AN EXPERIMENTAL INVESTIGATION OF STEAM-EJECTOR REFRIGERATOR: THE ANALYSIS OF PRESSURE PROFILE ALONG EJECTOR.

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In order to develop the performance of an ejector refrigerator, a better understanding in flowing and mixing characteristic through the ejector is significant. In this study, a 3kW steam ejector refrigerator was constructed. The static pressure profile along the ejector at various operating conditions was measured. The analyzed experimental results introduce three new parameters. Using these data, the flowing characteristic through the ejector was well understood and clearly explained.

1 Introduction

Ejector refrigeration seems to be the most appropriate large scale refrigeration system for the present energy and environmental situation. It can utilize low-grade waste heat from power plants, incinerators or industrial processes. Jet-refrigeration system has simple construction, few moving parts and no chemical corrosion. Moreover, water; a most environmentally friendly substance can be used as working fluid. One weak point is its low cooling capacity and COP. If this problem can be solved, ejector refrigerator will become a serious competitor to other types of refrigeration.

In order to improve performance of an ejector, the flowing characteristic through an ejector should be clearly explained. Many researchers have been attempting to explain this by establishing some theory and assumption, without experimentally approved. In this study, the small scaled steam-ejector refrigerator was constructed. The pressure profile along ejector axis was measured at various operating conditions. The tested results will be used to describe the phenomenon which takes place in an ejector.

2 Background

A schematic diagram of a steam ejector is shown in Figure 1. High-pressure primary steam enters the primary nozzle, through which it expands to produce a low-pressure region at the exit plane. The high velocity primary steam draws and entrains the secondary vapour into the mixing chamber and the flow speed is supersonic. A normal shock wave is then produced in the mixing chamber throat, creating a compression effect, and the flow speed is reduced to subsonic value. Further compression of the fluid is achieved as the mixed stream flows through the subsonic diffuser.

![Figure 1: A typical Steam-Ejector.](image)

![Figure 2: Schematic Diagram of Ordinary Ejector Refrigeration Cycle](image)
A typical steam ejector refrigeration system is shown in Figure 2. This system is similar to a conventional vapor compression system, except that the mechanical compressor is replaced by liquid recirculation pump, boiler and ejector. Briefly, as heat is added to boiler, high pressure and high temperature is evolved to produce the motive fluid for the ejector (2). The ejector draws a secondary vapour from the evaporator (3). This causes the water to evaporate at low temperature and produce useful refrigeration. The ejector exhausts the mixed vapour to the condenser (4) where it is liquefied (5). The liquid water is returned to the boiler via a pump (1) whilst the remainder is throttled to the evaporator (6), thus complete the cycle.

The work required by a liquid refrigeration pump is less than 1 percent compared to overall energy used by a boiler. So, the mechanical pump work can be omitted. COP of the system can be expressed as,

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COP = \frac{\text{heat absorbed at the evaporator}}{\text{heat input at the boiler}}
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3 Experimental Setup

The schematic diagram of an experimental steam ejector refrigerator is shown in Figure 3. This test facility consists of 8 principal components: steam boiler, evaporator, superheater, condenser, receiver tank, pumping system, pressure manifold and ejector. 8kW and 4kW electric heaters were used as simulated heat source and cooling load at steam boiler and evaporator respectively. In order to provide the superheated primary fluid, at different levels, a 500W superheater was installed before the motive steam entering the primary nozzle of ejector. The vapour refrigerant from the ejector was condensed at water cooled shell and coil type condenser. The cooling water running in the cooling coils was taken from the 13 kW water chiller. The boiler and superheater were covered by glass fiber wool with aluminum foil backing to prevent the thermal loss from the machine. The evaporator shell was well insulated, by neoprene foam rubber, from unexpected heat gain from the environment. The required data and operating conditions at each vessel can be measured and controlled separately by a data acquisition board connected to the personal computer.

