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Performance Comparison of Ejector Expansion Refrigeration Cycle with Throttled Expansion Cycle Using R-170 as Refrigerant

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Abstract- Since long time, it has been noticed that refrigerators are the devices which work almost 365 days round the clock; hence objective of energy efficiency improvement attracts much. There are several ways of improving the performance of a vapor compression refrigeration cycle. Use of an ejector as expansion device is one of the alternative way. The advent of new component 'Ejector' into refrigeration system opened the new era of research. The vital component, which decides the effective operation of the ejector expansion refrigeration system, is the ejector. Hence, design of an ejector and analyses of its physical and operational parameters have drawn special attention. The thermodynamic analysis of natural refrigerant (R 170) based vapour compression refrigeration cycles is presented in this article using a constant pressure mixing ejector as an expansion device. Using ejector as an expansion device, R 170 yields a maximum COP improvement of 24.12 percent.

Index Terms- Coefficient of performance, Comparison, Ejector expansion cycle, R 170, Vapour compression cycle.

I. INTRODUCTION

The throttling device in a refrigeration system normally serves two purposes. One of the thermodynamics function is expanding the liquid refrigerant from the condenser pressure to the evaporator pressure. The other one is the control function which may involve the supply of the liquid to the evaporator at the rate at which it is evaporated. Irreversibility associated with throttling is major issue in vapour compression refrigeration cycle. There are different ways to reduce the throttling losses in the refrigeration cycles. Use of ejector as an expansion device by replacing the throttling valve in the conventional vapor compression refrigeration cycle is a promising alternative to reduce the throttling losses or the expansion irreversibility in the refrigeration. Because of its simple structure, ease of manufacturing, no moving parts, low cost and low maintenance requirements, the use of two- phase ejector has become an important cycle modification recently. Ejector reduces the compressor work by raising the suction pressure to a level higher than that of which in turn improves COP of the system. It also enables to reduce size of the evaporator.

In 1990, an analysis is performed by Kornhouser [1] on the Ejector Expansion refrigeration cycle to investigate the performance improvement on vapor compression refrigeration (VCRC). Eight refrigerants were used, R11, R12, R113, R114, R500, R502, R22 and R717. According to this paper, refrigerant R502 has given the highest coefficient of performance improvement and the COP improvement using R12 was 21% over the basic cycle. In 1995, according to Domanski [2] the theoretical COP of the ejector-expansion refrigeration cycle was very much sensitive to the ejector efficiency. In 1998, Nakagawa and Takeuchi [3] research was concluding that the longer divergent part provides a longer period of time for the two-phase flow to achieve equilibrium. With this result, using longer length of the divergent part of the motive nozzle, higher motive nozzle efficiency was determined. In 2007, based on the second law of thermodynamics Yari and Siriousazar [4] worked on performance of transcritical CO2 refrigeration cycle with ejector-expansion. They found ejector is improving the optimum second-law efficiency by 24.8% as compared to conventional system and 16% as compared to internal heat exchanger system. In 2007, an analysis is given by Deng et al. [5] on a transcritical CO2 ejector-expansion refrigeration cycle that uses an ejector as the main expansion device instead of an expansion valve. He concluded for the given working conditions, the ejector was improving the maximum COP by 18.6% compared to the internal heat exchanger system and 22% compared to the conventional system. In 2007 according to simulation work by Nehdi et al. [6] on performance of the vapour compression cycle using ejector as an expander, it has been found that the geometric parameters of the ejector design have noticeable effects on the system performance. He achieved the maximum COP for Optimum geometric area ratio around 10. For the given operating conditions of evaporator temperature, 5ºC and condenser temperature, 40 ºC, In 2008, Yari, M. [7] conducted Exergetic analysis of the vapour compression refrigeration cycle using ejector as an expander, in this research the effects of evaporating temperature and condensing temperature on the COP, second law efficiency and exergy destruction in various component were calculated, and also summarized that the COP and second law efficiency of the ejector-compression is about 16% greater than that for the vapour compression cycle and total exergy destruction of the vapour compression cycle was about 24 % higher than that for the ejector-compression cycle. According to J Sarkar [8] - [9] research on the performance improvement with three natural refrigerants namely, ammonia, propane and isobutene, it has been determined that maximum performance improvement using ejector can be achieved in the case of isobutane, whereas ammonia giving minimum performance improvement. In 2012, J Sarkar [10] provided a detailed literature review on two-phase ejectors and their applications in vapor compression refrigeration and heat pump systems. In this paper he described wide theory on characteristics of both subcritical and transcritical vapor compression systems with various cycle configurations.

