THE EFFECT OF OPERATING PARAMETERS ON THE PERFORMANCE OF A STEAM EJECTOR REFRIGERATION SYSTEM WITH BAROMETRIC CONDENSER

تأثير عوامل التشغيل على أداء نظام التبريد بالأبواق البخارية مع مكثف بارومتري

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الخلاصة:

يتضمن البحث دراسة عملية لتأثير عوامل التشغيل على أداء نظام التبريد بالأبواق البخارية مع مكثف بارومتري السعة التبريدية للنظام ٧ طن تبريد (٢٤,٤٢ كيلوات) حيث تم إجراء التجارب عند ضغط بخار من ٦٠٠ إلى ٨٠٠ كيلوبسكال ، درجة حرارة مكثف من ٣٠ إلى ٤٨ درجة مئوية ودرجة حرارة تبخير من ٦ إلى ١١ درجة مئوية . كما تم تجهيز فونية البخار بمنظومة ميكانيكية للتحكم في وضع مخرج فونية البخار داخل غرفة الخلط.

وقد بينت التجارب العملية في هذا البحث أن نسبة الجر ومعامل الأداء للنظام دالة في ضغط بخار الغلاية ، درجة حرارة المكثف ودرجة حرارة المبخر. وقد أظهرت النتائج العملية أن معامل الأداء يزداد بزيادة ضغط البخار ودرجة حرارة التبخير بينما يقل مع زيادة درجة حرارة المكثف. كما أظهرت النتائج أيضا ضرورة تحريك فونية البخار ناحية مدخل البوق عند زيادة درجة حرارة المكثف وثبات ضغط البخار ودرجة حرارة المبخر.

وقد تم إستنتاج معادلة تجريبية لابعدية لحساب معامل الأداء كدالة في ضغط البخار وكل من ضغط المكثف والمبخر.

ABSTRACT

The main concern of this work is to investigate experimentally the effect of operating parameters and primary nozzle position on the performance of the steam ejector refrigeration system with barometric condenser. In this study, a steam ejector refrigeration system of 7.0 TR (24.42 kW) cooling capacity was used. The boiler pressure of 600 to 800 kPa, condenser temperature of 30°C to 48°C and, evaporator temperature of 6°C to 11°C are used. The primary nozzle is constructed so that it can be movable inside the mixing chamber

The experimental results of this work show that the entrainment ratio and coefficient of performance were found to be functions of steam-generator pressure, condenser temperature (back pressure) and evaporator temperature. The coefficient of performance increases with increasing of boiler pressure, evaporator temperature and decreasing of condenser temperature. An increase in COP (cooling capacity) can be achieved by retracting the nozzle from the mixing chamber as the condenser pressure (temperature) increase without changing either the evaporator or boiler temperatures. In practice, the nozzle position may be automatically controlled by monitoring the saturated temperatures and pressures in the boiler, evaporator and condenser.

A new experimental correlation was proposed for predicting the coefficient of performance related to the motive steam pressure, condenser pressure and evaporating pressure.

Keywords: Steam Ejector, Barometric Condenser- Jet refrigeration.

Nomenclature

A	Area of cross section , m^2	v	Specific volume, m^3/kg
COP	Coefficient of performance	x	Steam dryness fraction
CR	Compression ratio, (P_c/P_e)	W	Specific work, kJ/kg
h	Specific enthalpy, kJ/kg	$\dot{m}$	Mass flow rate, kg/s
P	Pressure, kPa	w	Flow entrainment ratio
T	Temperature, °C		

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Greeks symbol

η	Efficiency
γ	Ratio of specific heat (c_p/c_v)

Subscripts

c	Condenser	m	Mixing
d	Diffuser	n	Nozzle
e	Evaporator	p	Primary stream (motive steam)
f	Saturated liquid	t	nozzle throat
g	Steam generator, Saturated vapor	a	Atmosphere conditions

I. Introduction

The first steam jet refrigeration system dates back to 1901 and was designed by LeBlance in France and Parsons in England, ASHRAE [1]. Since then, researches has been ongoing for better understanding the performance characteristics of ejectors and improving their efficiency. One development is that by Munday and Bagster [2], who put forward a new ejector theory by introducing a hypothetical "effective area" somewhere in the mixing section for the flow of secondary vapor to explain the complicated mixing process of primary and secondary fluids in ejectors. This explains the choking phenomenon in the secondary stream. This theory was later used by Huang et al. [3], for the analysis of their experimental data. They also introduced the concept of "critical back pressure" to define the operating region for jet refrigeration systems.

A recent study by Decker [4], expanded the application of steam jet refrigeration into vacuum cooling used in paper bleaching operations and in pharmaceutical and chemical industries. In addition to providing chilled water, steam jet systems may also be used for quick chilling of process fluids and for direct vacuum chilling of foods. Flash cooling may not be practical with other types of refrigeration systems because the large volume of the evaporated gas at low pressure requires very large equipment. Therefore steam jet refrigeration systems complete vigorously with mechanical vapor compression in these applications.

The most recent studies [5-8], based on previous experimental results, demonstrated that of steam jet refrigeration system could provide a cost effective way of supplying cooling for building air-conditioning systems by utilizing low-grade waste heat currently rejected by district heating systems and combined heating and power schemes during summer months.

