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Experimental and theoretical analysis of condensation over nanoengineered surfaces

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Abstract

In this thesis a study of condensation over nanoengineered surfaces is presented. Firstly, a state of the art of the condensation process is introduced, starting from the classical filmwise condensation to the dropwise condensation. Several techniques are developed to change the wettability of an aluminum specimen, making it respectively superhydrophilic, hydrophobic and superhydrophobic. The wetting properties of the surfaces are changed by means of chemical treatments and a discussion about their influence on condensation mode will be undertaken. Surfaces are tested in the experimental apparatus assembled for the investigation of the condensation process over vertical surfaces existing at the Two-Phase Heat Transfer Laboratory of the Department of Industrial Engineering of Padova University. During tests it has been achieved filmwise condensation mode as well as dropwise condensation mode, although for a limit period of time. Experimental data are presented and compared to those achieved over an untreated specimen. It has been found that the hydrophobic specimen promotes slip-driven filmwise condensation, and several considerations are developed to explain this singularity. Furthermore, thanks to experimental data and literature research, it was possible to formulate theoretical analysis of the slip-driven process over hydrophobic-treated surfaces, which is not covered by the classical theory for vertical filmwise condensation.

Chapter 1

Condensation

Condensation is a thermodynamic process that converts vapor phase to liquid phase of a substance and it is one of the most important heat transfer operations. Condensation is widely used in industrial processes, for example heat and energy production, chemical industries, air conditioning and so on, therefore it is a very important issue to be investigated and this is why many researches are focused on it. There are different ways in which condensation could occur, as long as the vapor phase meet a surface with a temperature that is below the dew point of the substance. These ways are:

- Filmwise condensation (FWC). The condensed liquid forms a continuous liquid film on the heat transfer surface. This type of condensation is the most common one, because is the easiest and most natural among all.
- Dropwise condensation (DWC). The condensed liquid forms drops on the heat transfer surface mainly because of its low wettability. Normally, DWC leads to much higher heat transfer coefficients (HTC) than FWC, thus many investigations on its promotion have been undertaken.
- Homogeneous condensation. Condensation could happen also in the vapor bulk for either an increase in pressure or a decrease in temperature. A well-known case of bulk vapor condensation is the one occurring in the low stages of steam turbines where condensate droplets can arise and can lead to blade erosion problems.

The aim of this chapter is to investigate and to model the FWC and the DWC on a vertical surface. The correspondence between theory and experimental data, gained by our experimental apparatus, will be discussed in chapter 5.

1.1 – Filmwise condensation

During FWC on vertical surfaces the condensing steam forms a continuous film of liquid over the surface where heat exchange occurs. In this mechanism of condensation there are various thermal resistances, among these, however, is worth considering only the resistance given by the condensate layer of surface liquid, being this much greater than the other. If the film of liquid moves in a laminar state, heat is transferred by conduction. Therefore, a thin condensate film presents less thermal resistance and thus higher heat transfer coefficient compared to a thicker film. The vapor can condense on a vertical surface in static or dynamic conditions. In the first case, the liquid film is influenced only by the force of gravity, in the second case even by the tangential forces due to the steam velocity. Other factors that affect the HTC, besides the speed of the steam, the thickness of the film and to the turbulence of the motion, are the presence of waves on the film, the drag and the deposition of the droplets, the presence of splashing of condensate and the degree of subcooling of the liquid.

1.1.1 – Gravity controlled condensation

We present the theory of Nusselt (1916) for the calculation of the heat transfer coefficient $\bar{\alpha}_{grav}$ during laminar film condensation in stagnant condition on vertical surfaces. This theory is valid under following hypothesis:

- the condensate forms a continuous film that flows under the effect of gravity;
- the steam has zero velocity;
- the fluid is in laminar regime;
- pure fluid at constant pressure;
- the temperature difference through the film of liquid is constant;
- the fluid is a Newtonian one;
- the properties of the film are constant.

The treated situation is illustrated in Figure 1.1, which points out how the condensate film thickness δ increases along the plate.



Fig. 1.1 – Gravity controlled condensation on a vertical surface

At a distance z, we may write a force balance for a unit width of condensate film between the shear force τ and the gravitational force on the element highlighted in Fig. 1.1:

$$\tau dz = (\rho_l - \rho_v)g(\delta - y)dz$$
(1.1)

where ρ_l is the liquid density, ρ_v is the vapor density, g is the gravity acceleration and y is the coordinate in the direction of the liquid. The shear force is related to the velocity gradient within the liquid film

$$\tau = \mu_l \frac{\mathrm{d}\mathbf{u}}{\mathrm{d}\mathbf{y}} \tag{1.2}$$

where μ_l is the liquid viscosity. Combing Eq. (1.1) and Eq. (1.2) it is obtained

$$\frac{du}{dy} = \frac{(\rho_l - \rho_v)g}{\mu_l} (\delta - y) \tag{1.3}$$

which may be integrated to give

$$u = \frac{(\rho_l - \rho_v)g}{\mu_l} (\delta y - \frac{y^2}{2}) + C$$
(1.4)

since u=0 at y=0, it is C=0. The mean velocity in the liquid film \bar{u} is given by

$$\overline{u} = \frac{1}{\delta} \int_0^\delta u dy = \frac{(\rho_l - \rho_v)g}{\mu_l \delta} \int_0^\delta \left(\frac{\delta y - y^2}{2}\right) dy = \frac{(\rho_l - \rho_v)g\delta^2}{3\mu_l}$$
(1.5)

defining Γ as the mass flow in the film per unit width, then it follows that

$$\Gamma = \rho_l \overline{u} \delta = \frac{\rho_l (\rho_l - \rho_v) g \delta^3}{3\mu_l}$$
(1.6)

and the local liquid film thickness is thus given by

$$\delta = \left[\frac{3\mu_l\Gamma}{\rho_l(\rho_l - \rho_v)g}\right]^{1/3} \tag{1.7}$$

 Γ increases down the surface as a result of the condensation process. If the local condensation rate per unit surface area of the liquid film is \dot{m} , then we can write

$$\frac{d\Gamma}{dz} = \dot{m} = \frac{\dot{q}}{r'} = \frac{\rho_l(\rho_l - \rho_v)g\delta^2}{\mu_l}\frac{d\delta}{dz}$$
(1.8)

where \dot{q} is the heat flux. The parameter r' is the heat transferred to the surface and includes both latent and sensible heat transferred in the liquid phase per unit width between T_{SAT} , the temperature of saturation, and T_{WALL} , the temperature of the wall. If the surface is maintained at a constant temperature T_{WALL} and the film is in laminar flow, then \dot{q} is calculated by considering the conduction process through the film, i.e.

$$\dot{q} = \frac{(T_{SAT} - T_{WALL})\lambda_l}{\delta} \tag{1.9}$$

where λ_l is the liquid thermal conductivity. Combining Eq. (1.8) and Eq. (1.9) we obtain

$$\delta^3 \frac{d\delta}{dz} = \frac{\mu_l \lambda_l (T_{SAT} - T_{WALL})}{\rho_l (\rho_l - \rho_\nu) gr'} \tag{1.10}$$

and integrating with the boundary condition that $\delta=0$ at z=0, we have

$$\delta^4 = \frac{4\mu_l \lambda_l (T_{SAT} - T_{WALL})z}{\rho_l (\rho_l - \rho_v) gr'} \tag{1.11}$$

The local liquid film heat transfer coefficient α_{Nu} is defined as

$$\alpha_{Nu} = \frac{\dot{q}}{T_{SAT} - T_{WALL}} \tag{1.12}$$

and from foregoing equations we obtain

$$\alpha_{Nu} = \frac{\lambda_l}{\delta} = \left[\frac{\lambda_l^3 \rho_l (\rho_l - \rho_v) gr'}{4\mu_l (T_{SAT} - T_{WALL})z}\right]^{1/4}$$
(1.13)

It's often convenient to define a mean heat transfer coefficient $\bar{\alpha}_{Nu}$ over the plate length L

$$\bar{\alpha}_{Nu} = \frac{1}{L} \int_0^L \alpha_{Nu} dz = 0.943 \left[\frac{\lambda_l^3 \rho_l(\rho_l - \rho_v) gr'}{\mu_l(T_{SAT} - T_{WALL})L} \right]^{1/4}$$
(1.14)

By considering the liquid flow rate Γ_L at a specific coordinate L from the top

$$\Gamma = \frac{\overline{\alpha}_{Nu}(T_{SAT} - T_{WALL})L}{r'} \tag{1.15}$$

 $\bar{\alpha}_{Nu}$ becomes

$$\overline{\alpha}_{Nu} = 0,925 \left[\frac{\lambda_l^3 \rho_l (\rho_l - \rho_v) g}{\mu_l \Gamma} \right]^{1/3}$$
(1.16)

It is also convenient to rewrite $\bar{\alpha}_{Nu}$ in terms of Reynolds number Re

$$\operatorname{Re} = \frac{D_h \Gamma}{\mu_l \delta} = \frac{4\Gamma}{\mu_l} \tag{1.17}$$

where D_H is the hydraulic diameter

$$D_h = \frac{4P\delta}{P} = 4\delta \tag{1.18}$$

thus it is

$$\bar{\alpha}_{Nu} = 1,47 \left[\frac{\lambda_l^3 \rho_l (\rho_l - \rho_v) g}{\mu_l^2} \right]^{1/3} \text{Re}^{-1/3}$$
(1.19)

The Nusselt theory is a simplified theory of condensation, thus for comparison with experimental processes, therefore a number of additional effects should be considered:

- Subcooling. Since there is a temperature gradient in the film of liquid, the liquid near the wall is subcooled and the mean temperature T_L of liquid is lower than the saturation one.
- Inertia. Vapor condensing over the liquid film should not be assumed to be static, because it has to accelerate to the film velocity leading to inertia effects, as studied by Chen (1961).

- Vapor superheat. If vapor is superheated it has to be cooled from his bulk temperature to the interface saturation temperature, adding to the process a thermal resistance.
- Interfacial waves. Falling liquid film forms waves that grow along the surface. Some experiments by Van de Walt and Kroger (1972) showed that experimental heat fluxes were 5 to 10% greater, because of waves presence, than those predicted from the Nusselt theory.

The profile temperature accounting for all previous effects is shown in Figure 1.2.



Fig 1.2 – Condensation of a superheated vapor

The heat transfer coefficient in gravity controlled condensation is thus calculated

$$\bar{\alpha}_{grav} = 1.15 \left[0.0206 \left(\frac{r' \mu_l}{\lambda_l (T_{SAT} - T_{WALL})} \right)^{0.5} + 0.79 \right] \bar{\alpha}_{Nu}$$
(1.20)

The multiplicative factor 1.15 was introduced by Baehr and Stephan (2004) to consider the enhancement of $\bar{\alpha}_{grav}$ due to film waves. The term between square parenthesis is used to consider the inertia effect and was added by Depew and Reisbig (1964). The liquid subcooling is finally accounted by calculating the thermodynamic proprieties of the condensate at the mean temperature

$$T_l = 0.75T_{WALL} + 0.25T_{SAT} \tag{1.21}$$

meanwhile only vapor proprieties are calculated at T_{SAT} (Rose, 1999).

1.1.2 – Shear controlled condensation

When there are high vapor velocities, the heat transfer coefficient α_{ss} is controlled by the shear of the vapor, indeed the interfacial shear becomes significant compared to the gravitational force on the liquid phase. The first aspect for the calculation of α_{ss} is the regime of motion of the liquid film. Laminar flow of the condensate is the most common situation. If the speed of vapor is high enough the effect of gravity on the two-phase process could be ignored, leading to the shear controlled condensation mode. A scheme of the process is illustrated in Figure 1.3.



Fig. 1.3 – Shear forces on a liquid film in laminar regime

The film thickness is δ , the interfacial shear stress is τ_i and the wall shear stress is τ_0 . If the interfacial shear stress dominates, then $\tau_i \approx \tau_0$ and we can write

$$\mu_l \frac{du}{dy} = \tau_0 = \tau_i \tag{1.22}$$

Where *u* is the liquid velocity within the film at a distance *y* from the wall. Integrating with the condition u=0 at y=0, we have

$$u = \frac{\tau_0 y}{\mu_l} \tag{1.23}$$

thus, the mean velocity in the liquid film is given by

$$\overline{u} = \frac{1}{\delta} \int_0^\delta u dy = \frac{\tau_0 \delta}{2\mu_l}$$
(1.24)

The mass flow per unit periphery Γ and the liquid film thickness δ are given by

$$\Gamma = \rho_l \overline{u} \delta = \frac{\rho_l \tau_0 \delta^2}{2\mu_l} \tag{1.25}$$

$$\delta = \sqrt{\frac{2\Gamma\mu_l}{\tau_0\rho_l}} \tag{1.26}$$

and the local liquid film heat transfer coefficient, deduced from Eq. 1.12 is given by

$$\alpha_{SS} = \frac{\lambda_l}{\delta} = \sqrt{\frac{\lambda_l^2 \tau_0 \rho_l}{2\Gamma \mu_l}}$$
(1.27)

A well detailed analysis of this phenomenon was formulated by Carpenter and Colburn (1951). If the vapor density is small and the condensation rate is low, then the shear stress is given by

$$\tau_0 = -\frac{D_h}{4} \left(\frac{dp}{dz}\right)_F \tag{1.28}$$

where $\left(\frac{dp}{dz}\right)_{F}$ is the gradient of pressure along the channel, which can be evaluated with the model by Friedel (1979).

The Friedel model uses a two-phase multiplier to correlate the two-phase pressure gradient with the one which would occur during single-phase liquid flow

$$\Phi_{\rm LO}^2 = \frac{\left(\frac{\rm dp}{\rm dz}\right)_{\rm f}}{\left(\frac{\rm dp}{\rm dz}\right)_{\rm LO}} \tag{1.29}$$

with the liquid-only pressure gradient defined as

$$\left(\frac{dp}{dz}\right)_{LO} = \frac{2f_{LO}G^2}{D_h\rho_l}$$
(1.30)

where G is the moisture specific mass flow rate. The two-phase multiplier is calculated as

$$\Phi_{\rm L0}^2 = E + \frac{3.24 \,\mathrm{F}\,\mathrm{H}}{\mathrm{Fr}^{0.045} \mathrm{We}^{0.035}} \tag{1.31}$$

where

$$E = (1 - x)^{2} + x^{2} \frac{\rho_{I} f_{VO}}{\rho_{v} f_{LO}}$$
(1.32)

$$F = x^{0.78} (1 - x)^{0.224}$$
(1.33)

$$H = \left(\frac{\rho_l}{\rho_v}\right)^{0.91} \left(\frac{\mu_v}{\mu_l}\right)^{0.19} \left(1 - \frac{\mu_v}{\mu_l}\right)^{0.7}$$
(1.34)

$$Fr = \frac{G^2}{gD_h\rho_H^2}$$
(1.35)

$$We = \frac{G^2 D_h}{\gamma \rho_H}$$
(1.36)

where γ is the superficial tension of the liquid and f_{LO} and f_{VO} liquid only and vapor only are friction coefficients calculated by Hagen-Poiseuille and Blasius (Churchill, 1977). The element ρ_H is the homogenous density defined by

$$\rho_{\rm H} = \left(\frac{x}{\rho_{\rm V}} + \frac{1-x}{\rho_{\rm L}}\right)^{-1} \tag{1.37}$$

1.1.3 – Condensation under combined gravity and shear control

For intermediate vapor velocities, gravity begins to have a significant effect on the shear stress phenomena. Thus, in downflow conditions, τ_i is lower than τ_0 as illustrated in Figure 1.4.



Fig. 1.4 – Shear stress profiles in downflow

Using the simplified approach of Butterworth (1983), the heat transfer coefficient in the presence of both gravity and shear stress forces is evaluated as

$$\bar{\alpha}_{th} = \sqrt{\alpha_{ss}^2 + \bar{\alpha}_{grav}^2} \tag{1.38}$$

The calculation of α_{th} concludes the theoretical model of filmwise condensation.

1.2 – Dropwise condensation

Since its discovery in 1930 by Schmidt *et al.*, dropwise condensation has attracted significant attention due to its superior heat transfer performance compared with filmwise condensation. It has been reported that dropwise condensation on a hydrophobic surface enhances heat transfer by an order of magnitude compared with filmwise one. This is due to from the fact that the absence of condensate film and the continuous surface renewal by falling large drops drastically reduce the global thermal resistance improving the heat transfer. Unlike filmwise condensation where the liquid condensate forms a continuous film over the surface, dropwise condensation can be described as a combination of random processes such as initial drop formation, drop growth, and departure of large drops. The necessary but not sufficient condition for promoting DWC is that a surface has to be hydrophobic or superhydrophobic. The fundamental parameters to describe the behavior of a liquid droplet over a surface are:

- θ_a , advancing contact angle. For a hydrophobic surface $\theta_a > 90^\circ$, while for a superhydrophobic one it is $\theta_a > 150^\circ$;
- θ_r , receding contact angle;
- $\Delta\theta = \theta_a \theta_r$, contact angle hysteresis. For a superhydrophobic surface it is $\Delta\theta < 10^\circ$.



Fig. 1.5 – Drop on a non-planar surface. θ_a and θ_r are shown.

These angles are measured experimentally with a technique that will be shown in a following chapter and, in literature, are evaluated at normal conditions ($T_{WALL}=25^{\circ}$ C, $P=P_{ATM}$). We know that a drop that is growing on a surface during DWC is not at $T=25^{\circ}$ C and we experimentally demonstrate that temperature changes significantly θ_a and θ_r .

