EXERGY ANALYSIS OF LOW TO MEDIUM TEMPERATURE ORGANIC RANKINE CYCLE

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In partial fulfillment of the requirement for the award of the Degree of

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STUDENT'S DECLARATION

I hereby certify that the work which is being presented in the major project entitled
"**EXERGY ANALYSIS OF LOW TO MEDIUM TEMPERATURE ORGANIC RANKINE CYCLE**" in partial fulfillment of the requirements for the award of the degree of Master of Technology in Thermal Engineering, submitted to the department of Mechanical Engineering, is an authentic record of my own work carried under the supervision of Dr. Akhilesh Arora, Associate professor of Mechanical Engineering Department, Indira Gandhi Delhi Technical University, Delhi and Sh. Naushad A.Ansari ,Assistant Professor of Mechanical Engineering Department, Delhi Technological University.

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CERTIFICATE

Date:

This is certify that the dissertation entitled "**Exergy Analysis of low to medium Temperature Organic Rankine cycles''** submitted by **Mr. DEEPAK TIWARI (2K10/Thr/M.Tech/PT)** in partial fulfillment for the award of the Degree of Master of Technology in Thermal Engineering of Delhi Technological university, is an authentic record of student's own work carried out by him under my guidance and supervision. He has completed his work with utmost sincerity and diligence.

The work embodied in this major project has not been submitted for the award of any other to the Best of my knowledge.

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Abstract

In the present work HFO-1234yf is used as a Refrigerant &working fluid in refrigeration and air conditioning and Rankine cycle respectively. The first order simulation has been carried-out for the five ORC models, are such as Trilateral, Saturated, Superheated, Sub-critical, and Supercritical. he thermal efficiency of the five ORC model is compared by using various working fluid such as (HFO-1234yf,HFC-134a,HFC-245fa,Ethanol,Isopentane) by varying expander inlet temperature(30-200^oC),at fixed condensation temperature(30^oC) assuming fixed expander isentropic efficiency(η_{is}=0.70) and fixed isentropic pump efficiency(η_p=0.60). The thermal efficiency has been observed maximum for the 1234yf has very good potential for working fluid for ORC application for low to medium temperature. It has zero global warming potential (GWP), zero ozone layer depletion (ODP).

The exergetic η and exergy destruction has also been carried out for these Organic Rankine cycle

CONTENTS

Chapter 3

THERMODYNAMIC MODELLING

Chapter4

RESULT AND DISCUSSION

Chapter 5

LIST OF FIGURES

NOMENCLATURE

HFO	Hydro -fluro -Olefin
HFC	Hydro- fluro-carbon
EES	Engineering equation solver
Ed	Exergy destruction (kw)
η	Efficiency (%)
T	Temperature (^{o}C)

Subscripts

cond	condenser
exp	expander
pump	pump
evap	evaporator
ex	exergetic
in	inlet to expander
1	Pump inlet state
2	Pump exit state
3	Expander inlet state
4	Expander exit state

Greek letters η (efficiency)

CHAPTER 1

INTRODUCTION

1.1 Description of Organic Rankine cycle

Heat recovery using organic Rankine cycles (ORCs) is considered to be an important method among renewable energy processes because of its capability to generate power from the available waste heat and natural heat sources, such as solar radiation, ocean thermal sources, geothermal sources, biomass, and waste heat from fuel combustion or industrial processes.

The word “organic” in ORC implies that the cycle employs organic substances as its working fluid.

In ORC power systems, the working fluid must be selected to maximize thermal efficiency for temperature range of heat source and long-term performance. Typical temperature range for ORC is from low temperatures of about ∼100 ^{o} C to medium temperatures of about∼ 350^{o}C Existing ORC power systems still tend to use refrigerants with relatively high global warming potential (GWP) and ozone depletion potential (ODP) as their working fluids due to limited options for the working fluid.

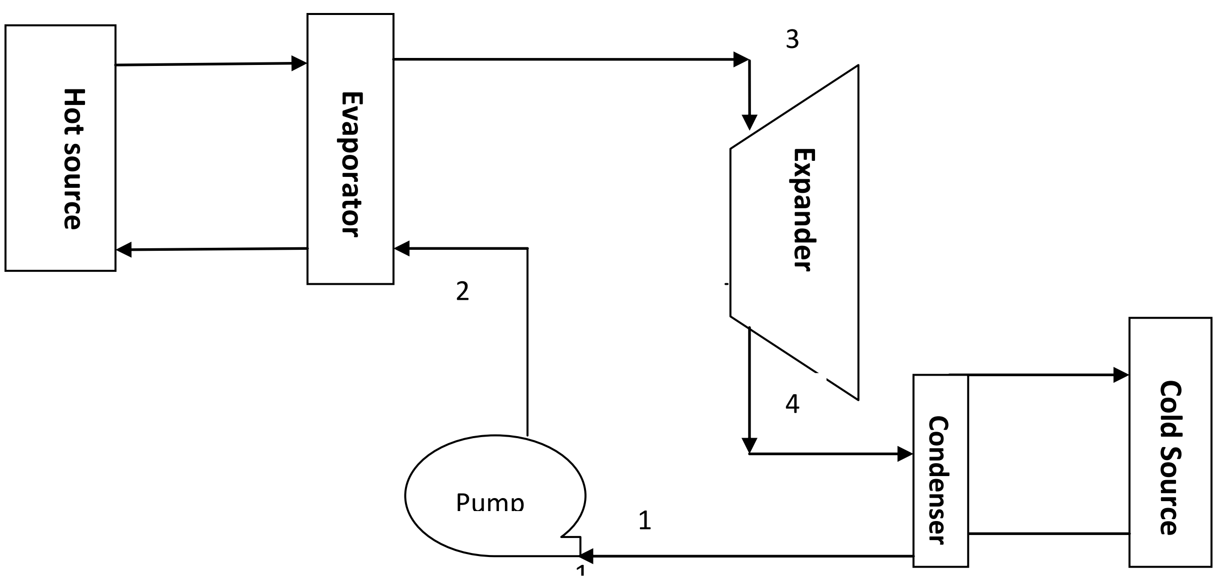


Fig. 1.1 **Basic Organic Rankine Cycle**

In the conversion of low temperature heat into electricity the greatest efficiency is obtained by using a Rankine cycle. A typical low temperature heat source is the waste heat in combustion engines and in industrial processes: hot gases from blast furnaces in the steel industry and from kilns in the ceramics industry, exhaust gases of diesel engines and gas turbines, hot liquids used to cool kilns or furnaces, etc. Organic Rankine cycle (ORC) is a Rankine cycle, where an organic fluid is used instead of water as the working fluid. Particularly in low temperature applications many benefits may be obtained by using ORC instead of steam Rankine process. Many commercial and test plants are made by using organic Rankine cycle. It is estimated that there are about 30 commercial ORC plants built before 1984 with an output over 100kW. The most common fluids used are refrigerants Rll (CC13F), R 113 (C2C13F3), R 114 (C2C12F4), toluene

(C6HsCH3) and fluorinol (CF3CH2OH), or fluor- inol mixed with water. The ORC-plant in general is treated in [1, 2].

1.2 ADVANTAGES

When the heat source is waste heat at a moderate inlet temperature, the best efficiency and highest power output is usually obtained by using a suitable organic fluid instead of water in the Rankine cycle. This is mainly because the specific vaporization heat of organic fluids is much lower than that of water. Thus the organic working fluid "follows" better the heat source fluid to be cooled. An example of this is shown in where the heat source is the exhaust gas of a gas turbine. In the ORC-process the exhaust gas can be cooled to a significantly lower temperature. This means, that more electric power can be produced from a given heat source.

1.3WORKING FLUIDS

As mentioned in the introduction section, five working fluids are used in this study: HFO-1234yf, HFC-134a, HFC-245fa, isopentane, and ethanol. The latter four are representative working fluids used in ORC applications and have been investigated by many other researchers. In addition to the HFC-134a applications mentioned in the introduction, HFC-245fa and iso-pentane are also used in ORC applications to recover low-temperature waste heat from industrial power plants.

Ethanol has also recently been used as an automotive ORC working fluid because of its ability to perform well at medium-temperature levels (200-300^{0} C) .The main advantage of using organic substances as working fluids is that their evaporation temperatures are relatively lower than steam (water vapor), which allows them to be vaporized or superheated by low or medium-temperature heat sources. In addition, using organic substances that are categorized as “dry” or “isentropic” fluids may extend the expander life; vapor expanded inside the expander tends to be dry under either saturated or superheated vapor conditions .Some important characteristics or properties of the working fluids simulated in this study.

The present first-order simulation was based on fundamental energy balance equations at the given expander and pump efficiencies. Therefore, actual performance should be verified by carrying out more detailed simulations with heat exchanger model and/or experiments. However, it is believed that the present results provide information on the fundamental performance of HFO- 1234yf as the first step toward considering ORC working fluids in terms of the temperature level.

The feasibility of HFO-1234yf as a working fluid was investigated through first-order simulations using five ORC models: trilateral, saturated, superheated, sub-critical, and supercritical cycle. The efficiency of HFO-1234yf was compared with that of other working fluids. The following results were obtained in this study:

1.4 SCOPE

1. The ORC performances using HFO-1234yf and HFC-134a as working fluids showed almost similar thermal efficiencies for all ORC models; thus, HFO-1234yf can replace HFC-134a as a suitable working fluid.

2. HFO-1234yf can be used for low- to medium-temperature ORC applications, even though its thermal efficiency was not always the best among the other simulated fluids depending on the temperature level.

3. The thermal efficiency of ORC with HFO-1234yf could be maximized approximately up to 8.8%-11.4% when the expander inlet temperature was 170^{o} C for a typical condensation temperature range (20-40^{o} C) in the supercritical cycle. However, a further increase in the expander inlet temperature to over 160 ^{o} C did not significantly boost the thermal efficiency.

4. Overall, HFO-1234yf is a potential working fluid for low- to medium-temperature heat sources of ORC applications (expander inlet temperature of up to 170^{o} C) if ODP, GWP, and safety are considered to have precedence over performance.

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 (WHR) system in two stage turbocharged HDD engine

The WHR is attained by introducing a Rankine cycle, which uses an organic substances 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 system utilizing R245fa as the working fluid is proposed and an experimental system is designed, constructed and tested. Both the evacuated solar collector and the flat plate solar collector are used in the experimental system, mean-while, a rolling-piston R245fa expander is also mounted in the system. The new designed R245fa expander works stably in the experiment, with an 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 solar collector, it is about3. 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 (ORCs) provide an alternative to traditional steam Rankine cycle for the conversion of low grade heat source into power, where conventional steam power cycle are 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. The customized cycle is demonstrated on a heat source derived from a fischer Tropsh reactor and its associated process. Analysis show a steam Rankine cycle can achieve a 20. 6% conversion for this heat source, whereas a simple organic Rankine using hexane as the working fluid can achieve a 20. 9% conversion efficiency.

Wang et al. [4] did the performance analysis of double organic Rankine cycle for discontinuous low temperature waste heat recovery. The optimal operations of several working fluids have been calculated by a procedure employing MATLAB and REFPROP.

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 point temperature difference has been employed to obtain a thermodynamic understanding of the ORC performance. Of all the working fluids investigated, some performances between each working fluid are rather similar. For a fixed low temperature heat source, the optimal operation condition should be mainly determined by the heat carrier of the heat source, and working fluids have limited influence.

Larjola [5] suggested that Electricity From industrial waste heat using high-speed organic Rankine cycle. In the conversion of low temperature heat into electricity the greatest efficiency is obtained in many cases by using an organic Rankine cycle. The ORC-process may be feasible also in high temperature applications, if the output is small. This paper deals with an ORC-design, in which a high-speed oil free turbo-generator-feed pump is used. The use of high-speed turbo-generator makes the ORC small, simple, hermetic and reduces significantly the maintenance expenses.

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 Organic Rankine (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 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 built and validated. The result shows that the external radiative or convective heat loss coefficient was built 3. 2 or 7. 0 w/m^2 when the ORC operated around 100°C.Radiative and convective heat loss due to connection between the expander and the support accounted for a large proportion of the total heat loss. The fitting relationship between heat loss and mean temperature difference were established. He also suggested that low conductivity material be embodied in the support of expander. Matters insulation for compact expander could be eliminated when the operation temperature is around 100°C.

Quoilin et al. [7] did Performance and design optimization of a low-cost solar organic Rankine cycle for remote power generation. Recent interest in small-scale solar thermal combined heat and power (CHP) 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 meeting this demand is parabolic trough solar thermal collector coupled with organic Rankine cycle (ORC) 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 though collectors, a storage tank, and a small-scale ORC engine 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 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).

Heberle 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 450 k.Series and parallel circuits of an organic Rankine cycle (ORC) and an additional heat generation was compared by second law analysis. The result shows that due to a combined heat and power generation, the second law efficiency of geothermal power plant can be significantly increased in comparison to power generation. The most efficient concept is series circuits with an organic working fluid that show high critical temperature like Isopentane. For parallel circuits and for power generation, fluid like R227ea with low critical temperature is too preferred.

Liu et al. [9] did investigation of two stage Rankine cycle for electric power plants.

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 working fluid with higher density than water, it is possible to reduce the installation size and to use air-cooled condenser. In order to search suitable working fluid for our application, nine potential candidates from four different organic fluid families and ammonia are tested. The performance of two stage Rankine cycle operating with those different working fluids is evaluated. The influences of design external temperature and the steam turbine outlet pressure on the system are analyzed. Optimal points are found at different cold source temperature and steam turbine outlet pressure for each fluid.

System efficiency can also be enhanced by introducing a regenerator for some of the selected working fluids.

Aleksandra& Borsukiewicz [10] suggested pumping work in the Rankine cycle& made

Calculation based result for the pumping work in the ORC system .Analysis has been carried out for 18 different organic fluids that can be used as working media in the subcritical ORC power plants. An attempt was made to find correlations between various thermo-physical properties of working fluids, specific work and power of the cycle. The simulation results allow a statement that the working substances with relatively low critical temperature have greater cycle pressure range for specified cycle temperature range than those with higher critical temperatures. The greater cycle pressure range contributes to a higher pumping power demand. Due to the fact that the specific pumping work does not result in explicit statement on the suitability of some working fluid to specified ORC power plant, a definition of the power decrease factor k is introduced.

Clemente et al. [11] proposed the Energy efficiency analysis of organic Rankine Cycle with scroll expanders for co-generative applications. They found Small scale Organic Rankine (ORC) system has been the object of a large number of studies in the last decade, because of the suitability for energy recovery and co-generative applications. The paper an ORC numerical model and its applications to two different case studies; the code has been obtained by combining a one-dimensional model of scroll machine and a thermodynamic model of a whole ORC system. Series production components, such as scroll compressor, from HVAC field have been first considered in order to reduce costs, because this is a critical issue for small scale energy recovery and cogeneration systems.

The detailed model of the scroll machine is applicable to calculate the performances of both a compressor and expander, as function of geometry of the device and the working fluid.

They found that model has been first tested and validated by comparing its outputs with experimental test on a commercial scroll compressor, then used to calculate the working curves of commercial scroll machine originally designed as compressor in the HVAC field, but operating as expanders. The model of the expander has been then integrated in the thermodynamic model of the ORC system. A series of comparison have been carried out in order to evaluate how the performances are influenced by scroll parameters, scroll geometry and working fluid for different applications. The result confirm the feasibility of small scale CHP systems with acceptable electrical efficiency, taking into account the low-temperature thermo source, small power output and the low-cost series production components employed.

Garcia-Rodriguez et al. [12] proposed Solar-powered Rankine cycle for fresh water production. The lack of access to electricity grid and fresh water strongly limits the development and the quality of life to many rural locations. The distributed solar power generation can be applied to many basic needs, not only electricity generation, but also desalination, cooling, heating, etc. For this reason it provides opportunity of social and economic development and therefore promoting employment. He also focused that on the analysis of distributed solar-powered generation systems for driving a reverse osmosis desalination process based on solar-heated Rankine cycles. Three different top temperature ranges are considered in order to consider medium to low temperature solar thermal collectors. Result presented in this paper points out the desalination system coupled to solar-powered organic Rankine cycle exhibit lower specific consumption of solar energy than solar distillation and solar photovoltaic reverse osmosis system. Therefore, there are interesting prospects for developing cost-effective solar desalination system based on such a technology although intensive experimental research is still needed.

SprouseIII and Depick [13] suggested the Review of Rankine cycle for internal combustion engine exhaust waste heat recovery.

Escalating fuel and future carbon dioxide emission limits are creating a renew interest in method to increase thermal efficiency of engine beyond the limit of in-cylinder techniques. One promising mechanism that accomplishes both objectives is the conversion of engine waste heat to a more useful form of energy, either mechanical or electrical.

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 the medium-grade energy of exhaust. Selection of the cycle expander and working fluid are the primary focus of the review, since they are regarded as having the largest impact on system performance. Result demonstrates a potential fuel economy improvement around 10% with modern refrigerators and advancements in expander technology.

Florian and Dieter [14] suggested exergy based fluid selection for a geothermal Organic Rankine cycle for combined heat and power generation. In the 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 organic Rankine cycle (ORC) and an additional heat generation were compared by second law analysis. Depending on operating parameter criteria for the choice of the working fluid were identified. The result shows that due to a combined heat and power generation, the second law efficiency of geothermal power plant can be significantly increased in comparison to power generation. The most efficient is series circuits with an organic working fluid that show high critical temperature like Isopentane. For parallel circuits and for power generation, fluids like R227ea with low critical temperature are too preferred.

Algieri and Morrne[15] suggested the comparative energetic analysis of high-temperature subcritical and transcritical Organic Rankine Cycle (ORC) in a biomass application in the sibari district. The present work aims to analyze the energetic performance of organic Rankine cycle (ORCs) for small-scale applications.

To this purpose, a parameter energy analysis has been performed to define the proper system configurations for a biomass power plant. Saturated and superheated conditions at the turbine inlet have been imposed and subcritical and transcritical cycle has been investigated. Furthermore, the effect of operating conditions and the impact of internal regeneration on system performed have been analyzed. Finally, the possible exploitation of biomass resulting from pruning residues of peach trees in the sibari district (Southern Italy) has been evaluated for configurations optimized during the energetic analysis. The analysis shows that ORCs represent a very interesting solution for small-scale and decentralized power production. Moreover, the result highlight the large influence of the maximum temperature and the significant impact of the inter regeneration on the power plant performances.

Hui-tao et al. [16] did the optimization of Low-Temperature Exhaust Gas waste heat Fueled Organic Rankine Cycle. Low temperature exhaust gas carrying large amount of waste heat released but steel- making process and many other industries, Organic Rankine Cycle are proven to be the most promising technology to recover the low-temperature waste heat, thereby to get more financial benefits for these industries. The exergy analysis of ORC units driven by low-temperature exhaust gas waste heat and charged with dry and isentropic fluid was performed, and intuitive approach with simple impression was developed to calculate the performances of the ORC unit. Parameter optimization was conducted with turbine inlet temperature simplified as the variable and exergy efficiency or power output as the objective function by means of penalty function and golden Section Searching algorithm based on formulation of the optimization problem. The power generated by optimized ORC unit can be nearly as twice as that generated by a non-optimized ORC model unit. In addition, cycle parametric analysis was performed to examine the effects of thermodynamic parameters on the cycle performed such as thermal efficiency and exergy efficiency.

It is proven that performance of ORC unit is mainly affected by the thermodynamic property of working fluid, the waste heat temperature, the pinch point temperature of the evaporator, the specific heat capacity of the heat carrier and the turbine inlet temperature under a given environment temperature.

Kosmadakis et al. [17] suggested the parametric theoretical study of a two-stage solar organic Rankine cycle for RO desalination. The present work concerns the parametric study of an autonomous; two stage solar organic Rankine cycle for RO desalination. The main goal of the current simulation is to estimate the efficiency, as well as to calculate the annual mechanical energy available for desalination in the considered cases, in order to evaluate the influence of various parameters on the performance of the system. The parametric study concerns the variation of different parameters, without charging actually the baseline case. The effect of the collectors slope and total number of evacuated tube collector used, have been extensively examined. The total cost is also taken into consideration and is calculated for different cases examined, along with the specific fresh water cost.

Tchanche et al. [18] Fluid selection for a low-temperature solar organic Rankine cycle. Theoretical performance as well as thermodynamic and environmental properties of few fluids has been comparatively assessed for use in low-temperature solar organic

Rankine cycle systems. Efficiencies, volume flow rate, mass flow rate, pressure ratio, toxicity, flammability, ODP and GWP were used for comparisons. Of 20 fluids investigated, R134a appears as the most suitable for small scale solar applications.R152a, R600a, R600 andR290 offer attractive performances but need safety precautions, owing to their flammability.

Shengjun et al. [19] did performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low temperature geothermal power generation. Organic Rankine cycle is a promising technology for converting the Low-grade energy to electricity. They also present an investigation on the parameter optimization and performance comparison of the fluids in subcritical ORC and transcritical power cycle in low-temperature binary geothermal power system. The optimization procedure was conducted with simulation program written in Mat lab using five indicators: thermal efficiency, exergy efficiency, recovery efficiency, heat exchanger area per unit power output (APR) and the level zed energy cost. With the given heat source and heat sink conditions, performances of working fluids were evaluated and compared under their optimized internal operation parameters. The optimum cycle design and the corresponding operation parameters were provided simultaneously. The optimum cycle design and the corresponding operation parameters were provided simultaneously. The results indicates that the choice of working fluids varies the objective function and the value of the optimized operation parameters are not all the same for different indicators.R123 in subcritical ORC system yields the highest thermal efficiency and exergy efficiency of 11. 1% and 54%, respectively. Although the thermal efficiency and exergy efficiency of R125 in transcritical cycle is 46. 4% and 20% lower than that of R123 in subcritical ORC, it provides 20.7% larger efficiency. And the LEC value is relatively low. More ever, 22032L Petroleum is saved and 74, 019kg CO_2 is reduced per year when the LEC value is used as the objective function. In conclusion, R-125 in transcritical power cycle show excellent economic and environmental performance and can maximized utilization of the geo thermal. It is preferable for the low-temperature geothermal ORC system.R41 also exhibits favourable performance except for its flammability.

Nobru et al. [20] presented Study on thermal efficiency of low –to medium-temperature organic Rankine cycle using HFO-1234yf.They found that heat recovery using organic Rankine cycle is considered to be an important method among renewable energy processes because of its capability to generate power from available waste heat and natural heat sources, such as solar radiation, ocean thermal sources, geothermal sources, biomass, waste heat from fuel combustion or industrial process.

Tchanche et al. [21] has proposed fluid selection for a low-temperature solar organic rankine cycle.

Vaja I et al [22] has suggested internal combustion engine bottoming with organic Rankine cycle.

Saleh B. et al [23] has suggested working fluid for low temperature organic cycle.

Gu et al. [24] has presented Theoretical and experimental Investigation of an organic Rankine cycle for a waste heat recovery system.

In ORC is power system, the working fluid must be selected to maximize thermal efficiency for temperature range of heat source and long –term performance.

Typical temperature range for orca is from low temperature of about=100°c to medium temperature of about =350˚c

Existing ORC power systems still tend to use refrigerants with relatively high global warming potential (GWP) and ozone depletion potential (ODP) as their working fluids due to limited option for the working fluid.

Spartz M& Minor B [25] has suggested, HFO-1234yf low GWP refrigerant a global sustainable solution for mobile air conditioning.

