Novel Vapor Compression Cycles with High Energy Efficiency Using Natural Refrigerants for Three Kinds of Appliances

Jianhua Wu
Xi'an Jiaotong University, China, People's Republic of, jhwu@mail.xjtu.edu.cn

Jiachen Li
Xi'an Jiaotong University, China, People's Republic of, lijiachen@stu.xjtu.edu.cn

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Jianhua WU*, Jiachen LI

School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an710049, China
*Corresponding author, Tel: 86-29-82663786; Email: jhwu@mail.xjtu.edu.cn

ABSTRACT

Natural refrigerants are eco-friendly substitutes for HCFCs and HFCs in vapor compression cycles, such as refrigeration or heat pump cycles. In this paper, three appliance systems are analyzed based on refrigerants characteristics for performance improvement, and novel and practical cycles are proposed and briefly analyzed. Dual-evaporator (DE) cycle for refrigerator can reduce the irreversible thermodynamic loss in one-stage cycle. Rotary compressors which have advantages in efficiency and dynamic balance capability are highly suitable to DE cycles. The improvement of theoretical refrigeration coefficient for DE cycle is up to 24.4% comparing with conventional one-stage cycle under refrigerator condition. Vapor injection heat pump (VIHP) cycle can efficiently improve the cycle performance especially under low ambient temperature. In the VIHP using R290 as refrigerant, the discharge temperature of compressor is usually not high enough for proper solubility of refrigerant in lubricant oil. Two techniques are applied to raise the discharge temperature, which brings higher performance and lubricating property to the compressor, and also brings the promotion to the cycle. The maximum improvements of theoretical heating coefficient and volumetric heating capacity are up to 17.8% and 40.7% respectively when evaporation temperature is -30℃. Heat pump cycle is suitable for electric vehicles because of its high theoretical heating coefficient of performance (ε_h) and heating capacity. In the heat pump cycle using CO2, combining with the fresh air heating progress in vehicle and wet compression, a particular cycle is proposed with much higher cycle performances. The ε_h and volumetric heating capacity can be at least 40% higher than that of conventional heat pump cycle when evaporation temperature is -30℃. And wet compression makes further improvements.

1. INTRODUCTION

In compliance with the Paris Agreement and the Kigali Amendment, hydrofluorocarbons (HFCs) refrigerants for vapor compression systems has a freeze date for each country, and need to be replaced finally. In addition, energy efficient devices for refrigerating or heat pump are in demand due to the increasing high energy price causing by its shortage. For these reasons, the development of refrigerating or heat pump cycles needs to be promoted. To achieve the further improvement of efficiency, the characteristics of natural refrigerants and device systems should be considered together, and hence the cycles can get redesigned combining with them.

In this paper, the progress of vapor compression cycles in three kinds of devices for refrigerating or heat pump will be briefly introduced. And then novel cycles will be designed. Finally necessary simulation or analysis will be conducted to prove the advancement of novel cycles comparing with each basic cycle.

2. DUAL-EVAPORATOR REFRIGERATOR CYCLE WITH ROTARY COMPRESSOR

2.1 Dual-Evaporator Refrigerator Cycle

Conventional refrigerator cycle is one-stage cycle which uses only one evaporator exchanging heat with refrigerating chamber and freezing chamber. The evaporation temperature is usually set on the basis of the temperature of freezing chamber. The thermodynamics of refrigeration cycle indicate that the efficiency is inversely proportional to the temperature difference between the condenser and the evaporator. The efficiency of one-stage cycle is low because of the large temperature difference between ambient and freezing chamber.

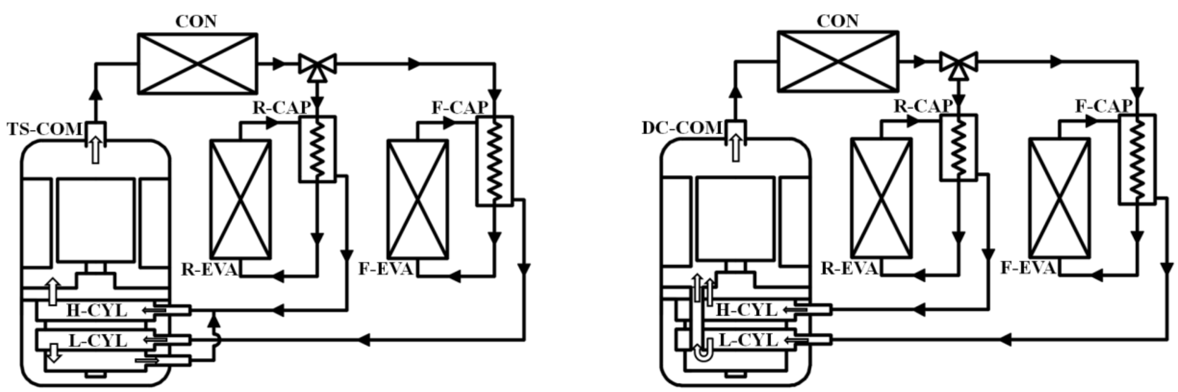
To improve the cycle efficiency, current novel cycles propose 2 evaporators connected in parallel, which can operate the refrigerating and freezing chambers separately (Mina *et al.*, 2015). Hence the cycle efficiency, especially the efficiency of refrigerating portion, gets greatly improved. Correspondingly the compression process needs to be changed into single-stage alternating compression, two-stage compression or parallel compression. These cycles are collectively known as dual-evaporator cycle.

