THE STUDY OF CO_2 TRANSCRITICAL CYCLE WATER SOURCE HEAT PUMP AND THE ANALYSIS OF THE APPLICATION

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ABSTRACT

The characteristics of CO_2 transcritical cycle heat pump such as the optimum high pressure, the maximum COP, the influence of evaporating temperature and outlet refrigerant temperature of gas cooler are analyzed in this paper. More over, through the analysis of expanding process, the ideal CO_2 expander demands are provided and the design technical direction of CO_2 is also pointed out. In addition, CO_2 transcritical cycle water source heat pump for hot water space heating is studied theoritically. By the comparison with the coal combustion boiler for space heating, we find that the first energy utilization rate of CO_2 transcritical cycle water source heat pump is higher. Furthermore, the combination of radiant ceiling panels and CO_2 transcritical cycle heat pump is analyzed and conclude that it is a nice and feasible two-win method in techniques. And by calculation and comparison, we obtain that CO_2 transcritical cycle with both expander and internal heat exchanger is the most suitable cycle to incorporate with the radiant ceiling panels if it is applied for both heating in winter and cooling in summer.

1. INTRODUCTION

Water source heat pump can not only meet the need of clean heating, but also has some advantages such as high energy efficiency, not being influenced by the air temperature of different seasons, no frost infection and effective application of waste heat. Therefore, it got fully developed from the middle of the 20^{th} century.

However, the conventional refrigerants such as Chlorofluorocarbons (CFCs) and Hydro chlorofluorocarbons (HCFCs) are now being regulated because of ozone depletion and global warming. Replacing (H)CFCs by HFCs and HFC mixtures still means using fluids with a large global warming potential. Simple logic indicates that substances present naturally in the biosphere, and for which the effects are long established, must be generally preferred as refrigerants (Lorentzen and Pettersen 1993). Carbon dioxide is one of these nature refrigerants.

Carbon dioxide was a commonly used refrigerant from the late 1800s and well into the 20^{th}

century. Owing to its complete harmlessness it was the generally preferred choice for usage on board ships. With the advent of the freones, the use of CO_2 was rapidly interrupted for the rapid loss of capacity at high cooling-water temperatures in tropics and the failure of the manufacturers to follow modern trends in CO_2 design towards more compact and price-effective high-speed types. The time is now ripe for application with present-day technology (Lorentzen 1994).

Carbon dioxide has many advantages as refrigerant in transcritical cycle such as Environmentally benign (ODP=0, GWP=1), safe (non-toxic & non-flammable), low cost, high volumetric capacity (compact systems), high thermal conductivity, low dynamic viscosity, compatible with normal lubricants and construction materials, low pressure ratio, low service cost (no reclamation), and considerable experience available. In addition, the heat rejection is in supercritical area, so the CO_2 discharge temperature of the compressor is high (can up to 100℃) and the gliding temperature is also high. Generally using conventional work medium the temperature of supplied hot water can only exceed to 50-55℃ at present. If raising the hot water temperature the condensing pressure would be too high, which will lead to the poor performance of the system. CO_2 transcritical cycle water source heat pump system, however, can rise the temperature of geothermal water, industrial waste water and the 30-60℃ underground water through the first heat pump temperature rising to more than 80℃ in order to supply space heat and other usages such as dehumidification. With the rapid increase of the city-heating load and the air-conditioned space, the system can partly replace the coal to supply heat so as to alleviate the city environment pollution and reduce the CO_2 emission.

