Influence of Operation Temperatures on Performance of a CO₂ Heat Pump Water Heating System*

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Abstract

An air-to-water CO₂ heat pump is used in combination with a domestic hot water storage tank, and the performance of this water heating system is affected significantly not only by instantaneous ambient air and city water temperatures but also by hourly changes in domestic hot water demand and temperature distribution in the storage tank. In this paper, the influence of operation conditions such as outlet water temperature during operation and inlet water temperature for shutdown of a CO₂ heat pump on the performance of a water heating system is investigated by numerical simulation. The trade-off relationship between the system efficiency and the volume of unused domestic hot water as criteria for the system performance is clarified in relation to the operation conditions. The influence of seasonal changes in ambient air and city water temperatures on the trade-off relationship is also clarified.

Key words: Heat Pump, Refrigerant, Carbon Dioxide, Thermal Storage, System Performance, Numerical Simulation

1. Introduction

Air-to-water heat pumps using CO₂ as a natural refrigerant have been developed and commercialized. They are expected to contribute to energy saving in domestic hot water (DHW) supply^{(1), (2)}.

Many theoretical and experimental studies have been conducted for the performance analysis only on CO₂ heat pumps^{(3)–(11)}. However, a CO₂ heat pump is used in combination with a DHW storage tank so that it can supply DHW even in the case of a sudden increase in DHW demand. The performance of this water heating system is affected significantly not only by instantaneous ambient air and city water temperatures but also by hourly changes in DHW demand and temperature distribution in the DHW storage tank. Therefore, it is necessary to conduct the performance analysis on water heating systems in consideration of these items. It takes much time to do it by experiment, and it is expected that numerical simulation enables one to do it very efficiently.

Few studies have been conducted for the performance analysis on water heating systems^{(12), (13)}. In these studies, simulation models have been proposed, and results obtained by numerical simulation have been compared with those by experiment. In addition, the influence of uncontrollable ambient conditions such as ambient air and city

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water temperatures on the system performance has been clarified. However, the system performance has been investigated under fixed operation conditions such as outlet water temperature during operation and inlet water temperature for shutdown of a CO₂ heat pump. These operation conditions are controllable and may affect the system performance significantly. For example, a high outlet water temperature during operation and a high inlet water temperature for shutdown increase the volume of DHW produced, which is suitable to avoid a shortage of DHW, but decrease the coefficient of performance (COP) of the CO₂ heat pump and consequently the system efficiency, and vice versa. Therefore, it is important to investigate which operation conditions should be adopted to improve the system performance.

The objective of this paper is to investigate the influence of the aforementioned operation conditions on the performance of a CO₂ heat pump water heating system by numerical simulation. A simulation model of the system is created by combining a simplified model of a CO₂ heat pump with a detailed model of a DHW storage tank, and the values of model parameters are estimated based on measured data for existing devices. This modeling results in a set of nonlinear differential algebraic equations, and it is solved by a hierarchical combination of the Runge-Kutta and Newton-Raphson methods. A numerical study is conducted for a typical daily profile of DHW demand, and the system performance is evaluated by changing the operation conditions. The trade-off relationship between the system efficiency and the volume of unused DHW as criteria for the system performance is investigated in relation to the operation conditions. The influence of seasonal changes in ambient air and city water temperatures on the trade-off relationship is also investigated. Discussion about the trade-off relationship is made based on hourly changes in temperature distributions in the DHW storage tank under various operation conditions.

2. CO₂ Heat Pump Water Heating System

Figure 1 shows the configuration of the CO₂ heat pump water heating system investigated in this paper. This system is composed of a CO₂ heat pump and a DHW storage tank. The CO₂ heat pump is composed of a compressor, a gas cooler, an expansion valve, and an evaporator. The system is equipped with a fan, a pump, and motors M1 to M3 as auxiliary machinery. Here, inlet and outlet water is defined as water at the inlet and outlet of the gas cooler, respectively. The system heats water using the refrigeration cycle of the CO₂ heat pump, stores DHW in the storage tank, and supplies it to a tapping site.

