Journal of Basic and Applied Engineering Research
Print ISSN: 2350-0077; Online ISSN: 2350-0255; Volume 1, Number 1; October, 2014 pp. 4-9

http://www.krishisanskriti.org/jbaer.html

Effect of Generator, Condenser and Evaporator Temperature on the Performance of Ejector Refrigeration System (ERS)

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Abstract: **Conventional vapor compression refrigeration system consumes a large quantity of high grade energy. This energy can be generated by the combustion of fossil fuels which produces air pollutants such as oxides of nitrogen and sulphur. To overcome this problem, ejector refrigeration system (ERS) which is powered by low grade thermal energy such as waste heat from industrial processes, exhaust emissions from internal combustion engines and solar energy, is believed to be most effective one. This paper presents a brief review of the working principle of the ejector refrigeration system and the performance of this system based on the works of different researchers available in the literature. The effects of different controlling parameters of the ejector refrigeration system such as generator temperature, evaporator temperature, and condenser temperature on the performance have been discussed.**

1. INTRODUCTION

In recent times most of the cooling and refrigeration system is based on the mechanical vapor compression system which fundamentally takes high quality form of energy (electrical energy) to power compressor unit. The energy required to operate such a process can be generated by the combustion of fossil fuels and thus contributes to the generation of air pollutants, such as oxides of nitrogen (NO_x) and sulphur (SO_x), carbon monoxide and carbon dioxide. These pollutants have adverse effects on human health and the environment. In addition, MVC (mechanical vapor compression) refrigeration and cooling cycles use chlorofluorocarbon compound (CFCs) which are not environment friendly. These, upon release, contribute to the destruction of the protective ozone layer in the upper atmosphere. Ozone depletion and possible global warming by halogenated chlorofluorocarbons have become international issues due to the potential harm to the environment. This tendency is likely to threaten the aims set by many developed and developing countries for the upcoming decades to develop an environmental friendly refrigeration system which can utilize energy in a most efficient way. Vapor ejector refrigeration (VER) has become a topic of interest for these reasons as it is a heat-operated system utilizing low-grade energy. Using low grade thermal energy (waste heat from industrial processes, exhaust emission of internal combustion engine) instead of electricity to operate a refrigerator can have important environmental benefits, especially when it is powered by a renewable energy source (e.g. solar thermal). Ejector refrigeration is one of the most promising technologies because of its relative simplicity and low capital cost when compared to other heat operated refrigeration system such as vapor absorption refrigeration system [1]. Furthermore, ejectors don't have moving parts, and thus require little maintenance and have a long life span. But the main disadvantages of this system are its low coefficient of performance (COP) which varies in the range (0.1–0.4) and its operation should be around critical points otherwise its COP deteriorates rapidly [2]. Even though their COP is very low, with careful design they can be serious competitors to other systems. This is especially important for those developed and developing country where a large portion of the electricity is consumed for the air conditioning and cooling operation.

2. WORKING PRINCIPLE OF EJECTOR REFRIGERATION

An ejector refrigeration system (ERS) employs an ejector to fulfill the function of a compressor. The ejector is widely known as a no-moving-part pump device or a non-mechanical compressor, requiring no maintenance and no lubrication [3]. The primary interest is the use of supersonic ejectors for performing thermal compression or refrigeration cycle performance maximization through thermal compression. The basic components of ERS are the evaporator, condenser, generator, pump, expansion valve and supersonic ejector as shown in the figure 1. Ejector cycle consists of high and low temperature sub cycles. In high temperature cycle waste heat is utilized to vaporize the refrigerant in the generator. Vapor then flows through the primary nozzle (convergent-divergent type) in which it is accelerated. In the constant throat section of the nozzle the velocity reaches to the sonic velocity, this phenomenon is known as the primary chocking of the primary or motive fluid. Then the low pressure created in the primary choking region helps to the entrainment of the secondary fluid from the evaporator. Then, the two fluids are mixed in the mixing chamber and pressure recovery takes place in the ejector [3]. Then the mixed vapor is fed to the condenser where heat is rejected to the environment.

