Modeling and comparison of different capacity modulation strategies with focus on seasonal performance

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Abstract. Over the past few years, demand for energy efficient HVAC systems has been increasing rapidly, and this trend is likely to continue in the future. There are various modulation technologies that can respond to load variations by switching between capacity levels (tandem, two-stage compressors), PWM control (digital scroll), speed modulation (variable-speed). These technologies can be compared using a single number figure of merit, IPLV.SI which expresses part-load efficiency based on weighted operation at various partial capacities. A model is proposed for a water chiller application having a nominal cooling capacity of 8kW. System performance has been calculated using a 10-coefficient compressor model provided by the manufacturer and the most well-known correlations for heat transfer and pressure drop in a brazed plate heat exchanger. Model validation is performed with experimental data obtained in a parallel experimental investigation. Single-speed and two-stage compressor have a comparable IPLV.SI while a variable-speed compressor has 20% higher IPLV.SI.

1. Introduction
HVAC&R systems are required to operate under different loads. The load can vary due to different ambient weather conditions, increased demand during certain time of the day or the systems are oversized to account for the quick pull down. Modulating compressor technologies are a good solution for systems operating under these varying loads. The modulation also provides precise temperature and humidity control.

The major limitation of any comparison study on the modulation strategies to the best of our knowledge is that these comparisons are either limited to only a few of the existing modulation strategies or a comparison between different data sets obtained with different systems using different refrigerants and operating conditions. This makes driving any insights on the relative effectiveness of these modulation strategies difficult.

As a first step, this paper proposes a model for the water chiller application having a nominal cooling capacity of 8 kW. This paper presents the results of modeling investigations into the part-load performance of single-speed and two-stage compressor. These results are validated with experimental data obtained in the parallel experimental investigation. The validated system model is used to predict the performance of a variable-speed compressor.

The model calculates IPLV.SI (Integrated Part Load Value), a single figure of merit used to compare modulation strategies in water chillers according to AHRI Standard 551/591 (2018) [1].

2. Modulation strategies being considered
The following are the different compressor modulation strategies being considered as part of this study. As a first step, this paper compares a single-speed, two-stage, and variable-speed compressor. All the compressors are scroll and have similar capacity and belong to the same generation.

2.1. Single-speed compressor
A single-speed compressor modulates capacity by cycling i.e., turning on and off. The on/off capacity control is the simplest method of adjusting the predetermined temperature (setpoint) using a thermostat. After the set temperature is reached, the thermostat stops the compressor and circulating refrigerant in the cycle. When the thermostat detects a rise in air temperature, it turns the compressor on [2].

2.2. Two-stage compressor
This is a commercially available two-stage compressor. It is different from conventional two-stage compressors that use a low-pressure stage and a high-pressure stage. Instead, it has two internal bypass ports which enable the system to run at 67%-part load capacity. This compressor can be operated either at a high stage (100% capacity) or a low stage (67% capacity). [3].

2.3. Variable-speed compressor
This compressor modulates its capacity by varying the speed of the motor [4]. The focus will be on the inverter driver variable-speed compressor with a brushless DC motor. The selected compressor can be operated between 20 Hz and 100 Hz.

3. AHRI 551/591 (2018) standard
AHRI 551/591 is used for the determination of the part-load performance of water chillers. The standard defines a single number part-load efficiency figure of merit called Integrated Part Load Value (IPLV.SI) calculated at part load rating conditions. These part load rating conditions are shown in Table 1. IPLV.SI is the weighted average of the COPR measured at these standard rating conditions as shown in Equation (1). These factors in Equation (1) are based on the weighted average of the most common building types and operations using average weather in 29 U.S cities.

\[
IPLV.SI = 0.01 \cdot A + 0.42 \cdot B + 0.45 \cdot C + 0.12 \cdot D
\]

\[A = COP_R \text{ at } 100\%\]
\[B = COP_R \text{ at } 75\%\]
\[C = COP_R \text{ at } 50\%\]
\[D = COP_R \text{ at } 25\%\]

