of energy per year (BCS Inc., 2008). This amount is about 30% of total energy consumed in the United States. This report also estimates that around 25-50% of that energy is lost to waste heats (BCS Inc., 2008). The report categorizes the waste heat based on the temperature of the waste products. The three waste heat groups are: low, medium, high temperature. Table 1 defines the temperature range for each source based on limited samples of industrial application. Table 1 also shows the amount of waste heat and work potential of each waste heat group (BCS Inc., 2008). The waste heat and work potential is based on the reference temperature of 25^{o} C.

	Temperature Range		Waste Heat (trillion BTU per year)		Work Potential (trillion BTU per year)
	ºF	ºC	77F [25ºC] *Reference*	300F [150ºC] *Reference*	77F [25ºC] *Reference*
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 waste heat

This data indicates that low temperature heat source results in 60 % of total waste heat source. It is estimated that 290 trillion BTU per year or 33% of this low temperature heat can be recaptured into useful works. Low temperature heat source recovery presents the largest opportunity to recover among other discarded heat.

1.1 Organic Rankine Cycle

Organic Rankine Cycle can generate power from natural and waste heat sources such as geothermal sources, biomass, solar radiation and waste heat coming from industrial processes and fuel combustion, solar radiation etc. Therefore, it is considered an important and promising method of renewable energy production process.

The organic rankine cycle uses an organic substance as a working fluid as is implied by the word "organic".

In organic rankine cycle power system, the working fluid selected should maximize the thermal efficiency considering the heat source and long term performance temperature range. The temperature range is from about ~100^o C to about ~350^o C.

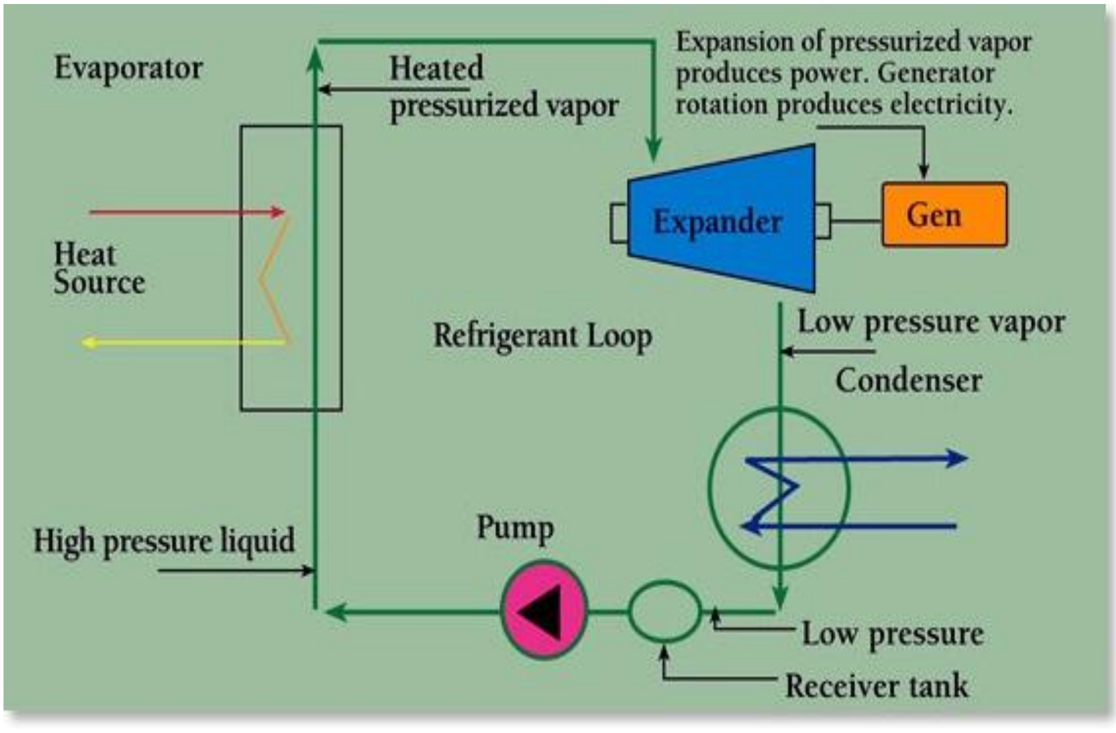

Fig. 1.1 Basic Organic Rankine Cycle

Rankine cycle gives the highest efficiency when low temperature heat is converted into electricity. The waste heat coming from combustion engines and processes in industries : hot gases coming from blast furnaces in steel industry and from the kilns in ceramic industry, exhaust gases coming from engines and turbines, hot liquids that are used to cool kilns and furnaces etc. The organic rankine cycle uses organic fluid as working fluid instead of water. Organic Rankine Cycle is used for establishment of many commercial plants and test plants. The refrigerants that are used most commonly in ORC include R123, R600, R11, R114 etc. These days the interest is focused on new refrigerants like R1234yf, R1234ze etc.

The net output, first and second law efficiency are the indicators of system performance besides the working fluid properties. As far as maximum pressure is concerned, the ORC can be subcritical or supercritical. The cycles can be employed with internal heat exchanger or without IHE. Certain discussions are continuing with some reports stating that exergy efficiency and power output stays unchanged while some stating that thermal efficiency rises but output is stationary. It is also stated that induction of IHE greatly reduces expansion ratio, evaporation pressure being below a certain limit.

1.2 Advantages :

Rankine Cycle gives the best efficiency and power output with an organic fluid as a working fluid instead of water when waste heat at moderate inlet temperature is used as a heat source. This is attributed to the fact that the specific vaporization heat of water is much higher than organic fluids. Thus, the heat source fluid that is to be cooled is followed better by the organic working fluid. In the ORC process more electric power can be generated from a heat source as the exhaust gas is cooled to a lower temperature.

1.3 WORKING FLUIDS

The following working fluids are used in this study: R123, R600, HFO-1234yf, HFC-134a, HFC-245fa, isopentane. Many researchers investigated these working fluids and are representative working fluids that are used in ORC. Not only HFC-134a is used but also HFC-245fa and iso-pentane are used in ORC to recover low temperature waste heat from the industrial plants.

Ethanol performs very well at medium temperature levels i.e. 200-300^o C hence, it is also recently being used as automotive ORC working fluid. The evaporation temperature of organic substances is relatively lower as compared to steam, this being the main reason and advantage of using them as working fluid in ORC. The low evaporation temperature makes it possible for them to vaporize or superheat by low or medium temperature heat sources. In addition to this, the expander life can be extended by using the organic substances which are categorized as dry or isentropic fluids. Under the saturated or superheated vapour conditions, the vapour that is expanded inside the expander tends to be dry.

expander and pump efficiencies. The actual performance has to be verified by performing detailed simulations for the model that employs heat exchanger.

The major focus today is kept on the subcritical cycle, but some of the research on supercritical cycle employing heat exchanger has also been done. The goa l of this study is to find the essential effects of Internal heat exchanger on the subcritical as well as supercritical organic rankine cycle. The performance evaluation parameters consists of net power output, thermal efficiency and also the exergy efficiency.

The study analyses the subcritical and supercritical model separately and for locating the pinch point and calculating hot stream outlet temperature, a nested iteration method is proposed. It is found in 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 dependent on a certain evaporation pressure.

1.4 Saturated Cycle Model :

It is the most basic cycle that can achieve a compact system design relative to acceptable thermal efficiencies when recovering low temperature waste heat.

The T-S diagram of the cycle is shown in the Fig. 1.2. The working fluid is heated in the evaporator at a constant pressure after isentropic pumping until it becomes saturated vapour which is then is followed by isentropic expansion process.

Depending on the working fluid, the phase condition of working fluid during the expansion process goes two-phase mixture for wet fluids, goes to superheated state for dry fluids and becomes saturated vapour for isentropic fluid. After the expansion process, isobaric condensation takes place in the condenser, which return fluid to saturated liquid state.

1.5 Trilateral Cycle Model :

It is a simple cycle as compared to conventional binary cycle, its performance is more competitive for low temperature heat sources, especially when a proper two phase expander is employed. Furthermore, it has a lower operational cost for equivalent sized componenets.

This cycle consists of two features, first being that the working fluid temperature profile tends to match the heat source temperature profile perfectly and the second one is that it operates at reasonable pressure The schematic diagram is shown in the Figure 1.3. The pumping pressure which is theoretically same as evaporation pressure lies below the critical pressure of working fluid, and the temperature at expander inlet is maintained at a target saturated temperature.

The cycle consists of isentropic pumping state of working fluid followed by isobaric heating of the working fluid to the bubble point. Isentropic expansion of working fluid to a two phase mixture is then induced and finally an isobaric condensation occurs to return the two phase mixture back to a liquid state.

The primary difference between trilateral and conventional cycle is that the working fluid remains a liquid as it leaves the evaporator and the fluid expansion through expander occurs entirely within the two-phase range.

