Experimental Study of the Behavior of a Hybrid Ejector-based Air-conditioning System with R134a

Hao Wang, Wenjian Cai, Youyi Wang, Jia Yan, Lei Wang

***Abstract*—A hybrid ejector-based air-conditioning system which combines a vapor compression cycle and a ejector refrigeration cycle was developed. The waste heat energy from automobile is applied as driven source towards ejector refrigeration cycle. Two ejectors with different mixing chamber diameters are applied separately for performance test and the system is operated under different working modes. The effect of generating, condensing and evaporating pressure on system performance are studied experimentally. The effect of ejector geometry parameters on ejector performance is also investigated. The performance comparison between two working modes is made and the results indicate that 1) the performance of proposed system is sensitive to the three pressures; 2) the coefficient of performance (COP) of hybrid ejector-based air-conditioning system is around 34% higher than that of conventional compressor based system which implies a potential energy saving ability of proposed hybrid system.**

***Keywords*—**Ejector, Performance evaluation, Low grade heat energy, Hybrid mode, Coefficient of performance

I. INTRODUCTION

Although ejector has been invented for decades which is earlier than compressor, the progress of ejector based refrigeration system is slow compared with vapor compression system. The reason for this is that the efficiency of vapor compression system is much higher than that of ejector refrigeration system. [1]-[5] However, due to the deteriorating environment caused by massive CO_2 emission and declining global storage of fossil fuel resources, ejector refrigeration system have regained research interests. The distinctive characteristic of ejector refrigeration system is its ability to utilize low grade heat energy which can be easily acquired from automobile waste gas, solar radiation, industrial process and geothermal energy. [6]- [6] Moreover, because of the application of ejector, the system possesses many advantages, such as passive device (no moving components), durable life span, little maintenance cost and high reliable. The main restriction of prevalence of ejector refrigeration system is its relative low efficiency compared to vapor compression system.

To have insightful comprehension on the effect of ejector in the whole system and to improve the efficiency of ejector refrigeration system, many studies have been conducted including performance evaluation towards variety of systems. Pounds [11] investigated the effect of nozzle size, axial nozzle location, high-temperature evaporator temperature and refrigeration temperature by conducting experiment of an ejector refrigeration system. The author found that an optimum nozzle location exists and could produce a maximum coefficient of performance (COP). The results showed that ejector refrigeration system can achieve a COP of 1.7 which demonstrated a very promising future. Yinhai Zhu [11] conducted experiments with an integrated ejector cooling system to analyze the system performance under design conditions. The results indicated that 9.1% COP improvement can be achieved using R22. The author then compared the system performance with the variations of 5 important parameters. The comparison illustrated that the refrigerant with high generator discharge temperature can improve the COP of the ejector cycle effectively. Yan Jia [12][13] studied the effects of ejector area ratio on system performance. An ejector with replaceable nozzle was applied in order to obtain the optimum area ratios under operation conditions. The consequence showed that the optimum area ratios fall in the range from 3.69 to 4.76 which are smaller than other previous studies. Also the operating conditions had great impact on the influence of area ratio. Chen Jianyong [14] investigated experimentally the influence of different temperatures on ejector refrigeration system performance using R123 and R141b. The results indicated that the condenser temperature had more influence than the generator and evaporator temperature on the area ratio and the entrainment ratio in the ejector. Additionally, the author concluded that area ratios need to keep up the pace with the variation of entrainment ratio as operating conditions are changed. Thus, a variable-geometry ejector seemed to be a very promising alternative to guarantee the optimum conditions of ejector

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refrigeration system.

The refrigerant used in the system is also of concerned. F.Wang [15] conducted a theoretical study on the evaluation of environment-friendly refrigerants with similar normal boiling points in ejector refrigeration cycle. Totally eight different kinds of refrigerants were studied comparatively. Entrainment ratios of ejector, system COP, pump power et al. of refrigerants were recorded and analyzed. The result shows that R134a and R152a have similar normal boiling points and give better performance and their COP were stable under different operation status. Besagni Giorgio [16] investigated the performance of heat driven ejector refrigeration system using refrigerant R134a, R600 and R601. The effect of generator, evaporator and condenser temperature over the entrainment ratio and COP had been studied under typical operating conditions of low grade energy sources. From the result obtained, it can be observed that the working fluids had a great impact on the ejector performance and each refrigerant had its own range of operating conditions. The optimum generating temperature for R134a is lower than others and is suitable for generator temperature between 70 and 100℃. Besides, R134a has several advantages over other refrigerants: 0 ozone depletion potential and relative lower global warming impact. Thus, R134a is chosen as the working fluid in our system.

In other studies, Boumaraf [17] presented a novel ejector expansion refrigeration cycle which applied two evaporators using ejector as a throttling valve. The constant area mixing model was applied to perform thermodynamic analysis of the new system. Both R134a and its potential substitute R1234yf was applied for investigation. It was found that the novel system had a significant improvement in COP for both R134a and R1234yf, moreover, the COP improvement of the new system using R134a was higher than that of R1234yf and was able to achieve more than 17% increase at certain condensing temperature.