Sun and Eames [9 & 10], have shown that the performance of an ejector-refrigeration cycle depends mainly on ejector design. The optimal ejector geometry is affected by operating temperatures. The use of variable-geometry ejectors is important in achieving optimal system performance.

A novel refrigeration cycle based on the combination of an absorption cycle with an ejector refrigeration cycle is studied by Sun et al. [11]. The combination brings together the advantages of absorption and ejector refrigeration systems and provides high COP for refrigeration and air-conditioning. The combined cycle is particularly suitable for utilising waste thermal energy. A computer simulation program was developed for the combined cycle and used to determine the performance of the system using LiBr H_2O for different generator, condenser and evaporator temperatures. Optimum operating conditions and ejector design data are also provided.

A computer model of an ejeetor, with particular reference to refrigeration applications, using methanol at different operating conditions as the working refrigerant was studied by Alexis and Katsanis [19].

Despite the above efforts, the performance characteristics of steam jet refrigeration systems are still not well understood; therefore an experimental investigation was carried out in the present work. The main concern of this work is to experimentally investigate the effect of operating parameters and primary nozzle position on the performance of the steam ejector refrigeration system with barometric condenser.

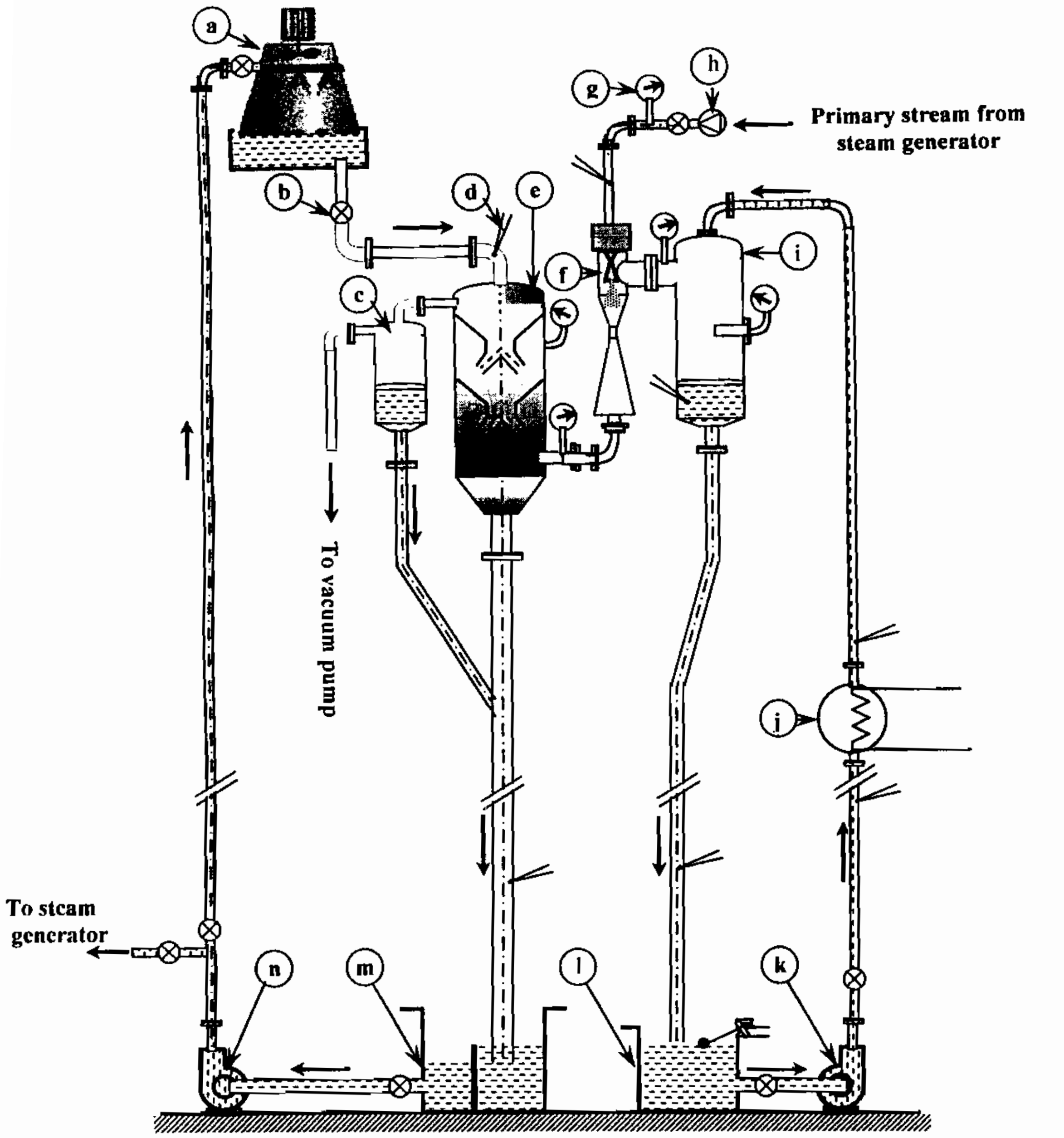
2. Experimental Apparatus

A schematic representation of the experimental setup is shown in Fig. 1. The apparatus consists mainly of a steam generator, an ejector (f), an evaporator (i), a barometric condenser (e), a cooling water tank (l), a hot-well tank (m), two liquid pumps (k and n), and measuring devices. The steam generator is a large capacity (8ton/h of steam) which used in industry process and a part of steam is using in this work. A controller is used to control the steam pressure. A vapor-liquid separator is installed. The evaporator is equipped with a 24-kW heater (j) controlled by a controller to within ±0.1°C. The exit mixture vapor from the ejector is condensed in a barometric condenser. The barometric condenser cost less to buy and install and they have many advantages. However, users should be aware that the barometric condenser is a direct contact design. The cooling water is mixed directly with the vapor to be condensed. The cooling water used in the condenser is provided by a cooling tower (a). A controller is used to control the condensing temperatures to within ±1°C, thus adjusting the back pressures of the ejector.