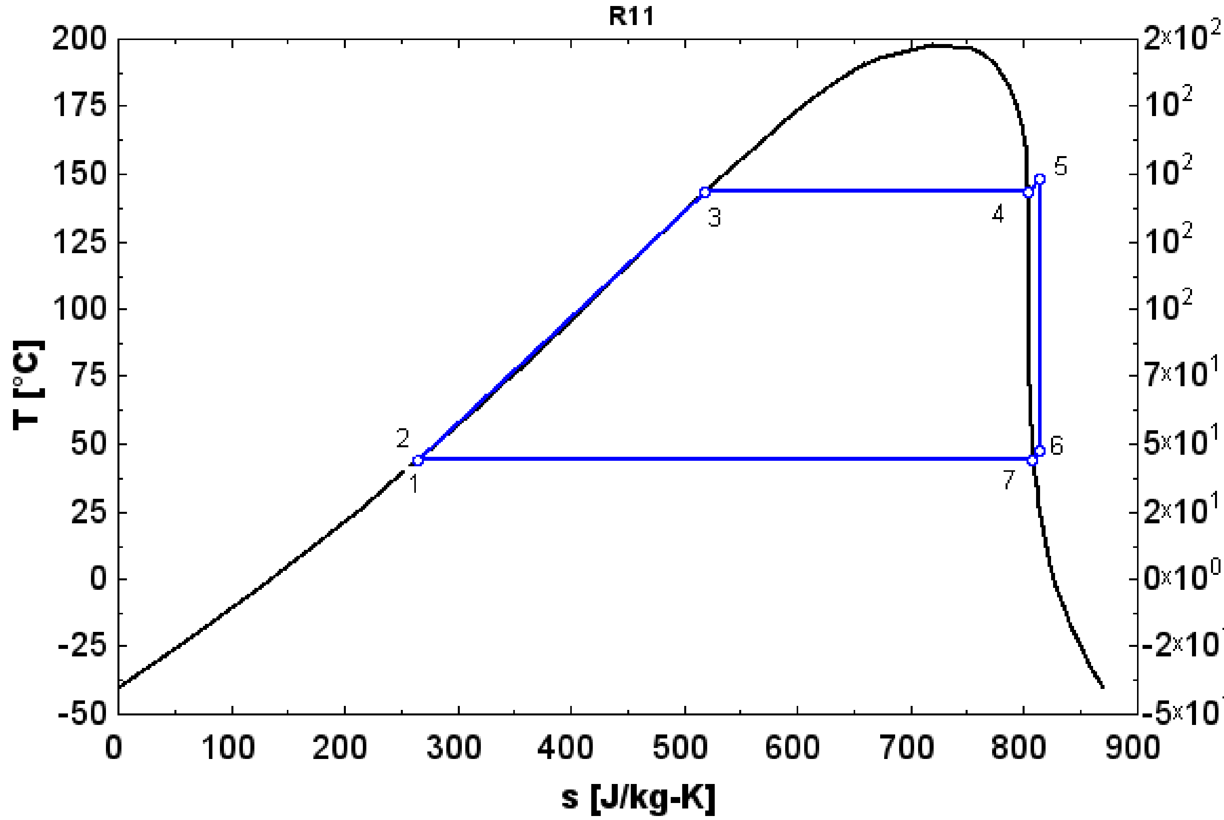

Fig. 1.2 T-S diag. for Saturated Cycle

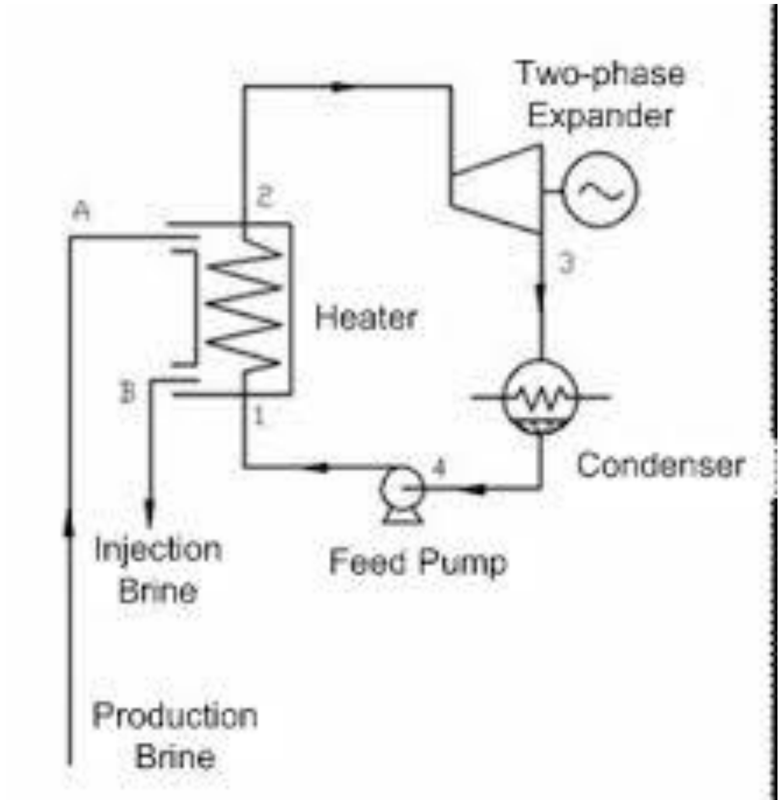

Fig. 1.3 Schematic diag. of trilateral system

1.6 Superheated Cycle Model :

As compared to trilateral and saturated cycle, the superheated cycle allows more heat to enter the ORC system and changes the working fluid state to superheated state. In some cases, the superheated cycle results in higher thermal efficiency than that of saturated cycle.

After the isentropic pumping process, heating process takes place until the working fluid phase changes to superheated state, the temperature being higher than saturated vapour state, the pressure level being kept the same. As the higher expander inlet temperature results in higher power production therefore the temperature is kept higher than the critical temperature at the expander inlet. The working fluid then passes through the expander with isentropic expansion and then condenses to saturated liquid at constant pressure.

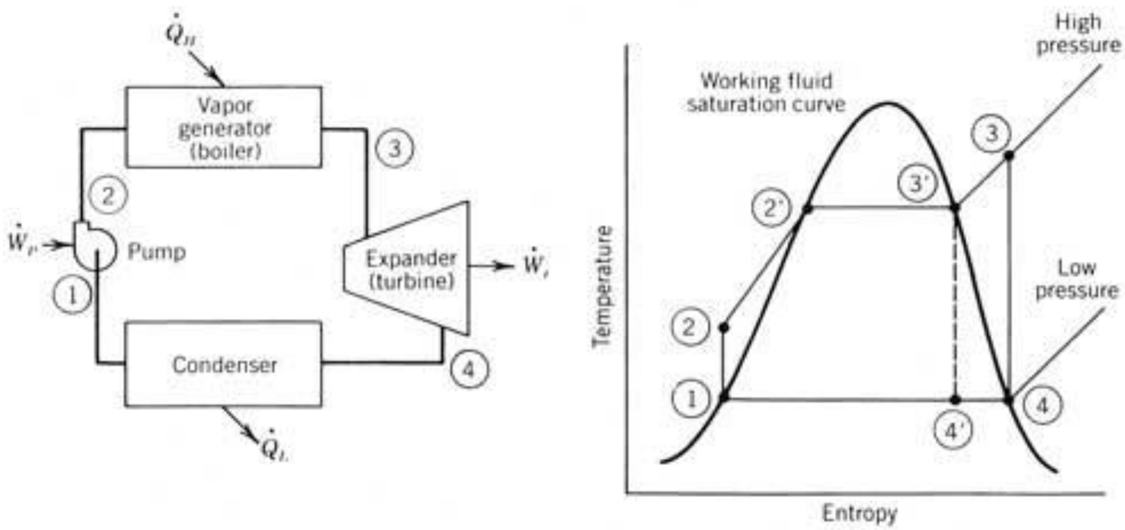

Fig. 1.4 Schematic diag. and T-S plot of Superheated System

1.7 <u>Subcritical Cycle Model</u> :

The subcritical cycle have almost operationl condition as in superheated cycle, however, the pumping pressure is kept higher than the pressure used for the superheated cycle but slightly lower than the critical pressure.

The T-S diagram is shown in the Figure 1.5 which looks like the superheated cycle. The processes involve isentropic pumping followed by isobaric heating, isentropic expansion and finally isobaric condensation.

The expander inlet temperature is kept higher than the critical temperature of working fluid after the expansion process in this cycle.

1.8 Supercritical Cycle Model :

In the supercritical Cycle, the expander inlet condition i.e. pressure and temperature are kept higher than the critical point of working fluid

The main advantage of the cycle is that the hot source temperature can be raised to a higher level that cannot be reached in other cycles.

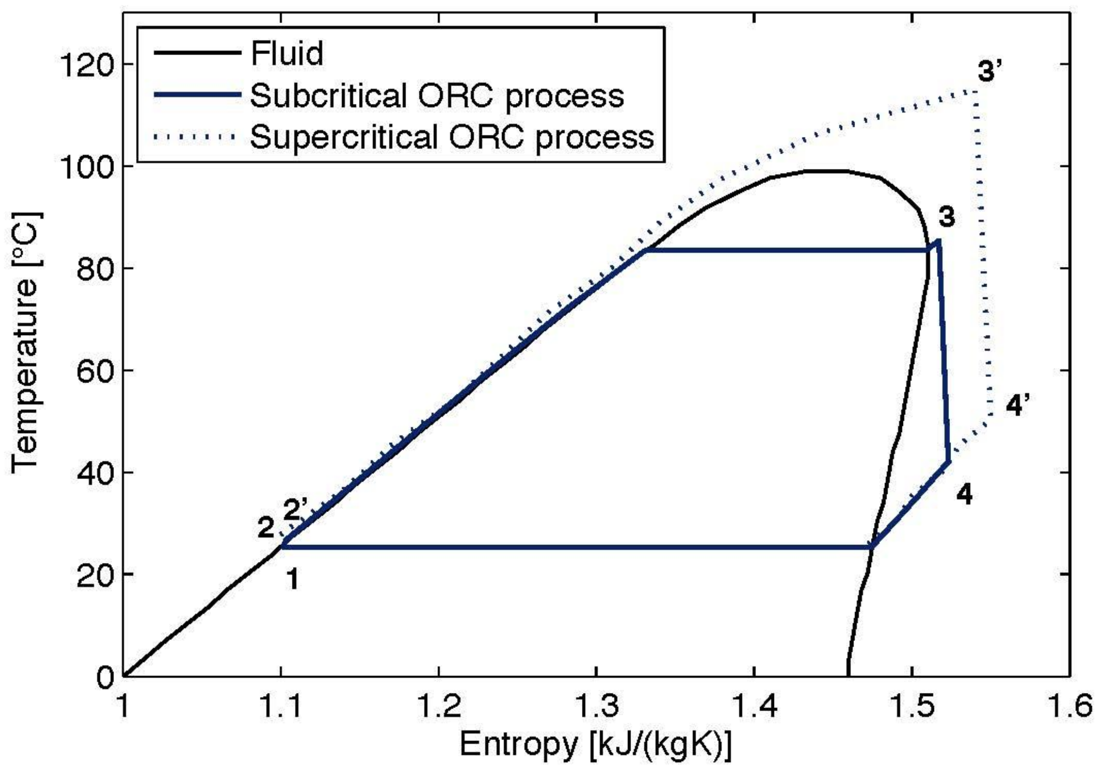

Fig. 1.5 T-s diag. showing subcritical and supercritical system

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] did the theoretical investigation and has suggested the feasibility of introducing a waste heat recovery system in two stage turbocharged HDD engine.

The WHR is attained by introducing a Rankine cycle, which uses an organic substance or directly water as working fluid depending on energetic performance considerations.

They also suggested an alternative for improving the overall thermal efficiency of Diesel engine consists of recovering the energy lost by means of a waste heat recovery(WHR) system. These solutions are based on adapting one of the turbochargers by removing its turbine and trying to recover the energy by rankine cycle. Finally, the turbine of the Rankine cycle supplies the recovered energy directly to the compressor of this turbocharger.

Wang. et al. [2] did the performance evaluation of a low temperature solar Rankine cycle system utilizing R245fa. They found that low-temperature solar Rankine cycle utilizing R245fa as the working fluid is proposed ad an experimental system is designed, constructed, and tested. Both the evacuated solar collector and flat plate solar collector are used in experimental system, meanwhile, a rolling piston R245fa expander is also mounted in the system. The new designed R245fa expander works stably in the experiment, with ana average expansion power output of 1.73kw and average isentropic efficiency of 45.2%. The overall power generation efficiency estimated is 4.2%, when the evacuated solar collector is utilized in the system, and with the condition of flat plate collector, it is about 3.2%.

J.et al. [3] suggested the method for customizing an organic rankine cycle to a complex heat source for efficient energy conversion, demonstrated on a Fischer Tropsch plant. Organic Rankine Cycle provide an alternative to traditional steam rankine cycle for the conversion of low grade heat source into power, where conventional steam power cycle is known to be inefficient. A large processing plant often has multiple low temperatures waste heat source available for conversion to electricity by a low temperature cycle, resulting in composite heat source with a complex temperature-enthalpy profile. Organic fluids are modeled using a pure substance database. The pinch analysis technique of forming composite curve is applied to analyze the effect of each building block on the temperature-enthalpy profile of the ORC heat requirement.

Wang et al. [4] did the performance ananlysis of double organic rankine cycle for discontinuous low temperature waste heat recovery. The optimal operation of several working fluidshve been calculated by a procedure employing MATLAB and REPROP.

The influence of outlet temperature of heat source on the net power output, thermal efficiency, power consumption, mass flow rate, expander outlet temperature, cycle irreversibility and exergy efficiency at a given pinch temperature difference has been employed to obtain a thermodynamic understanding of the ORC performance.

Larjola [5] suggested that electricity from industrial waste heat using high speed organic rankine cycle. In the onversion of low temperature heat into electricity the greatest efficiency is obtained in many cases by using an Organic Rankine Cycle. The ORC process my be feasible also in high temperature applications. This paper deals with an ORC design, in which a high speed oil free turbogenerator feed pump is used.

Li et al. [6] did evaluation of external heat loss from a small scale expander used in organic rankine cycle. With the scaling down of the ORC, the engine shaft power is not oly determined by the enthalpy drop in the expansion process but also the external heat loss from the expander. Theoretically and experimentally support in evaluating small scale expander heat loss is rare. They presented a quantitative study on the convection, radiation, and conduction heat transfer from a kw-scale expander. A mathematical model is buit and validated.