Nomenclature

D	Diameter (mm)	*Subscripts*	
L	Length (mm)	t	Nozzle throat
h	Specific enthalpy (kJ/kg)	e	Nozzle exit
s	Specific entropy (kJ/kg·K)	m	Mixing chamber
T	Temperature (℃)	f	Diffuser
P	Pressure (bar)	A,B	Ejector A, B
W	Power consumption (kW)	com	Compressor
$\dot{m}$	Mass flow rate (kg/s)	con	Condenser
n	coefficient	eva	Evaporator
Q	Heat load (kW)	SF	Secondary flow
c	coefficient	WF	Working flow
η	Ideal isentropic efficiency	sat	Saturation condition
A	Area (m^2)	s	Secondary flow
γ	Specific heat ratio	m	Mixing flow
ψ	Isentropic efficiency	isen	isentropic
R	Ideal gas constant	p	Primary flow
V	Velocity (m/s)		
H	Heat transfer coefficient	*Superscripts*	
v	Kinematic viscosity	*e*,n	Coefficient
μ	Absolute viscosity		
k	Thermal conductivity		
b	Coefficient		
ρ	Density (kg/m^3)		

Chen Lin [18] proposed an experimental investigation of the adjustable ejector in a multi-evaporator refrigeration system. Two parameters namely pressure recovery ratio (PRR) and relative pressure recovery ratio (RPRR) are measured to evaluate the system performance. The results show that the proposed system can be a promising approach to save energy significantly. Moreover, the decreasing of the primary cooling load will lead to the reduction of both PRR and RPRR. Meibo Xing [19] explored the extent of COP improvement of a novel vapor-compression refrigeration cycle with mechanical sub-cooling using an ejector. The author developed a mathematical model to predict the performance of the cycle by using R404A and R290. The results suggest that the ejector sub-cooled cycle is better than that of the conventional cycle. The novel cycle would achieve volumetric refrigeration capacity improvement of 11.7% with R404A and 7.2% with R290. Furthermore, the COP improvement of novel cycle was 9.5% with R404A and 7.0% with R290. In addition, the improvement of COP and cooling capacity majorly depended on the operation conditions of the ejector. Huang [20][21] which was capable of making use of waste energy from vapor refrigerant under superheated state. From the result they obtain, 15.4% COP improvement can be realized averagely while the maximum can reach to 24.5%. Kursad Ersoy [20] applied an ejector for expansion purpose instead of an expansion valve for expansion work recovery in

a vapor compression cycle. The author experimentally investigated the COP values for both conventional and ejector systems under same external conditions. The data obtained demonstrated that the system with ejector as the expander had better performance than the conventional one and the COP improvement was 6.2%-14.5% within an error of approximately 10%. Currently, no literature has been found on attempting to connect ejector and evaporator with compressor directly.

To fill this blank, we have proposed a hybrid system in which the evaporator outlet is connected directly to ejector secondary inlet via a variable speed compressor. A series of experiment tests are conducted in order to 1) investigate the effect of primary pressure, evaporating pressure and back pressure on COP and entrainment ratio; 2) to investigate the ejector mixing chamber effect on COP by using ejectors with different geometry parameters and 3) to compare the performance of both general mode and hybrid mode. The results show that the performance of the proposed system is very sensitive to three pressures of cycle and relatively promising COP improvements are achievable which imply a huge potential in the future.

II. Experiment Description

As shown in Figure 1, the proposed system is consisted of two sub-cycles: a vapor compression cycle and an ejector refrigeration cycle. The system can be operated under two different working modes, namely general (or compressor only) mode and hybrid mode (combination of compressor and ejector), by switching control valves installed in the refrigeration loop. Under the hybrid mode, compressor is in series connected with the ejector secondary inlet which is the evaporator outlet.

The rational for such arrangement is that the suction effect generated by ejector helps reducing the compression ratio of the compressor which in turn reduces the power consumption of compressor. In addition, the diffuser section of ejector will decelerate the discharge refrigerant and increases its pressure to the condensing pressure which helps improve the efficiency of condenser. Thus the overall COP of system can be improved.

A photograph of experimental platform is shown as in Fig.1, the whole system is consisted of following components:

- A generator;
- An electric evaporator;
- An air-cooled condenser;
- An ejector;
- A liquid receiver;
- A liquid pump;
- An electronic expansion valve (EEV);
- A compressor;
- A monitoring and control system using LabVIEW.

The schematic diagram of the system is shown in Fig.2.

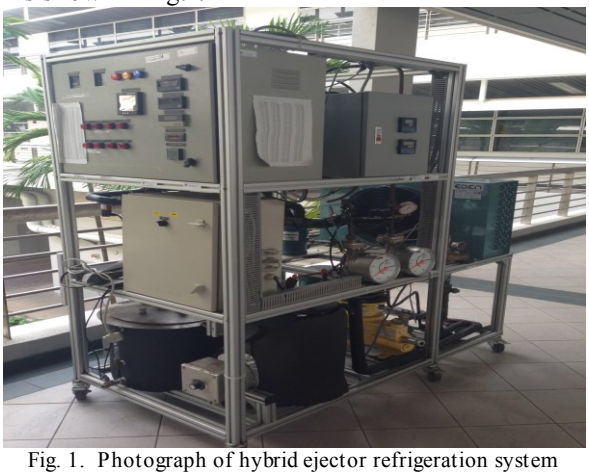

Fig. 1. Photograph of hybrid ejector refrigeration system

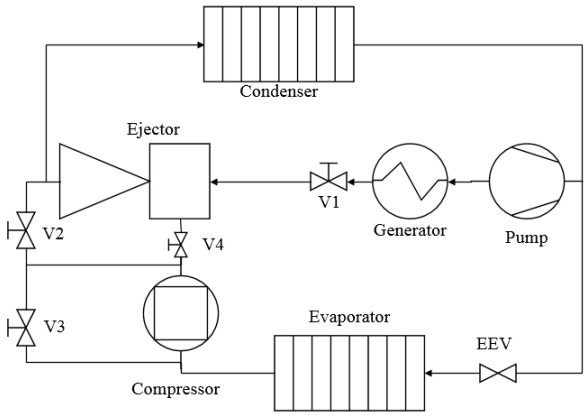

Fig. 2. Schematic diagram of hybrid ejector refrigeration system

TABLE I. OPERATION INFORMATION FOR DIFFERENT MODES

Different Modes	Operation details
General Mode	V1, V3, V4 off, V2 on
Ejector Mode	V2 off, V1, V3, V4 on
Hybrid Mode	V2, V3 off, V1, V4 on

The *p-h* chart of hybrid mode operation is shown in Fig.3.

