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Annual performance assessment of heat pump water heaters applying various refrigerants

Takaoki Suzuki[a], Zheng Ge[a], Muhamad Yulianto[b] Yoichi Miyaoka[c], Seiichi Yamaguchi[c], Kiyoshi Saito[c,*]

[a]Dept. of Applied Mechanics, Waseda University, Shinjuku-ku, Tokyo, 169-8555, Japan
[b]Waseda Research Institute for Science and Engineering, Waseda University, Shinjuku-ku, Tokyo, 169-8555, Japan
[c]Interdisciplinary institute for thermal energy conversion engineering and mathematics, Waseda University, Shinjuku-ku, Tokyo, 169-8555, Japan

Abstract

In order to help prevent global warming, greenhouse gas emissions must be reduced. Equivalent CO_2 emissions can be reduced by using heat pump water heaters instead of conventional gas boilers or electric-resistance water heaters. We focused on the differences in annual performance of heat pump water heaters using different refrigerants, including those that contained CO_2 and R32. When assessing different refrigerants, the effect of different climates in relation to the specific application area was taken into consideration. To this end, water supply and ambient temperatures were varied according to the particular climatic conditions to calculate annual performance. As a result, the effect of different refrigerants and climates on the annual performance of heat pump water heaters was revealed.

Keywords: Annual performance; Climate; Thermodynamic property; Refrigerants; Simulation;

Nomenclature

COP_h	Coefficient of Performance (heating)
c_p	Isobaric mass heat capacity kJ/kgK
E	Power consumption kWh
G	Mass flow rate kg/s
h	Specific enthalpy kJ/kg
K	Constant
P	Pressure MPa
$\dot{Q}$	Heating capacity kW
Q	Amount of heat transferred to water kWh
s	Specific entropy kJ/kgK
T	Temperature ° C
$\dot{W}$	Power input kW
ΔT	Temperature difference ° C
η	Efficiency

Subscripts

ad	Adiabatic
amb	Ambient
annual	Annual
ave	Average
con	Condenser
comp	Compressor
eva	Evaporator
i	i th hourly data in time series
pinch	Pinch
R	Refrigerant
series	time series
SH	Super heat
vap	Saturated vapor
W	Water
winlet	Water inlet

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* Corresponding author. Tel.: +81-3-5286-3259; fax: +81-3-5286-3259.
E-mail address: saito@waseda.jp

1. Introduction

To build a sustainable society and address recent environmental problems such as increasing energy demand and global warming, it is necessary to save energy. Widely applied heat pump technologies, which include highly energy-saving refrigeration, air-conditioning, and heating technologies, can contribute to energy conservation in society and alleviate environmental problems. The vapor compression heat pumps occupy an important position in energy conservation due to their energy saving performance and high versatility.

However, many refrigerants used in vapor compression heat pumps have adverse effects on the environment. In order to protect the environment, international efforts are being made to switch from conventional refrigerants with high environmental impact to refrigerants with low global warming potential (GWP). Various types of low GWP refrigerants, including mixed refrigerants, have been proposed with varying thermodynamic properties, transport properties, and environmental performance. However, the performance of a heat pump system optimized for newly developed refrigerants has not been clearly understood; thus, it is necessary to select a suitable refrigerant and to evaluate and compare the performance of each new refrigerant.

To evaluate and compare the performance of different heat pump systems, systems designed for different types of refrigerants must be considered. However, the typical drop-in test that is currently performed on new refrigerants is not sufficient to clarify the performance of the system. During drop-in tests, it is common that component specifications like dimensions of heat exchangers are not suitable for the target refrigerants. To deal with this problem, a heat pump performance simulation model, where the heat transfer performance of heat exchanger is considered by relying on the pinch temperature approach, is developed.

This study aims to establish a simulation-based performance evaluation method for vapor compression heat pumps. This method evaluates the energy consumption of heat pumps and compares their performance in various operating conditions and regions. The proposed evaluation method is developed with the aim of easily selecting suitable refrigerants and establishing system configuration guidelines.

2. Research target

This report covers heat pump water heaters using various refrigerants. A simulator was developed to evaluate the annual performance of heat pump water heaters. This simulator was built to compare the performance of different refrigerants, even when there was no actual machine designed for new refrigerants.

2.1. Heat pump water heaters

Compared to conventional electric-and gas water heaters, heat pump water heaters can release several times more heat than the input energy. An example of a widely used residential heat pump water heaters in Japan is the EcoCute. The EcoCute uses a supercritical CO_2cycle, enabling a highly efficient hot water supply under specific operating conditions. The CO_2 refrigerant has the advantages of a low GWP and high safety. However, there are disadvantages such as special measures being required for the refrigerant circuit due to high operating pressure and decreasing performance under conditions of higher inlet water temperature.

The performance of a heat pump varies in relation to the climatic conditions such as the ambient air temperature and the inlet water temperature. Accordingly, the beneficial effect of implementing a heat pump water heater can be expected to vary depending on the region. In addition, as suggested by the example of the EcoCute, the most suitable refrigerant for residential heat pump water heaters may vary depending on the region and application. Therefore, it is critical to assess the performance of residential heat pump water heaters using low GWP refrigerants in different climatic regions.

