Major Project - II

THERMODYNAMIC ANALYSIS OF MODIFIED VAPOUR COMPRESSION REFRIGERATION SYSTEM USED FOR HEATING WITH ALTERNATING REFRIGERANTS

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of the requirements for the award of the degree of

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By

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CANDIDATE'S DECLARATION

I hereby declare that the work which is being presented in this dissertation entitled "**THERMODYNAMIC ANALYSIS OF MODIFIED VAPOUR COMPRESSION REFRIGERATION SYSTEM USED FOR HEATING WITH ALTERNATING REFRIGERANTS**" in the partial fulfillment of the requirements for the award of the **degree of Master of Engineering with specialization in Thermal Engineering**, submitted to the Department of Mechanical Engineering ,Delhi Technological University, is an authentic record of my own work carried out under the supervision of **Mr. Naushad A Ansari , Assistant Professor**, Mechanical Engineering Department, Delhi technological university, Delhi.

The matter embodied in this dissertation has not been submitted by me for the award of any other degree.

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CERTIFICATE

This is to certify that the above statement made by the candidate is correct to the best of my knowledge.

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ABSTRACT

The present study is focused on a simple thermodynamic and heat transfer analysis of modified vapour compression refrigeration (MVCR) system with various operating variables using R-134a, R-1234yf and R-1234ze as the refrigerants in modified vapour compression. In the present work emphasis is on waste heat recovery from condenser using air as a cooling fluid of a vapour compression refrigeration system utilized in air conditioning system or ice plants. The study covering the quantity and nature of low grade heat, an estimation of the quantity and amount of recoverable waste heat to meet the requirement from a refrigeration / air conditioning system. The modified vapour compression System consist of a waste heat recovery device called Canopus heat exchanger, with separate cooling of condenser.

A computer program has been made in Engineering Equation Solver to evaluate the performance of integrated vapour compression System. Therefore, The effects of the pressure drop and fouling in the evaporator, condenser, Canopus heat exchanger is neglected. Results indicate that the there is an increase in the overall coefficient of performance and cooling capacity using Canopus heat exchanger in the simple vapour compression refrigeration system. A comparison of between the properties of refrigerants R-134a, R-1234yf and R-1234ze is also presented.

The computational analysis has allowed the determination of the best energetic performance of the system. The analysis can represent a real system with a reasonable accuracy and is useful for future research work of whole System.

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NOMENCLATURE

Notations

CHE	Canopus Heat Exchanger
COP	Coefficient of Performance
VCR	Vapour Compression System
TR	Tonn of Refrigeration
MVCR	Modified Vapour Compression System
WHRS	Waste Heat Recovery System
TEWI	Total Equivalent Warming Impact
LCCP	Life Cycle Climate Performance
HCFCs	Hydro-Chlorofluorocarbons
HFC	Hydro-Flouro Carbon
CFC	Chloro-flouro carbon
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
IPCC	Intergovernmental Panel on Climate Change
ASHRAE	American Society of Heating, Refrigerating and Air Conditioning Engineers
RAC	Refrigeration And Air-Conditioning
m	Mass flow rate
t	Temperature
EES	Engineering Equation Solver
W	Work Input
Q	Heat
h	Enthalpy
HE	Heat Exchanger
EES	Engineering Equation Solver
W	Work Input
v	Velocity

C	Specific Heat

Subscripts

comp	Compressor
cps	Canopus
evap	Evaporator
isen	Isentropic
R	Refrigerant
svcr	Simple Vapour Compressor System
a	air
1	State point exit to Evaporator
2	State point exit to compressor
3	State point exit to Canopus Heat exchanger
4	State point exit to condenser
5	State point exit Expansion valve
6	State point entry to condenser cooling fluid
7	State point exit to condenser cooling fluid
8	State point entry to Canopus cooling fluid
9	State point exit to Canopus cooling fluid

CHAPTER- 1

INTRODUCTION

Since the dawn of the Industrial Age, the accelerating pace of industrialization has seen a rapid and exponential decrease in the availability of natural energy sources. While certain lobby groups have warned of an energy crisis since the mid-twentieth century, the scale of the crisis has gained prominence in the consciousness of the moguls of industry in the last decade or so. Hence, since the beginning of the twenty-first century, energy conservation has become a major feature of interest in most industrialised countries. The economics of saving energy versus wasting it has driven industrial activists to pay more attention to energy conservation. This awareness started in the mid-1970s, when the oil producing countries used the oil price as a weapon against countries that supported Israel. The continuing rise in the price of oil and other finite energy sources, coupled with the evidence of the contribution of man to global warming, have, in the years since the signing of the Kyoto Protocol on Climate Change, sparked innovation aimed at energy conservation. Alternative energy sources include technologies such as hydroelectric, wave, solar energy, and nuclear power.

The implementation of energy conservation requires that all the possibilities of counteracting any potential loss of energy must be considered. This includes reducing heat losses from furnace, thermal insulation, repair of steam leaks in power plants, direct heat losses from refrigeration or air conditioning system and all other practices that may be implemented rapidly and, preferably at low cost.

Waste heat recovery systems have been used for many decades, particularly in power generation and energy intensive industries (Reiter, 1983). They conserve energy by storing and reusing available waste heat. They transfer energy from sources of waste heat to energy conversion systems, by using various types of heat recovery equipment. They reduce energy consumption which results in significant cost savings (Reiter, 1983) WHRSs form an important part of the methods developed to increase the efficiency of various energy saving systems.

Waste heat recovery systems are totally dependent on the temperature of the waste heat stream. A waste heat stream is a source of thermal energy generated by means of mechanical, electrical or any other form of industrial process, including chemical reactions, fuel combustion, furnace heat, and nuclear radiation decay, for which no useful application has been found. Unless utilized, the waste heat is dumped into the environment without being utilized for economically beneficial purposes. The attractive aspect of waste heat is the value of the estimated cost saving, if losses are reduced. However, not all waste heat can be fully recovered. A significant amount of heat may be recovered by using specially designed heat recovery equipment, depending on the temperature of the heat source and the nature of the working fluid used in the system. WHRS equipment should be able to process waste heat at any temperature from chilled cooling water to the high-temperature waste heat of a nuclear reactor . Usually high-temperature waste heat leads to more efficient energy recovery and cost effectiveness. Typical examples of WHRSs are preheating of fuel-air mixtures, space heating, and pre-heated boiler feed water or process water.

For high-temperature energy recovery, a cascade of waste heat recovery equipment may be used to ensure that the maximum amount of heat is recovered. An important application of waste heat equipment is in the use of high-temperature waste heat for air preheating or low temperature waste heat for process feed water heating or steam generation.

During the last few decades the rapidly growing country like India and China has been accompanied by more building which in turn generates more energy and environment related problems. Accordingly, the energy consumption of buildings is growing year after year. As the demand for air conditioning increased greatly during the last decade, large demands of electric power and uncertain availability of fossil fuel have led to a surge of interest in the efficient energy application in air conditioning system. Thus the efficient use of energy has become a major issue in the refrigeration and air conditioning domains. Gains thanks to the use of equipment producing the same amount of cold with reduced energy consumption benefit everyone: the user, who of course has lower energy expenses, and the atmosphere, which absorbs less carbon dioxide (CO_2). Renewed interest in energy efficiency can be explained by the fact that the overall global warming impact of refrigerating equipment is now considered and reflected in TEWI (Total Equivalent Warming Impact) or LCCP (Life Cycle Climate Performance) concepts.

The use of energy-efficient refrigerating installations certainly contributes to overall reductions in or limiting of global warming, but the manner in which such gains are measured needs careful consideration. Emissions reductions will probably not be tallied under HFCs. Instead, they will probably be lost in a sea of figures on electrical consumption and the refrigeration and air conditioning practitioners who have achieved emissions reductions probably won't even get any credit for their efforts. Whatever method is used in tallying emissions reductions, it is vital to continue to address this issue and to ensure that the efforts of the refrigeration profession as a whole are widely publicized.

The rejected (sensible and condensation) heat from air conditioning systems is a readily available energy source, that can be used to produce low temperature hot water for washing and bathing [1-4]. Energy consumption of building space heating, air conditioning, and household sanitary water will continue to increase with economic prosperity. Air conditioning system is widely used in building cooling, however there are several problems associated with utilizing the air conditioning system. A great deal of useful waste energy, which can be used for other purposes, is directly dissipated to the environment. This dissipated heat not only wastes energy, but also causes severe pollution in the surrounding area. It is observed that plentiful waste condensing heat from traditional air conditioning system is directly exhausted to the environment.

Refrigeration and air conditioning systems have a major impact on energy demand with roughly 30% of total energy consumption in the world. The depletion of the world fossil fuel reserves coupled with the global environmental problems stimulated the search for alternative sources, and demand for gradual phase out of halogenated fluids has been increasing at an alarming rate. Most of industrial process uses a lot of thermal energy by burning fossil fuel to produce steam or heat for the purpose. After the processes, heat is rejected to the surrounding as waste. This waste heat can be converted to useful refrigeration by using a heat operated refrigeration system, such as an absorption refrigeration cycle. Thus electricity purchased from utility companies for conventional vapour compression refrigerators can be reduced.

With fossil fuels fast depleting, it is imperative to look for refrigeration systems that require less high-grade energy for their operation. The continuous depletion of the ozone layer, which shields the earth's surface from ultra voilet radiation, has resulted in a series of international treaties demanding a gradual phase out of halogenated fluids. Human activities have considerably increased the concentration of greenhouse gases (CFC, hydro-chlorofluorocarbons, carbon dioxide, methane, nitrous oxide) that determine the earth's surface and atmosphere warming that might adversely affect the natural ecosystem. There are a number greenhouse gases that are released into atmosphere as a result of human activities, but carbon-dioxide at 84% in 2005 is by far the most abundant. Over the last hundred years , the mean temperature have increased by 0.3 to 0.6ºC, and doubling the amount of carbon dioxide in the atmosphere is likely to yield a further temperature increase from 1.5 to 4.5ºC. Warming temperatures are believed to melt glaciers and ice sheets, which further contribute to the swelling oceans. Some argue that these changes are not directly attributable to human-caused global warming, but these observed trends have raised concerns that global warming may already be having an impact on the environment and human life. In particular, it is well known that the greenhouse effect resulting from an operating plant is not a secondary matter. Climate change caused by human activities threatens to affect the climate, habitat, health and economy of virtually all the countries of the world. The chair of the Intergovernmental Panel on Climate Change (IPCC) recently confirmed the influence of such emissions on the Earth's climate. In particular, it is well known that the greenhouse effect resulting from a refrigeration plant is not a secondary matter.

A large amount of low-temperature heat source which is going as waste heat is available from superheat horn of a vapour compression refrigeration system. If these low temperature heat sources can be used, it will not only improve the overall system energy efficiency, but decrease the heat pollution to the environment as well.

Refrigeration and air-conditioning (RAC) plants remove heat from the space to be conditioned and pumping it to the higher temperature sink. Added to this heat which is gathered from the space, there is work added by the compressor in a vapour compression system and the total heat is rejected to the environment. Heat rejected from the air-conditioning system is the superheat of the refrigerant vapour and a portion of latent heat of the refrigerant fluid. Heat rejected from refrigeration and air-conditioning plants is of low grade quality. Due to the high costs associated with the recovery of such heat and the availability of alternate means for meeting low grade heat requirements, low grade waste heat is generally rejected to the atmosphere. However, in view of the negative values that are being placed on waste or reject heat streams into the atmosphere by statutory bodies in developed countries, disposal of reject heat into the atmosphere is becoming even more costly.

A rough estimate of waste heat available from an RAC system indicates that about 3 -5 kW of waste energy is rejected to the environment for every kilowatt of energy expended by the compressor [5] Recovery of this energy will contribute to the saving of overall energy costs. However, the potentiality of such energy recovery has not been considered while designing RAC systems in most of the installed units, obviously due to limited avenues where the recovered energy could be put to use. The low grade quality of heat available as waste energy limits the areas of utilization. Therefore, a comprehensive study covering the quantity and nature of requirement of low grade heat, an estimation of the quantity and amount of recoverable waste heat to meet the requirement from a refrigeration and air-conditioning system will have to be carried out. Some of the issues which are involved in this direction are matching of the cooling demand and availability of low grade heat. Unlike industrial reject heat, which is available in larger quantities and at higher temperatures, the reject heat from RAC systems has a fluctuating profile of availability. While the availability of reject heat from industry is dependent on fixed operating parameters, the reject heat from RAC systems is dependent on ambient weather conditions. Against industrial applications, apart from internal cascading of low grade heat sources to low temperature processes within the plant, there are low temperature heat requirements in grain drying and food processing.

The scope of this work is limited to the collection of waste heat from RAC plants and utilization of that waste heat in food processing, drying, space heating etc. This modified vapuor compression refrigeration (VCR) system utilizes the waste heat from superheated horn of the VCR system through an additional heat exchanger known as the Canopus heat exchanger (CHE) incorporated between the compressor and condenser of a VCR system. This system is aimed at increasing cooling capacity, increasing energetic performance and reducing running cost. A simple thermodynamic and heat transfer analysis is carried out to evaluate the heat recovery and the operating conditions for different component temperatures and effectiveness of the CHE. The effects of the inlet water temperature have also been studied.

CHAPTER- 2

LITERATURE REVIEW

Kaushik and Singh et al. [5] presented an investigation of the feasibility of heat recovery from the condenser of a vapour compression refrigeration (VCR) system through a Canopus heat exchanger (CHE) between the compressor and condenser components. The presence of the CHE makes it possible to recover the superheat of the discharged vapour and utilize it for increasing the temperature of the external fluid (water) removing heat from the condenser. The effects of the operating temperatures in the condenser and evaporator for different inlet water temperatures and mass flow rates on the heat recovery output and its distribution over the condenser and CHE (the fraction of the condenser heat available through the CHE), available outlet water temperature and heat recovery factor had all been studied and optimum operating parameters for feasible heat recovery had been ascertained. The parametric results obtained for different working fluids, such as R-22, R-12, R-717 and R-500, had been presented. They found that, in general, a heat recovery factor of the order of 2.0 and 40% of condenser heat can be recovered through the Canopus heat exchanger for a typical set of operating conditions.

Aprea and Renno et al. [6] has gone through the experimental studies of performances of a vapour compression refrigeration plant using as working fluids R22 and its substitute R417A (R125/R134a/R600, 46.6/50/3.4% in mass). This type of plant is applied to a commercially available cold store, generally adopted for preservation of foodstuff. Its working is regulated by on/off cycles of the compressor, operating at the nominal frequency of 50 Hz, imposed by the classical thermostatic control. The problems related to the replacement of the fully halogenated chlorofluorocarbons and of the partially halogenated hydrochloroflourocarbons have been only partially solved. As for R22, there is no unanimous solution for its substitution. In this paper, the refrigerant fluid experimentally tested as a substitute for R22 is the non-azeotropic mixture R417A. R417A is non-ozone depleting long term replacement for R22, which can be used in new and existing direct expansion systems using traditional R22 lubricants. R417A provides an easy solution to the legislation requirement, since it presents an ozone depletion potential equal to zero, and it does not require a change of lubricant type, equipment redesign or significant modifications. The experimental analysis had allowed the determination of the best energetic performances of R22 in comparison with those of R417A in terms of the coefficient of performance, exergetic efficiency, exergy destroyed in the plant components and other variables characterizing the refrigeration plant performance.

Reddy-Panwar-Kaushik et al. [7] their study is intended to deal with the waste heat recovery from the industrial refrigeration and air-conditioning system by introducing Canopus heat exchanger. There is a considerable amount of low-grade heat available in large-capacity systems. To recover this low-grade heat, a Canopus heat exchanger is introduced between compressor and condenser components. They have gone through the system feasibility with various operating parameters and its effect on heat recovery factor and overall COP of the system. The parametric results obtained for different eco-friendly working fluids, such as R-134a and R-507a which is the combination of R-143a and R-152a and are found suitable for both industrial and domestic applications have been presented. They found that, in general, overall COP of the system is improved without affecting the actual performance of the system. The potential of low-grade heat availability is increased with increasing cooling capacity.

Rahman and Rahman et al. [8] They presented the performance of the recently developed integrated space condition and domestic water heating device. A conventional split type air conditioner is modified to reclaim the superheated portion of the heat leaving the compressor to be utilized to heat up water for domestic purposes. The experimental investigation revealed that this device could heat up water from room temperature to 82ºC within 9 hours of operation (8:00 am to 5:00 pm) at the same time cool down the room air to the desired level. By using this type of energy recovery device, compressor efficiency can be improved and at the same time, hot water for domestic purposes can continuously be obtained free of charge. The end result is expected to be faster cooling and prolonged compressor life. This system is simple yet affordable and able to save water heating cost and environment friendly, i.e., less heat is rejected to the environment.

Sathiamurthi and Srinivasan et al. [9] studied and analyses the feasibility of retrofitting the waste heat recovery system for hot water generation. They attempted to recover the heat rejected by a 16TR air-conditioning system used in the college computer centre. The waste heat is used for producing hot water for bathing of 600 students in the hostel. The demand and supply and storage requirements are attempted. Such a system is fabricated and preliminary experiments are carried out. The study has shown that such a system is technically feasible and economically viable. The proposed waste heat based hot water system (HWS) produces hot water by recovering waste heat from the conventional air conditioning system. The layout of the proposed of the HWS is shown in Fig. 4.

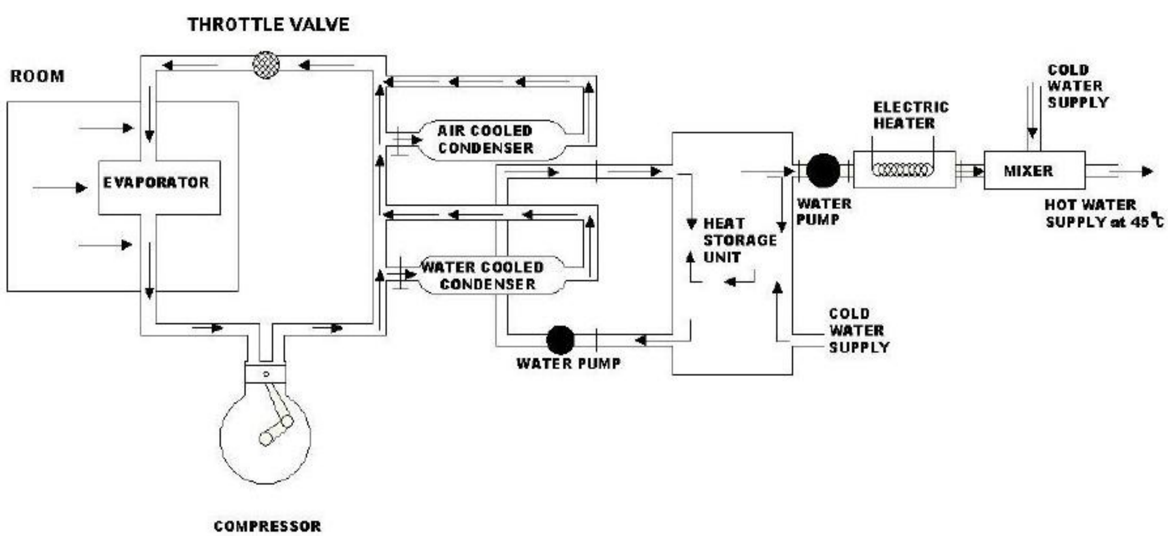

Fig 1: Layout of the proposed waste heat based hot water system

A 16 TR air conditioning system is taken for study. The system is used for air-conditioning the computer centre of the college. The centre is used by students during 9.00 am to 9.00 pm on working days and 9.00 am to 5.00 pm during holidays. The college has 30 such computer centres to cater the needs of 5000 applied sciences, engineering and management students. The college has 6 hostels, (4 for gents and 2 for ladies) and in each hostel about 600 students are staying. The proposed HWS uses waste heat from one 16 TR air conditioner fitted in one of its computer centers to one such hostel with 600 students. Heat recovery from refrigeration plants has been analysed.

Herbas, Dalvi and Parise et al. [10] they presented a study on the alternatives for recovering the amount of energy from the condenser to produce hot water, or warm air. A basic thermodynamic analysis is carried out, showing the most appropriate conditions for heat recovery. A simulation model is also employed for the performance prediction of cases where temperature requirements are present. Data from an experimental unit are also presented and compared with predicted results. Tests were performed over a range of condenser and evaporator temperatures and mentioned that considerable reduction on the energy consumption can be obtained, particularly when similar heating and cooling loads are involved. Also, from the experimental results, it has been found that lower condensing temperatures held higher energy conversion ratio's. However, if the required hot water temperature is well above the condensing temperature, most of the heating load will be left to the conventional heater, thus reducing the ECR. In short, a considerable amount of heat can be recovered from refrigeration plants although at low temperature levels. In practice, it will be difficult to find systems capable at operating with heating loads close to the refrigeration load to 1. The present analysis has been restricted to refrigeration load values greater than 1- For lower heating loads, although exceeded by the condenser output, there would be the temperature requirement to be met. Further research is necessary.

Abu-Mulaweh et al. [11] resented the design and performance of a thermosiphon heat recovery system that recovers heat rejected from an air conditioner is described by presenting some experimental test data. His paper presents an experimental setup that was designed, developed, and constructed in understanding the basic heat transfer processes by utilizing real life applications such as using waste heat from a window type air conditioner to heat water for residential and commercial use. Heat recovery from an air conditioner by thermosiphon is attractive because it eliminates the need for a circulating pump. He carried out the results indicate that the design of the thermosiphon heat recovery system was a success. The schematic diagram of the system shown in the fig 5.

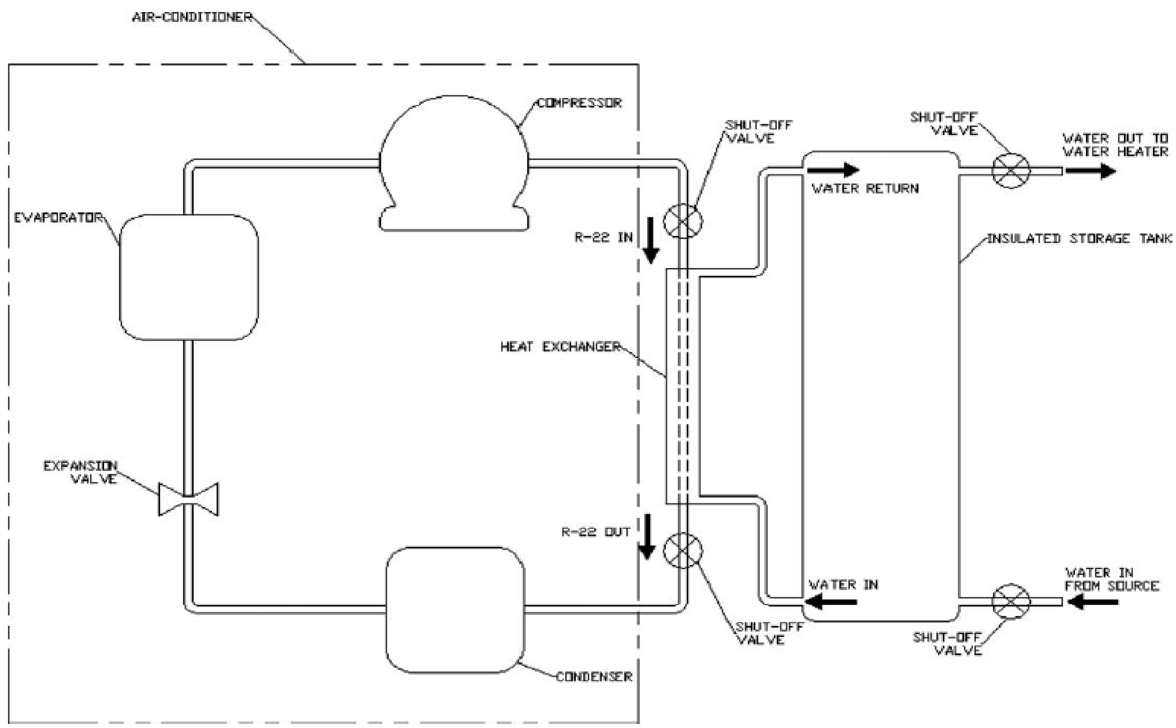

Fig 2: Schematic of the thermosiphon heat recovery system.