There are three main types of dual-evaporator cycles distinguish by different compression processes i.e. alternating single-stage compression, two-stage compression and parallel compression. Alternating single-stage compression cycle is approximate to conventional one-stage cycle. Hence when one evaporator operates to refrigerate, another has to be idle. Furthermore the compressor could only reach its highest efficiency under one of evaporator conditions. On the contrary, the last two types of dual-evaporator cycles can operate two parallel evaporators synchronously. Their compression types can be realized by one compressor with two cylinders, or by two individual compressors. Due to the limited system space, cost control of manufacturing and noise reduction, compressor with two cylinders is more appropriate.

In this case, it is necessary to carry out the next research. Reciprocating compressor, rotary compressor and scroll compressor are the most three frequent compressors in domestic appliances. In rotary compressor, two cylinders can be superposed to implement the function in demand. Besides, the superposition structure brings particular mechanical balance and only a little growth of compressor size.

2.2 Rotary Compressor for Dual-Evaporator Cycle

To realize two types of dual-evaporator cycles with rotary compressor, two-stage rotary compressor and dual-cylinder rotary compressor are proposed. Meanwhile the corresponding cycles with two types of rotary compressors are designed and shown in Figure 1a and 1b respectively.



a. TSCDE cycle with two-stage rotary compressor b. PCDE cycle with dual-cylinder rotary compressor

Figure 1: Schematic for a TSCDE cycle and PCDE cycle

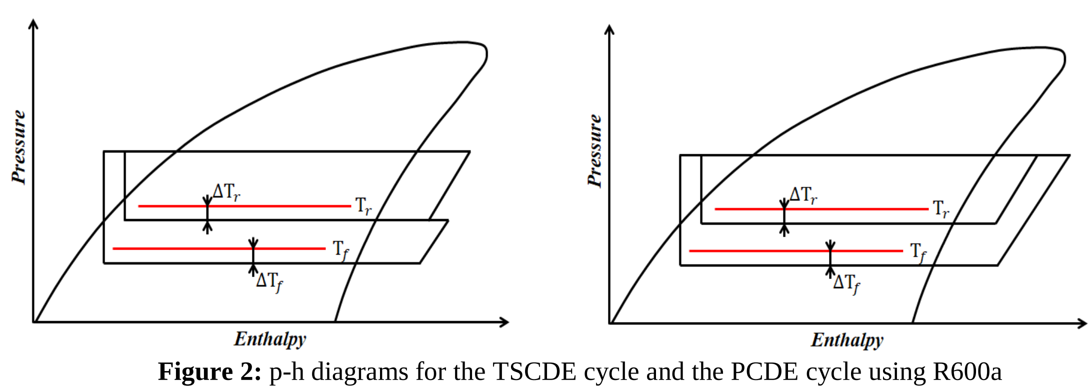


Figure 2: p-h diagrams for the TSCDE cycle and the PCDE cycle using R600a

In the two-stage compression dual-evaporator (TSCDE) cycle shown in Figure 1a, low pressure cylinder (LCYL) and high pressure cylinder (HCYL) are piled together and working for different compression missions. Meanwhile the non-adiabatic capillaries are placed in pipeline to constitute the internal heat exchangers and improve the cycle performance. Different from TSCDE cycle, in the parallel compression dual-evaporator (PCDE) cycle shown in Figure 1b, two cylinders suck refrigerant gas under different suction pressures but compress to the same discharge pressure.

The novel cycles are designed using R600a as refrigerant, due to its eco-friendliness and good property (Joybari *et al.*, 2013). Figure 2 shows the p-h diagrams for a TSCDE cycle and a PCDE cycle. The red lines mean the chamber temperatures. Both cycles have small temperature differences between evaporator and chamber ambient in each chamber. In addition, the pressure difference of low pressure cylinder in PCDE cycle is higher than that in TSCDE cycle, which causes higher compression work and lower volumetric efficiency.