2.2. Low GWP refrigerants

In order to mitigate global warming, the production and use of refrigerants with high GWPs are limited, and the use of low GWP refrigerants is promoted. Currently, hydrofluorocarbons (HFCs) are popular refrigerants; however, they often have a high GWP. Various alternative refrigerants have been proposed to satisfy conflicting requirements, and evaluated for different application cases by simulations and experiments. [1] [2] [3]

However, the heat pump system tested with an alternative refrigerant was designed for operating with the existing refrigerant. Therefore, when a new refrigerant is proposed, the system is not operated in a state that is suitable for the specific refrigerant. Namely, performance assessments performed when other refrigerants are applied to a system designed for a different working fluid are not providing sufficient information for guiding the choice on which new refrigerant to use.

3. Method

The performance of the heat pump water heater was evaluated for various refrigerants. Hot water supply demand and climate vary depending on the region and season. To evaluate the energy saving and environmental performance of heat pump water heaters, it was necessary to evaluate the performance throughout the year.

Therefore, an annual performance simulator for heat pump water heaters was constructed, which used the annual climatic conditions and hot water demand in each region as input data. Then, the annual performance could be calculated by simply specifying the refrigerant. This feature made it easy to evaluate a variety of new refrigerants that did not have actual equipment and could serve as guidelines for refrigerant selection.

3.1. Simulator overview

In the annual performance simulator, the specifications of the water heater, the climatic conditions, the demand for hot water supply, etc. are provides as the input and the annual operating state fluctuations and the annual performance are the output. Heat pump water heaters are expected to operate throughout the year. Thus, in order to judge the applicability of low GWP refrigerants to water heaters, it is important to evaluate the performance of water heaters throughout the year. Therefore, the annual power consumption of the water heater and the annual average COP, along with the time series data of the operating state, such as compressor discharge temperature, are given as the output. Fig. 1 shows the input/output relationship of the annual performance simulator. The subscript i is used to represent certain hourly data in the time series. The annual performance simulator includes a simulator that analyzes the thermodynamic cycle.

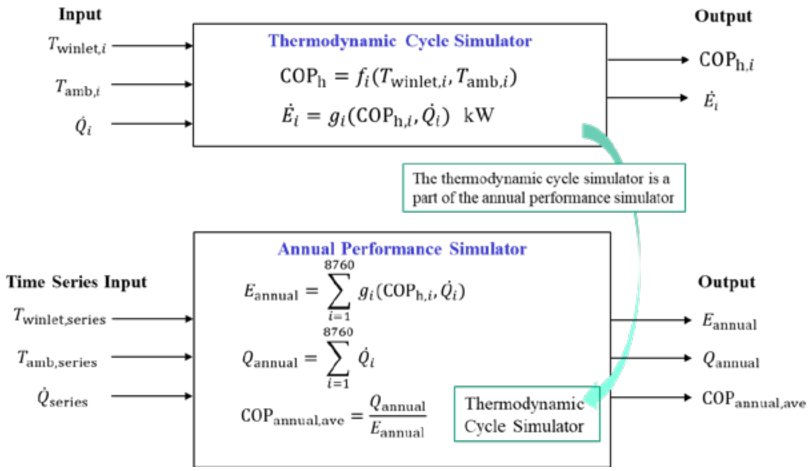


Fig. 1. Annual performance simulator overview

3.2. Thermodynamic Cycle Simulator

In order to design heat pump equipment suitable for the region and application, it is necessary to compare candidate refrigerants and select an appropriate refrigerant. To evaluate the annual performance in each region when a new refrigerant is applied to the heat pump water heater, commonly, a considerable amount of time is spent setting the design parameters for various new refrigerants and performing simulations and experiments.

Therefore, in this report, a mathematical model that emphasizes the calculation speed and ease of setting the design parameters of the equipment was developed for thermodynamic cycle analysis and incorporated into the annual performance simulator. This section describes this mathematical model.

Fig. 2 shows the flow diagram of the heat pump water heater and the corresponding Ph diagram of the heat pump cycle. The numbering of each point indicated by the arrows in Fig. 2 is used throughout this paper.

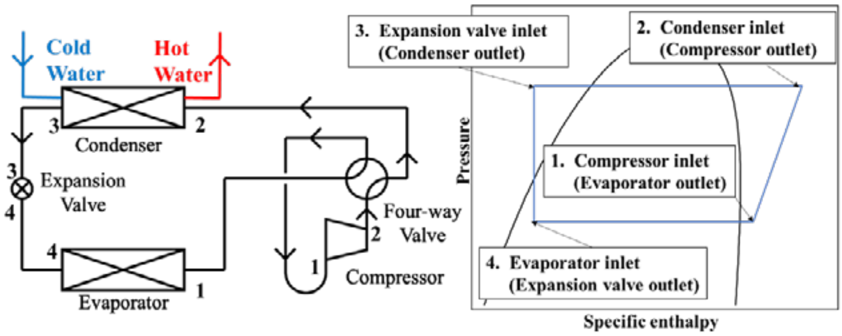


Fig. 2. Heat pump cycle flow diagram and Ph diagram

The heating coefficient of performance (COP) of the heat pump cycle is defined by the following formula.

