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A Simulation for Predicting the Refrigerant Flow Characteristics Including Metastable Region in Non-Adiabatic Capillary Tubes

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Abstract

The capillary tube/suction line heat exchanger (SLHX) is widely used in small refrigeration systems. The refrigerant flowing in the SLHX experiences frictional and accelerational head losses, flashing, and heat transfer simultaneously. The simulation of refrigerant flow through SLHX is important since this will help engineers analyze and optimize the SLHX incorporated in a refrigeration system. The present SLHX model is based on conservation equations of mass, momentum and energy. Also a meta-stable model is included. All these equations are solved simultaneously. In this paper, HFC-134a refrigerant flow through a non-adiabatic capillary tube is simulated. The simulation results are discussed but not validated against experimental measurements yet.

1. Introduction

Expansion device in vapor compression refrigeration cycle controls the refrigerant flow rate along with vapor refrigerant compressor. The capillary tube is widely used as an expansion device in small refrigeration systems. The capillary tube is usually made from a long hollow copper tube with an internal diameter from 0.05 and 1.0 mm. The capillary tube may be soldered to a suction-line tube, which is in between evaporator and compressor, to form a counter-flow heat exchanger as shown in Figure 1. This is called the SLHX and widely used in domestic refrigerators. This SLHX may improve the performance of a vapor compression refrigeration cycle.

The flow phenomena that are expected to take place inside the capillary tube of a SLHX include subcooled liquid, meta-stable liquid, flashing, condensation due to heat transfer, single-phase and two-phase wall friction, etc as can be seen in the Figure 1. The flow behavior inside capillary tubes and the effect of capillary tube on vapor compression refrigeration cycle have been studied extensively over the past half century. Koizumi and Yokoyama (1980) studied about characteristics of refrigerant flow in capillary tube. Chen et al. (1990) developed a correlation for meta-stable flow of R-12 through capillary tubes. Dirik et al. (1994) carried out both numerical and experimental studies on adiabatic and non-adiabatic capillary tubes with R-134a. Peixoto and Bullard (1994) developed a simulation and design model for capillary tube-suction line heat exchanger. Mezavila and Melo (1996) developed a simulation model for non-adiabatic capillary tubes using R-134a. Bansal (1998) presented a homogeneous two-phase flow model. Wongwises et al. (2003) predicted the refrigerant flow characteristics including meta-stable region in adiabatic capillary tubes. Recently, Garcia-Valladares (2007) developed the simulation model of non-adiabatic capillary tubes considering meta-stable region. However, there have been little attempts to simulate refrigerant flow characteristics in a SLHX with consideration of meta-stable region. A reliable SLHX model is required in simulating refrigeration cycle, which will play an important role in enhancing energy efficiency of domestic refrigeration systems. In this regard, the present work aims to develop a SLHX model which takes meta-stable region into consideration. The present model can be applied to analyses of the refrigerant flow characteristics inside a SLHX and prediction of the incremental length of the non-adiabatic capillary tube due to meta-stable phenomenon.

2. Physical Models

The refrigerant flow prediction through capillary tube of a SLHX is very complicated because single & two-phase frictional head loss, flashing, and heat transfer take place simultaneously. In order to simplify the problem, a SLHX model is developed based on steady-state homogeneous two-phase flow model. No entrained oil is assumed in the model. The SLHX forms a count-flow heat exchanger.

The capillary tube can be divided into three sections according to the physical contact with suction line: inlet adiabatic region, non-adiabatic region, outlet adiabatic region. On the other hand, the refrigerant may exist in each region at different physical state such as subooled liquid flow, meta-stable liquid flow, and two-phase flow. Any possibility of the refrigerant status in each region should be considered for simulation.