Fig. 1 Configuration of CO₂ heat pump water heating system

3. Numerical Simulation

3.1 Simulation model

A simplified static model is adopted for the CO₂ heat pump⁽¹²⁾. Although the CO₂ heat pump includes the aforementioned four components, they are not taken into account explicitly, and it is expressed by one model. Figure 2 shows the model of the CO₂ heat pump. The mass flow rates and temperatures of water at the inlet and outlet $\dot{m}_{\text{HPi}}$, $\dot{m}_{\text{HPo}}$, T_{HPi} , and T_{HPo} , COP η_{HP} , heat output $\dot{Q}_{\text{HP}}$, power consumption $\dot{W}_{\text{HP}}$, and ambient air temperature T^a are adopted as basic variables whose values are to be determined. The mass and energy balance relationships as well as the energy input and output relationship are adopted as basic equations to be satisfied. The remaining equations to be considered are approximate functions of the power consumption and COP, and they are expressed in relation to the ambient air and inlet water temperatures for each discrete value of the outlet water temperature. Although detailed explanation about this modeling is omitted here, the modeling results in a set of nonlinear algebraic equations, which is expressed by

$$f_{\text{HP}}(\mathbf{y}_{\text{HP}}(t), t) = \mathbf{0} \quad (1)$$

at each time t , where f_{HP} is the vector for the aforementioned equations, and $\mathbf{y}_{\text{HP}}$ is the vector for the aforementioned variables.

A detailed dynamic model is adopted for the DHW storage tank^{(12), (13)}. The objective of this paper is to analyze the performance of the water heating system, and is not to analyze the three-dimensional temperature distribution in the DHW storage tank. Therefore, a one-dimensional simulation model is used to analyze the performance of the water heating system in a reasonable computation time. Figure 3 shows the model of the DHW storage tank. To consider the one-dimensional vertical temperature distribution in the DHW storage tank, it is vertically divided into J control volumes with the same volume, in each of which the water temperature is assumed to be uniform. It is also assumed that the heat transfer $\dot{Q}_{\text{ST}j}$ ($j = 1, 2, \dots, J$) occurs by water flow and heat conduction as well as heat loss from the tank surface. The mass flow rates and temperatures of water for each control volume $\dot{m}_{\text{ST}j}$ ($j = 1, 2, \dots, J-1$) and $T_{\text{ST}j}$ ($j = 1, 2, \dots, J$) are adopted as basic variables whose values are to be determined. In addition, the mass flow rates and temperatures of water at the inlet and outlet of the top and bottom of the DHW storage tank $\dot{m}_{\text{STi}}^t$, $\dot{m}_{\text{STo}}^t$, $\dot{m}_{\text{STi}}^b$, $\dot{m}_{\text{STo}}^b$, T_{STi}^t , T_{STo}^t , T_{STi}^b , and T_{STo}^b are adopted as variables. The mass and energy balance relationships for each control volume are adopted as basic equations to be satisfied. Although detailed explanation about the modeling is omitted here, the modeling results in a set of nonlinear differential algebraic equations, which is expressed by

Fig. 2 Model of CO₂ heat pump

Fig. 3 Model of DHW storage tank

$$f_{ST}(x_{ST}(t), x'_{ST}(t), y_{ST}(t), t) = 0 \quad (2)$$

where f_{ST} is the vector for the aforementioned equations, x_{ST} is the vector for the variables with their derivatives: the temperatures of water for all the control volumes, y_{ST} is the vector for all the other variables without their derivatives, x'_{ST} is the derivative of x_{ST} with respect to time t .

At the connection points between the CO₂ heat pump and DHW storage tank, connection conditions are taken into account to equalize the values of the corresponding variables. The outlet water temperature is given as an operation condition. The city water temperature and the mass flow rate of DHW from the storage tank to the tapping site are given as boundary conditions. The ambient air temperature is given as an ambient condition.

3.2 Solution method

The aforementioned modeling for the performance analysis by numerical simulation is conducted by a building block approach as follows: The component models for the CO₂ heat pump and DHW storage tank, and the substance model for water are defined independently; The system model is composed of the component and substance models as well as the connection, operation, boundary, and ambient conditions⁽¹⁴⁾.