$$\mathrm{COP_h} = \frac{\dot{Q}_{\mathrm{con}}}{\dot{W}_{\mathrm{comp}}} \tag{1}$$

To clarify the mass and energy balance of each component of a heat pump, the balance equations for each component are described below. The mass balance equation and energy balance equation in the compressor are

$$G_{\mathrm{R1}} = G_{\mathrm{R2}} \tag{2}$$

$$\dot{W}_{\mathrm{comp}} = G_{\mathrm{R2}} h_{\mathrm{R2}} - G_{\mathrm{R1}} h_{\mathrm{R1}} \tag{3}$$

The compressor suction temperature and compressor suction enthalpy are given by

$$T_{\mathrm{R1}} = T_{\mathrm{Rvap,eva}} + \Delta T_{\mathrm{SH}} \tag{4}$$

$$h_{\mathrm{R1}} = h_{\mathrm{R1}}(T_{\mathrm{R1}}, P_{\mathrm{eva}}) \tag{5}$$

For zeotropic refrigerants, $T_{\mathrm{Rvap,eva}}$ is given by the evaporator dew point. The compressor adiabatic efficiency is

$$\eta_{\mathrm{comp}} = \frac{h_{\mathrm{R1}} - h_{\mathrm{R2,ad}}}{h_{\mathrm{R1}} - h_{\mathrm{R2}}} \tag{6}$$

Using equation (6), the compressor discharge enthalpy can be calculated by

$$h_{\mathrm{R2}} = h_{\mathrm{R1}} + \frac{h_{\mathrm{R2,ad}} - h_{\mathrm{R1}}}{\eta_{\mathrm{comp}}} \tag{7}$$

η_{comp} is a constant and is given to the simulator as an input. The value of η_{comp} is assumed from typical value of actual compressor adiabatic efficiency. Here, in order to calculate $h_{\mathrm{R2,ad}}$ by the equation of state, the compressor discharge entropy of the reversible process is used.

$$s_{\mathrm{R2,ad}} = s_{\mathrm{R1}} \tag{8}$$

$$h_{\mathrm{R2,ad}} = h_{\mathrm{R2,ad}}(P_{\mathrm{con}}, s_{\mathrm{R2,ad}}) \tag{9}$$

The same formula is used for the condenser in the subcritical cycle and the gas cooler in the transcritical cycle. As heat pump water heaters are the target of this study, a mathematical model of a counter-flow heat exchanger is constructed. The mass balance and energy balance equations of the refrigerant in the condenser (or, gas cooler) are

$$\boldsymbol{G}_{\mathbf{R2}} = \boldsymbol{G}_{\mathbf{R3}} \tag{10}$$

$$\dot{\boldsymbol{Q}}_{\mathbf{con}} = \boldsymbol{G}_{\mathbf{R2}}\boldsymbol{h}_{\mathbf{R2}} - \boldsymbol{G}_{\mathbf{R3}}\boldsymbol{h}_{\mathbf{R3}} \tag{11}$$

For the water,

$$\boldsymbol{G}_{\mathbf{W2}} = \boldsymbol{G}_{\mathbf{W3}} \tag{12}$$

$$\dot{\boldsymbol{Q}}_{\mathbf{con}} = \boldsymbol{G}_{\mathbf{W2}}\boldsymbol{c}_{\boldsymbol{p}\mathbf{W}}\boldsymbol{T}_{\mathbf{W2}} - \boldsymbol{G}_{\mathbf{W3}}\boldsymbol{c}_{\boldsymbol{p}\mathbf{W}}\boldsymbol{T}_{\mathbf{W3}} \tag{13}$$

The condenser (or, gas cooler) refrigerant outlet temperature is

$$\boldsymbol{T}_{\mathbf{R3}} = \boldsymbol{T}_{\mathbf{W3}} + \boldsymbol{\Delta T}_{\mathbf{3}} \tag{14}$$

The mass balance and energy balance equations for the expansion valve are as follows.

$$\boldsymbol{G}_{\mathbf{R3}} = \boldsymbol{G}_{\mathbf{R4}} \tag{15}$$

$$\boldsymbol{G}_{\mathbf{R3}}\boldsymbol{h}_{\mathbf{R3}} = \boldsymbol{G}_{\mathbf{R4}}\boldsymbol{h}_{\mathbf{R4}} \tag{16}$$

The mass balance and energy balance equations in the evaporator are

$$\boldsymbol{G}_{\mathbf{R4}} = \boldsymbol{G}_{\mathbf{R1}} \tag{17}$$

$$\dot{\boldsymbol{Q}}_{\mathbf{eva}} = \boldsymbol{G}_{\mathbf{R1}}\boldsymbol{h}_{\mathbf{R1}} - \boldsymbol{G}_{\mathbf{R4}}\boldsymbol{h}_{\mathbf{R4}} \tag{18}$$

The refrigerant dew point temperature in the evaporator is

$$\boldsymbol{T}_{\mathbf{Rvap,eva}} = \boldsymbol{T}_{\mathbf{amb}} - \boldsymbol{\Delta T}_{\mathbf{eva}} \tag{19}$$

