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Experimental Investigation of Ejector-Expansion Refrigeration System Using Refrigerant R-134a

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Abstract: Integration of two-phase flow of ejector in simple vapor compression refrigeration (VCR) system is the most effective way to improve the coefficient of performance (COP) of the system. In the present paper, the dimensions of two-phase flow ejector are newly defined and applied experimentally. Results are evaluated performance characteristics. For these objectives, experimental set up is fabricated. Published experimental data and present results shows improvement in present results. COP of the system increased by 7.72 % – 16.58 % over basic cycle as evaporation temperature changes from -10^0 C to 10^0 C. And COP increased by 8.8 % - 4.11 % over basic cycle as condensation temperature ranges from 30^0 C-50^0 C.
Keywords: COP, two-phase flow ejector, ejector-expansion refrigeration system

Nomenclature

VCR	Vapor Compression Refrigeration
COP	Coefficient of Performance
EEC	Ejector Expansion Cycle
BC	Basic Cycle
EERS	Ejector Expansion Refrigeration System
RE	Refrigeration Effect
WD	Work Done
M	Mass Flow
η	Isentropic Efficiency
TR	Tonnage of Refrigeration
ω	Entrainment Ratio
h	Specific Enthalpy
u	Velocity
P	Pressure
ρ	Density
a	Area
D	Diameter
L	Length
Subscript	
p	Primary
s	Secondary
tot	Total
mn,e	Motive Nozzle Exit
ss	Suction Section
ms	Mixing Section
dns	Diffuser Nozzle Section

I. INTRODUCTION

Refrigeration systems are more electricity consumption devices in India. As India is developing country the refrigeration devices like refrigerator, air conditioner, heat pumps, freezers, etc are also in developing mode. This research implies to the refrigerator.

Most often we use vapor compression refrigeration system into it and which is very old technique of refrigeration, also it is not so energy efficient.

In simple VCR system, the expansion process is isenthalpic (where enthalpy remains constant) but isenthalpic process reduces the cooling capacity of evaporator due to energy loss in throttling process. And in EERS, with the help of ejector the expansion process converts into isentropic expansion. Such cycle is called as ejector-expansion refrigeration system. Use of two-phase ejector into the simple VCR system is one of the most effective technique to increase COP of the system by increasing cooling capacity and decreasing the work of the compressor [1],[3].

There is classification of two-phase ejector as a) constant pressure and b) constant area ejector [2] Motive nozzle of ejector located within suction nozzle before constant area section known as constant pressure ejector, However the motive nozzle exit plane located within constant area section known as constant area ejector.

The effect of diameter variation of the motive nozzle, on the COP, primary mass flow rate, secondary mass flow rate, compressor pressure ratio, cooling capacity, discharge temperature [4].

Using constant-area ejector, a simulation program was developed and performance of the system investigated using EES software. The system COP increased by 87.5% as evaporation temperature changes from -10^0 C to 10^0 C [9].

A two-phase constant area ejector flow model was used, the improvement ratio in COP rises whereas ejector area ratio drops. The minimum COP improvement ratio in the investigated field was 10.1% while its maximum was 22.34%. [8].

Presents a modelling procedure and numerical approach of liquid-vapor ejectors for refrigeration [7]. Performance comparison of R-1234yf with R134a was observed in the ejector-expansion refrigeration system [6].

II. SYSTEM DESCRIPTION

The system consists of compressor, a condenser, an expansion valve, an evaporator and two phase flow ejector, and separator are inserted to recover the expansion losses and to increase the COP as shown in Figure 1. The saturated vapor from the separator is sucked and compressed to a high pressure. , superheated vapor at discharge of the compressor is condensed by passed through condenser. The high pressure saturated liquid comes out from the condenser is called as primary flow of refrigerant. The liquid refrigerant from separator is expanded in the expansion valve and then passed through evaporator and evaporated by absorbing heat from cooled objects. The vapor from evaporator comes into suction nozzle of ejector is called secondary flow. Primary and secondary mass flow are mixed into the mixing chamber results a mixture of intermediate pressure at the outlet of the ejector and then passed to the separator. Figure 1 shows by pass line indicated by red line which completes simple VCR cycle by reversing the three valves attached to the ejector. And black lines completes the ejector-expansion refrigeration cycle when valves reverses their positions. On the basis of these two cycles, further experimentation is completed with the performance characteristics.

III. SYSTEM ASSUMPTIONS

The following assumptions are made in the system analysis, which is carried out on the basis of mass, energy and momentum conservation equations.

A. All system components are well insulated,
B. Flow is steady state one-dimensional,
C. Two-phase flow mixture are assumed to be in
D. equilibrium flow,
E. All velocities at inlet and outlet are zero i.e. at stagnation conditions,
F. Friction losses are defined in terms of efficiencies,
G. Separator is 100% efficient,
H. No pressure changes in all system components except compressor, an expansion valve and ejector.

IV. OPERATING CONDITIONS

Evaporator temperature ranges from -10^0 C to 10^0 C when condenser temperature fixed at 40^0C.

Condenser temperature increases from 30^0 C to 50^0 C when evaporator temperature constant at 0^0C. And System cooling capacity is of 1TR.

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V. EJECTOR DESIGN

Ejector is designed on the basis of design procedure published in the open literature [8], [9]. Ejector has three important parts such as motive nozzle, the suction nozzle, and the mixing section. Ejector design used in this paper is put special attention on the ejector diameters rather than length. Table I shows the dimensional comparison of ejector-parameters.

