Performance prediction of ejector refrigeration system using R1234yf as working fluid

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Abstract. An ejector compression system for cooling is introduced which utilizes low-grade energy freely available from renewable resources like solar energy, bio waste etc. In this paper the performance of system is evaluated applying the conservation principles of mass and energy on ejector considering it as a control volume. The behavior of the ratio of mass flow rate of fluid from evaporator to the mass flow rate of fluid from generator, COP and cooling effect by varying the condenser/evaporator temperature have been observed. The refrigerant R1234yf which has low ODP and GWP has been used and the analysis has been done with the help of Engineering Equation Solver (EES) Software. Maximum COP of the system is found to be 0.3458 at the evaporator temperature 10°C, condenser temperature 30°C and generator temperature 70°C.

1. Introduction

Refrigeration and air conditioning systems are becoming essential commodities for a comfortable human life. The conventional vapor compression refrigeration system (VCRS) typically uses a huge quantity of high-grade energy in the form of electricity for the compression process. However due to advancements in technologies, some alternate refrigeration systems are also being devised which consume less or no electricity for producing cooling effect. One of the promising alternative is ejector refrigeration system (ERS) that works on low grade energy. Even the energy obtained at low temperature from the solar thermal systems, bio-gasifiers, Industrial waste etc. can be effectively used in ERS. Simple construction, low maintenance, operation cost and capital cost are the other advantages of ejector refrigeration system in addition to eco-friendly system. The only challenge with this system is low performance as compared to the conventional VCRS. Using renewable energy resources instead of electricity produced by fossil fuels leads to reduction of emission of greenhouse gases [1] and thus environment friendly. K.F. Fong et al. [2] performed experiment on solar-assisted desiccant cooling system. Mehmet Bilgili [3] described the VCRS which worked on solar energy and achieved COP between 3 to 4.

A basic ejector refrigeration system consists of generator, evaporator, condenser, pump, throttling valve and ejector. The compressor is replaced by ejector [4] and thus there is no moving parts except a small pump which consumes negligible amount of electrical energy as compared to the compressor of vapor compression refrigeration system. An ejector is made in four sections namely convergent-divergent nozzle, suction chamber, mixing chamber and a diffuser. The high-pressure and high-
temperature vapor (named as primary fluid) exiting from generator expands to low pressure at the exit of the convergent-divergent nozzle. This pressure gets lesser than the pressure in the evaporator, thus the refrigerant vapor in the evaporator (named as secondary fluid) gets entrained into the ejector. Thereafter mixing of the primary and secondary fluid takes place in the constant area section of the ejector. A normal shock wave occurs which increases the pressure and thus reduces the velocity of the mixed refrigerant lower than the sonic velocity before the subsonic diffuser section. Diffuser recovers the pressure of the working fluid and make it equal to the back pressure of the condenser. The refrigerant gets condensed in the condenser and after expansion in the throttle valve, the secondary fluid produces cooling effect in the evaporator and the remaining primary fluid travels to the generator through pump and the cycle continues.

The ejector, was conceived by Sir Charles Parsons in 1901 for deaerating the condenser of a steam engine. However, it was first time used by Maurice Leblanc in steam jet cooling system in the 1910, [5]. Several researchers developed the one-dimensional ejector system by applying theories of fluid dynamics to the primary and secondary flow with the help of mathematical models [6,7]. An ejector cooling system was developed by Huang et al. [8] for refrigerant R141b and developed empirical relation to predict the geometry of the ejector for a required performance [9]. Experimental performance and their characteristics of an ejector refrigeration system by using HCFC-123 [10] refrigerant was performed by Sun DW and Emes IW [11,12]. Mathematical relation of the area-ratio of ejector at different location was suggested by the different researchers [13,14]. Mani & Sankarlal [15], designed, fabricated and studied the effect of performance parameters on the working of an Ejector Refrigeration System using Ammonia as refrigerant. The results reflected that entrainment ratio and the COP improve with the increasing generator and evaporator temperature, but decrease with increasing condenser temperature. Further, Mani & Selvaraju [16] compared environment friendly refrigerants in ejector refrigeration system and obtained the influence of different parameters. Among all the working fluids selected, R134a gave the best performance. Tyagi & Murty [17] analyzed an ejector refrigeration system with low grade thermal energy obtained from waste steam and exhaust flue gases from automobiles to maintain the generator temperature in the range of 70°C to 85°C and determined the COP between 0.08 to 0.33 with R-11 as refrigerant in the ejector refrigeration system. The evaporator temperature was varied from -3°C to 18°C.

