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Parametric Study of a Vapor Compression Refrigeration Cycle Using a Two-Phase Constant Area Ejector

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***Abstract*—**There are several ways of improving the performance of a vapor compression refrigeration cycle. Use of an ejector as expansion device is one of the alternative ways. The present paper aims at evaluate the performance improvement of a vapor compression refrigeration cycle under a wide range of operating conditions. A numerical model is developed and a parametric study of important parameters such as condensation (30-50°C) and evaporation temperatures (-20-5°C), nozzle and diffuser efficiencies (0.75-0.95), subcooling and superheating degrees (0-15K) are investigated. The model verification gives a good agreement with the literature data. The simulation results revealed that condensation temperature has the highest effect (129%) on the performance improvement ratio while superheating has the lowest one (6.2%). Among ejector efficiencies, the diffuser efficiency has a significant effect on the COP of ejector expansion refrigeration cycle. The COP improvement percentage decreases from 10.9% to 4.6% as subcooling degrees increases by 15K.

***Keywords*—**Numerical modeling, R134a, Two phase ejector, Vapor compression refrigeration system.

I. Introduction

NOWADAYS Egypt suffers from electricity crisis. The demand for electricity in Egypt has been growing at an average rate around 6% annually over the last years, since the early 2000s. The electricity consumption in the field of refrigeration and air conditioning systems is about 23% of the total electricity consumption. In order to reduce this demand a high energy efficiency systems should be adopted. The most commonly used system in refrigeration and air conditioning industry is a vapor compression refrigeration system. There are several ways of enhancing the performance of a vapor compression refrigeration cycle. The use of the heat exchanger for sub-cooling and superheating is a conventional method. In the recent time, several researchers apply inverter and control method to regulate the motor rotation of compressor according to cooling load in the cooled compartment [1]. Due to no moving parts, low cost, simple structure and low maintenance requirements, the use of two- phase ejector has become a promising cycle modification recently. Typical vapor compression refrigeration cycle uses capillary tube, thermostatic expansion valve and other throttling devices to reduce refrigerant pressure from condenser to evaporator. Theoretically, the pressure drop is considered as an isenthalpic process (constant enthalpy). However, isenthalpic process causes a decrease in the evaporator cooling capacity due to energy loss in the throttling process. To recover this energy loss, an ejector can be used to generate isentropic condition in the throttling process and the cycle is called as ejector expansion refrigeration cycle (EERC) [2].

In 1990, Kornhouser [3] was the first researcher who performed a numerical analysis on the EERC to investigate the performance improvement on vapor compression refrigeration cycle (VCRC). Seven refrigerants have been investigated, namely R11, R113, R114, R500, R502, R22 and R717. According to his results, refrigerant R502 was the highest performance improvement and the COP improvement in using R12 was 21% over the conventional cycle. Bilir and Ersoy [4] performed a computational analysis on the performance improvement of ejector expansion refrigeration cycle over conventional VCRC similar to that of Kornhauser [3]. Using refrigerant R134a, the COP improvement of the expansion cycle over standard cycle is 10-22%. Moreover, the COP improvement increases when the condenser temperature increases. This means that the use of ejector instead of an expansion valve is more advantageous in the air-cooled condensers than that of water-cooled condensers. Sarkar [5]-[6] performed the performance improvement of three natural refrigerants namely, ammonia, propane and isobutane. The results revealed that maximum performance improvement using ejector can be achieved in case of isobutane, whereas minimum performance improvement can be achieved for ammonia. Furthermore, the COP improvement over basic expansion cycle increases due to increase in pressure lift ratio with the increase in condenser temperature and decrease in evaporator temperature. A numerical analysis of geometric ejector effects on the performance of the system using twenty synthetic refrigerants is presented by Nehdi et al. [7]. They introduced a geometric area ratio and ratio of mixing chamber to primary nozzle throat area. It was concluded that the maximum COP is obtained when the optimum area ratio is around 10. For optimum area ratio, refrigerant R141b achieved the highest COP. The COP improvement over the conventional cycle is 22%. Fong et al. [8] simulated a solar electric driven ejector vapor compression chiller for space conditioning using R22, R134a and R410A. Coefficient of performance of the chiller was increased for all the three refrigerants as compared with conventional system. However, R134a was the most significant. The literature review revealed

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that many investigators interested with the performance improvement of ejector expansion refrigeration system. However, most of these studies are for the standard saturated cycle. The effect of subcooling or superheating degrees as well as the ejector internal efficiencies is no more discussed.

