15TH IEA HEAT PUMP CONFERENCE
MAY 26–29, 2026 | HOFBURG
VIENNA 2026
DECARBONISATION THROUGH INNOVATION

Simple method for estimating energy efficiency of industrial heat pumps available on the Japanese market

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Abstract

Industrial heat pumps are recognized as an important technology for decarbonizing industries due to their high energy efficiency and ability to be electrified with low carbon electricity. In Japan, various types of industrial heat pumps have been commercialized to accommodate a wide range of industrial heating processes. This diversity requires time and effort when examining the applicability of heat pumps, which is one of the factors hindering their introduction.

This study aims to develop a unified and simple method for estimating the energy efficiency of industrial heat pumps. First, energy performance data on industrial heat pumps available in the Japanese market are collected and analyzed. Then, based on the regression analysis, two types of correlations between Lorenz efficiency and logarithmic mean temperature lift are developed. The first-type correlation can estimate the average COP (coefficient of performance) of industrial heat pumps with single-stage cycles with an accuracy of approximately within ±10%. On the other hand, the second-type correlation represents the highest performance of existing technologies including two-stage cycles and cascade cycles. Both correlations can be calculated by inputting 6 parameters: heat source medium, heat sink medium, heat source inlet and outlet temperatures, and heat sink inlet and outlet temperatures.

Keywords: industrial heat pump; temperature lift; COP; Lorenz efficiency

1. Introduction

According to International Energy Agency [1], heat pumps, powered by low-emissions electricity, are the central technology in the global transition to secure and sustainable heating. To expand the application area of such heat pumps, technological development and commercialization of high-temperature heat pumps are advancing for various industrial heating processes such as cleaning, sterilization, drying, distillation, and concentration [2]. However, while industrial heat pump technologies have matured, they remain in the early stages of adoption. Therefore, efforts contributing to technology deployment are necessary, not just technological development. Industrial heating processes are diverse, requiring the selection of heat pumps suited to each process. Yet, a methodology for this has not been established.

When considering the introduction of industrial heat pumps, the process begins with concept screening – determining which heat sources should heat be recovered form and which heat demands should heat be delivered to. At this stage, those who conduct the concept screening estimate the effects of introducing the selected integration concepts. The effects depend heavily on the coefficient of performance (COP). After selecting the product model, those who conduct the concept screening can obtain the COP listed in the product catalog. However, during the concept screening stage before specific product is determined, it is cumbersome to refer to the COP of individual products.

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Therefore, this study attempts to develop an equation that can easily reproduce the COP of industrial heat pump products with a certain degree of accuracy. Regarding the required accuracy for cost estimation, this study assumes a range of approximately ±30% to ±10%, corresponding to Class 5 (Concept Screening) to Class 3 (Budget Approval) in the classification recommended by AACE International [3].

For those conducting concept screening, equations that do not require access to internal heat pump information are convenient. Arpagaus [4] proposed the following COP correlation equation based on a total of 41 data points from 14 industrial heat pump products in Europe.

$$COP = 68.455 {\Delta T_{\mathrm{lift}}}^{-0.76} \tag{1}$$

Here, the temperature lift is defined as the simple temperature lift, calculated as the difference between the heat sink outlet temperature and the heat source inlet temperature. The temperature range in the data is 30-130 K. The coefficient of determination (R^2) is 0.78, confirming a good fit to the data.

Similarly, Schlosser et al. [5] proposed the following COP correlation equation.

$$COP = a \cdot (\Delta T_{\mathrm{lift}} + 2b)^{c} \cdot \left(T_{\mathrm{H,o}} + b\right)^{d} \tag{2}$$

Here, each fitting parameter (a, b, c, d) has a different value depending on the type of refrigerant and the configuration of the heat pump system. For high-temperature heat pumps, based on a total of 43 data points from 8 industrial heat pump products in Europe, a is decided as 1.9118, b is 0.044189, c is -0.89094, and d is 0.67895. The application range is the heat source inlet temperatures of 25-110°C, the heat sink outlet temperatures of 80-160°C, the temperature lift of 25-95 K. The R^2 is 0.95, confirming a good fit to the data.

Both the existing correlations use only external heat pump information such as heat source and heat sink temperatures. This is convenient for those conducting concept screening. However, the correlation by Arpagaus does not consider the effect of temperature level on the COP. In addition, both correlations directly express the COP as the first law efficiency, and thus do not account for the constraints of the second law of thermodynamics. Furthermore, both correlations apply only to water source heat pumps.

This study proposes a simplified correlation equation for the energy efficiency that does not use internal heat pump information, like the existing correlations. However, unlike previous research, it attempts a method using the second law efficiency, Lorenz efficiency rather than directly expressing the first law efficiency, COP. That way, it attempts a highly generalized method applicable even to heat pumps with large temperature glides. Note that data for creating the correlation uses performance data from Japanese industrial heat pump products.

