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System design and analysis of the trans-critical carbon-dioxide automotive air-conditioning system

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Abstract: As an environmentally harmless and feasible alternate refrigerant, CO_2 has attracted worldwide attention, especially in the area of automobile air-conditioning (AAC). The thermal property of CO_2 and its trans-critical refrigeration cycle is very different from that of the traditional CFC or HCFC system. The detailed process of CO_2 system thermal cycle design and optimization is described in this paper. System prototype and performance test bench were developed to analyze the performance of the CO_2 AAC system.

Key words: Automotive air-conditioning(AAC), Carbon-dioxide, Trans-critical cycle.
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INTRODUCTION

After the Kyoto Protocol was signed, the worldwide refrigeration and air-conditioning industry faced up to great changes in the environmental problem. For automotive air-conditioning (AAC), the current refrigerant, R-134a, has been regarded as restricted-use-refrigerant by the Kyoto Protocol because of its high potential for global warming. Finding a new alternative refrigerant is an urgent issue for the AAC industry since the R-134a refrigerant in the AAC system easily leaks. The new refrigerant should be environmentally problem free, effective and easily available.

After 1992, when Lorentzen brought forward the theory of CO_2 trans-critical refrigeration cycle and built a CO_2 AAC prototype for test (Pettersen *et al*., 1993), the CO_2 system attracted worldwide attention. Current research in many countries showed that the CO_2 system had refrigeration performance very close to that of R-12 or R-134a (Lorentzen, 1995), and many other benefits as a refrigerant, such as no environmental harm, easy availability, and low cost. University of Illinois tests showed that the COP (coefficient of performance) of CO_2 AAC system was fairly equal or superior to the R134a system under some special work conditions (Yin *et al*., 1999). Soak and cool down test in vehicle also showed that the CO_2 AAC system had superior performance (Takahashi, 2000).

However, the trans-critical cycle of the CO_2 system is a different concept from the traditional vapor compression refrigeration cycle, and has different refrigerant property, thermal cycle and work pressure. Detailed system thermal cycle analysis, design and optimization are introduced in this paper. A domestic first CO_2 AAC system prototype and performance test bench was built for system experimental analysis.

DESCRIPTION OF TRANS-CRITICAL AAC SYSTEM

Fig.1 is the concept of the CO_2 AAC system and its thermal cycle in the log P-H diagram. The function of the compressor, evaporator and valve is almost the same as that of the traditional AAC system. High temperature and high pressure refrigerant is cooled down in a gas cooler. Similar process happens in the condenser, but in the gas cooler, the temperature of CO_2 is higher than its critical temperature, 31.1 ℃, so the CO_2 is always in gas state without condensation. That is the greatest difference between the CO_2 system and the traditional R134a system.

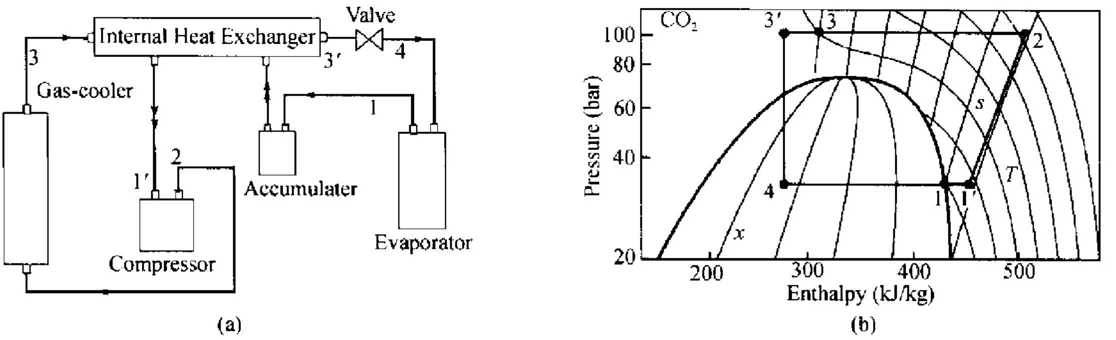

Fig.1 Concept of CO_2 trans-critical AAC system and its thermal cycle Log P-H Diagram

(a) CO_2 AAC system layout; (b) CO_2 AAC system thermal cycle in Log P-H diagram

1 – Evaporator outlet; 1′ – Compressor inlet; 2 – Compressor outlet;

3 – Gas cooler outlet; 3′ – Valve inlet; 4 – Valve outlet

Another character of the CO_2 system is that the liquid fraction behind the valve is smaller than that in the R134a system, so an internal heat exchanger is utilized to cool down the refrigerant to the valve with cold gas from the accumulator. The liquid fraction at the evaporator inlet increases and so does the system COP.