CHAPTER- 3

REFRIGERANTS

HCFCs (hydro-chlorofluorocarbons) and CFCs (chlorofluocarbons) have been applied extensively as refrigerants in air conditioning and refrigeration systems from 1930s as a result of their outstanding safety properties. However, due to harmful impact on ozone layer, by the year 1987 at Montreal Protocol it was decided to establish requirements that initiated the worldwide phase out of CFCs. By the year 1992, the Montreal Protocol was improved to found a schedule in order to phase out the HCFCs. Moreover in 1997 at Kyoto Protocol it was expressed that concentration of greenhouse gases in the atmosphere should be established in a level which is not intensifying global warming ozone layer. Subsequently it was decided to decrease global warming by reduction of greenhouse gases' emissions [12].

As a consequence of this protocol even new developed HFCs refrigerants like R-134a should be gradually phased out due to their high global warming potentials. Hence in order to meet the global ecological goals, conventional refrigerants should be replaced by more environmental friendly and safe refrigerants in a way the energy efficiency also is improved.

3.1 Fluid Selection

In refrigeration and air conditioning systems selection of an appropriate working fluid is one of the most significant steps for a particular application. Low global warming potential has been inserted to the long list of desirable criteria of refrigerant's selection. In fact, environmental characteristics of refrigerants are becoming the dominant criteria provided that their thermodynamic behaviors and safeties are favorable as well.

3.1.1 Chemical and thermo-physical properties

Generally, thermodynamic and transport properties of refrigerants are the key factors in refrigerant's selection as they determine the performance of the system. The desirable thermodynamic properties are a normal boiling point slightly less than target temperature and, thereby, an evaporating pressure higher than atmospheric pressure. The other favorable characteristics are, low liquid viscosity, high heat of vapourization, modest liquid density and slightly high gas density. It is worthwhile to mention that high heat of vapourization and gas density lead to higher capacity with a specific compressor in a refrigeration system. High liquid thermal conductivity intensifies heat transfer and results in smaller required heat exchangers.

Low viscosity also causes low pressure drop in the heat exchangers. Smaller pressure ratio leads lower compression work and improve COP of the system [13].

In view of the fact that boiling point and gas density are influenced by the pressure, operating pressure is a factor to choose a suitable refrigerant for a particular application.

Selected refrigerant should be also chemically stable under operation condition while it shouldn't decompose nor react with material in the system.

3.1.2 Environmental impact and safety aspects

Environmental effects are the main problems of common refrigerants so that non environmental friendly impacts of CFCs and later on HCFCs brought about them to be phased out despite of being stable, non-flammable and non-toxic (comparing to Sulfur Dioxide and other refrigerants used before the introduction of CFCs). Ozone depletion potential (ODP) and global warming potential (GWP) are the significant factors demonstrate the direct impact of refrigerants in case of any leakage or releasing to the surroundings. However, using low GWP refrigerants are not the only efficient way to reduce greenhouse gas emissions. In fact it is probable to choose a low GWP refrigerant but still raise total greenhouse gas emissions. When the low GWP refrigerant causes more energy use and fuel consumption actually there are larger indirect emissions. Therefore in developing the low GWP refrigerants always energy efficiency of the system must be studied and its indirect climate impacts should be considered besides its direct emissions. Life cycle climate performance (LCCP) helps to consider overall potential of greenhouse gas emission of the system including materials, transportation, and operation, production, recycling, servicing and end-of-life. The LCCP (Life Cycle Climate Performance) study is one of the main steps in evaluation of the cradle-to-grave global warming impact of any refrigerant. Furthermore, toxicity and flammability are the determining factors to select suitable refrigerant for any application. Low toxicity and flammability are the most desirable aspects in safety and health studies.

Although HC refrigerants have highly flammable characteristics according to the standards of ASHRAE(American Society of Heating, Refrigerating and Air Conditioning Engineers) as a negative specification, they have not only several preferable specifications such as zero ozone depletion potential, very low global warming, non-toxicity, and higher performance than other refrigerant types

3.2 Safety Classification of Refrigerants ASHRAE Standard 34-2010.

As per ASHRAE (American Society of Heating, Refrigerating and Air Conditioning Engineers)

- Toxicity classification
  - Class **A**: No known toxicity at < 400 ppm
  - Class **B**: Evidence of toxicity at < 400 ppm
- Flammability Classification
  - Class **1** - No Flame Propagation
  - Class **2** - LFL > 0.10 kg/m^3 and heat of combustion < 19 MJ/kg
  - Class **3** - LFL ≤ 0.10 kg/m^3 and heat of combustion ≥ 19 MJ/kg
- New 2L flammability classification
  - Maximum burning velocity of ≤ 10 cm/s (3.9 in./s) when tested at 23.0°C (73.4°F) and 101.3 kPa (14.7 psia).

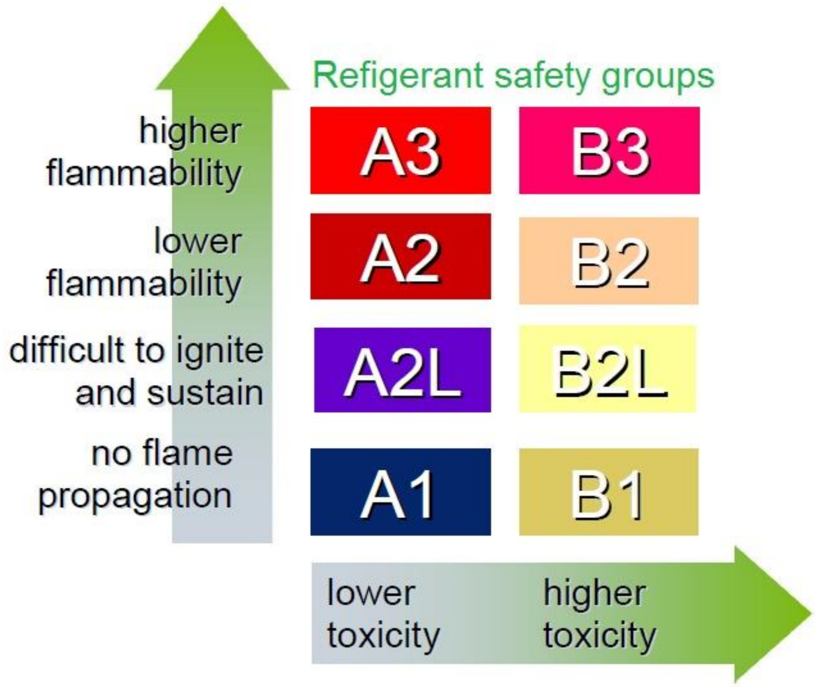

Fig 3: latest Designation and Safety Classification of Refrigerants (ASHRAE Standard 34-2010)

3.3 REFRIGERANTS USED

The results carried out have been displayed using some commonly used eco-friendly refrigerant R134a and the newly developed low global warming potential refrigerants R1234yf and R1234ze in industries and domestic purposes.

3.3.1 Refrigerant R-1234ze

1,3,3,3-Tetrafluoropropene (R-1234ze) is a hydrofluoroolefin. It was developed as a "fourth generation" refrigerant to replace R-134a and as a blowing agent for foam and aerosol applications. The use of R-134a is being phased out because of its high global-warming potential. R-1234ze has zero ozone-depletion potential and a low global-warming potential (GWP = 6).

Its thermodynamic behavior is similar to the R-134a. It is non-flammable which make it safe in refrigeration application. In table 1 some of its fundamental properties have been written [9].

Table 1: Thermodynamic properties For R-1234ze (E) [14].

Chemical formula	Trans, $CHF=CHCF_3$
Molecular weight (kg/kmol)	114.04
GWP	6
ODP	0
Vapour Pressure at 25°C[Mpa]	0.49
Atmospheric Life Time [day]	18
Vapour Pressure at 50°C[MPa]	1.080
Critical Temperature[°C]	79
Critical Pressure[MPa]	3.632
Critical density[kg/m3]	486
Normal Boiling Point[°C]	-20
Flame limit	None to 30°C
Lower Flammability Limit[vol.% in air]	-
Upper Flammability limit[vol.% in air]	-
Ignition Temperature[°C]	288 - 293
Minimum Ignition Energy[mJ]	No Ignition at 20°C, 61,000-64,000 at 54°C
Liquid Density at 25°C[kg/m3]	1180
Acentric Factor	0.296

3.3.2 Refrigerant R-1234yf

2,3,3,3-Tetrafluoropropene, or R-1234yf, is a hydrofluoroolefin with the formula $CH_2=CFCF_3$. It has been proposed as a replacement for R-134a as a refrigerant in automobile air conditioners. HFO-1234yf is the first in a new class of refrigerants acquiring a global warming potential (GWP) rating 335 times less than that of R-134a (but still 4 times higher than the alternative substitute carbon dioxide) and an atmospheric lifetime of about 400 times shorter.

It has a good compatibility with existing technology that leads to fast global adoption. Its system performance is similar to R-134a and has comparable COP and cooling capacity to those of R134a. Being thermally stable is another notable property of this new refrigerant which makes it a promising candidate to substitute R-134a.

Table 2 is a summary of the properties of R-1234yf and R-134a and it can be seen this new developed refrigerant has close thermodynamic properties to R134a despite of its higher flammability [15].

Table 2: Thermodynamic properties For R-1234yf .

Chemical formula	$CF_3CF=CH_2$
Safety Class	A2L
GWP	4
ODP	0
Molar mass [kg/kmol]	114.04
Critical pressure [MPa]	3.382
Critical Temperature [°C]	95
Normal Boiling Point [°C]	-29
Atmospheric Life Time [year]	<0.05 (11 day)
Lower Flammability Limit [vol.% in air-23ºC]	6.2
Upper Flammability limit [vol.% in air-23ºC]	12.3
Auto ignition Temperature [°C]	405
Minimum Ignition Energy [mJ]	5000-10000
Heat of Combustion [kJ/g]	10.7
Acute Toxicity Exposure Level [ppm]	101,000
Molecular Weight [kg/kmol]	114
Vapour Density at 25°C [kg/m3]	4.7508
Vapour Cp at 25°C [kJ/kgK]	0.92811

3.3.3 Refrigerant R-134a

Tetrafluoroethane (R134a) is a blend component for refrigeration. It also a propellant for aerosol and a blowing agent for extruded polystyrene foams. It replaces the CFC R12 (dichlorodifluoromethane) and in few years the HCFC R22 (chlorodifluoromethane). Table 3 shows the properties of R-134a.

Table 3: Properties of refrigerant R-134a.

Chemical formula	CH_2FCF_3
Safety Class	A1
GWP	1430
ODP	0
Molar mass [kg/kmol]	102.3
Critical pressure [MPa]	4.0593
Critical Temperature [°C]	102
Normal Boiling Point [°C]	-26
Atmospheric Life Time [year]	14
Lower Flammability Limit [vol.% in air-23ºC]	-
Upper Flammability limit [vol.% in air-23ºC]	-
Auto ignition Temperature [°C]	>750
Minimum Ignition Energy [mJ]	-
Heat of Combustion [kJ/g]	4.2
Acute Toxicity Exposure Level [ppm]	50,000
Molecular Weight [kg/kmol]	102
Vapour Density at 25°C [kg/m3]	4.2439
Vapour Cp at 25°C [kJ/kgK]	0.85118

3.3.4 Comparison of Properties

Table 4 showing the comparison of the refrigerants selected also figure 2 shows the vapour pressure diagram of some different refrigerants including the R-1234ze.

Table 4: Comparing properties of different refrigerants

Refrigerants	Chemical composition	Molecular weight [g/mol]	Critical temp. [°C]	Critical pressure [MPa]	Normal Boiling Point[°C]	Safety class	ODP	GWP
Ammonia	N-H3	17.02	132.3	11.28	−33.34	B2	0	0
R1234ze (E)	Trans, $CHF=CHCF_3$	114.04	79	3.632	-20	A2L	0	6
R1234yf	$CF_3CF=CH_2$	114	95	3.382	-29	A2L	0	4
R134a	CH_2FCF_3	102	101.1	4.059	-26	A1	0	1430

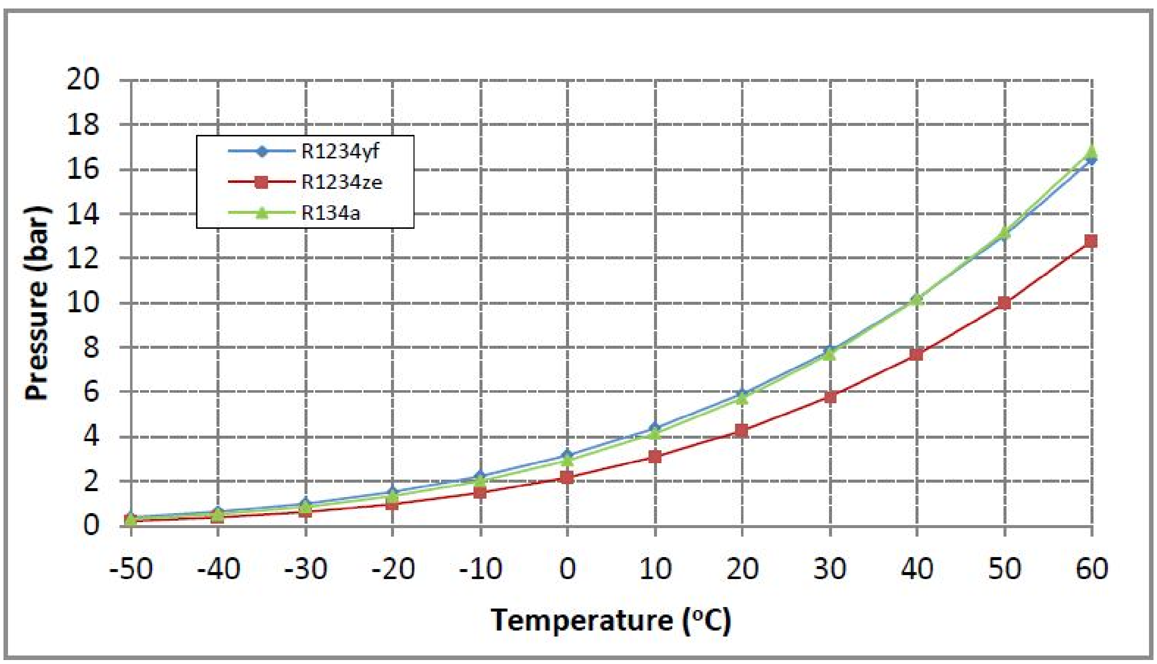

Fig 4: Vapour pressure diagram for R-1234ze, R-1234yf and R-134a.

CHAPTER- 4
VAPOUR COMPRESSION REFRIGERATION SYSTEM

4.1 Simple VCR system

A vapour compression refrigeration (VCR) cycle is a work absorbing thermodynamic system based on the reversed heat engine principle. It consists of a mechanical compressor, a condenser, an expansion device for throttling, and an evaporator as illustrated in Fig. 3.

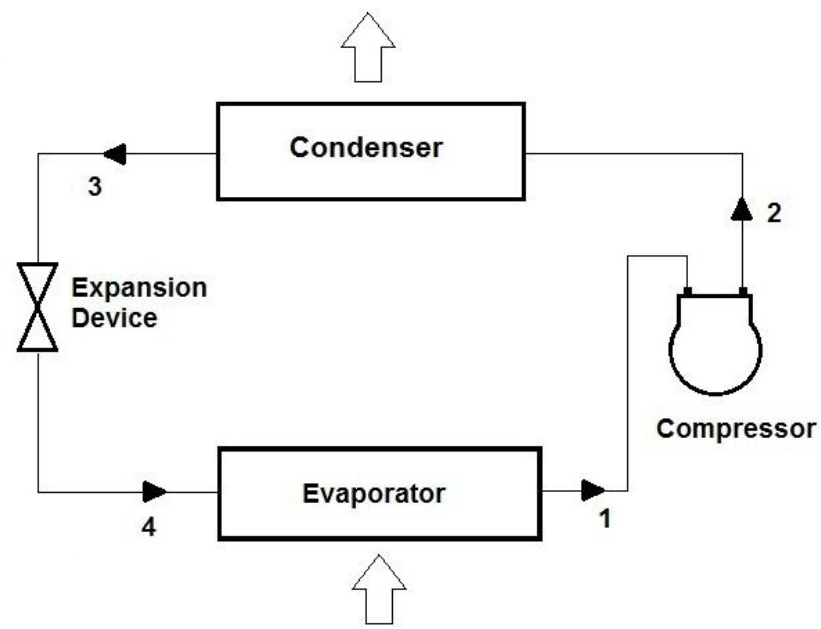

Fig 5: Schematic diagram of simple vapour compression refrigeration system.

Processes involve in the cycles shown in the figure 3 are as follows:

- 1–2 Adiabatic compression of the refrigerant's vapour to the pressure of the condenser.
- 2–3 Constant-pressure rejection of heat, causing desuperheating and condensation of the refrigerant.
- 3–4 Throttling (i.e., isenthalpic expansion) of the refrigerant's saturated/subcooled, liquid from the condenser's to the evaporator's pressure.
- 4–1 Constant-pressure heat absorption, causing evaporation of the refrigerant to produce saturated/superheated vapour.

CHAPTER- 5

THERMODYNAMAIC MODELING OF MODIFIED VCR SYSTEM

In this chapter a computational model was developed for carrying out the analysis of the system using Engineering Equation Solver software. The various properties of the selected refrigerants in the state points are taken from the EES. To analyze the possible realistic performance, a detailed thermodynamic analysis of modified vapour compression refrigeration (VCR) system has been carried out by ignoring the kinetic and potential energy change.

5.1 System Description

The present work is on this modified vapour compression refrigeration (MVCR) system, where Canopus heat exchanger is employed between condenser and compressor. The removal of heat from both the heat exchanger i.e. Canopus HE and condenser is carried out by air. Working fluids with desirable thermodynamic characteristics, such as ammonia and R-134a, R-1234ze and R-1234yf are used as refrigerants. In the thermodynamic cycle, the saturated vapour of the refrigerant coming from the evaporator is drawn by the compressor and compressed adiabatically at the expense of input work W_{comp} (process 1-2). The compressed refrigerant vapour is directed firstly into the Canopus heat exchanger (process 2-3) where sensible heat (Q_{cps}) is removed than to condenser where it condenses (process 3-4) at constant pressure as a result of the removal of heat of condensation (Q_{cond}), with an external air or water cooling medium. The liquid refrigerant coming from the condenser undergoes adiabatic expansion through the expansion valve, accompanied by a drop in pressure at constant enthalpy (process 4-5). This assumption of isenthalpic expansion, rather than isentropic expansion is good from the point of view of practical convenience and economic constraints. However, this may be a cause of energy loss during throttling. The liquid refrigerant under reduced pressure evaporates in the evaporator, thereby absorbing heat (Q_{evap}) from the space to be cooled. The low pressure vapour from the evaporator is compressed by the compressor and the cycle is repeated.

The air temperature rise in the condenser is below the temperature of condensation of the refrigerant vapour. After compression, the superheated refrigerant vapour transfers a large amount of heat to the water stream through the Canopus heat exchanger which has been given its name "Canopus" heat exchanger on the basis of the project concerned undertaken in the U.S.A., as mentioned in the available literature [16]. This is a standard liquid-vapour heat exchanger where heat is transferred from the superheated vapour to the coolant fluid air. A counter-flow arrangement is used in which the incoming air stream comes into contact with the higher temperature superheated refrigerant vapour leaving the compressor and attains a temperature much higher than the condensing temperature of the vapour.

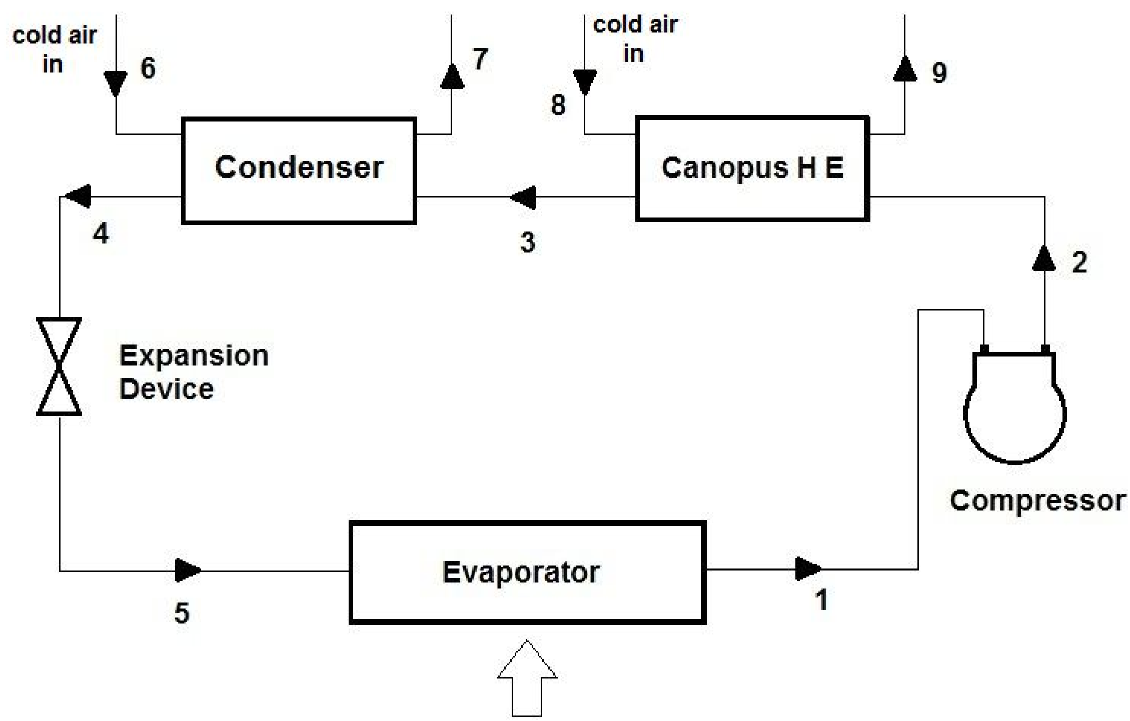

Fig 6: Schematic diagram of modified vapour compression refrigeration system

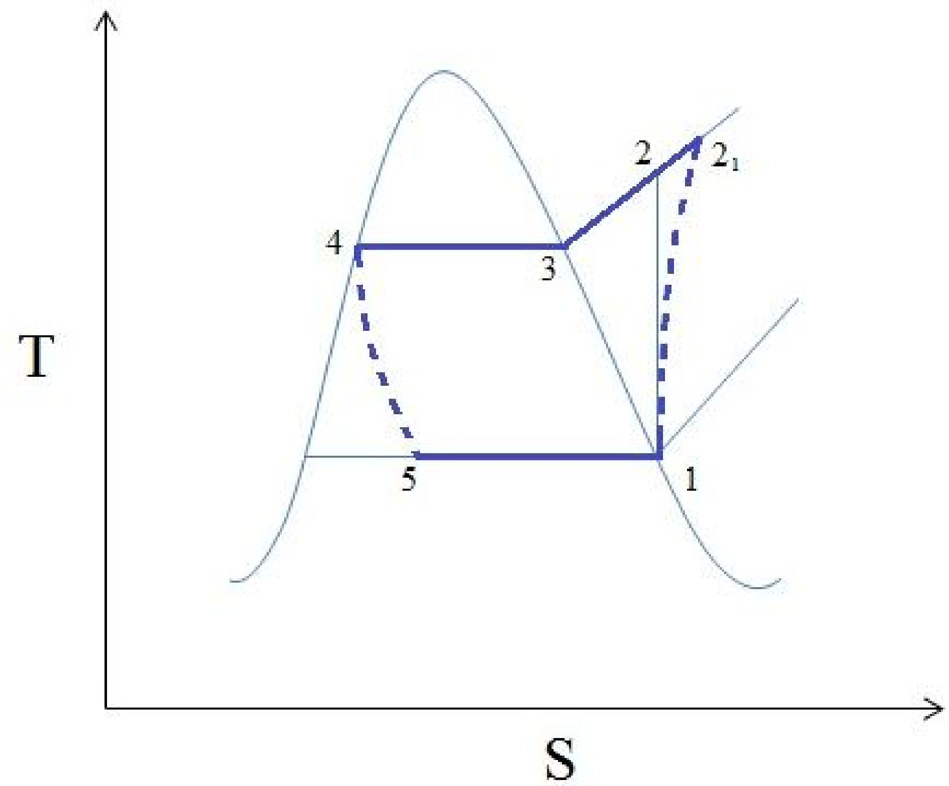

Fig 7: Temperature(T) vs Entropy (S) diagram of modified VCR system

The processes involved in the cycle (see figure 7) for present system is as follows:

(1–2) → Adiabatic compression of the vapour refrigerant.

