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Experimentation and Cycle Performance Prediction of Hybrid A/C system Using Automobile Exhaust Waste Heat

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Abstract: A hybrid air-conditioning (A/C) system is proposed which consists of two cycles: 1) an ejector cycle driven by exhausted waste heat; and 2) a compressor A/C cycle. The system can operate under three modes: compressor, hybrid and ejector mode. Under the hybrid mode, the ejector driven by waste heat reduces the compression ratio (CR) of the compressor and boosts the compressor discharge pressure to the condenser pressure. The governing equations are derived based on energy and mass balances for each component of the system. The performance of the hybrid A/C system under compressor mode and hybrid mode is first analyzed theoretically at design conditions. Then it was tested experimentally through variations of 1) primary pressure from 11 to 23 bar; 2) evaporation pressure from 2.5 to 4.5bar; and 3) condensation pressure from 9 to 12.5 bar, respectively. The experimental results show that the hybrid system is feasible, and can significantly enhance the performance of the automobile A/C systems, 35.2% COP improvement at the automobile idle conditions and more than 40% COP improvement when the automobile speeds over 80km/h conditions. The increase of primary pressure and evaporation pressure has positive effect on system performance while the increase of condensation pressure has negative effect. The hybrid A/C system has the potential to be adopted in automobiles for the advantages of low cost, durable operation and better energy efficiency.

Key words: automobile air-conditioning system; ejector; exhaust waste heat; hybrid system; compression ratio; COP.

1. Introduction

Air conditioner (A/C) system is an important component of automobiles whose utilization provides comfort to occupants. However, it significantly increases the energy consumption of automobiles and decreases their performances. Lambert et al. reported that mechanical compressors of A/C systems could increase fuel consumption of automobiles by 12–17% [1]. The fossil fuels burned by automobiles emit waste heat, CO₂ and other harmful particles which have been causing long-term damages to the environment.

Generally, an engine for an automobile operates at maximum 35% thermal efficiency which means it may release about 65% of the energy as waste heat through coolant, exhaust gases and engine compartment warm-up, almost twice as much as its mechanical output. During the operation of a typical driving cycle, the engine efficiency cannot always reach its maximum. As the operating efficiency decreases, the wasted energy increases, thus representing a larger energy

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potential for recovery and utilization [2, 3]. For instance, an average fuel economy of a car in the US is near 0.09 liter of gasoline per kilometre, and a representative vehicle could be the Ford Taurus with a 3.0 L engine and a maximum output power of 115 kW. Data shows that the waste heat available for a 115 kW engine varies from 20 to 400 kW across the engine map, with an average value of 23 kW over the Federal Test Procedure cycle [4]. Temperatures of waste heat can range from 200 °C surface temperature to 600 °C gas temperature. The recovery of exhaust waste heat for automobile cooling system is, therefore, a promising way for energy saving. Among the various techniques for utilization of waste heat or low grade energy to A/C system, ejector based A/C system has been extensively studied as it has fewer moving parts (no compressor), very low in wear and significantly durable [5].

The technique of ejector refrigeration cycle was developed by Maurice Leblanc in 1910[6], even before vapor-compression refrigeration cycle. However, the ejector refrigeration system has not been widely adopted due to its low COP [7]. To overcome the drawback of low COP of ejector based systems, many studies are focused on optimization and improvement of system performance and ejector efficiency. Sun analysed the effect of ejector geometries on system performance, including flow rates and entrainment ratio for a 5kW steam-jet refrigerator [8]. Ziapour and Abbasy [9] integrated the heat pipe with an ejector which resulted in a compact and high performance system. Yan et al [10] investigated the key geometry parameters on the performance including the area ratio, the ratio of primary nozzle exit position, the length of constant-area mixing section to primary nozzle diameter, the converging angle of constant-pressure mixing section, the ratio of primary nozzle exit position and the length of constant-area mixing section to the diameter of constant-area mixing section. Zhu and Jiang [11] designed a bypass ejector with an annular cavity in the nozzle wall, and their results showed that the primary mass flow rate in the bypass ejector is always about 20% less than that in the conventional ejector and the maximum improvement in the entrainment performance of 31.5%. Chen et al. [12] made an evaluation on the optimum performance of ejectors in refrigeration system and concluded that a variable-geometry ejector seems a very promising alternative to ensure that the ejector refrigeration system operates at its optimum conditions. They also performed a detailed study on ejector working characteristics in terms of refrigeration efficiency, ejector entrainment ratio, and irreversibilities in each ejector component (nozzle, mixing chamber and diffuser) using R141b, R245fa and R600a as the working fluids[13]. They found