The equations for the CO₂ heat pump are static, while those for the DHW storage tank are dynamic. Therefore, the modeling of the system results in a set of nonlinear differential algebraic equations, which is expressed by

$$\begin{cases} f(x(t), x'(t), y(t), t) = 0 \\ x(t_0) = x_0 \end{cases} \quad (3)$$

where f is the vector for all the equations composed of f_{HP} of Eq. (1) and f_{ST} of Eq. (2) as well as the connection, operation, boundary, and ambient conditions, x is the vector for the variables with their derivatives composed of x_{ST} in Eq. (2), y is the vector for all the other variables without their derivatives composed of y_{HP} in Eq. (1) and y_{ST} in Eq. (2), x' is the derivative of x with respect to time t , and x_0 is the initial value of x at the initial time t_0 . It is solved by a hierarchical combination of the Runge-Kutta and Newton-Raphson methods⁽¹⁴⁾.

3.3 Validation of simulation model

In advance of its application to a numerical study, the simulation model has been validated as follows.

An important thing is to investigate the validity of the one-dimensional simulation model for the DHW storage tank. The three-dimensional numerical simulation has also been conducted for another study⁽¹⁵⁾. The temperature distribution obtained by the one-dimensional numerical simulation has been different locally from that obtained by the three-dimensional one, especially near the inlet and outlet of the top and bottom of the DHW storage tank. However, since the flow rate of water through the DHW storage tank is small enough, the temperature distribution obtained by the one-dimensional numerical simulation has coincided well globally with that obtained by the three-dimensional one. In addition, it has coincided well globally with that obtained by the experiment, as described below. Therefore, the one-dimensional simulation model has been validated.

It is also important to determine the number of control volumes for the DHW storage tank J appropriately in relation to the sampling time interval for the Runge-Kutta method, so that the hourly change in the temperature distribution in the DHW storage tank due to the heat transfer by water flow and heat conduction indicates a real one. For this purpose, J has been set at 120, 200, and 240 for the sampling time interval of 10 s, and the hourly

change in the temperature distribution obtained by the numerical simulation has been compared with that by the experiment. It has been concluded that the hourly changes in the temperature distribution for $J = 200$ and 240 are in good agreement with each other as well as that by the experiment, and $J = 200$ has been used in the previous study on the comparison between the numerical simulation and experiment⁽¹²⁾.

Another important thing is to solve the set of nonlinear differential algebraic equations accurately. As aforementioned, the Runge-Kutta method is adopted at the upper level to solve the differential equations. This method has the 3rd order accuracy. In addition, the Newton-Raphson method is adopted at the lower level to satisfy the nonlinear algebraic equations accurately at each sampling time. Therefore, it is concluded that this solution method for the set of nonlinear differential algebraic equations has high accuracy.

4. Numerical Study

4.1 Conditions

A numerical study is conducted for four days in summer, mid-season, and winter, on each of which a typical daily profile of DHW demand repeats. This is because the results on the 4th day are similar to those on the 3rd day, and the periodic steady state appears.

Table 1 shows the specifications of the CO₂ heat pump water heating system. The rated heat output of the CO₂ heat pump is set at 4.50 kW. The volume of the DHW storage tank is set at 370 L. The values of model parameters included in the equations are estimated based on measured data for existing devices. As an example, Fig. 4 shows measured values and approximate functions for the power consumption and COP of the CO₂ heat pump in relation to the ambient air and inlet water temperatures for the outlet water temperature of 65 °C. Here, each value is relative to its measured one under the ambient air and the inlet/outlet water temperatures of 16, 15, and 65 °C, respectively. As aforementioned, the number of control volumes for the DHW storage tank is set at $J = 200$, and the sampling time interval for the Runge-Kutta method is set at 10 s. The values of model parameters have also been validated by the previous study on the comparison between the numerical simulation and experiment⁽¹²⁾.