2. THE CHARACTERISTICS OF CO_2 TRANSCRITICAL CYCLE HEAT PUMP

2.1 The optimum high pressure and the maximum COP

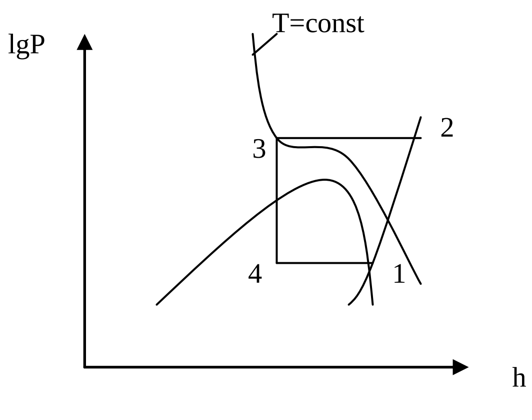

Figure1 LgP-h diagram for the transcritical cycle

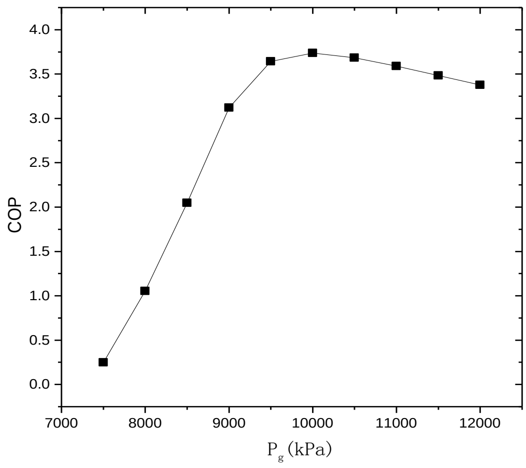

Figure2 Influence of P_g on COP

As shown in Figure1, the heat absorption and the heat rejection process of CO_2 are in two-phase and supercritical areas respectively. In conventional subcritical cycles, the enthalpy of the condenser outlet liquid is only the function of temperature. While in the supercritical area of the transcritical cycle, the pressure and the temperature are two independent variables, which determine the fluid enthalpy at the same time. It is illustrated clearly from the S-shape isothermal line in the supercritical area in Figure 1. Assuming a constant evaporating temperature T_4 and a constant refrigerant temperature at the gas cooler outlet T_3, variation of the high pressure P_3 leads to a change in refrigeration capacity, the shaft power and the coefficient of performance COP. The COP has a maximum value depending on P_3, because of the S-shape of the isotherm line and the straight compression line (Kauf 1999). With the rising of P_3, the isothermal line becomes steep, which demonstrates that the increase speed of the refrigeration capacity reduces with the enhancement of the pressure. But the isotropic line is almost a straight line, so the shaft power increases nearly linearly with the enhancement of the pressure.

Figure 2 shows the theoretically calculated COP at different pressure P_g when T_4=10℃ and T_3=40℃. At the beginning COP increases quickly, and then increases gently, at last it becomes decreasing. When the pressure is about 10 Mpa, the system got the maximum COP.

2.2 The influence of T_{evap} and T_3 on the performance of the cycle

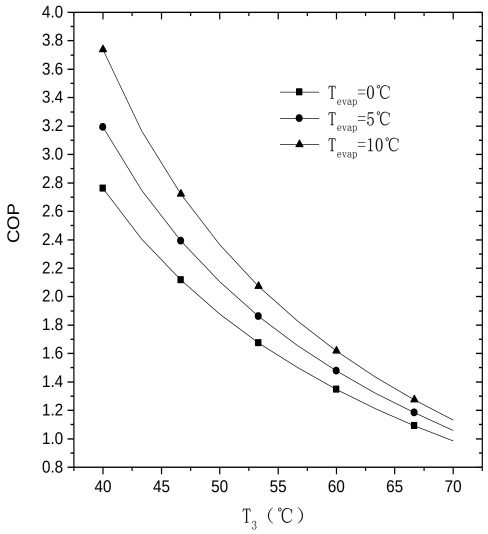

Figure 3 COP at different T_3 and different T_{evap}

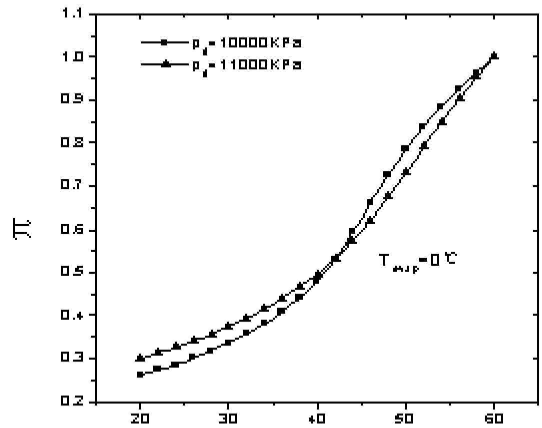

Figure 4 The loss ratio at different T_3

Figure 3 illustrates the COP at different T_3 and different evaporating temperature T_{evap}. COP decreases with the dropping of T_{evap} and decreases very quickly with the rising of T_3. The primary reason is the throttling loss increases sharply with the rising of T_3. Figure 4 demonstrates that the throttling losses of CO_2 transcriticle cycle at different T_3 when the evaporation temperature is 0℃. π refers to the loss ratio of the throttling loss when the refrigerant temperature before throttling is T_3 to the throttling loss when that is 60℃. The throttling valve loss occupies the largest part in all the losses of the cycle (Ma et al. 2001).