(2–3) → Constant-pressure heat rejection process in Canopus HE.

(3–4) → Constant-pressure heat rejection process in Condenser.

(4-5) → Isenthalpic expansion process.

(5-1) → Constant-pressure heat absorption process (phase change from liquid to vapour)

(6-7) → Air temperature rise in condenser.

(8-9) → Air temperature rise in Canopus heat exchanger

5.1.1 Advantages of the modified VCR system:

- The waste heat rejected by the vapour compression system is recovered through Canopus HE is utilized in food processing, drying, space heating etc. Heat rejected by the condenser can be recovered to heat the working fluid (air).

- Recovery of waste heat further contributes in conservation of energy as well as it would be helpful in reduction of global warming.

- Since waste heat is being utilized therefore there will be increase in overall capacity of the system and increase in COP.

- Air is abundant in atmosphere and doesn’t need any external circuit to warm any space or thing means can directly fed to get the desired results.

- Maintenance and operating cost of air circulating circuit in less.

5.2 Thermodynamic Modeling

In an open flow system, there are three types of energy transfer across the control surface namely working transfer, heat transfer, and energy associated with mass transfer and/or flow. The first law of thermodynamics or energy balance for the steady flow process of an open system is given by:

$$\sum Q + m\left(h_i + \frac{V_i^2}{2} + gZ_i\right) = m\left(h_o + \frac{V_o^2}{2} + gZ_o\right) + W_c \tag{1}$$

For steady state flow, the energy balance for a thermal system can be estimated by using Eq. 1. Cycle diagram of vapour compressor refrigeration is illustrated in Fig. 6; T–S cycle diagram is presented in Fig. 7.

Actual compressors take more work than isentropic compressors. Isentropic Efficiency of Compressor is given by:

$$\eta_{isen} = \frac{\text{Isentropic work in compression}}{\text{Actual work in Compression}} = \frac{(h_{21} - h_1)}{(h_2 - h_1)} \qquad (2)$$

The actual enthalpy h_2 can be evaluated from the compressor isentropic efficiency, which is defined as the ratio of the isentropic compression work to the actual compression work.

Refrigerating effect is calculated by using

$$Q_{evap} = m_R * (h_1 - h_5) \quad \text{or} \qquad (3)$$

$$Q_{evap} = 3.517 * TR \qquad (4)$$

Heat rejected by condenser to the surrounding is given by

$$Q_{cond} = Q_{evap} + W_c \qquad (5)$$

Coefficient of Performance of simple VCR system i.e. no heat recovery device is installed is given by

$$COP_{svcr} = \frac{Q_{evap}}{W_{comp}} = \frac{m_R(h_1 - h_5)}{m_R(h_2 - h_1)} \qquad (6)$$

Heat rejected by the condenser to the surroundings is given by

$$Q_{cond} = m_R * (h_3 - h_4) \qquad (7)$$

Heat removed through the Canopus HE is given by

$$q_{Cps} = C_a * (t_9 - t_8) \qquad (8)$$

Heat absorbed by the cooling air through the condenser is given by

$$q_{Ca} = C_a * (t_7 - t_6) \tag{9}$$

The heat removal factor (HRF) is given by the relation:

$$HRF = (h_2 - h_4) / (h_3 - h_4) \tag{10}$$

Overall COP of the VCR system with Canopus heat exchanger for combined cooling and heating is given by the relation:

$$COP_{cps} = \frac{Q_{Evap}}{W_{comp}} = (Q_{evap} + (q_{cps} * m_a)) / W_c \tag{11}$$

5.3 Simulation Study

Simulation was performed to evaluate the integrated Refrigeration system with the following assumptions:

1. The flow through all the components is under steady state.
2. The pressure drop due to friction within the refrigeration system can be neglected, except through the expansion valve.
3. The fluid streams in the piping between the components and the heat exchangers are adiabatic.
4. Assuming the kinetic and potential energy change as zero in the system.

5.4 Current applications of refrigeration

Most widely-used current applications of refrigeration are for the air-conditioning of private homes and public buildings, and the refrigeration of foodstuffs in homes, restaurants and large storage warehouses. The use of refrigerators in our kitchens for the storage of fruits and vegetables has allowed us to add fresh salads to our diets year round, and to store fish and meats safely for long periods. Dairy products are constantly in need of refrigeration, and it was only discovered in the past few decades that eggs needed to be refrigerated during shipment rather than waiting to be refrigerated after arrival at the grocery store. Meats, poultry and fish all must be kept in climate-controlled environments before being sold. Refrigeration also helps keep fruits and vegetables edible longer.

The theoretical analysis of the MVCR system has been carried out for the following applications. These are the evaporator temperatures of vapour compression refrigeration system.

1) chilling of water : -2°C
2) Storage of milk : -5°C
3) Ice silo : -10°C
4) Ice cream : -20°C
5) Fishery item : -25°C
6) Cold storage & deep freezer : -30°C

5.5 Selection criterion for input parameters and their values

- Depending upon the six different applications of the system the evaporator temperature has been varied in the range of -3 to -35°C in step increment of 3°C.
- Normal ambient conditions were the deciding factor for the temperature range of condenser and absorber, which is 40°C to 50°C in step of 1°C.

Analysis of the Integrated Refrigeration System shown in Fig.6 has been carried out in this work is based on the following input parameters:

Refrigerants	R134a, R1234ze and R1234yf
Effectiveness of canopus HE	$\varepsilon = 0.80$
Condenser Temperature	$T_4 = 313 – 323$ K
Evaporator Temperature	T1= 243 - 270 K
Air inlet temperature	T_6, T_8= 283 K
Tonns of refrigeration	TR=10
Compressor efficiency	$\eta_{comp} = 0.75$
Mass flow rate of cooling fluid (air)	$m_a = 1.3$ Kg/s

By carrying out the thermodynamic analysis of the system for the conditions stated above the values of temperature T, pressure P, and concentration X at various state points of the cycle have been obtained. The computer program developed in EES [17] is used to compute various the performance parameters of the system such as cooling capacity of MVCR system, coefficient of performance of the combine system COP_{cps}, quantity of heat removed in Canopus HE q_{cps}, compressor work W_c etc.

5.6 Solution Methodology in Engineering Equation Solver (EES)

EES is a software package developed by Dr. Sanford Klein of the University of Wisconsin. EES incorporates the programming structures of C and FORTRAN with a built-in iterator, thermodynamic and transport property relations, graphical capacities, numerical integration, and many others useful mathematical functions. By grouping equations that are to be solved simultaneously, EES is able to function at a high rate of computational speed. Ammonia-water mixture properties are calculated in EES using the correlation developed by Ibrahim and Klein (1993). There are two major differences between EES and existing numerical equations solving programs. First, EES automatically identifies and groups equations that must be solved simultaneously. This feature simplifies the process for the user and ensures that the solver will always operate at optimum efficiency. Second, EES provides many built in mathematical and thermo-physical property functions useful for engineering calculations. The basic function provided by Engineering Equation Solver (EES) is the numerical solution of non-linear algebraic and differential equations, EES provides built in thermodynamic and transport property functions for many fluids including water, dry and moist air. Included in the property database are thermodynamic properties for R134a, R1234ze and R1234yf mixtures. Any information between quotation marks [“] or braces [{}] is an optional comment. Variable names must start with a letter.

A code containing a good library of working fluid properties suitable for heat pumps is the Engineering Equations Solver (EES) [17]. Here the user must write the equations governing the cycle and make sure the set is well-defined. In the case of a non-linear set of equations, the user must check the results to make sure that the mathematical solution is also a physical one. In one form or another, the user has to do a fair amount of programming to lead the simulator toward convergence to the correct solution.

5.7 Computer Program and Its Output

The computer program for the thermodynamic analysis of the system developed in EES has been given in Appendix A along with its flow diagram for computation procedure.

5.8 Performance Calculations

The performance of the system has been calculated by using the above said computer program using refrigerant R-134a, R-1234ze and R-1234yf and the corresponding results for both the refrigerants have been obtained. The results of one sample calculation have been shown in Appendix A.

CHAPTER 6

RESULTS AND DISCUSSION

In order to have a numerical appreciation of heat recovery from a vapour compression refrigeration system, a computer program has been developed in Engineering Equation Solver (version 9.223), using the set of input parameters given in previous chapter.

In the present study first of all observed that it is more relevant to use R1234ze, R1234yf, and R-134a as the refrigerant in modified VCR as value of ODP and GWP for these refrigerants is much less as compared to other refrigerants like R-407c, R-410a, R-32, R-22. From thermodynamic point of view, ammonia is also one of the best options so that its heat transfer coefficient is higher than R22, R11, R12 and R502 also from the point of availability, economic and environment, ammonia is better suited. But due to its toxicity and flammability it cannot be used for house hold appliances.

The effect of evaporator temperature and condenser temperature on air inlet temperature, and mass flow rate with constant effectiveness of the Canopus heat exchanger on various heat recovery parameters are presented in Table 5-22.

It reveals that:

(i) As the condenser temperature increases, T_7 and T_9 increases, ratio of refrigerant to water mass flow rates increases, the heat removal factor (HRF) also increases.

(ii) As evaporator temperature increases, the discharged temperature T_7 and T_9, the refrigerant mass flow rate and heat removal factor (HRF) decreases.

(iii) For increase in condenser temperature, the amount of heat removed in Canopus is always greater than heat removed in condenser and both decreses with increase in evaporator temperature for refrigerant R-134a.

almost remain constant, the ratio of the refrigerant to water mass flow rates decreases, while the fraction of the condenser heat removed by the CHE (FR) and heat removal factor (HRF) increases.

As cooling capacity increases, the COP of VCR with Canopus heat exchanger remain same for particular TE, mW, and mR whereas overall COP of the system considerably increases by introducing CHE.

Using our computer program we have calculated the results for a particular set of data. The effect of input parameter on condenser heat output and its distribution over the condenser and Canopus heat exchanger, the fraction of condenser heat available through CHE, the available water temperature and the heat recovery factor have been studied by changing each of the operating parameters while keeping the others constant. The parametric variations of the input variables have been undertaken to study their effects on the condenser heat recovered through the Canopus HE (CPS) as a function of evaporator temperature for different working fluids, namely R1234ze, R1234yf, and R-134a.

Effects of different temperatures i.e. Evaporator and Condenser, on different parameters COP_{cps}, COP_{svcr}, T_7, T_9, heat removed in Canopus HE (q_{cps}), heat removed in Condenser (q_{ca}), heat removal factor and m_R for R1234ze, R1234yf, and R-134a have tabulated below:

6.1 Tables for Refrigerant R-134a

Table 5. Effect of evaporator temperature (T_1) on different parameters for R-134a (at $T_c = T_4 = 313K$, $\varepsilon=0.80$, $\eta_c=0.75$, $m_a=1.3$ kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	318.4	322.7	35.89	32.04	1.12	2.05	4.771	0.2836
Run 2	246	317.9	321.8	35.06	31.56	1.111	2.194	5.037	0.2793
Run 3	249	317.4	320.9	34.25	31.09	1.102	2.352	5.33	0.2752
Run 4	252	316.9	320	33.46	30.64	1.092	2.527	5.653	0.2712
Run 5	255	316.4	319.2	32.7	30.21	1.082	2.722	6.011	0.2674
Run 6	258	315.9	318.3	31.96	29.79	1.073	2.938	6.409	0.2637
Run 7	261	315.5	317.5	31.24	29.39	1.063	3.182	6.856	0.2601
Run 8	264	315.1	316.8	30.54	29	1.053	3.456	7.359	0.2567
Run 9	267	314.6	316	29.87	28.63	1.043	3.769	7.929	0.2534
Run 10	270	314.2	315.3	29.21	28.26	1.034	4.126	8.582	0.2502

Table 6. Effect of evaporator temperature (T_1) on different parameters for R-134a
(at $T_c = T_4 = 315$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	318.8	324.6	37.63	32.4	1.162	1.95	4.664	0.2906
Run 2	246	318.3	323.7	36.8	31.9	1.153	2.086	4.923	0.2861
Run 3	249	317.7	322.8	35.98	31.42	1.145	2.234	5.206	0.2818
Run 4	252	317.2	321.9	35.19	30.96	1.137	2.398	5.518	0.2777
Run 5	255	316.7	321.1	34.43	30.51	1.128	2.58	5.863	0.2737
Run 6	258	316.3	320.2	33.68	30.08	1.12	2.782	6.245	0.2698
Run 7	261	315.8	319.4	32.96	29.66	1.111	3.008	6.672	0.266
Run 8	264	315.4	318.7	32.26	29.26	1.103	3.261	7.151	0.2624
Run 9	267	314.9	317.9	31.59	28.87	1.094	3.549	7.692	0.259
Run 10	270	314.5	317.2	30.93	28.5	1.085	3.876	8.307	0.2556

Table 7. Effect of evaporator temperature (T_1) on different parameters for R-134a
(at $T_c = T_4 = 317$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	319.2	326.5	39.37	32.78	1.201	1.856	4.556	0.298
Run 2	246	318.7	325.6	38.53	32.26	1.194	1.983	4.807	0.2933
Run 3	249	318.1	324.7	37.71	31.76	1.187	2.123	5.082	0.2888
Run 4	252	317.6	323.8	36.92	31.28	1.18	2.277	5.384	0.2844
Run 5	255	317.1	323	36.15	30.82	1.173	2.447	5.716	0.2802
Run 6	258	316.6	322.1	35.4	30.37	1.166	2.635	6.083	0.2762
Run 7	261	316.1	321.3	34.68	29.94	1.158	2.845	6.492	0.2723
Run 8	264	315.6	320.6	33.98	29.53	1.151	3.08	6.949	0.2685
Run 9	267	315.2	319.8	33.3	29.13	1.143	3.345	7.463	0.2648
Run 10	270	314.8	319.1	32.65	28.74	1.136	3.646	8.045	0.2613

Table 8. Effect of evaporator temperature (T_1) on different parameters for R-134a (at $T_c = T_4 = 319$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	319.7	328.4	41.1	33.17	1.239	1.765	4.447	0.3059
Run 2	246	319.1	327.5	40.26	32.63	1.234	1.886	4.691	0.301
Run 3	249	318.5	326.6	39.43	32.12	1.228	2.017	4.958	0.2962
Run 4	252	318	325.7	38.64	31.62	1.222	2.162	5.249	0.2916
Run 5	255	317.4	324.9	37.86	31.14	1.216	2.321	5.57	0.2872
Run 6	258	316.9	324	37.12	30.68	1.21	2.497	5.924	0.2829
Run 7	261	316.4	323.2	36.39	30.23	1.204	2.693	6.316	0.2788
Run 8	264	316	322.5	35.69	29.81	1.197	2.912	6.753	0.2749
Run 9	267	315.5	321.7	35.01	29.39	1.191	3.157	7.243	0.2711
Run 10	270	315.1	321	34.36	28.99	1.185	3.434	7.795	0.2674

Table 9. Effect of evaporator temperature (T_1) on different parameters for R-134a (at $T_c = T_4 = 321$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	320.1	330.3	42.83	33.58	1.275	1.679	4.337	0.3143
Run 2	246	319.5	329.4	41.98	33.02	1.271	1.793	4.575	0.3091
Run 3	249	318.9	328.5	41.15	32.49	1.267	1.917	4.833	0.304
Run 4	252	318.3	327.6	40.35	31.97	1.262	2.053	5.116	0.2992
Run 5	255	317.8	326.8	39.57	31.47	1.257	2.203	5.425	0.2946
Run 6	258	317.3	325.9	38.82	31	1.253	2.368	5.766	0.2901
Run 7	261	316.8	325.1	38.1	30.53	1.248	2.551	6.143	0.2858
Run 8	264	316.3	324.3	37.39	30.09	1.243	2.754	6.561	0.2816
Run 9	267	315.8	323.6	36.71	29.66	1.238	2.982	7.029	0.2776
Run 10	270	315.3	322.9	36.06	29.25	1.233	3.238	7.553	0.2738

Table 10. Effect of evaporator temperature (T_1) on different parameters for R-134a (at $T_c = T_4 = 323$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	320.6	332.2	44.55	34.01	1.31	1.597	4.226	0.3232
Run 2	246	320	331.3	43.69	33.43	1.307	1.704	4.457	0.3177
Run 3	249	319.3	330.4	42.86	32.87	1.304	1.822	4.708	0.3124
Run 4	252	318.8	329.5	42.06	32.34	1.301	1.95	4.982	0.3073
Run 5	255	318.2	328.6	41.28	31.82	1.297	2.091	5.281	0.3024
Run 6	258	317.6	327.8	40.52	31.32	1.294	2.246	5.61	0.2977
Run 7	261	317.1	327	39.8	30.85	1.29	2.417	5.972	0.2931
Run 8	264	316.6	326.2	39.09	30.39	1.287	2.607	6.374	0.2888
Run 9	267	316.1	325.5	38.41	29.94	1.283	2.818	6.82	0.2846
Run 10	270	315.6	324.7	37.76	29.52	1.279	3.055	7.32	0.2805

6.2 Tables for Refrigerant R-1234ze

Table 11. Effect of evaporator temperature (T_1) on different parameters for R-1234ze
(at $T_c = T_4 = 313$ K, $\varepsilon=0.80$, $\eta_c=0.75$, $m_a=1.3$ kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	321.7	317.8	31.47	34.99	0.8995	1.739	3.762	0.3274
Run 2	246	320.9	316.8	30.6	34.3	0.8922	1.877	4.001	0.321
Run 3	249	320.2	315.9	29.79	33.65	0.8854	2.029	4.264	0.3149
Run 4	252	319.5	315.1	29.04	33.02	0.8794	2.197	4.555	0.309
Run 5	255	318.8	314.3	28.33	32.41	0.8741	2.383	4.878	0.3033
Run 6	258	318.2	313.6	27.68	31.83	0.8697	2.589	5.239	0.2979
Run 7	261	317.6	312.9	27.08	31.27	0.866	2.821	5.644	0.2927
Run 8	264	317	312.3	26.53	30.73	0.8632	3.081	6.103	0.2876
Run 9	267	316.4	311.8	26.02	30.21	0.8612	3.377	6.625	0.2827
Run 10	270	315.8	311.2	25.55	29.71	0.86	3.715	7.223	0.2781

Table 12. Effect of evaporator temperature (T_1) on different parameters for R-1234ze
(at $T_c = T_4 = 315$ K, $\varepsilon=0.80$, $\eta_c=0.75$, $m_a=1.3$ kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	322.3	319.4	32.94	35.56	0.9263	1.656	3.672	0.3366
Run 2	246	321.5	318.5	32.07	34.84	0.9206	1.786	3.904	0.3298
Run 3	249	320.8	317.6	31.27	34.16	0.9154	1.929	4.159	0.3233
Run 4	252	320	316.7	30.52	33.5	0.9109	2.087	4.441	0.3171
Run 5	255	319.3	316	29.82	32.87	0.9071	2.261	4.752	0.3112
Run 6	258	318.7	315.3	29.17	32.27	0.9041	2.454	5.099	0.3054
Run 7	261	318	314.6	28.58	31.68	0.902	2.669	5.488	0.2999
Run 8	264	317.4	314	28.03	31.12	0.9006	2.91	5.926	0.2946
Run 9	267	316.8	313.4	27.53	30.59	0.9	3.183	6.421	0.2895
Run 10	270	316.2	312.9	27.06	30.07	0.9	3.493	6.986	0.2846

Table 13. Effect of evaporator temperature (T_1) on different parameters for R-1234ze
(at $T_c = T_4 = 317$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	323	321	34.39	36.15	0.9514	1.576	3.581	0.3463
Run 2	246	322.1	320.1	33.54	35.4	0.9473	1.7	3.807	0.3391
Run 3	249	321.4	319.2	32.74	34.69	0.9437	1.835	4.054	0.3323
Run 4	252	320.6	318.4	31.99	34.01	0.9407	1.983	4.327	0.3257
Run 5	255	319.9	317.6	31.3	33.35	0.9385	2.146	4.628	0.3195
Run 6	258	319.2	316.9	30.66	32.72	0.937	2.326	4.962	0.3134
Run 7	261	318.5	316.2	30.07	32.12	0.9363	2.527	5.335	0.3076
Run 8	264	317.9	315.6	29.53	31.53	0.9363	2.751	5.753	0.3021
Run 9	267	317.2	315.1	29.03	30.97	0.9371	3.003	6.225	0.2967
Run 10	270	316.6	314.6	28.57	30.44	0.9386	3.288	6.76	0.2915

Table 14. Effect of evaporator temperature (T_1) on different parameters for R-1234ze
(at $T_c = T_4 = 319$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	323.7	322.6	35.85	36.78	0.9745	1.5	3.488	0.3567
Run 2	246	322.8	321.7	35	36	0.9721	1.617	3.709	0.3491
Run 3	249	322	320.8	34.2	35.25	0.9702	1.744	3.949	0.3418
Run 4	252	321.2	320	33.46	34.54	0.9688	1.884	4.213	0.3349
Run 5	255	320.4	319.2	32.77	33.85	0.9681	2.037	4.505	0.3283
Run 6	258	319.7	318.5	32.14	33.2	0.9681	2.206	4.827	0.3219
Run 7	261	319	317.9	31.55	32.57	0.9689	2.393	5.185	0.3158
Run 8	264	318.3	317.3	31.02	31.96	0.9704	2.602	5.586	0.3099
Run 9	267	317.7	316.7	30.52	31.38	0.9726	2.836	6.035	0.3043
Run 10	270	317.1	316.2	30.07	30.82	0.9755	3.099	6.543	0.2989

Table 15. Effect of evaporator temperature (T_1) on different parameters for R-1234ze (at $T_c = T_4 = 321$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	324.4	324.2	37.3	37.45	0.9958	1.427	3.395	0.3677
Run 2	246	323.5	323.3	36.45	36.63	0.9951	1.538	3.61	0.3597
Run 3	249	322.6	322.4	35.66	35.85	0.9948	1.658	3.844	0.352
Run 4	252	321.8	321.6	34.93	35.1	0.995	1.79	4.1	0.3446
Run 5	255	321	320.9	34.24	34.38	0.9959	1.934	4.382	0.3376
Run 6	258	320.3	320.2	33.61	33.7	0.9975	2.093	4.693	0.3309
Run 7	261	319.5	319.5	33.03	33.04	0.9998	2.268	5.037	0.3244
Run 8	264	318.8	318.9	32.5	32.41	1.003	2.463	5.421	0.3182
Run 9	267	318.2	318.4	32.01	31.81	1.006	2.68	5.851	0.3123
Run 10	270	317.5	317.9	31.56	31.22	1.011	2.924	6.334	0.3066

Table 16. Effect of evaporator temperature (T_1) on different parameters for R-1234ze (at $T_c = T_4 = 323$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	325.2	325.8	38.74	38.16	1.015	1.357	3.301	0.3796
Run 2	246	324.2	324.9	37.9	37.3	1.016	1.462	3.51	0.371
Run 3	249	323.3	324	37.12	36.48	1.018	1.576	3.738	0.3628
Run 4	252	322.5	323.2	36.39	35.69	1.019	1.7	3.986	0.355
Run 5	255	321.6	322.5	35.71	34.94	1.022	1.836	4.259	0.3476
Run 6	258	320.8	321.8	35.09	34.23	1.025	1.985	4.56	0.3405
Run 7	261	320.1	321.2	34.51	33.54	1.029	2.15	4.892	0.3336
Run 8	264	319.4	320.6	33.98	32.88	1.034	2.331	5.26	0.3271
Run 9	267	318.7	320	33.5	32.25	1.039	2.534	5.671	0.3208
Run 10	270	318	319.5	33.05	31.65	1.045	2.76	6.132	0.3148