VI. MATHEMATICAL MODEL

A. Motive Nozzle

Using definition of motive nozzle's isentropic efficiency (η_{mne}), the enthalpy of fluid at the motive nozzle exit (h_{3a}), followed by expression (1),

$$h_{3a} = h_3 (1 - \eta_{mne}) + \eta_{mne} h_{3a,is} \quad (1)$$

Where, $h_{3a,is}$ = enthalpy of motive nozzle at the end of isentropic expansion and η_{mne} is the isentropic efficiency of the motive nozzle exit.

Applying energy equation to the motive nozzle, the velocity at motive nozzle exit expressed as follows:

$$= \sqrt{2(h_3 - h_{3a})} \quad (2)$$

Using the principle of conservation of mass, the area of motive nozzle section is calculated by following equation,

$$a_{3a} = \frac{m_{wt}}{\rho_{3a} u_{3a} (1+\omega)} \quad (3)$$

Where, m_{wt}= total mass flow

ω = Entrainment Ratio (Ratio of secondary mass flow rate to the primary mass flow rate)

B. Suction Nozzle Section

Applying similar expression applied to the motive nozzle for calculations of the suction nozzle.

$$h_{7a} = (1 - \eta_{ss}) h_7 + \eta_{ss} h_{7a,is} \quad (4)$$

$$u_{7a} = \sqrt{2(h_7 - h_{7a})} \quad (5)$$

$$a_{7a} = \frac{m_{wt}}{\rho_{7a} u_{7a} (1+\omega)} \quad (6)$$

C. Constant Area Mixing Section

using principle of conservation of momentum in a constant mixing section, the velocity of fluid in the mixing section can be calculated by following equation,

$$u_{3m} = P_b(a_{3a} + a_{7a}) + \frac{1}{1+\omega} u_{3a} + \frac{1}{1+\omega} u_{7a} - P_{3m} a_{3m} \quad (7)$$

Where, a_{3m} = Cross section of mixing section

By applying the principle of conservation of energy, enthalpy of the fluid at the exit of the mixing section can be calculated by following equation,

$$h_{3m} = \frac{1}{1+\omega}(h_3 + \omega h_7) + \frac{u^2{}_{3m}}{2} \quad (8)$$

For corrected calculations below expression should be satisfy.

$$a_{3m} u_{3m} \rho_{3m} = m_{wt} \quad (9)$$

D. Diffuser Section

Using the principle of conservation of energy, the enthalpy of the diffuser exit can be calculated by using below expression:

$$h_4 = \frac{h_3 + \omega h_7}{1+\omega} \quad (10)$$

The isentropic enthalpy of the diffuser exit can be calculated from below expression,

$h_{4,is} = \eta_d(h_4 - h_{3m}) + h_{3m}$(11)

Where, η_d = isentropic efficiency at diffuser exit

E. COP and COP Improvement

$Q_{RE} = \frac{m_s}{m_p + m_s}(h_7 - h_6)$(12)

$W.D._{comp} = \frac{m_p}{m_p + m_s}(h_2 - h_1)$ (13)

$COP_{EEC} = \frac{Q_{RE}}{W.D._{comp}}$ (14)

COP Improvement in percentage

$\% \text{ COP} = \frac{COP_{EEC} - COP_{BC}}{COP_{BC}}$ (15)

\Pressure Lift

$P_L = \frac{P_4}{P_7}$ (16)

Entrainment Ratio (ω):

$\omega = \frac{m_s}{m_p}$ (17)

