Theoretical Study of a Carbon Dioxide Double Loop System

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ABSTRACT

In the current research, a carbon dioxide double loop system is proposed. The system contains of two sub systems: a CO_2 power subsystem and a CO_2 refrigeration subsystem. The power subsystem is able to utilize the energy from the low-grade heat source to produce power. The power is then transferred to the refrigeration subsystem, partly or totally covering the power consumption of the compressor. Furthermore, it is also possible to take advantage of the temperature glides of both subsystems' heat rejection processes to produce hot water. Engineering Equation Solver (EES) is employed to analyze the system performance. The results show that the proposed system is a very promising way to provide cooling, heating and hot water in a more efficient way comparing to traditional systems.

1. INTRODUCTION

Heating, cooling and hot water supply consume a large amount of energy produced by burning fossil fuels, which causes well-known environmental problems as global warming, environment pollution, etc. At the same time, the most commonly used refrigerants in conventional air conditioning systems (i.e. synthetic refrigerants as CFCs, HCFCs) are strong climate gases themselves. Therefore, with increasing concern on environmental issues and energy shortages, a lot of research is needed to provide a more energy efficient and environmental benign way to provide heating, cooling and hot water supply.

Carbon dioxide (CO_2) is an environmental benign natural working medium, which has no ozone depleting potential (ODP) and negligible global warming potential (GWP=1). Furthermore, it is also inexpensive, non-explosive, non-flammable and abundant in the nature. With the awareness of more and more severe environmental problems caused by using synthetic refrigerants (i.e. CFC's ODP problem and HFC's GWP problem), CO_2 has gained increasingly interest ever since 1990s (Lorentzen, 1989). Besides that, carbon dioxide also has a great potential to be used as a working medium in power cycles to utilize the energy in low-grade heat sources. This is mainly due to its cycle's heating process is taken place in the supercritical region, where its temperature profile can match the heat source temperature profile better than other working fluids. Therefore, "pinching", which is commonly encountered in the heat exchanger for other working fluids and limit the cycle performance, can be avoided (Chen *et al.*, 2005; Chen *et al.*, 2006).

In the current research, a carbon dioxide double loop system is theoretically analyzed. The so-called carbon dioxide double loop system is a natural combination of a carbon dioxide power subsystem and a carbon dioxide refrigeration subsystem, which are running in parallel (Granryd, 2005). This system adopts the advantages of both the CO_2 power subsystem and CO_2 refrigeration subsystem. It is also possible to take the advantage of the temperature glides of both systems' heat rejection processes. Furthermore, the double loop system arrangement is flexible, due to the reason that two subsystems work in parallel but separated. In this way, the two subsystems can have different mass flows. Consequently, the power part's mass flow can be adjusted according to the heat source situation to provide the optimum power production, while the cooling part mass flow is adjusted to meet the cooling demand. Furthermore, the two subsystems' working pressure can also be adjusted freely to achieve the optimum system working condition.

Figure 1 shows the schematic layout of the CO_2 double loop system, which contains two parts: power part (upper part) and refrigeration part (lower part). The power part is composed of 5 main parts: a pump, a gas heater, an expansion machine, a gas cooler and an internal heat exchanger. The refrigeration parts contains of 4 main parts: a compressor, a gas cooler, an expansion valve and an evaporator. Furthermore, an internal heat exchanger is also included in the system to ensure the refrigerant vapour is slightly superheated (I.e. 5°C superheat) before enters the compressor. Due to its low critical temperature (31.1 °C, 73.8 bar), carbon dioxide refrigeration cycle generally works as a transcritical cycle, while carbon dioxide power cycle can work as either a supercritical cycle or a transcritical cycle. Comparing the two different power cycles, the transcritical power cycle will produce more power and achieve a higher thermal efficiency than the supercritical power cycle for a certain heat source temperature. However, due to the low critical temperature of carbon dioxide, the CO_2 transcritcal power cycle needs a very cold heat sink to be able to reject the heat in subcritical region, which may be difficult to realize in reality unless cold water is available. Therefore, in the current research, the combination of a carbon dioxide supercritical power cycle and a carbon dioxide transcritical refrigeration cycle is investigated. Furthermore, to simply the system, one gas cooler is assumed to be used for the whole double loop system, thus same gas cooler outlet temperature should be achieved for both sub-systems. The T-S diagram of the corresponding cycles is schematically showed in Figure 2.

