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Influence of ambient temperatures on performance of a CO_2 heat pump water heating system

Ryohei Yokoyama[a,*], Takeshi Shimizu[a], Koichi Ito[b], Kazuhisa Takemura[c]

[a]Department of Mechanical Engineering, Osaka Prefecture University, 1-1 Gakuen-cho, Naka-ku, Sakai, Osaka 599-8531, Japan
[b]Advanced Research Institute for Science and Engineering, Waseda University, 3-4-1 Okubo, Shinjuku-ku, Tokyo 169-8555, Japan
[c]Research and Development Department, Kansai Electric Power Co., Inc., 3-11-20 Nakoji, Amagasaki, Hyogo 661-0974, Japan

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Abstract

In residential applications, an air-to-water CO_2 heat pump is used in combination with a domestic hot water storage tank, and the performance of this system is affected significantly not only by instantaneous ambient air and city water temperatures but also by hourly changes of domestic hot water consumption and temperature distribution in the storage tank. In this paper, the performance of a CO_2 heat pump water heating system is analyzed by numerical simulation. A simulation model is created based on thermodynamic equations, and the values of model parameters are estimated based on measured data for existing devices. The calculated performance is compared with the measured one, and the simulation model is validated. The system performance is clarified in consideration of seasonal changes of ambient air and city water temperatures.

Keywords: Heat pump; Refrigerant; CO_2; Thermal storage; System performance; Thermodynamics; Cycle analysis; Numerical simulation

1. Introduction

Air-to-water heat pumps using CO_2 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 of CO_2 heat pumps [3–11]. However, in residential applications, a CO_2 heat pump is used in combination with a DHW storage tank. Therefore, the performance of this system is affected significantly not only by instantaneous ambient air and city water temperatures but also by hourly changes of DHW consumption and temperature distribution in the DHW storage tank. It takes much time to analyze the performance by experiment, and it is expected that numerical simulation enables one to analyze the performance efficiently.

The objective of this paper is to analyze the performance of a CO_2 heat pump water heating system by numerical simulation. A simulation model of the system is created based on thermodynamic equations, and the values of model parameters are estimated based on measured data for existing devices. The model 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. The calculated performance is compared with the measured one, and the validity of the simulation model is investigated. In addition, the performance is analyzed in consideration of seasonal changes of ambient air and city water temperatures, and the influence of these ambient temperatures on performance criteria is investigated.

2. CO_2 heat pump water heating system

Fig. 1 shows the configuration of the CO_2 heat pump water heating system investigated in this paper. This system is composed of a CO_2 heat pump and a DHW storage tank (ST). The CO_2 heat pump is composed of a compressor

*Corresponding author. Tel.: +81 72 254 9229; fax: +81 72 254 9904.
E-mail address: yokoyama@ese.me.osakafu-u.ac.jp (R. Yokoyama).

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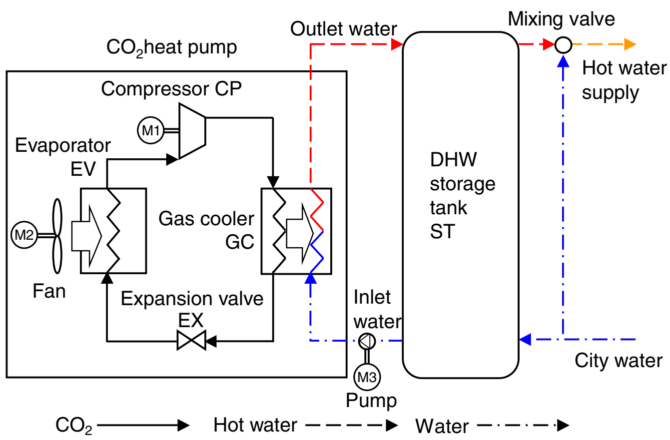

Fig. 1. CO_2 heat pump water heating system.

(CP), a gas cooler (GC), an expansion valve (EX), and an evaporator (EV). The system is equipped with a fan, a pump, and motors (M1∼3) as auxiliary machinery. Here, inlet and outlet water is defined as water at the inlet and outlet of the GC, respectively. The system heats water using the refrigeration cycle of the CO_2 heat pump, stores DHW in the ST, and supplies it to a tapping site.