If a compressor cannot be unloaded to 25%, 50%, or 75% load point, then the compressor is run at the minimum step of unloading at the condenser entering water shown in Table 3 for 25%, 50%, or 75% capacity points as required. Once the \( \text{COP}_{\text{test}} \) is calculated at these conditions using Equation (2), it is degraded to \( \text{COP}_R \) using the Equation (3), (4), (5), and (6).

\[
\text{COP}_{\text{test}} = \frac{\dot{Q}_{ev,avg}}{\dot{W}_{cp}}
\]

\[
\text{COP}_R = \frac{\text{COP}_{\text{test}}}{G_D}
\]

\[
G_D = (-0.13 \cdot LF) + 1.13
\]

\[
LF = \frac{\text{(%Load)}(Q_{ev 100\%})}{(Q_{ev min\%load})}
\]
\[
\% Load = \frac{(\text{Part load net capacity})}{(\text{Full load rated net capacity})}
\]

(6)

Table 1: AHRI 551/591 part load conditions for IPLV.SI

| Condition | Part load ratio (%) | Condenser Inlet/Outlet (°C) | Evaporator Inlet/Outlet (°C) |
|-----------|---------------------|-----------------------------|-------------------------------|
| A         | 100                 | 30/35                       | 12/7                          |
| B         | 75                  | 24.5/*                      | */7                           |
| C         | 50                  | 19/*                        | */7                           |
| D         | 25                  | 19/*                        | */7                           |

Table 1 shows the part load conditions. As seen in the table, condenser inlet, outlet, and evaporator inlet, outlet are mentioned for A condition while for the B, C, and D conditions, only the condenser inlet and evaporator outlet temperature are mentioned. 20% ethylene glycol by volume in water (Water Ethylene Glycol mixture, WEG) is used as the secondary fluid. Standard required that the condenser and evaporator WEG flow rate used for the A condition be used for the B, C, and D conditions.

4. System model

R410A is the refrigerant used in this study. The properties of the refrigerant and Water Ethylene Glycol (WEG) mixture are calculated using REFPROP [5] database. The effects of oil circulation are not considered in the model.

4.1. Compressor

The compressor model is developed using the ARI 10-coefficients provided by the manufacturer. Mass flow rate and compressor energy consumption are expressed as a function of the evaporation temperature \( T_{ev} \) and the condensation temperature \( T_{co} \) for a rated superheat. The variable-speed compressor is a 20-coefficient model expressed as a function of compressor speed \( RPM \), \( T_{ev} \) and \( T_{co} \)

\[
\dot{m}_{\text{rated}} = M_0 + M_1 \cdot T_{ev} + M_2 \cdot T_{co} + M_3 \cdot RPM + M_4 \cdot T_{ev} \cdot T_{co} + M_5 \cdot T_{ev} \cdot RPM + M_6 \cdot T_{co} \\
\cdot RPM + M_7 \cdot T_{ev}^2 + M_8 \cdot T_{co}^2 + M_9 \cdot RPM^2 + M_{10} \cdot T_{ev} \cdot T_{co} \cdot RPM + M_{11} \\
\cdot T_{ev}^2 \cdot T_{co} + M_{12} \cdot T_{ev}^2 \cdot RPM + M_{13} \cdot T_{ev}^3 + M_{14} \cdot T_{co}^2 \cdot T_{ev} + M_{15} \cdot T_{co}^2 \\
\cdot RPM + M_{16} \cdot T_{co}^3 + M_{17} \cdot RPM^2 \cdot T_{ev} + M_{18} \cdot RPM^2 \cdot T_{co} + M_{19} \cdot RPM^3
\]

