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NUCLEATE BOILING IN WATER FOR DIFFERENT PRESSURES

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ABSTRACT

In this paper, the most common calculation methods for nucleate boiling are presented and compared to experimental results. Especially for low pressures, the predictions of the heat transfer coefficients are far away from the experiments. The calculation method of Gorenflo (1984) turns out to be the most accurate one. New investigations are carried out for water and organic fluids in order to improve this method especially for low pressures. Therefore, an existing standard apparatus for pool boiling at ambient pressure is prepared to fit the new requirements for variable pressures.

The setup of the test apparatus is presented, as well as the first results of the measurements. These results show that further influencing parameters have to be taken into account for the boiling mechanisms, such as the surface tension of the fluid, the microstructure of the surface and its corresponding wettability.

1. INTRODUCTION

Heat transfer in pool boiling is mainly used in flooded evaporators as in heat pumps, refrigerating technique, air conditioning and within the process industry. The trend towards a better understanding of the fundamentals of bubble formation is supported by new developments in computer calculations and in measurement techniques. However, little progress has been made in the development of theoretically based predictive methods. A literature survey shows that numerous influencing parameters for the heat transfer have been identified. Anyhow, it is not clear how the certain influence parameters can be taken into account. The reason is that the existing experimental data is often incomplete, for example in regard to the properties of the heating element, namely roughness parameters and the wettability, which are rarely documented. Moreover, more thermophysical properties have to be considered, such as the surface tension and the slope of the vapor pressure curve of the fluid. This paper is taking care of the two last mentioned parameters.

2. STATE OF THE ART – EMPIRICAL CORRELATIONS

There are different calculation methods for the prediction of the heat transfer in nucleate boiling. For practical applications, they are often adapted to special features, for example enhanced tubes.

In table 1, the most popular calculation methods for nucleate boiling are presented. Afterwards, the calculations are compared to numerous experimental results from the literature.

Nishikawa (1982) introduced a correlation that was determined by curve fitting of experimental results, see eq. (1). The reduced pressure $p^* = p/p_c$, the surface roughness $R_{p,old}$ of the heating surface, the molar mass M and the critical properties p_c and T_c of the evaporating fluid as well as the heat flux q are identified as influencing parameters for the heat transfer. The surface roughness $R_{p,old}$ is calculated by an older standard and can be updated by equation (2).

The correlation of Cooper (1984) was also fitted to experimental results, see eq. (3). Cooper identifies the same influencing parameters for the heat flux as Nishikawa did, except for p_c and T_c .

Gorenflo (1984) undertakes a strict separation of the influencing parameters for the heat transfer. The different parameters (p^* , q , R_a , wall material and thermophysical properties of the fluid) are separated as factors, as shown in table 1.

In this correlation, water is supposed to be a special fluid with its own characteristics in contrast to the organic fluids. However, a literature review shows that this assumption is based on only few experiments in the early 1970s, and especially for low pressures, there is a great deviation within these experimental results. Consequently, new comparative experiments must be carried out for water and other (organic) fluids to either verify or disprove the assumptions.

Table 1: Calculation methods for the heat transfer coefficients:

