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A Review on Novel Air Conditioning System Using Alternative Refrigerants

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Abstract: The present work aims to design a constant-area two phase ejector. Also the performance characteristics of ejector expansion refrigeration cycle are evaluated. In order to achieve this, a simulation program is developed in Engineering Equation Solver (EES) software. In a refrigeration cycle, the pressure drop is considered as isenthalpic process where the enthalpy remains constant. However, isenthalpic process causes a decrease in the evaporator cooling capacity due to energy loss in the throttling process. An efficiency enhancing alternative was proposed to recover this energy loss, which uses an ejector that can be used to generate isentropic condition where the entropy remains constant in the throttling process. Such a cycle is called as ejector expansion refrigeration cycle (EERC). The EERC is variant of the standard vapor compression cycle in which an ejector is used to recover part of the work that would otherwise be lost in the expansion valve.

Keywords — ***Keywords: 2 phase ejector, Ejector expansion refrigeration cycle, Air- conditioning, Nozzle, Diffuser, Throat.***

I. INTRODUCTION

Ejector is a device that uses the enthalpy released during the expansion of a high pressure fluid to provide compression work to low pressure fluid. The high pressure (motive or primary) fluid is expanded through a nozzle to low pressure and high velocity and is mixed with a low-pressure (suction or secondary) fluid. The two fluids do not have to be of the same composition but often are for refrigeration applications. The mixed flow is then decelerated in the diffuser, and the momentum is converted to static pressure. Thus, the net effect of the ejector is a pressure increase of the suction fluid provided by the expansion of and mixing with the motive fluid. The ejector can theoretically approach an isentropic expansion because of the enthalpy of the motive stream is converted to kinetic energy, which is then transferred to the suction stream, ultimately reducing the enthalpy of the motive stream as it passes through the ejector. An isentropic expansion represents the ideal limiting case when no irreversibility is present and no entropy generation occurs.

II. LITERATURE REVIEW

Kornhauser et al. was the first to analyze the performance of the standard ejector cycle. According to his study the COP of the ejector cycle was strongly dependent on the specified efficiencies of the ejector components and that the maximum COP improvement achieved by the ejector cycle compared to cycle with isenthalpic expansion ranged from 12 to 30 % depending on the refrigerant. A similar study was performed by Domanski in the year 1995 with 38 different refrigerants, predicting up to 80% COP improvement. Nehdi et al. numerically proved that replacing expansion valve with a two-phase ejector improved COP up to 22 % in a vapour compression system using R141b as refrigerant. Bilir and Ersoy theoretically found that refrigerantR134a gives 22.3% improvement for ejector expansion refrigeration cycle as compared with basic cycle. It was also found experimentally that use of ejector instead of throttle valve for R134a refrigeration cycle improves COP by 7.34%-12.87%. Jahar Sarkar presented thermodynamic analysis and comparison of ammonia, propane and Iso-butane based vapour compression refrigeration cycles using constant area ejector an expansion device. Optimization of ejector geometric parameter based on the maximum cooling COP and performance improvement for different operating conditions was studied. COP improvements were strongly dependent on refrigerant properties as well as on operating conditions. Refrigerant isobutene yield maximum COP of 21.6%. Whereas 17.9% and 11.9 % improvement was observed for propane and ammonia respectively. Sumeru et al. presented study that provides experimental result of a novel cycle using an ejector as an expansion device in split type air conditioner. By studying the nozzle diameter and the diffuser angle, it was found that a properly designed ejector can increase the system

COP over 14.5%.The result of this study showed that the increment of the COP is due to increase in ambient temperature were more influenced by the decrease in the input power of the compressor than that of the cooling capacity.

Table No. 1 The Summary of numerical works on the Standard ejector systems

Author	Year	COPimp	Working Fluid
Kornhauser	1990	21%	R11,R12,R22 ,R113,R114, R500,R502,R 717
Liu et al.	2002	11.9% -21.6%	R600a,R717, R290
Nehdi et al.	2007	22%	±20 refrigerants
Deng et al.	2007	22%	CO2
Bilir and Ersoy	2009	10.1% - 22.34%	R134a
Sarkar	2010	11.9%-21.6%	R290, R600a, R717.
Sumeru et al.	2013	1.99%-23.03%	R22