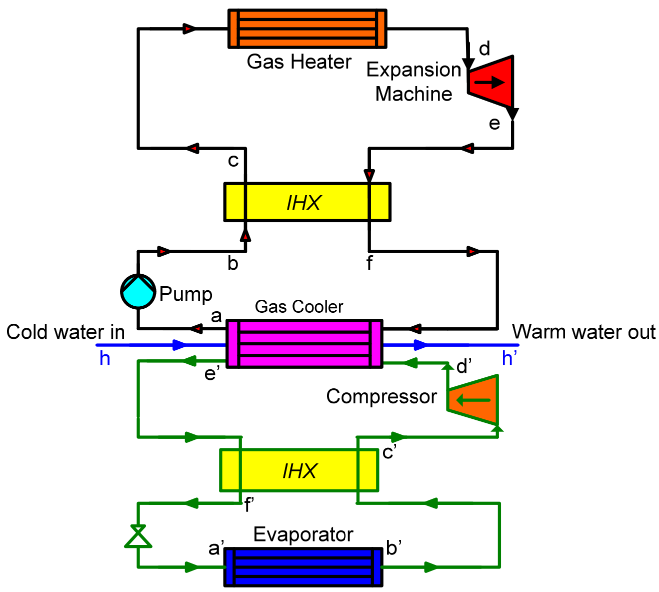

Figure 1 Schematic of solar driven carbon dioxide double loop system

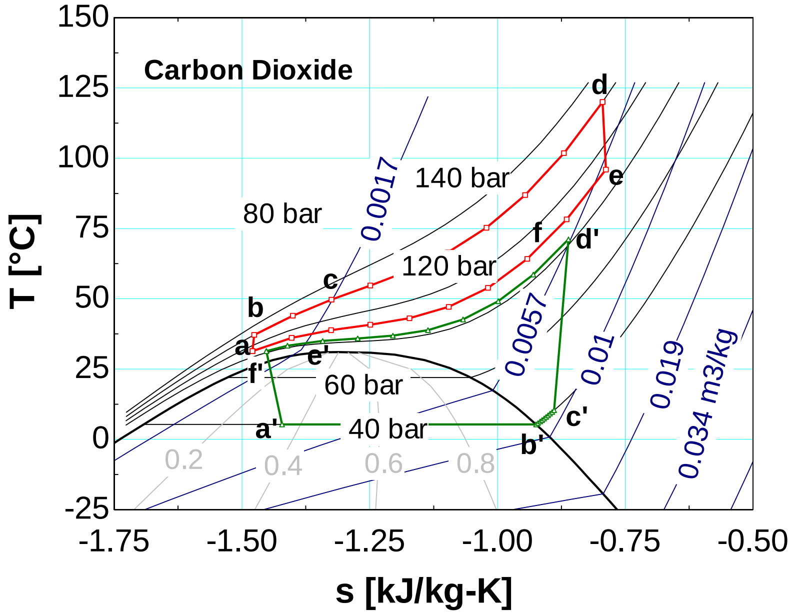

Figure 2 Schematic T-S diagram of a carbon dioxide supercritical double loop cycle

Two efficiencies can be employed to evaluate the double loop system performance. For the power subsystem, the system thermal efficiency is defined as the ratio of the net work output to the heat input (eq. 1).

$$\eta_{th} = \frac{W_{net}}{Q_{input}} = \frac{W_{exp.} - W_{pump}}{Q_{input}} = \frac{(h_d - h_e) - (h_b - h_a)}{(h_d - h_c)} \qquad (1)$$

For the cooling subsystem, the COP of a vapour compression refrigeration system is traditionally defined as eq 2.