In order to represent the phenomenon associated with this type of condensation, numerous researchers have attempted to establish mathematical models to be utilized for designing dropwise condensers. One of the earliest dropwise condensation models was proposed by Rose (1966), who combined a calculation for the heat transfer through a single drop with a calculation for the drop size distribution. Its single drop heat transfer model considered the conduction resistance, the vapor-liquid interfacial matter transfer, the promoter layer resistance, and the resistance due to the convex liquid surface. For the drop size distribution, it has been used the concept of the surface area fraction occupied by drops equal to or larger than the drop size of interest. Based on this model, advanced models have then been presented. Rose himself (2002) employed a population balance theory to evaluate the transient change of local drop size distribution by taking into account the two mechanisms of small drop's growth: direct condensation onto the drop and coalescence with neighboring drops. This enabled a more accurate prediction of drop size distribution for small drops, which was previously difficult to experimentally examine. However none of the previous models adequately represents a condensation occurring on a surface whose hydrophobic feature leads to a contact angle greater than 90°. They assumed that the shape of drops is fixed to be hemispherical, i.e., the contact angle is held constant at 90° and it makes the previous models improper for enhanced hydrophobic surfaces, which generate a water-contact angle greater than 90°. Also the assumption of a constant θ_a of the growing droplet is eliminated by new evidences. Furthermore, also the superficial morphology of the tested surface plays a crucial role, in DWC mechanisms are different from a flat to a hierarchical surface (see Fig. 1.6). The most evident difference is the state in which the droplet hangs on the surface:

- Wenzel state. Here the drop completely wets the surface, this involves a lower mobility of the droplet and thus a lower HTC. In a flat surface a drop anchor itself in this way. In Fig. 1.9 the wenzel W case e) is visually explained.
- Cassie-Baxter state. The droplet is suspended on the hierarchical surface, it could happen in particular cases, one of them is for the presence of non-condensable gases. See Fig. 1.9 the suspended S case c).
- Hybrid state. The droplet partially wets the surface. See Fig. 1.9 the partially wetted PW case d).



Fig. 1.6 – Scanning electron microscope of a)Copper-based nano-textured superhydrophobic surface (photo by Torresin et al. (2013)) and b)Aluminum flat hydrophobic surface.

The parameters previously introduced are accounted in the model by Wang *et al.* (2013), which will be used for theoretical analysis of the dropwise condensation process.

In the following pages the behavior of a droplet on a surface, the dynamics that insist on the condensing drop and the heat transfer model applied to a flat surface, that is the simplest case to study, will be presented. We will discuss, moreover, about the transition between DWC to FWC.

1.2.1 – Surface wettability

The study of how a liquid spreads over a solid is very important for the analysis of mobility of the droplet and as consequence to the determination of the heat transfer mode. Wetting is the ability of a liquid to stay in contact with a solid surface, resulting from intermolecular interactions when the two are brought together. The degree of wetting (wettability) is determined by a force balance between adhesive and cohesive forces: adhesive forces between a liquid and solid cause a liquid drop to spread across the surface, while cohesive forces within the liquid cause the drop to ball up avoiding contact with the surface. For the comprehension of this relationship three equations are fundamental. The first is

$$\gamma_{sg} = \gamma_{ls} + \gamma_{lg} \cos \theta_a \tag{1.39}$$

where γ_{sg} is the superficial tension between solid and gas, γ_{lg} is the superficial tension between liquid and gas and γ_{ls} is the superficial tension between solid and liquid. Eq. (1.39) is the Young equation and it links superficial tensions of the three phases, gas, liquid, and solid, with the contact angle. Here the contact angle is substitute by the advancing contact angle. The second one is:

$$w_a = \gamma_{sg} + \gamma_{lg} - \gamma_{ls} \tag{1.40}$$

that explains the specific adhesive energy (w_a) . Combining Eq. (1.39) with Eq. (1.40) we obtain

$$w_a = \gamma_{lg} (1 + \cos \theta_a) \tag{1.41}$$

In this way w_a is simple to calculate, in fact θ_a is experimentally determined and γ_{lg} is easily found in literature. The third equation is:

$$w_c = 2\gamma_{lg} \tag{1.42}$$

that shows the specific cohesive energy (w_c). While γ_{lg} is constant, the superficial energy of the solid (γ_{ls}) could be changed by properly treating the substrate, thus θ_a changes. The three typical configurations of a water drop (volume=1µL) sitting over an aluminum substrate are here reported:





Fig. 1.7 – Different configuration between a drop (Volume= $l \mu L$) and aluminum treated: a)superhydrophilic, b)hydrophobic and c)superhydrophobic. On the left side is presented the shape that assumes a drop on that specific surface with the dimensions reported on the axis, in m, of the graphs. On the right side is explicated the force balance between cohesive and adhesive forces and the resulting force (F_R) in μN .

The graphs in Fig. 1.7 are modeled with MATLAB software. The drop is modeled as a spherical cap. The Eqs. (1.41) and (1.42) are calculated as a force, therefore the free energies are multiplied by the wet perimeter, i.e. where the liquid wets the surface. The resulting force (F_R) is defined as the difference between adhesive and cohesive force, if it is positive $w_c > w_a$ and thus the drop does not wet the surface (hydrophobicity), otherwise $w_c < w_a$ and thus the drop wets the surface (hydrophilicity). Three cases showed in Fig. 1.7:

- a) $F_R < 0$. The liquid wet the surface, this is the case of etched aluminum that has high surface free energy. In the example $\theta_a = 20^\circ$.
- b) $F_R > 0$. The liquid does not completely wet the surface, this is a border line case, which happens when the aluminum surface undergoes a specific treatment and becomes hydrophobic. It has to be noticed that the drop is almost hemispheric and

this is the case of a treated flat polished surface. Here $\theta_a = 100^\circ$ and the drop has a low mobility.

c) $F_R >> 0$. Here the droplet don't wet the surface and has high mobility. In this case aluminum is treated to be superhydrophobic, presenting $\theta_a = 160^{\circ}$.

The three cases were not chosen randomly but with knowledge of the facts, since they will be investigated later on in this thesis. It is important to underline that the figure describe the behavior of a droplet on a planar surface, i.e. in a steady state.

1.2.2 – Vertical surface: droplet mobility

In the experimental section presented in this thesis condensation on a vertical surface is studied, thus the forces that interact with the droplet and therefore its behavior (droplet mobility). In this paragraph θ_r is introduced as a main factor, in fact it determines how much a drop is anchored on the surface, i.e. it determines its mobility. At a macroscopic level there are three principal forces on a droplet resting on a specimen (Fig. 1.8).



Fig. 1.8 – Droplet on a vertical surface subjected to different forces. a) Superhydrophobic surface. b) Hydrophobic surface.

A force balance was proposed by Wier and McCarthy (2006) without considering the drag force. In our experimental apparatus droplets are subjected also to a specific vapor mass flow rate thus it must be considered. In Fig. 1.8 two cases with the same θ_a (~150°) are shown:

a) Superhydrophobic surface. The hysteresis contact angle is, at least, less than 10°, so θ_r is about 140°.

b) Hydrophobic surface. θ_r is even less than 90°, which reflects in a low droplet mobility.

Since the droplet is moving the adhesive force (F_A) , hence similar to a friction force, is now defined as

$$F_A = k\gamma_{lg} |\cos\theta_r - \cos\theta_a| P_w \tag{1.43}$$

where P_w is the wet perimeter and k is a non-dimensional factor that describes the nonperfect circular shape of the droplet perimeter, thus is always bigger than 1 moreover, for a hydrophobic surface it is higher than the one of a superhydrophobic surface, since the same volume is distributed on a major area (see Fig.1.7). The adhesive force is applied in the center line at the interface between the solid and the liquid. The gravity force (F_G) is determined as

$$F_G = \rho_l V g \tag{1.44}$$

where V is the drop volume. This force is, on the other hand, applied on the barycenter of the spherical cap. The drag force (F_D), caused by the vapor velocity, is define as

$$F_D = \frac{1}{2} C_d \rho_v v_v^2 A_c \tag{1.45}$$

where A_c is the lateral area of the droplet, v_v is the vapor velocity and C_d is the drag coefficient

$$C_d = \frac{10}{\sqrt{Re_v}} \tag{1.46}$$

 Re_v is the Reynolds number of the vapor. Measuring θ_a and θ_r experimentally it is possible to calculate for which radius (*r*) the droplet departs from the surface, by imposing the following force balance

$$F_R = F_A - F_G - F_D \tag{1.47}$$

As expected, for a hydrophobic surface r is higher than the one of a superhydrophobic surface, at the same system conditions Rose (2002) proposed an empirical formula to determine the departing radius

$$r = K_3 \left(\frac{\gamma_{lg}}{\rho_{lg}}\right)^{0.5} \tag{1.48}$$

 K_3 is a factor depending on the superficial characteristics that yields the hydrophobicity to the surface. This factor is proposed to be equal to 0.3 for hydrophobic surface, instead for superhydrophobic surface is about 1.

1.2.3 – Mechanisms during DWC

DWC presents a complicated, still not well understood, mechanism of formation, growth, coalescence and slipping off droplets which is highly dependent on the surface characteristics. These mechanisms are fundamental for modelling the heat transfer and have been studied, both experimentally and theoretically, by several authors: Rose (1966, 1998, 2002), Tanaka (1975), Wu and Maa (1976), Kim and Kim (2011), Ma *et al.* (2012) and Wang *et al.* (2013). The droplet formation has been suggested by many authors to be a nucleation phenomenon related to the number of nucleation sites on the surfaces (N_S), that are around $10^8 - 10^{12} \text{ N}_S/\text{cm}^2$. We speak about a droplet since it has a thermodynamically radius viable and it is shown by

$$r_{min} = \frac{2T_{SAT}\gamma_{lg}}{r'\rho_l(T_{SAT} - T_{SUP})} \tag{1.49}$$

After the formation of a droplet, it starts to growth with direct impingement of the vapor molecules on the liquid-vapor interface, this happens because the temperature of the droplet is lower than T_{SAT} . Droplet continues growing and it is still anchored to the surface, with a different behavior depending on the surface properties. Wetting modes are summarized in Fig. 1.9. The following formula is proposed by Wang *et al.* (2013) to estimate the droplet wetting mode

$$E = \frac{-1}{f_r * \cos \theta_a} \tag{1.50}$$

where f_r is the roughness factor and it is defined by

$$f_r = 1 + \frac{\pi dh}{l^2} \tag{1.51}$$

When E>1 the contact line can overcomes the energy barrier to depin and a W droplet is formed (Fig. 1.9(e)). If E<1 complete depinning is not possible and the droplet grows upward over the top of the pillar array forming a PW Cassie droplet (Figs. 1.9(c)) and 1.9(d)).



Fig. 1.9 – Schematics of the modeled structured surface. (a) side view and (b) top view of the structure.
Dimensions h, d, and l represents the pillar height, diameter and center-to-center spacing, respectively.
Schematics shows the (c) S, (d) PW, and (e) W morphologies.(Wang et al. 2013)

Eqs. (1.49) and (1.50) show that the energy surface, thus θ_a , and the structure of the surface, thus f_r , are both fundamental to determine the wetting morphology and it points out that a surface has to be well built to optimize, and to promote dropwise condensation mode. These conclusions are also reported on the work of Liu et al. (2010). Moreover, droplet coalescence introduces a further length scale dependency on the emergent droplet wetting morphology. Wang et al. (2013) considers that when the average condensing droplet spacing (l_c) approaches the pillar spacing l, the formation of local pinning barriers is disrupted. While it may be energetically favorable to form Cassie droplets (E<1), this may not be possible due to droplet coalescence between two adjacent unit cells which bypasses the pinning barriers associated with the pillar sides. This effect results in the formation of thermodynamically unfavorable W droplets which are highly pinned to the surface. To avoid formation of W droplets on Cassie-Baxter stable surfaces, a critical droplet separation distance of at least 2l must be maintained. When a droplet reaches the critical radius r starts to fall down, it is remembered that we are referencing to DWC on a vertical surface. Fig. 1.10 shows environmental scanning electron microscope (SEM) images of the DWC process.



Fig. 1.10 – *SEM images of dropwise condensation at different times for a* $Cu(OH)_2$ *nanostructured surface. The coalescence of drops on the superhydrophobic surface, shown at 60 s (a) and 336 s (c), causes the drops to depart from the surface, as seen in the vacant regions of figures (b) at 64 s and (d) at 342 s. (Dietz et al. 2010)*

Falling drops fall very quickly after starting to move. By collecting other drops in their path, the moving droplets grow and thus sweep diverging tracks, causing lower regions on the surface to be swept more frequently. This ensures that only drops near the top of the surface can reach the size at which gravity and drag exceed the net surface tension restraining force.

Another factor that influence the condensation phenomena is the presence of noncondensable gases (NCG) in the experimental section. A wide research on this theme has been carried out by Ma *et al.* (2012). In their experimental section they tested two different treatment on copper, i.e. one to make it superhydrophobic and the other to make it hydrophobic, over which they condensed steam with and without NCG. The behavior of the condensate droplets during DWC on superhydrophobic surface with and without NCG is shown in Fig. 1.11, W_{NCG} is the mole fraction of NCG in the steam-air mixture.



Fig. 1.11 – DWC with and without NCG. (a) $W_{NCG}=0$, $q=149 \text{ kW/m}^2$, $T_{WALL}=96.3 \text{ °C}$; (b) $W_{NCG}=35\%$, $q=9.70 \text{ kW/m}^2$, $T_{WALL}=83.5 \text{ °C}$; (c) $W_{NCG}=50\%$, $q=7.93 \text{ kW/m}^2$, $T_{WALL}=76.8 \text{ °C}$; (d) $W_{NCG}=75\%$, q=4.61 kW/m^2 , $T_{WALL}=60.2 \text{ °C}$; (e) $W_{NCG}=90\%$, $q=1.44 \text{ kW/m}^2$, $T_{WALL}=40.6 \text{ °C}$ ($\Delta T=5 \text{ K}$; P=110-112 kPa). (Ma et al. 2012)

It can be seen that the contact line of the condensate droplet is not circular for the pure steam DWC, and the coverage area of condensate droplets appears darker, indicating that the condensate droplets stand in the Wenzel mode. On the contrary, the condensate droplets look much more circular and brighter for DWC of steam-air mixture, denoting the Cassie-Baxter mode. Comparing the images shown in Fig 1.11 it is possible to see that when the NCG concentration increases, there is a transition from the Wenzel to Cassie-Baxter wetting mode. During steady-state condensation of steam in the presence of NCG the liquid-solid contact area looks much more uniform. The droplets show contrasted light and shadow in a mixed Wenzel-Cassie- Baxter wetting mode. It is mainly due to the transition from the condensate sinkage mode to a mixed Cassie-Baxter-Wenzel mode. The darker regions of the condensate droplets due to the diffused reflection show that the wetting Wenzel mode exists, and the brighter regions due to some reflection occurring at the liquid-air interface indicate that the Cassie- Baxter mode with air trapped underneath exists as well. As a comparison, the wetting behavior of condensate droplets on a hydrophobic surface was also visually studied. Different from the wetting behavior of condensate droplets on the superhydrophobic surface, no brighter spots, as shown in Fig. 1.12, were found during the coalescence of the condensate droplets on the hydrophobic surface. From the experimental results of the wetting behavior presented above, it is demonstrated that NCG can be trapped in the cavities of the hierarchical structures composing the roughness of the superhydrophobic surface by the condensate droplets, during condensation in the presence of NCG. This wetting mode may result in a high mobility of the condensate droplets.



Fig. 1.12 – Coalescence behavior of condensate droplets on hydrophobic surface during steam-air vapor mixture condensation process (W_{NCG} =75%, T_{WALL} =59.6 °C, ΔT =5 K, P=107.0 kPa, q=4.36 kW/m²). (Ma et al. 2012)

1.2.4 – Heat transfer models

Fig. 1.13 shows the results obtained by several authors during the analysis of FWC phenomenon. As it can be seen, high uncertainty still characterizes this research field for many reasons. During dropwise condensation the vapor–surface temperature difference is very small and in many cases its magnitude may be comparable with the uncertainty in its measurement so that misleading inferences can be drawn when plotting a quantity involving its reciprocal. Moreover, during dropwise condensation mode the coolant-side resistance dominates the process and the heat flux is usually adjusted by varying the coolant flowrate, thus the temperature difference is a dependent quantity. There is little doubt that the correct results lie in the shaded region with lower temperature differences. Error in the other data is primarily due to the presence of NCG in the vapor and, in some cases, to insufficient accuracy in the measurement of the surface temperature. The extreme sensitivity of the dropwise condensation heat transfer coefficient to even minute quantities

(a few ppm) of air in the steam has been demonstrated in several investigations, as previously shown.



Fig. 1.13 – Heat transfer measurements for dropwise condensation of steam at near-atmospheric pressure: 1,Schmidt et al. (1930); 2, Nagle et al. (1935); 3, Gnam (1937); 4, Fitzpatrick et al. (1939); 5, Shea andKrase (1940); 6, Fatica and Katz (1949); 7, Kirschbaum et al. (1951); 8, Hampson and Ozisik (1952);9, Wenzel (1957); 10, Welch and Westwater (1961); 11, Le Fevre and Rose (1964); 12, Kast (1963);13, Le Fevre and Rose (1965); 14, Tanner et al. (1965a); 15, Citakoglu (1966); 16, Griffith and Lee(1967); 17, Citakoglu and Rose (1968); 18, Graham (1969); 19, Wilmshurst and Rose (1970); 20, Tanasawa and Ochiai (1973); 21, Aksan and Rose (1973); 22, Stylianou and Rose (1980); 23, Ma et al.(1994); 24, Leipertz and Koch (1998)

A simple model for the dropwise condensation process has been presented by Rose (1966), but it has some intrinsic limits. The essence of the approach is to combine an expression for the heat transfer through a drop of a given size with an expression for the average distribution of drop sizes and to integrate overall drop sizes to obtain, for a given vapor–surface temperature difference, the heat flux for the surface. The crucial factors in the expression for heat transfer through a drop (q_b) are conduction, surface curvature and interface temperature drop. A related approach by Tanaka (1975) includes detail of the coalescence process. In the theory of Le Fevre and Rose (1966) the heat flux through the base of a hemispherical, drop with radius r (see Eq. (1.48)) is given by

$$q_b = \frac{(T_{SAT} - T_{SUP}) - \frac{2\gamma_{lg}T_{SAT}}{r\rho_l r'}}{K1\frac{r}{\lambda_l} + K2\frac{0.627T_{SAT}S+1}{0.664r'^2\rho_{\nu}\varsigma-1} \left(\frac{RT_{SAT}}{2\pi}\right)^{0.5}}$$
(1.52)

where ς is the ratio of the principal specific heat capacities, *K1* is a constant introduced by Rose (1966) concerning the conduction in a drop (*K1=2/3*) and *K2* is a constant introduced by Rose (1966) and it is the ratio between the base and the surface area of a drop (*K2=1/2*). The second term in the numerator on the right-hand side of equation (1.52) is the amount by which the vapor must be subcooled in order to condense on the convex liquid surface. The first term in the denominator is the conduction resistance, and the second term accounts for interphase mass transfer as well as possible promoter layer resistance (included in *K2*). For the distribution of drop sizes, Rose (1966) used

$$f = 1 - \left(\frac{r_{var}}{r}\right)^{1/3}$$
(1.53)

where f is the fraction of surface area covered by drops having a base radius greater than r_{var} . Equation (1.53) indicates that no area is covered by drops larger than the largest, and that, as the drop radius approaches zero, the fraction of area covered by all larger drops approaches 1. Equation (1.53) may be written as

$$A(r_{var})dr_{var} = \frac{1}{3} \left(\frac{r_{var}}{r}\right)^{-1/3} \frac{dr_{var}}{r}$$
(1.54)

Equations (1.52) and (1.54) were combined and the average heat flux for the surface (in terms of the temperature difference) was obtained by integration overall the drop radii:

$$q = \frac{1}{3r^{1/3}} \int_{r_{min}}^{r} q_b r_{var}^{-2/3} dr_{var}$$
(1.55)

for r_{min} see Eq. (1.49). A graph giving the theoretical prediction for dropwise condensation of steam at different pressures is given in Fig. 1.14.