2. Lorenz efficiency

Lorenz efficiency is the ratio of the actual COP to Lorenz COP as shown in Eq. (3). The Lorenz COP is expressed using logarithmic mean temperatures as shown in Eq. (4).

$$\eta_{\mathrm{Lorenz}} = \frac{COP}{COP_{\mathrm{Lorenz}}} \tag{3}$$

$$COP_{\mathrm{Lorenz}} = \frac{\bar{T}_{\mathrm{H}}}{\bar{T}_{\mathrm{H}} - \bar{T}_{\mathrm{C}}} = \frac{(T_{\mathrm{H,o}} - T_{\mathrm{H,i}})/\ln{(T_{\mathrm{H,o}}/T_{\mathrm{H,i}})}}{(T_{\mathrm{H,o}} - T_{\mathrm{H,i}})/\ln{(T_{\mathrm{H,o}}/T_{\mathrm{H,i}})} - (T_{\mathrm{C,i}} - T_{\mathrm{C,o}})/\ln{(T_{\mathrm{C,i}}/T_{\mathrm{C,o}})}} = \frac{\bar{T}_{\mathrm{H}}}{\Delta\bar{T}_{\mathrm{lift}}} \tag{4}$$

In actual heat pumps, the Lorenz efficiency is less than 1 due to various losses such as necessity of temperature differences for heat exchange, temperature glide mismatch between refrigerant and heat source/sink, compressor efficiency, throttling loss, heat loss, pressure drop. To understand these loss factors, Jensen et al. [6] factorized the Lorenz efficiency for single-stage compression heat pump cycle as follows.

$$\eta_{\mathrm{Lorenz}} = \eta_{\mathrm{HT}} \cdot \eta_{\mathrm{is,cmp}} \cdot (1 - f_{\mathrm{ref}}) + \left(1 - \eta_{\mathrm{is,cmp}} - f_{\mathrm{Q}}\right) \cdot \frac{\Delta\bar{T}_{\mathrm{lift}}}{\bar{T}_{\mathrm{H}}} \tag{5}$$

$$\eta_{\mathrm{HT}} = \frac{1 + \dfrac{\Delta\bar{T}_{\mathrm{r,H}} + \Delta\bar{T}_{\mathrm{p,H}}}{\bar{T}_{\mathrm{H}}}}{1 + \dfrac{\Delta\bar{T}_{\mathrm{r,H}} + \Delta\bar{T}_{\mathrm{p,H}} + \Delta\bar{T}_{\mathrm{r,C}} + \Delta\bar{T}_{\mathrm{p,C}}}{\Delta\bar{T}_{\mathrm{lift}}}} \tag{6}$$

$$\eta_{\mathrm{is,cmp}} = \frac{W_{\mathrm{is,cmp}}}{W_{\mathrm{cmp}}}, \qquad f_{\mathrm{ref}} = \frac{W_{\mathrm{is,exp}}}{W_{\mathrm{is,cmp}}}, \qquad f_{\mathrm{Q}} = \frac{Q_{\mathrm{loss}}}{W_{\mathrm{cmp}}} \tag{7, 8, 9}$$

In this equation, necessity of temperature difference for heat exchange is considered in $\Delta\overline{T}_{\mathrm{p,H}}$ and $\Delta\overline{T}_{\mathrm{p,C}}$, which are pinch point temperature differences in condenser and evaporator. Temperature glide mismatch is considered in $\Delta\overline{T}_{\mathrm{r,H}}$ and $\Delta\overline{T}_{\mathrm{r,C}}$, which are temperature differences between refrigerant and heat sink or heat source, excluding $\Delta\overline{T}_{\mathrm{p,H}}$ and $\Delta\overline{T}_{\mathrm{p,C}}$. Compressor efficiency is expressed as $\eta_{\mathrm{is,cmp}}$, which is a compressor isentropic efficiency. Throttling loss is considered in f_{ref}, defined as the ratio between the expansion work and the compression work for isentropic processes. Heat loss is considered in f_{Q}, defined as the ratio between the compressor heat loss and the compression work.