The thermocouples types K are installed in the desired locations for temperature measurement. Calibration of the thermocouples indicates that the accuracy is to within ±0.1°C. Absolute vacuum devices (type DVR2, Vacuubrand digital) with accuraey of 1mbar are installed in the inlet of the secondary flow and the exit of the ejector to measure their absolute vacuum pressures (g). The inlet pressure of the ejector is measured by a gauge pressure. The primary flow rate is measured by a differential pressure-type flow meter (h) that uses the theory of the conservation of energy in a fluid flowing through a pipe of specific design (Samson 4Z). After calibration the accuracy of this flow meter is ±1.0% of the reading value. The secondary flow rate is not measured directly. Instead, a marked glass tank was installed to make up water to the cooling water tank. The time period for the make up water between two marks of the glass tube was measured by a digital timer. Then the flow rate can be simply calculated by dividing a known volume between the two marks into time period.

2.1 The ejector Assembly

Figure 2 shows the configuration of the ejector used, which is the most important compartment of the system. The corresponding geometric dimensions are also listed in Fig. 2. According to the work of Huang et. al [3], the location of the primary nozzle significantly affects the system performance, and the optimum position of the primary nozzle exit is at the entrance plane of the mixing section. It was designed using a semi-empirical method described by Eames and Aphornratana [12]. The body of the ejector (9) as shown in Fig. (2) was manufactured from stainless steel pipe 158 mm in outer diameter fitted with flanges at both ends to provide a connection with the condenser and a location for the primary nozzle support plate (1). A stainless steel suction manifold 155 / 147 mm outer / inner diameters (2) was welded to the ejector body and provided with a flange to connect it to the evaporator. Flanged connections were sealed with O-seals (11) to provide an airtight construction. The primary nozzle assembly, shown in Fig. 2, was made in three pieces; the nozzle (5) itself, a spacer (4) and a screwed connector (3) to connect the assembly to the support flange (1). The diffuser assembly consisted of a mixing section (6), a diffuser throat section (7),



a. Cooling tower
b. Control valve
c. Condensate separator
d. Thermocouple
e. Barometric condenser
f. Steam ejector
g. Pressure gauge
h. Steam flowmeter
i. Evaporator
j. . Heating load
k. Cooling water circulating pump
l. Cooling water tank
m. Hot-well tank
n. Hot water pump

Fig. 1. Schematic view of ejector-refrigeration system (not to scale).

a subsonic diffuser section (8) and a tail piece (10). These were manufactured in aluminum. Their outside diameters were machined to provide a push fit into the ejector body and O-seals (11) were fitted to prevent steam from leaking back from the condenser to the evaporator. Location lugs were provided to ensure alignment between the separate pieces. Figure 2 also shows the nozzle exit position (NXP) datum.

