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A new ejector refrigeration system with an additional jet pump

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Abstract

A new ejector refrigeration system (NERS) with an additional liquid–vapor jet pump was proposed. The jet pump was used to decrease the backpressure of the ejector, and then the entrainment ratio and the coefficient of performance (COP) of the new system could be increased. The theoretical analysis and simulation calculation was carried out for the new system. The comparison between NERS and conventional ejector refrigeration system (ERS) was made under the same operating condition. The variation of the new system's COP with generator temperature and backpressure was discussed for two refrigerants: R134a and R152a. The calculation results show the COP of NERS can be improved more effectively and that happens at the cost of more pump work. From the point of view of exergy, the new system is of higher exergetic efficiency and feasible.

Keywords: Ejector refrigeration; Ejector; Jet pump

1. Introduction

Ejector refrigeration system (ERS) powered by low-grade energy has been studied since the mid-1950s. Compared with other refrigeration systems, ERS has some special advantages such as the simplicity in construction, high reliability and low cost. However, the coefficient of performance for the conventional ERS is significantly lower than that for others. This has restricted its wide applications.

In recent years, people have tried to find wider application for ejector refrigeration systems in refrigeration and air conditioning field by directly utilizing low-grade thermal energy, such as solar energy and waste heat. Eames and Aphornratana conducted the research on a small-scale steam-jet refrigerator, in which an ejector with movable primary nozzle was used [1,2]. Sun's studies showed that the use of variable-geometry ejector might achieve optimum performance under various operating condition [3]. Chang and Chen investigated a steam-jet refrigerator using petal nozzle and results indicated it might elevate the performance of an ejector [4]. A combined ejector-vapor compression refrigeration systems as a mechanically efficient way to increase the COP of ERS was proposed and studied [5–7]. In addition, Arbel and Sokolov also proposed combined systems, using the booster sub-cycle as compression-enhanced way to increase the COP of ERS [8]. Despite of the higher COP in this combined cycle, additional compressor actually makes systems more complicated and higher cost due to compressor itself.

This present study takes a new approach to enhance the COP of ERS. For this purpose, a jet pump (liquid–vapor type) is introduced into the conventional ERS (CERS), and the new ERS (NERS) is proposed. By the jet pump the backpressure of the ejector is decreased, the entrainment ratio is increased and then the COP of the system is improved. The new system can utilize solar energy or waste heat more effectively and alleviate the energy tension in some extent. A

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Nomenclature

COP	coefficient of performance	e	evaporator
Ex	exergy (kW)	eje	ejector
h	enthalpy (kJ/kg)	g	generator
m	mass flow rate (kg/s)	jet	jet pump
P	pressure (Pa)	m	mixing section of ejector
P_b	backpressure (Pa)	mf	mixed fluid
Q	heat load (kW)	n	nozzle
s	entropy (kJ/kg K)	n1	inlet of nozzle
t	temperature (°C)	n2	outlet of nozzle
T	thermodynamic temperature (K)	p	pump
T_0	ambient temperature (K)	pf	primary fluid
u	velocity (m/s)	s	isentropic procedure
η	efficiency	sf	secondary fluid
μ	entrainment ratio, $\mu = m_{sf}/m_{pf}$		

Subscripts

c	condenser
d	diffuser section of ejector

general description of the new system is presented and further analysis on its performance is conducted based on mathematical model. And the comparison between NERS and the ERS is made.

2. General system description

Fig. 1 shows a basic CERS. It mainly consists of a generator, an evaporator, a condenser, expansion device, ejector and circulating pump. In this system, the vapor (primary fluid) from the generator at high pressure flows through the nozzle of the ejector and entrains the vapor (secondary fluid) from evaporator at low pressure. The primary fluid and secondary fluid then mix in the mixing section and recover a pressure in the diffuser. The combined fluid flows to condenser where it condenses. And then the condensate is divided into two parts: one is pumped back to the generator, and the other flows through the expansion device and enters the evaporator, where it is evaporated to vapor. The vapor is finally entrained into the ejector again, thus finishing the ejector refrigeration cycle. Within one cycle, when the system obtains heat from heat source in the generator and work input from the pump, it produces the cooling effect in the evaporator and dissipates the heat to the environment through the condenser.

