ENHANCEMENT OF REFRIGERATION PERFORMANCE WITH AN EJECTOR

By

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CERTIFICATION OF APPROVAL

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Approved by,

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(Dr. Aklilu Tesfamichael Baheta)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

January 2015

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

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Wani James Ramadan

ABSTRACT

The effective use of energy is an important issue for society considering the future increasing demand, restricted use of fossil fuels and associated environmental problems. This project describes a mathematical model of the ejector refrigeration cycle where the ejector is the main component and replaces the expansion valves in the refrigeration cycle. Modelling equations for ejector analysis were developed using one dimensional steady conditions using conversation of mass, conversation of energy and conversation of momentum equations. The refrigerant used for the cycle is R134a. Using Engineering Equation Solver (EES), a computer simulation was developed to solve the modeling equations under the optimum values of Ejector area ratio (AR) =14, entrainment ratio (ω) = 0.54, nozzle efficiency (η_n) = 90%, diffuser efficiency (η_d) = 85% at operating temperatures of evaporator (T_{evap}) = -5^oC, compressor temperature of (T_{comp}) = 80^oC and condenser temperature (T_{cond}) = 30^oC. With the above operating conditions, the ejector refrigeration cycle achieved a COP of 5.141 compared to COP of 4.609 for the conventional refrigeration cycle. The result shows that ejector refrigeration cycle offers better COP compared to conventional refrigeration cycle.

Keywords: Refrigeration, ejector, coefficient of performance (COP), Engineering Equation Solver (EES), R134a refrigerant.

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CHAPTER 1: INTRODUCTION

1.1 Background of the study:

Refrigeration refers to the process of cooling and maintaining a temperature of the system below that of its surrounding to cool the system or space to the required temperature achieved by transferring heat from a cooler low-energy reservoir to a warmer high-energy reservoir. Ejector refrigeration system is a type of refrigeration system where an ejector is introduced to minimize losses and energy consumption. Its construction, maintenance and installation are relatively cheaper compared to that of traditional refrigerator. One of the most known important applications of refrigeration are the preservation of perishable food products by storing them at low temperatures, providing thermal comfort to people by means of air conditioning extracting heat under controlled conditions described [1].

1.1.1 Ejector:

The ejectors are devices where high-velocity primary fluid mixes with a second fluid stream by mean of momentum and energy transfer; the mixture is then discharged into a region of higher pressure than the source of the secondary fluid. Ejectors can be operated with both incompressible fluids (liquid) and compressible fluids (gas and liquid) depending on applications. With the compressible fluid, the nozzle is supersonic hence supersonic approach that allows greater conversion of primary flow pressure to secondary flow pressure head increase is adopted. Figure 1.1 shows schematic diagram of the ejector refrigeration cycle.

The ejector refrigeration loop consists of two subsystems: the power subsystem, and the refrigeration subsystem. In the power subsystem, the refrigerant flows through the compressor, the condenser, the ejector, the separator and lastly flows back to the compressor to supply high pressure motive fluid. In the refrigeration subsystem, the refrigerant flows through the separator, the throttle valve, the evaporator, the ejector and then back to the separator to supply the required cooling capacity [2]. The motive fluid is first accelerated to supersonic velocity in the convergent–divergent nozzle, which entrains the evaporated fluid from the evaporator and the two fluids mix together in the mixing chamber. In the diffuser, the velocity of the mixed fluid is stepped down and the pressure is lifted to the condenser pressure.

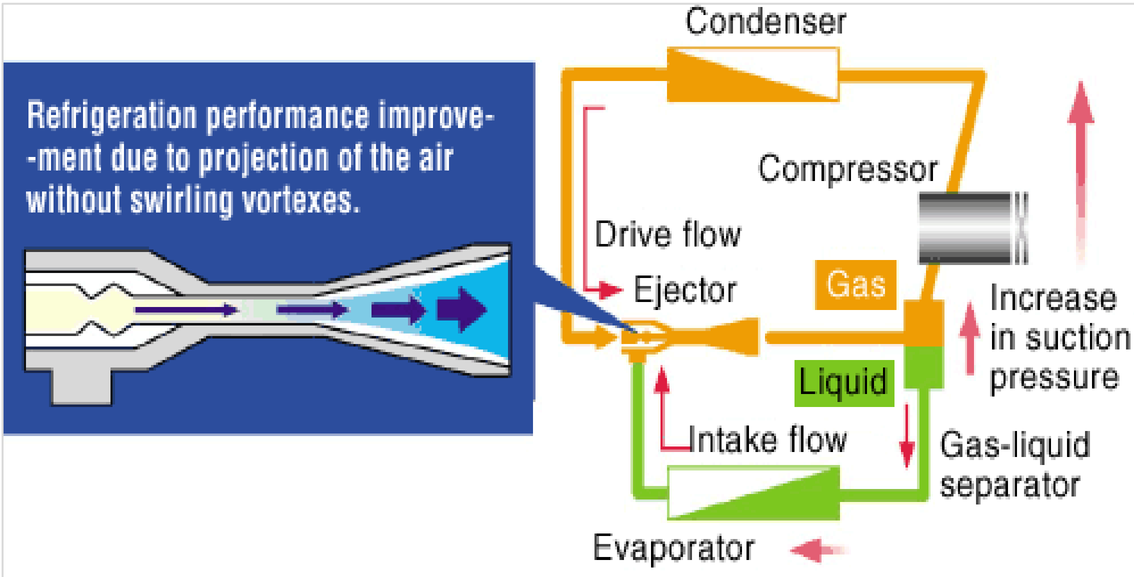

Figure 1.1: Schematic diagram of an Ejector refrigeration cycle [4]

1.1.2 Ejector Development

A brief historical and present development of ejector evaluation started in 1858 when Henri Giffard invented injector to feed water into the reservoir of a steam engine boiler. His idea was later used by Sir Charles Pearson in 1901 inventing an ejector used to remove air from the condenser of steam engine and in 1910, Maurice Leblanc invented the first steam refrigeration system. In 1931, N.H Gay expanded Maurice Leblanc idea to utilize a two-phase ejector instead of expansion valve in the refrigeration cycle. And finally in 2003, DENSO developed the first world ejector cycle that increases the coefficient of refrigerator and air conditioning performance resulting in significant energy savings [1] and [2].

1.2 Problem statement

1.2.1 Problem identification

In the present days, the world experiences a lot of problems mainly energy and environmental crisis as well as the natural disaster. The use of refrigeration contributes substantially in excessive energy consumption and reducing energy consumption will not only contributes towards solving the energy crisis but also reducing the adverse effects on human health and the environmental crisis resulting from the emission of carbon dioxide, nitrogen oxide, sulfur dioxide, etc. from the power generation [3].

Throttling losses and higher power consumption are the major drawbacks of the conventional refrigeration system. In order to reduce throttling losses, ejector is introduced to minimize losses and the energy consumption. Hence, reducing energy consumption and increasing COP, improve environmental preservation thus significantly energy and cost savings (cost effective).

1.2.2 Significant of the Project

The ability of the ejector refrigeration system to produce refrigeration using low grade energy such as waste heat, solar heat, and geothermal energy and including its benefits as in simple construction, no moving parts, no lubrication required give a significant advantages of a reliable system. Not only that but also, it is feasible and economically cost effective as maintenance cost and installation cost are minimized due to its electrical energy saving potential. This project would enhance savings in term of energy and the cost as the result of the reduced energy consumption.

1.3 Objectives of the Project

The main objectives of this project are:

i. to study the enhancement of refrigeration performance with an ejector and compare it to conventional refrigeration cycle.

ii. to carry out parametric analysis of refrigeration cycle with an ejector.

1.4 Scope of the project

This study focused on enhancement of refrigeration performance using an ejector based on relevant mathematical modelling equations:

- Steady state conditions
- Two-phase ejector
- R134a refrigerant selected.

CHAPTER 2: LITERATURE REVIEW

This chapter consists of two parts: the first part introduces the basic of the ejector, parameters influencing the performance. The second part would discuss refrigerants effect on refrigeration performance obtained from comprehensive, relevant literatures.

A typical ejector model is shown in figure 2.1. It comprises of a motive nozzle, a suction chamber, a mixing section and a diffuser. The principle of ejector operation is the conversion of internal energy and pressure associated with workflow of motive fluid stream to kinetic energy. The nozzle is of converging-diverging design to accelerate the working fluid from subsonic to supersonic velocity. The throat is the minimum sectional area of the nozzle where the maximum mass flow rate is determined when the throat is choked. The area ratio of the nozzle exit to the throat is a paramount factor in determining the required velocity at its exit depicted [4].

The mixing can occur at constant pressure and constant area respectively. Shock takes place at the nozzle's constant area section to reduce the mixing velocity from supersonic to subsonic because velocity cannot be reduced below sonic velocity in a converging region. With constant area mixing, mixing occurred in the constant area section and shock takes place before diffuser entrance if fluid is supersonic after mixing. At the diffuser section, the compression with conversion of kinetic energy into enthalpy occurs due to the divergent conical shape of the diffuser. It is also noted that better ejector performance is attained theoretically when constant pressure mixing section was used as seen in the publicized literature [4]. However, experimental data correspond reasonably with constant area analysis although ejector configurations depict constant pressure mixing scenario as the best said.

The constant pressure design model initially was developed and applied in designing and evaluating the performance of various ejectors. Later, modifications were made especially to reduce losses within the ejector and the mixing motive and entrain streams.

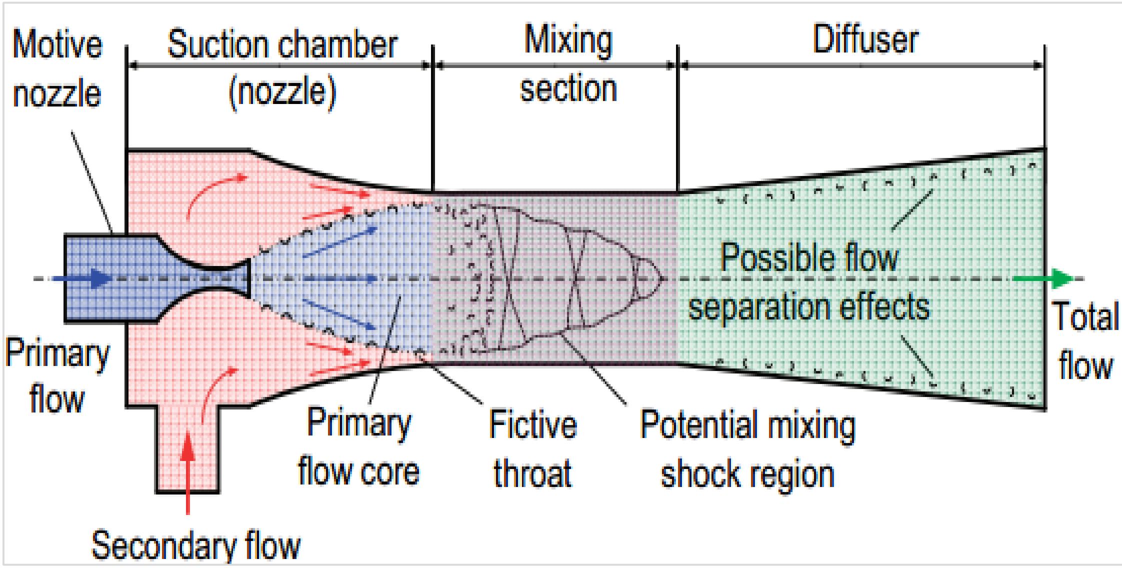

Figure 2.1: Schematic diagram of a typical two-phase ejector (Stefan Elbel, 2006)

The figure 2.2 illustrates the pressure and velocity distribution along the ejector section.