II. DESCRIPTION OF THE SYSTEM

The layout of the ejector expansion vapour compression refrigeration cycle in figure 1 and the corresponding Pressure−Enthalpy diagram is shown in figure 2.

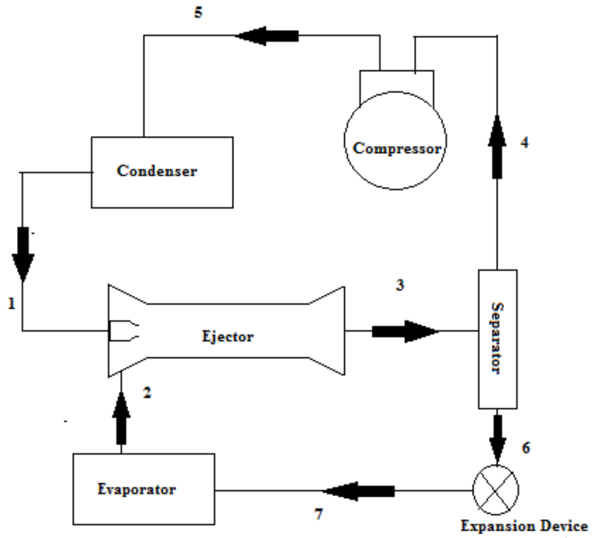

Figure1. Ejector Expansion Vapour Compression Refrigeration Cycle

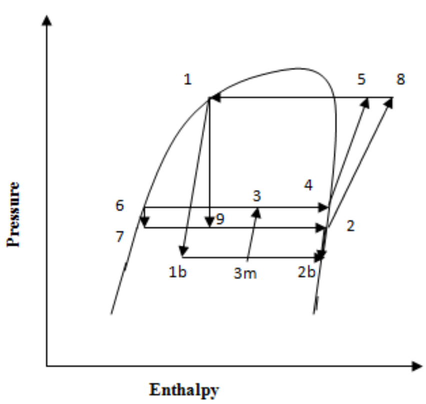

Figure2. Pressure Enthalpy Diagram

The primary flow from the condenser (state 1) and the secondary flow from the evaporator (state 2) are expanding through primary and secondary nozzles, respectively (1–1b and 2–2b) to mixing chamber pressure, mixing at constant pressure (1b, 2b–3m). The mixed flow is discharged through the diffuser (3m–3) of the ejector and then separated in forms of vapour (state 4) and liquid (state 6) so that this ratio matched with the inlet ratio of primary and secondary flows. Then the liquid circulates through the expansion valve (6–7) and evaporator (7–2), whereas the vapour flows into the compressor (4-5) and the condenser (5-1). There are three ejector parameters, entrainment ratio (secondary mass flow to primary mass flow), PLR (diffuser exit pressure to secondary nozzle inlet pressure), and geometric area ratio (total nozzle exit area to primary nozzle exit area), which greatly affect the system performance.

III. ANALYSIS OF THE SYSTEM

The ejector expansion vapour compression refrigeration cycle has been modeled based on following conservation laws and equations:

- Conservation of mass,
- Conservation of momentum,
- Conservation of energy

The following assumptions have been used to simplify the theoretical model and set up the equations per unit total ejector flow,

1. The refrigerant will be at all times in thermodynamic quasi-equilibrium.
2. There is negligible Pressure Drop
3. No wall Friction.
4. Steady state one-dimensional model.
5. Thermodynamic processes in compressor, expansion valve and ejector area assumed to be adiabatic
6. No heat transfer with the surrounding for the system except in the condenser.
7. The flow across the throttle valve is isenthalpic.
8. The refrigerant condition at the evaporator outlet is saturated vapour and condenser outlet is saturated liquid.
9. The vapour condition from the separator is saturated vapour and the liquid coming from the separator is saturated liquid.

A. Nomenclature

A	cross-sectional area (m2)
V	fluid velocity (m/s)
COP	coefficient of performance
h	specific enthalpy (kJ/kg)
P	pressure (kPa)
PLR	pressure lift ratio
qe	specific cooling effect (kJ/kg)
t	temperature (◦C)
wc	specific work (kJ/kg)
x	vapour quality
η	isentropic efficiency
μ	entrainment ratio
Φ	ejector area ratio