out that operating conditions and ejector component efficiencies have significant influence on ejector behaviour, and different refrigerants perform distinctively different in the ejector refrigeration system. Kasperski and Gil [14] carried out a series of calculations of the ejector refrigeration cycle for nine hydrocarbons for the generator temperature between 70 and 200 °C, with assumed temperatures of evaporation 10 °C and condensation 40 °C. Results showed that each hydrocarbon has its own maximal entrainment ratio at its individual temperature of optimum.

In other studies, Sokolov and Hearshgal [15] introduced a booster assisted ejector configuration by using mechanical power to enhance the secondary pressure without changing the refrigeration temperature. The booster assisted ejector cycle is very similar to the conventional ejector cycle. The low-pressure ratio mechanical-driven compressor is placed between the evaporator outlet and the ejector suction line. Therefore, the ejector can have a higher suction pressure, which increases its performance. The assisted ejector configuration, or named as hybrid ejector, was further researched by Sun and Eames [16, 17]. The authors conducted mathematical simulations using R718 and R134a as the refrigerant in the ejector and the vapor compression sub-cycle, respectively. They achieved a maximum theoretical COP improvement of more than 50%. Gokthun [18] conducted an analysis of maximum possible COP of a solar powered hybrid ejector-vapor compression cycle in terms of Carnot efficiency. The increased COP of the conventional vapor compression system can be achieved by increasing the sub-cooling level before entering the expansion device. The author employed energetic optimization technique to investigate the optimal performance of focusing collector driven, irreversible Carnot ejector-vapor compression cascaded cooling and heating systems. The study derived the optimal operating temperature of the solar collector and the maximum overall COP of the cooling and heating modes of the combined system. Huang et al. [19] proposed another combined ejector-vapor compression machine which allows the waste heat from the superheated vapor refrigerant in the compression sub-cycle to drive the ejector sub-cycle. The cooling effect obtained from the ejector-cooling device is used to cool the liquid condensate of compression sub-cycle to a sub-cooled state so that the COP of the system can increase. Their analytic and experimental results verified the feasibility of the system. The study claimed an average of 15.4% and a maximum of 24.5% COP improvement. Their subsequent studies on solar-assisted A/C system were reported in references [20, 21]. Yan et al. [22] described a combined ejector-vapor compression cycle that

uses working fluid R134a and air-cooled condensers for both sub-cycles. Their test results show that the system with certain operating conditions results in relatively high COP improvements (15.9-21.0%). Vidal and Colle [23] studied a combined compressor-ejector refrigeration cycle to balance the large pressure difference between low evaporation pressure and high primary pressure. They utilized an intercooler to integrate the ejector based refrigeration cycle and the compressor based cycle to improve system COP. Zhu and Jiang [24] proposed a hybrid vapor compression refrigeration system with an integrated ejector cooling cycle which was driven by the waste heat from the condenser. Their results showed the COP improvement of 9.1% for R22 system. Though there are many studies on the compressor-ejector combined system, few are concerned with waste heat from exhaust by taking the real operation of an engine into account. The variation in waste heat will require different modes for compressor and ejector combined system.

Fig. 1 Schematic diagram of the proposed hybrid system

In order to compensate the change in exhaust heat under different operation conditions, a hybrid A/C system was proposed, shown in Fig. 1, which consists of two cycles: 1) an ejector cycle driven by exhausted waste heat, and 2) a compressor A/C cycle. The system can operate under three modes: compressor, hybrid and ejector mode. Under the hybrid mode, the ejector secondary inlet which is connected in series with the discharge side of compressor to reduce the compression ratio (CR) of the compressor and using the pressure recover property to boost the

compressor discharge pressure to the condenser pressure. The overall system COP is consequently improved through the reduction of compressor CR. The governing equations are derived based on energy and mass balances for each component including compressor, generator, evaporator, condenser and pump in the system. The performance of the designed hybrid A/C system apparatus is first analyzed for the on-design conditions based on the governing equations, and then tested through variations of 1) primary pressure from 11 bar (evaporating temperature -42.97 °C for R134a) to 23 bar (evaporating temperature 73.74 °C for R134a); 2) evaporate pressure from 2.5 bar (evaporating temperature -4.28 °C for R134a) to 4.5 bar (evaporating temperature 12.48 °C for R134a); and 3) back pressure from 9 bar (evaporating temperature 35.53 °C for R134a) to 12.5 bar (evaporating temperature 47.9 °C for R134a), respectively. Improvement of COP and CR of hybrid mode over compressor mode were studied under these conditions.