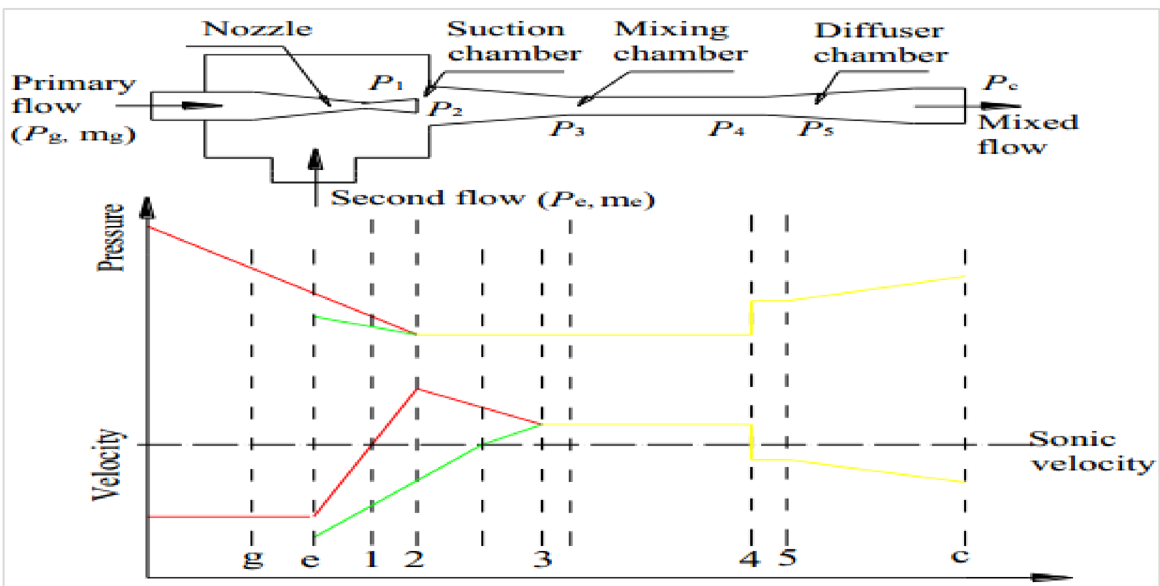

Figure 2.2. Velocity and pressure along the ejector [5]

2.1 Theory of the ejector refrigeration system:

One-dimensional ejector theory was first introduced in 1950 by Keenan based on ideal gas law in conjunction with the principles of the conservation of mass, momentum, continuity and energy [5]. It has been used as a theoretical basis in ejector design for the last fifty years and has been very useful in describing a logical mathematical model of ejector operation. Thermodynamics properties of real gases were employed to eliminate analytical errors unpredicted and found widespread application of ejectors in industrial settings to form a vapor compression heat pump to drive heat compressor which then became common in air conditioning and refrigeration in hotels. He also pointed out that other refrigerants were tested and performed better. His work and [6] concurred and found 20% improvement of the system performance (COP) using halocarbon refrigerants. Experimental study of refrigerant R134a didn't only show enhancement refrigeration performance but also environmentally friendly (refer in appendix D. 2).

In addition, he demonstrated how the performance of the refrigeration system can be improved using two-phase ejector to reduce the inherent throttling losses associated with the use of an expansion valve [7]. Their work, however, corresponded with [8].

The refrigeration cycle using a two-phase ejector where two main improvements were made. First, the cooling capacity increases due to large specific enthalpy difference across the evaporator in comparison to a system having isenthalpic expansion valve and secondly the COP of the system was improved mainly due to reduction in compressor work. Compression reduction increases suction pressure of the compressor by the ejector, hence, compressor work reduced as a result of higher compressor efficiencies thus decreasing compression ratio.

The concept of two-phase ejector systems, transportation refrigeration system by DENSO study resulted in a very high significant improvement of the cooling capacity and coefficient of performance by 25% to 45% and 45% to 65% respectively. However in this study, no definite details, refrigerant used and methodology to achieve results experimentally or computationally were given. Ejector advantages of less additional cost and lower weight were noted.

In a related research, experimental study found COP enhancement of 20% over conventional trans-critical R744 systems [9]. On the same note, simulation modelling of trans-critical R744 two-phase ejector system by Kornhauser's approach resulted in 16% COP improvement.

Mathematical model of one-dimensional of the ejector was developed using modelling equations to govern the flow and thermodynamics based on the constant-area ejector flow model. The theoretical results show that the COP of the ejector cycle is better than the conventional system [10].

Furthermore, Kornhauser's approach to study the effect of ejector on refrigeration performance was used and considerable increase in COP and reduction in throttling losses were also achieved. Also, the experimental results with needle extended into the nozzle throat of the motive nozzle instantaneously improved the COP and cooling capacity by 7% and 8% compared to expansion valve system. The use of ejector also offered reduction in evaporator pressure drop, increase heat transfer coefficient and improved refrigerant distribution in the evaporator.

Figure 2.3 shows results conducted with evaporator temperature (15-15^{o}C), condenser temperature (20^{o}C – 60^{o}C), and $\eta_n = 0.90$, $\eta_m = \eta_d = 0.85$. The results obtained were plotted and it is clearly seen that the COP increases with evaporator temperature and decreases with condenser temperature.

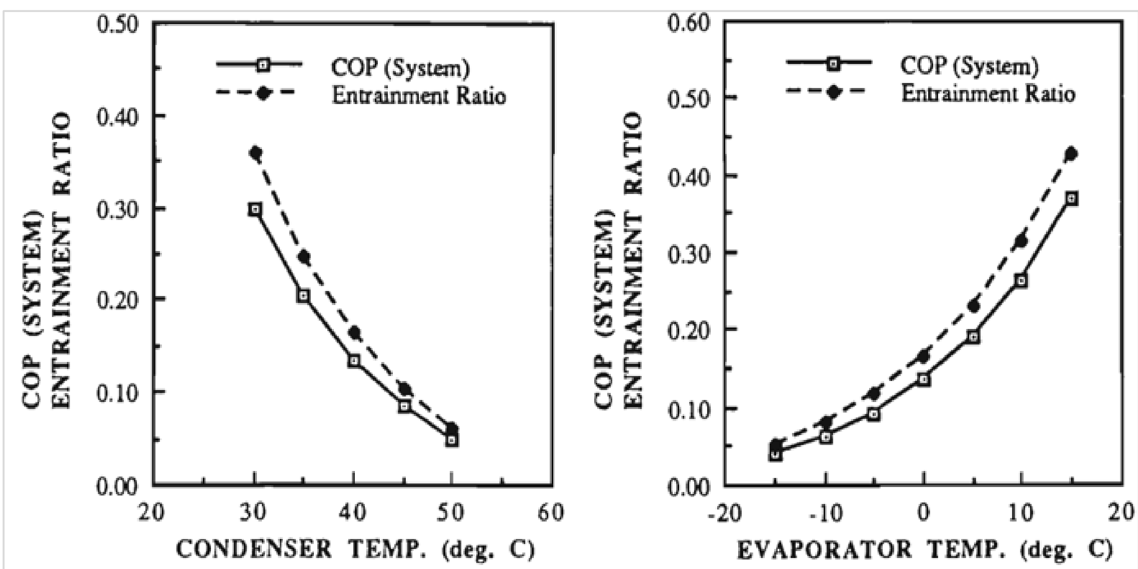

Figure 2.3: Variations in condenser and evaporator temperatures with respect to entrainment ratio (ω) and the coefficient of performance (COP_{ERC}) [4]

On the other hand, experimental study of R134a working fluid was evaluated using evaporator temperature range (6^{o}C, 10^{o}C, 14^{o}C), primary stream temperature range (110^{o}C, 120^{o}C & 120^{o}C), primary stream pressure (143 kPa, 200kPa, 270 kPa) and the condenser pressure (3.5 kPa to 7.0 kPa) and COP improvement of 5.6% was obtained at a primary stream pressure of 270kPa and 15^{o}C, 12.4% at a primary stream pressure of 270kPa and 15^{o}C evaporator temperature and 2.7% at primary stream pressure of 200kPa, and 10^{o}C evaporator temperature [22]. This experimental work was however, validated in comparison with sun (1997) and Chunnanond (2004) as depicted in the figure 2.4.

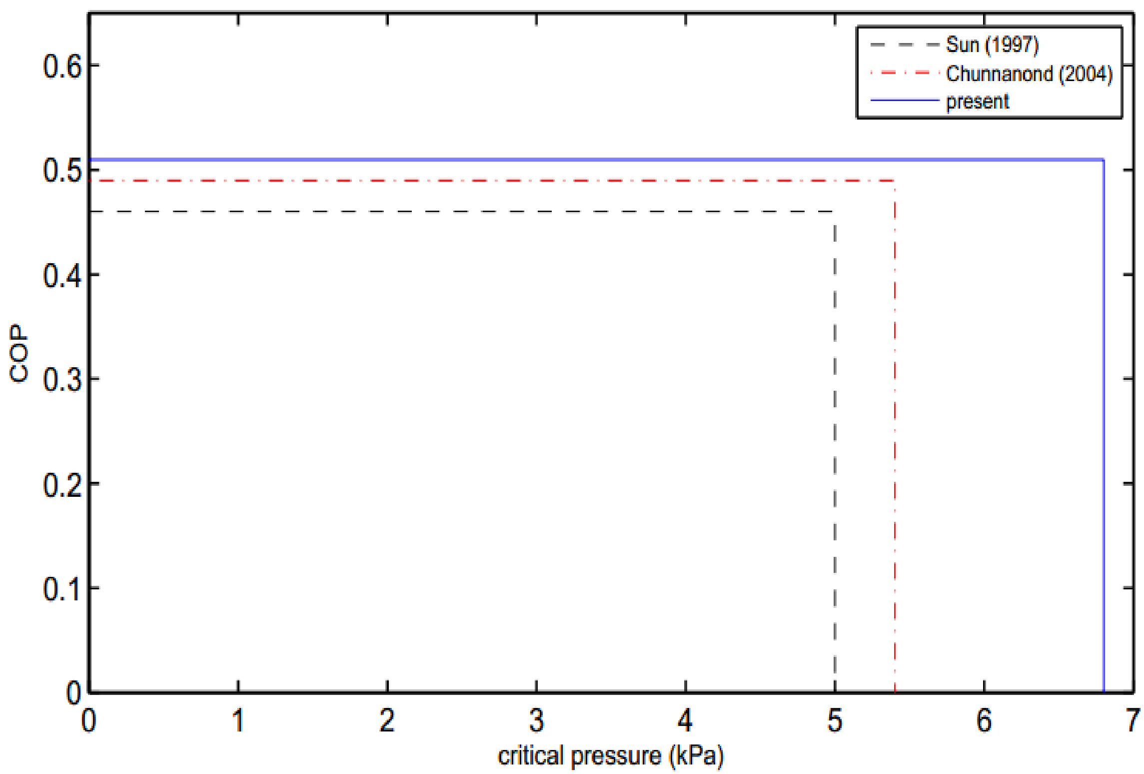

Figure 2.4: Comparison of Al-Doori experimental results (2013) and those of Sun (1997) and Chunnanond (2004)

2.2 Refrigerants (working fluids)

Besides ejector geometrical influence on the system performance, refrigeration performance also depends on thermodynamic properties of the refrigerants used [3]. The properties and characteristics considered in selecting working fluid for refrigeration application include: environment aspects like ozone depletion potential (ODP), atmospheric Lifetime, Global warming potential (GWP), total equivalent warming impact (TEWI), and life cycle climate performance (LCCP). Besides the above, other properties are summarized in the table 2.1.

Table 2.1: Refrigerants properties selection

Ozone- and environment friendly	Nonreactive and non-depletive with the lubricating oils of the compressor
Low boiling temperature	Nonacidic in case of a mixture with water or air
Low volume of flow rate per unit capacity	Chemically stable,
Vaporization pressure lower than atmospheric pressure	Suitable thermal and physical properties (e.g., thermal conductivity, viscosity)
High heat of vaporization	Commercially available
Nonflammable and nonexplosive	Easily detectable in case of leakage
Noncorrosive and nontoxic	Low cost

Many studies have been conducted to investigate refrigerant's effect on the performance of ejector refrigeration systems. Ejector refrigeration performance using R-12 was conducted and the analysis found 21% COP increase over the conventional cycle under the same operating conditions. Theoretical investigation of synthetic refrigerants was also conducted using ejector and COP increment of 8.6% was obtained compared to the conventional refrigeration system [11]. Similarly the effect of natural refrigerants (ammonia, propane and isobutane) was carried out, and found that propane got 26.1% COP improvement, whereas isobutane got 22.8% COP, and ammonia was 11.7% COP using ejector as an expansion device in the refrigeration cycle [12].