Using expander to replace the throttling valve is a very effective way to decrease the loss and improve the COP of the cycle (Ma 1999). The cycle with expander is approach to the ideal Carnot cycle, and there is no throttling loss theoretically. So the COP is higher than the conventional cycle. Moreover, the expanding ratio of CO_2 transcritical cycle is small (only 2-4), and the expanding power is large (25%-30% of the compressor shaft power). Therefore, the replacement is applicable from both theoretical and technical points of view.

2.3 Large gliding temperature in the heat rejection process

The large gliding temperature in the heat rejection process can match well with the heat media, and satisfy the stage application of the heat pump.

3. THE ISSUES IN THE EXPANDER DESIGN

In the CO_2 transcritical cycle, the inlet refrigerant of the expander is in the supercritical area. So the density resembles to the liquids and it has good mobility. First, the supercritical fluid changes to the high-pressure liquid. Then there are boiling cores form, which produce bubbles in the high-pressure liquid. The bubbles grow up and expand to output mechanical power. According to the process, the ideal expander should meet the demands as follows:

a. High pressure (up to 15Mpa) resistant
b. Can work in gas-liquid two phase area and stand for both liquid-slugging and gas-erosion
c. Small volume
d. Hermetically sealed
e. High entropy efficiency more than 60%
f. Work stably and availably for high velocity
g. Enough intensity

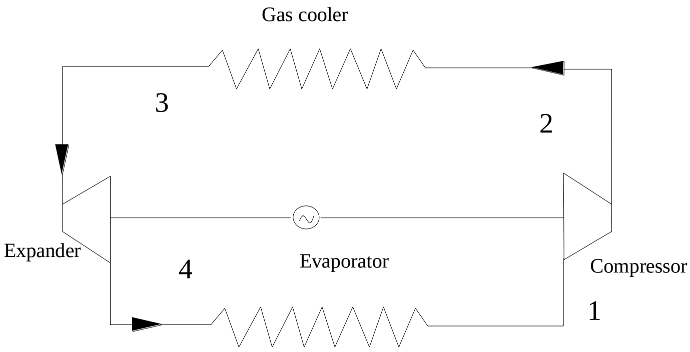

Figure 5a Diagram of CO_2 transcritical cycle with expander

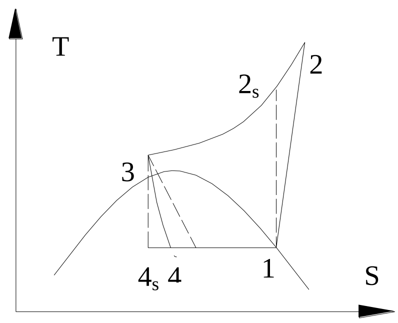

Figure 5b T-S Diagram of CO_2 transcritical cycle with expander

To improve the hermetical performance of the expander in the high pressure we plan to design the expander and the compressor at the same shaft as shown in Figure5a. Another advantage of this kind of design is that the compressor can use the recovery power effectively. If the mechanical construction can achieve technically, it is even better to incooperate the expander and the compressor into one shell to become a semi-hermetical or hermetical expander-compressor.

4. CO_2 TRANSCRITICAL CYCLE WATER SOURCE HEAT PUMP APPLICATIONS

4.1 CO_2 transcritical cycle water source heat pump for hot water space heating

In the calculation of the CO_2 transcritical cycle water source heat pump applying to hot water space heating, we assume that the outlet refrigerant of the evaporator is saturated vapor; the isentropic efficiency of the expander is 65%; the isentropic efficiency of the compressor is

80%; other irreversible loss can be neglected; the temperature difference between T_3 and return water temperature is 5℃; the COP and high pressure of the system is the optimum value at the operation condition. When the supply water and return water temperature are 95℃ and 55℃ respectively (standard high temperature heating supply) and when T_3 is 60℃, the calculation results are shown in Figure 6 and Figure 7.

