Performance simulation of CO₂ transcritical cooling system with mechanical subcooling cycle using low GWP refrigerants

Sun-Ik NA, Gu Hwang KANG, Min Soo KIM[*][*]

Department of Mechanical and Aerospace Engineering, Seoul National University, Seoul 08826, Republic of Korea

Abstract

The conventional automobile cooling systems using CO_2 as a refrigerant are hard to achieve proper cooling performance at hot summer season. In order to enhance the of CO_2 cooling systems, a mechanical subcooling cycle is proposed for application to vehicles with limited installation space. To analyze performance of the system, numerical models were built to predict the COP by changing the design parameters and heat exchanger arrangement. In addition, the maximum COP is determined by using an optimal method. The results show that the COP improves when applying the subcooling cycle, and the maximum COP is obtained based on the thermodynamic states at the outlets of the gas cooler and the sub-cooler.

Keywords: Low GWP; Carbon dioxide (CO₂); Mobile air-conditioning systme(MAC); Heat pump; Mechanical subcooling cycle

1. Introduction

In order to restrain global warming, F-gases with GWPs greater than 150 are prohibited that use it for vehicles by the mobile air conditioning system (MACs) Directive of EU. Also, the Kigali amendment to the Montreal Protocol and F-gas regulation of EU are required to cut the usage of the high GWP refrigerants. With these tightened environmental regulations, the development of the refrigerant cooling system using low GWP refrigerants become increasingly important.

The carbon dioxide (CO_2) have been attracted because its GWP is just 1. Also, the CO_2 boosted attention since the German Association of the Automotive Industry (VDA) officially announced that suitable air-conditioning systems for a future vehicle would apply CO_2 as a refrigerant in 2007. Since then, Kim *et al.* [1] deal with the effects of the operating parameters on the cooling performance for an automotive air conditioning system. A refrigeration system using CO_2 as a refrigerant, which is cheap and easily available, is a good choice if its performance is comparable to that of other systems. In this sense, a dedicated mechanical subcooling system is introduced in this study in an attempt to improve the performance of the CO_2 cooling system. The dedicated mechanical subcooling cycle in the CO_2 cooling system is an additional vapor compression cycle which uses a small charge amount of the low-GWP refrigerant. Thornton *et al.* [2] explored the optimum value of the subcooling evaporator temperature by developing a numerical model for an ideal subcooling cycle in consideration of supermarket applications. Khan and Zubair [3] presented that the performance of a system is related to the refrigerant saturation temperature of the sub-cooler and is enhanced as compared to a simple CO_2 cooling cycle. Despite the dedicated subcooling cycle has been studied, there are relatively few studies on automobile applications with it.

In this paper, in order to obtain the maximum performance of the mobile air conditioning system (MACs), the numerical model is built, and performance analysis is conducted using an optimal method. Also, the influence of the low GWP refrigerants such as R-290, R-600a, R-1270, R-32, R-1234yf, and R-1234ze(E) on the subcooling cycle is investigated. In addition to the above points, this paper investigates the impact of the install configuration of heat exchangers on system efficiency.

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[*] Corresponding author. Tel.: +82 2 880 8362; fax: +82 2 880 6046.
E-mail address: minskim@snu.ac.kr (M. S. Kim) (Corresponding author).

2. System configuration and modeling

2.1. System configuration

The refrigeration system with mechanical subcooling consists of two as shown in Fig. 1(a). Each cycle uses a compressor, expansion valve, and heat exchangers. These two cycles are coupled with each other *via* a sub-cooler. The main cycle, as indicated by the bold line in Fig. 1(a), cools down the automotive cabin through heat exchange at the evaporator using CO_2. Meanwhile, the other cycle, named as the subcooling cycle, operates for the purpose of decreasing the quality of the CO_2 entering the evaporator of the main cycle by absorbing the heat at the sub-cooler.
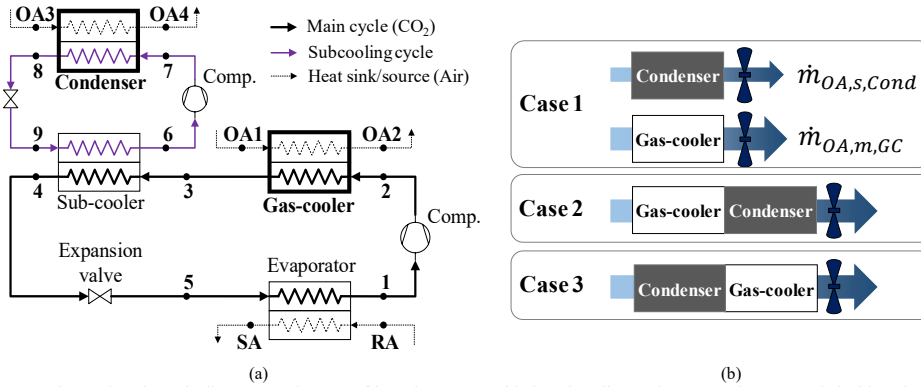