In a related research to investigate refrigerant's performance, experimental work found that system performance improves when high molecular weight fluid (refrigerants) are used [13]. Various refrigerants such as water, R11, R12, R13, R21, R123, R142b, R134a, R152a, RC318 and R500 [4]. It was found that R-12 got the highest entrainment ratio and COP but again R-12 is CFC, hence, its use is banned. However, water, on the other hand, has an increased COP, thus a serious competitor to others but water is limited by low evaporator temperature, high vacuum and low thermodynamic performance, based on the compressor expander model [5]. On the other hand, analysis on environmentally friendly refrigerants: R-123, R-134a, R-152, and R-717 was conducted [8]. Refrigerants R-134a and R-512a are recommended for temperature of 70^{o}C – 80^{o}C and ammonia suitable for temperature above 90^{o}C. Based on the findings, ejector refrigeration systems with halocarbon refrigerants are considered suitable than systems using water because halocarbons provide cooling temperature below zero degree Celsius and low boiling temperature and even higher temperature.

Air and water established the basis for both experimental and mathematical model for ejector cycle development in a number of refrigerants studies to investigate ejector performance initiating [15]. However, considering conventional restrictions. Some of these refrigerants are phase-out due to the demand for non-inflammable, non-toxic, non-phase-out, and non-supercritical state refrigerants. These refrigerants were reduced to R-134a and R-245fa. R-245fa, on the other hand, exhibits high primary fluid temperature capability (154^{o}C) and the highest thermodynamic performance [16]. Therefore, this leaves R-134a as the best option for ejector refrigeration application due to its good thermophysical properties and thermodynamic performance along with its availability (refer to appendix D. 2).

Regardless of the efforts put forth to enhance ejector refrigeration performance, many researchers still found discrepancy results with relatively low COP of ejector refrigeration less than 0.2. Studies [18] and [19] on refrigerants (R11, R12, R123, R22, R113, R114, R142, or R142b) conducted, got low COP compared with the conventional system. The main problem was the design of the ejector [19]. Further explanation was based on the mixing type such as constant area mixing and constant pressure mixing according to nozzle position [20]. These, however, affect the mixing of the primary and the entrained fluids resulting in low performance. The problem of the low COP of the ejector refrigeration cycle has drawn criticism towards its technology, hence, hindering the development and the commercialization of ejector refrigeration system.

With all the literatures analyzed, it was deduced that the use of ejector in refrigeration systems enhances coefficient of performance (COP) of the refrigeration.

CHAPTER 3: METHODOLOGY

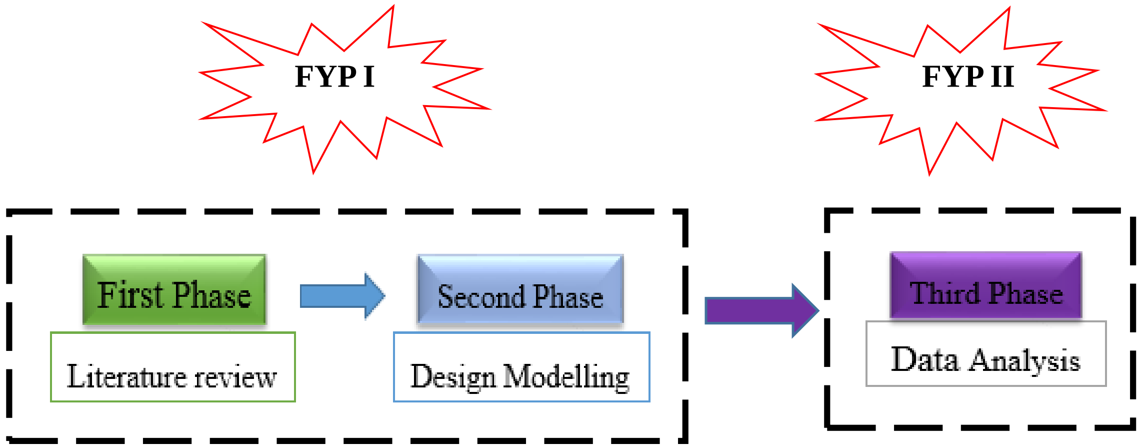

Figure 3.1: phases of the methodology

3.1 First Phase:

Literature review was conducted to gather information about the ejector refrigeration system phenomena, modeling of ejector and information based on performance analysis.

3.2 Second Phase:

Mathematical model using the first principle of thermodynamics and 1-D dimensional analysis was developed. Modeling ejector refrigeration cycle requires understanding of ejector phenomena as well as refrigeration analysis where the mode of operation for both components of ejector and refrigeration were analyzed.

3.3 Modelling Equations:

The governing equations are derived based on figure 3.2, the schematic diagram of the ejector refrigeration cycle and the P-h diagram.

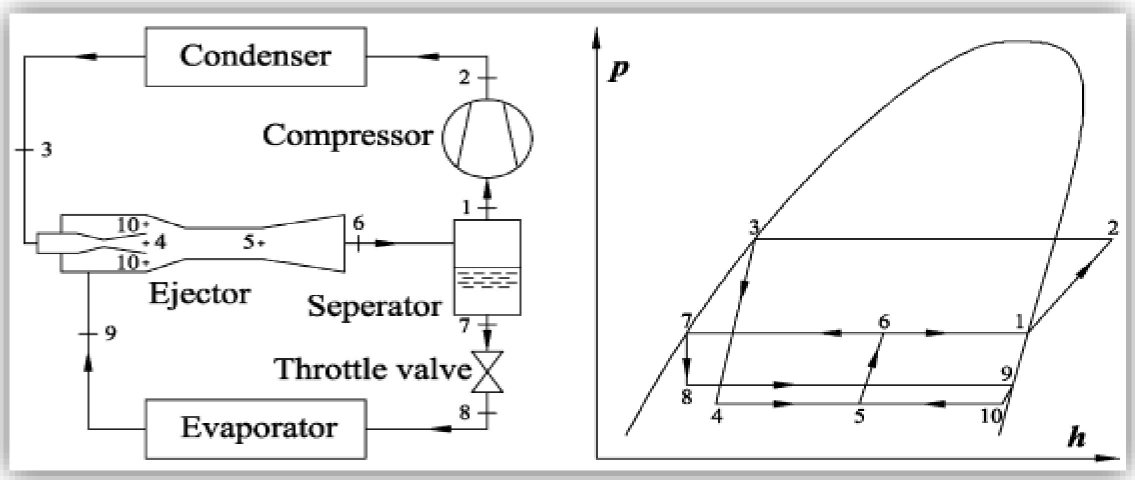

Figure. 3.2: Schematic and p–h diagram of the ejector-expansion refrigeration cycle [20]

Condenser model analysis:

Refrigerant leaving the condenser is considered as a saturated liquid [14].

$$P_3 = P_C = P\ (T_{C,}\ x{=}0) \tag{1}$$

$$h_3 = h\ (T_C,\ x{=}0) \tag{2}$$

$$S_3 = s\ (T_e,\ x{=}0) \tag{3}$$

Evaporator model analysis:

Refrigerant leaving the evaporator is considered as saturated vapor.

$$P_9 = P_e = P\ (T_e,\ x{=}1) \tag{4}$$

$$h_9 = h\ (Te,\ x{=}1) \tag{5}$$

$$s_9 = s(T_e,\ x{=}1) \tag{6}$$

$$P_4 = P_{9\,-}\ \Delta P \tag{7}$$

But the pressure drop, ΔP, in the evaporator and condenser is assumed negligible [9]

$$h_{4s} = h(h_3 - \eta_n\ (h_3 - h_{4s}) \tag{8}$$

Velocity of motive fluid at the nozzle exit, $V_4 = \sqrt{2(h_3 - h_4)}$, (9)

$$A_4 = \frac{1}{(1+\omega)(\rho 4 \omega 4)} \quad \text{as explained [22]} \tag{10}$$

Nozzle model analysis:

$$P_{10} = P_9 - \Delta P, \text{ and pressure drop is assumed negligible} \tag{11}$$

$$h_{10s} = h\,(P_{10}, s_9) \tag{12}$$

And h_{10}s is the enthalpy at state 10 when isentropic compression process is considered [20].

$$h_{10} = h_9 - \eta_n\,(h_9 - h_{10s}), \text{ and } \eta_n = (h_9 - h_{10})/\,(h_9 - h_{10s}) \tag{13}$$

$$V_{10} = \frac{\omega}{(1+\omega)\rho 10 V10} \tag{14}$$

Mixing chamber analysis:

At the ejector mixing chamber, the mixing is done at constant pressure and friction losses are negligible [20].

$$P_5 = P_4 = P_{10} \tag{15}$$

$$V_5 = \sqrt{\eta_m}(\frac{1}{(1+\omega)} V4 + \frac{\omega}{(1+\omega)} V_{10} \tag{16}$$

$$h_5 = \frac{1}{(1+\omega)}\left(h4 + \frac{V4^2}{2}\right) + \frac{\omega}{1+\omega}\left(h10 + \frac{V10^2}{2}\right) - \frac{V5^2}{2} \tag{17}$$

$$S_5 = s\,(P_5, h_5) \tag{18}$$

Diffuser model analysis:

At the exit of the diffuser, the conservation of energy equation is given by:

$$h_6 = h_5 + \frac{V5^2}{2} \tag{19}$$

$$h_{6s} = h_5 + \eta_d\,(h_6 - h_5), \text{ and } \eta_d = (h_{6s} - h_5)/\,(h_6 - h_5) \text{ as quoted from [4]} \tag{20}$$

Where η_d is the diffuser efficiency and h_3s represents the enthalpy obtained at state 3 under isentropic process through the diffuser [16].

$$P_6 = P\,(h_{6s,}\, s_5) \tag{21}$$

$$x_6 = \frac{1}{(1+\omega)}$$

Separator analysis:

In this study, the liquid-vapor separator is considered 100% efficient.

$$h_1 = h\,(P_{6,}\, x{=}1) \tag{22}$$

$$h_7 = h\,(P_{6,}\, x{=}0) \tag{23}$$

Compressor model analysis**:**

The motive enthalpy at state 2s for isentropic process is given by:

$$h_{2s} = h\,(P_{4,}\, s_1) \tag{24}$$

And assuming isentropic compression process, the actual enthalpy at state 2 becomes:

$$h_2 = h_1 + \frac{h2s - h1}{\eta_{comp}} \quad \text{as used by [11]} \tag{25}$$

Where η_{comp} refers to the isentropic efficiency of the compression process (1-2) as cited in [20]

$$\eta_{comp} = 0.874\text{-}0.0135\left(\frac{Pc}{Pe}\right) \tag{26}$$

Throttle valve analysis:

$$h_8 = h_7 \tag{27}$$

The rate of work of the compressor is expressed by:

$$W_{comp.m\_dot} = m_{p\_do}\,(h_2 - h_1) \tag{28}$$

The refrigeration capacity, Q_{evap}, is calculated as:

$$Q_{evap.m\_dot} = m_s\_dot\,(h_9 - h_8) \tag{29}$$

The Coefficient of Performance (COP) of the ejector refrigeration cycle is defined as:

$$COP = Q_{evap.ms\_dot}/W_{comp.mp\_dot} \tag{30}$$

$$\text{Root mean square (rms)} = \frac{n(\varepsilon xy-(\varepsilon x)(\varepsilon y))}{\sqrt{[n\varepsilon x^2-(\varepsilon x^2)][n\varepsilon y^2-(\varepsilon y)^2]}} \tag{31}$$

The data collection methodology is summarized in figure 5.

