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A Theoretical Analysis of Air Cooling System Using Thermal Ejector under Variable Generator Pressure

Mohamed Ouzzane, Mahmoud Bady

***Abstract*—**Due to energy and environment context, research is looking for the use of clean and energy efficient system in cooling industry. In this regard, the ejector represents one of the promising solutions. The thermal ejector is a passive component used for thermal compression in refrigeration and cooling systems, usually activated by heat either waste or solar. The present study introduces a theoretical analysis of the cooling system which uses a gas ejector thermal compression. A theoretical model is developed and applied for the design and simulation of the ejector, as well as the whole cooling system. Besides the conservation equations of mass, energy and momentum, the gas dynamic equations, state equations, isentropic relations as well as some appropriate assumptions are applied to simulate the flow and mixing in the ejector. This model coupled with the equations of the other components (condenser, evaporator, pump, and generator) is used to analyze profiles of pressure and velocity (Mach number), as well as evaluation of the cycle cooling capacity. A FORTRAN program is developed to carry out the investigation. Properties of refrigerant R134a are calculated using real gas equations. Among many parameters, it is thought that the generator pressure is the cornerstone in the cycle, and hence considered as the key parameter in this investigation. Results show that the generator pressure has a great effect on the ejector and on the whole cooling system. At high generator pressures, strong shock waves inside the ejector are created, which lead to significant condenser pressure at the ejector exit. Additionally, at higher generator pressures, the designed system can deliver cooling capacity for high condensing pressure (hot season).

***Keywords*—**Air cooling system, refrigeration, thermal ejector, thermal compression.

I. Introduction

RECENTLY, thermal ejectors have received a lot of interest in cooling system industry. Such interest can be attributed to the energy consumption of conventional compressors, which represents a considerable load on electrical grids, particularly when the cooling demand is high. Additionally, their simple geometry and reduced cost make them very attractive for many applications. The thermal ejector is a passive component used for thermal compression in cooling and refrigerating systems. It can be driven by low-grade heat sources, such as solar collectors, geothermal energy, industrial processes, and waste heat, instead, of high-grade electric energy [1], [2].

The ejector function in the cooling system is the same as the compressor in the conventional systems. However, in the ejector-cooling system, the ejector is considered as the key component of the whole system. It is composed of a nozzle, a mixing section, and a diffuser. During the operation, a high-pressure driving flow, which is the primary stream, enters the nozzle, wherein its flow velocity increases. The driving flow reaches sonic velocity at the throat and accelerates into a high-velocity flow with low pressure at the nozzle exit. In such time, a low-pressure flow, which is the secondary stream, enters the ejector from the suction-flow inlet. The flow is then accelerated towards the mixing section. Then, the two flows are completely mixed inside the mixing section, where a part of the kinetic energy from the primary stream is transferred to the secondary stream. The kinetic energy of the mixed flow converts to pressure energy in a diffuser.

In this paper, a FORTRAN program of two parts is developed in order to carry out the investigation. The first part deals with the ejector design, where the computation progress and control is based on Mach number increments down the subsonic primary inlet. The second part of the program is concerned with the simulation. In such case, the ejector geometry is known, and the physical parameters of operation and performance are to be determined. After validation, the program is used to estimate the cooling system parameters and the profile of ejector parameters, such as pressure, Mach number, as presented in the next sections.

II. Literature Review

Extensive experimental and theoretical investigations on thermal ejectors and their operation have been carried out during the last few decades. However, its modelling still represents a serious problem not yet completely resolved because of its highly complex flow field structure. Ridha et al. [3] studied the conjugate effects of ejector performance characteristics, the activation pressure-temperature conditions at the generator and the interaction with the compressor on refrigeration systems. Besides the conventional compression cycle, they selected three configurations: a hybrid ejector compressor booster and two cascade compressor ejector cycles. Dahmani et al. [4] presented a design methodology for simple ejector refrigeration systems of fixed cooling capacity. They carried out their investigation on four refrigerants (R134a, R152a, R290, and R600a). Ouzzane and Aidoun [5] derived a local mathematical model and computer programs for ejector studies in refrigeration cycles, one program for optimal ejector design and the other for simulation with more in-built flexibility. The model is based on Munday and Bagster's theory [6] and isentropic flow in the nozzles and the diffuser. In another study by Cardemil and Colle [7], a new

Mohamed Ouzzane is with Faculty of Engineering, the Islamic University of Medina, Saudi Arabia (email: m_ouzzane@iu.edu.sa).