Fig. 1.14 – Theoretical results of Rose (1966) for dropwise condensation of saturated steam at different vapor temperatures (Rose (2000))

The previous model can be properly applied to hydrophobic flat surfaces, but it fails when applied to micro/nano structured superhydrophobic surfaces. Three main inconsistencies arise when applying the previous model to condensation on structured surfaces:

- Droplet wetting morphology cannot be predicted (i.e., Cassie, Wenzel or partially wetting);
- droplet contact angle can vary during droplet growth;
- droplet size distribution is not valid for droplets with non-constant contact angles and with surfaces undergoing coalescence-induced droplet jumping as the main mode of droplet removal.

Wang *et al.* (2013) included in their model the different wetting modes (see Fig. 1.9) and more phenomena that attend on DWC. The structure of the model is similar to the one of Rose (2002) model, thus for each wetting morphology (W morphology corresponds to the flat surface) they calculated the heat transfer by a single drop that change contact angle while growing and the corresponding droplets size distribution on the surface. Some results are reported in Fig. 1.15 and Fig 1.16.



Fig. 1.15 – (a) Condensing droplet apparent contact angle θ_a as a function of coalescence length (lc/l) and ratio of pillar diameter to center-to-center spacing (d/l). (b) Condensing droplet departure radius r (in figure \hat{R}) as a function of coalescence length (lc/l) and ratio of pillar diameter to center-to-center spacing (d/l) (Wang et al. 2013).

In Fig 1.14 (a) for d/l > 0.36, the PW droplet morphology is favored; however, droplet jumping is not possible due to the high solid fraction and high contact line pinning to the surface structure. For $l_c/l < 2$ (not-shown), liquid films and pinned W droplets are formed due to droplet merging within the unit cell of the structure. In (b) is shown that regimes of

W droplet have higher departure radii than PW droplets due to higher surface adhesion and contact angle hysteresis. From Fig.1.15 is evident that:

- a best structured surface for the heat transfer, that may be hydrophobic instead of superhydrophobic exists;
- flat surfaces that present W wetting morphology has r higher than other configurations, i.e. minor heat fluxes at the same ΔT .



Fig. 1.16 – Normalized overall steady-state surface heat flux q/q_{max} , as a function of coalescence length (lc/l) and ratio of pillar diameter to center-to-center spacing (d/l) for (a) $h=1\mu$ m, (b) $h=2\mu$ m (Wang et al., 2013).

Fig.1.16 shows that scaling down the surface structure ((b) to (a)) enhances performance due to the reduced micro/nanostructure thermal resistance. q_{max} was determined from examining the peak heat flux in all cases, which occurred for the smallest scale structure (a), $q_{max}=342.12$ kW/m². Fig.1.16 reiterates the two previous point and furthermore adds another important question that will be illustrated in the next paragraph. There is a maximum heat flux that could be transfer by a superhydrophobic, or hydrophobic, surface beyond which FWC happens.

1.2.5 – Transition from DWC to FWC

Transition from dropwise to filmwise condensation mode in shown in Fig. 1.17.



Fig. 1.17 – Condensation of aniline on a PTFE-coated copper surface (a) using air as coolant fluid, (b) to (f) using water as coolant fluid with a flowrate increasing from (b) to (f). Vapor temperature is equal to 113 °C. (Wilmshurst, 1979)

As Fig. 1.17 suggests, the transition from DWC to

FWC is due to the increasing of the heat flux, thus it exists a maximum heat flux which can be sustained during DWC. It is properly specified that the maximum heat flux is closely related to the surface as well as to the condensing substance. In Rose (2002) there is a list of peak heat fluxes for different substances and how those are linked to the saturation temperature. Hence we will present several considerations about this phenomenon on both structured and flat surfaces.

1.2.5.1 – Structured surfaces

Figure 1.18 shows SEM images of a copper and an aluminum nanostructured surfaces, treated with different techniques to be superhydrophobic.



Fig. 1.18 – SEM images of a) Copper superhydrophobic structured surface (Torresin et al. 2012), b) Aluminum superhydrophobic structured surface (Bisetto et al. 2014).

There are clearly two different structures. The copper one presents a high density of nanowires, instead the aluminum one presents peaks and canyons, this fact depends on the treatment that the specimen withstands. The behavior of a droplet sitting on the two morphologies is now investigated (see Fig.1.9). If we take Eq. (1.40) and we multiply it by the area between the solid and the liquid

$$W_a = (\gamma_{sg} + \gamma_{lg} - \gamma_{ls})A_{ls} \tag{1.56}$$

we obtain the adhesive energy that is necessary to the droplet to roll off.


Fig. 1.19 – Sketch of a droplet in the two morphologies

There are two considerations that can be drawn from Fig. 1.19:

- *W_a* in case a) is lower than in case b), a droplet sitting in a canyon is more anchored to the surface due to the increment of *A_{ls}*.
- External forces could not be sufficient to dry canyons. Therefore for capillarity they will be covered by condensate, compromising surface properties.

In Wang *et al.* (2013) model there is an interesting comparison between structured and flat surfaces reported in Figure 1.20.



Fig. 1.20 – Structured and flat surface heat flux q" as a function of intrinsic promoter coating contact angle hysteresis $\Delta \theta$. Inset: Droplet departure radius (\hat{R}), as a function of intrinsic flat surface contact angle hysteresis ($\Delta \theta$) (Wang et al. (2013)).

Figure 1.20 shows that if the contact angle hysteresis is reduced for a flat surface (θ_a =120°, θ_r =variable), the overall heat flux is increases due to the reduction in the droplet departing diameter. In addition, the results show that at low contact angle hysteresis, the flat surface

has better performance than a superhydrophobic surface. However, as the length scale of the superhydrophobic surface is reduced, the required hysteresis to maintain enhancement for the flat surface is also reduced. Therefore, structured surfaces with relatively large structure scale features may not be advantageous when compared to a flat hydrophobic surface. Nevertheless, in a flat surface is difficult to obtain low $\Delta\theta$, e.g. lower than 50°.

1.2.5.2 – Flat surfaces

The physic phenomena that stay at the base of the transition from DWC to FWC are reassumed by Rose (2002). The first was that film condensation could result from total coverage or flooding by moving/falling drops at some distance along/down the condensing surface. The second was the suggestion that the time taken for drops of given size to coalesce, governed by surface tension, inertia and viscosity, should be constant, while the growth rate of drops increases with increasing in condensation rate; at sufficiently high condensation rates, the heat transfer area would be covered more quickly than it was made available, resulting in transition. The third possible mechanism was that, with increasing vapor–surface temperature difference, the number of active nucleation sites increases that much that, in some locations, the active sites are effectively in contact with neighbors, giving rise to a wetted patch. Anyhow, the physic explanation for the phenomenon is still not understood.

Tanasawa and Utaka (1983) studied the peak heat flux on a copper surface over which steam condensation occurs. The peak heat flux for steam at atmospheric pressure exceeded 10 MW/m^2 when the vapor–surface temperature difference was around 20 K. The study of Utaka (1987) demonstrated that the heat peak flux depends also to the overall cooling-side conductance, thus the material of the specimen.

As it will be shown later, in our case, FWC is completely developed on a flat hydrophobic treated surface, thus a new type of condensation with a new model to calculate the heat transfer coefficient have been analyzed.

1.3 – Slip driven condensation

At the macroscopic level, it is well accepted that the boundary condition for a viscous fluid at a solid wall is the one of no-slipping, i.e., the fluid velocity matches the velocity of the solid boundary. While the no-slip boundary condition has been proven experimentally to be accurate for a number of macroscopic flows, it remains an assumption that is not based on physical principles. In fact, nearly two hundred years ago Navier proposed a general boundary condition that incorporates the possibility of fluid slip at a solid boundary. Navier's proposed boundary condition assumes that the velocity, called slip velocity u_s (see Figure 1.21), at the solid surface is proportional to the shear stress at the surface

$$u_s = \beta \frac{\partial u_z}{\partial y} \tag{1.57}$$

where β is the slip length or slip coefficient. The agreed physic meaning of β is explained by Fig. (1.21).



Fig. 1.21 – *Slip length* β *over a surface (P. G. de Gennes, 2002).*

If $\beta=0$ then the generally assumed no-slip boundary condition is obtained. If β is finite, fluid slip occurs at the wall, but its effect depends upon the length scale of the flow. For example, the solution for Stokes flow between two infinite parallel plates with the boundary conditions of no shear stress at the centerline and Navier's hypothesis (Eq. (1.57)) at the wall, yields

$$u_{s} = \frac{h^{2}}{2\mu} \left(-\frac{\partial p}{\partial z} \right) \left[\left(1 - \left(\frac{y}{h} \right)^{2} \right) + \frac{2\beta}{h} \right]$$
(1.58)

where 2h is the distance between the two plates, μ is the viscosity, and $\left(-\frac{\partial p}{\partial z}\right)$ is the

fluid pressure gradient. The first term in the brackets is the standard solution for pressuredriven Stokes flow between two infinite parallel plates with no slip, while the second term, ${}^{2\beta}/{}_{h}$, represents an additional velocity associated with the general boundary condition

given by Eq. (1.57). If β is finite, as *h* decreases, the specific weight of the second term increases. Thus, at the micro- and nano-length scales this term can be significant and can dominate the process. In literature we found both analytical and experimental procedures to determinate this slip factor, otherwise the slip velocity (u_s), but this phenomenon is still difficult to evaluate. It was proved, anyway, that a slip boundary exists when the surface is hydrophobic or superhydrophobic, as showed first by Tretheway and Meinhart (2001) and lately also by Bushan *et al.* (2009). Different attempts were made to estimate the slip velocity, but only on stationary single phase flow between infinite and planar plates. Results converge in this two factors:

- u_s is about 10% of the free stream velocity, that is measured at h;
- the free stream velocity is influenced by the slip velocity only in microscale, i.e. it was demonstrated that *h* has to be inferior than 1 mm.

We tried to understand, besides, what is the physical phenomena that is beyond the experimental evidences induced by several research teams. Hence the most accredited theories are presented:

- 1. micro-nano bubbles grow between solid and liquid, thus the liquid could slip on the solid surface (de Gennes, 2002);
- 2. a low-viscosity zone is present at the proximity of the solid surface, thus, in this zone, the liquid presents an higher velocity than usual (Granick *et al.*, 2003).

Hsieh *et al.* (2009), Tretheway and Meinhart (2004) and others have done several experiments to prove and understand this phenomenon and its cause, but no one, to the best of the author's knowledge, studied it in a two-phase heat transfer. Here we analyze only the second hypothesis, since during pure steam condensation there are no non-condensable gases that can stand between the solid and the liquid. To really comprehend the second

hypotesis we may consider the free energy balance and how it varies from a condition of hydrophilic surface to an hydrophobic one. Considering the adhesion energy (w_a) it is

$$w_a = \gamma_{sg} + \gamma_{lg} - \gamma_{ls} \tag{1.59}$$

where γ_{sg} is the superficial tension between solid and gas, γ_{lg} is the superficial tension between liquid and gas and γ_{ls} is the superficial tension between solid and liquid (see Fig. 1.22).



Fig. 1.22–Interaction between superficial tensions on a liquid volume.

The concept beyond this dissertation is that the bond between the liquid and the solid, given by Eq.(1.59), decreases with the decreasing of the solid energy surface and this permits to the liquid to slip.



Fig. 1.23 – Scheme of intermolecular bond between solid-liquid (w_a) and liquid-liquid.

1.3.1 – Slip driven condensation model

We applied the slip factor to the Nusselt model (1916) for condensation under gravity control and to the Carpenter and Colburn model (1951) for condensation under shear force caused by vapor.

The starting point is Eq. (1.4) that determine the velocity gradient along the condensate layer. It has to be integrated trough δ and the boundary condition is that at y=0, u=0 and so C=0. This is no longer true, indeed we know that for a hydrophobic surface there is a slip velocity thus at y=0 $u=u_s$ and so $C=u_s$.

Firstly it will be shown the modified Nusselt model, then the Carpenter and Colburn modified one.

<u>1.3.1.1 – Gravity controlled condensation</u>

The modified model starts with the following equation

$$u = \frac{(\rho_l - \rho_v)g}{\mu_l} \left(\delta y - \frac{y^2}{2} \right) + u_s \tag{1.60}$$

The mean velocity along the liquid film \bar{u} is given by

$$\bar{u} = \frac{1}{\delta} \int_0^\delta u dy = \frac{(\rho_l - \rho_\nu)g}{\mu_l} \frac{1}{\delta} \int_0^\delta \left(\delta y - \frac{y^2}{2} \right) dy + \frac{1}{\delta} \int_0^\delta u_s dy = \frac{(\rho_l - \rho_\nu)g}{3\mu_l} \delta^2 + u_s$$
(1.61)

Now it is calculated the condensate mass flow rate per unit width, as

$$\Gamma = \bar{u}\delta\rho_l \tag{1.62}$$

And if it is evaluated the increasing of Γ along the surface (z axis), we obtain

$$\frac{d\Gamma}{dz} = \frac{\rho_l(\rho_l - \rho_v)g}{\mu_l} \delta^2 \frac{d\delta}{dz} + \rho_l u_s \frac{d\delta}{dz}$$
(1.63)

Remembering Eqs. (1.8) and (1.9) then we can write

$$\frac{(T_{SAT} - T_{WALL})\lambda_l}{\delta r'} = \frac{\rho_l(\rho_l - \rho_v)g}{\mu_l} \delta^2 \frac{d\delta}{dz} + \rho_l u_s \frac{d\delta}{dz}$$
(1.64)

and

$$\frac{(T_{SAT} - T_{WALL})\lambda_l}{r'}dz = \frac{\rho_l(\rho_l - \rho_v)g}{\mu_l}\delta^3 d\delta + \rho_l u_s \delta d\delta$$
(1.65)

Integrating with the boundary condition that $\delta=0$ at z=0, we have

$$\frac{(T_{SAT} - T_{WALL})\lambda_l}{r'} z = \frac{\rho_l(\rho_l - \rho_v)g}{4\mu_l} \delta^4 + \frac{\rho_l u_s \delta^2}{2}$$
(1.66)

Now, for simplicity, we call

$$a = \frac{(T_{SAT} - T_{WALL})\lambda_l}{r'}z \tag{1.67}$$

$$b = \frac{\rho_l(\rho_l - \rho_v)g}{4\mu_l} \tag{1.68}$$

$$c = \frac{\rho_l u_s}{2} \tag{1.69}$$

and we can rewrite Eq. (1.66) as follow

$$b\delta^4 + c\delta^2 - a = 0 \tag{1.70}$$

If we call $\delta^2 = x$, Eq. (1.70) could be reduced

$$bx^2 + cx - a = 0 \tag{1.71}$$

that is a second grade equation that has two solutions

$$x_1 = \frac{-c + \sqrt{c^2 + 4ab}}{2b} \ x_2 = \frac{-c - \sqrt{c^2 + 4ab}}{2b}$$

 x_2 has to be eliminated because at the numerator it has two negative terms, so it isn't a real solution for the liquid thickness. Thus remembering (1.70) we obtain

$$\delta = \left[\frac{-\frac{\rho_{l}u_{s}}{2} + \sqrt{\left(\frac{\rho_{l}u_{s}}{2}\right)^{2} + 4\frac{\rho_{l}(\rho_{l}-\rho_{v})g(T_{SAT}-T_{WALL})\lambda_{l}}{4\mu_{l}}}}{2\frac{\rho_{l}(\rho_{l}-\rho_{v})g}{4\mu_{l}}}\right]^{1/2}$$
(1.72)

Obviously the other negative root is eliminated. As in Eq. (1.13) we calculate the HTC driven by slippery as

$$\alpha_s = \frac{\lambda_l}{\delta} \tag{1.73}$$

Now at α_s all the considerations exposed in the first paragraph of this chapter are applied, thus we write

$$\alpha_{grav,s} = 1.15 \left[0.0206 \left(\frac{r'\mu_l}{\lambda_l (T_{SAT} - T_{WALL})} \right)^{0.5} + 0.79 \right] \alpha_s$$
(1.74)

It has to be noticed that $\alpha_{grav,s}$ is a local coefficient and it is appropriate to calculate the mean heat transfer coefficient along the specimen. Thus

$$\bar{\alpha}_{grav,s} = \frac{1}{L} \int_0^L \alpha_{grav,s} \, dz \tag{1.75}$$

<u>1.3.1.2 – Shear controlled condensation</u>

The shear stress caused by vapor on the liquid film is evaluated gathering the case with no slip factor, but remembering that the vapor has to persists on the thickness due to the slip boundary. Thus in the following Eq. that comes from Eq. (1.27)

$$a_{ss,s} = \frac{\lambda_l}{\delta} = \sqrt{\frac{\lambda_l^2 \tau_0 \rho_l}{2\Gamma \mu_l}} \tag{1.76}$$

is present Γ that is calculated by Eq. (1.6) where δ is substitute by Eq. (1.72). As usually it is determined the mean coefficient

$$\bar{\alpha}_{ss,s} = \frac{1}{L} \int_0^L \alpha_{ss,s} \, dz \tag{1.77}$$

<u>1.3.1.3 – Condensation under combined gravity and shear control</u>

We still use the simplified approach of Butterworth (1983), so the HTC is estimated from

$$a_{th,s} = \sqrt{\alpha_{ss,s}^2 + a_{grav,s}^2} \tag{1.78}$$

<u>1.3.1.4 – Slip velocity appraisal</u>

In previous paragraphs we presented the new condensation model that accounts for a slip velocity at the interface between solid and liquid, but we did not explain how to calculate this velocity. As told before, in literature, there are several methods to experimentally estimate u_s but it is impossible to apply those kind of instruments (e.g. µPIV) to our experimental section for the time being. However different research teams (Tretheway and Meinhart, (2002) and Hsieh and Lin (2009)) found a correlation between u_s and the free stream velocity u_{free} :

$$u_s = 0.1 u_{free} \tag{1.79}$$

Meanwhile Granick *et al.* (2003) explicit that the slip velocity increases with the increasing of the mass flow rate. Thus, on principle, we had to understand how to define u_{free} , in fact in their experiments the authors have a single phase flow and u_{free} is the velocity at the center between the two planar plate that is the highest velocity that the liquid reaches. In our case there is not a symmetric distribution of velocities; we could also define the u_{free} velocity as the velocity that the condensate layer reaches at the interface between the liquid and the vapor ($u_{interface}$). The second important conclusion that comes out from previous researches is the reliance of the slippery condition on the thickness of the liquid, i.e. with the increasing of the thickness, the influence of a slip velocity decreases. The model combines this two factors in the evaluation of the slip length from which the slip velocity is calculated. The second phenomenon is modeled as shown in Fig. 1.24. At the correspondence of the maximum condensate thickness, the slip coefficient assumes a minimum value, whereas β assumes the higher value where δ is the lowest.



