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# Simulation on the Performance of CarbonDioxide and HFC-125 Heat Pumpsfor Medium-andHigh-Temperature Heating

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***Abstract*—**In order to compare the performance of the carbon dioxide and HFC-125 heat pumps for medium-and high-temperature heating, both heat pump cycles were optimized using a simulation method. To fairly compare the performance of the cycles by using different working fluids, each cycle was optimized from the viewpoint of heating COP by two design parameters. The first is the gas cooler exit temperature and the other is the ratio of the overall heat conductance of the gas cooler to the combined overall heat conductance of the gas cooler and the evaporator. The inlet and outlet temperatures of secondary fluid of the gas cooler were fixed at 40/90°C and 40/150°C.The results shows that the HFC-125 heat pump has 6% higher heating COP than carbon dioxide heat pump when the heat sink exit temperature is fixed at 90ºC, while the latter outperforms the former when the heat sink exit temperature is fixed at 150ºC under the simulation conditions considered in the present study.

***Keywords*—**Carbon Dioxide, HFC-125, Transcritical, Heat Pump

## I. Introduction

SINCE many industrial and household applications need medium-andhigh-temperature heating, a medium-and high-temperature heat pump can be a good technical and economical option. Regarding this, a great deal of efforts, e.g., improving component performance and devising a novel cycle, has been made to increase a performance of heat pumps.

Proper selection of the working fluid also plays a significant role in increasing performance. A carbon dioxide, which is non-toxic, non-flammable and is compatible to normal lubricants and common construction materials, has been considered as a successful working fluid for a water heater application that provides hot water up to 90°C [1].One of the reasons why a carbon dioxide is preferred for a water heater application is the benefit of temperature glide matching between the supercritical carbon dioxide and water in a gas cooler [2].However, its working pressure is very high and products are still under development. In this study, a HFC-125, which also forms a transcritical heat pump cycle with a lower working pressure than the carbon dioxide, was considered as a working fluid as well as the carbon dioxide for a heat pump that provides 90°C and 150ºC. Even though a HFC-125 is used a lot in refrigeration and heat pump applications as a component of the zeotropic mixture HFC-407C or HFC-410A, studies on a HFC-125 transcritical heat pump are rare [3].

The main objective of this study is to compare the performance of the carbon dioxide and HFC-125 heat pumps for medium-and high-temperature heating.

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For this purpose, heat pump cycles using carbon dioxide and HFC-125 were optimized in terms of heating COP by simulation method. After that, they were compared at the point of their maximum performance.

## II. Thermodynamic Analysis of the Cycle

Fig. 1 is a schematic of the heat pump cycle in this study. A working fluid leaves the gas cooler (state point 1 in Fig. 1; SP1), is expanded to a low pressure (SP2), and then is heated to a superheated vapor (SP3). After compression through the compressor to a high pressure (SP4), the vapor is cooled (SP1).

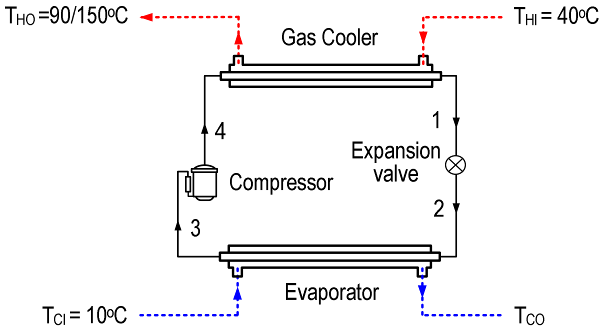


Fig. 1 Schematic diagram of a heat pump cycle

For the cycle simulation, it is assumed that the system reaches a steady state. Pressure drop and heat loss in each component are neglected. All heat exchangers are assumed to be in counter-flow configuration. The energy and exergy balances at the gas cooler and the evaporator are

$$Q_{GC} = \dot{m}_r(h_4 - h_1) = \dot{m}c_{p,H}(T_{HO} - T_{HI}) = (UA)_{GC}\,\Delta T_{m,GC} \quad (1)$$

