

CO₂ Transcritical Cycle for Ground Source Heat Pump

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ABSTRACT

This paper gave a schematic of the CO₂ transcritical cycle for ground source heat pump system and compared it with the cycles of R22 and R134a. The volumetric efficiency and isentropic efficiency is also taken into consideration in this comparison. The results indicated that CO₂ ground source heat pump system is equivalent to the conventional refrigerants cycles in coefficient of performance in high temperature heat sink. The paper analyzed the operating characteristics of CO₂ ground source heat pump system when both the temperature and the flow of water are changed and also discussed the capacity control methods of the CO₂ ground source heat pump system.

1. INTRODUCTION

Ground source heat pump systems(GSHP) have become increasingly popular in heating and cooling applications. These systems have been recognized to provide viable, environment-friendly alternatives to conventional unitary. In recent years, with the keenness of the conflict between resource and environment, the research and application of ground source heat pumps develop quickly.

Cane et al (2000) and Martin et al (2000) surveyed commercial and institutional buildings with ground source heat pumps systems throughout the United States and Canada and investigated maintenance and service costs. The ground loop heat exchangers used in closed loop systems can be either horizontal or vertical in configuration. Most of them are vertical. Vertical ground loop heat exchangers typically consist of high density

polyethylene pipe U-tubes inserted into deep boreholes. The ground source heat pump systems were applied widely in heating and cooling for building areas from 1394m² to 33444m² and the capacities of heat pump units are from 25ton(88kW) to 1400ton(4924kW). Typical per unit capacity is about 12kW. The air-conditioning system circuits can be water to water system or a water to air system. Most of these systems have heat recovery units. Contrast to the conventional air-conditioning systems, the ground source heat pump can make significant contributions to reductions in electrical energy usage and allow more effective demand-side management.

However, when ground source heat pump systems are applied in air-conditioning systems, the higher condensing temperature and great temperature difference are needed. This is the main obstacle for heat pump cycle with the chlorofluorocarbons (CFCs), hydrochlorfluorocarbons(HCFCs) and hydrofluorocarbons(HFCs) synthetic refrigerants. The volumetric refrigeration capacity of them, such as R22 and R134a, falls rapidly with the increase of the condensing temperature. Furthermore, HFCs are substances which are not reactive with ozone, but they have a high global warming potential and must be phase-out by the 'Kyoto Protocol'.

As consequence, reusing the natural substances as working fluid is a very safety choice for the environment and human health in a long term, which is called as "complete solution" for refrigerant alternative by G. Lorentzen(1993). Carbon dioxide is a substance which has a background of successful use as a refrigerant and has been proved to be not harmful to world environment. In the fields of heat pumps, Carbon dioxide is regarded as the most potential candidates for its good properties.

The high rejection temperature of the compressor and the great temperature glide of the gas cooler in the CO₂ transcritical cycle is very suitable for heat recovery. Ground source heat pumps with conventional refrigerants can only get hot water temperature no higher than 60°C, whereas CO₂ transcritical cycle heat pumps can supply 90°C water or even higher. So the later one has a wide range of application. It can be said that CO₂ is an ideal refrigerant for ground source heat pump systems.

In recent years, there were many investigations about the CO₂ transcritical cycle. Pettr Neksa et al (1998) reported the CO₂ heat pump water heater. Robinson and Groll (1998) simulated and analyzed the single-stage compression CO₂ transcritical cycle with turbine. Rozfentsev and Wang (2001) focused on the some design features of a CO₂ air conditioner. F.Kauf (Kauf 1999) and S.M.Liao et al (Liao 2000) provided the correlation of the optimal rejection pressure in the CO₂ transcritical cycle respectively. The authors of this paper(1998,2001) also studied on the CO₂ transcritical cycle in water to water heat pump systems and the combination system of the desiccant cooling and the CO₂

transcritical cycle.

This paper analyzed the CO₂ transcritical cycle ground source heat pump system with the U-tube ground heat exchanger and compared it with the R22 and R134a system. The effects of the volumetric efficiency and isentropic efficiency are taken account into this comparison. The operating characteristics of the CO₂ transcritical cycle ground source heat pump system are also studied when the outlet temperature and mass flow of hot water is changed

2. CYCLE ANALYSIS

2.1. SCHEMATIC OF CARBON DIOXIDE GSHP

The schematic of the CO₂ transcritical cycle ground source heat pump system is showed in Fig.1.