3. Ejector Cycle

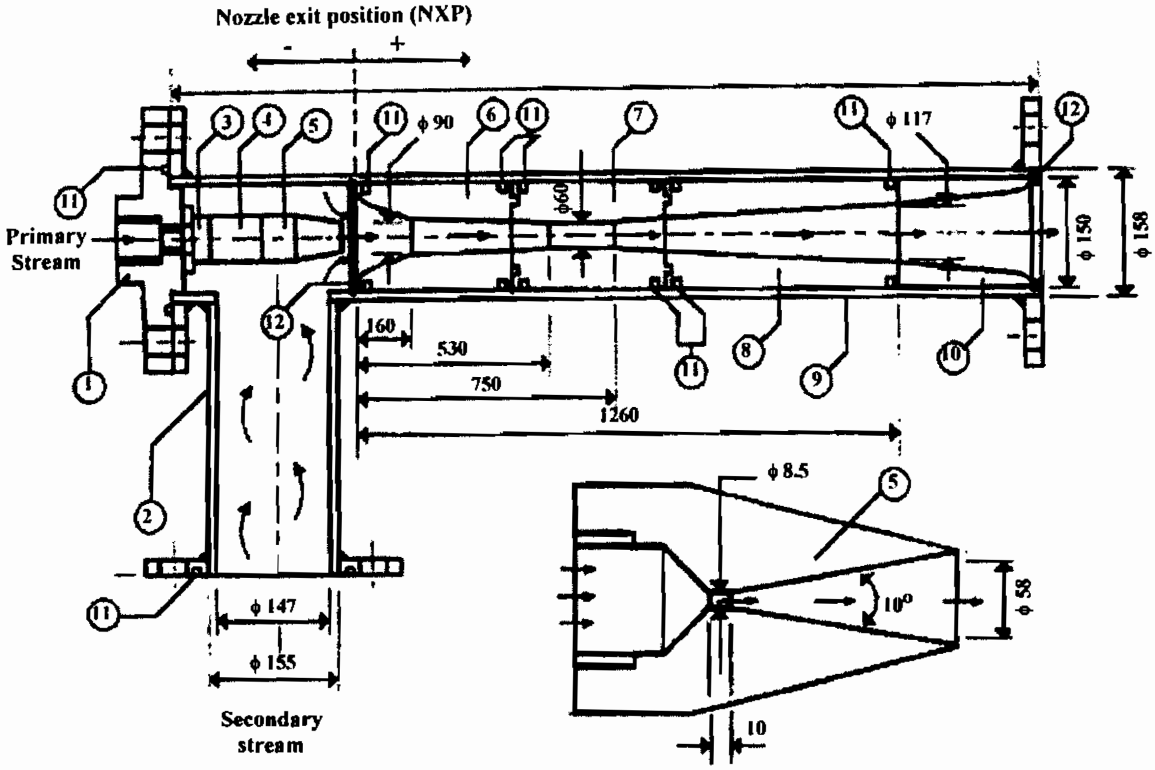
Figure (3) shows a schematic diagram of an ejector-refrigeration system with ejector components. An ejector cycle is similar to a mechanical vapour-compression system except that the compressor is replaced by a liquid pump, boiler and ejector. This cycle is sometimes referred to as an ejector-compression refrigeration system. Operation at two different pressure levels for vapour-compression systems is changed to three different pressure levels for ejector cycle.

The vapor (primary or driving fluid) flows through the primary convergent-divergent nozzle and creates a vacuum at the nozzle exit where the vapor (secondary or driven fluid) from the evaporator is entrained into the ejector. Both streams then mix and form a single supersonic stream. The stream then undergoes a transverse shock in the constant-area section and diffuses in the diffuser section until its pressure increases to the condenser pressure. Emerging from the ejector, the fluid undergoes a temperature reduction in the regenerator and then condenses in the condenser with the condensation heat being rejected to the environment. Finally, the condensate is partly pumped to the regenerator and flows back to the boiler and partly expands via a throttling valve and evaporates in the evaporator to produce the necessary cooling effect. The evaporated vapor is entrained by the ejector to complete the cycle. Ejector performance is measured by the entrainment ratio, which is defined as the mass-flow rate ratio of the secondary flow to that of primary flow.

4- Experimental Procedure

At the start of every experimental run, all air was evacuated from the inside of the system using a vacuum pump and open the cooling to flow water to condenser until its pressure is equal to the saturation pressure corresponding to the water temperature at the evaporator. The high-pressure vapor (primary flow) enters the primary nozzle of the ejector and induces the low-pressure vapor (secondary flow) from the evaporator. The mixture vapor formed is condensed in the condenser and flows down into the water receiver by gravity.

After evacuation the cooling water tank was filled with water. Once the required generator steam pressure had been achieved, the supply valve was opened causing the ejector to operate. During evaporator cooling refrigerant water from the bottom of the vessel was circulated through the heater coil (j in Fig. 1) and its flow was adjusted until the desired evaporation pressure was obtained. The condenser cooling water flow was then adjusted to provide the required ejector back pressure. When the system was operating in a steady state condition, the pressure, temperature and water level readings were recorded from the generator, evaporator and condenser vessels respectively. The batch-type experimental system is designed to run at least 1 hour. After this time a further set of readings was taken and results were calculated. The entrainment ratios were determined from the measured water content changes in the evaporator over the measured steady state operating time of the experiment and primary steam flow rate. The water flow rate at the condenser was used to check the results.



1. Nozzle support flange
2. Suction manifold
3. Nozzle connector
4. Spacer
5. Primary nozzle
6. Mixing section
7. Diffuser throat section
8. Subsonic diffuser section
9. Ejector body
10. Tail piece
11. 'O' seals
12. Fixing rings

Fig. 2. Sectional drawing of the test steam ejector (Dimensions in mm)

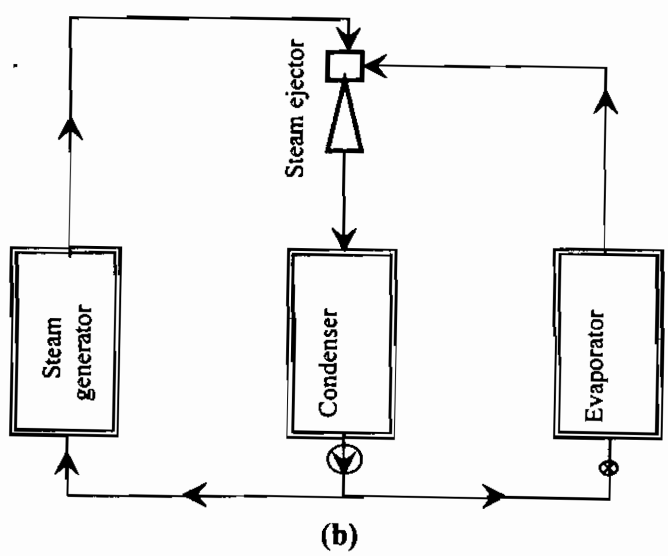


Fig. 3. Schematic view of the ejector cycle

5. Results and Discussion

The ejector performance is defined by the mass flow rate ratio of the secondary flow to primary flow, which is called entrainment ratio, w, i.e;

