Steady-State Numerical Simulation Of A Vapor Compression Heat Pump System As An Effective Method To Predict Its Performance

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Steady-State Numerical Simulation of a Vapor Compression Heat Pump System as an Effective Method to Predict its Performance

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ABSTRACT

During the last few decades a strong driving force has been directed to the development of simulation models and methodologies for vapor compression systems and its main components. These models are highly useful as an effective method to predict the system performance and the thermodynamic processes of the vapor compression system under different operating conditions or different component’s properties, as well as refrigerant replacements. This paper presents the mathematical model of a heat pump and numerical steady-state simulations with refrigerant R407C compared to experimental data. Using the information from manufacturer’s data and real properties of scroll compressor, brazed plate heat exchangers and additionally built-in liquid-vapor heat exchanger, the mathematical model of the thermodynamic and heat transfer process in its different components is developed in order to predict the operating conditions of the heat pump. With numerical simulations the effect of: the liquid-vapor heat exchanger; evaporator, condenser and desuperheater constructive properties; different temperatures of the external fluids at condenser’s and evaporator’s inlets; the effect of degree of superheat and degree of subcooling, as well as refrigerant replacement are analyzed. The final goal is to predict the thermodynamic states, pressure drops, heat transfer rates and system performance of the heat pump and to identify the heat pump configuration which will result in the most adequate operating conditions.

1. INTRODUCTION

In refrigeration and air-conditioning systems numerical simulations have recently increased due to new regulations resulting in further refrigerant replacements. The model of any refrigeration or heat pump system can be useful tool for performance prediction and analysis of design (redesign) of system components. With these models evaluation of the design of future refrigeration devices, or prediction of performance for existing heat pump or refrigeration system with new refrigerant can be done.

Mathematical modelling of vapor compression systems requires the combination of thermodynamic and heat transfer calculations to prepare the model as a useful tool for such analyses. The models can be used by engineers to predict the interactive effect of the components size on the system design and its operational characteristics. The capability to predict these effects can enhance the ability of engineers to make wise decisions in the design process and thereby shortening the design cycle, as pointed out by Quiao et al. (2010).

The aim of this paper is to present the mathematical model of a heat pump, by modelling the thermodynamic and heat transfer features across main components, and achieving simulation results of thermodynamic states, pressure drops, heat transfer rates and system performance prediction for different operating conditions, as well as influence of variable geometry on the concept design.
2. DESCRIPTION OF THE HEAT PUMP AND ITS MAIN COMPONENTS

The considered heat pump is a liquid-water heat pump for heating and cooling and also for hot water production (Figure 1). The heat pump is reversible and includes a scroll compressor, brazed plate heat exchanger (condenser, evaporator and desuperheater), a coaxial liquid-vapor heat exchanger, a four way valve and a thermostatic expansion valve. The refrigerant is the zeotropic mixture R407C.

![Figure 1: Schematic representation of the heat pump with liquid-vapor heat exchanger (winter-heating mode)](image)

(Numbers in square brackets are related to refrigerant states in the model description)

The compressor is a hermetic scroll compressor with a displacement of 9.44 m³/h for a rotational speed of 2900 rpm. The condenser and the evaporator are identical brazed plate heat exchangers with 20 effective plates, whereas the desuperheater has 30 effective plates.

The liquid – vapor heat exchanger is used with the goal of analyzing the effect on the system performance of using this heat exchanger. According to the Figure 1, when the liquid – vapor heat exchanger is used (not used) the valves A1, A2, B1 and B2 are opened (closed), and the valves A3 and B3 are closed (opened).

The four way valve is used for the reversion between winter or summer mode. In this paper only the winter mode of the heat pump work is considered, so the complete function of this valve is not used.

3. THERMODYNAMIC AND HEAT TRANSFER MODEL OF THE HEAT PUMP

The mathematical model takes into account specific data and dimensions of components with goal to predict the real operating conditions of the heat pump. The following general assumptions are made:

1. The heat losses in refrigerant pipes are negligible (pipes are insulated);
2. Heat transfer between the heat exchangers and surroundings is negligible;
3. Pressure drops in the refrigerant pipes between main components are negligible.