Therefore, the present paper aims at evaluate the performance improvement of a vapor compression refrigeration system when a two phase ejector is used instead of expansion device over a wide range of operating conditions. Effects of condensation and evaporation temperatures, motive, suction and diffuser efficiencies as well as subcooling and superheating are investigated.

II. System Description and Analysis

Fig. 1 shows a schematic diagram of ejector expansion refrigeration cycle (EERC) and the corresponding p−h diagram. The primary flow from the condenser (state 1) and the secondary flow from the evaporator (state 2) are expanding through primary and secondary nozzles, respectively (1-1b and 2-2b) to mixing chamber pressure, mixing at constant pressure (3m). The mixed flow is discharged through the diffuser (3m-3) of the ejector and then separated in forms of vapor (state 4) and liquid (state 6) so that this ratio matched with the inlet ratio of primary and secondary flows. Then the liquid circulates through the expansion valve (6-7) and then evaporates in the evaporator (7-2), whereas the vapor circulated through the compressor (4-5) and then condensate in the condenser (5-1). In this way, the compressor inlet pressure in this system is relatively higher than that in a basic cycle and hence less work is used to operate the compressor in the EERC. For constant pressure mixing ejector, the primary nozzle exit located within the suction nozzle in front of the constant-area section and the static pressure is assumed to be constant through the mixing process.

The ejector expansion vapor compression refrigeration cycle has been modeled based on the mass, momentum, and energy conservations. To simplify the theoretical model and set up the equations per unit mass flow rate at the ejector exit, the following assumptions have been made.

- Neglect the pressure drop in the condenser, evaporator, separator, and the connection tubes.
- No heat transfer with the environment for the system except in the condenser.
- Both the motive stream and the suction stream reach the same pressure at the inlet of the constant pressure mixing section of the ejector.
- Kinetic energies of the refrigerant at the ejector inlet (nozzle inlets) and outlet (diffuser outlet) are negligible.

A. Motive Nozzle

Fig. 2 shows a schematic diagram of a constant area ejector. By using the definition of motive nozzle's isentropic efficiency the specific enthalpy of the primary fluid at the nozzle exit is given by the following expression:

$$h_{1b} = h_1(1-\eta_n) + \eta_n h_{1b,is} \quad (1)$$

where, h_{1b}, is the corresponding enthalpy of the motive stream at the end of the isentropic expansion process and ηn is the isentropic efficiency of the motive nozzle. Using the energy equation for motive nozzle, the speed at the nozzle exit can be found as:

$$u_{1b} = [2(h_1 - h_{1b})]^{0.5} \quad (2)$$

From the principle of conservation of mass, the area of the motive stream at the inlet of constant area mixing section is given by:

$$a_{1b} = \frac{v_{1b}}{u_{1b}}\left(\frac{1}{1+w}\right) \quad (3)$$

where w is the ratio of the entrained flow rate to the primary flow rate.

B. Suction Nozzle

Similar to the motive nozzle analysis made above, the following equations can be derived;

$$h_{2b} = h_2(1-\eta_s) + \eta_s h_{2b,is} \quad (4)$$
$$u_{2b} = [2(h_2 - h_{2b})]^{0.5} \quad (5)$$
$$a_{2b} = \left(\frac{v_{2b}}{u_{2b}}\right)\left(\frac{w}{1+w}\right) \quad (6)$$

where h_{2b} is the corresponding enthalpy of the suction stream at the end of the isentropic expansion process and η_n is the isentropic efficiency of the suction nozzle.