ΔT_{eva} is given to the simulator as a positive constant. This is used to calculate the compressor suction temperature in equation (4). The refrigerant evaporation pressure is calculated from the equation of state as follows.

$$\boldsymbol{P}_{\mathbf{eva}} = \boldsymbol{P}_{\mathbf{eva}}\left(\boldsymbol{T}_{\boldsymbol{Rvap,eva}}\right) \tag{20}$$

We focus on the pinch temperature in the condenser in order to build a model that determines the state of the refrigerant in the condenser from the temperature conditions of the hot water inlet and outlet. There is at least one pinch temperature in the condenser, usually two. It is impossible from the viewpoint of heat transfer to have a pinch temperature below 0 ° C. Therefore, a constraint condition that takes a positive finite value as the pinch temperature is set. K_1 and K_2 are positive constants. The inequality constraints on the condenser refrigerant outlet pinch temperature ΔT_3 and the condenser internal pinch temperature $\Delta T_{\mathrm{con,pinch}}$ (see Fig. 3) are represented by

$$\Delta T_3 \geq K_1 \tag{21}$$

$$\boldsymbol{\Delta T}_{\mathbf{con,pinch}} \geq \boldsymbol{K}_{\mathbf{2}} \tag{22}$$

Here, $\Delta T_{\mathrm{con,pinch}}$ is the pinch temperature in the condenser that can exist other than ΔT_3, and is defined as follows using the temperatures $T_{\mathrm{R,con}}$ and $T_{\mathrm{W,con}}$ of the refrigerant and hot water, respectively, that exchange heat with each other. [4]

$$\boldsymbol{\Delta T}_{\mathbf{con,pinch}} = \boldsymbol{T}_{\mathbf{R,con,pinch}} - \boldsymbol{T}_{\mathbf{W,con,pinch}} \tag{23}$$

The simulator input values of K_1, K_2 and ΔT_{eva} are assumed from typical value of domestic heat pump hot water heaters. The heat exchange and pinch temperature in the condenser will be explained using Fig. 3 with the heat exchange on the horizontal axis and the temperature on the vertical axis.

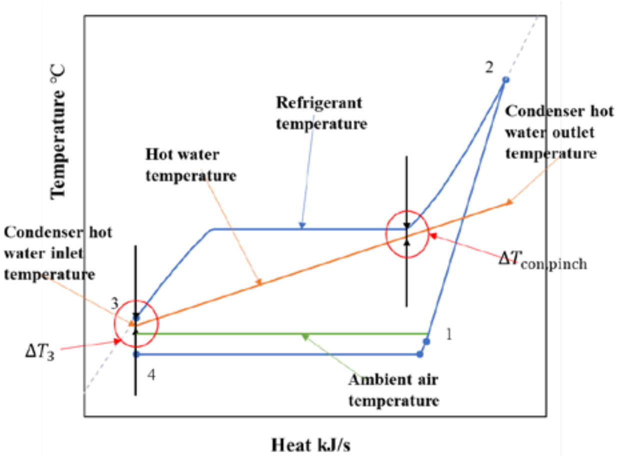


Fig. 3. Temperature heat diagram

From the mathematical model of the condenser model described above, the absolute value of exchanged heat of both fluids in the condenser is equal. Therefore, the horizontal width of the thermal composite line of water and the thermal composite line of refrigerant are equal, and the start and end points are at the same position on the horizontal axis. As the flow rates of both fluids are directly related to the heat capacity, the curves (including straight lines) representing the heat exchange process of both fluids can be scaled with respect to the horizontal axis, depending on the mass flow rate of each fluid. The relationship between heat exchange and temperature is expressed in Fig. 3.

$$T_{\mathrm{R,con,pinch}} = T_{\mathrm{R,con,pinch}}\left(P_{\mathrm{con,pinch}}, h_{\mathrm{con,pinch}}\right) \tag{24}$$

$$T_{\mathrm{W,con}} = \frac{T_{\mathrm{W2}} - T_{\mathrm{W3}}}{G_{\mathrm{W2}}h_{\mathrm{W2}} - G_{\mathrm{W3}}h_{\mathrm{W3}}}\left(G_{\mathrm{W}}h_{\mathrm{W}} - G_{\mathrm{R}}h_{\mathrm{R}}\right) + T_{\mathrm{W3}} \tag{25}$$

Under these conditions, the COP maximization problem, with the condenser pressure as the optimization parameter, is solved. A cycle is constructed using the solution of this maximization problem as the condenser pressure P_{con}.

4. Discussion

4.1. Fundamental characteristics of heat pump water heater

Several refrigerants are evaluated using the thermodynamic simulator in order to clarify similar and significantly different characteristics. As R410A refrigerant, which is currently widely used in heat pumps, has a high GWP of 1924, alternative low GWP refrigerants have been proposed. R32 and R446A are compared as substitute refrigerants for R410A. In addition, R134a and propane are also included in the comparison to understand the characteristics that differ significantly depending on the refrigerant.

Japan's average climatic conditions are adopted as reference conditions, and the COP, compressor discharge temperature, and refrigerant mass flow of each refrigerant compared. The standard conditions are an inlet water temperature of 16.5 ° C, outlet water temperature of 65 ° C, and an ambient air temperature of 16 °C. Fig. 4 shows the results of the comparison.