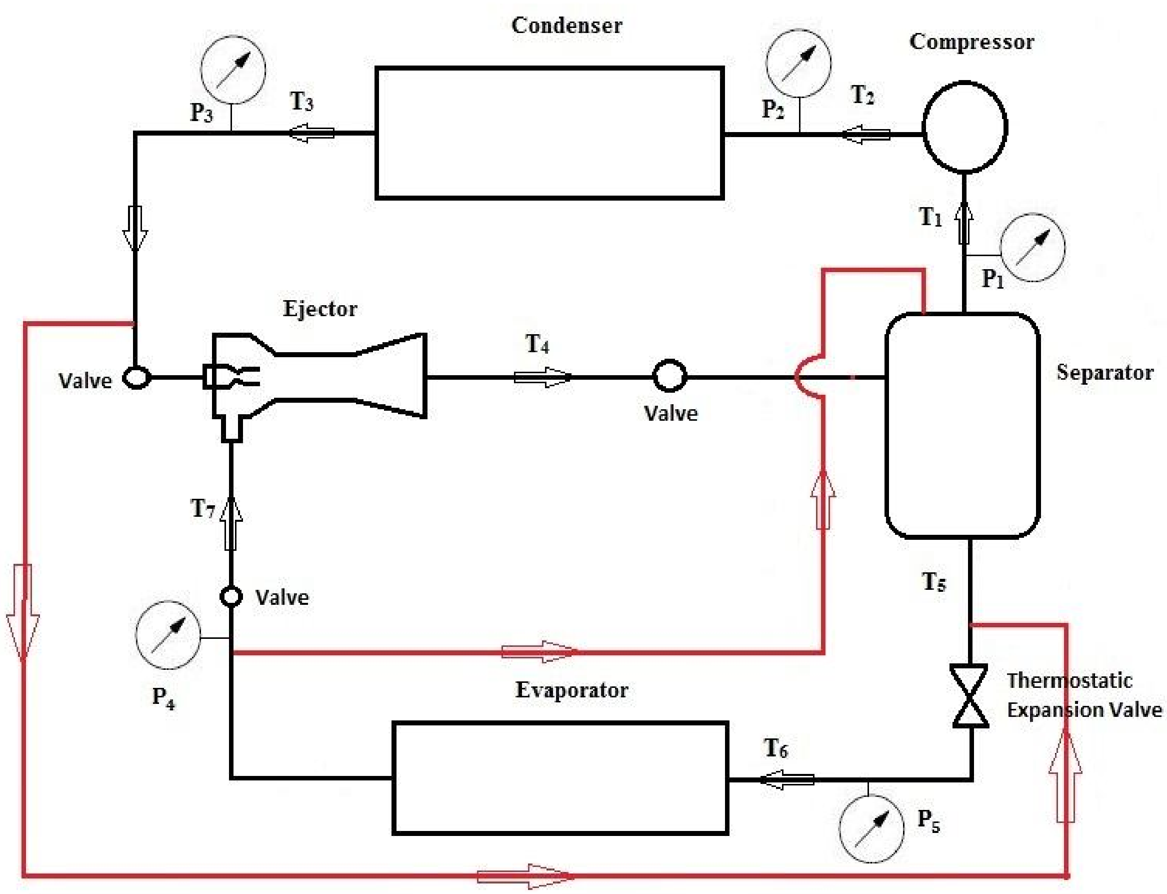

Figure 1Schematic of Ejector-Expansion Refrigeration System

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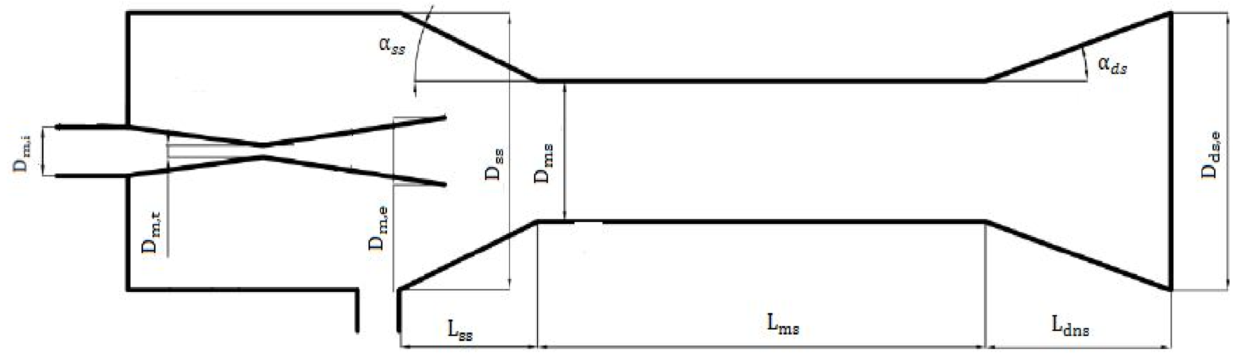

Figure 2Ejector Parameters

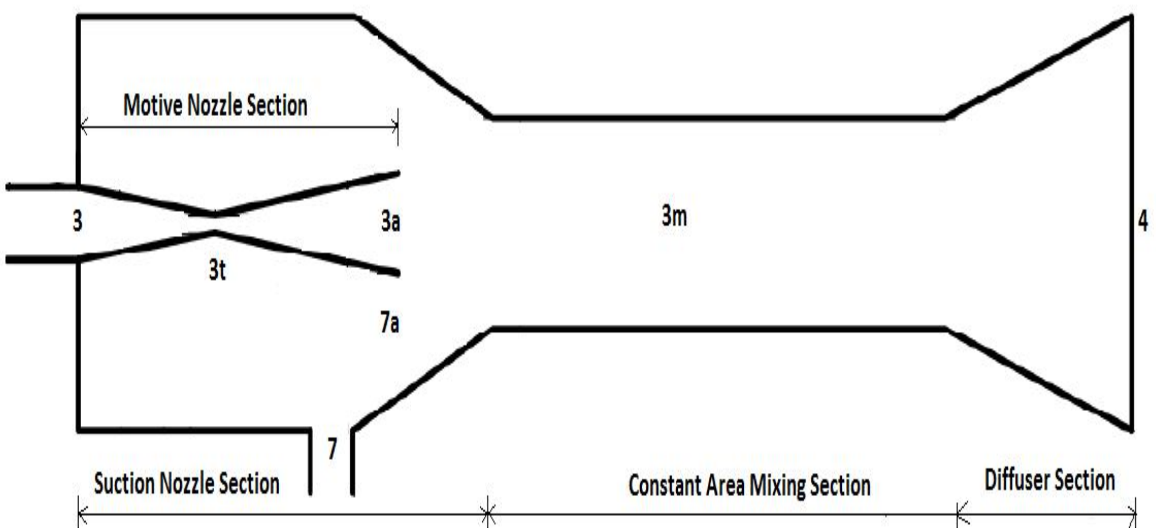

Figure 3Ejector Sections

Table 1 Ejector Dimensions
All Dimensions are in mm

Literature Studied	Refrigerant Used	$D_{m,i}$	$D_{m,t}$	$D_{m,e}$	$D_{s,s}$	D_{ms}	$D_{ds,e}$	L_{ss}	L_{ms}	L_{ns}	α_{ss}	α_{ds}
Disawas and Wongwises (2004)	R-134a	6	0.9	1.8	30	10	22	30	110	90	18.4	3.8
Chaiwongsa and Wongwises (2007)	R-134a	6	1.0	2.5	30	10	22	30	110	90	18.4	3.8
Chaiwongsa and Wongwises (2008)	R-134a	6	0.9	3.8	30	10	22	30	110	90	18.4	3.8
Present Literature	R-134a	6	2	4	25	9	16	25	55.2	62	21	3.5

Table 1 shows dimensions of the ejector used in previous work and present work. Present paper introduces the complete new dimension of the ejector with the help of mass and moment conservation equations stated in the literature. These dimension are not used previously and also shows better results than previous work.