(7)

\[
W_{cp} = P_0 + P_1 \cdot T_{ev} + P_2 \cdot T_{co} + P_3 \cdot RPM + P_4 \cdot T_{ev} \cdot T_{co} + P_5 \cdot T_{ev} \cdot RPM + P_6 \cdot T_{co} \cdot RPM \\
+ P_7 \cdot T_{ev}^2 + P_8 \cdot T_{co}^2 + P_9 \cdot RPM^2 + P_{10} \cdot T_{ev} \cdot T_{co} \cdot RPM + P_{11} \cdot T_{ev}^2 \cdot T_{co} \\
+ P_{12} \cdot T_{ev}^2 \cdot RPM + P_{13} \cdot T_{ev}^3 + P_{14} \cdot T_{co}^2 \cdot T_{ev} + P_{15} \cdot T_{co}^2 \cdot RPM + P_{16} \cdot T_{co}^3 \\
+ P_{17} \cdot RPM^2 \cdot T_{ev} + P_{18} \cdot RPM^2 \cdot T_{co} + P_{19} \cdot RPM^3
\]

(8)

where \( M_0-M_{19} \) and \( P_0-P_{19} \) are constant coefficients

AHRI 540 (2015) recommends a mass flow correction while calculating mass flow rate for a different superheat from rated superheat. The change of superheat has been reported to have negligible effect on compressor power consumption [6].

\[
\dot{m}_{\text{corrected}} = \{1 + F_v\left(\frac{\dot{m}_{\text{rated}}}{\dot{m}_{\text{corrected}}} - 1\right)\} \cdot \dot{m}_{\text{rated}}
\]

(9)

where:

\[ F_v = \text{Volumetric efficiency correction factor} \]

- a value of one (1) is used as an approximation

\[ \dot{m}_{\text{corrected}} = \text{Refrigerant mass flow rate at suction condition, kg/s} \]
\[ \dot{m}_{\text{rated}} = \text{Refrigerant mass flow rate at rated superheat, kg/s} \]
\[ v_{\text{corrected}} = \text{Specific volume at suction condition, m}^3/\text{kg} \]
\[ v_{\text{rated}} = \text{Specific volume at rated condition, m}^3/\text{kg} \]

Compressor efficiencies are used to validate the compressor model with the experimental setup. Isentropic compressor efficiency is a ratio between the isentropic work \( W_{\text{isen}} \) and real compressor power consumption \( W_{\text{cp}} \)

\[ \eta_{\text{isen}} = \frac{W_{\text{isen}}}{W_{\text{cp}}} \]  \hspace{1cm} (10)

where \( W_{\text{isen}} = \dot{m}_{\text{corrected}} \cdot (h_{\text{cp,isen}} - h_{\text{cp,i}}) \)

Compressor volumetric efficiency is a ratio of the actual delivered gas volume to the theoretic swept volume of the compressor

\[ \eta_{\text{vol}} = \frac{v_{\text{corrected}} \cdot \dot{m}_{\text{corrected}}}{V_{\text{dis}} \cdot f} \]  \hspace{1cm} (11)

where \( V_{\text{dis}} \) is the displacement volume of the compressor and \( f \) is the operating frequency of the compressor expressed in \([m^3/rev]\) and \([rev/s]\) respectively.

Compressor isentropic and global efficiencies can be expressed as a function of the pressure ratio \( \tau \) between condensation and evaporation pressure

\[ \eta_{\text{isen}} = A + B \cdot \tau + C \cdot \tau^2 \]  \hspace{1cm} (12)
\[ \eta_{\text{vol}} = D + E \cdot \tau \]  \hspace{1cm} (13)

where \( A- E \) are regression coefficients.

4.2. Brazed plate heat exchanger

The heat transfer between WEG and refrigerant occurs in Brazed Plate Heat Exchangers (BPHE). Each BPHE is divided into finite elements along the flow of the fluid, pressure drop, and heat transfer correlations are applied to this element. End plate effects are neglected in BPHE model.