C. Constant Area Mixing Chamber

Applying the principle of conservation of momentum in a constant mixing chamber, the speed of mixed stream at the exit of mixing chamber is calculated from the following equation:

$$u_{3m} = p_b(a_{1b} + a_{2b}) + \frac{1}{1+w}u_{1b} + \frac{w}{1+w}u_{2b} - p_{3m}a_{3m} \quad (7)$$

Applying the principle of conservation of energy, the enthalpy of the mixed stream at the exit of mixing chamber can be found from:

$$h_{3m} = \frac{1}{1+w}(h_1 + wh_2) - \frac{u_{3m}^2}{2} \quad (8)$$

For unit flow rate of ejector at the exit of a constant area mixing chamber, (9) should be verified [9],

$$\frac{a_{3m}u_{3m}}{v_{3m}} = 1 \quad (9)$$

D. Diffuser

The enthalpy of the stream at the diffuser exit can be found by applying the principle of conservation of energy throughout the ejector,

$$h_3 = \frac{h_1 + wh_2}{1+w} \quad (10)$$

The exit isentropic enthalpy from the diffuser is given as:

$$h_{3s} = \eta_d(h_3 - h_{3m}) + h_{3m} \tag{11}$$

where η_d is the isentropic efficiency of the diffuser. Quality of the fluid at the diffuser exit x_3 is found with the diffuser exit pressure p_3 and enthalpy h_3. On the other hand, to maintain cycle continuity, the quality of the stream leaving the ejector (12) should be approved [10],

$$x_3 = \frac{1}{1+w} \tag{12}$$

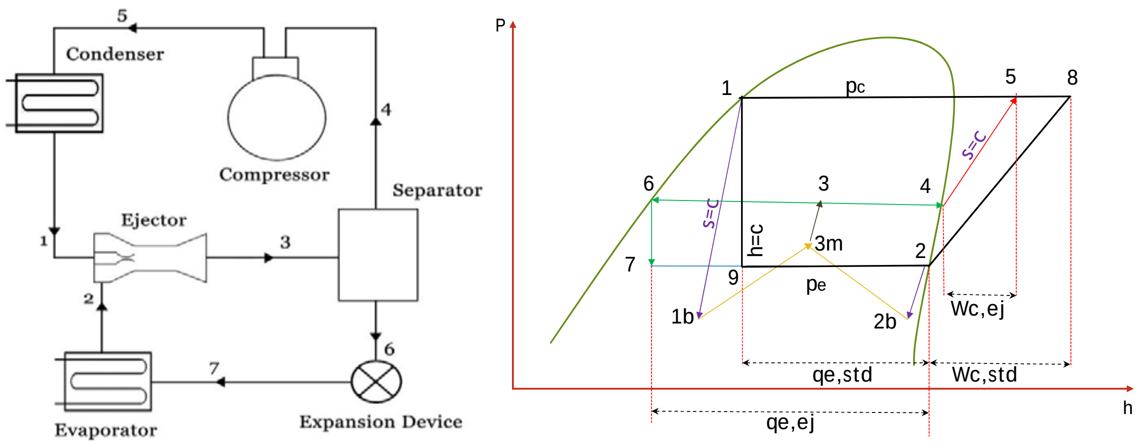

Fig. 1 Schematic diagram of ejector expansion refrigeration cycle and p-h diagram