B. Subscripts

B	basic cycle
c	compressor
ej	ejector

C. Equations

Using above assumptions, following equations are set up

$$h_1 = h_{1b} + V_{1b}^2/2 \tag{1}$$

$$h_2 = h_{2b} + V_{2b}^2/2 \tag{2}$$

Nozzle outlet enthalpies are given by

$$h_{1b} = h_1 - \eta_n[h_1 - h(P_{2b}, s_1)] \tag{3}$$

$$h_{2b} = h_2 - \eta_n[h_2 - h(P_{2b}, s_2)] \tag{4}$$

Primary and secondary nozzle exit area is given by

$$A_{1b} = 1/(1+\mu)\rho_{1b} V_{1b} \tag{5}$$

$$A_{2b} = 1/(1+\mu)\rho_{2b} V_{2b} \tag{6}$$

The ejector area ratio of this system is given by

$$\Phi = (A_{1b} + A_{2b}) / A_{1b} \tag{7}$$

Velocity and enthalpy at the outlet of the constant pressure mixing section of the ejector are given by

$$V_{3m} = (1/(1+\mu)) V_{1b} + (\mu/(1+\mu)) V_{2b} \tag{8}$$

$$h_{3m} = (1/(1+\mu))(h_{1b} + V_{1b}^2/2) + \frac{\mu}{1+\mu}(h_{2b} + V_{2b}^2/2) - V_{3m}^2/2 \tag{9}$$

For the diffuser section

$$h_{3m} + V_{3m}^2/2 = h_3 \tag{10}$$

Whereas diffuser exit enthalpy can be found by

$$h_3 = h_{3m} + [h(P_4, s_{3m}) - h_{3m}]/\eta_d \tag{11}$$

The Ejector overall energy balance is given by

$$\frac{1}{1+\mu} h_1 + \frac{\mu}{1+\mu} h_2 = h_3 \tag{12}$$

The specific compressor work can be found by

$$w_{c,ej} = \frac{1}{1+\mu}(h_5 - h_4) \tag{13}$$

Cooling effect can be found by

$$q_{e,ej} = \left(\frac{\mu}{1+\mu}\right)(h_2 - h_7) \tag{14}$$

The COP of the ejector expansion cycle can be determined by

$$COP = \frac{q_{e,ej}}{w_{c,ej}} \tag{15}$$

The COP of the corresponding basic vapour compression cycle is given by

$$COP_B = \frac{h_2 - h_1}{h_8 - h_2} \tag{16}$$

And the COP improvement is given by

$$\Delta COP = \frac{COP - COP_B}{COP_B} \tag{17}$$

IV. COMPUTATIONAL METHOD

A computer MODEL is developed to solve the above equations using MATLAB SIMULINK. The model takes following input data type of refrigerant, evaporation temperature, condensation temperature, and refrigerant properties. Refrigerant (R 170) is used as a working fluid into the system. The simulation procedure is as follows;

1. First of all calculating thermodynamic properties at state points 1 and 2. For the given motive and suction efficiencies Specific enthalpies and specific volume at states 1b and 2b are calculated. Velocities of refrigerant stream at the corresponding states are calculated by using (1) and (2).
2. Starting calculation with assuming an iterative value of entrainment ratio (1> μ >0) than fluid velocity and specific enthalpy at the exit of constant pressure mixing section (state 3m) are calculated using (8) and (9).
3. Using (10) to (12) and known diffuser efficiency, specific enthalpy, pressure vapour quality at state 3 are calculated by applying effective iteration technique, other properties are also calculated.
4. If the condition $(1 + \mu)\, x3 \cong 1$ is not satisfied then steps 8–9 will be repeated by feeding a new value of μ until the condition is satisfied.
5. Calculate thermodynamic properties at states 4, 6, and 7. Also by using known compressor isentropic efficiency the properties of state 5 are calculated
6. The various performance parameters like wc,ej, qe,ej, COP, COPB and ΔCOP are calculated Using (14) to (17).

V. RESULTS

Using ejector as an expansion device, R 170 yields a maximum COP improvement of 24.12 percent. For given evaporator temperature 208K, condenser temperature 242K and Refrigerant, calculations were performed to observe effect of mixing temperature. It was found that, mixing temperature is attributed to change in entrainment ratio which in turn affects Coefficient of performance (COP). With increase in mixing temperature Coefficient of performance (COP) first increases as entrainment ratio increases, after certain optimum value it decreases as entrainment ratio decreases. Mixing temperature is also attributed to ejector area ratio, so by changing ejector areas we can select corresponding mixing temperature.