2 Hybrid A/C system

2.1 System description

The designed hybrid system mainly consists of a compressor, a condenser, an evaporator, a generator, a pump, an ejector and an Electronic Expansion Valve (EEV). It can operate under three modes including: 1) compressor mode; 2) hybrid mode; and 3) ejector mode. The three operation modes can be switched through open and close of the valves as listed in Table 1.

Table 1 Valve status under different operation mode

Mode	Valve Open	Valve Close	Operation conditions
Compressor Mode	V1, V4	V2, V3	Only compressor works at automobile starting period
Hybrid Mode	V1,V3	V2,V4	Both compressor and ejector work when certain amount of exhaust heat recovered
Ejector Mode	V2,V3	V1,V4	Only ejector works for small cooling load and available exhaust heat

In the compressor mode, the compressor works same as the normal automobile cooling system during the periods for engine shut down and initial start phase when there is no enough exhaust

The thermodynamic cycle of the hybrid system is shown as in pressure enthalpy diagram of Fig. 2. 1-3-4-5-1 is the compressor mode; 4-6-7-2-3-4 and 4-5-1-2 is the hybrid mode. In the triangle $\Delta 123$, in particular, the dotted line $\overline{13}$, solid line $\overline{12}$ denote the compressor operation process in compressor mode and in hybrid mode, respectively. The solid line $\overline{72}$ and $\overline{23}$ roughly denote the ejector operation process. The pressure value of point (1) and (3) are equal to evaporation pressure and condensation pressure, respectively. It is obvious that the compression process $\overline{12}$ under the hybrid mode needs less work compared with compression process $\overline{13}$ under compressor mode.

The governing equations for each component of the hybrid system are derived based on energy and mass balances.

(1) Evaporator

Given Q_e , T_e and ΔT_e , the mass flow rate of the evaporator is m_2 :

$$m_2 = Q_e / (h_1 - h_5) \quad (1)$$

Since the throttling process is isenthalpic, then

$$h_5 = h_4 = f_{sat}(T_c) \quad (2)$$

Assume that the refrigerant at the outlet of the evaporator is super-heated ΔT_e , then

$$h_1 = f(T_1, P_1) \quad (3)$$

where

$$T_1 = T_e + \Delta T_e \quad (4)$$

$$P_1 = f_{sat}(T_e) \quad (5)$$

$$s_1 = f(T_1, P_1) \quad (6)$$

The pressure drop in heat exchanges is not considered, and the system pressure drop due to values has also been neglected.

(2) Compressor

Assuming the compression is a non-isentropic process in compressor [24], then

$$W_c = m_2(h_{2'} - h_{1'}) \quad (7)$$

Where m_2 is the refrigerant mass flow rate, calculated from eq. (1). Let η_c be the isentropic efficiency of the compression process and assuming the compression process (1')-(2',s) is an isentropic process and process (1')-(2') is the actual compression process. State (1') is same as (1). Then

$$h_{2'} = h_{1'} + (h_{2',s} - h_{1'}) / \eta_c = h_1 + (h_{2',s} - h_1) / \eta_c \quad (8)$$

The refrigeration enthalpy at state (2') for isentropic process is:

$$h_{2',s} = f(s_1, P_{2'}) \quad (9)$$

In the compressor mode, the discharge pressure of the compressor $P_{2'} = P_c$, then:

$$h_{2',s} = f(s_1, P_c) \quad (10)$$

(3) Ejector

Given the entrainment ratio of the ejector, the primary mass flow rate can be evaluated as:

$$m_1 = \frac{m_2}{\mu} \quad (11)$$

The enthalpy of the primary working fluid at the ejector inlet is:

$$h_7 = f(T_7, P_7) \quad (12)$$

The pressure and temperature of the primary working fluid is P_7, T_7 .

$$P_7 = P_g \quad (13)$$

$$T_7 = T_g + \Delta T_g \quad (14)$$

ΔT_g is the superheat at the outlet of the generator.