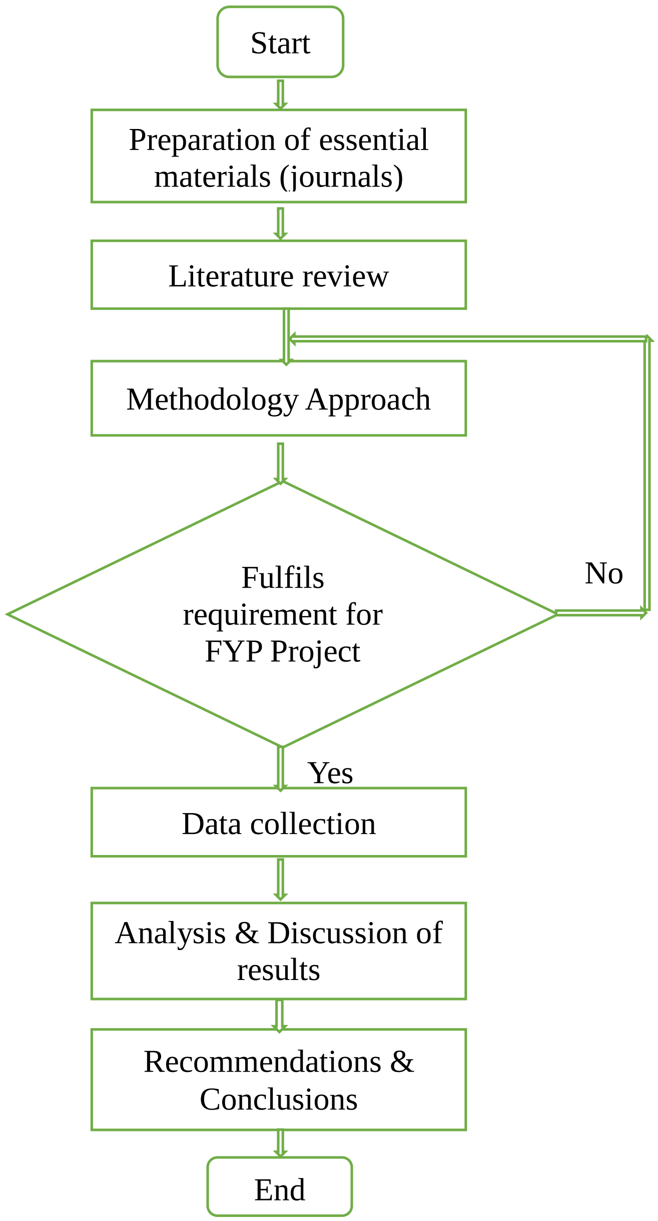

Figure 6: Summary of the project methodology

3.4 **Key Milestone:**

All the project activities were broken down as outlined in table 3.1 considering the following key milestones.

- Ejector Model:
- Refrigeration system model:
- Validation:

Table 3.1 Gantt chart: Timeline for FYP 1

No	Work/Detail	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Project Title selection														
2	Ejector refrigeration Preliminary research														
3	Extended Proposal completion														
4	Proposal defense presentation														
5	Project modelling														
6	Interim Draft report														
10	Interim Report completion														

Key Milestone

Process

Table 3.2 Gantt chart: Timeline for FYP II

No	**Tasks/Week**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**
1	modelling procedure continues														
2	Data gathering & work progress														
3	Progress Report completion														
4	Pre-SEDEX Poster Presentation														
5	Data analysis & report writing														
6	Finish Dissertation Draft & Technical Paper														
7	Submission of Dissertation														
8	Viva presentation														

	Process
	Key Milestone

3.5 Design input conditions

In this study, the ejector refrigeration system was analyzed according to the following input design conditions as in table 3.3. The simulation program was carried out and the simulated results presented in chapter 4.

Table 3.3: Design input conditions of the refrigeration system [10]

Parameters	Temperature (ºC)	Pressure (bar)
Compressor	80	7.7
Condenser	30	7.7
Evaporator	-5	1.6

Parameters	Specifications
Ejector area ratio	14
Ejector diffuser efficiency	0.85
Nozzle efficiency	0.90
Mixing chamber efficiency	0.85
Entrainment ratio	0.53
Refrigerant	R 134a
Primary mass flow rate	0.005773 kg/s
Secondary mass flow rate	0.01089 kg/s

Refrigerant R-134a was used in this project because of its zero Ozone Depletion Potential (ODP=0), relatively good thermophysical properties and readily availability regardless of its DWP=1300).

3.6 Validation

In this section, the results obtained were compared with K. Ganesh Babu and K. Ravi Kumar [21]. The mathematical model was developed based on 1-D dimensional analysis and the calculations were solved with EES to obtain the COP and analyze the parameters of the ejector refrigeration cycle. To validate the model, the results obtained were compared with [21] conducted using R134a refrigerant. K. Ganesh Babu and K. Ravi kumar input design conditions include: evaporator temperature (T_e) = -15^{o}C, condenser temperature (T_c) = 30^{o}C, Diffuser efficiency (η_d) = 85%, nozzle efficiency (η_n) = 85%, area ratio (AR) = 14, and entrainment ratio (ω) = 0.53 with evaporator pressure = 1.64 bar, condenser pressure = 7.7 bar and mixing camber pressure = 2.794 bar. These conditions correspond with the present model. In their study, 4.649 COP of the ejector refrigeration cycle and 3.733 COP of conventional refrigeration cycle were obtained respectively. The percentage enhancement of refrigeration cycle was 19.7% relative to conventional refrigeration cycle. Using empirical approach, root mean square (rms= 0.96) equation (31), relative to K. Ganesh babu and K. Ravi Kumar. The figures 3.4 and 3.5 illustrate COP obtained by K. Ganesh Babu and K. Ravi study and that achieved by the present study.

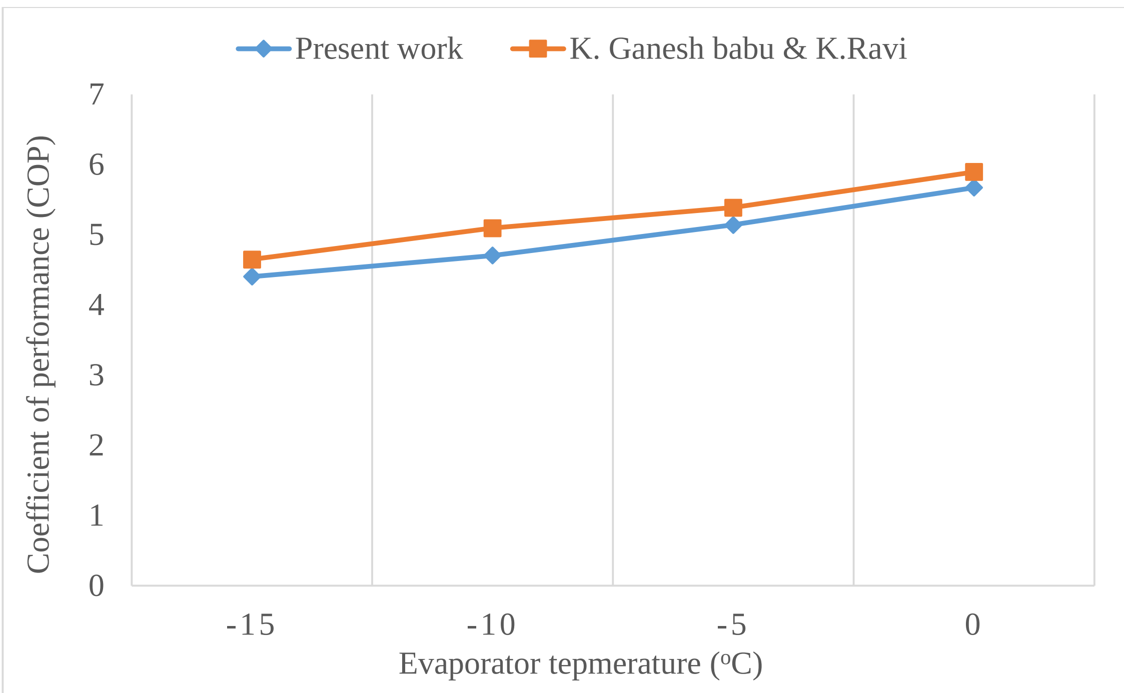

Figure 3.4: COP results for both K. Ganesh Babu & K. Ravi and the present work with respect to evaporator temperature

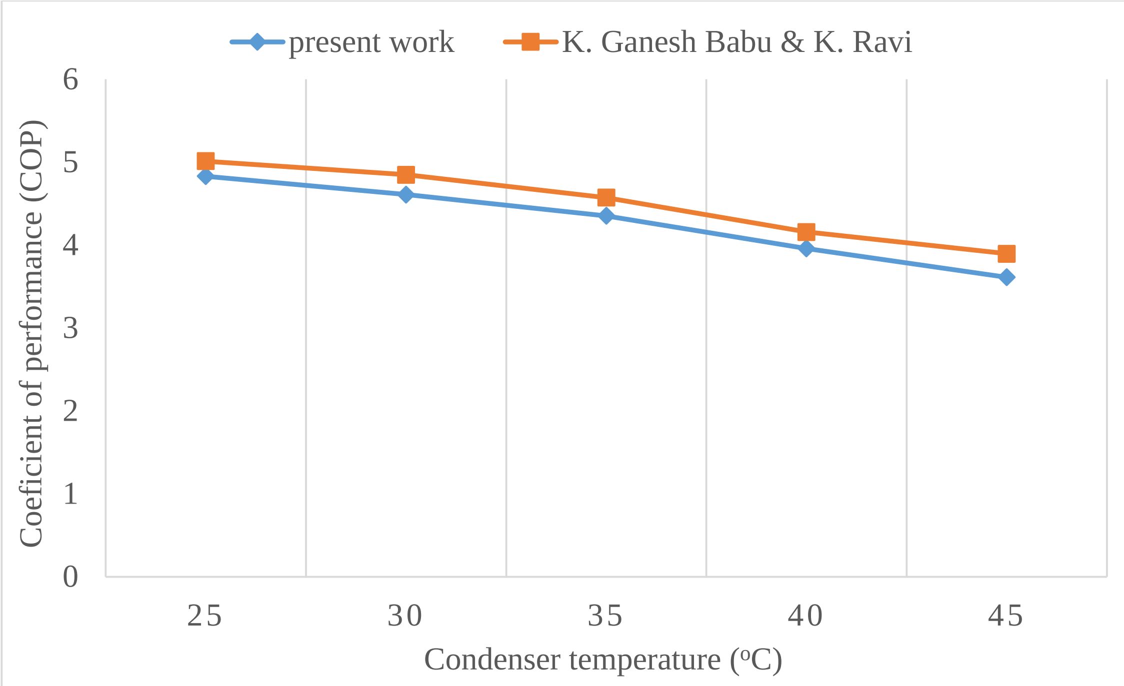

Figure 3.5: COP results for both K. Ganesh Babu & K. Ravi study and the present work with respect to condenser temperatures

Nevertheless [17], [18] and [19] demonstrated a relatively a lower COP at critical pressure of 7.3kPa. On the same note, the present study is also compared to [4] and [11] as illustrated in figures 2.1 and 2.4. It can be seen that this study corresponds well with the previous literatures as discussed above, hence, the validity of the mathematical model is confirmed.

CHAPTER 4: RESULTS AND DISCUSSION

4.1 Introduction:

This chapter presents the results obtained for ejector refrigeration cycle following the stated methodology and the design input conditions as in table 3.3 extracted from literature [10] and [21]. The parameters such as evaporator temperature, condenser pressure, compressor temperature, and nozzle and diffuser efficiencies were analyzed. From the analysis, their effect on COP was deduced. This project was compared with K. Ganesh and K. Ravi [21] study to ascertain the validation of the model. The study was conducted with evaporator temperature (-15^{o}C to 15^{o}C), condenser temperature range (25^{o}C-50^{o}C), compressor temperature (60^{o}C-100^{o}C) and refrigerant R134a was used. The results obtained are tabulated in the tables 4.1, 4.2, 4.3, 4.4, and 4.5. The calculations were obtained using Engineering Equation Solver (EES) because EES contains thermodynamic properties for all the refrigerants built in it.

4.2 Comparison of Ejector Analyses (Constant area method and constant pressure method)

This study adopted the constant area method for the analysis since it is mostly used by researchers than the constant pressure [4]. It's clearly seen in figure 4.1 that both constant area and constant pressure increases with increasing evaporator temperature as evaporator affects positively on entrainment ratio as such better ejector performance and entrainment ratio was greatly obtained using the constant pressure method than the constant area method which corresponds to the statement in reference [13]. However, the ejector performance also shows constant area method having better results than the constant pressure method when design conditions were varied as illustrated in figures 4.1, 4.2 and 4.3. Unlike figures 4.2 and 4.3; both methods, constant area and constant pressure increase with decreasing condenser temperature and vice versa. This trend acts positively on ejector performance since it increases the entrainment ratio. On the other hand, entrainment ratio and COP decrease as the condenser temperature increases per COP expression (equation 31) and the figures 4.2 and 4.5.