Total heat transfer coefficients \( k \) for all heat exchangers are calculated according to the following equation:

\[
k_{\text{heat exchanger}} = \frac{1}{\alpha_{\text{refrigerant}} + \frac{c_{\text{wall}}}{\lambda_{\text{wall}}} + \frac{1}{\alpha_{\text{second flow medium}}}}.
\]

3.1 Hermetic scroll compressor

From the manufacturer’s data two efficiencies were defined as a function of the pressure ratio \( RC \) to predict the refrigerant mass flow rate and electric power for the compression process.
The volumetric efficiency \( \eta_{\text{vol}} \), was obtained by a simple regression analysis, with a coefficient of determination \( R^2 \) of 96.5%:

\[
\eta_{\text{vol}} = 1.04783 - 0.028886 \cdot RC.
\]  

(2)

The overall efficiency \( \eta_{\text{ov}} \), was obtained as a higher order polynomial fit as a function of pressure ratio, with a coefficient of determination \( R^2 \) of 91.6%:

\[
\eta_{\text{ov}} = 0.257014 + 0.366384 \cdot RC - 0.0951739 \cdot RC^2 + 0.00679058 \cdot RC^3.
\]  

(3)

Pressures at the compressor’s outlet and at inlet define the pressure ratio \( RC = p_{\text{out}} / p_{\text{in}} \). The refrigerant enthalpy and temperature at the compressor outlet were obtained using the next two equations:

\[
h_{[2]} = \frac{h_{[2]i} - h_{[1]i}}{\eta_{\text{in}}} + h_{[1]i} \rightarrow T_{[2]} = f \left( h_{[2]i}; p_{[2]} \right),
\]  

(4)

\[
\eta_{\text{in}} = 0.96 - 0.00046 \cdot N + 9.4 \cdot 10^{-8} \cdot N^2 + 0.07 \cdot RC - 0.0018 \cdot RC^2.
\]  

(5)

where the presented coefficients given in equation (5) are obtained from the literature Zhou et al. (2010), Shafiei et al. (2013) and Leducq et al. (2003). The refrigerant mass flow rate \( m_{\text{ref}} \) is calculated from the compressor displacement \( V_{\text{comp}} \), the volumetric efficiency \( \eta_{\text{vol}} \) and the refrigerant density at the compressor inlet \( \rho_{[1]} \):

\[
m_{\text{ref}} = \dot{V}_{\text{comp}} \cdot \eta_{\text{vol}} \cdot \rho_{[1]}.
\]  

(6)

The electric power of compressor \( W_{\text{comp,elec}} \) is obtained from the overall efficiency and the isentropic power:

\[
W_{\text{comp,elec}} = \frac{m_{\text{ref}} \cdot \left( h_{[2]i} - h_{[1]i} \right)}{\eta_{\text{ov}}} = \frac{\dot{W}_c}{\eta_{\text{ov}}},
\]  

(7)

The idea is to use only compressor manufacturer data about overall efficiency in order to calculate the compressor electric power. The manufacturer overall efficiency data is based on the compressor isentropic power, so the total engaged electric power is obtained by dividing isentropic power of the compressor with overall efficiency, equation (3).

3.2 Thermostatic expansion valve

The thermostatic expansion valve is assumed to be perfectly insulated; therefore, an isenthalpic process \( h_{[0]} = h_{[1]} \) is considered. It is assumed that the thermostatic valve controls a constant superheat degree at the evaporator’s outlet.

3.3 Evaporator

The total heat transfer rate of evaporation process is related by applying the energy balance for the refrigerant flow on one side flow, and propylene-glycol water mixture in the counter current flow on the opposite side of the brazed plate heat exchanger, given by the following equations:

\[
\dot{Q}_e = \dot{m}_{\text{ref}} \cdot \left( h_{[1]i} - h_{[1]} \right),
\]  

(8)

\[
\dot{Q}_e = \dot{m}_{\text{PGW}} \cdot c_{\text{pgw}} \cdot \left( T_{\text{PGW,exin}} - T_{\text{PGW,exout}} \right).
\]  

(9)