**Figure 1**: Counter flow in a discrete element

For each discrete element shown in Fig. 1, heat transfer is calculated using LMTD approach shown in Eq. (14) and (15) and energy balance between the two fluids shown in Eq. (16).

\[ Q = UA \cdot \text{LMTD} \]  \hspace{1cm} (14)
\[ \text{LMTD} = \frac{(T_{\text{h},i} - T_{\text{c},o}) - (T_{\text{h},o} - T_{\text{c},i})}{\ln((T_{\text{h},i} - T_{\text{c},o})/(T_{\text{h},o} - T_{\text{c},i}))} \]  \hspace{1cm} (15)

where \( UA \) is the overall heat transfer coefficient, \( \text{LMTD} \) is the log mean temperature difference. Eq. (15) represents \( \text{LMTD} \) calculation for a counter flow heat exchanger.

\[ Q = \dot{m}_c \cdot c_v \cdot (T_{\text{c},o} - T_{\text{c},i}) = \dot{m}_h \cdot c_h \cdot (T_{\text{h},i} - T_{\text{h},o}) = \dot{m}_{\text{ref}} \cdot (\Delta h) \]  \hspace{1cm} (16)

where \( c \) is the specific heat \([kJ/(kg K)]\), \( \dot{m} \) is the mass flow rate \([kg/s]\) and \( \Delta h \) is the refrigerant enthalpy.

The procedure to calculate the heat transfer in a discrete element involves taking an initial guess of the one of the outlet/inlet temperatures. Then the other outlet/inlet temperatures and \( Q \) are found using Equation (16). Next, a new value of \( \dot{Q} \) is determined using Equation (14). If the newly calculated heat flux matches the previous one, the element computation is complete. Otherwise, \( \dot{Q} \) from Equation (14) becomes the new estimate for Equation (16) and another computation sequence is performed [7].

\[ \frac{1}{UA} = \left( \frac{1}{ht_{\text{weg}}A_{\text{weg}}} + \frac{1}{ht_{\text{ref}}A_{\text{ref}}} \right) \]  \hspace{1cm} (17)
where \( htc_{weg} \) and \( htc_{ref} \) are the heat transfer coefficients of WEG and refrigerant respectively.

Correlations are used to predict \( htc_{weg} \) and \( htc_{ref} \). These correlations change according to the phase of the refrigerant and whether it is a single phase or two phase heat transfer.

### 4.2.1. Single phase heat transfer

Eq. (18) is used for the WEG and refrigerant heat transfer coefficients [8]. This correlation is valid for

\[ 50 \leq Re \leq 8000; \ 2 \leq Pr \leq 290. \]

\[
Nu = (-1.342 \times 10^{-4} \cdot \beta^2 + 1.808 \times 10^{-4} \cdot \beta - 0.0075) 
\cdot Re^{(-7.956 \times 10^{-5} \cdot \beta^2 + 9.687 \times 10^{-3} \cdot \beta + 0.3155}) \cdot Re^{\phi / \beta} \cdot Re^{\gamma / \beta} \cdot Pr^{1/3} \cdot \left( \frac{\mu}{\mu_w} \right)^{0.14}
\]

where \( Nu \) is the Nusselt number, \( Re \) is Reynolds number, \( Pr \) is Prandtl number and \( \mu \) is the dynamic viscosity. The other factors in the Equation (18) are dependent on BPHE geometry. \( \beta \) is the herringbone angle measured with vertical, \( \phi \) is the enlargement factor and \( \gamma \) is corrugation profile aspect ratio of the BPHE.

\[
Nu = htc \cdot D_h / k_w
\]

where \( D_h \) is the hydraulic diameter and \( k_w \) is the thermal conductivity. Heat transfer coefficient is calculated using Equations (18) and (19).

\[
Re = \frac{\rho V D_h}{\mu}
\]

\[
V = \frac{m}{\rho n Wb}
\]

where \( \rho \) is density, \( V \) is the velocity inside the BPHE, \( n \) is the channel number for corresponding side, \( W \) is the width of the BPHE and \( b \) is the corrugation depth.