VII. RESULTS

This chapter covers the detail results and discussion of the experimentation conducted during work. The various system affected parameters are analyzed through the graphical representation. Each system affecting parameter is discussed based on the analysis of results obtained from the measured and estimated parameters.

Experimentation is carried out the effect of evaporation and condensation temperature on system parameters by using ejector expansion cycle such as pressure lift, compressor power, COP, entrainment ratio, etc. are discussed.

Table 2 represents the comparison between present work results and previous work results. From this table it is clear that the newly developed ejector shows the better results than previous work. Table 2 also shows the percentage deviation of COP with respect to previous work.

Table 2 Comparison between present work results and previous results

Literature	Refrigerant	COP Value	% Deviation
Praitoon Chaiwongsa (2007) [4]	R-134a	5.93	5.5882
Somchai Wongwises (2004) [2]	R-134a	5.21	17.0514
Li Zhao [14]	Mixture R134a/R-143a	4.24	32.4948
Nagihan Bilir	R-134a	5.47	12.9119
M. Hassanain [9]	R-134a	5.87	6.5435
Present Literature	R-134a	6.281	0

The main motive of the reported graphs are to see how the different parameters will vary with respect to temperature and then optimized them at maximum COP value. We can easily understand how the different parameters vary with respect to temperature in Table 3 and Table 4.

Table 3 Variation of Different Parameters with respect to Evaporator Temperature

T_e (^{0} C)	$M_{p\ (Kg\ s\text{-}1)}$	$M_{s\ (Kg\ s\text{-}1)}$	ω	Pressure Lift	Compressor Power (kW)	COP (BC)	COP (EEC)	COP%
-10	0.07	0.0494	0.06879	1.15	3.26	2.8312	3.05	7.7281
-5	0.06859	0.05023	0.712	1.133	2.9272	3.192	3.4708	8.7343
0	0.06737	0.05102	0.7359	1.103	2.5056	3.651	3.9969	9.4741
5	0.06612	0.05189	0.7621	1.085	2.18	4.282	4.7332	10.5371
10	0.06494	0.0528	0.789	1.073	1.7016	5.53	6.281	13.5804

A. Effect of Evaporation Temperature on System Performance

Figure 4 shows the relationship between primary mass flow rate (M_p), secondary mass flow rate (M_s) and COP of the system. The primary mass flow rate and secondary mass flow rate is the function of compressor power and refrigerant effect respectively and it directly affected COP of the system. Primary mass flow rate decreases by 7.79% when evaporation temperature raises from -10^0 C to 10^0 C and hence compressor power decreases. Due to decrease in primary mass flow rate, secondary mass flow rate increases by 6.88%. Finally, refrigerant effect increases and hence COP of the system increases.

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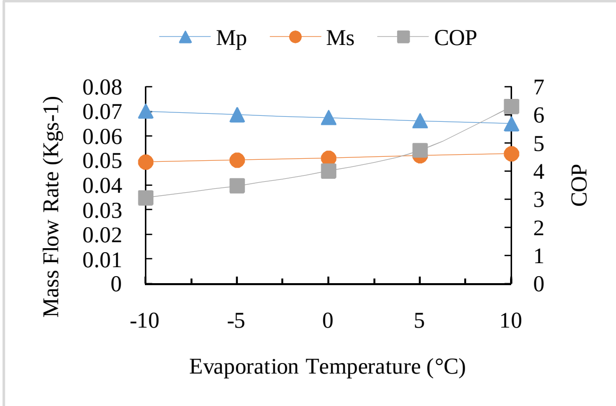

Figure 4 Effect of Evaporation Temperature on Mass Flow Rate and COP

Figure 5 shows the effect of evaporation temperature on system coefficient of performance, entrainment ratio. Entrainment ratio is the ratio of secondary mass flow rate to the primary mass flow rate. When evaporation temperature raises from -10^0 C to 10^0 C, primary mass flow rate decreases and secondary mass flow rate increases and hence entrainment ratio increases by 15.2%. Entrainment ratio is a function of compressor power and refrigerant effect hence it directly affect the COP of the system. Clearly from figure 5, rate of COP increases by 49.53% as entrainment ratio increases as evaporation temperature changes from -10^0 C to 10^0 C.

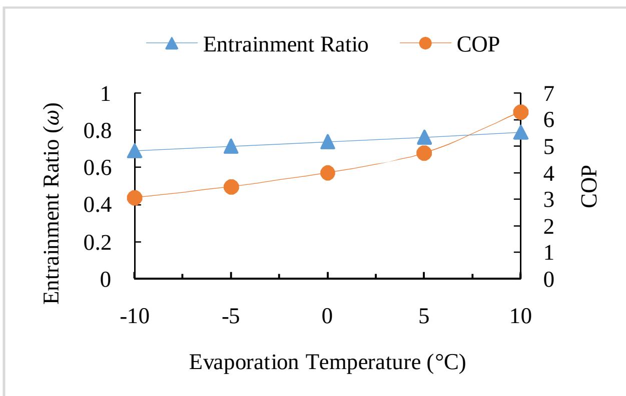

Figure 5 Effect of Evaporation Temperature on Entrainment Ratio and COP

Figure 6 shows that the effect of evaporation temperature on the compressor power and pressure lift. Pressure lift is the ratio of evaporator pressure to separator pressure. Evaporation pressure is always greater than separator pressure and ratio is always becomes greater than unity. As evaporation temperature increases from -10^0 C to 10^0 C, the separator pressure increases and hence there is decrease in pressure lift as shown in figure 6. As primary mass flow rate decreases with decrease in compressor work by 13% when evaporation temperature increases from -10^0 C to 10^0 C.