Fig. 1.24 – *Correspondence between* δ *and* β *.*

Once β is estimated, the dependency by the velocity interface has to be added. First of all, it is explained how the $u_{interface}$ is evaluated.

The gravity force, as the shear stress, induces a velocity on the liquid film that is calculated by

$$u_{grav} = \frac{(\rho_l - \rho_v)g}{\mu_l} \left(\delta^2 - \frac{\delta^2}{2}\right) \tag{1.80}$$

for the gravitational component and

$$u_{shear} = \frac{\tau_0 \delta}{\mu_l} \tag{1.81}$$

for the shear stress. It has to be noticed that both velocities are evaluated at δ , calculated by Eq. (1.72), therefore at the maximum thickness of the liquid film where the speed is maximum. Thus these factors, u_{grav} and u_{shear} , are estimated in the normal condition of no slip, and the $u_{interface}$ becomes

$$u_{interface} = u_{grav} + u_{shear} \tag{1.82}$$

Now it is possible to correct the slip coefficient with a factor (*F*) that depends on the $u_{interface}$, therefore β is firstly calculated as explained in Fig. 1.24 and then it is multiplied by the corrective factor. As the velocity at the interface between the liquid and the vapor increases, also *F* increases (Fig. 1.25).



Fig. 1.25 – *Correspondence between* F *and* $u_{interface}$.

Where $u_{interface}$ is equal to 0 the factor assumes a value equal to 1, thus β depends only by the condensate layer thickness, whereas the $u_{interface}$ becomes higher, β is more influenced by this phenomenon. Once β is estimated, u_s is simply determined (see Eq. (1.57)).

Nomenclature

 $A = \operatorname{area}, \mathrm{m}^2$

 A_c = droplet lateral area, m²

 A_{ls} = droplet wet area, m²

C = integral constant, -

 $C_d = \text{drag coefficient}, -$

 c_p = specific heat capacity, J kg⁻¹ K⁻¹

d = pillar diameter, m

 D_h = hydraulic diameter, m

 $\frac{d}{dy}$ = gradient along y axis, m⁻¹

 $\frac{d}{dz}$ = gradient along z axis, m⁻¹

dz = derivate of length along z axis, m

 $\left(\frac{dp}{dz}\right)_F$ = two-phase frictional pressure gradient, Pa m⁻¹

E = coefficient, -

F = coefficient, -

 F_A = droplet moving adhesive force, N

 $F_{adhesive}$ = droplet static adhesive force, N

 $F_{cohesive}$ = droplet static cohesive force, N

 $F_D = \text{drag force, N}$

 F_G = gravity force, N

 F_R = droplet static resulting force, N

Fr = Froude number, -

 f_r = the roughness factor, -

f = fraction of surface area covered by drops, -

G = moisture specific mass flow rate, kg s⁻¹

H = coefficient, -

h = pillar height, m

 $g = \text{gravity acceleration, m s}^{-2}$

k = non-circular perimeter factor, -

 $K_1 = \text{factor}, -$

 $K_2 = \text{factor}, -$

 $K_3 = \text{factor}, -$

L =length, m

l = pillar center-to-center spacing, m

 l_c = average condensing droplet spacing, m

 $\dot{m} = \text{mass flux}, \text{kg s}^{-1}$

P = perimeter, m

p =pressure, Pa

$$Pr = Prandtl number, -$$

 P_w = droplet wet perimeter, m

 \dot{q} = heat flux, W m⁻²

 q_b = heat flux through the base of a hemispherical drop, W m⁻²

R= gas constant, J K⁻¹ mol⁻¹

- r' =latent heat of vaporization, J kg⁻¹
- r = departure droplet radius, m
- r_{min} = droplet minimum radius, m
- r_{var} = droplet variable radius, m
- Re = Reynolds number, -
- T =temperature, K
- u =velocity, m s⁻¹
- $u_s = \text{slip velocity, m s}^{-1}$
- \overline{u} = mean velocity, m s⁻¹
- V = droplet volume, m³
- x = vapor quality, -
- y = perpendicular axis, m
- W_a = adhesive energy, J
- w_a = specific adhesive energy, N m⁻¹
- w_c = specific cohesive energy, N m⁻¹
- We = Weber number, -
- z =longitudinal axis, m

Greek symbols

- α = heat transfer coefficient, W m⁻² K⁻¹
- $\overline{\alpha}$ = mean heat transfer coefficient, W m⁻² K⁻¹

 $\beta = \text{slip factor, m}$

 γ_{sg} = superficial tension between solid and gas, N m⁻¹

 γ_{lg} = superficial tension between liquid and gas, N m⁻¹

 γ_{ls} = superficial tension between solid and liquid, N m⁻¹

 δ = thickness, m

 $\Delta\theta$ = contact angle hysteresis, °

 ΔT_{ml} = logarithmic mean temperature difference, K, °C

 ς = ratio of the principal specific heat capacities, -

 Φ^2 = two-phase multiplier, -

- ε '= turbulent flow diffusivity, m³ s⁻¹
- θ_{ad} = advancing contact angle, °
- θ_{re} = receding contact angle, °
- λ = thermal conductivity, W m⁻¹ K⁻¹
- μ = dynamic viscosity, Pa s
- ρ = density, kg m⁻³

 $\tau =$ shear stress, Pa

 τ_i = shear stress at the liquid-vapor interface, Pa

- τ_0 = shear stress at the solid-liquid interface, Pa
- Γ = condensate mass flux per unit with, kg s⁻¹

