A Review on R-290 in Split-type Inverter Air Conditioner using Ejector based Refrigeration Cycle

Mayur Hamjade, Priya Zine, Tejasvi Sonawane, Dr. K. V. Mali

Mechanical Engineering Department, MAEER's MIT College of Engineering, Pune, Maharashtra, India.

Abstract

This paper presents an experimental study and determination of performance of a split-type inverter air conditioner based on refrigeration cycle using ejector. Instead of conventional expansion device ejector recovers the expansion process loss by generating isentropic expansion process and also by increasing the cooling capacity and decreasing the power consumption. Hence the objective of proposed research is to replace the capillary tube with an optimized ejector and also compare the performance of R-22 with R-290. It is expected to considerably increase COP as well as reduce the emission of greenhouse gases in the environment. Laws of conservation of mass, momentum and energy equation would be used to determine the dimensions of ejector. With the use of ejector and R-290 the COP of an AC is expected to increase by about 32.90% was claimed by K. Sumeru et al. [1].

Keywords: Split type A.C., Ejector, Design, Entrainment ratio, Propane, Montreal Protocol.

1. Introduction

Split-type air conditioners have a wide range of use in domestic and commercial buildings. It is not a hidden fact that air conditioners (A.C.) consume huge amount of energy to maintain indoor air temperature. Air conditioning systems use approximately 49% of the total energy consumption of a building [1]. As a result, even a small improvement in the performance of the system will generate a significant impact on energy saving. Use of an ejector as an expansion device is one of the alternative ways of improving the performance of a refrigeration cycle. A typical standard Vapor compression refrigeration cycle uses a capillary tube, a thermostatic expansion valve and other throttling devices to reduce refrigerant pressure from condenser section to the evaporator section. Theoretically, the pressure drop in conventional expansion devices is considered as an isenthalpic process (constant enthalpy). An isenthalpic process causes decrease in the evaporator coolingcapacity because of the energy loss in the throttling process. This loss costs a lot in the C.O.P of the standard VCRC. To recover this energy loss, isentropic (constant entropy) is required in the expansion process. An ejector can generate isentropic conditions in the throttling process. An ejector is a device that uses the expansion of a high-pressure fluid to entrain and provide compression power to fluid at a lower pressure. A high-pressure fluid (the motive fluid), enters the ejector and expands through a converging-diverging nozzle (the motive nozzle), to a low pressure and high velocity. At the same time, a separate stream of fluid (the suction fluid), at a much lower pressure than the motive fluid, enters the ejector and accelerates through the suction nozzle, which is generally only converging. The two streams of fluid meet in the mixing section of the ejector, where momentum is transferred from the motive fluid to the suction fluid. The pressure at which the two fluids meet at the beginning of the mixing section is called the mixing pressure, and this pressure must be lower than the ejector suction pressure so that suction to could occur. The above process results in the reduction in losses and simultaneously increases the COP.

According to Montreal Protocol (1987) and Kyoto Protocol(1997) certain CFC's and HCFC's have been banned in developed countrie and are going to be banned in developing countries until 2030. So it has been a demand of time to search for promising alternative refrigerants. R-290 can prove as a better alternative for the above refrigerants.

2. Literature Survey

Karhouser et al. [8] performed the thermodynamic analysis of the standard vapour compression refrigeration cycle using an ejectoras an expansion device for the first time. His study was constrained to one dimensional model. He obtained COP_{imp} up to 21% over the standard cycle.

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A Numerical analysis to evaluate the effect of ejector geometry on system performance using multiple refrigerants was performed by Nehdi et al. [5]. He obtained COP_{imp} up to 22% over the standard cycle. Experimental performance of window air conditioner using R-290 was evaluated by Devotta [3]. Under drop-in conditions, the cooling capacity with R-290 was 6.6% lower and EER was higher by 7.9% compared to R-22. Due to lower mass flow rate of R-290 pressure drops in both condenser and evaporator were found lower. Chang [4] reported experimental results for several compressor speeds and showed that the cooling capacity using R-290 is about 14% lower and EER is marginally higher than R-22. Corberan [5] showed that R-290 practically provides around 5% higher energy efficiency and 10% lower capacity due to improvement in compressor efficiency under drop-in conditions. The heat transfer coefficients in evaporator are higher.