3. Numerical simulations

3.1. Outline of modeling

For each component of the system, the mass flow rates $\dot{m}$ of CO_2 and water, the pressure p of CO_2, and the temperatures T of CO_2 and water are adopted as variables whose values are to be determined. In addition, the mass, pressure, and energy balance relationships are adopted as basic thermodynamic equations to be satisfied. If necessary, other relationships are adopted. The characteristics of the CO_2 heat pump are expressed by static equations, while those of the ST are expressed by dynamic ones. The values of model parameters included in the equations are estimated based on measured data for a CO_2 heat pump and a ST commercialized recently. At the connection points between the adjacent components, connection conditions are taken into account to equalize the values of the corresponding variables. Boundary conditions are taken into account to set the values of the corresponding variables. If necessary, ambient conditions are taken into account.

3.2. Component models

The following equations are taken into account for the components. A variable for each component is denoted by the subscript of the corresponding component symbol. In addition, variables at the inlet and outlet of each component are denoted by the subscripts i and o, respectively.

3.2.1. Compressor

A model of the CP is shown in Fig. 2. The ideal characteristics of the CP are expressed by an adiabatic change. In addition, the adiabatic efficiency is introduced to express the real characteristics in comparison with the ideal ones. Here, no heat loss from the CP is assumed. The mass and energy balance relationships for the real characteristics are expressed by

$$\left.\begin{aligned}&\dot{m}_{\mathrm{CPi}} = \dot{m}_{\mathrm{CPo}}\\&\dot{m}_{\mathrm{CPi}}h(p_{\mathrm{CPi}}, T_{\mathrm{CPi}}) + \dot{W}_{\mathrm{CP}} = \dot{m}_{\mathrm{CPo}}h(p_{\mathrm{CPo}}, T_{\mathrm{CPo}})\end{aligned}\right\}, \tag{1}$$

where $\dot{W}_{\mathrm{CP}}$ is the power consumption of the CP, and h is the specific enthalpy of CO_2. The energy and entropy balance relationships for the ideal characteristics by the adiabatic change are expressed by

$$\left.\begin{aligned}&\dot{m}_{\mathrm{CPi}}h(p_{\mathrm{CPi}}, T_{\mathrm{CPi}}) + \eta_{\mathrm{CP}}(p_{\mathrm{CPo}}/p_{\mathrm{CPi}})\dot{W}_{\mathrm{CP}}\\&\quad = \dot{m}_{\mathrm{CPo}}h(p_{\mathrm{CPo}}, T^{\mathrm{ad}}_{\mathrm{CPo}})\\&\dot{m}_{\mathrm{CPi}}s(p_{\mathrm{CPi}}, T_{\mathrm{CPi}}) = \dot{m}_{\mathrm{CPo}}s(p_{\mathrm{CPo}}, T^{\mathrm{ad}}_{\mathrm{CPo}})\end{aligned}\right\}, \tag{2}$$

where s is the specific entropy of CO_2, and the superscript ad denotes the adiabatic change. Since the adiabatic efficiency η_{CP} is considered to change with the pressure ratio $p_{\mathrm{CPo}}/p_{\mathrm{CPi}}$, η_{CP} is expressed as a function with respect to $p_{\mathrm{CPo}}/p_{\mathrm{CPi}}$ as follows:

$$\eta_{\mathrm{CP}}(p_{\mathrm{CPo}}/p_{\mathrm{CPi}}) = a_1 + b_1(p_{\mathrm{CPo}}/p_{\mathrm{CPi}}) + c_1(p_{\mathrm{CPo}}/p_{\mathrm{CPi}})^2, \tag{3}$$

where a_1, b_1, and c_1 are the coefficients of the quadratic equation. In addition, the relationship between the power consumptions of the CP and its driving motor is expressed by

$$\eta_{\mathrm{M1}}\dot{W}_{\mathrm{M1}} = \dot{W}_{\mathrm{CP}}, \tag{4}$$

where $\dot{W}_{\mathrm{M1}}$ and η_{M1} are the power consumption and efficiency of the driving motor, respectively.

3.2.2. Gas cooler

A model of the GC is shown in Fig. 3. The GC is regarded as a counter-flow heat exchanger. Here, it is necessary to consider a strong nonlinearity in the function of the specific enthalpy with respect to the temperature for the real CO_2 gas. For this purpose, the GC is divided into N control volumes with the same heat exchange area, in each of which the specific heat of CO_2 is assumed to be constant, and the rate of heat exchange between CO_2 and water is expressed as the product of the overall heat transfer coefficient, heat exchange area, and log-mean temperature difference. The pressure of CO_2 is assumed to be constant throughout the GC. Each control volume from the inlet of CO_2 is identified by the number $n = 1, 2, \ldots, N$,

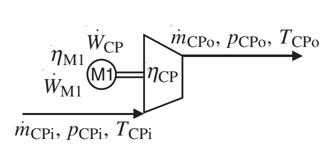

Fig. 2. Model of the compressor.