Quoilin et al. [7] did performance and design optimization of a low cost solar organic Rankine cycle for remote power generation. Recent interests in small scale solar thermal combined heat and power system has coincided with the demand growth for distributed electricity supplies in area poorly served by centralized power stations. One potential technical approach to meet this demand is parabolic trough solar thermal collector coupled with Organic Rankine Cycle heat engine. He also describes the design of solar organic rankine cycle being installed in Lesotho for rural electrification purpose. The system consists of parabolic trough collectors, a storage tank, a small scale ORC engine scroll expanders. A model of each component is developed which allows sizing the different components of cycle and evaluates the performance of system.

Herberle et al. [8] did exergy based fluid selection for a geothermal Organic Rankine cycle for combined heat and power generation. In this study the option of combined heat and power generation was considered for geothermal resources at a temperature level below 450k. Series and parallel circuits of an organic rankine cycle and an additional heat generation was compared by second law analysis. The result shows that the second law efficiency of geothermal power plant can be significantly increased in comparison to power generation.

Liu et al. [9] did investigation of two stage Rankine cycle for electric power plant. A two stage Rankine cycle for power generation is presented in this paper. It is made of water steam Rankine Cycle and Organic Rankine bottoming Cycle. By using an organic fluid with density higher than water. The performance of two stage Rankine cycle operating with those different working fluids is evaluated. System efficiency can also be enhanced by introducing a regenerator for some of working fluids.

Aleksandra & Borsukiewicz [10] suggested pumping work in rankine cycle and made calculation based result for the pumping work on ORC. Analysis has been carried out for 18 different organic fluids that can be used a working fluids in the subcritical ORC system. An attempt was made to find correlations between various thermo-physical properties of working fluids, specific work and power of cycle.

Clemente et al. [11] proposed the energy efficiency analysis of organic rankine cycle with scroll expanders for co-generative applications. The model of scroll machine is applicable to calculate the performances of both a compressor and expander, as fuction of geometry of device and working fluid.

Garcia-Rodriguez et al. [12] proposed solar powered Rankine cycle for fresh water production. He also focused that on the analysis of distributed solar powered generation systems for driving a reverse osmosis desalination process. Results point out tht the desalination system coupled to soalr powered organic Rankine cycle exhibit lower specific consumption of solar energy than solar distillation.

Sprousell and Depick [13] suggested the review of Rankine Cycle for internal combustion engine exhaust waste heat recovery. This paper review the history of internal combustion engine exhaust waste heat recovery focusing on organic Rankine cycle since this thermodynamic cycle works well with medium grade energy of exhaust. Selection of cycle expander and working fluid are primary focus of the review. Results demonstrate a potential fuel economy improvement around 10%.

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

Algieri and Morrne [15] suggested the comparative energetic ananlysis of high temperature subcritical and transcritical Organic Rankine cycle in a biomass application in the Sibari district. The present work aims to analyze the energetic performance of ORC's for small scale applications.

Hui-tao et al. [16] did the optimization of Low temperature exhaust gas waste heat fuelled organic Rankine cycle. Cycle parametric analysis was performed to examine the effects of thermodynamic parameters on the cycle.

Kosmadakis et al. [17] suggested the parametric theoretical study of a to stage solar orgnic Rankine cycle for RO desalination. The present work concerns the parametric study of an autonomous two stage solar ORC for Ro desalination. The main aim is to estimate the efficiency as well as to calculate nnual mechanical energy available for desalination.

J.P. Roy and Ashok Mishra [18] This paper presents an analysis of regenerative Organic Rankine Cycle (ORC), based on parametric optimization using R-123 and R-134a during superheating at a constant pressure of 2.50 MPa under realistic conditions. The aim was to select a better working fluid on the basis of obtained system efficiency, turbine work output, irreversibility rate and second law efficiency under applied fixed and variable heat source temperature conditions. R-123 has been found a better working fluid than R-134 for converting low grade heat to power. A computer program has been developed to parametrically optimize and compare the system and second law efficiency, turbine work output, system mass flow rate, irreversibility rate and irreversibility ratio with increases in turbine inlet temperature (TIT) under different heat source temperature conditions to obtain the optimum operating conditions while using R-123 as the working fluid during superheating at various turbine inlet pressures for the utilization of the waste heat sources of temperatures above 150 °C. The calculated results reveal that an inlet pressure of 2.70 MPa gives the maximum system efficiency, turbine work output and second law efficiency with minimum irreversibility rate, irreversibility ratio and system mass flow rate up to a TIT in the range of 165 °C–250 °C.

W. Li et al. [19] Research in the paper sets some initial conditions, boundary conditions, and hypotheses for a mathematic model. Three kinds of pure fluid and one mixture were selected as working fluids and their parametric changes were calculated under different evaporating temperatures. When organic Rankine cycle (ORC) is equipped with internal heat exchanger (IHE), its parameters varies while using different working fluids.

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

Karellas et al. [21] has presented a subcritical cycle and simulated it under the same operation conditions as superheated cycle, however, the pumping pressure is kept higher than the pressure used for superheated cycle but slightly lower than the critical pressure for working fluids.

V. Lemort et al. [22] This paper first presents the results of an experimental study carried out on a prototype of an open-drive oil-free scroll expander integrated into an ORC working with refrigerant HCFC-123. By exploiting the overall expander performance measurements, the eight parameters of a scroll expander semi-empirical model are then identified. The model is able to compute variables of first importance such as the mass flow rate, the delivered shaft power and the discharge temperature, and secondary variables such as the supply heating-up, the exhaust cooling-down, the ambient losses, the internal leakage and the mechanical losses. The maximum deviation between the predictions by the model and the measurements is 2% for the mass flow rate, 5% for the shaft power and 3 K for the discharge temperature. The validated model of the expander is finally used to quantify the different losses and to indicate how the design of the expander might be altered to achieve better performances. This analysis pointed out that the internal leakages and, to a lesser extent, the supply pressure drop and the mechanical losses are the main losses affecting the performance of the expander.

A.Schuster et al. [23] The use of organic working fluids for the realization of the so called Organic Rankine Cycle (ORC) has been proven to be a promising solution for decentralized combined heat and power production (CHP). The process allows the use of low temperature heat sources, offering an advantageous efficiency in small-scale applications. This is the reason why the number of geothermal and biomass fired power plants based on this technology have been increased within the last years. The favourable characteristics of ORC make them suitable for being integrated in applications like solar desalination with reverse osmosis system, waste heat recovery from biogas digestion plants or micro-CHP systems. In this paper, the state of the art of ORC applications will be presented together with innovative systems which have been simulated in a process simulation environment using experimental data. The results of the simulation like efficiencies, water production rates or achievable electricity production cost will be presented and discussed.

F.Herbele & D. Bruggemann [24] In this study the option of combined heat and power generation was considered for geothermal resources at a temperature level below 450 K. Series and parallel circuits of an Organic Rankine Cycle (ORC) and an additional heat generation were compared by second law analysis. Depending on operating parameters criteria for the choice of the working fluid were identified. The results show that due to a combined heat and power generation, the second law efficiency of a geothermal power plant can be significantly increased in comparison to a power generation. The most efficient concept is a series circuit with an organic working fluid that shows high critical temperatures like isopentane. For parallel circuits and for power generation, fluids like R227ea with low critical temperatures are to be preferred.

J.P. Roy, M.K. Mishra, A. Mishra [25] Parametric optimization and performance analysis of a waste heat recovery system based on Organic Rankine Cycle, using R-12, R-123 and R-134a as working fluids for power generation have been studied. The cycles are compared with heat source as waste heat of flue gas at 140 °C and 312 Kg/s/unit mass flow rate at the exhaust of ID fans for 4 × 210 MW, NTPC Ltd. Kahalgaon, India. Optimization of turbine inlet pressure for maximum work and efficiencies of the system along the saturated vapour line and isobaric superheating at different pressures has been carried out for the selected fluids. The results show that R-123 has the maximum work output and efficiencies among all the selected fluids. The Carnot efficiency for R-123 at corrected pressure evaluated under similar conditions is close to the actual efficiency. It can generate 19.09 MW with a mass flow rate of 341.16 Kg/s having a pinch point of 5 °C, First law efficiency of 25.30% and the Second law efficiency of 64.40%. Hence selection of an Organic Rankine Cycle with R-123 as working fluid appears to be a choice system for utilizing low-grade heat sources for power generation.

J.Li, G.Pei, Y.Z.Li, J.Ji [26] With the scaling down of the Organic Rankine Cycle (ORC), the engine shaft power is not only determined by the enthalpy drop in the expansion process but also the external heat loss from the expander. Theoretical and experimental support in evaluating small-scale expander heat loss is rare. This paper presents a quantitative study on the convection, radiation, and conduction heat transfer from a kW-scale expander. A mathematical model is built and validated. The results show that the external radiative or convective heat loss coefficient was about 3.2 or 7.0 W=K$m2 when the ORC operated around 100 _C. Radiative and convective heat loss coefficients increased as the expander operation temperature increased. Conductive heat loss due to the connection between the expander and the support accounted for a large proportion of the total heat loss. The fitting relationships between heat loss and mean temperature difference were established. It is suggested that low conductivity material be embodied in the support of expander. Mattress insulation for compact expander could be eliminated when the operation temperature is around 100 _C.

S. Quoilin et al [27] Recent interest in small-scale solar thermal combined heat and power (CHP) power systems has coincided with demand growth for distributed electricity supplies in areas poorly served by centralized power stations. One potential technical approach to meeting this demand is the parabolic trough solar thermal collector coupled with an organic Rankine cycle (ORC) heat engine. The paper describes the design of a solar organic Rankine cycle being installed in Lesotho for rural electrification purpose. The system consists of parabolic though collectors, a storages tank, and a small-scale ORC engine using scroll expanders. A model of each component is developed taking into account the main physical and mechanical phenomena occurring in the cycle and based on experimental data for the main key components. The model allows sizing the different components of the cycle and evaluates the performance of the system. Different working fluids are compared, and two different expansion machine configurations are simulated (single and double stage).

N. Yamada, M.N.A. Mohamad, T.T. Kien [28] HFO_1234yf is a new refrigerant with a nearly zero global warming potential (GWP) and ozone depletion potential (ODP); it exhibits thermodynamic properties similar to HFC_134a. The potential of HFO_1234yf as a working fluid for organic Rankine cycles (ORC) is elucidated through a first-order simulation. A fundamental thermodynamic model of ORC with five types of cycles trilateral, saturated, superheated, sub-critical, and supercritical used to compare the thermal efficiency of HFO1234yf with that of other working fluids. HFO_1234yf was found to offer a thermal efficiency that was comparable to that of HFC_134a. This paper provides a useful map that clearly shows the best possible thermal efficiency among the five types of cycles for various expander inlet and condensation temperatures. The highest thermal efficiency range (8.8%e11.4%) was obtained when the supercritical ORC was used at an expander inlet temperature of 170 _ 10 _C and a condensation temperature range of 20e40 _C for the given pump and expander efficiency. It is concluded that HFO_1234yf is a potential working fluid for ORC applications, especially for those with low- to medium-temperature heat sources.

F.A. Al-Sulaiman, I. Dincer, F. Hamdullahpur [29] This part I of the study presents the thermoeconomic optimization formulations of three new trigeneration systems using organic Rankine cycle (ORC): SOFC-trigeneration, biomass-trigeneration, and solar-trigeneration systems. A thermoeconomic modeling is employed using the specific exergy costing (SPECO) method while the optimization performed using the Powell’s method to minimize the product cost of trigeneration (combined, cooling, heating, and power). The results help in understanding how to apply the thermoeconomic modeling and thermoeconomic optimization to a trigeneration system.