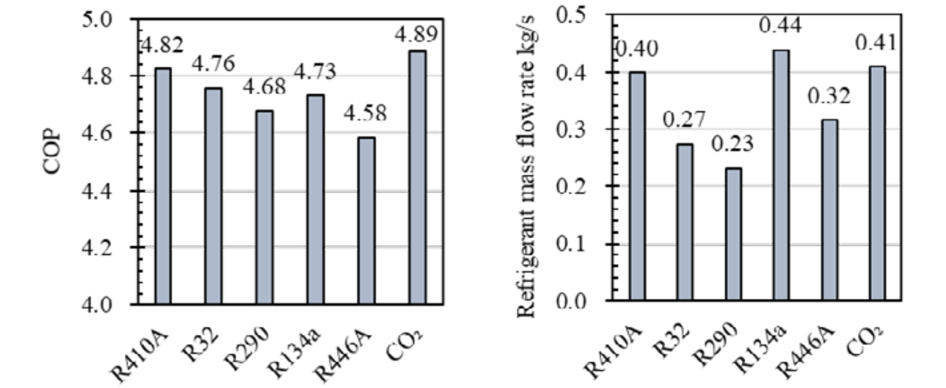
The refrigerant with the highest COP (4.89) is CO_2, and the refrigerant with the lowest COP (4.58) is R446A. CO_2 refrigerant is a transcritical cycle under these operating conditions. R410A has the highest COP of the subcritical cycle refrigerants. The compressor discharge temperature is the highest for R32, at 113.2 °C. R290 is the refrigerant with the lowest discharge temperature of 82.6 °C. R134a also has a low temperature of 83.7 °C. The mass flow rate of R134a is the largest, but R410A and CO_2 feature similar values. Conversely, R290 has the smallest mass flow rate, and the value of R32 is close to it.

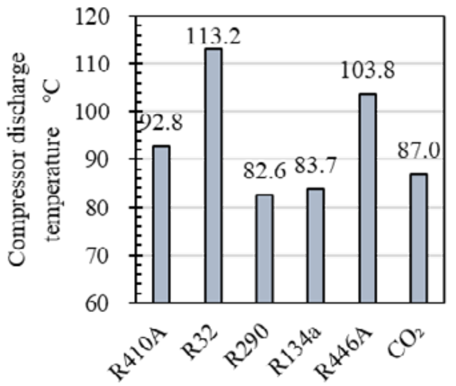
A parameter study is conducted for COP, compressor discharge temperature, and refrigerant flow temperature. Fig. 5 shows the results for the feed water temperature, Fig. 6 shows the results for the condenser outlet water temperature, and Fig. 7 shows the results for the ambient air temperature.

First, we focus on COP. With one exception, all refrigerants show similar trends for water supply, hot water, and ambient air temperature. Generally, the COP decreases with respect to the feed water temperature and condenser outlet water temperature and rises with respect to the ambient air temperature. In the region where the feed water temperature is low, the COP of the CO_2 refrigerant is higher than that of other refrigerants. However, when the feed water temperature exceeds 20 °C, the COP of other refrigerants may increase. The COP of the CO_2 refrigerant is the lowest at around 30 °C.

Next, we focus on the compressor discharge temperature. Although the discharge temperature values differ between refrigerants, there is no significant difference in the tendency for parameter changes except for CO_2. The compressor discharge temperature with CO_2 refrigerant rises more than other refrigerants when the feed water temperature is high and the ambient air temperature is low.

Finally, the refrigerant mass flow rate is described. Here, the CO_2 refrigerant also shows a different tendency. Compared with other refrigerants, the increase in mass flow rate is remarkable when the feed water temperature and the ambient air temperature become high.





Calculation condition (Standard condition, defined in text)	
Ambient air temperature	16 °C
Water inlet temperature	16.5 °C
Water outlet temperature	65 °C
ΔT_3	3°C
$\Delta T_{con,pinch}$	3°C

Fig. 4. Refrigerant comparison under standard conditions

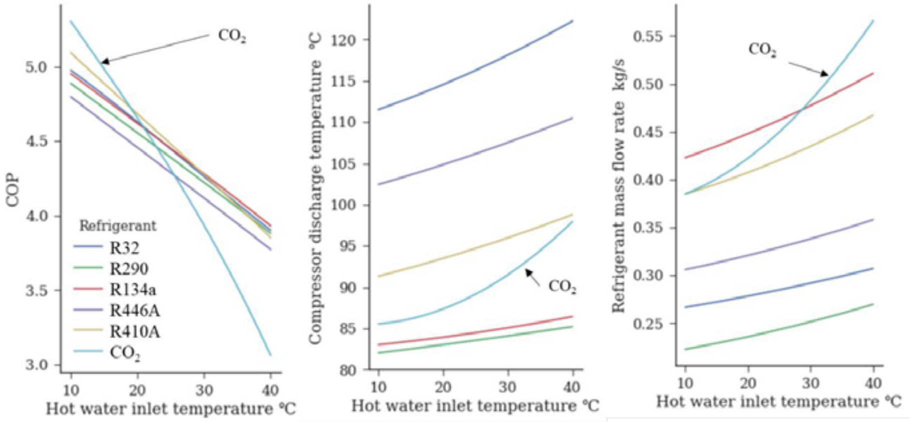


Fig. 5. Parametric study on influence of hot water inlet temperature on COP, Compressor discharge temperature, and Refrigerant mass flow rate