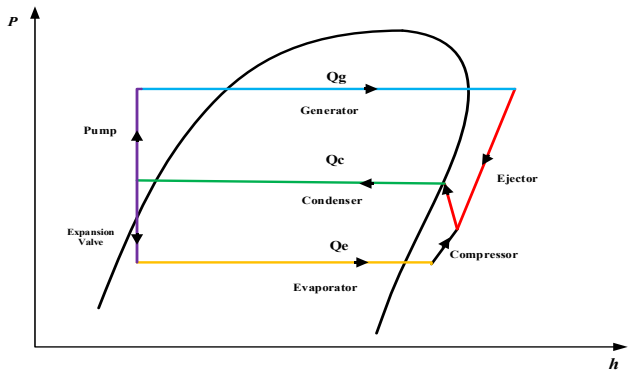

Fig. 3. *p-h* chart of hybrid mode operation

The generator consists of a vapor collection tank, a tubular heater immersed in the tank and a temperature-controlled conduction oil bath. The oil bath is heated by an electric heater with maximum power output of 8 kW. The temperature of the hot oil is maintained at 120℃ with deviation of ±0.5℃ by a PID controller. The electric evaporator with maximum power output of 3 kW is installed to accurately measure the system cooling capacity. The air-cooled condenser has rated condensing load of 15.6 kW and the vapor refrigerant is from the diffuser of ejector. The liquid pump is located at the bottom of the platform in case of cavitation. The speed of the pump is adjusted via a control device. The EEV is impelled by a step motor which is controlled by a PID controller. The inverter compressor is applied after the evaporator in order to raise the secondary flow pressure to designed value. To avoid heat losses and gains, all tubes are covered with thick glass fiber wool that has an aluminum foil layer on its outer surface. The monitoring and control panel are built based on LabVIEW.

In addition, eight hand valves are installed which allow the system being operated under different modes; two flow meters with different measuring range are placed at certain position to accurately measure the primary and secondary mass flow rate; several pressure and temperature sensors installed in the system are showed in following table. Since the range of generating pressure is 10~23bar, the pressure sensor with range from 0~40bar is applied at ejector primary side. These sensors are widely used among industrial fields for their high accuracy and steady performance. Therefore, no further calibration is needed.

TABLE II. DETAILS OF MEASUREMENT DEVICES

Different Sensors	Instrument	Unit of measurement	Accuracy	Range
Temperature	PT1000 platinum resistance sensors	℃	±0.05	-50~150
Pressure	Huba OEM piezoelectric pressure transmitters	Bar	±0.01	0~15
Pressure	Huba OEM piezoelectric pressure transmitters	Bar	±0.02	0~40
Flow rate	Yokogawa RAMC rotor flow meters	L/h	±0.32	0~20
Flow rate	Yokogawa RAMC rotor flow meters	L/h	±1.92	0~120

The experimental plant was usually brought to steady state condition within 40mins after the inputs were changed. Data collection process was begun as soon as the system stabilized. The refrigerant enthalpies are determined from measured temperature and pressure values according to NIST database. The total expanded uncertainties for the airside heat transfer rate measurements are ±1.5%. The total expanded uncertainties for the refrigerant side heat transfer rate measurements are ±3.5% for the condenser and ±5% for the generator, with the biggest contributor being the uncertainties in the thermodynamic property estimation calculations [23].

Fig.4 and Fig.5 show the dimensions of two ejectors used in the experiments. The detail parameters are listed as follow:

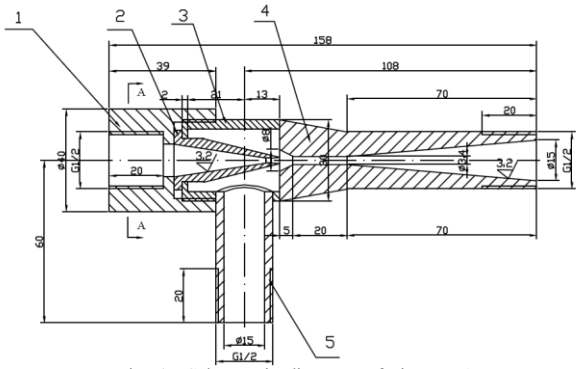

Fig. 4. Schematic diagram of ejector A

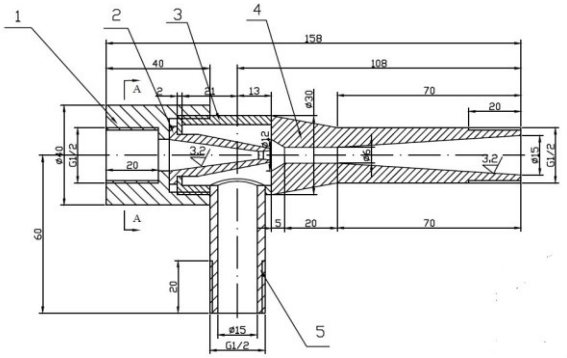

Fig. 5. Schematic diagram of ejector B

- Diameter at nozzle throat D_{tA}=1.6mm, D_{tB}=3mm.
- Diameter at nozzle exit D_{eA}=2.3mm, D_{eB}=3.8mm.
- Mixing chamber diameter D_{mA}=3.4mm, D_{mB}=6mm.
- Diffuser exit diameter D_{fA}=D_{fB}=15mm.
- The distance between nozzle exit and the mixing chamber inlet NXP_A=NXP_B=5mm.
- Mixing chamber length L_{mA}=L_{mB}=20mm.
- Diffuser length L_{fA}=L_{fB}=70mm.
- Nozzle divergent section length L_{nA}=4mm, L_{nB}=5mm.

The ejector consists of four parts: *The nozzle, the suction chamber, the mixing chamber and the diffuser*.