SYSTEM THERMAL CYCLE DESIGN AND OPTIMIZATION

1. Input parameter

The design is for a car AC, the refrigeration capacity is 5 kw, and the design condition is the reference of standard JISD1618-86, as follows:

Evaporator air inlet dry-bulb temperature: 27 ℃;
Evaporator air inlet web-bulb temperature: 19.5 ℃;
Gas cooler air inlet temperature: 35 ℃;
Compressor rotation speed: 1800 rev/min;
Air velocity of the gas cooler: 4.5 m/s;
Air velocity of the evaporator: 2.5m/s;
Evaporation temperature: 0 ℃;
Temperature difference of gas cooler refrigerant outlet and air inlet: 5 ℃;
Assumed compressor entropy efficiency: 0.7;
Volume efficiency: 0.6.

2. Optimization of gas cooler pressure

In the gas cooler, the state of CO_2 is supercritical; the temperature and pressure are independent; and the isothermal curve above critical point is "S" shaped, so there is one optimal pressure in the gas cooler. When other parameters are identical, the system COP reaches maximum at a certain gas cooler pressure (Fig.2).

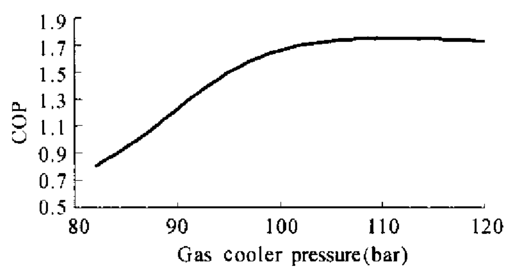

Fig.2 COP vs pressure in gas cooler

The optimal pressure in the gas cooler differs inversely with the heat capacity of the internal heat exchanger. Table 1 lists some calculation results of optimal gas cooler pressure vs. internal heat exchange.

Table 1 shows that larger internal heat exchanger capacity means lower optimal gas cooler pressure, and higher compressor outlet temperature. Too high compressor outlet temperature will cause problem of hose and oil. The data marked "*" in Table 1 is chosen as design parameter of the CO_2 thermal cycle. Table 2 summarized the refrigerant parameters in the designed thermal cycle.

DEVELOPMENT OF SYSTEM PROTOTYPE AND PERFORMANCE TEST BENCH

Based on current domestic material and manufacture level, the CO_2 trans-critical AAC system prototype was designed and developed.

Table 1 Optimal gas cooler pressure vs heat capacity of internal heat exchanger

Internal heat exchange capacity (kJ/kg)	0	2	20	25*	40	54
Optimal gas cooler pressure (bar)	106	105	103	103	101	100
System COP	1.89	1.9	1.95	1.97	2.01	2.05
Compressor outlet temperature (℃)	99.36	100.2	115.5	120.7	134.1	148.2

Table 2 System thermal cycle parameters

Position	T (℃)	P (bar)	Enthalpy (kJ/kg)	Fraction
Outlet of evaporator	0	34.8147	732.95	1
Inlet of compressor	16.11	34.8147	757.95	Super heat
Outlet of compressor	120.6	103	832.72	Super heat
Outlet of gas cooler	40	103	610.98	Super heat
Inlet of evaporator	0	103	585.98	Super heat
Inlet of expansion valve	35.08	34.8147	585.98	0.38

1. Heat exchangers

Based on the property of CO_2, small inner diameter tubes are good for heat transfer and pressure resistance. Tube-fin type was chosen since its material is easily available in China. Internal heat exchanger is reversed-flow tube-in-tube heat exchanger. Fig.3 is the photo of the gas cooler and evaporator.

Fig.3 Photo of gas cooler and evaporator

2. Compressor

A reciprocating compressor was chosen for the CO_2 system because of its easy control of leakage through the gap between the piston and the cylinder (Jurgen *et al*., 1998). Fig.4 is the structure of such kind of compressor.