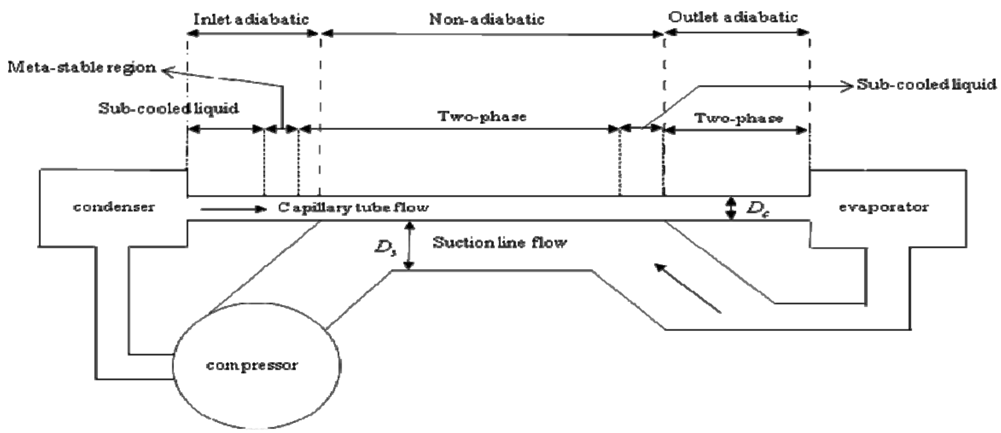

Fig. 1. Schematic of capillary-suction line heat exchanger.

The conservation equations of mass, momentum, and energy are required to predict the refrigerant flow behavior. Since steady-state is assumed, the mass flow rate in the capillary tube and suction line should be the same as a constant.

$$\dot{m}_C = \dot{m}_S = \dot{m} \tag{1}$$

where, subscript *C* and *S* represent capillary tube and suction line pipe, respectively. The momentum conservation equation which is applicable to both of capillary tube and suction line pipe can be written as

$$-\frac{dp}{dz} = f\frac{G^2 v}{2D} + G^2\frac{dv}{dz} + \frac{g\sin\theta}{v} \tag{2}$$

The terms on the right hand side represent wall shear, momentum, and elevation change of refrigerant, respectively. The wall shear is evaluated based on Darcy friction factor such as $\tau_w = f\rho V^2/8 = fG^2 v/8$. In order to make use of Eq.(2) for head loss calculation, friction factor (f) correlation is required. For single-phase flow, Churchill's equation [6] is used. This equation takes wall roughness effect into account, which may have large influence in case of small diameter tubes like a capillary.

$$f_{sp} = 8\left[\left(\frac{8}{\mathrm{Re}_{sp}}\right)^{12} + \frac{1}{(A+B)^{1.5}}\right]^{1/12} \tag{3}$$

Where

$$A = \left(2.457\ln\left[\frac{1}{(7/\mathrm{Re}_{sp})^{0.9} + 0.27\varepsilon/D}\right]\right)^{16} \tag{4}$$

$$B = \left(\frac{37530}{\mathrm{Re}_{sp}}\right)^{16} \tag{5}$$

The single-phase flow Reynolds number is defined as follows:

$$\mathrm{Re}_{sp} = \frac{GD}{\mu} \tag{6}$$

For two-phase flow, friction factor is evaluated using Lin et al. [7] correlation which is as follows:

$$f_{tp} = \phi_{lo}^2 f_{sp} \left(\frac{v_{sp}}{v_{tp}} \right) \tag{7}$$

where f_{sp} is calculated by Eq. (3) and two-phase multiplier, ϕ_{lo}^2, is given by

$$\phi_{lo}^2 = \left[\frac{\left(\frac{8}{\mathrm{Re}_{tp}} \right)^{12} + \frac{1}{(A_{tp} + B_{tp})^{1.5}}}{\left(\frac{8}{\mathrm{Re}_{sp}} \right)^{12} + \frac{1}{(A_{sp} + B_{sp})^{1.5}}} \right]^{\frac{1}{12}} \left[1 + x \left(\frac{v_v}{v_1} - 1 \right) \right] \tag{8}$$

The two-phase flow Reynolds number (Re_{tp}) can be calculated by the same definition as Eq.(6) only with replacement of single-phase viscosity (μ) by two-phase viscosity (μ_{tp}). For two-phase viscosity, an average two-phase viscosity of McAdam' model is used.

$$\frac{1}{\mu_{tp}} = \frac{x}{\mu_v} + \frac{1-x}{\mu_l} \tag{9}$$

The energy conservation equation for steady-state, steady flow, vapor-phase refrigerant flow through suction-line can be written as follows:

$$d\dot{Q} = \dot{m}di = \dot{m}C_p dT \tag{10}$$

It was assumed that there is no work and changes in kinetic energy and potential energy are negligible. The heat transfer rate in Eq. (10) is obtained from a relationship given by

$$d\dot{Q} = UA(T_c - T_s) \tag{11}$$

where, T_c, T_s are average temperature of capillary tube and suction-line in a control volume. The temperature and thermodynamic properties of a control volume are assumed to be constant. The uncertainties induced by this assumption are minimized if we make control volume size as small as possible. The above equation is associated with overall thermal conductance, UA, which is obtained from the following thermal resistance relationship.

