ANALYSIS OF NEW REGENERATIVE VAPOUR COMPRESSION REFRIGERATION SYSTEM USING EJECTOR

THESIS
SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR THE AWARD OF THE DEGREE
OF
MASTER OF TECHNOLOGY
IN
THERMAL ENGINEERING

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2K12/THR/23

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CERTIFICATE

This is to certify that dissertation titled **"ANALYSIS OF NEW REGENERATIVE VAPOUR CUMPRESSION REFRIGERATION SYSTEM USING EJECTOR"** submitted by **VIPIN KUMAR (2K12/THE/23)** in partial requirements for the award Degree of Master Of Technology (Thermal Engineering) at the Delhi Technological University, Delhi is an authentic record of the student own work carried out by me under our guidance and supervision. To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any Degree or Diploma.

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ACKNOWLEDGEMENT

The writing of this dissertation has been one of the most significant academic challenges I have ever had to face; without GOD's blessing and support, patience and guidance of the following people, this study would not have been completed. It is to them I owe my deepest gratitude.

- **Dr. R.S. Mishra, Professor,** Mechanical Engineering Department, DELHI TECHNOLOGICAL UNIVERSITY (Formerly Delhi College of Engineering) for their initiative in this field of research, valuable guidance, encouragement and affection for the successful completion of this work. Their sincere sympathies and kind attitude always encouraged me to carry out the present work firmly.
- **Mr. Naushad Ansari, Assistant Professor,** Mechanical Engineering Department, DELHI TECHNOLOGICAL UNIVERSITY (Formerly Delhi College of Engineering) for their invaluable guidance, patient review and their continuous inspiration .
- **Dr. Naveen Kumar,** HOD, Mechanical Engineering Department , DELHI TECHNOLOGICAL UNIVERSITY (Formerly Delhi College of Engineering), New Delhi, for providing me with the best facilities in the department and timely suggestions.

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M.TECH. (THERMAL)

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Nomenclature

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A	Area (m^2)
COP	Coefficient of performance
d	Diameter (m)
h	Specific enthalpy (kJ/kg)
L	length (m)
M_a	Mach number
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (bar)
$\dot{Q}$	Heat transfer rate (kW)
T	Temperature (°C)
V	Velocity (m/s)
η	Efficiency
μ	Co-efficient of discharge
ρ	Density (kg/m^3)
k	Adiabatic co-efficient
w	Ejection Ratio
U	Velocity

V	Mean sh. Volume
e	Energy consumed
H	pressure difference
a	speed of sound

Subscripts

v	vapour
f	fluid
mb	motive stream at constant area mixing
m	mixing stream
b	basic cycle
d	diffuser
p	pump
c	compressure
mix	mixing chamber

ABSTRACT

In present study a mathematical model is developed in EES for a new regenerative vapour compression refrigerating system. The model is then used to determine the work of compressor, work of pump, refrigerating capacity of the system, thermal performance of the system and COP of the new regenerative vapour compression system using R134a and R152a as a refrigerant. And then this result is compare with basic vapour compression cycle.The basic purpose of using this cycle is to use the regenerative use of potential energy of ejector two phase expansion flow. which is going to lost in expansion valve. The new features of the process which are essential are:

1.The compression is done in two step and throttling is also done in two step. First compress in compressor and secondly in ejector.

2.Use a jet device as a second step compression and compressor a initial step, where Compression and throttling are combined.

3. The ratio of a working fluid at the first and second step of compression is controlled. In this proposed system, the compressor compresses the vapor only to 50-60% of the final pressure, while the additional compression is achieved by a jet device using the working fluid's internal potential energy of flow. Therefore, the amount of mechanical energy is significantly reduced required by a compressor , resulting in the increase of efficiency (either COP or EER). The novelty of the cycle is in the equipment and in the way the multi-staging is accomplished.

In this proposed system, approximately 50-60% of the final pressure is compress by the help of compressor. And the remaining compression is provided in a jet device called ejector using internal potential energy of the working fluid flow. Therefore, the amount of mechanical energy is significantly reduced Required by a compressor and a coefficient of performance (COP) can be theoretically increased by a ratio reaching 1.4-1.5.

CHAPTER 1

INTRODUCTION

Air-conditioning and refrigeration systems are extensively used in air-conditioning and cooling applications. Improved system performance will reduce energy consumption as well as reduce green house gases emissions. Decreases of the condensation temperature and increases of the evaporation temperature or the liquid condensate subcooling will improve the COP of the refrigeration system. These improvements are limited in practice since these temperatures depend on the environmental temperature and operating conditions. According to the second law of thermodynamics (Clausius statement), heat cannot be transferred from low temperature body to high temperature body without an aid of external agency. Thus, heat rejected at the condenser is equal to the sum of heat absorbed in the evaporator and compressor work equivalent heat. The refrigerant at the compressor outlet is usually quite warm (usually 80-110°C for R134a and R22 air-conditioning systems); thus, a large amount of energy must be rejected to the environment in the condenser. This waste heat energy can be utilized to increase the refrigeration system performance. An ejector cooling system driven by low-grade heat energy can effectively use the waste heat to improve the system COP. An ejector based cooling system offers several advantages, such as no moving parts in the ejector, efficient utilization of the waste heat and low cost This thesis describes a refrigeration system that combines a basic vapor compression refrigeration cycle with an ejector cooling cycle. The ejector cooling cycle is driven by the waste heat from the condenser of the vapor compression refrigeration cycle. The ejector secondary flow is compressed first by means of a booster to ensure that the ejector works at the right conditions. The obtained cooling capacity from the ejector cycle is directly fed into the evaporator of the vapor compression refrigeration cycle. The entire refrigeration system performance is simulated to analyze the effects of the condensation temperature, the evaporation temperature, the pressure ratio, the pressure lift ratio and the ejector area ratio on the system performance. The results are compared with a basic refrigeration system.

Simple vapour compression cycle

The vapor-compression cycle is used in most household refrigerators as well as in many large commercial and industrial refrigeration systems. Figure 1 provides a schematic diagram of the components of a typical vapor-compression refrigeration system.

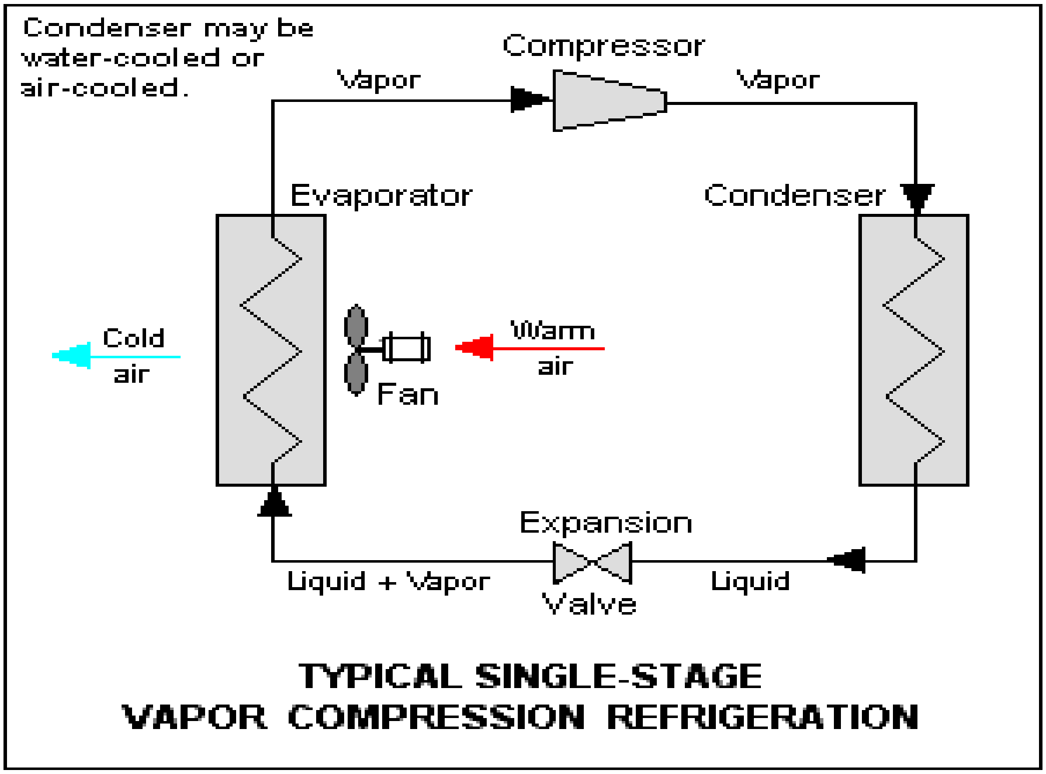


Fig. 1 Simple vapor compression cycle

In simple vapour compression cycle basically there are four devices are udsed.

Compressor: In compressor low temperature and low pressure refrigerant from evaporator compresses to high pressure

Condenser: Heat is removed at constant at constant pressure and phase change takes place

Expansion Valve: Expansion of refrigerant at constant enthalpy

Evaporator: Evaporator is the space from which heat is to be removed .

$$\text{COP}=\frac{Refrigerati\ on\ effect}{Work\ input\ \ to\ compressor} \qquad 1a$$

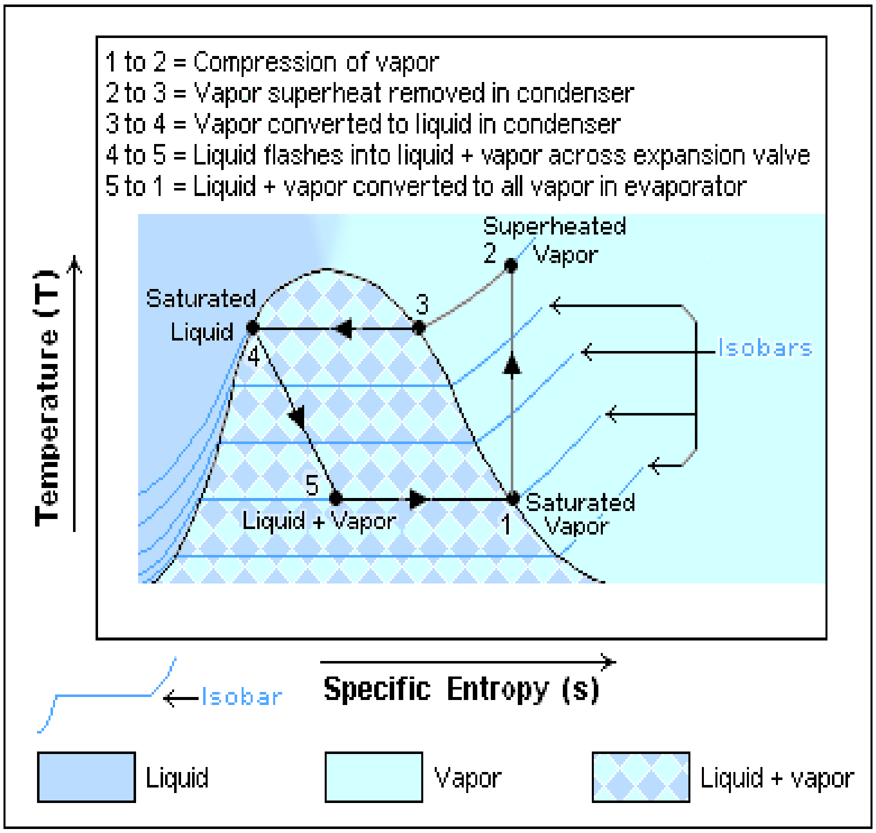


Fig. 2. T-S diagram of simple V-C cycle

From the above discussion it is clear that compressor is only device in simple vapour compression system which takes energy input. So by reducing this work input we can increase the COP of the system.