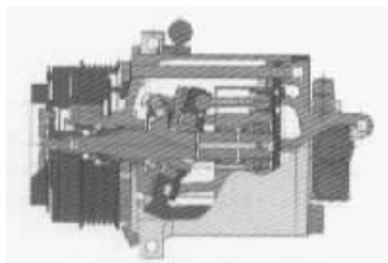

Fig.4 Structure of reciprocating CO_2 compressor

SYSTEM TEST BENCH AND TEST RESULT

1. CO_2 AAC system performance test bench

The CO_2 system working pressure is much higher than that of the R134a system. So the only special equipment for such a test bench is high-pressure sensor. The DRUCK 0-160 bar and 0-300 bar pressure sensors were chosen for low and high-pressure measurement. The air side equipment was the same as standard calorimeter used for vehicle AAC systems. Fig.5 gives the layout of the CO_2 system performance test bench.

2. Test result

Table 3 lists the air side test results of the CO_2 AAC system at compressor speed of 1800 rev/min:

Table 3 Air side test results

Parameter	Test data
Evaporator air inlet dry bulb temperature (℃)	29.0
Evaporator air inlet wet bulb temperature (℃)	21.5
Evaporator air outlet dry bulb temperature (℃)	9.0
Evaporator air outlet wet bulb temperature (℃)	7.5
Evaporator air flow (m^3/s)	0.101
Evaporator air side capacity (kW)	4.46
Gas cooler air inlet temperature (℃)	36.6
Gas cooler air outlet temperature (℃)	41.8
Gas cooler air side capacity (kW)	7.20

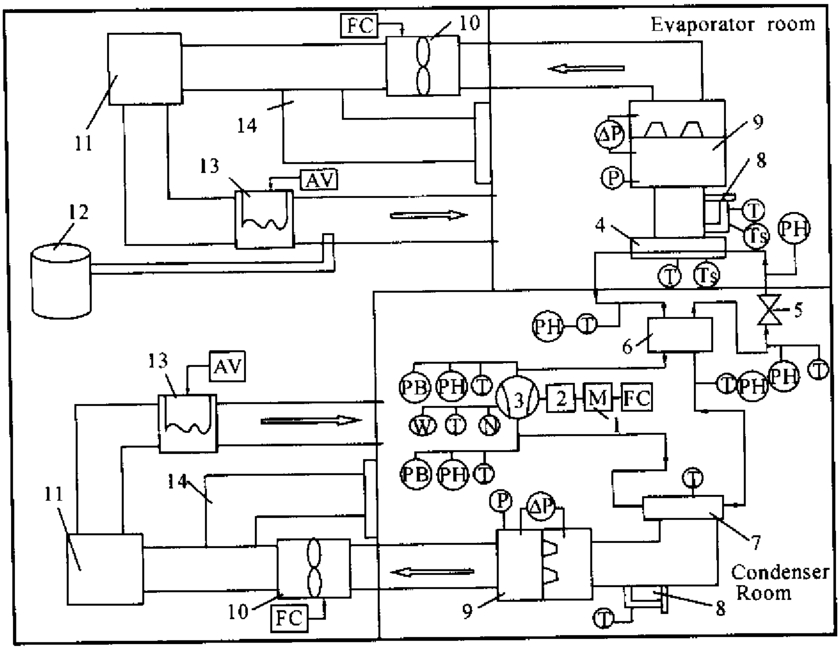

Fig.5 Performance test bench of CO_2 system

1. Motor; 2. Torque meter; 3. Compressor; 4. Evaporator; 5. Expansion valve; 6. Internal heat exchanger; 7. Gas cooler; 8. Sample tube; 9. Nozzle pressure meter; 10. Fan; 11. Air-conditioner; 12. Humidity controller; 13. Heater; 14. Air tunnel

Table 4 lists the refrigerant side results of the CO_2 AAC system

Table 4 Refrigerant side test results

Position	T (℃)	P (bar)
Outlet of evaporator	1.8	34.5
Inlet of compressor	10.2	31.1
Outlet of compressor	107.1	100.6
Outlet of gas cooler	44.6	96.3
Inlet of evaporator	4.2	37.2
Inlet of expansion valve	40.1	94.3

The power of the compressor motor is 2.7 kw. Assuming the efficiency of the power transfer from the motor to the compressor is 0.97, the system COP is around 1.7. The data is different from designed value, because of:

1) Pressure drop in heater transfer and pipes;

2) Effect of oil in system;

3) Different compressor volume efficiency and entropy efficiency from assumption in design stage.

CONCLUSIONS

Development of the CO_2 AAC system is introduced in this paper. The CO_2 thermal cycle is optimized based on the character of CO_2 trans-critical cycle. The domestic first CO_2 system prototype and test bench were developed. The capacity in the performance test reached 4.5 kW under the design condition, which was close to the design target of 5 kw.

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