Mahmoud Bady is with Faculty of Engineering, the Islamic University of Medina, Saudi Arabia, on leave from Assiut University, Assiut 271516, Egypt (corresponding author, email: mfbady@iu.edu.sa).

theoretical ejector model was developed for the performance evaluation of vapor ejectors operating in the critical mode. The model was derived based on the 1-D methodology and made use of real gas equations.

When the ejector is working under variable operating conditions, Jia and Wenjian [8] evaluated the influence of the area ratio on the entrainment ratio, COP and cooling capacity by replacing different sized nozzles. Varga et al. [9] numerically investigated a variable area ratio ejector with a removable needle and found that the entrainment ratio improved 77% compared to a fixed area ratio ejector at a low enough back pressure. Chen et al. [10] developed a two-dimensional theoretical model to study a variable-geometry ejector (VGE) and evaluate its effect on the cycle performance. They reported that the VGE is feasible for unstable heat-source utilization where it can be adjusted to its design point to obtain high efficiency. Sag and Ersoy [11] designed an ejector to reduce the throttling losses of a refrigeration system. Their proposed system obtained an optimal performance that had a 5-13% higher COP than the traditional system. Li et al. [12] carried out an investigation of the variable area ratio ejector on a multi-evaporator refrigeration system. The experiments indicated that energy saved was raised to 112% by the variable area ratio ejector compared to a conventional system. Other experimental results were introduced by Aphornratana and Eames [13] who showed the benefit of using an ejector with a primary nozzle that was moved axially in the cylindrical mixing chamber. They reported that; for a given ejector geometry and fixed condenser and evaporating temperatures; there exists an optimum temperature of the primary vapor which maximizes the entrainment ratio and the COP. Fenglei et al. [14] carried out an experimental investigation to study the performance of an ejector refrigeration system with refrigerant R134a. The effects of operating parameters and area ratio on the ejector performance were investigated. They concluded that the ejector performance is immediately changed by varying the ejector operational mode which is determined by the relation between the actual condensing temperature and the critical condensing temperature.

In the previous studies, the effect of generator pressure on the ejector-based cooling system still needs more focus in order to understand the ejector effect on the system performance under variable cooling loads. The present paper represents a step towards more investigations in the application of control techniques on the cooling system which uses thermal compression ejectors, instead of conventional compressors.

III. Description of the Cooling System with Ejector

Fig. 1 shows a schematic representation of the system under consideration, where the refrigerant is heated in the generator through solar energy (or low-grade energy source). The superheated vapor at state 3 is condensed by rejecting heat Q_{con} to a heat sink, which is normally ambient air or water. At 4, the exit from the condenser, the working fluid is assumed to be saturated liquid (quality x_4 = 0). Part of it (the secondary fluid $\dot{m}_s$) is throttled to low pressure at state 7 and evaporated by receiving heat from another fluid stream. The cooling of this stream represents the useful effect of the system Q_{evap}. At state 2, the exit from the evaporator, the working fluid is assumed to be superheated. Another part of the working fluid at state 5 (the primary fluid $\dot{m}_p$) is pumped to high pressure and superheated from 5 to 1 in the generator by receiving low-grade heat Q_{gen}. The high-pressure vapor at state 1 mixes with the secondary stream at state 2 in the ejector, where the exit mixture pressure is the condenser pressure. The mixing process of the two streams is complicated since they mix irreversibly and are compressed through a series of shocks in a constant area chamber. Fig. 2 shows the geometrical parameters of the thermal ejector.