(a)   (b)

Fig. 1. The schematic diagram; (a) the CO_2 refrigeration system with the subcooling cycle. Two cycles are coupled with each other *via* a sub-cooler. (b) The layout cases according to the configuration of heat exchangers

For the refrigeration system using mechanical subcooling, it is necessary to install an additional condenser as compared to the conventional simple CO_2 cycle. This requirement is not a severe problem for a stationary system; however, for a vehicle, it should be considered that limited space is available for heat exchange with ambient air. Thus, the simulation cases were divided according to the heat exchanger configuration, as shown in Fig. 1(b). Case 1 is the case with two heat exchangers, the gas cooler of the main cycle and the condenser of the subcooling cycle, and they are located parallel with their faces to the outside of the vehicle, and the fans are operated separately. Meanwhile, in cases 2 and 3, the two heat exchangers are installed in a serial configuration, both cases have only one fan, and the air mass flow rate is the same through the two heat exchangers. Case 2 is that the gas cooler is located in front of the condenser of the subcooling cycle. Case 3 is that the gas cooler and the condenser are counterchanged compared with case 2.

2.2. System configuration

The energy balance equations for compressors and heat exchangers such as the gas cooler, sub-cooler, evaporator of the main cycle, and the condenser of the subcooling cycle are given in the following Eqs. (1)-(6):

$$\dot{W}_{m,Comp} = \dot{m}_{CO_2}(h_2 - h_1) = \dot{m}_{CO_2}(h_{2i} - h_1)\eta_{m,Comp} \qquad (1)$$
$$\dot{W}_{s,Comp} = \dot{m}_s(h_7 - h_6) = \dot{m}_s(h_{7i} - h_6)\eta_{s,Comp} \qquad (2)$$
$$\dot{Q}_{m,Evap} = \dot{m}_{CO_2}(h_1 - h_5) = \dot{m}_{RA}(h_{RA} - h_{SA}) \qquad (3)$$
$$\dot{Q}_{m,GC} = \dot{m}_{CO_2}(h_2 - h_3) = \dot{m}_{OA,m,GC}(h_{OA2} - h_{OA1}) \qquad (4)$$
$$\dot{Q}_{s,Cond} = \dot{m}_s(h_7 - h_8) = \dot{m}_{OA,s,Cond}(h_{OA4} - h_{OA3}) \qquad (5)$$
$$\dot{Q}_{SC} = \dot{m}_{CO_2}(h_3 - h_4) = \dot{m}_s(h_6 - h_9) \qquad (6)$$
$$COP_{c,AC} \equiv \frac{\dot{Q}_{m,Evap}}{\dot{W}_{m,Comp} + \dot{W}_{s,Comp}} \qquad (7)$$

If the compressor performance is calculated by using a constant isentropic efficiency in Eqs. (1) and (2), the results might be different from actual situations because it considers only the inlet and outlet states of the compression process. The polytropic efficiency is known as it has the advantage of allowing real property effects to be included in a natural way because the polytropic efficiency calculates and integrates the work for

each differential step of the compression process. The polytropic efficiency can be translated analytically into an isentropic efficiency for a specific pressure ratio [4], as shown in Eq. (8).

$$\eta = \frac{W_{12,i}}{W_{12}} = \frac{\left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)} - 1}{\left(\frac{P_2}{P_1}\right)^{\frac{1}{\eta_{PC}}\left(\frac{k-1}{k}\right)} - 1} \tag{8}$$

The *UA* method is used in the model to obtain the results regarding the effect of varying the size of each heat exchanger. Because the *UA* values can represent the sizes of the heat exchangers [5-7].