Use of Ejectors

Since its invention in the early twentieth century, the gas to gas, or vapor-to-vapor ejector, has found wide application in industries for the processes of evacuation, refrigeration, and solid powder transportation etc., or in modern jet planes for thrust augmentation. Air and steam are the common working fluids of an ejector. The study of refrigerant (CFCs, HCFCs and HFCs) ejectors for air-conditioning or refrigeration applications started in the mid-1950s for utilizing low-grade energy such as solar or waste heat energy as the heat source. The operation of a gas-to-gas or vapor-to-vapor ejector results mainly from the gas-dynamic effect and the momentum exchange of two gaseous streams (primary and secondary or entrained streams) inside the ejector. Two choking phenomena exist in the ejector performance [1]: one in the primary flow through the nozzle and the other in the entrained or suction flow. The entrained flow rate or the entrainment ratio (entrained-to-primary flow ratio v . *m_* s=*m_* p) of an ejector is affected by many factors. The physical phenomena involve supersonic flow, shock interactions, and turbulent mixing of two streams inside the ejector enclosure. It is so complicated that the design of an ejector to date still heavily relies on trials-and-errors methods although a number of gas-dynamic theories for ejector analysis were developed by several researchers

Ejector Working Principle

As outlined in Figure 1, a typical ejector consists of a motive nozzle, a suction chamber, a mixing section, and a diffuser. The working principle of the ejector is based on converting internal energy and pressure related flow work contained in the motive fluid stream into kinetic energy. The motive nozzle is typically of a converging-diverging design. This allows the high-speed jet exiting the nozzle to become supersonic

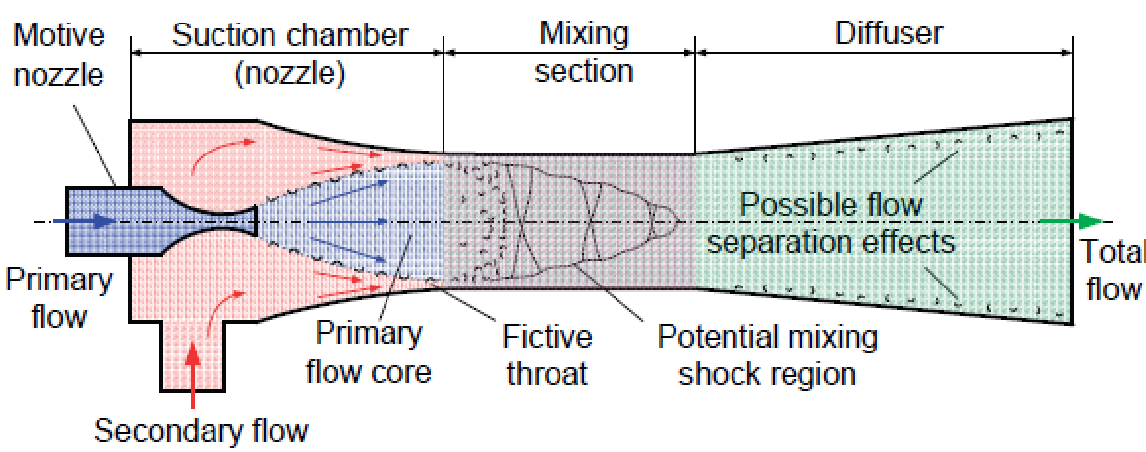
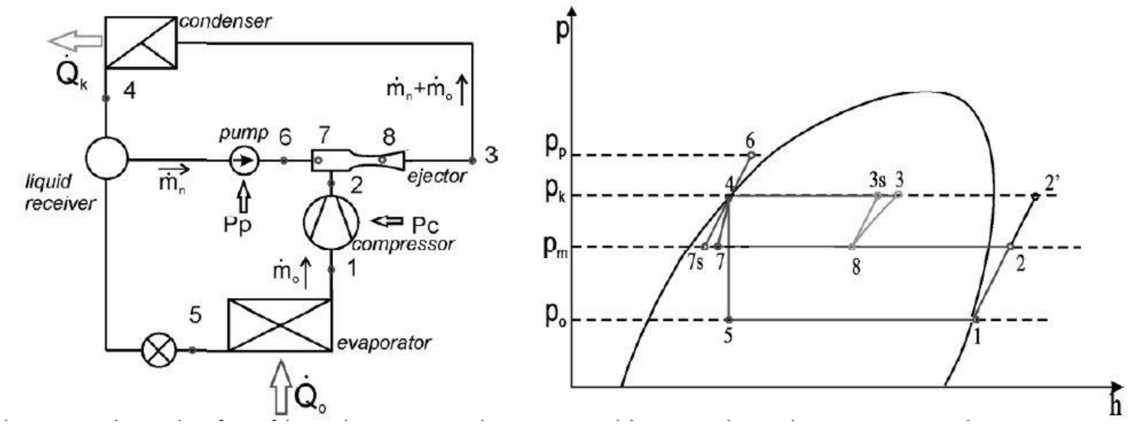


Fig-3: Ejector

Depending on the state of the primary fluid, the flow at the exit of the motive nozzle might be two-phase. Flashing of the primary flow inside the nozzle might be delayed due to thermodynamic and hydrodynamic non-equilibrium effects. The high-speed jet starts interacting with the secondary fluid inside the suction chamber. Momentum is transferred from the primary flow which results in an acceleration of the secondary flow. An additional suction nozzle can be used to pre-accelerate the relatively stagnant suction flow. This helps to reduce excessive shearing losses caused by large velocity differences between the two fluid streams. Depending on the operating conditions both the supersonic primary flow and the secondary flow might be choked inside the ejector. Due to static pressure differences it is possible for the primary flow core to fan out and to create a fictive throat in which the secondary flow reaches sonic condition before both streams thoroughly mix in the subsequent mixing section. The mixing section can be designed as a segment having a constant cross-sectional area but often has a tapered inlet section. Most simulation models either assume mixing at constant area associated with pressure changes or mixing at constant pressure as a result of changes in cross-sectional area of the mixing section. The mixing process is frequently accompanied by shock wave phenomena resulting in a considerable pressure rise. The total flow at the exit of the mixing section can still have high flow velocities. Thus, a diffuser is used to recover the remainder of the kinetic energy and to convert it into potential energy, thereby increasing the static pressure. Typically, the total flow exiting the diffuser has a pressure in between that of the primary and the secondary streams entering the ejector. Therefore, the ejector acts as a motive-flow driven fluid pump used to elevate the pressure of the entrained fluid. The two major characteristics which can be used to determine the performance of an ejector are the suction pressure ratio and the mass entrainment ratio. The suction pressure ratio is defined as the ratio of diffuser exit pressure to the pressure of the suction flow entering the ejector. The mass entrainment ratio is defined as the ratio of suction mass flow rate to motive mass flow rate. A well-designed ejector is able to provide large suction pressure ratios and large mass entrainment ratios at the same time.

New Regenerative cycle : In new generative cycle we compress the 50 -60 % part of desired pressure and the remaining pressure rise is achieved in ejector as a second step compression.



Working principle of new regenerative vapor compression refrigeration system is as follow:

1. Low temperature and low pressure refrigerant from evaporator goes to the compressor

2. In compressor refrigerant compresses to 50 to 60 % of condenser pressur.

3. Remaining pressure rise is achieved in jet device.

Refrigerant Used

A **refrigerant** is a substance or mixture, usually a fluid, used in a heat pump and refrigeration cycle. In most cycles it undergoes phase transitions from a liquid to a gas and back again. Many working fluids have been used for such purposes. Fluorocarbons, especially chlorofluorocarbons, became commonplace in the 20th century, but they are being phased out because of their ozon depletion effects. Other common refrigerants used in various applications are ammonia, sulfur dioxide, and non-halogenated hydrocarbons such as propane.

The ideal refrigerant would have favorable thermodynamic properties, be noncorrosive to mechanical components, and be safe, including free from toxicity and flammability. It would not cause ozone depletion or climate change. Since different fluids have the desired traits in different degree, choice is a matter of trade-off.

The desired thermodynamic properties are a boiling point somewhat below the target temperature, a high heat of vaporization, a moderate density in liquid form, a relatively high density in gaseous form, and a high critical temperature. Since boiling point and gas density are affected by pressure, refrigerants may be made more suitable for a particular application by choice of operating pressures. Refrigerants such as ammonia (R717), carbon dioxide (R744) and non-halogenated hydrocarbons preserve the ozone layer and have no (ammonia) or only a low (carbon dioxide, hydrocarbons) global warming potential. They are used in air-conditioning systems for buildings, in sport and leisure facilities, in the chemical/pharmaceutical industry, in the automotive industry and above all in the food industry (production, storage, retailing). In these settings their toxicity is less a concern than in home equipment.

Emissions from automobile air conditioning are a growing concern because of their impact on climate change. From 2011 on, the European Union will phase out refrigerants with a global warming potential (GWP) of more than 150 in automotive air conditioning (GWP = 100 year warming potential of one kilogram of a gas relative to one kilogram of CO_2).This will ban potent greenhouse gases such as the refrigerant HFC-134a—which has a GWP of 1410—to promote safe and energy-efficient refrigerants.