Author	Correlation for heat transfer coefficient α
Nishikawa (1982)	$\alpha = 31,4 \frac{p_c^{0,2}}{M^{0,1} \cdot T_c^{0,9}} \cdot (8 \cdot R_{p,old})^{\frac{1-p^*}{5}} \cdot \frac{(p^*)^{0,23}}{(1-0,99 \cdot p^*)^{0,9}} \cdot q^{0,8} \quad (1)$ <p>The surface roughness $R_{p,old}$ is calculated by an older standard and can be updated by</p> $R_{p,old} = \frac{R_a}{0,4} \quad (2)$
Cooper (1984)	$\alpha = C \cdot q^{0,67} \cdot p^{*(0,12-0,2 \cdot \lg[R_{p,old}])} \cdot (-\lg p^*)^{-0,55} \cdot M^{-0,5} \quad (3)$ <p>with $C = 95$ (copper) or $C = 55$ (stainless-steel)</p>
Gorenflo (1984)	$\alpha = \alpha_0 \cdot F(p^*) \cdot F\left(\frac{q}{q_0}\right) \cdot F_{WR} \cdot F_{WM} \quad (4)$ <p>with</p> $F(p^*) = 1,2 \cdot (p^*)^{0,27} + \left(2,5 + \frac{1}{1-p^*}\right) \cdot p^* \quad (5) \quad \text{for organic fluids}$ <p>or</p> $F(p^*) = 1,73 \cdot (p^*)^{0,27} + \left(6,1 + \frac{0,68}{1-(p^*)^2}\right) \cdot (p^*)^2 \quad (6) \quad \text{for water}$ <p>and</p> $F\left(\frac{q}{q_0}\right) = \left(\frac{q}{q_0}\right)^{n(p^*)} \quad (7)$ <p>with</p> $n(p^*) = 0,9 - 0,3 \cdot (p^*)^{0,3} \quad (8) \quad \text{for organic fluids or}$ $n(p^*) = 0,9 - 0,3 \cdot (p^*)^{0,15} \quad (9) \quad \text{for water}$ <p>and</p> $F_{WR} = \left(\frac{R_a}{R_{a0}}\right)^{2/15} \quad (10) \quad \text{and} \quad F_{WM} = \left(\frac{\lambda \cdot \rho \cdot c}{\lambda_o \cdot \rho_o \cdot c_o}\right)^{0,25} \quad (11)$ <p>α_0 is a reference value at standardized conditions, that means pool boiling on a copper surface with $R_{a0} = 0,4 \mu\text{m}$ at $p_0^* = 0,1$, $q_0 = 20000 \text{ W/m}^2$. It can be considered as a factor for the thermophysical properties of the fluid.</p>
Author	Correlation for α_0
Gorenflo, Kotthoff (2005)	$\alpha_0 = \left(3,6 \cdot \frac{\left(\frac{dp}{dT}\right)_{VPC}}{\sigma} \right)_{p^*=0,1}^{0,6} \quad (\alpha_0 \text{ in kW/m}^2; dp/dT \text{ in bar/K}) \quad (13)$

3. TEST APPARATUS

As shown in the correlations in chapter 2, many different parameters influence the heat transfer coefficient, such as the reduced pressure p^* , the heat flux q , the roughness and material of the heating surface and the thermophysical properties of the fluid. The heat transfer coefficient itself is defined as the quotient of the heat flux and the wall superheat

$$\alpha = \frac{q}{\Delta T} = \frac{q}{T_W - T_S} \quad (14)$$

Consequently, all the above parameters have to be determined before or during the experiments. The boiling experiments are carried out in a recently installed test rig. It is based on the standard boiling apparatus as suggested by Gorenflo and Goetz (1982) and has been completely rebuilt in order to fit the new requirements of the oncoming experiments. Those can now be carried out for pressure ranges between 0 and 50 bar and temperatures from -50°C to $+200^\circ\text{C}$. The general design of the test rig is presented in Luke (2008) (conference paper #2400).

The whole circulation loop is placed inside a conditioned chamber wherein the temperature is adjusted to the saturation temperature T_S of the boiling fluid. The chamber temperature is regulated by a PID-controlled cartridge heater, which reheats a special cooling fluid (silicon oil) after the heat removal in a thermostat.

The centerpiece of the test facility is a natural circulation loop for the test fluid, including the evaporator and condenser. The emerging vapor flows to the condenser, which is located above the evaporator. Once the fluid is liquefied again, it flows through a downpipe and re-enters the evaporator at its bottom. The temperature of the condensate is measured at the entrance of the evaporator. In case that the condensate is subcooled, it can be heated up to saturation temperature by a preheater. The evaporator is made of stainless steel. It has got a volume of approximately 12 liters and a diameter of 300mm. Inside the vessel, the temperature distribution is measured by four resistance thermometers in the liquid and vapor phase. The saturation pressure is measured in the vapor phase by precisely calibrated pressure transmitters. The measurement signals are evaluated with a high precision voltmeter and converted by a PC.

The electrically heated test device is located in the middle of the evaporator so that the boiling mechanism can easily be visualized through the glass windows. This allows high speed video recordings of the bubble formation on the test tube. The high speed video recordings are done through additional windows within the walls of the climate chamber without disturbing the thermal equilibrium. The electrical power to the boiling surface (up to 3500W DC) is controlled by a manually operated voltage converter and measured by a power transducer.

The recent tests are carried out with a copper tube, which is designed as described in Luke (2008) (conference paper #2400). The superheat of the tube wall is directly measured by miniature thermocouples type K with one junction located in the saturated liquid below the test tube and the other junction in the test tube; about two millimeters below its surface. Each thermoelectric voltage is conditioned by its own amplifier with a gain of 1000 whereby the thermocouples are directly soldered with the amplifier in order to avoid distorting thermal contact resistances at this point. Afterwards, the amplified signals are sent to an analog-to-digital converter, where they are measured and computed and finally forwarded to the PC.