F.A. Al-Sulaiman, I. Dincer, F. Hamdullahpur [30] In this part II of the study, three new trigeneration systems are examined. These systems are SOFC-trigeneration, biomass-trigeneration, and solar-trigeneration systems. This study reveals that the maximum trigeneration-exergy efficiencies are about 38% for the SOFC-trigeneration system, 28% for the biomass-trigeneration system and 18% for the solar-trigeneration system. Moreover,the maximum cost per exergy unit for the SOFC-trigeneration system is approximately 38 $/GJ, for the biomass-trigeneration system is 26 $/GJ, and for the solar-trigeneration system is 24 $/GJ. This study reveals that the solar trigeneration system offers the best thermoeconomic performance among the three systems. This is because the solar-trigeneration system has the lowest cost per exergy unit. Furthermore, the solar trigeneration system has zero CO_2 emissions and it is based on a free renewable energy source.

2.2 Objectives of Present Work :

- The performances of IHE in subcritical and supercritical ORC systems are explored.
- The paper has set a new model and taken acid dew point into consideration.
- A modified maximum heat exchange in IHE calculating method has been given.
- A new approach calculating pinch point position has been put forward.
- It gives useful insights for design and operation in applications of IHE to ORC.

CHAPTER – 3

SYSTEM DESCRIPTION

3.1 Subcritical ORC

Fig. 3.1 shows that the basic configuration of ORC system which is composed of an evaporator , an expansion device, a condenser, a pump and a cooling cycle. The condenser and the evaporator are both described as three-stage models by Quoilin and Wei. The thermodynamic processes of the subcritical ORC system corresponding to the numbers as depicted in Fig. 3.1 are illustrated on T-S diagram in Fig. 3.2. The curve which is connecting the points 5, 9, 10 and 4 is known as the saturation curve. The quantities P_i, T_i, s_i, and h_i denote respectively the pressure, the temperature, specific entropy, specific enthalpy at state point i.

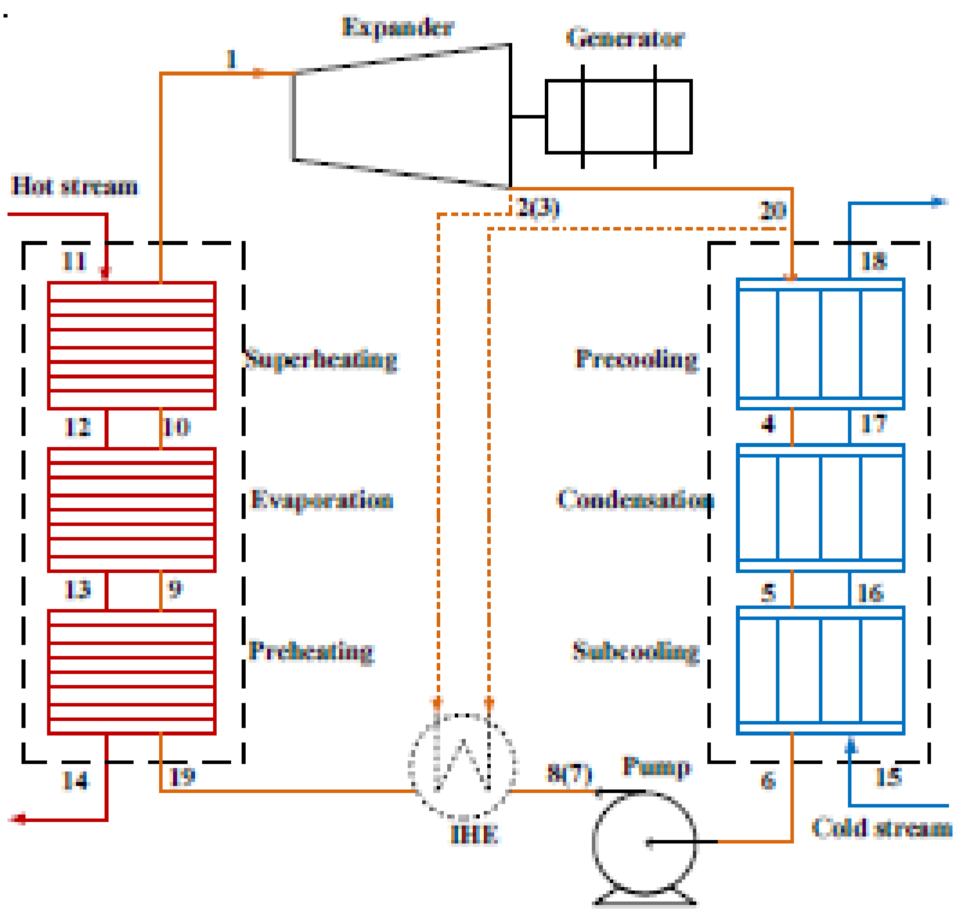

Fig. 3.1 Schematic diagram of Subcritical ORC System with IHE

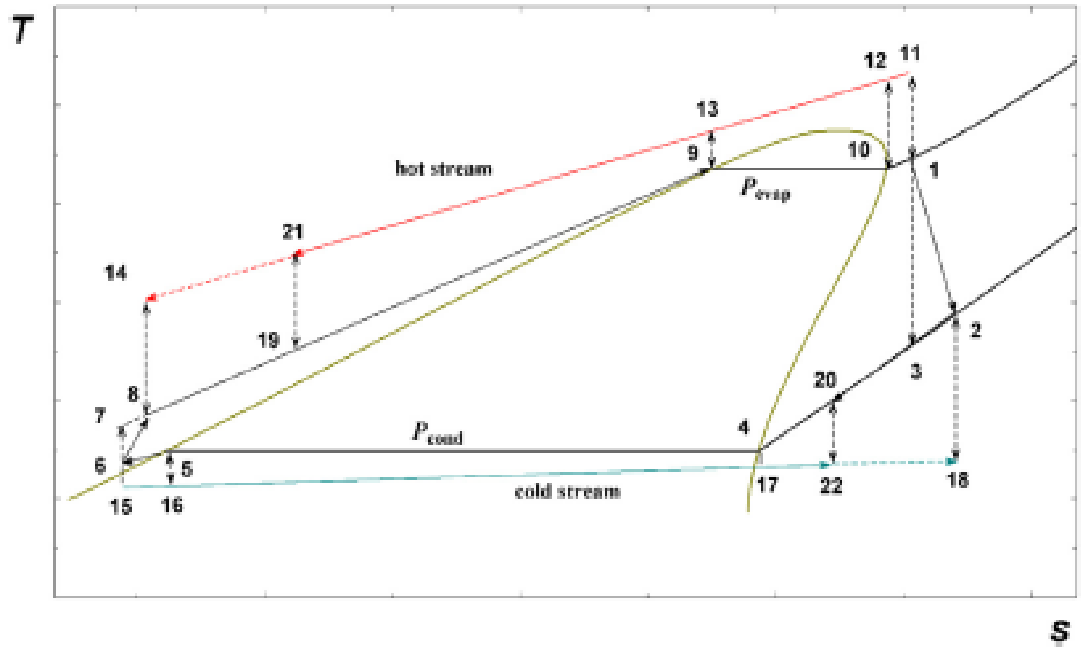

Fig. 3.2 T-S diagram of the Subcritical ORC System

At point 1, the working fluid having temperature $T_1 = T_{evap} + T_{sup}$ and the pressure being P_1 is the superheated vapour from where it enters the expander and undergoes an expansion until reaches at point 2 where it is having pressure P_2 and temperature T_2. During the expansion, the expander delivers the work W_T. At point 2, the working fluid enters into the condenser in which it releases the specific heat Q_{cond} to the cold stream and returns to the subcooled state point 6 at constant pressure. At the state point 6 in cycle, the working fluid is the subcooled liquid with temperature $T_6 = T_{cond} - T_{sub}$ and the corresponding vapour pressure is $P_6 = P_2$, both the temperature and the pressure being the lowest ones in the Organic Rankine Cycle. Then the fluid undergoes adiabatic compression and reaches the pressure $P_8 = P_1$ which is the highest pressure in the Organic Rankine Cycle and the temperature being T_8 at the state point 8. The compression requires work W_p. From the point 8, the liquid flows through the evaporator at the constant pressure till it reaches at its overheated temperature which is the highest temperature in the Organic Rankine Cycle at state point 1. Processes 1-2 and 6-8 are real ones whereas the processes 1-3 and 6-7 are ideal cases in which $s_1 = s_3$, $s_6 = s_7$.

When IHE is proposed, the liquid working fluid before entering the evaporator exchanges heat with the gas working fluid coming out of expander in the IHE. The liquid working fluid gets heated from T_8 to T_{19} whereas the gas working fluid gets cooled from T_2 to T_{20}.

As it is very well known that if we know any two parameters out of the four parameters T_i, P_i, h_i, s_i of the state point than the other two parameters can be determined. And for doing all these thermodynamic calculations use of Engineering Equation Solver Ver. 8 is done. This software is easy and flexible to use and has the advantage of ready to use optimization techniques and fluid properties. It makes use of the same equation of state as used in REFPROP-NIST. This is a versatile software and a very useful tool.

3.2 Supercritical ORC

In a supercritical organic rankine cycle, the process that occurs in evaporator is a complex two phase process, simply implying that a three stage model cannot be used. However the other facilities are same with almost no differences among them. The schematic diagram is similar to the diagram shown in Fig. 3.3 with the only difference that an evaporator is replaced by a one-stage model. Fig. 3.3 shows the supercritical system with IHE and the thermodynamic processes on the T-S diagram are shown in Fig. 3.4.

Due to the complexity of two phase change, there occurs a difficulty in locating the position of pinch point. In a subcritical cycle, by default the pinch point is assumed between point 9 and point 13, which in most cases is tenable, and the added condition that the temperature difference between T_{11} and T_1 is too large. However, in a supercritical cycle, trends of working fluid evaporation progress and constant pressure lines of the heat source are similar, hence it becomes difficult to identify the pinch point position qualitatively.

Some previous studies did their simulation by assuming the outlet temperature of the heat source or by assuming the location of pinch point, or setting and defining effectiveness of internal heat exchanger at a constant value of €. It simplified the calculation while it would also lead to the consequence that the temperature difference of the pinch point would be less or more than the value which is already set, which is not in conformity with the real condition in heat exchanger. The given minimum heat transfer temperature difference ΔT_{pp} is used for calculating the effectiveness of Internal heat exchanger i.e. €, but the derivation of reversing process is not real. Here, an iteration method is suggested through which the pinch point position in the supercritical cycle can be located. Once ΔT_e is set, the working fluid's evaporation zone is divided equally into N number of parts in the direction of entropy.