### 4.2.2. Boiling

Han et al. developed the following correlation to model the boiling in R410A [9]. Nusselt number is given by

\[
Nu = Ge_1 \cdot Re_{eq}^{Ge_2} \cdot Bo_{eq}^{0.3} \cdot Pr^{0.4}
\]

where \( Re_{eq} \) is equivalent Reynolds number, \( Bo_{eq} \) is the equivalent Boiling number, \( Ge_{eq} \) is the equivalent mass flux and the coefficients \( Ge_1 \) and \( Ge_2 \) are functions of the BPHE geometry and are given by the below Equations (23), (24), (25) and (26).

\[
Re_{eq} = \frac{Ge_{eq} D_h}{\mu Q}
\]

\[
Bo_{eq} = \frac{AG_{eq} \Delta h_{fg}}{Q}
\]

\[
Ge_{eq} = \left( \frac{m}{Wbn} \right) \left[ 1 - x + x \left( \frac{\rho_f}{\rho_g} \right)^{0.5} \right]
\]

\[
Ge_1 = 2.81 \left( \frac{p_{corr}}{D_h} \right)^{-0.041} \cdot \beta^{-2.83}
\]

\[
Ge_2 = 0.746 \left( \frac{p_{corr}}{D_h} \right)^{-0.082} \cdot \beta^{0.61}
\]

where \( p_{corr} \) is BPHE pitch, \( x \) is vapor quality and \( \Delta h_{fg} \) is enthalpy of vaporization.

### 4.2.3. Condensation

Han et al. developed a similar correlation for the condensation heat transfer. The proposed Nusselt number has the following form [10].

\[
Nu = Ge_1 \cdot Re_{eq}^{Ge_2} \cdot Pr^{1/3}
\]
where the coefficients $G_{e_1}$ and $G_{e_2}$ are functions of the BPHE geometry. These are different from the ones given by Equations (26) and (27).

$$G_{e_1} = 11.22 \left( \frac{P_{corr}}{D_h} \right)^{-2.83} \cdot \beta^{-4.5}$$  \hspace{1cm} (29)

$$G_{e_2} = 0.35 \left( \frac{P_{corr}}{D_h} \right)^{0.23} \cdot \beta^{1.48}$$  \hspace{1cm} (30)

4.3. Solving algorithm

![Flow chart](image)

**Figure 2**: Algorithm for the solving the system model

A sequential iteration algorithm was designed and implemented as shown in the flow chart in Figure 2 [11]. Newton-Raphson method was used to calculate the iteration step. First, the condenser and evaporator WEG boundary conditions, as well as the required superheat and subcooling are provided as inputs. The WEG boundary conditions depend on the condition being modelled. If it is A condition, the inlet and outlet WEG temperatures are specified for the condenser and evaporator. For the B, C and D conditions, WEG inlet temperature and mass flow rate are provided as inputs.

The system parameters such as compressor suction ($P_{cp,ri}$), discharge pressure ($P_{cp,ro}$), evaporator inlet pressure ($P_{ev,ri}$), compressor speed (RPM) (for a variable-speed compressor) are initialized. Starting with the compressor modules, discharge state and refrigerant mass flow rate are determined. These are input to the condenser and subcooler modules. Once, these two BPHE modules converge, they output subcooling temperature at the subcooler outlet ($x_{sc,ro}$). The discharge pressure of the compressor is updated to match the required subcooling ($x_{subcoa}$).

Then, expansion valve (modelled as an isenthalpic process) outputs the evaporator inlet temperature and pressure. The evaporator model outputs the superheat at the compressor inlet ($x_{ev,ro}$). Evaporator
inlet pressure is updated to match the required superheat ($x_{SUc}$). Then the suction pressure i.e. evaporator outlet pressure (calculated with the pressure drop calculation) is compared with the initially guess compressor suction pressure. The iteration is stopped if these two pressures match, if not the suction pressure is changed and model reiterates. In case of a variable-speed compressor, ($Q_{ev}$) is compared to the ($Q_{req}$) and the compressor speed is changed accordingly.