$$w = \frac{m_s}{m_p} \tag{1}$$

The system coefficient of performance is measured by the coefficient of performance (COP), which is defined as the ratio of cooling load at the evaporator and the energy consumption at the boiler. The COP value can be expressed as:

$$COP = \frac{Q_c}{Q_b + W_{pumps}} = \frac{m_s(h_{g,e} - h_{f,c})}{m_p(h_{g,b} - h_{f,c}) + W_{pumps}} \tag{2}$$

Where,
Q_c = cooling load at the evaporator,
Q_b = energy input to the boiler and
W_{pump} = work required by the pump.

Obviously, the system performance is closely related to the ejector performance, and hence the ejector performance characteristics also represent the characteristics of the system.

5.1. Validation of the Experimental Results

Before reporting to the main results of the present work, mention will be made of relevant auxiliary experiments. To demonstrate the validity of the experimental apparatus, and to maintain chocking of the primary flow in the nozzle, the pressure ratio of the evaporator over the boiler should be less than the critical pressure ratio of steam which equal 0.5376 (critical pressure ratio = $\left(\frac{2}{k+1}\right)^{k/k-1}$). In all the experiments the pressure ratio was always much lower than 0.5376. Therefore the primary nozzle always operated in choked conditions. Under chocking conditions, the mass flow rate through the nozzle is not affected by condenser and evaporator conditions and can be expressed by:

$$m_p = A_t P_b \sqrt{\frac{\eta_n k}{RT_b}} \left(\frac{2}{k+1}\right)^{(k+1)/[2(k-1)]} \tag{3}$$

$$m_p = f(P_p, T_p)$$

The variation in saturated steam flow through the primary nozzle with generator pressure is shown in Fig. (4). When these experimental results are compared with the theoretical results calculated using one-dimensional compressible flow theory [15], it can be seen that the calculated values agree very well with the measured data. It is concluded from the figure that the present experimental results always operated in chocked conditions and the error in mass flow in most cases is less than 4%.

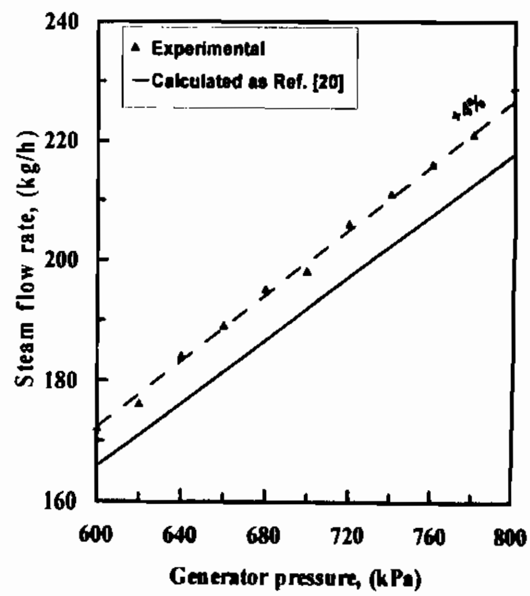


Fig. (4) Comparison between experimental and calculated steam flow through primary nozzle with generator pressure.

Also Figure (5) shows a deviation between present measurements and calculated values using the model in Reference [13] for evaporator temperatures and entrainment ratio at 8.0 bar boiler temperature and 39°C condenser temperature. It is concluded from the figure that the present experimental results always agree with the calculated results in chocked conditions within -11%.

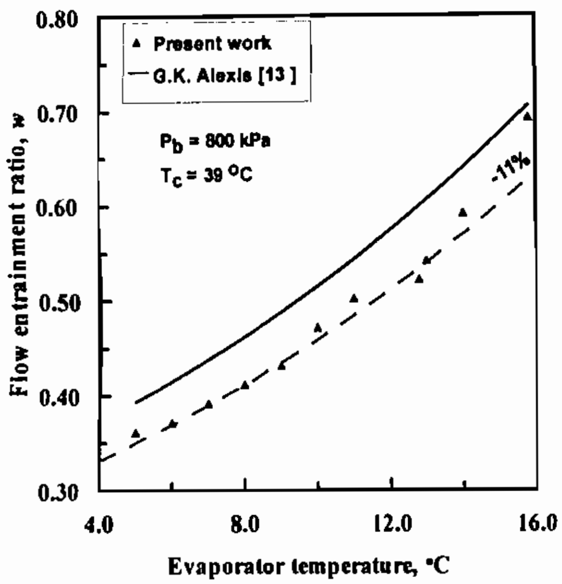


Fig. (5) Deviation between present work and and calculated model in Refrence [17].