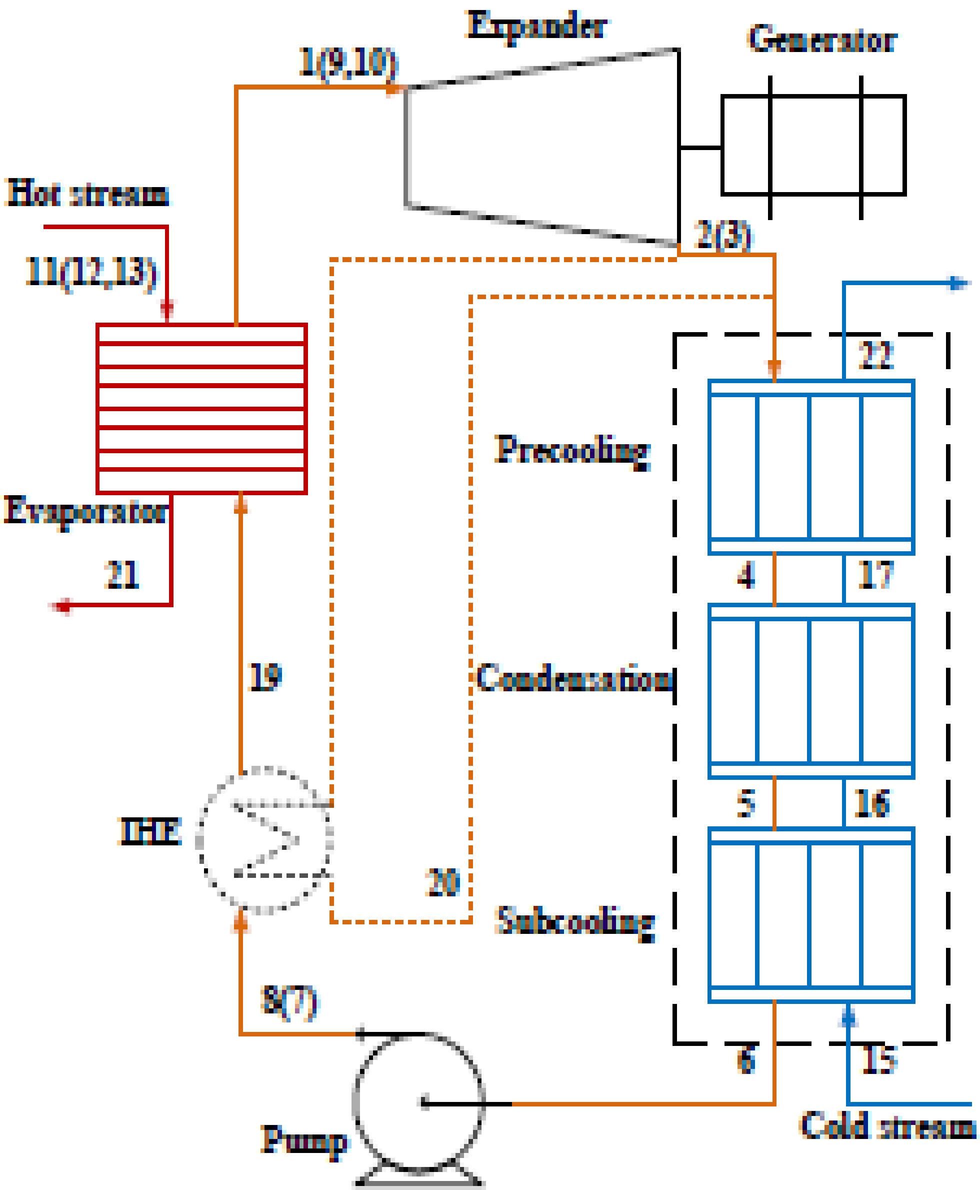

Fig. 3.3 Schematic diagram of supercritical ORC System with IHE

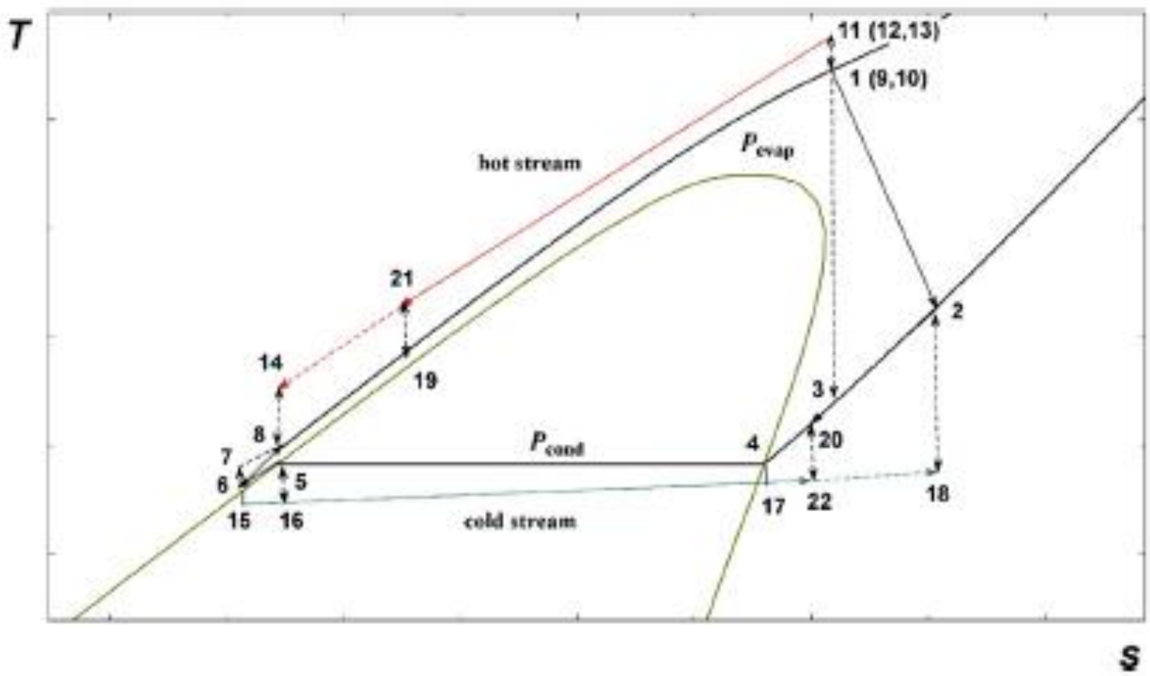

Fig. 3.4 T-S Diagram of Supercritical ORC

The lowering of the outlet temperature of heat source is used to find the minimum gap between the working fluid evaporation isobar and the heat source isobar. When this equals ΔT_{pp}, the outlet temperature can be determined and the position of pinch point is N. Fig. 3.5 demonstrates this method and Fig. 3.6 is the flow diagram which is written as per this method. The application of this approach includes the calculation of outlet temperature of heat source in supercritical systems, which strictly suggests that ΔT_{pp} should be at the set value of 10^{o} C.

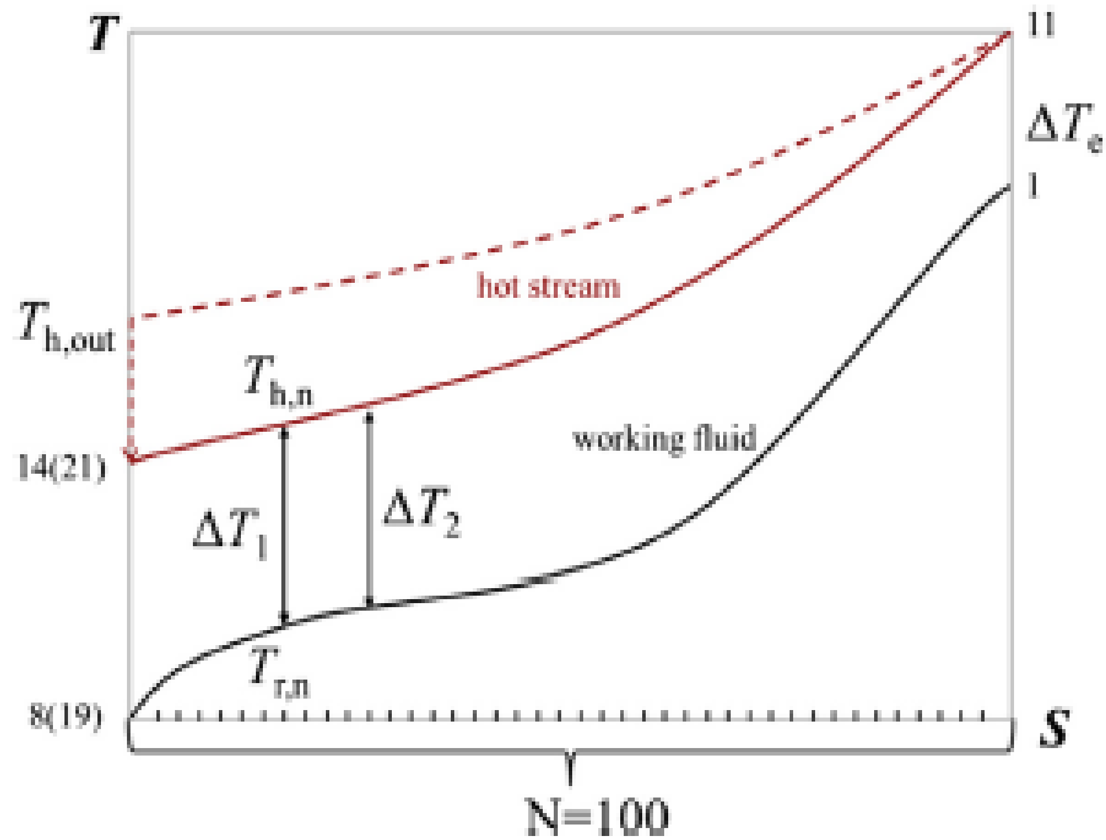

Fig. 3.4 Schematic diag. of Iteration Method for Supecritical ORC Pinch Point Locating

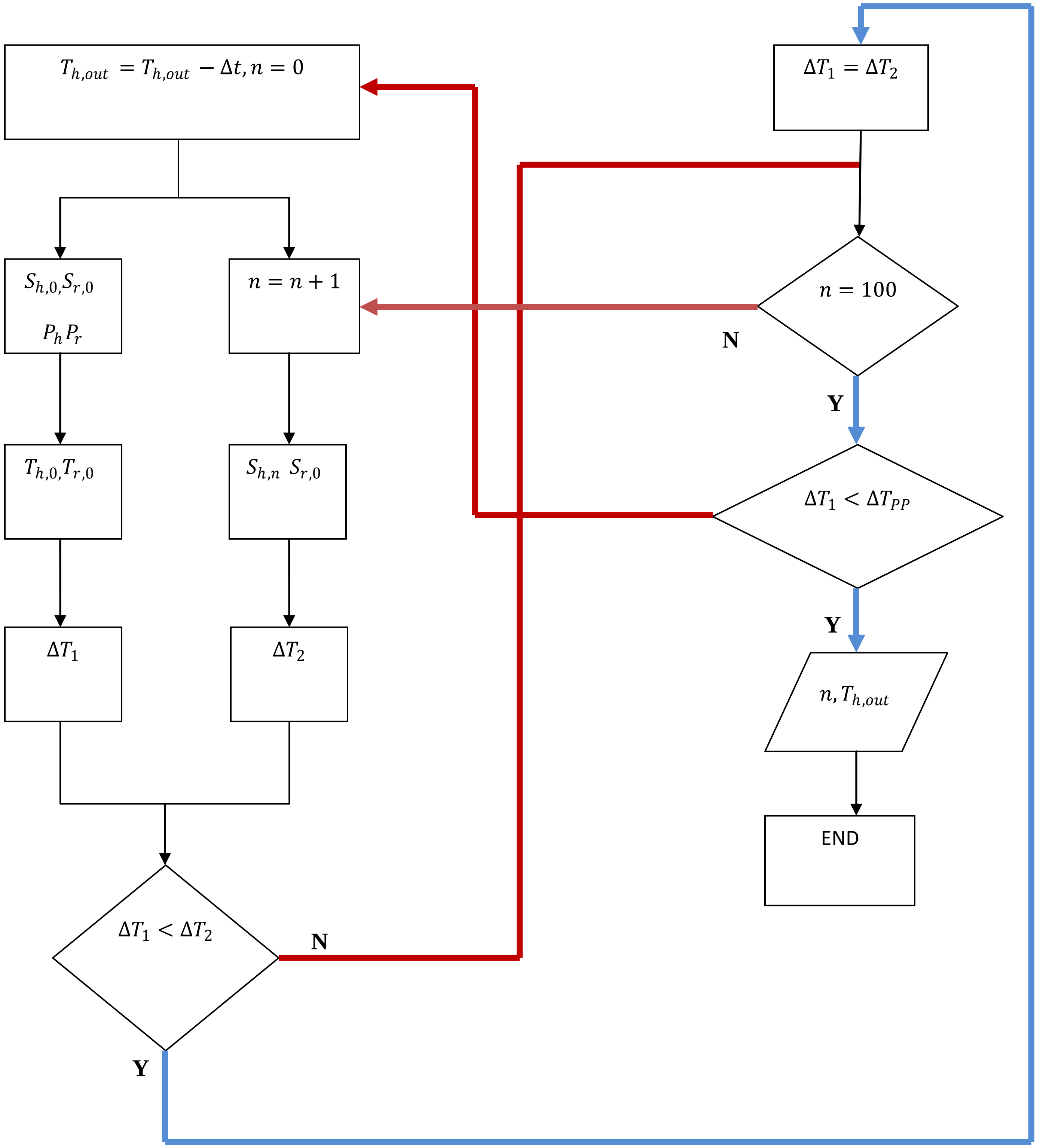

Fig. 3.6 Flow Chart of the Pinch Point and Hot Stream Outlet Temperature Calculation Program

3.3 Global Model

The system safety, economic efficiency of ORC system and environmental protection is greatly influenced by the choice of working fluids. The widely used working fluids includes the applicable fluids with good thermodynamics, safety and economic characteristics, environmental protection, such as fluids with low liquid specific heat, viscosity, toxicity, flammability, ozone depletion potential, global warming potential, and price.