The working principle of the ejector is briefly described below:

The primary flow (sometimes referred as motive flow) enters the ejector with a subsonic velocity. It then goes through a converging-diverging section. During the converging part, it is accelerated to high velocity and the pressure is reduced based on Bernoulli's principle. The velocity reaches sonic with Mach number equals to 1 at the nozzle throat and the flow chokes. When it continues passing through the diverging part, the velocity increases again and reaches supersonic. A very low pressure region (sometimes referred as vacuum region) is generated in the suction chamber due to this high speed primary flow. Therefore, the secondary flow from the evaporator is entrained by this suction force. The two flows then mix, compress and will generate several

shocks in the mixing chamber [24]. Generally speaking, there are two kinds of mixing chamber, a constant pressure type and a constant area type. Each one has its own character, but the constant area type is more famous because of its simplicity of manufacturing. The shock causes a sudden reduction in velocity and thereby an abrupt increase in pressure. As the mixing flow emerges from mixing chamber, it goes through another pressure recovery process in the diffuser. The kinetic energy is converted into potential energy. Finally, the pressure of mixture reaches the corresponding condensing pressure.

The design operating conditions of the system are: condensing pressure 9bar, evaporating pressure 3.5bar and generating pressure 22bar. The design cooling capacity is 1.5kW. The experiment conditions are listed in table III. Particularly, the tests were carried out from 9th Aug 2013 to 2nd May 2014 on every weekday 2pm in Singapore and the test duration was 3 hours per day.

TABLE III. OPERATING CONDITIONS OF THE SYSTEM

Operating Condition	Value
Ambient temperature (℃)	33~34
Condensing pressure (bar)	8~12
Evaporating pressure (bar)	2.5~4.5
Generating pressure (bar)	10~25
Compressor frequency (Hz)	30~50
EEV opening ratio (ω)	30%~60%

III. SYSTEM PERFORMANCE ANALYSIS

The proposed hybrid system is a combination of a conventional vapor compression cycle and an ejector cycle. The governing equations are based on conservation of energy and mass for each component in two sub-cycles.

A. Compressor

Since the compression process happens rapidly, it is often assumed to be adiabatic. Based on NIST database, the enthalpy and entropy of the refrigerant at compressor inlet are determined by the compressor inlet temperature and pressure as[26]:

$$h_{com,inlet}, s_{com,inlet} = f(T_{com,inlet}, P_{com,inlet}) \quad (1)$$

The enthalpy of refrigerant at compressor outlet for the isentropic process is:

$$h_{com,outlet,isen} = f(s_{com,outlet,isen}, P_{com,outlet}) \quad (2)$$

where $s_{com,outlet,isen} = s_{com,inlet}$

The actual enthalpy at compressor outlet is:

$$h_{com,outlet} = h_{com,inlet} + (h_{com,outlet,isen} - h_{com,inlet}) \Big/ \eta_c \quad (3)$$

where η_c is the isentropic efficiency of compression process.

The power consumption of compressor can be calculated as:

$$W_{com} = \dot{m} \quad , - h_{com,inlet}) \quad (4)$$

The heat transfer process mainly happens within two components: condenser and evaporator. Since the operation principle of condenser and evaporator are similar, only condenser is selected as an illustration. A condensing process majorly consists of two loops: the refrigerant loop and secondary fluid loop. In the refrigerant loop, the heat is transferred from refrigerant fluid to secondary fluid through metal tube wall due to temperature difference between the two fluids in the manner of conduction. The amount of heat transfer can be obtained by multiplying the refrigerant mass flow rate with enthalpy difference between inlet and outlet. In the secondary fluid loop, the heat transfer is a single-phase forced convection process. Based on Newton's cooling law, the convective heat transfer can be expressed as:

$$Q = HA\Delta T \tag{5}$$

where H is the average convective heat transfer coefficient, A is the condenser surface are and ΔT is the average temperature difference between metal tube wall and the secondary fluid, respectively.

The convective heat transfer H is usually determined empirically by a set of dimensionless fluid properties. For instance, the heat transfer coefficient of forced convection process can be expressed as:

$$\frac{Hd}{k} = C(\frac{Vd}{v})^e(\frac{\mu c_p}{k})^n \tag{6}$$

where d is characteristic length, v is the kinematic viscosity, μ is the absolute viscosity, c_p is the specific heat at constant pressure, C,e and n are coefficients.

Since mass flow rate can be written as:

$$\dot{m} \tag{7}$$

Substituting equation (7) into (6), after rearrangement and combination, we can obtain:

$$H = b\dot{m} \tag{8}$$

where under steady state condition, the viscosity, thermal conductivity and specific heat are approximately constants. According to the principle of energy balance, the heat transfer of refrigerant side is equal to that of secondary fluid side. Combining the heat transfer of two loops and applying the Levenberg-Marquardt method, the H can be determined.

B. Condenser

The condenser applied is air-cooled type. The enthalpy of refrigerant at condenser inlet is determined as:

$$h_{con,inlet} = f(T_{con,inlet}, P_{con,inlet}) \tag{9}$$

Since the condensing process is assumed to be isobaric, the pressure remains constant, also the refrigerant is assumed to be saturated liquid at condenser exit:

$$h_{con,outlet} = f_{sat}(P_{con,outlet}) \tag{10}$$

where $P_{con,outlet} = P_{con,inlet}$

The heat load for condenser is then:

$$Q_{con} = \dot{m} \qquad -h_{con,outlet}) \tag{11}$$

C. Evaporator

According to the hybrid evaporator model, the cooling capacity can be determined as [26]:

$$Q_{eva} = \frac{\dot{m}}{1 + c_2 \frac{\dot{m}}{\dot{m}}} \cdot \tag{12}$$

where c_1,c_2 and e are constant coefficients to be determined, h_{vapor} and $h_{\text{eva,inlet}}$ are the specific saturated vapor enthalpy and refrigerant enthalpy at evaporator inlet, respectively. $T_{\text{SF,inlet}}$ and T_{sat} are secondary flow inlet temperature and saturated temperature of refrigerant, $\dot{m}_{\text{s}}$ and $\dot{m}_{\text{SF}}$ are mass flow rate of refrigerant and secondary flow.