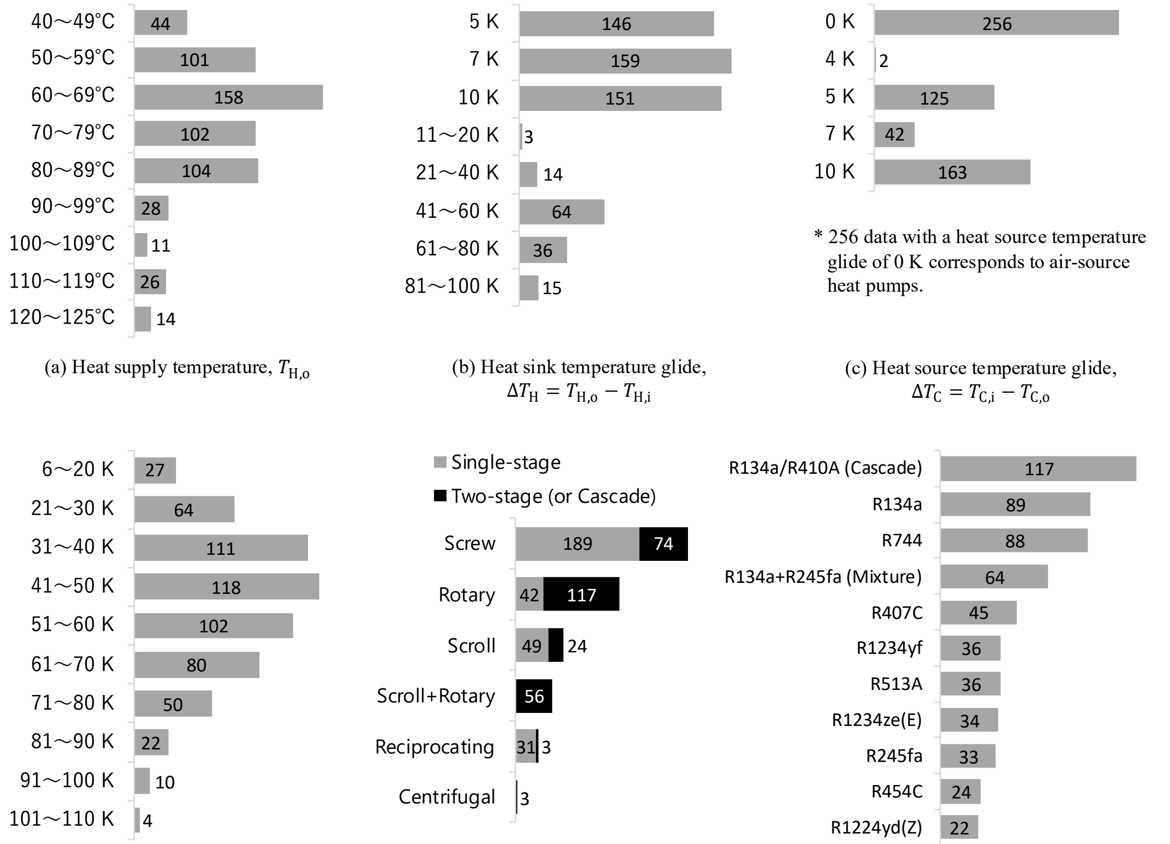
Creating an equation suitable for concept screening based on Eq. (5) is not straightforward, but Eq. (5) is useful for understanding the characteristics of the Lorenz efficiency. Equation (5) will be referenced during the data analysis described in Chapter 4.

3. Data Collection

In creating the correlation equation, performance data of industrial heat pump products in Japan is used. The main specifications of heat pumps are as follows.

- Type of heat pumps: electrically driven vapor compression cycle heat pumps
- Heat source media: water, air
- Heat sink media: hot water, hot air
- Heat supply temperature: 55°C or higher at rated conditions

Here, the data includes heat pumps employing various technologies without restrictions on compressor type, refrigerant type, or cycle configuration. Furthermore, to ensure data reliability, only models with published data for three or more temperature conditions are included. Meeting these conditions, the author surveyed commercially available industrial heat pumps as of the end of 2023, collecting data for 25 models totaling 588 data points.



(a) Heat supply temperature, $T_{\mathrm{H,o}}$

(b) Heat sink temperature glide, $\Delta T_{\mathrm{H}} = T_{\mathrm{H,o}} - T_{\mathrm{H,i}}$

(c) Heat source temperature glide, $\Delta T_{\mathrm{C}} = T_{\mathrm{C,i}} - T_{\mathrm{C,o}}$

(d) Logarithmic mean temperature lift, $\Delta\overline{T}_{\mathrm{lift}} = \overline{T}_{\mathrm{H}} - \overline{T}_{\mathrm{C}}$

(e) Compressor type

(f) Refrigerant type

Figure 1. Data distributions.

Figure 1 shows the distribution of the collected data. The heat supply temperature (heat sink outlet temperature) covers a wide range from 40°C to 125°C, and various types of refrigerants are used accordingly. The temperature glide in heat sink is up to 100 K, while the temperature glide in heat source is 10 K or less. For air-source heat pumps, the air inlet-outlet temperature difference is small when properly designed, and since air outlet temperature is generally not listed in product catalogue, the temperature glide is set to 0 K. The logarithmic mean temperature lift range is from 6 K to 110 K. The data includes models equipped with compressors such as screw, rotary, scroll, reciprocating and centrifugal types.

4. Data Analysis

4.1. Correlation between COP and temperature lift

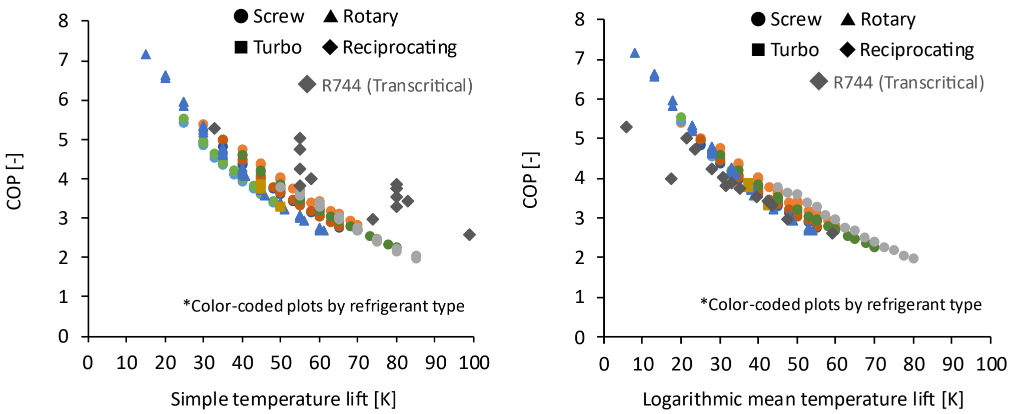
As the first step in the analysis, we will examine the correlation between COP and temperature lift, focusing on water-source hot water supply heat pumps with single-stage compression.