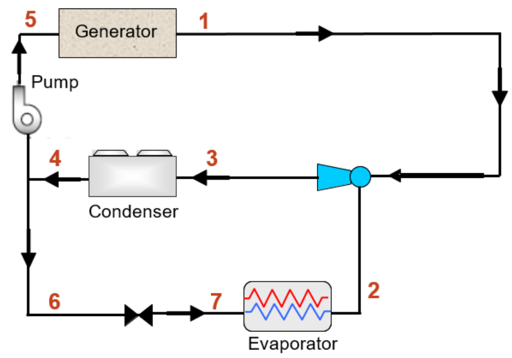

Fig. 1 Schematic of the proposed cooling system with an ejector activated by solar energy

IV. Mathematical Model of the Cooling System with Ejector

The construction of well-designed mathematical models of the ejector has become the key subject of many studies. Many mathematical models, found in the literature, have been developed and employed to analyze, develop and design ejectors [1]. These models include CFD simulations, global model, and the numerical models. Although CFD simulations give detailed information concerning pressure, velocity, Mach number…etc., the mathematical analysis using 1-D numerical modelling with computer programs represents a simple method of the flow mixing investigation if the appropriate conditions and equations are considered.

Certainly, the mathematical description of the flow inside the ejector is complex. Besides the conservation equations of mass, energy and momentum, the gas dynamic equations, state equations, isentropic relations as well as some appropriate assumptions need to be used to assist in the description of the flow and mixing in the ejector. Accordingly, to simplify the modeling, without loss of generality, the following main assumptions are applied:

1) The flow inside the ejector is steady and one dimensional.
2) Ejector inner wall is adiabatic.
3) The mixing of the primary and secondary streams in the ejector occurs at constant pressure.

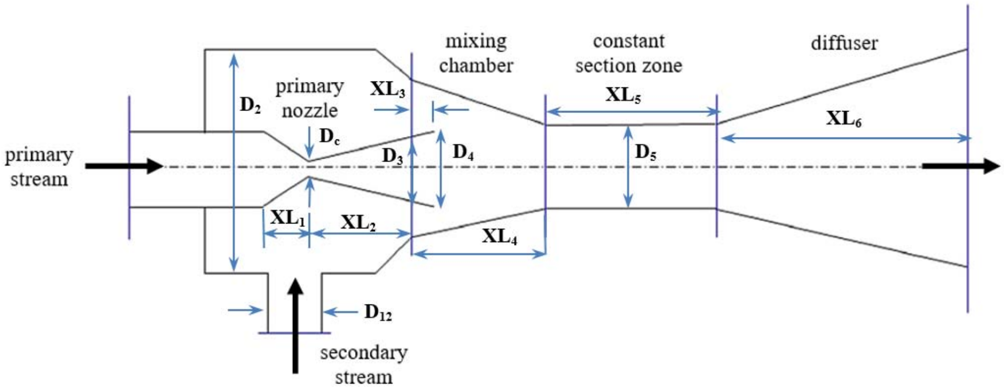

Fig. 2 Schematic diagram of ejector geometry

4) Primary and secondary streams preserve their identity over some distance following the exit from their respective nozzles, before mixing takes place.
5) The effects of frictional in the nozzles and the diffuser and mixing losses in the mixing chamber are taken into account by using coefficients introduced into the isentropic relations.
6) The pressure drop and heat loss in the piping system are neglected.

Generally, the ejector performance and geometry are expressed in terms of the entrainment ratio (ω), the compression ratio (τ) and the area ratio (A_r) defined as:

$$\omega = \frac{\dot{m}_s}{\dot{m}_p} \tag{1}$$

$$\tau = \frac{p_3}{p_2} \tag{2}$$

$$A_r = \frac{A_m}{A_t} \tag{3}$$

where A_m is the cross-section of the cylindrical mixing chamber and A_t is the throat area of the primary nozzle.