Sparted also reported that HFO-1234yf has excellent environmental properties

They also reported that toxicity and thermodynamic properties.

Gu Z&Sato H [26] has proposed supercritical cycle for geothermal binary design.

Tanka K et al [27] reported- Critical temperature, critical density, and critical pressure.

Takizawa et al. [28] reported HFO-1234yf mildly flammable and could be a global solution to mobile air conditioning systems.

The sponsors of SAE CRP1234 [29] concluded that HFO-1234yf can be used to globally replace refrigerant used in future mobile (cars and light truck) air conditioning system.

Those aspects of HFO-1234yf should allow it to consider as a potential ORC working fluid. He never, thus far, there have been no studies regarding the use of HFO-1234yf as an ORC working fluid.

In this study , the theoretical thermal efficiency of ORC using HFO-1234yf as the working fluid was simulated and compared with that of ORC using there working fluid to verify the feasibility and performance characteristics of HFO-1234yf.

Five working fluids-HFO-1234yf, HFC-134a, HFC-245faed, ethanol, and iso-pentane-were considered for five types of thermodynamic ORC model –trilateral, saturated, superheated, sub-critical, and supercritical.

Zyhowski Gj. [30] proposed the opportunities for HFC-245f organic Rankine cycle appended to distributed power generation system.

Wei D et al. [31] did his performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery.

Chacrtegui R.et al. [32] has proposed Alternative ORC bottoming cycle for combined cycle power plants.

Blok K.et al. [33] proposed opties Voor verbetering van de energie-efficientie.

In addition to the HFC-134a application mentioned in the introduction, HFC-245fa and isopentane are also used in ORC application to recover low –temperature waste heat from Industrial power plant.

Ringer J et al. [34] used Rankine cycle for waste heat recovery of IC engine.

Ethanol has also recently been used as an automotive ORC working fluid because of its ability to perform well at medium-temperature levels.

Desai NB. et al. [35] has proposed integration of rankine cycle, energy.

The main advantage of using organic substances as working fluid is that their evaporation temperature are relatively lower than steam, which allow them to be vaporized or superheated by low or medium-temperature heat sources.

In addition, using organic substances that are categorized as dry or isentropic fluid may extend the expander life; vapor expanded inside the expander tends to be dry under either saturated or super heated vapor conditions.

Brown BW et al.[36] did their Flow sheet simulation of the trilateral cycle in which it is compared as simple cycle as conventional binary cycle, its performance is more competitive for low-temperature heat sources, especially when a proper two- phase expander is employed.

Smith IK et al [37] developed the trilateral flash cycle system with design of high-efficiency two phase expander.

Dipippo R. [38] has proposed ideal thermal efficiency for geothermal binary plants

Zamfirescu&Dincer [39] described two attractive features of this cycle the working fluid temperature profile tends to match the heat source temperature profile perfectly, and it operates at reasonable pressure so its implementation is economically feasible for low-power applications.

Dai Y et al. [40] did his parametric optimization and comparative study of organic Rankine Cycle for low grade waste heat recovery in saturated cycle was selected because it is the most basic cycle that can achieve a compact system design relative to acceptable thermal efficiencies when recovering low temperature waste heat.

Yamamoto T. et al. [41] suggested Design and testing of the organic Rankine cycle.

Karellas et al. [42] has presented sub-critical cycle ever simulated in this study have almost similar operation condition to superheated cycle; however, the pumping pressure is kept higher than the pressure used for the superheated cycle but slightly lower than the critical pressure for working fluids.

2.2 Conclusions of Literature survey

- An ample literature is available on organic Rankine cycle where organic substances as working fluid have been used that their evaporation temperatures are lower than steam (water vapor), the advantage which allow them to be Vaporized or superheated by low or medium-temperature heat sources.
- Five fundamental thermodynamic models of ORC with five types of cycle-trilateral, saturated, superheated, sub-critical and supercritical have been studied & VIZ to compare the thermal efficiency of HFO-1234yf have been compared with that of other working fluids, HFO-1234yf was found to a thermal efficiency that was comparable to that ofHFC-134a.
- In a recent study on low to medium temperature ORC cycles using HFO-1234yf (Reference)

- It also provides a useful map that clearly shows the best possible thermal efficiency among the five types for various expander inlet and condensation temperatures.
- The highest thermal efficiency range (8. 8%-11. 4) was obtained when the supercritical ORC was used at an expander inlet temperature of170±10°c and a condensation temperature range 20-40°c for the given pump and expander efficiency.
- It is also concluded that HFO-1234yf is potential working fluid for ORC applications, especially for those with low-to medium-temperature heat sources.

2.3 Gaps in Literature survey.

From whole literature survey it is found that there are different type's working fluids (R1234yf, R-245fa, Ethanol, Isopentane, R-134a, Iso-pentane) that are used in different types of cycles such as, Trilateral, Saturated, Supercritical, Superheated & first law analysis has been carried out.

But no study has been done on exergy analysis of these cycles using R1234yf.

2.4 Objective of present work

Based on the gaps discussed above in the literature survey there are following objectives of present work are as under:

(1) To Exergy Analysis of rankine cycle by determining exergy destruction in different components such as, pump, evaporator, expander, condenser.

(2) To find the first law efficiency under different working parameters.

(3) To find the exergetic efficiency of Organic rankine cycle, for saturated, Trilateral, supercritical, superheated with various working fluids.

CHAPTER 3

THERMODYNAMIC MODELLING

The Basic ORC simulation model as shown in here are four main components: expander, condenser, and pump.

The operation principles are similar to those of the Rankine cycle, the working fluid, which is initially in a liquid state, is delivered by the pump to the evaporator at a designated pressure and mass flow rate.

3.1 Basic ORC model

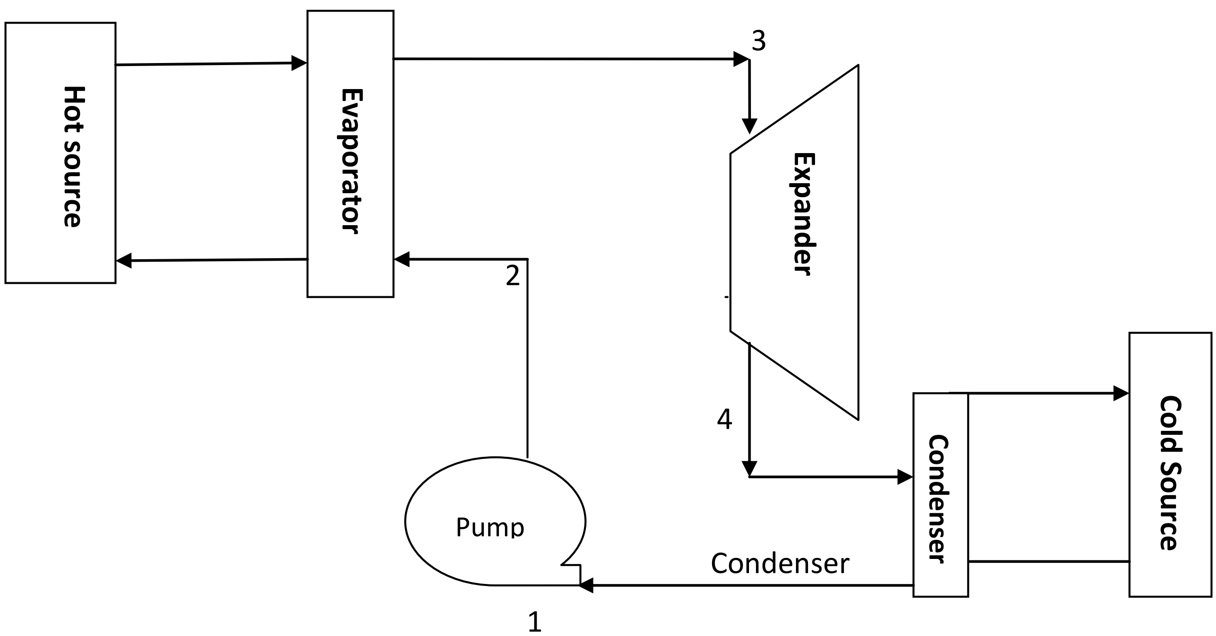


Fig.3.1. Basic ORC model

The working fluid is heated to a specified temperature level inside the evaporator, and the high-temperature and high-pressure liquid from evaporator expands inside the expander to produce rotational power. The low-pressure fluid from the expander is then condensed inside the condenser to the liquid state before the liquid is re-circulated by the pump.

The thermodynamic ORC models were examined for five working fluids in this study, different cycle path should be determined for each combination of ORC model and working fluid.

Temperature-entropy (T-S) diagram are best way to represent these path Figs.2and3, and 4 show the T-S diagrams of five ORC models for ''wet'', ''dry'' and' ''isentropic'' fluids, respectively.

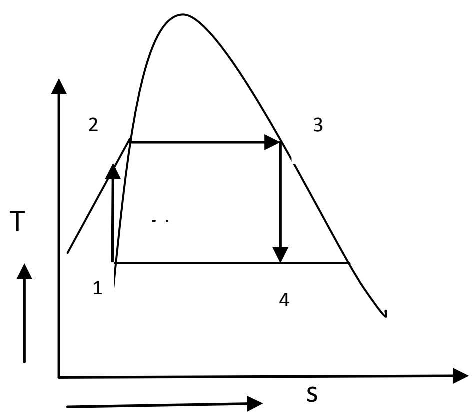


Fig.3.2.Saturated ORC

3.2Saturated cycle model:

It is 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 for the saturated cycle represented by the cycle path 1-2-3-4-1.

The working fluid is heated in the evaporator at a constant pressure after isentropic pumping (1-2) until it becomes saturated vapor (2-3);this is then followed by isentropic expansion process (3-4).

Depending on the type of working fluid, the phase condition of the working fluid during the expansion process goes two-phase mixture (4), for wet fluids, goes to superheated state (4) for dry fluids and becomes saturated vapor (4) for isentropic fluid.

After the expansion process, isobaric condensation takes place in the condenser, which returns the fluid to saturated liquid state. (4-1)

3.3Trilateral cycle model:

The trilateral ORC was chosen for this study because it is a simple cycle; compared to conventional binary cycle, its performance is more competitive for low-Temperature heat sources, especially when a proper two-phase expander is employed.

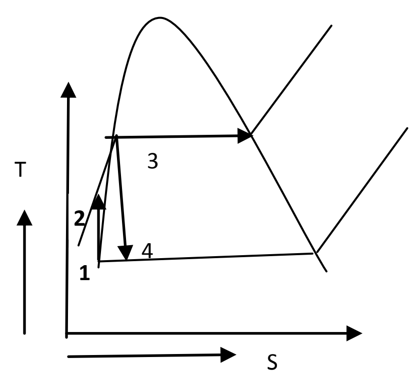


Fig.3. 3 (Trilateral cycle model)

Furthermore, it has a lower operational cost for equivalent-sized components.

Zamfirecu described two attractive features of this cycle: the working fluid temperature profile tends to match the heat source temperature profile perfectly, and it operates at reasonable pressure so its implementation is economically feasible for low-power applications.

In this study, the expected T-s diagram of the trilateral ORC model is represented by cycle path 1-2-3-4-1. The pumping pressure, which is theoretically the same as evaporation pressure, lies below the critical pressure of the working fluid, and the temperature at the expander inlet is maintained at a target saturated temperature.

The cycle consists of isentropic pumping state (1-2) of a working fluid the state, following by isobaric heating of the working fluid to the bubble point (2-3). Isentropic expansion of the working fluid to a two-phase mixture (3-4) is then induced, and finally an isobaric condensation occurs to return the two-phase mixture back to a liquid state (4-1).

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

At constant condensation temperature, thermal efficiency sharply increases when the expander inlet temperature rises from T_{in}=50^{o}C to 170+± 10^{o}C:it then decreases slowly when the expander inlet temperature further increases up to 200^{o}C.

3.4Superheated cycle model:

In comparison to the trilateral and saturated cycle, the key operation of the superheated cycle is to allow more heat to enter the ORC system and changing the working fluid state to superheated state.

In some cases, the superheated cycle results in higher thermal efficiency than that of the saturated cycle are represented by cycle path 1-2-3-4-1.

After the isentropic pumping process (1-2), the heating process (2-3) takes place until the working, fluid phase changes to superheated state: the temperature is higher than in saturated vapor state while the pressure level is kept the same.

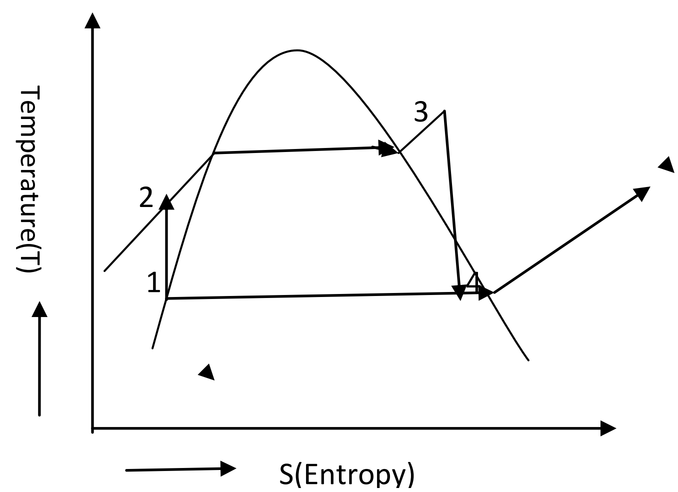


Fig. 3.4.T-S diagram for simple ORC Superheated

As the higher expander inlet temperature results in higher power production, a temperature higher than the critical temperature at expander inlet (3).

The working fluid then passes through the expander with isentropic expansion (3-4): it then condenses to saturated liquid (4-1) at constant pressure.

3.5Sub-critical cycle model

The sub-critical cycles simulated in this study have almost similar operational condition in the superheated cycles, however, the pumping pressure is kept higher than the pressure used for the superheated cycle but slightly lower than the critical pressure.

This study used a pressure level of 94% of the critical pressure.

This pressure level was assumed following the methodology presented by karallas for sub-critical ORC studies.

But a value slightly below theirs was found in this study.

The T-s diagram of this cycle (represented by cycle path. 1-2-3-4-1. looks like the superheated cycle: (1-2), is isentropic pumping; state (2-3) is isobaric heating; state (3-4) is isentropic expansion; and state (4-1) is isobaric condensation.

Similar to the superheated cycle, the expander inlet temperature for this cycle is kept higher than the critical temperature of the working fluid phase after the expansion process: a wet region in two-phase mixture or superheated state

3.6Supercritical cycle model:

In the supercritical cycle, the expander inlet condition inlet condition (pressure and temperature) are kept higher than the critical point of the working fluid.

The main advantage of the supercritical cycle is that the hot –source temperature can be raised to an ever higher level than the other cycles.

According to the Carnot efficiency, the thermal efficiency of this cycle of this cycle can be increased even further reported that propane and HFC-134a are appropriate working fluid for the supercritical cycle geothermal binary design.

In the present simulation, the pumping expanded through isentropic expansion (3-4) to either the superheated (4) or two-phase mixture (4) state.

Through isobaric condensation (4-1), the working fluid is then condensed into saturated liquid.

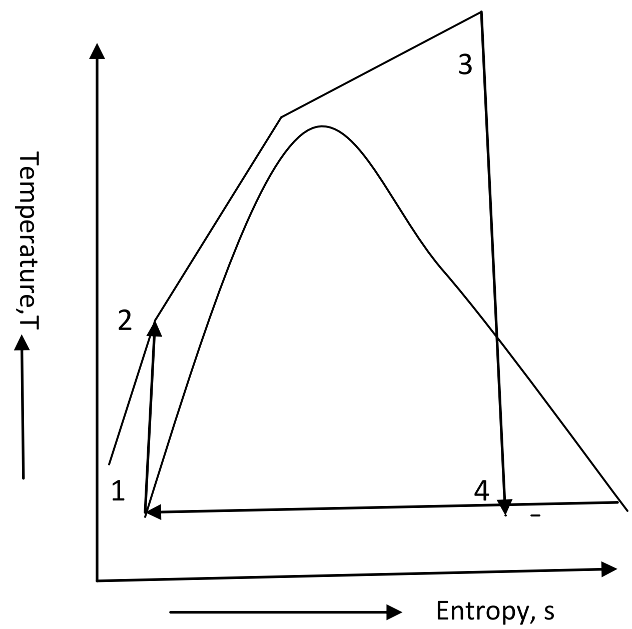


Fig.3.6. **Temperature-Entropy diagram for Supercritical ORC cycle**

In addition, the thermodynamic properties of the selected working fluid can restrict the cycle to operating at specific pressures or temperature.

Furthermore, the pump and expander were assumed to operate below 100% efficiency in the present simulation.

Therefore, the state point at the pump and expander outlet was determined from the given pump's performance.

In this chapter the detailed explanation of various cycles fundamental equations are introduced that are based on the energy balance of the ORC model according to the first & second law analysis.

Further input parameters are also specified for calculation:

3.7 Governing equation

The general energy balance equation for all existing cycle (Saturated, Trilateral, Supercritical, Superheated, and Subcritical) is as follow:

The pump isentropic efficiency and pumping power are defined as

$$\eta_p = \frac{h_{2s-}h_2}{h_2 - h_1} \qquad \text{.......... (1)}$$

$$W_P = \dot{m} * (h_2 - h_1) \qquad \text{.......... (2)}$$

<u>For the expander</u>

Isentropic efficiency (η_e)

$$\eta_{e=} \frac{(h_{3-}h_{4)}}{(h_3 - h_4)} \qquad \text{.......... (3)}$$

$$W_e = \dot{m} * (h_{3-}h_4) \qquad \text{.......... (4)}$$

<u>The heat input in the working fluid through Evaporator</u>

$$Q_{in} = \dot{m} * (h_3 - h_2) \qquad \text{.......... (5)}$$

<u>Thermal Efficiency of the ORC</u>

$$\eta_{th} = \frac{w_{e-}w_P}{Q_{in}} \qquad \text{.......... (6)}$$

$$\eta_{th} = \frac{(h_{3-}h_3) - (h_2 - h_1)}{Q_{in}} \qquad \text{.......... (7)}$$

3.8 Exergy Analysis

The second law of thermodynamics assists in evaluating the performance of the system based on exergy, which always decreases owing to irreversibility. Exergy is the measure of usefulness, quality or potential of a stream to cause change and an effective measure of the potential of a substance to impact the environment (Dincer, 2003). Exergy balance for a control volume undergoing steady state process is expressed as (Lee and Sherif, 2001)

$$\dot{E}D_i = \sum(\dot{m}e)_{in} - \sum(\dot{m}e)_{out} + \left[\sum\left(\dot{Q}(1-\frac{T_O}{T})\right)_{in} + \sum\left(\dot{Q}(1-\frac{T_O}{T})\right)_{out}\right] \pm \sum\dot{W}$$

Where EDi represents the rate of exergy destruction occurring in the process in the component under consideration.

The first two terms on the right hand side represent exergy of streams entering and leaving the control volume. The third and fourth terms are the exergy associated with heat transfer Q from the source maintained at constant temperature T and is equal to work obtained by Carnot engine operating between T and T_0. The last term is the mechanical work transfer to or from the control volume.

Second law performance of the system can be measured in terms of exergetic efficiency where Tr is the temperature of the space to be cooled. Exergetic efficiency can also be expressed in terms of total exergy destruction and exergy supplied to the system, i.e.

The Exergetic efficiency shows the percentage of the fuel exergy provided to a system that is found in the product exergy.

<u>Exergy destruction in evaporator:</u>

$E_{devap} = (h_2\text{-}T_O{*}S_2)\text{-}\ (h_3\text{-}T_0{*}S_3)+q_s{*}[1\text{-}T_O/\ (T_3+273)]$ .... (8)

<u>Exergy destruction in expander:</u>

$E_{dexp} = (h_3\text{-}T_O{*}S_3)\text{-}\ (h_4\text{-}T_O{*}S_4)\text{-}\ (h_3\text{-}h_4)$ .... (9)

<u>Exergy destruction in condencer:</u>

$(h_4\text{-}T_O{*}S_4)\text{-}\ (h_1\text{-}T_o{*}S_1)$ .... (10)

<u>Exergy destruction in pump:</u>

$E_{dpump}= (h_1\text{-}T_o{*}S_1)\text{-}\ (h_2\text{-}T_o{*}S_2)+\ (h_2\text{-}h_1)$ .... (11)

$Edtotal = E_{devap} + E_{dexp} + E_{dcond} + E_{dpump}$ .... (12)

<u>Exergetic Efficiency</u>

$\eta_{ex}=1\text{-}E_{dtotal}/E_{inpu}$ .... (13)

$$E_{input} = q_S*[1-T_O/(T_3+273)]+W \quad \text{..........} (14)$$

$$\eta_{ex1} = W_T/E_{input} \quad \text{..........} (15)$$

3.9 Input condition:

The input to the Thermodynamic model of the Organic Rankine cycle expander inlet temperature (T_3 in°C_), Fixed condenser temperature (T_4 =30^{o}C),

Fixed expander isentropic efficiency (η_T=75%), Fixed pump isentropic efficiency (60%).

Pump pressure ratio ($\frac{p_2}{p_1}$=2.5, 4), for superheated cycle.

The thermodynamic properties of various working fluids at various state points are calculated using program developed in EES.

The values of input at a design point are given in Table and these are referred from the work of Yamada et al. [20].

Expander inlet Temperature(T_3 in^{o}c)	30-200
Fixed condenser Temperature(T_4in^{o}C)	30
Isentropic Expander efficiency(η_T)	75
Isentropic pump Efficiency(η_p)	60
Pump pressure ratio($\frac{p_2}{p_1}, in\ MPa$)	2.5 -4
Working fluids	R1234yf,R123,R245fa,Ethanol,Isopentane

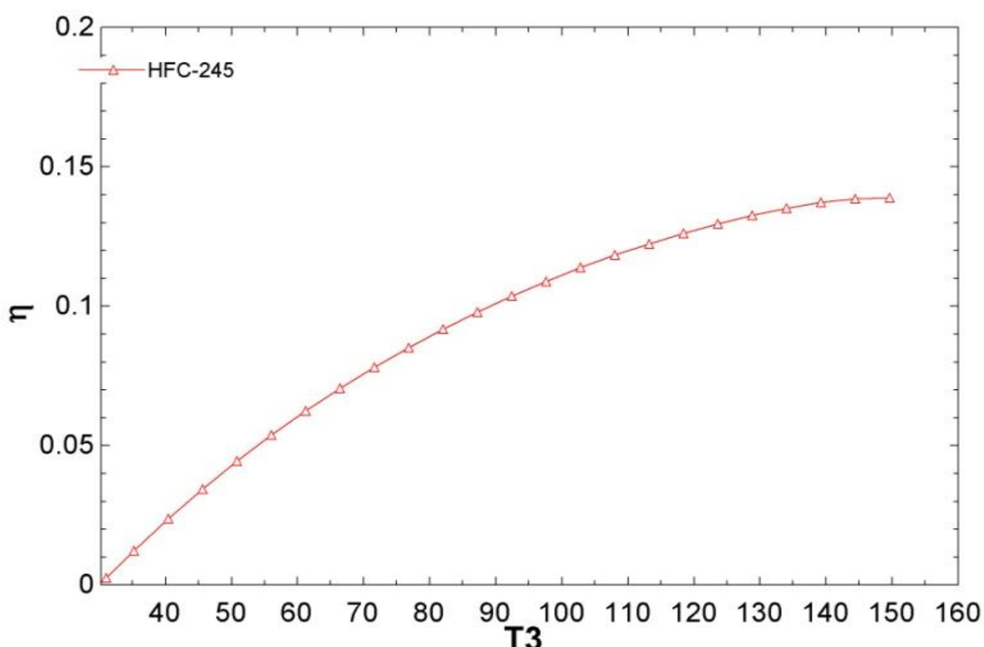
CHAPTER 4

RESULT AND DISCUSSION

The chapter deals with the result and discussion for simple ORC cycle with various working fluids at different Expander inlet temperature at fixed condenser temperature (30^oC). Isentropic Expander efficiency (η_T=75%), Isentropic pump Efficiency (η_p=60%)

4.1. Effect of Expander inlet Temperature on Thermal (η) Efficiency

The Figs. 4.1 to 4.20 show the effect of expander inlet temperature on thermal efficiency.