From the viewpoint of the first energy utilization rate, we can compare the heating supply methods of coal combustion boiler and the heat pump (Hong 2000).

The first energy utilization rate of coal combustion boiler heating can be calculated as:

$$m_b = \frac{1000\times 3600}{H\cdot\eta} = \frac{1000\times 3600}{29260\times 0.6} = 205kg/1000kW\cdot h \tag{1}$$

The average efficiency of coal combustion boilers for heating in China is η=55%-60%. In this calculation, we assume η=60%. H is the thermal capacity of standard coal, we assume H=29260kJ/kg.

The first energy utilization rate of heat pump heating can be calculated as:

$$m_{hp} = \frac{1000\times b}{COP} = \frac{1000\times 0.4}{2.0} = 200kg/1000kW\cdot h \tag{2}$$

$b = 0.4kg/kW\cdot h$, which is the standard coal consummation of power plants at present. By the comparison, we conclude that the first energy utilization rate of the heat pump is smaller than that of the coal combustion boiler for heating supply unless the COP is larger than 2. Figure 7 shows that the COP is larger than 2 at evaporating temperature range of 5 to 20℃. Therefore, the CO_2 transcritical cycle water source heat pump is better than the coal combustion boiler for hot water space heating at the aspect of first energy utilization rate.

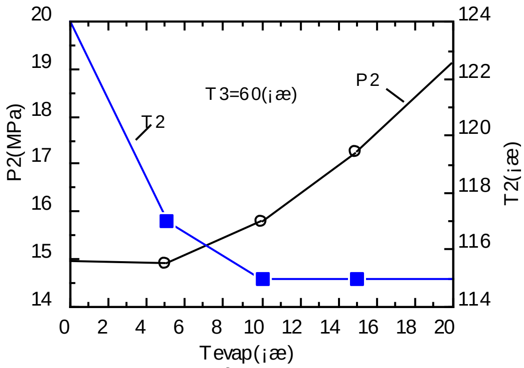

Figure 6 Inlet refrigerant pressure and temperature of the gas cooler at different evaporating temperature

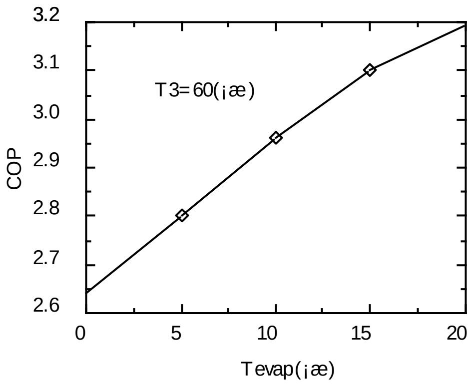

Figure 7 COP of the system at different evaporating temperature

4.2 A combination of radiant ceiling panels and CO_2 transcritical cycle in air conditioning system

From the analysis above, we admit that CO_2 transcritical cycle has advantages in space heating. However, it also has the disadvantage of low COP in air-conditioning in summer because of the great quantity of discharge heat of the gas cooler. The combination of radiant ceiling panels and CO_2 transcritical cycle heat pump is a nice and feasible two-win method in techniques. Applying radiant technology in Air conditioning systems of buildings has some comprehensive advantages in several aspects such as comfort, energy conversation and space saving. However, we have to solve the problem of humidity to avoid condensation of the panels, if we apply the radiant ceiling panels in the air conditioning system in humid climate. Using CO_2 as refrigerant in the transcritical cycle heat pump also possesses the merits in environment (ODP=0, GWP=1), but its efficiency is relatively low. A good improving way is to recover the discharge heat. Utilizing the heat of discharge gas as heat source of desiccant equipment can not only save the cost of the heating power, but also can improve the energy utilization rate of the CO_2 transcritical cycle heat pump (Zha et al. 2001). Moreover, the temperature of the water flowing in radiant ceiling panels is relatively high, so the evaporating temperature of the CO_2 transcritical cycle heat pump can be increased. Therefore, the COP of the system can also increase. In addition, the radiant ceiling panels can incorporate with the room fitment and decoration so that the radiant ceiling panels can not only be used as the room ceil, but also supply heating in winter and cooling in summer. The combination system is also adaptable to the place where has restrict on noise, vibration and stability of the room temperature. The schematic picture of the system is provided as Figure 8.