The evaporation heat transfer process is observed in two parts, boiling heat transfer and heat transfer of superheated phase. Dutto et al. (1991) and Fernando et al. (2004) suggested a model with the logarithmic mean temperature difference for both parts. The same method is applied in Longo and Gasparella (2007a) for the vaporization process in this commercial heat exchanger types:
\[ \Delta T_{E,ln} = \left[ \frac{\dot{Q}_{\text{boil}}}{\Delta T_{E,ln,boil}} \right] + \left[ \frac{\dot{Q}_{\text{sup}}}{\Delta T_{E,ln,sup}} \right], \]  
\[ \Delta T_{E,ln,boil} = \ln \left( \frac{T_{\text{PGW,E,in}} - T_{\text{E,in+1}}}{T_{\text{PGW,E,out}} - T_{\text{E,in+1}}} \right), \]  
and for superheating zone:

\[ \Delta T_{E,ln,sup} = \ln \left( \frac{T_{\text{PGW,E,in}} - T_{\text{E,in}}} {T_{\text{PGW,E,out}} - T_{\text{E,in}}} \right). \]

The total heat transfer coefficient of the boiling part of the evaporator \( k_{E,boil} \) and total heat transfer coefficient of the superheating part of the evaporator \( k_{E,sup} \) are calculated according to equation (1).

The temperature of the propylene-glycol water mixture between the superheating zone and the boiling zone \( T_{\text{PGW,E,m}} \) is calculated from an energy balance:

\[ T_{\text{PGW,E,m}} = T_{\text{PGW,E,in}} - \dot{m}_{\text{PGW}} \cdot c_{\text{PGW}} \cdot \left( \frac{T_{\text{E,in+1}} - T_{\text{E,in}}} {\dot{m}_{\text{PGW}} \cdot c_{\text{PGW}}} \right). \]

The heat transfer coefficient on the refrigerant side for the boiling process \( \alpha_{E,ref,boil} \) is calculated from Copper (1984). The heat transfer coefficient of the propylene-glycol water mixture \( \alpha_{E,PGW} \) and of the superheating phase (refrigerant single phase – vapor) in the evaporator are calculated from Thonon (1995).

### 3.4 Condenser

In general, the condensation process in this type of equipment can be divided into three zones: vapor desuperheating, condensation and liquid subcooling. However, in our case, the refrigerant side surface of the condenser was always lower than the refrigerant dew point temperature. As a result, only two zones (condensation and liquid subcooling) are considered in the analysis.

The total heat transfer in the condenser is given by the three following equations:

\[ \dot{Q}_{\text{C, total}} = \dot{m}_{\text{ref}} \cdot (h_{[4]} - h_{[5]}), \]  
\[ \dot{Q}_{\text{C, total}} = \dot{m}_{\text{C,W}} \cdot c_{\text{C,W}} \cdot (T_{\text{C,W, out}} - T_{\text{C,W, in}}), \]  
\[ \dot{Q}_{\text{C, total}} = \dot{Q}_{C} + \dot{Q}_{C,\text{sub}}. \]

The surface of the condenser is also divided into two zones:

\[ A_{\text{C, total}} = A_{C} + A_{\text{C, sub}}. \]

For the condensation process it is considered that the subcooling region starts somewhere in the “middle” of the condenser; then:
The logarithmic mean temperature difference for the condensation process is calculated by the following equation:

$$\Delta T_{\text{C,ln}} = \frac{(T_{(i=1)} - T_{\text{W,in}})}{\ln \left(\frac{T_{(i=1)} - T_{\text{W,ref}}}{T_{(i=0)} - T_{\text{C,W}}}\right)}.$$  \hspace{1cm} (24)

Similarly,

$$\Delta T_{\text{C,sub,ln}} = \frac{(T_{\text{ref}} - T_{\text{W,in}})}{\ln \left(\frac{T_{\text{ref}} - T_{\text{C,W}}}{T_{[i]} - T_{\text{W,ref}}\right)}.$$  \hspace{1cm} (28)

In this paper a correction factor \( F_i \) for the condensation of R407C has been included as suggested by Shah and Focke (1988). The correction factor is calculated as a function of the number of transfer units, according to Mancin et al. (2011). The heat transfer coefficient for the water side \( \alpha_{\text{C,W}} \) is calculated from Thonon (1995) and the computation approach suggested by Mancin (2011) is used for the refrigerant side.