One of the most promising alternatives is CO_2 (R-744). Carbon dioxide is non-flammable, non-ozone depleting, has a global warming potential of 1. R-744 can be used as a working fluid in climate control systems for cars, residential air conditioning, hot water pumps, commercial refrigeration, and vending machines.R12 is compatible with mineral oil, while R134a is compatible with synthetic oil that contains esters.GM has announced that it will start using "hydrofluoroolefin", HFO-1234yf, in all of its brands by 2013 Dimethyl ether (DME) is also gaining popularity as a refrigerant, but like propane, it is also dangerously flammable.

Under Section 608 of the EPA Clean Air Act it is illegal in the United States to knowingly release HFC-134a refrigerants into the atmosphere.

When refrigerants are removed they should be recycled to clean out any contaminants and return them to a usable condition. Refrigerants should never be mixed together outside of facilities licensed to do so for the purpose of producing blends. Some refrigerants must be managed as hazardous waste even if recycled, and special precautions are required for their transport, depending on the legislation of the country's government.

Various refrigerant reclamation methods are in use to recover refrigerants for reuse.

Refrigerant by class and R number

Refrigerants may be divided into three classes according to their manner of absorption or extraction of heat from the substances to be refrigerated:

- Class 1: This class includes refrigerants that cool by phase change (typically boiling), using the refrigerant's latent heat.
- Class 2: These refrigerants cool by temperature change or 'sensible heat', the quantity of heat being the specific heat capacity x the temperature change. They are air, calcium chloride brine, sodium chloride brine, alcohol, and similar nonfreezing solutions. The purpose of Class 2 refrigerants is to receive a reduction of temperature from Class 1 refrigerants and convey this lower temperature to the area to be air-conditioned.
- Class 3: This group consists of solutions that contain absorbed vapors of liquefiable agents or refrigerating media. These solutions function by nature of their ability to carry liquefiable vapors, which produce a cooling effect by the absorption of their heat of solution. They can also be classified into many categories.

The R-# numbering system was developed by Dupont corporation (which owns the Freon trademark) and systematically identifies the molecular structure of refrigerants made with a single halogenated hydrocarbon. The meaning of the codes is as follows:

- Subtracting 90 from the concatenated numbers of Carbon, hydrogen and fluorine atoms, respectively gives the assigned R#.
- Remaining bonds not accounted for are occupied by chlorine atoms.
- *A suffix of a lower-case letter a, b, or c* indicates increasingly unsymmetrical isomers.
- As a special case, the R-400 series is made up of zeotropic blends (those where the boiling point of constituent compounds differs enough to lead to changes in relative concentration because of fraction distillation) and the R-500 series is made up of so-called azeotrops blends. The rightmost digit is assigned arbitrarily by ASHRAE, an industry organization.

For example, R-134a has 2 carbon atoms, 2 hydrogen atoms, and 4 fluorine atoms, an empirical formula of tetrafluoroethane. The "a" suffix indicates that the isomer is unbalanced by one atom, giving 1,1,1,2-Tetrafluoroethane. R-134 (without the "a" suffix) would have a molecular structure of 1,1,2,2-Tetrafluoroethane—a compound not especially effective as a refrigerant.

The same numbers are used with an R- prefix for generic refrigerants, with a "Propellant" prefix (e.g., "Propellant 12") for the same chemical used as a propellant for an aerosol spray, and with trade names for the compounds, such as "**Freon** 12". Recently, a practice of using HFC- for hydrofluorocarbons, CFC- for chlorofluorocarbons, and HCFC- for hydro chlorofluorocarbons has arisen, because of the regulatory differences among these groups

<u>Notable blends</u>

Below are some notable blended HFC mixtures. There exist many more (see list of refrigerants). All R-400 (R-4xx) and R-500 (R-5xx) hydroflurocarbons are blends, as noted above.

- **R-401A** is a HCFC zeotropic blend of R-22, R-152a, and R-124. It is designed as a replacement for R-12.
- **R-404A** is a HFC "nearly azeotropic" blend of 52 wt.% R-143a, 44 wt.% R-125, and 4 wt.% R-134a. It is designed as a replacement of R-22 and R-502 CFC. Its boiling point at normal pressure is -46.5 °C, its liquid density is 0.485 g/cm^3.
- **R-406A** is a zeotropic blend of 55 wt.% R-22, 4 wt.% R-600a, and 41 wt.% R-142b.
- **R-407A** is a HFC zeotropic blend of 20 wt.% R-32, 40 wt.% R-125, and 40 wt.% R-134a.
- **R-407C** is a zeotropic hydrofluorocarbon blend of R-32, R-125, and R-134a. The R-32 serves to provide the heat capacity, R-125 decreases flammability, R-134a reduces pressure.
- **R-408A** is azeotropic HCFC blend of R-22, R-125, and R-143a. It is a substitute for R-502. Its boiling point is -44.4 °C.
- **R-409A** is azeotropic HCFC blend of R-22, R-124, and R-142b. Its boiling point is -35.3 °C. Its critical temperature is 109.4 °C.

- **R-410A** is a near-azeotropic blend of R-32 and R-125. The US Environmental Protection Agency recognizes it as an acceptable substitute for R-22 in household and light commercial air conditioning systems. It appears to have gained widespread market acceptance under several trade names.
- **R-438A** another HFC blended replacement for R-22, with five components: R-32, R-125/R-134a, R-600, and R-601a, blended in respective ratios 8.5+.5,-1.5%; 45±1.5%; 44.2±1.5%; 1.7+.1,-.2%; 0.6+.1,-.2%. The mean ‘’mo’’lecular weight of this mix is 99, resulting in the tradename ISCEON MO99 from manufacturer DuPont (a line of blended HFC products developed initially by Rhodia, and sold to DuPont).
- **R-500** is an azeotropic blend of 73.8 wt.% R-12 and 26.2 wt.% of R-152a.
- **R-502** is an azeotropic blend of R-22 and R-115.

In my work I am working on R134a and R152a as a refrigerant, which have very low GDP and is are very environment friendly.

CHAPTER 2

LITRATURE REVIEW

Literature Review

Many theories and experiments have been done to reduce the power consumption and increase the COP of the vapor compression refrigeration system. There are so many methods to increase the COP of the vapor compression system introducing the ejector is one of them. There are some theories and experiments which I reviewed for my study as follws:

Vapor compression refrigeration systems typically utilize expansion valves or other throttling devices to lower the pressure of liquid refrigerant and deliver it to the evaporator. In a typical refrigeration cycle, the expansion valve lowers the refrigerant pressure by 5-7 times. The reason for lowering the pressure is to allow the refrigerant to evaporate at certain desired low temperature. However, the process of throttling is isenthalpic, which means that the kinetic energy produced during the pressure reduction is dissipated and eventually wasted. Therefore, it is desirable to recover this kinetic energy to increase the efficiency of the entire refrigeration cycle. One method to accomplish this was developed by a group of scientists from City University (London) [7] The literature search has revealed that the principal method to accomplish this task in the past was using the ejector instead of the throttling valve. The velocity increase in the throat of the ejector device is used to entrain the refrigerant exiting the evaporator by momentum exchange. The following diffuser section of the ejector re-compresses the refrigerant by slowing down the mixed stream. Through the action of an ejector, the compressor suction pressure is therefore higher than it would be in a standard cycle, resulting in less compression work thus improvement in cycle efficiency.

The first theoretical principles of the ejector were elaborated by Parsons in 1900 while the first prototype was built by Leblanc (1910). Further improvements were introduced by Gay in 1931 [9]. Ejectors were first applied for refrigeration cycles by Heller in 1955 for absorption systems and by Badylkes in 1958 for vapor compression systems [8]. In the USA, the first application was reported by Kemper in 1966, but only patent is in existence while no experimental or theoretical background have been published. Following up on this early work, Kornhauser [10] has conducted a theoretical analysis and showed that the ideal ejector cycle resulted in 21% efficiency as compared with standard vapor compression cycle. The prototype unit was built, however its performance was much less than the ideal and reached at maximum only 5% using working fluids CFCs/ HCFCs/ HFCs. This was attributed to shortcomings in the design of the ejector, specifically too simplified two-phase flow model assumed in the design Latest work on ejectors had concentrated on using them in transcritical CO_2 systems where high pressures allow for better recovery of the kinetic energy [11],[12], [13]. Detailed investigations were presented in [12], in particular a constant pressure mixing model for the superheated vapor ejector was established and the thermodynamic analysis of the ejector expansion for transcritical CO_2 was performed. It was found that the COP (Coefficient of Performance) of the transcritical CO_2 cycle with an ejector can be improved by as much as 16% over the basic transcritical CO_2 cycle for typical A/C operation conditions. However, only theoretical model is presented in the subject reference with no supporting practical experiments.

Yinhai zhu and peixue jiang, (2012) [1] developed a model which combine the vapor compression system with ejector cooling cycle. The waste heat of condenser invapor compression system is utilized to drive the ejector cooling cycle. In this system evaporator gets the additional cooling from ejector cooling cycle and this shows that there is an increase in refrigeration effect of combined cycle and finaly increase in COP. This system shows the result for high compressor discharge temperature COP is improved by 9.1%.

Jialin yu and Huazhao, (2007) [2] investigated a naval auto cascade refrigeration cycle with an ejector. The ejector is used to recover the some available work to increase the compression suction pressure this enables the compressor to operate at lower pressure ratio, which in turn improves the cycle efficiency. In this study they use the refrigerant as a mixture of R23/R134a. In this study they operated at condenser pressure of 40^0C, the evaporator inlet temperature -40.3^0C, and mass fraction of r23 is 0.15 , the pressure ratio of the ejector reaches to 1.35, the pressure ratio of compressor is reduced by 25.8% and COP is increased by 19.1% over the conventional auto cascade refrigeration cycle.

A khalil and E. Elgendy, (2011), [3] developed a mathematical model to design R134a ejector and to predict the performance characteristics of vapor jet refrigeration system over a wide range of investigated parameters.

Arbel and Sokolov (2004)[4] presented a theoretical study of a solar driven combined VCR-VER using R142b as a working fluid. The study compared the performance of the system with previous studies developed by Sokolov, where R113 was used. They showed out not only technical but also ecological improvements by using R142b. At this time use of R113 is prohibited.

L. Kairouni, M. Elakhdar, E. Nehndi and N. Bouaziz [5] developed a improved cooling cycle for a conventional multi-evaporator simple compression system utilizing ejector for vapour pre compression is analyzed. The ejector increase the refrigeration cycle consists of multi evaporators. The COP of novel cycle is better than the conventional system.