Jung [6] reported that the heat transfer coefficients for R-22 were lower compared to R-290. Lee [4,5] proposed non- dimensional heat transfer correlations for condensation and for evaporation which shows good agreement with experimental.

Fernando [6] reduced R-290 charge by using minichannel aluminum heat exchanger and suggested modified correlations for both condensation and evaporation which agrees within 15% with experimental.

3. Ejector based refrigeration cycle

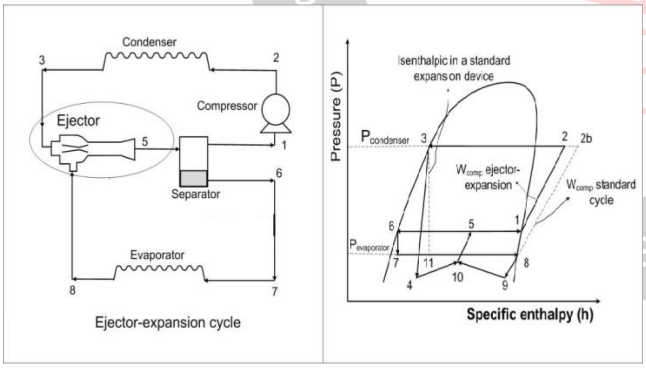

Figure 3.1

(a) Ejector based refrigeration cycle
(b) Ph diagram for ejector based refrigeration cycle and standard cycle[1]

Figure 3.1(a) represents the schematic diagram of ejector based refrigeration cycle. Figure 3.1(b) shows Ph diagram for the standard V.C.R.C and ejector based refrigeration cycle. In fig 3.1 (b) points 8-2b-3-11-8 represents the standard vapor compression refrigeration cycle. Point 3 to point 11 represents the isenthalpic throttling process, while the isentropic throttling process occurs from point 3 to point 4. There are two flows in the ejector based refrigeration cycle: primary flow and secondary flow. The compressor circulates primary flow through the condenser, ejector and separator (point 1, 2, 3, 4, 10, 1), whereas the secondary flow is circulated in the evaporator, ejector and separator (point 6, 7, 8, 9, 10, 5, 6). The primary and secondary flows mix at the constant-area and diffuser (point 10, 5). The sudden change in pressure and velocity of the refrigerant from point 9 - 10 is attributed to normal shock, induced downstream ofthe constant area mixing section. This shock causes a compression effect and thus a sudden drop in the refrigerant flow speed is achived [7]. As shown in Figure 3.1(b), the pressure at point 1 is higher than that of the suction pressure in the standard cycle (point 8). This means that the compressor work of the ejector based refrigeration cycle is lower than that of the standard V.C.R.C. Further, the refrigerant passes through the capillary tube in process 6-7 where the expansion of refrigerant takes place at constant enthalpy and pressure decreases. The refrigerant passes through the evaporator at constant pressure where enthalpy increases in process 7-8. In process 8-9 the vapor refrigerant again enters into the ejector where there is slight decrease in the enthalpy and pressure. At constant area mixing chamber, pressure increases but enthalpy decreases from 9-10. Again it passes through the diffuser and the cycle continues.

4. Design and Simulation

The ejector based refrigeration cycle has been designed by applying mass, mumentum and energy conservation equations. To simplify the design the following considerations were made:

i. The pressure drop in condenser, evaporator, separator and the tube connections has been neglected.

ii. Heat transfer from system to surrounding takes place only through the condenser.

iii. The refrigerant conditions are saturated at the evaporator and condenser outlets.

iv. The vapor stream and liquid stream from the separator are both saturated.

v. Isenthalpic flow takes place across the expansion valve or throttle valve.

vi. At the inlet of the constant area mixing section of the ejector both the motive stream and the suction stream reach the same pressure.

vii. The expansion efficiencies of the motive stream, suction stream and diffuser of the ejector are given constants.

viii. At the ejector inlet and outlet kinetic energies of the refrigerant can be neglected.