### 3.4 Liquid – Vapor heat exchanger

An energy balance for the liquid and vapor side of this heat exchanger gives the following equations:

$$\dot{Q}_{\text{LVHX}} = \dot{m}_{\text{ref}} \cdot (h_{[i]} - h_{[g]}) \hspace{1cm} (29)$$

$$h_{[i]} - h_{[g]} = h_{[3]} - h_{[6]} \hspace{1cm} (30)$$

For the liquid phase, the heat transfer coefficient is calculated from the widely known correlations for friction factor and Nusselt number of Petukhov (1970) and Gnielinski (1976), respectively. The heat transfer coefficient for the vapor flow in the concentric annular duct was calculated according to Petukhov and Roizen (1964) and Petukhov (1970), using a modified form of the Gnielinski (1976) correlation for turbulent flow.

### 3.5 Desuperheater

In this paper the mathematical model of the desuperheater was simplified. It is considered that during hot water production the refrigerant will not condense. The heat exchanger is modelled a single phase flow heat exchanger in both sides using the correlations given by Thonon (1995).

### 3.7 Pressure drops in the refrigerant circuit

The frictional pressure drops on the refrigerant side in the evaporator, condenser and desuperheater are calculated based on accordance to Collier and Thome (1996). The total pressure drop \( \Delta P_{E,i} \) is calculated from the frictional \( \Delta P_{E,f,i} \), momentum \( \Delta P_{E,m} \), gravity and the manifolds and ports \( \Delta P_{E,p} \) pressure drops:

$$\Delta P_{E,i} = \Delta P_{E,f,i} + \Delta P_{E,m} + \Delta P_{E,g} + \Delta P_{E,p}.$$  \hspace{1cm} (31)

The momentum and gravity pressure drops are estimated by the homogeneous model for the two-phase flow as follows:
\[ \Delta P_{\text{v,E}} = \frac{m_{\text{v,E}}^2}{2 \rho_{\text{v,E}}} \]  
\[ \Delta P_{\text{g,E}} = g \cdot \rho_{\text{g,E}} \cdot H_e, \]  

where \((v_{\text{L,E}})\) and \((v_{\text{v,E}})\) are the specific volume of evaporator’s liquid and vapor phase, \((\Delta X_e)\) is the vapor quality change between inlet and outlet at the evaporator. The average two-phase density between inlet and outlet is calculated by following equation:

\[ \rho_{\text{m,E}} = \frac{X_{\text{m,E}} + \left(1 - X_{\text{m,E}}\right)^{-1}}{\rho_{\text{v,E}}} \]  

where \((X_{\text{m,E}})\) is the average vapor quality between inlet and outlet of the evaporator.

The pressure drops in the inlet and outlet manifolds and ports are empirically estimated, in accordance with Shah and Focke (1988), as follows:

\[ \Delta P_{\text{v,d}} = 1.5 \cdot \frac{m_{\text{v,\text{in,E}}}^2}{2 \rho_{\text{v,E}}}. \]  

The frictional pressure drop calculation was based on the experimental data from Longo and Gasparella (2007-a) and Longo and Gasparella (2007-b), the frictional pressure drops of refrigerant during the evaporation process in the brazed plate heat exchanger:

\[ \Delta P_{\text{f,I}} = 1553 \cdot \frac{m_{\text{v,\text{in,E}}}^2}{2 \rho_{\text{v,E}}}. \]  

In the condenser, a similar procedure was followed. For calculating the frictional pressure drops the Longo (2009) correlation was used. The same model is set for the desuperheater where, in this case, the momentum pressure rise is set to zero because it is considered that refrigerant will not condense (no phase change).

Pressure drops in the liquid-vapor heat exchanger are also included in the mathematical model the friction factor for the liquid and vapor sides (concentric annular duct) are calculated (for turbulent flow) from Petukhov (1970) and Gnielinski (1976).

The pressure drop in the 4 way valve of 14000 Pa is obtained from manufacturer data.