$$\frac{1}{UA} = \frac{1}{h_c D_c \pi dz} + \frac{\ln(D_{c,o} / D_c)}{2\pi k_{c,w} dz} + \frac{\delta}{k_{solder} w dz} + \frac{\ln(D_{s,o} / D_s)}{2\pi k_{s,w} dz} + \frac{1}{h_s D_s \pi dz} \tag{12}$$

Each term of right side hand of Eq. (12) represent capillary-side convective thermal resistance, conductive thermal resistance of capillary tube wall, conductive thermal resistance of soldering, conductive thermal resistance of suction line wall, and suction line-side convective thermal resistance, respectively. For single-phase flow, the convective heat transfer coefficient is evaluated by Gnielinski's equation[9] as

$$Nu = \frac{(f/8)(R_e - 1000)P_r}{1 + 12.7(f/8)^{0.5}(P_r^{2/3} - 1)} \tag{13}$$

For two-phase flow, the convective heat transfer coefficient on capillary tube wall is assumed to be infinite because the main thermal resistance of the heat exchanger region lies on the suction line side. The validity of this assumption has been confirmed by Mezavila et al. k_{solder}, δ, and w denote thermal conductivity, thickness and width of soldering joint. In practice, tin soldering is applied only one side as shown in Figure 2. In this work, however, soldering joint is assumed to be symmetric as shown in Figure 3.

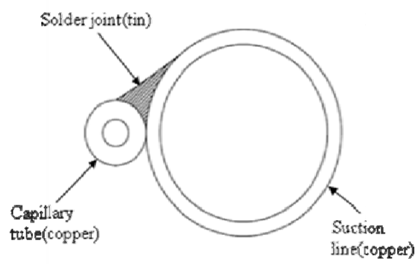

Fig. 2. SLHX soldering in practice

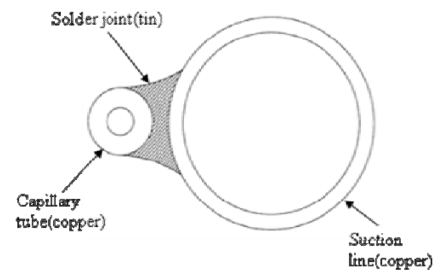

Fig. 3. SLHX soldering in model

The meta-stable region due to the delay of vapourization, the point of inception of vapourization may not occur at the end point of sub-cooled liquid region. It has been observed from some experiments that the vapourization does not occur at the intersection of pressure and temperature lines in the liquid region but delays downstream from this point. This is due to the fact that the formation of the first vapour bubble requires a finite amount of superheat. Chen et al.[10] studied the flow of R12 through capillary tubes and attempted to develop a method based on the classic nucleation theory to predict the delay of vapourization. The correlation is as follows:

$$\frac{(P_s - P_v)\sqrt{kT_s}}{\sigma^{3/2}} = 0.679(\frac{v_G}{v_G - v_L})\,\mathrm{Re}_L^{0.914}(\frac{\Delta T_{sub}}{T_c})^{-0.208}(\frac{D}{D'})^{-3.18} \tag{14}$$

The range of operating conditions which are as follows:

$0.464\times10^4 < \mathrm{Re} < 3.74\times10^4$, $0 < \Delta T_{sub} < 17°C$, $0.66 < D < 1.17mm$

Where D' is the reference length ($\sqrt{kT_s/\sigma}\times10^4$), k is the Boltzman constant (1.380662×10^{-23}), P_v is the pressure at flashing point and T_c is the critical temperature of refrigerant. The thermodynamic properties used to determine P_v are evaluated at the saturated state.

3. Numerical Solution

The present model uses the section by section method. This method divides whole capillary tube and suction line into successive multiple sections as shown in Figure 4. Heat transfer and thermodynamic properties of each control volume are calculated successively by an iterative method. The length of control volumes used in the simulation model is 1 mm. The thermodynamic and transport properties of refrigerant are evaluated using REFPROP V7.0 [11].