Expander Inlet Temp ($T3^oC$)

Fig.4.1.Thermal Efficiency For Saturated cycle with,HFC-245fa

4.2. Effect of Expander inlet temperature (T3) on Efficiency (η) using is pentane in Saturated cycle:

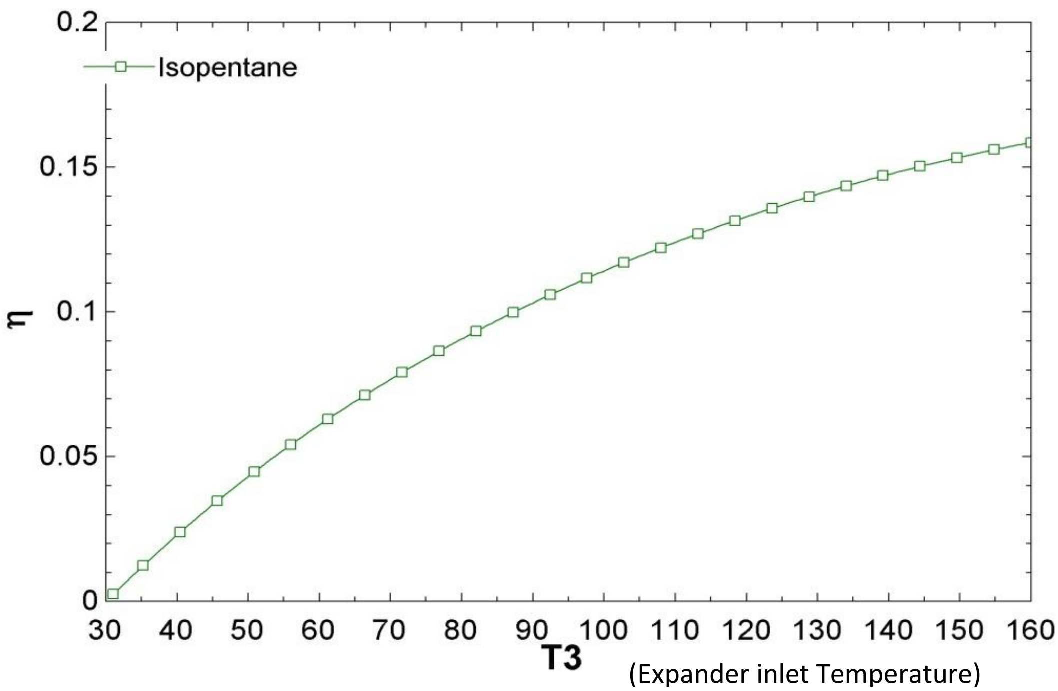


Fig.4.2.Thermal Efficiency For Saturated cycle with,Isopentane

4.3. Effect of Expander inlet Temperature on efficiency using (HFC-134a) in Saturated cycle:

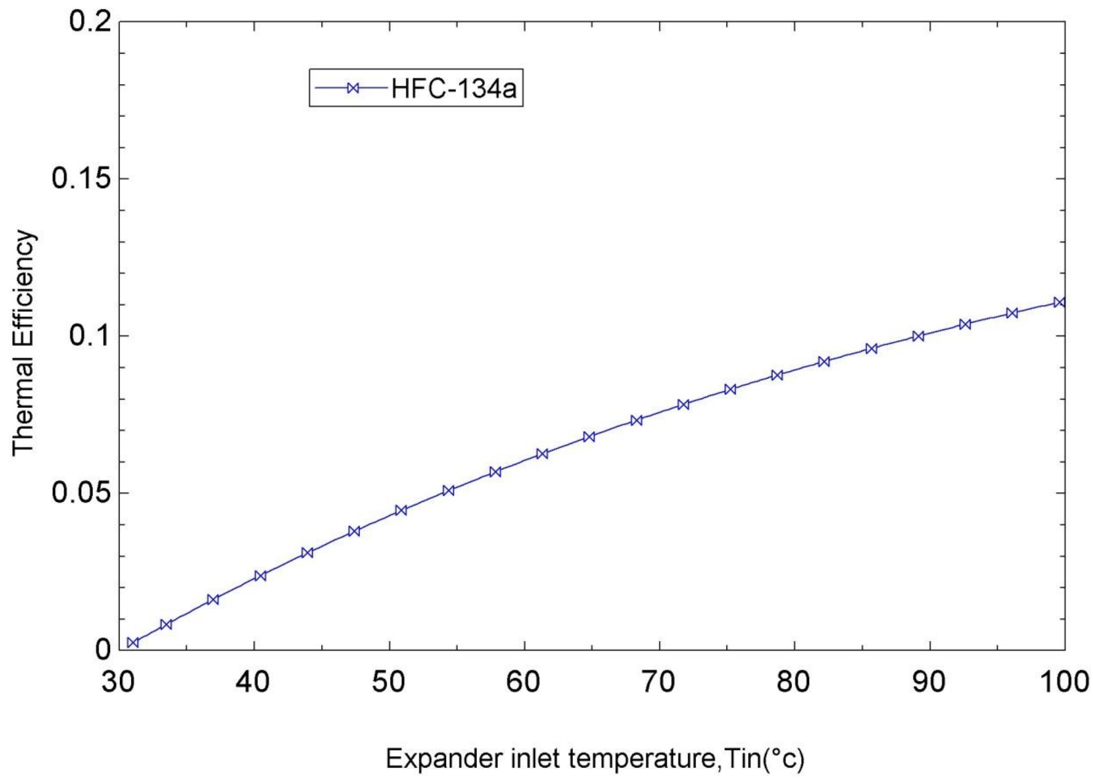


Fig.4.3.Thermal Efficiency for Saturated cycle with, HFC-134a

4.4. Effect of Expander inlet Temperature on efficiency using HFO-1234yf in saturated cycle

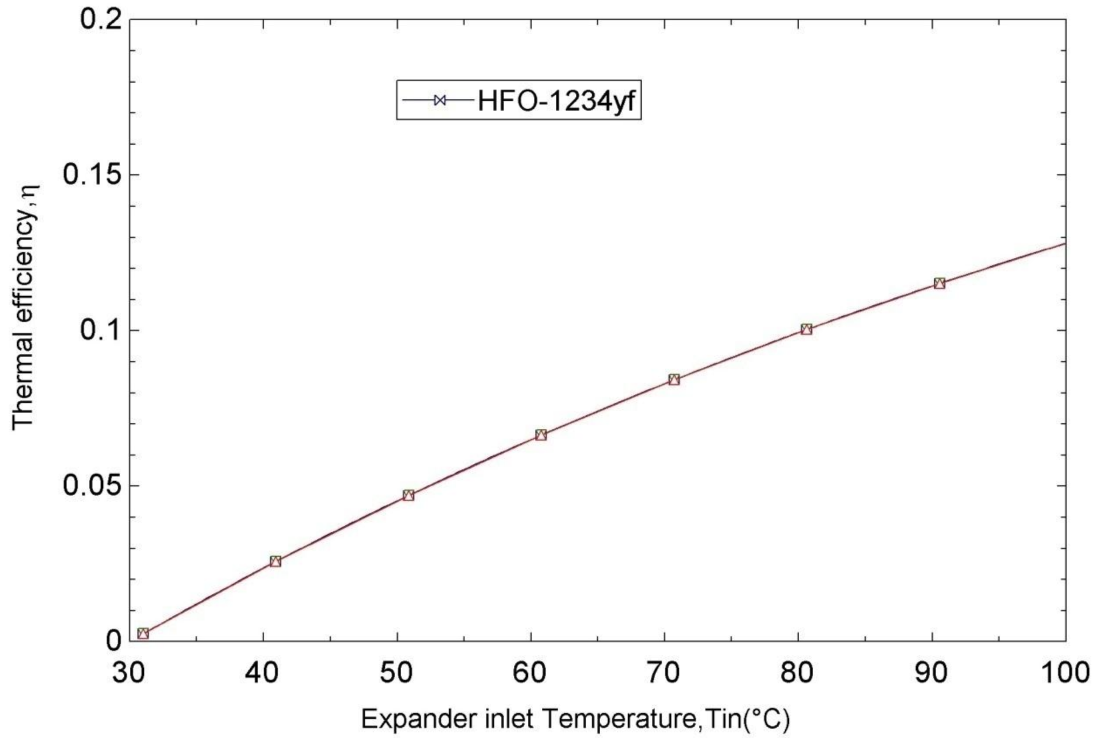
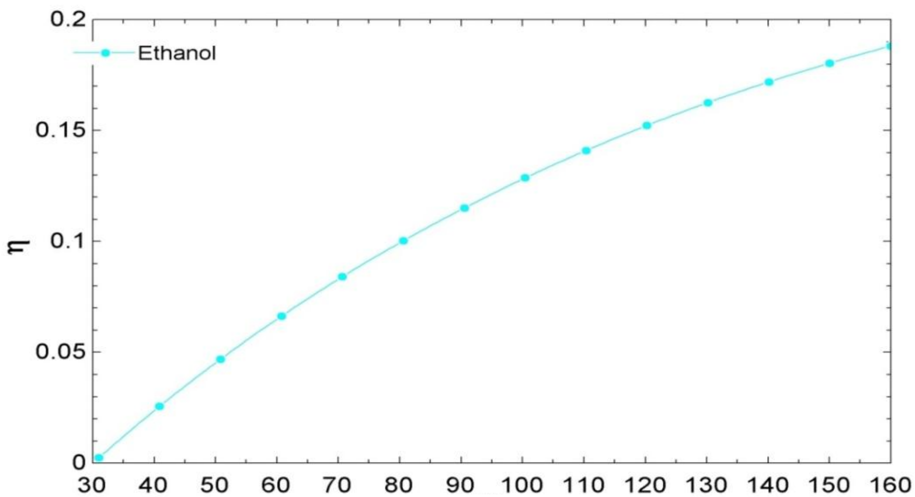


Fig.4.4.Thermal Efficiency for Saturated cycle with, HFO-1234yf

4.5. Effect of Expander inlet Temperature on efficiency using (Ethanol) in saturated cycle:



(Expander inlet Temperature°C)

Fig.4.5.Thermal Efficiency For Saturated cycle with,Ethanol

4.6.Relative Comparisions of Thermal Efficiency by using various working fluid in Saturated ORC model

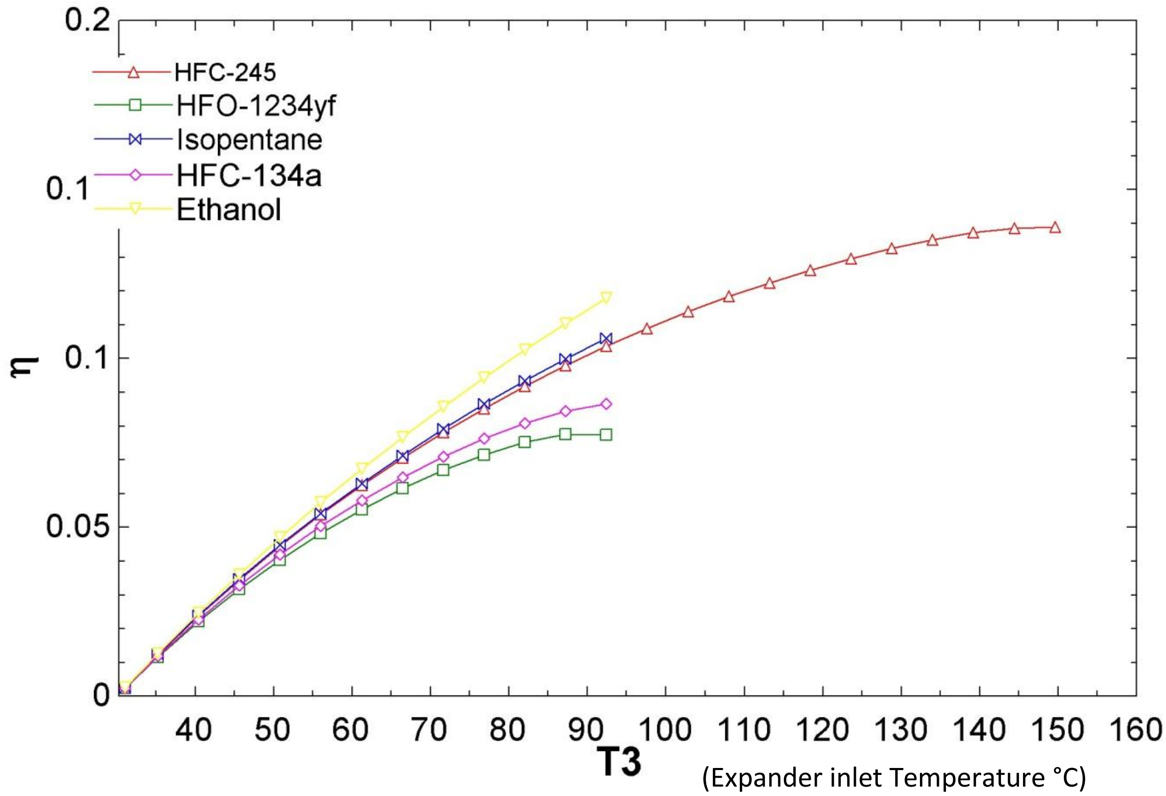
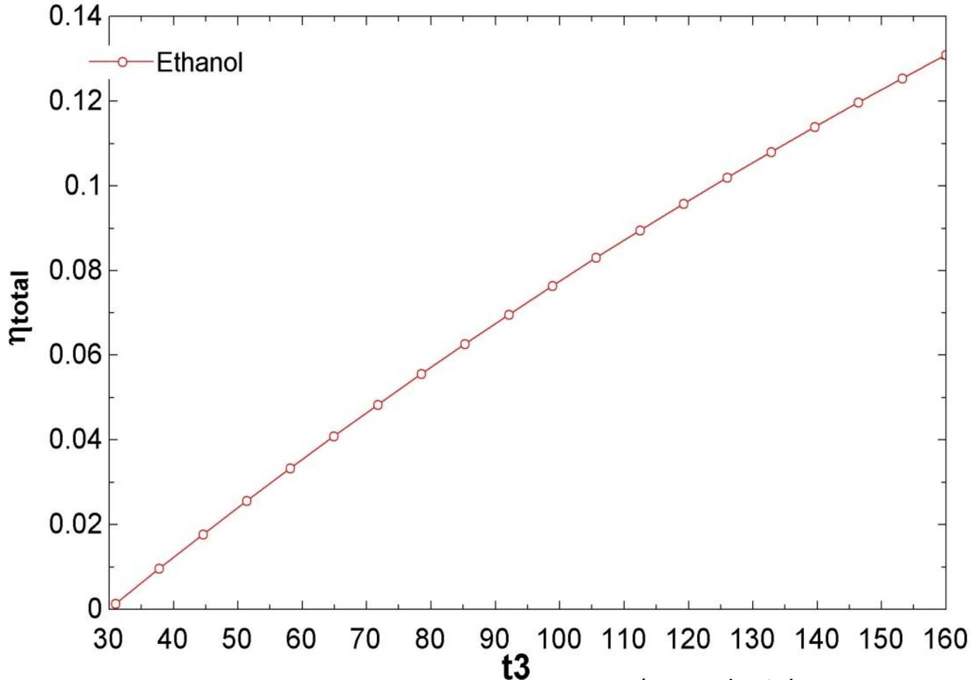


Fig.4.6.Comparision of Thermal Efficiency For Saturated cycle with,(HFC-245fa,HFO-1234YF,Isopentane,HFC-

Fig 4.6 shows the variation of thermal efficiency with expander inlet temperature for Saturated ORC for various working fluids.

It has been observed that Ethanol shows higher thermal efficiency, HFO-1234yf shows lower thermal efficiency for expander inlet temperature (30-160).

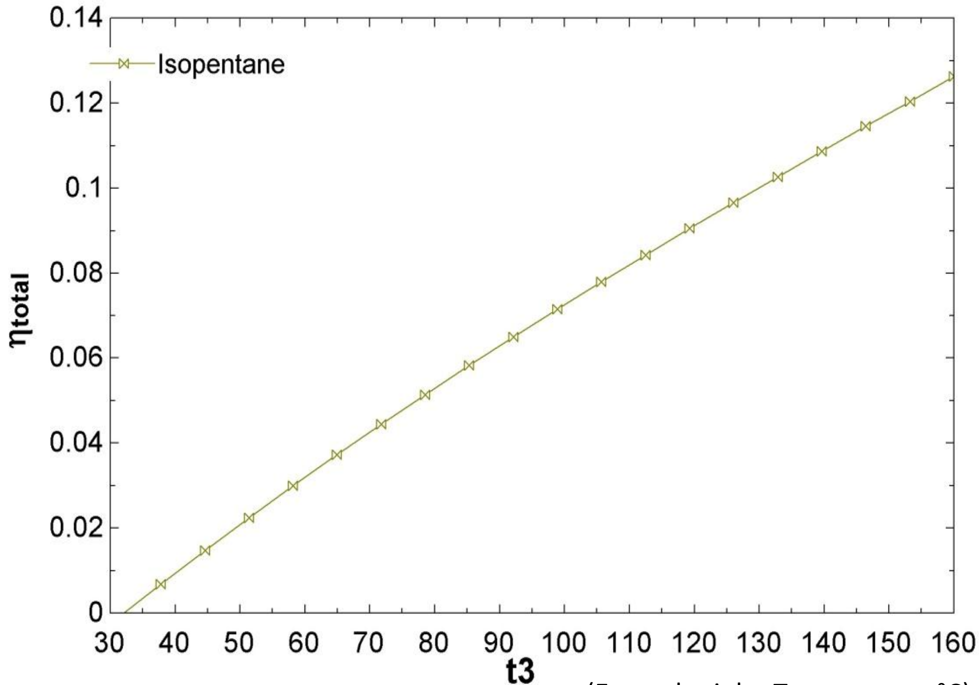
4.7. Effect of expander inlet temperature on Thermal Efficiency in Trilateral cycle by using Ethanol:



(Expander inlet Temperature °C)

Fig.4.7.Thermal Efficiency For Trilateral cycle with,Ethanol

4.8. Effect of expander inlet temperature on Thermal Efficiency in Trilateral cycle by using is pentane:



(Expander inlet Temperature °C)

Fig.4.8.Thermal Efficiency For Trilateral cycle with Isopentane

4.9 .Effect of Expander inlet temperature on Thermal Efficiency in Trilateral by using HFC-245fa

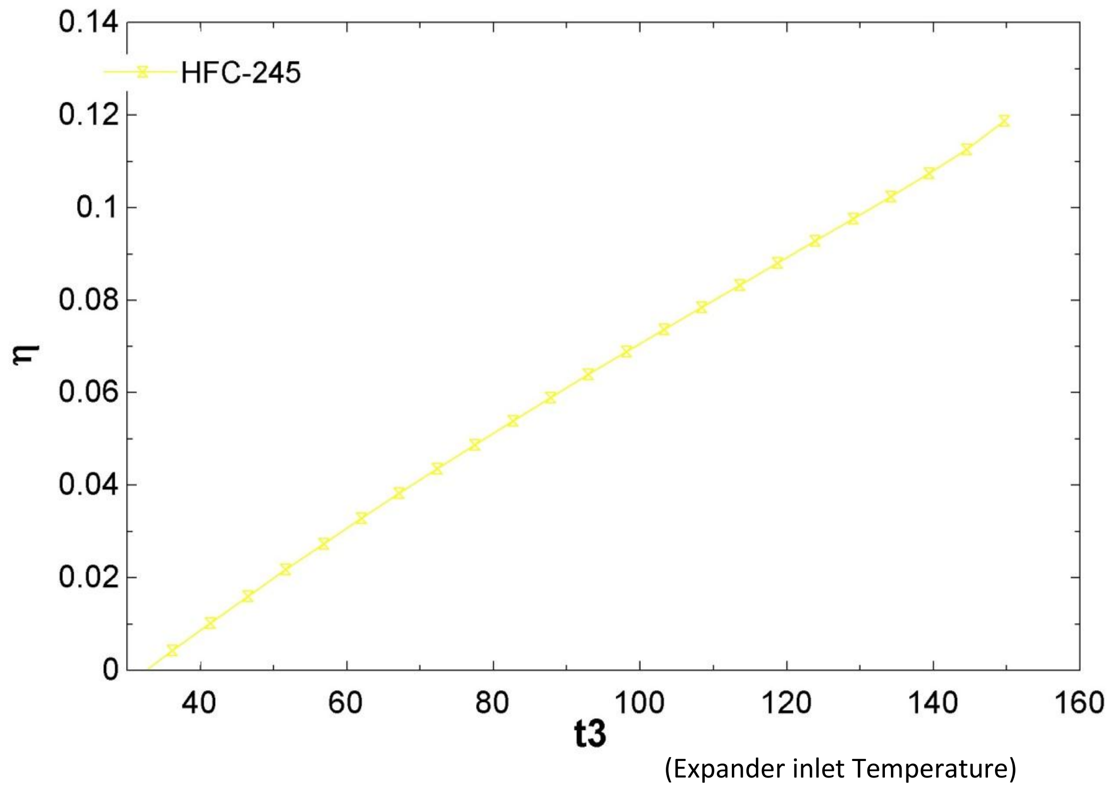
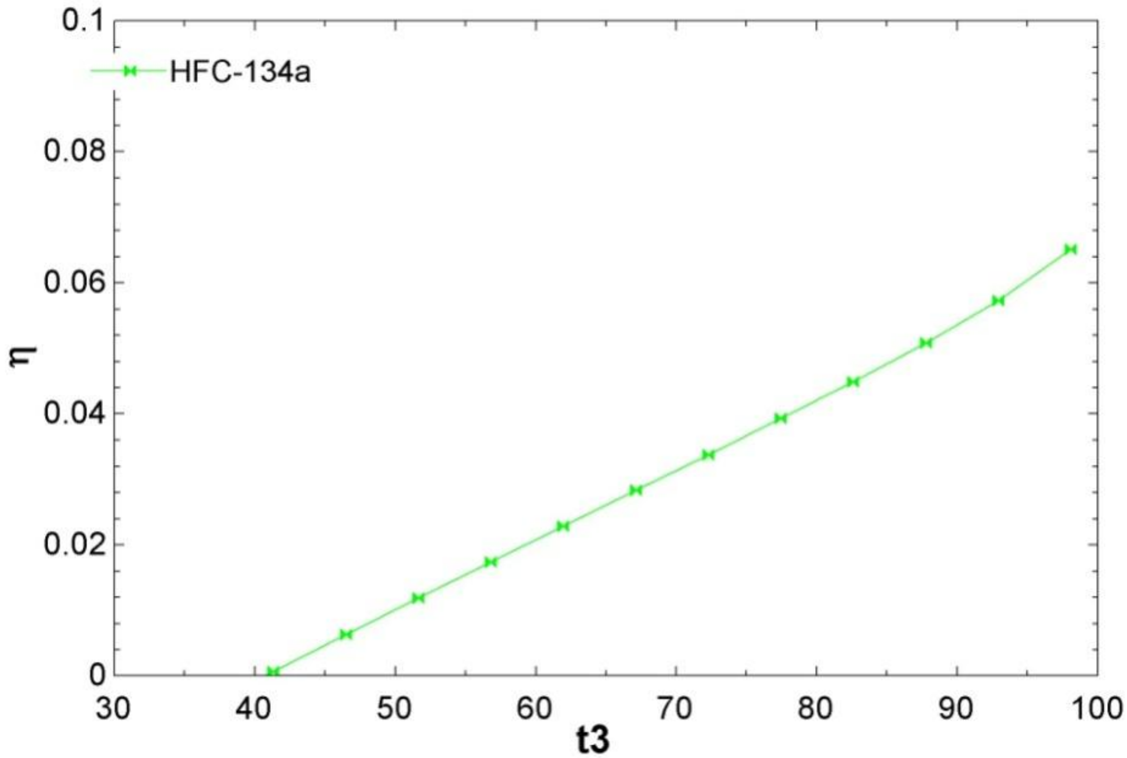