### 3.8 System performance of the heat pump

The coefficient of performance \((COP)\) is calculated from the cooling or heating capacity (winter/summer mode) and the total electric power consumption for the heat pump:

\[ COP_{\text{inst}} = \frac{\dot{Q}}{W_{\text{TOTAL,elec}}}. \]  

The total power consumption includes the compressor and three pumps: for the propylene-glycol water mixture, for the heating water in the condenser, and also for the desuperheater (water side). For all three pumps the needed electrical power is obtained from the manufacturer data as a function of the volumetric flow rate:

\[ W_{\text{ele,pump}} = a \cdot q_v + b \cdot q_v + c, \]  

\[ W_{\text{TOTAL,elec}} = \dot{W}_{\text{comp,elec}} + \dot{W}_{\text{elec,pump,PGW}} + \dot{W}_{\text{elec,pump,H}} + \dot{W}_{\text{elec,pump,DE}}. \]

### 4. VALIDATION OF THE MODEL

For the validation of the mathematical model some preliminary tests were performed on the experimental station with the heat pump described in the section 2. The experimental facility has been equipped with a supervisory control and data acquisition (SCADA). During each experiment steady state conditions were reached within a period of 2-3 hours while data recording during the next 30 minutes was done for measured temperatures and pressures.
A set of 19 steady state test during different operating conditions with R407C as refrigerant were performed in order to validate the model. The inputs used in order to validate the model were: temperature and volumetric flow rate of the propylene – glycol water mixture (“PGW”) at the evaporator inlet, temperature and volumetric flow rate of water at the condenser inlet, degree of superheat controlled by thermo-expansion valve and subcooling degree measured at condenser outlet. All the experiments were performed without the liquid-vapor heat exchanger. Figures 2 and 3 compare some experimental and simulation results. It can be seen that the accordance between experimental and simulation results are reasonable. As a result, the model can be used as an effective tool for the heat pump analysis.

Figure 2: Deviation between simulated and experimental values of temperatures and pressures of refrigerant R407C

Figure 3: Deviation between simulated and experimental values of total input power for the heat pump (electric power of compressor and three pumps)

5. SIMULATION RESULTS AND DISCUSSION

Across simulation results, the functionality of the model as well as the different possibilities for using the model are presented in this section. The simulations were done with (LVHX-ON) and without (LVHX-OFF) liquid-vapor heat exchanger. For all the analyses the following input parameters were set: degree of superheat of 3°C; subcooling degree of 2°C; temperature and volume flow rate of the propylene-glycol water mixture at the evaporator inlet of 8°C and 42 lit/min, respectively; temperature and volume flow rate of water at the condenser inlet of 30°C and 35 lit/min, respectively. In these results the use of the desuperheater for hot water production was not considered.

Degree of superheat and degree of subcooling
Figure 4 shows the effect of the degree of superheat and the degree of subcooling on the COP, using or not the LVHX. It can be seen that the higher the degree of superheat or degree of subcooling, the lower the COP. Also, it is concluded that the use of the LVHX is not recommended from an energy efficiency point of view (COP).

Second flow medium temperatures at the inlet of the evaporator and the condenser
Figure 5 shows that, as expected, the model predicts that increasing the temperature of the water at the condenser inlet leads to lower COP’s whereas increasing the temperature of the propylene-glycol water mixture at the
evaporator inlet leads to higher values of COP. Figure 5 also shows that the relationship between these variables is nearly linear.

Figure 4: Influence of degree of superheat and degree of subcooling on the system performance (COP) when “LVHX” is in and out of operation

Figure 5: Influence of water temperature on condenser inlet and propylene-glycol water mixture at evaporator inlet on the system performance (COP), when LVHX is used and not used

Effective plates in the evaporator and condenser

Figure 6 shows the influence on the system performance (COP) and heat exchanger pressure drops when the number of effective plates in the evaporator and condenser are varied between 8 and 58. As expected, increasing the number of plates leads to higher COP values and lower pressure drops in the heat exchangers. Results show that the actual heat pump design (with 20 effective plates in the evaporator and condenser) has some margin of improvement; for example if the number of plates is increases from 20 to 30, then the COP increases from 3.48 to 3.66. The final decision on the number of plates should also consider a cost/economics function.