5.2 Effect of the Nozzle Exit Position (NXP)

The NXP is defined here as the distance from the exit plane of the primary nozzle to the entry plane of the second conical part of the mixing chamber as shown in Fig. 2. A series of tests was carried out to determine the effect of changes in the nozzle exit position on the achievable flow entrainment ratio. The primary nozzle was arranged so that it could be moved axially to vary the NXP value by means of a screw mechanism. Before each test run the condenser pressure was adjusted to the desired value by varying the cooling water flow. The ejector entrainment ratio was held at a constant value throughout by fixing the generator pressure and evaporator heat input. During these experiments the primary nozzle was incrementally moved away from the mixing chamber entrance. After each change in NXP, the system was given time to stabilize before data readings were recorded. The results of experimental data are shown in Fig. (6). The optimum NXP value is considered to be that which gives the greatest secondary flow. It can be seen from Fig. (6) that the optimum NXP/*D* position increased from -0.17 to approximately +0.26 when the condenser temperature was decreased 34 °C to 42°C. In practice this means that, as the condenser pressure (ambient temperature) increases, the primary nozzle should be moved further into the mixing chamber in order to maintain the desired evaporator pressure.

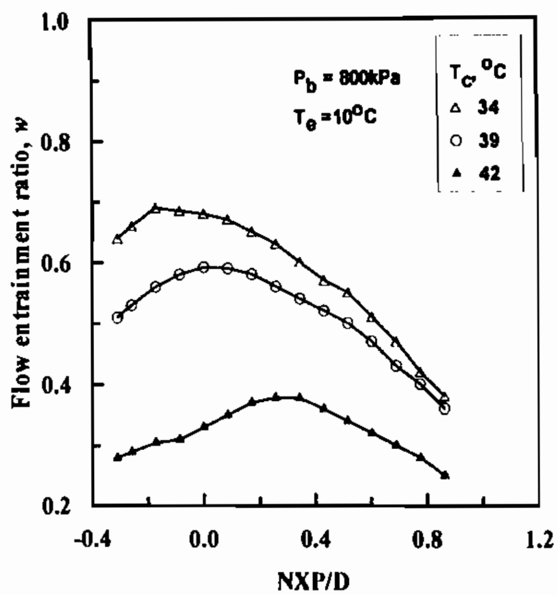


Fig. (6) Variation of flow entrainment ratio with nozzle exit position (NXP) at different condenser temperatures.

5.3 Effect of Condenser Temperature

The effect of condenser temperature on entrainment ratio at different evaporator temperature is shown in Fig. (7).

It can be seen from the figure that the increase of condenser temperature decreases the entrainment ratio at all evaporating temperatures. This is because the higher the condenser temperature the lower the temperature difference between steam and condensed water (LMTD), and consequently the amount of condensate is small. This leads to inerease the back pressure of the eondenser and vapor load to the downstream ejector.

5.4 Effect of Evaporating Temperature

The influence of evaporator temperature under different condenser temperatures on entrainment ratio and coefficient of performance are shown in Figs. (8) and (9) for constant boiler pressure. It can be seen from the figures that the entrainment ratio and coefficient of performance increases with increasing evaporator temperature.

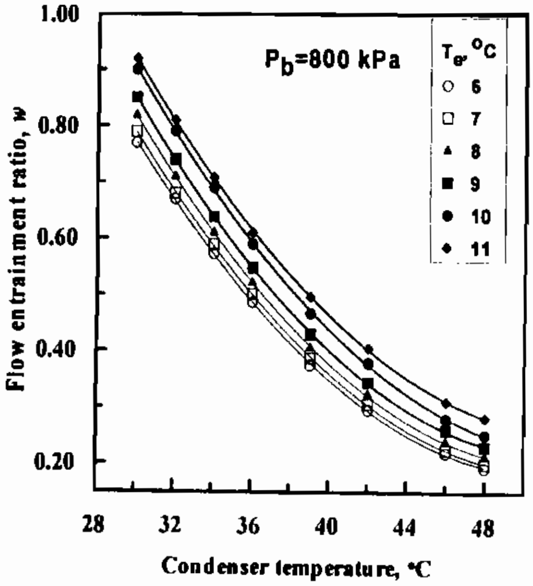


Fig. (7) Effect of condenser temperature on flow entrainment ratio at diferent evaporating temperatures.

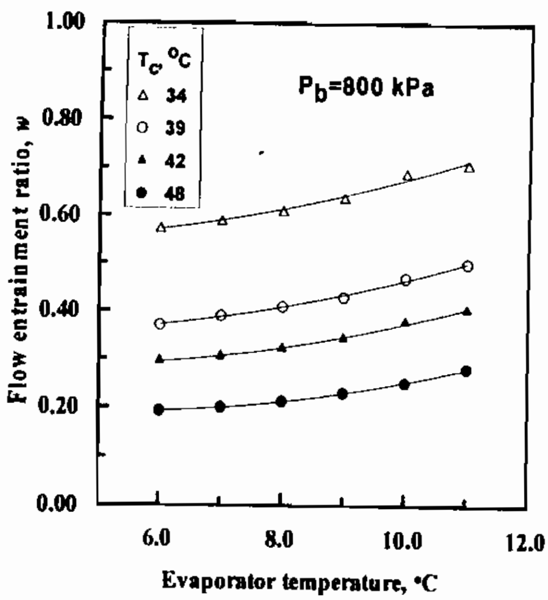


Fig. (8) Effect of evaporator temperature on Flow entrainment ratio at diferent condenser temperature.