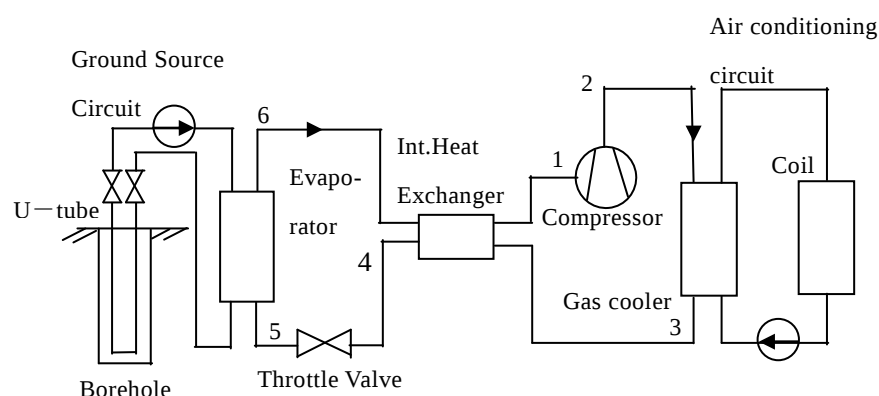


Fig.1 flow chart of the CO₂ transcritical cycle ground source heat pump

This system is composed of three parts, which are a closed loop systems with an U-tube ground heat exchanger, a CO₂ transcritical cycle heat pump unit and the terminal units of air-conditioning system. The U-tube pipe diameters are in the range from 20mm to 32mm and the U-tube are accordingly closed spaced in the borehole. The depth of boreholes are in the range from 30m to 120m. The boreholes have typical diameters of 75mm to 150mm. The working fluid of the ground source circuit is water with outlet temperature 25—35°C in summer and 0—10°C in winter. The CO₂ transcritical cycle system has one compressor and an internal heat exchanger. The optimum operating pressure of this system is affected by the temperature of the gas cooler outlet. Depending on the customers' needs, the heat pump units can combine with the terminal units of air-conditioning system as a CO₂- water system or a CO₂-air one.

The media in terminal units can be cooled passing through the evaporator of heat pump in summer and heated by the gas cooler in winter. The closed loop ground heat

exchanger rejects heat of the gas cooler to the soil in summer and supplied heat to the evaporator of heat pump as a low temperature heat source in winter.

2.2 COEFFICIENT OF PERFORMANCE

The coefficient of performance of the CO₂ transcritical cycle ground source heat pump systems, as shown in Fig.1, is calculated and analyzed in the design conditions of summer and winter and the comparison of heat pump cycles of CO₂, R22 and R134a are carried out at the same conditions.

According to the characteristics of ground source heat pumps, the evaporating temperature is set to be 0°C in winter and 5°C in summer. The coefficient of performance of the heat pumps are investigated when the outlet temperature of the CO₂ gas cooler and condensing temperature of R22 and R134a are variable with the water temperature of the gas cooler or condenser

$$\text{The coefficient of performance of heat pump in summer is } cop = \frac{q_e}{w} \quad (1)$$

$$\text{The coefficient of performance heat pump in winter is } cop_h = \frac{q_h}{w} \quad (2)$$

The change of volumetric efficiency and isentropic efficiency, caused by the variety of compression ratio, is taken account into the analysis of the coefficient of performance. The compressor is assumed to be a reciprocating type.

The volumetric efficiency and isentropic efficiency is calculated in the light of the equation (3) and (4) for the CO₂ transcritical cycle.

$$\text{The volumetric efficiency is } \lambda = 0.994 - 0.108 \left(\frac{p_r}{p_s} \right)^{0.714} \quad (3)$$

$$\text{The isentropic efficiency is } \eta = 0.977 - 0.092 \left(\frac{p_r}{p_s} \right)^{0.714} \quad (4)$$

The volumetric efficiency and isentropic efficiency is calculated in the light of the equation (5) and (6) for the R22 and R134a cycle.

$$\text{The volumetric efficiency is } \lambda = 0.94 - 0.085 \left[\left(\frac{p_r}{p_s} \right)^{\frac{1}{m}} - 1 \right] \quad (5)$$

$$\text{The isentropic efficiency is } \eta = \eta_i \cdot \eta_m = (\lambda_T + ct_0) \cdot \eta_m \quad (6)$$