Based upon above criterion, the working fluids R600 and R123 are chosen for subcritical and supercritical cases separately. The basic environmental and thermodynamic properties of the above mentioned fluids are shown in Table 3.1.

Working Fluid	t_{cr} (^{o}C)	P_{cr} (MPa)	t_{fp} (^{o}C)	t_{bp} (^{o}C)	Safety	ODP	GWP
R123	183.7	3.668	-107.2	27.79	B1	0.012	120
R600	152	3.796	-138.3	-0.5208	A3	0	-20

Table 3.1 Basic thermodynamic and environmental properties of R123 and R600

The following assumptions are made in the analysis in order to simplify the computing and complicacy which are neither the key points nor they affect the accuracy of results coming in the model suggested.

- The whole system is under equilibrium state.
- The specific heat of the hot stream is replaced with one the one of hot air.
- The heat losses and leakages through the pipeline and components are ignored.

Some unchanged parameters involved in the calculations are shown in Table 3.2

Parameter	**Sign**	**Unit**	**Value**
Environment Temperature	T_o	^{o}C	15
Low Grade Heat flow temperature	$T_{h,in}$	^{o}C	200
Cooling water temperature	$T_{c,in}$	^{o}C	15

Condensation Temperature	T_{cond}	^oC	30
Low Grade Heat Mass Flow Rate	$q_{v,h}$	Kg/s	1
Superheat Degree	T_{sup}	^oC	3
Degree of Subcooling	T_{sub}	^oC	1
Pinch Point	ΔT_{pp}	^oC	10
Temperature difference at hot stream inlet of Evaporator	ΔT_e	^oC	15
Acid dew point	T_{ad}	^oC	110
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
Efficiency of cycling pump	η_{cpump}	1	0.8
Total circulation loop Flow resistance	dP_{if}	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 that are used for the different steps of analysis to be performed. They are presented in order of the showed system performances.

4.1 Energy Analysis

The first law of thermodynamics is the basis for the energy analysis. The final results includes the net output and the thermal efficiency. As already mentioned in the assumptions, their value is dependent only on evaporation pressure i.e. $P_{evap} = P_1 = P_{8.}$ Following are the set of equations for different components.

4.1.1 Subcritical ORC

The equation for Evaporator:

$$Q_{evap} = \dot{m}_h (h_{11} - h_{14}) = \dot{m}_r (h_1 - h_8) \qquad (1)$$

The equation for the expander:

$$\eta_{T,I} = h_2 - h_1 / h_3 - h_1 \qquad (2)$$

$$w_T = \eta_{T,m} * (h_1 - h_2) \qquad (3)$$

where $\eta_{T,m}$ is the mechanical efficiency of expander.

The equation for Condenser:

$$Q_{cond} = \dot{m}_c * (h_{18} - h_{15}) = \dot{m}_r * (h_2 - h_6) \qquad (4)$$

The Equation for fluid pump:

$$W_p = \int vdP \text{ (limits from } P_6 \text{ to } P_8) / \eta_P \quad (5)$$

Considering the effect of small liquid compressibility, the above is simplified as

$$W_p = \int vdP \text{ (limits from } P_6 \text{ to } P_8) / \eta_P = (v_8+v_6/2) * (P_8-P_6) / \eta_P \quad (6)$$

$$P_8 - P_6 = dP_{rf} + (P_{evap} - P_{cond}) \quad (7)$$

$$\eta_P = \eta_{P,i} * \eta_{P,m} \quad (8)$$

$$\eta_{P,i} = (h_6 - h_7) / (h_6 - h_8) \quad (9)$$

Equation for the cycle Pump :

$$W_{cpump} = dP_{cf} \{(v_{15} + v_{16}) / 2\} / \eta_{cpump} \quad (10)$$

Equation for the net system output:

$$\dot{W}_{sys} = -(-\dot{m}_{r*}w_T + \dot{m}_{r*}w_p + \dot{m}_{c*}w_{cpump}) \quad (11)$$

Equation for finding the Cycle Thermal Efficiency :

$$\eta_{th} = \dot{W}_{sys} / Q_{evap} \quad (12)$$

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

$$Q_{evap} = \dot{m}_h * (h_{11} - h_{21}) = \dot{m}_r * (h_1 - h_{19}) \quad (13)$$

$$Q_{cond} = \dot{m}_c * (h_{22} - h_{15}) = \dot{m}_r * (h_{20} - h_6) \quad (14)$$

For the Internal Heat Exchanger :

$$h_{20} - h_2 = h_8 - h_{19} \quad (15)$$

4.1.2 Supercritical ORC

Since all the used formulas in supercritical case are same as that of the subcritical case hence suggesting that the energy analysis in both the cases are almost same. As it is observed in the fig. 4.1, at the critical point the specific heat of working fluid changes dramatically in case of supercritical cycle. These variation characteristics are shown in the temperature range that is considered in simulation which is assumed constant in case of subcritical cycle. Then there is required a modification in the cycle to take these variations into account. The following equation represents the heat between two streams in internal heat exchanger :

$$Q_{IHE} = \dot{m}_h * (h_{19} - h_8) = \dot{m}_r * (h_2 - h_{20}) \quad (16)$$

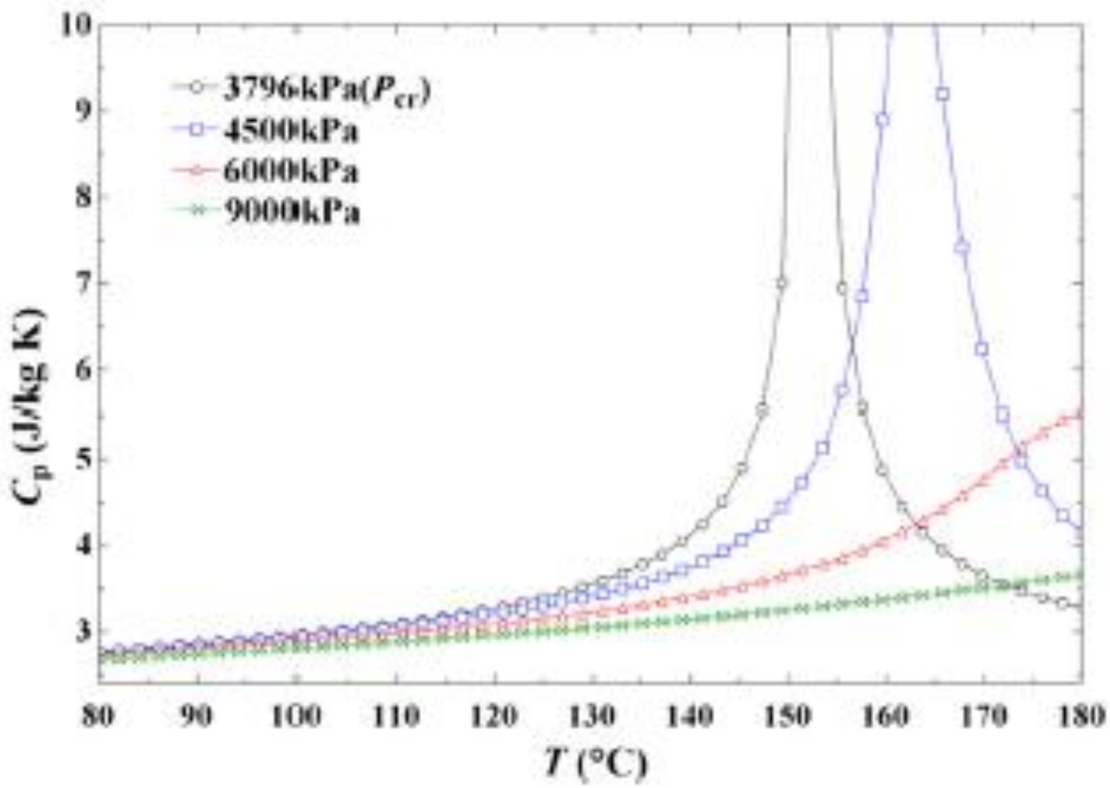

Fig. 4.1 Isobaric specific heat of R600 versus temperature at different pressures

Either of the following situation occurs when minimum heat transfer temperature difference i.e. ΔT_{pp} is taken into account which in turn depends on which f the two streams in internal heat exchanger has smaller heat capacity. The two situations includes :

Either,

$T_{20} \rightarrow T_8 + \Delta T_{pp}$ i.e. T_{20} approaches towards $T_8 + \Delta T_{pp}$

Or,

$T_{19} \rightarrow T_2 - \Delta T_{pp}$ i.e. T_{19} approaches towards $T_2 - \Delta T_{pp}$

Thus, the smaller of the following two quantities gives the maximum heat exchange i.e. $Q_{IHE,max}$:

$$Q_{IHE} = \dot{m}_r * (h_2 - h_{20}) \text{ when } T_{20} \rightarrow T_8 + \Delta T_{pp} \quad (17)$$

$$Q_{IHE} = \dot{m}_h * (h_{19} - h_8) \text{ when } T_{19} \rightarrow T_2 - \Delta T_{pp} \quad (18)$$

Also, the condition i.e. $T_2 \geq T_8 + \Delta T_{pp}$ should also be taken into account under the same constraint.