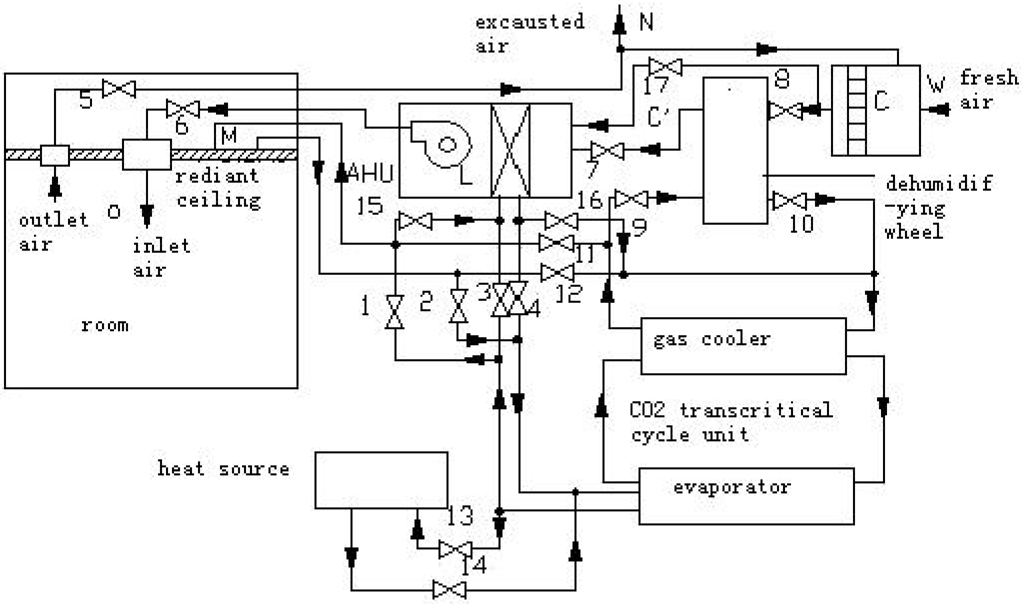

Figure 8 The schematic picture of the combination system of radiant ceiling panels and CO_2 transcritical cycle heat pump

The system consists of CO_2 transcritical cycle heat pump unit, dehumidifying unit, air mixing unit, air handling unit(AHU) and radiant ceiling panels unit. The system can supply both heating and cooling in a year. In summer, the valve 1-10 open, 11-17 close. Due to the high humidity in summer, the mixture of the fresh air and the return air enter the dehumidifying wheel, then pass through the heat exchanger to lower the temperature and to be dehumidified before entering the air-conditioned room. The heat exchanger is cooled by the evaporator of the CO_2 transcritical cycle heat pump and the dehumidifying wheel is heated by the gas cooler of the CO_2 transcritical cycle heat pump. The radiant ceiling panels is also cooled by the evaporator of the CO_2 transcritical cycle heat pump.

In winter, the valve 11-17open, 1-10 close. The dehumidify unit stops working. The radiant ceiling panels and air-handling unit are both heated by the gas cooler of the CO_2 transcritical cycle heat pump. The operation will need low temperature heat source such as ground water and the surface water in warm area.

For the calculation of the CO_2 transcritical cycle water source heat pump, we assume that the outlet refrigerant of the evaporator is saturated vapor; the isentropic efficiency of the expander is 65%; the isentropic efficiency of the compressor is 80%; the superheat of the internal heat exchanger is 15℃; Other irreversible loss can be neglected. We can compare (a)the CO_2 transcritical cycle with internal heat exchanger(Figure 9), (b)CO_2 transcritical cycle with expander(Figure 5) and (c)CO_2 transcritical cycle with both expander and internal heat exchanger. To save energy and to avoid dew condensing on the radiant ceiling panels, we let the evaporating temperature 10℃. When T_3 is 40℃, the calculation results are shown in Table 1.