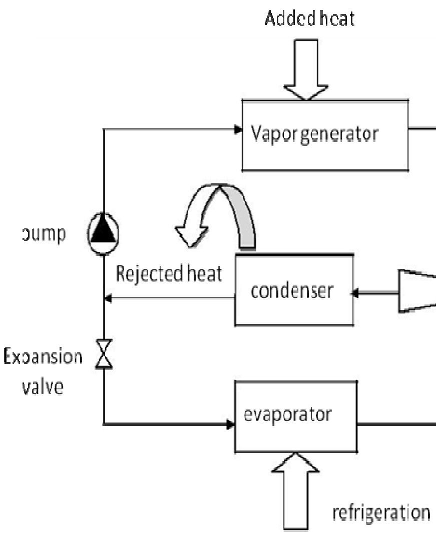

Figure 1. Schematic diagram of a typical ejector refrigeration system [4]

Then the liquid from the condenser is divided into two streams. One goes to the evaporator where it evaporates to produce the required refrigeration effect. Before evaporation it passes through the expansion valve in the same way as it is done in case of vapor compression refrigeration system. Rest of the liquid goes to the generator by the mechanical pump and completes the cycle.

3. WORKING PRINCIPLE OF EJECTOR

Ejector is basically consists of three parts, namely, primary nozzle, mixing chamber and diffuser. Variations in the stream velocity and pressure as a function of location inside the ejector are shown in Fig. 2 and are explained in this section. Primary fluid enters the nozzle as the subsonic flow. When it is passed through the converging section its pressure decreases and velocity increases in a linear fashion. Its velocity reaches to sonic velocity at the throat section of the nozzle. This is known as the choking of the primary fluid. As the primary fluid reaches sonic velocity at the throat and then flows through the diverging section its velocity reaches to the supersonic velocity at the nozzle exit point (NXP). At the NXP, motive steam pressure becomes lower than the secondary vapor and then the secondary steam entrains the mixing chamber. Several attempts have been made to understand the mixing characteristics of the two fluids. The primary fluid spreads towards the ejector wall and forms a shear layer between the two fluids [5].This layer impedes mixing of the fluids up to some distance. Mixing of these two fluids start when the strength of this layer becomes weak. A large portion of irreversibilities is associated with this mixing process due to an energy loss from the shear force developed at the interface of two fluid surfaces [6]. It is assumed that the mixing of the fluids in the ejector occurs at a constant pressure before entering into constant area chamber [5]. Then the mixture goes through a shock inside constant area section of the diffuser. The shock is associated with the increase in the mixture pressure and decrease of the mixture velocity to subsonic condition [7]. As the subsonic mixture flows through the diverging section further increase of the pressure occurs. At the exit of the diffuser, mixture pressure is slightly greater than the back pressure of the condenser.

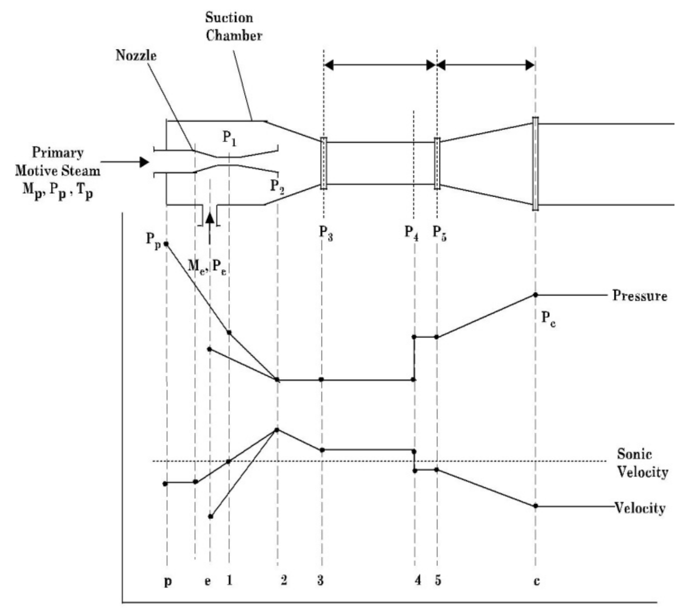

Figure 2. Typical ejector geometry, pressure and velocity profiles along ejector length. [7]