Here, c=0.0025, η_m — efficiency of friction, it is assumed to be 0.9.

$$\lambda_T = \frac{T_s}{aT_c + b(T_s - T_e)} \quad a=1 \sim 1.15, \quad b=0.25 \sim 0.8 \quad (7)$$

In order to unify the comparative standard of different refrigerants cycles, the equivalent condensing temperature is introduced to the analysis of the performance coefficient.

$$T_{eq,c} = \frac{\int_c T ds}{\int_c ds} = \frac{\int_c T ds}{\Delta s} \quad (8)$$

The optimal rejection pressure of CO₂ transcritical cycle is calculated according to the equation (9) .

$$p_{opt} = (2.778 - 0.0157t_e)t_c + (0.381t_e - 9.34) \quad (9)$$

2.3 RESULTS

The relation between the equivalent condensing temperature and the COP in summer and COP_h in winter of four heat pump cycles is shown in Fig.2 and Fig.3, which are the single-stage compression CO₂ cycle with a internal heat exchanger, the single-stage compression CO₂ cycle with an expander, R22 cycle and R134a cycle. The changes of volumetric efficiency and volumetric refrigeration capacity of R134a with various condensing temperature are demonstrated in Fig.4.

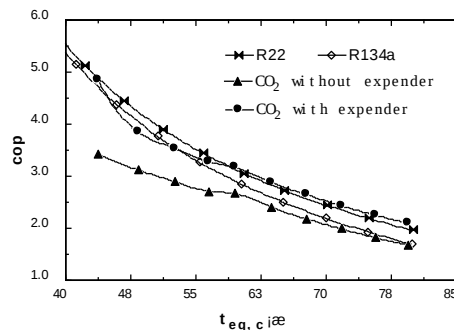


Fig.2 Relation between COP in summer and equivalent temperature

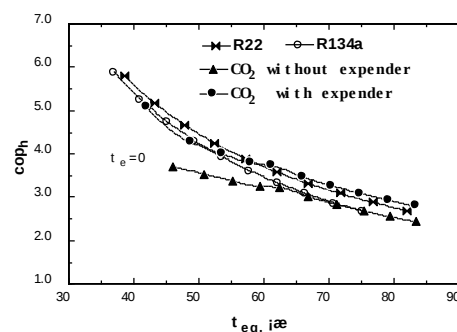


Fig.3 Relation between COPh in winter and equivalent temperature

It is illustrated in Fig.2 and Fig.3 that the coefficient of performance of R22 cycles is better than that of both CO₂ cycles without expanders and R134a cycles within the whole given scope of equivalent condensing temperature. There is a great gap between the coefficient of performance of CO₂ cycles without expander and that of R22 cycles and R134a cycles when the equivalent condensing

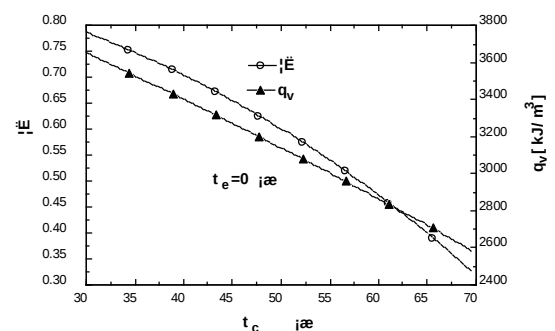


Fig.4. R134a λ 、 q_v at different equivalent condensing temperature

temperature is low. With the raise of equivalent condensing temperature, this gap becomes small. When the equivalent condensing temperature is over 65°C, the coefficient of performance of CO₂ cycles without turbines is close to that of R134a cycles. It can be indicated from Fig.2 and Fig.3 that the coefficient of performance of CO₂ cycles with expander is fairly close to that of R22 cycles despite the equivalent condensing temperature is low too. Moreover, the former is even higher than the later when the equivalent condensing temperature is super to 55°C.

As shown in Fig.4, the coefficient of performance of conventional refrigerants, especially R134a, decreases quickly with the equivalent condensing temperature rising up. The reason is its compression ratio increases and both the volumetric efficiency and the volumetric refrigeration capacity fall fleetly with the raise of equivalent condensing temperature. Although the rejection pressure is high in the CO₂ transcritical cycle, the compression ratio is small and the volumetric efficiency is still high. So the coefficient of performance CO₂ transcritical cycle can be equal to that of the conventional refrigerants when the former one is applied to those systems which need high temperature water such as air-conditioning systems and tap water heating systems.