All of the heat exchangers within the cycle were modeled to have countercurrent flow and were sliced into many segments so as to calculate the thermal properties of the fluid flowing inside. The *UA* values of a heat exchanger were calculated by summing the *UA* values of a segment as shown in the following Eqs. (9)-(12):

$$\sum_{k=0}^{n} UA_k = \sum_{k=0}^{n} \frac{\dot{Q}_k}{LMTD_k} \tag{9}$$

$$LMTD_k = \frac{\Delta T_k - \Delta T_{k+1}}{\ln(\Delta T_k/\Delta T_{k+1})} \tag{10}$$

$$\Delta T_k = T_{H,k} - T_{C,k} \tag{11}$$

$$\Delta T_{k+1} = T_{H,k+1} - T_{C,k+1} \tag{12}$$

In order to obtain the maximum COP, the generalized pattern search method, which is a direct search algorithm, was used in this paper. The direct search is a suitable method for solving non-linear optimization problems because it is not required any information about the gradient of the objective function. The simulation was performed through several iterations until the variables were converged within the error bound.

The simulation model was developed using the MATLAB program, and the thermal properties were loaded from the REFPROP database [8]. The operating condition and constraints were set up for calculating a reference result. In order to compare the calculation results under various conditions, the cooling load of the evaporator in the main cycle was kept at 5 kW. The equivalent thermal conductance of the gas cooler ($UA_{m,GC}$) and the evaporator ($UA_{m,Evap}$) were designed 500 W/K in the main cycle. In the subcooling cycle, the equivalent thermal conductance of the condenser ($UA_{s,Cond}$) and the sub-cooler (UA_{SC}) were sat as 100 W/K.

3. Performance analysis

3.1. Optimal operating conditions

According to the literature [9, 10], the cooling performance is depended on the thermal state of the CO_2 at the gas cooler outlet, i.e., state 3 in Fig. 1(a). To find the optimum operation condition, new variables are introduced as following Eqs. (13, 14).

$$\Delta T_{3-4} \equiv T_3 - T_4 \tag{13}$$

$$\Delta T_{3-OA1} \equiv T_3 - T_{OA1} \tag{14}$$

The CO_2 temperature difference between the inlet and outlet of the sub-cooler is defined as shown in Eq. (13). It is proportional relation with the heat capacity of the sub-cooler. And the temperature difference between the CO_2 and air at the gas cooler outlet (ΔT_{3-OA1}) is defined as Eq. (14). The CO_2 pressure of the gas cooler is mainly affected by ΔT_{3-OA1} and the equivalent thermal conductance of the gas cooler ($UA_{m,GC}$). To explore the influence of ΔT_{3-4} and ΔT_{3-OA1}, the simulation was conducted under the same condition except the values of ΔT_{3-4} and ΔT_{3-OA1}. As a result, the COP changed in a concave downward shape according to two variables of ΔT_{3-OA1} and ΔT_{3-4}, which means that the maximum point exists according to not only the temperature and pressure of the gas cooler outlet, but also the decreased temperature by the subcooling cycle.

3.2. Refrigerant for the subcooling cycle

To investigate the effect by the refrigerant type of the subcooling cycle, the model covers the refrigerants with not only natural refrigerants but also R-1234yf, R-1234ze(E), R-152a, and R-32 which are regarded as promising refrigerants. Fig. 2 displays the results of the simulation according to the ratio of the refrigerant flow rate (ϕ) which is defined as ratio of the subcooling cycle refrigerant mass flow rate ($\dot{m}_s$) to the CO_2 mass flow

rate in the main cycle($\dot{m}_m$), i.e., $\phi \equiv \dot{m}_s / \dot{m}_m$. The result reveals the reveal that the type of refrigerant in the subcooling does not show a significant difference. Even though the cases of R-600, R-601a, and R-610 were obtained higher efficiency, these refrigerants would be not recommended because the evaporating pressures are lower than an atmosphere. Even if there are minute leakage, air (non-condensation gas) and moisture can enter the system. Then injected non-condensation gas drops the cooling capacity and moisture cause corrosive and harmful to system components [11].
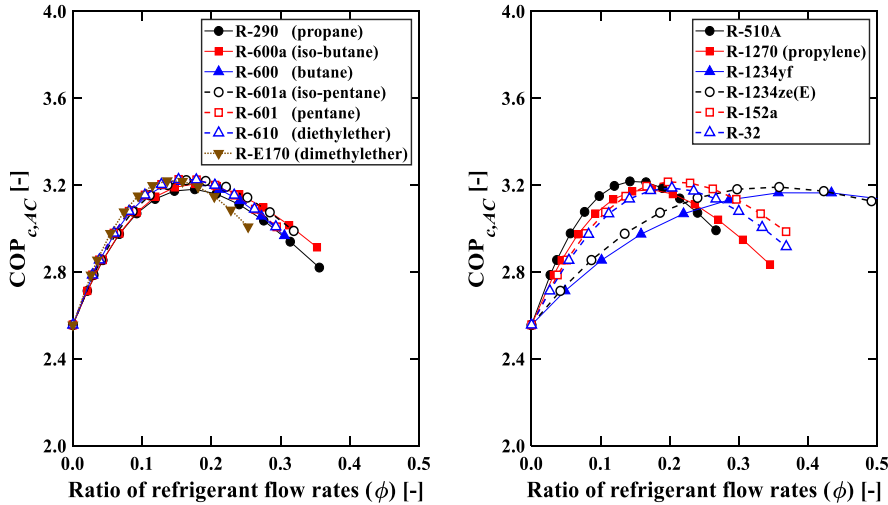