6.3 Tables for Refrigerant R-1234yf

Table 17. Effect of evaporator temperature (T_1) on different parameters for R-1234yf
(at $T_c = T_4 = 313$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	323.1	314.8	28.76	36.25	0.7933	1.633	3.369	0.3919
Run 2	246	322.2	314.1	28.11	35.46	0.7927	1.767	3.603	0.3833
Run 3	249	321.4	313.4	27.5	34.7	0.7926	1.915	3.861	0.3751
Run 4	252	320.6	312.8	26.94	33.97	0.793	2.078	4.147	0.3672
Run 5	255	319.8	312.2	26.42	33.28	0.794	2.259	4.465	0.3597
Run 6	258	319.1	311.7	25.94	32.61	0.7955	2.461	4.821	0.3525
Run 7	261	318.4	311.2	25.51	31.98	0.7977	2.687	5.221	0.3457
Run 8	264	317.7	310.8	25.11	31.37	0.8004	2.942	5.672	0.3391
Run 9	267	317	310.4	24.74	30.78	0.8037	3.232	6.187	0.3328
Run 10	270	316.4	310	24.41	30.22	0.8075	3.563	6.777	0.3267

Table 18. Effect of evaporator temperature (T_1) on different parameters for R-1234yf
(at $T_c = T_4 = 315$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	323.8	316.4	30.2	36.94	0.8175	1.547	3.273	0.4051
Run 2	246	322.9	315.7	29.55	36.1	0.8185	1.673	3.501	0.3959
Run 3	249	322	315	28.95	35.31	0.82	1.812	3.751	0.3872
Run 4	252	321.2	314.4	28.4	34.54	0.822	1.965	4.028	0.3788
Run 5	255	320.4	313.8	27.88	33.82	0.8246	2.134	4.334	0.3709
Run 6	258	319.6	313.3	27.41	33.12	0.8277	2.322	4.676	0.3632
Run 7	261	318.9	312.8	26.98	32.46	0.8313	2.533	5.058	0.3559
Run 8	264	318.2	312.4	26.59	31.82	0.8355	2.768	5.489	0.349
Run 9	267	317.5	312	26.23	31.21	0.8403	3.035	5.977	0.3423
Run 10	270	316.9	311.6	25.9	30.63	0.8456	3.338	6.533	0.3359

Table 19. Effect of evaporator temperature (T_1) on different parameters for R-1234yf (at $T_c = T_4 = 317$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	324.7	318	31.64	37.68	0.8396	1.464	3.177	0.4194
Run 2	246	323.7	317.3	30.99	36.79	0.8424	1.584	3.398	0.4096
Run 3	249	322.7	316.6	30.4	35.95	0.8456	1.714	3.641	0.4002
Run 4	252	321.9	316	29.85	35.15	0.8493	1.858	3.908	0.3913
Run 5	255	321	315.4	29.35	34.39	0.8534	2.016	4.204	0.3828
Run 6	258	320.2	314.9	28.88	33.66	0.8581	2.192	4.532	0.3747
Run 7	261	319.4	314.5	28.46	32.96	0.8633	2.387	4.899	0.3669
Run 8	264	318.7	314	28.07	32.29	0.8691	2.606	5.31	0.3595
Run 9	267	318	313.6	27.71	31.66	0.8753	2.852	5.773	0.3524
Run 10	270	317.3	313.3	27.39	31.05	0.8821	3.131	6.3	0.3456

Table 20. Effect of evaporator temperature (T_1) on different parameters for R-1234yf (at $T_c = T_4 = 319$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	325.5	319.6	33.07	38.46	0.8598	1.385	3.078	0.4349
Run 2	246	324.5	318.9	32.44	37.53	0.8644	1.498	3.293	0.4243
Run 3	249	323.5	318.2	31.85	36.64	0.8693	1.621	3.529	0.4143
Run 4	252	322.6	317.6	31.31	35.79	0.8747	1.756	3.788	0.4047
Run 5	255	321.7	317.1	30.81	34.99	0.8805	1.904	4.073	0.3956
Run 6	258	320.8	316.6	30.35	34.22	0.8868	2.069	4.389	0.387
Run 7	261	320	316.1	29.93	33.49	0.8936	2.251	4.741	0.3787
Run 8	264	319.3	315.7	29.54	32.79	0.9009	2.454	5.134	0.3708
Run 9	267	318.5	315.3	29.19	32.13	0.9088	2.681	5.575	0.3633
Run 10	270	317.8	314.9	28.88	31.49	0.917	2.938	6.074	0.3561

Table 21. Effect of evaporator temperature (T_1) on different parameters for R-1234yf
(at $T_c = T_4 = 321$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	326.4	321.1	34.5	39.3	0.878	1.309	2.978	0.4517
Run 2	246	325.4	320.5	33.88	38.31	0.8843	1.415	3.187	0.4403
Run 3	249	324.3	319.8	33.3	37.37	0.891	1.531	3.416	0.4295
Run 4	252	323.3	319.2	32.76	36.48	0.8982	1.658	3.667	0.4193
Run 5	255	322.4	318.7	32.27	35.63	0.9057	1.798	3.942	0.4095
Run 6	258	321.5	318.2	31.82	34.82	0.9137	1.952	4.247	0.4002
Run 7	261	320.6	317.7	31.4	34.05	0.9222	2.122	4.584	0.3914
Run 8	264	319.8	317.3	31.02	33.32	0.9311	2.311	4.961	0.383
Run 9	267	319.1	316.9	30.68	32.62	0.9405	2.522	5.382	0.3749
Run 10	270	318.3	316.6	30.37	31.95	0.9504	2.759	5.856	0.3673

Table 22. Effect of evaporator temperature (T_1) on different parameters for R-1234yf
(at $T_c = T_4 = 323$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s)

	T_1	T_7	T_9	q_{cps}	q_{ca}	HRF	COP_{svcr}	COP_{cps}	m_R
Run 1	243	327.4	322.7	35.94	40.2	0.8939	1.235	2.877	0.4701
Run 2	246	326.3	322	35.32	39.15	0.9022	1.336	3.08	0.4578
Run 3	249	325.2	321.4	34.74	38.15	0.9107	1.445	3.302	0.4461
Run 4	252	324.1	320.8	34.22	37.2	0.9197	1.565	3.545	0.435
Run 5	255	323.1	320.3	33.73	36.31	0.929	1.696	3.811	0.4246
Run 6	258	322.2	319.8	33.28	35.46	0.9388	1.84	4.104	0.4146
Run 7	261	321.3	319.3	32.88	34.65	0.9489	1.999	4.429	0.4051
Run 8	264	320.5	318.9	32.5	33.87	0.9596	2.176	4.79	0.3961
Run 9	267	319.6	318.6	32.17	33.14	0.9706	2.372	5.192	0.3875
Run 10	270	318.9	318.2	31.86	32.44	0.9821	2.591	5.643	0.3793

6.4 Results for Refrigerant R-134a

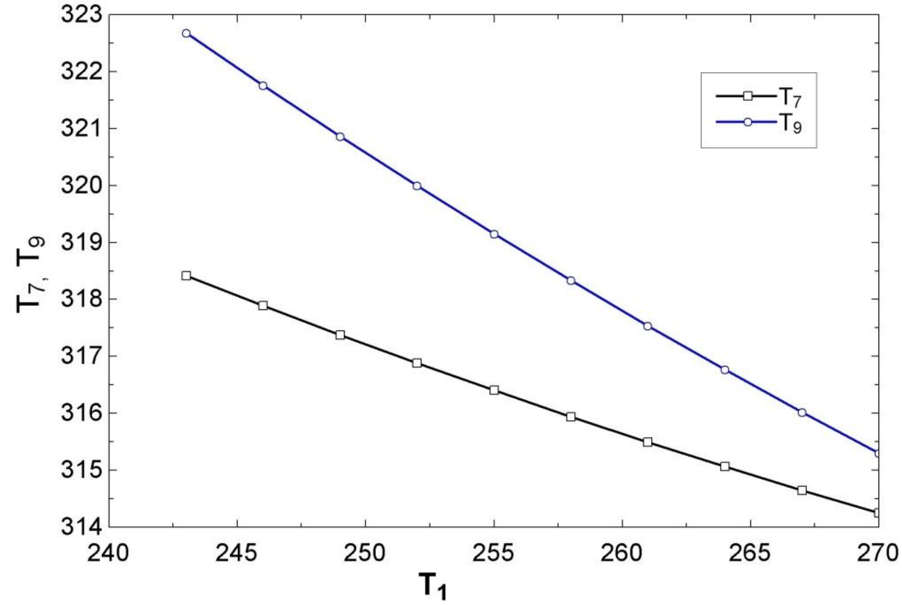

Fig. 8: Variation in T_7 and T_9 temperature. with varying evaporator temp. (at $T_c = 313$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

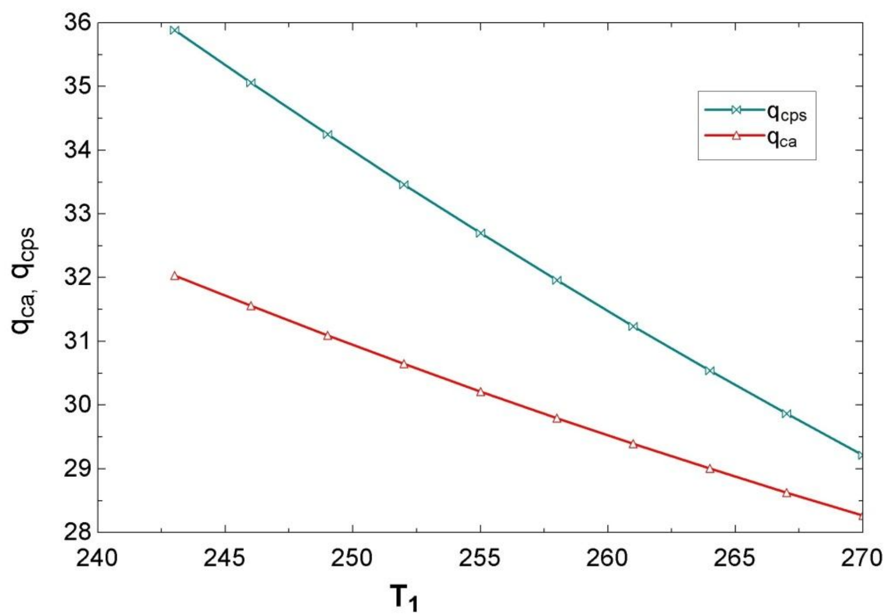

Fig. 9: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at $T_c = 313$ K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R134a)

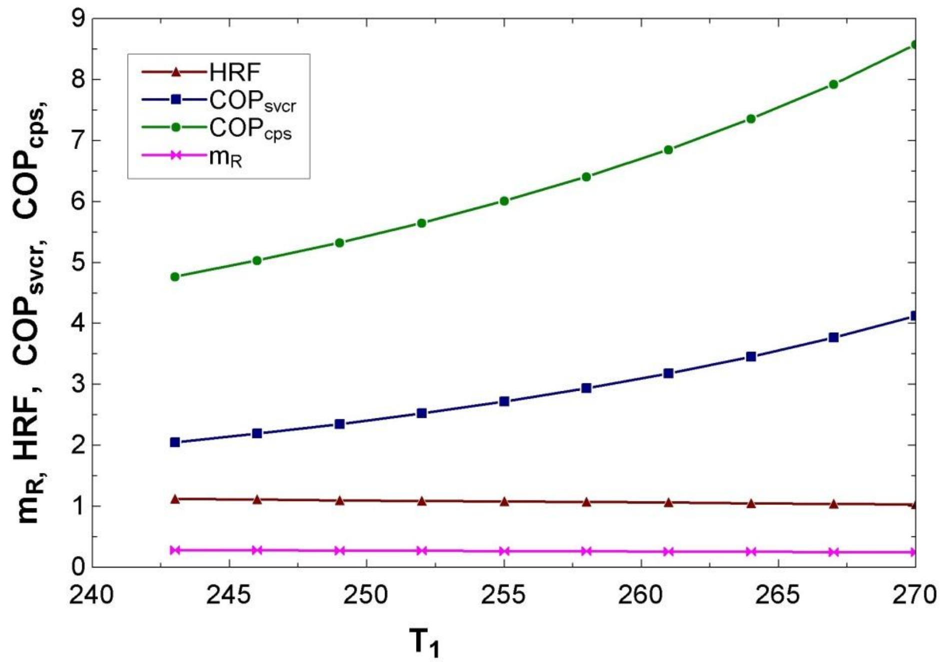

Fig. 10: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at $T_c = 313$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

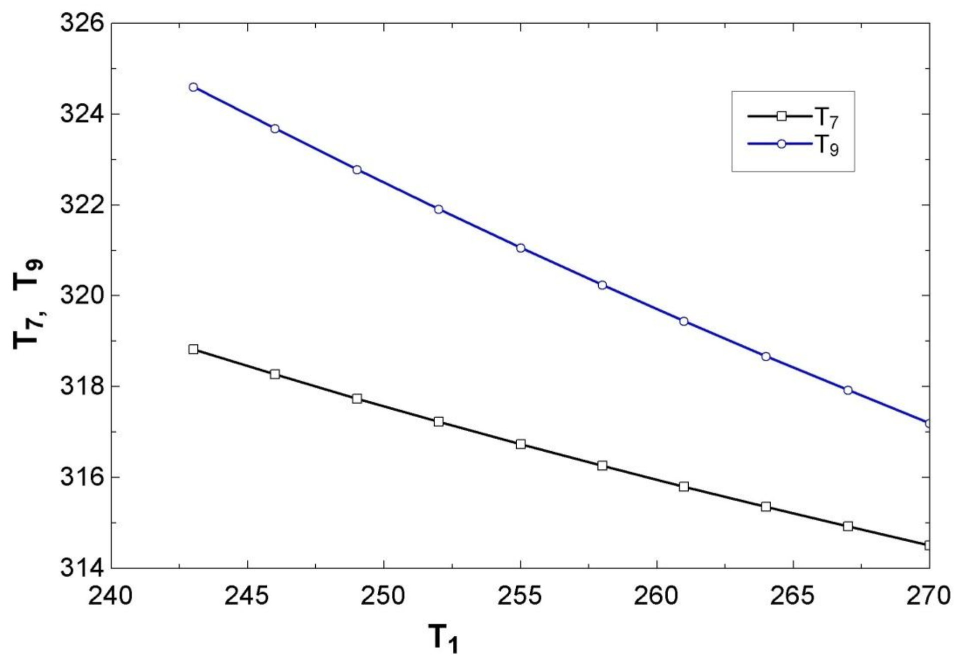

Fig. 11: Variation in T_7 and T_9 temperature. with varying evaporator temp. (at $T_c = 315$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

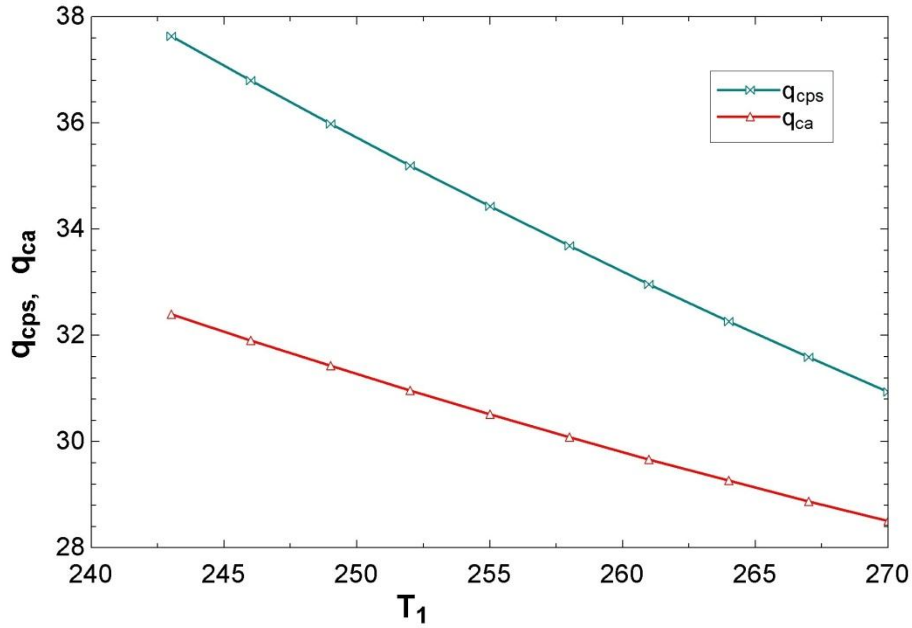

Fig. 12: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 315 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R134a)

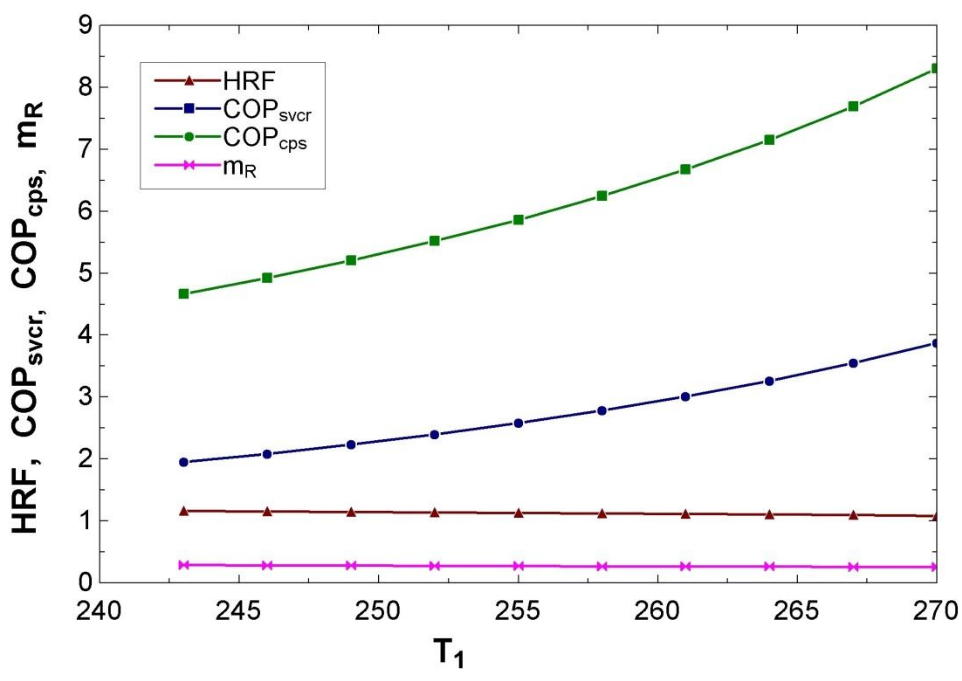

Fig. 13: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 315 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

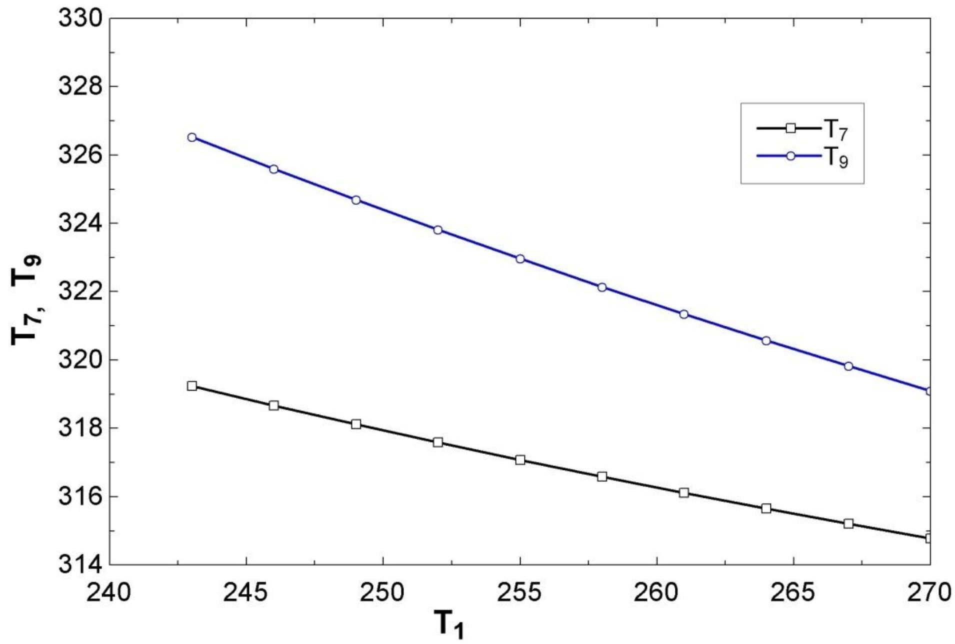

Fig. 14: Variation in T_7 and T_9 temperature. with varying evaporator temp. (at T_c = 317 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

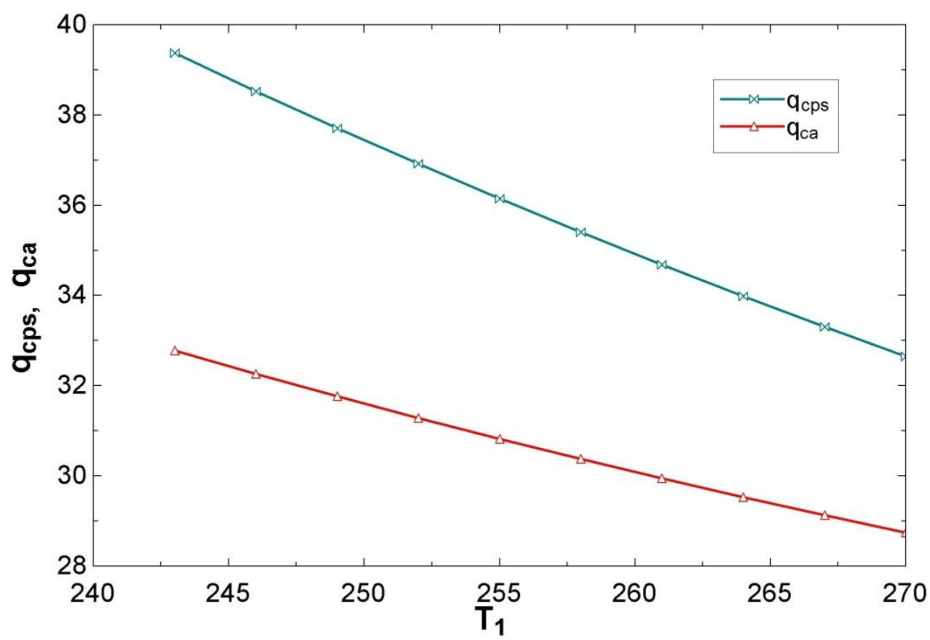

Fig. 15: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 317 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R134a)

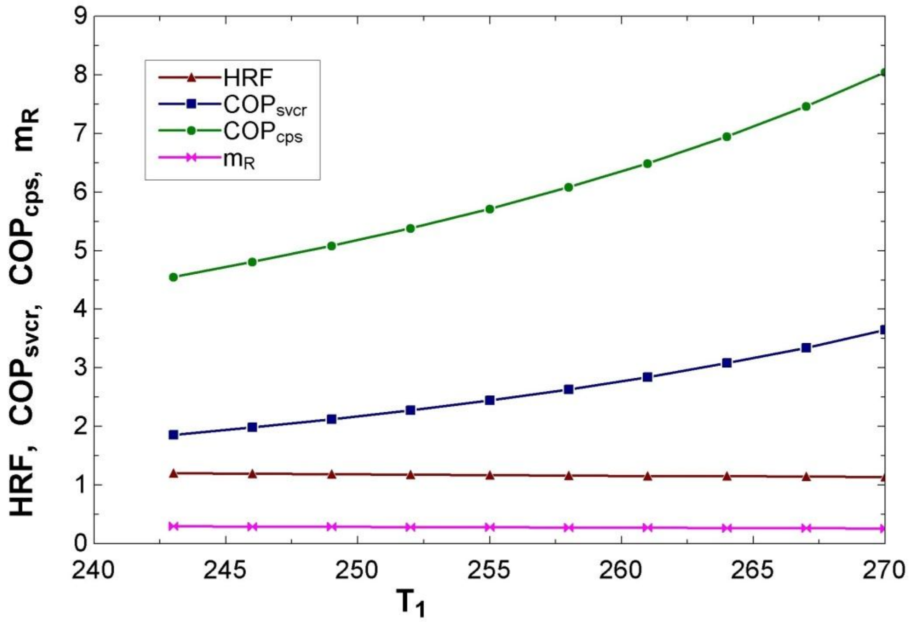

Fig. 16: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 317 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