Figure 2 shows a graph plotting COP against temperature lift. For the temperature lift on the horizontal axis, Fig. 2(a) uses simple temperature lift, while Fig. 2(b) uses logarithmic mean temperature lift. Each figure displays 10 sets of data, with symbols indicating compressor type and colors indicating refrigerant type.

Overall, it is observed that the COP tends to decrease as the temperature lift increases. However, for a model with R744 (Transcritical) in Fig. 2(a), significantly different COP data exists even for the same simple temperature lift. Furthermore, compared to other models, it tends to exhibit a higher COP under the same temperature lift conditions. This is because, being a transcritical cycle, even with the same temperature lift, the heat sink inlet temperature differs; the lower the heat sink inlet temperature, the higher the COP becomes.

On the other hand, when plotted against logarithmic mean temperature lift as shown in Fig. 2(b), the R744 (Transcritical) model also exhibit a tendency for COP to decrease as the temperature lift increases, and the COP has similar values with other models. Note that two data points for R744 show significantly lower COP compared to the others. These two points correspond to the heat source outlet temperature of 32°C, where the evaporation temperature is near R744's critical temperature of 31°C. This temperature condition would cause the lower COP compared to the other models.

Thus, it is evident that by using the logarithmic mean temperature lift, COPs with various models – including those with transcritical cycles – can be uniformly organized.



(a) Simple temperature lift (b) Logarithmic mean temperature lift

Figure 2. Correlation between COP and temperature lift.

4.2. Correlation between Lorenz efficiency and logarithmic mean temperature lift

Generally, even for the same temperature lift, the COP varies depending on the temperature level. However, by using Lorenz efficiency instead of COP, the influence of the temperature level can be mitigated.

Figure 3 shows a graph plotting Lorenz efficiency against logarithmic mean temperature lift. Focusing on models equipped with screw compressors, it can be observed that, except for the models in orange, there is no significant difference in Lorenz efficiency even between models operating at different temperature levels using different refrigerants. The model in orange uses a zeotropic mixed refrigerant, which exhibit slightly higher Lorenz efficiency compared to models using other pure refrigerants. However, the difference in Lorenz efficiency is less than 0.05. If this difference is acceptable, a correlation equation independent of refrigerant type can be developed.

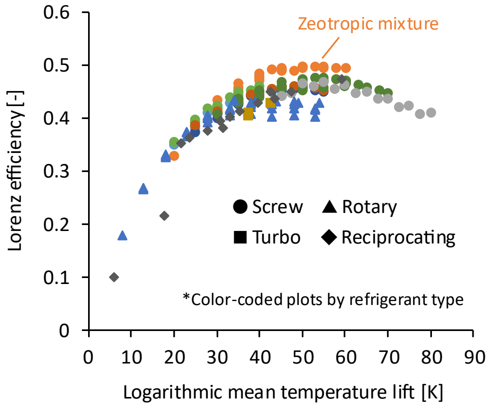


Figure 3. Correlation between Lorenz efficiency and logarithmic mean temperature lift

4.3. Influence of compressor optimization

Most water-source hot water supply heat pumps with single-stage compressors commercialized in Japan, as shown in Fig. 3, have the maximum Lorenz efficiency at a logarithmic mean temperature lift of approximately 50 K. However, some products equip single-stage compressors with a small internal volume ratio to achieve higher efficiency in smaller temperature lifts, while others adopt two-stage compression cycles or cascade cycles to achieve higher efficiency in larger temperature lifts. This paper distinguishes between "standard performance" models, where the Lorenz efficiency peaks at a temperature lift of around 50 K, and "high performance" models designed for higher efficiency in specific small or large temperature lift ranges.

Sections 4.1 and 4.2 focused solely on standard performance models. In this section, as the second stage of analysis, we expand the scope to include high performance models equipped with compressors optimized for specific pressure ratios.

All data shown in Fig. 4 are from models equipped with screw compressors. Based on compressor structure, they are classified into three types: standard performance machines, high performance machines for low pressure ratio, high performance machines for high pressure ratio. The plots include 7 standard performance machines with different refrigerants and temperature levels, 2 high performance machines for low pressure ratio, and 3 high performance machines for high pressure ratio.

The Lorenz efficiency of the standard performance machines peaks at a logarithmic mean temperature lift of around 50 K, whereas the Lorenz efficiency of the high performance machines for low pressure ratio peaks at around 30 K. However, there is no significant difference in the peak values, both being approximately 0.48.

On the other hand, for the high performance machines for high pressure ratio, the Lorenz efficiency reaches its maximum at around 70 K, though the peak is not sharp. Furthermore, the maximum Lorenz efficiency value is approximately 0.55, which is higher that that of standard performance machines.

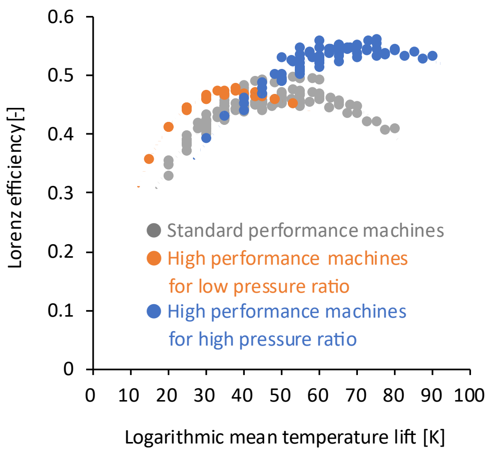


Figure 4. Influence of compressor optimization.