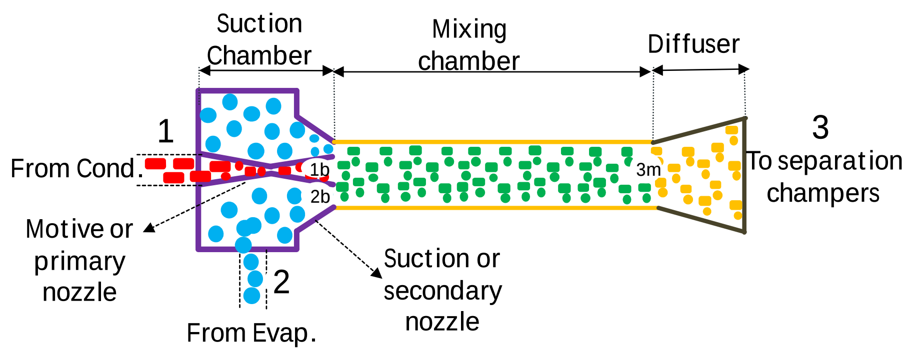

Fig. 2 Schematic diagram of constant area ejector flow model

E. Performance Characteristics of EERC

The performance of EERC is characterized by cooling capacity, compressor power and coefficient of performance. The cooling capacity is calculated as:

$$Q_e = \frac{w}{1+w}(h_2 - h_7) \tag{13}$$

However, the compressor power consumption is given as:

$$P_{com} = \frac{1}{1+w}(h_5 - h_4) \tag{14}$$

Cooling coefficient of performance (COP_{ej}) of EERC is:

$$COP_{ej} = \frac{Q_e}{P_{com}} \tag{15}$$

Performance improvement ratio according to the basic cycle COP_{im} is calculated from:

$$COP_{im} = \frac{COP_{ej} - COP_{std}}{COP_{std}} \tag{16}$$

where COP_{std} is the COP at same evaporator and condenser temperature of a basic refrigeration cycle.

F. Computational Procedure

A computer program is written to solve the above equations using EES software [11]. The model input data are refrigerant type, evaporation temperature, condensation temperature, degrees of superheating and subcooling and motive, suction and diffuser efficiencies and compressor isentropic efficiency. Refrigerant R134a is used as a working fluid. The simulation procedure used in the present investigation is as follows;

1. Thermodynamic properties at state points 1 and 2 are calculated. Specific enthalpies and specific volume at states 1b and 2b are calculated by given both motive and

suction efficiencies. Velocities at the corresponding states are calculated by using (2) and (5).
2. An iterative value of entrainment ratio ($1>w>0$) is assumed and both fluid velocity and specific enthalpy at the exit of constant pressure mixing section (state 3m) are calculated using (7) and (8).
3. Using (10) to (12) and given diffuser efficiency, specific enthalpy, pressure and vapour quality at state 3 are calculated by effective iteration technique and then other properties are also calculated.
4. If the condition of (12) is not satisfied, steps 2–3 will be repeated by using a new value of x_3 until the condition is satisfied.
5. Properties at states 4, 6, and 7 are calculated. Then the properties of state 5 are calculated using compressor isentropic efficiency.
6. Using (13) to (16), the performance parameters Q_e, P_{com} , COP_{std}, COP_{ej} and COP_{im} are calculated. A flow chart of the simulation program is presented in Fig. 3.

III. Results and Discussion

The effects of important parameters such as condensation and evaporation temperatures, nozzle and diffuser efficiencies, subcooling and superheating degrees are shown in Figs. 5-8. It should be noted that when one of them varied, the other parameters remain constant at a practical value. Both ranges and fixed values of the considered parameters are presented in Table I.

A. Model Verification

Model verification is performed on the available literature data for R-134a by Bilir and Ersoy [4]. Predicted improvement percentage of COP using the present model is compared with the theoretical data of Bilir and Ersoy for a wide range of condensation (35-50°C) and evaporation (-25-5°C) temperatures in Fig. 4. It can be stated that, the present model simulation results have a good agreement with the published data. An average error percentage is 2.8% over the entire range of the considered data.

B. Effect of Condensation Temperature

Fig. 5 [A] shows variation of coefficient of performance for both basic vapor compression refrigeration cycle (VCRC) and ejector expansion refrigeration cycle (EERC). As a condensation temperature increases both COPs are decreased but with different rates. Basically, increasing of condensation temperature led compressor power to increase and cooling capacity to decrease therefore COP decrease. However, the rate of decrease in EERC (49%) is slower than in VCRC (52%). Moreover, COP of EERC at higher condensation temperature is better than COP of VCRS which confirm the efficient use of EERC in air cooled condenser as compared with VCRC [4]. As condensation temperature increases from 30 to 40°C, the improvement ratio in COP increases from 5.2% to 12% as shown in Fig. 5 [B]. This trend can be attributed to lower pressure lift of EERC than VCRC.