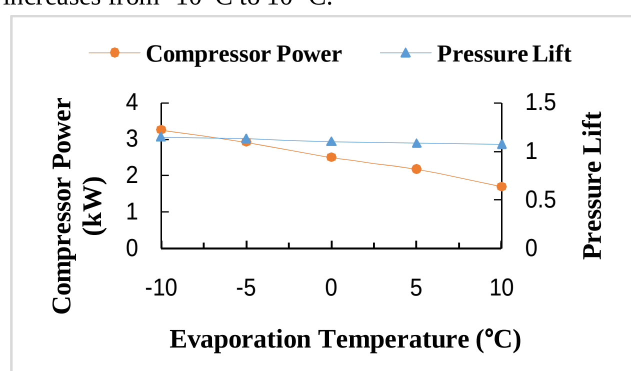

Figure 6Effect of Evaporation Temperature on Compressor Power and Pressure Lift

Figure 7 shows the comparison of ejector-expansion cycle (EEC) and basic cycle (BC) with COP and rate of COP (COP %) of the system. As evaporation temperature increases from -10^0 C to 10^0 C, the COP of the both basic cycle and ejector-expansion cycle increases. The rate of COP (COP %) is maximum of the higher evaporation temperature when evaporation temperature changes from -10^0 C to 10^0 C as shown in fig. 7. EEC has greater potential to save work in throttling process over BC. The COP of the EEC increases from 7.72% to 13.58% over BC.

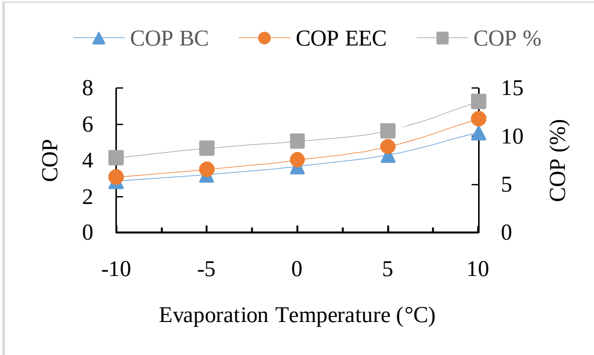

Figure 7 Effect of Evaporation Temperature on COP of System

B. Effect of Condensation Temperature on System Performance

Table 4 Variation of Different Parameters with respect to Condenser Temperature

T_c (^{0} C)	$M_{p\ (Kg\ s-1)}$	$M_{s\ (Kg\ s-1)}$	ω	Pressure Lift	Compressor Power (kW)	COP (BC)	COP (EEC)	COP%
30	0.0612	0.0506	0.82	1.0689	1.468	5.887	6.7177	14.1107
35	0.06415	0.05078	0.769	1.0862	1.7106	5.18	5.7587	11.1718
40	0.06741	0.05098	0.7359	1.1034	2.067	4.2926	4.7595	10.8768
45	0.07101	0.05122	0.703	1.1379	2.177	4.1	4.4924	9.5707
50	0.07514	0.0515	0.6699	1.1724	2.304	3.903	4.4924	8.8009

Figure 8 shows the relationship between primary mass flow rate, secondary mass flow rate and COP of the system. The primary mass flow rate and secondary mass flow rate is the function of compressor power and refrigerant effect respectively and it directly affected COP of the system. Primary mass flow rate increase by 8.49% when condensation temperature raises from 30^0 C to 50^0 C and hence compressor power decreases. Due to decrease in primary mass flow rate, secondary mass flow rate decrease by 2.64%. Finally, refrigerant effect increases and hence COP of the system increases.

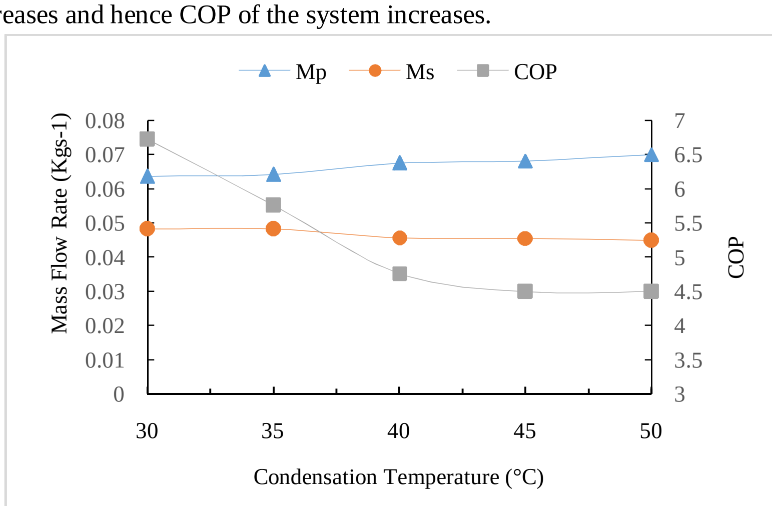

Figure 8 Effect of Condensation Temperature on Mass Flow Rate and COP

Figure 9 shows the effect of condensation temperature on system, entrainment ratio and coefficient of performance. When condensation temperature raises from 30^0 C to 50^0C, primary mass flow rate increase and secondary mass flow rate decreases and hence entrainment ratio decreases by 22.4%. Entrainment ratio is a function of compressor power and refrigerant effect hence it directly affect the COP of the system. Clearly from figure 9, rate of COP decreases by 56% as entrainment ratio increases as evaporation temperature changes from 30^0 C to 50^0 C.