4. COP OF EJECTOR REFRIGERATION SYSTEM

Coefficient of performance (COP) is the normal performance index used in any refrigeration system. Keenan and Kneman [8] presented an ejector model to analyze air ejector. Their analysis of 1D model was based on ideal gas dynamics and the conservation of mass, momentum and energy. In their study, they considered the ejector with constant area mixing chamber excluding the diffuser section. In this model, heat and friction losses were not considered. In subsequent study Keenan *et al.* [9] extended their analysis considering constant pressure mixing process. No frictional or heat losses was considered in this analysis also. Munday and Bagster [10] developed a new ejector theory in which they assumed that the primary fluid went out without mixing with the secondary fluid and mixing process started beyond the hypothetical throat ("effective area") with an uniform pressure. They also concluded that stem jet ejector should be designed in most prevailing condition unless it is severe to achieve greater overall efficiency. Eames *et al.* [11] modified the Keenan's model considering the irreversibilities associated with the primary nozzle, mixing chamber and diffuser with their associated efficiencies. Huang *et al.* [12] carried out a study based on constant pressure mixing and the flow was assumed to be in chocking condition at the throat. They also carried out an experiment using 11 ejectors and R141b as the working fluid to verify the theoretical results obtained from the 1D model based on gas dynamics. Designing of ejector at constant pressure mixing process is much more common in different literature because the theoretical validation in this analysis with experimental result is more acceptable as error is less. The cycle efficiency is measured by the coefficient of performances (COP). COP is given by COP = (Refrigeration effect)/ (heat input to the generator + power consumption by the pump). As the power input to the pump is much less than the heat input to the generation then the pump work can be neglected, then

$$COP = \frac{\dot{m}_e (h_{fg})_e}{\dot{m}_p (h_{fg})_p} \quad (1)$$

where, h_{fg} is the latent heat of vaporization and $\dot{m}$ is the rate of vapor generation. If the primary and secondary fluids are the same, then Eq. (1) is reduced to

$$COP = \frac{\dot{m}_e}{\dot{m}_p} = \omega(\text{entrainment ratio}) \quad (2)$$

If the primary and secondary fluids are different, the latent heats of vaporization will be different than the COP and entrainment ratio will be different.

5. EJECTOR REFRIGERATION SYSTEM PERFORMANCE

The performance analysis of ejector refrigeration system depends on different parameters under which system works. The most important parameters include generator temperature, evaporator temperature, condenser temperature and pressures of the primary and secondary fluids. This paper presents a review report from the existing literature of the effects of these parameters on the system performance.

5.1 Effect of generator temperature COP

Yapici *et al.* [13] measured the performance of an ejector refrigeration system at constant evaporator temperature and condenser pressure, for different generator temperature in the range of 82°C to 105°C. R123 was selected as the working fluid and six ejectors with area ratios varying from 6.56 to 11.45 were used. From the experimental investigation of steam jet ejector, Chunnanod and Aphornratana [14] found that COP increased from 0.26 to 0.5 when the generator temperature was varied from 120°C to 140°C. Selvaragu and Mani [15] also found the same type of trend of the COP variation with the generator temperature. They have provided the reason behind it as the generator temperature of the given primary vapor increases; flow entrainment of the secondary vapor tends to increase because of the increase in the enthalpy of primary vapor. Cizungu *et al.* [16] have done a theoretical investigation on two-phase ejectors based on two components NH_3-H_2O and separate investigation have also been carried out on single component NH_3. From the analysis of single component system same type of variation was obtained i.e., COP was improved with the increase of generator temperature. Also, it was observed that COP reached a maximum value for a specific area ratio.

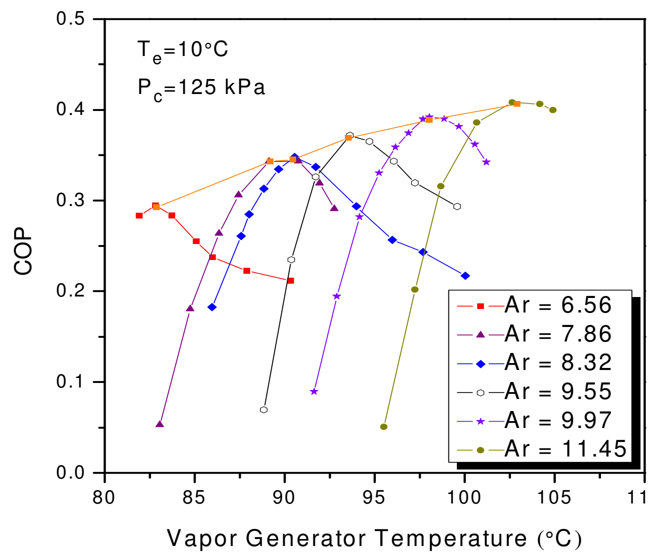

Figure 3. Performance curves for various area ratios and the optimum COP curve [13]