Fig.4.9.Thermal Efficiency For Trilateral cycle with,HFC-245fa

4.10 .Effect of expander inlet Temperature on Thermal Efficiency in Trilateral by using HFC-134a



(Expander inlet temperature°C)

Fig.4.10.Thermal Efficiency For Trilateral cycle with,HFC-134a

4.11 Effect of expander inlet Temperature on Thermal Efficiency in Trilateral by using HFC-1234yf

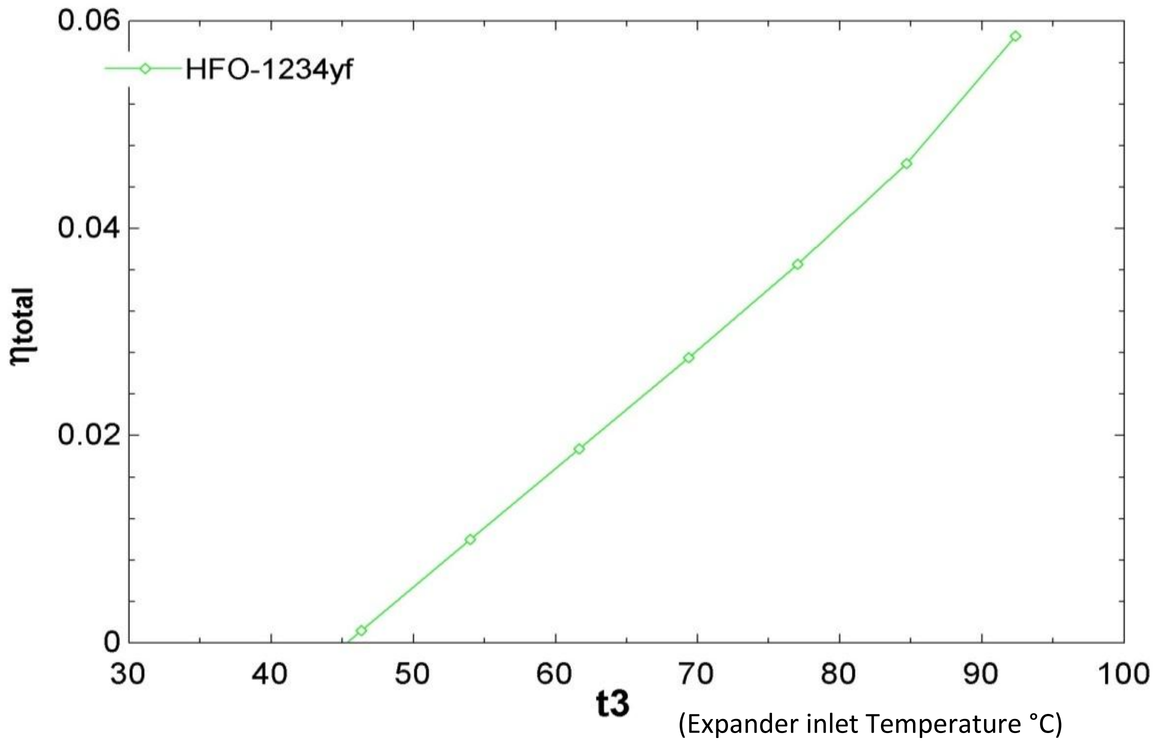


Fig.4.11.Thermal Efficiency For Trilateral cycle with,HFC-1234yf

4.12. Relative comparison of Trilateral by using various working fluids:

Fig 4.17 shows the variation of thermal efficiency with expander inlet temperature for super critical cycle for various working fluids.

It has been observed that Ethanol shows higher thermal efficiency, HFO-134a shows lower thermal efficiency for expander inlet temperature (30-160).

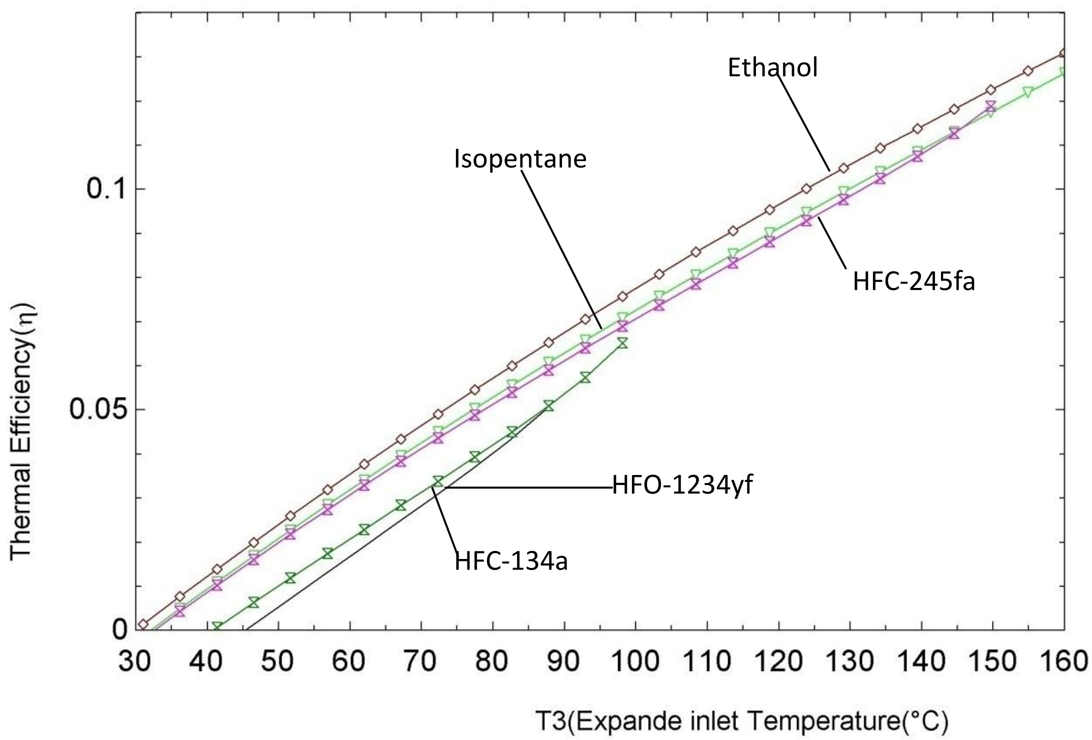


Fig.4.12.Comparision of Thermal Efficiency For Trilateral cycle with,(HFC-245fa,HFO-1234YF,Isopentane,HFC-134a,Ethanol)

4.13. Effect of Expander inlet Temperature on Thermal Efficiency in Super critical cycle by using HF0-1234yf

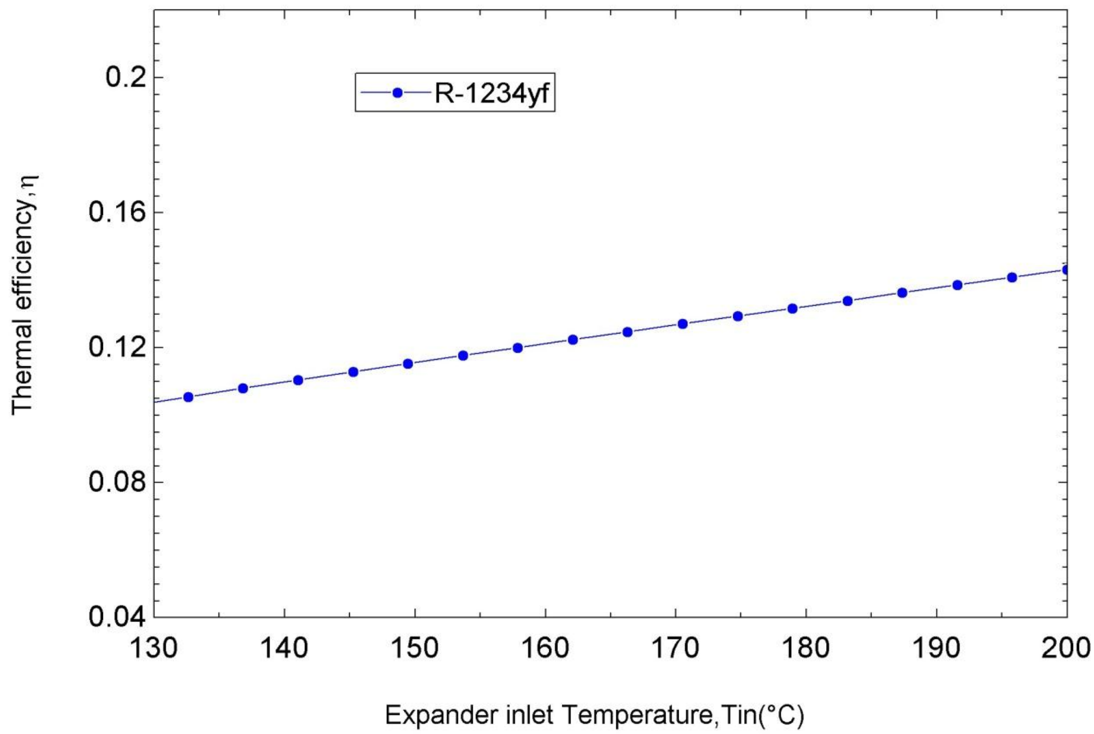


Fig.4.13.Thermal Efficiency For Supercritical cycle with,HFC-1234yf

4.14. Effect of Expander inlet Temperature on Thermal Efficiency in Supercritical by using HFC-134a

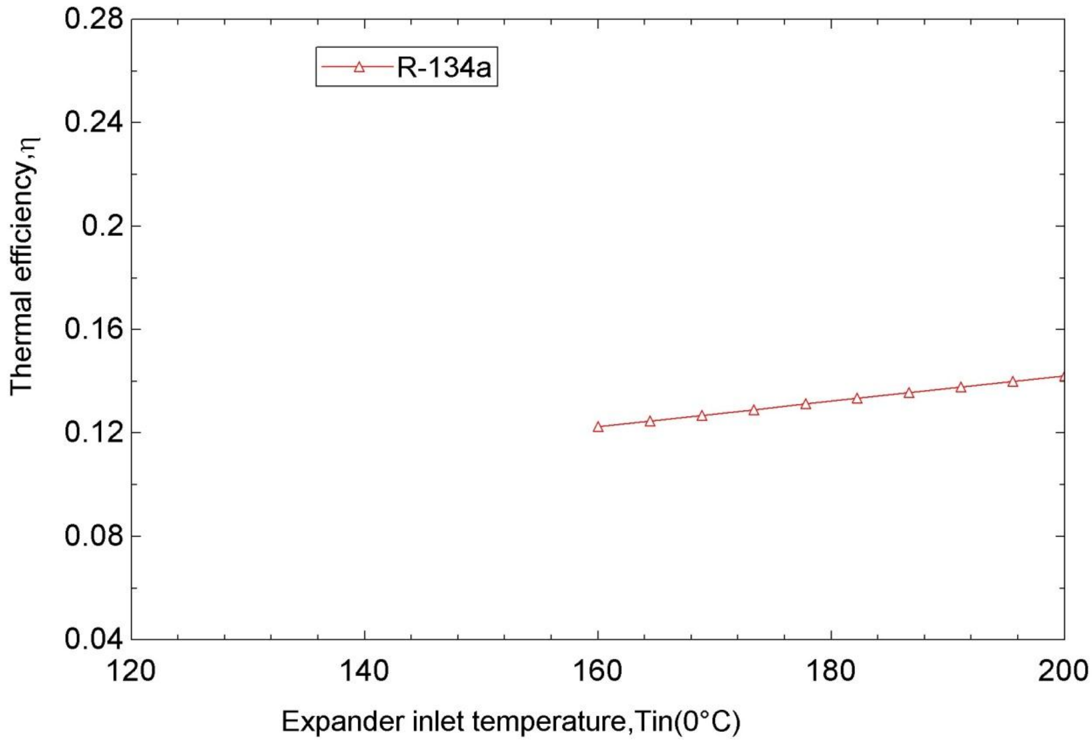


Fig.4.14.Thermal Efficiency For Supercritical cycle with,HFC-134a

4.15. Effect of Expander inlet Temperature on Thermal Efficiency in supercritical cycle by using HFC-245a

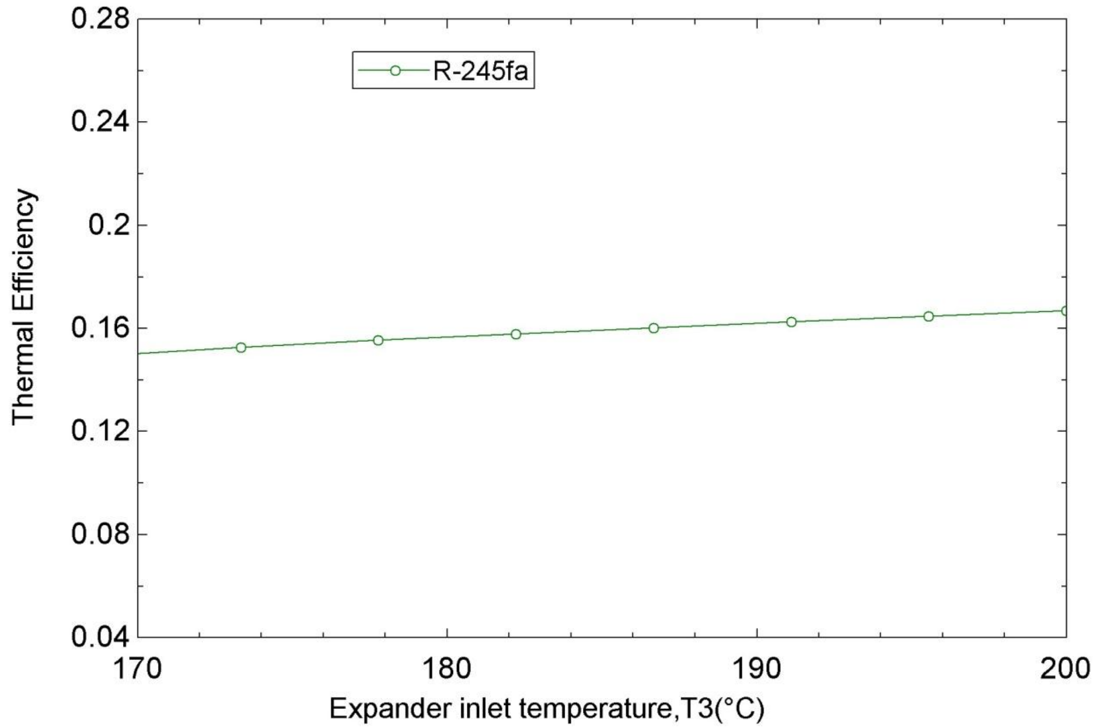


Fig.4.15.Thermal Efficiency For Supercritical cycle with,HFC-245

4.16 .Effect of Expander Inlet temperature on Thermal Efficiency of Supercritical by using Isopentane

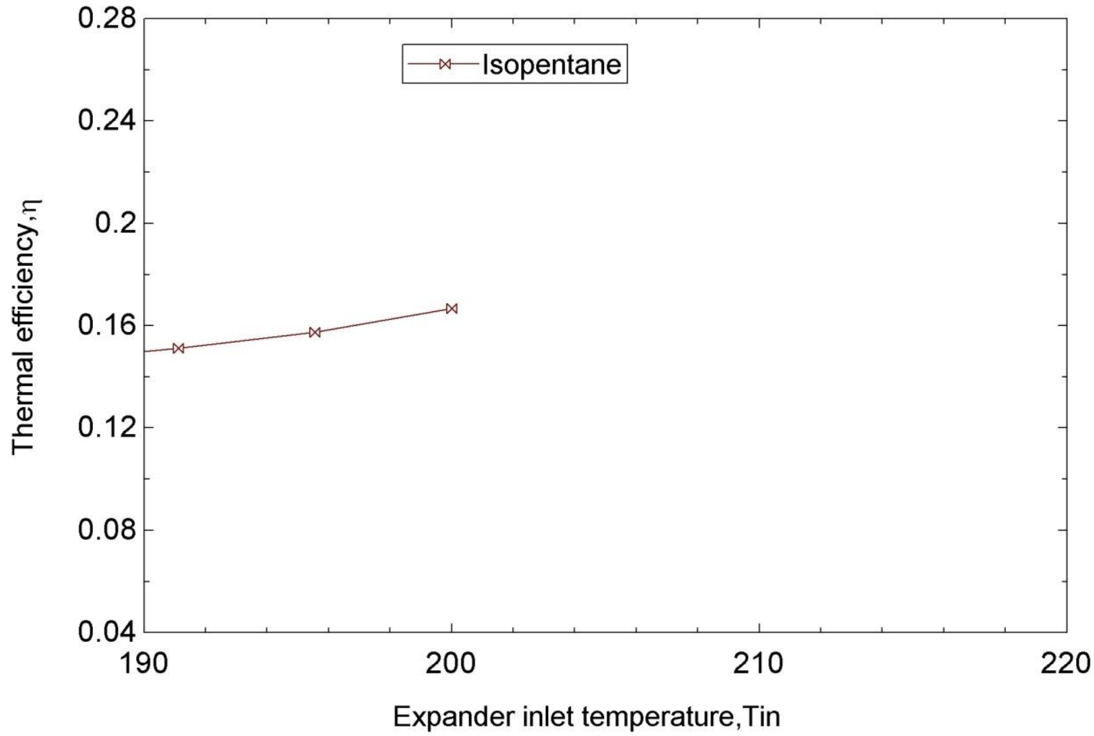


Fig.4.16.Thermal Efficiency For Supercritical cycle with Isopentane

4.17. Relative Comparison of Thermal Efficiency for various working Fluids in Supercritical cycle.

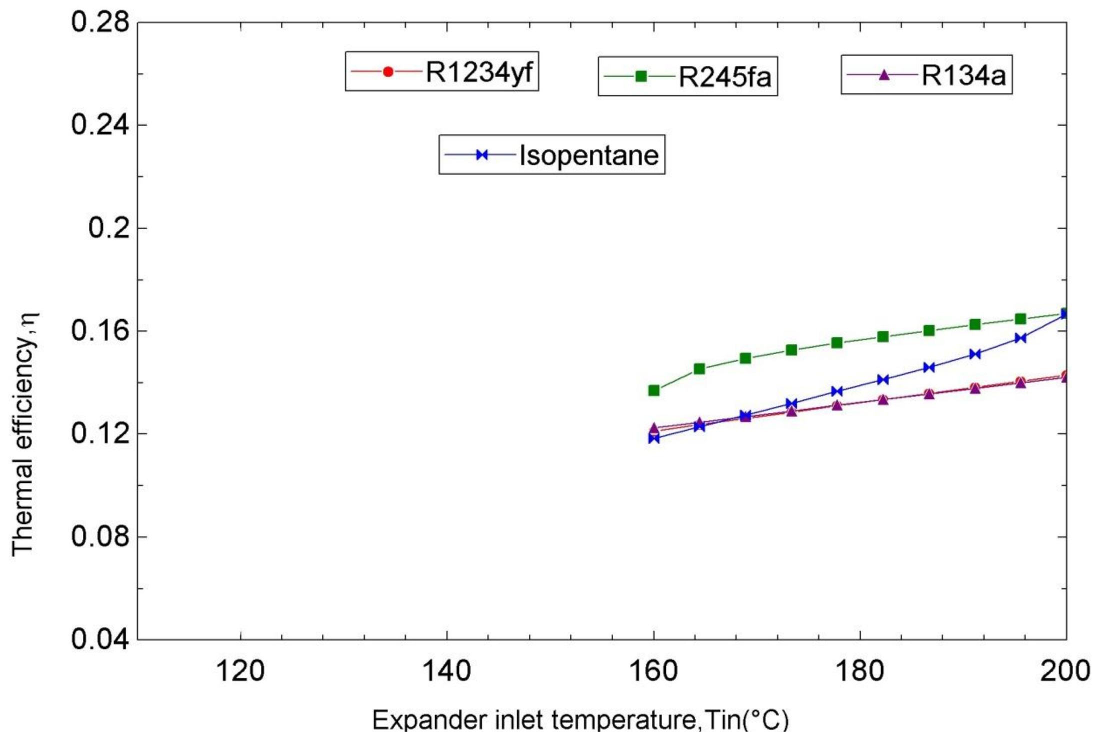


Fig.4.17 **Comparison of Thermal Efficiency For Supercritical cycle with,(HFC-245fa,HFO-1234YF,Isopentane,HFC-134a,Ethanol)**

Fig 4.17 shows the variation of thermal efficiency with expander inlet temperature for super critical cycle for various working fluids.

It has been observed that R-245fa shows higher thermal efficiency, R-134a shows lower thermal efficiency for expander inlet temperature (100-200).