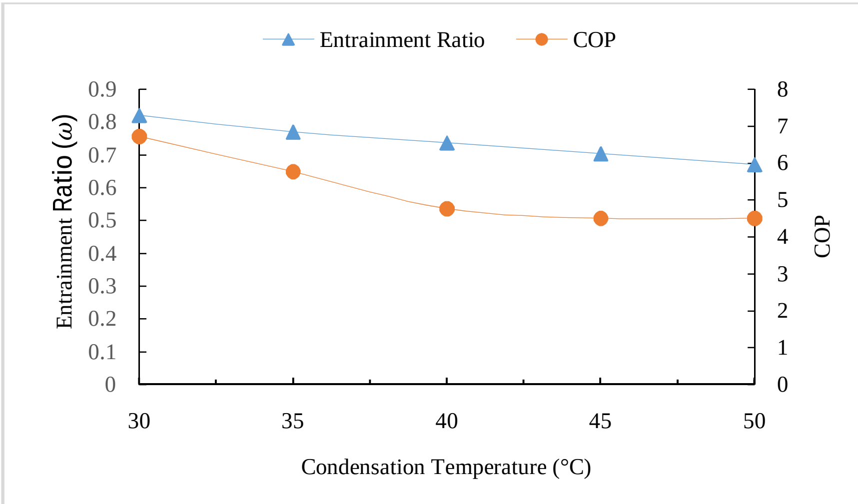

Figure 9 Effect of Condensation Temperature on Entrainment Ratio and COP

Figure 10 shows that the effect of condensation temperature on the compressor power and pressure lift. As condensation temperature increases from 30^0 C to 50^0 C, the separator pressure decreases and diffuser exit pressure increases hence there is increase in pressure lift as shown in figure 10. As primary mass flow rate increase with increase in compressor work by 56.94% when condensation temperature increases from 30^0 C to 50^0 C.

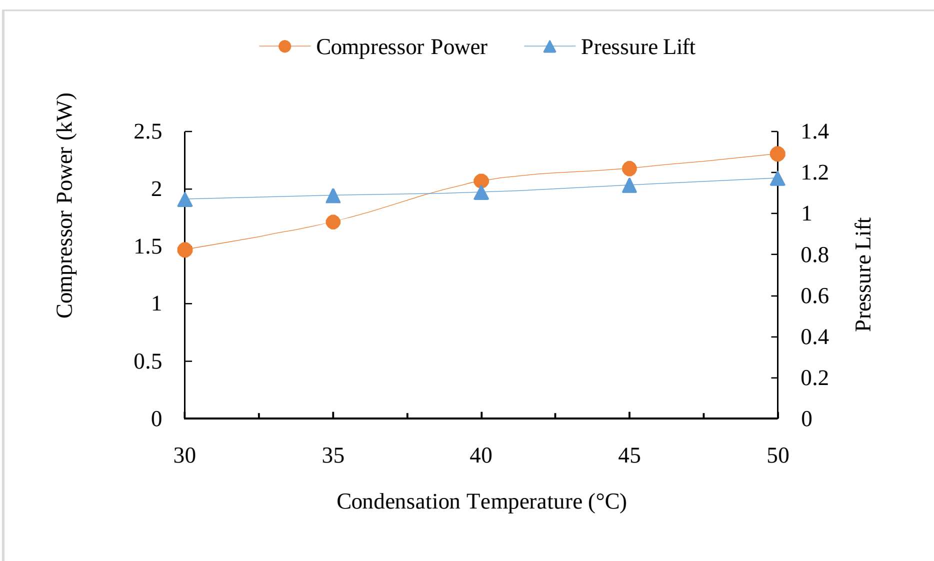

Figure 10 Effect of Condensation Temperature on Compressor Power and Pressure Lift

Figure 11 shows the comparison of ejector-expansion cycle (EEC) and basic cycle (BC) with COP and rate of COP (COP %) of the system. As condensation temperature increases from 30^0 C to 50^0 C, the COP of the both basic cycle and ejector-expansion cycle decreases. The rate of COP (COP %) is maximum of the lower condensation temperature, when condensation temperature changes from 30^0 C to 50^0 C as shown in figure 10. EEC has greater potential to save work in throttling process over BC. The COP of the EEC decreases from 14.11% to 8.8% overBC.