5. Results
The results from the system model are validated using the experimental data from a parallel study. The experimental results from a single-speed and two-stage are used for the system validation. Then the system model is used to estimate the performance of a variable-speed compressor.

5.1. Model validation
In a parallel experimental study, R410A Water Ethylene Glycol (WEG) chiller was built to compare the different modulation strategies.

![Compressor isentropic efficiency](image3)

![Compressor volumetric efficiency](image4)

**Figure 3:** Compressor isentropic efficiency  
**Figure 4:** Compressor volumetric efficiency

![Evaporator cooling capacity prediction](image5)

![System COP prediction](image6)

**Figure 5:** Evaporator cooling capacity prediction  
**Figure 6:** System COP prediction

Fig. 3 and Fig. 4 show the results of the regression fit compressor efficiencies as a function of pressure ratio ($\tau$) with the manufacturer data. The experimental data is also shown on the same curve with solid points. Experimentally determined compressor efficiencies have a maximum difference of 11% compared to the regression fit curves.

The modelled cooling capacity and COP are compared with the experimentally determined cooling and COP in Fig. 5 and Fig. 6. Most of the predictions are within ±10%.

5.2. Single-speed compressor, two-stage and variable-speed compressor part load performance
The part load performance of single-speed and two-stage compressor are shown in Fig. 7 and Fig. 8. A single-speed compressor can only operate at one stage. As the percent load changes from 100% to 25%, the standard requires a reduced condenser WEG inlet temperature, this reduces the condensing temperature and pressure which in turn causes capacity to increase and compressor power to reduce in
a single-speed compressor. A two-stage compressor can be operated at either high stage or low stage. For the A condition, the two-stage compressor is operated at a high stage. For the B, C, and D conditions, as the required load reduces, the compressor is operated at the low stage. The low stage cannot match capacity at C, D condition, thus incurring cycling losses. The shaded region is a representation of the cycling losses, which are higher in a single-speed compressor.

![Figure 7. Variation of capacity, power with percent load for single-speed compressor](image)

![Figure 8. Variation of capacity, power with percent load for two-stage compressor](image)

![Figure 9. Variation of capacity, power with percent load for variable-speed compressor](image)

The part load performance of variable-speed compressor is shown in Fig. 9. The compressor can vary its speed between 100 Hz and 20 Hz, thus able to match the capacity with the load. The power consumption reduces from A to D because of the lower operation speed.

5.3. **Comparison between single-speed, two-stage and variable-speed compressor**

Fig. 10 shows the $COP_{test}$ given by Equation (2). All three compressors have similar capacity and power consumption at the 100% load, thus the $COP_{test}$ are around the same. It is interesting to note that at the 75% load, though the two-stage compressor is operating at the low stage, the variable-speed compressor is running at lower frequency, the single-speed compressor has higher $COP_{test}$ than two-stage. However, the $COP_{test}$ for the variable-speed compressor is significantly higher at the lower percent loads because of reduced capacity and reduced compressor power. The $COP_{test}$ are 2% higher for a single-speed compressor at the B, C and D conditions compared to a two-stage compressor.

Fig. 11 shows the $COP_R$ given by Equation (3). Except for the 100% load when all the compressors have similar performance, the variable-speed compressor has a higher $COP_R$ than other two compressor. This trend can also be graphically explained by the cycling losses represented in Fig. 7, 8 and 9. $COP_{test}$ is higher for a single-speed compressor compared to a two-stage compressor at B, C and D conditions. However, only single-speed compressor undergoes cycling losses at B condition. Though both the two-stage and single-speed compressor undergo cycling losses at C and D conditions, two-stage compressor has lower degradation factor because of its operation in low stage which leads to similar $COP_R$ in both these compressors.
Figure 10. Comparison of \(COP_{\text{test}}\) between single-speed, two-stage and variable-speed compressor

IPLV.SI is calculated using the \(COP_R\) from Fig. 11 and Equation (1). Single-speed and two-stage compressors have an IPLV.SI of 4.0 while the variable-speed compressor has an IPLV.SI of 4.8. This 20% difference can be explained by the higher \(COP_R\) of the variable-speed compressor at 25%, 50% and 75% load which have the higher weighting factors than 100% load in Equation (1). Though \(COP_{\text{test}}\) are higher for a single-speed compressor compared to a two-stage compressor, IPLV.SI are the same because of higher cycling losses incurred by a single-speed compressor.