Table 1 Specifications of CO₂ heat pump water heating system

Equipment	Specification	Value
CO ₂ heat pump	Rated heat output	4.50 kW
DHW storage tank	Volume	370 L
	Height	1.43 m
	Diameter	0.56 m
	Overall heat transfer coefficient	0.80 W/(m ² ·°C)
	Number of control volumes J	200

Fig. 4 Measured values and approximate functions for performance characteristics of CO₂ heat pump

Table 2 shows the ambient air and city water temperatures in summer, mid-season, and winter as the ambient and boundary conditions, respectively, which is prescribed by the Japan Refrigeration and Air Conditioning Industry Association (JRA)⁽¹⁶⁾. These values are assumed to be constant throughout the four days in each season. The hourly changes in the flow rate and temperature of DHW demand are given on each day as shown in Fig. 5, which is also prescribed by JRA⁽¹⁶⁾. Here, the height of each vertical line means the flow rate, as indicated. The temperature is shown above each vertical line. In addition, the thickness of each vertical line means the duration. These changes are assumed to be the same in all the seasons. As the boundary condition, the mass flow rate of DHW from the storage tank to the tapping site is calculated using these values. As the initial condition, the temperature of water in all the control volumes of the DHW storage tank is set at the city water temperature at 0:00 on the 1st day.

The system is assumed to be operated in the charging and tapping modes independently during the nighttime and daytime, respectively. In the charging mode, the outlet water temperature of the CO₂ heat pump is set at a certain value as the operation condition. This value is changed by 5 °C within the range between 65 and 85 °C. In addition, the CO₂ heat pump is started up at 2:00 and is shut down with a shutdown condition satisfied at an appropriate time before 7:00. The shutdown condition is that the inlet water temperature of the CO₂ heat pump attains a certain value. This value is changed by 10 °C within the range between 30 and 60 °C. The numerical simulation is conducted for all the combinations of the aforementioned values for the outlet water temperature during operation and the inlet water temperature for shutdown of the CO₂ heat pump.

4.2 Results and discussion

4.2.1 Influence of operation temperatures

First, the influence of the outlet water temperature during operation and the inlet water temperature for shutdown of the CO₂ heat pump on the system performance is investigated in mid-season.

Figure 6 shows the relationship between the system efficiency and the volume of unused DHW on the 4th day in mid-season for all the combinations of the values for the

Table 2 Ambient air and city water temperatures in each season

Season	(Unit: °C)	
	Ambient air	City water
Summer	25	24
Mid-season	16	17
Winter	7	9

Fig. 5 Hourly changes in flow rate and temperature of DHW demand

Fig. 6 Trade-off relationship between system efficiency and volume of unused DHW (4th day, mid-season)

outlet water temperature during operation and the inlet water temperature for shutdown. Here, the system efficiency is defined as the ratio of the heat output for DHW supply to the power consumption by the system, and the volume of unused DHW is defined as the one with a temperature of 42 °C obtained by mixing the DHW with temperatures higher than 42 °C in the storage tank and the city water at 24:00 after the end of DHW supply. In addition, relative values are used for these performance criteria, because the system performance of existing devices has still been improved by the technological development. Here, the value of each performance criterion is relative to that on the 1st day under the operation conditions that the outlet water temperature during operation is 85 °C, and that the inlet water temperature for shutdown is 60 °C. Figure 6 indicates a clear trade-off relationship between the system efficiency and the volume of unused DHW. In addition, in spite of different combinations of the values for the inlet and outlet water temperatures, and consequently different temperature distributions in the DHW storage tank, the trade-off relationship can almost be expressed by one curve.

The temperature distributions in the DHW storage tank are investigated for cases A to D chosen as shown in Table 3. Figures 7 (a) to (d) show the hourly changes in the temperature distributions in the DHW storage tank in the charging mode on the 4th day in mid-season for cases A to D, respectively. The profiles of temperature distributions for the layer of water with middle temperatures between the lowest and highest ones are different for the low and high inlet water temperatures for shutdown. For cases A and C with the low inlet water temperature for shutdown, the layer of water with middle temperatures is divided into two parts: The gradient of the temperature distribution, or the temperature change per height is small in the lower part of the layer; It is large in the upper part of the layer. On the other hand, for cases B and D with the high inlet water temperature for shutdown, the layer of water with middle temperatures is divided into three parts: The gradient of the temperature distribution is large in the lower part of the layer; It is small in the middle part of the layer; It is large again in the upper part of the layer. It is interesting that the system efficiency and volume of unused DHW have almost same values for cases B and C, although the profiles of temperature distributions for these cases are quite different.