2.3 Cycle and Compressor Analyses

Brief cycle performance analyses of TSCDE cycle and PCDE cycle are carried out based on cycle simulation under the conventional refrigerator conditions. Meanwhile the conventional one-stage (OS) cycle is taken as basis for comparison. The theoretical refrigeration coefficients for TSCDE and PCDE cycles can be expressed as

$$\varepsilon_{r,TSCDE} = \frac{Q_r + Q_f}{W} = \frac{m_r \cdot \Delta h_r + m_f \cdot \Delta h_f}{m_r \cdot \Delta h_{lc} + (m_r + m_f) \cdot \Delta h_{hc}} \tag{1}$$

$$\varepsilon_{r,PCDE} = \frac{Q_r + Q_f}{W} = \frac{m_r \cdot \Delta h_r + m_f \cdot \Delta h_f}{m_r \cdot \Delta h_{hc} + m_f \cdot \Delta h_{lc}} \tag{2}$$

The conditions and results for three cycles are shown in Table 1.

Table 1: The conditions and results for three refrigerator cycles

Cycles	OS	TSCDE	PCDE
Conditions			
Condensation temperature (℃)	40		
Refrigerating evaporation temperature (℃)	-25	-5	-5
Freezing evaporation temperature (℃)		-25	-25
Minimum ΔT for heat exchanger capillary (℃)	5		
Refrigerating capacity ratio	1:1		
Results			
Theoretical refrigeration coefficient (W/W)	3.12	3.89	3.88
Improvement percentage (%)		24.4	24.3

Figure 3 shows the variety of crankshaft torque in one circle for two types of rotary compressors above and one-stage rotary compressor. Due to the 180 phase difference between two cylinders, the torque loads are dislocation superposed. Consequently, vibration is greatly reduced than one-stage rotary compressor. Moreover the vibration of dual-cylinder rotary compressor is a little less than two-stage rotary compressor.

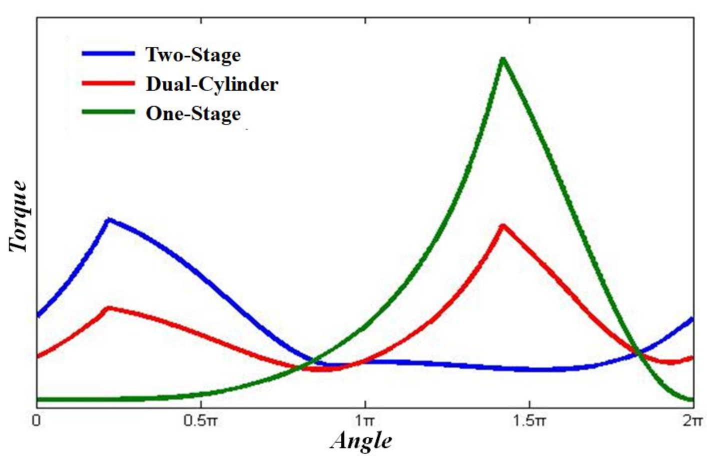


Figure 3: Variety of crankshaft torque for three types of rotary compressors

In general, rotary compressors realize the high performance of dual-evaporator cycles. The improvements of theoretical refrigeration coefficient for TSCDE cycle and PCDE cycle are up to 24.4% and 24.3% comparing with conventional one-stage cycle. Besides, the torque fluctuations are less because of the superposition of compressive loads, which is benefit to vibration and noise reduction, and higher mechanical efficiency.

3. R290 AIR-SOURCE HEAT PUMP CYCLES FOR COLD REGIONS

3.1 Air-Source Heat Pump System and R290

A well-designed heat pump system is, theoretically, the ideal solution for household heating to achieve energy saving and environmental protection for its high coefficient and heating capacity (Chua *et al.*, 2010). And air-source heat pump (ASHP) is the most conventional system for small or medium residential housing.

Conventional refrigerants used in heat pump are R22, R134a and R410A, which belong to HCFCs or HFCs with high ozone depletion potential (ODP) and global warming potential (GWP). They are going to be substituted out of environmental responsibility and policy. As natural refrigerant, R290 (Propane) is one of the most potential replacements in ASHP system for residential space heating. It is relatively stable and compatible with refrigeration equipment materials, miscible and soluble with many existing lubricating oils, and also nontoxic for human. Besides R290 possesses better thermal transfer coefficient and lower pressure drops, which lead to smaller size or higher efficiency of heat exchanger comparing with R22 (Eric, 2001).

However, the performance especially the heating capacity of ASHP declines rapidly with the ambient temperature reducing. Furthermore, with increase of pressure ratio causing by low ambient temperature, the discharge temperature increases and threatens the safe operation of compressor (Redón *et al.*, 2014).