4.18. Effect of Expander inlet temperature on Thermal Efficiency by using HFO-1234yf in Superheated cycle at pump pressure Ratio (γ=2.5)

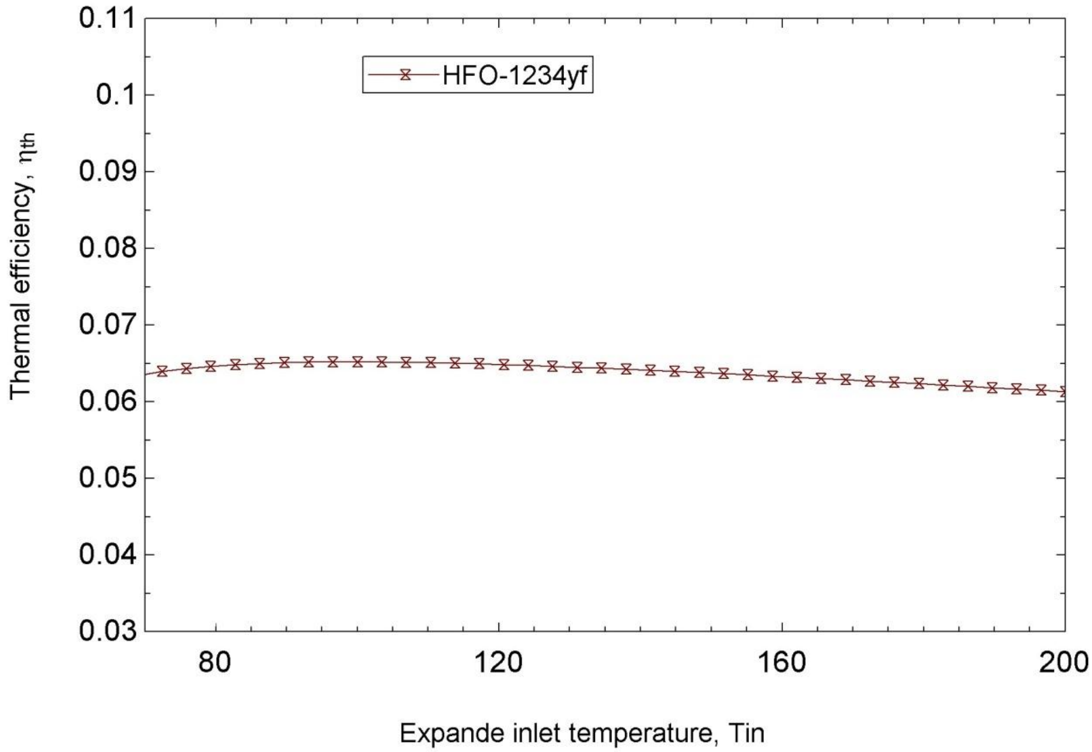


Fig.4.18 Thermal Efficiency For superheated cycle with,HFC-1234yf

4.19. Effect of Expander inlet on Thermal Efficiency in superheated cycle usinR-134a at pump pressure Ratio (γ=2.5)

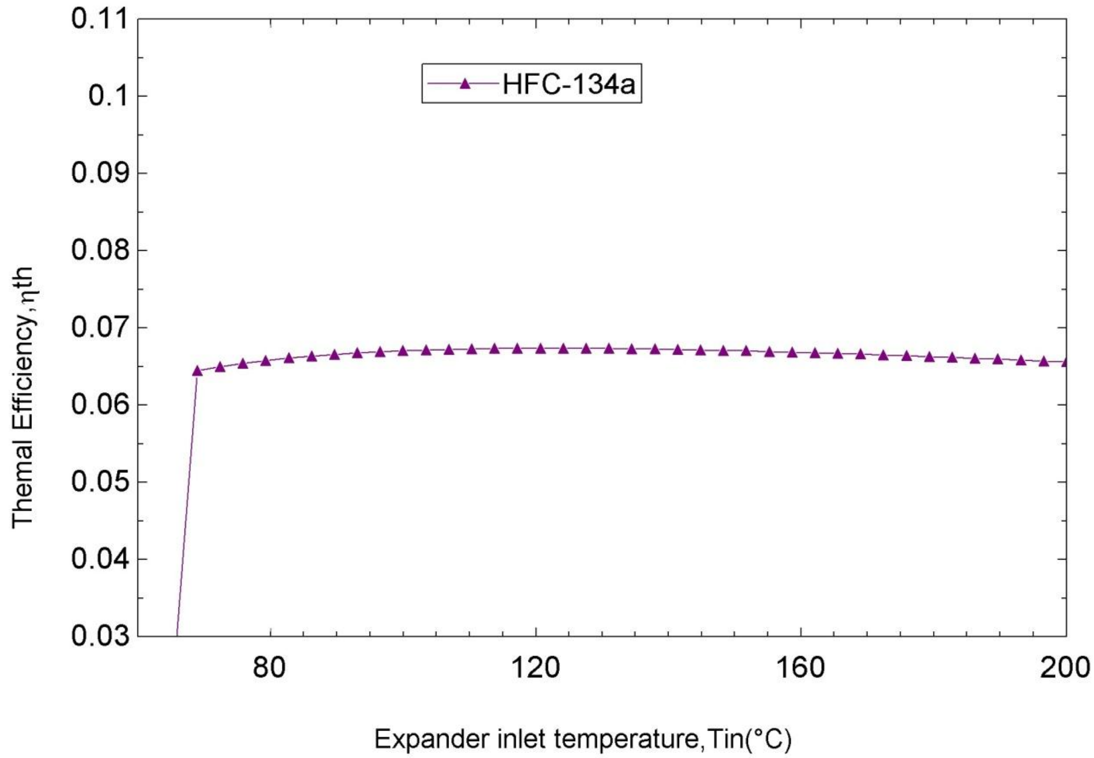


Fig.4.19Thermal Efficiency Forsuperheated cycle with,HFC-134a

4.20. Effect of Expander inlet Temperature on thermal efficiency in superheated cycle by using Iso-pentane at pump pressure Ratio (γ=2.5)

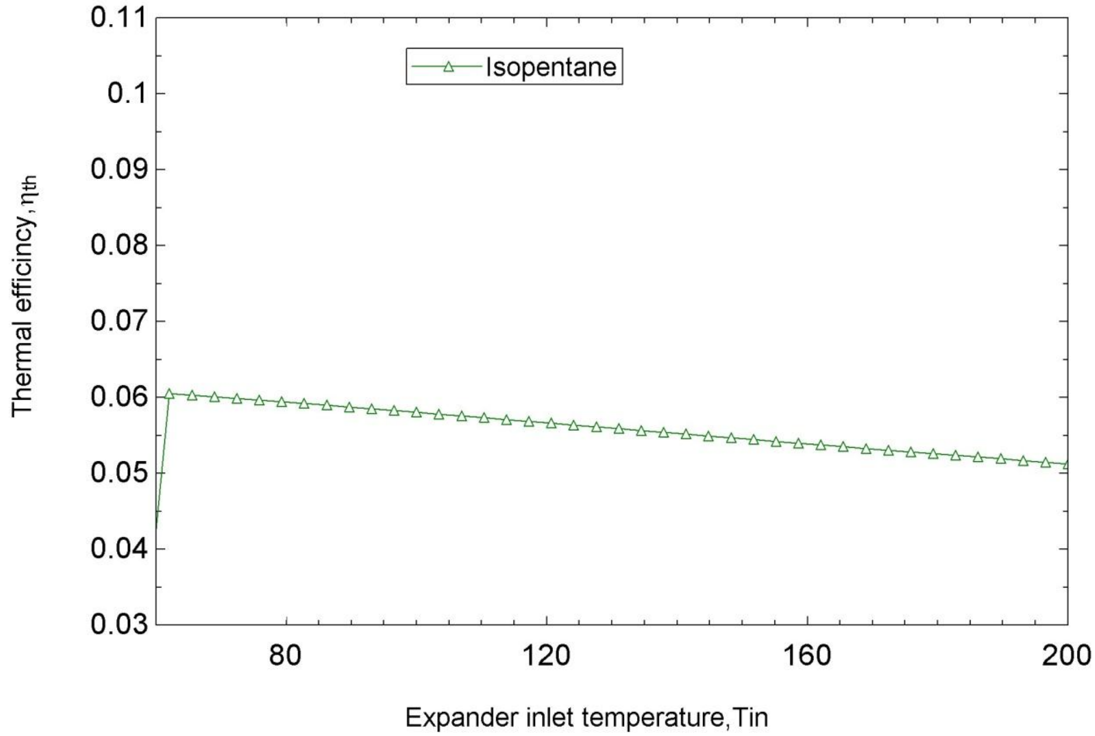


Fig. 4.20 Thermal Efficiency For superheated cycle with,Isopentane

4.21. Effect of expander inlet temperature in superheated cycle using ethanol at pump pressure Ratio (γ=2.5)

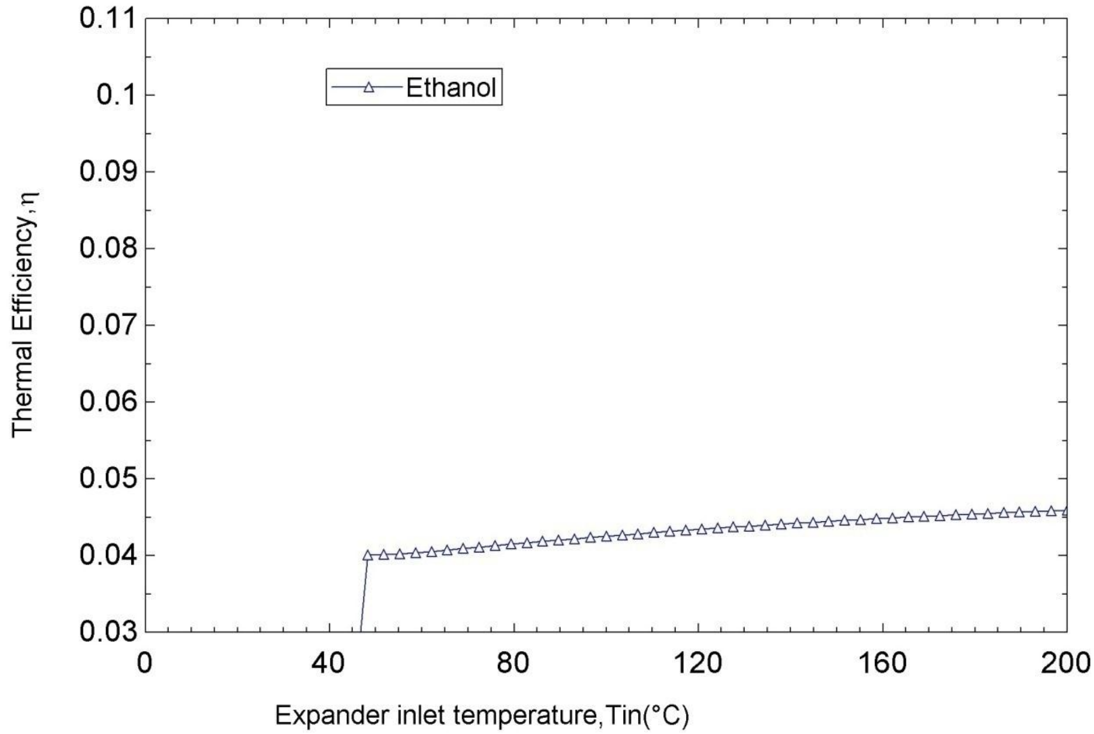


Fig. 4.21 Thermal Efficiency For superheated cycle with,HFC-1234yf

4.22 Effect of Expander Inlet Temperature on Thermal Efficiency in Superheated Cycle in super heated cycle pump pressure ratio (γ=2.5)

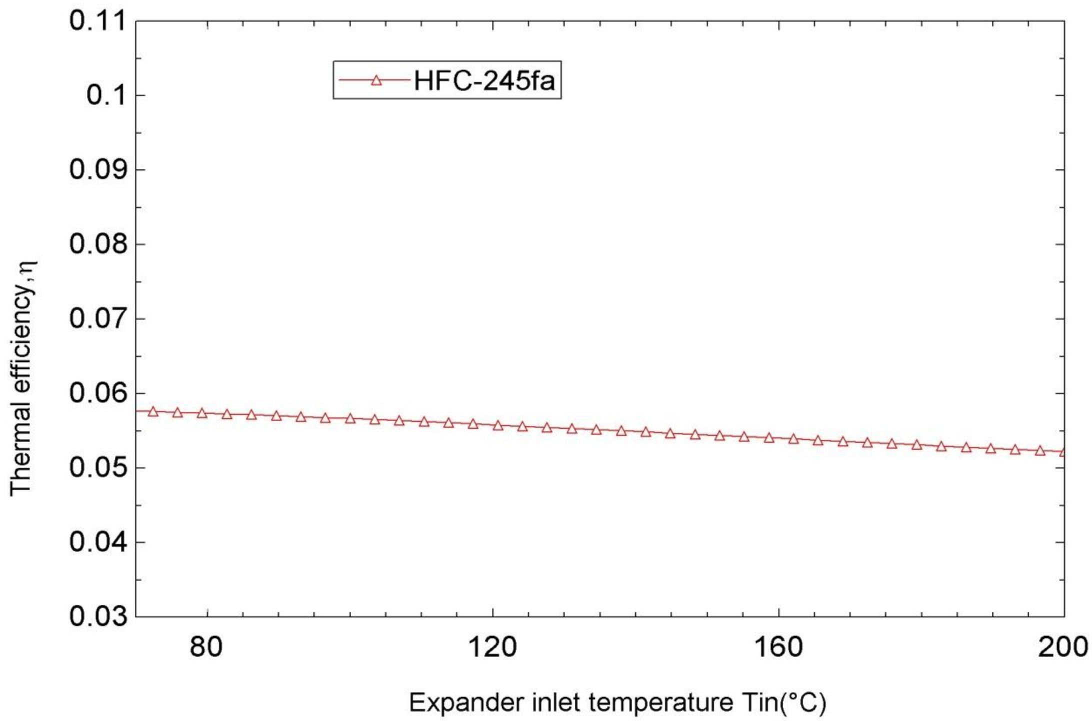


Fig. 4.22 Thermal Efficiency For superheatedcycle with,HFC-245fa

4.23 Relative comparision of thermal efficiency by using various working fluid in super-heated cycle pump pressure Ratio (γ=2.5)

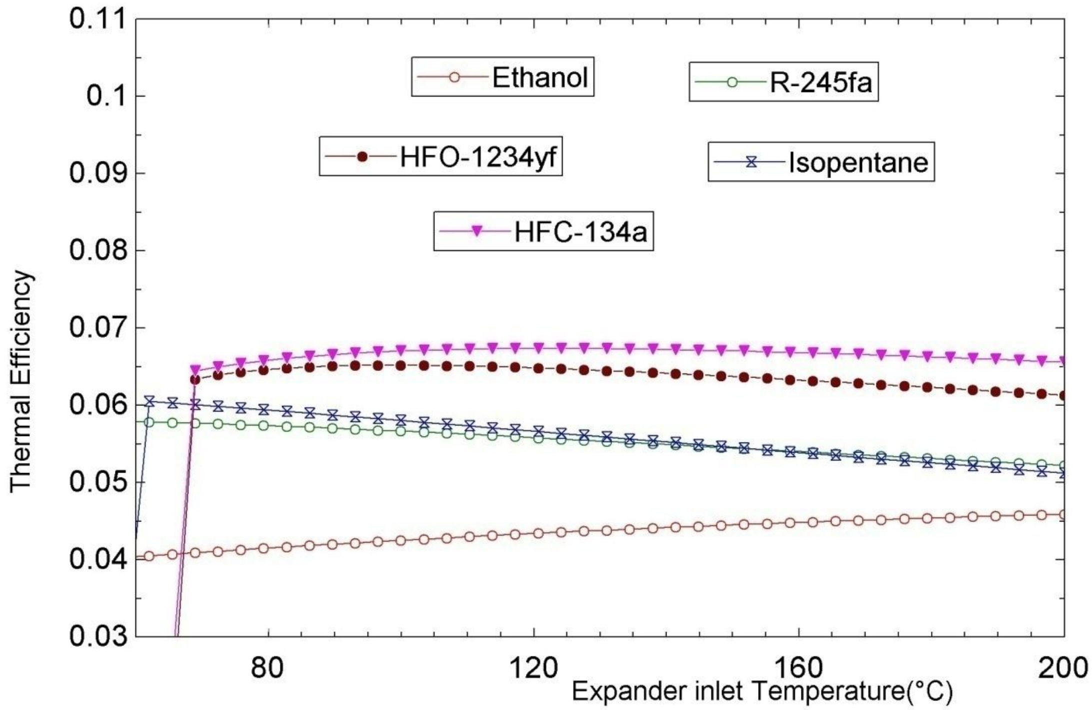


Fig. 4.23 **Comparison of Thermal Efficiency For superheated cycle with, (HFC-245fa, HFO-1234YF, Isopentane, HFC-134a, and Ethanol)**

Fig 4.23 shows the variation of thermal efficiency with expander inlet temperature for super heated cycle for various working fluids.

It has been observed that HFC-134a shows higher thermal efficiency, Ethanol shows lower thermal efficiency for expander inlet temperature (70-200).

From above Thermal efficiency distibution it is observed that at pressure ratio (γ=2. 5) HFC-134a shows highest thermal Efficiency.

At Expander inlet Temperature 68˚C the Thermal Effciency varies from 6. 4 to 6. 6% of HFO-134a.

At Expander inlet temperature 68 to 200˚C The Thermal Efficiency Varies From 6. 4 to 6. 1% by using HFO-1234yf.

HFC-134a and HFO-1234yf has showed maximum thermal Efficiency (6. 7%) at Expander inlet temperature 122˚C and 6. 6% at 92˚C, respectively.

HFC-245 and Isopentane shows the Thermal Efficiency in The middle, while ethanol showed the lowest thermal Efficiency.

TrilateraL cycle:

In Trilateral ORC Cycle with HF0-1234yf shows the Lowest Thermal Effiency (0-5. 4%) for the Temperature range (30-90˚C).

HFC-134a has approximately 0. 1% higher thermal Efficiency than HFO-1234yf at The same Temperature level (50-75˚C).

As compared to other working fluid at the same Expander inlet Temperature HFO-1234yf shows a thermal Efficiency 1%, 2% and 1. 5% Lower than that of HFC-245fa, Isopentane, and ethanol, respectively.

Ethanol shows Highest Thermal Efficiency Range (9. 9% to18. 7%) in the Temperature Range (80 to160˚C).

Saturated Cycle

In saturated Rankine cycle model by using HFO-1234yf shows Lowest Thermal Efficiency (0-8%) as compared to other working fluid in the temperature Range (30-90$^{\circ}c$).

Super critical ORC model.

From Comparative study in supercritical cycle by using different Types of The Working Fluid, it is clear that HFO-1234yf produced (η_{th}=8. 7to 9%) fo the temperature range (105 to 155$^{\circ}c$) and then decreased to 8. 8% at 200˚C.

For the temperature range 155˚C the super critical cycle has maximum thermal efficiency (7. 8-10%).

HFO-1234yf shows lower thermal efficiency than HFC-134a which is less than 1. 7%.

4.24 Exergetic Efficiency Comparision of superheated cycle by using various working fluid in superheated cycle.

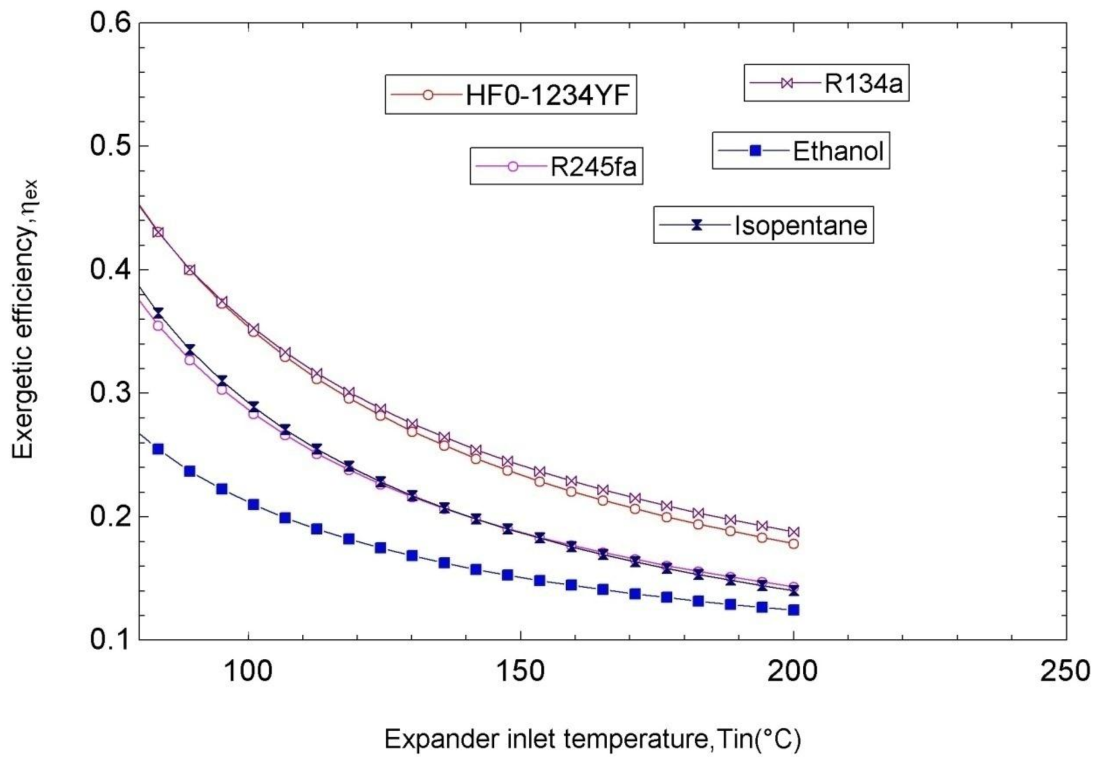


Fig. 4.24 **Comparison of Exergetic Efficiency For superheated cycle with, (HFC-245fa, HFO-1234YF, Isopentane, HFC-134a, and Ethanol)**

Fig 4.24 shows the variation of Exergetic efficiency with expander inlet temperature for saturated cycle for various working fluids.

R-134a shows higher exergetic efficiency, Ethanol shows lower thermal efficiency for expander inlet temperature (90-250).

4.25 Exergetic Efficiency Comparision of various Working Fluid in Saturated Cycle:

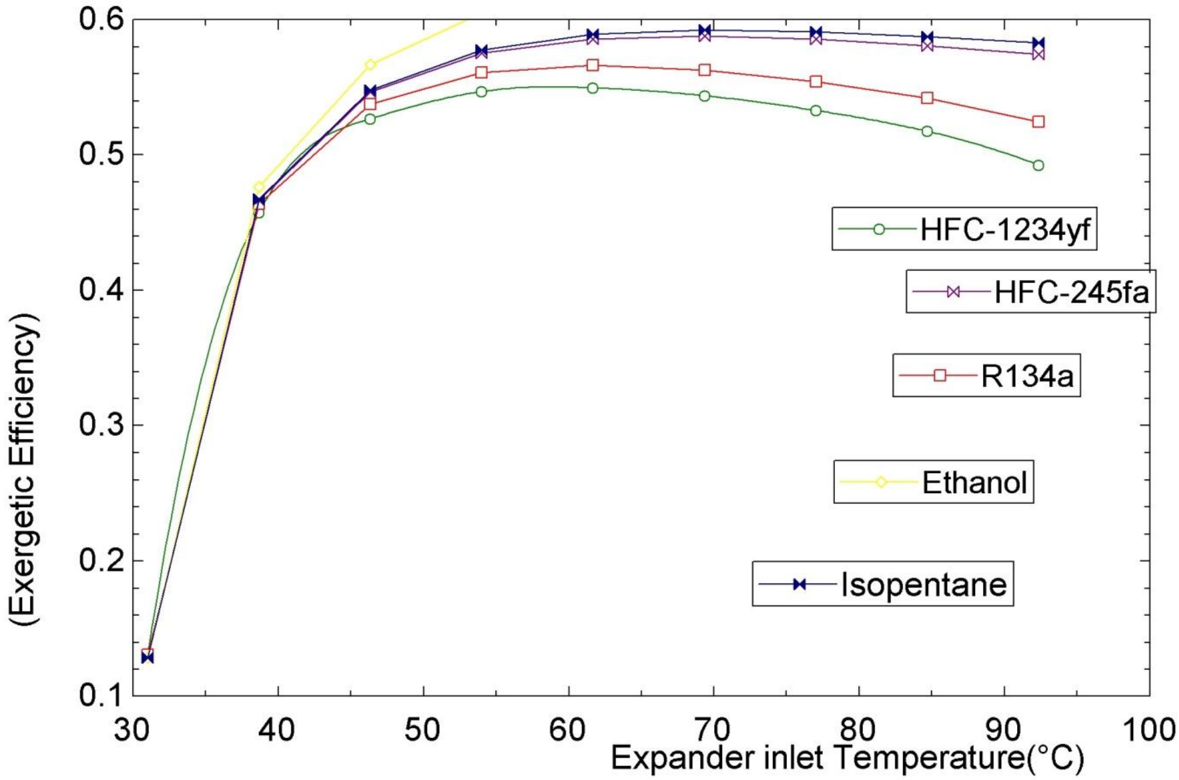


Fig. 4.25 **Comparison of Exergetic Efficiency for Saturated cycle with, (HFC-245fa, HFO-1234yf, Isopentane, HFC-134a, Ethanol)**

Fig 4.25 shows the variation of Exergetic efficiency with expander inlet temperature for saturated cycle for various working fluids.