The cooling capacity of the system is calculated by:

$$\dot{Q}_{evap} = \dot{m}_s (h_2 - h_7) \tag{4}$$

The generator and pump powers are calculated respectively by:

$$\dot{Q}_{gen} = \dot{m}_p (h_1 - h_5) \tag{5}$$

$$\dot{W}_p = \dot{m}_s (h_5 - h_4) \tag{6}$$

Similarly, the condenser heat rate is given by:

$$\dot{Q}_{con} = (\dot{m}_s + \dot{m}_p)(h_3 - h_4) \tag{7}$$

V. Solution Procedure

The simulation program is written to predict the behavior of a fixed geometry ejector, in response to imposed inlet conditions. The input parameters for this program are the ejector dimensions, generator temperature (T_{gen}) and evaporator temperature (T_{evap}). The program output data are the primary flow rate ($\dot{m}_p$)), the secondary flow rate ($\dot{m}_s$), the entrainment ratio (ω), the pressure ratio (τ)... etc. Modulation functions are embedded in this program such that refrigerant flow rates at an inlet are self-adjusting according to external operating constraints. In this way, ejector operation and performance can be analyzed under different conditions, including off-design situations.

For simulation, the ejector geometry is known, and the physical parameters of operation and performance are to be determined. Since the constitutive equations being of coupled, non-linear type, an iterative procedure given by the flowcharts shown in Fig. 3 is applied to simulate the base case ejector in off-design conditions. For more details concerning the mathematical model and the solution technique, refer to the previous investigation carried by Ouzzane et al. [5].

VI. Results and Discussion

In this section, the theoretical model is validated against the published work at first. Then, the results of the mathematical model are introduced, where the parametric analysis, as well as, the performance of the cycle is investigated.

A. Validation of the Proposed Model

The theoretical model used in this study is based on the one developed previously by Ouzzane et al. [5] with a small adjustment of certain factors. At this time, the model has been validated using measurement data obtained by Huang et al. [15] for R141 b refrigerant. For comparison purposes, the experimental and theoretical data are presented in the same figure to show the variation of the entrainment factor versus the saturated temperature at the exit of the ejector. It has been found that the trends are similar and the agreement between experimental and calculated data is satisfactory since the discrepancies in the region of off-design do not exceed 13%. Recently, an experimental work carried out by Fenglei et al. [13] on an ejector operating under different modes using the same refrigerant (R134a) as our work has been published. Such paper provided an interesting result and enough information that can be used for validation. The ejector experimented consists of two interchangeable main parts; nozzle and ejector body including mixed chamber and diffuser. The authors combined two different nozzles (A and B) with three bodies (A, B and C) to test different ejectors with different section ratios (A-A, A-C, B-A, and B-B). Based on these data, the two ejector tools developed in the present work; design tool and simulation tool have been validated.