$$E_{D,GC} = \dot{m}_H(e_{HI} - e_{HO}) + \dot{m}_r(e_4 - e_1) \quad (2)$$

$$Q_E = \dot{m}_r(h_3 - h_2) = \dot{m}c_{p,C}(T_{CI} - T_{CO}) = (UA)_E\,\Delta T_{m,E} \quad (3)$$

$$E_{D,E} = \dot{m}_C(e_{CI} - e_{CO}) + \dot{m}_r(e_2 - e_3). \quad (4)$$

The compressor isentropic efficiency can be expressed as

$$\eta_s = (h_{4s} - h_3)/(h_4 - h_3). \quad (5)$$

The power input to the compressor is

$$W = \dot{m}_r(h_4 - h_3). \quad (6)$$

The exergy destruction rates in compression and expansion processes are

$$E_{D,Comp} = \dot{m}_r T_0 (s_4 - s_3) \tag{7}$$

$$E_{D,Exp} = \dot{m}_r T_0 (s_2 - s_1). \tag{8}$$

Then, the heating COP is determined as $Q_{GC}/W$ .In this study, in order to perform a practical evaluation of the cycle's performance, the total overall conductance (TOC), $(UA)_{GC} + (UA)_E$ ,which can be regarded as a measure of the size of heat exchangers constituting the cycle, was given.The heat pump cycle has many design parameters: temperature, pressure at each part of the cycle, mass flow rate, and so on. In this study, two independent variables are selected to maximize heating COP.The first isthe gas cooler exit temperature $T_1$ and the other is the $(UA)_{GC}/((UA)_{GC}+(UA)_E)$ value, or the ratio of the overall heat conductance of the gas cooler to the combined overall heat conductance of the gas cooler and the evaporator.For a simulation, the following conditions are also given:

(1)Heat source inlet temperature $T_{CI}$ =10°C and the thermalcapacitancerate $\dot{m}c_{p,C}$ = 29.3 kW/K.
(2)Heat sink inlet and exit temperature $T_{HI}$ =40°C and $T_{HO}$ =90/150°C at $\dot{m}c_{p,H}$ = 4.18/2.09 kW/K.
(3)The TOC $(UA)_{GC}+(UA)_E$ is fixed at 30 kW/K.
(4)The isentropic efficiency for the compressor $\eta_s$ is 0.7.
(5)The evaporator exit superheat is 5°C.

The thermodynamic properties of the working fluids are calculated by REFPROP 8.0[4].Once the two independent variables are given, the cycle performance can be found as shown in Fig. 2. First, we assume the gas cooler pressure $P_4$ and the evaporator pressure $P_2$ . From the evaporator pressure and given evaporator exit superheat, evaporator exit state (SP3) can be determined. By using an isentropic efficiency for the compressor, compressor exit state (SP4) is determined. Then the working fluid mass flow rate $\dot{m}_r$ can be determined by(1). Assuming that the expansion process is isenthalpic, the evaporator inlet state (SP2) can be determined. From(3), the heat sink exit temperature $T_{CO}$ is determined. After this process, we can determine $(UA)$ values for two heatexchangers from(1) and (3), where the mean temperature difference $\Delta T_m$ can be expressed as (9) by assuming the constant overall heat transfer coefficient *U*[5]-[6].