Theoretical Background of Two-Phase Flow

The innovation presented in this report relies on the principle of two-phase flow, therefore as a part of this project, we have conducted an extensive study of the literature related to this subject. It needs to emphasize that considerable portion of research on the subject has been conducted outside of the USA (Holland, Russia, China), therefore we included non-English literature search as well, especially in Russian, German and Polish languages. In addition to literature study, the Principal Investigator has attended seminars and conferences, in particular Refrigeration and A/C Conference at Purdue U. in July 2004 and International HVAC Conference in Orlando, FL, Feb. 2005. Several trips to universities and telephone discussions with scientists were also undertaken. In particular, PI has traveled to UMass to discuss the theoretical aspects of ejector design with Prof. D. Schmidt and to U. of RI to conduct discussions with Prof. A. Lucia both recognized experts on the subject.

Our main objective in this study was to determine if any previous research was able to describe the conditions for sustained critical flow in two-phase mixtures. The advantages of critical flow are apparent – the independence of flow rate from the discharge conditions (downstream pressure and geometry), in a broad range of counter-pressures. The successful conclusion of this project could provide the fundamentals for design of hydraulic equipment with higher efficiency, for example pumps with flat characteristics, heat exchangers, ideal mixers and proportioners. Similarly, the ejector design, which was proposed for this project, relies on the condition of sustained critical flow. Our conclusion was that even though many researchers present various results and data from their studies, the governing parameters associated with critical flow in two-phase medium, such as gas content, bubble radius, pressure ratios, flow ratios, etc. remain to be investigated.

It is known that gas-liquid flow has often a non-equilibrium character, which demonstrates itself by occurrence of pulsation of pressure, density, velocity and temperatures for both phases. Under certain conditions, these processes cause undesired effects, such as water hammer, vibrations of pipelines, perturbation of circulation modes and heat exchange, while under different conditions, occurrence of two-phase flow improves the heat exchange and increases the efficiency for many components of energy systems. Therefore, various theoretical schemes were considered in which the dynamic properties of two-phase stream or “bubbly liquids” could be profitably used in the design of propulsion devices. The underlying idea is that expansion of a compressed gas bubble-liquid mixture might be an efficient way to produce the momentum necessary for thrust.

Models exist for the prediction of unsteady two-phase flashing flows in variable cross section ducts and valves, including Homogeneous Equilibrium Model (HEM) and Homogeneous Relaxation Model (HRM) [2], [6]. None of those models however show the existence of a critical condition without making assumptions, which are unrealistic and cannot be justified experimentally. For example, HRM assumes small relative velocities in relation to speed of acoustic waves in medium. Despite this, the critical condition does exist in two-phase flow and can be very well observed in practice as shown by a number of studies, including the observation of this author on water-air mixture using the transparent nozzle (this experiments is described in details further in this report). The sudden change from misty and milky flow to bubbly flow is apparent at certain flow conditions, such as velocity, backpressure and volumetric content of each phase. One of fundamental publications on the subject, by Wallis [2] acknowledges the presence of critical flow but simultaneously confirms that the mathematical model for such condition is extremely complicated since it has to consider not only criticality in one location, but also may include parts of the upstream system. Concluding, it is possible to state that two-phase mixture, if it is sufficiently homogeneous, has completely different properties than each of its component. The most important is the reduction of speed of sound as first observed in 1941 (Wood) [1], which brings the possibility of supersonic flow at moderate velocities: 20-50 m/s.

The majority of research on two-phase flow has been devoted to nuclear reactor behavior under LOCA conditions. A multitude of computer codes were developed with different levels of accuracy and correctness of the underlying models and assumptions. The most comprehensible laboratory experiments to date were carried out in France under MOBY DICK program in 1980's and included study of steady state critical flow in nozzles at medium to high pressures ("steam-water choked flow")[6]. The nozzle model was selected due to its importance in simulation of small breaks in pipe. The results were widely disseminated throughout the scientific community; however, these experiments were not consequential in changing the way for two-phase flow engineering design. Based on our study, it appears that the detailed simulation of fast transient two-phase flow is yet an unresolved problem in spite of its practical importance and the progress in research in last several decades. Such fast transient appears in many industrial applications and processes with variety of initial and boundary conditions, different fluid and different thermodynamic conditions.

In the engineering practice, the dynamic of fluids is described by two fundamental properties: viscosity and compressibility. Specifically for liquid, the viscosity and Reynolds number are determining properties, as speed of liquids is almost always slower than their sonic speed. On the other hand, for gases, which often move with speeds near their sonic speed, a Mach number or compressibility becomes the determining factor for calculations. The situation changes drastically for two-phase mixtures. To determine the dynamics of such flow the existing models still consider modified Reynolds number and viscosity, but traditionally compressibility is not utilized in these calculations. This is the great disadvantage of existing theoretical approach to analysis of two-phase flow because ignoring the influence of Mach number in two-phase flow leads to pipelines vibrations, intensification of waves, and possibly also inaccuracy in predicting LOCAL conditions in nuclear reactors [3]. Such situation is hard to understand since many publicized research results point to lower speed of sound in two-phase flow and consequently its higher compressibility.

The newer research shows that the two-phase medium is more compressive than the gas and consequently, the speed of sound in two-phase mediums is much lower than that in the homogeneous gas or liquid. By starting from the volumetric content of gas in gas-liquid mixture and introducing certain assumptions, i.e. no slip between phases and isothermal nature of the flow, Van Wijngaarden [1] derives his fundamental formula for the speed of sound in two-phase mixture:

$$a^2 = p / \rho_f \, \beta(1-\beta) \tag{1}$$

Where a is the speed of sound, p is the pressure, ρ_f is the density of the liquid phase and β is the volume occupied by the gas in a unit volume of the mixture. The formula contains the result that, unless ® is very close to either zero or unity, the speed of sound in the two-phase mixture is lower than speed of sound in pure gas. A minimum exists for ® = 0.5, in which case, at a pressure of 1 bar, a mixture of air and water has a sound velocity of 20 m/s. Similar results were obtained for two-phase homogeneous mixture, such as water-steam, liquid refrigerant-vapor refrigerant, etc. These results were confirmed by a number of researchers in USA and abroad. Consequently, a handful of articles published in 1980's and 90's [3], [5] present diagrams for speed of sound vs. ® for two-phase mixture. The general character of such diagrams is shown in Figure 1 below.

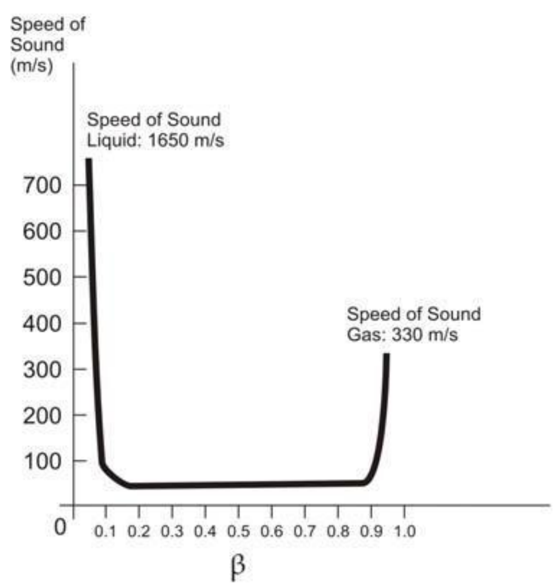


Figure 4. Sonic speed for two-phase flow

Van Wijngaarden [1] describes also the process when the vapor is accelerated in a jet device to a velocity to or slightly above the sonic velocity. With this increase in velocity, the energy of thermal motion of the molecules is converted into kinetic energy of flow and accompanied by a simultaneous decrease of internal energy. This increased kinetic energy is then converted in the jet device into potential energy in form of higher pressure. Such conversion is realized by deceleration of supersonic vapor flow by introducing a liquid component of higher density. The mixing of two phases, vapor and liquid, leads to the decrease in local sound velocity according to the diagram of Figure 1 and the concomitant creation of a "shock wave" with consequent increase of pressure. The pressure ratio achieved by such shock wave can be calculated from the formula given by Campbell & Pitcher [1] for isothermal process:

$$p_2 / p_1 = M^2 \qquad 2$$

where M is the Mach number

For adiabatic process, the formula given by Fisenko [3] takes slightly different form:

$$p_2 / p_1 = 1 + k \beta M^2 \qquad 3$$

Where k is the adiabatic coefficient: $k = c_p/c_v$

The above theory brings about the possibility of obtaining the supersonic flow in Laval nozzles and this can be considered in propulsion devices. Among others, Witte [4] investigated the efficiency of a propulsion device based on injection of compressed air bubbles in the throat section of a nozzle and observed the pressure jump associated with the supersonic flow. Indeed, this was further confirmed by both computer modeling and laboratory experiments.

Reassuming, the state-of-an-art study presented above has concluded the following:

1.Critical flow in two-phase media such as air-water and steam-water were observed by a handful of researchers (including this author), however the theoretical model remains to be developed. Under critical flow conditions the flow rate from the system is independent of the conditions in the receiver (especially pressure).

2.The speed of sound in two-phase mixture is much lower than in any of its components, thus the supersonic flow can exist in such mixtures at relatively low velocities, i.e. 20-50 m/s.

3.The supersonic flow in Laval nozzles produces “shock waves”, thus increasing the outlet pressure proportionally to the square of Mach number achieved in the nozzle. Shock waves of this nature were observed in experiments and called either “mixing shocks” or “condensing shocks”, however the exact condition for their presence are not described in the available literature.

4. The amount of research on the above subject is limited and no practical applications were found. It is possible that this project might be the first attempt to bring the practical use of two-phase flow theory.

5. It appears that one of research directions in this and next phases of this project has to be definition of conditions for a sustained critical flow in two-phase mixture of liquid and vapor R22 refrigerant. No previous research was found on this specific subject.