Ma *et al.* [18] reported that maximum cooling capacity and COP was obtained at boiler temperature 92.8°C and 90°C respectively. When the generator temperature exceeded the optimum value, there was an increase in the heat supply to the generator and the decrease in the COP [16] of the system. The results of the experimental work of Yapici *et al.* [13] have been presented in Fig. 3 as a case study. It can be clearly observed that in every area ratio there is an optimum generator temperature where maximum COP can be obtained after that it deteriorated rapidly. Also it was found that with increasing the area ratio COP also increased.

5.2 Effect of evaporator temperature on COP

Selvaraju and Mani [15] have done an experiment for the investigation of performance of VERS using R134a as working fluid. Experimental investigations by Pounds *et al.*

[4] have proposed that maximum efficiency can be varied from 0.4 to 1.2 when evaporator temperature varies from 5°C to 15°C at the constant generator and condenser temperature. Yapici *et al.* [13] reported that at constant generator and condenser temperature, system COP could reach from 0.29 at 10°C to 0.4 at 15°C. Behavior of ammonia (R717) through an ejector with a low temperature thermal source, have been investigated by Rogdakis and Alexis [17] and found that the performance of the ERS was a cubic function of evaporator temperature. Ma *et al.* [18] reported that that as the evaporator temperature was increased from 6°C to 13°C, the entrainment ratio increased from 0.064 to 0.498 at constant generator temperature of 90°C. They proposed that with the increase of evaporator temperature the evaporator pressure became high and therefore a small amount of motive vapor was sufficient to create necessary suction and entrained the required quantity of secondary vapor in the ejector.

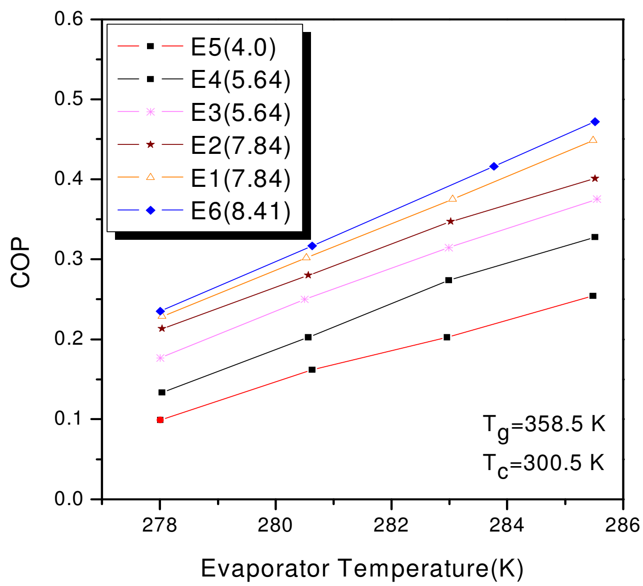

Figure.4. Variation of COP with evaporation Temperature [15]

Chunnanond and Aphornratana [14] reported that when boiler or generator temperature remained constant at 130°C, COP value changed from 0.29 to 0.49 when evaporator temperature was varied from 5°C to 15°C. It was further seen that ejector with higher area ratio was capable of more secondary fluid entrainment than the ejector with the lower area ratio, so the ejector with lower area ratio required higher evaporator temperature for the required refrigeration effect [19]. The work of Selvaraju and Mani [15] has been considered as case study for this case and the relevant results have been presented in Fig.4. A generator temperature of 358.5K and a condenser temperature of 300.5K were chosen as the operating conditions for this study where the evaporator temperature was varied from 275K to 285.5K. It has been found that for the given generator and condenser temperature, COP is increased with increase in evaporator temperature.