Fig. 2. The COP comparison with various refrigerants versus the ratio of refrigerant flow rate under the conditions.
($UA_{m,GC}$ = 500 kW/K, $UA_{s,Cond}$ = 100 kW/K, ΔT_{3-OA1} = 3℃)

While the optimum ratios of the refrigerant flow rate (ϕ_{opt}) of the hydrocarbon and R-32 were calculated similar values, the ϕ_{opt} of the R-1234yf and R-1234ze(E) were about twice. This result is obtained because the specific latent heat of vaporization of R-1234yf and R-1234ze(E) is roughly half with hydrocarbons'. In other words, the R-1234yf and R-1234ze(E) require a higher mass flow rate than hydrocarbons and R-32 to obtain the maximum COP.

3.3. Performance variation according to the heat exchanger configuration

In order to investigate the performance of the refrigeration system, the analysis was conducted by changing the configuration of heat exchangers. This analysis would be particularly necessary for automotive applications where the installation space for the mobile air conditioning system (MACs) is limited.

In case 1, the parallel configuration with two fans are applied with different capacities or operated at different revolution speeds. The air temperature of the gas cooler of the main and the condenser of the subcooling cycle is 35℃ in both cases, i.e., $T_{OA1} = T_{OA3}$ =35℃. The air mass flow rate of the gas cooler in the main cycle, $\dot{m}_{OA,m,GC}$, is determined to reach the air temperature from 35 to 50℃ in the gas cooler. Meanwhile, the air mass flow rate of the condenser in the subcooling cycle, $\dot{m}_{OA,s,Cond}$, is relatively calculated by the relative ratio of the air mass flow rate, as shown below in Eq. (15).

$$\psi \equiv \dot{m}_{OA,s,Cond} / \dot{m}_{OA,m,GC} \qquad (15)$$

As a result, the COP was lower when it was placed in series than when it was placed in parallel. In case 2, the discharged air from the gas cooler is blowing into the subcooling cycle in order to form a higher condensation temperature than case 1, thereby increasing the pressure ratio of the subcooling cycle compressor and lowering the COP. In case 3, the air from the condenser of the subcooling cycle flows into the gas cooler of the main cycle. The increased air temperature of the gas cooler inlet leads to the raised pressure of the gas cooler and the reduced efficiency of the system. The results of case 3, in which the condenser of the subcooling cycle is arranged first, are not better than those of case 2. The first reason is that the performance is more sensitive to the pressure ratio of the main cycle due to the fact that the CO_2 refrigerant mass flow rate is greater

than the subcooling cycle refrigerant mass flow rate. Another reason is the thermodynamic property of the CO_2, as the thermodynamic state of the CO_2 gas cooler outlet exists above the CO_2 critical point. Near the critical point, the thermodynamic properties change drastically. For this reason, not even a slight difference from the optimum condition will affect the COP more sensitively.