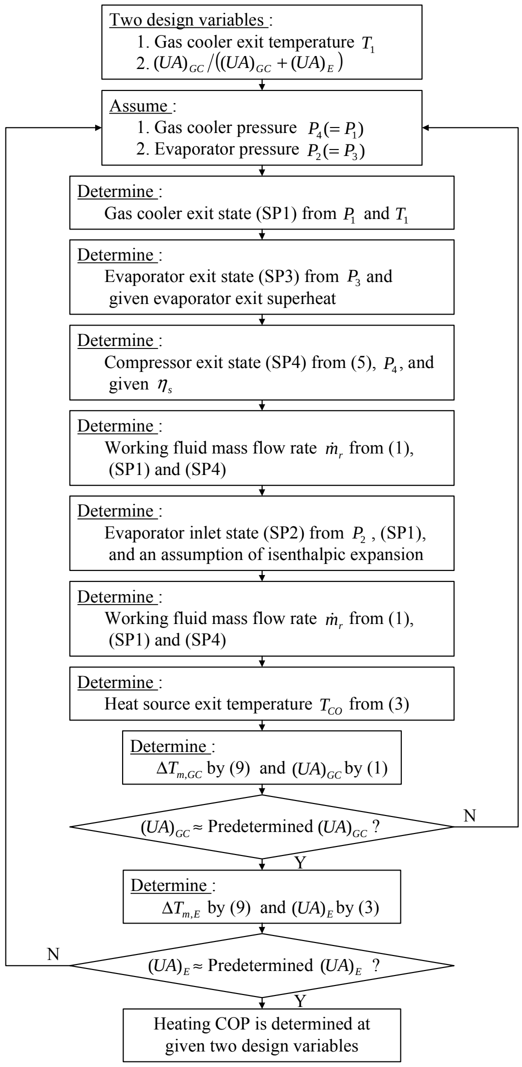

Fig. 2 Calculation procedure

$$\frac{1}{\Delta T_m} = \frac{1}{Q_i}\int_0^{Q_i} \frac{dQ}{\Delta T(Q)} \tag{9}$$

After obtaining $(UA)_{GC}$ and $(UA)_E$ values, the initially assumed $P_4$ and $P_2$ are renewed.Then the above procedure is repeated until calculated $(UA)_{GC}$ and $(UA)_E$ values reach predetermined $(UA)_{GC}$ and $(UA)_E$ values from the given TOC and $(UA)_{GC}/((UA)_{GC}+(UA)_E)$ value. Once the iterationsarecompleted for given conditions, the heating COP is determined. Then, the exergy destruction rate in each component can also be determined [7].

## III. Results and Discussion

Fig. 3 shows the heating COP variations of carbon dioxide heat pump cycle over the change of two independent variables when a heat sink exit temperature $T_{HO}$ =90°C.

Once $(UA)_{GC}/((UA)_{GC}+(UA)_E)$ is given, optimal $T_1$ is capable of maximizing the heating COP. The reason why the

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optimum combination of $(UA)_{GC}/((UA)_{GC}+(UA)_E)$ and $T_1$ exists is as follows. If $T_1$ is too high, the evaporator inlet quality increases too much, causing the cycle performance to decrease. If $T_1$ is too low, since the temperature difference at the gas cooler exit ($T_1$ - $T_{HI}$) decreases, the gas cooler inlet temperature ($T_4$) increases to meet $\Delta T_m$,which is determined by predetermined $Q_{GC}$ and $(UA)_{GC}$.This increases a gas cooling pressure ($P_4$), which in turn decreases the cycle performance. These two competing effects allow optimal $T_1$ value to exist.

Meanwhile, when $(UA)_{GC}/((UA)_{GC}+(UA)_E)$ becomes too small, a gas cooling pressure becomes too high. To the contrary, when $(UA)_{GC}/((UA)_{GC}+(UA)_E)$ is too high, an evaporating pressure becomes toolow, causing the cycle performance to decrease. As we have seen, the heating COP is determined by two independent variables under the given conditions.Therefore, it is necessary to optimize the two variables in order to maximize the heating COP. For this purpose, the pattern search algorithm (PSA)was employed in this study. The PSA, a method for solving optimization problems, does not require any information about the gradient of the objective function [8].In this study, the PSA was implementedby using Matlab 2009a [9].Table I shows the optimization results of carbon dioxide and HFC-125 heat pumps.