C. Effect of Evaporation Temperature

A comparison between coefficient of performance of both EERC and VCRC for different evaporation temperature is presented in Fig. 6 [A]. Obviously, COPs of both cycles is direct proportional to evaporation temperature as a result of decreasing cooling effect and increasing compressor specific work as evaporation temperature increases. It should be noted, COP of EERC is higher than standard cycle at any application temperature (air conditioning, refrigeration and freezing). Moreover, the improvement in COP increases as evaporation temperature decreases (See Fig. 6 [B]). As evaporation temperature decreases from 5 to -20, the improvement ratio increases by 84% which proves the high energy efficiency of EERC in freezing applications more than air conditioning application. This high rate of improvement is attributed to the lower pressure ratio of EERC than VCRC.

D.Effect of Motive, Suction and Diffuser Efficiencies

Fig. 7 shows variations of COP of both EERC and VCRC as well as performance improvement ratio and pressure ratio with nozzles and diffuser efficiencies. Nozzles and diffuser efficiencies varied within a practical values (0.75 - 0.95). It should be noted, as nozzles and diffuser efficiencies increase COP of EERC increase with 1.4 to 2%. In fact, nozzles and diffuser efficiencies have a little effect of COP of EERC. Moreover, a constant value of 85% for nozzles and diffuser efficiencies is reasonable for simulation process. It can be observed that the improvement percentage is highly sensitive to diffuser efficiency followed by motive nozzle efficiency and suction nozzle efficiency in that order (see Fig. 7 [B]). As nozzles and diffuser efficiencies increase from 0.65 to 0.85, the improvement ratio in COP increases from 7.1% to 9.3% as an average value. The improvement in COP is attributed mainly to the increase in cooling effect because the pressure ratio is constant as shown in Fig. 7 [B].

E. Effect of Subcooling and Superheating Degrees

Fig. 8 [A] illustrates a comparison between COP of EERC and VCRC for a wide range of both subcooling and superheating degrees. Clearly as subcooling degree increases both COPs are increased but with different rates. In fact, increasing of subcooling degrees led the cooling capacity to increase therefore COP increase. However, the rate of increase in COP for VCRC (14%) is much higher than those of COP of EERC (9%). Variation of improvement percentage in COP and pressure ratio is shown in Fig. 8 [B]. This figure demonstrates that the performance improvement percentage decreases from 10.9% to 4.6 as subcooling degrees increases by 15K. Hence, a comparison of both systems in theoretical work using saturated conditions gives better results while in reality the subcoolİng degrees let this improvement to vanish. It should be mentioned that superheating has a little effect of on both COPs and consequently performance improvement percentage.

IV. Conclusions

In the present work, a numerical model is developed based on mass, momentum and energy conservation equations to investigate the effect of operating condition parameters on the performance improvement of EERC as compared with VCRC. The effect of evaporation and condensation temperatures, nozzle and diffuser efficiencies as well as subcooling and superheating degrees is discussed. Based on the simulation results of the following conclusions are presented:

- Condensation temperature has the highest effect of the performance improvement ratio. As condensation temperature increases from 30 to 50°C, the performance improvement ratio is doubled.
- The use of EERC is most efficient with air cooled condenser and in freezing applications.
- The improvement percentage is highly sensitive to diffuser efficiency followed by motive nozzle efficiency and suction nozzle efficiency in that order.
- The rate of increase in COP of VCRC (14%) is much higher than those of COP of EERC (9%) as subcooling degrees increases with 15K.
- Superheating at the evaporator exit has the lowest effect of the COPs of both cycles and improvement percentages.