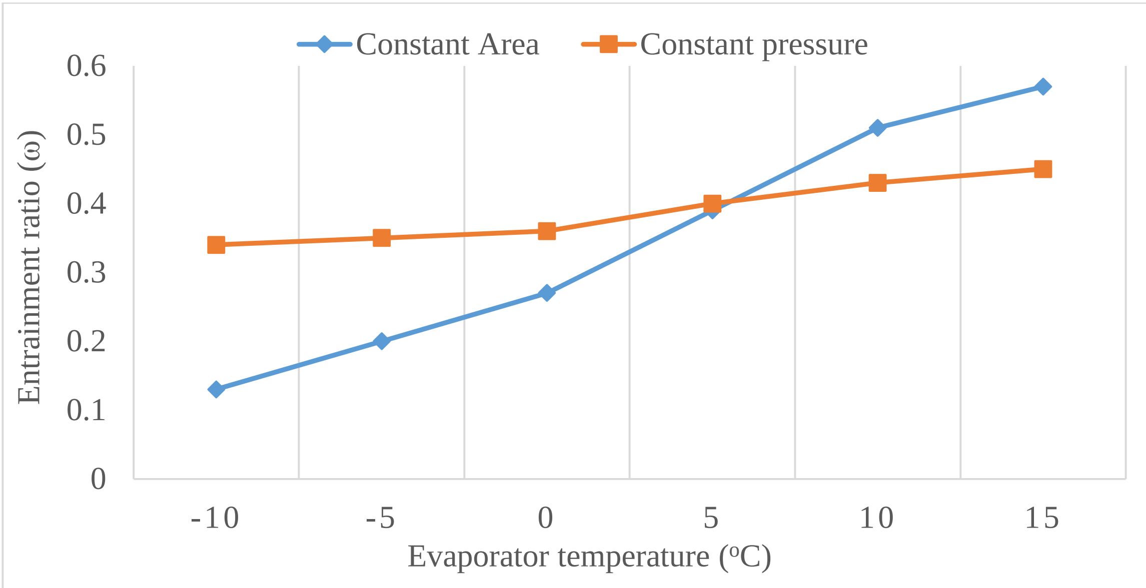

Figure 4.1: Shows correlation of the constant area ratio and constant pressure with respect to entrainment ratio when generator temperature is varied.

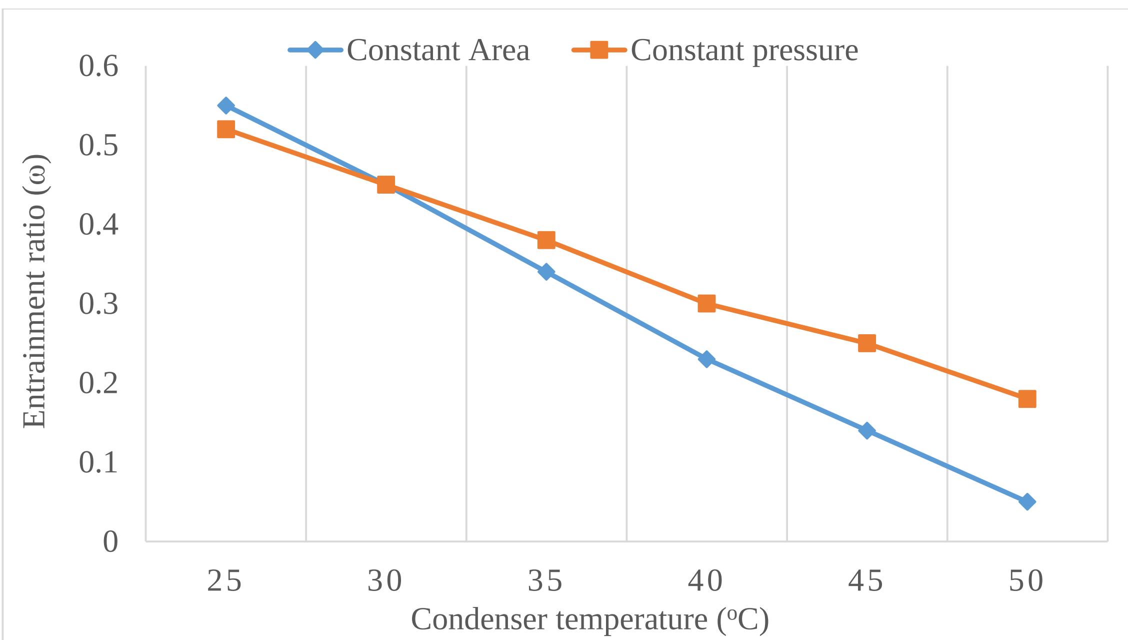

Figure 4.2: Shows relationship of the constant area ratio method and constant pressure

method with respect to entrainment ratio when condenser temperature is varied.

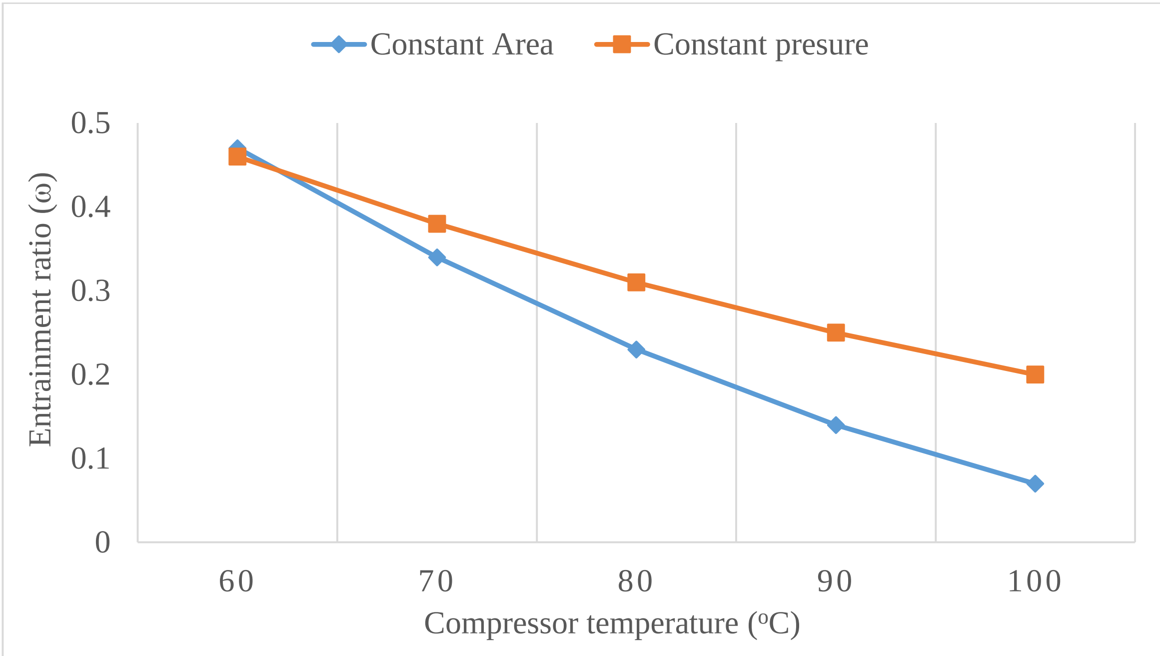

Figure 4.3: Shows correlation of the constant area ratio and constant pressure with respect to entrainment ratio when compressor temperature is varied.

4.3 Parametric Analysis to enhance ejector design

In ejector refrigeration system design, there are three crucial design parameters to consider; the temperatures of the evaporator, condenser and the Compressor (generator). This section presents how these parameters affect the system performance. The study was simulated by varying one of the parameters while the other two remained fix on the input design conditions described in table 3.3 and the results were obtained by the means of the simulation program (EES). The results are presented in tables 4.1, 4.2, 4.3 and the figures 4.4, 4.5 and 4.6 respectively.

Table 4.1: Variation of Evaporator Temperature, T_e (T_{com}=80°C, T_c=30°C)

Condenser temperature (°C)	Evaporator temperature (°C)	Compressor Temperature (°C)	COP of CRC	COP of ERC

30	-15	80	3.868	4.403
30	-10	80	4.128	4.704
30	**-5**	**80**	**4.609**	**5.141**
30	0	80	5.064	5.572
30	5	80	5.599	5.945
30	10	80	5.875	6.24
30	15	80	6.157	6.552

Table 4.2: Variation of Condenser Temperature, T_c; (T_e= -5°C, T_{com}=80°C)

Condenser temperature (°C)	Evaporator temperature (°C)	Compressor temperature (°C)	COP of CRC	COP of ERC
25	-5	80	4.898	5.332
30	**-5**	**80**	**4.609**	**5.141**
35	-5	80	4.44	4.852
40	-5	80	4.266	4.559
45	-5	80	4.135	4.312
50	-5	80	4.003	4.465

Table 4.3: Variation of Compressor Temperature, T_{com}; (T_e= -5°C, T_c= 30°C)

Condenser Temperature (°C)	Evaporator Temperature (°C)	Compressor Temperature (°C)	COP of CRC	COP of ERC
30	-5	60	4.942	5.513
30	-5	70	4.759	5.308
30	**-5**	**80**	**4.609**	**5.141**
30	-5	90	3.956	4.367
30	-5	100	3.753	4.141

4.3.1 The effect of evaporator temperature on the COP

The figure 4.4 shows the effect of evaporator temperature ($T_{e)}$ on COP of the ejector refrigeration cycle at constant condenser temperature (T_c=30^{o}C) and compressor temperature (T_{com}=80^{o}c) respectively. The COP increases with increase in evaporator temperature and the mass flow rate. On the other hand, this phenomena could also be explained in terms of the pressure difference between the nozzle exit and the evaporator (P_9-P_4). The pressure difference increases as evaporator temperature increases which also increases the refrigeration effect (Q_{evap}) and reduces compressor work (W_c). Evaporator temperature reduces irreversibility due to the reduction of the work input into the compressor and the heat rejection from the condenser.

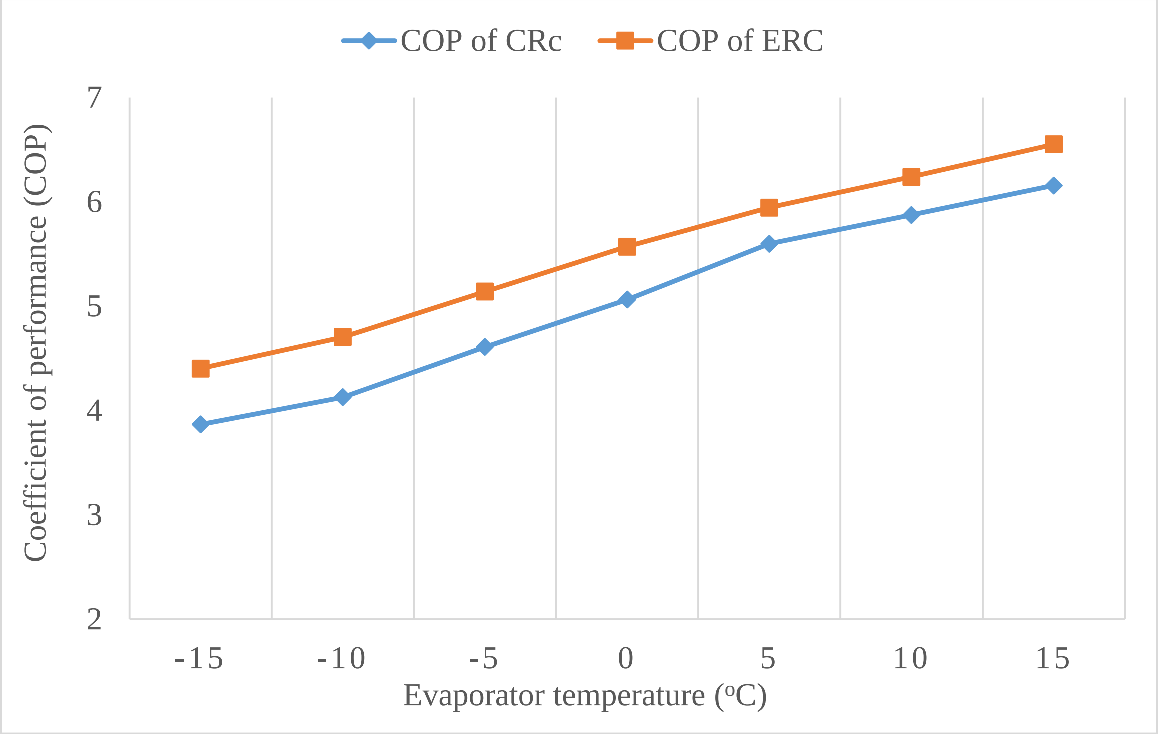

Figure 4.4: Comparison of COP with variation in Evaporator temperature (^{o} C)

4.3.2 The effect of condenser temperature on the COP

Figure 4.5 represents a trend of COP with respect to condenser temperature for both conventional refrigeration cycle and ejector refrigeration cycle. It is clear that COP decreases with increasing condenser temperature because increase in condenser temperature increases the enthalpy of refrigerant at the inlet to the evaporator meanwhile the evaporator enthalpy remains constant at a constant evaporator temperature hence causing low entrainment ratio and COP.