HFO-1234yf shows lower exergetic efficiency, Isopentane shows higher thermal efficiency for expander inlet temperature (30-100).

4.26 .Exergetic Efficiency Comparison of Working Fluid in Trilateral (ORC):

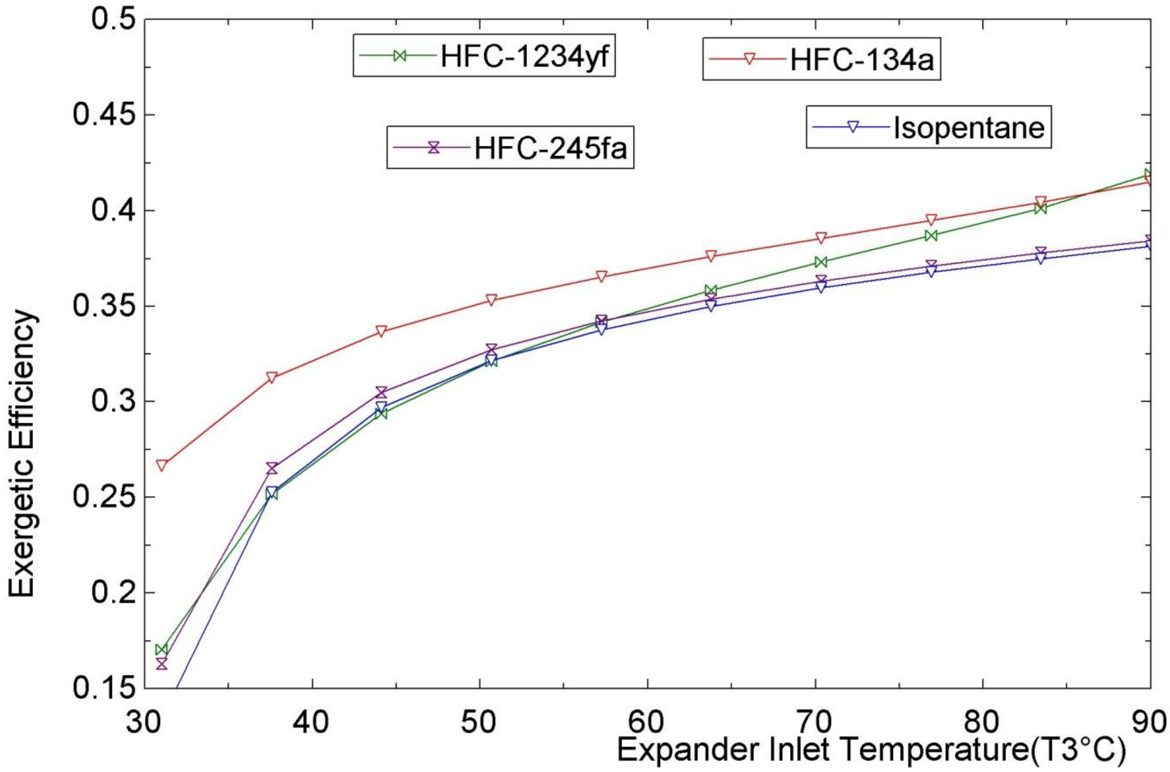


Fig. 4.26 **Comparison of Exergetic Efficiency for Trilateral cycle with, (HFC-245fa, HFO-1234YF, Isopentane, HFC-134a, Ethanol)**

Fig 4.26 shows the variation of Exergetic efficiency with expander inlet temperature for trilateral cycle for various working fluids.

HFO-134a shows higher exergetic efficiency, Isopentane shows lower thermal efficiency for expander inlet temperature (30-90).

4.27. Exergetic Efficiency comparison in supercritical (0RC) cycle by using different working fluid.

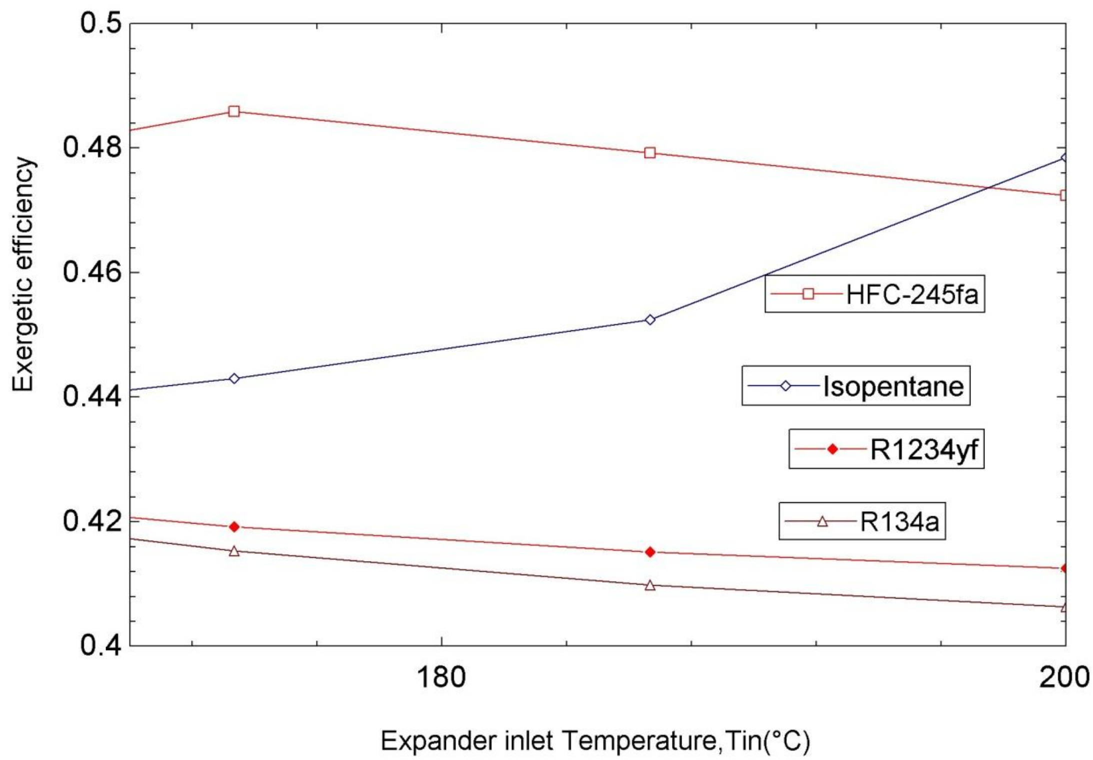


Fig. 4.27 **Comparison of Exergetic Efficiency for Supercritical cycle with, (HFC-245fa, HFO-1234YF, Isopentane, HFC-134a, Ethanol)**

Fig 4.27 shows the variation of Exergetic efficiency with expander inlet temperature for supercritical cycle for various working fluids.

HFO-245fa shows higher exergetic efficiency, R-134a shows lower thermal efficiency for expander inlet temperature (120-200).

4.28 Exergy destruction in expander by using R-1234yf in supercritical cycle.

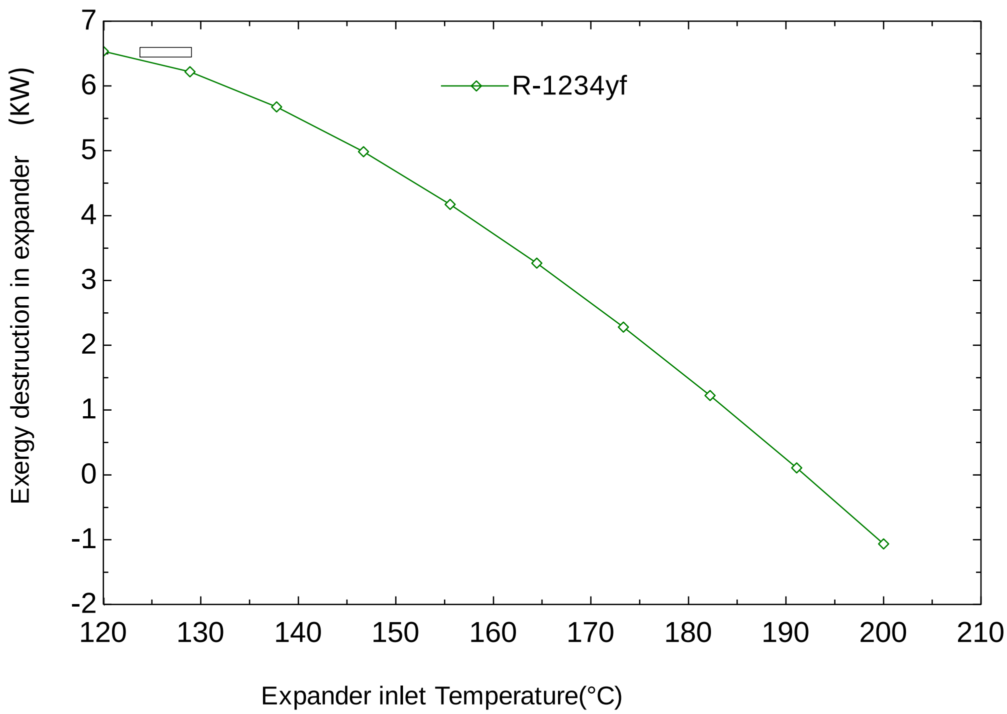


Fig.no.4.28 Exergy destruction of expander in super critical cycle with R-1234yf

4.29 Exergy destruction in condenser by using R-1234yf in supercritical cycle

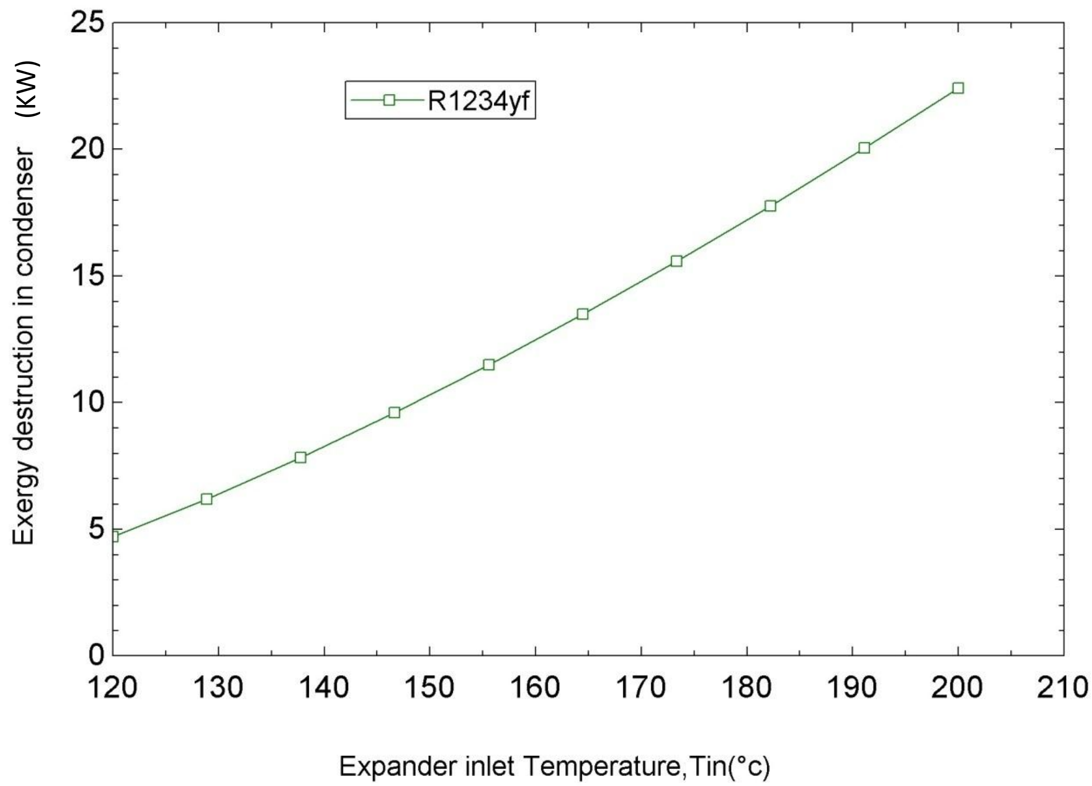


Fig.no.4.29 Exergy destruction of condenser in super critical cycle with R1234yf

4.30 **Exergy destruction of Evaporator by using R-1234yf in supercritical cycle**

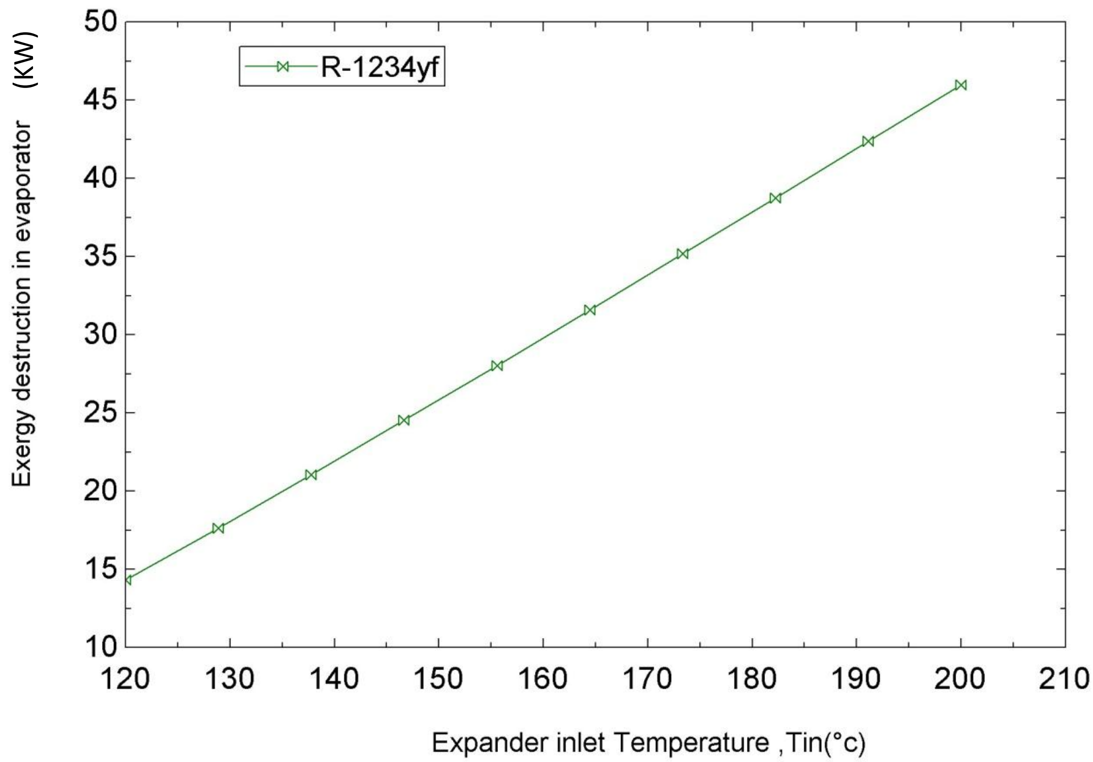


Fig.no.4.30 Exergy destruction of condenser in super critical cycle with R-1234yf

4.31 Total Exergy destruction in super critical cycle by using R-1234yf

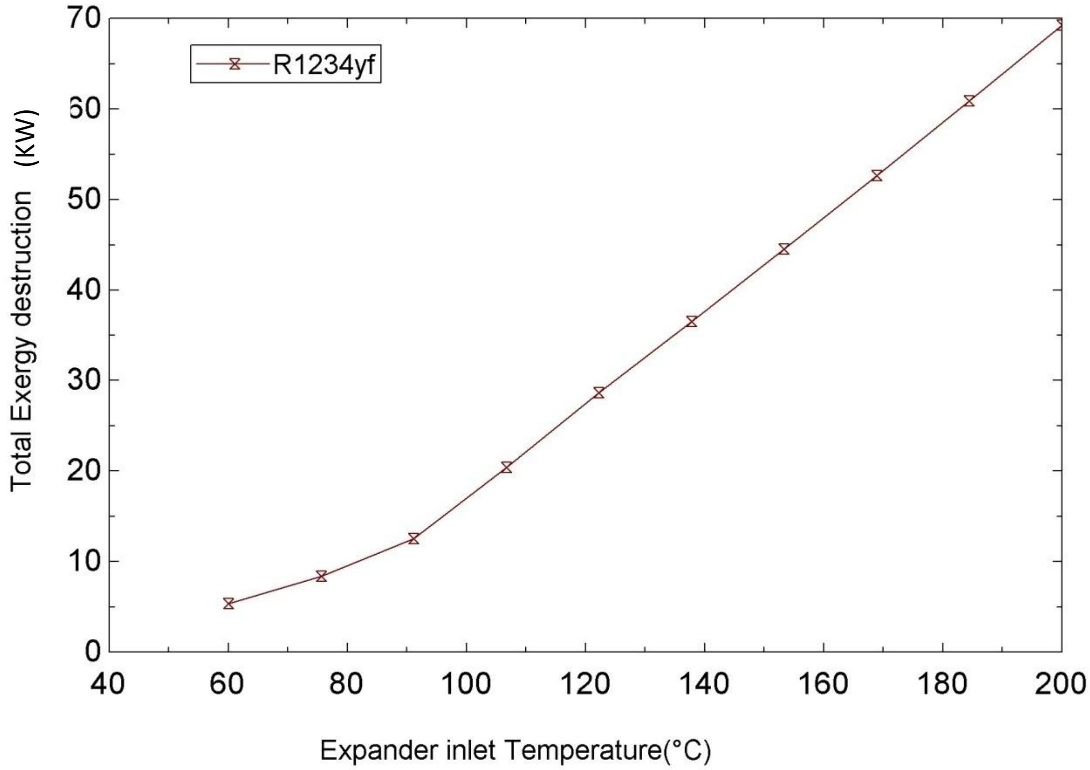


Fig.no.4.31Total exergy destruction of condenser in super critical cycle with R-1234yf