Based on the above assumptions the following design procedure could be considered.

The total mass flow rate of the system for which the ejector has to be designed is calculated.

Dryness fraction at point 5 is considered as (x_5). Entrainment ratio (z) is calculated as

$$z = \frac{1 - x_5}{x_5}$$

The pressure at point 4 (motive nozzle exit) should be less than the discharge pressure for the refrigerant for entrained. P_4 has to be assumed and iterated for maximum COP.

Velocities at the inlet of motive nozzle and suction nozzle are determined as

$$u_3 = m_4/(\rho_3 a_3)$$

$$u_8 = m_9/(\rho_8 a_8)$$

Dimensions of motive nozzle are calculated by using energy conservation and continuity equation as:

$$u_4 = \sqrt{2\left(h_3 + \frac{u_3^2}{2}\right) - h_4}$$

$$a_4 = \frac{a_3 u_3}{u_4}$$

$$d_4 = \sqrt{\frac{4a_4}{\pi}}$$

Dimensions of motive nozzle are calculated by using the same equation used for motive nozzle. The geometric ejector parameter for the ejector is given by

$$q = \frac{a_4 + a_9}{a_4}$$

The dimensions and properties for mixing section are calculated by applying mass, momentum and energy conservation equation as follows:

$$h_{10} = \frac{(m_4 h_4 + m_9 h_9)}{m_1}$$

$$u_{10} = \sqrt{2\left\{\left[\frac{1}{1+z}\left(h_4 + \frac{u_4^2}{2}\right)\right] + \left[\frac{z}{1+z}\left(h_9 + \frac{u_9^2}{2}\right)\right]\right\} - h_{10}}$$

$$P_{10} = \frac{\left[P_9(a_4 + a_9) + \left(\frac{1}{1+z}u_4\right) + \left(\frac{z}{1+z}u_9\right) - u_{10}\right]}{a_4 + a_9}$$

$$\rho_{10} = \frac{(\rho_4 a_4 u_4) + (\rho_9 a_9 u_9)}{u_{10}(a_4 + a_9)}$$

Consider the diffuser angle as 7° [8], then the dimensions and properties of diffuser section are calculated by using energy conservation and continuity equation, geometric relations.

$$h_5 = h_{10} + \eta_2\left(\frac{u_{10}^2}{2}\right)$$

$$u_5 = \sqrt{[2(h_{10} - h_5)] + u_{10}^2}$$

$$a_5 = \frac{(a_4 + a_9)u_{10}}{u_5}$$

$$d_5 = \sqrt{\frac{4a_5}{\pi}}$$

$$P_5 = \frac{P_{10}}{\eta_2}$$

$$l = \frac{(d_5 - d_9)}{\tan(7^\circ)}$$

5. Dimensions of Ejector

Sr.No.	Description	Symbol	Value
1	Motive nozzle throat diameter	d_t	1.13mm
2	Diameter of primary nozzle exit	d_{et}	12.7mm
3	Diameter of mixing chamber exit	d_m	12.7mm
4	Length of mixing section	L_m	59mm
5	Length of diffuser section	L_{diff}	8.5mm
6	Diameter of diffuser section	d_{diff}	19.12mm

6. Performance Evaluation

The cooling COP of the ejector based refrigeration cycle can is calculated by

$$COP = q_e/w_c$$

and the cooling COP of corresponding ejector based refrigeration cycle is given as

$$COP_e = (h_8 - h_3)/(h_{2b} - h_8.)$$

The improved COP using ejector can be evaluated by

$$COP_i = (COP - COP_e)/COP_e$$

7. Conclusion

The literature survey done regarding the ejector based refrigeration cycle directed us to a conclusion that the idea of replacing capillary tube could lead to the following results:

1. Increasing the coefficient of performance.
2. Decrease in emission level of CFC's.
3. Achievement of lower air conditioning temperature.

Further, the ejector design process have been initiated using mass, momentum and energy equations which would be proceeded with experimental testing of the ejector air conditioning system.

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