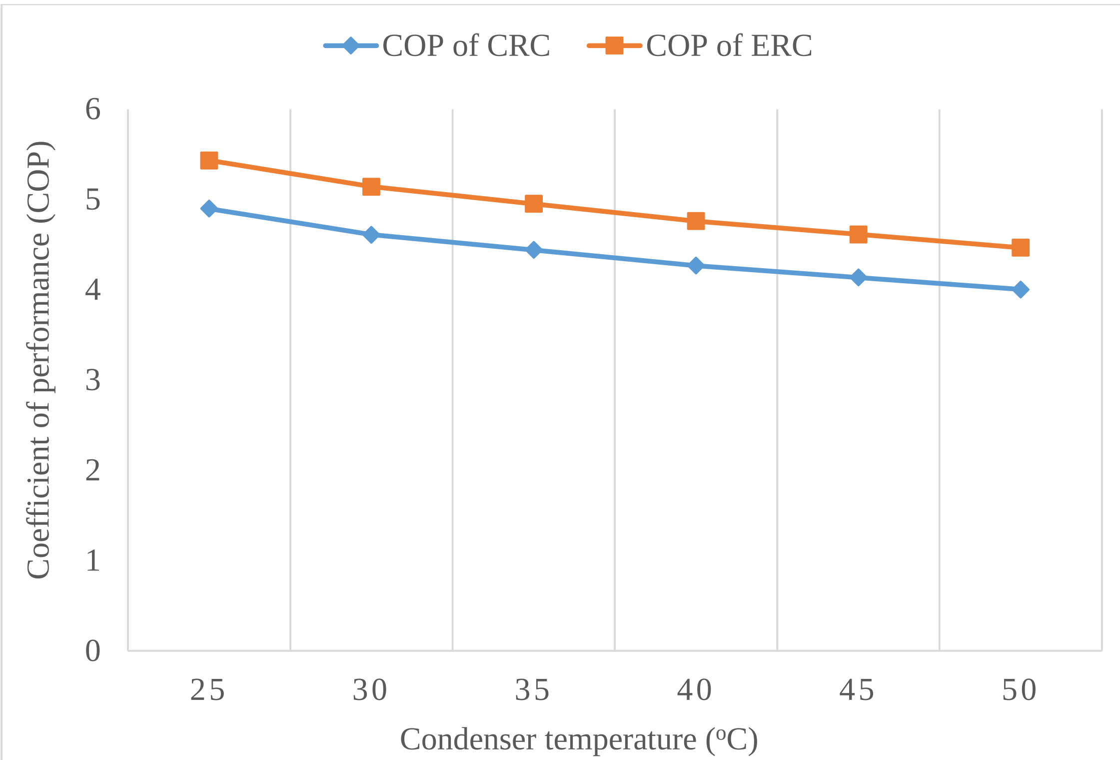

Figure 4.5: Comparison of COP by changing condenser temperature (° C)

4.3.3 The effect of compressor temperature on the COP

Figure 4.6 depicts variation of compressor temperature (T_{com}) from 60°C to 100°C at constant evaporator and condenser temperatures. From figure 4.6, the COP decreases with increasing compressor temperatures. As the compressor temperature increases, pressure and enthalpy differences between the compressor inlet and outlet increases. Mass rate in the compressor and the compressor work as well increase causing COP decrement per equation 31.

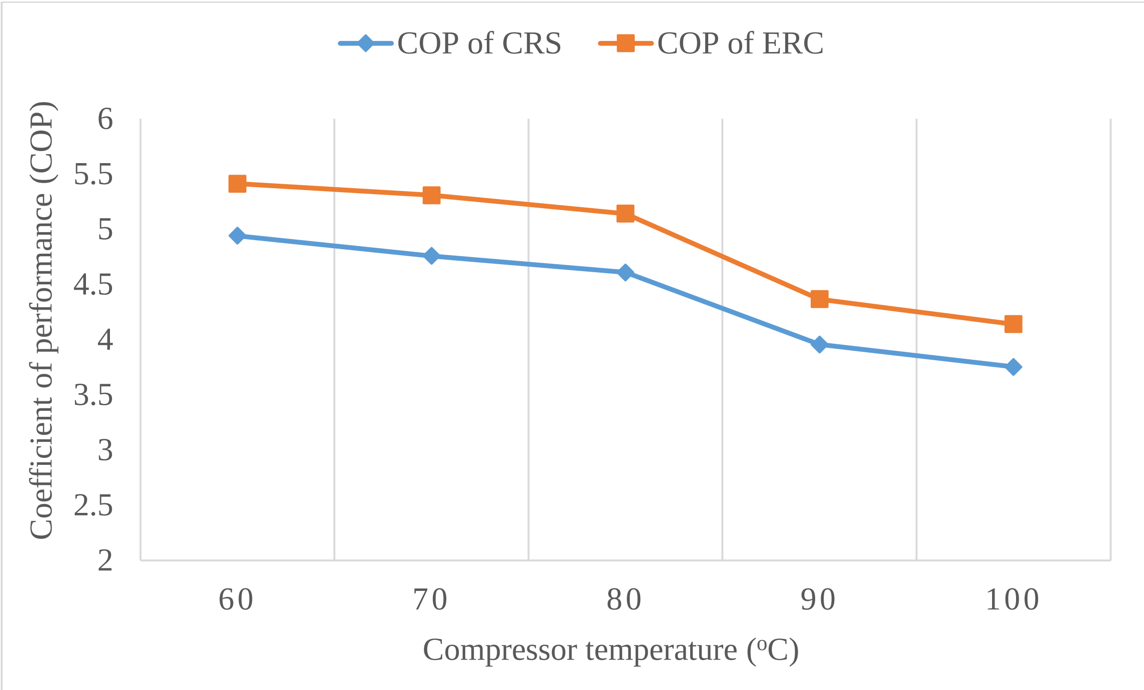

Figure 4.6: Comparison of COP with variation in compressor temperature (° C)

4.4 The entrainment ratio (ω) and refrigeration capacity (Q_{evap}) analysis on COP

Figure 4.7 shows COP with respect to entrainment ratio and it indicates that COP increases with increase in entrainment ratio. Increase in entrainment ratio means that the mass flow rate from the evaporator is also increasing. Increasing mass flow rate in the evaporator automatically increase the cooling capacity as well the coefficient of performance (COP) increases as in figure 4.8.

Meanwhile figure 4.8 shows COP increment with refrigeration capacity increase because increase in pressure difference reduces the compressor work. Reduction of compressor work increases COP as illustrated by equation (31). However, compressor work reduction also increases refrigeration capacity and as explained entrainment ratio also increased since COP and entrainment ratio are directly proportional. Thus, refrigeration capacity increases with increase in entrainment ratio figure 4.8.

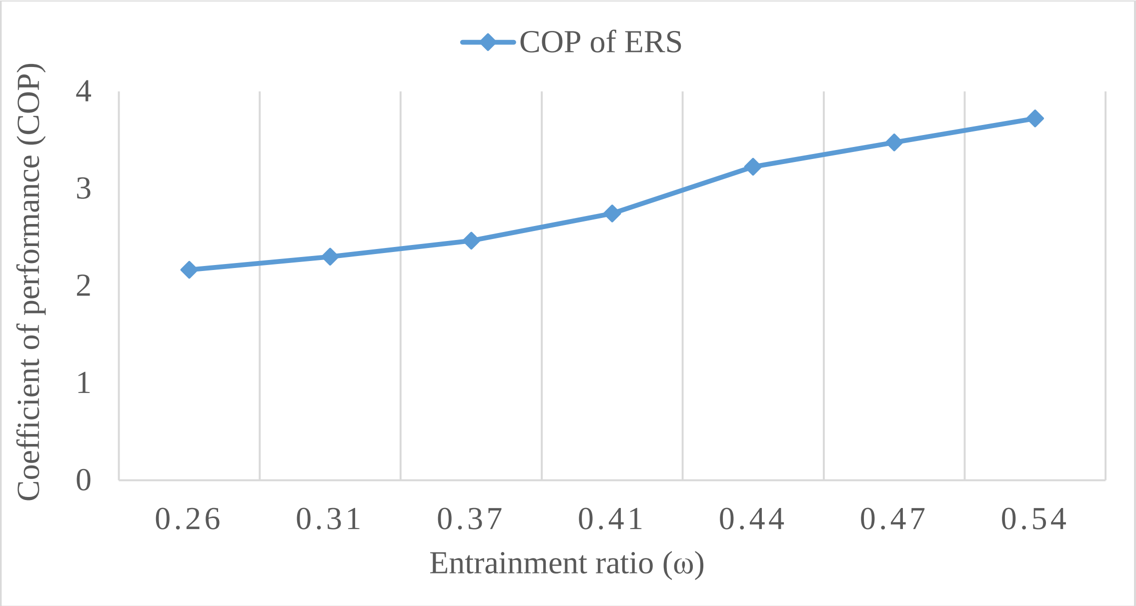

Figure 4.7: Graph of COP against entrainment ration (ω)

T_e = -5^{o}C, P_{evap} = 1.6 bar, T_c = 30^{o}C and P_c = 7.7 bar

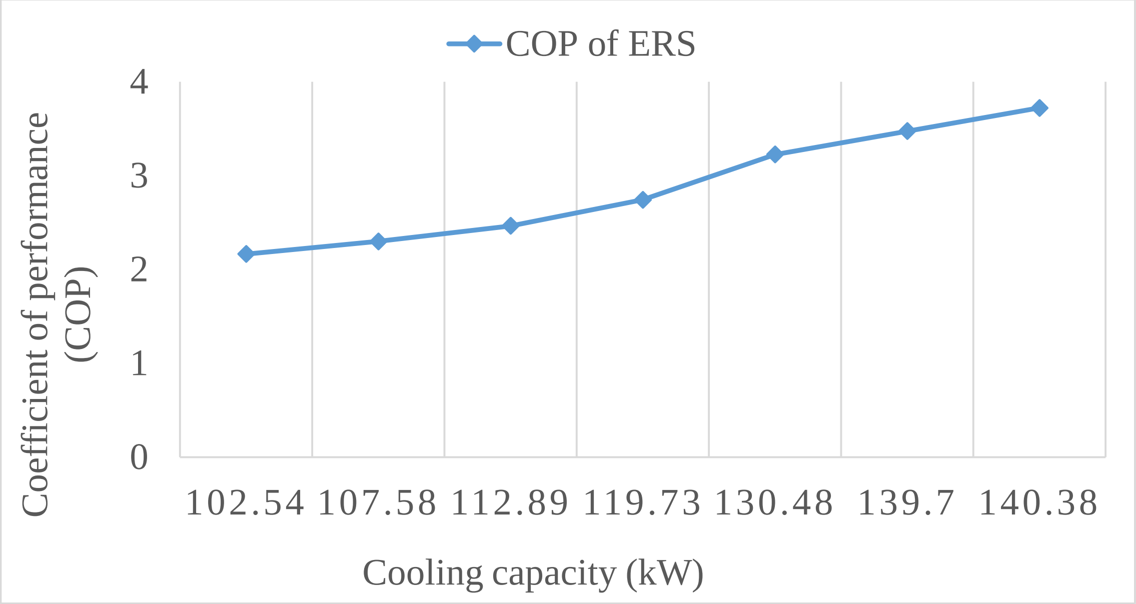

Figure 4.8: Graph of COP against refrigeration capacity

T_e = -5°C, P_{evap} = 1.6 bar, T_c = 30°C and P_c = 7.7 bar

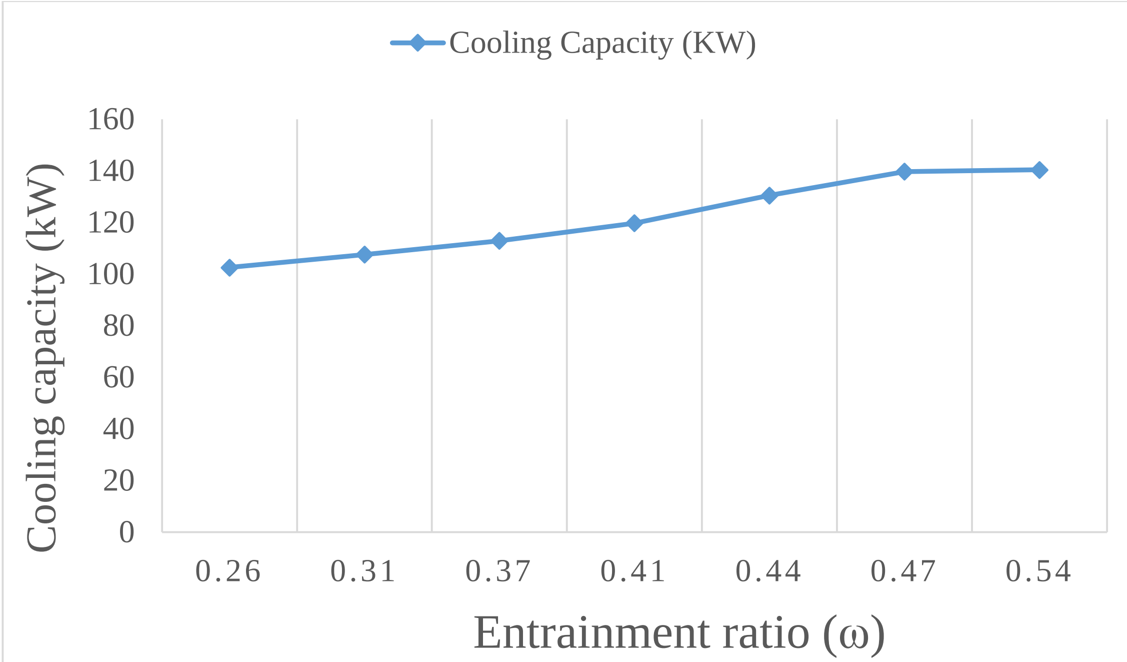

Figure 4.9: Graph of cooling capacity with respect to entrainment ratio (ω)