The temperature can be obtained by the sensors installed. The specific enthalpy of refrigerant at evaporator inlet can be obtained through:

$$h_{eva,inlet} = f(T_{eva,inlet}, P_{eva,inlet}) \quad (13)$$

D. Pump

The pump is not an isentropic process. The actual refrigerant enthalpy at the pump outlet can be obtained as:

$$h_{pump,outlet} = h_{pump,inlet} + (h_{pump,outlet,isen} - h_{pump,inlet}) \Big/ \eta_{pump} \quad (14)$$

where η_{pump} is the ideal isentropic efficiency of pump process. The thermodynamic conditions of refrigerant at pump inlet are the same as that at condenser outlet.

The enthalpy of the refrigerant at pump outlet for an isentropic process is:

$$h_{pump,outlet,isen} = f(s_{pump,outlet,isen}, P_{pump,outlet}) \quad (15)$$

where $s_{pump,outlet,isen}=s_{con,outlet}$

The power consumption of the liquid pump can be calculated as:

$$W_{pump} = \dot{m}(\quad let - h_{pump,inlet}) \quad (16)$$

E. Ejector

The performance of ejector is influenced by both its geometric parameters and operation conditions. The entrainment ratio which is defined as the ratio of secondary mass flow rate to primary flow rate is usually applied to evaluate the ejector performance. The primary flow is often assumed to be choked in the primary nozzle, and under that condition as well as ideal gas assumption, the mass flow rate of primary flow can be given by [27]:

$$\dot{m} \quad \frac{\gamma\psi_p}{R_g T_p})^{\frac{1}{2}} (\frac{1+\gamma}{2})^{\frac{1+\gamma}{2-2\gamma}} \quad (17)$$

where P_p and T_p are the stagnation pressure and temperature of the primary flow at the nozzle inlet, A_t is the area of nozzle throat, R_g is the ideal gas coefficient, γ is the specific heat ratio of vapor refrigerant and ψ_p is the isentropic coefficient of compressible flow, respectively.

The mass flow rate of secondary flow is more complicated to determine than primary flow. Based on shock circle model, it can be expressed as [28]:

$$\dot{m} \quad \frac{pm,inlet}{R_g T_{sm,inlet}} [\frac{nD_m^2}{4n+4}\left(1-\frac{D_{pm,inlet}}{D_m}\right)^{1+\frac{1}{n}} - \frac{nD_m^2}{8n+4}\left(1-\frac{D_{pm,inlet}}{D_m}\right)^{2+\frac{1}{n}}] \quad (18)$$

where P_s is the stagnation pressure of the secondary flow at ejector inlet, $V_{pm,inlet}$ and $T_{sm,inlet}$ are the primary flow velocity and secondary flow temperature at mixing chamber inlet, D_m and $D_{pm,inlet}$ are the diameter of mixing chamber and diameter of primary flow choke circle at mixing chamber inlet, respectively.

Hence, the entrainment ratio can be obtained as:

$$\omega = \frac{\dot{m}}{\dot{m}} \quad (19)$$

The performance of the refrigeration system can be evaluated based on the coefficient of performance (also short as COP) which is defined as the cooling capacity generated to the power consumption the system need. The coefficient of performance of the proposed hybrid system is:

$$COP_{hybrid} = \frac{Q_e}{W_{pump} + W_{com} + W_{gen}} \quad (20)$$

The power consumption of generator is obtained by power meter installed.

And for the coefficient of performance of the conventional vapor compression system, it is expressed as:

$$COP_{general} = \frac{Q_e}{W_{com}} \quad (21)$$

IV. EXPERIMENT RESULTS AND DISCUSSION

The major objective of this experiment is to investigate the effect of condenser, evaporator and generator pressure on the system performance with ejectors of different mixing chamber diameter. The controlling variables method is applied that means when investigating one particular parameter, the others remain constant. For instance, when investigating the back pressure effect on entrainment ratio, both generating pressure and evaporating pressure are kept constant. Also, the comparison between conventional vapor compression cycle and proposed hybrid cycle is of our interest. The operating conditions for two cycles are the same, only the structure difference is investigated. In order to mitigate the influence of random error, each experimental point is the average value of 50 experiment data. Moreover, based on accuracy of experimental apparatus and principle of error transfer, the relative error for entrainment ratio is 6.7%, for cooling capacity is 10.1% and for COP is 13.3%.

A. Effect of back pressure on entrainment ratio

The condensing pressure (back pressure) is closely related to entrainment ratio and is affected by ambient condition which is impossible to adjust compared with evaporating and generating pressures.

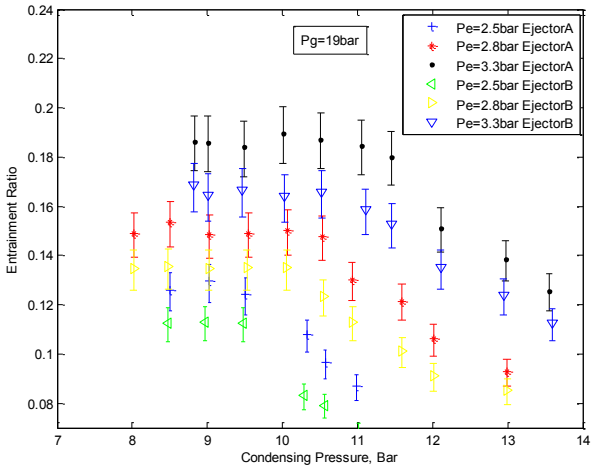

Fig. 6. Variation of entrainment ratio with condensing pressure under constant generating pressure