CHAPTER 3

THEMODYNAMIC ANALYSIS OF THE REGENERATIVE VAPOUR COMPRESSION CYCLE

3.Description of the New Refrigeration Cycle

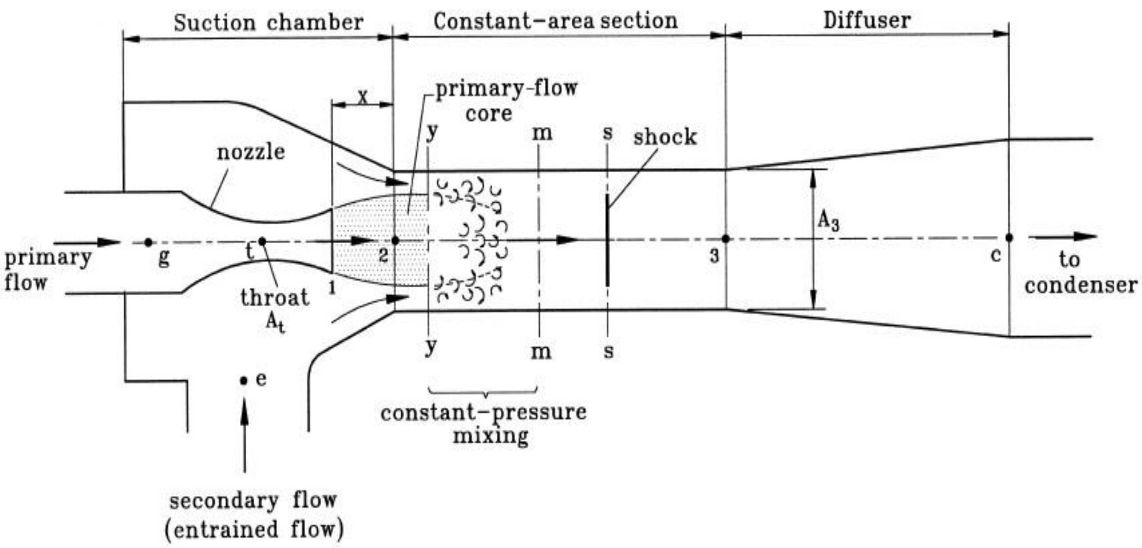
3.1 Technical Concept

The nature of the innovation presented in this report lies in the application of supersonic ejector to increase the efficiency of vapor compression refrigeration cycle. We decided early in the stage of development that the working medium would be R22, the refrigerant that is widely available and relatively inexpensive. For that reason, all designing was performed for this refrigerant. Following are novel concepts and explanation how do they differ from previously reported ejectors for refrigeration cycles.

1.The ejector is used as a second stage compressor in the cycle. In this capacity it is lowering the compression ratio, but not by increasing the suction pressure as in known previous systems but by decreasing the discharge pressure from the compressor. The disclosed ejector is used as a second stage compressor in the cycle.

2.The design of the ejector is based on the theory of two-phase flow, which considers the previously explained lowering of speed of sound in two-phase mixture with subsequent creation of the shock, which increases the pressure on the outlet of the ejector.

3.In our ejector, the motive stream is the liquid refrigerant while the suction stream is the compressed vapor refrigerant from the compressor.



The Figure 5: shows the design of the supersonic ejector:

The principle of the ejector operation, presented above is utilized to construct the cooling/refrigeration system shown in Figure 3. In this new system, the mechanical compressor compresses the vapor to approximately 50-60% of the final pressure. Additional compression is provided by the ejector device explained above using internal

compressor and is sent to the ejector where it mixes with the liquid flow coming from the separator, located after the condenser. The flow of working medium is then directed to the condenser where it is cooled by transferring the heat to the high-temperature receiver. The application of this invention improves the efficiency of the thermal transformation by means of lowering the need for energy to run the compressor.

potential energy of the working fluid flow. Therefore, the amount of mechanical energy required by a compressor is significantly reduced. The principle of the proposed system as shown in Figure 3 includes the main piping circuit (1), containing the evaporator (2), a compressor (3), an ejector device (4), a condenser (5), a separator tank (6), an intermediate heat exchanger (7) and an expansion valve (8). The circulation of a liquid phase of the working medium is provided by the additional liquid line (10 and 11), and a pump (9). The evaporator (2) absorbs the heat from source (12), while the condenser (5) is connected to the heat sink – high temperature heat receiver (13). It needs to note that the device as above can be used also for heating and in this capacity it can operate as a heat pump.

The working medium kept at low pressure vaporizes in the evaporator with using the heat energy of low-temperature source. Further, the working medium is compressed in the

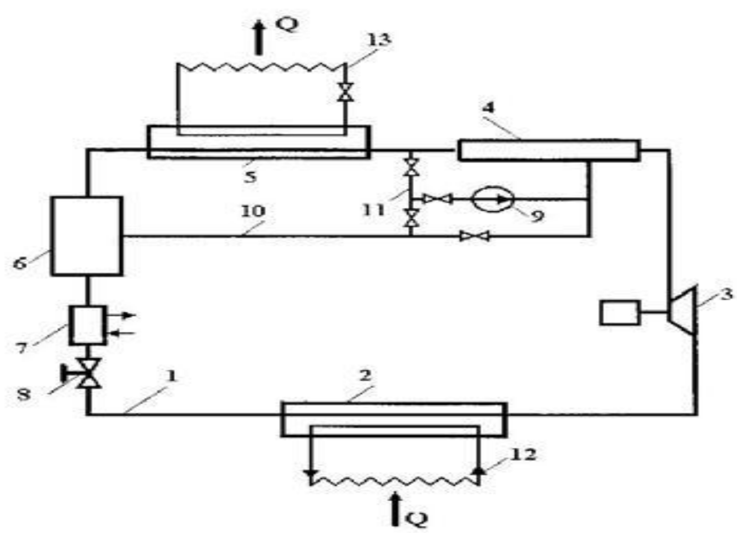


Figure 6. Schematic of the new refrigeration system/heat pump

The above purpose is realized by the method of heat transformation that includes evaporation of the part of working medium at lowered pressure with utilizing the thermal energy of the low-temperature source, mixing two parts of the working medium in a jet device, cooling the flow of working medium with transfer of its thermal energy to high-temperature receiver, and dividing the working medium to two parts.

The other specifics of the new device are:

-additional connection of ejector device to the circuit after the condenser in order to regulate the temperature of the ejector device.

- Installing a pump for liquid pumping into the side piping (feedback circuit)
- Installing a separation tank between cooler and expansion valve.
- Installing of the additional cooler (heat exchanger) before the expansion valve.

In the proposed method as opposed to known methods, the compression of the working medium is replaced by the compression of the vapor part of the working medium in the first stage in a compressor and then in the second stage in a vapor-liquid ejector device. In the ejector device, liquid-vapor mixture achieves supersonic velocity, which causes the sudden increase of the pressure (shock wave) with simultaneous condensation of the vapor and increase of the temperature.

The proposed method can be realized by utilizing the traditional working mediums such as low temperature boiling fluids, the same as used by heat pumps and home refrigerators, for example R12, R22, R134a etc. or their mixtures with mineral or synthetic oils, water, etc.

Further explanation of the new proposed refrigeration cycle is presented on the p-h diagram in Figure 4 below. It needs to emphasize that the presentation as in Figure 4 is conditional and serves the purpose of illustration since the exact presentation of these processes is rather difficult because they are not stationary and have the variable masses of the working fluid. In Fig.4 the following processes of change in the state of a working fluid are depicted:

1-2 - evaporation of a part of the working fluid;

2-3 – compression of vapor in the compressor (the first step);

3-4-8 – mixing of vapor and liquid parts of the working medium in the ejector;

4-5 - compression of the working medium in the ejector (the second step);

5-6 - isobaric cooling of the liquid working medium;

6-7- compression of a part of the cooled liquid working medium by the pump;

7-8- expansion of this part of the cooled liquid working medium in the ejector;

6-1 – throttling of the evaporating part of the working fluid.

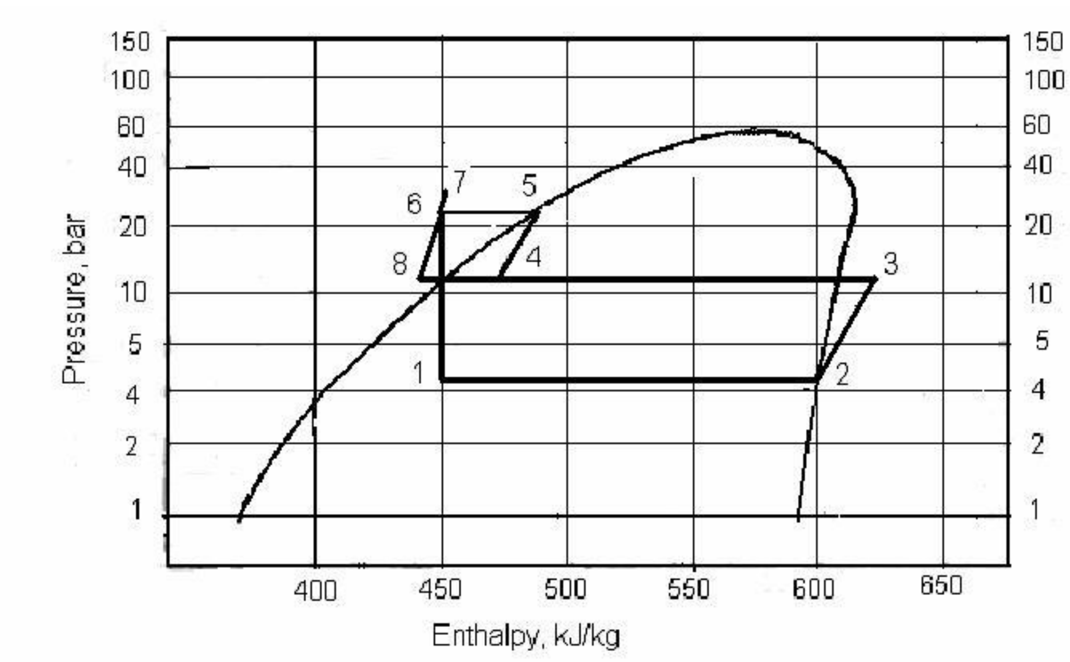


Figure 7. P-h diagram of the new refrigeration cycle with a two-phase ejector for R134a refrigerant

The technical concept proposed for the realization of the considered project is based on the following preconditions: - expediency of the creation of the geothermal heat pumps for the average climatic conditions as in USA and specifically New England; - the productivity of the created prototype should correspond to the equipment being in greatest demand in the market for similar services.

3.2. Theoretical Model of the New Refrigeration Cycle with a Two-Phase Ejector

A theoretical model has been developed to evaluate the capabilities of the two-stage refrigerating system with a vapor-liquid ejector. The main distinguishing feature of such model from the similar models of the vapor-ejector or gas-ejector refrigeration cycles is the use of a liquid flow as ejecting (motive stream) agent and of the vapor phase as ejecting medium (suction stream). At present the issues of using the vapor-liquid flows in the refrigeration cycles appear not yet to be reflected sufficiently enough in the scientific and technical literature [3].