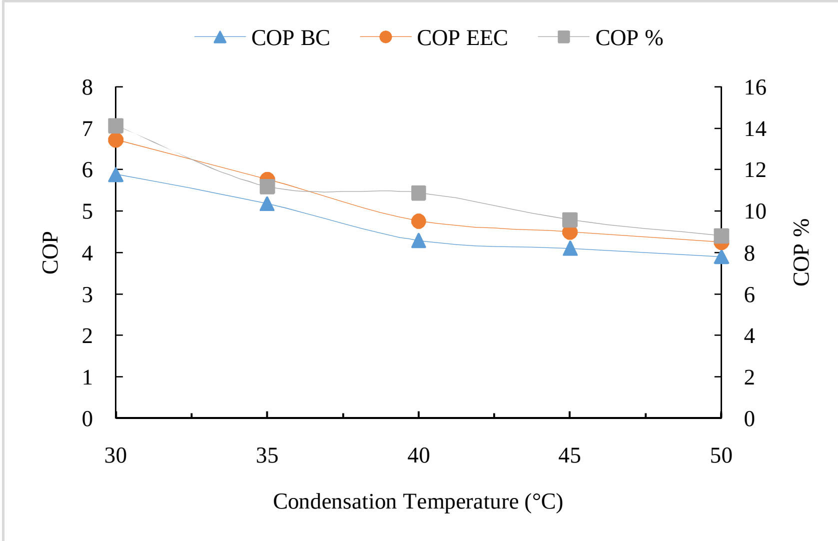

Figure 11 Effect of Condensation Temperature on COP of System

VIII. CONCLUSION

In this paper, newly designed constant area ejector is tested by taking effect of evaporation temperature ranges from -10^0 C to 10^0 C and condensation temperatures ranges from 30^0 C to 50^0 C and ejector expansion refrigeration system performance are investigated. Also performance of ejector-expansion refrigeration system is presented in comparison with simple vapor compression refrigeration system under the above operating conditions. During test, the ambient temperature was maintained at 27° C. The calculations were done by using P-h chart provided. From this experimentation, following conclusions were made, which helps the researchers for future studies:

A. As the evaporation temperature changes from -10° C to 10° C, COP and entrainment ratio increases by 105.93% and 14.55% respectively. And the ejector diameters such as motive nozzle section diameter, motive section diameter, suction section diameter, mixing section diameter decreases by 1.41%, 38.09%, 21.81%, 22.78% respectively,

B. The system COP and entrainment ratio decreases by 49.53% and 22.4% as the condensation temperature varies from 30°C to 50°C. However, Ejector diameters such as motive nozzle section diameter, suction nozzle diameter, and mixed section diameter decreases by 1%, 14.86%, 32.89%, respectively and motive nozzle section diameter increases by 1.26%.,

C. The COP of ejector expansion cycle has improvement from 7.72% to 13.58% over basic cycle when the evaporation temperature increases from -10° C to 10° C.

D. As condensation temperature increases from 30° C to 50° C the COP of the ejector expansion cycle has improvement from 8.8% to 14.11% over basic cycle.

E. As the evaporation temperature increases from -10° C to 10° C, primary mass flow rate decreases by 7.22% and secondary mass flow rate increases by 6.88% as total mass flow rate decreases by 1.41%.

F. The primary mass flow rate increases by 8.49% and secondary mass flow rate decreases by 6.97% as total mass flow rate increases by 2.64%.

G. As the evaporation temperature increases from -10° C to 10° C, the compressor power decreases by 47.8%. However, pressure lift decreases by 6.69

H. The compressor power and pressure lift increases by 56.94% and 9.68% respectively as condensation temperature increases from 30° C to 50° C.

IX. ACKNOWLEDGMENT

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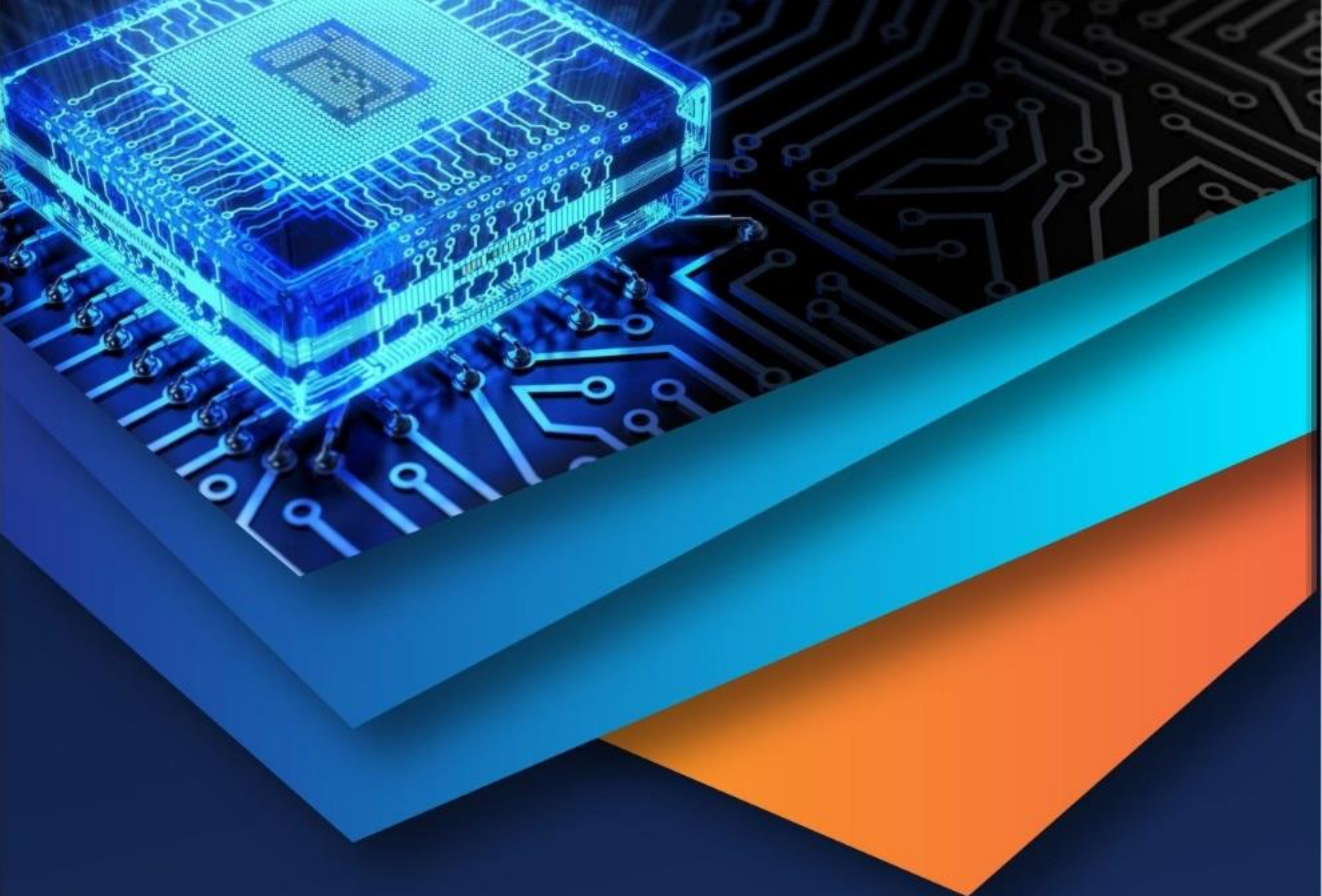
REFERENCES