When the heat sink exit temperature $T_{HO}$ is fixed at 90°C, each optimized cycle becomes transcritical cycle as shown in Figs. 4 and 5. From the viewpoint of a heating COP, the HFC-125 cycle shows better performance of 2.66, while the carbon dioxide cycle has a relatively low performance of 2.50. The main reason for this is that theexergy destruction rate during expansion process ($E_{D,Exp}$) of carbon dioxide cycle is 16 kW, which is higher than that of HFC-125 cycle. The carbon dioxide cycle's high $E_{D,Exp}$ value also increases an evaporator inlet quality ($x_2$) to 0.42, while that of HFC-125 cycle is at about 0.40. One of the major differences between the two cycles is their working pressure.

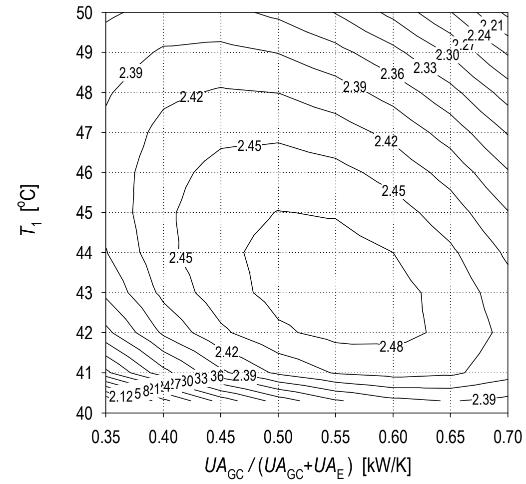


Fig. 3 Heating COP of carbon dioxide heat pump over the change of two independent variables when $T_{HO}$ =90°C

At the maximum heating COP condition, the gas cooling pressure of the carbon dioxide cycle is 14,956 kPa,while that of the HFC-125 cycle is 5,804 kPa, which provides opportunitiesfor advantages in safety and cost.Carbon dioxide cycle is superior to HFC-125 cycle as far as a compression ratio is concerned. The compression ratio of the carbon dioxide cycle is 4.5, while that of HFC-125 cycle is 9.4, which is rather higher than ordinary design.

TABLE I
PERFORMANCE OF CARBON DIOXIDE AND HFC-125 HEAT PUMPS UNDER THEIR MAXIMUM HEATING COP CONDITIONS

| | | $T_{HO}$ =90°C | | $T_{HO}$ =150°C | |
|---|---|---|---|---|---|
| | | $CO_2$ Transcritical | HFC-125 Transcritical | $CO_2$ Transcritical | HFC-125 Transcritical |
| $P_{crit}$ | kPa | 7,377.3 | 3,617.7 | 7,377.3 | 3,617.7 |
| $(UA)_{GC}$ | kW/K | 16.5 | 16.9 | 18.5 | 19.8 |
| $(UA)_E$ | kW/K | 13.5 | 13.1 | 11.5 | 10.2 |
| $T_1$ | °C | 43.1 | 43.5 | 44.9 | 47.7 |
| $T_2$ | °C | -1.7 | -2.5 | -3.0 | -3.5 |
| $T_3$ | °C | 3.3 | 2.5 | 2.0 | 1.5 |
| $T_4$ | °C | 143.9 | 109.7 | 192.2 | 164.0 |
| $P_1$ (=$P_4$) | kPa | 14,955.8 | 5,803.8 | 22,813.1 | 19,297.7 |
| $P_2$ (=$P_3$) | kPa | 3,328.9 | 620.1 | 3,220.1 | 600.0 |
| $P_1/P_2$ | - | 4.5 | 9.4 | 7.1 | 32.2 |
| $T_{CO}$ | °C | 5.7 | 5.5 | 5.8 | 6.1 |
| $\dot{m}_r$ | kg/s | 0.86 | 1.6 | 0.79 | 1.5 |
| $x_2$ | - | 0.42 | 0.4 | 0.38 | 0.5 |
| $\Delta T_{m,GC}$ | °C | 12.7 | 12.4 | 12.5 | 11.6 |
| $\Delta T_{m,E}$ | °C | 9.3 | 9.9 | 10.6 | 11.3 |
| $\Delta T_{min,GC}$ | °C | 3.1 | 3.5 | 4.9 | 7.7 |
| $\Delta T_{min,E}$ | °C | 6.7 | 7.5 | 8.0 | 8.5 |
| $Q_{GC}$ | kW | 209.3 | 209.3 | 230.2 | 230.2 |
| $Q_E$ | kW | 125.5 | 130.6 | 122.5 | 114.8 |
| $W$ | kW | 83.8 | 78.7 | 107.7 | 115.4 |
| $E_{D,E}$ | kW | 3.9 | 4.4 | 4.4 | 4.4 |
| $E_{D,GC}$ | kW | 8.4 | 6.3 | 5.9 | 4.3 |
| $E_{D,Comp}$ | kW | 15.6 | 15.8 | 18.1 | 20.5 |
| $E_{D,Exp}$ | kW | 16.0 | 12.9 | 20.2 | 26.4 |
| $COP_H$ | - | 2.50 | 2.66 | 2.14 | 1.99 |