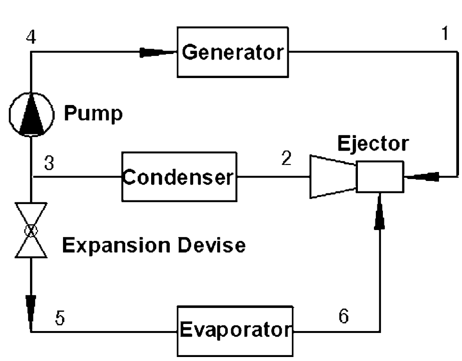

Fig. 1. A conventional ejector refrigeration system.

The ejector is the key component in the ejector refrigeration cycle. The ejector generally consists of four parts: a nozzle section for a primary flow and a suction chamber for the secondary flow, a mixing section for the primary flow and secondary flow to mix, a throat section (constant area section) in which the mixed fluid normally undergoes a transverse shock and a pressure rise, and a diffuser section for the mixed fluid to recompress to the back pressure. Generally, a jet pump is similar to an ejector in configuration. Fig. 2 shows the schematic configuration of an ejector (or a jet pump) and the variation of pressure in the ejector or the jet pump. In this figure the real line and the dashed line represent the variation of pressure in the ejector or the jet pump, respectively.

The COP of the CERS is highly dependent on the entrainment of ejector (the ratio of secondary flow rate to primary flow rate). The entrainment is relative to the primary flow inlet state, secondary flow inlet state and mixing outlet state of the ejector. The three primary, secondary and backpressures, are main factors that affect the entrainment. Some research results showed the entrainment increases with the decreasing backpressure when primary pressure and secondary pressure are constant [9,10]. These pressures are respectively the

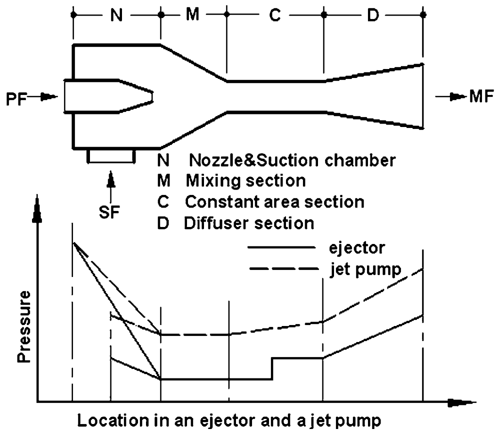

Fig. 2. The variation of pressure in ejector or jet pump.

functions of condensing temperature, generating temperature and evaporating temperature. Usually these temperatures depend on operating condition and are fixed within a range. The optimum operating condition could not make more improvement for COP.

The NERS developed by author is shown in Fig. 3. In this system, an additional jet pump is put between ejector and condenser. This additional jet pump is applied to entrain the mixing vapor from ejector, which is as secondary flow of jet pump. Working fluid of jet pump (primary flow) is from the dividend part of circulating pump outlet liquid. Clearly, an additional jet pump could make backpressure of the ejector lower, and enhance entrainment. As we know, the work consumed by circulating pump is very small due to the incompressibility of liquid. Compared with the compressor as booster, jet pump has more advantages such as simple construction and low cost. This could be as a mechanically efficient way to increase the COP of NERS. Details of new cycle performance analysis are discussed as follows.