3. OPERATING CHARACTERISTICS OF THE HEAT PUMP

Theoretical suction volume of the semi-hermetic reciprocating compressor, used in the CO₂ transcritical cycle ground source heat pump system showed in Fig.1, is assumed to $V_n=2.7\text{m}^3/\text{h}$ and rotary speed is $r=1450\text{rpm}$. The design conditions of the ground source heat pump is assumed that the inlet water temperature of the gas cooler is $t_{w,i}=30^\circ\text{C}$ and the outlet water temperature of the gas cooler is $t_{w,o}=50^\circ\text{C}$, water flow of the cycle is $G_w=0.103\text{kg/s}$. The analysis involves operation characteristics of the system when temperature and flow of the outlet water change, but temperature of the inlet water keeps constant. At the same time, on the assumption that refrigerant temperature of the gas cooler outlet is 32°C and evaporating temperature is 0°C . The equations of heat balance are as follows.

$$Q_h = G_w c_{p,w} (t_{w,o} - t_{w,i}) = q_h G_r \quad (10)$$

$$Q_e = \frac{q_v v}{3600} = q_e G_r \quad (11)$$

The volumetric efficiency and the isentropic efficiency can be computed respectively by the mean of the equation (3) and (4).

3.1 CHARACTERISTIC AT DIFFERENT OUTLET WATER TEMPERATURE ($t_{w,o}$)

When the load of air-conditioning systems or hot water supplying systems changes, the temperature control can be adapted to regulate the outlet water temperature of the heat pump to match the actual load and the water flux of the cycle is constant at one time. Meanwhile, adjustment of the operating pressure and the suction volume of the heat pump system can be achieved by mean of regulating the throttle open extent so as to keep the balance between the discharging heat of the gas cooler and actual load of the air-conditioning system. The values of COP_h and the operating pressure at different outlet water temperatures are shown in Fig5. When the outlet water temperature increases, the relative operation pressure raises and COP_h decreases as a result. When $t_{w,o}=50^\circ\text{C}$, the rejecting gas pressure is $p_r=79.556\text{bar}$, $COP_h=3.663$ and when $t_{w,o}=55^\circ\text{C}$, the rejecting gas pressure is $p_r=104.354\text{bar}$, $COP_h=3.269$.

3.2 CHACTERISTIC OF THE HEAT PUMP WITH CHANGES OF WATER FLUX (G_w)

The air-conditioning system can accomplish the adjustment by flux control with the changes of actual load. CO_2 transcritical cycle heat pump can meet the actual need by regulating the throttle open extent to control the operation pressure and the suction volume. The result is shown in Fig.6. With the increase of the mass flow of the air-conditioning system, COP_h of the heat pump falls and the operating pressure rises.

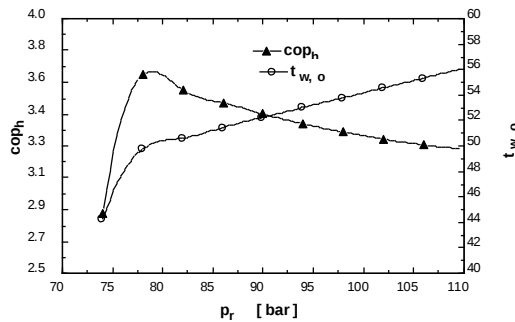


Fig.5 Relation of COP_h , discharging gas pressure and outlet water temperature

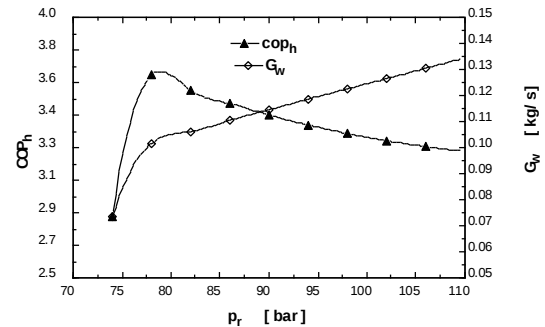


Fig.6 Relation of COP_h , discharging gas pressure and water flux

3.3 CAPACITY CONTROL METHODS

The above analysis indicates that the CO_2 transcritical cycle heat pump system can control the operating pressure to match the load by regulating the throttle. Additionally, the

capacity control can be achieved by varying the revolution of the compressor when the temperature of the gas cooler outlet and operating pressure of the compressor remain constant. The relation between the revolution of the compressor and the load is showed in Fig.8 when the outlet temperature of the gas cooler is 32°C and the supply and return water temperature of the air-conditioning system keeps constant. At the same time, the operating pressure p_r and COP_h are constants.