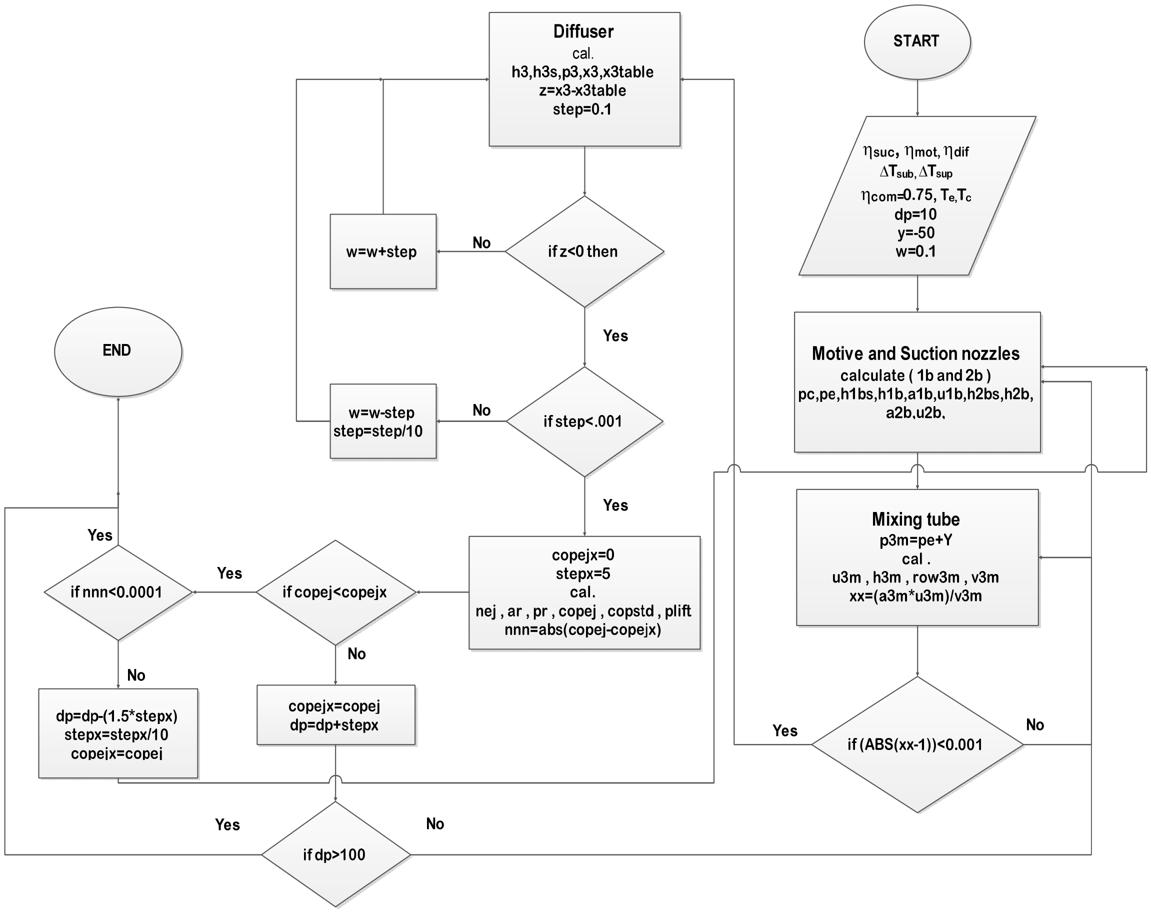

Fig. 3 Flow chart of the simulation program

TABLE I
CONSIDERED PARAMETER RANGES

Parameter	Condensation temperature (°C)	Evaporation temperature (°C)	Motive, suction and diffuser efficiencies	Subcooling and superheating degrees
Range	30-50	-20 - 5	0.75-0.95	0- 15
Fixed value	40	5	0.85	5