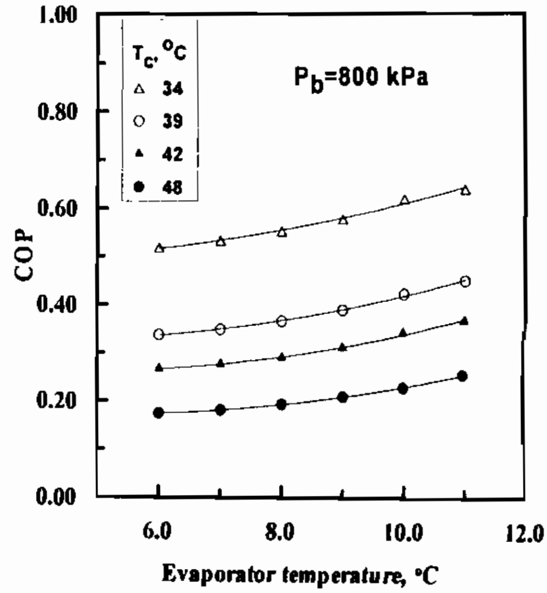


Fig. (9) Effect of evaporator temperature on cofficient of performance (COP) at diferent condenser temperatures.

Also, Figs. (10) and (11) show the effect of evaporator temperature on entrainment ratio and coefficient of performance for different boiler pressures (motive steam pressures).

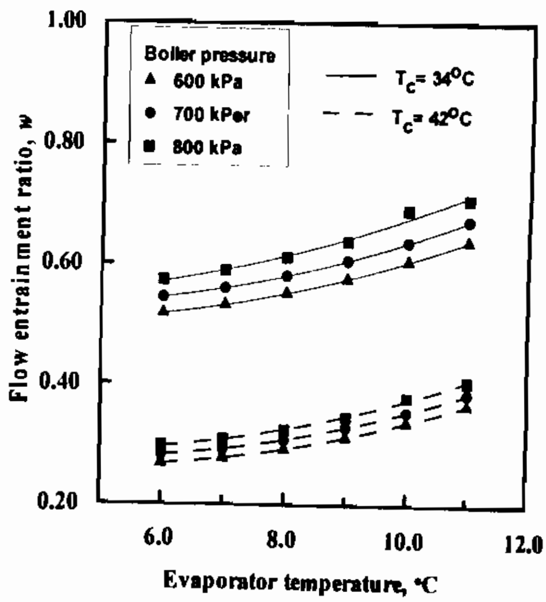


Fig. (10) Effect of evaporator temperature on flow entrainment ratio at diferent boiler pressure.

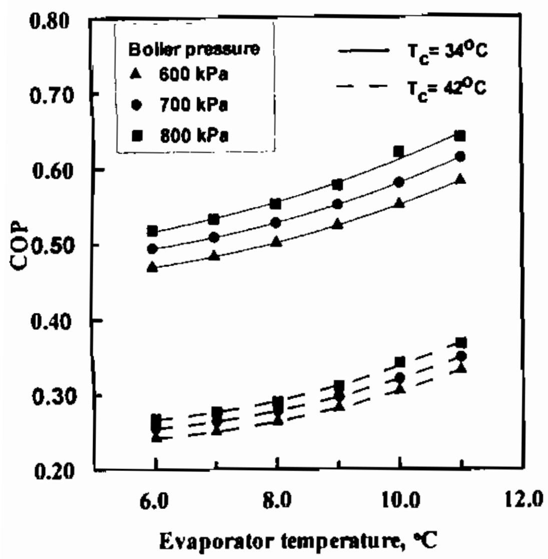


Fig. (11) Effect of evaporator temperature on cofficient of performance at diferent boiler pressures.

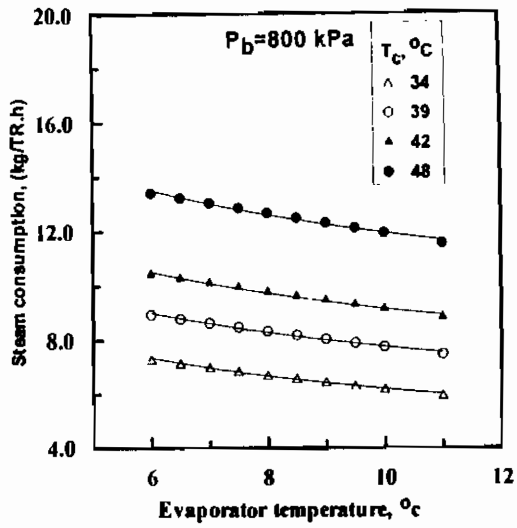


Fig. (12) Relation between Steam consumption per ton refrigeration and evaporator temperature at different condenser temperatures.

It can be seen from these figures that both of the entrainment ratio and coefficient of performance increase with increasing evaporator temperature and increasing boiler pressures.

This is due to the decrease of motive steam flow rate and its enthalpy as the pressure of steam decreases. This is because the decrease in the pressure of motive steam decreases the motive steam flow rate, its enthalpy and the energy required to the suction of the required quantity from the evaporator vaporized flashed steam.