Due to this setup, the superheat has to be recalculated by taking into account the temperature difference between the thermocouple and the surface of the tube.

The thermal heat leaving the tube has to be calculated by the overall electrical power consumption of the heating element and the energy losses between the power supply and the heating element.

Steady state conditions are obtained by adjusting the heat removed in the condenser to the power input of the heating surface and by regulating the air temperature in the climate chamber to T_S . The condenser is cooled down by a second coolant circuit.

4. FIRST RESULTS

The comparison of experimental data for boiling water to the above calculation methods for water boiling on copper surfaces with $R_a = 0,4\mu\text{m}$ is shown in figure 1.

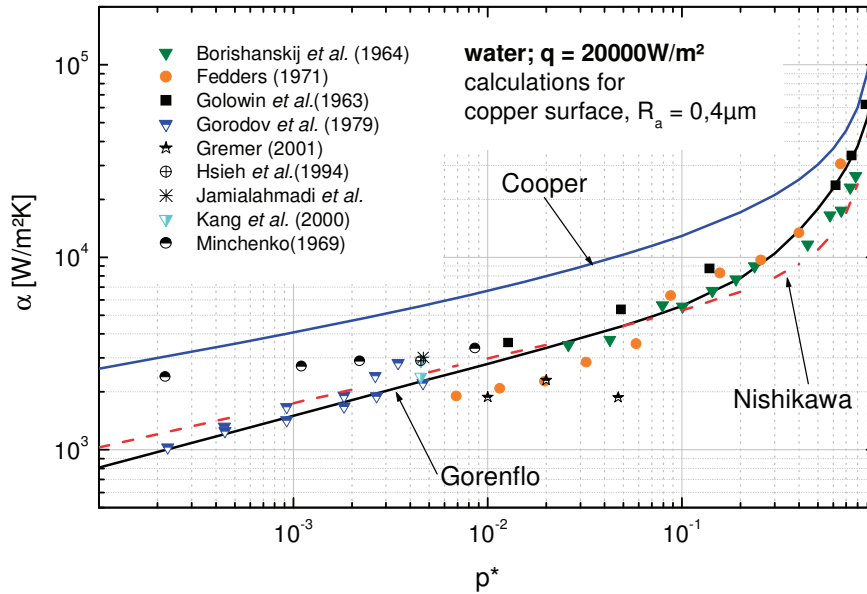


Figure 1: Comparison of experimental data for water calculated with the methods of Nishikawa (1982), Gorenflo (1984) and Cooper (1984).

The Gorenflo equations predict a stronger rise of the heat transfer coefficient than Nishikawa. Both methods seem to be more accurate for water than the correlations of Cooper. Especially for intermediate and high reduced pressures, the Gorenflo correlation shows a good agreement with the experimental results. For low pressures there is a great deviation within the experimental results.

Cooper overestimates the heat transfer coefficient within the whole pressure range. This may lead to significant problems during the design of the heat exchangers: The overestimation for low pressures results into an undersized design of the heat exchanger and consequently, the performance will be unsatisfactory.

