# A Theoretical Analysis of Air Cooling System Using Thermal Ejector Adapted to Operation Conditions for Control Strategy

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**Abstract**

The thermal ejector is a passive component used for thermal compression, activated by heat (waste or solar), applied mainly for cooling and refrigerating. Nowadays, it is of interest to many researchers and engineers worldwide. The present study introduces a theoretical analysis of the cooling system which uses a gas ejector thermal compression. In such work, the ejector performance is adapted according to the operation conditions of the cooling system in order to attain a control strategy to satisfy the required cooling load with acceptable performance. Theoretical models are developed and applied for the design and simulation of the ejector. 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. These models coupled with the equations of the other components (condenser, evaporator, pump, and generator) are used to analyze the performance of the cooling system. Two FORTRAN programs are developed to carry out the investigation; one for the ejector design and the other is for the simulation purpose. 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. So, it is considered as the key parameter in this investigation to evaluate the cycle performance. The effectiveness of the model is verified by comparing the calculated results with experimental data available in the literature. Then, the simulation results have been used to propose a control strategy to select the appropriate ejector for a given operating condition, where multiple parallel ejectors are used in the system. From the study results, it was found that; for generator pressures lower than the design pressure, the ejector is working very well, where the cycle performance parameters equal to or lower than the required values by the system design. At high generator pressures, strong shock waves inside the ejector are occurred, which leads to significant condensing pressure at the ejector exit (condenser inlet). At such high pressure, the designed system has the ability to deliver cooling capacity for high condensing pressure during hot seasons.

**Keywords**

Air cooling system, refrigeration, thermal ejector, thermal compression.

## تحليل نظري لنظام تبريد الهواء باستخدام القاذف الحراري المتكيف مع ظروف التشغيل لاستراتيجية التحكم

القاذف الحراري هو عنصر سلبي يستخدم للضغط الحراري ، ويتم تنشيطه بالطاقة الحرارية سواء الطاقة المهدرة أو الطاقة الشمسية، ويستخدم بشكل أساسي في دوائر التبريد والتكييف حيث أصبح القاذف في الوقت الراهن محط اهتمام العديد من الباحثين والمهندسين في جميع أنحاء العالم وبالتالي تقدم الدراسة الحالية تحليلًا نظريًا لدائرة تبريد تستخدم قاذف غازي للضغط الحراري حيث يتم تغيير أداء القاذف وفقًا لظروف تشغيل نظام التبريد من أجل تحقيق استراتيجية تحكم تضمن الحصول علي حمل التبريد المطلوب بأداء مقبول وقد تم تطوير نماذج رياضية نظرية وتطبيقها في عملية تصميم ومحاكاة القاذف إلى جانب معادلات حفظ الكتلة والطاقة وكمية التحرك ،كما تم تطبيق المعادلات الديناميكية للغاز، ومعادلات الحالة ، والعلاقات القياسية وكذلك بعض الافتراضات المناسبة لمحاكاة التدفق والخلط داخل القاذف وتم استخدام هذه النماذج إلى جانب معادلات الاجزاء دائرة التبريد الأخرى مثل المكثف ، المبخر ، المضخة ، والمولد لتحليل أداء نظام التبريد ولعمل هذه الدراسة قام الباحثون بتطوير برنامجين بلغة الفورتران $FORTRAN$ لإجراء الدراسة ؛ أحدهما لتصميم القاذف والآخر لعملية المحاكاة وتم حساب خواص مائع التبريد (فريون 134– أ) باستخدام معادلات الغاز الحقيقية ومن بين العديد من المتغيرات ، اعتبر المؤلفون أن ضغط المولد هو حجر الزاوية في دورة التبريد ولذلك تم استخدامه كمتغيررئيسي في هذه الدراسة لتقييم أداء الدورة و تم التحقق من فعالية النموذج الرياضي المستخدم من خلال مقارنة النتائج المحسوبة مع البيانات التجريبية المتاحة في الدراسات القديمة وعلي هذا الاساس تم استخدام نتائج المحاكاة لاقتراح استراتيجية تحكم لتحديد القاذف المناسب لحالة تشغيل معينة ، حيث يمكن استخدام قاذفات متوازية متعددة في نظام التبريد و قد أظهرت نتائج الدراسة أنه عندما يكون ضغط المولد أقل من ضغط التصميم ، يعمل القاذف بشكل جيد للغاية ، حيث تكون معاملات أداء الدورة مساوية أو أقل من القيم المطلوبة من النظام.... أما عند الضغوط الاعلي فأنه تحدث موجات تصادمية قوية داخل القاذف، مما يؤدي إلى ضغط عالي عند مخرج القاذف (مدخل المكثف) وبالتالي، عند هذا الضغط العالي، فإن نظام التبريد تكون لديه القدرة على توفير قدرة تبريد خلال المواسم الحارة.

**Nomenclature:**

| | |
|---|---|
| A | Cross sectional area ($m^2$) |
| COP | Coefficient of performance (--) |
| D | Diameter (m) |
| h | Enthalpy (J/kg.K) |
| M | Mach number (--) |
| $\dot{m}$ | Mass flow rate (kg/s) |
| p | Pressure (kPa) |
| Q | Heat rate (W) |
| s | Entropy (J/K) |
| T | Temperature (°C) |
| u | Speed (m/s) |
| $V_{son}$ | Sonic speed (m/s) |
| $\dot{W}$ | Power (kW) |
| XL | Section length (m) |
| ρ | Density ($kg/m^3$) |
| ω | Entrainment ratio (--) |
| φ | Coefficient of friction due to mixing |
| τ | Compression ratio (--) |

## 1. Introduction

Recently, thermal ejectors have received a lot of interest in the 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 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.