To solve these problems, the most usual techniques for performance enhancing are internal heat exchanger, vapor injection and variable speed compression, and for discharge temperature control are liquid injection, oil injection and two-stage compression with intermediate cooling(Long and Yiqiang, 2018). Meanwhile those techniques are almost all researched basing on conventional refrigerants like R22, R134a and R410A. The researches of R290 ASHP system for cold regions are few, and in urgent need to be conducted.

3.2 Problem Research on R290 Air-Source Heat Pump Cycle

The major problem of R290 as refrigerant is flammability, which has led to higher demand of safety for commercial systems and limitation of charge quantity for domestic systems (Palm, 2008). And the charge limitation would be able to lead to the insufficiency of refrigerant in cycle circulation.

On the other side, the problems of ASHP with conventional refrigerants get somewhat different when using R290. Due to the large-gradient isentropic curve, the high discharge temperature problem causing by high pressure ratio would not be destructive. With the adoption of vapor injection technique to improve system performance, the discharge temperature is further reduced, which would cause problems in other aspects.

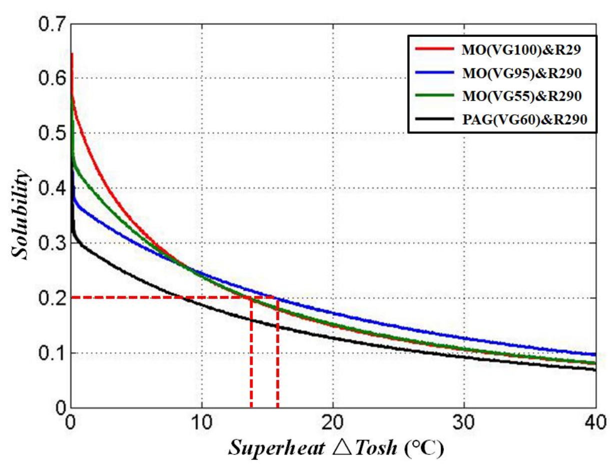


Figure 4: Effect of ΔTosh on solubility of R290 in four kinds of lubricant oil

Synthesizing the problems showing above, the selection of lubricant oil for compressor or system becomes a comprehensive problem. For rotary compressor with high back pressure, which is widely applied in domestic heat pump air conditioner, the low discharge temperature causes the higher solubility of refrigerant in oil sump, so that the lack of refrigerant gets more serious. In the vapor injection heat pump system, the discharge temperature is lower further, and the insufficiency phenomenon is more serious.
Figure 4 shows the relationship between ΔTosh and solubility of R290 in four kinds of lubricant oil in common use, including three kinds of mineral oil (MO) and one kind of PAG oil. To limit the solubility less than 20%, the ΔTosh of MO needs to attain about 15℃. The ΔTosh means the temperature difference between equivalent temperature of oil sump and saturation temperature for discharge pressure. And according to correlative research, equivalent temperature for oil sump is approximately equal to isentropic discharge temperature. However, the discharge temperature tends to be not high enough causing by physical property of R290.

To sum up, in order to solve the problems above, the cycle needs to be redesigned to raise the discharge temperature and still keep the high performance, or the lubricant oil needs to be researched further to adapt to R290 heat pump. By the way, for PAG oil, the demand of discharge temperature is much lower, but the insolubility will cause the oil return problem which has not been studied fully.

3.3 Techniques and Cycle Performance Analyses

In order to raise the discharge temperature of heat pump system using R290 as refrigerant, two techniques are adopted respectively. The novel cycles will be introduced in detail of schematics and p-h diagrams. And then some brief performance analyses will be conducted.

The first effective method is proposing an internal heat exchanger in vapor injection heat pump cycle. A vapor injection heat pump cycle using economizer as vapor generator is taken as an example. The schematic and p-h diagram for internal heat exchanger vapor injection heat pump (IHE-VIHP) cycle are shown in Figure 5. The red line and the blue line in p-h diagram indicate the heat exchange relationships of internal heat exchanger (IHX) and economizer (ECO) respectively.