In this cooling system, ejector is an essential component which entrain the secondary fluid from the evaporator with the help of flow energy of primary fluid [18]. The literature reveals that the potential of the ejector refrigeration system (ERS) need to be exploited and further demand for a better and effective ejector refrigeration system and hence the motivation for current work. Novelty of the work lies in the performance analysis of a heat assisted ejector refrigeration system using newly introduced refrigerant R1234yf which has zero ODP and negligible GWP.

2. System Description

Figure 1 presents the line diagram of ejector refrigeration system. The proposed system comprises a power cycle and a refrigeration cycle as discussed further. Power cycle can be visualized between generator and condenser. Heat is supplied to the liquid refrigerant in the generator and thus high-pressure saturated vapor obtained is expanded in the convergent-divergent nozzle of the ejector. Expansion in the nozzle produces low-pressure at exit of the primary nozzle which works as reason to entrain secondary fluid from the evaporator. The entrained secondary fluid and primary fluid get mixed in the constant area section and then the diffuser section increases its pressure. Thus, ejector can be considered as a work producing device which sucks the refrigerant from the evaporator and raises its pressure like compressor of the vapor compression system. The mixture after getting condensed in the condenser again travel to the generator through pump and thus completes the power cycle. The refrigeration after producing cooling effect in the evaporator gets sucked by the ejector and ejector increase its pressure and then latent heat is rejected in the condenser. The liquid refrigerant after expanding in the expansion valve again takes the cooling load in the evaporator and thus the refrigeration cycle continues. P-h diagram for the system is shown in Figure 2.
3. Thermodynamic analysis
Analysis of the system has been done based on the following assumptions

- Nozzle, diffuser and entrainment efficiency are 90%, 90% and 80% respectively.
- Vapor exiting the evaporator is dry and saturated.
- Motive vapor is saturated at the nozzle entry.
- Pressure losses in the whole system are considered to be negligible.
- Combined mass flow rate of primary and secondary fluids in the system is 2 kg/s.
- Kinetic energies at state points 1, 10 and 4 are neglected.

3.1. Governing Equations
Mass, momentum and energy conservation have been used to analyses the system. The number in subscript denote various state points as shown in figure 2. Considering the ejector as a control volume and the process taking place in it as adiabatic, the balanced energy equation for the ejector is

$$m_1 h_1 + m_{10} h_{10} = (m_1 + m_{10}) h_4$$ (1)

$$m_1 h_1 + m_{10} h_{10} = (m_1 + m_{10}) (h_3 + 0.5 V_3^2)$$ (2)

In the convergent-diverging nozzle, the velocity of the incoming motive vapor increases while expanding through it.

$$h_1 = h_2 + \frac{V_2^2}{2g}$$ (3)

The ratio of the actual enthalpy difference to the isentropic enthalpy difference is the efficiency of the nozzle and is expressed as:

$$\eta_n = \frac{h_1 - h_2}{h_3 - h_4}$$ (4)

The ratio of mass flow rate of the secondary fluid from evaporator to the mass flow rate of primary fluid from the generator is called the Entrainment ratio ($\mu$)

$$\mu = \frac{m_{10}}{m_1}$$ (5)
In mixing chamber, the primary fluid and secondary fluid mix to obtain sufficient amount of kinetic energy. Entrainment efficiency is the ratio of the resulting mixture kinetic energy to the energy available to motive vapor.

\[ \eta_e = \frac{(m_1 + m_{10})KE_3}{m_1 KE_2} \] (6)

The ratio of the enthalpy difference that occurred between the entrance to exit stagnation pressure to the kinetic energy after mixing at sate ‘4’ is called diffuser efficiency.

\[ \eta_d = \frac{h_{4'} - h_3}{h_4 - h_3} \] (7)