The most important feature in the ejector-cooling system is that; it can use renewable sources of energy to drive the generator. Solar and wind energies represent the most promising energy resources to drive heat-recovery systems, as they are easily accessible and cheap compared to other renewable energy sources. However, the supply of these energies is unstable, which represents a serious problem in the regulation and stabilization of the ejector-cooling systems powered by these energy sources. Accordingly, there is a serious need to design a control technique within the cycle components that can operate the cooling system according to variable operating conditions. From this perspective, the main objective of the current research is to enhance the performance of the ejector-cooling system by controlling the flow conditions with variable operating conditions.

To apply a control technique, the present study proposes the use of multiple parallel ejectors in the cooling system, where only one ejector is in operation. The other ejectors are switched off during the operation of ejector 1 (Ej-1). If -for any variation in the operating conditions of the cycle- the working ejector is unable to attain the required cooling capacity, the refrigerant flow to that ejector is stopped, and the flow path is switched to one of the other ejectors. The control depends basically on the obtained cooling load of the system compared to the desired one, and the cycle the coefficient of performance (COP).

## 2. 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 et al. [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 et al. [7], a new 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, Yan et al. [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 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 et al. [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 et al. [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.

However, in the previous studies, no literature was found concerning the application of control strategy on the ejector flow according to specific operating condition through the application of the multi-ejector system. The present paper appears to be the first step towards more investigations in the application of control techniques on the cooling system which uses thermal compression ejectors, instead of conventional compressors.

## 3. Description of the cooling system with ejector

Figure 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 6 and evaporated by receiving heat from another fluid stream. The cooling of this stream represents the useful effect of the system (cooling capacity $Q_{evp}$). At state 2, the exit from the evaporator, the working fluid is assumed to be saturated vapor (quality $x_2 = 1$). The remaining part of the working refrigerant 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.

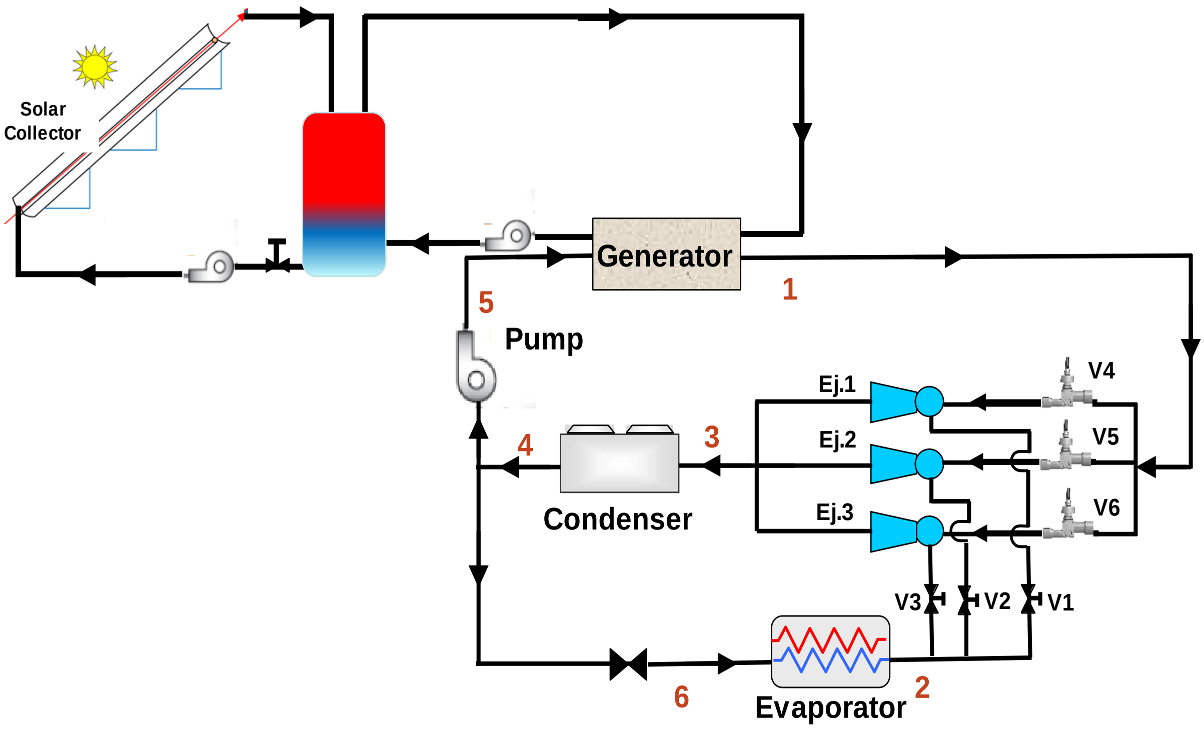



Figure 1: Schematic of the proposed cooling system with an ejector activated by solar energy

## 4. Mathematical Model of the cooling system with an 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. These models include CFD simulations, global models, and 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. The kinetic energy at the inlets of the primary and secondary flows and at the exit of the diffuser are negligible.

3. Ejector inner wall is adiabatic.
4. Uniform pressure at the position of the mixing section under optimal operating conditions.
5. Primary and secondary streams preserve their identity over some distance following the exit from their respective nozzles, before mixing takes place.
6. The effects of frictional in the nozzles and the diffuser and mixing losses in the mixing chamber are taken into account by using constant coefficients introduced into the isentropic relations.