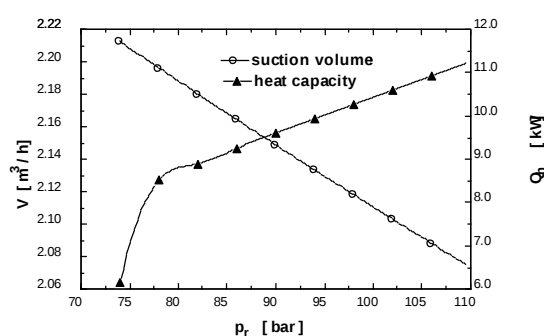


Fig.7 Relation of suction volume, heat and pressure

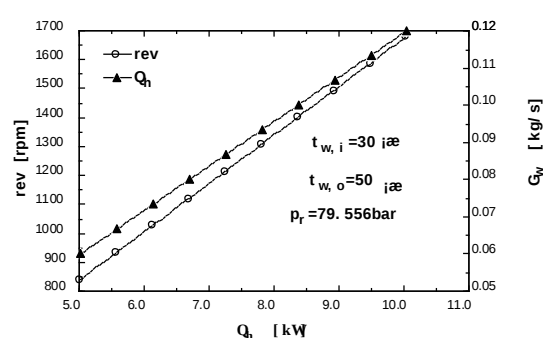


Fig.8 Relation of revolution, water flow and heat

Comparing Fig.7 with Fig.5 and Fig.6, the COP_h falls and the operating pressure rises up when the throttle in the system is regulated with the actual load increasing. But the declining range of COP_h is not obvious. Contrast to varying the revolution of the compressor, regulating the throttle is relatively suitable to the secure operation of the compressor and is easy to achieve. This characteristics of the CO_2 transcritical cycle is completely different to that of the conventional refrigerant cycles.

In fact, the transport energy consumption of the closed loop with U-tube ground heat exchanger and the air-conditioning system are on a large proportion to that of the CO_2 transcritical cycle ground source heat pump system. It is significant to decrease this part of energy consumption when the practical load changed. Therefore, the feasible capacity control methods of CO_2 ground source heat pump systems are to vary the pump or fan speed of heat source and heat sink. Whereas, the throttle of CO_2 heat pump unit must be controlled to fit for the change of the heat load.

4. CONCLUSION

When CO_2 transcritical cycle ground source heat pump system is applied in the air-conditioning systems and the tap water heating systems which the high water outlet temperature need to supply, it can be achieved the same performance as the R134a cycle. The CO_2 cycle with the expander is even superiority to the conventional refrigerant cycles.

The CO₂ transcritical cycle heat pump can be controlled by regulating throttle simply. Since there is the great temperature glide in the gas cooler, this system can be used in the heat recovery systems that require water with great temperature difference or with high temperature. Due to the increase of the temperature difference of heat sink circuit, the transport energy consumption of the system can be decreased effectively. On the other hand, because of the volumetric refrigeration capacity of CO₂ much higher than conventional refrigerants, it should be possible to produce systems compact and cost efficient. The analysis above suggested that CO₂ transcritical cycle for ground source heat pump is a desirable system in the applications of air conditioning and tap water heating.

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NOMENCLATURE

		Subscripts	
$c_{p,w}$	water specific heat (kJ/kg.K)	c	gas cooler or condenser
G_w	water flow rate in gas cooler (kg/s)	e	evaporator
G_r	refrigerant flow rate (kg/s)	eq	equivalent
p	pressure (bar)	h	heat
q	specific refrigerating or heating effect (kJ/kg)	in	inlet
qv	volumetric refr. capacity (kJ/m ³)	opt	optimal
Q	heat flux (kW)	out	outlet
r	revolution of compressor (rpm)	r	rejection
t	temperature (°C)	s	suction
T	Kelvin temperature (K)	w	water
w	specific work (kJ/kg)	Greek	
V_n	ideal suction volume (m ³ /h)	λ	volumetric efficiency
V	actual suction volume (m ³ /h)	η	isentropic efficiency

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