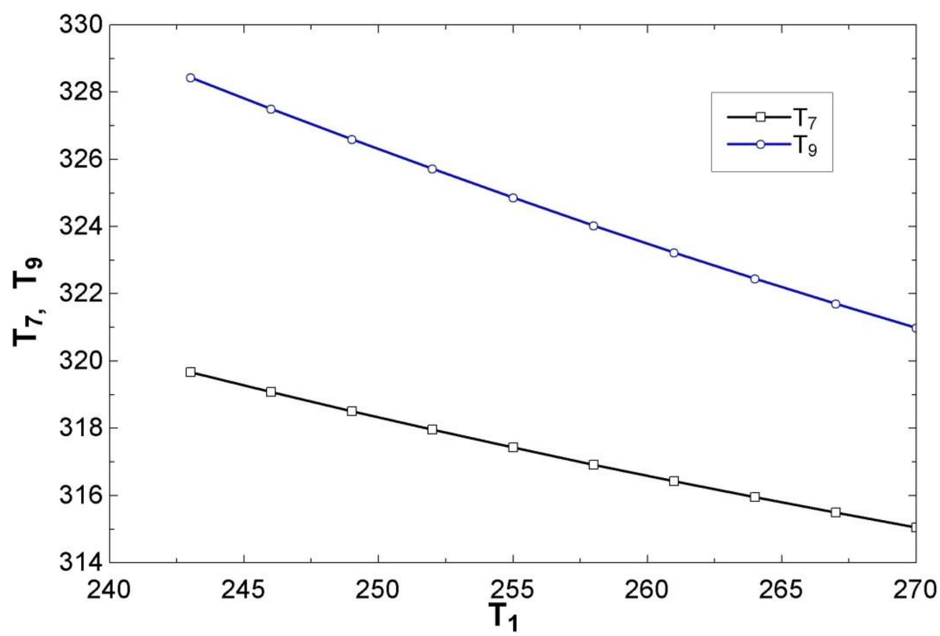

Fig. 17: Variation in T_7 and T_9 temperature. with varying evaporator temp. (at T_c = 319 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

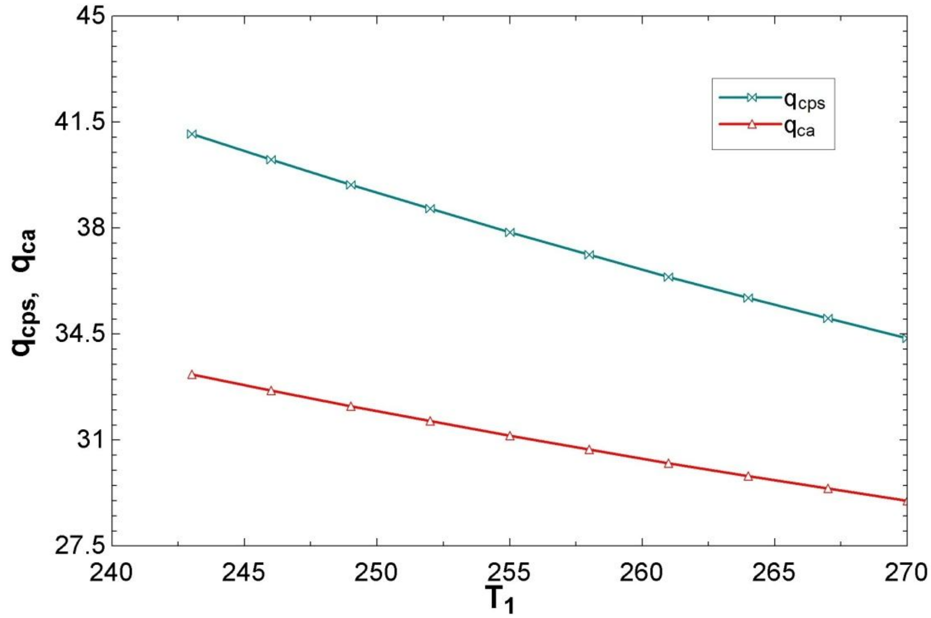

Fig. 18: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at $T_c = 319$ K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R134a)

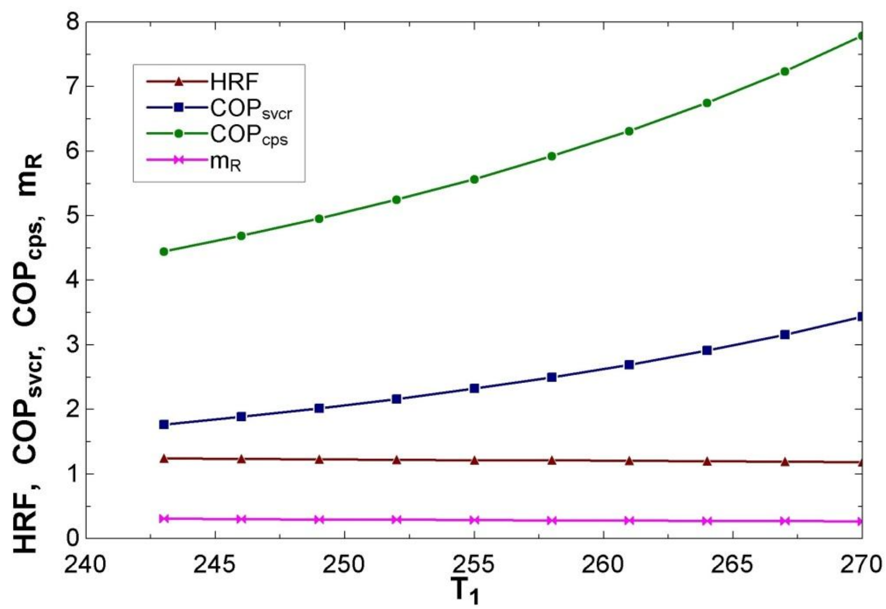

Fig. 19: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at $T_c = 319$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

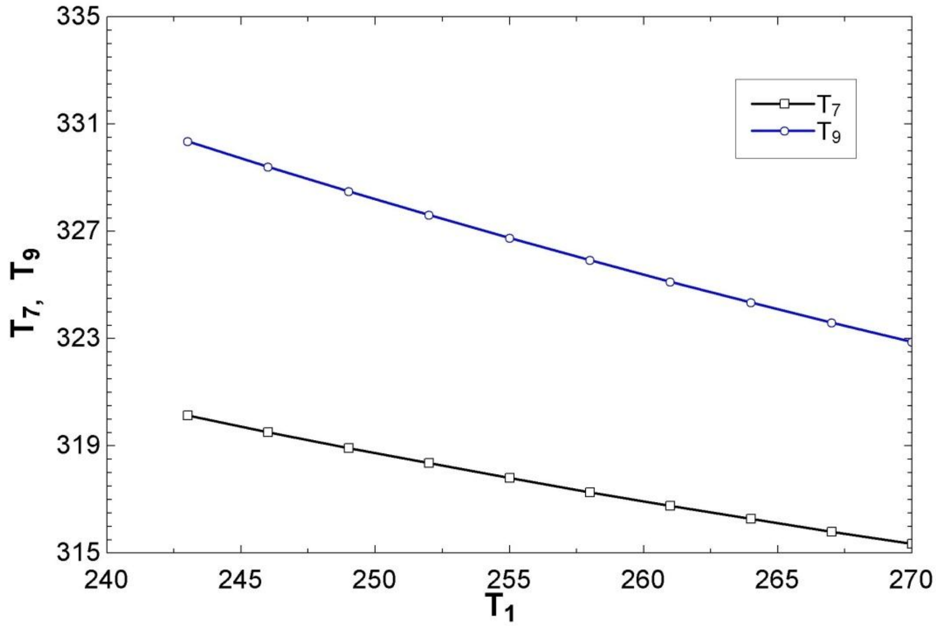

Fig. 20: Variation in T_7 and T_9 temperature. with varying evaporator temp. (at $T_c = 321$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

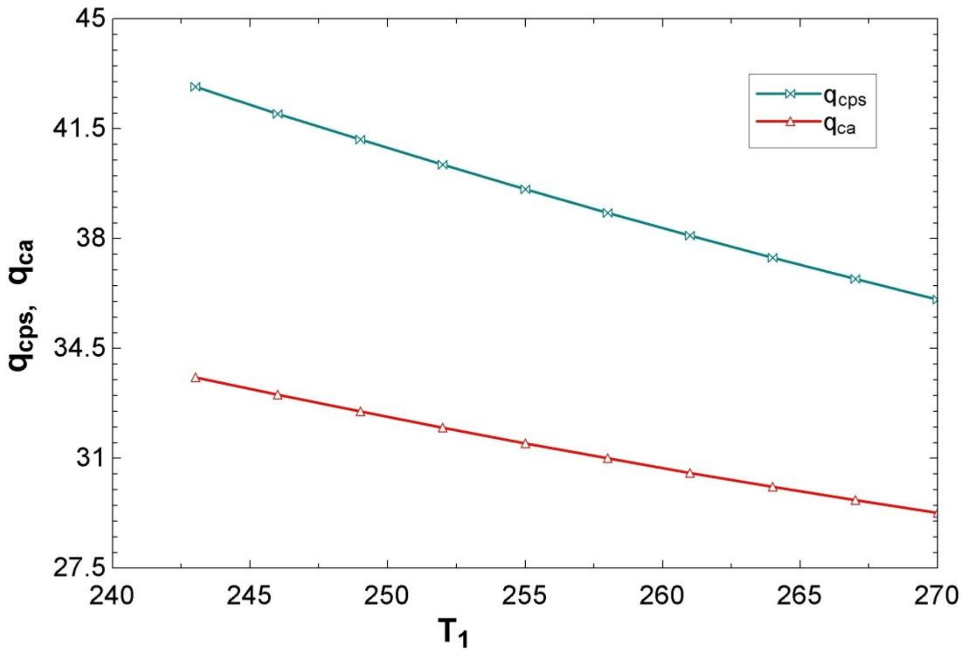

Fig. 21: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at $T_c = 321$ K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R134a)

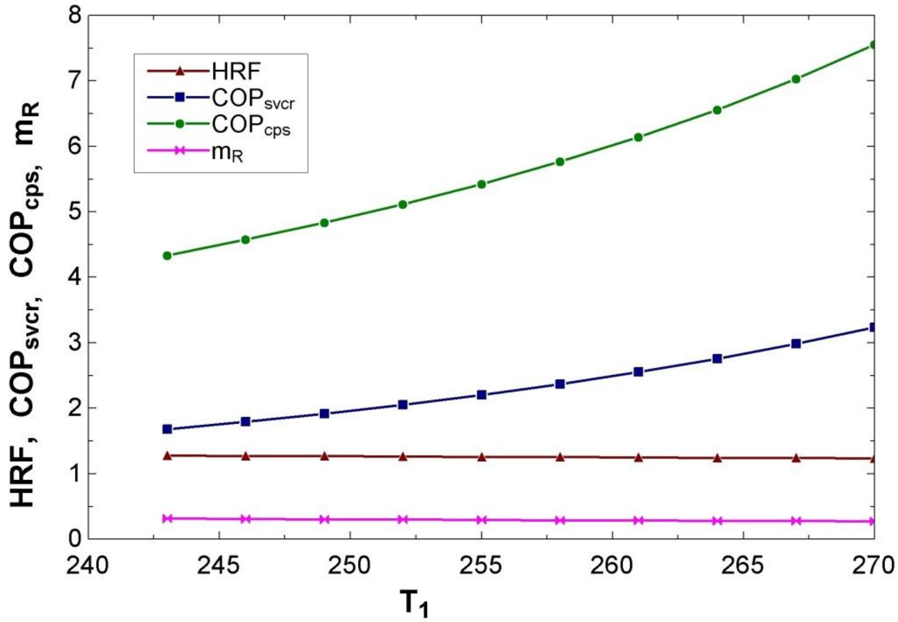

Fig. 22: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 321 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

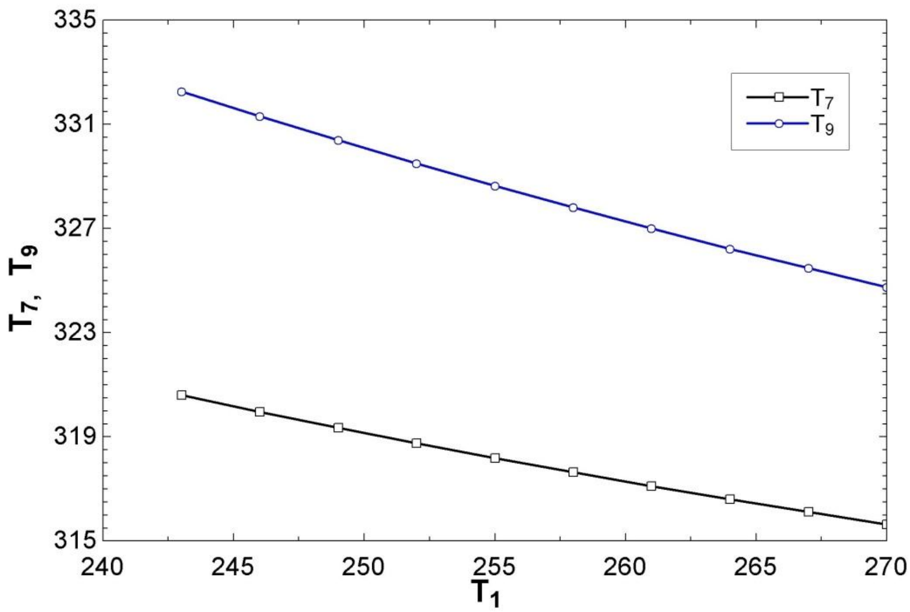

Fig. 23: Variation in T_7 and T_9 temperature. with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

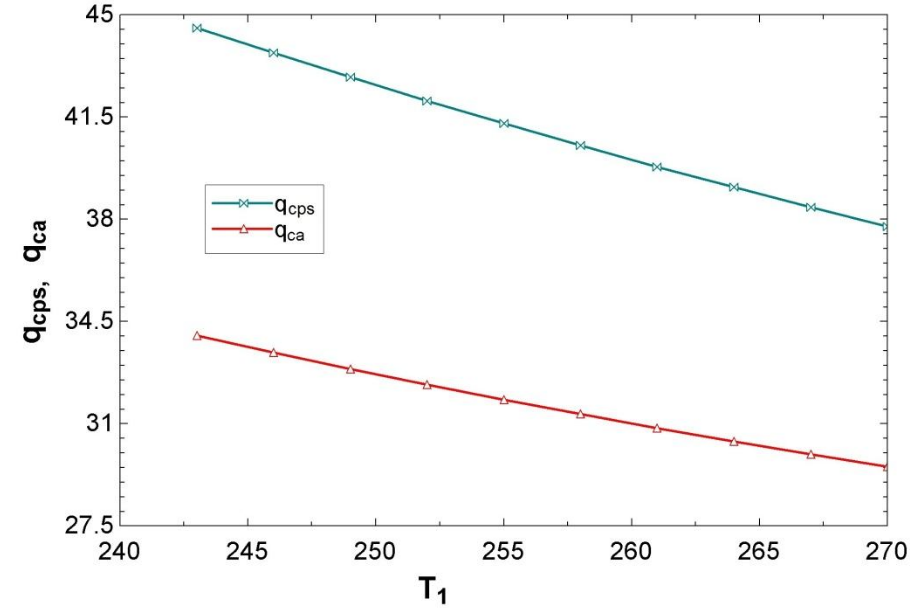

Fig. 24: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R134a)

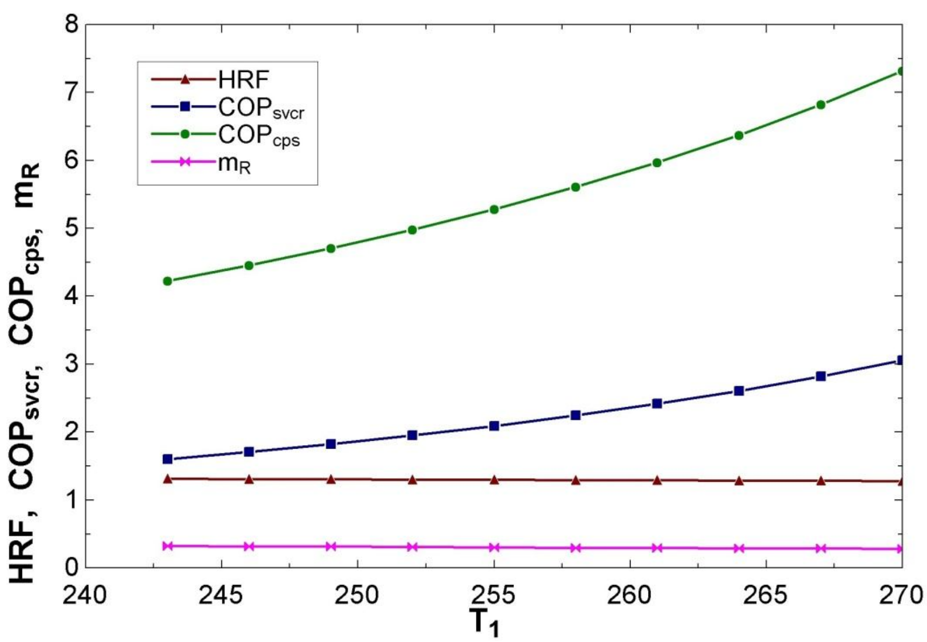

Fig. 25: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R134a)

6.5 Results for Refrigerant R-1234ze

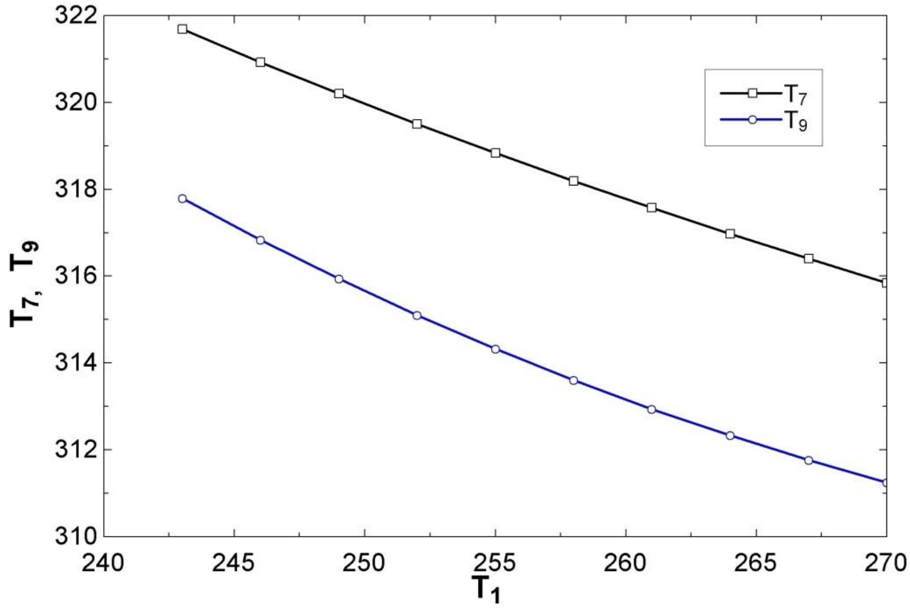

Fig. 26: Variation in T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at $T_c = 313$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

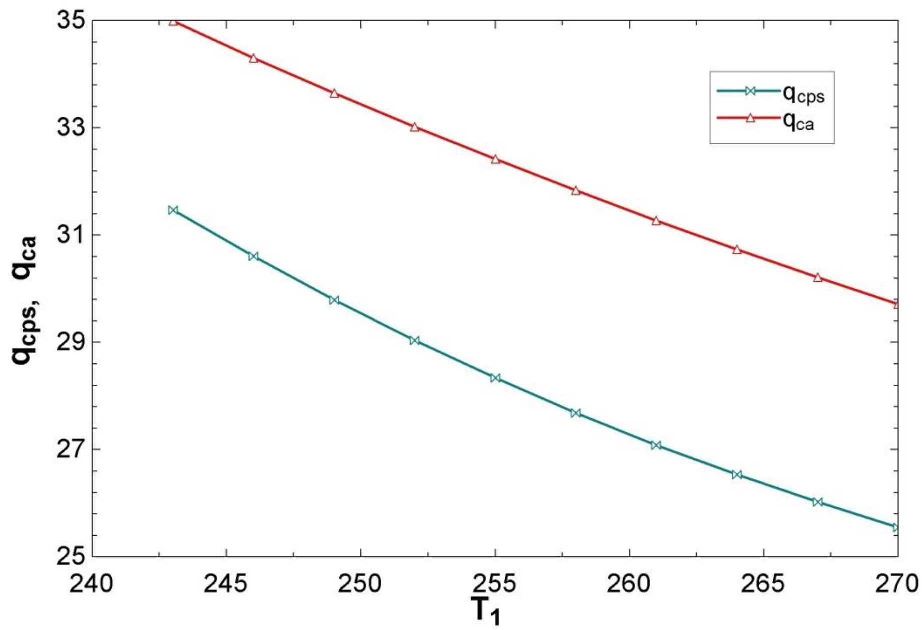

Fig. 27: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at $T_c = 313$ K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234ze)

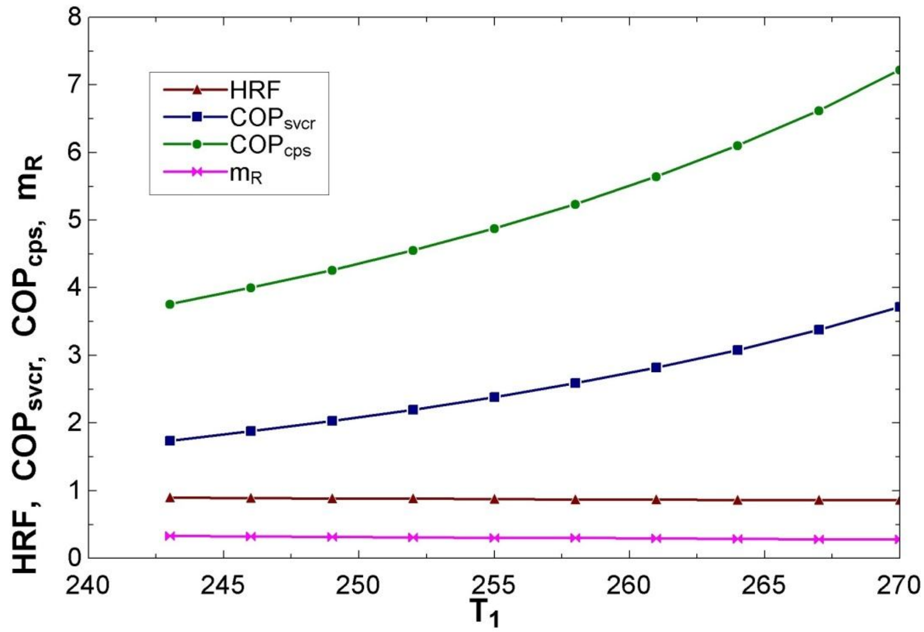

Fig. 28: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at $T_c = 313$ K, $\varepsilon=0.80$, $\eta_c=0.75$, $m_a=1.3$ kg/s, R1234ze)