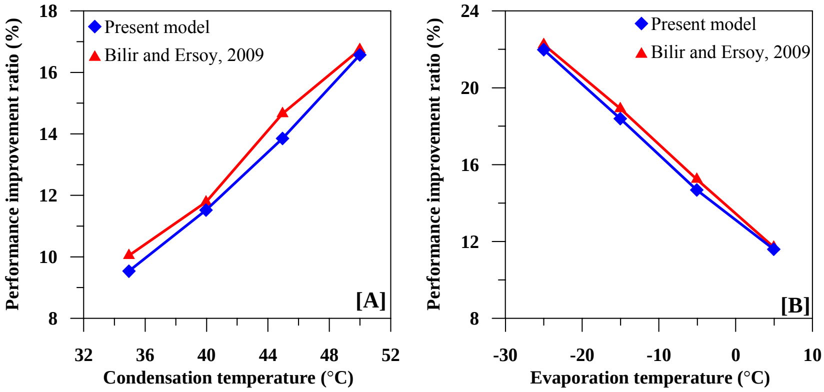

Fig. 4 Comparison between present model and previous model

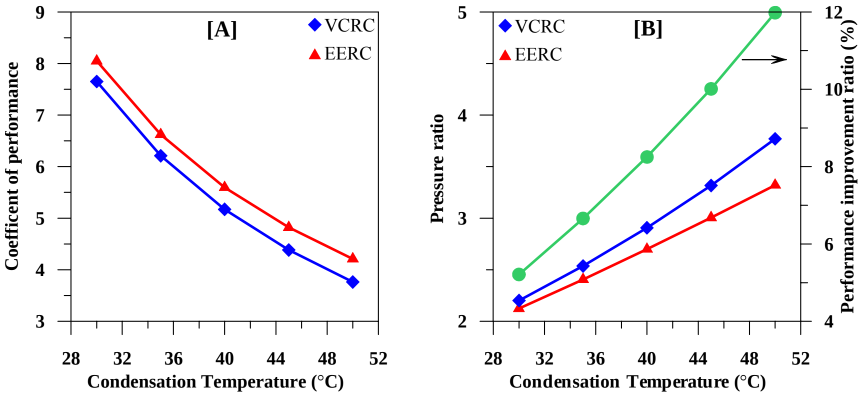

Fig. 5 Variation of coefficient of performance, performance improvement ratio and pressure ratio with condensation temperature

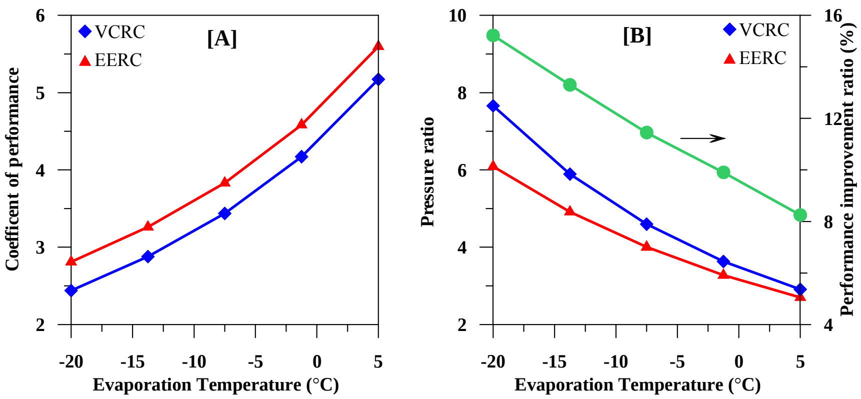

Fig. 6 Variation of coefficient of performance, performance improvement ratio and pressure ratio with evaporation temperature

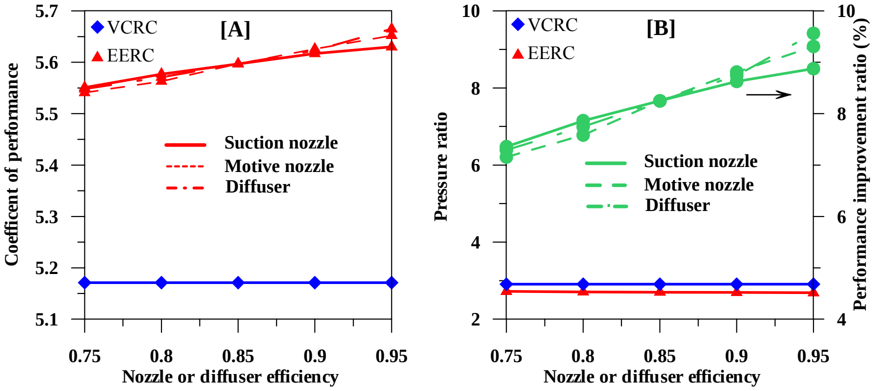

Fig. 7 Effect of nozzle and diffuser efficiencies on coefficient of performance, performance improvement ratio and pressure ratio