According to energy balance, the flow enthalpy at the outlet of the ejector can be obtained:

$$h_3 = \frac{m_1 h_7 + m_2 h_2 - E_\delta}{m_1 + m_2} \quad (15)$$

Where E_δ is the total kinetic energy loss of the primary flow and the secondary flow together in the ejector, and h_2 is the same as $h_{2'}$.

When the pressure, temperature and mass flow rate of both the primary working fluid and the secondary fluid, and the back pressure (condensation pressure P_c) is known, the ejector can be designed. Detailed structure and parameters of the ejector can be found in Ref [10].

(4) Condenser

The heat load of condenser under compressor mode is:

$$Q_{c,comp} = m_2(h_{3'} - h_4) \quad (16)$$

Assume no sub cooling at the outlet of the condenser,

$$h_4 = f_{sat}(T_c) \quad (17)$$

Under hybrid mode, however, the heat load function of condenser differs from that of the compressor mode,

$$Q_{c,hybr} = (m_1 + m_2)(h_3 - h_4) \quad (18)$$

(5) Pump

The work consumed by pump is

$$W_{pump} = m_1(h_6 - h_4) \quad (19)$$

Assuming that the pump undergoes a non-isentropic process, then

$$h_6 = h_4 + (h_{4,s} - h_4)/\eta_p \quad (20)$$

Where η_p is the isentropic efficiency of the pump. The refrigeration enthalpy for isentropic process is:

$$h_{4,s} = f(s_4, P_g) \quad (21)$$

$$s_4 = f_{sat}(T_c) \quad (22)$$

(6) Generator

The heat absorbed by refrigerant in generator can be described as:

$$Q_g = m_1(h_7 - h_6) \quad (23)$$

Assuming that η_{exch} is the heat exchanger efficiency, then the heat needs to be provided to the generator is

$$Q_{ex} = Q_g/\eta_{exch} \quad (24)$$

This heat can also be expressed as:

$$Q_{ex} = F_{fuel}\eta_{conv}\eta_{ex} \quad (25)$$

Where, F_{fuel} is the fuel consumed per hour of the automobile engine, η_{conv} is fuel efficiency converting to energy and η_{ex} is the energy efficiency transferring to coolant water and exhaust.

2.3 System performance parameters

The parameters used to evaluate system performance are:

(1) Compression ratio: defined as the ratio of the discharge pressure to the suction pressure of the compressor.

In compressor mode, the compression ratio CR_c is

$$CR_c = \frac{P_{disc}}{P_{suct}} \quad (26)$$

While in hybrid mode, the compression ratio CR_h is

$$CR_h = \frac{P_{disc,hybr}}{P_{suct}} \quad (27)$$

where P_{disc} , P_{suct} and $P_{disc,hybr}$ denote the discharge pressure of compressor in compression mode, the suction pressure of compressor in compression mode and hybrid mode, and the discharge pressure of compressor in hybrid mode, respectively. Let ΔCR and $\Delta CR\%$ be the compression ratio decrease and percent decrease, respectively, then

$$\Delta CR = CR_c - CR_h \quad (28)$$

$$\Delta CR\% = \frac{CR_c - CR_h}{CR_c} \times 100 \quad (29)$$

- (2) Ejector pressure lift ratio: defined as the ratio of the condenser pressure to the evaporation pressure of the.

$$CLR = \frac{P_c}{P_e} \quad (30)$$

- (3) System COP: defined as the ratio of the produced refrigeration to the energy input into the refrigeration cycle including compressor, evaporator, condenser, pump, and expansion valve. In compressor mode, the COP_c is

$$COP_c = \frac{Q_e}{W_c + W_{fan}} \quad (31)$$

In hybrid mode, the COP_h is

$$COP_h = \frac{Q_e}{W_c + W_{pump} + W_{fan}} \quad (32)$$

Where W_c , W_{pump} and W_{fan} are the input energy required by the compressor, pump and condenser fan, respectively. The COP improvement ΔCOP and COP percent improvement $\Delta COP\%$ are

$$\Delta COP = COP_h - COP_c \quad (33)$$

And

$$\Delta COP\% = \frac{COP_h - COP_c}{COP_c} \times 100\% \quad (34)$$

2.4 Design conditions

The cooling load Q_e given is 3.2053 kW, values of all the efficiencies are set as 0.9[25], and other given design parameters are listed in Table 2. The NIST library [26] is adopted to simulate the thermodynamic properties. Based on the above equations, the calculate process as shown in Fig 3, and the calculated results for the hybrid system are also listed in Table 2.