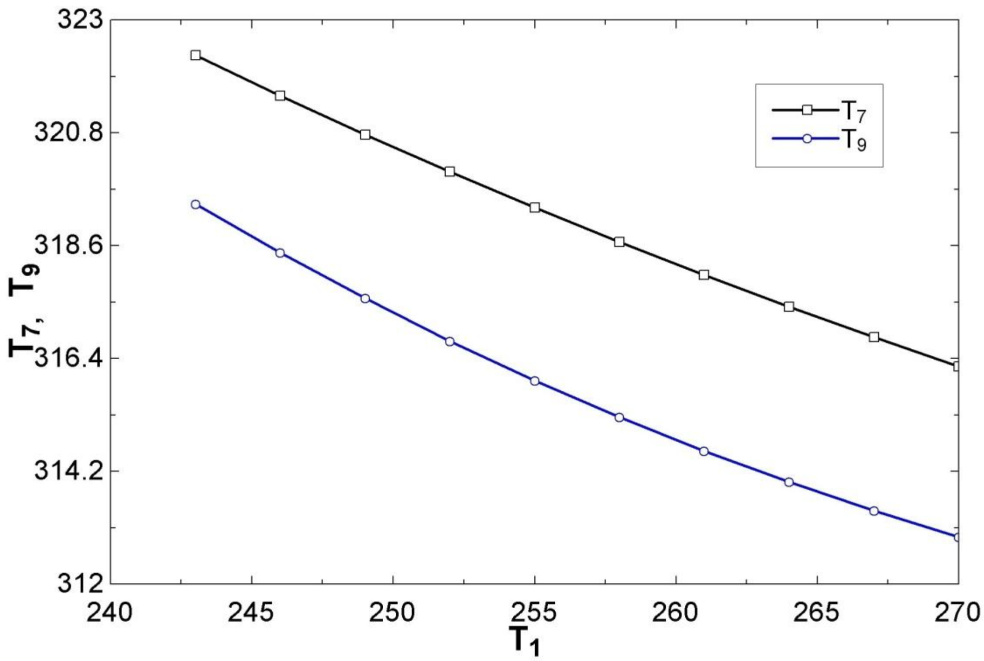

Fig. 29: Variation in T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at $T_c = 315$ K, $\varepsilon=0.80$, $\eta_c=0.75$, $m_a=1.3$ kg/s, R1234ze)

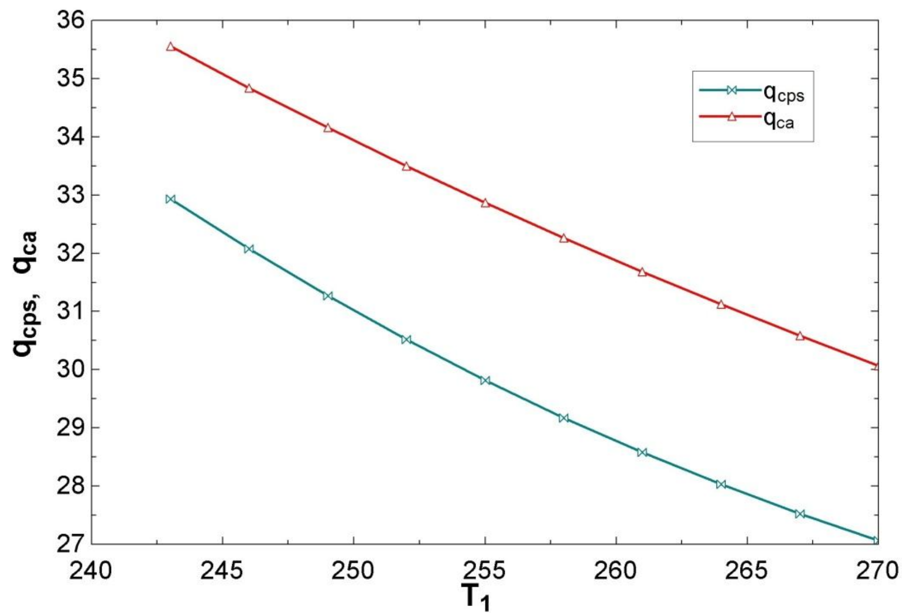

Fig. 30: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 315 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234ze)

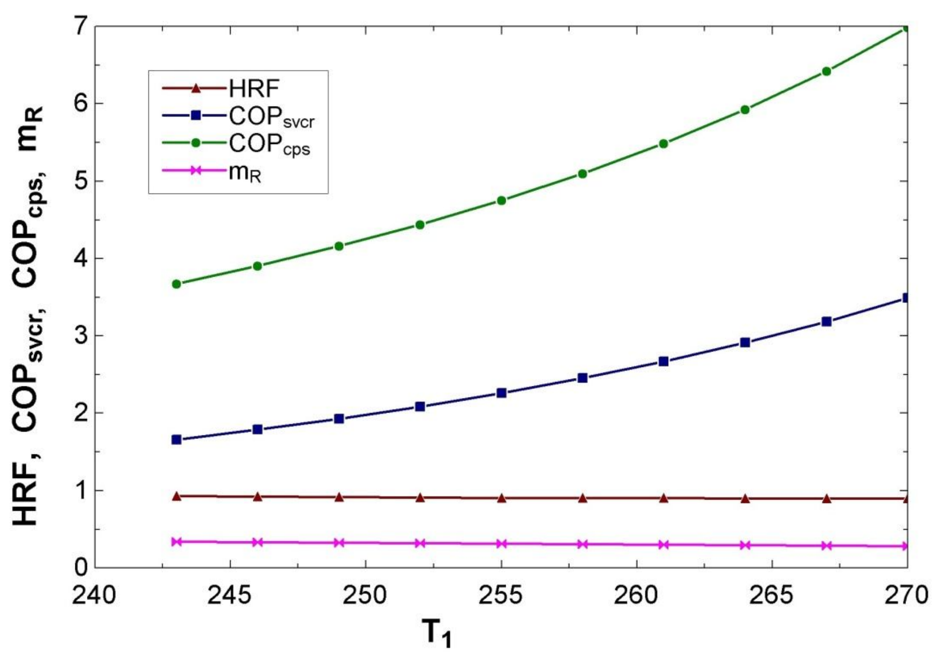

Fig. 31: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 315 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

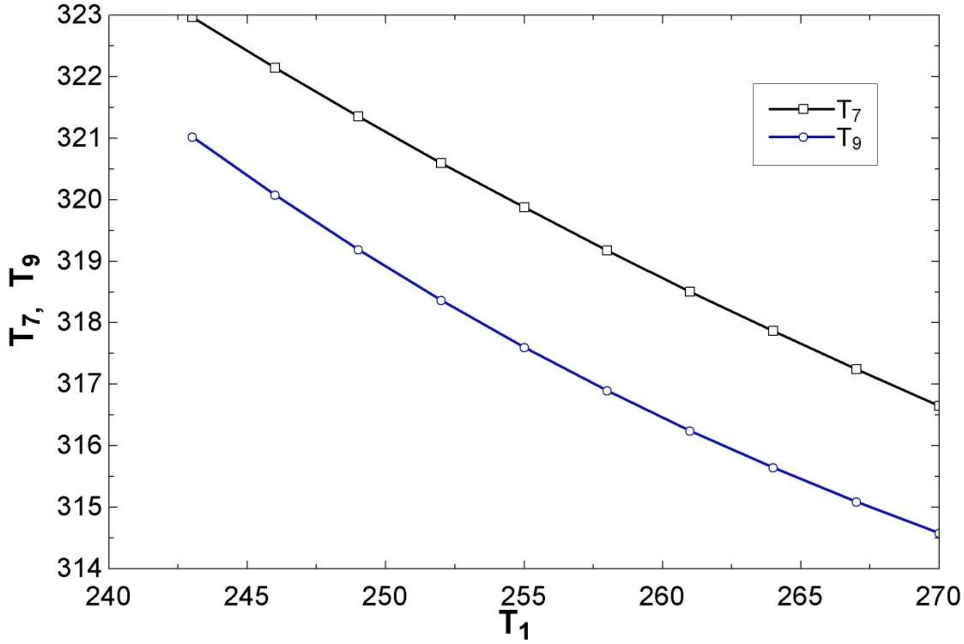

Fig. 32: Variation in T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at $T_c = 317$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

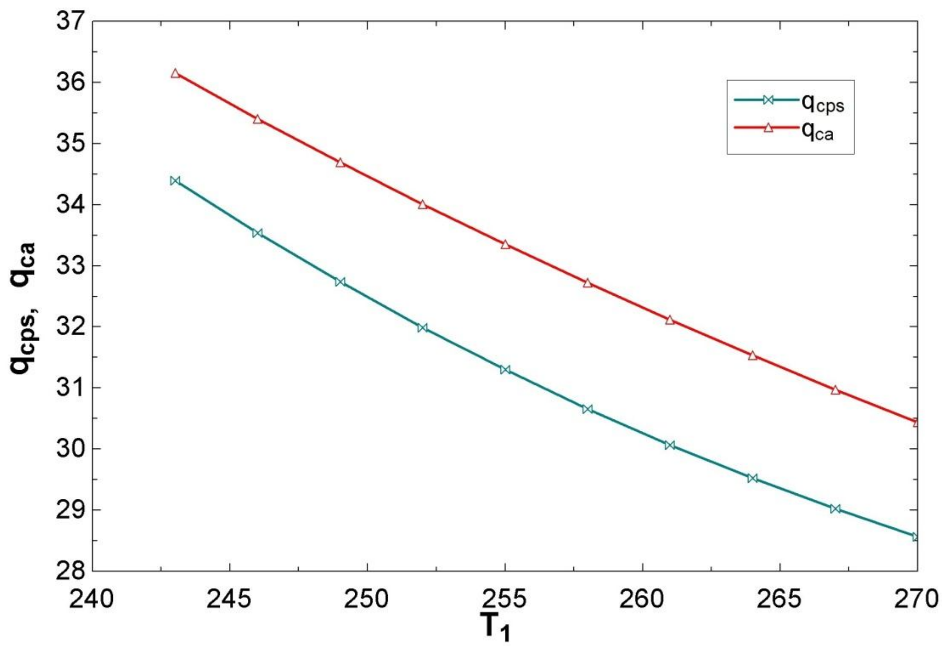

Fig. 33: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at $T_c = 317$ K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234ze)

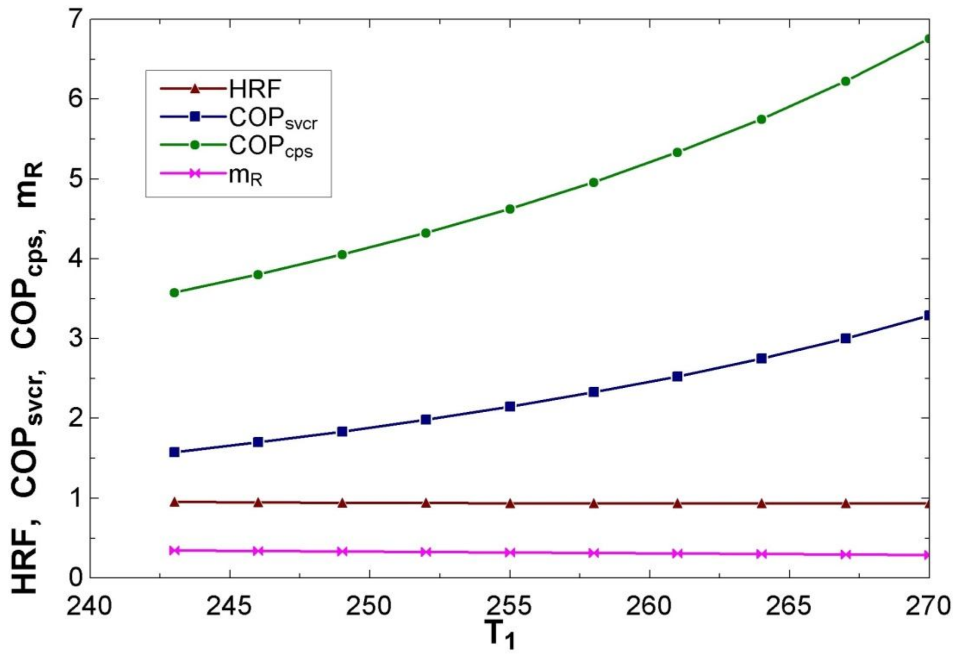

Fig. 34: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 317 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

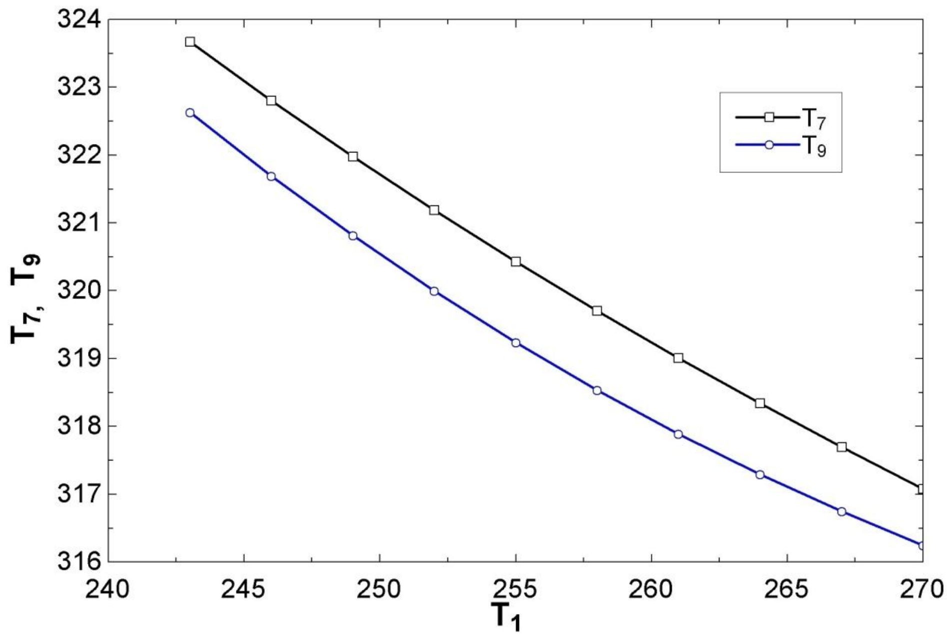

Fig. 35: Variation in T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 319 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

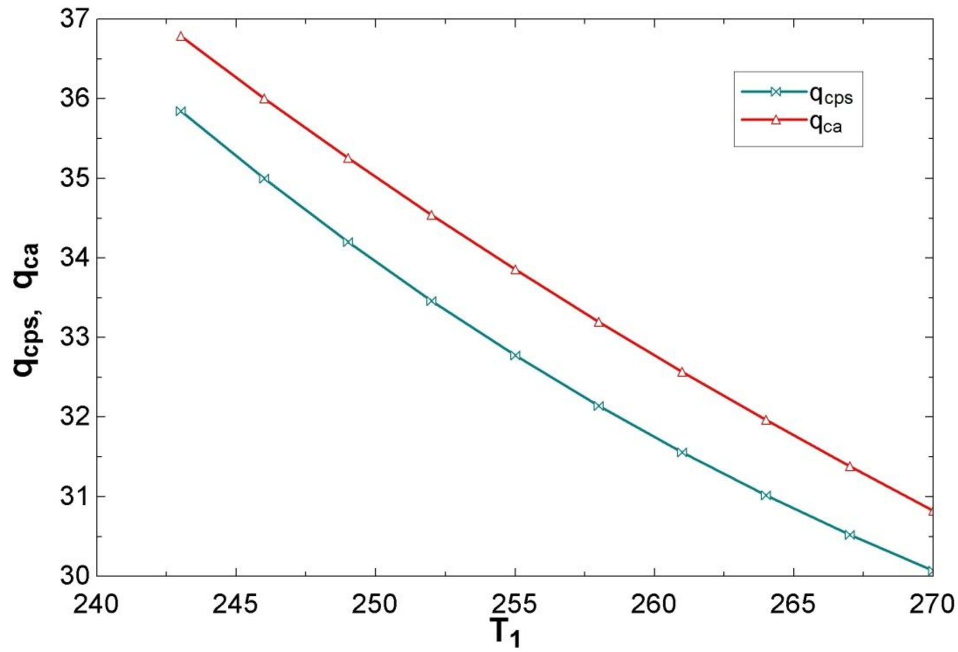

Fig. 36: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 319 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234ze)

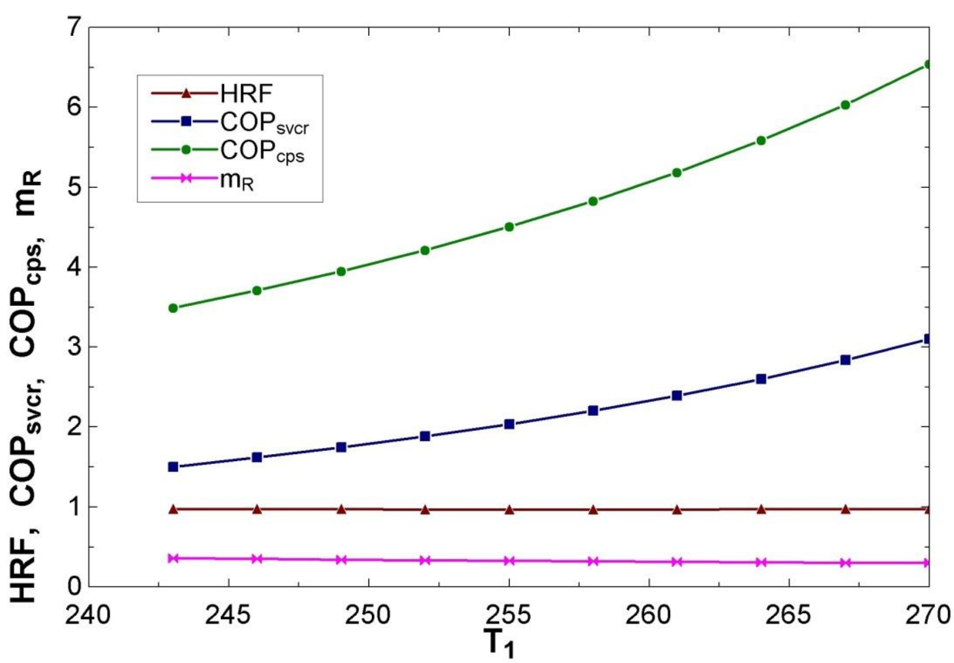

Fig. 37: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 319 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

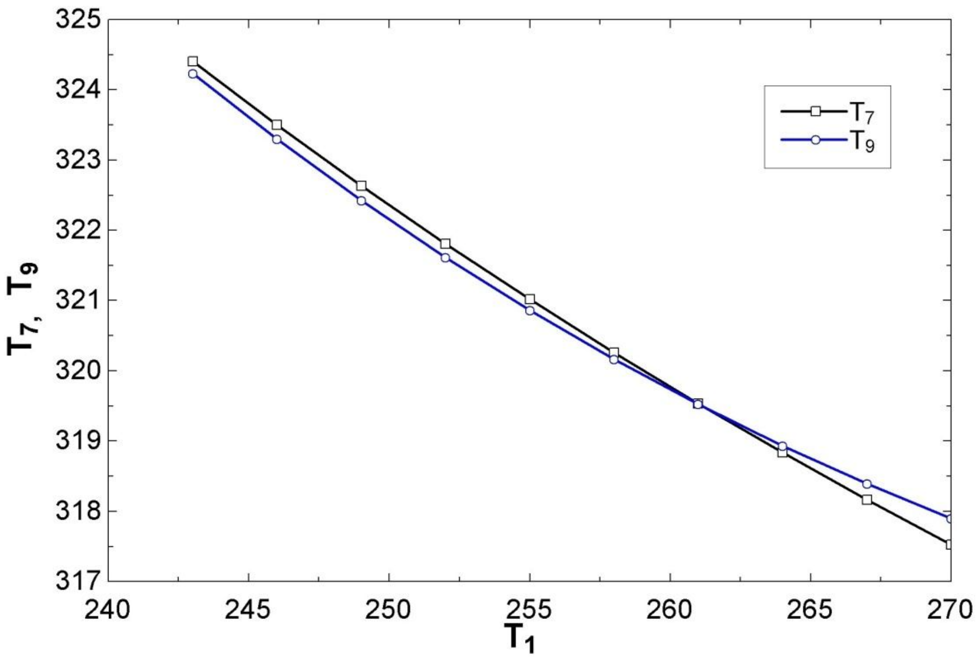

Fig. 38: Variation in T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 321 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

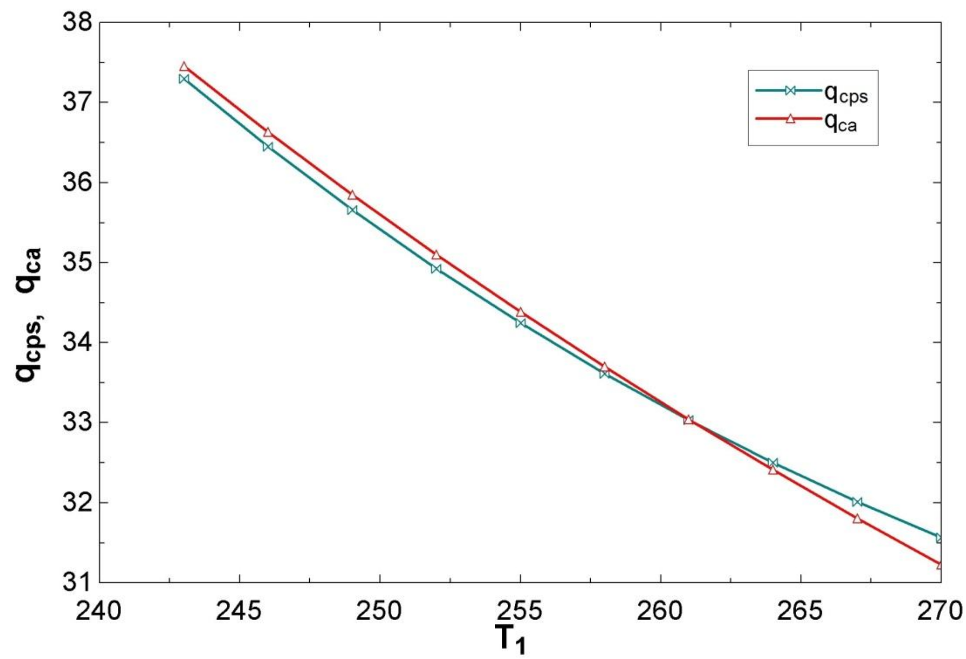

Fig. 39: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 321 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234ze)

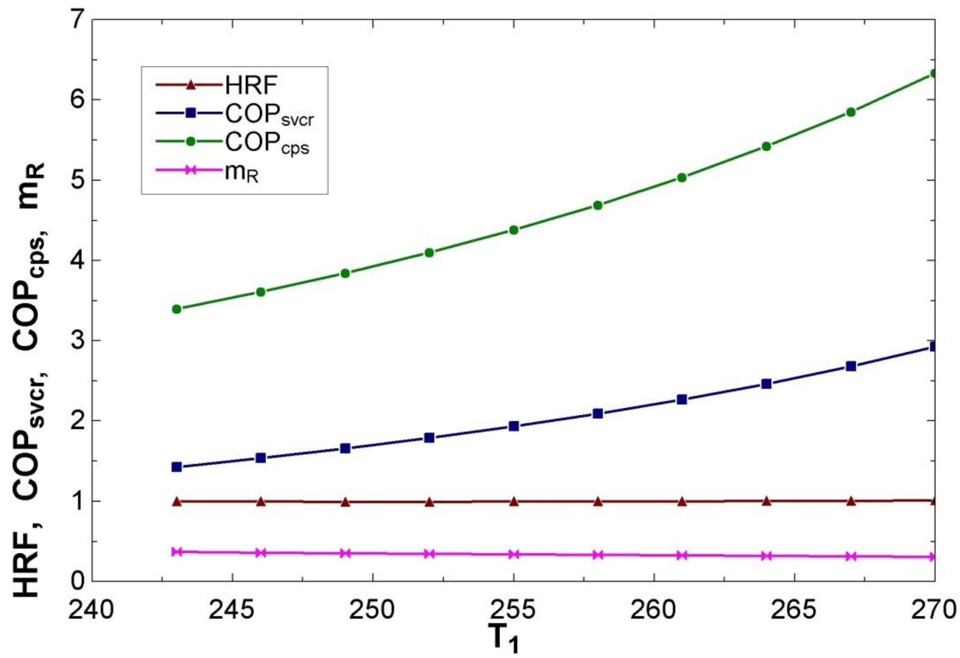

Fig. 40: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at $T_c = 321$ K, $\varepsilon=0.80$, $\eta_c=0.75$, $m_a=1.3$ kg/s, R1234ze)