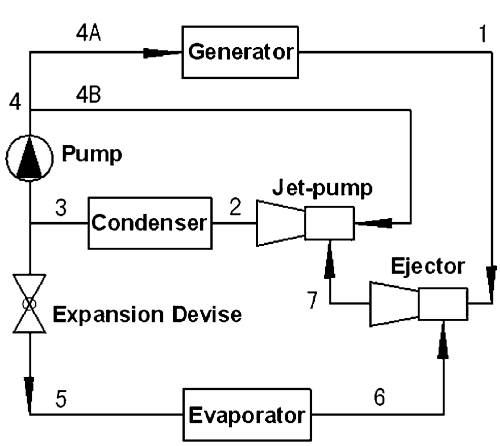

Fig. 3. A new ejector refrigeration system.

3. Theoretical analysis and simulation calculation

In the present system, the main ejector is a common vapor–vapor ejector and the jet pump is like a type of liquid–vapor ejector. They have similar structure and working process. Generally, there are two basic approaches for ejector analysis used by researchers [11–13]. In the present study, for simplicity the one-dimensional constant pressure mixing theory developed by Keenan et al. [14] is applied to both ejector and jet pump although two-phase mixing flow through the jet pump is more complicated. The following assumptions are made for the analysis:

(1) The flow inside the ejector and jet pump is in steady state and one-dimensional.
(2) The compression and expansion process in ejectors are isentropic or friction losses are defined in terms of the isentropic efficiency in the nozzle, mixing section, diffuser and circulating pump.
(3) Mixing in the mixing section of ejectors occurs at constant pressure and complying with the conversion of energy and momentum; the inner walls of ejectors are adiabatic, namely no heat losses.
(4) Normal transverse shocks may occur at any plane in the throat section.
(5) A homogenous two-phase flow mixing is idealized for the jet pump inside.

Based on the above assumptions, the mass, momentum and energy equations are applied to sections for both the ejector and the jet pump and then the following equations are obtained.

In the nozzle section, when inlet velocity of primary flow $\mu_{\mathrm{pf,n1}}$ is negligible, the exit velocity of primary flow is expressed according to energy balance equations as

$$u_{\mathrm{pf,n2}} = \sqrt{2\eta_{\mathrm{n}}(h_{\mathrm{pf,n1}} - h_{\mathrm{pf,n2,s}})} \tag{1}$$

where

$$h_{\mathrm{pf,n1}} = f(T_{\mathrm{pf,n1}}, P_{\mathrm{pf,n1}}) \tag{2}$$

$$s_{\mathrm{pf,n1}} = f(T_{\mathrm{pf,n1}}, P_{\mathrm{pf,n1}}) \tag{3}$$

$$s_{\mathrm{pf,n2,s}} = s_{\mathrm{pf,n1}} \tag{4}$$

$$P_{\mathrm{pf,n2,s}} = P_{\mathrm{m}} \tag{5}$$

$$h_{\mathrm{pf,n2,s}} = f(s_{\mathrm{pf,n2,s}}, P_{\mathrm{pf,n2,s}}) \tag{6}$$

In the mixing section, when inlet velocity of secondary flow is negligible, the average velocity of mixed flow is written according to momentum equations:

$$u_{\mathrm{mf,m}} = u_{\mathrm{pf,n2}}\sqrt{\eta_{\mathrm{m}}}/(1+\mu) \tag{7}$$

The enthalpy of mixed flow is written according to energy equation as

$$h_{\mathrm{mf,m}} = (h_{\mathrm{pf,n1}} + \mu h_{\mathrm{sf}})/(1+\mu) - u_{\mathrm{mf,m}}^2/2 \tag{8}$$

where

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

$$s_{mf,m} = f(h_{mf,m}, P_{mf,m}) \tag{10}$$

In diffuser section, the mixed fluid converts the kinetic energy into pressure energy. When the exit velocity of the mixed fluid is neglected, the actual exit enthalpy of the mixed fluid is written according to energy balance equation as

$$h_{mf} = h_{mf,m} + u_{mf,m}^2/2 \tag{11}$$

In addition, according to the definition of isentropic efficiency for compression in diffuser section, the actual exit enthalpy of the mixed fluid is also expressed as