To simplify the theoretical model of the refrigeration cycle with a vapor-liquid injector, the following assumptions that are analogous to the ones in the paper [12] are made:

1. Neglect the pressure drop in the condenser and evaporator and in the connection tubes.
2. No heat losses to the environment from the system, except for the heat rejection in the condenser.
3. The vapor stream from the separator is a saturated vapor and the liquid stream from the separator is a saturated liquid.
4. The flow across the expansion valve or the throttle valves is isenthalpic.
5. The compressor has a given isentropic efficiency.
6. The evaporator has a given outlet superheat and the condenser has a given outlet temperature.
7. The flow in the ejector is considered a one-dimensional homogeneous equilibrium flow.
8. Both the motive stream and the suction stream reach the same pressure at the inlet of the constant area mixing section of the ejector. There is no mixing between the two streams before the inlet of the constant area mixing section.
9. The expansion efficiencies of the motive stream and suction stream are given

constants. The

diffuser of the ejector also has a given efficiency.

Using these assumptions, the equations of the ejector expansion R22 cycle have been stated. Assuming that the pressure before the inlet of the constant area mixing section of the ejector is P_b and the ejection ratio of the ejector (ratio of mass flows of vapor $m_v$and liquid m_f) is

$$w = m_v / m_f \qquad 4$$

The motive stream follows an isentropic expansion process from pressure $P_1$to pressure $P_b$before

it enters the constant area mixing section, or otherwise the value of entropy $S_i$for the moving stream in the point 7 and in the point 8 are equal:

$$S_7 = S_8 \qquad 5$$

The corresponding enthalpy h_8 of the moving stream at the end of the isentropic expansion process can be determined using the P-h diagram for R22 or by equation

$$h_7 – h_8 = (p_7 – p_8) / \rho \qquad 6$$

Further, applying the conservation of energy across the expansion process, the velocity of the motive stream at the inlet of the constant area mixing section is given by equation:

$$U_{mb} = \mu\sqrt{2}(h_8 – h_7) = \mu\sqrt{2}gH \qquad 7$$

where g= 9.81 m/ s^2,$\propto$ is a coefficient of discharge and H is pressure difference (P_7– P_8) of the motive stream expressed in meters of a liquid column. With using a P-h diagram we can find the specific volume for both the motive stream in the point 8, V_8, and the suction stream in the point 3, V_3, as well as the same for their mixing in the point 4,V_4 .In the case of two-phase flows, the cross-section area of the ejector mixing section per unit total ejector flow rate, a_m, can be determined by equation:

$$a_m = V_m / u_{mb} \qquad 8$$

in which V_m = 2/ (ρ_4+ ρ_5) is the mean specific volume of the vapor-liquid mixture at the ejector mixing section and ρ_4, ρ_5 is the density of the vapor-liquid mixture in the states corresponding to points 4 and 5 on the P-h diagram of Figure 4.

The method employed here for calculating the cross-section area of the mixing channel is characteristic of similar techniques of two-phase ejector calculation given by Fisenko [3]. From the known values of the velocity of the mixing stream and across mixing section area(cross-section area of the cylindrical channel at the mixing chamber outlet), it is possible to calculate the pressures of the working fluid flow at the mixing chamber outlet, P_{mix}, and at the ejector outlet after the diffuser, P_d. In this event the following equations were applied:

$$P_b\, a_m + (1 /_{(1+w))} * u_{mb} + (1 /_{(1+w))} * u_{sb} = P_m\, a_m + u_{mix} \qquad 9$$

$$P_d = P_{mix} + \rho\, (u_{mix}^2 - u_d^2) / 2 \qquad 10$$

the former being the momentum conservation equation, whereas the latter is the energy conservation equation in the form of Bernoulli equation. In these equations u_{mb}, u_{sb} – velocities of the liquid and vapor flows (motive and suction) at the mixing section inlet, u_{mix}, u_d -mixture flow velocity at the diffuser inlet and outlet.

It needs to emphasize that in our case, the mixture velocity in the mixing chamber has to be somewhat higher than local sonic speed of this two-phase flow because in this case the efficiency of the vapor-liquid ejector increases [3]. In its turn, according to the known data [1], the speed of sound propagation α in a two-phase medium can be as low as only 20-50 m/s and for its estimate one can apply the equation:

$$\alpha^2 = kP / \rho_{mix} \;\; or\; a^2 = P // \rho_f \beta (1 - \beta) \qquad 11$$

where k is isentropic coefficient, P, ρ_{mix} is pressure and density of the two-phase flow, ρ_f is density of the liquid phase and β is the volumetric content of vapor in the mixture.

The quantity of energy ℓ_p, consumed by a pump in compressing a working fluid is calculated with the formula

$$\ell_p = m_f\, \Delta P_{7\text{-}6}/(\rho_{mix}\eta) = m_f(h_7 - h_6)/(\rho_{mix}\eta) \qquad 12$$

where η is efficiency (coefficient of efficiency) of the pump.

The quantity of energy needed for the compression of the vapor flow m_v by the compressor with

the performance η_c is determined by the expression

$$\ell_c = m_v(h_3 - h_2)/\eta \qquad 13$$

Other energy characteristics of the cycle are defined as follows:

- Refrigerating capacity of the system Qo

$$Q_o = m_v(h_2 - h_1) \qquad 14$$

- Thermal performance Q_h

$$Q_h = (m_f + m_v)(h_5 - h_6) = (m_f + m_v)\, c_p(T_5 - T_6) \qquad 15$$

- The compression work ℓ done by the compressor and the pump

$$\ell = \ell_c + \ell_p \qquad 16$$

The coefficient of performance (COP) of the two phase ejector cycle can be determined by:

$$COP = Q_h / \ell \qquad 17$$

For the basic one-step refrigeration cycle operating in the same temperature range, the evaporator heat capacity Q_{bo} and the condenser heat capacity Q_{bh} are given by:

$$Q_{bo} = m_v(h_{b2} - h_b) \qquad 18$$

$$Q_{bh} = m_v (h_{b3} - h_{b6}) \qquad 19$$

The compressor work of the same basic cycle operating without using the ejector is found by:

$$\ell_b = m_v (h_{b3} - h_{b2}) \eta \qquad 20$$

where h_{bi} are the enthalpies of the corresponding points in the P-h diagram cycle of Fig. 5 where a comparison is shown between one step compression conventional cycle and the new cycle. Then, the performance of the basic refrigerant cycle with the same temperature range is given by:

$$COP_b = Q_{bh} / \ell_b \qquad 21$$

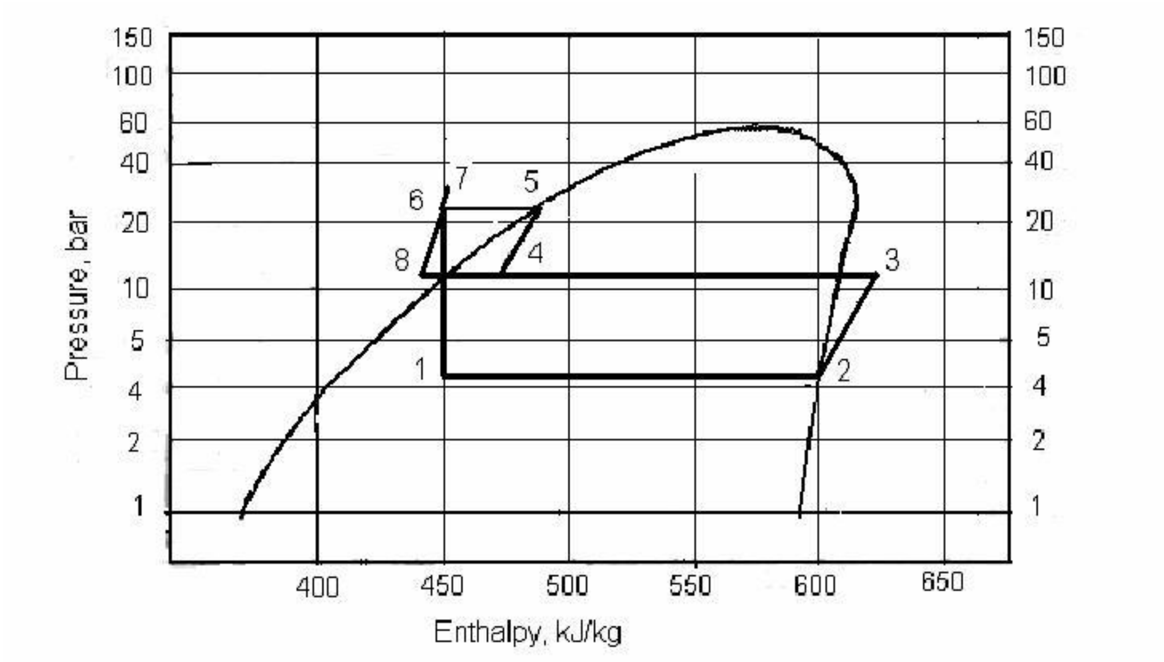


Figure 8. Comparison of P-h diagrams of the new refrigeration cycle with a two-phase ejector, Cycle 1 (points: 1-2-3-4-5-6-1 and 6-7-8-4) and the traditional cycle Cycle 2: (the point: 1-2-3 -6-1)

The relative performance of the two-phase ejector refrigerant cycle is defined as:

$$R = \mathrm{COP} / \mathrm{COP}_b \qquad 22$$

Using the above theoretical model, the relative performance of the two-phase ejector R22 cycle in comparison with a similar traditional cycle with the one-stage compression in the same temperature range can be estimated. The P-h diagrams of these cycles are presented in Fig. 5, while the properties of the refrigerant in the characteristic points of these diagrams are tabulated in Table 1. In carrying out calculations it has been assumed that coefficient of efficiency of the hydraulic pump and compressor is equal to 0.8, and corresponding values (magnitudes) of the evaporator capacity for both cycles under consideration are identical. For the case studied which corresponds to the data of Table I, the ejector displays following characteristics:

- Velocity of outflow of the motive fluid from the ejector nozzle

$$U_{mb} = \mu\sqrt{2(h_7 - h_8)} = 0.86\sqrt{2*2*10^3} = 54.4 \text{ m/s} \quad 23$$

- Cross-section area of the mixing nozzle

$$A_m = 2/(\rho_4 + \rho_5)\, u_{mb} \quad 24$$

Then, considering that u_{mb} = *54.4* m /s, $u_{sb} \approx$ *10* m /s, w= 0.11, ΔP= *1.3* MPa , P_b= *1.19* MPa and u_{mix}= 20 m /s, through the equation (9) we find the value of the mixture pressure P_{mix}= *25.16* MPa. The specific energy characteristics of the cycle with a vapor-liquid ejector (cycle 1) in comparison with the traditional cycle (cycle 2) are given in Table II.