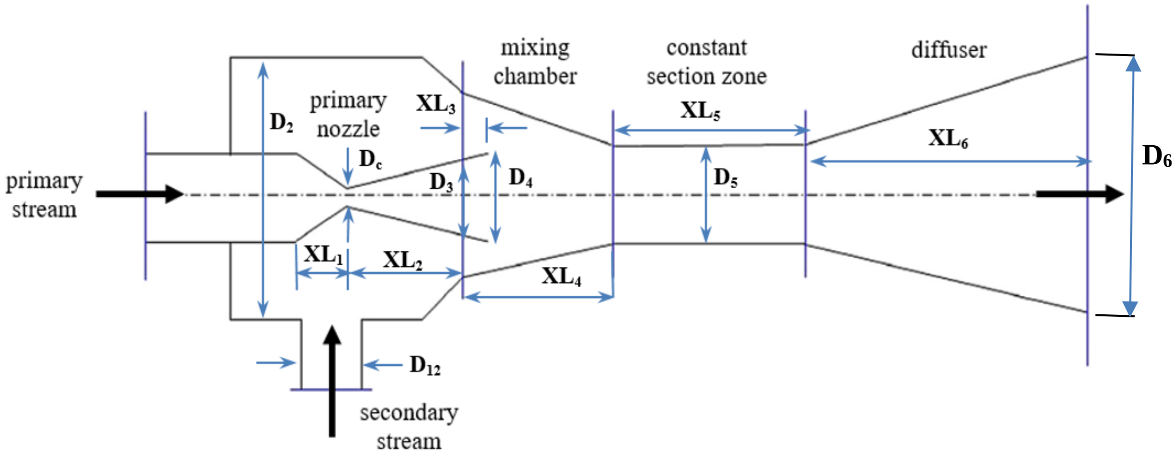



Figure 2 Schematic diagram of ejector geometry.

The fundamental conservation equations of momentum, energy, and mass are applied to elementary control volumes in the different zones of the ejector (primary and secondary nozzles, the mixing chamber, the constant section zone, and the diffuser), as shown in Fig. 2. In such case, for an elementary control volume, shown in Fig. 3, the following equations are applied:

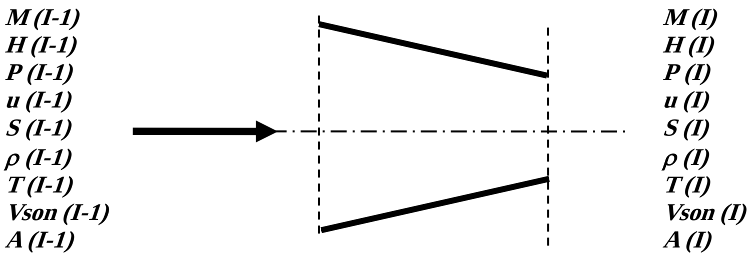



Figure 3 Elementary control volume.

Energy conservation:

$$h(I)+\frac{u(I)^2}{2}=h(I-1)+\frac{u(I-1)^2}{2} \tag{1}$$

Mass conservation:

$$\rho(I)*u(I)*A(I)=\rho(I-1)*u(I-1)*A(I-1) \quad (2)$$

Isentropic compression and expansion:

$$s(I)=s(I-1) \quad (3)$$

Momentum conservation:

$$\rho(I)*u^2(I)+p(I)=\rho(I-1)*u^2(I-1)+p(I-1) \quad (4)$$

where (*I-1*) is the inflow section and (*I*) is the outlet flow section.

For the mixing process, the following equation is applied with the coefficient of friction due to mixing φ:

$$\varphi\left[p_1A_1+p_2A_2+u_1\dot{m}_1+u_2\dot{m}_2\right]=pA_m+u_m\dot{m}_t \quad (5)$$

where $p_1 = p_2$ and $\dot{m}_t = \dot{m}_1 + \dot{m}_2$

The momentum equation is needed when shock conditions are present, particularly in the constant section zone or during off-design operation. The calculation procedure varies somewhat, depending on the calculation being performed. For design, the optimization criterion is the critical flow in the nozzles (double shocking in the primary and secondary convergent sections, sonic conditions). The conservation equations stated above are applied along the same general lines for both the design and the simulation procedures.

Generally, the ejector performance and geometry are expressed in terms of the entrainment ratio ($\omega$), the compression ratio ($\tau$) and the area ratio ($A_r$) defined as:

$$\omega=\frac{\dot{m}_s}{\dot{m}_p} \quad (6)$$

$$\tau=\frac{p_3}{p_2} \quad (7)$$

$$A_r=\frac{A_m}{A_t} \quad (8)$$

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

The performance of the cooling system as a whole is determined through COP. For systems using an ejector activated by an external heat source, the COP is given by:

$$COP=\frac{\dot{Q}_{evap}}{\dot{W}_p+\dot{Q}_{gen}} \quad (9)$$

where:

$$\dot{Q}_{evap} = \dot{m}_s (h_2 - h_6) \quad (10)$$

$$\dot{Q}_{gen} = \dot{m}_p (h_1 - h_5) \quad (11)$$

$$\dot{W}_p = \dot{m}_s (h_5 - h_4) \quad (12)$$

Similarly, the condenser heat is given by:

$$\dot{Q}_{con} = (\dot{m}_s + \dot{m}_p)(h_3 - h_4) \quad (13)$$

In the case of not considering the generation heat rate when solar/waste energy is used as the heat source, the COP is estimated as:

$$COP = \frac{\dot{Q}_{evap}}{\dot{W}_p} \quad (14)$$

## 5. Solution Procedure

Simulation and design of ejectors are two requirements of our modelling which cannot be obtained by the same calculation procedure. Some steps are specific to design while others are applied for simulation. Based on these considerations, two program versions are developed: one for design and the other is for simulation.