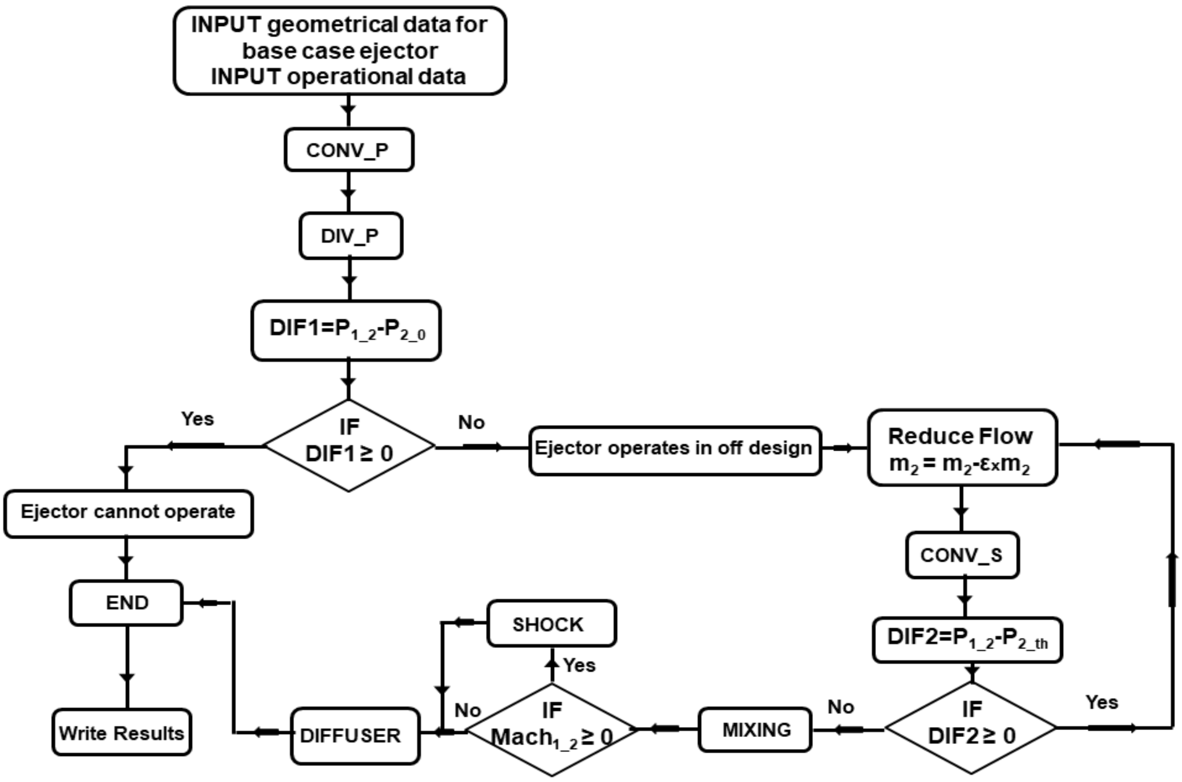

Fig. 3 Flowchart of the main iterative calculation steps

TABLE I
COMPARISON OF EJECTOR DESIGN DATA AGAINST PUBLISHED EXPERIMENTAL RESULTS [13]

D_{1c}		D_5		D_{12}		D_3		T_{cond} (°C)	
Exp	Cal	Exp	Cal	Exp	Cal	Exp	Cal	Exp	Cal
2.09	1.97	4.16	3.98	2.70	3.07	12.90	12.73	32.0	35.3

Table I presents the results related to the ejector design tool for an ejector with area ratio A_r = 3.96. The different geometrical parameters compared here are diameters of the throat, divergent, mixing chamber and the exit of the diffuser. The saturation temperature at the exit of the ejector, presented in the last column of the table is also an output parameter used for comparison. From the table, it can be seen that the agreement between the actual ejector sizes and the calculations is very satisfactory. However, the difference for the diameter of the divergent D_{12} is a little bigger. At this location (exit of the nozzle), the mixing of the two streams starts. This process is the most complicated part for the modeling because of the complexity of multiple physical phenomena including sound shock waves and high intensity of frictions. On the other hand, the mixing process does not happen immediately after the nozzle exit at a constant section, but it occupies a certain length which depends on many parameters and it is very difficult to estimate its value.

For the saturation temperature at the exit of the ejector presented in the last column of the table, it is clearly shown that the theoretical model overestimates this parameter due to certain assumptions applied in this study. The simulation ejector tool has also been validated by the experimental data presented by Fenglei et al. [13]. The comparison concerned the variation of the entrainment ratio versus the saturation temperature at the exit of the ejector operating under the following conditions: T_g = 75 °C and T_{evap} = 15 °C. Fig. 4 shows that the trends of the entrainment ratio ω versus the condenser pressure for both simulation and measurements are similar. In the region of the off-design conditions, a right shift of around 2 kPa is observed in the calculated data due to the same reason as for the ejector design tool. In general, it can be concluded that; we developed two strong tools able to reflect with good accuracy the behavior of thermal ejectors.