$$h_{mf} = h_{mf,m} + (h_{mf,d,s} - h_{mf,m})/\eta_d \tag{12}$$

where

$$h_{mf,d,s} = f(s_{mf,d,s}, P_{mf,d,s}) \tag{13}$$

$$P_{mf,d,s} = P_b \tag{14}$$

$$s_{mf,d,s} = s_{mf,m} \tag{15}$$

It is known the performance of ejectors are evaluated by their entrainment ratio μ. Based on above Eqs. (1), (7), and (12), it can be derived as

$$\mu = \sqrt{\eta_n\eta_m\eta_d((h_{pf,n1} - h_{pf,n2,s})/(h_{mf,d,s} - h_{mf,m}))} - 1 \tag{16}$$

When the inlet state parameters of primary fluid, secondary fluid and backpressure are given, the value of entrainment ratio μ could be found using iteration calculation.

For the whole cycle performance, the basic equations obtained from the conversion law for energy are written as follows:

$$\text{For evaporator:} \quad Q_e = m_e(h_6 - h_5) \tag{17}$$

$$\text{For generator:} \quad Q_g = m_g(h_1 - h_4) \tag{18}$$

$$\text{For condenser:} \quad Q_c = m_c(h_2 - h_3) \tag{19}$$

$$\text{For circulating pump:} \quad W = m_p(h_4 - h_3) \tag{20}$$

The system performance is evaluated by the coefficient of performance COP, which is the ratio of the cooling effect to the gross energy input of the cycle. It is written as follows:

$$\text{COP} = Q_e/(Q_g + W) \tag{21}$$

4. Simulation results and discussion

Usually, condensing temperature and evaporating temperature depend on refrigeration request and environmental temperature, but generating temperature lies on the potential temperature of heat resource. The former two temperatures can only vary in a limited range but the latter one can change more widely. Therefore, in the following analysis, more emphasis is laid on the impacts that are made by generating temperature and backpressure.

The following simulation results are based on the isentropic efficiencies of the ejectors, which are assumed to be $\eta_n = 0.85$, $\eta_d = 0.85$, $\eta_m = 0.95$. Considering the important function of the circulating pump in the new system, pump efficiency has also been taken into account in the simulation by assuming isentropic efficiency $\eta_p = 0.75$. The cooling capacity of the refrigeration systems is taken to be 1000 W for seeing about COP of them.

R134a and R152a are singled out as working refrigerants. Their thermodynamics properties are calculated based on PR equation of state, and the calculating program is written with Fortran Language.

4.1. *Effect of secondary pressure and backpressure on the entrainment ratio*

Secondary pressure and backpressure are two main factors that effect on the entrainment ratio of an ejector. When secondary pressure increases or backpressure decreases, the entrainment ratio will be increased, but there exist some differences. Fig. 4 shows the effect of secondary pressure and backpressure on the ejector. In the figure the comparison is made on the basis of the equal relative changes of those two pressures. Here the reference entrainment ratio is calculated when the generating temperature is 85 °C, the condensing temperature is 35 °C and the evaporating temperature is 5 °C. P_0 and μ_0 are representative of corresponding backpressure (or secondary pressure) and entrainment ratio, while P and μ stand for variational values. The coordinates are zero-dimensional, (a) represents backpressure curve and (b) represents secondary pressure. From the figure, it is clear that the decrease of backpressure can make the

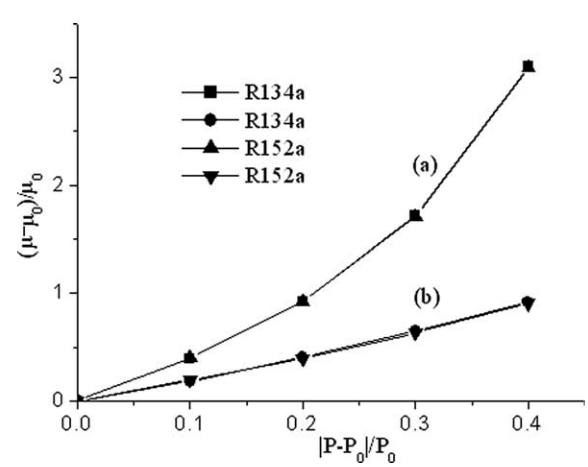

Fig. 4. Variation of relative change rate of ejector entrainment ratio with secondary pressure or backpressure.