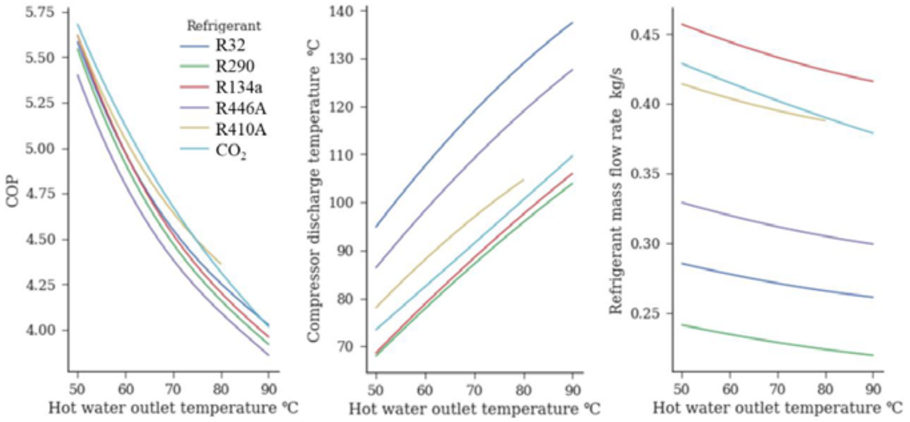


Fig. 6. Parametric study on influence of hot water outlet temperature on COP, Compressor discharge temperature, and Refrigerant mass flow rate

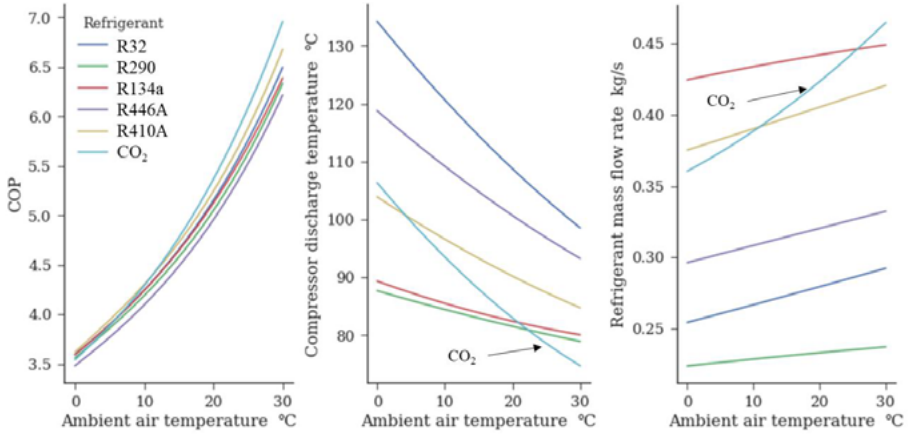


Fig. 7. Parametric study on influence of ambient air temperature on COP, Compressor discharge temperature, and Refrigerant mass flow rate

4.2. Annual performance evaluation

A simulation is conducted to investigate the effect of differences in area, climate, and refrigerant on the annual performance of heat pump water heaters. The condenser hot water outlet temperature is fixed at 65 °C, and the ambient air temperature, water inlet temperature, and hot water demand were input in time series. In this report, the supply water temperature is given by a direct relation to the ambient temperature, and the same daily pattern is used for the hot water demand in each region. The hot water demand pattern is based on table B.6 of JIS C9220:2011 Annex B. [5]

The target refrigerants are CO_2 and R32, which have been used as refrigerants for domestic hot water heaters in Japan. Sapporo, Tokyo, and Naha are selected as three representative cities for Japan's high, middle, and low latitudes. These three cities are indicated in Fig. 8. The climatic data on the ambient air temperature are from 2018 [6]. Fig. 9 shows the ambient temperature fluctuations in these three cities. For ease of understanding, the trend of annual fluctuation in Fig. 9 uses monthly averaged data, while input data to the simulator refer to the hourly data.