$$COP = \frac{Q_{cooling}}{W_{basic}} \qquad (2)$$

Where, $Q_{cooling}$ is the cooling capacity of the cooling system, W_{basic} is the required compression work of the compressor. For the double loop system, the power part of the double loop system produces power. The power is then transferred to the cooling part to cover the compressor work. Since the power produced by the double loop system's power part is gained from the low-grade heat source, when solar thermal or low-grade heat is employed as a heat source, the power produces in the power part will be energy produced "free of charge". Therefore, the system can achieve the required cooling capacity with less energy demand for the compression work, thus save the fuel and lower the emissions. Consequently, the "new COP" of the double loop system is called "COP $_{double}$" and it can be defined by eq. 3[1].

$$COP_{double} = \frac{Q_{cooling}}{W_{double}} = \frac{Q_{cooling}}{W_{basic} - W_{powerpart}} \qquad (3)$$

1 This definition is only useful for a comparison with a conventional system where the extra work, Woutput increases the apparent COP. For a truly heat driven system where $W_{basic} = W_{output}$ the definition gives an infinite COP.

Where $Q_{cooling}$ is the required cooling capacity, W_{double} is the new compression work after taking away the energy gained by the double loop's power part, W_{basic} is the original compression work of the cooling cycle, W_{output} is the work output from the combined cycle power part, i.e. the "free" energy gained from low-grade heat source.

2. BASIC SYSTEM PERFORMANCE

The system is modelled in EES to analyze the system thermodynamic performance. Several assumptions are needed to analyze the system performance:

- The pump efficiency is assumed to be 0.8 in the current study based on the work by Tadano *et al.* (2000) on CO_2 hermetic compressors, which was under the similar working conditions as the current cycle.
- The CO_2 expansion machine isentropic efficiency is assumed to be 0.85 based on research by Nickl *et al.* (2003) and Huff *et al.* (2003).
- The compressor's isentropic efficiency is assumed to be 0.75 according to the research done by Rozhentsev and Wang (2001).
- The power part internal heat exchanger is assumed to have 0.9 effectiveness, based on Boewe *et al.*'s research (2001). The cooing part internal heat exchanger is assumed to ensure 5 °C superheat after evaporator.
- The gas cooler of the double loop system is assumed to have 85% efficiency when heating up the hot water.
- To simplify the basic system analysis, the same gas cooler pressure is assumed for both the double loop system's power subsystem and the refrigeration subsystem.
- The system is assumed to be well insulated and the heat losses are neglected.

Furthermore, unlike traditional refrigeration system, carbon dioxide refrigeration system partly works in the supercritical region, in which the working fluid's temperature is independent of pressure. Therefore, there is an optimum gas cooler pressure to achieve maximum COP.

Liao *et al.* (2000) proposed a correlation to predict the optimum heat rejection pressure in terms of evaporation temperature and the gas cooler's outlet temperature, which is expressed as eq. (4).

$$p_{opt} = (2.778 - 0.0157 t_e) t_{gco} + (0.381 t_e - 9.34) \tag{4}$$

Based on eq. (4), the optimum heat rejection pressure for the proposed working condition will be 83 bar and this pressure is used for both subsystems in the basic system performance analysis.

The more detailed system simulation parameters are listed in table 1

Table 1 Carbon dioxide double loop system basic operating conditions

Simulation Parameters	Value	Unit
Evaporator pressure	40	bar
Evaporation temperature	5.3	°C
Refrigerant mass flow	290	Kg/h
Superheat after evaporator	5 (fixed value)	K

Gas cooler pressure	83	bar
Gas cooler outlet temperature	35[2]	°C
Gas heater pressure	120	bar
Expansion inlet temperature	120	°C
Compression efficiency	75%	-
Expansion efficiency	85%	-
Pump efficiency	80%	-
Power part IHX effectiveness	0.9	-
Cooling waler inlet temperature	15	°C
Cooling waler mass flow rate	540	Kg/h

Assuming the double loop system's power part has the same mass flow as the cooing part (i.e. 290 kg/h), the system performances in Table 1 described working conditions are showed in Table 2. It can be noticed from Table 2 that if the low-grade heat source as solar thermal or waste heat is used, the proposed double loop system can improve the basic refrigeration system's COP with 34 %.