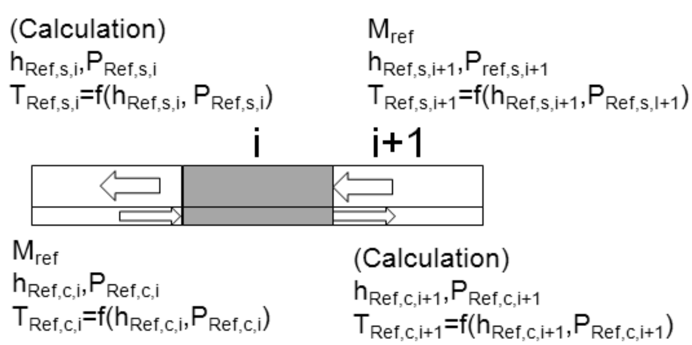

Fig. 4. The schematic of the SLHX for section-by-section calculation

Figure 5 shows the flowchart of numerical solution process. The calculation process proceeds along the direction of the refrigerant flow through capillary tube. First determine the phase of refrigerant and location (adiabatic or non_adiabatic) from input data (geometry, mass flow rate, temperature, pressure, enthalpy). Then the other properties of refrigerant are calculated from REFPROP V 7.0 database. The properties of i^{th} control volume are the input data of $(i+1)^{th}$ control volume. Each control volume needs to be calculated successively until it reaches the other end of a SLHX. An iteration process is required in order to get solutions for each control volume. It is because the enthalpy, pressure, and transport properties of a control volume are interdependent. Because the form of the capillary tube and the suction line heat exchanger is the lateral counter flow heat exchanger, the properties of the first control volume of the suction line are assumed in the solution process. After completion of a solution process for all control volumes, the assumed values are compared with the calculated value. If convergence criterion in not met, the values assumed or calculated in the pre-step are replaced by the newly calculated values. This iteration goes on until convergence criterion is met. The geometric and operational parameters of SLHX used in this simulation are listed in Table 1.