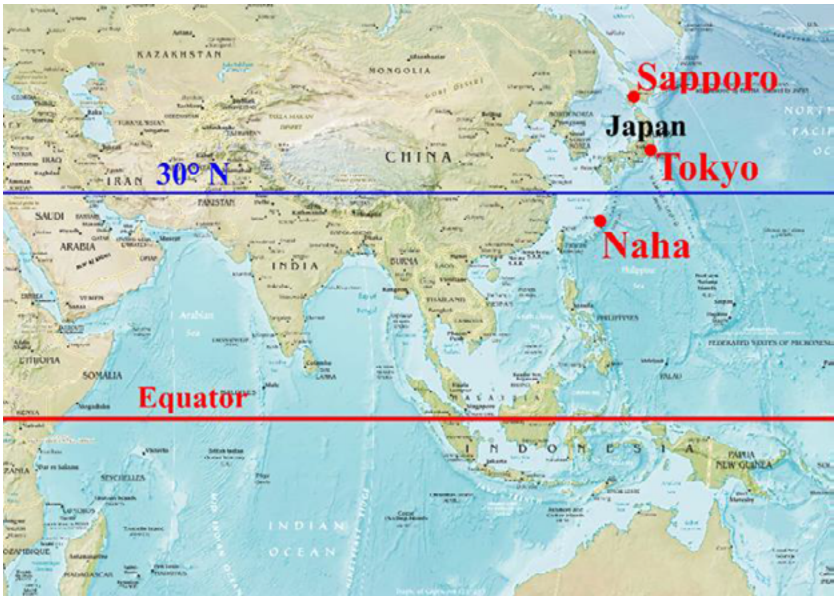


Fig. 8. Location of the target cities [7]

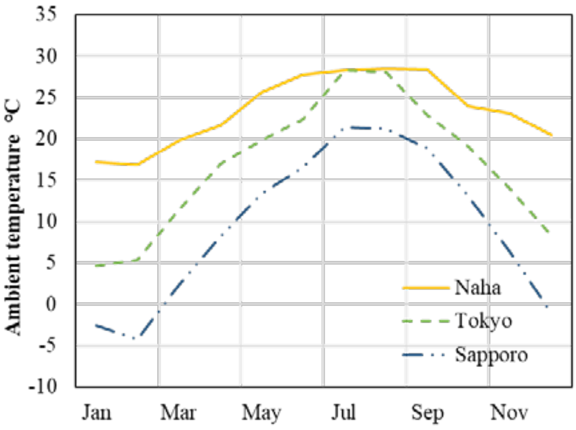


Fig. 9. Annual ambient temperature at Sapporo, Tokyo, and Naha

Fig. 10 shows the evaluation results of the annual power consumption and annual average COP of the water heater. Generally, the annual power consumption decreases and the annual average COP tends to increase as the latitude of the region decreases. When the CO_2 refrigerant is used as a reference, the annual power consumption of the R32 refrigerant increased by a maximum of 2.53%, and annual average COP decreased by 2.46%. Focusing on the regional characteristics, the annual power consumption in Sapporo was approximately twice that of Naha. This is not only because of the lower heat load due to the higher ambient temperature of Naha, but also because the COP is higher here than in Sapporo almost throughout a year, as described later.

Fig. 11 shows the annual COP change. The graph on the left is for the CO_2 refrigerant, and the right is for the R32 refrigerant. For all refrigerants, it is demonstrated that the COP is higher in summer than in winter. Naha has shown that it can operate at a high COP throughout the year and there is almost no COP difference between the two refrigerants.

To compare the trend of the COP variation for each refrigerant in detail, Fig. 12 shows the COP fluctuation for the first week of 2018 and Fig. 13 shows that of the first week of the second half of same year. Thus, the difference in COP trend for each refrigerant is not significant. According to Fig. 7, regarding the ambient air temperature, the performance difference is considered to increase in the hot season and regions. However, as shown in Fig. 5, CO_2 has a significantly larger COP drop than other refrigerants with respect to the water inlet temperature. Accordingly, the COP of CO_2 refrigerant is slightly higher in any of these three regions during winter, however, the COP of R32 refrigerant come to almost same at Tokyo and Naha in summer, as shown in Fig. 12 and Fig. 13. Therefore, it can be considered that the COP is reversed with other refrigerants in hot seasons and regions.