Table 2 Carbon dioxide double loop system's performance under the basic operating conditions

Performance Parameters	Value	Unit
Double loop power part thermal efficiency (without IHX)	4.77%	-
Double loop power part thermal efficiency (with IHX)	7.48%	-
Basic refrigeration system COP	3.09	-
Double loop system COP_{double}	4.13	-
Water outlet temperature	60.8	°C
System cooling capacity	9.76	kW
Power of hot water production	25.1	kW

3. DISCUSSION

As mentioned before, one advantage of carbon dioxide double loop system is that both subsystems can have different gas cooler pressures. Therefore, the gas cooler pressure of the power subsystem can be different from the refrigeration system's optimum gas cooler pressure. For traditional power cycles, the heat rejection pressure should be kept as low as possible to achieve a high thermal efficiency. However, for supercritical power cycles, low heat rejection pressure does not always lead to a high efficiency. Instead, there is an optimum heat rejection pressure for a certain cycle working condition.

If the other basic system working conditions are kept constant, the power subsystem performance can be plotted against different gas cooler pressure in Figure 3.

[2] This temperature is the temperature before the IHX. The real gas cooler outlet temperature is the temperature after providing 5°C superheat at evaporator outlet (I.e. 33.39 °C in the current case).

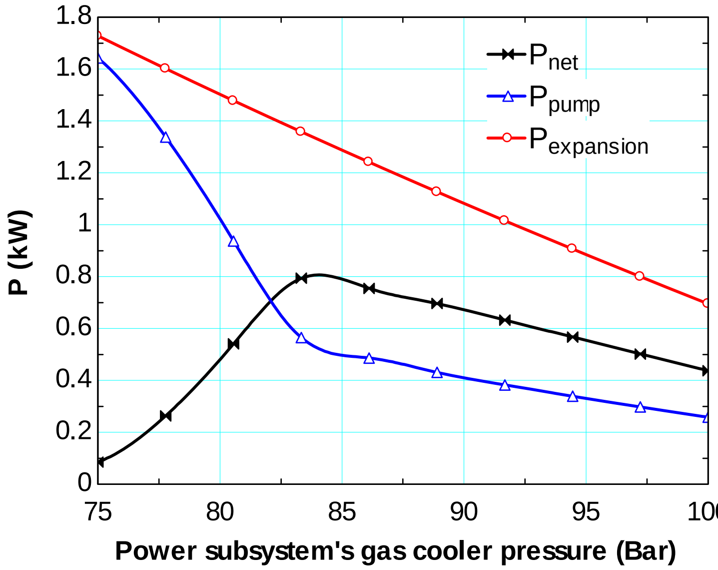

Figure 3. Performance of the double loop system's power subsystem *vs.* its gas cooler pressure

As shown in Figure 3, the pump work of the power subsystem decreases with increasing gas cooler pressure. Furthermore, the decrease is more obvious at the pressure up to 80 bar and then becomes less noticeable at even higher pressures. Meanwhile, the expansion work also decreases with increasing gas cooler pressure. Therefore, there is an optimum gas cooler pressure for the double loop system's power subsystem, which appears when the difference between its expansion work and its pump work reaches the maximum.

By keeping the optimum refrigeration subsystem's gas cooler pressure, the double loop system's COP (COP_{double}) is plotted against different gas cooler pressures of power subsystem at different gas heater pressures (Figure 4).