Ejector design requires the following main input parameters: the cooling capacity ($Q_{evap}$), the entrainment ratio ($\omega$), generator pressure ($p_g$), and the evaporator temperature and pressure ($p_{evap}$) and ($T_{evap}$). Then, the program then determines the ejector geometry and the dimensions such as the secondary stream inlet diameter ($D_{12}$), critical diameter ($D_{1c}$), constant section zone diameter ($D_5$) as well as ejector exit pressure ($p_{exit}$). This design corresponds to a unique set of conditions, which is the optimal design point. If outside conditions vary, the ejector will not operate optimally. This new condition will be off-design and cannot be handled correctly by this program version.

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 mass flow rate ($\dot{m}_p$), the secondary mass flow rate ($\dot{m}_s$), the entrainment ratio ($\omega$), the pressure ratio ($\tau$), ..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.

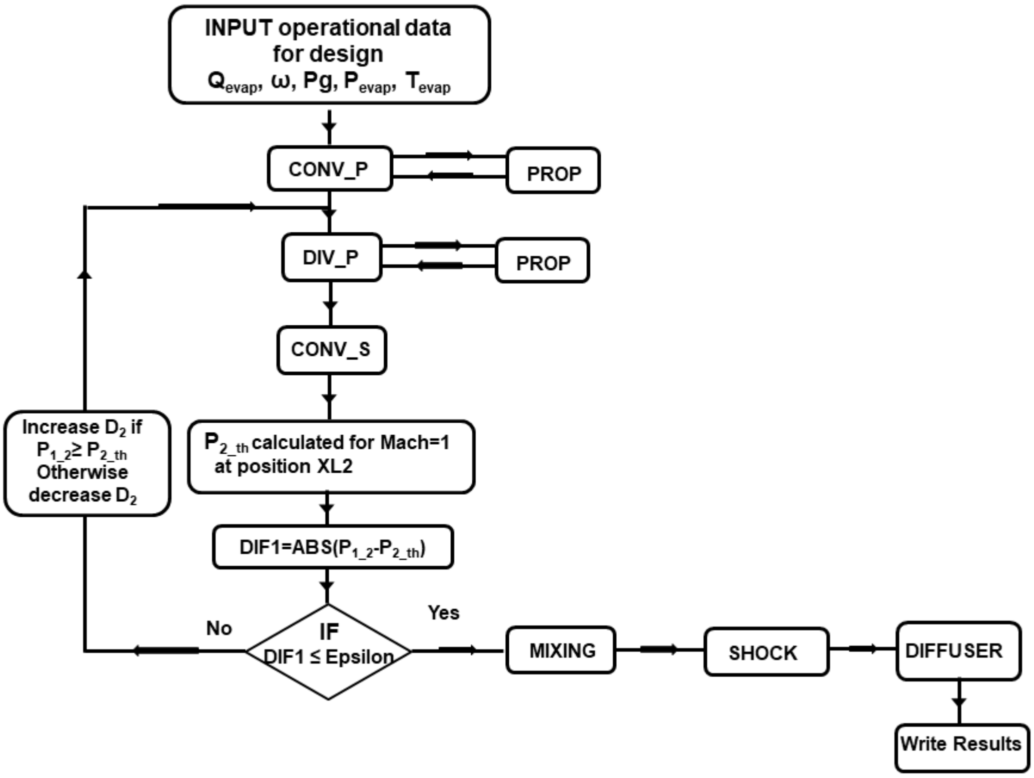
Based on the above system of equations, two different versions of FORTRAN programs are developed; Version A (for design) and Version B (for simulation). For ejector design, the computation progress and control is based on Mach number increments down the subsonic primary convergent until choking occurs (critical condition). At this point (M = 1), $XL_1$ and $D_c$ showed in Fig. 2, are determined. Primary flow progresses through the divergent $XL_2$ and the fictive expansion cone $XL_3$. The final position of the primary stream before mixing is $X = XL_1 + XL_2 + XL_3$, where the prevailing pressure is $Pc_1$. Secondary flow starts at $X = XL_1$ and is accelerated to its critical conditions ($M_2 = 1$) at $X = XL_1 + XL_2 + XL_3$. Its pressure condition is $Pc_2$. In order for mixing to proceed, critical pressures must match. For this purpose, a pressure test is performed such that:

If $Pc_2 > Pc_1$, then $D_2$ is increased

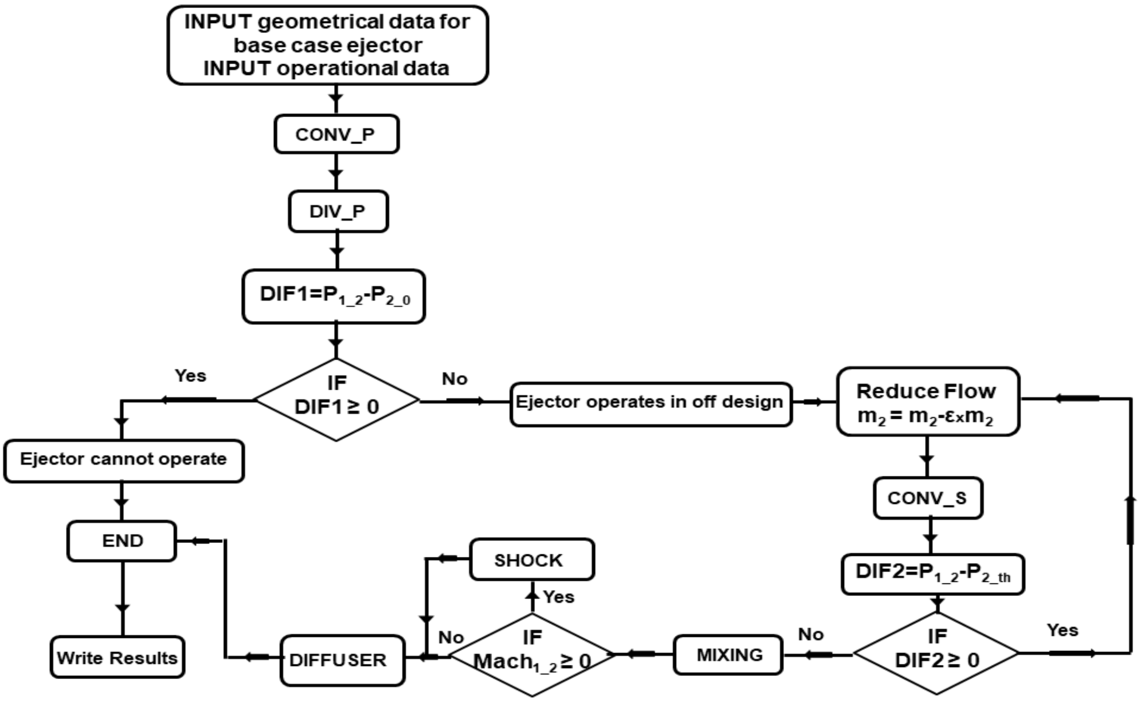
If $Pc_2 < Pc_1$, then $D_2$ is decreased

The calculation is repeated until the pressures are close and $Pc_2 \geq Pc_1$, in which case $D_2$ is determined. At this point, the flow is supersonic, as it enters the convergent mixing section, where it is slowed down to $M \cong 1$. Then, $XL_4$ and $D_5$ are determined. The length of the constant section zone, $XL_5$ is determined empirically. In this section, a shock wave is produced with a corresponding pressure (and temperature) increase resulting in subsonic flow at the diffuser entry. Additional isentropic compression of the flow takes place in the diffuser to velocity just high enough to flow to the condenser. Values of $XL_6$ and $D_6$ are thus determined. Figure 4a shows the solution procedure followed by the authors to design the ejector.

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 flow-charts shown in Fig. 4b 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].




(a)




(b)

Figure 4 Flowchart of the main iterative calculation steps; a) for design, and b) for simulation.

## 6. 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 are investigated.

### 5.1 Validation of the proposed model

The theoretical model used in this study is based on the one developed previously by the co-author of this paper 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 don’t 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.

Table 1 presents the results related to the ejector design tool for two different ejectors having two different area ratios respectively: ejector 1 ($A_r$ = 2.96) and ejector 2 ($A_r$ = 2.77). for calculations, the input data, presented in the first column, are the cooling capacity ($Q_{evap}$) the entrainment ratio (ω), the temperature of the generator ($T_g$) and the saturation temperature in the evaporator ($T_{evap}$). The different geometrical parameters compared 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.

Table1. Comparison of ejector design data against published experimental results [13]

| Ejector | $D_{1c}$ | | $D_5$ | | $D_{12}$ | | $D_3$ | | $T_{cond}$ (°C) | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Exp. | Cal. | Exp. | Cal. | Exp. | Cal. | Exp. | Cal. | Exp. | Cal. |
| <u>Ejector 1</u><br>$A_r$ = 3.96<br>$Q_{evap}$ = 2.0 kW<br>$\omega$ = 0.45<br>$T_g$ = 75 °C<br>$T_{evap}$ = 15 °C | 2.09 | 1.97 | 4.16 | 3.98 | 2.70 | 3.07 | 12.90 | 12.73 | 32.0 | 35.3 |
| <u>Ejector 2</u><br>$A_r$ = 2.77<br>$Q_{evap}$ = 0.55 kW<br>ω = 0.08<br>$T_g$ = 75 °C<br>$T_{evap}$ = 10 °C | 2.50 | 2.44 | 4.16 | 4.37 | 3.30 | 4.04 | 12.90 | 13.12 | 34.1 | 36.8 |

From the table, it can be seen that; the agreement between the actual ejector sizes and the calculations is very satisfactory. However, the difference in 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 of the modeling because of the complexity of multiple physical phenomena including sound waves and high intensity of frictions. On the other hand, the mixing process doesn't 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's clearly shown that the theoretical model overestimates this parameter due to the 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 for ejector 1 operating under the following conditions: $T_g$ = 75 °C and $T_{evap}$ = 15 °C. Figure 5 below 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 the ejectors.