The conducted theoretical analysis of the two-phase ejector for R22 cycle, despite the assumed simplifications of the model, permits to conclude that the application of vapor-liquid jet devices opens up new opportunities for a substantial improvement of the refrigeration cycle efficiency.

CHAPTER 4

RESULTS AND DISCUSSION

Results and discussion

TABLE-I

Qualities and quantity of R134a and R152a Refrigerant in characteristics points on the diagram

Points	P (bar)	ti (°C)	Si (kJ/kgK)	Xi
1	4.2	-5	4.07	0.22
2	4.2	-5	4.76	1
3	11.9	35	4.77	1
4	11.9	30	4.21	0.1
5	24.2	55	4.25	~0.01
6	24.2	40	4.15	0
7	30	41	4.16	0
8	11.9	30	4.14	~0.02
3''	24.2	85	4.76	1

Performance of regenerative cycle with R134a

For the basic cycle of vapor compression refrigeration system using R134a as arefrigerant COP is=2.935, and for the regenerative cycle using ejector as second step compression COP is=3.589 which is higher than basic cycle of vapour compression system. COP is increased by 22%. Where as work of compression is also reduced.

Performance of regenerative cycle with R152a

For the basic cycle of vapor compression refrigeration system using R152a as arefrigerant COP is=3.012, and for the regenerative cycle using ejector as second step compression COP is=3.601 which is higher than basic cycle of vapour compression system. COP is increased by 16%. Where as work of compression is also reduced.

Effect of Mach no. On U_{mb}

As the mach no. in mixing chamber is increasing the velocity in mixing chamber is also increasing this increase in velocity results in increase in exit pressure of diffuser of ejector.

Table -2

Effect of mach no. on Umb	
M	U_{mb}
1	20
1.2	26.2
1.4	29.4
1.6	30.1
1.8	32.5
2	34.01
2.2	38.4
2.4	42.6
2.6	45.4
2.8	50.6
3	55.3

Effect of mixing velocity on diffuser exit pressure

As the velocity in mixing chamber is increases the pressure at exit of diffuser in also increases because by the increase of velocity of fluid in mixing chamber there is increase in kinetic energy of fluid, this increase in kinetic energy converts in to pressure rise in diffuser.

Table-3

Effect of Umb on ejector exit pressure	
U_{mb}	Pressure(MPa)
20	1.19
25	3.2
30	4.9
35	5.62
40	6.81

Effect of evaporator temperature on cooling capacity

Table-4

Effect of Te on cooling capacity	
Te(0C)	Cooling capacity(kW)
-10	4.8
-8	4.87
-6	4.92
-4	4.94
-2	4.98
0	5.01
2	5.05
4	5.09
6	5.13
8	5.17
10	5.19

Effect of evaporator temperature on entrainment ratio

Table-5

Effect of Te on entrainment ratio	
Te(^{0}C)	ω
-10	3
-8	3
-6	3
-4	3
-2	3
0	3
2	3
4	3
6	3
8	
10	3