4.32. Exergy destruction of Expander in saturated cycle by using HF0-1234yf.

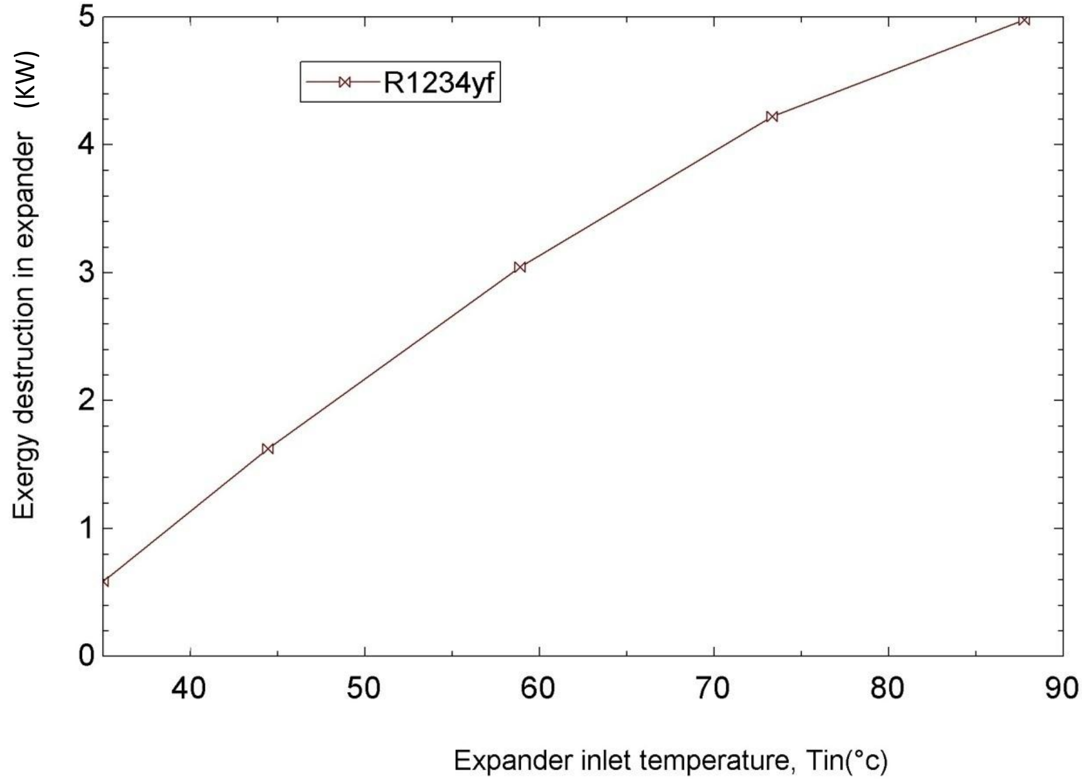


Fig.no.4.32 Exergy destruction of Evaporator in saturated cycle with R-1234yf

4.33 Exergy destruction of condeser in saturated cycle by using HF0-1234yf.

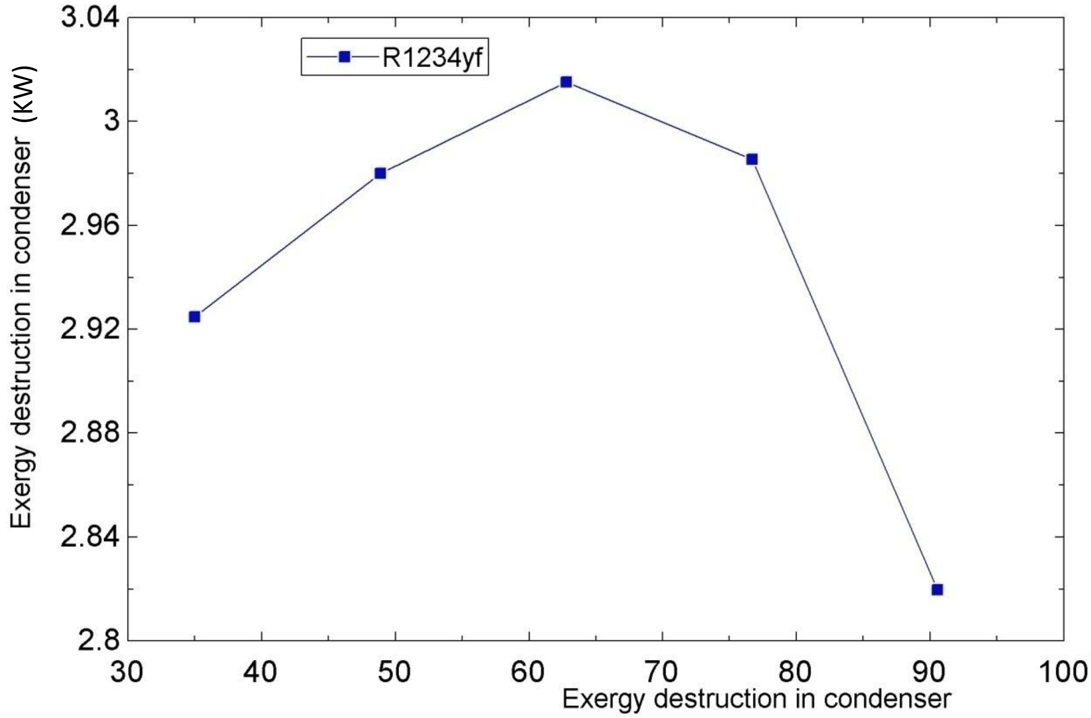


Fig.no.4.33 Exergy destruction of Expander in saturated cycle with R-1234yf

4.34. Exergy destruction of Evaporator in saturated cycle by using HF0-1234yf.

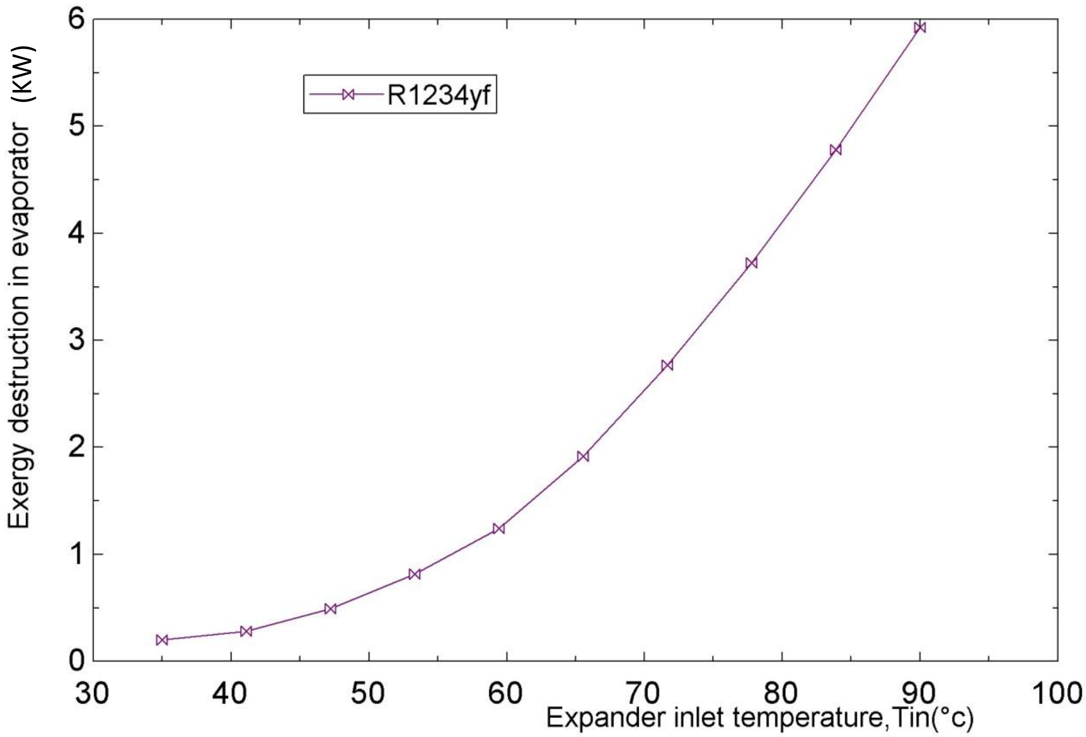


Fig.no.4.34 Exergy destruction of Evaporator in saturated cycle with R-1234yf

4.35.Exergy destruction of pump in saturated cycle by using HF0-1234yf.

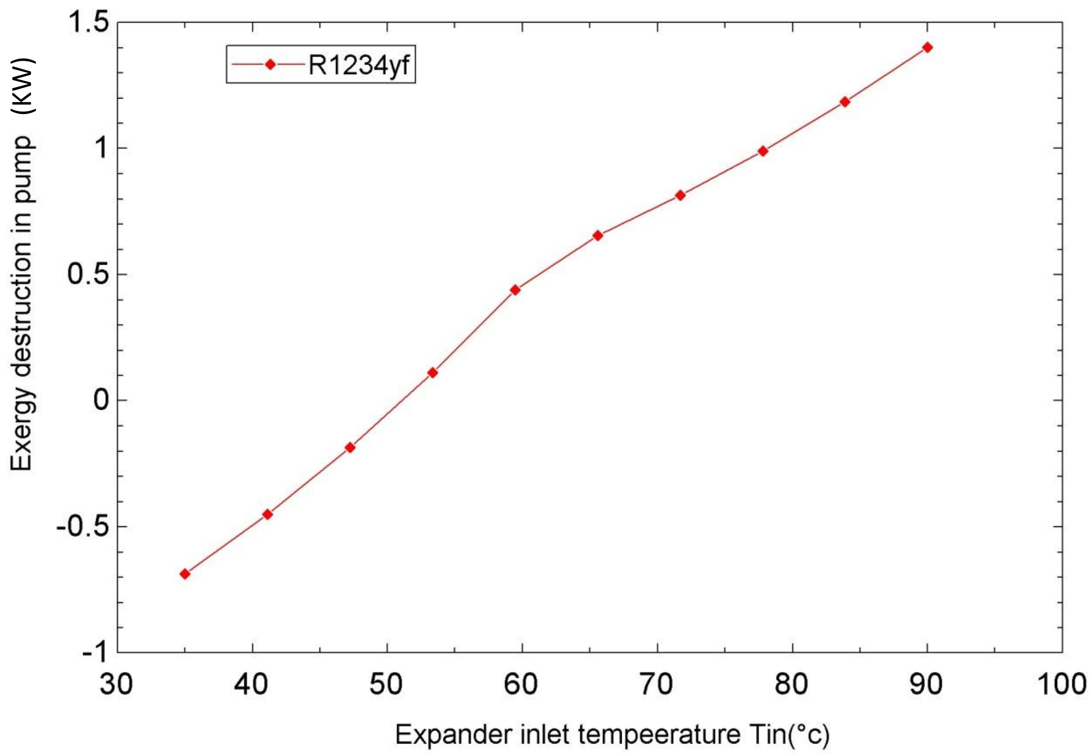


Fig.no.4.35 Exergy destruction of pump in saturated cycle with R-1234yf

4.36.Total exergy destruction of Saturated rankine cycle by using HF0-1234yf.