4.2 Exergy Analysis

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

The exergy analysis can be done by calculating the specific flow exergy at exit and entry of sources and also at each state point in cycle. The following expression gives the quantity :

$$e = h - h_0 + T_0 (s_0 - s) \quad (19)$$

where the subscript 0 represents the environmental conditions which are considered to be 15^o C and atmospheric pressure. Following are the equations for different components :

Equation for Evaporator :

$$\acute{\dot{E}}_{d,evap} = \dot{m}_h * (e_{11} - e_{14}) + \dot{m}_r * (e_8 - e_1) \tag{20}$$

Equation for Expander :

$$\acute{\dot{E}}_{d,T} = \dot{m}_r * (e_1 - e_2) - \dot{W}_T \tag{21}$$

Equation for condenser :

$$\acute{\dot{E}}_{d,cond} = \dot{m}_r * (e_2 - e_6) + \dot{m}_c * (e_{15} - e_{18}) \tag{22}$$

Equation for working pump :

$$\acute{\dot{E}}_{d,P} = \dot{W}_P + \dot{m}_r * (e_6 - e_8) \tag{23}$$

Equation for cooling cycle pump :

$$\acute{\dot{E}}_{d,cpump} = \dot{W}_{cpump} + \dot{m}_c * (e_{15} - e_{18}) \tag{24}$$

Expression for system total exergy destruction rate :

$$\acute{\dot{E}}_{d,tot} = \acute{\dot{E}}_{d,evap} + \acute{\dot{E}}_{d,cpump} + \acute{\dot{E}}_{d,P} + \acute{\dot{E}}_{d,cond} + \acute{\dot{E}}_{d,T} \tag{25}$$

The overall exergy efficiency is calculated as :

$$\eta_{ex} = 1 - \{\acute{\dot{E}}_{d,tot} / \dot{m}_h * (e_{11} - e_{14})\} \tag{26}$$

If internal heat exchanger is considered, then Eqn. 20 and 22 is changed to

$$\acute{\dot{E}}_{d,evap} = \dot{m}_r * (e_{19} - e_1) - \dot{m}_h * (e_{21} - e_{11}) \tag{27}$$

$$\acute{\dot{E}}_{d,cond} = \dot{m}_r * (e_{20} - e_6) - \dot{m}_c * (e_{22} - e_{15}) \tag{28}$$

Expression for internal heat exchanger :

$$\acute{\dot{E}}_{d,IHE} = \dot{m}_r * (e_8 - e_{19}) - \dot{m}_r * (e_{20} - e_2) \quad (29)$$

The equations 25 and 26 are modified as :

$$\acute{\dot{E}}_{d,tot} = \acute{\dot{E}}_{d,evap} + \acute{\dot{E}}_{d,cpump} + \acute{\dot{E}}_{d,P} + \acute{\dot{E}}_{d,cond} + \acute{\dot{E}}_{d,T} + \acute{\dot{E}}_{d,IHE} \quad (30)$$

$$\eta_{ex} = 1 - \{\acute{\dot{E}}_{d,tot} / \dot{m}_h * (e_{11} - e_{21})\} \quad (31)$$

CHAPTER – 5

RESULTS AND DISCUSSIONS

5.1 Subcritical ORC

It I observed from the Fig. 5.1 that the power output of subcritical system is not affected by the internal heat exchanger and the maximum power obtained is $\dot{W}$ = 11.3 KW at an evaporator pressure P_{evap} = 1374 KPa. As the evaporating pressure increases, the power output increases and after achieving the maximum value it starts decreasing. In this process, the focus is to be put on that the power output of low evaporation pressure section is less than that of high pressure section. Fig. 5.2, 5.3, 5.4 shows that within the simulated evaporating pressure range, the exergy efficiency and thermal efficiency are at its peak whereas the outlet temperature of heat source is monotonously increasing. When the internal heat exchanger is used, the same change trend is observed in the three indicators, however they get improved at different levels.

Acid dew point T_{ad} is to be regarded in case of utilization of low grade heat especially in case of industrial waste heat recovery. Consideration of T_{ad} introduces another function of IHE. In Fig. 5.4 the acid dew point is represented by red line such that it have two intersections with curves of heat source outlet temperature i.e. at 1993 KPa and 2323 KPa. There are two ranges in the right part of two intersections Range A (1993-3600 KPa) and Range B (2323-3600 KPa) which respectively represent the range of evaporating pressure in accordance with the requirement $T_{h,out} > T_{ad}$, as illustrated also in Fig. 5.1, 5.2, 5.3. The achievable net output, exergy efficiency and thermal efficiency can clearly be interpreted by the above steps. Fig. 5.1 shows that the division of range results in different maximum output of 9.264 KW and 10.47 KW with the output gap of $\Delta\dot{W}$ = 0.846 KW. The maximum net output of the system I indirectly increased by this. The division of range also changes the ranges of exergy and thermal efficiency. Fig. 5.2 shows that the achievable maximum values of thermal efficiency i.e. 12.4% and 11.03% are not influenced by the division of acid dew point. Whilst, after division, the maximum value reached by Range A is 40.15% at an evaporator pressure P_{evap} = 2350 KPa on the original curve whereas it is not so for Range B, it reaches 37.91% at P_{evap} = 2132 KPa which is displayed in Fig. 5.3

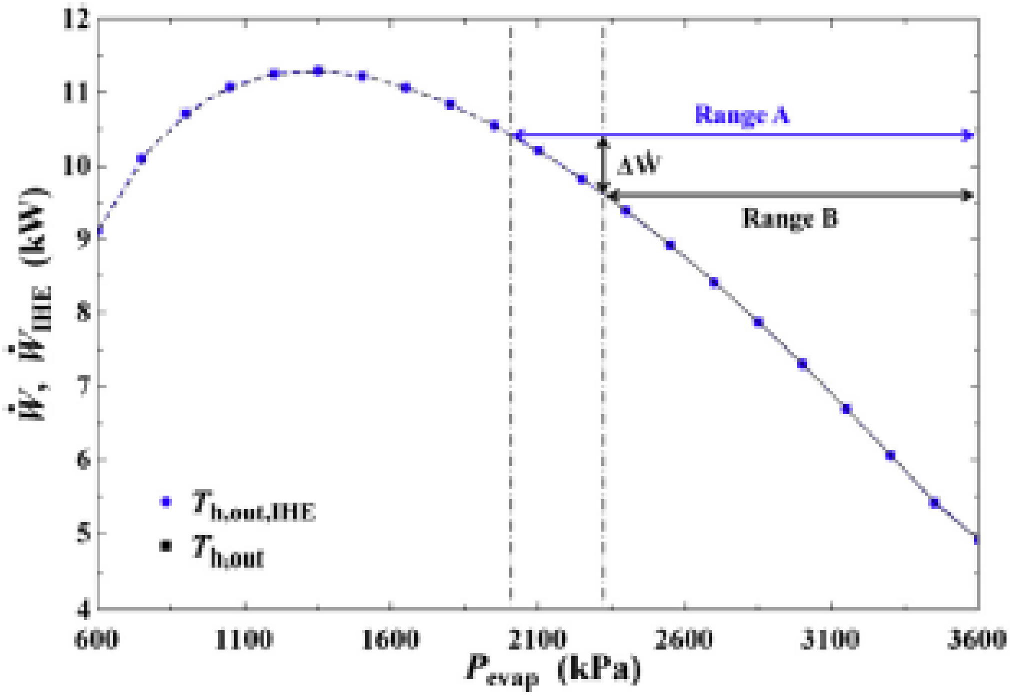

Fig. 5.1 System Net Power Output of Subcritical ORC Operating on R123

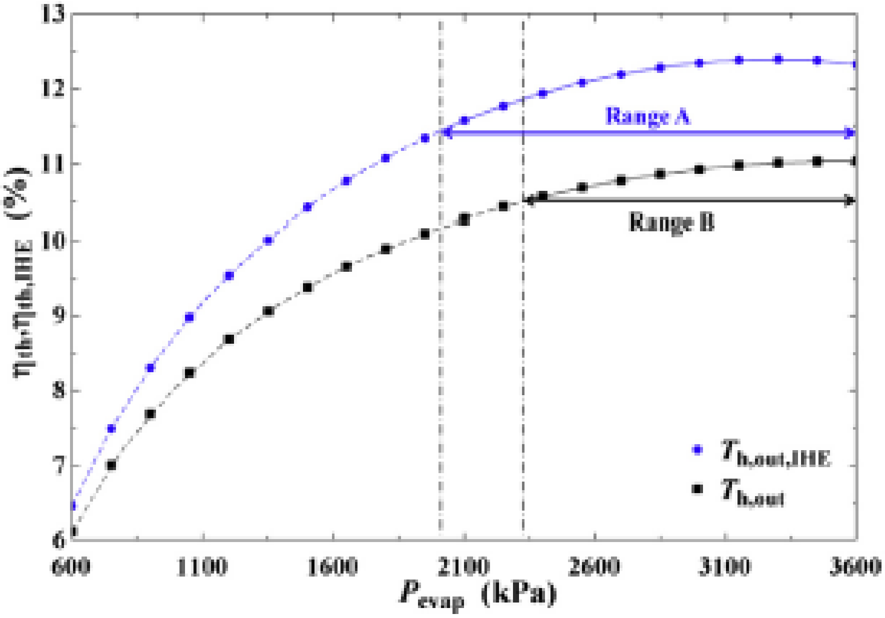

Fig. 5.2 Thermal Efficiency of Subcritical ORC Operating on R123

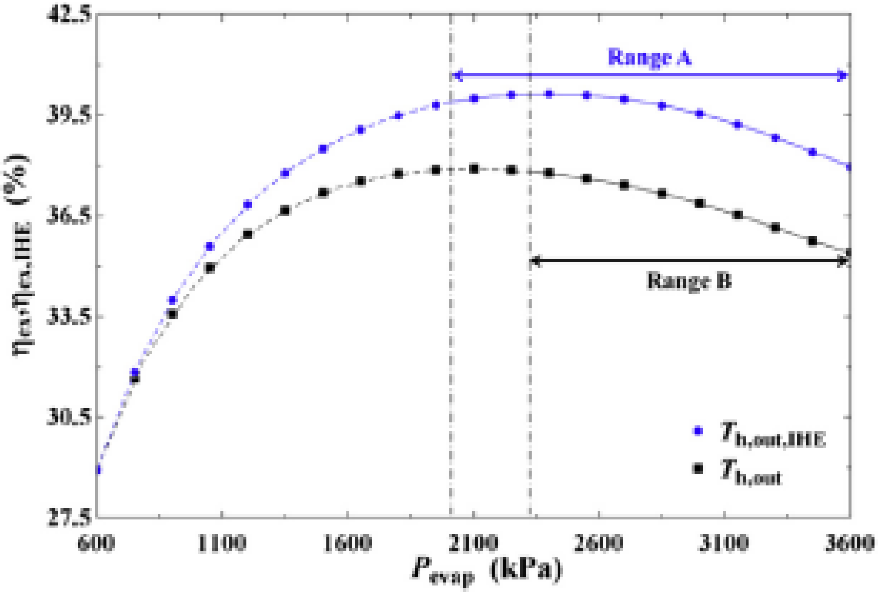

Fig. 5.3 Exergy Efficiency of the Subcritical ORC Operating on R123

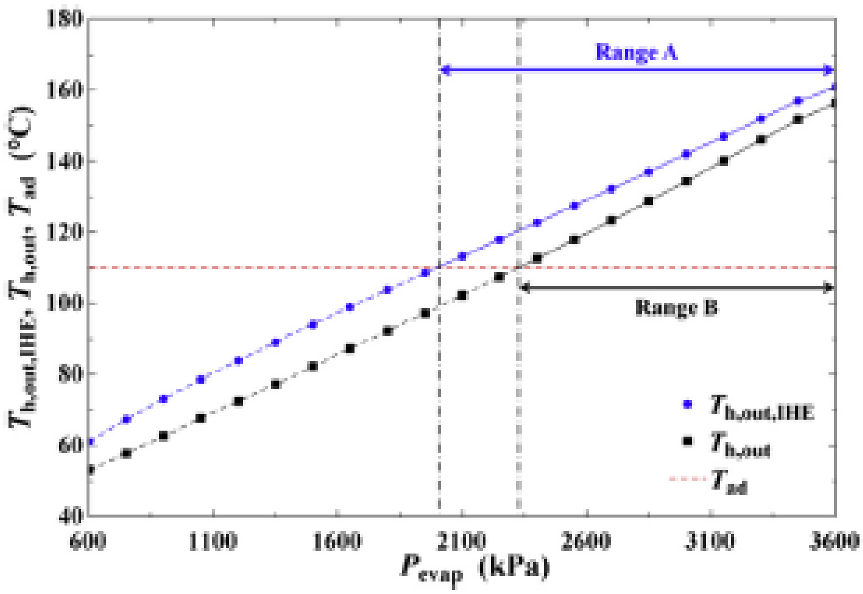

Fig. 5.4 Outlet Temperature of the Hot Stream Operating on R123

5.2 Supercritical ORC

A supercritical organic rankine cycle system is a different system from a subcritical system. In a supercritical system, using the constraint $T_2 \geq T_8 + \Delta T_{pp}$ a Range C is derived. As the evaporating pressure is very high as compared to that in subcritical case, T_8 reaches $T_2 - \Delta T_{pp}$ or even higher than this, ensuring that the system performance would not be worsened by the IHE. In this Range, the thermal efficiency, exergy efficiency and the outlet temperature are improved by the use of IHE as shown in Fig. 5.6, 5.7, 5.8. Also, the system net output does not change and remains same as in subcritical cycle, shown in Fig. 5.5 being the maximum $\dot{W}$ = 13.34 KW at evaporator pressure P_{evap} = 7500 KPa. The system performance deteriorates when the internal heat exchanger is used out of this range, suggesting that it is not applicable outside this range. The maximum values of thermal efficiency, exergy efficiency, and the outlet temperature of heat source lies near the low pressure stage in the simulation of supercritical ORC without IHE as depicted in Fig. 5.6, 5.7, 5.8. However, they generally have a decreasing nature with the increase in evaporating pressure. When the system is equipped with internal heat exchanger than the three indicators keeps on monotonous decrease. On the right border of Range C, there exists the break points of each curve turning up and then the curves coinciding with the curves without internal heat exchanger.