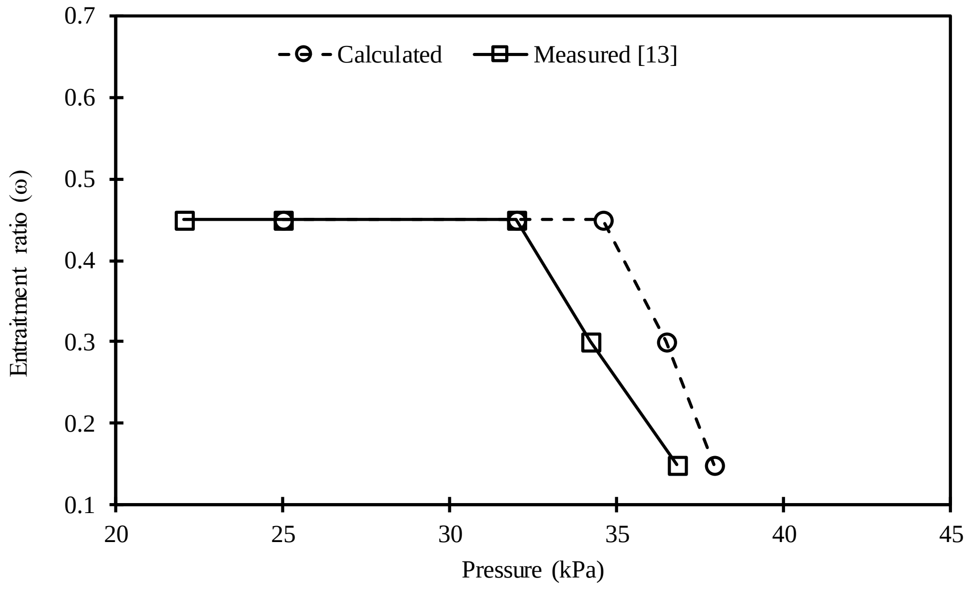



Figure 5 Comparison of entrainment ratio ω versus pressure from calculated and measured data.

## 5.2 Generator pressure effect

For this study, a base-case is considered, where the input parameters are shown in Table 2.

Table 2. Input parameters and operational conditions for the base-case of ejector 1 design (Ej-1)

| $Q_{evap}$ (kW) | ω | $T_{gen}$ (sat) (°C) | $p_{gen}$ (kPa) | $\dot{m}_1$ (kg/s) | $T_{evap}$ (sat) (°C) | $p_{evap}$ (kPa) | $\dot{m}_2$ (kg/s) |
|---|---|---|---|---|---|---|---|
| 20.0 | 0.4 | 90.0 | 2116.8 | 0.268 | 15.0 | 488.0 | 0.107 |

The design program used these input data to determine the corresponding ejector dimension as presented in Table 3.

Table 3. Dimensions of ejector 1 (Ej-1).

| Convergent (primary) | Divergent (primary) | Convergent (mixture) | Constant section | Diffuser |
|---|---|---|---|---|
| $D_1$ =18.00 | $D_c$ = 6.4 | $D_2$ = 12.63 | $D_5$ = 11.90 | $D_5$ =11.90 |
| $D_c$ = 6.40 | $D_3$ = 9.64 | $D_5$ = 11.90 | - | $D_6$ = 32.5 |
| $XL_1$ = 26.8 | $XL_2$ = 28.0 | $XL_3$ = 14 | $XL_5$ = 71.00 | $XL_6$ = 150.0 |
| Angle =14° | Angle = 3° | Angle = 1.5° | - | Angle = 4° |

The effect of generator pressure ($p_g$) on the cooling capacity ($Q_{evap}$), coefficient of performance of the system, the generator heat load ($Q_{gen}$) and the condenser saturation temperature ($T_{sat}$) are presented in Fig. 6 through Fig. 9. In these figures, there are two curves, the dashed curve, which represents the design results, and the continuous curve which represents the simulation results of the ejector base-case (Ej-1). In such analysis, two different zones are considered; Zone I, for the case when $(p_g)_{system} \leq (p_g)_{optimal\text{-}design}$, and Zone II, for the case, when $(p_g)_{system} > (p_g)_{optimal\text{-}design}$.

Figure 6 shows the variation of the evaporator cooling load with the generator pressure. Two important observations can be made: firstly, the cooling load decreases when the generator pressure increases, and secondly, almost identical values of $Q_{evap}$ are obtained for both design and simulation in Zone I. The cooling system, in that case, operates very well. This is mainly due to the low condensing temperature (cold weather).

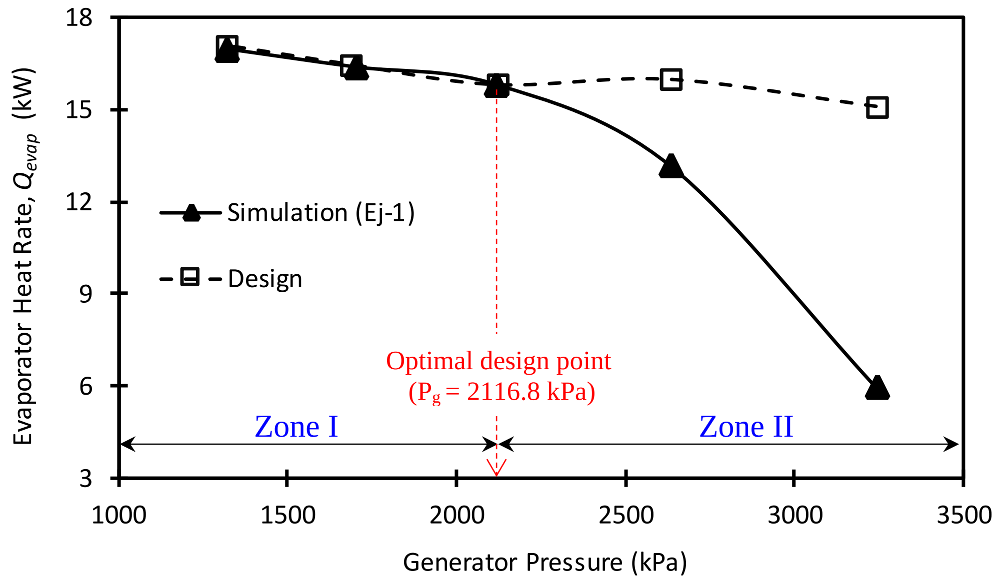



Figure 6. Cooling capacity versus the generator pressure.

On the other hand, increasing $p_g$ beyond the optimal point leads to a considerable decrease in the evaporator cooling capacity. Accordingly, the cooling system cannot satisfy the required evaporator load with acceptable performance. In such a case, increasing the generator pressure increases the mixing pressure at the exit of the primary and secondary nozzles. Thus, the system is not able to deliver the required cooling capacity, since the mass flow rate of the system is lower than that of the optimal design case. Accordingly, a control valve is needed in such a situation to switch the refrigerant to either Ej-2 or Ej-3 depending on the required cooling capacity of the system.