6. Conclusions

This study compared a single-speed, two-stage and a variable-speed compressor using a system model. All the compressors have similar capacity and belong to the same generation. The system model was validated with results from a parallel experimental study and then the model is used to estimate the performance of a variable-speed compressor. AHRI 551/591 was used to determine the IPLV.SI for these compressors. A variable-speed compressor has the highest IPLV.SI of 4.8 because of its modulation. Single stage compressor has a higher \(COP_{\text{test}}\) than a two-stage compressor but it is more penalized for the cycling losses which causes both the compressors to have similar IPLV.SI of 4.0.

7. Nomenclature

**Symbols**

- \(A\) heat transfer area \([\text{m}^2]\)
- \(b\) corrugation depth \([\text{m}]\)
- \(B_o\) boiling number
- \(B_{\text{PHE}}\) brazed plate heat exchanger
- \(c\) specific heat \([\text{kJ/kg-K}]\)
- \(C_d\) degradation coefficient [-]
- \(COP\) coefficient of performance [-]
- \(D_h\) hydraulic diameter \([\text{m}]\)
- \(f\) compressor frequency \([\text{s-1}]\)
- \(F_v\) volumetric efficiency factor [-]
- \(G\) mass flux \([\text{kg/s m}^2]\)
- \(G_{e_1}, G_{e_2}\) Han – Lee – Kim correlation coefficients
- \(h\) enthalpy \([\text{kJ/kg}]\)
- \(htc\) heat transfer coefficient \([\text{W/m}^2\text{K}]\)
- \(IPLV.SI\) integrated part load value [-]
- \(k\) thermal conductivity \([\text{W/mK}]\)
- \(LF\) load factor [-]
- \(\dot{m}\) mass flow rate \([\text{kg/s}]\)

**Greek**

- \(\beta\) herringbone angle \([\text{[^\circ]}]\)
- \(\gamma\) Corrugation profile aspect ratio [-]
- \(\epsilon\) error [-]
- \(\eta\) efficiency [-]
- \(\mu\) dynamic viscosity \([\text{Pa-s}]\)
- \(\rho\) density \([\text{kg/m}^3]\)
- \(\tau\) compression ratio [-]
- \(\phi\) enlargement factor [-]
- \(\Delta\) difference [-]

**Subscripts**

- \(\text{avg}\) average
- \(c\) cold-side
- \(\text{co}\) condenser
- \(\text{cp}\) compressor
- \(\text{dis}\) displacement
- \(\text{eq}\) equivalent
\[ M_0 - M_{19} \quad \text{ARI compressor coefficients} \]
\[ Nu \quad \text{Nusselt number [-]} \]
\[ P_0 - P_{19} \quad \text{ARI compressor coefficients} \]
\[ p_{corr} \quad \text{heat exchanger pitch} \]
\[ Pr \quad \text{Prandtl number [-]} \]
\[ Q \quad \text{heat transfer rate [kW]} \]
\[ Re \quad \text{Reynolds number [-]} \]
\[ T \quad \text{temperature [°C]} \]
\[ U \quad \text{overall heat transfer coefficient [W/m}^2\text{K]} \]
\[ v \quad \text{specific volume [m}^3\text{/kg]} \]
\[ V \quad \text{fluid velocity in BPHE [m/s]} \]
\[ W \quad \text{BPHE width [m]} \]
\[ W_{E} \quad \text{water ethylene glycol} \]
\[ x \quad \text{vapor quality [-]} \]

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