Experimental investigation of an ejector refrigeration system using R-134a

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Abstract: In the present study, the performance of ejector refrigeration system is investigated using R-134a as working fluid. Six ejectors of different geometries are used to experimentally evaluate the coefficient of performance, cooling capacity, pressure lift and critical condenser pressure for fixed generator and evaporator temperature, while varying the condenser temperature. It is observed that an optimum area ratio of the ejector is required to obtain better performance and is found that a higher value of area ratio gives maximum COP for the specified conditions but for a lower value of critical condenser pressure. The critical condenser pressure for the ejector of area ratio 10.08 comes out to be 778.9 kPa whereas it is 916.72 kPa for the ejector of area ratio 6.451 while working at generator and evaporator temperature of 80 °C and 15 °C respectively. Further, the ejector having a shorter length of constant area section is found to excel. It is also concluded that for lower evaporator pressure, higher pressure lift is obtained from the same ejector.

Keywords: Ejector, refrigeration, Vapor compression, experiments.

1. Introduction
Refrigeration and air-conditioning are now essential requirements for producing human comfort and various other cooling applications. Cooling system working on vapor compression cycle consumes a major fraction of the electrical energy, about 40 % during peak hours, produced in the world [1]. The ejector refrigeration system (ERS) is an emerging technology that does not require electricity for its operation rather utilize heat energy for producing refrigeration effect and the required heat energy can be provided by using solar thermal systems, industrial waste heat, or various other renewable resources. Figure 1 shows a schematic diagram of the ERS.

![Figure 1 Ejector refrigeration system](image-url)
The key component of ERS is ejector and figure 2 shows the basic details of an ejector. Ejector consists of mainly four parts: primary nozzle, suction chamber, constant area section, and diffuser section. The primary nozzle of convergent–divergent type is used to accelerate and expand the primary fluid coming out from the generator. Consequently, the expansion of the primary refrigerant creates low-pressure at the exit of the primary nozzle and it works as a motive to entrain low-pressure vapor (also known as secondary fluid) from the evaporator. Both the primary and secondary fluids get completely mixed in the constant area section of the ejector at constant pressure. The speed of the mixed fluid reduces to subsonic region due to the generation of normal shock wave in the constant area section and thereafter in the subsonic diffuser, the recovery of the pressure is achieved.

Various researchers analyzed and predicted the performance of ejector using various mathematical models. Keenan et al. [2] first developed constant pressure mixing theory. Huang et al. [3] formulated 1-dimensional model to analyze the behavior of a single phase ejector working in double choking mode against critical condenser condition. They experimentally validated their model using eleven different geometry ejectors. R-141b was used as refrigerant in the system. Chou et al. [4] proposed a similar 1-D model to predict the maximum mass flow rate of secondary fluid entrained by the ejector while operating in double choking mode. Selvaraju & Mani [5] developed a mathematical model to predict the performance of ejector for many environment-friendly refrigerants at various operating conditions. Angelino & Invernizzi [6] analyzed a Rankine cycle powered ejector refrigeration system. Zhu et al. [7] developed a shock circle model for predicting the performance more accurately with an assumption of uniform radial pressure and non-uniform velocity distribution. Chen et al. [8] carried out 1-D investigation of the performance of ejector for sub-critical conditions also. Kumar and Sachdeva [9] proposed 1-D model to calculate all the dimensions of an ejector working at fixed operating conditions. They introduced a few novel concepts of Prandtl’s mixing length, Prandtl-Meyer expansion wave, Kelvin-Helmholtz instability, and Baroclinic effect to calculate various diameters, mixing length, nozzle exit position etc. to their 1-D model. Although, mathematical analysis has its own importance, but still experimental investigations are required to evaluate the performance of any system in real conditions. A very few researchers had experimentally analyzed the performance of an ejector refrigeration system. Brunner et al. [10] designed a variable flow ejector for re-circulating hydrogen in the fuel cell system to increase its performance. The mass flow rate of the ejector was controlled electronically and it exhibited excellent performance compared to Ballard’s specification for recirculation flow rate. Butrymowicz et al. [11]
investigated an ejector refrigeration system for generator temperature less than 75 °C, while operating with Isobutane in the system. They analyzed the ejectors for off-design conditions as well and found that ERS can effectively compete with absorption system at low generator temperature conditions. Zhu and Jiang [12] carried out an experimental and numerical investigation of the shock wave characteristic on the entrainment ratio of an ejector. They captured the structure of the shock wave in the mixing chamber using optical Schlieren technique. The experimental observations of the ejector refrigeration system are still needed to understand and improve its performance characteristics. The present paper is about an experimental investigation on ERS using six different ejector geometries working at the same generator temperature. The novelty of this work lies in the performance analysis of an indigenously designed experimental set up of ERS for different ejector geometries. The main focus is on the analysis of critical condenser pressure obtained with different ejectors.