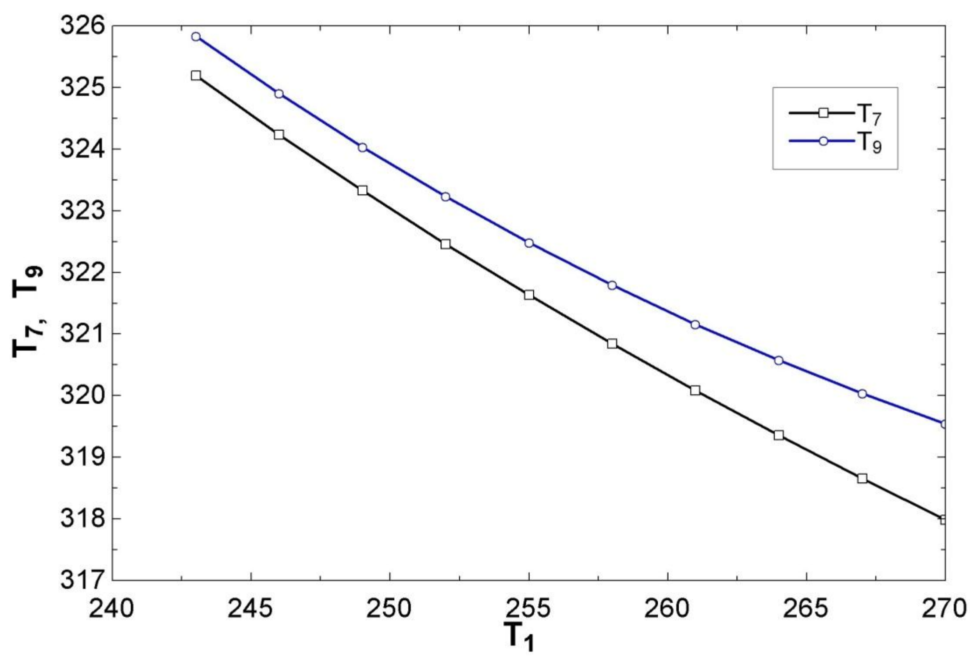

Fig. 41: Variation in T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at $T_c = 323$ K, $\varepsilon=0.80$, $\eta_c=0.75$, $m_a=1.3$ kg/s, R1234ze)

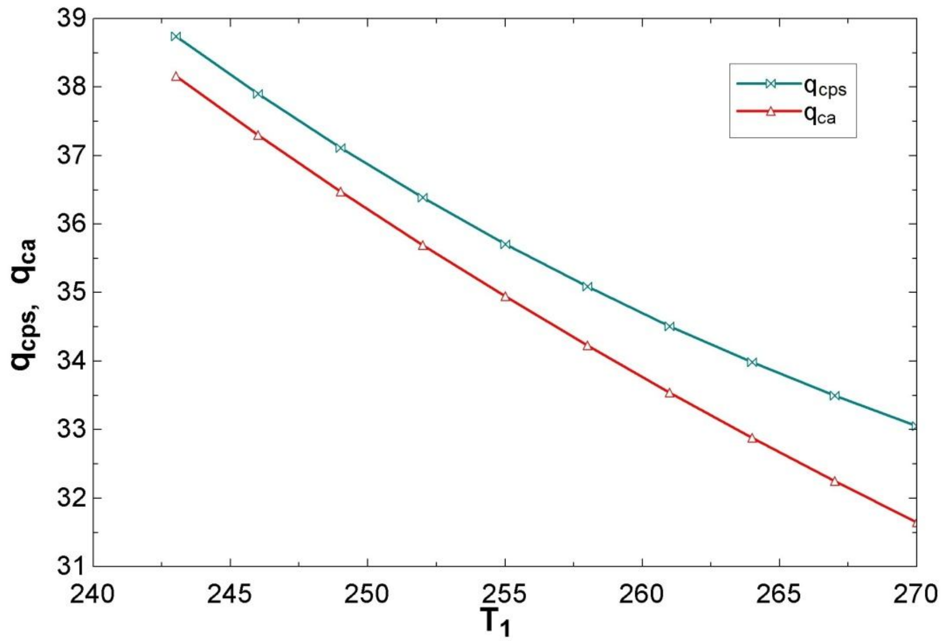

Fig. 42: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234ze)

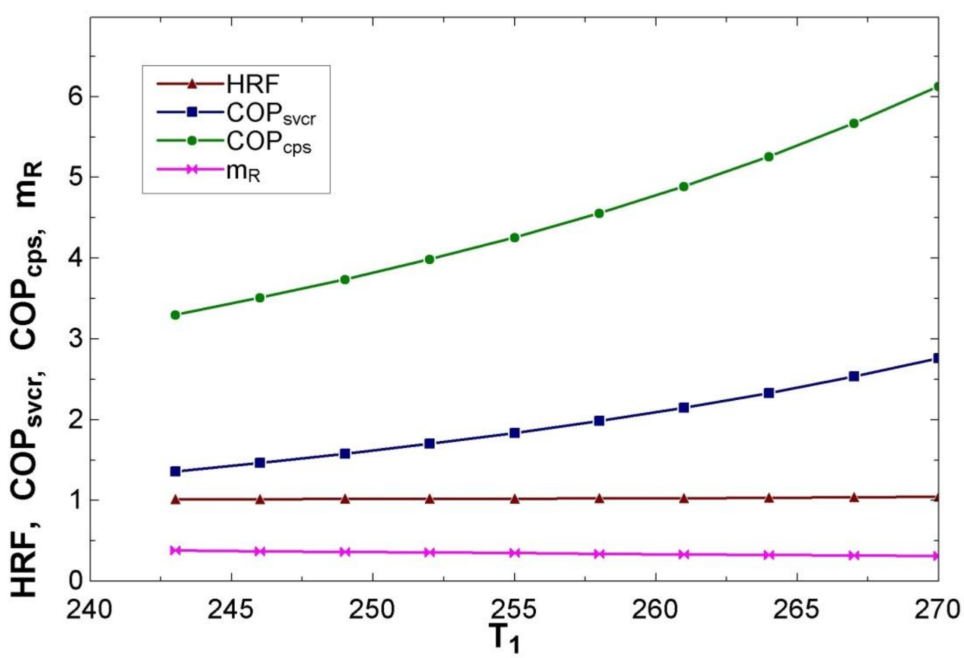

Fig. 43: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234ze)

6.6 Results for Refrigerant R-1234yf

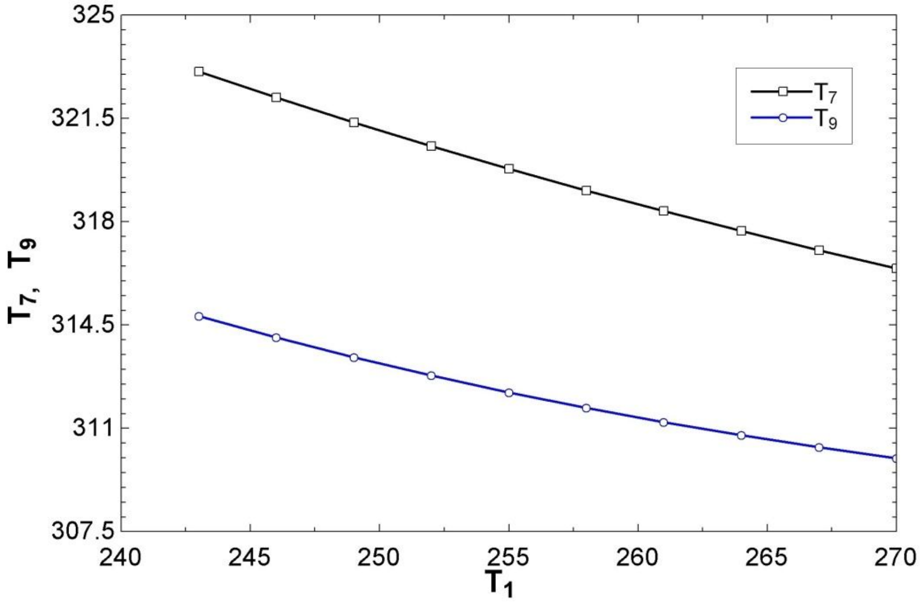

Fig. 44: Variation T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 313 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

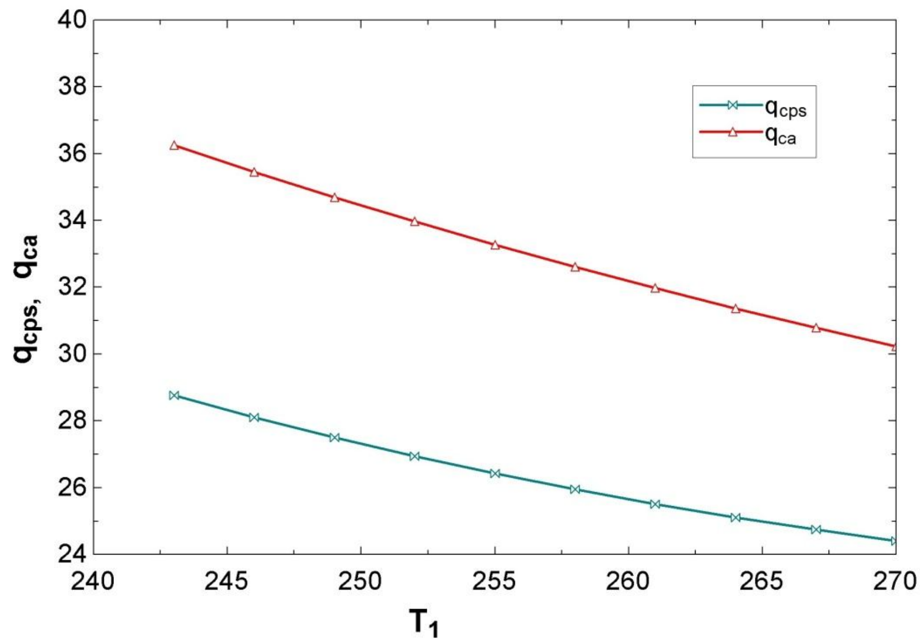

Fig. 45: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 313 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234yf)

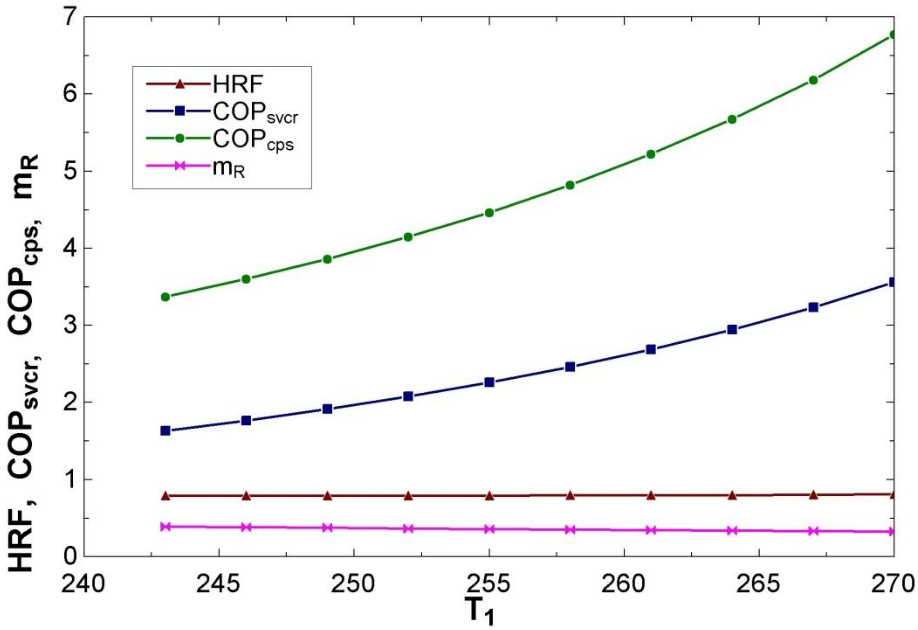

Fig. 46: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 313 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

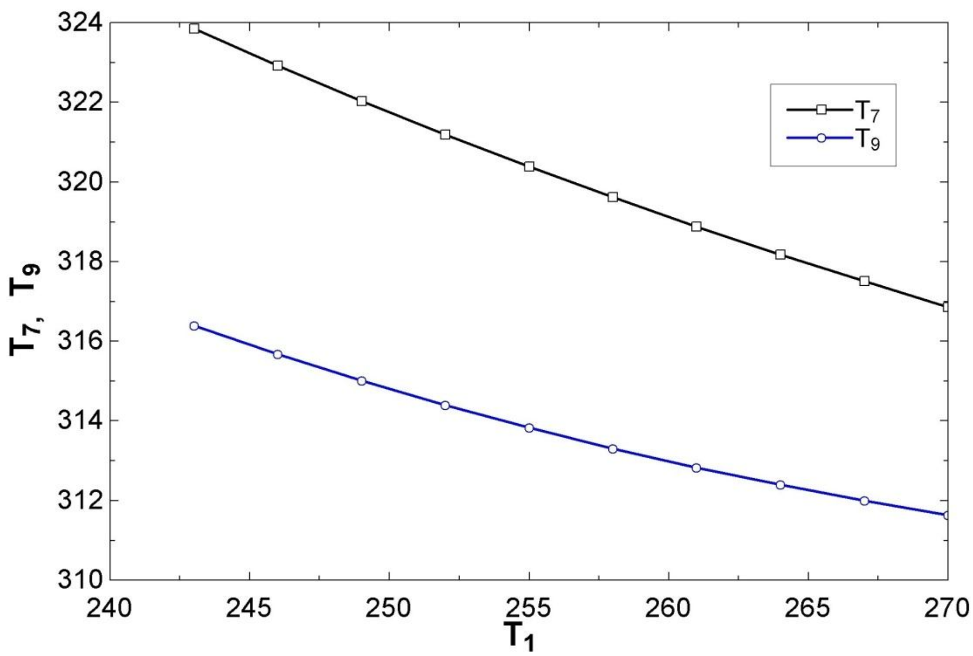

Fig. 47: Variation T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 315 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

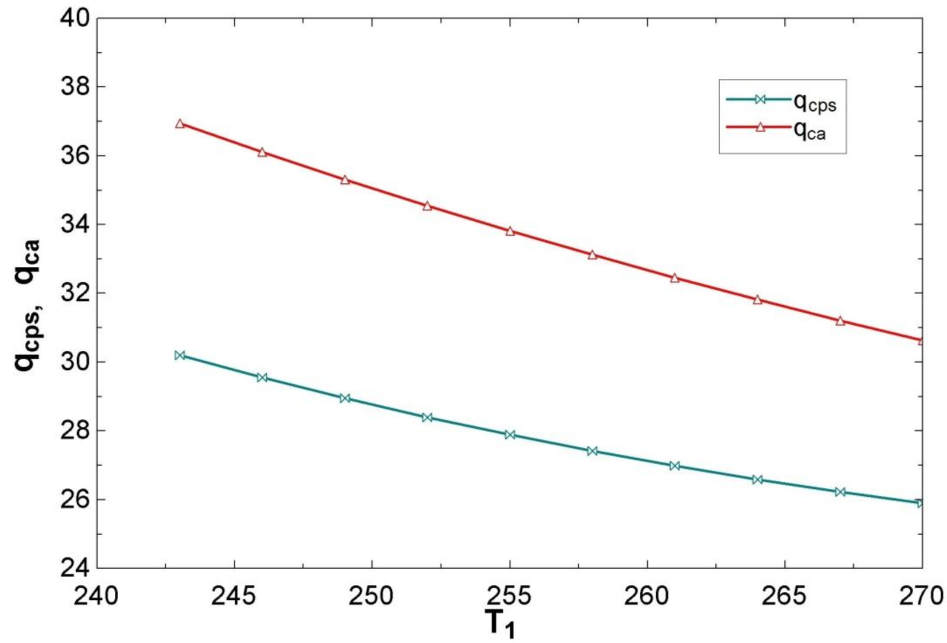

Fig. 48: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at $T_c = 315$ K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234yf)

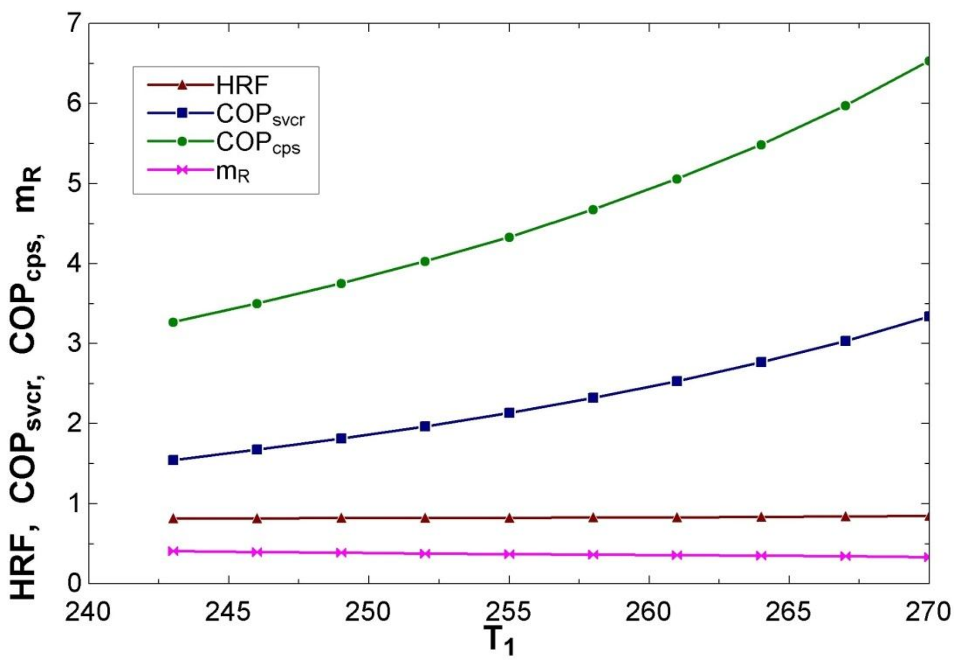

Fig. 49: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at $T_c = 315$ K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

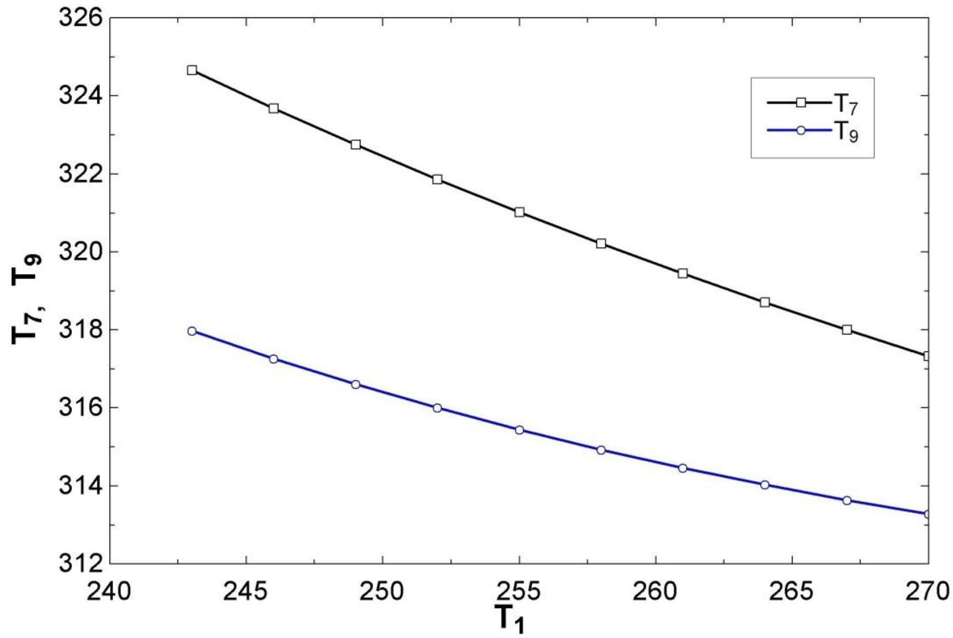

Fig. 50: Variation T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 317 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

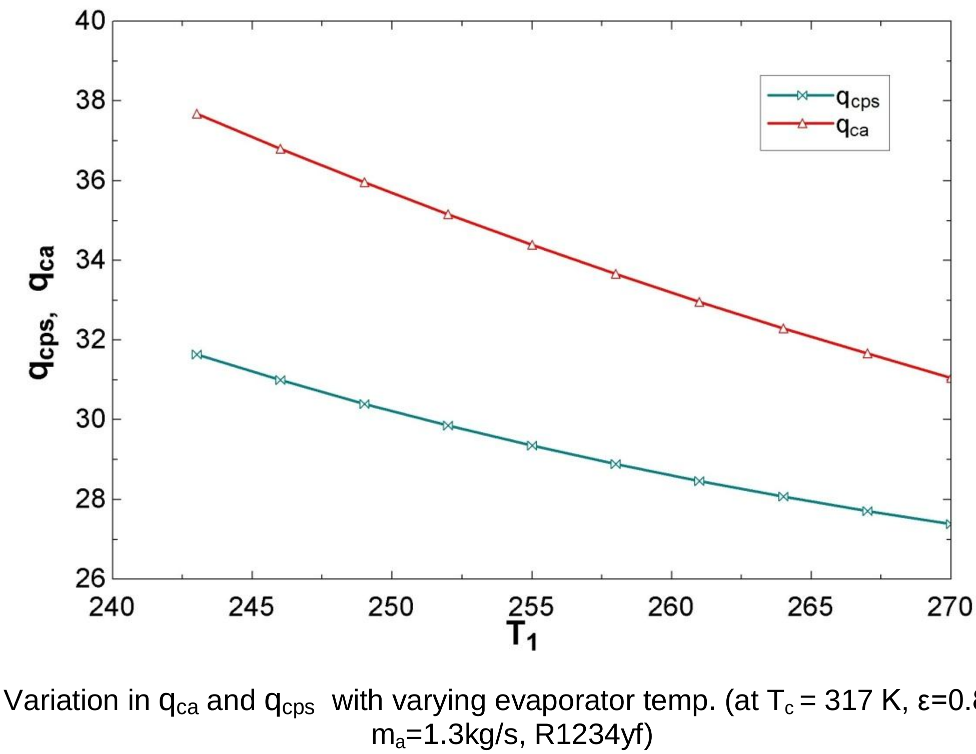

Fig. 51: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 317 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234yf)

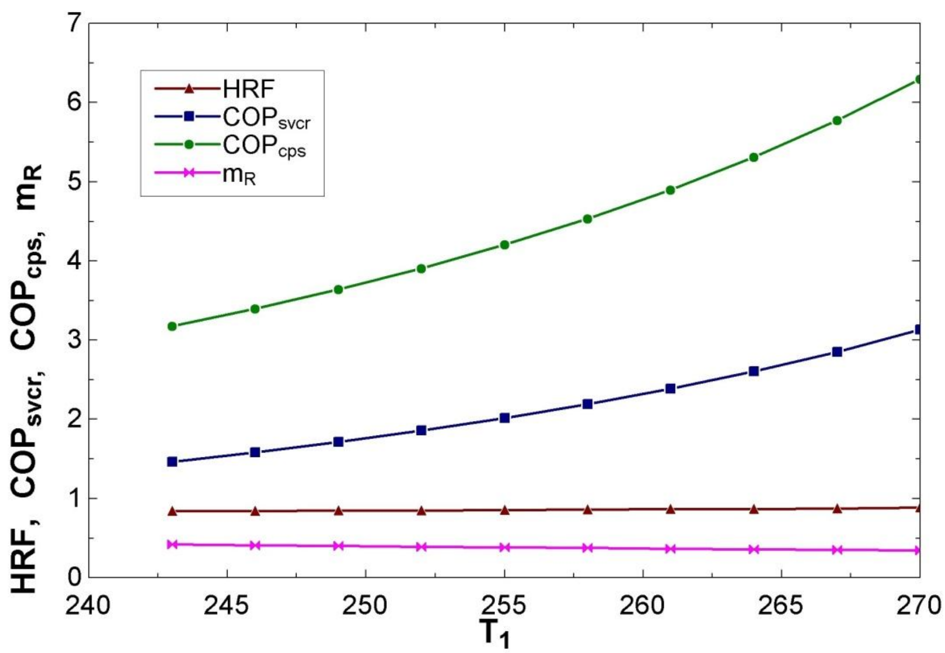

Fig. 52: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 317 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