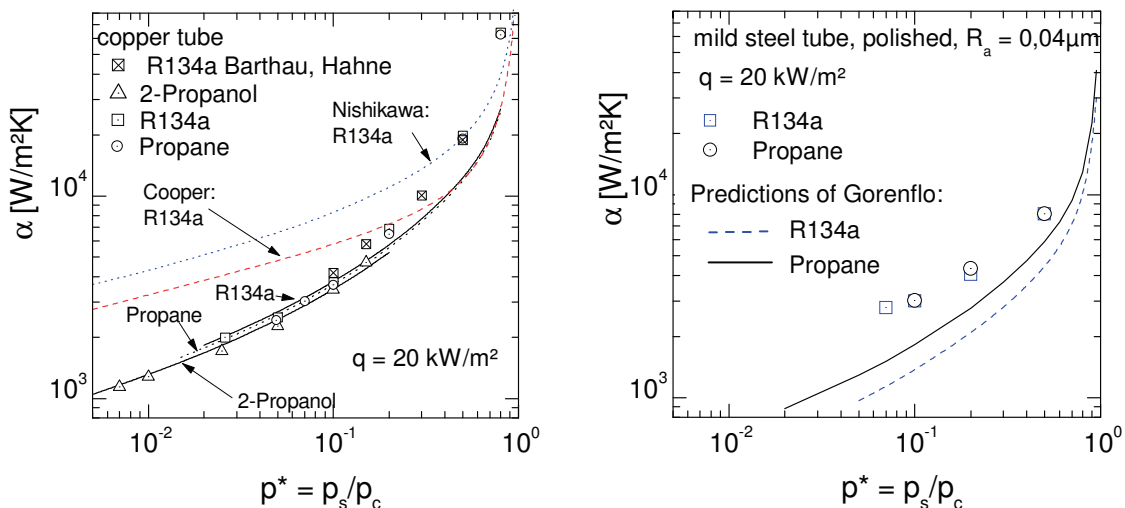


Figure 2a and 2b: Experimental data for 2-propanol, Propane and R134a boiling on a copper tube (2a) and mild steel tubes (2b) and the comparison to the above calculation methods.

In figure 2a the results of Luke (2003) for the heat transfer experiments are presented for the three fluids R134a, Propane and 2-Propanol boiling on a copper surface with $R_a=0,4\mu\text{m}$. The data agree well with the results obtained by Barthau and Hahne (2001) for R134a (squares with dash) boiling on a gold-plated, fine sandblasted copper tube. The heat transfer coefficients for the three substances are very similar if compared at the same normalized pressure p^* . As we will see later, this is supposed to happen because of the similar surface tensions of the fluids at these pressures, see figure 3.

In addition to the experiments, the predictions of Gorenflo, Nishikawa and Cooper are shown for R134a. For the benefit of clarity, only the more accurate Gorenflo predictions are presented for Propane and 2-Propanol.

In figure 2b, the latest measurements for Propane and R134a boiling on a mild steel tube are presented. The significant difference of the heat transfer coefficients between the copper surface and the mild steel surface is caused by the different materials of the heating elements and the resulting differences in wettability and surface energy. However, the interactions between the thermophysical properties of the fluid and the heating surface has not been investigated in detail so far. The exceptional position of water that is stated by Gorenflo could be a result of this fact, because water has a very high surface tension compared to all other fluids (see figure 3) and consequently, the interaction between fluid and surface is different, too. For the same reduced pressures, the surface tensions of Propane, R134a and 2-Propanol are very similar to each other (as well as the heat transfer coefficients, as already shown in figure 2). They are represented by the dashed line in figure 3. On the other hand, the surface tension of water differs significantly and reaches values that are three times higher than those of the other fluids.

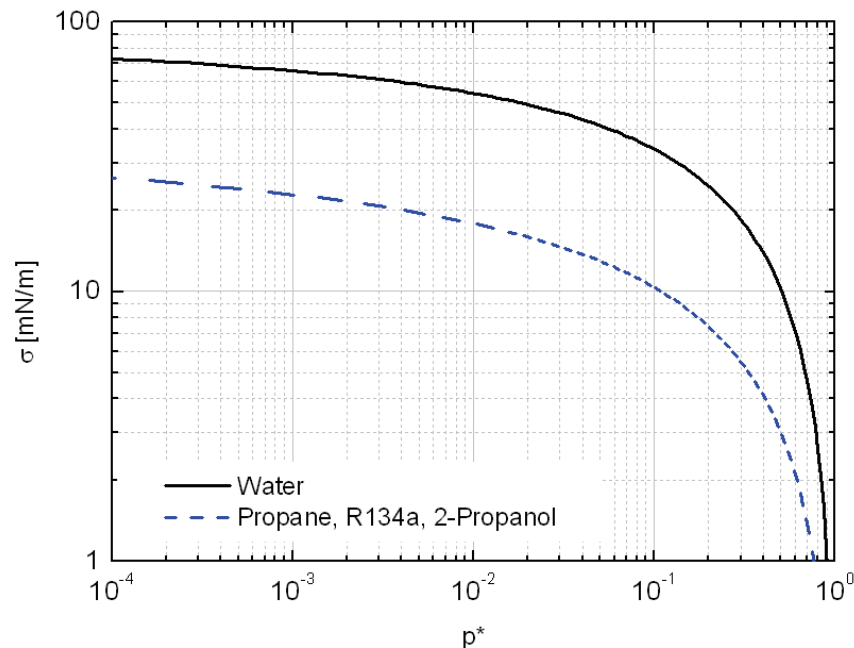


Figure 3: Comparison of the surface tension of water, propane, R134a and 2-propanol at the respective saturation temperature

It has already been shown in various earlier publications, that the slope of the vapour pressure curve (VPC) as well as the surface tension σ are the two most important properties of the fluid for nucleate pool boiling heat transfer, see for example Bier (1977), Gorenflo (2004), Kotthoff (2008). However, this influence has not been considered in the calculation methods so far. Gorenflo (2004) is the only one who includes this parameter in a new correlation for the calculation of α_0 , see (13).