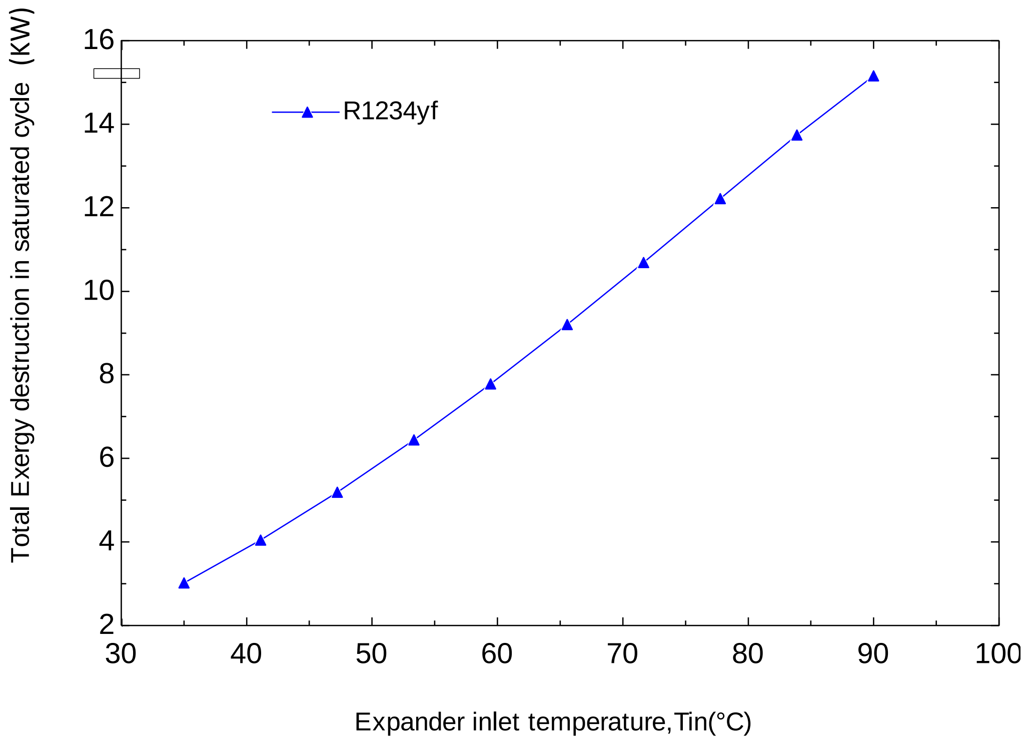


Fig.no.4.36 Total exergy destruction of pump in saturated cycle with R-1234yf

4.37 Exergy destruction of expander in superheated cycle for HFO-1234yf.

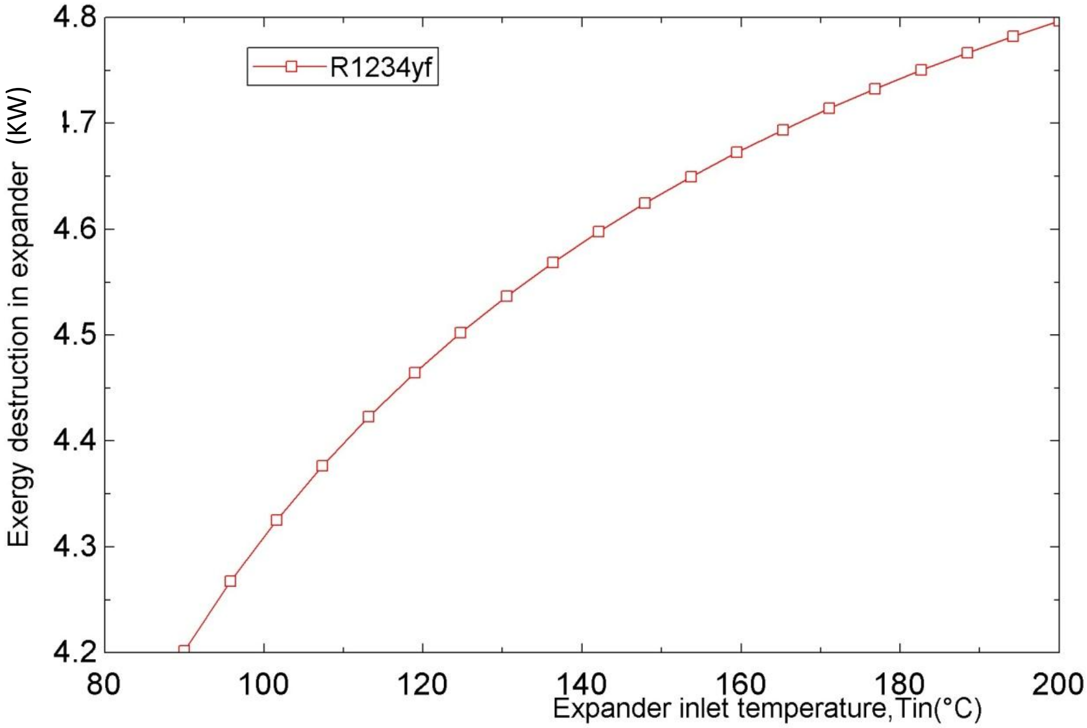


Fig.4.37 Exergy destruction of expander in superheated cycle with R-1234yf

4.38. Exergy destruction of condenser in superheated cycle for HFO-1234yf.

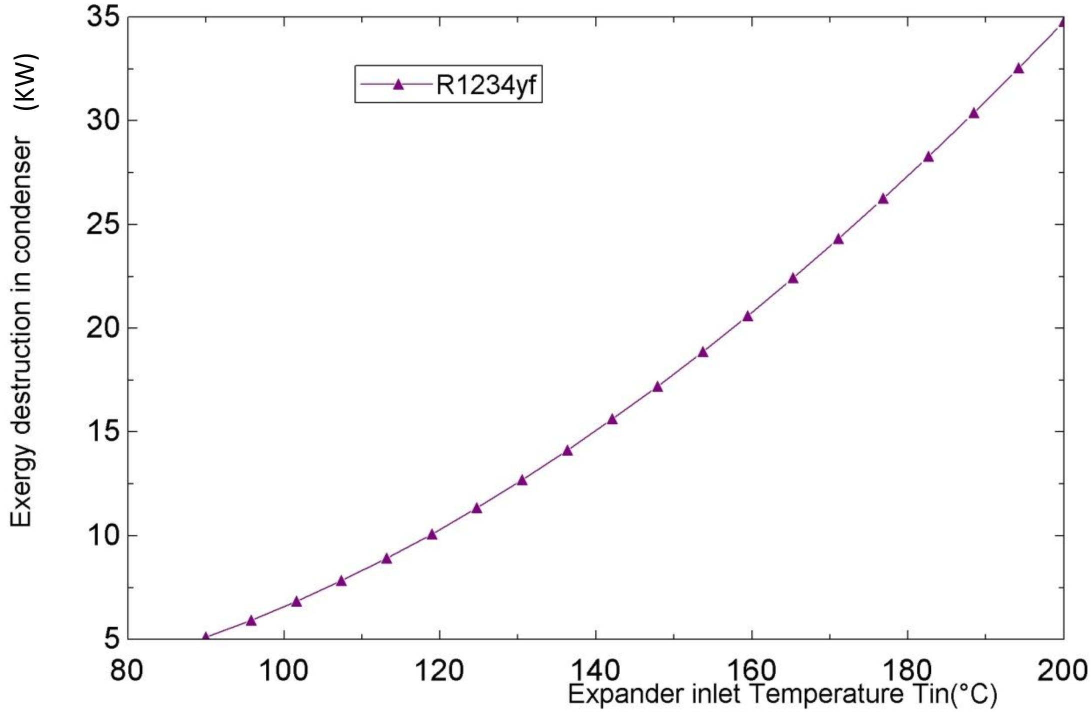


Fig.4.38 Exergy destruction of condenser in superheated cycle with R-1234yf

4.39. Exergy destruction of evaporator in superheated cycle for HFO-1234yf

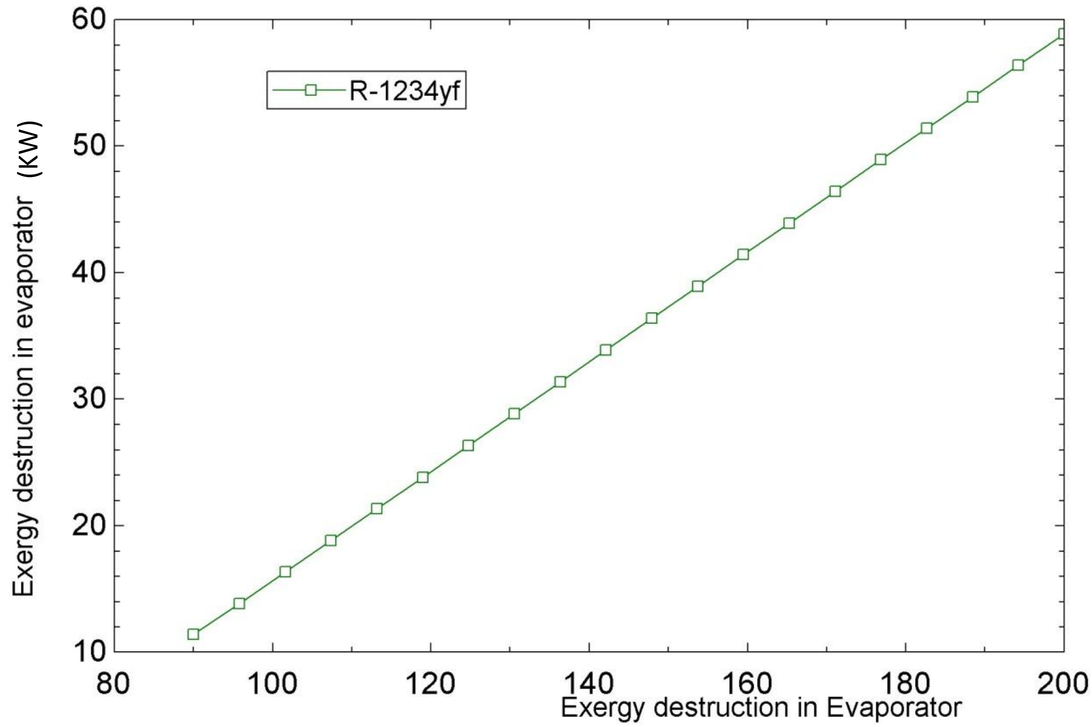


Fig.4.39Exergy destruction of Evaporator in superheated cycle with R-1234yf

Total exergy destruction in superheated cycle for HFO-1234yf

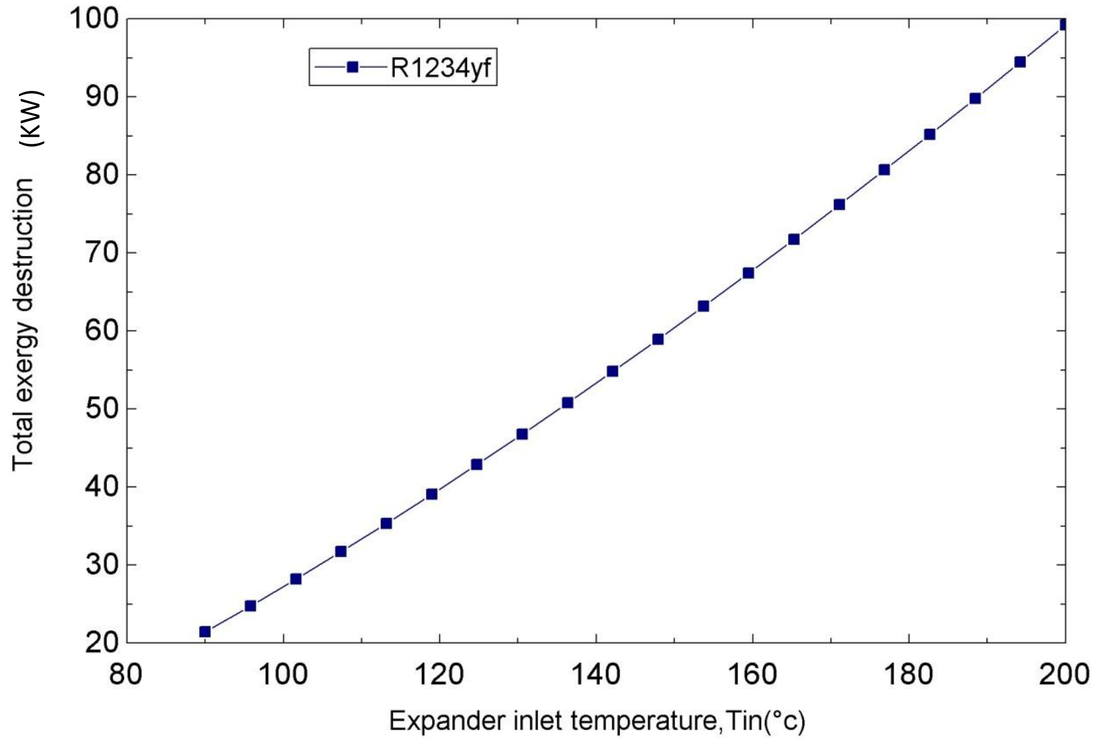


Fig.4.40Total exergy destruction in superheated cycle with R-1234yf

4.41. Exergy destruction of expander in trilateral cycle.

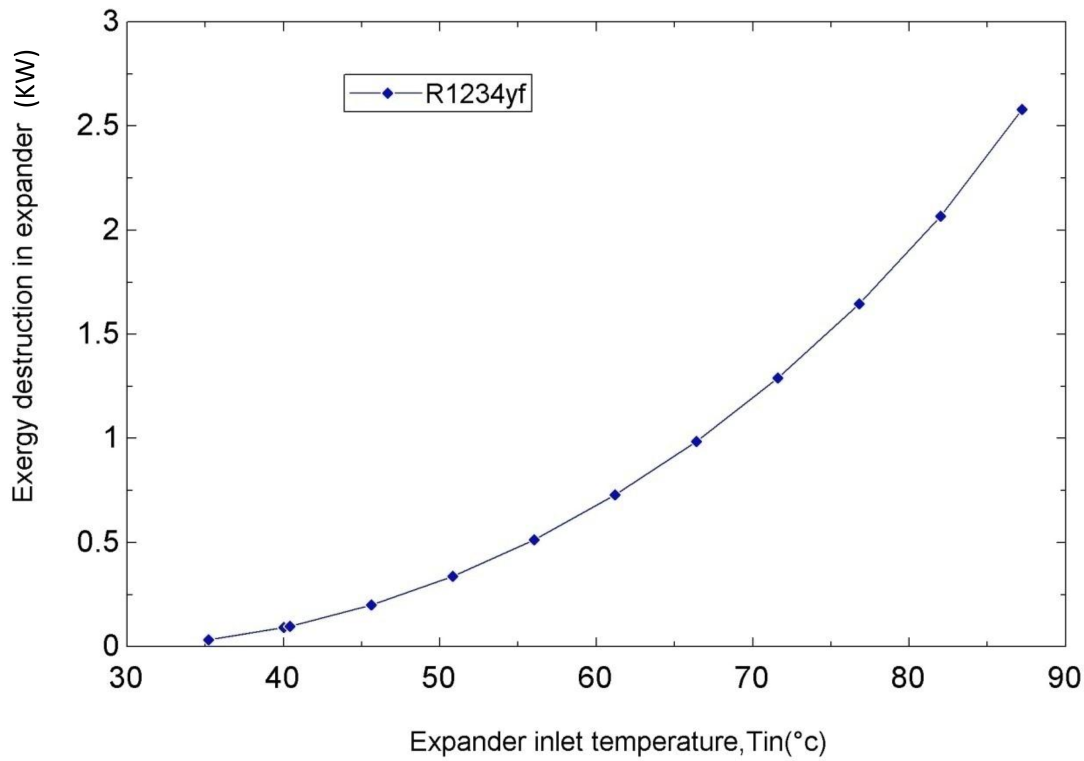


Fig.4.41Exergy destruction of expander in trilateral cycle with R-1234yf

4.42. Exergy destruction of condenser in trilateral cycle.

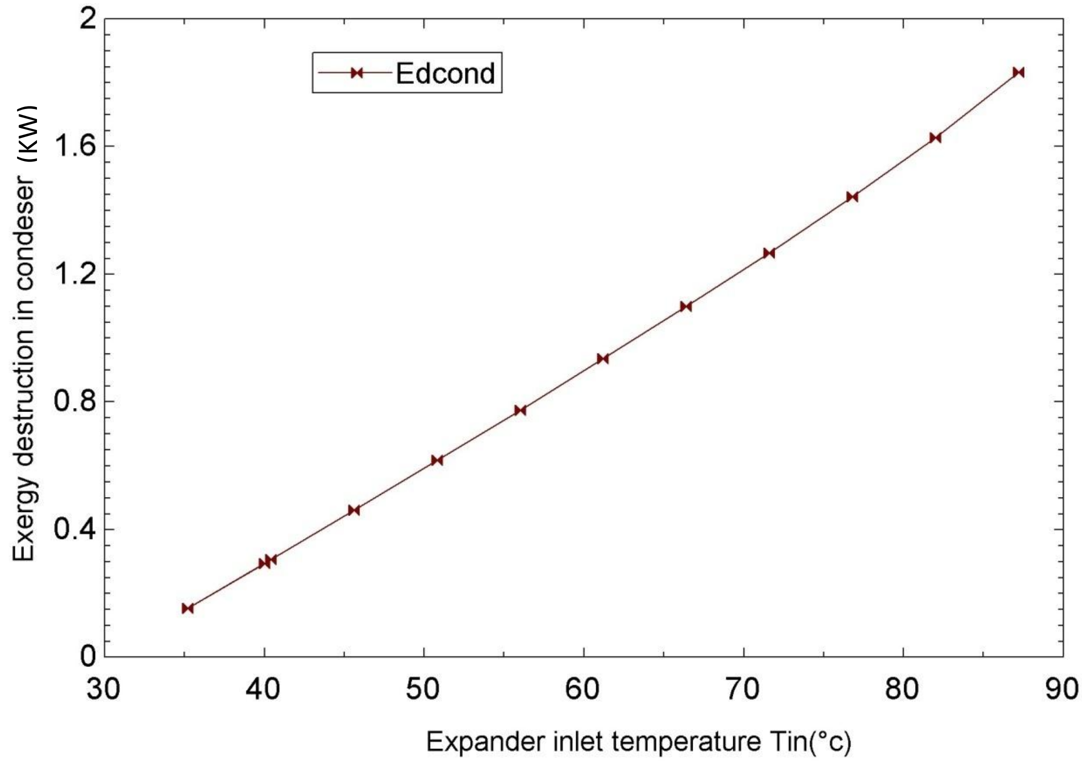


Fig.4.42 Exergy destruction of condenser in Trilateral cycle with R-1234yf

4.43 Exergy destruction of Evaporator in Trilateral cycle.

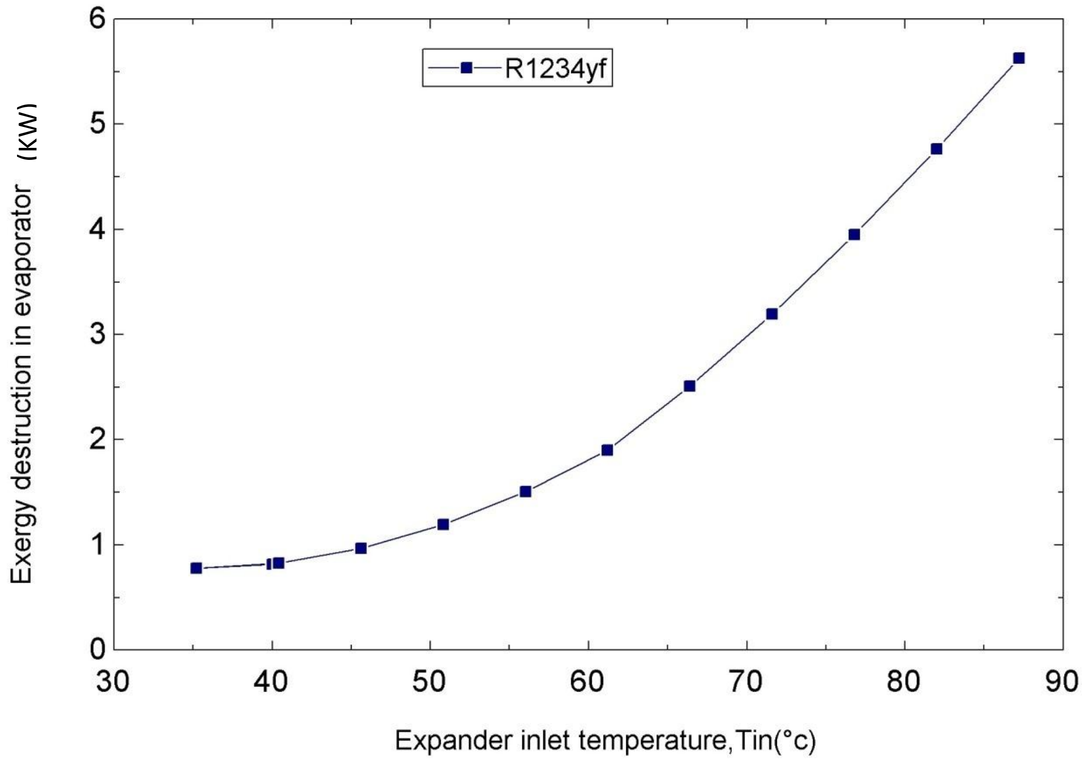


Fig.4.43 Exergy destruction of evaporator in Trilateral cycle with R-1234yf

4.44 Exergy destruction of pump in trilateral cycle.

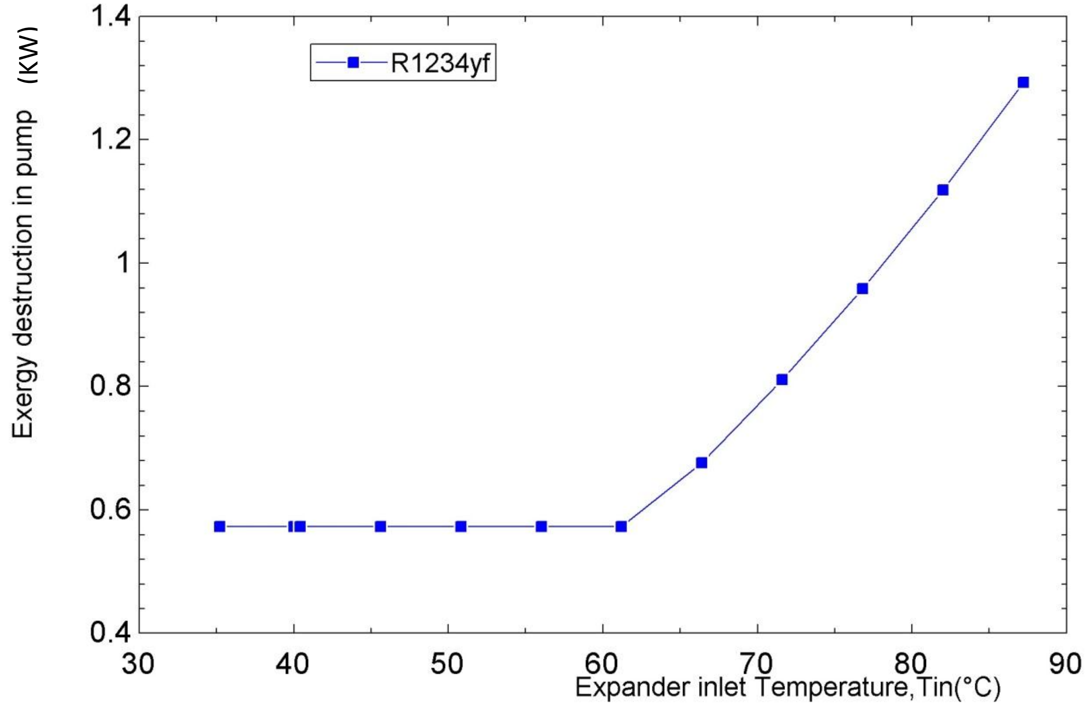


Fig.4.44 Exergy destruction of pump in Trilateral with R1234yf

4.45. Comparision of exergy destruction in various components (expander, evaporator, condenser, pump) in saturated cycle using HF0-1234yf.

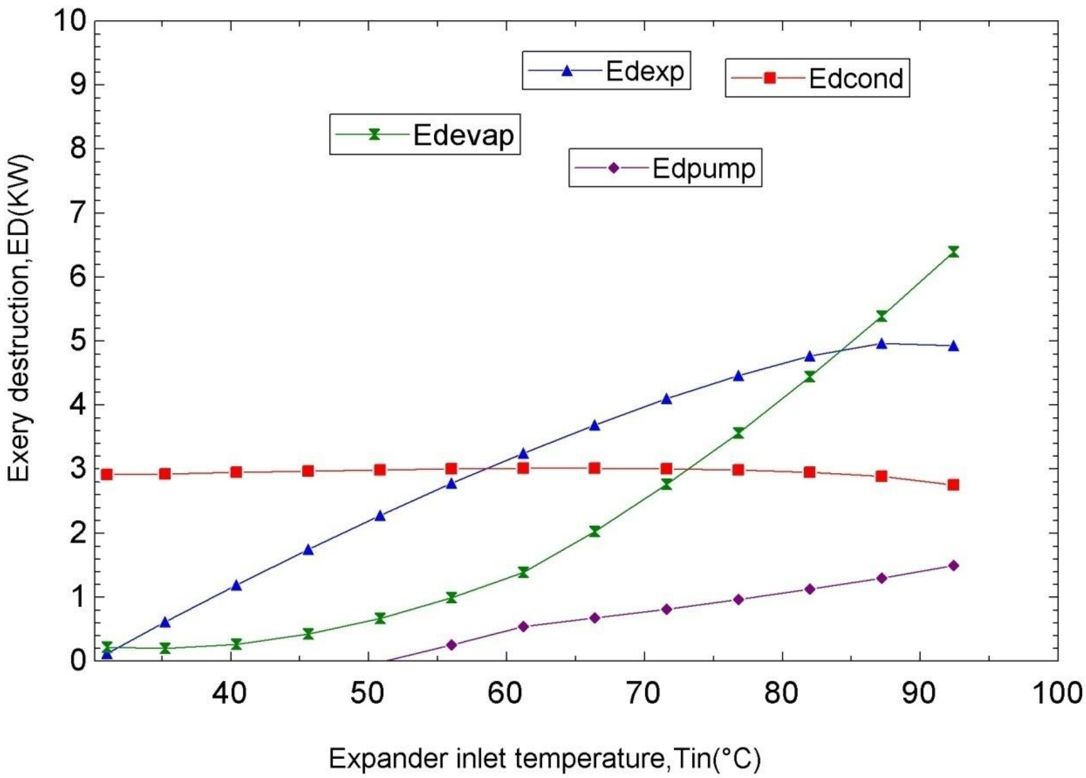


Fig.4.45 comparision of Exergy destruction in saturated cycle

Fig.4.45 shows the variation of Exergy destruction with expander inlet temperature (90-190) for super heated cycle for HFO-1234yf working fluid.

The exergy destruction in evaporator is higher as compare to expander, pump, condenser and pump shows the lower exergy destruction.

It has been observed that as the temperature decreases the exergy destruction in evaporator, pump, expander, condenser decreases.

4.46 Comparison of exergy destruction in various components (expander, pump, evaporator, condenser) in supercritical cycle using HFO-1234yf.

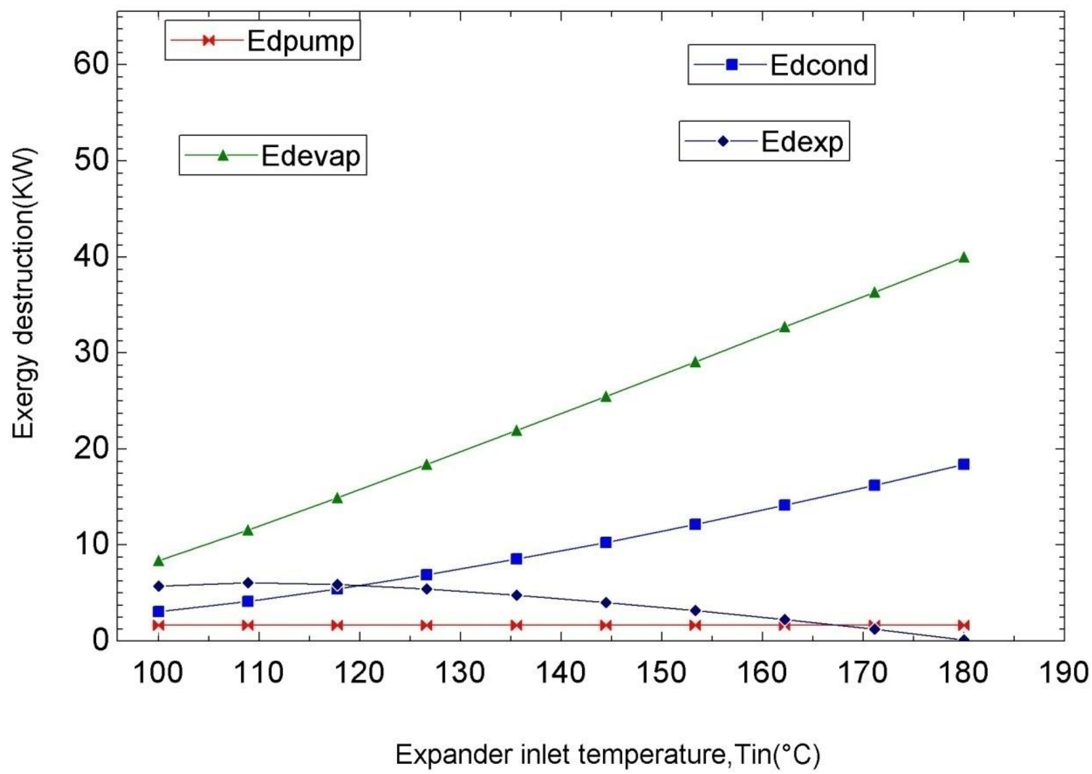


Fig.4.46comparision of Exergy destruction in supercritical cycle

Fig.4.46 shows the variation of Exergy destruction with expander inlet temperature (90-190) for super heated cycle for HFO-1234yf working fluid.

The exergy destruction in evaporator is higher as compare to expander, pump, condenser and pump shows the lower exergy destruction.

It has been observed that as the temperature decreases the exergy destruction in evaporator, pump, expander, condenser decreases.

4.47. Comparison of exergy destruction in various components (expander, pump, evaporator, and condenser) in superheated cycle using HFO-1234yf.

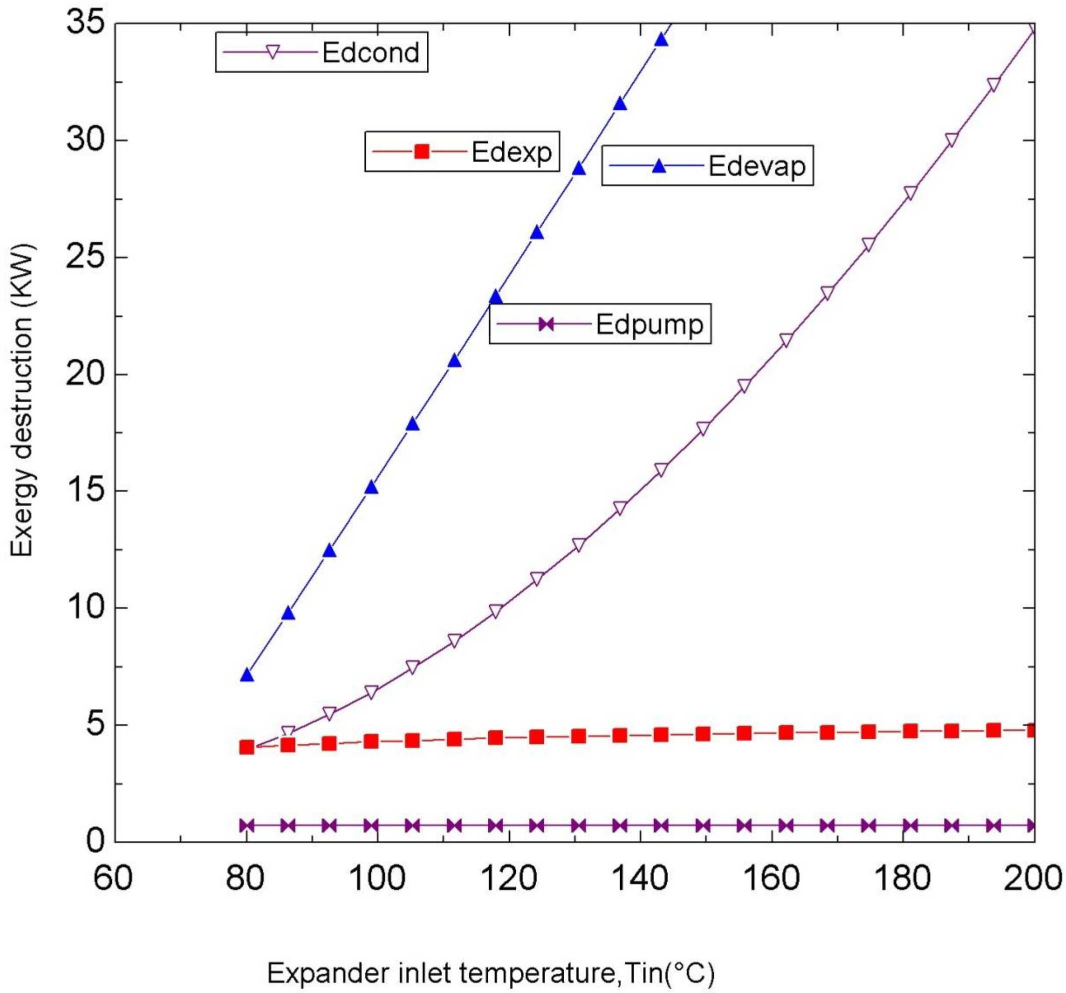


Fig.4.47comparision of Exergy destruction in superheated cycle

Fig.4.47 shows the variation of Exergy destruction with expander inlet temperature (60-200) for super heated cycle for HFO-1234yf working fluid.

The exergy destruction in evaporator is higher as compare to expander, pump, condenser and pump shows the lower exergy destruction.

It has been observed that as the temperature decreases the exergy destruction in evaporator, pump, expander, condenser decreases.

4.48 Comparison of exergy destruction in various components (expander, pump, evaporator, condenser) in Trilateral cycle using HFO-1234yf.

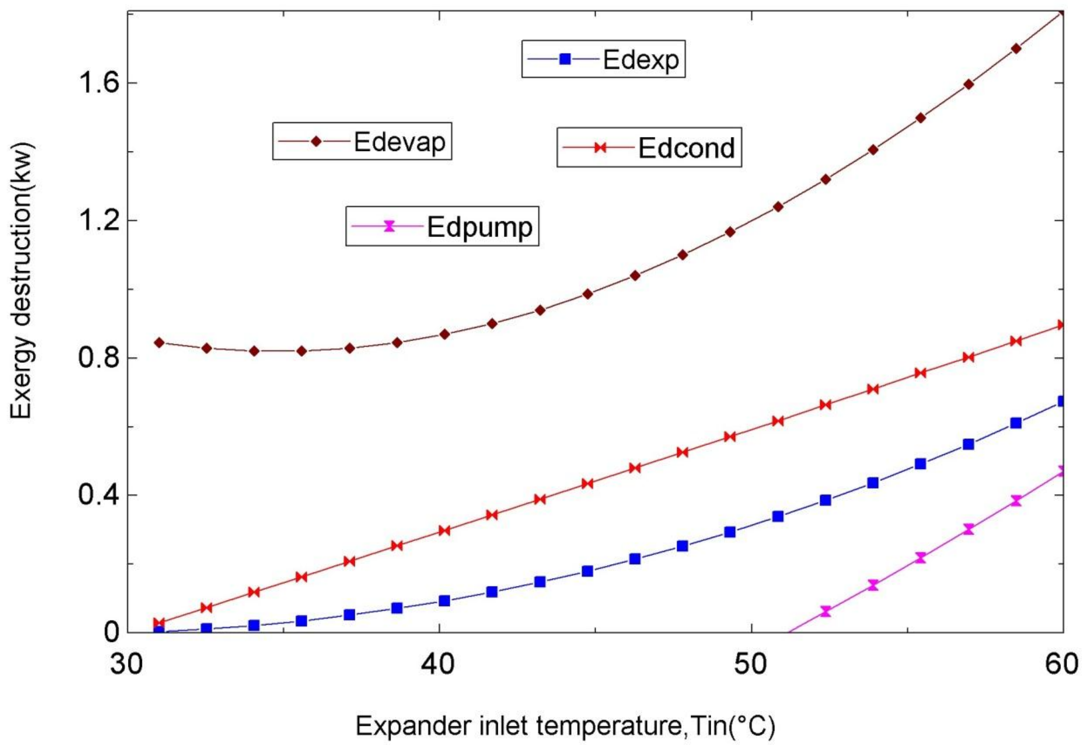


Fig.4.48 comparision of Exergy destruction in Trilateral cycle

Fig.4.48 shows the variation of Exergy destruction with expander inlet temperature for Trilateral cycle for HFO-1234yf working fluid.

The exergy destruction in evaporator is higher as compare to expander, pump, condenser and pump shows the lower exergy destruction.

It has been observed that as the temperature decreases the exergy destruction in evaporator, pump, expander, condenser decreases.

CHAPTER-5

CONCLUSIONS AND SCOPE FOR FUTURE WORK

5.1. Conclusions

On the basis of result obtained from the thermodynamic model, following conclusions are drawn-

HFO-1234yf used as working fluid in five ORC models through first order simulation are such as ,saturated, superheated, sub-critical and supercritical cycle. The first law efficiency of HFO-1234yf has compared with that of other working fluids (R-245fa, R-134a, Isopentane, and Ethanol) the results are obtained with help of simulation in this study: HFO-1234yf can be used for low-to medium-temperature ORC applications; even though it's thermal efficiency was not always the best among the other simulated fluids depending on the temperature level. The thermal efficiency of ORC with HFO-1234yf could be maximized approximately up to 8.8%-11.4% when the expander inlet temperature was 170^{o} C for a typical condensation temperature range (20-40^{o} C) in the supercritical cycle. However, a further increase in the expander inlet temperature to over 160 ^{o} C did not significantly boost the thermal efficiency. Overall, HFO-1234yf is a potential working fluid for low- to medium-temperature heat sources of ORC applications (expander inlet temperature of up to 170^{o} C) if ODP, GWP, and safety are considered to have precedence over performance. For supercritical cycle with HFO-1234yf the exergy destruction is highest in evaporator and lowest in pump. For saturated cycle with HFO-1234yf the exergy destruction in evaporator is highest and lowest in pump. For superheated cycle the exergy destruction in evaporator is highest in evaporator is highest and lowest in pump. For trilateral cycle with HFO-1234f the exergy destruction in evaporator is highest lowest in pump

Fig 4.24 shows the variation of exergetic efficiency in **super heated** cycle with expander inlet temperature with various working fluids (R-134a, HFO-1234yf, R-245fa, Isopentane, and Ethanol). At the expander inlet temperature (80-250^{o}C) the R-134a shows higher exergetic efficiency (.4-.5) in all working fluid. Ethanol shows lower thermal efficiency (0.2-0.3) in all working fluid.HFO-1234yf shows (0.406) exergetic efficiency which is second highest after R-134a in all working Fluid. Fig 4.25 shows exergetic efficiency comparison of various working fluids in **saturated cycle.** At the expander inlet temperature (30-100^{o}C) Ethanol shows higher exergetic efficiency (0.6) in all working fluids (HFO-1234yf, HFC-245fa, R-134a).HFO-1234yf shows lower exergetic efficiency (.405) in all working fluids.Fig4.26 shows Exergetic efficiency comparison of various working fluids in **trilateral cycle**. At the expander inlet temperature (30-90^{o}C) HFO-1234yf shows higher exergetic efficiency (.35-0.4) Isopentane shows lower thermal efficiency (0.3-0.35).Fig 4.27 shows Exergetic efficiency comparison of various working fluid in **supercritical cycle.** At the expander inlet temperature (80-200^{o}C) HCC-245 shows higher exergetic efficiency which is above to 0.48.

R134a shows lower Exergetic efficiency which is below to (0.42)

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