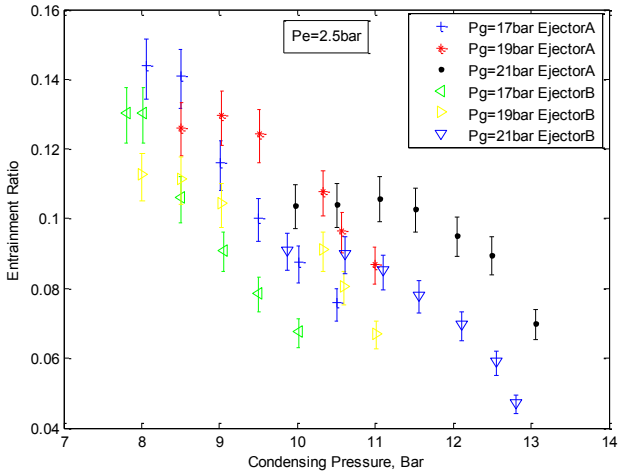

Fig. 7. Variation of entrainment ratio with condensing pressure under constant evaporating pressure

The experiments are conducted under different conditions so that the relationship between critical back pressure and evaporating

(or generating) pressure can be obtained. In order to more intuitively reveal the variation trend of entrainment ratio under different operation conditions, experimental points are connected with lines. Fig.6 shows Variation of entrainment ratio with condensing pressure under constant generating pressure, it can be seen that 1) under constant evaporating (P_e) and generating pressure (P_g), entrainment ratio remains constant before condensing pressure increases to a certain value, 2) as condensing pressure increases, the entrainment ratio drop sharply. The reason for this critical condensing pressure is that when condensing pressure (P_c) is lower than critical pressure, the mixed flow is choked within mixing chamber. Therefore, the mass flow rate of secondary flow is independent from condensing pressure. Only the increase of evaporating pressure can raise the secondary flow rate. That is why there is a horizontal part at left side of the line. When condensing pressure is higher than critical pressure, the mixed flow is no longer choked and the secondary mass flow varies with condensing pressure. The higher the condensing pressure is, the more resistance there will be at the outlet of ejector. The pressure difference between ejector inlet and outlet that generates motive force is reduced. So the entrainment ratio decreases sharply as a result. Moreover, continually increasing condensing pressure will cause reverse flow circumstance in ejector which should be highly prohibited. Also from Fig.6, it can be found that when evaporating pressure increases, both entrainment ratio and critical pressure value increase. Since the increment of evaporator pressure leads to the increase of pressure difference between secondary inlet and vacuum region, more secondary flow will be sucked into ejector due to this larger pressure difference. Thus the entrainment ratio increases with increase of evaporator pressure. Besides, the evaporator pressure augment results in larger momentum that is capable of dealing with higher condensing pressure. Hence, when evaporator pressure increases, the critical pressure value will increase that offers a wider and safer operation range for condenser. So the increases of evaporating pressure will result in a better performance of ejector with sacrifice of evaporator operation range.

The variation of entrainment ratio with the condensing pressure under constant evaporating pressure is illustrated in Fig. 7. It is observed that 1) under constant evaporating pressure, increasing generating pressure causes entrainment ratio to decrease and 2) critical pressure value will increase with increment of generating pressure. The reason for this phenomenon is obvious that when generating pressure increases, the primary mass flow rate will increase. Since the diameter of mixing chamber is fixed and the mixed flow is choked within mixing chamber when condensing pressure lower than critical value, an increment of primary flow rate will cause secondary flow rate to decrease which leads to a reduction of entrainment ratio. The numerator which represents secondary flow rate decreases and the denominator which represents primary flow increases, together the entrainment ratio decreases. As secondary flow goes through evaporator and related to cooling capacity, this also leads to a decrease of cooling capacity and COP. However, a higher generating pressure will causes the momentum of the mixed flow to increase. Thus, the critical condenser pressure is increased. Higher generating pressure requires more power from liquid pump or higher temperature produced by generator heater. In terms of system efficiency, it is beneficial to have the system operated under lower generating pressure.

Both Fig.6 and Fig.7 indicate that the entrainment ratio of ejector A is higher than that of ejector B under same operating conditions. The reason for this performance is that the mixing chamber diameter of ejector B is 6mm which is 76.4% larger than 3.4mm of ejector A while the nozzle throat diameter of ejector B is 3mm which is 87.5% bigger than 1.6mm of ejector A. So under the same operating conditions, the ejector B ought to be able to entrain more secondary flow due to larger nozzle throat, however, since the mixing chamber diameter of ejector B is only 76.4% increase from A, this will limit the mass flow rate of secondary flow. As a result, both primary flow and secondary flow will increase when applying ejector B, the increment amount of primary flow is higher than that of secondary flow which leads to decrease of entrainment ratio.

For better performance, the ejector with higher entrainment ratio is desired. Thus, ejector A is chosen to conduct the following experiments.

B. Effect of primary and secondary pressure on entrainment ratio

The evaporating pressure on system performance is tested for both entrainment ratio and COP. Fig.8 depicts how the entrainment ratio varies with evaporating pressure.

From the figure, it can be concluded that for the constant condensing and generating pressures, the entrainment ratio increases along with the evaporating pressure. With the increase of evaporating pressure, the pressure difference between ejector secondary inlet and vacuum region increases, thus a small amount of primary motive flow is sufficient to generate enough suction force which entrains the secondary flow into ejector. Therefore, at higher evaporating pressure value, the entrainment ratio is larger. On the other hand, an increase of generating pressure will cause the entrainment ratio to decrease. Similar trend can be observed from variation of COP with evaporating pressure which will be shown later.