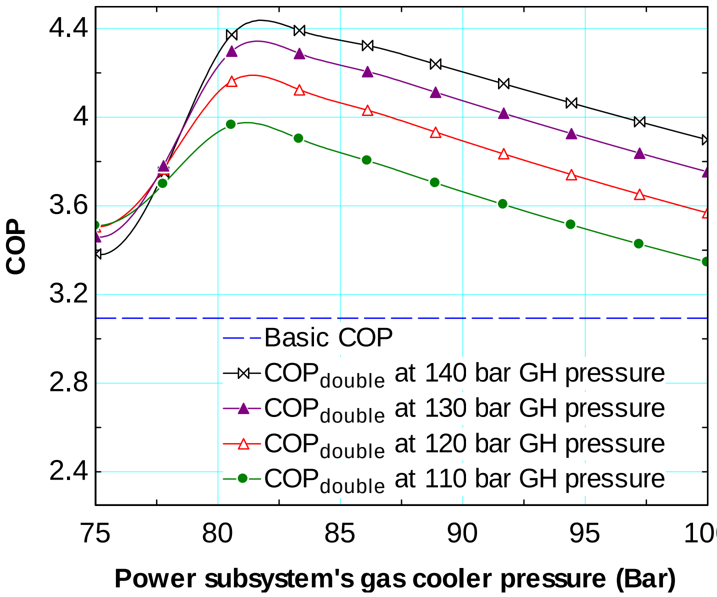

Figure 4 Basic refrigeration system's COP and double loop system's COP_{double} *vs.* different gas cooler pressures at different gas heater pressures.

From Figure 4, one may notice that with the contribution from the system's power part, the proposed double loop system can achieve a much higher COP than the basic carbon dioxide refrigeration system can. For a certain system working condition, there is an optimum power subsystem's gas cooler pressure, which enables a maximum COP for the double loop system. Furthermore, the simulation results also show that the optimum gas cooler pressure for the power subsystem is independent of its gas heater pressure.

Due to the reason that the power part of the double loop system works as a supercritical system, its gas heater pressure also influences the system performance. While keeping other basic system working conditions constant, the double loop system's COP (COP_{double}) is plotted against different gas heater pressures at different gas cooler pressures and different expansion inlet temperatures (Figure 5).