4.4. Influence due to heat exchange with air

Up to the previous section, we have analyzed water-source hot water supply heat pumps. In this section, as the third stage of analysis, we will examine air-source hot water supply heat pumps and water-source hot air supply heat pumps.

Figure 5 shows the Lorenz efficiency for air-source hot water supply heat pumps and water-source hot air supply heat pumps equipped with single-stage compressors. Overall, the Lorenz efficiency is lower than that of water-source hot water supply heat pumps shown in Fig. 3. The reason for this would be that heat exchange with air involves a larger temperature difference compared to water, resulting in lower system heat transfer performance corresponding to η_{HT} in Eq. (6).

Furthermore, referring to Eq. (6), air-source hot water supply heat pumps have a larger $\Delta\overline{T}_{\mathrm{p,C}}$ compared to water-source heat pumps. Similarly, hot air supply heat pumps have a larger $\Delta\overline{T}_{\mathrm{p,H}}$ compared to water supply heat pumps. Theoretically, the efficiency η_{HT} should differ between air-source hot water supply heat pumps and water-source hot air supply heat pumps. However, Figure 5 shows that there is no significant difference in Lorenz efficiency between the two. Therefore, a common correlation equation can be created for both air-source hot water supply heat pumps and water-source hot air supply heat pumps.

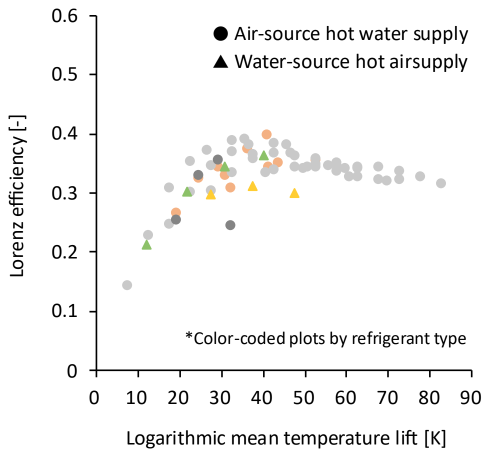


Figure 5. Influence due to heat exchange with air.

5. Creation of Correlation Equations

5.1. Guidelines derived from data analysis

Based on the analysis results in Chapter 4, the following six guidelines are derived for creating correlation equations.

- By organizing data with the relationship between Lorenz efficiency and logarithmic mean temperature lift, it is possible to derive the same correlation equation for transcritical cycles with large temperature glides as for subcritical cycles with small temperature glides.
- The influence of refrigerant differences results in approximately a 10% performance variation due to factors such as improved temperature glide matching achieved by adopting zeotropic mixed refrigerants. However, if this variation is acceptable, a correlation equation independent of refrigerant type can be developed.
- For standard compressors (those achieving maximum Lorenz efficiency at a logarithmic mean temperature lift of around 50 K), differences between compressor types are small. This allows the creation of a correlation equation independent of compressor type.
- High performance machines with compressors optimized for low pressure ratio or high pressure ratio have different maximum point of Lorenz efficiency from standard performance machines. Therefore, it is preferable to create a different correlation equation between standard performance machines and high performance machines.
- A common correlation equation can be created for both air-source hot water supply heat pumps and water-source hot air supply heat pumps.

- Lorenz efficiency of air-source hot water supply heat pumps is lower than that of water-source hot water supply heat pumps. Therefore, it is preferable to develop a different correlation equation between air-source hot water supply heat pumps and water-source hot water supply heat pumps.

5.2. Determination of correlation equations

A correlation equation was derived from the relationship between Lorenz efficiency and logarithmic mean temperature lift. The equation's form was determined to be that of Eq. (10) as a function satisfying the following three conditions.

- Lorenz efficiency reaches a maximum at a certain logarithmic mean temperature lift.
- Lorenz efficiency is 0 when logarithmic mean temperature lift is 0 K.
- Lorenz efficiency does not diverge when logarithmic mean temperature lift is infinite.

$$\eta_{\text{Lorenz}} = a \cdot \Delta\bar{T}_{\text{lift}} \cdot (b + c \cdot \Delta\bar{T}_{\text{lift}})^{d} \quad (10)$$

Here, it must be $a > 0$ and $d < -1$.

The parameters (*a*, *b*, *c*, *d*) in Eq. (10) are set to four different combinations based on the performance level of machines and the combination of heat source and heat sink media. The parameters are listed in Table 1.