T_e = -5°C, P_{evap} = 1.6 bar, T_c = 30°C and P_c = 7.7 bar

4.5 Sensitivity analysis on the effect of Nozzle and Diffuser Efficiencies

Besides temperatures affecting COP, other design parameters taken into consideration were: the nozzle efficiency and the diffuser efficiency. In this analysis, it is shown how the nozzle and the diffuser affect the ejector and the COP sensitive parameters that also affect COP are the diffuser and nozzle efficiencies. The one efficiency was varied while the other one was remained fix. The other parameters also being fixed in the input design conditions as in table 3.3. The results are shown in tables 4.4, 4.5 and 4.6 and the figures 4.10, 4.11 and 4.12 respectively.

Table 4.4: Sensitivity analysis of Diffuser efficiency effect on COP and Compressor work

Diffuser Efficiency	Compressor work	COP of ERC
10	25.4	5.27
20	25.22	5.32
30	25.06	5.367
40	24.94	5.419
50	24.7	5.471
60	24.548	5.54
70	24.28	5.573
80	24.09	5.632
90	23.95	5.689
100	23.75	5.73

Table 4.5: Sensitivity analysis of Nozzle efficiency effect on the COP

Nozzle efficiency (η_n)	COP_{ERC}
10	78.64
20	38.87
30	24.68
40	20.42
50	16.08
60	12.86
70	10.32
80	7.86
90	6.6
100	5.46

4.5.1 The effect of nozzle efficiency on the COP

The effect of the nozzle efficiency was examined by varying its efficiency from 0.1 to 1.0 with 0.85 diffuser efficiency. From the table 4.5, it can be noted that increase in nozzle efficiency reduces the COP for the reason that there is reduction of pressure at the nozzle exit which in turn also increases the pressure ratio of the compressor. Thus, increase in compressor pressure ratio increases the compressor work which as a result decreases the COP of the system. For system enhancement, the nozzle efficiency should be less than or equal to the diffuser efficiency.

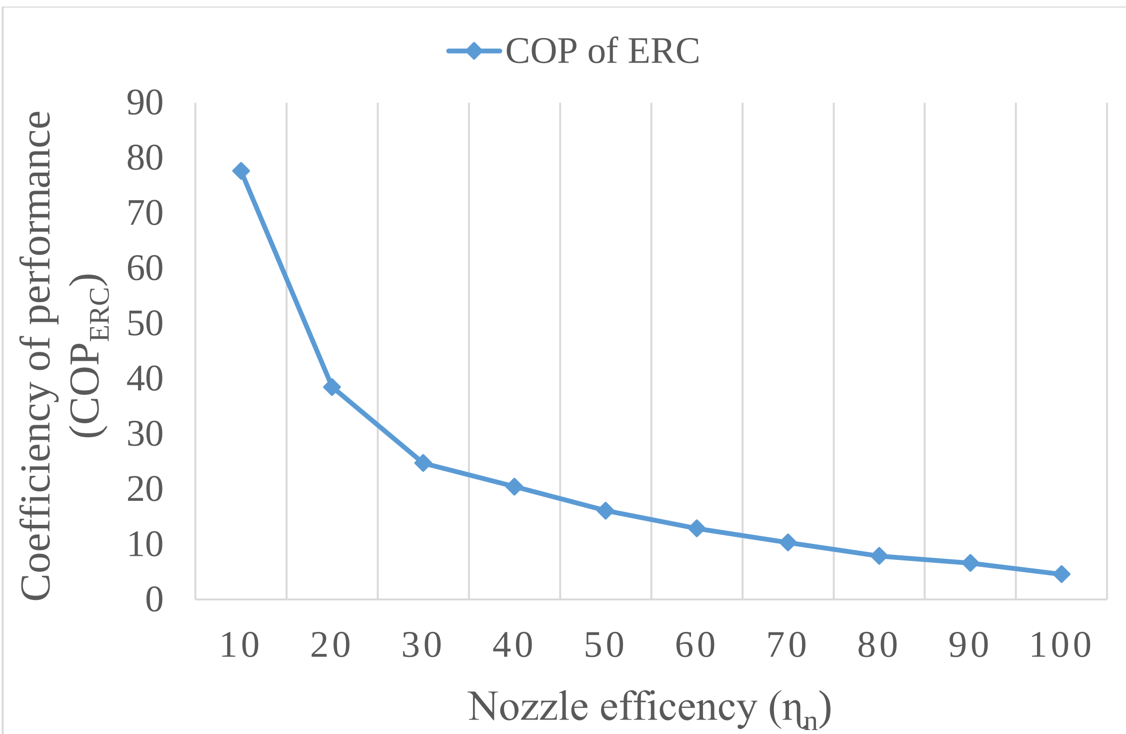

Figure 4.10: Graph of nozzle efficiency against Coefficient of performance (COP) at T_e = -5°C, T_c= 30°C, T_c= 80°C and η_d = 85%

4.5.2 The effect of Diffuser efficiency on the COP

Figure 4.10 indicates that increment in diffuser efficiency reduces the compressor work because increase in diffuser efficiency increases the pressure at the outlet of diffuser which decreases the pressure ratio of the compressor. The Compressor pressure ratio also decreases the compressor work. Figure 4.11, graph of COP against diffuser efficiency and it's clearly seen that reduction in compressor work substantially increases the COP (equation 31).

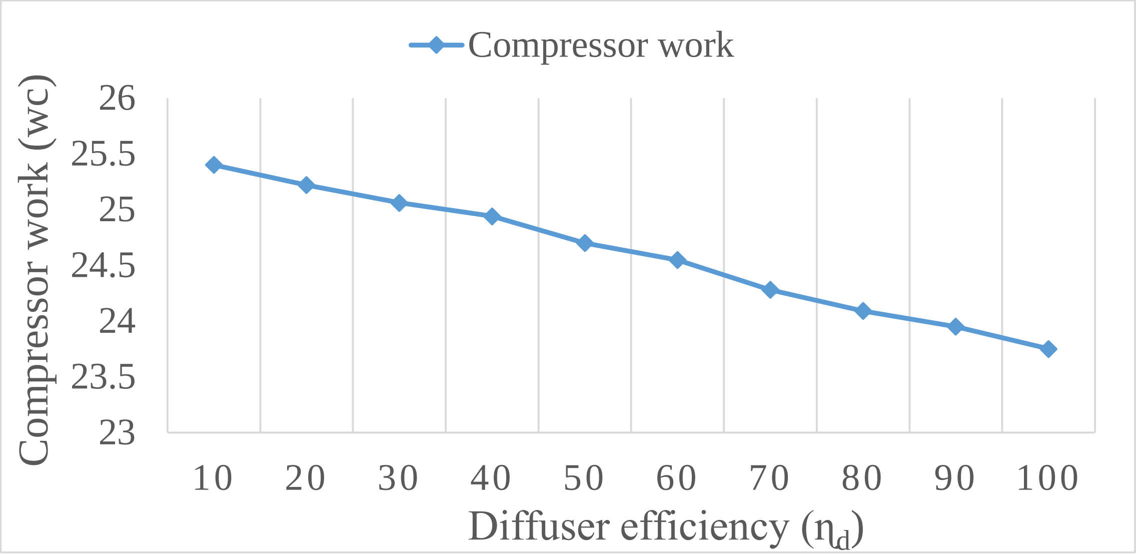

Figure 4.11: Graph of diffuser efficiency (η_d) against compressor work (W_C)

(T_e = -5ºC, T_c= 30ºC, T_c= 80ºC and η_n = 90%)

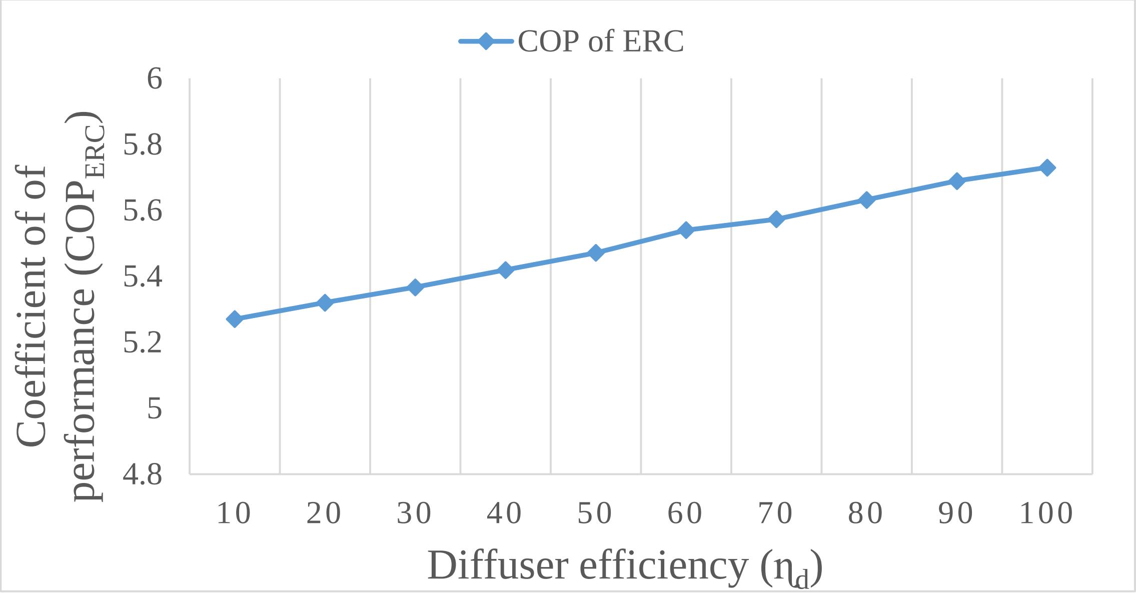

Figure 4.12: Graph of diffuser efficiency against Coefficient of performance (COP)

(T_e = -5ºC, T_c= 30ºC, T_c= 80ºC and η_n = 90%)

4.6 Coefficient of performance (COP) Calculations for both conventional and Ejector refrigeration cycles:

Coefficient of performance (COP_{CRC}) = m-dot*(h [4]-h [1])/m-dot* (h [1]-h [4])

COP_{CRC} = 4.609

Coefficient of performance (COP_{ERC}) = Q_{evap}.m_dot/$Q_{comp.}$m_dot

= ω ((h$_9$ – h$_8$)/ (h$_2$ – h$_1$))

$COP_{\mathbf{ERC}}$ = 5.141

Percentage improvement of refrigeration system with an ejector is expressed as

Percentage improvement = (COP_{ERC} - COP_{CRC})/ COP_{ERC}

= ((5.141-4.609)/5.141) * 100

= 10.35%

The COP increase of the ejector refrigeration cycle over the conventional refrigeration cycle is 10.35%.