2. Experimental Investigation

An actual experimental setup of ERS, photograph shown in figure 3, is fabricated using R-134a as a refrigerant. The test bench consists of four main parts: generator, condenser, evaporator, and ejector. Various measuring instruments like pressure gauges, temperature sensors & indicator, voltmeter, ammeter are used to accurately measure the different variables.

2.1. Experimental apparatus

Figure 4 shows the schematic of the experimental setup. It consists of two main cycles. One is the power cycle comprising generator, pump, and condenser, which is used to produce high-pressure vapors of the refrigerant R-134a. Another cycle is refrigeration cycle that includes an expansion device and evaporator. Both the cycles coordinate each other through ejector and the condenser. Ejector utilizes the energy of power cycle and produces refrigeration effect in the evaporator.

2.2. Ejector

The ejectors are made-up of brass in three interchangeable sections i.e. the suction chamber, constant area section, and diffuser section. The suction chamber and diffuser are common to all the ejectors. The convergent-divergent primary nozzle is placed perfectly in line with the center of the constant area section. Three primary nozzles of different throat and exit diameters but same inlet diameter are considered in the analysis. Moreover, two constant area sections of different diameter and length have been used to obtain six different geometries of the ejector. The ejector geometries analyzed are named A to F as shown in table 1.

2.3. Heat exchangers

The tube in shell type heat exchangers are employed as generator, condenser, and evaporator. Copper tube of 6.35 mm diameter is used in the generator and the condenser; whereas the evaporator is made up of 9.525 mm copper tube. The length of the copper tube in the evaporator, generator, and the condenser is 15.24, 24.384 and 30.48 meters respectively. These lengths are turned in the spiral form and then are placed in different plastic containers insulated by 38.1 mm thick industrial insulation. The copper tube in all these heat exchangers is used to circulate refrigerant while the plastic container carries water as external fluid. A 4.5 kW electric heater with a current controller device is used to heat 40% concentrated glycol solution which further provides heat to the refrigerant of the system. A mechanical stirrer is used to maintain a uniform temperature of glycol solution. All connecting pipes used in the test facility are insulated by 9 mm thick industrial pipe insulation to minimize heat interaction with the surrounding.
Figure 3 Photograph of the experimental setup

Table 1 Ejector geometries used in the experiments

| Sr. No | Ejector | Primary nozzle geometry | Secondary nozzle geometry | Area Ratio |
|--------|---------|-------------------------|---------------------------|-----------|
|        |         | Inlet Area (A₁₁) (mm²) | Throat Area (A₂₁) (mm²)  | Exit Area (A₃₁) (mm²) | Divergence Angle (θ) (Degree) | Area of CAS (A₄) (mm²) | Length of CAS (L) (mm) | NXP |         |
| 1      | A       | 31.67                   | 3.14                      | 10.293                   | 21.6                       | 31.67                   | 80                          | 2.54 | 10.081   |
| 2      | B       | 31.67                   | 3.14                      | 10.293                   | 21.6                       | 31.67                   | 120                         | 2.54 | 10.081   |
| 3      | C       | 31.67                   | 4.91                      | 15.975                   | 21.12                      | 31.67                   | 80                          | 2.54 | 6.45     |
| 4      | D       | 31.67                   | 4.91                      | 15.975                   | 21.12                      | 31.67                   | 120                         | 2.54 | 6.45     |
| 5      | E       | 31.67                   | 7.07                      | 20.589                   | 21.2                       | 67.201                  | 80                          | 2.54 | 9.51     |
| 6      | F       | 31.67                   | 7.07                      | 20.589                   | 21.2                       | 67.201                  | 120                         | 2.54 | 9.51     |

2.4. Measuring instruments
J-type thermocouples with an accuracy of ±0.1 °C are used to measure the temperature at various locations. Bourdon-tube type pressure gauges calibrated for R-134a refrigerant are used to measure pressure in the test facility having an accuracy of ±2 PSI. Voltage and current supplied to the heating element placed inside the evaporator and the generator are measured using analog voltmeter and ammeter with an uncertainty of ±5 V & ±0.05 A respectively. A current controller is used to control the amount of current supplied to the heating element in the evaporator and the generator. Total uncertainty in experimental setup for COP of the system is found to be less than 5%.