Table No. 2 -The Summary of experimental works on the Standard ejector systems

Authors	Year	COPimp	Working Fluid
Harrell and Kornhauser	1995	3.9-7.9%	R134a
Menegay and Kornhauser	1996	3.2-3.8%	R12
Disawas and Wongwises	2004	Higher than standard cycle	R143a
Ozaki et al.	2004	20%	CO_2
Wongwises and Disawas	2005	Higher than standard cycle	R134a
Deng et al.	2007	22%	CO_2
Chaiwongsa and Wongwises	2007	Higher than standard cycle	R134a
Elbel and Hrnjak	2008	7%	CO_2
Elbel	2011	7%	CO_2
Lucas and Koehler	2012	17%	CO_2

III. WORKING OF TWO-PHASE EJECTOR

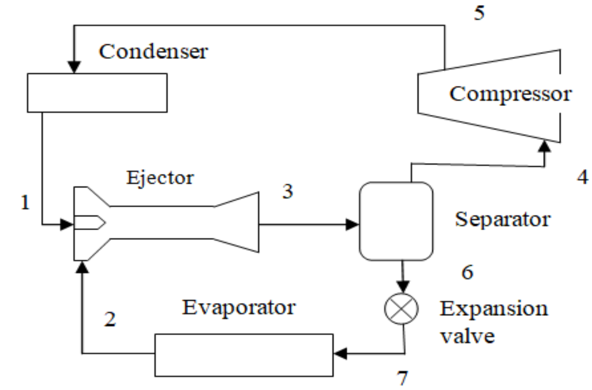
Figure 1a shows the schematic diagram of EERS and Figure 1b shows its corresponding p-h diagram. The EERS contains the same components as that of conventional vapor compression refrigeration system such as compressor, a condenser, an expansion device and an evaporator. However, the ejector and a separator is incorporated into it to recover the expansion process losses and to increase suction pressure of compressor. The saturated vapor from the separator is sucked (point 4 and is compressed to higher pressure and temperature. The compressed vapors are allowed to pass through the condenser (point 5) and the superheat vapor is condensed rejecting the heat to the surrounding (Qcon). The saturated liquid at high pressure exit from condenser (point 1) is called motive or primary stream. Again the saturated liquid from the separator is expanded (point 7) in the expansion device and passed through the evaporator where the heat is received from the cooled space or object (Qeva). The low pressure saturated vapor exit from the evaporator (point 2) is called secondary stream. Now the primary and secondary flows are mixed in the constant area ejector and the mixture of liquid and vapor exits the ejector at higher pressure than the evaporator pressure.

The schematic of a constant area two phase ejector is shown in fig.2 The ejector is mainly consists of two converging and diversion nozzle. The motive flow exit from the condenser is expanded in the motive nozzle (point 1) to a certain back pressure (Pb). The Pb is lower than the evaporator pressure (Peva). Due to this pressure difference the secondary flow is entrained inside and expanded through the suction nozzle (P2b) where both the streams are mixed together at the entrance of constant area mixing chamber where the pressure is assumed to be higher than the back pressure. The phenomenon of chocking takes place in the constant area section where the pressure increases and the velocity decreases (point 3m). Finally in the diffuser, the pressure further increases and at the outlet of the ejector (point 3). The pressure is called as the separator pressure (Psep).

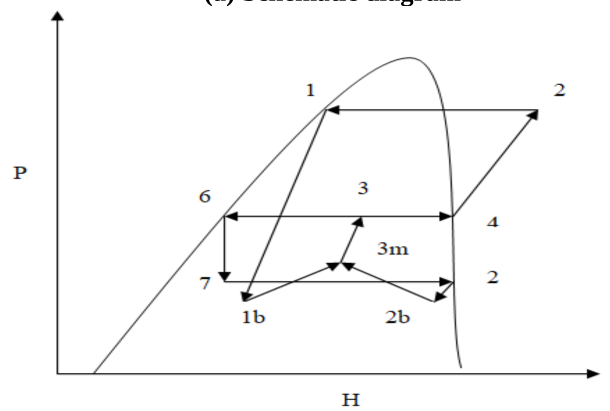
IV. SYSTEM ANALYSIS

The system analysis is done considering the following assumptions;

- The flow is considered as steady state one dimensional flow.
- The separator is assumed to be 100% efficient.
- Primary and secondary fluids upstream velocities are zero.
- The refrigerant leaving the evaporator and condenser is in 100% saturated condition.
- The friction losses are considered in terms of isentropic efficiencies.



(a) Schematic diagram



(b) p-h diagram

V. ANALYSIS

The ejector analysis is done by using set of governing equations like conservation of mass, momentum and energy. Applying these governing equations to each element of the ejector some equations are developed.

-Motive nozzle

From the pressure enthalpy diagram of EERS. The sp. enthalpy of motive fluid at the exit of nozzle is given by;

$$h_{1b} = h_1 (1 - \eta_{mn}) + \eta_{mn}\, h_{1b,is} \qquad (1)$$

where h_{1b} is the sp. enthalpy in (kJ/kg) of primary fluid and η_{mn} is the isentropic efficiency of motive nozzle.