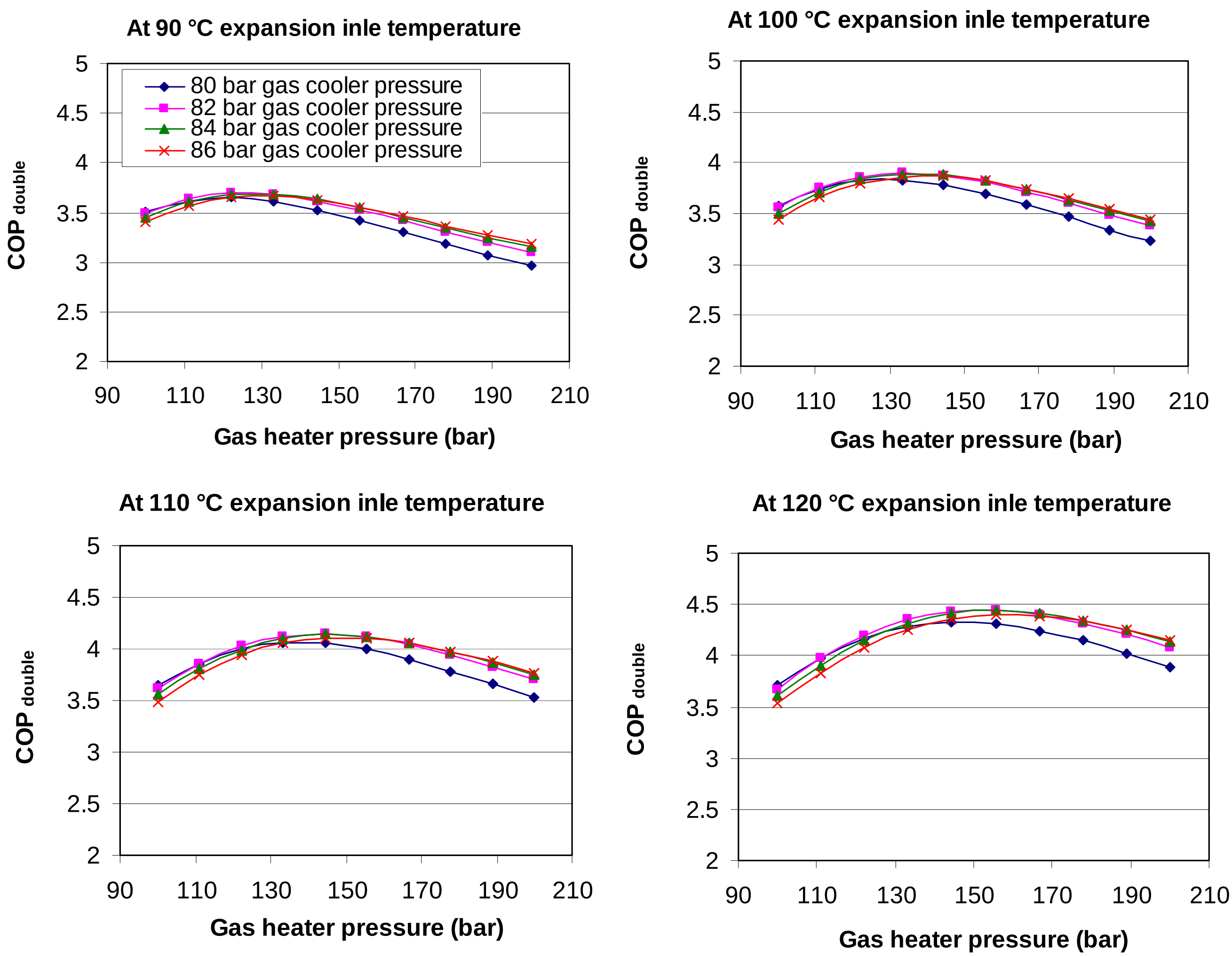

Figure 5 Double loop system's COP against different gas heater pressures at different gas cooler pressures and different expansion inlet temperatures.

As shown in Figure 5, for a certain expansion inlet temperature and a certain gas cooler pressure, there is an optimum gas heater pressure, which enables the maximum COP. Furthermore, for a certain gas cooler pressure, the higher the expansion inlet temperature is, the higher the optimum gas heater pressure will be. It can also be noticed in the figure that at the optimum gas heater pressure, the optimum gas cooler pressure is almost constant (e.g. around 82 bar for the current basic system working condition). Furthermore, the power subsystem's gas cooler pressure has very limited influence on the double loop system's COP_{double} at optimum gas heat pressure.

The influences of the compressor, expansion machine and pump's isentropic efficiencies on the system performance are also studied in the current paper. Maintaining the basic carbon dioxide double system working conditions constant and changing one parameter at a time, the double loop system's COPs are plotted against the efficiency of the pump, the compressor and the expansion machine respectively in comparison with a basic carbon dioxide refrigeration system (Figure 6).