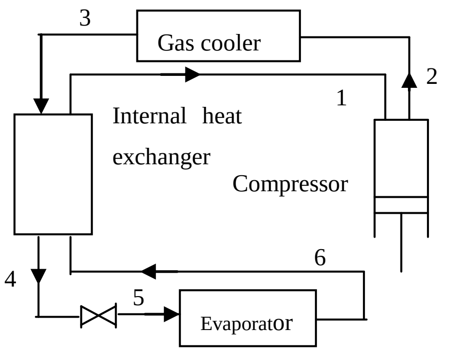

Figure9a Diagram of CO_2 transcritical cycle with internal heat exchanger

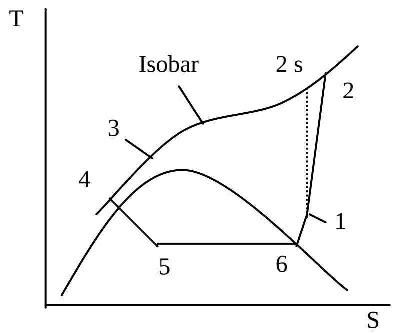

Figure9bT-S diagram of CO_2 transcritical cycle with internal heat exchanger

Table1. The calculation results of the (a)(b)(c) three kinds of cycles

	P_2 (MPa)	T_2 (℃)	COP_h	COP_r	P_2 (MPa)	T_2 (℃)	COP_h	COP_r	P_2 (MPa)	T_2 (℃)	COP_h	COP_r
a	8	74.1	2.6	1.6	10	94.7	4.1	3.1	11	103.8	3.9	2.9
b	9.5	74.4	5.0	4.0	12.5	93.9	4.4	3.4	13.5	100.7	4.2	3.2
c	8	74.1	3.6	3.5	10	94.7	4.8	4.4	11	103.8	4.5	4.0

P_2, T_2: The pressure and temperature of the inlet refrigerant of the gas cooler

COP_h, COP_r: the heating COP and cooling COP of the cycles

It is demonstrated that cycle b has the highest COP when T_2 is less than 80℃. When it is used for heating by radiant ceiling panels only in winter, cycle b is the most suitable one. However, when it is used for cooling by radiant ceiling panels in summer, b is not suitable. Because dehumidify unit must be working in humid area such as Southeast China in summer. Generally the inlet temperature of regeneration air of air-conditioning dehumidifier must be higher than 80℃. T_2 should be higher than 90℃ since the discharge heat of the gas cooler is used to heat the regeneration air. Both cycle a and c can meet the need. But cycle c has the highest COP when T_2 is higher than 94℃ and it is more close to the optimum pressure. Therefore, cycle c is the most suitable one if it is used to both heating in winter and cooling in summer.

When T_2 is 103.8℃, COP_h of cycle c can exceed to 4.5 and COP_r can exceed to 4.0. The dehumidify cooling COP based on the heat capacity for the regeneration air is 0.5 (Wang et al. 2000). If the dehumidifying unit is taken into consideration, the combination efficiency can be calculated as:

In summer：COP_r=4.0+0.5(1+4.0)=6.5 (3)

In winter ：COP_h=4.5

Both of them are higher than the efficiency of the conventional refrigerant cycle.

The radiant heat capacity can be calculated as:

$$Q_r = F_{12} A_{eff} \varepsilon_1 \varepsilon_2 \sigma (T_1^4 - T_2^4) \tag{4}$$

Where, $T_2 = (T_i + T_o)/2$

F_{12}： View factor

ε_1： Emissivity of surrounding surface

ε_2： Emissivity of radiator surface

σ ： Stefan-Boltzman constant [5.57×10^{-8} W/(m^2． K^4)]

T_1： Average temperature of surrounding surfaces [K]

T_2： Mean radiator surface temperature [K]

T_I： Water inlet temperature [K]

T_o： Water outlet temperature [K]

In winter, the radiant ceiling removes the heat load of the room, while air handling unit (AHU) removed the fresh air load. The heat load distribution of the gas cooler can be regulated according to the fresh air rate. In summer, the radiant ceiling removes the sensible load of the room, the dehumidify wheel removes the latent load and AHU removes both part of sensible load and part of latent load.

5. DISCUSSION

Although the calculated COP of the CO_2 transcritical cycle heat pump applied to hot water space heating is higher than 2.6, the practical COP may be lower than the calculated one. Because in the calculation we assume that the outlet refrigerant of the evaporator is saturated vapor; the isentropic efficiency of the expander is 65%; the isentropic efficiency of the compressor is 80%; other irreversible loss can be neglected. However, there still exist room to improve the energy application efficiency. As it is shown in Figure 3 that with the rising of T_3, the COP decreases very quickly. In the calculation, T_3 is set to 60℃ to match the standard high temperature heating supply (95℃/55℃). If we apply the stage energy, for instance, using the 55℃ return water to supply heat though radiant floor, the energy efficiency can be greatly improved.