CHAPTER 5: CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

The model to enhance ejector refrigeration performance was developed using the input design conditions in table 3.3. With the above input condition, 4.609 COP_{CRC} of conventional refrigeration cycle and 5.141 COP_{ERC} of ejector refrigeration cycle were obtained representing 10.35% improvement of the refrigeration performance by an ejector. The 10.35% performance enhancement represents potential saving in terms of energy and cost. This verifies the objective of this study which was to study the enhancement of refrigeration performance with an ejector compare to conventional refrigeration cycle.

From the parametric analysis of ejector refrigeration cycle, the following parameters: temperatures of evaporator, condenser and compressor including nozzle efficiency, diffuser efficiency and the entrainment ratio were evaluated. It was noted that the COP depends largely on evaporator, condenser and compressor temperatures (figures 4.4, 4.5 and 4.6). It also depends on entrainment ratio (figures 4.7, 4.8 and 4.9) as well as nozzle efficiency and diffuser efficiency (figures 4.10, 4.11 and 4.12) and also depends on the ejector design and working fluid. This fulfils the second objective of this study which was to carry out parametric analysis of refrigeration cycle with an ejector.

5.2 Recommendations

1. The similar performance analysis of the ejector refrigeration system with different refrigerants should be conducted and performance comparison are made to ascertain practicability of use of the ejector technology.

2. An experimental validation of the methodology should be considered to carry out some tests on ejector components to study its behavior experimentally and compared to its modelled behavior.

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APPENDIX A:

A. 1: CONVENTIONAL REFRIGERATION CYCLE EES SIMULATION PROGRAM

```
"Conventional Refrigeration cycle Input data"
T[1]=15
T[3]=30
T[2]=80
P[1]=1.64
P[2]=7.706
comEff=0.8
m_dot=0.01089

"Evaporator"
P[1]=P[4]
h[1]=Enthalpy(R134a,T=T[1],x=1)
Q_evap=m_dot*(h 1]-h 4])

"Condenser"
P[3]=P[2]
H [3]= Enthalpy(R134a,T=T[3],x=0)
Q_cond=m_dot*(h [2]-h [3])

"Valve"
H [4]=h[3]
x4=quality(R134a, h=h[4], P=P[4])

"Compressor"
x[1]=1
h2s=Enthalpy (R134a, T=T [2], P=P [2])
s[1]=Entropy(R134a, T=T[1], x=x[1])
comEff=(h2s-h[1])/(h[2]-h[1])
w_com=m_dot*(h [2]-h [1])

COP=abs (Q_evap/w_com)
```

A. 2 EQUATIONS IN THE MAIN PROGRAM

There are a total of 21 equations in the Main program.

Block	Rel. Res.	Abs. Res.	Units	Calls	Time(ms)	Equations
0	0.000E+00	0.000E+00	OK	1	0	**T[1]**=-5
0	0.000E+00	0.000E+00	OK	1	0	**T[3]**=30
0	0.000E+00	0.000E+00	OK	1	0	**T[2]**=80
0	0.000E+00	0.000E+00	OK	1	0	**P[1]**=1.64
0	0.000E+00	0.000E+00	OK	1	0	**P[2]**=7.706
0	0.000E+00	0.000E+00	OK	1	0	**comEff**=0.8
0	0.000E+00	0.000E+00	OK	1	0	**m_dot**=0.01089
0	0.000E+00	0.000E+00	OK	1	0	**x[1]**=1
0	0.000E+00	0.000E+00	OK	4	0	P[1]= **P[4]**
0	0.000E+00	0.000E+00	OK	4	0	**h[1]**=Enthalpy(R134a,T=T[1],x=1)
0	0.000E+00	0.000E+00	OK	4	0	**P[3]**=P[2]
0	0.000E+00	0.000E+00	OK	4	0	**h[3]**=Enthalpy(R134a,T=T[3],x=0)
0	0.000E+00	0.000E+00	OK	4	0	**h[4]**=h[3]
0	0.000E+00	0.000E+00	OK	4	0	**x4**=quality(R134a,h=h[4],P=P[4])
0	0.000E+00	0.000E+00	OK	4	0	**h2s**=Enthalpy(R134a,T=T[2],P=P[2])
0	0.000E+00	0.000E+00	OK	4	0	**s[1]**=Entropy(R134a,T=T[1],x=x[1])
0	0.000E+00	0.000E+00	OK	4	0	comEff=(h2s-h[1])/(**h[2]**-h[1])
0	0.000E+00	0.000E+00	OK	4	0	**w_com**=m_dot*(h[2]-h[1])
0	0.000E+00	0.000E+00	OK	4	0	**Q_evap**=m_dot*(h[1]-h[4])
0	0.000E+00	0.000E+00	OK	4	0	**Q_cond**=m_dot*(h[2]-h[3])
0	0.000E+00	0.000E+00	OK	4	0	**COP**=abs(Q_evap/w_com)

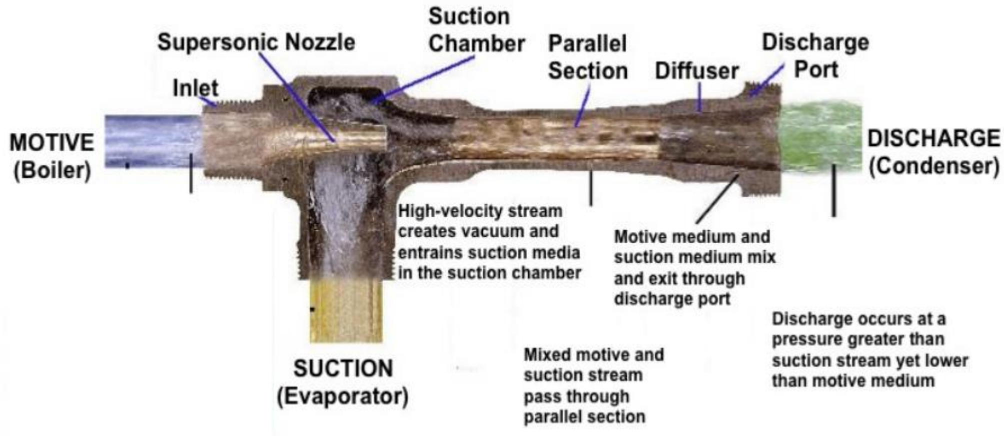
Variables shown in bold font are determined by the equation(s) in each block.

A. 3 EJECTOR REFRIGERATION CYCLE EES SIMULATION PROGRAM

```
"Input Design Conditions"
T1=-5
T2=80
T3=30
P9=1.64
diff_P=0.5
P3=7.7
u=0.53
AR=14
eff-mn=0.90
eff_d=0.85
eff-sn=0.85
eff_ms=0.85
A4=2.56
P10=P4

"At The condenser outlet"
P3=Pc
h3=Enthalpy(R134a, T=T3, x=0)
s3=Entropy(R134a, T=T3, x=0)

"At Evaporator Outlet"
P9=Pe
Te=T1
h9=Enthalpy(R134a, T=T1, x=1)
s9=Entropy(R134a, T=T1, x=1)

"At the nozzle Outlet"
P4=P9-diff_P
h4s=Enthalpy(R134a, P=P4,s=s3)
h4=h3-eff_mn*(h3-h4s)
c4=(2*(h3-h4))^0.5
A4= 1/(1+u)*d4*c4

"At the suction nozzle Outlet"
d4=d10
P10=P9-diff_P
h10s=Enthalpy(R134a, P=P10, s=s9)
h10=h9-eff-sn*(h9-h10s)
c10=(2*(h9-h10))^0.5
A10=u/(1+u)*d10*c10
```

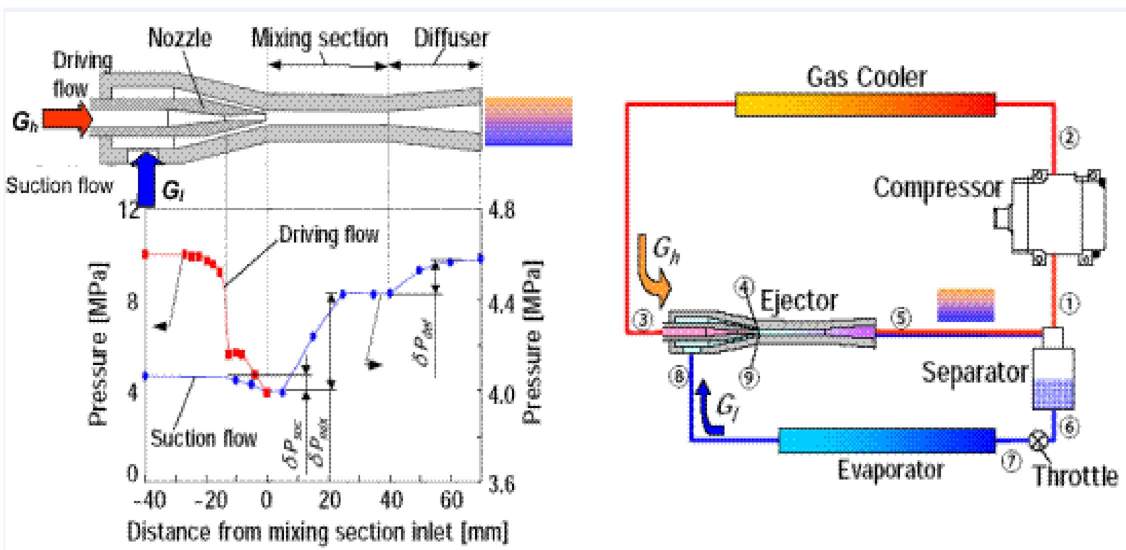
```
"At the mixing section"
P5=P4
c5=eff_ms^0.5*(c4/(1+u)+(u*c10^2)/(1+u))
h5=(h4+c4^2/2)/(1+u)+u*(h10+c10^2/2)/(1+u)-c5^2/2
s5=Entropy(R134a, P=P5, h=h5)

"At the diffuser outlet"
h6=h5+(c5^2)/2
h6s=h5+eff_d*(h6-h5)
P6=Pressure(R134a, h=h6s, s=s5)
x6=1/(1+u)
AR=(A4+A10)/A4

"At the separator outlets"
h1=Enthalpy(R134a, P=P6, x=1)
h7=Enthalpy(R134a, P=P6, x=0)

"At the compressor outlet"
P2=P3
h2s=Enthalpy(R134a, P=P2, T=T2)
h2=h1+(h2s-h1)/eff_com
eff_com=0.874-0.0135*(P3/P9)

"At the throtle valve outlet"
h8=h7
W_comp=u*(h2-h1)/(1+u)
Q_evap=u8*(h9-h8)/(1+u)
COP=Q_evap/W_comp
```

B. 1 EJECTOR MODE OF OPERATIONS



Principles of Ejector operation (curtesy of Penberthy)

C. 2 VELOCITY AND PRESSURE DISTRIBUTION IN EJECTOR REFRIGERATION CYCLE



Pressure distribution within the ejector (Curtesy of DENSO)

D. 2 THERMOPHYSICAL PROPERTIES OF REFRIGERANT R134A

General:	
Name	1,1,1,2-tetrafluoroethane
Formula	CH_2F-CF_3
Main application fields (in compliance with the legislation in force)	domestic refrigeration; refrigerated transport; air conditioning
Molar mass	102,0 kg/kmol

Thermophysical properties:	
Normal boiling point (at 0.1013 MPa)	−26,1°C
Critical temperature	101,1°C
Critical pression	4,06 MPa

Properties at 0°C (at saturation)*			
	Unit (SI)	Liquid	Vapour
Pression	MPa	0,29	0,29
Volume massique	dm^3/kg	0,77	69,31
Specific heat capacity • at constant pressure • at constant volume	 kJ/(kg K) kJ/(kg K)	 1,34 0,88	 0,90 0,76
Viscosity	10^{-6} Pa s	271,08	10,73
Thermal conductivity	W/(m K)	0,092	0,012
Surface tension	N/m	0,012	
Heat of vaporization	kJ/kg	198,6	

* These data are derived from the brochure Thermodynamic and physical Properties of R134a published by the IIR. You can order it on line.

Environmental properties:	
ODP (R11=1)	0
GWP (CO2=1)	1300

The GWP used as a reference here is the GWP of CO2 over an integration period of 100 years.

Safety Group : A1