5.3 Effect of condenser temperature on COP

Yen *et al.* [20] have studied the effect of condenser temperature on the system COP using CFD simulation. Eames *et al.* [16] also found maximum COP as 0.38 when boiler and evaporator temperature remained constant and the back pressure was at 36 mbar and beyond this pressure performance detoriated rapidly. A main cause for the rapid decrease of the COP at the critical condenser pressure was described by Allouche *et al.* [21]. They showed that with the increase of the condenser pressure, the position of the secondary shock wave was shifted from the diffuser section to the constant area mixing section. When the position of the shock wave reached the inlet of the mixing section, secondary fluid stopped to entrain and malfunctioning of the ejector started after this.

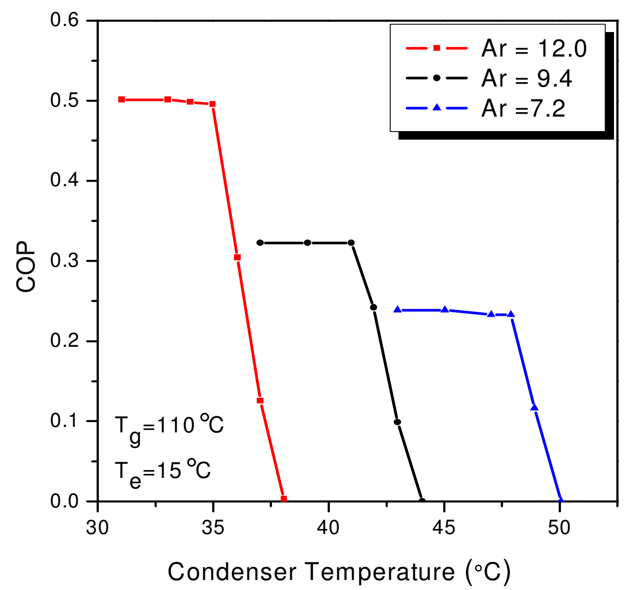

Figure 5. Variation of COP with condenser temperature [20]

Depending on the analysis of an axi-symmetric CFD model where water was used as working fluid and powered by solar energy, Verga *et al.* [21] showed that entrainment ratio increased with the area ratio but critical back pressure decreased. They proposed that for a condenser temperature of 32°C the optimal area ratio would be approximately 21 with entrainment ratio of 0.36. However, the same ejector would fail to operate at 35°C and in this case the optimal area ratio would be 16.3 with entrainment ratio of 0.28. Pounds *et al.* [4] also proposed that with the increase of condenser pressure at a constant generator and evaporator temperature, COP increased at first and then decreased rapidly at the critical condenser pressure.

Selvaraju and Mani [15] investigated the performances of vapor ejector refrigeration system with R134a as a working fluid and found that for different sets of ejector with different area ratios, system COP increased with condenser temperature and area ratio. The work of Yen *et al.* [20] has been chosen as the case study here and the results have been presented in Fig. 5. The working fluid of this analysis is R245fa and the k-ε turbulence 2D steady flow model was used for the theoretical analysis. From the result it was observed that for a particular ejector and the given generator and evaporator temperature the COP remained constant first with an increase in condenser temperature. But the COP decreased rapidly after a temperature, which is the critical point for the given generator temperature. It was further observed that the ejector with lower area ratio could be able to achieve higher critical condenser temperature.

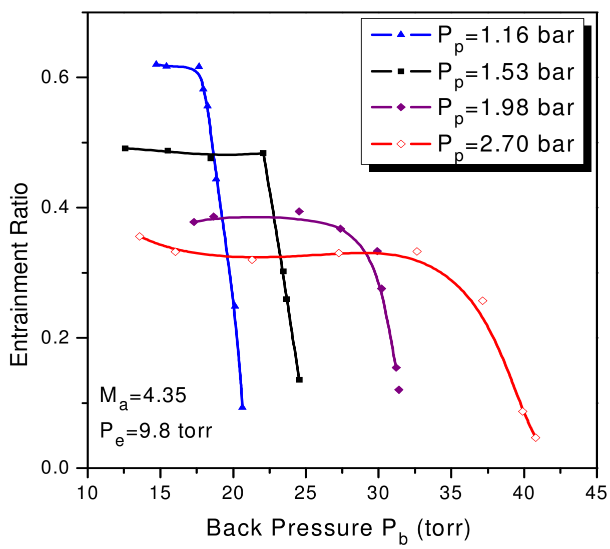

Figure 6. Effect of pressure of primary flow on the entrainment ratio [23]