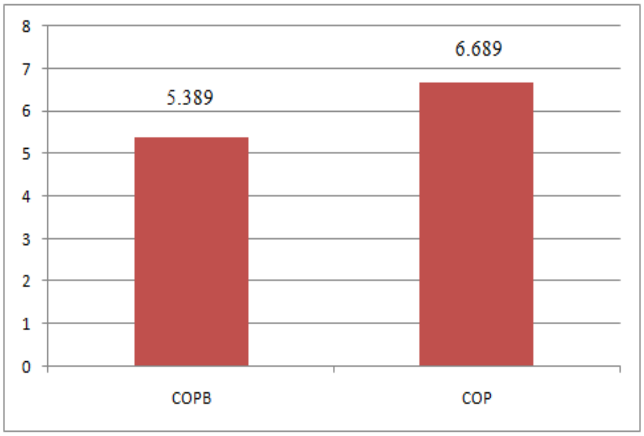

Figure3. Performance Comparison of Ejector Expansion Cycle with Throttled Expansion Cycle

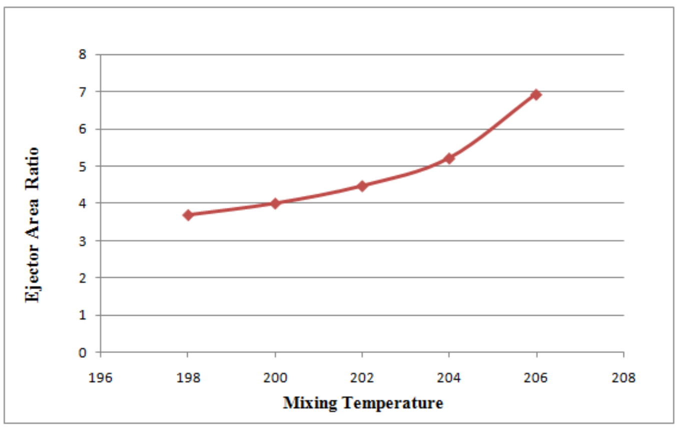

Figure4. Mixing Temperature v/s Ejector Area Ratio

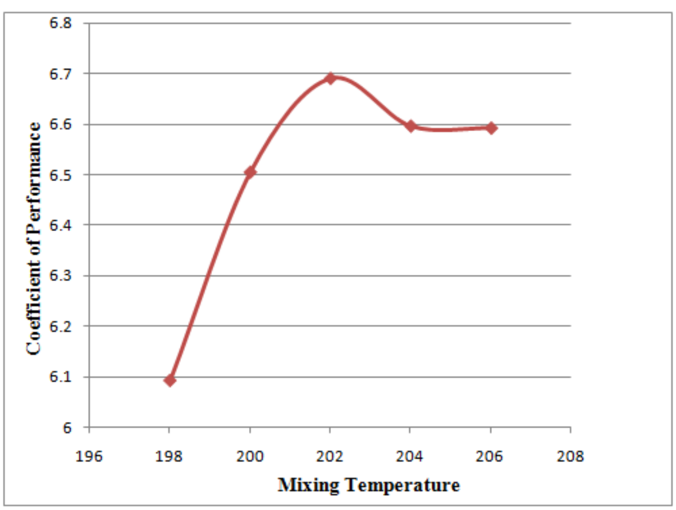

Figure5. Mixing Temperature v/s Coefficient of Performance

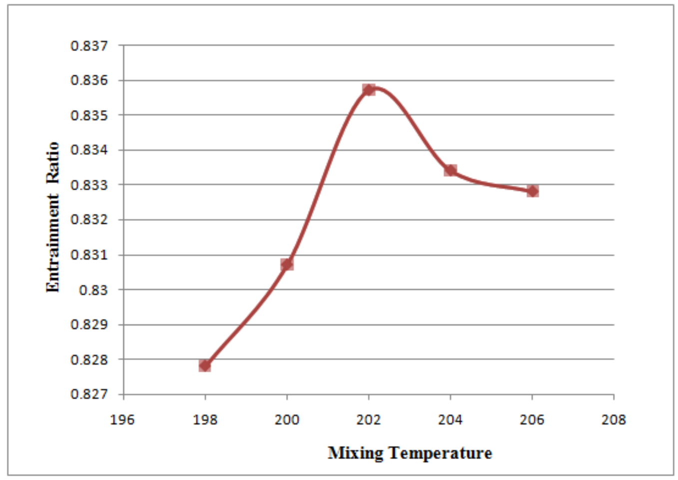

Figure6. Mixing Temperature v/s Entrainment Ratio

VI. CONCLUSION

On the basis of above analysis it has been found that an ejector refrigeration system is suitable for experimental studies, since it require ejector a simple component that significantly improves system performance. By keeping same evaporator temperature and condenser temperature the ejector expansion refrigeration cycle provides increased coefficient of performance, decreased compressor displacement, and decreased compression ratio, reduced evaporator size as compared with a standard vapor compression cycle. This may open new era of the search for non CFC, refrigerants. Deeper scrutiny is required for understanding of various processes within the two-phase ejector. Implementations of the ejector expansion cycle for practical uses have not yet been worked out. Further research is required in this field.

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