entrainment ratio acquire more improvement under the circumstances of the equal relative changes. This also indicates that the new system with additional jet pump arranged behind the ejector can improve the performance of the ejector much better.

4.2. Effect of generating temperature on refrigeration systems performance

The following results are obtained when condensing temperature $t_c = 35$ °C, evaporating temperature $t_e = 5$ °C, backpressure $P_b = 7.5$ bar for R134a and $P_b = 6.5$ bar for R152a.

Fig. 5 shows the variation of entrainment ratio μ_{eje}, μ_{jet} with generating temperature t_g. As t_g rises, μ_{eje}, μ_{jet} all increase. This results from that an increase in t_g causes generating pressure namely primary pressure of ejector and jet pump to increase, and then the performance of them is improved. Figs. 6 and 7 show the variation of generator load and pump work with generating temperature rising. From Fig. 6, we can see in the NERS the heat load of generator drops as t_g rises. On the one hand, for entrainment ratio μ_{eje}, μ_{jet} increase, the flow rate of refrigerant in the cycle is caused to decrease when cooling capacity is kept at 1000 W. On the other hand, the heat load of unit mass flow rate is also decreased as t_g rises. When generating temperature varies in the range of 80–100 °C, heat energy consumed in the new system is less than and only accounts for 52–64% of that in the conventional system. In addition, NERS consumes more pump work than CERS, and the trend of pump work is not obvious. For small entrainment ratio of jet pump in the NERS, the flow rate is increased and then pump work is increased. As generating temperature t_g rises, the decrease of flow rate and the increase of unit mass pump work together make the whole pump work not change conspicuously.

Fig. 8 shows variation of COP with generator temperature t_g. As t_g rises, COP of NERS will increase, and the R152a's COP curve is higher than R134a's. It is clear that COP of NERS is much greater than that of CERS. When $t_g = 95$ °C, COP of R152a and R134a

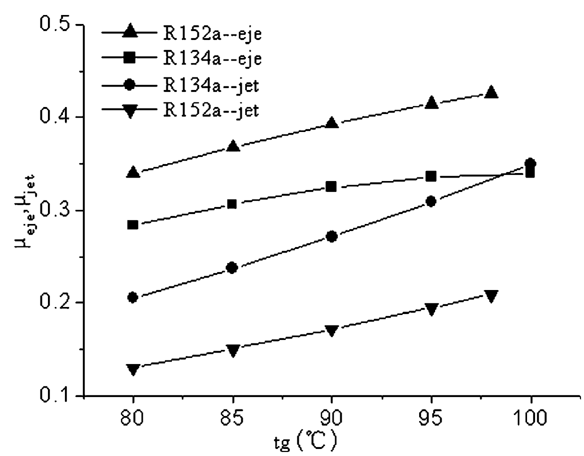

Fig. 5. Variation of entrainment ratio with generating temperature.

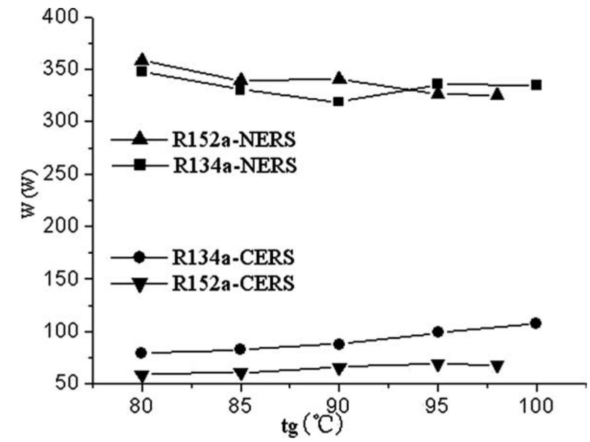

Fig. 7. Variation of pump work with generating temperature.