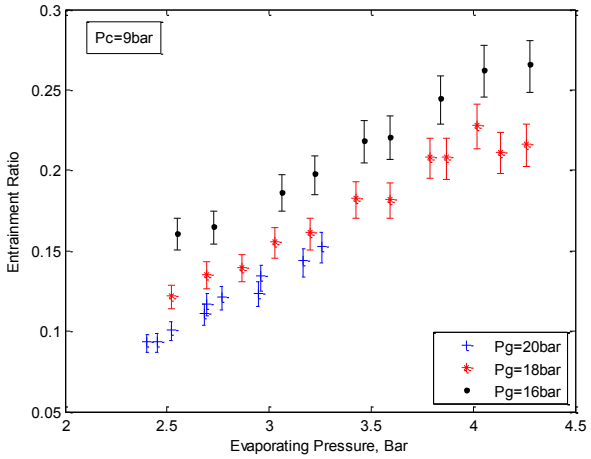

Fig. 8. Variation of entrainment ratio with evaporating pressure under constant condensing pressure

In order to investigate the relationship between system performance and generating pressure, the experiment is conducted under following conditions: the P_c is constant at 9 bar, the P_e is constant at 3 and 4 bar while the P_g is varied from 10~23 bar and the system is operating under hybrid mode.

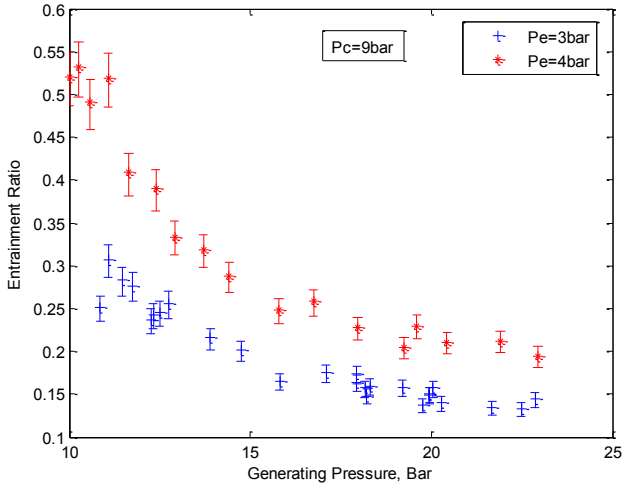

Fig. 9. Variation of entrainment ratio with generating pressure under constant condensing pressure

The variation of entrainment ratio with generating pressure under constant condensing pressure is illustrated in Fig. 9. It can be seen that the entrainment ratio under hybrid mode is decreasing along with the increasing of generating pressure. Since under hybrid mode, compressor helps compensate suction effect even when generating pressure is not high enough. Although the entrainment ratio seems to be quite satisfying, it is due to the small primary mass flow rate, the actual cooling capacity is not at its optimum. As generating pressure keeps increasing, both the primary mass flow rate and the suction effect generated by ejector are increasing proportionally; this will help mitigate the compressor loads which in turn reduce the power consumption of compressor.

C. Effect of different pressure on COP

Since COP is an important parameter to evaluate the system performance and also in order to improve the efficiency of the system, the influence of evaporating and generating pressure on COP is studied. The condensing pressure is not studied because its adjustment is not practical as previous two pressures in real application.

From the experiment data, it can be seen that COP increases with the increment of evaporating pressure under all three generating pressures. From previous results, it is known that increase of evaporator pressure leads to increase of entrainment ratio and finally results in cooling capacity augment. Also, under constant generating and condensing pressure, a higher evaporating pressure is able to be entrained by a lower generating pressure which means higher evaporating pressure requires lesser power consumption of liquid pump or generator heater. This will together help improve the COP of the system. In order to achieve higher COP value, the system should be run at higher evaporating pressure condition. However, in real application, a lower evaporating temperature, which also means lower evaporating pressure, is desired. So it has to be a balance between achieving higher COP and desired evaporating temperature.

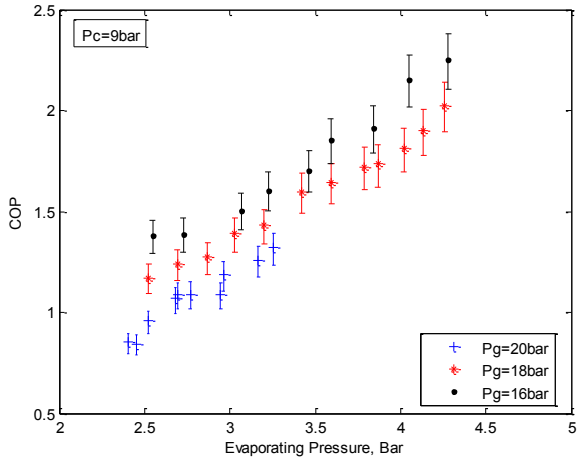

Fig. 10. Variation of COP with evaporating pressure under constant condensing pressure

Additionally, the variation of COP with generating pressure is investigated. Under hybrid mode, for given evaporating and condensing pressure, COP is increasing with the increment of generating pressure. Although the entrainment ratio is decreasing with the increment of generating pressure, it does not mean that COP will follow the same pattern. At higher generating pressure, the mass flow rate of primary flow is higher; the suction force generated by primary flow is large enough to achieve sufficient entrain effect under relative lower secondary pressure condition. Thus the compress ratio decreases which leads to a reduction in compressor power consumption. If more exhaust heat is collected and recovered, the generating pressure is higher which results in a higher COP.