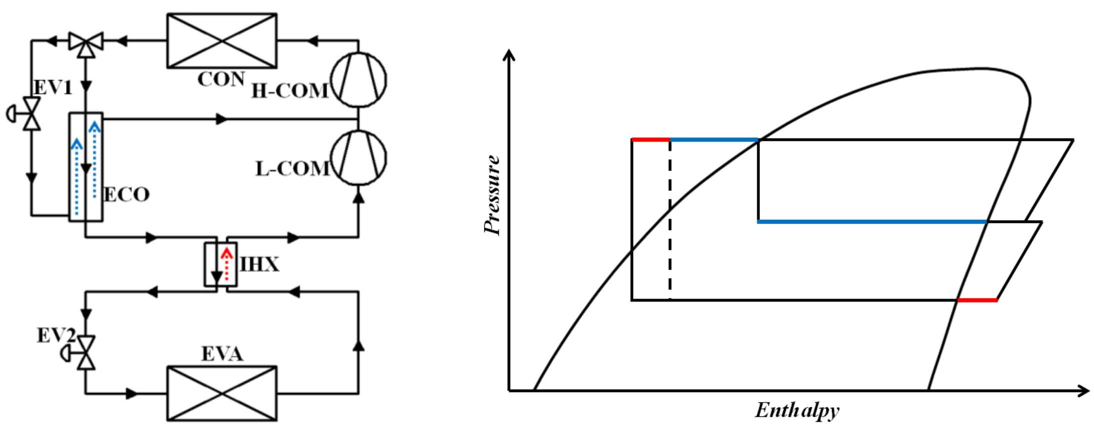


Figure 5: Schematic and p-h diagram for IHE-VIHP cycle

Another technique is based on conventional vapor injection heat pump cycle using economizer as vapor generator. The temperature of injection vapor is controlled superheating condition close to outlet temperature of condenser, distinguishing from saturated vapor injection in usual. Figure 6 shows the schematic and p-h diagram of superheating vapor injection heat pump (SHVIHP) cycle, and indicates that the injection vapor is fully superheated at the outlet of economizer. This cycle implements particular control strategy on conventional cycle, which can effectively avoid the complexity and cost increase.

Cycle performance analysis is based on cycle simulation of the theoretical heating coefficient (ε_h), volumetric heating capacity (qv) and isentropic discharge temperature (Td). Meanwhile three conventional heat pump cycles are taken as reference cycles, including single-stage heat pump (HP) cycle, single-stage heat pump with internal heat exchanger (IHE-HP) cycle and vapor injection heat pump (VIHP) cycle using economizer as vapor generator.

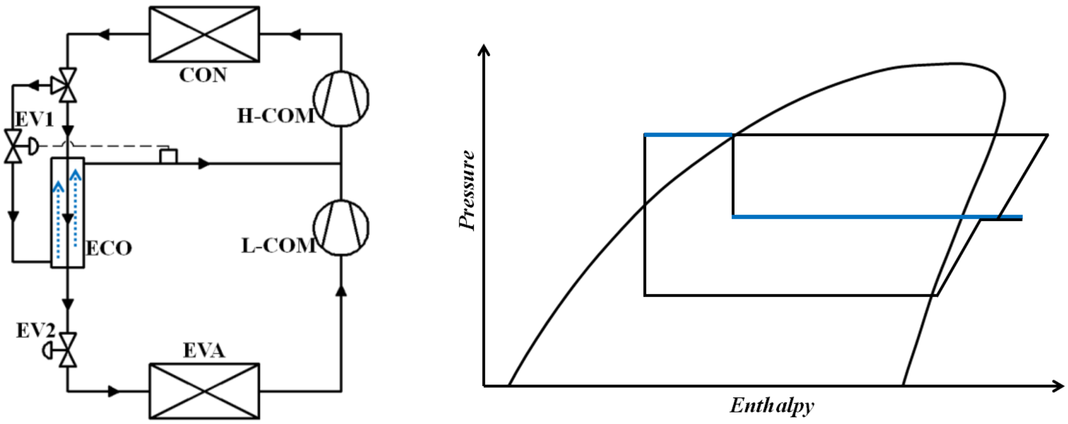


Figure 6: Schematic and p-h diagram for SHVIHP cycle

Table 2 lists the thermodynamic conditions based on household heating in cold region, and the simulation results as follow. Different conditions are expressed by different evaporation temperatures (Te). In addition, the Rqv means relative volumetric heating capacity comparing with HP cycle. The dqv is the decline ratio of qv under -30℃ Te, comparing with 0℃. The theoretical heating coefficients (ε_h) for HP and IHX-HP can be expressed as

$$\varepsilon_h = \frac{Q_{cond}}{W} = \frac{m \cdot \Delta h_{cond}}{m \cdot \Delta h_c} \tag{3}$$

And the ε_h and qv for all vapor injection cycles can be expressed as

$$\varepsilon_h = \frac{Q_{cond}}{W} = \frac{\left(1 + x_{inj}\right) \cdot m \cdot \Delta h_{cond}}{m \cdot \Delta h_{lc} + \left(1 + x_{inj}\right) \cdot m \cdot \Delta h_{hc}} \tag{4}$$

$$q_v = \frac{Q_{cond}}{V} = \frac{\left(1 + x_{inj}\right) \cdot m \cdot \Delta h_{cond}}{v \cdot m} \tag{5}$$

Table 2: Thermodynamic conditions and simulation results

Conditions		
Ambient temperature (℃)	7	-20
Evaporation temperature (℃)	0	-30
Condensation temperature (℃)	50	
Saturation temperature for injection vapor (℃)	20	5
Superheat for non IHE system (℃)	3	
Minimum temperature difference in IHX and ECO (℃)	5	