Table 1. Parameters of correlation equations

Equation	Performance	Heat source / Heat sink	*a*	*b*	*c*	*d*
#1	Standard	Water / Water	2.65×10^{-4}	2.82×10^{-1}	1.97×10^{-3}	-3.67
#2	Standard	Water / Air, Air / Water	2.43×10^{-4}	3.16×10^{-1}	2.37×10^{-3}	-4.06
#3	High	Water / Water	1.70×10^{-4}	2.61×10^{-2}	7.45×10^{-4}	-1.50
#4	High	Water / Air, Air / Water	6.10×10^{-2}	1.51	8.68×10^{-3}	-2.98

5.3. Data fitting

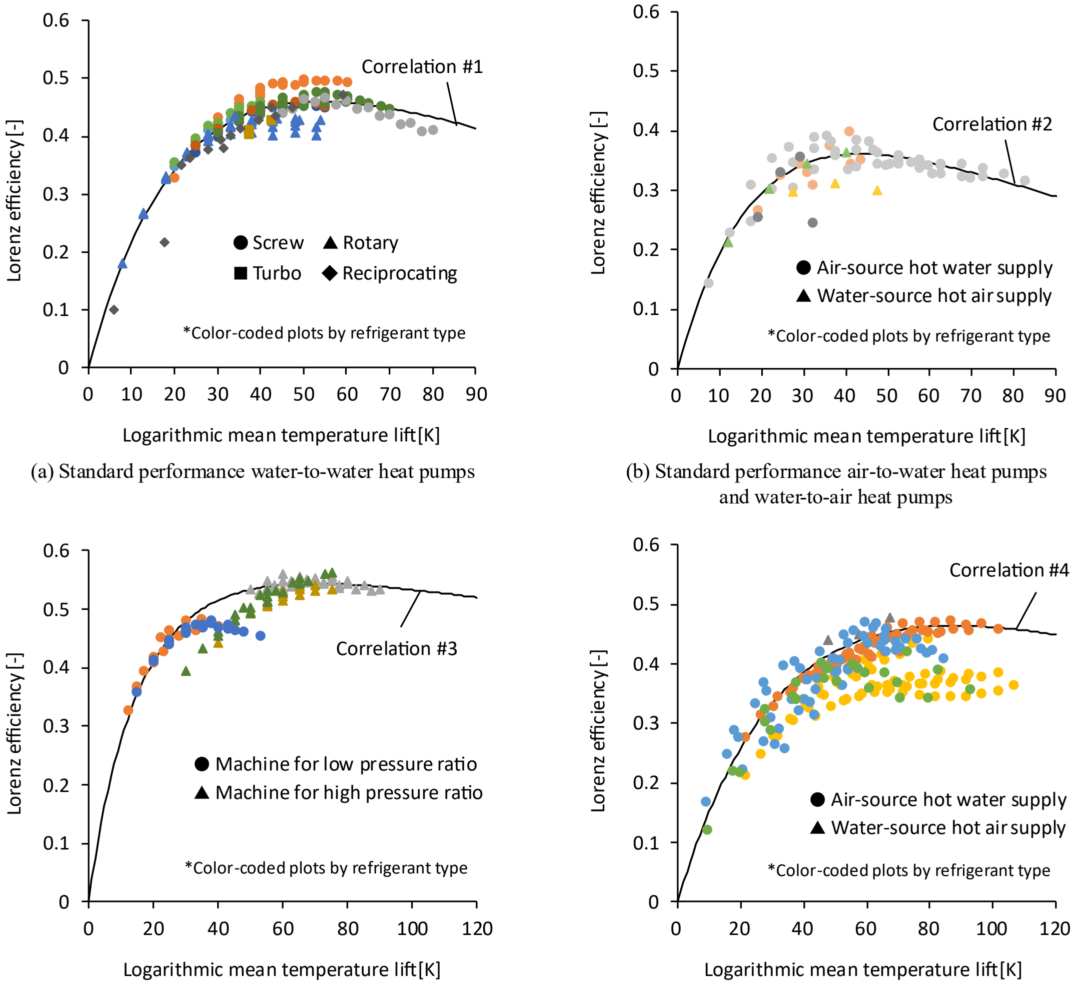
Figure 6 shows a graph illustrating the fitting results of the four correlation equations (#1, #2, #3, #4) with the catalog data.

As can be confirmed from Figs. 6 (a) and (b), correlation equations #1 and #2 for standard performance machines represent average performance.

For high-performance water-source hot water supply heat pumps shown in Fig. 6 (c), correlation equation #3 is developed to fit machines designed for low pressure ratios under conditions with small temperature lifts (approximately 30 K or less) and to fit machines designed for high pressure ratios under conditions with large temperature lifts (approximately 60 K or more). Therefore, this correlation equation #3 represents the maximum achievable performance of existing technology by selecting the appropriate technique according to the temperature lift.

For high-performance air-source hot water supply heat pumps and water-source hot air supply heat pumps shown in Fig. 6 (d), performance differences between product models are relatively large. Correlation equation #4 is formulated to represent the maximum performance when either heat source or heat sink medium is air.

The following points should be noted when applying these correlation equations. While correlation equations #1 and #2 represent the average performance of heat pumps equipped with standard single-stage compressors, correlation equations #3 and #4 are fitted to represent the highest achievable performance of existing technology. For concept screening of industrial heat pump integration, it is recommended to use the average performance calculated by correlation equations #1 and #2 for preliminary calculations, while treating the top-runner performance calculated by correlation equations #3 and #4 as a reference value. Therefore, it should be noted that the COP calculated by correlation equations #3 and #4 represents the performance of a relatively high performance subset of products and may overestimate the performance of other products.