Effect of evaporator temperature on COP with R134a

Table-6

Te(^{0}C)	COP
-10	3.39
-8	3.42
-6	3.49
-4	3.541
-2	3.612
0	3.684
2	3.725
4	3.812
6	3.924
8	4.125
10	4.249

Effect of evaporator temperature on COP with R152a

Table-7

Te(^{0}C)	COP
-10	3.421
-8	3.479
-6	3.524
-4	3.645
-2	3.754
0	3.869
2	3.991
4	4.186
6	4.287
8	4.401
10	4.625

Fig. 9: Effect of mach no. on velocity of mixing chamber

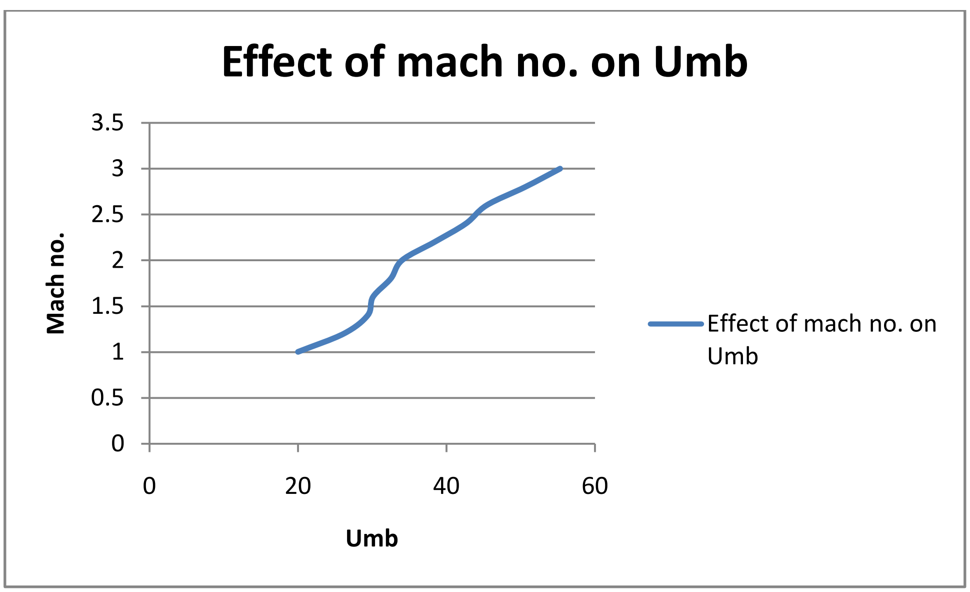


Fig. 10: Effect of evaporator temperature on entrainment ratio

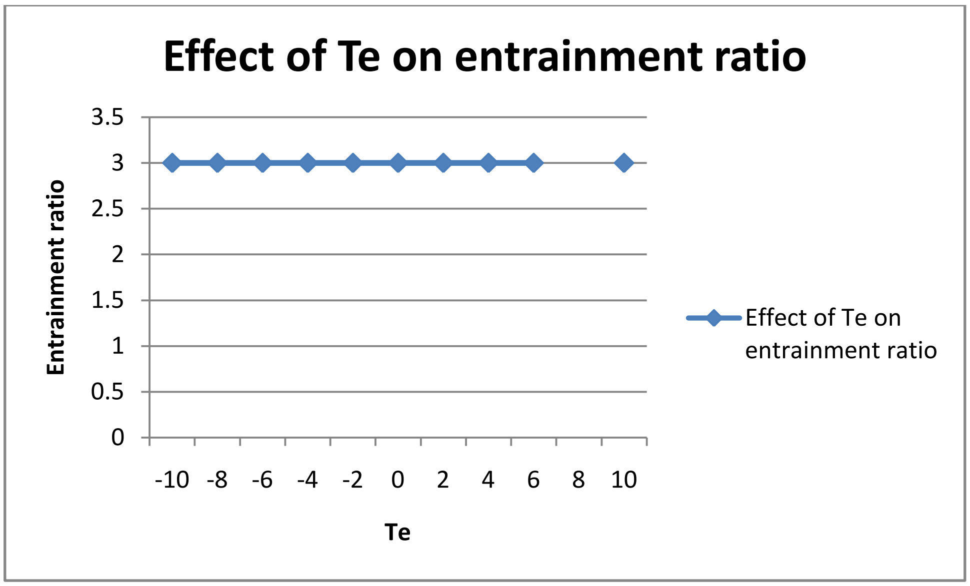


Fig. 11:Effect of mixing chamber velocity on diffuser exit pressure

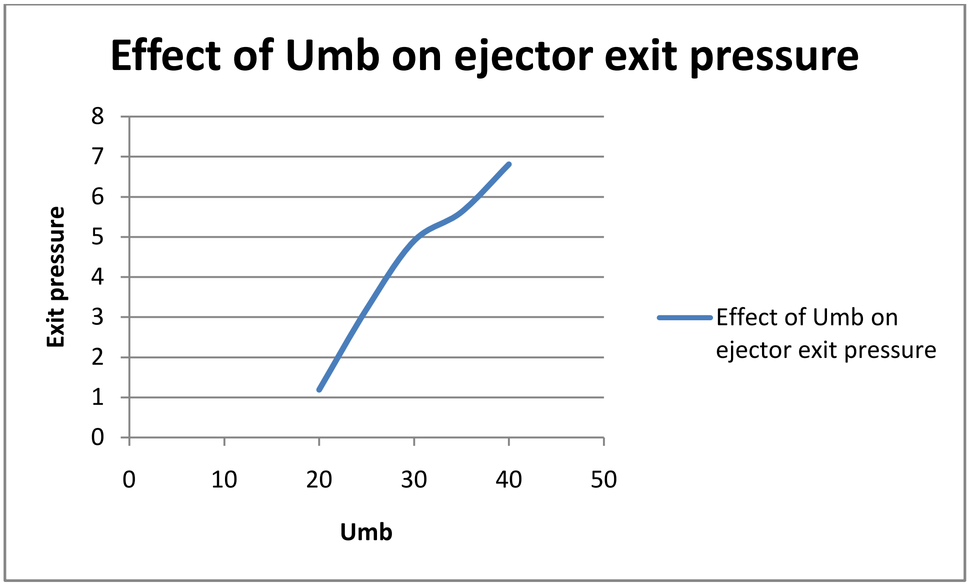


Fig-12: Effect of evaporator temperature on COP with R134a

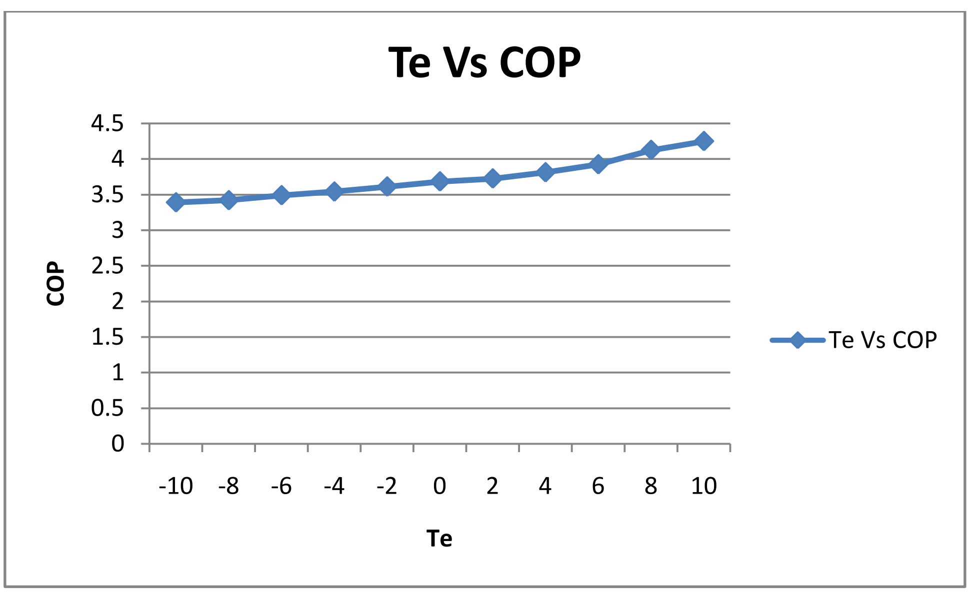


Fig-13: Effect of evaporator temperature on cooling capacity

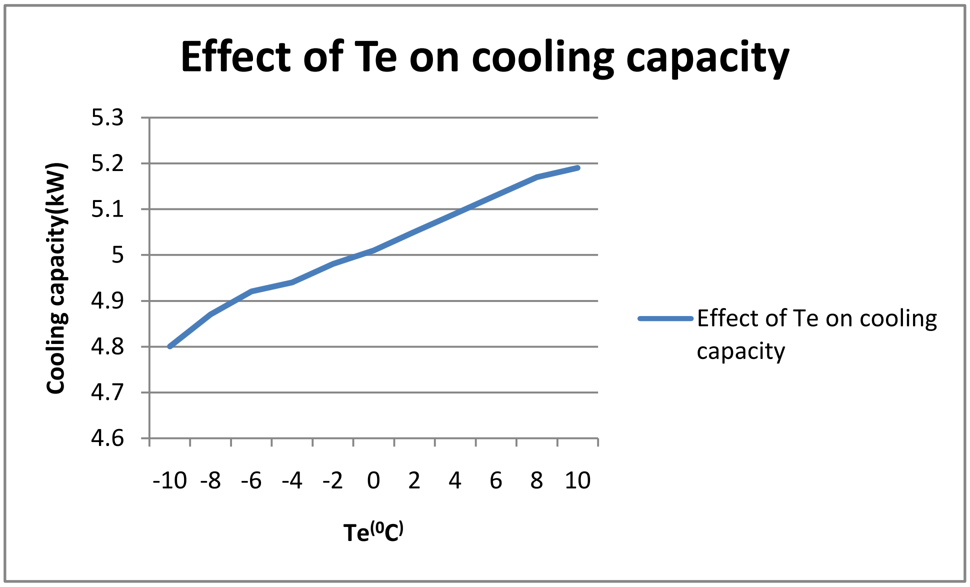
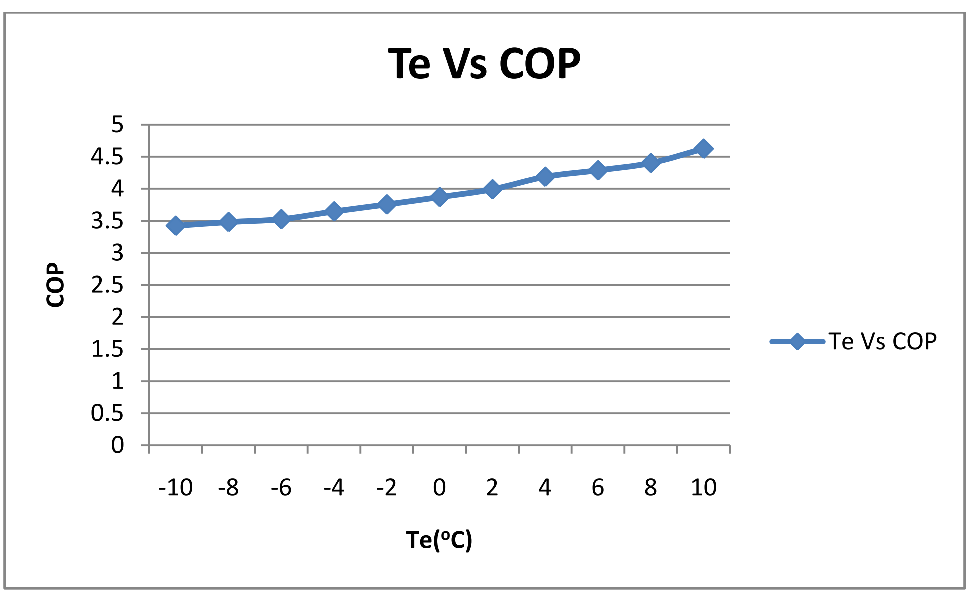


Fig-14: Effect of evaporator temperature on COP with R152a



CHAPTER 5

CONCLUSIONS AND SCOPE FOR FUTURE REASEARCH

Conclusions

In present work performance analysis of a new regenerative vapor compression cycle is done with R134a and R152a as a refrigerant. This discussion and analysis of the obtained results permits the following remarks:

1. COP of the regenerative cycle is higher than the conventional vapour compression cycle.

2. COP in the case of R152a is higher than R134a

3. By using the ejector as a second step compression work of compression is reducing so we can reduce the power consumption.

4. Cost of running the refrigeration system is also reducing.

5.COP of both the refrigerant is increasing with increase in velocity of fluid in mixing chamber of ejector

6. By the decrease of evaporator temperature COP of both the refrigerant is decreasing

7. As the mach number is increasing the pressure at the exit of diffuser is also increasing

8. As the condenser temperature is increasing the COP of both the refrigerant is inscreasing

Scope for future work

1. Performance analysis this new regenerative vapour compression system with experimental system.

2. This analysis can be done for other refrigerant

3. Find out the optimum Pressure rise in compressor and In ejector

4. Designing a best ejector for the purpose of new regenerative vapour compression refrigeration system

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ANNEXURE

ANNEXURE

Program for calculating the COP and different properties of new regenerative vapor compression cycle using ejector

Program for refrigerant R152a

```
h_1=Enthalpy(R152a,T=T[1],P=P[1])
T[1]=-5
P[1]=400.2
h_2=Enthalpy(R152a,T=T[2],P=P[1])
T[2]=-5
P[2]=400.2
h_3=Enthalpy(R152a,T=T[3],P=P[3])
T[3]=35
P[3]=1100.9
h_4=Enthalpy(R152a,T=T[4],P=P[4])
T[4]=30
P[4]=1100.9
h_5=Enthalpy(R152a,T=T[5],x=x[5])
T[5]=55
P[5]=2400.2
x[5]=0.01

h_6=Enthalpy(R152a,T=T[6],P=P[6])
T[6]=40
P[6]=2400.2
h_7=Enthalpy(R152a,T=T[7],P=P[7])
T[7]=41
P[7]=3000
h_8=Enthalpy(R152a,T=T[8],P=P[8])
T[8]=30
P[8]=1100.9
 rho_1=Density(R152a,T=T[1],P=P[1])
rho_2=Density(R152a,T=T[2],P=P[2])
rho_3=Density(R152a,T=T[3],P=P[3])
rho_4=Density(R152a,T=T[4],P=P[4])
rho_5=Density(R152a,T=T[5],P=P[5])
rho_6=Density(R152a,T=T[6],P=P[6])
rho_7=Density(R152a,T=T[7],P=P[7])
rho_8=Density(R152a,T=T[8],P=P[8])
```

```
w=m_v/m_f


u_b=m*(2*(h_7-h_8))**0.5
a_m=v_m/u_b
v_m=2/(rho_4+rho_5)
p_b*a_m+(1/(1+w))*u_b+(w/(1+w))*u_s=p_m*a_m+u_x
p_d=p_m+(rho_5*((u_x**2)-(u_d**2)))/2

u_s=10
w=0.11
p_b=1000.19
u_x=20
m_f=7.154
p_d=2444.2
m=0.9

e_p=m_f*(h_7-h_6)/(rho_7*n_p)
n_p=0.8
e_c=m_v*(h_3-h_2)/n_c
n_c=0.8
q_0=m_v*(h_2-h_1)
q_h=(m_f+m_v)*(h_5-h_6)
e=e_c+e_p
COP=q_h/e
```

Program for the refrigerant R134a

```
h_1=Enthalpy(R134a,T=T[1],P=P[1])
T[1]=-5
P[1]=400.2
h_2=Enthalpy(R134a,T=T[2],P=P[1])
T[2]=-5
P[2]=400.2
h_3=Enthalpy(R134a,T=T[3],P=P[3])
T[3]=35
P[3]=1100.9
h_4=Enthalpy(R134a,T=T[4],P=P[4])
T[4]=30
P[4]=1100.9
h_5=Enthalpy(R134a,T=T[5], x=x[5])
T[5]=55
P[5]=2400.2
x[5]=0.01

h_6=Enthalpy(R134a,T=T[6], P=P,[6])
T[6]=40,
P[6]=2400.2,
```

```
h_7=Enthalpy(R134a,T=T[7],P=P[7])
T[7]=41,
P[7]=3000,
h_8=Enthalpy(R134a,T=T[8],P=P[8])
T[8]=30
P[8]=1100.9
 rho_1=Density(R134a,T=T[1],P=P[1])
rho_2=Density(R134a,T=T[2],P=P[2])
rho_3=Density(R134a,T=T[3],P=P[3])
rho_4=Density(R134a,T=T[4],P=P[4])
rho_5=Density(R134a,T=T[5],P=P[5])
rho_6=Density(R134a,T=T[6],P=P[6])
rho_7=Density(R134a,T=T[7],P=P[7])
rho_8=Density(R134a,T=T[8],P=P[8])
```

```
w=m_v/m_f
```

```
u_b=m*(2*(h_7-h_8))**0.5
a_m=v_m/u_b
v_m=2/(rho_4+rho_5)
p_b*a_m+(1/(1+w))*u_b+(w/(1+w))*u_s=p_m*a_m+u_x
p_d=p_m+(rho_5*((u_x**2)-(u_d**2)))/2

u_s=10
w=0.11
p_b=1000.19
u_x=20
m_f=7.154
p_d=2444.2
m=0.9

e_p=m_f*(h_7-h_6)/(rho_7*n_p)
n_p=0.8
e_c=m_v*(h_3-h_2)/n_c
n_c=0.8
q_0=m_v*(h_2-h_1)
q_h=(m_f+m_v)*(h_5-h_6)
e=e_c+e_p
COP=q_h/e
```