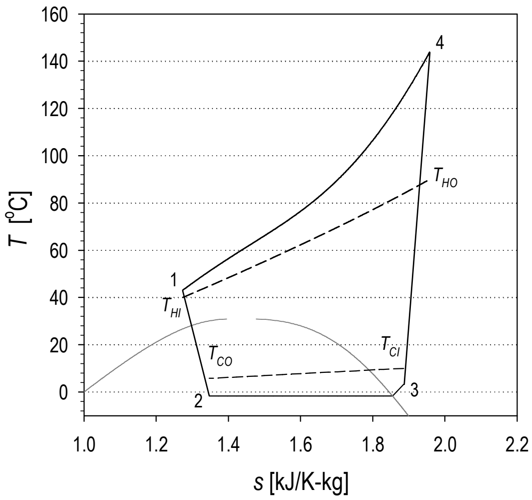


Fig. 4 Optimized carbon dioxide cycle on a T-s diagram when $T_{HO}$ =90°C

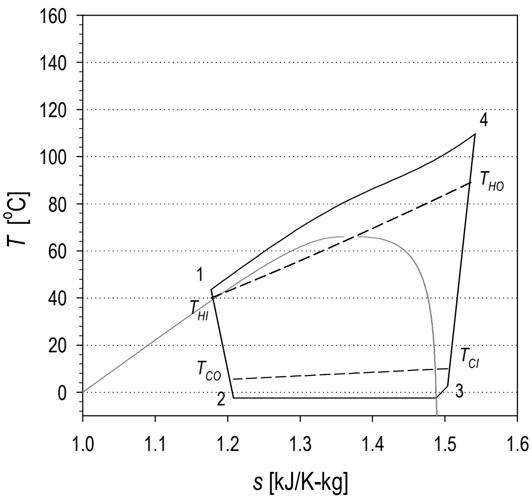


Fig. 5 Optimized HFC-125 cycle on a T-s diagram when $T_{HO}$ =90°C

On the other hand, exergy destruction rates during compression process or evaporation process show no significant difference between the two cycles under the simulation conditions considered in the present study.

Meanwhile, when nk exit temperature $T_{HO}$ is fixed at 150°C, the carbon dioxide cycle outperforms the HFC-125 cycle. In this case, HFC-125 cycle's high $E_{D,Exp}$ value accounts for the poor performance of HFC-125 cycle. Furthermore, HFC-125 cycle's high compression ratio of about 32 can be a technical barrier from the viewpoint of real implementation.

## IV. Conclusion

In order to compare the performance of the carbon dioxide and HFC-125 heat pumps for medium-and high-temperature heating, both heat pump cycles were optimized in terms of heating COP by simulation method. When the heat sink exit temperature is fixed at 90ºC, the HFC-125 heat pump has 6% higher heating COP than carbon dioxide heat pump. The main reason for this is that the exergy destruction rate during expansion process of carbon dioxide cycle is higher than that of HFC-125 cycle. However, exergy destruction rates during compression process or evaporation process show no significant difference between the two cycles under the simulation conditions considered in the present study. For a high temperature heating, the carbon dioxide cycle looks promising due to HFC-125 cycle's high compression ratio and relatively low performance.

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