Results						
Te		HP	IHX-HP	VIHP	IHX-VIHP	SHVIHP
0℃	ε_h (W/W)	5.01	5.29	5.50	5.55	5.51
	qv (kJ/m^3)	3113.5	3330.4	3918.6	3940.6	3916.3
	Td (℃)	58.1	97.0	57.0	69.1	63.4
-30℃	ε_h (W/W)	2.88	3.20	3.36	3.40	3.38
	qv (kJ/m^3)	1222.7	1361.6	1723.9	1721.6	1721.1
	Rqv (%)		11.4	41.0	40.7	40.7
	Td (℃)	65.0	133.1	61.8	75.7	71.8
	dqv (%)	-60.7	-59.1	-56.0	-56.3	-56.1

As is shown in Table 2, the ε_h and qv for IHE-VIHP and SHVIHP cycle keep the highest level among five cycles. As compared to HP cycle, the improvements of ε_h are up to 17.8% and 16.8% respectively when evaporation temperature is -30℃, and the improvements of qv are both up to 40.7%.

In addition, the Tds of IHE-VIHP and SHVIHP are almost kept between 65-75℃ for proper solubility and viscosity. The Tds of HP and VIHP cycles are close to the condensation temperature, which would cause high solubility and low viscosity. And to achieve higher performance, the Td of IHE-HP is much higher, which meanwhile would cause safety problems.

In general, the IHE-VIHP and SHVIHP cycles are able to control the discharge temperature for proper solubility of R290 in lubricant oil. Meanwhile they own the highest performance including ε_h and qv, and the improvements are up to 16.8% and 40.7% comparing with conventional HP cycle. By the way, the performance of IHE-VIHP cycle can be further improved by optimum medium temperature design. And Reverse-cycle defrosting can also be realized in IHE-VIHP with four-way reversing valve.

4. NOVEL CO2 HEAT PUMP CYCLE FOR ELECTRIC VEHICLE

4.1 Heat Pump System for Electric Vehicle and CO2

Electric vehicle (EV) is a potential replacement for conventional vehicle with internal combustion engine. However the energy density of battery unit is still not high enough comparing with fossil fuels, which limits the widespread use of EVs.

Heating ventilation air conditioning (HVAC) system is an essential portion of vehicle to insure safe driving and maintain passengers comfortable. However it consumes large part of electric quantity of EV, especially in winter. Besides proportional fresh air is sucked into HVAC system and mixed with return air, to control air temperature, humidity, and oxygen concentration, so that extra heating load is brought, and the system performance reduces further.

For positive temperature coefficient (PTC) electric heater which is widely employed in EVs, correlational research indicated that the reduction percentage of driving range was up to 50.0% (Jong *et al.*, 2013). Air source heat pump (ASHP) is the alternatively choice with higher efficiency (Zhaogang, 2014). However the performance decline of ASHP under low ambient temperature conditions has become the major problem due to multiple ambient for vehicles.

Carbon dioxide (CO2) is a kind of natural refrigerant with 0 ODP and 1 GWP usually using in ASHP. As the critical temperature is as low as 31.2℃, refrigerant for heating is working in super-critical state, which owns extremely good thermodynamic properties. Besides the performance decline problem of ASHP under low ambient temperature is slighter than system with other refrigerants. However the problems of CO2 as refrigerant are high pressure ratio and difference, and high discharge temperature.

Therefore, it is necessary to redesign the ASHP for EV, combining the features of CO2 and HVAC system. Moreover, Vorster and Meyer’s (2000) researches indicated that wet compression can increase the COP of CO2 systems, which can be employed in EV for further improvement.

4.2 Preheating and Wet Compression Heat Pump Cycle

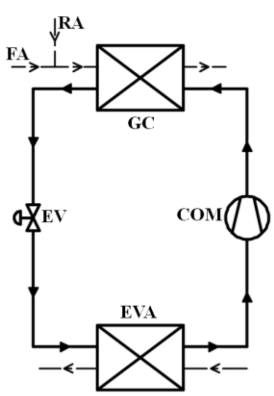


Figure 7: Schematic for HP cycle

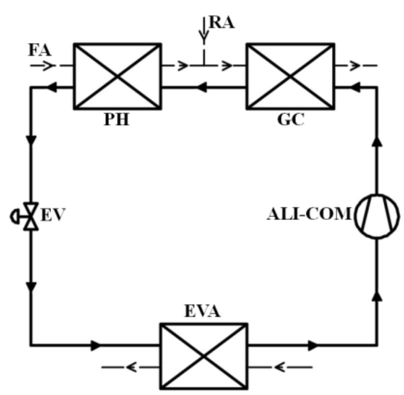


Figure 8: Schematic for PHWC-HP cycle

Conventional single-stage heat pump (HP) cycle using CO2 as refrigerant is taken as the basic cycle for novel heat pump cycle newly designed. The schematic for HP cycle is shown in Figure 7. Note that the GC is used to heat the mixture air composed of fresh air outside and return air inside.