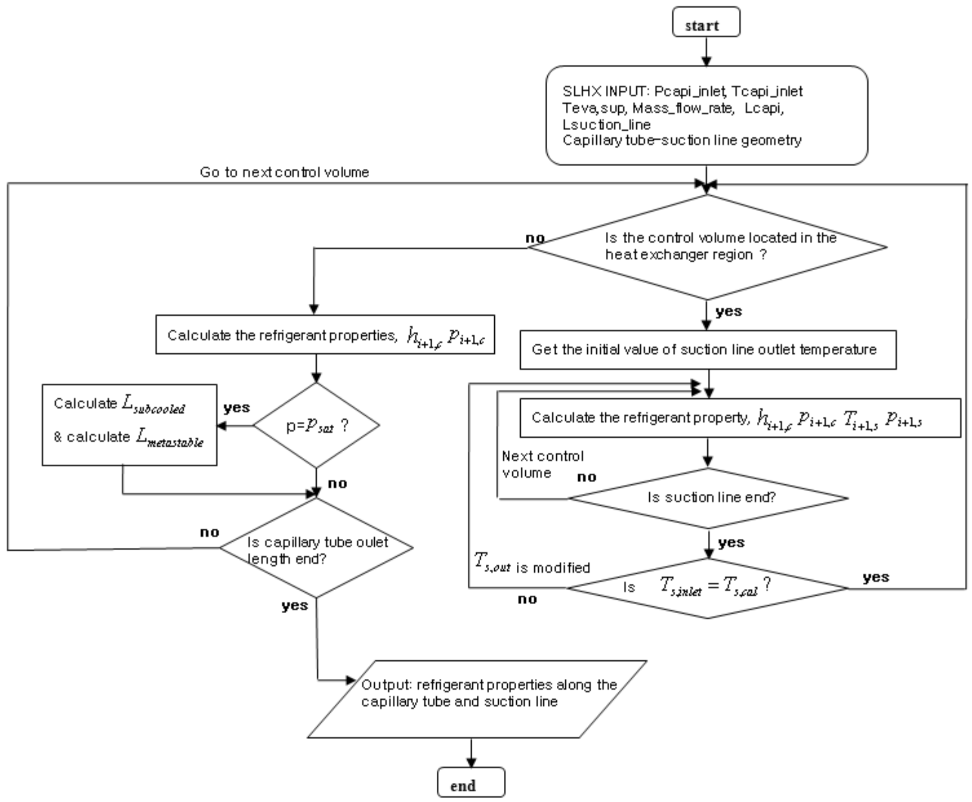

Fig. 5. Flowchart for SLHX calculations

Table 1 Geometric and operational parameters of SLHX simulated

Parameter	Value(unit)	Parameter	Value(unit)
Capillary tube inlet length (no metastable)	1.0 (m)	Roughness of capillary tube inner side	3.81 (μm)
Heat exchanger length	0.93 (m)	Roughness of suction line inner side	3.81 (μm)
Capillary tube outlet length	0.42 (m)	Mass flow rate	2 (kg/hr)
Capillary tube internal diameter	0.83 (mm)	Condenser outlet temperature	304.1 (K)
Capillary tube external diameter	2.0 (mm)	Sub-cooling temperature	1 (K)
Suction line internal diameter	6.0 (mm)	Super-heated temperature	4 (K)
Suction line external diameter	9.0 (mm)	Soldering joint width	4 (mm)
Soldering joint material	tin	Soldering joint thickness	1 (mm)

4 Results and Discussions

The calculation results for refrigerant flow through a non-adiabatic capillary tube are plotted on a Mollier diagram of Figure 6. A refrigerant having 1K subcooling flows into the capillary tube entrance. Figure 6 shows enthalpy is kept constant while pressure decreases in both of inlet adiabatic region and outlet adiabatic region. This plot also shows an enthalpy decrease in the non-adiabatic region. Figure 7 shows the variation in quality along the tube length. In the inlet adiabatic region, the quality starts to increase from the location around 300mm. The pressure decreases as the refrigerant flows along the capillary tube and reach this location to start flashing. A pressure variation trend along the capillary tube is plotted in Figure 8. The refrigerant pressure is on the decrease as it flows through the inlet adiabatic region at a subcooled liquid state. During the initial period that refrigerant remains at subcooled liquid state the pressure decreases in a linear manner. This plot shows a meta-stable region retards the change of refrigerant state from subcooled liquid to a two-phase flow so that increased pressure loss is delayed. When flashing starts the pressure drop is enhanced. Figure 9 shows a temperature variation along the capillary tube length. The temperature remains at constant level during subcooled liquid flow but starts to decrease when it starts flashing at around 300mm. In the non-adiabatic region, the quality is on the decrease. It means that the condensation due to heat transfer is stronger than flashing due to pressure drop in this region. This heat transfer leads to a slowdown of capillary temperature decrease and an increase in suction line temperature as shown in Figure 9. Note that the flow directions of capillary flow and suction line flow are opposite. Further heat transfer leads to a zero-quality or liquid subcooling at around 1800mm. This introduction of an additional subcooled liquid region results in an increase in capillary tube length required to make the same pressure difference. The subcooled liquid flows out of the non-adiabatic region and enter the outlet adiabatic region. In this region the refrigerant will experience throttling without heat transfer. So the refrigerant starts flashing and the quality increases. We can see the gradient of quality change in the outlet adiabatic region is larger than that in the inlet adiabatic region. It is because the vapor specific volume in outlet adiabatic region is larger than that in the inlet adiabatic region due to pressure change. Furthermore, acceleration pressure loss becomes larger in this region. That is why significant portion of pressure drop occurs in the outlet region.

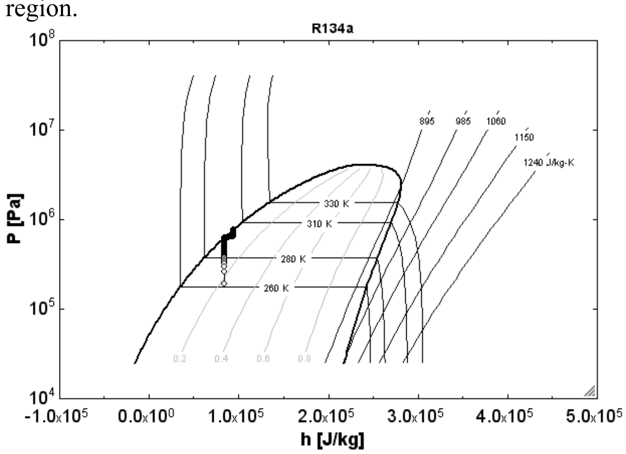

Fig. 6. The P-h diagram of the refrigerant flow through capillary tube.

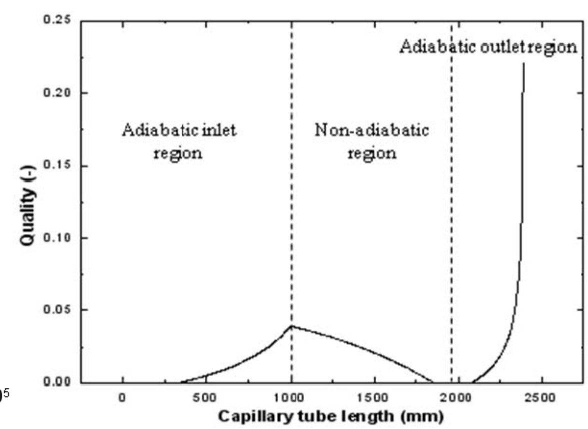

Fig. 7. Variation of the quality along the capillary tube.