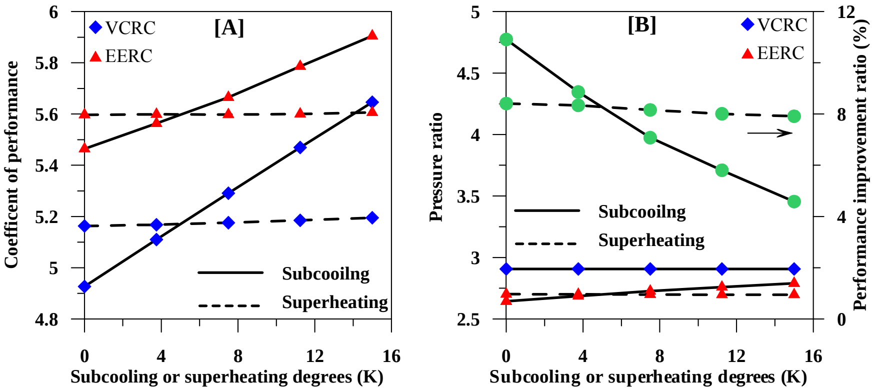

Fig. 8 Effect of subcooling and superheating degrees on coefficient of performance, performance improvement ratio and pressure ratio

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REFERENCES

[1] C. Aprea, R. Mastrullo, C. Renno, "Fuzzy control of the compressor speed in a refrigeration plant," International Journal of Refrigeration 2004, 27(6), pp. 639-648.

[2] J. Sarkar, "Ejector enhanced vapor compression refrigeration and heat pump systems - A review," Renewable and Sustainable Energy Reviews 2012, 16 (9), pp. 6647-6659.

[3] A.A Kornhauser, "The use of an ejector as a refrigerant expander," In: Proceedings of the USN/IIR-Purdue refrigeration conference 1990, West Lafayette, IN, USA, pp. 10–9.

[4] N. Bilir, H.K Ersoy, "Performance improvement of the vapour compression refrigeration cycle by a two-phase constant area ejector," International Journal of Energy Research 2009, 33(5),pp. 469–80.

[5] J. Sarkar, "Performance characteristics of natural refrigerants based ejector expansion refrigeration cycles," Proceedings of the IMechE, Part A: Journal of Power and Energy 2009,223, pp. 543–50.

[6] J. Sarkar, "Geometric parameter optimization of ejector-expansion refrigeration cycle with natural refrigerants," International Journal of Energy Research 2010,34(1), pp. 84–94.

[7] E. Nehdi, L. Kairouani, M. Bouzaina, "Performance analysis of the vapour compression cycle using ejector as an expander," International Journal of Energy Research 2007, 31(4), 364–75.

[8] K.F Fong, C.K Lee, T.T Chow, "Improvement of solar-electric compression refrigeration system through ejector-assisted vapour compression chiller for space conditioning in subtropical climate," Energy Buildings 2011,43, pp. 3383–90.

[9] D. Li, E.A Groll, "Transcritical CO2 refrigeration cycle with ejector-expansion device," International Journal of Refrigeration 2005, 28, pp. 766–773.

[10] J.Q Deng, P.X Jiang, T. Lu, W. Lu, "Particular characteristics of transcritical CO2 refrigeration cycle with an ejector," Applied Thermal Engineering 2007, 27, pp. 381–388.

[11] S.A Klein. Engineering Equation Solver (EES). F-Chart Software 2013, Madison: Wisconsin.