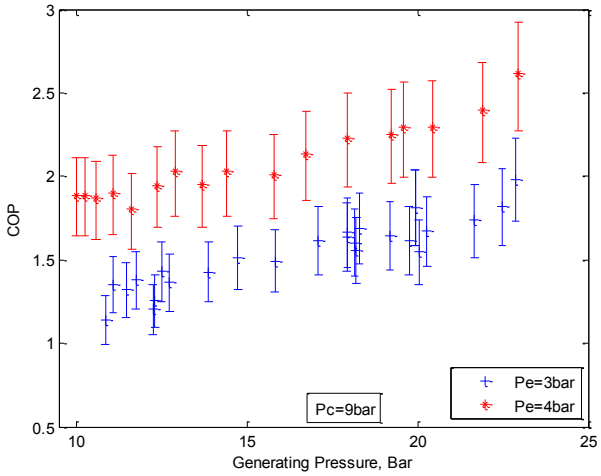

Fig. 11. Variation of COP with generating pressure under constant condensing pressure

D. Performance comparison between two modes

The comparison of cooling capacity, power consumption and COP between general mode and hybrid mode are illustrated in Fig.12, 13, 14, respectively. From the three figures, it can be seen that the application of ejector in hybrid refrigeration cycle (hybrid mode) can indeed improve the system performance over conventional vapor compression cycle (general mode). From equation (16) and (17), it can be observed that there are two more power consumed components (liquid pump and generator heater) within hybrid mode. However, the power consumption of hybrid mode is lower than that of general mode. The reason is that the employment of passive device like ejector which utilizes the waste heat energy to generator suction force shares part of the responsibility of the compressor, particularly, the operation principle of mixing chamber in some extent are similar to a compressor which reduces the energy consumption of mechanical compressor in return. This reduced amount of energy is larger than the power consumption which liquid pump and heater bring along. Thus, the total power consumption of hybrid mode is lower than that of general mode. Additional, the use of ejector enhances the entrain effect of secondary flow that related to cooling capacity. Therefore, the cooling capacity of hybrid mode is higher than that of general mode. All these factors together contribute to the COP improvement of hybrid refrigeration system. Given a scrutiny towards the result, it can be found that the cooling capacity is averagely 27.3% higher under hybrid mode than that of general mode; the power consumption is averagely 20.3% lesser under hybrid mode. As a consequence, the COP improvement of hybrid mode is about 34% which meets the rough estimation; however, there still remains a distance from setting goal. Since ejector is capable of utilizing waste source energy and this kind of energy is

free in a sense. If the generator heater of proposed system is powered by low grade heat energy, such as automobile exhaust, geothermal or solar energy, the power consumption of system will be further reduced. Thus 34% improvement is not the ultimate goal. The improvement can also be realized from an appropriate control strategy and optimization algorithm which will be studied in the future.

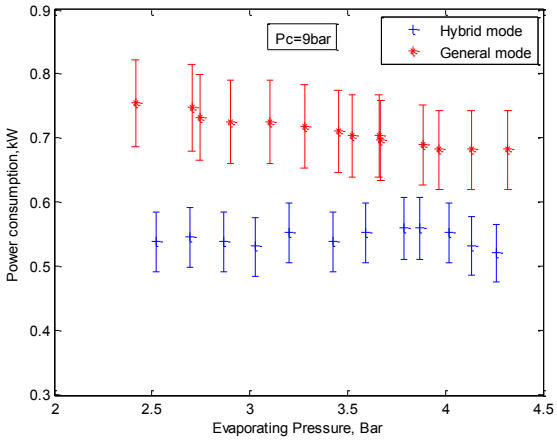

Fig. 12. Power consumption between hybrid mode and general mode

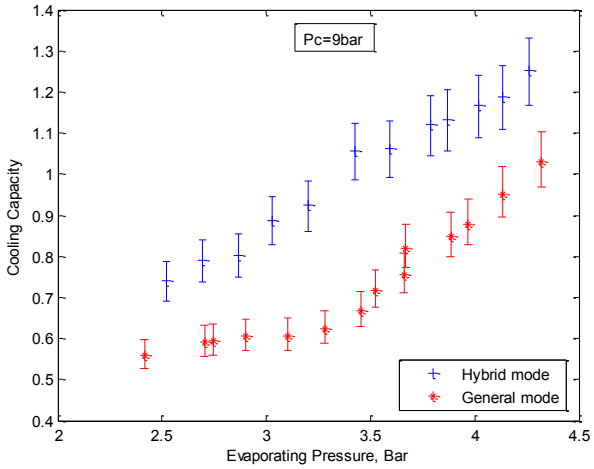

Fig. 13. Cooling capacity between hybrid mode and general mode

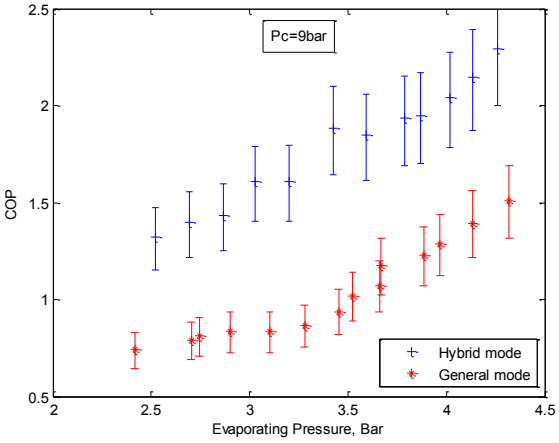

Fig. 14. COP between hybrid mode and general mode

V. CONCLUSIONS

In this work, the effect of geometrical parameter on ejector performance and the effect of condenser, evaporator and generator pressure on the hybrid system performance was investigated experimentally. The performance comparison between general mode

and hybrid mode was presented. Two performance indexes namely COP and entrainment ratio, were introduced to evaluate the performance improvement of the hybrid ejector-based air conditioning system. The primary findings are as follows:

1) The performance of ejector is sensitive to its area ratio and an ejector with area ratio of 4.51 gives better performance than the one of 4 in terms of critical pressure and entrainment ratio.
2) The performance of hybrid system is sensitive to evaporating and generating pressure. A higher evaporating pressure leads to larger entrainment ratio and COP while a higher generating pressure only results in higher COP.
3) The compressor applied which enlarge the pressure difference between ejector secondary inlet and evaporator outlet indeed improve the overall performance and the COP improvement can reach up to 34% which indicated the potential of energy saving in hybrid ejector refrigeration system.

ACKNOWLEDGMENT

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