When the acid dew point is taken into consideration, Range A gets figured out in the same way as shown in Fig. 5.8. Also, as the supercritical case is much more in match with the heat source than a subcritical case, leading to a lower outlet temperature of heat source in supercritical system, outlet temperature without IHE being lower than the T_{ad}, makes the Range B nonexistent. In other words, only a part of evaporating pressure range i.e. 4000-5080 KPa meet the condition $T_{h,out} > T_{ad}$ by the use of IHE. The system attains the maximum values of indicators $T_{h,out}$ = 119.2^{o} C, η_{ex} = 45%, η_{th} = 13.43% at an evaporator pressure P_{evap} = 4000 KPa and the maximum power output being 11.55 KW at an evaporator pressure P_{evap} = 5080 KPa.

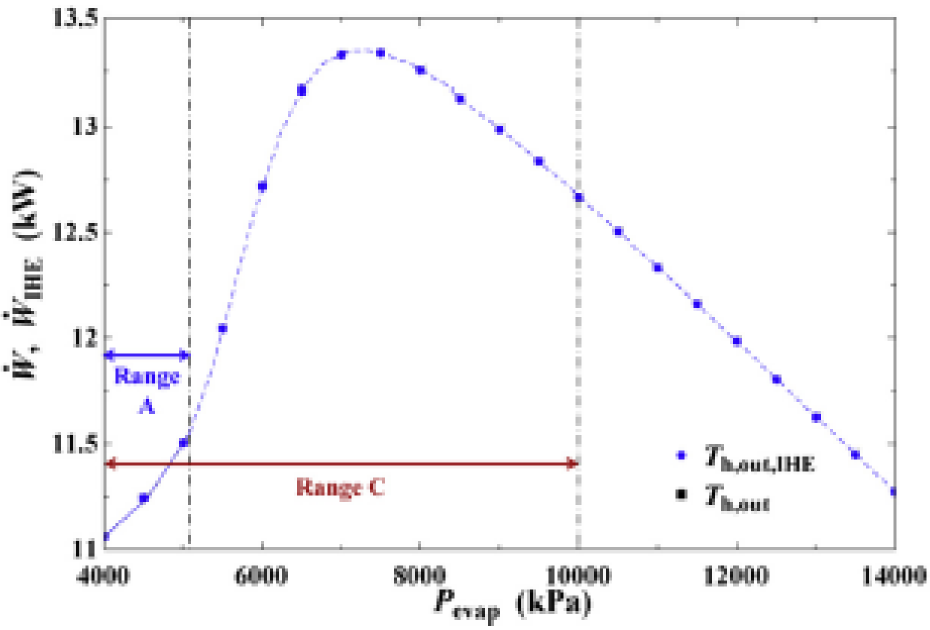

Fig. 5.5 System Net Power Output of supercritical Cycle Operating on R600

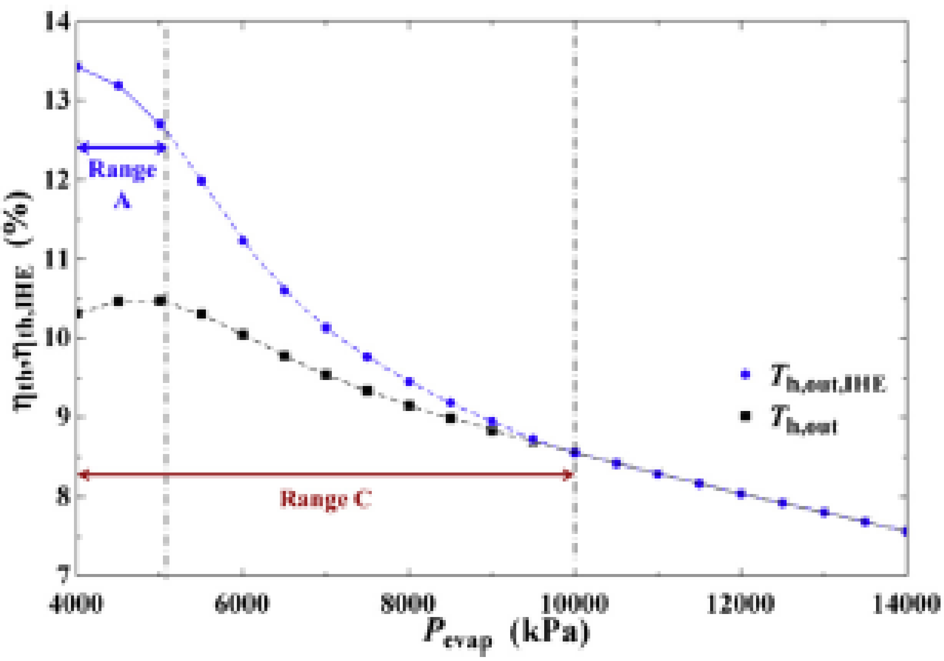

Fig. 5.7

Fig. 5.6 Thermal Efficiency of Supercritical Cycle Operating on R600

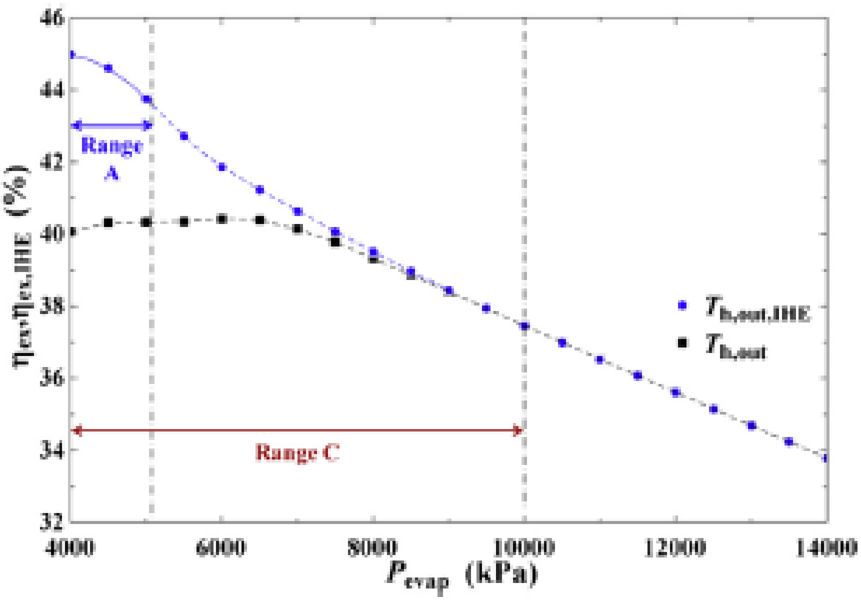

Fig. 5.7 Exergy Efficiency of Supercritical Cycle Operating on R600

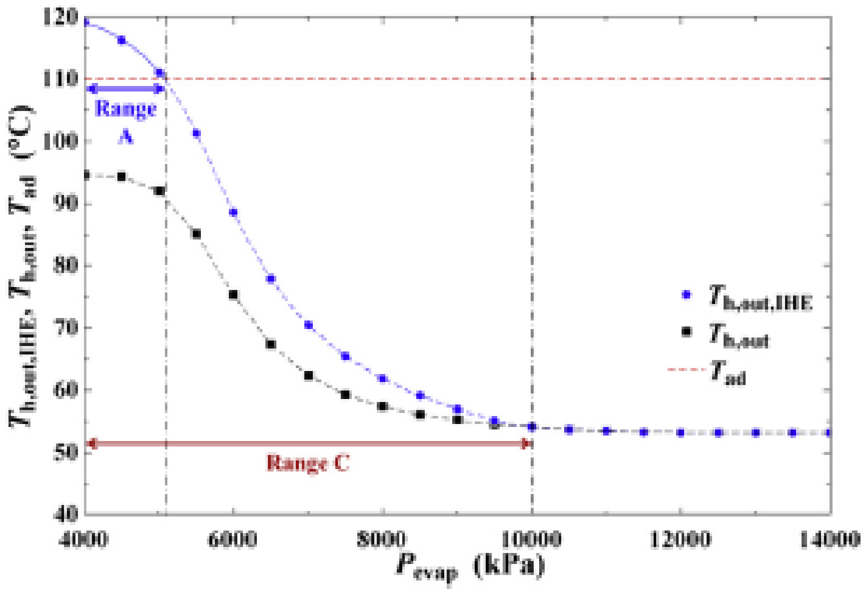

Fig. 5.8 Outlet Temperature of Hot Stream of Supercritical Cycle Operating on R600

CHAPTER – 6

CONCLUSION

In this study, the simulation of subcritical and supercritical organic rankine cycle system with or without internal heat exchanger using respectively R123 and R600 is done which illustrates the impacts of internal heat exchanger on ORC systems. The mass flow rate taken into account for analysis is 1 kg/s and the heat source temperature is taken as 200^o C. The temperature difference of 10^o C is set as the minimum heat transfer temperature difference of heat exchanger. When a supercritical system is modeled, a modified method for maximum heat exchange in an internal heat exchanger is given, which is attributed by the fact that in the supercritical system when working conditions lies near critical point conditions of the working fluid, then the specific heat of the fluid should not be considered as constant, which is different from subcritical system. Also, a new approach is suggested for calculating the location of pinch point and outlet temperature of heat source in subcritical system which specifies that the minimum temperature difference of evaporator for heat transfer should be equal to the set value of 10^0 C. And, when acid dew point is considered, it is observed that the internal heat exchanger improves the net output and justifies the applicability of system. The main consequences are:

- The internal heat exchanger always increases the exergy efficiency, thermal efficiency and outlet temperature of heat source in subcritical ORC system. However, there is no change in the net output of the system which remains the same.
- The supercritical ORC system consists of a Range C in which the thermal efficiency, outlet temperature of heat source and exergy efficiency improves by the application of internal heat exchanger, keeping the system net output to be the same. The system performance degrades when an internal heat exchanger is used outside of this range.
- When acid dew point T_{ad} is considered , the net output of system is increased indirectly in a subcritical system on the application of internal heat exchanger while in supercritical system, the evaporation pressure starts obeying the rule $T_{h,out} > T_{ad}$. It is found that the applicability of system is strengthened by the application of IHE.

6.1 Future Scope

- A comprehensive analysis of different upcoming working fluids comparison is to be done in both subcritical d supercritical systems that are equipped with IHE.
- More economical indicators like UA values and heat exchanger area is to be investigated.

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