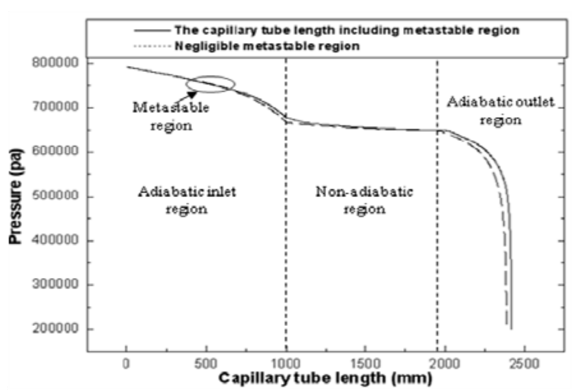

Fig. 8. Pressure change along the SLHX

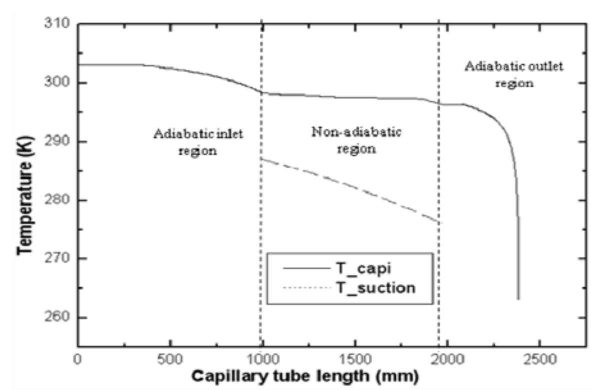

Fig. 9. Temperature variation along the SLHX

5 Concluding Remarks

A set of equations was established that can be used for prediction of refrigerant flow characteristics inside a adiabatic and non-adiabatic capillary tube. The equations are based on conservation of mass, momentum, and energy of refrigerant. The present model was applied to a SLHX in which heat transfer takes place from capillary tube side to suction line side. The inclusion of meta-stable region model allows an increase in capillary tube length for the same pressure difference and a change in capillary exit temperature. It was noteworthy that the refrigerant flow at two-phase state may turn into a subcooled liquid state as it flows forward through a capillary tube due to heat transfer. It is caused by the fact that the entropy reduction due to heat transfer was larger than the entropy generation due to capillary tube wall friction. This subcooled liquid turns into two-phase flow again as it flows further through adiabatic capillary region. This phase change during expansion process leads to an increase in capillary tube length required and a change in exit temperature.

Nomenclature		
A	cross sectional area	(m^2)
C_p	specific heat	(kJ/kg K)
D	diameter	(m)
f	friction factor	(-)
G	mass flux	(kg/s m)
g	gravity acceleration	(m/s^2)
h	specific enthalpy or heat transfer coefficient	(kJ/kg) or (kW/m^2 K)
k	conductivity	(kW/m.k)
L	length	(m)
$\dot{m}$	mass flow rate	(kg/s)
Nu	Nusselt number	(-)
Pr	Prandtl number	(-)
p	pressure	(pa)
q	heat flux	(kJ/ m^2)
Re	Reynolds number	(-)
T	temperature	(K)
V	velocity	(m/s)
U	overall heat transfer coefficient	(kW/K)
υ	specific volume	(m^3/kg)
w	specific work or width of solder joint	(kJ/kg) or (m)
x	vapour quality	(-)
z	distance from capillary tube inlet	(m)
θ	inclination angle	(deg)
ε	wall roughness	(mm)
δ	soldering joint thickness	(m)
ρ	density	(kg/m^3)
τ	shear stress	(N/m^2)
σ	surface tension	(N/m)
simul	simulation	

Subscripts	
c	capillary tube
cond	condenser
evap	evaporator
in	inlet
HX	heat exchanger region
j	soldering joint
l	liquid
o	outside
out	outlet adiabatic region
s	suction line
sp	single-phase
sub	sub-cooling
sup	superheating
tp	two-phase
v	vapour
w	wall

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