In order to clarify the influence of the surface tension, water is chosen as test fluid due to its high surface tension. First tests show that the surface tension of water can be approximately halved with the help of surfactants, such as Triton X 100 or Lutensol, see figure 4.

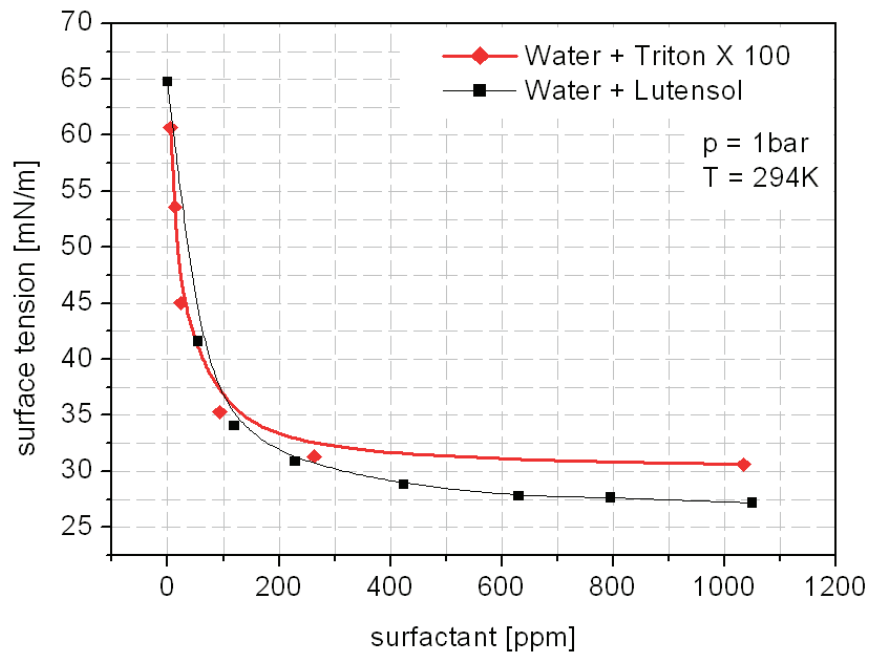


Figure 4: Variation of the surface tension of water with decreasing fraction of two surfactants

Based on the assumption that the surface tension is halved while the slope of the vapor pressure curve remains stable due to the small amount of the surfactant, there will be a remarkable increase of the heat transfer coefficient. According to Gorenflo, α_0 of water could approximately increase by the factor of 1,5 with the help of surfactants.