(a) Standard performance water-to-water heat pumps

(b) Standard performance air-to-water heat pumps and water-to-air heat pumps

(c) High performance water-to-water heat pumps

(c) High performance air-to-water heat pumps and water-to-air heat pumps

Figure 6. Data fitting of correlation equations developed by the present study.

5.4. Validation of correlation equations for standard performance machines

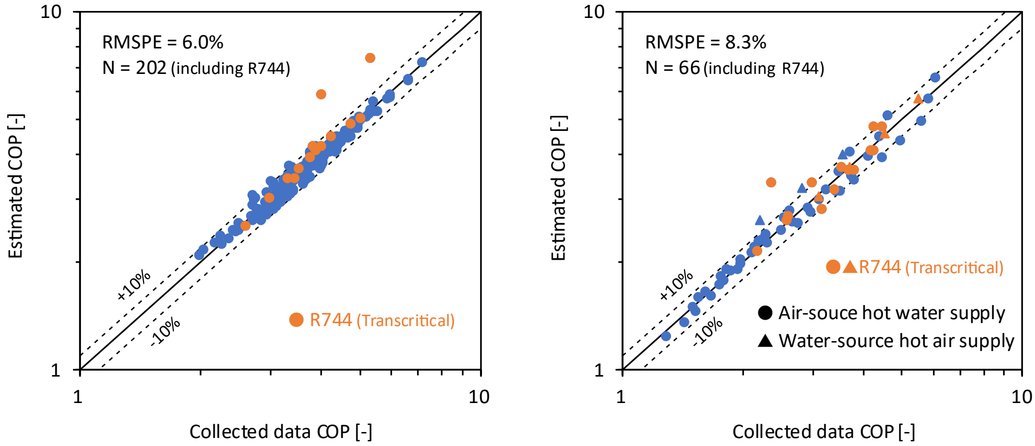
For the goodness-of-fit of correlation equations, the root mean square percentage error (RMSPE) defined by the following equation is used as an indicator.

$$RMSPE = 100\sqrt{\frac{1}{N}\sum_{j=1}^{N}\left(\frac{COP_{\mathrm{cal},j} - COP_{\mathrm{data},j}}{COP_{\mathrm{data},j}}\right)^2} \tag{11}$$

Here, COP_{data} is the COP of the collected data from product catalogs, COP_{cal} is the COP calculated using the correlation equations, and N is the number of data points.

For standard performance water-source hot water supply heat pumps in Fig. 7, including R744 (Transcritical) data, correlation equation #1 shows good fit with an RMSPE of 6.0%. Furthermore, for 96% (194 data points) of the data, the deviation from the correlation equation is within ±10%. However, the correlation overestimates the data for two points in R744. As described in Figure 2 (b), these data were obtained under conditions where the evaporation temperature was near the critical temperature of R744, which would explain why the COP is lower compared to other conditions. Since these two points represent operating conditions that are inherently unfavorable when using R744, excluding them results in an RMSPE of 4.2%.

Figure 7 (b) plots both air-source hot water supply heat pumps and water-source hot air supply heat pumps together. Correlation equation #2 shows good fit with an RMSPE of 8.3%. Furthermore, for 80% (53 data points) of the data, the deviation from the correlation equation is within ±10%.



(a) Correlation equation #1 for standard performance water -to-water heat pumps

(b) Correlation equation #2 for standard performance air-to-water heat pumps and water-to-air heat pumps

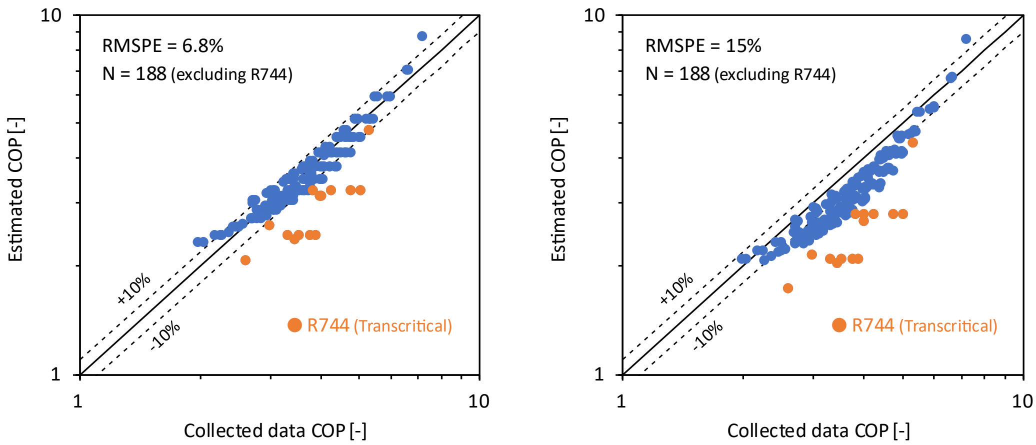
Figure 7. Validation of correlation equations developed by the present study.

As described above, correlation equations #1 and #2 for standard performance machines can calculate the average COP of industrial heat pump products equipped with standard single-stage compressors with an accuracy of approximately ±10%.

5.5. Evaluation of existing correlation equations

Existing correlation equations by Arpagaus, Eq. (1) and by Schlosser et al., Eq. (2) are also evaluated. It is necessary to keep in mind that both equations were developed based on industrial heat pump products in Europe, while collected data are for products in Japan.