Fig. 3 The flowchart for calculation

The calculation results show that 16.56kW of exhaust waste heat is used in this hybrid system for the design condition, which is equivalent to the heat available at idle condition of a normal compact car, representing the minimum heat recovered for the car running at 80 kilometer per hour, which can be as high as 34.3 kW. The COP of the hybrid mode is 38.26% high over that of compressor mode while the compression ratio of compressor decreased from 2.63 to 1.78.

Table 2 System on-design performance

		Compressor mode	Hybrid mode
Given design parameters	$Q_e(\text{kW})$	3.2053	3.2053
	$T_e(^\circ\text{C})$	7	7
	$\Delta T_e(^\circ\text{C})$	10	10
	$T_c(^\circ\text{C})$	45	45
	$T_g(^\circ\text{C})$	-	83
	$\Delta T_g(^\circ\text{C})$	-	20
Calculated results	$Q_g(\text{kW})$	-	16.56
	$P_g(\text{bar})$	-	18
	$P_e(\text{bar})$	3.5	3.5
	$Q_c(\text{kW})$	4.34	19.71
	$P_c(\text{bar})$	9.2	9.2
	$M_1(\text{kg/s})$	-	0.1133
	$M_2(\text{kg/s})$	-	0.02266
	CR	2.63	1.78
	COP	2.60	3.59
	$\Delta COP\%$	-	38.26

3 Experimental layout and apparatus

The designed schematic diagram for testing the hybrid automobile A/C system is shown as in Fig. 4. The Picture of the designed experimental apparatus is shown in Fig. 5. The generator is an oil tank heated by a heater simulating the automobile exhaust heat source. The high pressure refrigerant steam was produced in the generator filled with heat transfer oil which has smoking point of 400°C . Evaporator coil is immersed in another tank filled by water which is heated by another heater simulating the cooling load measured by a power meter. R134a is selected as the refrigerant in the hybrid A/C experimental apparatus. The compressor used is a hermetic piston compressor with a variable speed inverter. The condenser is a fan-coil type, where the compressed steam will be cooled by ambient air blown by a fan. The pump used for pumping the refrigerant to the generator is a plunger pump. An electronic expansion valve is used for the throttling process, which adjusts the mass flow rate according to evaporator pressure.

In the system performance test, the parameters of temperature, pressure and refrigerant flow rate were captured by NI data acquisition system and processed by Labview software. The instruments used are listed in Table 3.

Table 3 Instruments used in the system

Instruments	Scale	Error
Pressure sensor	0~30bar	0.5%
Temperature sensor (PT1000)	-20~400°C	0.3°C
Flow meter	0-140L/h	0.5%
Power meter	0-100Kw	0.5%
Frequency transformer	-	0.1Hz

The uncertainties on all the parameters measurement quantities (pressures, temperatures, mass flow rates, etc.) have been determined according to the Guide of Uncertainties on Measurements (GUM) [27].

Fig. 4 The experimental layout of the hybrid A/C system

Fig. 5 Photo of experimental apparatus of the hybrid A/C system

4 Results and Discussions

4.1 Effect of primary pressure on system performance

Under the hybrid mode, varying the primary flow pressure from 11 bar (evaporating temperature 42.97 °C for R134a) to 23 bar (evaporating temperature 73.74 °C for R134a) while keeping evaporation pressure at 3.5 (evaporating temperature 5.03 °C for R134a) bar and condensate pressure at 9 bar (evaporating temperature 35.53 °C for R134a), the effect of primary pressure on the system performance was tested, as relationship of pressure-enthalpy shown in Fig.6 and effect of primary pressure on the discharge pressure shown in Fig.7. It can be seen that the discharge pressure of the compressor decreases in a nearly parabola fashion with the increase of the primary pressure. It drops slowly at first but quickly in the later period as the primary pressure increases. It decreased to 7.1bar (evaporating temperature 21.2 °C for R134a) and 6.7 bar (evaporating temperature 25.23 °C for R134a) for the idle and 80km/h conditions, respectively.