Subscripts

l = liquid

LO =liquid only

F =friction

g = gas

```
GRAV= gravitational
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Nu = Nusselt

S = slip

s = solid

SAT = saturation

SS = shear stress

WALL = surface

v = vapor

Chapter 2

The experimental apparatus

Herein it will be described the experimental apparatus used to study the condensation phenomenon on flat and hierarchical structured surfaces. Firstly the design of the apparatus will be presented and secondly the measurement and acquisition system. Later on, the data reduction technique and the uncertainty analysis for the experimental results is presented.

2.1 – Components of the experimental apparatus

The experimental setup is schematically shown in Fig. 2.1.



Fig. 2.1 – Scheme of the experimental apparatus. P = Pressure transducer, T = Thermocouple, dT = Thermopile, CFM = Coriolis mass Flow Meter, MFM = Magnetic mass Flow Meter, MF = MechanicalFilter.

The system consists of four main components: the boiling chamber, the test section, the cooling water loop and the secondary condenser. We will describe the paraphernalia following the vapor flow loop from the boiling chamber to the boiling chamber.

2.1.1 – Heating zones

Steam is generated in a cylindrical stainless steel evaporator with an internal diameter of 0.2 m and height 0.3 m (Fig. 2.2), capable of working up to a relative pressure of 3 bar; to avoid potentially dangerous conditions the chamber is equipped with a safety valve which is triggered at 3 bar. The chamber contains electrical resistances (heating elements) connected in parallel with each other, that can provide a maximum power of 4 kW to the fluid.



Fig. 2.2 – Boiling chamber.

As noticed from the figure, all the external parts of the chamber are well insulated to minimize thermal losses. The chamber also has a level indicator in order to know the height of the liquid head inside of it, since it is important to maintain the electrical resistances always wetted by the liquid for safety reasons. The adjustment of the input power is handled through a separate electrical panel (Figure 2.3) connected to the heating elements, meanwhile the electrical power supplied to vaporize the liquid inside the boiling chamber is measured by a power analyzer NORMA 4000(Fig. 2.17).



Fig. 2.3 – *Electrical panel for the regulation of the power in the boiling chamber.*

The subcooled liquid entering the chamber is vaporized in the boiler and then it flows through a stainless steel tube having an internal diameter equal to 0.019 m and a length equal to 2 m up to the measurement section. The connection between the boiler and the test section was well insulated to ensure that the vapor remains at the saturated state when it reaches the test section. To avoid formation of condensate on the walls of the vapor line the tubes are heated up to saturation temperature by means of an electrical heater installed around the pipe (wall temperature is checked through a T-type thermocouple) for a length of 0.5 m. The input voltage of the heating elements is controlled by a variable transformer VARIAC (Figure 2.4).



Fig. 2.4 – VARIAC for the regulation of the power of the heating elements.

It was made an upgrading of the heating system during the experimental tests because that length was not enough to prevent the formation of condensate along the line. Thus, now, also the other 1.5 m of pipeline is heated.

2.1.2 – Test section

The test section consists in a PEEK channel that is fitted with the test surface of a rectangular plain aluminum substrate, the top of which can be nanoengineered and functionalized to obtain superhydrophobicity in order to perform tests during dropwise condensation. Vapor pressure and temperature are measured at the inlet of the measuring section by means of a differential pressure transducer (coupled with an absolute one for ambient pressure evaluation) and a T-type thermocouple, respectively. The metallic specimen is cooled on the backside by a water bath, which allows precise control of coolant temperature difference between the inlet and the outlet is measured by means of a three-junction copper-constantan thermopile. To assure precise evaluation of heat flux, the coolant mass flow rate is measured by means of a Coriolis effect mass flow meter.



Fig. 2.5 – Sketch of the test section.

The experimental section has been made to measure the heat transfer coefficient during the condensation of steam on nanostructured surfaces and meanwhile allowing direct visualization of the process. It basically consists in a condensation chamber made of PEEK, an advanced thermoplastic material, with excellent mechanical characteristics, excellent

resistance to chemical corrosion and a large working range in terms of temperatures. A fundamental characteristic of this material is to have a very low thermal conductivity $(\lambda_{PEEK}=0.25 \text{ W/(m K)})$: this means that the dispersions towards the external environment are minimal and ensuring monodimensionality of heat flux along the specimen. The section consists of a rectangular channel having dimensions equal to 160 mm x 30 mm x 5 mm dug in the block of PEEK (hydraulic diameter d_h=0.0086 m). One side of the channel is covered by two glass plates that allow the visualization of the process, with an intermediate chamber to increase the thermal insulation. The front glass is equipped with an electric heater to avoid the condensation of the steam on the internal glass and thus to permit to observe the two-phase process. On the opposite side there is a hole where the sample is placed. The thickness of the sample is equal to 10 mm and dimensions of the condensation surface are 50 mm x 20 mm and for the analysis of the heat exchange it is instrumented with four thermocouples. The metal sample, which in all the tests is aluminum, is positioned between the two main parts of the test section, the first consists in the steam channel block, the other one in the cooling channel block. The latter consists in a rectangular duct having dimensions equal to 160 mm x 20 mm x 5 mm with a hole for the accommodation of the back side of the sample. Between the vapor side and the cooling side two intermediary frames are positioned for gaskets and thermocouples. Fig. 2.5, Fig. 2.6 and Fig. 2.7 report drawing and photos of the test section.



Fig. 2.6 – Schematic of the test section.



Fig. 2.7 – Photo of the test section components before assembling.

The cooling system is used to adjust the surface temperature of the test sample, thus to control the saturation to wall temperature difference. The cooling water flows in countercurrent with respect to the direction of the steam inside the test section. The position of the input and output water channels, as well as the channel length, were determined in order to have a uniform velocity profile of the fluid in the cooling duct. In the area below the sample is necessary that the velocity profiles of the cooling water are fully developed and will not suffer the turbulence caused by the input and output channels. For this purpose, simulations were made with codes CFD calculation to evaluate the optimum geometry, in order to have a heat exchange efficiency and a temperature profile as possible one-dimensional along the sample when it is cooled. The variable parameter was the channel length and thus we obtained 160 mm long.

2.1.3 – Secondary condenser

Downstream of the test section, the uncondensed vapor and condensed liquid pass through a secondary water condenser which completes the condensation process and returns the subcooled liquid to the boiler through a gravity drain. Inlet and outlet water temperatures are acquired by means of T-type thermocouples while water mass flow rate is measured by a magnetic flow meter. The secondary condenser is made from a brass cylindrical body (internal diameter 0.1 m) containing a coil inside which the cooling water flows. In Figure 2.8 we can see a graphical representation. The cooling water flowing through the coil is aqueduct water at a temperature of about 15 ° C. It is permitted the adjustment of the water flow thanks to two valves, one between the coil and the magnetic flowmeter and the other after the coil. The spiral consists of a copper tube of internal diameter 6 mm, having a thickness equal to 1 mm, and it is about half meter long. To check the complete condensation of the vapor it is installed in the pipeline a liquid indicator.



Fig. 2.8 – Internal and external view of the secondary condenser.

Since it has been decided not to install a circulating pump, several considerations have been made in order to assure the natural convection of the fluid. First of all, the dimensions of the evaporator have been properly defined. This is very important because the electrical heaters must always be wet by the water. To guarantee the liquid return in the boiler, some evaluations have been made in order to establish the right position for the secondary condenser: the total pressure drop due to the sum of the frictional pressure losses in the vapor line and in the test section, plus the local pressure drops due to abrupt geometry changes have been calculated. With a maximum mass flow rate equal to 5.57 kg/h a value of 4340 Pa has been obtained, corresponding to a required height between the top of the boiling chamber and the bottom of the post-condenser equal to 0.443 m.

2.1.4 – Auxiliary components

The apparatus is composed by other components that maintain stable and appropriate work conditions, as it is could be seen in Fig. 2.1. To regulate the system pressure, a hydraulic accumulator (Fig. 2.9) is installed in the liquid line downstream the post-condenser. Moreover, for testing a sample over several days, the hydraulic accumulator is used to maintain the system in overpressure overnight, thereby avoiding the need for DI water refilling. A precise needle valve (Fig. 2.10) is used to regulate the liquid flow at the inlet of the boiling chamber in order to achieve stable conditions during the tests. In between the hydraulic accumulator and the needle valve a mechanical filter and a liquid indicator are installed. Before entering the boiling chamber the temperature of the subcooled liquid is measured by means of a T-type thermocouple. A filling line is also installed: when the system has to be filled, after proper vacuuming, water from a supplying tank is pumped inside the test rig using a centrifugal pump. All the components of the test rig, with the exception of the test section, are made by stainless steel in order to avoid contamination of the fluid. Steam components, boiling chamber and the stainless steel line are very well insulated in order to avoid heat losses to the ambient.



Fig. 2.9 – Hydraulic accumulator.



Fig. 2.10 – Needle valve.

When performing condensation heat transfer tests, it is crucial to prevent any noncondensable gases from entering into the flow loop in order to avoid any potential errors in the measurement of heat transfer coefficient. To this end, before filling the flow loop with de-ionized (DI) water, the air in the test rig is pumped out using a vacuum pump. Moreover, tests are performed maintaining the flow loop in overpressure compared to external ambient, preventing any entry of non-condensable gases into the flow loop. To definitely remove non-condensable gases, maybe dissolved in the filling water, we use a dedicated valve located in the upper part of the post-condenser. This procedure is made when the water is heated to the saturation temperature and the saturated steam starts to circulate in the system and it is repeated several times.

A full view of the experimental apparatus is shown in Figure 2.11.



Fig. 2.11 – *Experimental apparatus.*

2.2 – Measurement system

The aim of the apparatus is to calculate the heat transfer coefficient during pure steam condensation and to pursue this goal it needs different kind of transducers which are presented below.

2.2.1 – Temperature transducers

There are two kind of temperature transducers in the system.

2.2.1.1 – T-type thermocouple

It is formed by two copper-constantan junctions, one positioned in the point where the temperature has to be measured and the other inside a proper reference instrument, which is equipped with a physical cell in which it is reproduced and maintained the reference temperature 0° C. Thermocouple has an intrinsic uncertainty of $\pm 0.05^{\circ}$ C. In the circuit are used both thermocouples in the metal sheath (stainless steel, external diameter 0.5 mm) that thermocouples sheathed in Teflon (external diameter 0.5 mm).



Fig. 2.12 – T-type thermocouple. Copper and constantan cables are shown.

Thermocouples were placed:

- At the entrance of the measuring section, in the steam circuit, for the measurement of the saturation temperature.
- In the measuring specimen (4 thermocouples), for the extrapolation of the surface temperature.
- At the entrance of the cooling circuit of the measuring section, to measure water inlet temperature.
- At the entrance and at the exit of the cooling circuit of the post-condenser, for the measurement of heat flux removed from post-condenser.

- Downstream of the post-condenser in the primary circuit, for the measurement of the degree of subcooling of the liquid at the exit of the post-condenser.
- At the inlet of the boiling chamber, for determining the mass flow rate of the liquid circulating in the circuit.
- On the wall of the vapor line, in two different positions, to verify that there is no condensation along the tube.

<u>2.2.1.2 – T-type thermopile</u>

A three junctions copper-constantan thermophile, with an intrinsic uncertainty of $\pm 0.03^{\circ}$ C has been installed. In the wells located in the inlet and outlet of the cooling circuit of the measuring section, measure the temperature gradient of the water, which removes heat from the condensing vapor flowing in the primary side.

2.2.1.3 - Calibration test

To calibrate the thermocouples and the thermopile a Heart Scientific Super Thermometer-II coupled with two RTD Standard AS115, that permits to obtain a measurement of the reference temperature with uncertainty of ± 0.002 K, or a resistance thermometer PT100, that permits to obtain a measure of reference temperature with uncertainty of ± 0.01 K were used. Figure 2.13 shows a photo of superthermometer and probes, respectively. The aim of the calibration is to obtain an error band for thermocouples lower than ± 0.05 K comparing to the reference value measured by the superthermometer. For the thermopile the desired error band is ± 0.03 K. The difference between the reference value measured by the superthermometer and the one read by the instrument gives the value of the correction that must be applied to the measured parameter.



Fig. 2.13 – RTD Standard AS115T probes on the left and Super Thermometer on the right

The calibration of the thermocouples was carried out for a range of temperatures varying between 20° C and 60° C by coupling the superthermometer with two RTD Standard

AS115 and between 60° C and 98° C by coupling it to the resistance thermometer PT100, because AS115 are not calibrated for temperatures above 60° C. The calibration of the thermopile was carried out for the same range of temperature of thermocouples but with a different procedure. Meanwhile one probe was inserted in a measuring well at a fixed temperature (10-25-40-55-70-85° C) the other one was in an another well where the temperature was been changed in a range of \pm 5° C respectively to the fixed temperature. An example of calibration is reported below.



Fig. 2.14 – Calibration curve for the thermopile at 55°C. In the x-axis it is shown the temperature difference measured by the instrument, while in the y-axis it is shown the difference between the values obtained with superthermometer and the thermopile.

The graph in Figure 2.14 shows the deviations between the temperature differences measured with the superthermometer and those measured by the thermopile. In the graph it is also reported the trend line of the experimental points, from which it is possible to derive the calibration equation of the thermopile.

2.2.2 – Pressure transducers

The experimental apparatus is equipped with two different kind of pressure transducers.

<u>2.2.2.1 – Druck differential pressure transducer</u>

The differential transducer (Fig. 2.15) is used to calculate the pressure difference between the external environment and the internal of the measurement section. The pressure line connecting the instrument to the pressure port is heated by an electrical resistance in order to remove any drops of condensate that may form inside the duct that could alter the measure. In our system a differential pressure transducer Druck D5071 with full scale 7 bar, coupled to an electronic converter Druck DPI 280 is used.

The uncertainty of the instrument is determined with the following relationship:

$\xi=\pm 0.04\%\, p_{fullscale}$

(2.1)



Fig. 2.15 – Relative pressure transducer (left side)

2.2.2.2 – Druck absolute pressure transducer

The absolute pressure transducer, having 1.2 bar full scale, is used in parallel with the differential one and it is connected directly to the external environment. Knowing the ambient pressure and the pressure difference Δp between the external environment and the condensing vapor is possible to evaluate the absolute pressure inside the primary

$$p_{vap} = p_{atm} + \Delta p \tag{2.2}$$

The uncertainty of the absolute pressure transducer is calculated as:

2.2.3 - Mass flow meters

There are two kind of mass flow transducers in our system.

2.2.3.1 - Coriolis mass flow meter Endress + Hauser PROMASS 80

The Coriolis mass flow meter (Figure 2.16), located along the cooling circuit of the measuring section is installed, for a precise evaluation of the heat flux removed by the cooling water. In this type of measuring system the fluid flows inside through a curved duct, which starts to oscillate developing a Coriolis force

$$\vec{F} = 2\dot{m}\vec{\omega}\wedge\vec{v} \tag{2.4}$$

where ω is the angular velocity, v is the fluid velocity and \dot{m} is the mass flow. The Coriolis force induces a deformation on the duct that is measured and thus the mass flow is calculated.

The uncertainty of the flow meter which measures the flow rate of the coolant is:

$$\xi = \pm 0.15\% \pm \left(\frac{0.1}{\dot{m}}\right)\%$$
(2.5)

with \dot{m} in kg/h.



Fig. 2.15 – Coriolis mass flow meter.

2.2.3.2 - Electromagnetic mass flow meter Danfoss MAG 1100

The flowmeter is located along the cooling line of the post-condenser (Figure 2.16). The tool generates in the liquid (which must be an electrical conductor) an alternating magnetic field through the electrodes. By measuring the voltage generated between two duct sections along the fluid way it is possible to derive its volumetric flow rate, since the speed of the liquid is linearly proportional to the measured voltage.

The uncertainty of the instrument is given by

$$\xi = \pm 0.25\% \ if \ v_{fluid} \ge 1.5 \ \text{ft/}_{\text{S}} \tag{2.6}$$

$$\xi = \pm \frac{0.41}{v_{fluid}} \% \ if \ v_{fluid} < 1.5 \ \text{ft/}_{\text{S}}$$
(2.7)



Fig. 2.16 – Electromagnetic flow meter.

2.2.4 – Power analyzer

The power analyzer Fluke Norma 4000 (Fig. 2.17) is used to measure the current and the voltage applied to the heads of electrical resistors placed inside the boiling chamber, and therefore the power supplied from the same to the liquid to vaporize. The instrument has an uncertainty in the measurement of the active power supplied in the boiling chamber equal to $\pm 0.1\%$ of the measurement.



Fig. 2.17 – Power analyzer Norma.

2.3 – Data reduction technique

For extrapolating the HTC the following parameters have to be calculated:

- Wall temperature of the specimen.
- Specific heat flux through the specimen (q).
- Specific mass flow rate of the vapor flowing in the experimental section.

With these parameters is possible to evaluate the heat transfer coefficient

$$HTC = \frac{q}{\Delta T_{ml}}$$
(2.8)

with ΔT_{ml} mean logarithmic temperature difference between the steam and the wall. In all the following formulas heat losses to the environment in the vapor line, in the boiling chamber and in the test section are neglected; this is justified by the fact that these components are well insulated.

2.3.1 – Superficial temperatures

The aluminum sample is embedded with four thermocouples, two at the inlet and two at the output with respect to the vapor flow, located at two different positions along the perpendicular axis (y) of the sample: one pair is located 1 mm below the top surface, the other 2.75 mm below the top surface. A sketch of the specimen prototype is shown in figure 2.18.



Fig. 2.18 – Sketch of the specimen. All the significant measures are reported.

Having two temperature measurements at two different depths along the sample, it is possible to extrapolate the superficial inlet and outlet temperatures

$$T_{WALL,IN} = T'_{IN} + (T'_{IN} - T''_{IN}) \frac{y_1}{y_2 - y_1}$$
(2.9)

$$T_{WALL,OUT} = T'_{OUT} + (T'_{OUT} - T''_{OUT})\frac{y_1}{y_2 - y_1}$$
(2.10)

where T' are the temperatures measured at $y_1 = I$ mm, while T'' are the temperatures measured at $y_2 = 2.75$ mm. Once superficial temperatures have been extrapolated the mean logarithmic temperature difference between the saturated steam and the wall can be calculated as

$$\Delta T_{ml} = \frac{T_{WALL,OUT} - T_{WALL,IN}}{\ln\left(\frac{T_{SAT} - T_{WALL,IN}}{T_{SAT} - T_{WALL,OUT}}\right)}$$
(2.11)

 T_{SAT} , the saturation temperature of the steam, is derived from the saturation pressure (measured) and it is verified through the T-type thermocouple which is located at the entrance of the test section in order to assess the possible presence of non-condensable gases.

2.3.2 – Heat flux

There are three different techniques to obtain the specific heat flux through the specimen.

The primary cooling system is regulated by a thermostatic bath (Termohaacke), which allows the circulation of water in the back of the metal sample. The mass flow rate of the refrigerant $\dot{m}_{coolant}$ is measured through the Coriolis mass flow meter, while the temperature measurements are given by a T-type thermocouple and the thermopile. To perform precise measurements of the water temperatures $\Delta T_{coolant}$ two measurement wells have been realized on the primary cooling circuit, one at the inlet and one at the outlet of the measuring section. The specific heat that is extracted by the cooling water from the condensing vapor, can be calculated as

$$q_{cooling} = \frac{\dot{m}_{coolant} c_{p,coolant} \Delta T_{coolant}}{A}$$
(2.12)

where $c_{p,coolant}$ is the specific heat of the coolant water, at the average temperature between the inlet and the outlet of the section and A is the specimen frontal area,