Enthalpy at ejector exit can be found using the expression as shown.

\[ h_4 - h_3 = \frac{V^2}{2g} \] (8)

Performance of the complete cycle is evaluated by the fundamental thermodynamic equations based on principle of conservation of energy for each component as follows:

Cooling effect

\[ Q_E = (h_{10} - h_8) * \dot{m}_{10} \] (9)

Heat rejected in condenser

\[ Q_C = (\dot{m}_1 + \dot{m}_{10}) * (h_4 - h_5) \] (10)

Constant pressure heat addition in the generator

\[ Q_G = \dot{m}_1 * (h_1 - h_9) \] (11)

The ratio of cooling effect of the system to the heat added in the generator is known as Coefficient of performance (COP)

\[ COP = \frac{Q_E}{Q_G} \] (12)

Cooling effect obtained can be obtained as heat rejected by the condenser to the evaporated heat.

\[ R = \frac{(\dot{m}_1 + \dot{m}_{10}) * (h_4 - h_5)}{(h_{10} - h_8) * \dot{m}_{10}} \] (13)

4. Model Validation

The thermodynamic equations used in the model are solved using Engineering Equation Solver. The present study on ejector refrigeration system has been validated with the results of Tyagi and Murty using the same refrigerants R-11 and R-113. Table 1 shows the comparison of this data. It shows that results obtained from both the models for same operating conditions differ by a maximum error of ±3.032%.

| Performance Parameter | Tyagi and Murty [17] | Current Model |
|-----------------------|----------------------|---------------|
| Generator Temperature | 70°C                 | 75°C          |
| Refrigeration effect  | 5.4375               | 4.3929        |
| \(\dot{m}_1/\dot{m}_{10}\) | 3.5625               | 2.7857        |
| COP                   | 0.2341               | 0.3051        |

Table 1. Comparing results with proposed model
5. Results and Discussion

Analysis of the ejector cooling system for refrigerant R1234yf having zero ODP and low GWP has been done in this work.

Variation of different parameters like coefficient of performance, entrainment ratio $\mu$ and cooling effect with generator temperature are shown in figure 3. It shows that entrainment ratio, COP and cooling effect increase with rise in generator temperature. This can be explained as the entrainment ratio increases, mass flow rate of secondary fluid increases and so does the COP and cooling effect.

![Figure 3. Effect of generator temperature on different parameters at $T_E=0^\circ$C, $T_G=30^\circ$C](image)

Figure 4 shows that an increase in evaporator temperature results an increase in COP, entrainment ratio and cooling effect. Higher evaporator temperature increases the suction pressure in the compressor, thus decrease the compressor work.

Figure 5 shows the variation of the same parameters with the condenser temperature while keeping generator and evaporator temperature constant. As the condenser temperature increases from 20-34 ($^\circ$C), the cooling effect, entrainment ratio, and COP decrease. Cooling effect and COP are directly dependent on entrainment ratio.

![Figure 4. Effect of $T_E$ on different parameters at $T_G=70^\circ$C & $T_c=30^\circ$C](image)
6. Conclusions

Primarily the variation of entrainment ratio, cooling effect and COP has been studied individually with respect to generator temperature, evaporator temperature and condenser temperature. Entrainment ratio is found to have significant impact on all the three parameters. The study is concluded as under:

- There is increase in cooling effect, entrainment ratio and COP with the increase in generator temperature.
- With the rise in evaporator temperature; cooling effect, entrainment ratio and COP increase.
- Cooling effect, entrainment ratio and COP increase with the decrease in condenser temperature.
- Maximum COP of the system is found to be 0.3458 for evaporator temperature 10°C, condenser temperature 30°C and generator temperature 70°C.

Nomenclature

g: Acceleration due to gravity
h: Enthalpy
ṁ: Mass flow rate
Q: Heating Effect
R: Refrigeration Effect
V: Velocity

Subscripts

C: Condenser
d: Diffuser
e: Entrainment
E: Evaporator
G: Generator
n: Nozzle

Greek symbols

\( \eta \): Efficiency
\( \mu \): Entrainment ratio
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