Fig.6 Pressure-enthalpy (evaporation pressure is 3.5bar, the condenser pressure is 9 bar and primary flow pressure from 11bar to 23bar)

Fig.7 effect of primary pressure on the discharge pressure

Fig.8 describes the relationships between the compression ratio CR, COP_h and the primary pressure of the ejector. It is observed that CR decreases and COP_h increases with the increase of the primary pressure. The COP and CR of the system under compressor mode working in design conditions were tested and chosen as the base. The COP_h and CR of the system under hybrid mode were compared with them and the resulted ΔCOP and ΔCR are also shown in Fig.8. The figure shows that both COP_h and ΔCOP increase with the primary pressure in a nearly parabola fashion, which means higher primary pressure provides more work in replace of the compressor. The system operates more efficiently at a higher primary pressure.

Corresponding to the idle condition, also the design condition, in Fig.8, compression ratio reduction of 20.1% and COP improvement of 35.2% are achieved, which is very close to the calculated results. Similarly, compression ratio reduction of 22.9% and COP improvement of 41.3% are obtained at the 80km/h running condition when more available exhaust heat can be recovered to increase the primary flow pressure.

Fig. 8 effect of primary pressure on the compression ratio and COP_h

4.2 Effect of evaporation pressure

To consider the effect of the evaporation pressure on system performance under both compressor mode and hybrid mode, the evaporation pressure is varied from 2.5 bar (evaporating temperature $-4.28\text{ }^{\circ}\text{C}$ for R134a) to 4.5 bar (evaporating temperature $12.48\text{ }^{\circ}\text{C}$ for R134a) while the condensation pressure is kept constant at 9 bar (evaporating temperature $35.53\text{ }^{\circ}\text{C}$ for R134a) and the ejector primary pressure is kept at 18 bar in the hybrid mode, as relationship of pressure-enthalpy shown in Fig.9.

Fig.9 Pressure-enthalpy (primary flow pressure is 18 bar, the condenser pressure is 9 bar and evaporation pressure from 2.5 bar to 4.5 bar)

Fig. 10 shows the influence of evaporation pressure on the discharge pressure of the compressor. Under compressor mode, the discharge pressure is constant value of 9 bar (evaporating temperature $35.53\text{ }^{\circ}\text{C}$ for R134a) because the discharge pressure is equal to the condensation pressure. Under hybrid mode, the discharge pressure changes from 6 bar (evaporating temperature $21.58\text{ }^{\circ}\text{C}$ for R134a) to 6.5 bar (evaporating temperature $24.22\text{ }^{\circ}\text{C}$ for R134a) when the evaporation pressure increase from 2.5 bar (evaporating temperature $-4.28\text{ }^{\circ}\text{C}$ for R134a) to 4.0 bar (evaporating temperature $8.93\text{ }^{\circ}\text{C}$ for R134a). The reduction of discharge pressure is around 3

bar (evaporating temperature 0.67 °C for R134a) to 2.5 bar (evaporating temperature -4.28 °C for R134a) lower than that of compressor mode.

Fig. 10 Effect of evaporation pressure on the discharge pressure

The variation of CR for this test is shown in Fig. 11. Although both compression ratios decrease in the same trend under these two modes as the evaporation pressure increases, the CR of the hybrid mode is 0.5 - 1 less than that of the compressor mode, indicating that the load of the compressor is decreased under hybrid mode. The value of ΔCR decreases slowly as evaporation pressure increases and tends to be a constant when evaporation pressure further increases.

Fig. 12 indicates the COP comparison between compressor mode and hybrid mode. Although both COPs increase with the evaporation pressure, the trend is not very similar. The COP_h increases more rapidly than COP_c , as the evaporation pressure increases. Thus the ΔCOP rises sharply with the evaporation pressure. Over the test range, the COP of hybrid mode is 0.3 - 1.1 higher than that of the compressor mode. From the above analysis, raising the evaporation pressure is more effective for the system performance improvement under hybrid mode than that under the compressor mode.

Fig. 11 Effect of evaporation pressure on compression ratio

Fig. 12 Effect of evaporation pressure on COP

4.3 Effect of the condensation pressure

The condensation pressure, i.e. the back pressure of the ejector under hybrid mode, is an important parameter affecting the compression ratio as the condensation pressure is determined by the high temperature near the surface of the automobile. Fig.13 and Fig.14 presents the variation of the condensation pressure from 9 bar (evaporating temperature 35.53 °C for R134a) to 12.5 bar (evaporating temperature 47.91 °C for R134a) on the discharge pressure under the hybrid mode, while the evaporation pressure and the primary pressure are constant. The results indicate that the discharge pressure of compressor increases linearly from 7 bar (evaporating temperature 26.71 °C for R134a) to 10 bar (evaporating temperature 39.39 °C for R134a), representing 2 bar (evaporating temperature -10.08 °C for R134a) to 2.5 bar (evaporating temperature -4.28 °C for R134a) of increase on discharge pressure.