The effect of generator pressure on the generator heat rate is presented in Fig. 7. The design values of $Q_{gen}$ are almost constant with an average value of 51.5 kW, while the simulation heat rate increases sharply as $p_g$ increases. In Zone I, the cooling system operates very well, where the $Q_{gen}$ is lower than the required value to activate the generator. When $p_g$ exceeds the optimal design value (Zone II), the difference between the design and simulation values becomes positive. In such a case, $Q_{gen}$ is considerably higher than the value required by the system, which is expected to decrease the COP of the cooling system, since the evaporator heat rate decreases with increasing $p_g$.

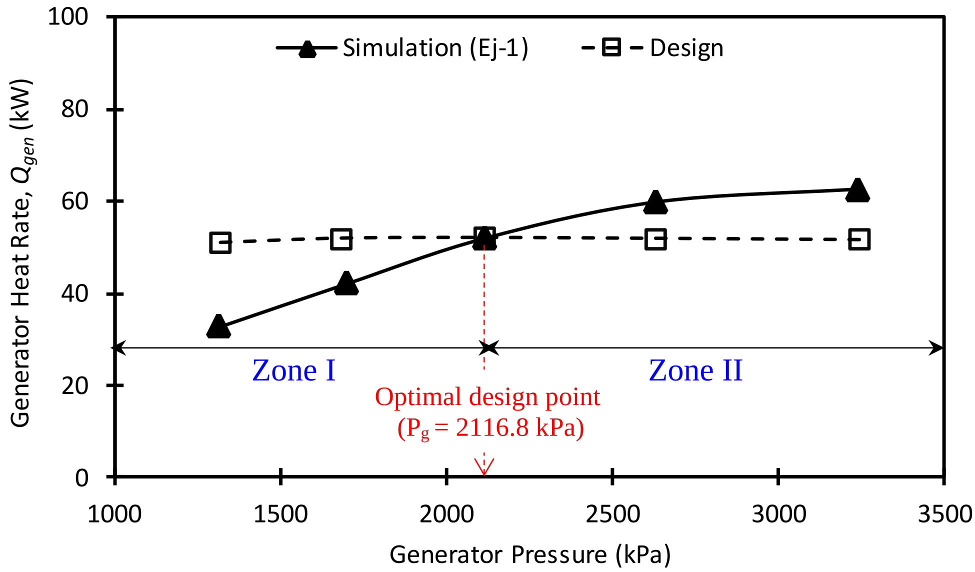



Figure 7. Variation of generator heat rate with the generator exit pressure

At the same time, the cycle COP variation with the generator pressure is illustrated in Fig. 8. The design COP is almost constant at a value of 0.3, while the simulation COP decreases as $p_g$ increases. Such a trend is expected to occur since $Q_{evap}$ decreases and $Q_{gen}$ increases considerably with the increase of $p_g$, while the pump power does not vary much. Generally, the COP behavior reflects the trends of $Q_{evap}$ and $Q_{gen}$ given in Figs. 6 and 7, respectively.

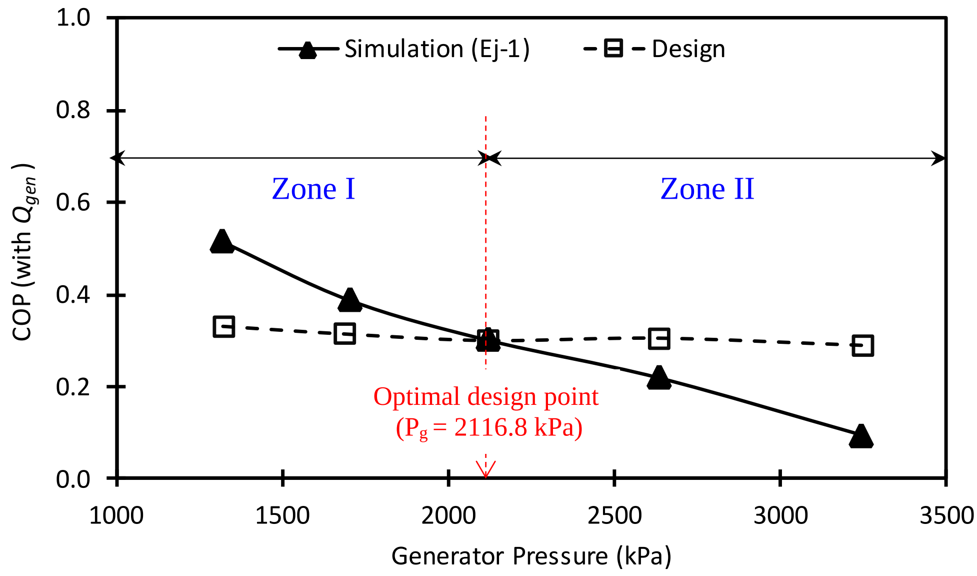



Figure 8. Variation of the cycle COP with the generator pressure.

Figure 9 shows the effect of generator pressure on the condenser saturation temperature. The trends of both design and simulation curves are similar, where they increase as the generator pressure increases.