3. Operating parameters and experimental procedure
The glycol solution is heated and maintained at 90 °C, whereas the temperature of the vapor refrigerant coming out from the generator is settled at 84 °C. To minimize the effect of heat loss from the vapor refrigerant while passing through the connecting pipe, superheating of 4 °C above the saturation condition
is maintained in all the experiments. The temperature of the refrigerant in the evaporator is kept constant and the experiments are conducted for 2 different evaporator temperatures i.e. 10 °C & 15 °C. Water is circulated in the condenser to take the latent heat of the refrigerant. This water after gaining heat from the condenser is drained into a large separate container.

**Figure 4** Schematic diagram of the experimental setup

4. **Results and discussions**

In present study, the ERS is experimentally investigated while varying the condenser temperature, ejector geometry and the evaporator temperature.

The energy consumed by the pump is neglected because of its low value as compared to energy input in the generator. Therefore, COP of the ERS is evaluated as follows:

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\text{COP} = \frac{\text{refrigeration effect}}{\text{Heat supplied to generator}}
\]  

(1)

The refrigeration effect produced is same as the electricity consumed by the submerged electric heater placed in the evaporator to maintain its temperature. Similarly, the amount of heat supplied to the generator is calculated. Figure 5 shows the variation of COP at different condenser temperature for all the 6 ejectors while keeping the evaporator at 10 °C.
The double choking phenomenon can be observed in the same figure. It also predicts that the critical condenser pressure is less for the ejectors of large area ratios. Increase in condenser pressure above the critical condenser pressure makes the system to run in single choking mode and the performance of the system decreases significantly. Higher COP is obtained for large area ratios but the working condenser pressure range of the system is less. These results are in the same fashion as obtained by the various researchers. It is also observed that ejector A provides more COP in comparison to the ejector B because it has shorter length of the constant area section. A long length of constant area section increases the frictional losses inside the ejector. Similar trends are obtained at an evaporator temperature of 15 °C as shown in figure 6. Increasing the evaporator temperature increases the COP of the system and at condenser temperature of 30.4 °C, the COP of the system is 12 percent more than the system running at 10 °C operating with ejector A. COP of the system remains constant for the condenser pressure less than the critical condenser temperature as the system operates in double choking mode.
Cooling effect obtained in the ERS at 10 °C evaporator temperature can be visualized in figure 7 and it is less for the ejector of low area ratio ejectors D and E while operating the system at critical condenser mode. Figure 8 predicts the cooling capacity of the system for the evaporator temperature of 15 °C with the condenser temperature. Similar trends are obtained but with the more cooling capacity at higher evaporator temperature. It is also observed that the ejector with less throat area i.e. ejector A provides more cooling effect.

**Figure 7** Variation of cooling capacity (W) with condenser temperature (°C) and 10 °C evaporator temperature

**Figure 8** Variation of cooling capacity (W) with condenser pressure (°C) and 15 °C evaporator temperature

As the primary nozzle throat area decreases, the mass flow rate of the primary fluid also decreases but this provides more passage for the secondary fluid and hence increases the entrainment ratio and cooling capacity. However, it decreases the critical condenser pressure of the system.
Pressure lift is the ratio of critical condenser pressure to the evaporator pressure. Higher pressure lift is required to run the system for a wide range of condenser pressure. Figures 9 depicts the pressure lift obtained in the ejectors for the evaporator temperatures of 10 °C and it is observed that the increase in length of constant area section decreases the pressure lift even if the area ratio of the ejector is same. Figures 10 shows the similar trend of the pressure lift obtained in the ejectors for the evaporator temperatures of 15 °C, however, it is found that the pressure lift obtained for lower evaporator temperature is higher.

5. Conclusions
The paper experimentally predicts the performance of an ejector refrigeration system (ERS) for fixed generator and evaporator conditions while varying the condenser temperature. It is found that an appropriate area ratio of the ejector is required to obtain maximum performance of the system. Large area
ratio gives better performance but at the expanse of critical condenser pressure of the system. It is also found that for the same area ratio if the length of the constant area section is large in comparison to the optimum length, the performance of the system decreases due to the higher frictional losses inside the ejector. Also, the pressure lift obtained from the ejector is more for lower evaporator temperature.

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Nomenclature

| Symbol | Description | Subscript | Description |
|--------|-------------|-----------|-------------|
| P      | Pressure (kPa) | p         | Primary fluid |
| T      | Temperature (°C) | s         | Secondary fluid |
| L      | Length of constant area section (mm) | evp       | Evaporator |
| A      | Area (mm²) | g         | Generator |
| NXP    | Nozzle exit position (mm) | s         | Secondary fluid |
| COP    | Coefficient of Performance | g         | Generator |
| ERS    | Ejector Refrigeration System |           |             |
| HSV    | Hand Shut Valve |           |             |
| V      | Voltage |           |             |