Geometries of experimental ejector are described in figure 1. The ejector was designed based on the methods provided in literature [1 and 2]. Along the ejector axis, the static pressure at each operating conditions was tapped and measured using the pressure transducer attached with the pressure manifold. This information can be used to analyze and explain the flowing and mixing characteristic of two streams through the ejector.
The experiments were done over the boiler temperature 110-150°C, evaporator 5-15°C, condenser pressure 25-60 mBar and superheated heat input of 0-180W. The variation of cooling capacity and COP agree with the study of Eames et al. [1]. From the experimental results, a pressure profile along the axis of ejector assembly can be plotted, as shown in Figure 5. This curve can clearly explain the flowing characteristic in the ejector. After primary fluid expanding and accelerating through the primary nozzle, it fans out into the mixing chamber without mixing with the secondary fluid, with some value of “expansion angle”. This angle was thought to be the function of boiler conditions and the saturation pressure of evaporator. This flow results the converging duct for the secondary fluid. At some cross section along this duct, speed of the secondary fluid was raised to sonic velocity with the expense of static pressure. This cross section was defined by Munday and Bagster [3] as the “effective area”. They also suggested that the mixing process begins after the secondary flow chokes. The “effective position” can be investigated by the point of lowest static pressure along the ejector, before it gradually increases. Velocity and pressure of the mixed flow remain constant while it passes through constant area throat section of the ejector. At a certain distance either in throat or diffuser section, called “shocking position”, transverse shock waves are induced creating a compression effect. Across this point, the velocity of mixed fluid sudden drops to subsonic value. A further compression of fluid is achieved as the flow passes through the rest of subsonic diffuser section.

From the experimental results shown in Figure 6, the system’s performance curve for the specified boiler and evaporator can be plotted. There are three regions: choked flow in the mixing chamber, unchoked flow and reversed flow. When the ejector operated under “critical condenser pressure”, COP and cooling capacity remains constant. It can be seen in Figure 7, curve A to E, that transverse shock moves backward to the mixing chamber as condenser pressure increases. Moreover, it is obvious that the entrainment of secondary fluid remains unchanged. Further increasing condenser pressure higher than critical value, the entrainment ratio begins to fall rapidly. The transverse shocking moved in the mixing chamber. As the secondary flow varies, the secondary flow is no longer choked. If the condenser pressure still increases, ejector loses its function completely; the flow will reverse back into the evaporator.
It was found that primary fluid always choking when expanded through the primary nozzle. Thus, critical mass flow of nozzle was dependent on upstream conditions only. From the experiment, it was found that, superheated levels have less effect on primary mass flow, compared to boiler conditions. This can be concluded that, there is only one parameter, boiler operating condition, can vary refrigerant mass flow through the primary nozzle.

From Figure 8, decreasing boiler pressure causes the cooling capacity and COP of the system to rise with expenses of critical condenser pressure. This can be explained by Figure 10. Higher pressure and at nozzle exit plane thus, lower expansion angle was shown when boiler temperature decreased. More secondary mass can be entrained via the resultant shorter and steeper converging duct. The shorter converging duct can be noticed by the moving upstream of effective position, in Figure 9. Therefore, COP and cooling capacity of the system increases with lower boiler temperature. However, decreasing boiler temperature decreases primary mass and momentum of the mixed steam. The shocking position moves upstream and the ejector can be operated at lower condenser pressure.
Higher value of cooling capacity, COP and critical condenser pressure can be achieved by increasing evaporator temperature, Figure 9. It was thought that the expansion wave from primary nozzle experienced more compression effect from a higher value of saturation pressure in the evaporator. So, the smaller value of expansion angle and longer converging duct were resulted. As seen in Figure 11, the effective position moved downstream with the greater value of evaporator pressure. More evaporated fluid can be extracted and passed through the longer converging duct. Therefore, a greater value of cooling capacity and COP of the system can be achieved. The shocking position moved downstream with increasing of momentum of mixed stream. So, the ejector has higher critical condenser pressure.

The superheated level of motive fluid, before entering primary nozzle, has no effect on COP and critical condenser pressure of the system. This conclusion confirms the recommendation of ESDU [2] that the superheater has no other advantage than to ensure that the refrigerant remains dry before entering the primary nozzle.

5 Conclusion

In this study, the 3kW steam ejector refrigerator was constructed. Static pressure along ejector axis was tapped and measured a various operating conditions. The experiments were done over the boiler temperature range of 110-150°C, evaporator 5-15°C, and condenser pressure 25-60 mBar. The analyzed experimental data introduces three new parameters, which are "expansion angle", "effective position" and "shocking position". These three parameters combining with the idea proposed in literature can clearly explain the flowing and mixing characteristic through the ejector. The better understanding of the ejector may introduce a new design method of the ejectors has better performance.

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References

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