Figure 8 shows the hourly changes in the COP of the CO₂ heat pump on the 4th day in mid-season for cases A to D. Here, the COP is defined as the ratio of the heat output to the power consumption by the CO₂ heat pump, and is relative to that at 2:00 for case C. The COP for cases A and B is larger than that for cases C and D at 2:00. This is because the outlet water temperature during operation for cases A and B is lower than that for cases C and D. The COP decreases to less than 50 % at the end of operation for cases B and D. This is because the inlet water temperature rises to 60 °C at that time. On the other hand, the COP is kept more than 80 % even at the end of operation for cases A and C. This is because the inlet water temperature is kept lower than 30 °C. As a result, the average COP for cases A and C is larger than that for cases B and D, respectively.

4.2.2 Influence of ambient and operation temperatures

Second, the influence of the ambient air and city water temperatures in all the seasons on the system performance is investigated in addition to that of the outlet water temperature

Table 3 Outlet water temperature during operation and inlet water temperature for shutdown in each case

Case	(Unit: °C)	
	Outlet water temp. during operation	Inlet water temp. for shutdown
A	65	30
B	65	60
C	85	30
D	85	60

during operation and the inlet water temperature for shutdown of the CO₂ heat pump.

Figure 9 shows the relationship between the system efficiency and the volume of unused DHW on the 4th day in all the seasons for all the combinations of the values for the outlet water temperature during operation and the inlet water temperature for shutdown. As the ambient air and city water temperatures increase, the trade-off relationship between the system efficiency and the volume of unused DHW rises. The trade-off relationship is located at the highest and lowest positions in summer and winter, respectively. This is because the COP of the CO₂ heat pump and the system efficiency increase with an increase in the ambient air temperature and decrease with an increase in the city water temperature, and the effect of the ambient air temperature is larger than that of the city water temperature. This is also because the heat output for DHW supply by the system decreases and the volume of unused DHW increases with an increase in the city water temperature. In addition, the trade-off relationship has larger changes in the system efficiency and the volume of unused DHW in summer than in winter. This is because of the following reasons.

Figure 10 shows the relationship between the system efficiency and the volume of

Fig. 7 Hourly changes in temperature distribution in DHW storage tank (charging mode, 4th day, mid-season)

Fig. 8 Hourly changes in COP (4th day, mid-season)

unused DHW on the 4th day in all the seasons for cases A to D. For cases A and C with the low inlet water temperature for shutdown, the ambient air and city water temperatures tend to affect the system efficiency more significantly than the volume of unused DHW. On the other hand, for cases B and D with the high inlet water temperature for shutdown, the ambient air and city water temperatures tend to affect the volume of unused DHW more significantly than system efficiency. These results can be understood based on the temperature distributions in the DHW storage tank as described later.

Figures 11 (a) to (d) show the hourly changes in the temperature distributions in the DHW storage tank in the charging mode on the 4th day in summer for cases A to D, respectively. Figure 12 shows the hourly changes in the COP of the CO₂ heat pump on the 4th day in summer for cases A to D. Figures 13 (a) to (d) show the hourly changes in the temperature distributions in the DHW storage tank in the charging mode on the 4th day in winter for cases A to D, respectively. Figure 14 shows the hourly changes in the COP of the CO₂ heat pump on the 4th day in winter for cases A to D.

According to Figs. 11 and 13, the temperature distributions in the DHW storage tank in summer and winter have features similar to those in mid-season. However, the gradient of the temperature distribution for the layer of water with middle temperatures in summer is smaller than that in mid-season, and that in winter is larger than that in mid-season. This is because, for example, since the city water temperature in summer is higher than that in mid-season, the heat output for DHW supply by the system is smaller, the period when the CO₂ heat pump is stopped is longer, and the heat transfer by heat conduction is relatively larger.