Figure 6: Influence on the system performance (COP) and pressure drops in the evaporator and the condenser by varying the number of effective plates in both heat exchangers
6. CONCLUSION

This paper covers the concept of the mathematical modeling of the vapor compression heat pumps and refrigeration systems. The modelling procedure considers the real dimensions and heat transfer features in all main components significant for the good prediction of the thermodynamic states and processes of the presented heat pump. The following conclusions are obtained:

- Presented mathematical model is an useful tool for the thermodynamic analysis of the vapor compression systems and it’s geometrical properties;
- Validation of the model gives the positive trend of compared temperatures and pressures, as also the total power of the heat pump (compressor and 3 pumps);
- Simulation results show that using the model it is possible to perform different analyses in many ways to predict the thermodynamic states, pressure drops, heat transfer rates and system performance of the heat pump and to identify the heat pump configuration which will result in the most adequate operating conditions;
- This kind of models can be useful tools to predict the working conditions for different refrigerants and also allow modifying some components in order to obtain the optimal working conditions.

NOMENCLATURE

\[
\begin{align*}
A & \quad \text{Area} \quad (\text{m}^2) \\
c_p & \quad \text{Specific heat capacity} \quad (\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}) \\
e & \quad \text{Thickness of the wall/tube} \quad (\text{m}) \\
g & \quad \text{Gravity} \quad (\text{m} \cdot \text{s}^{-2}) \\
h & \quad \text{Specific enthalpy} \quad (\text{J}\cdot\text{kg}^{-2}) \\
h' & \quad \text{Specific enthalpy in the “middle of the condenser} \quad (\text{J}\cdot\text{kg}^{-2}) \\
H & \quad \text{Plate height (from center of connectors)} \quad (\text{m}) \\
k & \quad \text{Overall heat transfer coefficient} \quad (\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}) \\
m & \quad \text{Mass flow rate} \quad (\text{kg}\cdot\text{s}^{-1}) \\
\dot{m}_{\text{th}} & \quad \text{Mass flux} \quad (\text{kg} \cdot \text{m}^{-2}\cdot\text{s}^{-1}) \\
N & \quad \text{Compressor rotation speed} \quad (\text{Hz}) \\
p & \quad \text{Pressure} \quad (\text{bar}) \\
\dot{Q} & \quad \text{Heat transfer rate} \quad (\text{W}) \\
q_v & \quad \text{Volumetric flow rate} \quad (\text{m}^3\cdot\text{s}^{-1}) \\
T & \quad \text{Temperature} \quad (\text{K}) \\
T' & \quad \text{Temperature in the “middle” of the condenser} \quad (\text{K}) \\
W & \quad \text{Power} \quad (\text{W}) \\
\end{align*}
\]

Greek symbols

\[
\begin{align*}
\alpha & \quad \text{Convective heat transfer coefficient} \quad (\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}) \\
\Delta p & \quad \text{Pressure drop} \quad (\text{bar}) \\
\Delta T & \quad \text{Temperature difference} \quad (\text{K}) \\
\Delta T_u & \quad \text{Logarithmic mean temperature difference} \quad (\text{K}) \\
\lambda & \quad \text{Thermal conductivity} \quad (\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}) \\
\rho & \quad \text{Density} \quad (\text{kg}\cdot\text{m}^{-3}) \\
\end{align*}
\]

Subscripts

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>Boil</td>
<td>Boiling (Boiling phase)</td>
</tr>
<tr>
<td>C</td>
<td>Condenser</td>
</tr>
<tr>
<td>DES</td>
<td>Desuperheater</td>
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<tr>
<td>E</td>
<td>Evaporator</td>
</tr>
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<td>elec</td>
<td>Electric</td>
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<tr>
<td>PGW</td>
<td>Propylene-glycol water mixture</td>
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<tr>
<td>ref</td>
<td>Refrigerant</td>
</tr>
<tr>
<td>sub</td>
<td>Subcooling (subcooling region)</td>
</tr>
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<td>Superheating (superheating phase)</td>
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in Inlet
inst Installation
is Isentropic
L Liquid
m Middle
V Vapor
W Water
(x=1) Dry saturated vapor
(x=0) Hot liquid

REFERENCES


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