Figure (12) illustrates the effect of evaporator temperature on steam consumption per ton refrigeration at different condenser temperature. It is seen from the figure that steam consumption decreases with the increase of evaporator temperature at the same condenser temperature. The figure shows also that the steam consumption increases with increasing the condenser temperature at the same evaporator temperature.

Finally Fig. (13) represents the relation between the COP of the system and the compression ratio, which is defined as the ratio between condenser and evaporator pressures, at different boiler temperature.

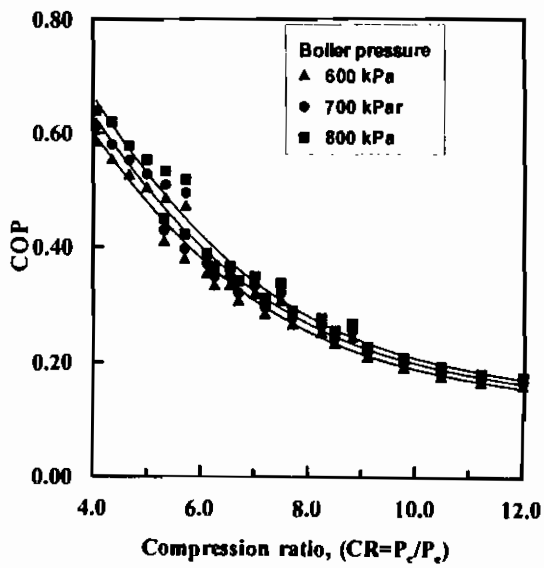


Fig. (13) Effect of pressure ratio on COP at diferent boiler pressures.

It is concluded from the figure that, for the same boiler pressure, COP decreases with the increase of compression ratio. It is seen also that the COP increases with boiler pressure at the same compression ratio. It is well known that the increase of compression ratio is due to the increase of condenser pressure or the decrease of evaporator pressure. The increase of condenser pressure (condenser temperature) decreases COP of the system as shown in Fig (9), also the decrease of evaporator pressure (evaporator temperature) decreases the COP as shown in Fig (9).

6. Correlation

An attempt was made to correlate the experimental results obtained in the present study for the given geometry. Such correlation is quit useful for the designers to calculate the COP of steam ejector for similar cases. The COP were correlated with the other relevant governing parameters such as motive steam pressure and compression ratio. The following correlation was obtained for the given geometry:

$$COP_{corr} = 1.952\left(\frac{P_b}{P_a}\right)^{0.347}\left(\frac{P_c}{P_e}\right)^{-1.27}, \qquad (4)$$

$600 \le P_b \le 800kPa, \quad 4.78 \le P_c \le 11.2kPa$

and $0.935 \le P_e \le 1.318kPa$

The above correlation predicts the COP values within an error of ±10 %. The above correlation is presented together with the experimental results in Fig (14).

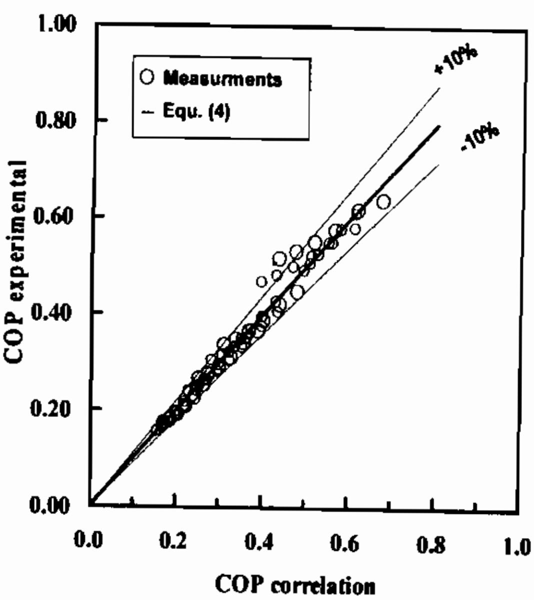


Fig. (14) Comparison between COP correlation and COP experimental.

7. Conclusion

This paper experimentally investigates the effect of operating parameters and primary nozzle position on the performance of steam ejector refrigeration system with barometric condenser. The following important conclusions are drawn:

1. The entrainment ratio and coefficient of performance were found to be functions of (in order of importance) steam-generator pressure, condenser temperature (back pressure) and evaporator temperature.
2. The use of an ejector with movable primary nozzle provides a more flexible operation than a totally fixed geometry unit. An increase in the cooling capacity can be achieved by retracting the nozzle from the mixing chamber as the condenser pressure falls without changing either the evaporator or boiler temperatures. In practice, the nozzle position may be automatically controlled by monitoring the saturated temperatures and pressures in the boiler, evaporator and condenser.
3. The COP, as well as the cooling capacity, increases with increasing generator pressure, but the condenser temperature decreases. In other words, this work yielded the fact that the highest operating efficiency can be achieved if the generator pressure increases with decreasing condenser temperature when the evaporator temperature is fixed.
4. A new experimental correlation was proposed for predicting the coefficient of performance as a function of motive steam pressure, condenser pressure and evaporating pressure for the tested limiting values and given geometry.

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