B. Flow Parameters inside the Ejector

Figs. 5 and 6 show the profiles of refrigerant pressure and Mach number inside the ejector for three different generator pressures: the base case of 2117 kPa, a lower pressure of 1318 kPa and higher pressure of 3244 kPa. The curves of the secondary streams are not shown in such figure to make the graph clearer. As shown in these figures, the pressure accelerates as it enters the convergent section of the nozzle and reaches the speed of sound at the nozzle throat. The speed is further increased while expanding through the divergent nozzle. At the nozzle exit, the primary fluid expands out with supersonic speed resulting in a low-pressure region, which allows a secondary fluid to be entrained into the suction chamber. During the mixing process of the two streams, the pressure is assumed to remain constant. By the end of the mixing section, the two streams are completely mixed, and due to the high-pressure region downstream, a shock wave is induced. This shock wave causes a major compression effect and a sudden drop in the flow speed from supersonic to subsonic. Further compression is achieved as the stream is brought to stagnation through the diffuser.

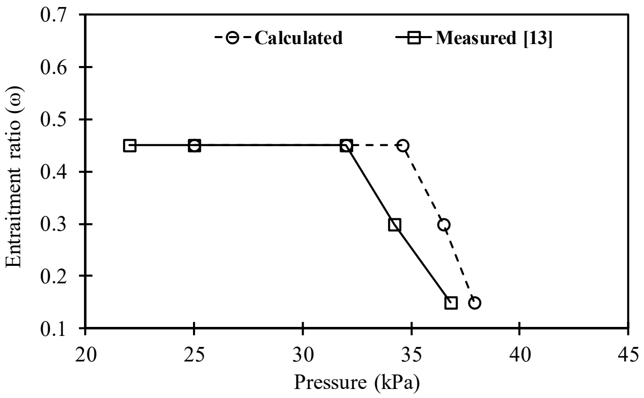

Fig. 4 Comparison of entrainment ratio ω versus pressure from calculated and measured data

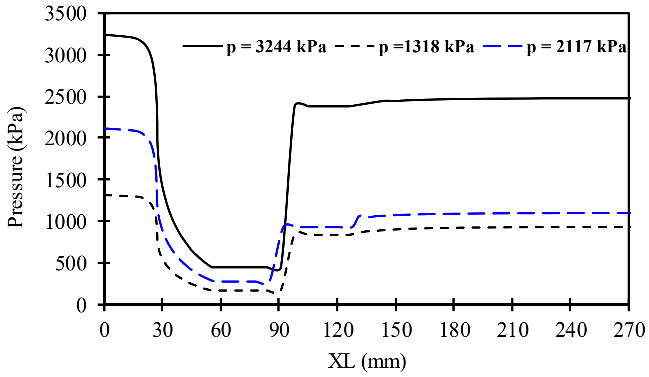

Fig. 5 The pressure profile inside the ejector for different generator pressures

Figs. 5 and 6 show that; the greater the generator pressure the stronger the pressure in the ejector. In the same time, the Mach number trends show a strong shock wave for high generator pressure compared with the low pressure. This is due to the difference between the ejector pressure in the constant cross-section zone and the exit pressure at the diffuser outlet. The two parameters; pressure and Mach number are developed in opposite directions since the decrease in pressure energy leads to increase the kinetic energy.

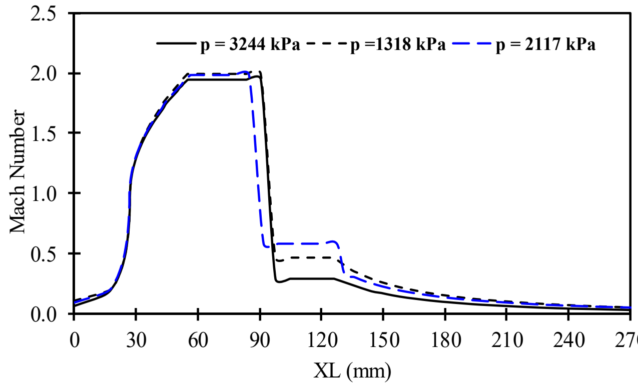

Fig. 6 The profile of Mach number inside the ejector for different generator pressures