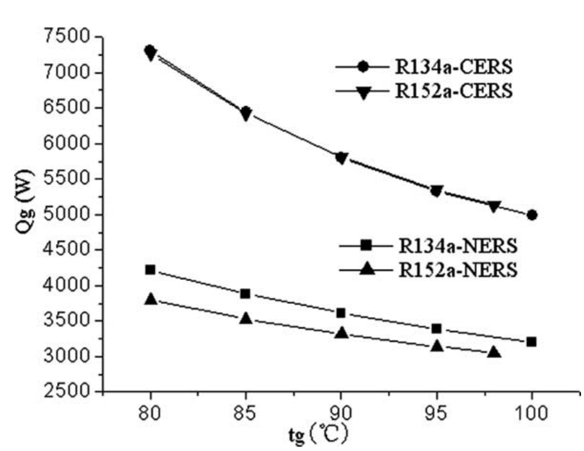

Fig. 6. Variation of generator's heat load with generating temperature.

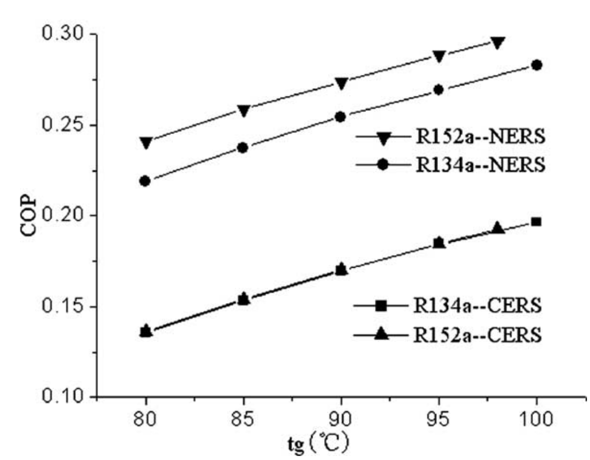

Fig. 8. Variation of COP with generating temperature.

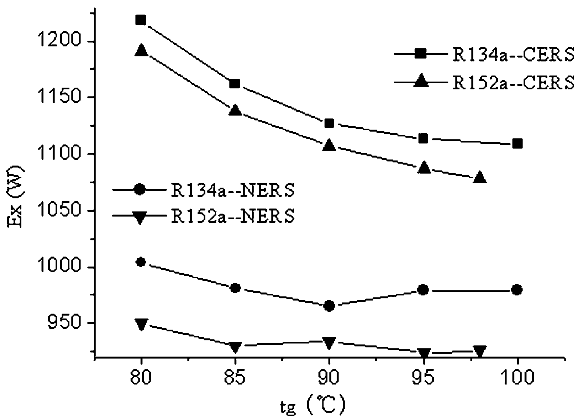

Fig. 9. Variation of input exergy with generator temperature.

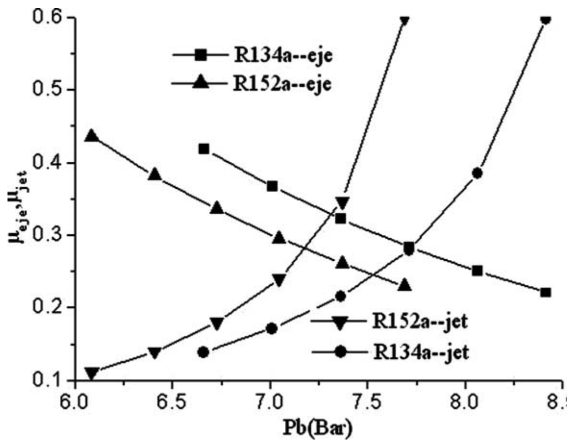

Fig. 10. Variation of entrainment ratio with backpressure.

in NERS increases respectively by 57.1% and 45.9% compared with that in CERS.