Now the velocity u_{1b} of motive stream in m/sec at nozzle exit is;

$$u_{1b} = [2(h_1 - h_{1b})]^{0.5} \qquad (2)$$

G_{1b} is mass flux in kg.m^2/sec is calculated from;

$$G_{1b} = \rho_{1b} u_{1b} \qquad (3)$$

Where ρ_{1b} is the density in kg/m^3 of fluid at motive nozzle exit.

Applying principle of conservation of mass, the area of motive stream can be estimated as follows;

$$a_{1b} = \frac{m_{tot}}{\rho_{1b} u_{1b}(1+\omega)} \qquad (4)$$

Where ω is the entrainment ratio which is the ratio of secondary mass flow rate (m_s) in kg/sec to that of primary mass flow rate (m_p).

$$\dot{m}_p = \frac{m_{tot}}{(1+\omega)} \qquad (5)$$

- Suction nozzle

In the similar way the equations are derived for the suction nozzle

$$h_{2b} = h_2 (1 - \eta_{sn}) + \eta_{sn}\, h_{2b,is} \qquad (6)$$

$$u_{2b} = [2(h_2 - h_{2b})]^{0.5} \qquad (7)$$

$$G_{2b} = \rho_{2b} u_{2b} \qquad (8)$$

$$a_{2b} = \frac{m_{tot}}{\rho_{2b} u_{2b}(1+\omega)} \qquad (9)$$

$$\dot{m}_s = \frac{m_{tot} \cdot \omega}{(1+\omega)} = \frac{Q_{eva}}{h_2 - h_7} \qquad (10)$$

wher h_{2b} is the sp. enthalpy of secondary stream and η_{sn} is the isentropic efficiency of suction nozzle.

-Mixing section

Now the velocity u_{3m} and mass flux G_{3m} at the constant area mixing chamber can be found by applying the conservation of momentum equation;

$$u_{3m} = \frac{P_b\,(a_{1b} + a_{2b})}{m_{tot}} + \frac{u_{1b}}{(1+\omega)} + \frac{u_{2b} \cdot \omega}{(1+\omega)} - \frac{P_{3m} \cdot a_{3m}}{m_{tot}} \qquad (11)$$

$$G_{3m} = \rho_{3m} u_{3m} \qquad (12)$$

By applying principle of conservation of energy, sp. enthalpy of the mixture is calculated as;

$$h_{3m} = \frac{h_1 + \omega . h_2}{(1+\omega)} - \frac{u_{3m}^2}{2} \qquad (13)$$

Now to get a constant flow rate at the exit of ejector, equation no 14 should be satisfied;

$$a_{3m} \cdot u_{3m} \cdot \rho_{3m} = \dot{m}_{tot} \qquad (14)$$

- Diffuser

By applying conservation of energy, the sp. enthalpy at exit of diffuser h_3 can be found by;

$$h_3 = \frac{h_1 + \omega . h_2}{(1+\omega)} \qquad (15)$$

Also, isentropic enthalpy at the same is;

$$h_{3,is} = \eta_d\,(h_3 - h_{3m}) + h_{3m} \qquad (16)$$

VI. EERS PERFORMANCE CHARACTERSTICS

The cooling capacity is given as;

$$Q_{eva} = \dot{m}_p\,(h_2 - h_7) \qquad (17)$$

Also the compressor power;

$$P_{com} = \frac{\dot{m}_p\,(h_{5,is} - h_4)}{\eta_{com}} \qquad (18)$$

Where η_{com} is the isentropic efficiency of the compressor

which is assumed to be 0.75

$$COP = \frac{Q_{eva}}{P_{com}} \qquad (19)$$

The entrainment ratio ω is;

$$\omega = \frac{\dot{m}_s}{\dot{m}_p} \tag{20}$$

Now the pressure lift we get by using the ejector is;

$$P_{lift} = \frac{P_{sep}}{P_{eva}} \tag{21}$$

Finally the ejector efficiency is calculated as;

$$\eta_{ej} = \omega \, \frac{(h_c - h_2)}{(h_A - h_B)} \tag{22}$$

VII. DESIGN OF EJECTOR

The main components of a two phase ejector are the motive nozzle, the suction nozzle, the constant area mixing section and the diffuser. To determine the ejector diameters is what the ejector design aims at. Whereas the lengths and the angles are considered as a percentage of ejector lengths and according to recommended angles.