$$A = 0.05 * 0.02 = 0.001 \text{ m}^2 \tag{2.13}$$

By means of the four thermocouples embedded in the specimen it is possible to evaluate the local inlet and outlet heat fluxes assuming one-dimensional conduction along the sample. The latter assumption is justified by the fact that the PEEK has a lower thermal conductivity compared to aluminum (λ_{PEEK} = 0.25 W/(m K), λ_{Al} = 200 W/(m K)). Then, appling the Fourier's law for thermal conduction, it is possible to obtain

$$q_{inlet} = \lambda_{Al} \frac{dT_{IN}}{dz} = \lambda_{Al} \frac{T_{SAT} - T_{WALL,IN}}{Z_2 - Z_1}$$
(2.14)

$$q_{outlet} = \lambda_{Al} \frac{dT_{OUT}}{dz} = \lambda_{Al} \frac{T_{SAT} - T_{WALL,OUT}}{Z_2 - Z_1}$$
(2.15)

where the inlet and the outlet temperatures are calculated respectively with Eqs. (2.9) and (2.10). Thus, the average heat flux along the specimen is

$$q_{ave} = \frac{1}{2}(q_{inlet} + q_{outlet}) \tag{2.16}$$
The heat supplied to the fluid in the boiling chamber is partially removed by the primary cooling system in the measuring section, and partially by the post-condenser. The heat extracted by the post-condenser can be calculated knowing the flow rate of cooling water in the secondary condenser $\dot{m}_{w,pc}$, measurement provided by the electromagnetic flow meter, and the temperatures at the inlet $T_{IN,pc}$ and the outlet of the coil $T_{OUT,pc}$, measures given by T-type thermocouples. Thus it is

$$Q_{post-condenser} = \dot{m}_{w,pc} c_{p,pc} (T_{OUT,pc} - T_{IN,pc})$$

$$(2.17)$$

where $c_{p,pc}$ is the specific heat of water, at the average temperature between the inlet and the outlet of the condenser. Knowing the heat provided by the electrical resistances Q_{BC} to vaporize the liquid in the boiling chamber, it is possible to extrapolate the specific heat flux along the specimen thought the thermal balance

$$q_{cooling} = \frac{Q_{BC} - Q_{post-condenser}}{A} \tag{2.18}$$

Agreement between the three methods to evaluate the removed heat flux has been checked during the tests and an example obtained with a vapor mass flow rate equal to $2 \text{ kg/(m}^2 \text{ s})$ is reported below.



Fig. 2.19 – Heat flux removed from the specimen versus mean logarithmic temperature difference between the saturated vapor and the sample surface. Data refer to a steam mass velocities $\dot{m}_{vap} = 2 [kg/(m^2 s)]$.

The agreement between the thermopile and the thermocouples method is very accurate, there is a variance lower than 6% and thus they are used to evaluate all the parameters in this thesis work. The third one is less accurate because it is not a direct measurement of the heat flux, it is used only to preliminary check the system thermal balance and for this reason it is not used to extrapolate data.

2.3.3 – Specific vapor mass flow rate

Knowing the temperature and the pressure of the liquid at the entrance of the boiling chamber, it is possible to calculate the enthalpy of the subcooled liquid at the evaporator inlet, as well as the latent heat of vaporization of water using REFPROP version 9.1. Measuring the heat supplied to the boiling chamber for vaporization of the liquid, it is possible to assess the mass flow of the fluid within the system as

$$\dot{m} = \frac{Q_{BC}}{h_v - h_{ls}} \tag{2.19}$$

where h_v and h_{ls} are respectively the enthalpies of saturated steam at the exit of the boiling chamber and of the subcooled liquid at the entrance of the boiler. It is then possible to calculate the specific vapor mass flow rate flowing in the test section as

$$G_{\nu} = \frac{\dot{m}}{s_c} \tag{2.20}$$

where S_c is the cross section of the channel, in our case $S_c = 0.005 * 0.030$ m. Acting on the electrical power supplied to the heating elements placed in the boiling chamber Q_{BC} it is thus possible to adjust the mass flow rate of the fluid flowing within the system, so it is possible to carry out tests with different specific steam mass flow tests.

2.3.4 – Outlet steam quality

This parameter is fundamental for the theoretical evaluation of the two-phase pressure gradient along the specimen, therefore for the calculation of α_{ss} (see Eq. (1.27) and followings). At the inlet it is supposed that $x_{in}=1$ and at the outlet x is evaluated imposing the test section thermal balance, as follows

$$Q = \alpha_{lab} \Delta T_{ml} A \tag{2.21}$$

$$h_{out} = h_{vap} - \frac{Q}{G} \tag{2.22}$$

where α_{lab} is the HTC experimentally calculated, ΔT_{ml} mean logarithmic temperature difference still experimentally calculated, A is the area of the channel and h_{vap} the vapor enthalpy. Once h_{out} , moisture outlet enthalpy, is evaluated x_{out} , the outlet vapor quality, can be calculated.

The vapor quality that used in Eq. (1.37) is the average mean between x_{out} and x_{in} .

2.4 – Experimental apparatus uncertainty

According to ISO Guide (1999) the uncertainty components are grouped into two categories "A" and "B", depending on the method of evaluation. Type A uncertainty obtained from a probability density function derived from the observed frequency distribution. The uncertainty of type B is obtained through a probability density function assumed on the basis of the degree of confidence in the occurrence of an event, and therefore it is estimated a priori on the basis of objective considerations. The combined uncertainty is used for results obtained from the values of other variables, and is indicated with u_c : this is the standard deviation associated with the result. Finally, the expanded uncertainty is obtained multiplying the combined uncertainty by a coverage factor. The purpose of it is to provide an interval in the neighborhood of the measurement result, which is expected to include the major portion of the distribution of values attributable to the measured value.

As regards the type A uncertainty, the best estimate of the expectation values of a quantity q, of which were made n independent observations q_k , all in the same experimental conditions, is the arithmetic mean or average value \overline{q} of the n observations

$$\bar{q} = \frac{1}{n} \sum_{k=1}^{n} q_k \tag{2.23}$$

Therefore for an input variable estimated by independent repeated observations, the arithmetic mean obtained from Eq. (2.23) is used as an estimate of the input. The individual observations differ due to random variations of the quantities of influence. The experimental variance of the observations is given by

$$s^{2}(q_{k}) = \frac{1}{n-1} \sum_{k=1}^{n} (q_{k} - \bar{q})^{2}$$
(2.24)

The experimental standard deviation of the mean is

$$\sigma^2(\bar{q}) = \frac{s^2(q_k)}{n} \tag{2.25}$$

And it is used as A type uncertainty of \bar{q} .

Regarding the type B uncertainty, which has not been obtained from repeated observations, uncertainty or the estimated variance are evaluated by scientific judgment based on all the information available on the variability of the input. These can be obtained from

- Data from previous measurements.
- The manufacturer's specifications.
- Data provided in calibration certificates.
- Uncertainties assigned to reference data taken from handbooks.

If a measurement has both type B uncertainty (u_b) and type A uncertainty (u_a) , the total uncertainty is determined by the formula

$$u_{tot} = \sqrt{u_a^2 + u_b^2} \tag{2.26}$$

Considering the case of input independent variables x_i , the uncertainty $u_c(y)$ of a derivate quantity y is obtained by the following formula

$$u_{c}(y) = \sqrt{\sum_{i=1}^{n} \left(\frac{df}{dx_{i}}\right)^{2} u^{2}(x_{i})}$$
(2.27)

where $u(x_i)$ is the uncertainty of the inlet quantity and f is the relating function

$$y = f(x_1, x_2, \dots, x_n)$$
(2.28)

Once the combined uncertainty is calculated, it is possible to calculate the extended uncertainty as

$$U = ku_c(y) \tag{2.29}$$

The coverage factor k is chosen according to the desired degree of confidence. For example, if we assume a Gaussian configuration, k refers to the following values:

- With a degree of confidence of 68.3% it is k=1.
- With a confidence level of 95.5% it is k=2.
- With a confidence level of 99.7% it is k=3.

The measures presented in this paper have all been subjected to analysis of the combined uncertainty. In particular the average uncertainty of the ΔT_{ml} between the vapor temperature and the specimen surface temperature is less than 1.3%; while the average uncertainty of the heat transfer coefficient is less than 8.2% referring to a confidence level equal to 95.5%.

Nomenclature

 $A = \text{area, m}^2$

- c_p = specific heat capacity, J kg⁻¹ K⁻¹
- D_h = hydraulic diameter, m
- F =Coriolis force, N
- G_v = specific mass flow rate, kg m⁻² s⁻¹
- HTC = heat transfer coefficient, W m⁻² K⁻¹
- $h = enthalpy, J kg^{-1}$
- k = coverage factor
- $\dot{m} = \text{mass flux}, \text{kg s}^{-1}$
- n = independent observations, -
- p =pressure, Pa
- Q = heat exchanged, W
- $q = \text{heat flux, W m}^{-2}$
- S_c = cross section of the channel, m²
- s = experimental variance
- T =temperature, K
- u_a = A-type uncertainty
- $u_b =$ B-type uncertainty

 u_c = combined uncertainty

- v = velocity, m s⁻¹
- x = vapor quality, -

- y = perpendicular axis, m
- z =longitudinal axis, m

Greek symbols

 α_{lab} = heat transfer coefficient experimentally measured, W m⁻² K⁻¹

- ΔT = temperature difference, K
- $\Delta T_{m/}$ = logarithmic mean temperature difference, K

 λ = thermal conductivity, W m⁻¹ K⁻¹

- ξ = instrument uncertainty
- σ = standard deviation
- ω = angular velocity, rad s⁻¹

Subscripts

- ' = correspondence to 1 mm depth respect to the specimen surface
- " = correspondence to 2.75 mm depth respect to the specimen surface

Al = aluminum

- *ave* = *average*
- *atm*= atmosphere
- BC = boiling chamber
- IN = inlet

OUT =outlet

pc = post condenser

PEEK, thermoplastic material

SAT = saturation

SS = shear stress

WALL = surface

vap = vapor

w = water

Chapter 3

The specimen

Three type of surfaces have been investigated in this Thesis:

- Superhydrophilic surfaces.
- Hydrophobic surfaces.
- Superhydrophobic surfaces.

The purpose of developing different treatments over a specimen, to obtain the aforementioned surfaces, is to study how wetting properties influence the condensation mode and performance. Processes are developed starting from the results presented in Bisetto *et al.* (2014).Surfaces wetting properties are investigated by means of contact angles analysis, while surface morphology is visually studied by digital field emission Scanning Electron Microscope (SEM).

SEM is mostly used for scanning the surface of a sample by means of an electron beam which is scanned over the surface of the specimen. This must be electrically conductive, in order to avoid surface charging which causes instability and decreases the resolution. The interaction of the electron beam with the atoms produces signals that are detected by a detector and the varying intensity of the signal is reproduced as an image on the screen. The following figures refer to an untreated aluminum sample.



Fig. 3.2 – *Photo of a flat polished aluminum specimen.*



Fig. 3.3 – SEM image of a flat polished aluminum specimen at different magnifications (1000x, 10000x, 20000x and 80000x).

Hence will be presented chemical processes used to change surface wettability and the wetting properties characterization.

3.1 – Materials

A high purity (AW 1050, minimum Al quantity 99.50%) aluminum plate was used for specimen fabrication.

Iron(III) Cloride (reagent grade 97%), Hydrogen Peroxide (30% w/w solution), Hexane(anhydrous, 95%), Tetrahydrofuran (≥99.9%) and 1H,1H,2H,2H-Perfluorooctyltriethoxysilane (98%) were purchased by Sigma Aldrich.

Sodium Hydroxide pellets and Polydimethylsiloxane (Sylgrad 184) were obtained by Merck and Dow Corning, respectively.

3.2 – Chemical treatments

Hence a detailed review of the treatments, that a sample undergoes, is exposed.

3.2.1 – Superhydrophilic specimen

The superhydrophilic specimen is mainly obtained etching the flat aluminium surface through different corrosive solutions.

Firstly the sample has to be cleaned, therefore it is immersed for 15' into Isopropanole (IPA), continuously stirring the solution through a sonication probe (Fig. 3.4), and dried into a nitrogen stream.



Fig. 3.4 – Sample immersed in IPA and stirred with the sonic probe.

After cleaning, the sample is immersed in a NaOH (1% w/w) aqueous solution for 15', to get rid of the superficial aluminum oxide (Fig. 3.5). This process is slightly corrosive as it can see from bubbling in Fig. 3.4 and it starts to develop an hierarchical structure on the surface. When extracting the sample from the solution this is dipped in IPA and deionized (DI) water to stop the reaction. In between the sample is dried with N_2 .



Fig. 3.5 – Sample immersed in the NaOH (1% w/w) aqueous solution and stirred with a sonic probe.

The final step is necessary for reaching the proper superficial roughness. The sample is immersed in a $FeCl_3$ – water (1 mol/L) corrosive solution for 7.5' (Fig. 3.6), and subsequently dipped into IPA and DI water before drying with nitrogen. During the immersion time the sample is etched achieving the desired morphology.



Fig. 3.6 – *Sample immersed in FeCl*₃ – *water (1 mol/L).*

Finally, to control the oxidation that otherwise will occur randomly in open atmosphere, the metallic specimen is immersed for 30' into a H_2O_2 (30% w/w) aqueous solution (Fig. 3.7), cleaned with DI water and again dried in N2.



Fig. 3.7 – *Sample immersed in* H_2O_2 (30% w/w).

The following figures show SEM analysis of the superhydrophilic roughened surface.



Fig. 3.8 – *SEM images of a superhydrophilic specimen, at different magnifications (1000x, 10000x, 20000x and 80000x).*

3.2.2 – Hydrophobic specimen

To realize an hydrophobic surface it is necessary to reduce the surface free energy and can be done by depositing a Self-Assembled Monolayer (SAM) over it. In this case, the process consists in spin-coating a n-Hexane – Perfluorooctyltriethoxysilane (FOTS) mixture (5% by volume) onto the sample at 800 rpm for 30 seconds. After the spin coating process, the sample is baked at 150°C for 30 minutes for final solvent evaporation and SAM stabilization (Fig. 3.9).



Fig. 3.9 – Specimen baked on an electrical heater.

Moreover, since the layer is extremely thin (few nanometers) and uniform, no appreciable differences on the morphology of the sample could be seen.

3.2.3 – Superhydrophobic specimen

Superhydrophobic surfaces can be produced by combining two factors: proper surface roughness and low surface free energy. These two elements allow water drops to sit over the surface with a quasi-spherical shape and to easily roll-off from it. The process herein used can be divided in three phases: the first phase consists in a controlled corrosion of the substrate, in order to obtain the proper superficial roughness, the second one is used to lower surface free energy and the third one is used to form a protective layer over the surface, in order to increase its stability in aggressive environments.

Proper morphology is obtained through the cleaning, de-oxidation, etching ad re-oxidation steps used for getting the superhydrophilic properties, as presented in Paragraph 3.2.1. After that, surface free energy is reduced forming a SAM over the substrate by immersing it in a FOTS-hexane (0.05% by volume) solution. Functionalization takes place at low temperature (7°C) for 1 hour and 45 minutes (Fig. 3.10). After that, the sample is rinsed in hexane and baked at 120°C for 45 minutes.



Fig. 3.10 – Specimen immersed in a n-Hexane –(FOTS) mixture (0.05% by volume) solution.

After the second step the sample is superhydrophobic, presenting very high advancing contact angle and very low contact angle hysteresis. However, it has been decided to introduce a third step, with the goal to enhance the resistance of the superhydrophobic properties in the presence of condensing vapor.

The superhydrophobic aluminum sample has been treated by dip coating it (Fig. 3.11) in a PDMS – FOTS – THF (Tetrahydrofuran) solution. The solution is prepared by combining 50 mg of PDMS (+10% curing agent) and 100 μ L of FOTS for each 10 mL of THF. The sample is dip coated inside the solution at 100 μ m/s before curing at 135°C for 30 minutes.



Fig. 3.11 – Specimen during the dip coating process. On the left side the image of the machine used for the process. On the right side an enlargement of the process where it can be seen the zone where the solution evaporates.

The following figures show SEM analysis of the superhydrophobic surface.



Fig. 3.12 – SEM images of a superhydrophobic specimen, at different magnifications (1000x, 5000x, 10000x and 20000x).

3.3 – Surface characterization

Surface wetting properties were analyzed by means of contact angles analysis. A setup for contact angles measurement has been organized at the Two-Phase Heat Transfer Laboratory of the Department of Industrial Engineering of Padova University. This consists of a compact CMOS camera (Thorlsabs GmbH ® DCC1545M) combined with a Thorlabs GmbH ® MVL7000 zoom lens and a LED light (Fig. 3.13). Typical spatial and time resolution of the system are 5 μ m/pixel and 30 fps respectively. Contact angles are measured using the standard sessile drop method, recording a water drop (backlight illuminated by the LED) expanding and contracting quasi-statically over the horizontally oriented surface of interest.



Fig. 3.13 – Setup for contact angles measurement. From the right to the left: CMOS camera (Thorlsabs GmbH ® DCC1545M) combined with a Thorlabs GmbH ® MVL7000 zoom lens, sample holder and the LED light.

Advancing (θ_a) and receding (θ_r) contact angles are evaluated by post-processing the videos fitting with a circle the drop profile near the contact point. Figure 3.14 shows an example of contact angle measurement. Contact angle hysteresis $(\Delta \theta)$ is calculated as the difference between θ_a and θ_r . Values of the contact angles reported in this Thesis are the average between at least five measurements for each sample in each condition, and the corresponding standard deviation is reported as experimental uncertainty.



Fig. 3.14 – Example of contact angle characterization. As a beginning, a frame of the moving drop is extracted from the video (a). Then the circle fitting the droplet profile near the contact point is created (b). Subsequently, the angle between the circle radius and the contact line is measured via software (c). Finally, the contact angle is geometrically evaluated (d).

The following chart reassumes the principal parameters for each type of surface.



Hydrophobic surface

Superhydrophobic surface



3.4 – Effect of surface temperature on the contact angles

Since superhydrophobic surfaces are investigated as DWC promoters, they are supposed to work at temperatures which could be different to ambient one, at which contact angles are usually evaluated. Thus, in the present work, influence of surface temperature on the wetting properties of the substrate is evaluated, by performing contact angle measurements at different sample superficial temperature. The superhydrophobic treated sample is heated up by means of an electrical heater at different superficial temperatures, while measuring contact angles as previously introduced. Fig. 3.15 shows the results of this analysis.



Fig. 3.15 - Advancing contact angle, receding contact angle and contact angle hysteresis as a function of wall superficial temperature. Tendency lines are reported.

It can be seen that the advancing contact angle is not influenced by the superficial temperature of the substrate, while the receding one decreases when increasing T_{WALL} . This leads to an augmentation of contact angle hysteresis, thus to a reduction of droplets mobility, the more the higher the wall temperature. This analysis underlines that in practical condensation conditions, as for example those achieved inside the experimental setup herein described (where vapor condenses at $T_{SAT} > 100^{\circ}$ C) wetting properties of the

surface are different than those evaluated in environmental conditions. Furthermore, since the surface specimen during the two-phase tests is never completely dried, the contact angles have been studied also maintaining the surface partially wetted to reproduce closely as possible the conditions achieved in the experimental section. The results are reported in Fig. 3.16.



Fig. 3.16 – Advancing contact angle, receding contact angle and contact angle hysteresis as a function of wall superficial temperature. Tendency lines are reported.

Comparing Fig. 3.15 and Fig. 3.16 is evident that the presence of liquid on the surface worsens the contact angles, thus it reduces the droplet mobility.

Nomenclature

T = temperature, K, °C

Greek symbols

 $\Delta\theta$ = contact angle hysteresis, °

 θ_a = advancing contact angle, °

 θ_r = receding contact angle, °

Subscripts

SAT = saturation

WALL = surface

Chapter 4

Experimental data

In this chapter experimental data acquired during the two-phase heat transfer over the treated aluminum surfaces will be presented. In particular, the three types of surfaces, characterized as shown in Chapter 3, are analyzed:

- Superhydrophilic surfaces.
- Hydrophobic surfaces.
- Superhydrophobic surfaces.

These data, moreover, will be compared with the results achieved over aluminum untreated surfaces.

4.1 – Test conditions

The tests have been carried out at different vapor flow rates, maintained constant during each test, by varying the temperature of the inlet cooling water. Typically the range of the cooling water temperature is from 10°C to 85°C. The variation of the temperature of the refrigerant leads to a variation of the sample surface temperature, thus the logarithmic mean temperature difference between the steam saturation temperature and the aluminum wall temperature is varied. The steam mass flux depends on the heat supplied to the fluid in the boiling chamber, then it is regulated by adjusting the electrical power supplied to the heating elements in the evaporator.

Data are acquired at a saturation temperature between 104°C and 107°C, which corresponds to a saturation pressure between 1.2 bar and 1.3 bar. The steam flux values used during experimental tests are the following:

- $\dot{m}_{vap} = 1.6 \frac{\text{kg}}{\text{m}^2\text{s}}$, that corresponds to $u_{vap} = 2 \frac{\text{m}}{\text{s}}$.
- $\dot{m}_{vap} = 3.2 \text{ kg/}_{\text{m}^2\text{s}}$, that corresponds to $u_{vap} = 4.6 \text{ m/}_{\text{s}}$.

- $\dot{m}_{vap} = 4.8 \text{ kg/}_{\text{m}^2\text{s}}$, that corresponds to $u_{vap} = 6.4 \text{ m/}_{\text{s}}$.
- $\dot{m}_{vap} = 5.1 \text{ kg/}_{\text{m}^2\text{s}}$, that corresponds to $u_{vap} = 7.2 \text{ m/}_{\text{s}}$.