Fig.13 Pressure-enthalpy (condenser pressure from 9 bar to 12.5 bar and evaporation pressure is 3.3 bar)

Fig. 14 Effect of condensation pressure on discharge pressure

Fig. 15 and Fig. 16 show the effect of the condensation pressure on compression ratio and COP of compressor mode and hybrid mode respectively. It can be seen that both compression ratios increase at the same rate while COP of hybrid mode drops more quickly than that of the compressor mode. Hence ΔCR is almost a constant value while ΔCOP lowers gradually as the condensation pressure increases. COP of the hybrid mode is improved 25.3% when the condensation pressure increases from 9 bar to 12.5 bar.

Fig. 15 Effect of condensation pressure on compression ratio

Fig. 16 Effect of condensation pressure on COP

5 Conclusions

This paper presented a hybrid A/C system by combining compressor refrigeration and ejector refrigeration cycles together, which works in three modes: compressor mode, ejector mode and hybrid mode. The heat recovered from automobile exhaust heat contributes to decrease the compression ratio and improve the system COP under hybrid mode. An experimental setup was designed and the system performances were systematically studied. The results indicate that the concept of the hybrid system design is feasible, and can significantly enhance the performance of the automobile A/C system. The main conclusions can be summarized as follows.

- 1) The experimental results indicate that 20.1% and 22.9% of compression ratio can be reduced under hybrid mode at automobile idle and 80km/h speed running conditions respectively.
- 2) The experimental results demonstrate that 35.2% COP increase is in accord with 38.26% of the on-designed simulation result at the automobile idle condition, and more than 40% COP improvement are obtained when the running speed over 80km/h condition under hybrid mode, when compared with compressor mode.
- 3) The increase of primary pressure and evaporation pressure has positive effects on system COP and CR, while the increase of condensation pressure has negative effect on the system COP and CR.
- 4) Raising the evaporation pressure is more effective for the system performance improvement under hybrid mode than that under the compressor mode.

Because of the low cost and durable operation of the ejector refrigeration cycle, the proposed hybrid A/C system has the potential to be used for automobile. The hybrid system has the advantage of better COP than that of the normal compressor A/C system under the same condition. The control and switch issues of the overall system are currently under investigation and the results will be reported later.

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Nomenclature

A	area of the ejector nozzle throat
COP	coefficient of performance
CR	compression ratio
E	energy
f	function, the rotation frequency of the compressor
h	specific enthalpy (kJ/kg)
m	mass flow rate (kg/s)
P	pressure (bar)
Q	quantity of heat (kJ)
R	gas constant (kJ/(kg · K))
T	temperature (°C)
V	valve
W	power (kW)

Greek letters

ψ	a coefficient related to isentropic efficient of the compressible flow in the nozzle
γ	the gas specific heat ratio
ρ	the refrigerant density
δ	kinetic energy loss
ω	the entrainment ratio

subscripts

<i>conv</i>	fuel efficient converting to energy
<i>disc</i>	discharge
<i>ex</i>	exhaust waste heat
<i>exch</i>	heat exchanger
<i>fan</i>	fan
<i>fuel</i>	fuel consumed
<i>h, hybr</i>	hybrid
<i>pump</i>	pump
<i>sat</i>	saturation
<i>s</i>	isentropic
<i>suct</i>	suction
<i>t</i>	throat

References:

- [1] M.A. Lambert, B.J. Jones, Automotive adsorption air conditioner powered by exhaust heat: part 1: conceptual and embodiment design, *Journal of Automobile Engineering* 220 (2006) 959-972.
- [2] M. Hosoz, M. Direk, Performance evaluation of an integrated automotive air conditioning and heat pump system, *Energy Conversion and Management* 47 (2006) 545-559.
- [3] L. Hugues, A.B. Talom, Heat recovery from automotive engine, *Applied Thermal Engineering* 29 (2009) 439-444.
- [4] V.H. Johnson, Heat-generated cooling opportunities in vehicles, Washington D.C, U.S.A: SAE Technical Papers (2002) 2002-01-1969.
- [5] S. He, Y. Li, R.Z. Wang, Progress of mathematical modelling on ejectors, *Renewable and Sustainable Energy Reviews* 13 (2009) 1760-1780.
- [6] W.F. Stoecker Steam-jet refrigeration. Boston MA, U.S.A: McGraw-Hill; 1958.