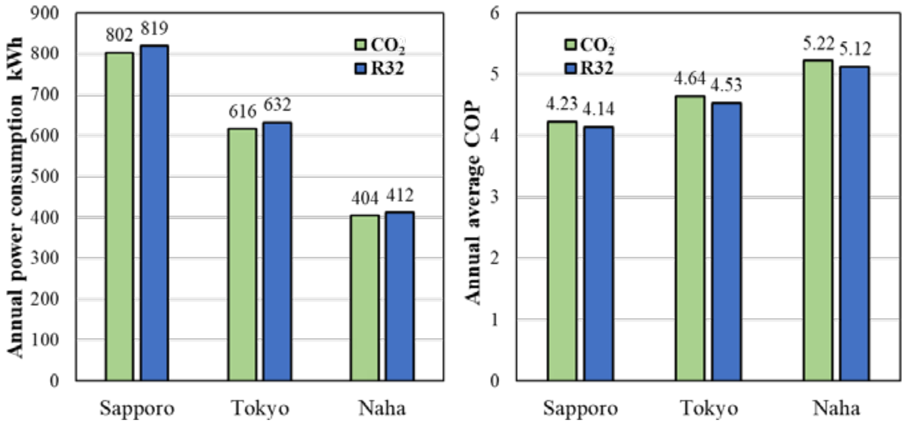


Fig. 10. Annual power consumption and annual average COP

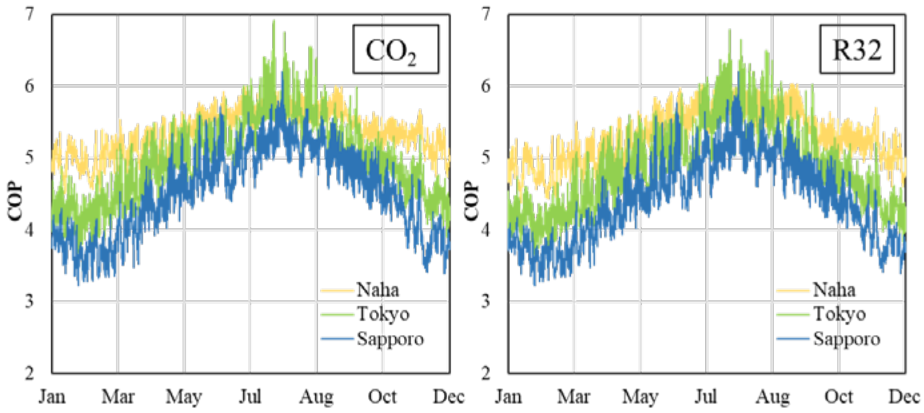


Fig. 11. Annual COP fluctuation (Left: CO_2, Right: R32)

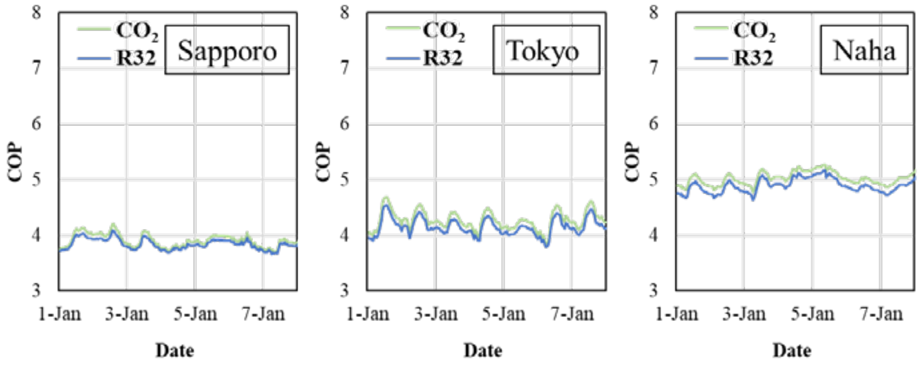


Fig. 12. Weekly COP fluctuation (Winter, Comparison by refrigerant)

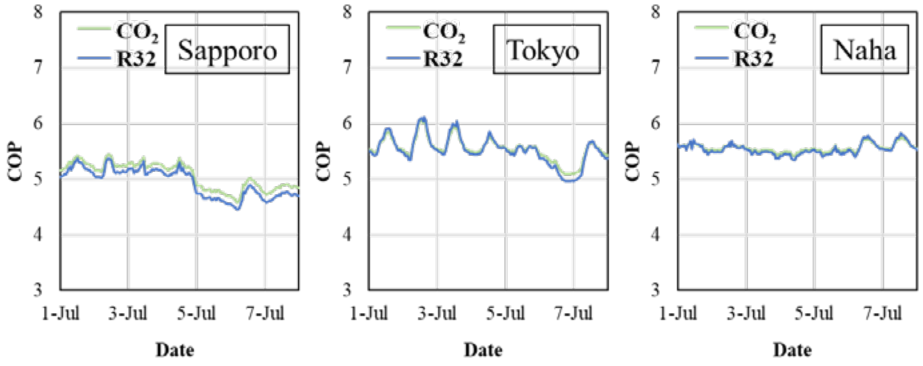


Fig. 13 Weekly COP fluctuation (Summer, Comparison by refrigerant)

5. Outlook for the future

Currently, in practice, a domestic heat pump water heater requires a water tank for the following two reasons. First of all, water usage during a shower is around 10 L/min for 10 to 20 minutes. Thus, the power output required in order to provide instant heating for a shower is about 10 to 20 kW. The higher the power output is, the more expensive the price of a heat pump is. As a result, it is impractical to use high output power heat pumps for domestic water heating. Secondly, due to the daily fluctuations of the electricity pricing in Japan, using electricity at night is cheaper and in order to lower the cost, heat pump systems are designed to operate and store hot water during that period.

This report focuses on the heat pump cycle itself rather than the water tank. However, using a water tank can result in the temperature of water supplied to condenser or gas cooler to be higher than that of the tap water. This is believed to have some influence to the system's performance. Other than that, the time of operation should also be carefully considered since factors like ambient air temperature, water supply temperature and heat loss can be different between day and night, and, thus, affect the system's efficiency. Therefore, in the future, we would like to build a model that considers the hot water storage tank and perform more accurate and versatile annual performance evaluation based on this model.

6. Conclusion

An annual performance simulator was constructed to compare the performance of various refrigerants and heat pump water heaters in different regions. The actual climate data of different Japanese cities were inputted into the simulator to compare the heater performance in different regions and with different refrigerants.

As a result, it was confirmed that the water heater performance was higher in lower latitude areas. In addition, it was suggested that the lower the latitude, the smaller the fluctuation of the driving condition, and the more stable the operation. The comparison of R32 and CO_2 suggests that the performance of the CO_2 refrigerant is higher in all regions of Japan. However, the results of the basic performance analysis suggest that the performance can be reversed in warmer areas, and in applications where the water supply temperature is high.

Acknowledgements

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