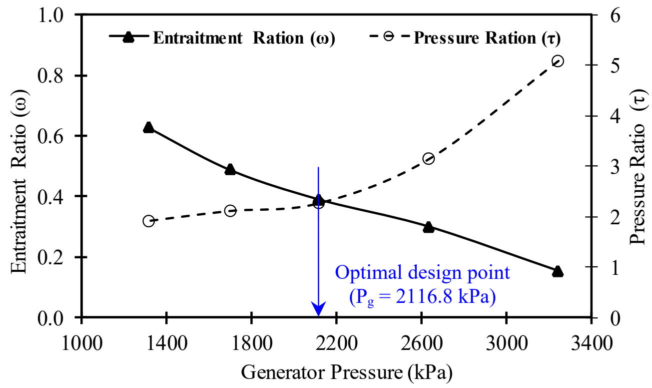

Fig. 7 Entrainment and pressure ratios versus the generator pressure

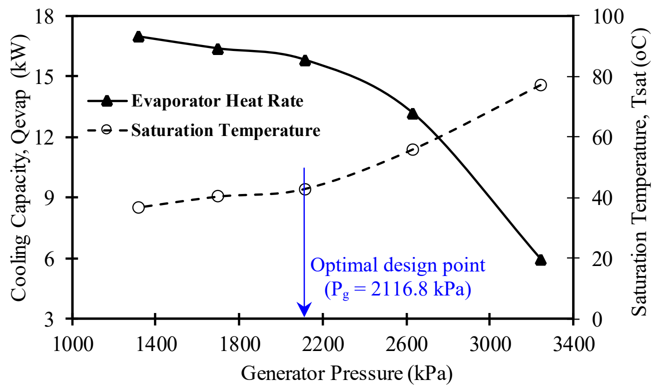

Fig. 8 Cooling capacity and saturation temperature of the generator pressure versus the generator pressure

At high generator pressure, the expansion in the mixing zone is great, which creates a low-pressure difference with the secondary inlet pressure (p_{mix} - p_2). This decreases the induced mass flow rate of the secondary stream (see Fig. 7), which in turn, decreases the cooling capacity of the system (see Fig. 8). This effect can be confirmed by the trend of the saturation temperature, given in Fig. 8. The condenser saturation temperature increases when the generator pressure increases. The required heat removal rate by the condenser, in this case, is higher than the design values. One important observation here is that; the system can operate very well with low generator pressures (up to the design value of 2117 kPa). However, to complete the analysis, the system is not independent of the ambient conditions. It is known that the greater the condenser saturation temperature, the lower the cooling capacity the system can deliver. For the air conditioning application, the condenser is located outside the building and interacts with the external ambient temperature. The condenser saturation temperature must be higher than ambient temperature in order to reject heat to the surrounding and then condenses the refrigerant. Such a condition is not easy to be attained since it depends on the weather conditions. Accordingly, the operating generator pressure of the cooling system is affected with the weather conditions (ambient temperature). This is why the applied generator pressure must be high enough to deliver the required cooling capacity.

VII. Conclusions

Based on theoretical model, two powerful ejector tools for design and simulation using FORTRAN have been developed and validated. These two codes can be used to study in details the effect of many parameters on the system performance. The simulation results show that:

1) The generator pressure has a great effect on:
   - the pressure ratio, and the entrainment ratio
   - the profile of ejector parameters, such as pressure, Mach number, enthalpy, etc.
   - the whole cooling system parameters, such as the cooling capacity, pump power, etc.
2) At high generator pressures, strong shock waves inside the ejector are occurred, which lead to significant condensing pressure at the ejector exit (condenser inlet). Additionally, at such high p_g, the designed system has the ability to deliver cooling capacity for high condensing pressure during hot seasons.

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