Figure 8 shows a comparison between the collected data COP and the COP calculated using the correlation equations by Arpagaus and by Schlosser et al. Both equations use a simple temperature lift, so it is confirmed that they do not apply to R744 (Transcritical), which has large temperature glides. However, excluding the R744 data, the correlation equation by Arpagaus fits well with an RMSPE of 6.8%.



(a) Correlation equation by Arpagaus, Eq. (1)

(b) Correlation equation by Schlosser et al., Eq. (2)

Figure 8. Evaluation of existing correlation equations for water-to-water heat pumps.

In addition, both equations by Arpagaus and by Schlosser et al. are formulated to directly fit COPs, not the second law efficiencies. In the equation by Arpagaus, the exponent for temperature lift is -0.76, while in the equation by Schlosser et al. it is -0,89094. When converting COP to Carnot efficiency, defined as the ratio between the COP and Carnot COP, the exponent for temperature lift becomes positive, making the relationship a monotonically increasing function of temperature lift. For reference, Fig. 9 shows the Carnot efficiency plotted against simple temperature lift, calculated using both equations by Arpagaus and by Schlosser et al. It can be observed that both Carnot efficiencies increase monotonically with temperature lift. Within the temperature lift range shown in Fig. 9, the Carnot efficiency remains below 1, that is no issues arise. However, as the temperature lift increases, there is a possibility that the calculated Carnot efficiency may exceed 1. Therefore, while these equations are valid when used within their proper application ranges, they inherently contain the potential to violate the second law of thermodynamics.

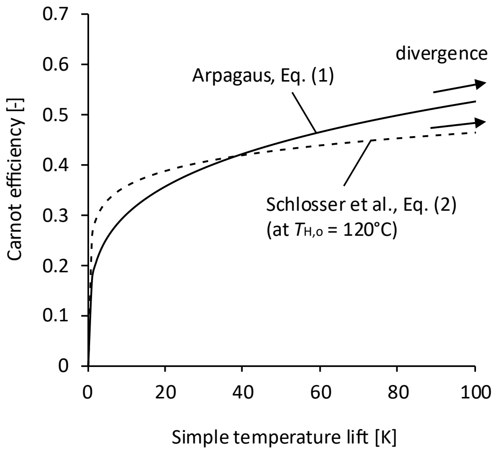


Figure 9. Characteristics of existing correlation equations by Arpagaus, Eq. (1) and by Schlosser et al., Eq. (2)

The correlation equation proposed in this paper differs from existing equations in two key aspects: its consideration of temperature glide and its reflection of the second law of thermodynamics. While the number of temperature data points required for COP calculation increases from two to four, transcritical cycle heat pumps with large temperature lifts can also be reproduced using a unified equation. Although less accurate than the correlation #1 developed in this study, the correlation equation by Arpagaus is useful considering it can be calculated more easily using a simple temperature lift, not a logarithmic mean temperature lift.

6. Conclusions

This study presented simplified correlation equations for calculating the coefficient of performance (COP) based on catalog performance data for industrial heat pump products in Japan. The correlation equations can be broadly categorized into two types. The first type equation targets standard performance machines, enabling calculation of the average COP for industrial heat pump products equipped with standard single-stage compressors with an accuracy of approximately ±10%. On the other hand, the second type equation targets high performance machines, representing performance levels achievable with existing technologies, including two-stage compression cycles and cascade cycles.

The correlation equations presented in this study are based solely on the correlation between Lorenz efficiency and logarithmic mean temperature lift. This can be calculated simply using the types of heat source and heat sink media and their inlet and outlet temperatures. Unlike existing equations based on the correlation between COP and simple temperature lift, the present equations are effective even for heat pumps with large temperature glides, such as transcritical cycle heat pumps.

The present correlation equations can be used to calculate average performance when considering the introduction of industrial heat pumps prior to determining a specific product model. Alternatively, it can serve as a benchmark indicator when developing new heat pumps.

However, the correlation equations depend on the technical characteristics of the products examined in this study. Therefore, when heat pumps with significantly different technical characteristics are commercialized in the future, the equation should be revised. Specifically, the different technical characteristics include vapor compression heat pumps using refrigerants (such as R718) with significantly different pressure ratios and temperature lifts relationships, and gas compression heat pumps employing reverse Brayton cycles or reverse Stirling cycles. Nevertheless, the shape of the correlation equation is consistent with the theoretical characteristics of heat pumps, so simply changing the parameters should make this equation valid.

Nomenclature

a, b, c, d	Fitting parameters [-]
COP	Coefficient of performance [-]
f	Factor [-]
Q	Heat [kJ]
$RMSPE$	Root mean square percentage error [%]
T	Temperature [K]
$\bar{T}$	Logarithmic mean temperature [K]
W	Work [kJ]

Greek symbols

Δ	Difference
η	Efficiency [-]

Subscripts

C	Heat source
cal	Calculation
Carnot	Carnot heat pump cycle
cmp	Compressor
data	Collected data
exp	Expansion device
H	Heat sink
HT	Heat transfer
i	Inlet
is	Isentropic process
lift	Temperature lift
Lorenz	Lorenz heat pump cycle
o	Outlet
p	Pinch point
ref	Refrigerant

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