By the calculation, we find cycle c is the most suitable cycle to the combination air conditioning system of radiant ceiling panels and CO_2 transcritical cycle. However, the initial cost of the system have to increase because the system not only affiliates an internal heat exchanger, but also applies a CO_2 expander though the efficiency of cycle c can be improved greatly because of the improved evaporating temperature and the recovered power of the expander. The price of internal heat exchanger is not high and some conventional cycle also often affiliates it to improve the efficiency. Therefore, the technical development and the cost reduction of the expander are crucial to the popular application of the system. We suppose that combining the function of the CO_2 compressor and the CO_2 expander in one part is the developing direction because the seal and the power recovery can be achieved easily.

The area control can also be achieved easily due to the adoption of radiant ceiling panels. But the thermal capacity of the radiant ceiling panels is relatively large, so the parameters control may be lagged behind. Further research is needed in the system matching and system control.

6. CONCLUSION

Because the CO_2 transcritical cycle is in the supercritical area, the pressure and the temperature are two independent variables, which determine the fluid enthalpy at the same time. And there exists an optimum high pressure and the maximum COP due to the S-shape isothermal line in the supercritical area.

- COP decreases with the dropping of T_{evap} and the COP decreases very quickly with the rising of T_3. The primary reason is the throttling loss increases sharply with the rising of T_3. And using expander to replace the throttling valve is a very effective way to decrease the loss and improve the COP of the cycle.
- The large gliding temperature in the heat rejection process of the CO_2 transcritical cycle heat pump can match well with the heat media, and satisfy the stage application of the heat pump.
- To improve the hermetical performance of the expander in the high pressure and to use the recovery power effectively, the expander and the compressor can be designed at the same shaft. If the mechanical construction can achieve technically, it is even better to incorporate the expander and the compressor into one shell to become a semi-hermetical or hermetical expander-compressor.
- The CO_2 transcritical cycle water source heat pump is better than the coal combustion boiler for hot water space heating at the aspect of first energy utilization rate.
- The combination of radiant ceiling panels and CO_2 transcritical cycle heat pump is a nice and feasible two-win method in techniques. And CO_2 transcritical cycle with both expander and internal heat exchanger is the most suitable one if it is used to both heating in winter and cooling in summer.

ACKNOWLEDGEMENTS

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REFERENCES

Hong F., 2000. Theoretical and experimental study on CO_2 transcritical cycle for water-to-water heat pump system, M.S. thesis of Tianjin University.

Kauf F., 1999. Determination of the optimum high pressure for transcritical CO_2-refrigeration

cycles, International Journal of Thermal Science, Vol. 38, pp. 325-330.
Lorentzen G. and Pettersen J., 1993. A new efficient and environmentally benign system for car air conditioning, International Journal of Refrigeration, Vol. 16, No. 1, pp. 4-12.
Lorentzen G., 1994. Revival of carbon dioxide as refrigerant, International Journal of Refrigeration, Vo. 17, No. 5, pp. 292-301.
Ma Y.T., Wang K.H., Yang Z. and Lu C.R., 1999. Thermodynamic analysis on CO_2 transcritical cycle with expander, Journal of Engineering Thermophysics, Vol. 20, No. 6, pp.662-665.
Ma Y.T., Wei D., Wang J.G., Zha S.T. and Lu C.R., 2001. A Theoretical analysis of the application of the two-phase flow screw expander in CO_2 transcritical refrigeration cycle, Journal of Engineering Thermophysics, Vol. 22, No. 2, pp. 137-140.
Wang J.G., Ma Y.T., and Wang K.H., 2000. A combination of CO_2 transcritical cycle and desiccant cooling, Chinese thermophysical academic conference in thermodynamics and energy application, pp. 473-477.
Zha S.T., Ma Y.T. and Wang J.G., 2001. A combination of radiant ceiling panels and CO_2 transcritical cycle, New Technical Development in Refrigeration and Air Conditioner, pp. 301-306, Shanghai Jiaotong University Press, Shanghai.