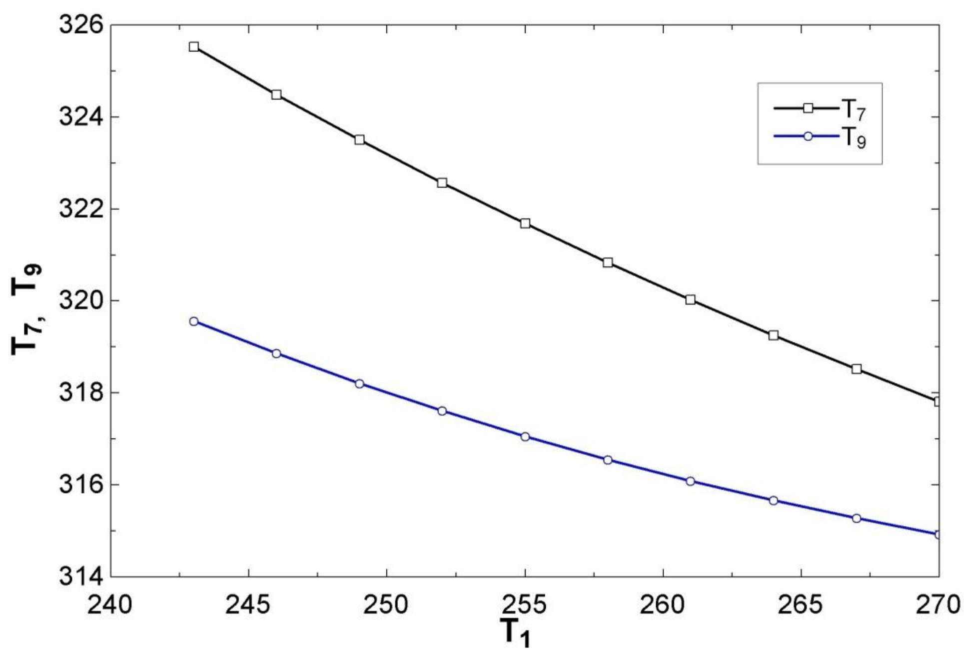

Fig. 53: Variation T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 319 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

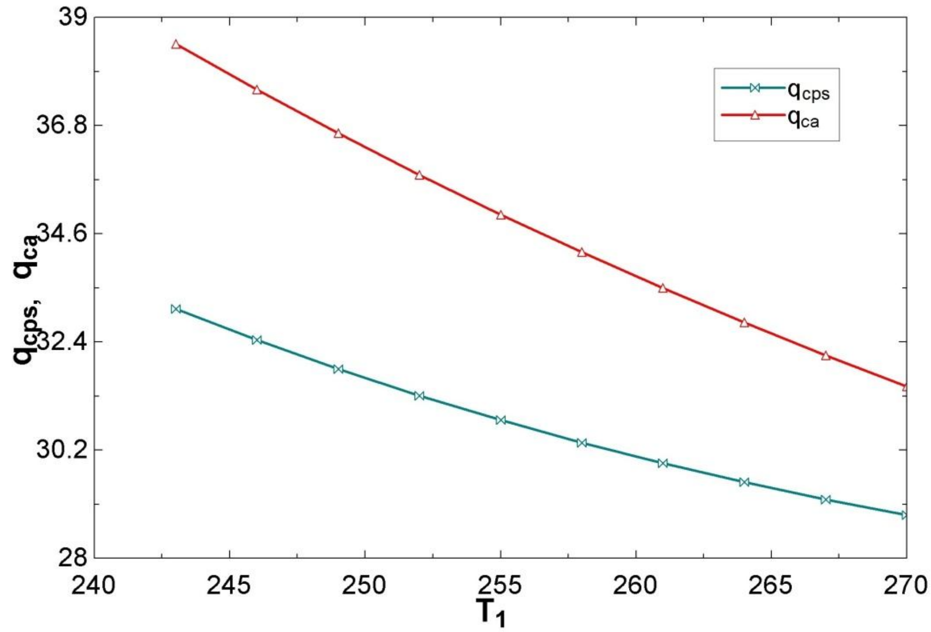

Fig. 54: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 319 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234yf)

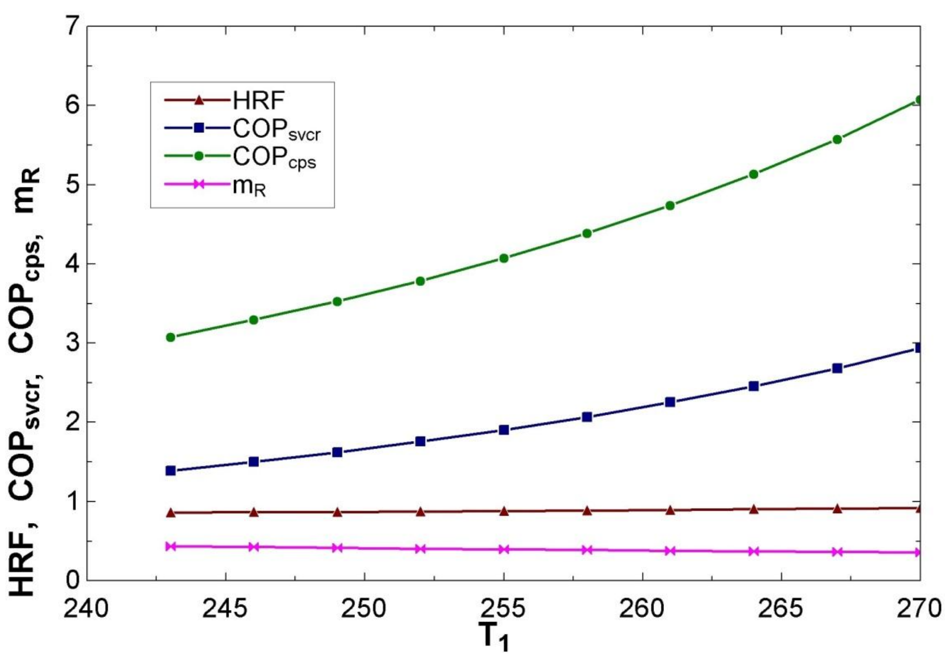

Fig. 55: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 319 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

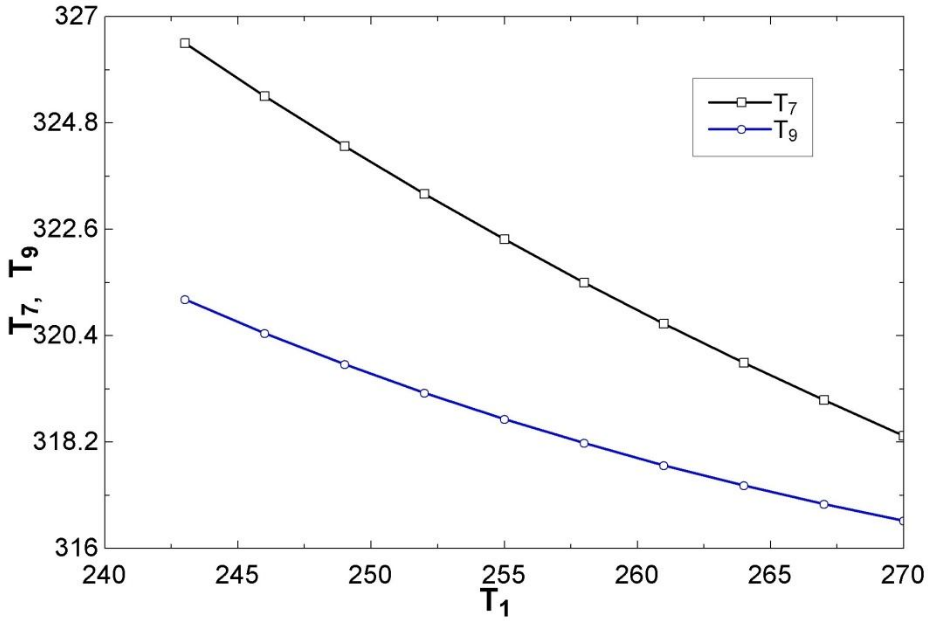

Fig. 56: Variation T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 321 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

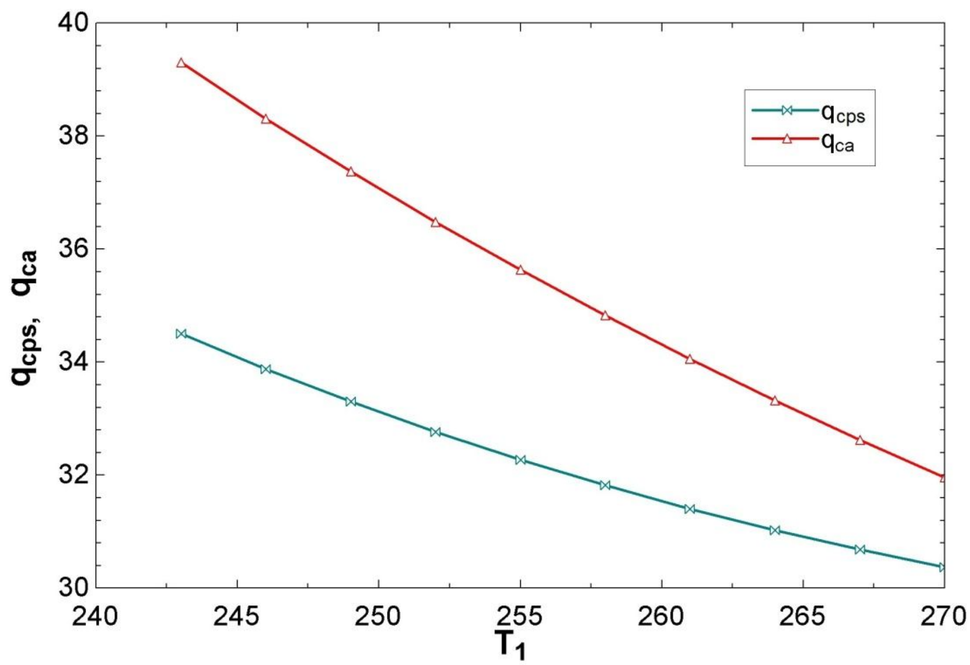

Fig. 57: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 321 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234yf)

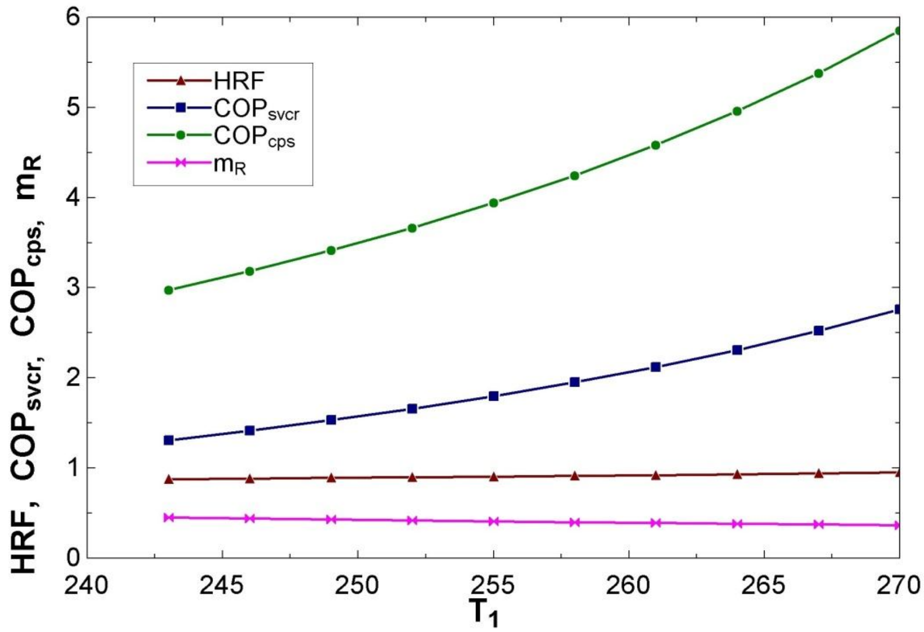

Fig. 58: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 321 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

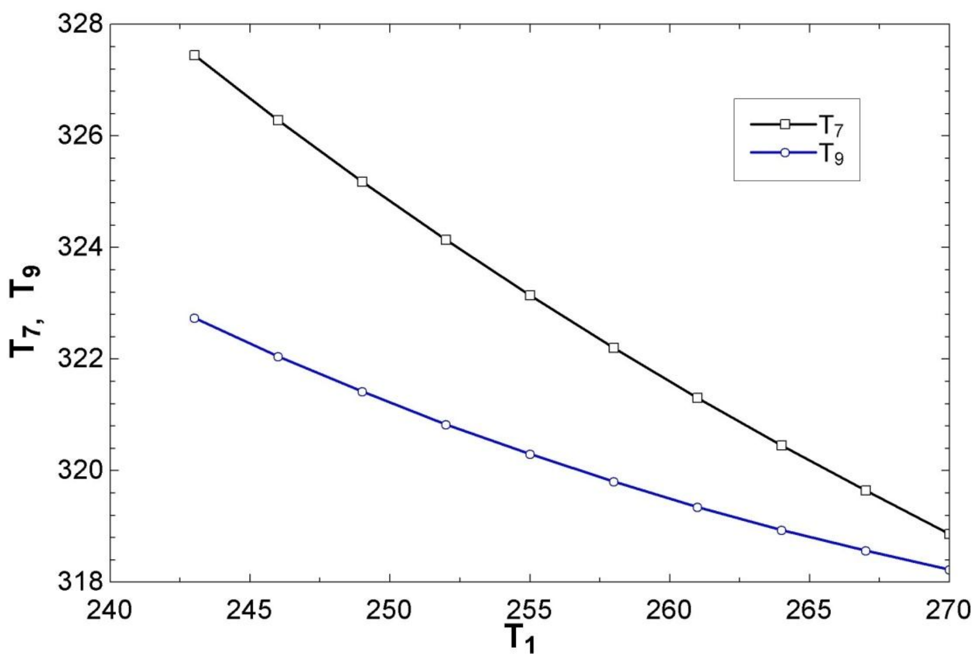

Fig. 59: Variation T_7 and T_9 (outlet temperatures) with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

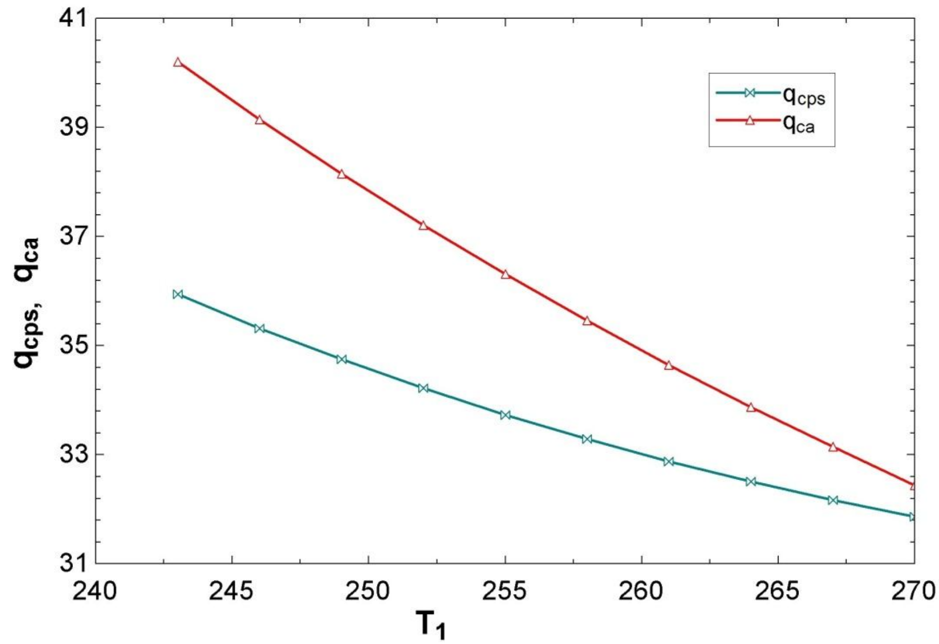

Fig. 60: Variation in q_{ca} and q_{cps} with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3kg/s, R1234yf)

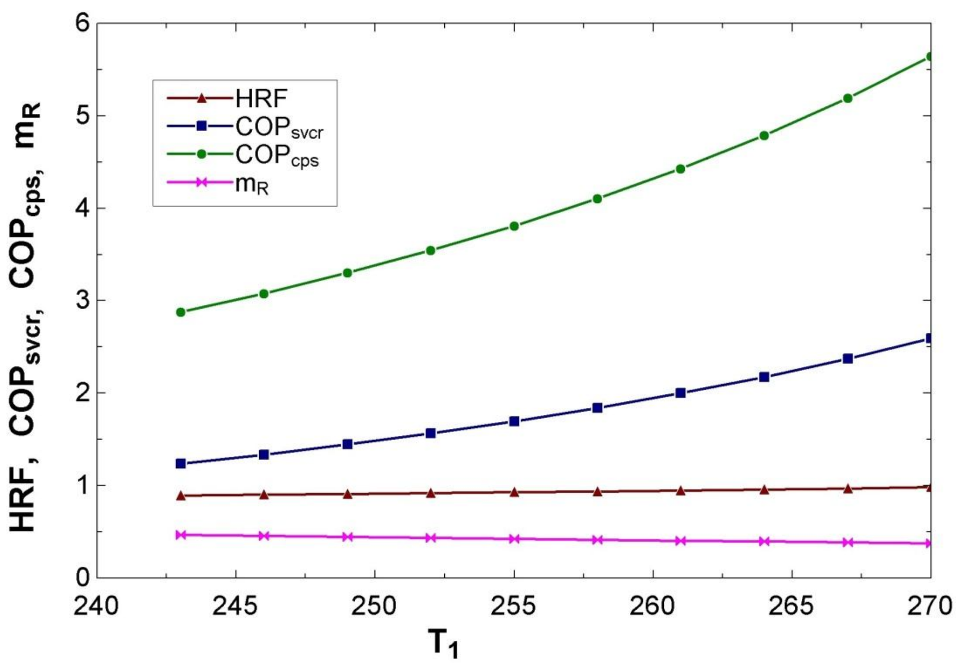

Fig. 61: Variation in COP_{svcr}, COP_{cps}, HRF and m_R with varying evaporator temp. (at T_c = 323 K, ε=0.80, η_c=0.75, m_a=1.3 kg/s, R1234yf)

CHAPTER 7

CONCLUSION

Present study is related to modified VCR system and focused on waste heat recovery from superheat horn of vapour compression system. A detailed thermodynamic analysis of the modified VCR system has been performed. The analysis has been done according to the applications of refrigeration. First of all results carried out by varying evaporator temperature of modified VCR system for most appropriate refrigerants R-134a, R-1234yf and R-1234ze.

At industrial level refrigeration and air-conditioning systems, considerable amount of low-grade heat is available as waste energy. The available waste heat can be utilized in drying or food processing sector by simply introducing a Canopus heat exchanger with existing system i.e. simple VCR system. The novelty of this Canopus HE is that it does not affect the COP of system in spite of increasing overall COP of the system. Hence, heat recovery through Canopus heat exchanger is feasible and can be maximized by selecting optimum water flow rate, inlet water temperature, suitable operating conditions, and working fluid.

On comparison of results for all three refrigerants it is concluded that for the same operating conditions, the R-134a yield better performance as compared to R-1234yf and R-1234ze. The Canopus heat exchanger options for heat recovery for low evaporator temperature R-134a giving higher heat removal factor whereas R-1234yf and R-1234ze giving higher heat removal factor at higher temperature.

CHAPTER 8

RECOMMENDATIONS FOR FUTURE WORK

1. Exergy analysis should be carried out for this modified vapour compression refrigeration system.

2. Thermo-economic optimization can be performed for optimal design of the system.

3. The analysis could be done using other better refrigerants of further low GWP and thermodynamic properties than R-134a, R-1234ze and R-1234yf.

4. The cooling fluid with higher specific heat gives better result i.e. the amount of heat removed in Canopus and condenser will be large.

CHAPTER 9

REFERENCES

1. Z. Sun, S. Li, “Analysis of energy reclamation of air-conditioning system”, Energy Conservation Technology, Vol-17(1994) 26-28.
2. G. Rong, “Mode of reusing Chiller’s heat energy in summer building”, Heat Energy Ventilated Air Conditioner 28(2) (1998) 27-29.
3. X. Wu, B. Xia, L. Lu, Y. Zhang, “Development and application of heat recovery unit”, Refrigeration and Air-Conditioning 1(6) (2001) 29-32.
4. X. Yang, “Analysis of thermal usage from condenser heat in middle and high standard hotels in summer”, Journal of Sichuan University of Science and Technology 21(2) (2002) 57-59.
5. S.C. Kaushik, M. Singh., Feasibility and Design studies for heat recovery from a refrigeration system with a Canopus heat exchanger, Heat Recovery Systems & CHP, Vol-15 (1995) 665-673.
6. C. Aprea, C. Renno, “Experimental comparison of R22 with R417A performance in a vapour compression refrigeration plant subjected to a cold store.” Energy Conversion and Management, Vol-45 (2004) 1807–1819.
7. V Siva Reddy, NL Panwar, SC Kaushik. “Exergetic analysis of a vapour compression refrigeration system with R134a, R143a, R152a, R404A, R407C, R410A, R502 and R507A”. Clean Technol Environ Policy, 2011. doi:10.1007/s10098-011-0374-0.
8. M. M. Rahman, H. Y. Rahman.”Performance of Newly Developed Integrated Space Conditioning and Domestic Water Heating Device” Journal of Energy & Environment, Vol-3 (2011), 23-27.
9. P. Sathiamurthi, PSS. Srinivasan, “Design and Development of Waste Heat Recovery Systemfor air Conditioning Unit” European Journal of Scientific Research, Vol-54 (2011), 102-110
10. T.B. Herbs, E. A. Dalvi and J. A. Parise. “Heat recovery from refrigeration plants: meeting load & temperature requirements”, Int. J. Refrig. Vol-13, (1990) 264-269.

11. H.I. Abu-Mulaweh, “Design and performance of a thermosiphon heat recovery system” , Applied Thermal Engineering, Vol-26, (2006) 471–477.
12. Eric Granryd, Refrigerant Cycle Data, “Thermophysical properties of refrigerants for applications in vapor compression systems”, Department of Energy Technology, KTH, Stockholm. 2005.
13. J. Roy Dossat, Roy, “Principles of Refrigeration”, New Jersey, Prentice Hall, Inc., (1991), ISBN: 0-13-717687-2.
14. R. Akasaka, 2010 “An application of the extended corresponding states model to thermodynamic property calculations for trans-1,3,3,3-tetrafluoropropene (HFO-1234ze(E))”, International Journal of Refrigeration, Vol-33, (2010) 907-914.
15. R. Akasaka, K. Tanaka, Y Higashi, “Thermodynamic property modeling for 2,3,3,3-tetraflouropropene(HFO-1234yf)”, International Journal of Refrigeration, Vol-33, (2010) 52-60.
16. CJ Andereen (editor) Recovering energy from refrigeration, energy technology 1998: 12–13.
17. Klein, S.A., Alvarado, F., 2012. Engineering Equation Solver, Version 9.223. F Chart Software, Middleton, WI.
18. M.A. Hammad, M.A. Alsaad, “The use of hydrocarbon mixtures as refrigerants in domestic refrigerators”, Applied Thermal Engineering, Vol-19 (1999) 1181–1189.
19. D. Jung, C.B. Kim, B.H. Lim, H.W. Lee, Testing of a hydrocarbon mixture in domestic refrigerators, ASHRAE Transactions, Vol-3 (1996) 1077–1084.
20. K.J. Park, T. Seo, D. Jung, “Performance of alternative refrigerants for residential air-conditioning applications”, Applied Energy, Vol-84 (2007) 985–991.
21. Air-Conditioning and Refrigeration Technology Institute (ARTI), Refrigerant Database, Calm, JM, July 2001, Arlington, USA.
22. K Fujima, Y Kato & T Nitawaki, Zero heat release nuclear cogeneration system using carbon dioxide as working fluid. Research Laboratory for Nuclear Reactors, Tokyo Institute of Technology 2-12-1, (2000).