- $\dot{m}_{vap} = 6.4 \text{ kg/}_{\text{m}^2\text{s}}$, that corresponds to $u_{vap} = 8.5 \text{ m/}_{\text{s}}$.

Where $\dot{m}_{vap} = 4.8 \text{ kg}/_{\text{m}^2\text{s}}$ and $\dot{m}_{vap} = 6.4 \text{ kg}/_{\text{m}^2\text{s}}$ were used only for the tests on the superhydrophilic specimen, meanwhile $\dot{m}_{vap} = 5.1 \text{ kg}/_{\text{m}^2\text{s}}$ was used just only for the tests on the hydrophobic one. As regards the boundary conditions, the input data more significant for the tests are:

- The power supplied to the boiling chamber.
- The saturation temperature of the steam.
- The mass flow rate of the water in the primary cooling system.
- The temperature of the inlet cooling water to the experimental section.

4.2 – Superhydrophilic specimen

Figure 4.1 reports the measured heat flux as a function of the wall subcooling degree, at different mass fluxes for the superhydrophilic treated specimen.



Fig. 4.1 – Heat flux versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the superhydrophilic sample. Data refer to different steam mass velocities \dot{m}_{vap} [kg/(m² s)]. Uncertainty bars are reported.

At a fixed vapor flux, the heat flux increases with the increase of the ΔT_{ml} , similarly, at a fixed ΔT_{ml} the heat flux increases with the increase of the mass flow rate. Trends are those obtained also for untreated samples, being the condensation mode still filmwise. Fig. 4.2 shows the heat transfer coefficient as a function of the wall subcooling degree.



Fig. 4.2 – HTC versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the superhydrophilic sample. Data refer to different steam mass velocities \dot{m}_{vap} [kg/(m² s)]. Uncertainty bars are reported.

At a fixed vapor flux, the HTC increases with the decrease of the ΔT_{ml} , meanwhile at a fixed ΔT_{ml} the heat flux increases while increasing of the mass flow rate. This behavior is typical of the FWC mode. The summary of the data acquired on the superhydrophilic sample is reported in Appendix A.

It is interesting to compare HTCs obtained on a superhydrophilic specimen and those achieved on an untreated one. This comparison is exposed in Fig. 4.3.



Fig. 4.3 – Ratio between HTC on the superhydrophilic specimen and the one on the untreated sample versus mean logarithmic temperature difference between the saturated vapor and the surface. Data refer to different steam mass velocities \dot{m}_{vap} [kg/(m² s)].

Figure 4.3 shows the ratio between the heat transfer coefficient obtained with the superhydrophilic and the untreated sample for two steam mass velocities (1.6 kg/(m² s) and 6.4 kg/(m² s), respectively) as a function of the mean logarithmic temperature difference between the steam and the wall. The comparison between the untreated and the superhydrophilic sample shows that the roughened surface presents lower heat transfer coefficient, thus lower heat flux, at the same \dot{m}_{vap} and ΔT_{ml} . This is due to the different wettability of the two samples. On the superhydrophilic specimen, the condensate is strongly attached on the substrate, thus gravity influence and shear stress influence on the liquid film thickness are reduced in comparison to a quasi-smooth surface. This is especially true at low vapor mass velocities, when the gravity force is dominant. When increasing the steam velocity, the shear stress becomes more relevant and it is able to compensate the negative effect of the high adhesion forces.

From this analysis it is possible to infer that condensation over a roughened surface which presents high wettability is less efficient that the process on a quasi-smooth surface.

4.2.1 – Analysis of the local HTCs

Fig. 4.3 shows that a superhydrophilic surface leads to lower heat transfer coefficients with respect to an untreated one. A further analysis of the local HTCs is necessary to understand why this happens and this is possible thanks to the four thermocouples within the specimen. The local superficial temperatures of the sample are experimentally calculated in two different point, thus it is possible to determine two local heat transfer coefficients. A pair of thermocouples is located at 2 mm from the top of the specimen at different depths, from now on they define the INLET parameters; the other pair is located at 2 mm from the bottom of the specimen at the same different depths and from now on they define the Same different depths and from now on they define the CUTLET parameters. The HTCs calculated over an superhydrophilic specimen are compared with those achieved over an untreated sample at the following vapor mass flow rates: 1.6 kg/(m^2 s) , 3.1 kg/(m^2 s) and 5.1 kg/(m^2 s) .



Fig. 4.4 – Ratio between the local HTC measured over superhydrophilic specimen and the corresponding one measured over the untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to \dot{m}_{vap} =1.6 kg/(m² s).



Fig. 4.5 – Ratio between the local HTC measured over superhydrophilic specimen and the corresponding one measured over the untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to \dot{m}_{vap} =3.1 kg/(m² s).



Fig. 4.6 – Ratio between the local HTC measured over superhydrophilic specimen and the corresponding one measured over the untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to \dot{m}_{vap} =5.1 kg/(m² s).

The previous figures shows that $\frac{HTC_{treated}}{HTC_{untreated}}$ ratio stays around a fixed value, equal to about 0.8, for all the vapor mass flow rates. The trend of HTC for each \dot{m}_{vap} is to

lightly decrease with the increasing of ΔT due to the increasing of the condensate layer thickness which adds more thermal resistance. Fig. 4.7 reassumes these results.



Fig. 4.7 – Ratio between the local HTCs measured over a superhydrophilic specimen and the corresponding one measured over an untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to different steam mass velocities \dot{m}_{vap} [kg/(m² s)].

Except for the INLET HTCs related to \dot{m}_{vap} =5.1 kg/(m² s), for which it seems that the vapor velocity manages to undo the roughness effect, the other parameters are quasi-constant. This could lead to think that the roughness influences equally the condensation process over the sample.

4.3 –Hydrophobic specimen

Figure 4.8 and figure 4.9 report the measured heat flux and the HTC as a function of the wall subcooling degree, at different vapor mass fluxes for the hydrophobic treated specimen.



Fig. 4.8 – Heat flux versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to different steam mass velocities \dot{m}_{vap} [kg/(m² s)]. Uncertainty bars are reported.



Fig. 4.9 – HTC versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to different steam mass velocities \dot{m}_{vap} [kg/(m² s)]. Uncertainty bars are reported.

The behavior is the same of the untreated sample, it is underlined that also on a hydrophobic surface a filmwise condensation is promoted. The summary of the data acquired on the hydrophobic sample is reported in Appendix B.

Next figure reports comparison between the heat transfer coefficients on the treated and the untreated samples.



Fig. 4.10 – *Ratio between the HTC measured over hydrophobic specimen and the one measured over the untreated sample versus mean logarithmic temperature difference between the saturated vapor and the surface. Data refer to different steam mass velocities* \dot{m}_{vap} [kg/(m² s)].

Figure 4.10 shows the ratio between the heat transfer coefficient obtained with the hydrophobic and the untreated sample for three steam mass velocities (1.5 kg/(m² s), 3.1 kg/(m² s) and 5.1 kg/(m² s), respectively) as a function of the mean logarithmic temperature difference between the steam and the wall. The comparison between the untreated and the hydrophobic sample shows that the untreated surface presents lower heat transfer coefficient, thus lower heat flux, at the same \dot{m}_{vap} and ΔT_{ml} . Two considerations could be done:

• $HTC_{treated}/HTC_{untreated}$ increases with the increasing of \dot{m}_{vap} , $HTC_{treated}$ from a

15% higher arrives to be 45% higher respect to $HTC_{untreated}$. The slip velocity seems to be strongly influenced by the velocity of the vapor.

• $HTC_{treated}/HTC_{untreated}$ decreases with the increasing of ΔT_{ml} at a fixed \dot{m}_{vap} .

The film of the condensate becoming thicker reduces the influence of the slip velocity, as it is reported in the literature.

4.3.1 – Analysis of the local HTCs

Hence a further analysis of the local HTCs is presented, they are compared to those achieved over an untreated sample to investigate how the slip velocity locally influence the heat exchange. In Paragraph 2.3.1 is reported how $T_{WALL,IN}$ and $T_{WALL,OUT}$ are calculated, thus it is possible to determine an INLET and an OUTLET heat transfer coefficient.



Fig. 4.11 – Ratio between the local HTC measured over hydrophobic specimen and the corresponding one measured over the untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to $\dot{m}_{vap} = 1.6 \text{ kg/(m}^2 \text{ s})$.


Fig. 4.12 – Ratio between the local HTC measured over hydrophobic specimen and the corresponding one measured over the untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to \dot{m}_{vap} =3.1 kg/(m² s).



Fig. 4.13 – Ratio between the local HTC measured over hydrophobic specimen and the corresponding one measured over the untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to $\dot{m}_{vap} = 5.1 \text{ kg/(m}^2 \text{ s})$.

Several considerations could be done from Fig. 4.11, Fig. 4.12 and Fig. 4.13:

• In Fig. 4.11 the data refer to $\dot{m}_{vap} = 1.6 \text{ kg/(m}^2 \text{ s})$. In this case, the shear stress induced by the vapor is negligible respect to the gravitational force. At the INLET, where the condensate layer is minimum, the $\frac{HTC_{treated}}{HTC_{untreated}}$ ratio is around 1.27, therefore slippery condition influences the filmwise condensation. As we move along the sample, the condensate layer increases in thickness, so at the OUTLET position there is a thicker liquid film respect to the INLET position. Here the $\frac{HTC_{treated}}{HTC_{untreated}}$ ratio is around 1, thus the slip factor no longer

influences the condensation.

• In Fig. 4.12 the data refer to $\dot{m}_{vap} = 3.1 \text{ kg/(m}^2 \text{ s})$, thus the condensate velocity is higher compared to the previous case. At the INLET the $\frac{HTC_{treated}}{HTC_{untreated}}$

ratio is around 1.51, meanwhile at the OUTLET position the ratio is about 1.12. Here the slip factor mainly affects the filmwise condensation respect to the previous case.

• In all the three cases, at the OUTLET position the HTCs tend to worsen with the increasing of the temperature difference between the saturated vapor and the surface. This happens because the condensate layer thickness increases with the increasing of ΔT , therefore the thermal resistance increases and thus the HTC worsens.

In Fig. 4.14 the $\frac{HTC_{treated}}{HTC_{untreated}}$ ratios at different mass flow rates are reassumed.



Fig. 4.14 – Ratio between the local HTCs measured over hydrophobic specimen and the corresponding one measured over the untreated sample versus the temperature difference between the saturated vapor and the surface. Data refer to different steam mass velocities \dot{m}_{vap} [kg/(m² s)].

The considerations previously done are summarized in Fig. 4.14:

• $HTC_{treated}/_{HTC_{untreated}}$ ratio increases with the increasing of the mass flow rate,

i.e. with the increasing of the velocity at the liquid-vapor interface.

• $HTC_{treated}/_{HTC_{untreated}}$ ratio decreases with the increasing of the temperature

difference between the saturated vapor and the surface, i.e. with the increasing of the condensate layer thickness.

Fig. 4.14 shows that
$$\left(\frac{HTC_{treated}}{HTC_{untreated}}\right)_{INLET}$$
 obtained with \dot{m}_{vap} =5.1 kg/(m² s) is lower than that obtained with \dot{m}_{vap} =3.1 kg/(m² s). In fact, the condensate thickness decreases with an increasing of \dot{m}_{vap} , therefore δ at the OUTLET tends to be closer to δ at

the INLET. This entails that the effect due to the thickness of the condensate layer influences in the same manner throughout the sample, as well as the effect due to the mass flow rate does. The sum of these two effects, thus, tend to be constant and this means having the same increment of HTC and a constant a slip factor along the specimen. This behavior gives as result an increasing of the mean HTC respect to an increasing of the mass flow rate (see Fig. 4.6).

4.4 – Superhydrophobic specimen

In Chapter 1 it was pointed out the transition from dropwise condensation mode to filmwise condensation mode. This process has been studied deeply since in the present analysis DWC over the superhydrophobic sample was promoted only for a limited period of time, over which condensation mode moved back to filmwise. Nevertheless, a few data was gained during DWC, and hereinafter they are discussed. Firstly, Fig. 4.15 presents a visual analysis of DWC phenomenon.



Fig. 4.15 – Droplets cycle on a superhydrophobic aluminum sample at about 10 K of surface subcooling and about 6.5 kg/(m² s) of vapor mass velocity. The arrow indicates the vapor flow direction.

The DWC cycle starts with nucleation of small drops, which form and grow over the surface ($\tau = 0$ s), and coalescence generating bigger droplets ($\tau = 0.07$ s). New drops continue to grow, due to continuous vapor condensation, eventually absorbing other small

droplets, until they reach a critical diameter ($\tau = 1.23$ s), over which they move from the surface because of combined gravity and shear stress forces ($\tau = 1.27$ s). The departing drops swept over the surface ($\tau = 1.27 - 1.44$ s), thereby wiping small droplets in their path. After the drop sweep, fresh drops start to nucleate ($\tau = 2.94$ s) and grow again, allowing the cyclic process to continue. As said, after a while, the filmwise condensation mode gradually substitutes the dropwise one as can be seen by analyzing Fig. 1.17. The transition between the two processes was acquired and it is presented in the following Figure, which shows the profiles of the four temperatures measured inside the specimen (named as in Paragraph 2.3.1) as a function of the acquisition time.



Fig. 4.16 – Temperature profiles inside the superhydrophobic specimen during the transition between dropwise and filmwise condensation mode. Saturation temperature is about 106°C, steam flows at about 3.2 $kg/(m^2 s)$.

When condensing in dropwise mode specimen temperatures are extremely high and close to each other, for the same orthogonal position, i.e. T'_{IN} is similar to T'_{OUT} and T''_{IN} is similar to T''_{OUT} . In Fig. 4.16 it is possible to notice a drastic reduction of the specimen temperatures, which is connected to the degradation of the condensation mode, at about 380 s. After a transition period of about 40 s the condensation mode stabilizes to filmwise, as it can be seen by the fact that the temperatures inside the specimen stabilize to lower

values than those before the transition. Moreover, after 420 s the specimen presents a high temperature gradient between the inlet and the outlet zones. This temperature variation is related to the presence of the condensate film over the surface, which is thicker the closer to the sample outlet, leading to a non-constant thermal resistance along the specimen which causes the temperature reduction. This is not the case when condensing in dropwise mode. Surface temperatures at the inlet and the outlet of the specimen are quite close to each other, leading to a smaller mean logarithmic temperature difference between the steam and the wall, thus to a higher heat transfer coefficient. This happens because the surface is continually wiped thanks to droplets motion, causing the superficial temperature of the specimen to be almost constant. Figure 4.17 reports the mean logarithmic temperature difference between the steam and the wall and the corresponding heat transfer coefficient during the transfer coefficient mode.



Fig. 4.17 - Mean logarithmic temperature difference between the steam and the wall (a) and heat transfer coefficient (b) as a function of the acquisition time during the transition between dropwise and filmwise condensation mode.

The Figure shows that when condensing in dropwise mode the steam-to-wall temperature difference is much lower than the one during FWC, which reflects in a much higher heat transfer coefficient. The mean value of HTC during DWC is $34 \text{ kW/(m}^2 \text{ K})$, meanwhile at the same conditions during FWC on a polished sample HTC is equal to $16 \text{ kW/(m}^2 \text{ K})$. Removed heat flux (Fig. 4.18) stays almost constant between the two condensation modes. This is because the main thermal resistance to the heat transport is located on the

convective and conduction side of the process, thus the heat flux is not substantially influenced by a variation of the two-phase heat transfer coefficient.



Fig. 4.18 – Specific heat flux as a function of the acquisition time during the transition between dropwise and filmwise condensation mode.

The superhydrophobic treated substrate was not able to sustain high-HTC dropwise condensation mode for more than a few minutes, over which a condensate film formed on the surface. This was not because of a degradation of the superhydrophobic properties of the sample, since when taken out from the setup it still presented high advancing contact angle and low contact angle hysteresis. The reason of the degradation of the condensation mode has been theoretically examined in Paragraph 1.2.5, but it is still not completely understood. In addition to that, after dropwise to filmwise transition occurs, the superhydrophobic surface performs as a superhydrophilic one, since the condensate layer is strongly pinned on the roughened surface. Thus, after the degradation of the condensation mode, the surface performs worse than an untreated one.

This analysis shows that there is no direct connection between contact angles and condensation mode. When condensing in the absence of air or other NCG, a surface with extremely high contact angles but non-adequate morphology will induce filmwise condensation mode, despite is high water repellency properties in open-atmosphere environments.

Nomenclature

HTC = heat transfer coefficient, W m⁻² K⁻¹

- \dot{m} = specific mass flow rate, kg m⁻² s⁻¹
- T =temperature, K

u = velocity, m s⁻¹

Greek symbols

 δ = thickness, m

- ΔT = temperature difference, K
- ΔT_{ml} = logarithmic mean temperature difference, K

 $\tau = time, s$

Subscripts

- ' = correspondence to 1 mm depth respect to the specimen surface
- " = correspondence to 2.75 mm depth respect to the specimen surface

IN = inlet

OUT = outlet

SAT = saturation

WALL = surface

vap = vapor

Chapter 5

Theoretical analysis

Hence the comparison between the models developed in Chapter 1 and the experimental data presented in Chapter 4 will be presented.

Firstly it is shown the theoretical analysis of the experimental data acquired on an untreated sample by Favaro C. (2014). The analysis has been done applying the Nusselt (1916) and the Carpenter and Colburn model (1951).



Fig. 5.1 – Calculated versus experimental heat transfer coefficient for the aluminum untreated specimen. Experimental uncertainties are also reported.

The majority of the data is in between $\pm 10\%$ error bands. However, with high vapor mass flow rate ($\dot{m}_{vap} = 5.1 \text{ kg/(m^2s)}$) the model tends to lose in accuracy, the corresponding

calculate HTCs stand in between -10% and -20% error bands. This happens because of the fact that the shear stress component becomes dominant, but this element is much more difficult to properly predict in comparison to gravitational one, since it includes two-phase frictional pressure losses.

Hereinafter it will be analyzed only the data acquired with vapor mass flow rates equal to $1.5 \text{ kg/(m}^2 \text{ s})$, $3.1 \text{ kg/(m}^2 \text{ s})$, $4.8 \text{ kg/(m}^2 \text{ s})$ for the superhydrophilic and $5.1 \text{ kg/(m}^2 \text{ s})$ for the hydrophobic since, as seen in Fig. 5.1, the models become less accurate with higher \dot{m}_{vap} .

5.1 – Superhydrophilic specimen

Over the aluminum superhydrophilic sample filmwise condensation is fully developed as reported in Paragraph 4.1, thus the model of FWC exposed in Paragraph 1.1 is applied. In the following figure the experimental data and calculated heat transfer coefficients for different vapor mass flow rates are compared. Uncertainty bars of the experimental data are also reported.



Fig. 5.2 – Calculated versus experimental HTC for the superhydrophilic specimen. Experimental uncertainties are also reported.

The theoretical model overestimates the experimental data and confirms that HTCs achieved over an untreated surface are higher than those achieved over and superhydrophilic one. Since the model was developed for an untreated surface, several considerations were made to best fit the experimental data. Following the evidences presented in Paragraph 4.2.1 analyzing the local HTCs and modifying the slip driven condensation model, a new theory has developed to describe the phenomenon. The concept of the slip factor β in this treatise could be transformed in a length where still the condensate velocity is equal to 0 (see Fig. 5.3). Considering the morphology of the superhydrophilic surface it could be supposed that the condensate strongly anchors over it (see Paragraph 3.3), thus the velocity of the condensate in this zone could be close to zero leading to lower HTCs. Therefore, the hydrophobic model is used to estimate the condensation parameters of a superhydrophilic surface by setting the slip factor equal to the sample superficial roughness.



Fig. 5.3 – Scheme of the condensate velocity along the perpendicular (y) axis of the specimen. In the left side the slip driven theory is reported, in the right side the same theory is applied to a superhydrophilic surface. The condensate velocity is supposed equal to 0 for the β length.

The following calculated data are obtained with these new considerations. The β value is set equal to 10 µm, a reasonable value for this kind of treatments (see Fig. 4.7).



Fig. 5.4 – Experimental and calculated HTCs versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to $\dot{m}_{vap=}1.6$ kg/(m² s). Experimental uncertainties are also reported.



Fig. 5.5 – Experimental and calculated HTCs versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to $\dot{m}_{vap}=3.1$ kg/(m² s). Experimental uncertainties are also reported.



Fig. 5.6 – Experimental and calculated HTCs versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to $\dot{m}_{vap}=4.8$ kg/(m² s). Experimental uncertainties are also reported.