The COP is improved when the air mass flow rate is high, but the power consumption of the fan should be considered for calculating the efficiency of the system. For this purpose, the system COP was newly defined as shown in Eq. (15) by adding the fan power consumption in the denominator.

$$COP_{c,sys} \equiv \frac{\dot{Q}_{m,Evap}}{\dot{W}_{m,Comp}+\dot{W}_{s,Comp}+\dot{W}_{Fan}} \tag{16}$$

The power consumption of a fan is strongly dependent on its specification and characteristics. Nevertheless, based on the fan laws [12], the power consumption of a fan is proportional to the cube of rotational speed when the density of the air and the diameters of the fans are constant, as shown in Eqs. (17)-(19):

$$\frac{\dot{W}_{Fan,\beta}}{\dot{W}_{Fan,\alpha}} = \left(\frac{D_{Fan,\beta}}{D_{Fan,\alpha}}\right)^5 \left(\frac{N_{Fan,\beta}}{N_{Fan,\alpha}}\right)^3 \left(\frac{\rho_{Fan,\beta}}{\rho_{Fan,\alpha}}\right) \tag{17}$$

$$\frac{\dot{V}_{Fan,\beta}}{\dot{V}_{Fan,\alpha}} = \left(\frac{D_{Fan,\beta}}{D_{Fan,\alpha}}\right)^3 \left(\frac{N_{Fan,\beta}}{N_{Fan,\alpha}}\right) \tag{18}$$

$$\frac{\dot{W}_{Fan,\beta}}{\dot{W}_{Fan,\alpha}} \cong \left(\frac{N_{Fan,\beta}}{N_{Fan,\alpha}}\right)^3 = \left(\frac{\dot{V}_{Fan,\beta}}{\dot{V}_{Fan,\alpha}}\right)^3 \cong \left(\frac{\dot{m}_{Fan,\beta}}{\dot{m}_{Fan,\alpha}}\right)^3 \tag{19}$$

In order to consider the fan power, a specific commercial radiator cooling fan was applied in the numerical model. It can blow about 0.89 kg/s by consuming 120 W at the 2,250 RPM. After considering the power consumption of fans, the system COPs are at their maximum points at a certain total air mass flow rate, as shown in Fig. 3. This result shows that fan power should be a design parameter of focus in the design of a MAC.
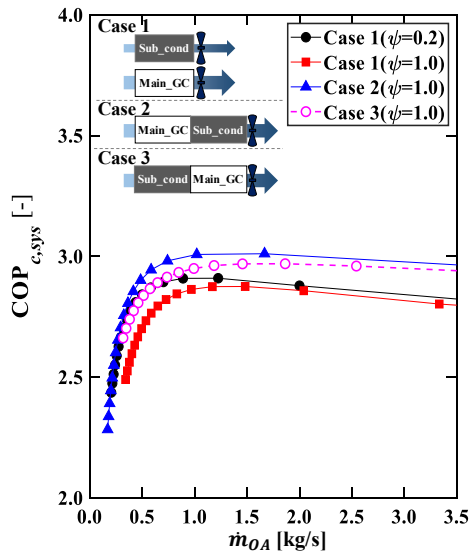

Fig. 3. The system performance ($COP_{c,sys}$) variation including the fan power consumption according to the total air mass flow rate. In contrast with $COP_{c,AC}$ which is calculated in Fig. 11, the $COP_{c,sys}$ is additionally considered energy consumption by the fan when obtaining the system efficiency. ($\dot{m}_{OA,m,GC}$).

4. Conclusions

This paper was conducted the thermodynamic analysis to investigate the performance of the CO_2 refrigeration system with the mechanical subcooling cycle using Low GWP refrigerants for a vehicle. The simulation was also conducted varied on the arrangement of the heat exchangers.

The results of this study show that the performance of CO_2 cooling system will be about 25% enhanced by using the subcooling cycle with the optimum operating condition. In order to obtain the optimum result, parameters such as mass flows of refrigerant, CO_2 temperatures and pressures at the gas cooler outlet and the sub-cooler outlet were variated. When the refrigerant mass flow rate in the subcooling cycle is optimum, the refrigerant type does not have a substantial effect on the performance. However, the systems using R-1234yf and R-1234ze(E) require a higher mass flow rate than hydrocarbons and R-32 to have the maximum COP. Because the specific latent heat of vaporization of R-1234yf and R-1234ze(E) is roughly half with hydrocarbons'.

Meanwhile, in order to implement the proposed mechanical subcooling system, heat exchangers would be additionally installed. If installation space is limited in a vehicle, design parameters and configuration should be considered about heat exchangers and fans. For this reason, performances were calculated according to heat exchanger arrangements. When two heat exchangers are installed serially, the gas cooler of the main cycle should be set to face outside in front of the condenser of the subcooling cycle in order to obtain higher COP. In addition, this study suggests that the power consumption of fans should be considered in the design of the mobile air conditioner system.

Acknowledgements

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