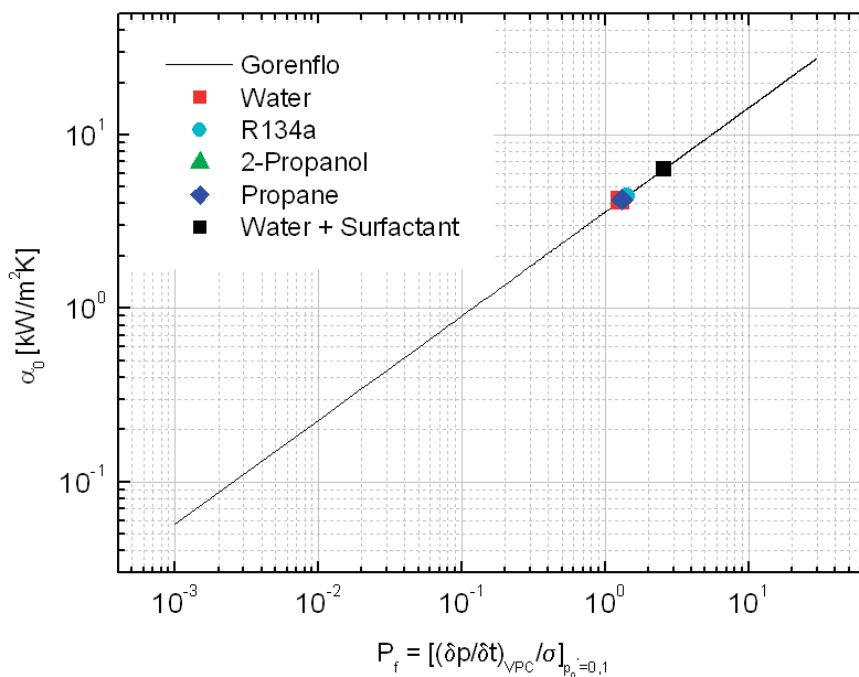


Figure 5: Variation of the heat transfer coefficient α_0 with the variation of the surface tension σ

Consequently, a general increase of the heat transfer coefficient α of about 50% would be possible, see figure 6. The main reason is the increasing amount of nucleation sites caused by the lower surface tension.

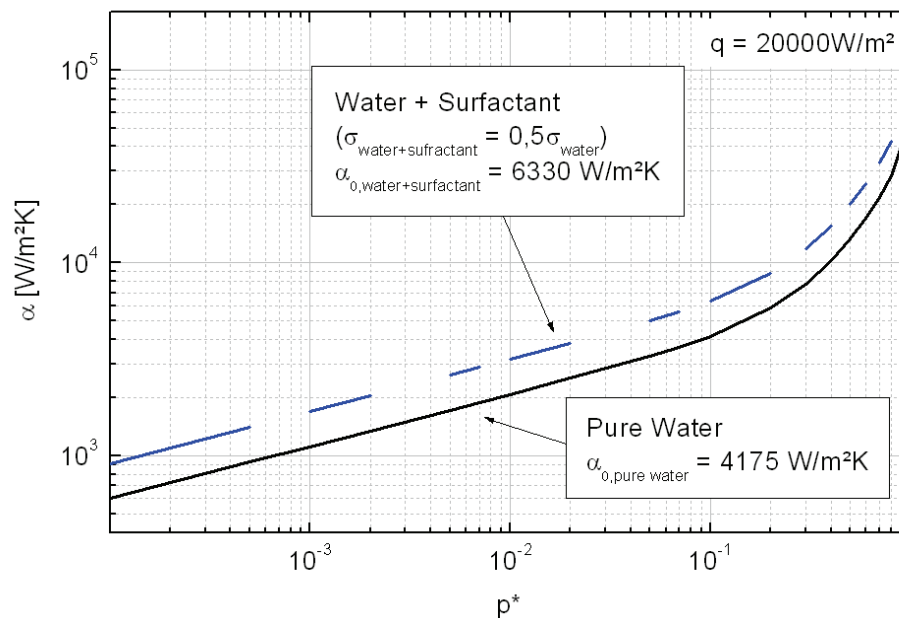


Figure 6: Possible increase of the heat transfer coefficient α by halving the surface tension σ . The predictions are calculated according to Gorenflo (1984) and Kotthoff/Gorenflo (2005)

5. CONCLUSION

The above investigations lead to the following conclusions:

- The surface tension of the fluid seems to be a key influence parameter for the heat transfer in boiling. For the same reduced saturation pressures p^* , the heat transfer coefficients are the same for Propane, R134a and 2-Propanol. These fluids have very similar surface tensions at the same reduced pressures. Do not expect to get an extension for submitting your paper.
- More experiments have to be carried out for water and water with surfactants in order to verify the results in chapter 4.
- More investigations have to be carried out in regard to the influence of the interactions between the fluid and the heating surface on the heat transfer; such as the wettability of the surface. In regard to the influence of the thermophysical properties of the fluid, the influence of the surface tension and the slope of the vapor pressure curve have to be taken into account.

NOMENCLATURE

A	area	(m ²)	q	heat flux	(W/m ²)
c	heat capacity	(J/kgK)	Q	heat energy	(W)
F(x)	function of x	(-)	T	temperature	(K)
M	molar mass	(kg/kmol)	T _c	critical temperature	(K)
n	exponent	(-)	α	heat transfer coefficient	(W/m ² K)
p	pressure	(N/m ²)	λ	heat conductance	(W/mK)
p _c	critical pressure	(N/m ²)	ρ	density	(kg/m ³)
p _s	saturation pressure	(N/m ²)	σ	surface tension	(N/m)
p* = p _s /p _c	reduced pressure	(-)			

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