It is obvious that NERS improves its COP at the cost of more pump work. In the new system both low-grade energy and high-grade electric work are utilized. Considering the grade of input energy and exergy concept, NERS can still save 12–20% input exergy when the comparison calculation is made based on the exergy analysis only for the input aspect of the whole system. Fig. 9 shows variation of input exergy with generator temperature and from that it is obvious that NERS is of higher exergetic efficiency than CERS. The input exergy is described as follows:

$$\mathrm{Ex} = \mathrm{Ex}_{Q_g} + W \tag{22}$$

where

$$\mathrm{Ex}_{Q_g} = (1 - T_0/T_g) \cdot Q_g \quad (T_0 = 298.15\ \mathrm{K}) \tag{23}$$

4.3. Effect of backpressure on NERS performance

The following results are obtained when generating temperature $t_g = 85$ °C, condensing temperature $t_c = 35$ °C, evaporating temperature $t_e = 5$ °C and yet P_b of ejector is variable.

The variation of entrainment ratio μ_{eje}, μ_{jet} with backpressure of ejector is shown in Fig. 10. An increase in P_b causes μ_{eje} to drop and yet μ_{jet} to increase. This illuminates the increase of P_b plays down the performance of ejector, which is similar to the analysis conclusion in References [6,7]. However, the increase of P_b, which is also the primary pressure of jet pump, enhances the entrainment ratio of jet pump.

Figs. 11 and 12 show the variation of generator's heat load Q_g and pump work W with backpressure. As P_b rises, Q_g increases and yet W decreases. For the drop of μ_{eje}, the flow rate of generator increases and then Q_g increases. In the same time, μ_{jet} increases. Thus, two entrainment ratios make the circulating flow rate decrease and then lead to the decrease of W. We can

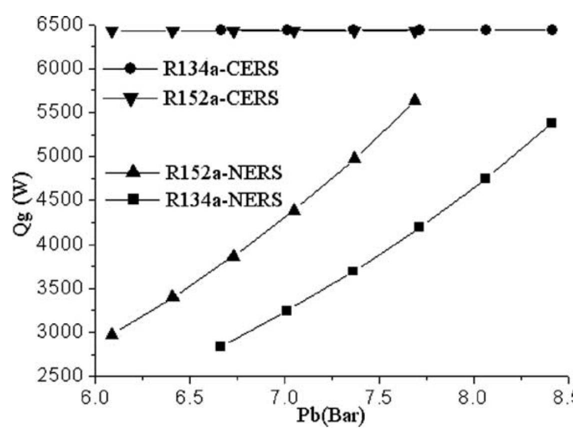

Fig. 11. Variation of generator's heat load with backpressure.

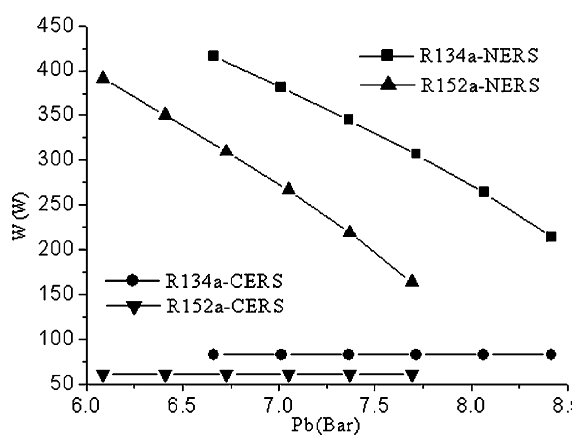

Fig. 12. Variation of pump work with backpressure.