5.4 Effect of the pressure of primary fluid on COP

Chang *et al.* [6] investigated the effect of the primary fluid pressure on the system performances. From their study it was found that entrainment ratio decreased with increasing the primary fluid pressure. From the investigation, it was found that entrainment ratio increased from 0.68 to 1.4 when primary pressure varies from 2.70 atm to 1.16 atm. Huang *et al.* [22] also stated that when the secondary pressure remained constant entrainment ratio varied from 0.35 to 0.62 where primary pressure was changed from 0.604 MPa to 0.400 MPa. This variation of entrainment ratio (indicator for COP) with pressure of the primary fluid has been shown in Fig. 6 as described by Chen and Sun [23]. The ratio of the pressure between the primary pressure and nozzle exit point (P_p/P_x) was always maintained constant .When P_p increased; P_x had to be increased to maintain the constant pressure ratio. So the pressure difference (P_e–P_x) decreased, then the entrainment of the secondary fluid decreased which deteriorate the entrainment ratio. Also P_p was pushing the secondary shock wave in the diffuser section hence increased the back pressure.

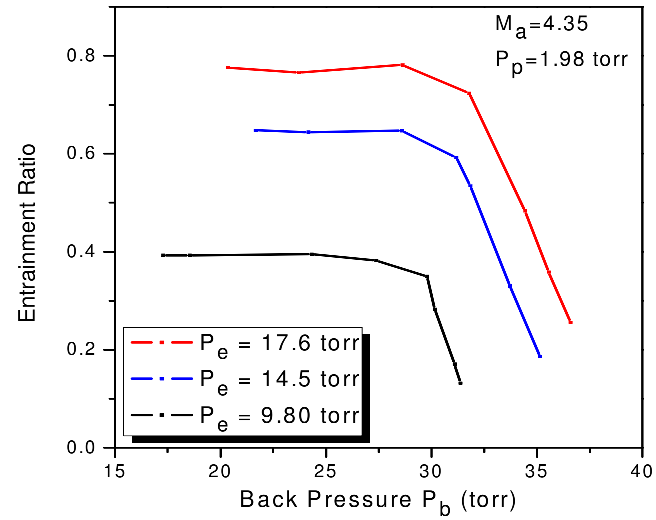

Figure 7. Effect of pressure of secondary flow on the entrainment ratio [23]

5.5 Effect of the pressure of secondary fluid on COP

From the study it was found that with the increase of the secondary fluid inlet pressure, the entrainment ratio and the back pressure both were increased. Chang *et al.* [6] stated that entrainment ratio reached to 2.2 when the pressure of the secondary fluid was at 17.54 torr whereas the entrainment ratio reduced to 1.2 when pressure decreased to 9.85 torr. Chen and Sun [23] proposed that the difference of pressure between the secondary fluid and the nozzle exit point (P_e-P_x) increased with the increase of the secondary fluid inlet pressure. Then the system COP increased as the entrainment of the secondary fluid increased. Also the secondary shock wave pushed further back to the downstream of the diffuser, then increasing the critical back pressure. For better understanding, the entrainment ratios vs. back pressure plots for different secondary fluid pressure have been shown in Fig. 7 from the work of Chen and Sun [23].

6. CONCLUSIONS

The following conclusions can be drawn from this review work. The co-efficient of performance (COP) of the ERS depends on the ejector configuration and operating temperatures of generator, evaporator and condenser. The optimal operating condition of the ejector is achieved when it is operated in critical mode. The COP increases with generator temperature up to the critical state. COP of the system increases with the evaporator temperature and area ratio. In this review it is found that condenser has much more influence than the generator and evaporator temperature on the area ratio and COP of the ejector. Finally, it can be stated that ejector based system are highly reliable, simple in construction, tolerance to a wide range of working fluid. However, low thermal COP and poor design performance must be improved for commercial use.

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