The throat diameter of motive nozzle is determined by the use of Henry and Fauske model which considers the metastable effect in the motive nozzle. The other cross sections are determined according to homogeneous equilibrium model.

$$G_c^2 = \left(\frac{x_0 v_{vt}}{n_t p_t}\right) + (v_{vt} - v_{l0}) \left[\frac{(1-x_0).N.ds_{lt}}{(s_{vt}-s_{lt})dp_t} - \frac{x_0.c_{pvt}.\left(\frac{1}{n_t}-\frac{1}{\gamma}\right)}{p_t\,(s_{v0}-s_{l0})}\right]^{-1} \tag{23}$$

And

$$(1-x_0)v_{l0}(p_0 - p_t) + \frac{x_0}{\gamma-1}(p_0 v_{v0} - p_t.v_{vt}) = \frac{[(1-x_0)v_{l0} + x_0 v_{vt}]^2}{2} G_c^2 \tag{24}$$

Where p_0 is the stagnation pressure in kPa of the fluid and the subscripts v, t and l signifies the vapour, throat and liquid refrigerant respectively. p_0 and x_0 will be considered as zero. With the help of following formulas the diameters are determined;

For the motive nozzle throat diameter;

$$D_{mn,t} = \sqrt{\frac{4\dot{m}_p}{\pi.G_c}} \tag{25}$$

$$D_{mn,e} = \sqrt{\frac{4.\dot{m}_p}{\pi.G_{1b}}} \tag{26}$$

For suction nozzle exit;

$$D_{sn,e} = \sqrt{\frac{4.\dot{m}_s}{\pi.G_{2b}}} \tag{27}$$

For constant area mixing section;

$$D_{ms} = \sqrt{\frac{4.\dot{m}_{tot}}{\pi.G_{3m}}} \tag{28}$$

To solve these equations a simulation program is developed in EES (Engineering Equation Solver) and for analyzing the performance parameters of ejector expansion refrigeration system. By giving the condensing and evaporating temperature limits we will get the nozzle throat diameters, the suction nozzle diameter, the mixing section diameter and also the diffuser exit diameter as the program output.

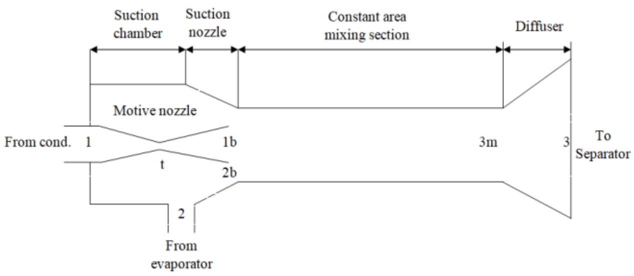


Figure 2 – Constant area mixing ejector

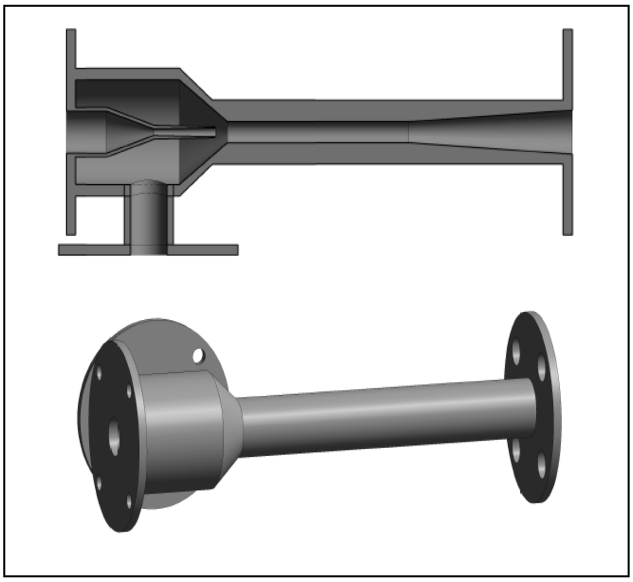


Figure 3 – Constant area two phase ejector 3D model.

VIII. CONCLUSION

- In this paper different studies done on ejector expansion refrigeration system have been studied and the benefit of using ejector instead of expansion device is known. There is no maintenance as there is no moving part and it increases the suction pressure of the compressor which ultimately reduces the compression work of compressor thus increasing the COP.
- There is a remarkable progress in the COP of the system as studied by various authors.
- A simulation program is developed for performance evaluation of the system and to determine the ejector dimensions for refrigerant R-22 for an air–conditioning application. By using the set of formulas, the diameters of the nozzle throat, suction nozzle, constant area mixing chamber and diffuser exit are obtained and on that basis a model is created.

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