- [7] S.J. Brown, P.A. Domanski, Review of alternative cooling technologies, *Applied Thermal Engineering* 64(1-2) (2014) 252-262.
- [8] D.W. Sun, Variable geometry ejectors and their applications in ejector refrigeration systems, *Energy* 21 (1996) 919-929.
- [9] B.M. Ziapour, A. Abbasy, First and second laws analysis of the heat pipe/ejector refrigeration cycle, *Energy* 35 (2010) 3307-3314.
- [10] J. Yan, W. Cai, Y. Li, Geometry parameters effect for air-cooled ejector cooling systems with R134a refrigerant, *Renewable Energy* 46 (2012) 155-163.
- [11] Y.H. Zhu, P.X. Jiang, Bypass ejector with an annular cavity in the nozzle wall to increase the entrainment: Experimental and numerical validation, *Energy* 68 (2014) 174-181.
- [12] J. Chen, H. Havtun, B. Palm, Parametric analysis of ejector working characteristics in the refrigeration system, *Applied Thermal Engineering* 69(1-2) (2014) 130-142.
- [13] J. Chen, H. Havtun, B. Palm, Investigation of ejectors in refrigeration system: Optimum performance evaluation and ejector area ratios perspectives, *Applied Thermal Engineering* 64(1-2) (2014) 182-191.
- [14] J. Kasperski, B. Gil, Performance estimation of ejector cycles using heavier hydrocarbon refrigerants, *Applied Thermal Engineering* 71(1) (2014): 197-203.
- [15] M. Sokolov, D. Hershtal, Enhanced ejector refrigeration cycles powered by low grade heat: part 1: systems characterization, *International Journal of Refrigeration* 13 (1990) 351-356.
- [16] D.W. Sun, Solar powered combined ejector-vapour compression cycle for air conditioning and refrigeration, *Energy Conversion and Management* 38 (1997) 479-791.
- [17] D.W. Sun, I.W. Eames, S. Aphornratana, Evaluation of a novel combined ejector absorption refrigeration cycle - I: computer simulation, *International Journal of Refrigeration* 19 (1996) 172-180.
- [18] S. Gokthun, Optimization of irreversible solar assisted ejector-vapor compression cascaded system, *Energy Conversion and Management* 41 (2000) 625-631.
- [19] B.J. Huang, V.A. Petrenko, J.M. Chang, C.P. Lin, S.S. Hu, A combined-cycle refrigeration system using ejector cooling cycle as the bottom cycle, *International Journal of Refrigeration* 24 (2001) 391-399.
- [20] B.J. Huang, J.H. Wu, H.Y. Hsu, J.H. Wang, Development of hybrid solar-assisted cooling/heating system, *Energy Conversion and Management* 51 (2010) 1643-1650.
- [21] B.J. Huang, Yen C.W., Wu J.H., Liu J.H., Hsu H.Y., V.O. Petrenko, J.M. Chang, C.W. Lu, Optimal control and performance test of solar-assisted cooling system, *Applied Thermal Engineering* 30 (2010) 2243-2252.
- [22] J. Yan, W. Cai, L. Zhao, Y. Li, L. Chen, Performance evaluation of a combined ejector-vapor compression cycle, *Renewable Energy* 55 (2013) 331-337.
- [23] H. Vidal, S. Colle, Simulation and economic optimization of a solar assisted combined ejector-vapor compression cycle for cooling applications, *Applied Thermal Engineering* 30(5) (2010) 478-486.

- [24] Y.H. Zhu, P.X. Jiang, Hybrid vapour compression refrigeration system with an integrated ejector cooling cycle, *International Journal of Refrigeration* 35 (2012) 68-78.
- [25] Zhu Y, Cai W, Wen C, et al. Simplified ejector model for control and optimization[J]. *Energy Conversion & Management*, 2008, 49(6):1424–1432.
- [26] NIST standard reference database 23: reference fluid thermodynamic and transport properties-REFPROP, version 9.1. U.S.A: National Institute of Standards and Technology; 2013.
- [27] Working Group 1 of the Joint Committee for Guides in Metrology, Evaluation of measurement data — Guide to the expression of uncertainty in measurement, version: JCGM 100: 2008.