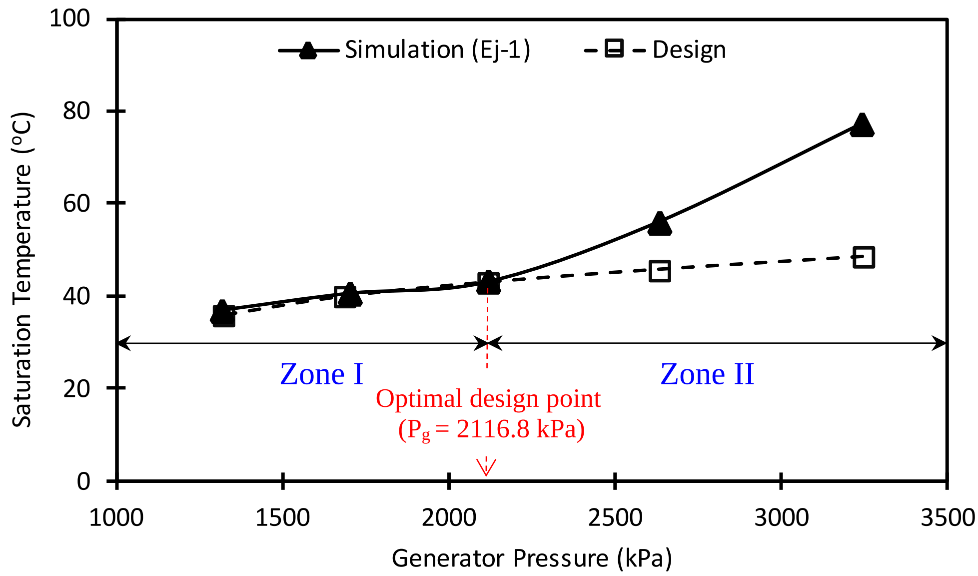



Figure 9. Saturation temperature of the generator pressure versus the generator pressure.

The trend of condenser saturation temperature reflects two important observations. First, the design and simulation values of the saturation temperatures are almost identical when $p_g$ is lower than the optimal design pressure of 2116.8 kPa (Zone I). In this case, the cooling system performs very well, where no extra heat is required to be removed by the condenser. Second, the 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. Such a result, confirms the need for switching the refrigerant flow to either Ej-2 or Ej-3.

One important observation from the above analysis is that; the system can operate very well with a generator pressure lower than the design pressure (Zone I). However, to complete the analysis, the system is not independent of the ambient conditions. Table 4 shows the condenser saturation temperature against the system cooling load at various generator pressures. Clearly, 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 ambient temperature must be lower than the condenser saturation temperature to be able to reject heat to the surroundings 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 by the weather conditions (ambient temperature) and not independent of it.

Table 4. Condenser saturation temperature versus the cooling load (Ej-1).

| $p_g$ (kPa) | Saturation temperature (°C ) | $Q_{evap}$ (kW) |
|---|---|---|
| 3244.2 | 77.2 | 5.97 |
| 2633.2 | 56.0 | 13.20 |
| **2116.8** | **43.0** | **15.84** |
| 1700.0 | 40.5 | 16.41 |
| 1317.9 | 36.9 | 17.00 |

From the above discussion, it is clear that a control strategy is required to attain the stable performance of the cooling system for any variation that may occur in the operating conditions. This can be done by adapting the ejector performance according to the operation conditions of the cooling system.

### 5.3 Control Strategy and Ejector Switching

The above results are obtained when ejector 1 (Ej-1) is the working ejector in the cooling system. For control purpose, some criteria should be followed to attain the required cooling capacity at the same evaporator temperature if the operating conditions for any reason are varied. In this regard, the authors suggest that the operating range of Ej-1 is when the required cooling capacity of the system does not exceed 10 % of the optimal design value. Thus;

$$\text{If } (Q_{evap})_{\text{system}} \leq 1.1\,(Q_{evap})_{\text{optimal-design}} \longrightarrow \text{Ej-1 is applied}$$

If such condition is not attained, ejector 2 (Ej-2) is applied, as:

If $(Q_{evap})_{\text{system}} > 1.1\,(Q_{evap})_{\text{optimal-design}} \longrightarrow$ Switch the flow to Ej-2

Similarly, in the case when the required cooling capacity becomes more than 20 % of the optimal design capacity, the flow is switched to ejector 3.

If $(Q_{evap})_{\text{system}} > 1.2\,(Q_{evap})_{\text{optimal-design}} \longrightarrow$ Switch the flow to Ej-3

Following these criteria, multiple ejectors can be added to the system to satisfy the required cooling capacity with any variation in the operating parameters. This control guarantees the stability of the system with any required cooling load.

## 6. **Conclusions**

In this paper, a mathematical model was developed to investigate the ejector-based cooling system performance and hence, enabled setting a control strategy that can satisfy the required cooling load with a stable performance. Two FORTRAN programs were developed to carry out the investigation; one for the ejector design and the other was for the simulation purpose. The generator pressure was considered as the main variable in the analysis. The simulation results have been functioned to propose a control strategy to select the appropriate ejector for a given operating condition, where multiple parallel ejectors can be used by the system. From the simulation results, the following conclusions arise:

1. The operating generator pressure of the cooling system is affected by the weather conditions (ambient temperature) and not independent of it. So, the selection of the operating $p_g$ depends basically on the weather condition.
2. For pressures lower than the optimal design pressure, Ej-1 is working very well, where the cycle performance parameters equal to or lower than the required values by the system design.
3. The range of operation of Ej-1 is when the required cooling capacity of the system does not exceed 10 % of the optimal design value. When $(Q_{evap})_{\text{system}}$ is in the range from 11 - 20 % of the design cooling load, the flow is switched to Ej-2. For $(Q_{evap})_{\text{system}}$ greater than 20% of the optimal design cooling load, Ej-3 will be the operating ejector. More ejectors can be added in the same way to the system, according to the desired cooling load by the system.

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