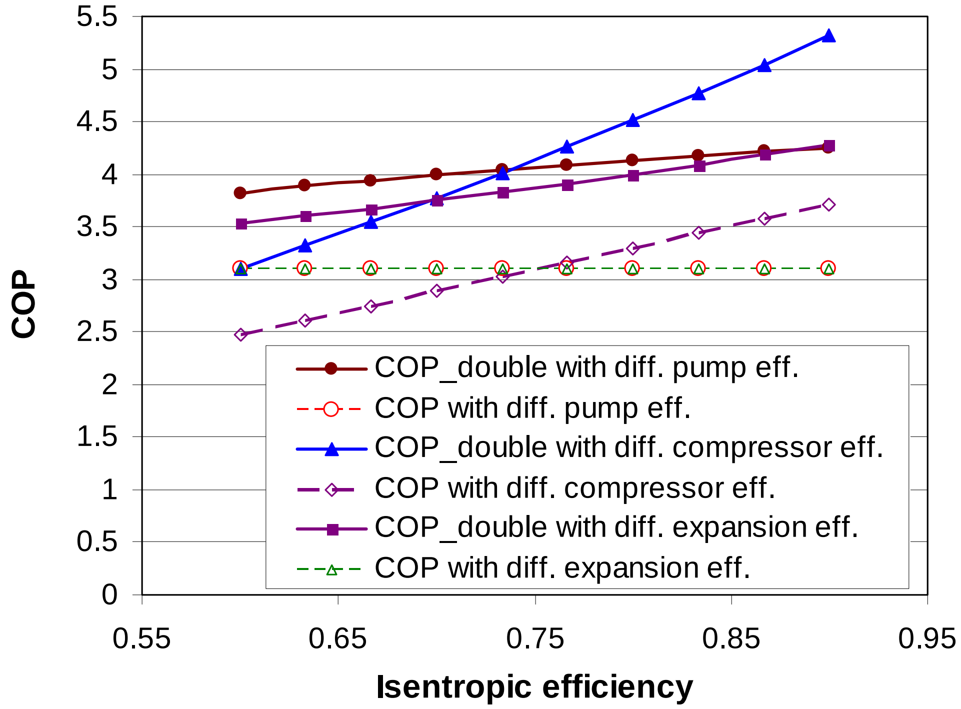

Figure 6 Double loop system's COP against different components' efficiencies

As shown in Figure 6, all the three components (pump, compressor and expansion machine) can improve the double loop system's COP. Comparing all the three components, the compressor has the most critical influence on the system's COP than the pump and expansion machine do. Furthermore, compared with a basic carbon dioxide refrigeration system, the compressor's influence on the system's COP is more crucial in a carbon dioxide double loop system than in a basic carbon dioxide refrigeration system.

4. CONCLUSION

In the current research, a carbon dioxide double loop system is proposed. The so-called carbon dioxide double loop system is a natural combination of a carbon dioxide power system and a carbon dioxide refrigeration system, which run in parallel. The system adopts the advantages of both the CO_2 power system and CO_2 refrigeration system. It is also possible to take the advantage of the temperature glides of the heat rejection processes in both subsystems in order to produce hot water. Furthermore, by using carbon dioxide as a working medium in power cycles, is able to utilize the energy from the low-grade heat source such as solar thermal and waste heat, to reduce the system energy demand, and to provide more efficient cooling/heating.

The system is modeled in EES to study its thermodynamic performance. The results show that under the pre-described basic system working condition, the system is able to increase the basic carbon dioxide refrigeration system for more than 30%.

By examining different system working parameters, the proposed carbon dioxide double loop system has both an optimum gas cooler pressure for the refrigeration subsystem and an optimum gas cooler pressure for the power subsystem. Furthermore, there is also an optimum gas heater pressure for a certain system working condition. For a certain gas cooler pressure, the higher the expansion inlet temperature is, the higher the optimum gas heater pressure will be. Nevertheless, at the optimum certain gas heater pressure, the optimum gas cooler pressure is almost constant regardless of expansion inlet temperature and its influence on the double loop system’s COP is negligible.

By plotting the double loop system’s COP against the efficiency of the pump, the compressor and the expansion machine respectively, it is found that the compressor has a more crucial influence on the system’s COP than the pump and the expansion machine do. Furthermore, the compressor’s influence on the system’s COP is more critical in a carbon dioxide double loop system than in a basic carbon dioxide refrigeration system.

The simulation results show that the carbon dioxide double loop system is a promising way to provide cooling, heating and hot water in a more environmental friendly and more efficient way than with traditional systems.

NOMENCLATURE

COP	Coefficient of Performance	(–)
EES	Engineer Equation Solver	(–)
GWP	Global Warming Potential	(–)
GH	Gas Heater	(–)
GC	Gas Cooler	(–)
h	Enthalpy	kJ/kg
IHX	Internal Heat Exchanger	(–)
ODP	Ozone Depleting Potential	(–)
ORC	Organic Rankine Cycle	(–)
P	Power	kW
Q	Cooling Capacity	kW
Q_{input}	Heat Input to the Power Subsystem	kW
W	Work	kW

Subscripts

a-f	power cycle route point
a’-f’	refrigeration cycle route point
basic	basic refrigeration system
c	gas cooler outlet
double	double loop system
e	evaporator
gco	gas cooler outlet
h-h’	points for water properties
opt	optimum

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