Figure 8 shows the schematic for the novel heat pump cycle named as preheating and wet compression heat pump (PHWC-HP) cycle. Distinguishing from basic HP cycle, novel cycle has a preheater (PH) to preheat the fresh air, and an anti-liquid impacting compressor (ALI-COM) to realize wet compression. In PHWC-HP cycle, fresh air is heated by preheater firstly. And after mixing, the mixture air is heated by GC. By the way, ATLI-COM operates basing on the advantages of rotary compressor or scroll compressor. Both types of compressors have crescent-shaped compression chamber and radial flexibility to reduce the liquid impact.

Figure 9 shows the p-h diagrams for HP cycle and PHWC-HP cycle, in dashed line and real line, respectively. Because of the preheating of fresh air, the enthalpy potential is entirely utilized, and the novel cycle possesses higher heating capacity. Moreover, the compression work is lower due to wet compression.

Figure 10 shows the T-h diagram for PHWC-HP cycle. The blue line means the constant line of discharge pressure, and is divided into two parts by separation point (SP) with different temperature gradients. The red lines mean the varieties of different air being heated. The steep part of blue line is appropriate to heat the fresh air (FA) from ambient temperature to return air temperature. Meanwhile the gentle part is appropriate to heat the mixture air (MA) with small temperature difference and large volume.

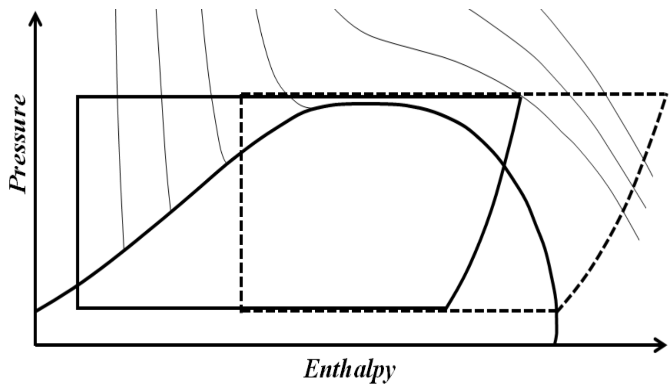


Figure 9: p-h diagrams for HP and PHWC-HP cycles

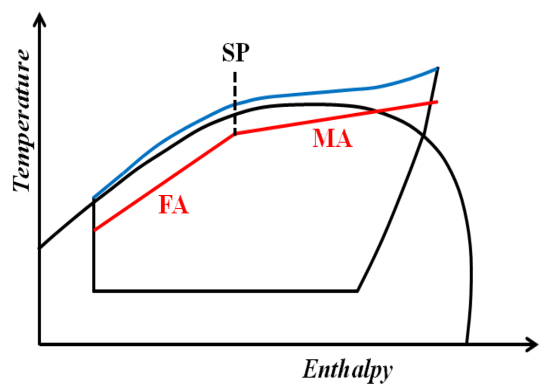


Figure 10: T-h diagram for PHWC-HP cycle

4.3 Performance Analyses and Evaluation

The performance analysis is based on cycle simulation in two steps. In the first step, the theoretical heating coefficient (ε_h) and volumetric heating capacity (qv) for HP and PHWC-HP cycles are simulated under saturation compression condition to evaluate the effect of preheater. And next step, PHWC-HP cycle is simulated under wet compression condition. The ε_h and qv for novel cycle can be express as

$$\varepsilon_h = \frac{Q_{gc} + Q_{ph}}{W} = \frac{m \cdot (\Delta h_{gc} + \Delta h_{ph})}{m \cdot \Delta h_c} \tag{6}$$

$$q_v = \frac{Q_{gc} + Q_{ph}}{V} = \frac{m \cdot (\Delta h_{gc} + \Delta h_{ph})}{v \cdot m} \tag{7}$$

Table 3: Saturation compression condition for the first step

Ambient temperature (℃)	-20~10
Evaporation temperature (℃)	-30~0
Discharge pressure (MPa)	7.5
Suction superheat (℃)	0 (saturation state)

Figure 11 shows the simulation results for the first step. Either the ε_h or the qv for PHWC-HP cycle keeps higher. The maximum improvements are both up to 40% when evaporation temperature is -30℃ because of the same compression process. As evaporation temperature is from 0℃ to -30℃, the decline percentages of ε_h for PHWC-HP and HP, are 39.8% and 47.5% respectively. And correspondingly, the decline percentages of qv are 41.0% and 48.6%.