The modified FWC model predicts the data well for low vapor flow rates, 1.5 and 3.1 $kg/(m^2 s)$, meanwhile for high vapor flow rate, 4.8 $kg/(m^2 s)$, the model tends to underestimate the heat transfer coefficient. As Fig. 4.7 shows, at high vapor mass flow rate the INLET HTC assumes a different behavior with respect to the others, probably because the higher vapor velocity influences also the condensate which is trapped in between the surface morphology. The results are reassumed in Fig. 5.7.



Fig. 5.7 – Calculated versus experimental HTC for the superhydrophilic specimen. Experimental uncertainties are also reported.

The model predicts the experimental results for $\dot{m}_{vap} = 1.6 \text{ kg/(m^2s)}$ and $\dot{m}_{vap} = 3.1 \text{ kg/(m^2s)}$ within $\pm 10\%$ error band. On the contrary, the highest \dot{m}_{vap} is underestimated. The hypothesis of a zone, corresponding to the superficial roughness where the condensate has a velocity equal to 0, could be a first step for the understanding of the phenomenon.

5.2 – Hydrophobic specimen

The aluminum treated hydrophobic specimen promotes filmwise condensation mode. However, if HTCs are paragoned with those obtained over an untreated sample (Fig. 4.10), higher performances are achieved, the more the higher is the vapor mass flow rate. Experimental data were firstly compared against FWC classic model. Results of this analysis are shown in Fig. 5.8.



Fig. 5.8 – Calculated versus experimental heat transfer coefficient for the hydrophobic specimen. Experimental uncertainties are also reported.

The model tends to underestimates the experimental data, which fall outside $\pm 10\%$ error bands, reaching a mean deviation equal to 22%. The behavior of the model is completely different if it is compared with Fig. 5.1, where the experimental data are well predicted for an untreated surface. Classic filmwise condensation theory is, thus, no longer adapt to describe the phenomenon. The enhancement of the condensation heat transfer coefficient is well explained if a slip velocity at the solid-liquid interface is introduced as it can be found in literature (Al-Jarrah *et al.*, 2008 and Pati *et al.*, 2013). The authors showed that a non-

zero slip length reflects on a non-zero slip velocity at the solid-liquid interface, leading to reduce condensate thickness and thus the increment of HTC. However, to the best of the author's knowledge experimental data have not ever gained for this mode of condensation and, furthermore, have not ever investigated the factors from which β depends. In fact, models in literature introduce a constant slip factor, meanwhile analyzing the experimental data (see Paragraph 4.3.1) it could be inferred that β is not constant along the sample and it depends from two factor: interfacial liquid-vapor velocity and condensate layer thickness. In Paragraph 1.3 it is reported the model that hereinafter is used to analyze the experimental data.

Firstly, it is shown how the slip factor and, thus, the slip velocity are calculated. Once u_s is determined, it is inserted in Eq. 1.60 and, thus, the heat transfer coefficient is obtained.



Fig. 5.9 – Slip factor β along the longitudinal axis of the specimen. Data refers to different steam mass flow rate and a cooling inlet water temperature equal to 25°C.

The slip coefficient increases with the increasing of \dot{m}_{vap} . In fact, this corresponds to an increasing of the shear stress phenomenon, thus to an increasing of the interfacial liquid-vapor velocity. For a fixed vapor mass flow rate, β decreases along the specimen and this happens because the increasing of the condensate thickness. Moreover, the β values assumed in the model are in between 11 µm and 2 µm, order of magnitude that is found in literature.

In the following figure the slip velocity values are shown.



Fig. 5.10 – Slip velocity u_s along the longitudinal axis of the specimen. Data refers to different steam mass flow rate and a cooling inlet water temperature equal to 25°C.

Once u_s is calculated, all the other parameters of the condensate layer are calculated by the model. The condensate layer thickness and the interface liquid-vapor velocity are shown in the following figures.



Fig. 5.11 – Interfacial liquid-vapor velocity along the longitudinal axis of the specimen. Data refers to different steam mass flow rate (a) $\dot{m}_{vap} = 1.6 \text{ kg/(m^2s)}$, b) $\dot{m}_{vap} = 3.1 \text{ kg/(m^2s)}$, c) $\dot{m}_{vap} = 5.1 \text{ kg/(m^2s)}$) and a cooling inlet water temperature equal to 25°C.

In Fig. 5.11 is reported the model with and without the slip factor, it is seen that the $u_{interface}$ is lightly higher with β . It is underlined that the maximum value that u_s assumes for each vapor mass flow rate (see Fig. 5.11) is less than 15% of the value at the corresponding z coordinate and at the corresponding mass flow rate. This percentage is, also, reported in experimental data found in literature.



Fig. 5.12 – Condensate layer thickness along the longitudinal axis of the specimen. Data refers to different steam mass flow rate (a) $\dot{m}_{vap} = 1.6 \text{ kg/(m^2s)}, b)\dot{m}_{vap} = 3.1 \text{ kg/(m^2s)}, c)\dot{m}_{vap} = 5.1 \text{ kg/(m^2s)})$ and a cooling inlet water temperature equal to 25°C.

Fig. 5.12 shows how the model calculates the film thickness with and without the slip factor, in slippery condition the thickness is lower and this explains the higher HTC. It is highlighted that at the INLET the condensate thickness with β is more separated from the one without β respect the OUTLET and this explains why the slip condition influences more the INLET HTC.

Thus, the heat transfer coefficients are calculated for the different mass flow rate with the considerations aforementioned and they are shown below.



Fig. 5.13 – Experimental and calculated HTCs versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to 1.6 kg/(m² s) specific vapor mass flow rate. Experimental uncertainties are also reported.



Fig. 5.14 – Experimental and calculated HTCs versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to 3.1 kg/(m² s) specific vapor mass flow rate. Experimental uncertainties are also reported.



Fig. 5.15 – Experimental and calculated HTCs versus mean logarithmic temperature difference between the saturated vapor and the surface during condensation over the hydrophobic sample. Data refer to 5.1 kg/(m² s) specific vapor mass flow rate. Experimental uncertainties are also reported.

As Fig. 5.13 and Fig. 5.14 evidence, the HTCs calculated are within the experimental uncertainty, meanwhile in Fig. 5.15, where the data refer to the highest mass flow rate, the model is less accurate. With the new model a mean deviation of 8.8% is reached, meanwhile with the classic model applied to the untreated specimen we have 7.8%.



Fig. 5.16 – Calculated versus experimental heat transfer coefficient for the hydrophobic specimen with the slip driven condensation model. Experimental uncertainties are also reported.

Fig. 5.16 shows that by introducing a non-zero slip velocity at the solid-liquid interface the tendency of the heat transfer coefficient can be satisfactory predicted.

By analyzing results provided by the model the reason of enhanced heat exchange can be found. When considering $u_s \neq 0$ the thickness of the condensate layer at the wall, thus the associated thermal resistance, decreases, due to the enhanced velocity gradient at the surface. However, this phenomenon needs further investigation.

Nomenclature

HTC = heat transfer coefficient, W m⁻² K⁻¹

- \dot{m} = specific mass flow rate, kg m⁻² s⁻¹
- T =temperature, K
- u = velocity, m s⁻¹
- y = perpendicular axis, m
- z =longitudinal axis, m

Greek symbols

- $\beta =$ slip factor, m
- δ = thickness, m
- ΔT = temperature difference, K
- ΔT_{ml} = logarithmic mean temperature difference, K

Subscripts

- calc = calculated
- exp = experimental
- S = slip
- SAT = saturation
- *WALL* = surface

vap = vapor

Concluding remarks

The purpose of this thesis is to study how the wetting properties of a material influence the two-phase heat transfer coefficient during condensation. The tested material is aluminum, since it is a material widely used in heat transfer industry.

Firstly, substrate wettability has been modified by proper chemical treatments. The procedure can be summarized in three steps: an etching process, where a proper roughness is induced on the polished surface, a functionalization process, where hydrophobic layer (FOTS) is deposited over the surface, and a coating process, where a protective layer is deposited over the surface. After the first step the substrate becomes superhydrophilic ($\theta_a = 12.3^\circ$, $\theta_r = 5.2^\circ$), while after the three steps the specimen becomes superhydrophobic ($\theta_a = 152.4^\circ$, $\theta_r = 148.3^\circ$). Moreover, if only the second step is applied over an untreated quasi-smooth sample hydrophobic properties are achieved ($\theta_a = 143.5^\circ$, $\theta_r = 43^\circ$).

Over both the superhydrophobic and the hydrophobic treated samples dropwise condensation is expected, meanwhile over the superhydrophilic one filmwise condensation should be promoted. Several tests were made on the three surfaces in the experimental apparatus, existing at the Two-Phase Heat Transfer Laboratory of the Department of Industrial Engineering of Padova University, and developed for steam condensation analysis, by varying the steam mass flow rate and the saturation to wall temperature difference.

The superhydrophilic sample promotes filmwise condensation and the high wetting properties of the substrate lead to lower heat transfer coefficient in comparison to an untreated sample; the relative decrease is higher at lower steam velocity.

The superhydrophobic sample does not promote a stationary dropwise condensation process, the phenomenon persists for few minutes then a transition to filmwise condensation occurs. The transition was investigated and analyzed since the specimen maintains his superhydrophobic properties once the test is ended. The first explanation for the process instability is that the low wetting property is a necessary but not a sufficient condition to promote dropwise condensation: in fact, a proper morphology configuration has to be reached. Secondly, a critical heat flux is introduced to explain why a superhydrophobic specimen does not promote dropwise condensation, i.e. there is a

maximum heat flow which can be transferred for a given surface beyond which the transition happens. In literature this transition is widely studied for copper surfaces, but it is still not clear for aluminum one. Different researches bond the critical heat flux with the conductivity of the material, meaning that the higher is the conductivity the higher is the critical heat flux. The phenomenon is still not completely understood and it needs further investigation. However, the heat transfer coefficient measured during dropwise condensation is double as compared to a polished sample at the same mean logarithmic temperature difference.

The hydrophobic sample does not promote dropwise condensation either. Moreover, the calculated heat transfer coefficients were compared to those achieved over an untreated sample and it results that an enhanced filmwise condensation happens. The HTCs resulted, with the increasing of the vapor mass flow rate, to be from 15% to 45% higher than those achieved over an untreated surface. Therefore a new model for this enhanced filmwise condensation were developed, since the classic model of Nusselt and Carpenter-Colburn can no longer be used to calculate the new heat transfer coefficients. A slip factor, as Navier suggested, has been introduced in the model to describe the slippery phenomenon at the solid-liquid interface. The slip boundary is experimentally analyzed in the literature, but only in single-phase flow and two dependences were found: the film thickness and the velocity of the flow. Slip-condensation over hydrophobic surfaces was thus modeled. Moreover, a model for superhydrophilic specimens was developed relating the slip coefficient to the substrate roughness. Both the models demonstrate a good agreement with the experimental data.

In conclusion, a new filmwise condensation mode has been experimentally and theoretically analyzed over an hydrophobic treated. Ongoing research is devoted to the analysis and the promotion of the dropwise condensation phenomenon.

Conclusioni

In questa tesi è stato presentato uno studio della condensazione su superfici nanostrutturate. Lo scopo è quello di capire come, al variare della bagnabilità di una superficie, venga influenzato il coefficiente dello scambio termico durante la condensazione. Il materiale preso in esame è l'alluminio essendo un materiale largamente impiegato nella realizzazione di scambiatori di calore.

Inizialmente è stata condotta un'ampia ricerca bibliografica per definire lo stato dell'arte sulla condensazione a film e a gocce su superfici verticali. Per quanto riguarda la condensazione a film, in questa tesi sono stati presentati il modello di Nusselt, che descrive la condensazione controllata dall'effetto della gravità, e il modello di Carpenter-Colburn, che descrive il processo di condensazione controllato dallo sforza tangenziale del vapore. Benché la condensazione a gocce sia conosciuta sino dagli anni '30 del secolo scorso, rimane tuttora un fenomeno non completamente compreso, dati i molteplici meccanismi che intervengono durante il processo. Sono stati poi presentati dei modelli che descrivono sia i meccanismi che lo scambio termico durante la condensazione a gocce, in particolare si ricorda il modello di Rose.

Il passo successivo è stato quello di modificare la bagnabilità di un campione in alluminio usando dei trattamenti chimici che possono essere riassunti in tre fasi principali: l'etching, con il quale si va a conferire una rugosità alla superficie a livello micro-nanometrico; la funzionalizzazione, attraverso la quale viene depositato un layer nanometrico idrofobico sul campione che quindi assume caratteristiche idrofobiche; il coating, con il quale viene depositato un layer protettivo sulla superficie, necessario per allungare la resistenza del trattamento. Per determinare la mobilità di una goccia su una superficie è necessario introdurre due parametri: l'angolo di contatto in avanzamento θ_a e l'angolo di contatto in recessione θ_r che determinano l'angolo tra interfaccia vapore-liquido e interfaccia liquidosolido di una goccia mentre viene fatta rispettivamente espandere e contrarre. Se un campione viene solamente immerso in una soluzione corrosiva (etching) questo assume caratteristiche superidrofiliche ($\theta_a = 12.3^\circ$, $\theta_r = 5.2^\circ$), mentre, se vengono applicati anche gli altri due processi, esso assume caratteristiche superidrofobiche ($\theta_a = 152.4^\circ$, $\theta_r = 148.3^\circ$). Un campione liscio sottoposto alla sola funzionalizzazione assume, invece, delle caratteristiche idrofobiche ($\theta_a = 143.5^\circ$, $\theta_r = 43^\circ$). Inoltre, è stato svolto un esperimento che prova come questi angoli, in particolare quello di recessione, varino al variare della temperatura della superficie del campione.

Una volta ottimizzati i processi necessari ad ottenere le tre tipologie di superfice, è stato possibile testarle nell'apparato sperimentale esistente nel laboratorio di scambio termico bifase dell'università di Padova e sviluppato appositamente per lo studio della condensazione del vapore su superfici verticali. Diverse prove sono state condotte per ogni superficie per determinare il coefficiente dello scambio termico al variare della portata del vapore e della differenza di temperatura saturazione-parete.

Si è osservato che il campione superidrofilico promuove la condensazione a film e i coefficienti dello scambio termico sono stati paragonati con quelli ottenuti su un campione non trattato. È risultato che un campione superidrofilico presenta coefficienti più bassi rispetto al campione non trattato, tanto più bassi quanto più bassa la velocità del vapore. Un'analisi dettagliata dei coefficienti dello scambio termico locale ha portato ad affermare che probabilmente questo peggioramento è dovuto proprio alla rugosità superficiale.

Il campione superidrofobico promuove una condensazione a gocce solo per un breve periodo, infatti il fenomeno persiste per alcuni minuti per poi passare ad una condensazione a film. La transizione è stata studiata e analizzata in quanto il campione mantiene le sue proprietà superidrofobiche una volta finito il test. La prima considerazione dedotta è che una bassa bagnabilità è una condizione necessaria ma non sufficiente per la promozione della condensazione a gocce: esiste, dunque, una morfologia superficiale ottima che deve essere realizzata per ottenere la condensazione a gocce. In secondo luogo, si evince che esiste un flusso termico critico oltre il quale avviene il passaggio alla condensazione a film, benché il campione rimanga superidrofobico. In letteratura questa transizione è ampiamente studiata per superfici in rame, mentre per l'alluminio non è ancora chiaro il fenomeno. Infatti, sembra che il passaggio sia influenzato dalla conducibilità termica del materiale, dunque superfici in rame promuovono condensazione a gocce per un intervallo maggiore di flussi termici rispetto a superfici in alluminio. Il fenomeno non è ancora del tutto compreso e ha bisogno di ulteriori indagini. Comunque, il coefficiente dello scambio termico misurato durante la condensazione a gocce è doppio rispetto a quello misurato in un campione non trattato alla stessa differenza di temperatura media logaritmica.

Mentre in letteratura si trova che campioni idrofobici in rame promuovono la condensazione a gocce, i campioni in alluminio testati nell'apparato sperimentale promuovono una condensazione a film. Anche in questo caso i coefficienti dello scambio termico sono stati confrontati con quelli ottenuti su una superficie non trattata e si è misurato un incremento dello scambio termico sul campione idrofobico. In particolare, aumentando la portata di vapore, si sono ottenuti valori dal 15% al 45% più elevati. È stata, dunque, effettuata un'analisi dei coefficienti dello scambio termico locale che ha poi portato a sviluppare un nuovo modello di condensazione, dato che i modelli classici di Nusselt e Carpenter-Colburn non erano sufficienti a spiegare l'aumento dei coefficienti. È stato introdotto, come già ipotizzato nel passato da Navier, un fattore di slittamento che tiene conto di una velocità non nulla all'interfaccia solido-liquido. In letteratura si trovano diverse evidenze sperimentali che verificano l'esistenza di questo fenomeno su superfici trattate idrofobicamente, ma solo in flusso monofase. Il fattore di slittamento è stato trovato dipendere dallo spessore del film di liquido e dalla velocità dello stesso. Verificate queste dipendenze trovate in letteratura con i dati sperimentali ottenuti dai test, è stato sviluppato un nuovo modello di condensazione introducendo questo fattore di slittamento nel modello classico della condensazione a film di Nusselt. La presenza di una velocità all'interfaccia liquido-solido comporta l'innalzamento dei coefficienti dello scambio termico rispetto alle condizioni di non slittamento e ciò sembra dipendere dal minor spessore del film di liquido, quindi dalla ridotta resistenza termica, che si viene a creare. Infine, è stato sviluppato anche un modello di scambio termico per il campione superidrofilico seguendo le considerazioni fatte per il campione idrofobico. Il fattore di slittamento è stata associato alla morfologia del substrato: esiste una zona, corrispondente alla rugosità, dove il film di liquido è fermo, intrappolato tra le asperità della superficie, con il risultato di avere uno spessore la cui conducibilità termica apparente è tra quella del liquido e quella dell'alluminio, perciò coefficienti dello scambio termico inferiori rispetto ad un campione liscio. Entrambi i modelli mostrano un buon accordo con i dati sperimentali.

In conclusione, una nuova modalità di condensazione a film è stata analizzata sperimentalmente e teoricamente su una superfice idrofobica. La sfida futura verterà nel riuscire a promuovere una condensazione a gocce stabile nel tempo.

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#	T sat [°C]	∆Tml [°C]	ṁ coolant [kg/s]	T inlet coolant [°C]	Q boiling chamber [W]	ṁ_vap [kg/(s m2)]	q coolant [kW/m2]	HTC [kW/m2K]
1	106.39	29.60	0.11	9.46	568.00	1.46	306.49	10.35
2	107.80	27.11	0.11	24.24	568.00	1.46	292.33	10.78
3	105.83	22.19	0.11	38.92	568.00	1.46	250.13	11.27
4	106.79	17.53	0.11	53.47	576.00	1.49	212.70	12.13
5	107.01	12.43	0.11	67.91	568.00	1.47	173.83	13.98
6	107.26	7.66	0.10	82.10	568.00	1.47	121.03	15.79
7	105.20	26.95	0.11	9.48	1208.00	3.12	308.73	11.46
8	105.78	24.56	0.11	24.22	1208.00	3.12	290.93	11.85
9	103.97	20.52	0.11	38.90	1208.00	3.12	250.17	12.19
10	104.06	15.86	0.11	53.50	1208.00	3.12	212.83	13.42
11	104.91	11.27	0.11	67.97	1208.00	3.12	167.79	14.89
12	106.21	6.68	0.10	82.23	1208.00	3.12	112.48	16.84
13	107.06	25.79	0.10	9.60	1832.00	4.71	341.20	13.23
14	106.67	22.80	0.10	24.29	1864.00	4.81	320.24	14.04
15	106.70	19.72	0.10	38.79	1840.00	4.74	286.49	14.53
16	107.01	15.62	0.10	53.40	1848.00	4.76	238.79	15.29
17	107.10	11.21	0.10	67.87	1856.00	4.79	185.71	16.57
18	107.21	6.85	0.10	82.11	1832.00	4.73	125.05	18.25
19	107.25	23.34	0.11	14.22	2456.00	6.31	335.63	14.38
20	106.35	20.92	0.11	24.28	2440.00	6.29	358.06	17.11
21	107.28	17.65	0.11	38.94	2440.00	6.60	309.73	17.55
22	107.13	14.41	0.11	53.51	2480.00	6.40	268.65	18.65
23	105.49	10.37	0.11	67.91	2480.00	6.39	202.29	19.51
24	107.74	5.65	0.11	82.17	2456.00	6.33	121.98	21.58

Appendix A

#	T sat [°C]	∆Tml [°C]	ṁ coolant [kg/s]	T inlet coolant [°C]	Q boiling chamber [W]	ṁ_vap [kg/(s m2)]	q coolant [kW/m2]	HTC [kW/m2K]
1	105.8	25.70	0.11	10	624.00	1.60	386.27	15.03
2	105.3	23.70	0.10	20	624.00	1.60	368.30	15.54
3	105.7	22.00	0.11	30	624.00	1.60	351.12	15.96
4	106	19.60	0.11	40	624.00	1.60	320.26	16.34
5	107.1	17.40	0.11	50	624.00	1.60	296.67	17.05
6	105.8	14.10	0.10	60	624.00	1.60	252.11	17.88
7	106.8	23.00	0.11	10	1168.00	3.10	424.58	18.46
8	107.2	21.50	0.11	20	1168.00	3.10	411.30	19.13
9	105.1	18.90	0.10	30	1168.00	3.10	375.92	19.89
10	107.2	17.60	0.10	40	1168.00	3.10	357.28	20.30
11	104.3	13.90	0.11	50	1168.00	3.10	292.18	21.02
12	105.6	12.10	0.10	60	1168.00	3.10	265.11	21.91
13	109.8	20.60	0.11	10	1885.00	5.10	490.90	23.83
14	109.9	19.50	0.11	20	1885.00	5.10	480.48	24.64
15	109.2	17.50	0.10	30	1885.00	5.10	456.58	26.09
16	109.3	15.80	0.10	40	1885.00	5.10	416.65	26.37
17	109	13.40	0.11	50	1885.00	5.10	371.85	27.75
18	108.8	11.00	0.10	60	1885.00	5.10	320.54	29.14

Appendix B

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