[1]. A. A. Kornhauser, the Use of an Ejector as a Refrigerant Expander, International Refrigeration and Air Conditioning Conference, Paper 82.
[2]. SomjinDisawas, SomchaiWongwises*, Experimental investigation on the performance of the refrigeration cycle using a two-phase ejector as an expansion device, International Journal of Refrigeration 27 (2004) 587–594.

[3]. SomchaiWongwises *, SomjinDisawas, Performance of the two-phase ejector expansion refrigeration cycle, International Journal of Heat and Mass Transfer 48 (2005) 4282–4286.
[4]. Praitoon Chaiwongsa, Somchai Wongwises*, Effect of throat diameters of the ejector on the performance of the refrigeration cycle using a two-phase ejector as an expansion device, International Journal of Refrigeration 30 (2007) 601-608.
[5]. Praitoon Chaiwongsa, Somchai Wongwises *, Experimental study on R-134a refrigeration system using a two-phase ejector as an expansion device, Applied Thermal Engineering 28 (2008) 467–477.
[6]. Huashan Li a,b, Fei Cao c, Xianbiao Bu a, Lingbao Wang a, XianlongWanga,Performance characteristics of R1234yf ejector-expansion refrigeration Cycle, Applied Energy 121 (2014) 96–103.
[7]. KhaledAmeur, ZineAidoun, Mohamed Ouzzane, Modeling and Numerical Approach for the Design and Operation of Two-phase Ejectors, Applied Thermal Engineering (2014), doi: 10.1016/j.applthermaleng.2014.11.022.
[8]. Nagihan Bilir and H. KursadErsoy, Performance improvement of the vapor compression refrigeration cycle by a two-phase constant area ejector, International journal of energy research, Int. J. Energy Res. 2009; 33:469–480.
[9]. M. Hassanain, E. Elgendy, M. Fatouh, Ejector expansion refrigeration system: ejector design and performance evaluation, International Journal of Refrigeration, (2015), doi:10.1016/j.ijrefrig.2015.05.018.
[10]. JaharSarkar, Geometric parameter optimization of ejector expansion refrigeration cycle with natural refrigerants, International Journal of Energy Research Int. J. Energy Res. 2010; 34:84–94.
[11]. F. Wang, D.Y. Li, Y. Zhou, Analysis for the ejector used as expansion valve in vapor compression refrigeration cycle, Applied Thermal Engineering (2015), doi: 10.1016/j.applthermaleng.2015.11.095.
[12]. J Sarkar, "Performance characteristics of natural-refrigerants based ejector expansion refrigeration cycles", Journal of Power and Energy, Proc. IMechE Vol. 223.
[13]. R. Yapici* and H.K. Ersoy, "Performance characteristics of the ejector refrigeration system based on the constant area ejector flow model", Energy Conversion and Management 46 (2005) 3117–3135.
[14]. Li Zhao[a,*] , XingyangYang[a], ShuaiDeng[a], Hailong Li[b], ZhixinYu[c], "Performance analysis of the ejector-expansion refrigeration cycle using zeotropic mixtures", International Journal of Refrigeration, 57 (2015) 197-207.
[15]. Jian-qiang Deng [a], Pei-xue Jiang [b,*], Tao Lu [b], Wei Lu [b], "Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector", Applied Thermal Engineering, 27 (2007) 381-388.
[16]. E. Elgendy, "Parametric Study of a Vapor Compression Refrigeration Cycle Using a Two-Phase Constant Area Ejector", International Journal of Mechanical, Aerospace, Industrial, Mechatronic and Manufacturing Engineering, Vol: 7, No: 8, 2013.
[17]. Fang Liu*, Yong Li, Eckhard A. Groll, "Performance enhancement of CO2 air conditioner with a controllable ejector", International Journal of Refrigeration, 35 (2012) 1604-1616.
[18]. Krzysztof Banasiak[a],*, Armin Hafner[b], Trond Andresen [b], "Experimental and numerical investigation of the influence of the two-phase ejector geometry on the performance of the R744 heat pump", International Journal of Refrigeration, 35 (2012) 1617-1625.
[19]. Gourav Patel* and Dr. SharadChaudhary**, "Performance Comparison of Ejector Expansion Refrigeration Cycle with Throttled Expansion Cycle Using R-170 as Refrigerant", International Journal of Scientific and Research Publications, Volume 4, Issue.



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