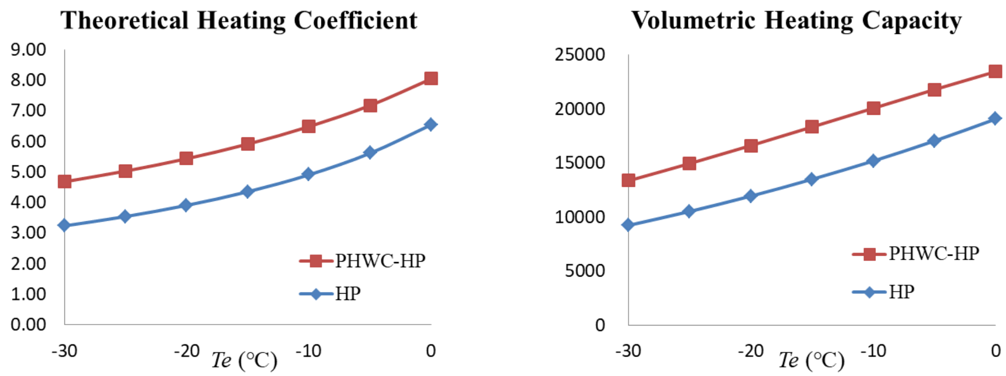


Figure 11: Effect of Te on ε_h and qv for HP and PHWC-HP cycles

For the second step, the condition of suction superheat is converted into suction liquid ratio from 0 to 0.2, and the evaporator temperature is set as -25℃ based on the first step condition. Figure 12 shows that with the suction liquid ratio increasing from 0 to 0.2, the ε_h increases by 9.1%, but the qv decreases by 5.7%.

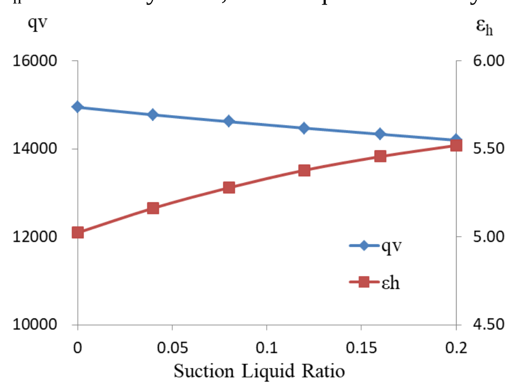


Figure 12: Effect of wet compression on ε_h and qv for PHWC-HP cycle

In general, these results explained well that the application of preheater can efficiently improve the ε_h and qv up to at least 40% for CO2 heat pump system. Besides wet compression improves the ε_h further, but causes the slight decrease of qv. By the way, wet compression can make contribution to discharge temperature control. And the suction liquid ratio can be adjusted with the change of fresh air ratio to attain optimum performance.

5. CONCLUSIONS

In this paper, the authors introduce the development of vapor compression cycles in three kinds of devices. Novel cycles are newly designed and necessary simulation or analysis is conducted comparing with each basic cycle.

For dual-evaporator refrigerator cycle using R600a, rotary compressors realize its high performance. The improvements of theoretical refrigeration coefficient comparing with one-stage cycle are up to 24.4% and 24.3% for different compressor structures under the conventional refrigerator condition. The torque fluctuations of rotary compressor using R600a are less than one-stage compressor, and are benefit to vibration and noise reduction.

For the IHE-VIHP cycle using R290 for domestic heat pump devices, the maximum improvements of theoretical heating coefficient and volumetric heating capacity are up to 16.8% and 40.7% respectively comparing to three basic cycles, when the evaporation temperature is -30°C. The discharge temperature is able to be controlled by internal heat exchanger for the proper solubility of R290 in lubricant oil.

For the PHWC-HP cycle using CO2 for electric vehicles, the potential energy is utilized to preheat the fresh air before mixing. The improvements of ε_h and volumetric heating capacity are up to at least 40% comparing with conventional heat pump cycle, when the evaporation temperature is -30℃. Besides, the wet compression improves the ε_h further and controls the discharge temperature.

These novel cycles newly designed in this paper are expected to be researched for further improvement, be investigated based on experiments, and finally be employed to relevant devices.

NOMENCLATURE

Subscript

COM	compressor	ε_r	theoretical refrigeration coefficient
CON	condenser	ε_h	theoretical heating coefficient
EVA	evaporator	qv	volumetric heating capacity
CAP	capillary	R-	refrigerating
EV	expansion valve	F-	freezing
SV	switching valve	H-	high pressure
GC	gas cooler	L-	low pressure
PH	preheater	TS-	two-stage
ECO	economizer	DC-	dual-cylinder
IHX	internal heat exchanger	ALI-	anti-liquid impacting
CYL	cylinder		

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