see the heat load in NERS is pretty lower than that in CERS, and NERS can save 12–56% heat energy. Of course, it is also clear that NERS consumes much more pump work W than CERS. However, NERS is better from the point of view of exergy shown in Fig. 13. When

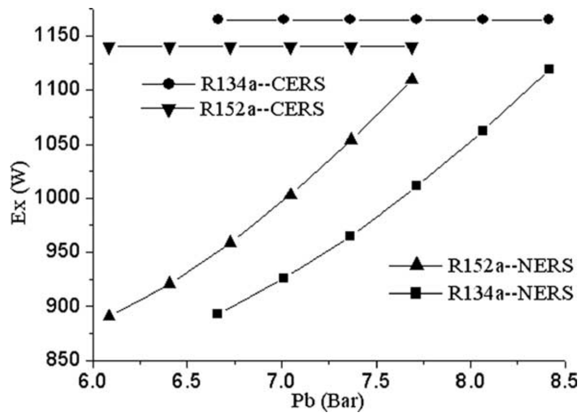

Fig. 13. Variation of input exergy with backpressure.

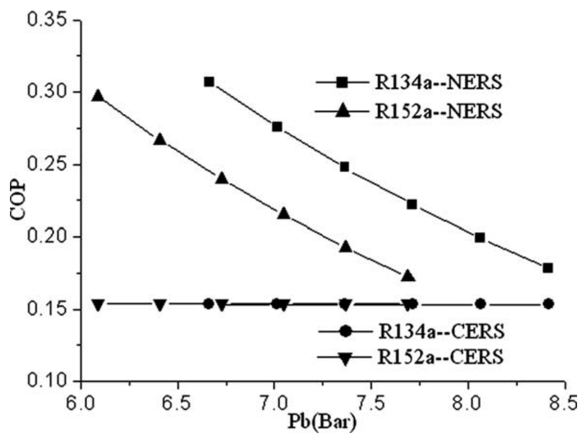

Fig. 14. Variation of COP with backpressure.

$P_b = 6.7$ bar, NERS can save 23% input exergy for R134a and 16% input exergy for R152a.

As P_b rises, the sum of Q_g and W in NERS increases, and that its COP drops as is shown in Fig. 14, where the COP curve of R134a is higher than that of R152a. In CERS COP is about 0.15 for those two refrigerants but in NERS their COP can reach above 0.30.

5. Conclusion

This paper proposed a new ejector refrigeration system (NERS). From the foregoing analysis, we can see NERS is feasible and there is an obvious improvement in COP.

(1) Ejector and jet pump are the key part. Entrainment ratios have a great effect on COP and the rise of COP depends on the coupling of those two entrainment ratios μ_{eje}, μ_{jet}. Jet pump as a type of liquid–vapor ejector functions weightily in the system. Therefore, in practical application jet pump of high efficiency and great entrainment ratio should be considered first in order to improve COP.

(2) When condensing temperature, evaporating temperature, backpressure of ejector and cooling capacity are kept unchangeable, as generating temperature rises, the heat load of generator decreases and COP of NERS increases. In the foregoing operating condition, when generating temperature is 95 °C, the COP of R152a and R134a in NERS is increased respectively by 57.1% and 45.9% compared with the conventional ejector refrigeration system (CERS).

(3) When generating temperature, condensing temperature, evaporating temperature and cooling capacity are kept unchangeable, as backpressure of ejector drops, generator's heat load decreases, pump work increases and COP increases. And when backpressure changes the COP of NERS can vary in a wide range and increase by one time compared with CERS. NERS improves its COP at the cost of more pump work. However, in the point of view of exergy the new system is much better. For example, in the foregoing operating condition, when $P_b = 6.7$ bar, NERS can save 23% input exergy for R134a and 16% input exergy for R152a. Therefore, in practice both COP and consumed pump work of the system should be taken into account integrally, that is to say, the system operating conditions should be optimized. And this is also the following research.

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