Generally, as described previously, the system efficiency and the volume of unused DHW increase with increases in the ambient air and city water temperatures. However, different features are found for different inlet water temperatures for shutdown. For cases A and C with the low inlet water temperature for shutdown, according to Figs. 11 (a) and (c) in summer, since the heat output for DHW supply is smaller but the difference between the inlet water temperature for shutdown and the city water temperature is smaller, the ratio of the operation time of the CO₂ heat pump with low values of the COP to the overall operation time is smaller, and since the gradient of the temperature distribution is smaller, the volume of unused DHW is not so larger. According to Figs. 13 (a) and (c) in winter,

Fig. 9 Trade-off relationship between system efficiency and volume of unused DHW (4th day, all seasons)

Fig. 10 Relationship between system efficiency and volume of unused DHW (4th day, cases A to D)

since the heat output for DHW supply is larger but the difference between the inlet water temperature for shutdown and the city water temperature is larger, the ratio of the operation time of the CO₂ heat pump with low values of the COP to the overall operation time is larger. This feature is also found in Figs. 12 and 14. Namely, for cases A and C, although the decreases in COP are small in summer and winter, the decrease rate in COP in winter is larger than that in summer. Therefore, for cases A and C, the ambient air and city water temperatures tend to affect the system efficiency more significantly than the volume of unused DHW.

On the other hand, for cases B and D with the high inlet water temperature for shutdown, according to Figs. 11 (b) and (d) in summer, since the difference between the inlet water temperature for shutdown and the city water temperature is smaller but the heat output for DHW supply is smaller, the ratio of the operation time of the CO₂ heat pump with low values of the COP to the overall operation time is larger, and since the gradient of the temperature distribution is not so smaller, the volume of unused DHW is larger. According to Figs. 13 (b) and (d) in winter, the difference between the inlet water temperature for shutdown and the city water temperature is larger but the heat output for

Fig. 11 Hourly changes in temperature distribution in DHW storage tank (charging mode, 4th day, summer)

Fig. 12 Hourly changes in COP (4th day, summer)

DHW supply is larger, the ratio of the operation time of the CO₂ heat pump with low values of the COP to the overall operation time is smaller. This feature is also found in Figs. 12 and 14. Namely, for cases B and D, although the decreases in COP are large in summer and winter, the decrease rate in COP in winter is almost equal to that in summer. Therefore, for cases B and D, the ambient air and city water temperatures tend to affect the volume of unused DHW more significantly than the system efficiency.

5. Conclusions

The influence of the operation conditions on the performance of a CO₂ heat pump water heating system has been investigated by numerical simulation. A simulation model of the system has been created by combining a simplified model of the CO₂ heat pump with a detailed model of a DHW storage tank, and the values of model parameters have been estimated based on measured data for existing devices. A numerical study has been conducted for a typical daily profile of DHW demand, and the system performance has been evaluated by changing the operation conditions.

Fig. 13 Hourly changes in temperature distribution in DHW storage tank (charging mode, 4th day, winter)

Fig. 14 Hourly changes in COP (4th day, winter)

The trade-off relationship between the system efficiency and the volume of unused DHW as criteria for the system performance has been clarified in relation to the outlet water temperature during operation and the inlet water temperature for shutdown. It has turned out that in spite of different combinations of the values for the inlet and outlet water temperatures, and consequently different temperature distributions in the DHW storage tank, the trade-off relationship can almost be expressed by one curve.

The influence of seasonal changes in ambient air and city water temperatures on the trade-off relationship has also been clarified. It has turned out that as the ambient air and city water temperatures increase, the trade-off relationship has larger changes in the system efficiency and the volume of unused DHW. It has also turned out that for cases with the low inlet water temperature for shutdown, the ambient air and city water temperatures tend to affect the system efficiency more significantly than the volume of unused DHW, and that for cases with the high inlet water temperature for shutdown, the ambient air and city water temperatures tend to affect the volume of unused DHW more significantly than the system efficiency.

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