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## Design criteria for coatings in next generation condensing economizers

Nikolaos Vourdas<sup>a</sup>, Hussam Jouhara<sup>b</sup>, Savvas A. Tassou<sup>b</sup>, Vassilis N. Stathopoulos<sup>a, \*</sup>

<sup>a</sup> *Laboratory of Chemistry and Materials Technology, Technological Educational Institute of Sterea Ellada, Psachna Campus, 34400, Greece*

<sup>b</sup> *Brunel University London, Institute of Energy Futures, Centre for Sustainable Energy use in Food chains (CSEF),  
Uxbridge, UB8 3PH, United Kingdom*

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### Abstract

Surface processing of the materials used for the heat exchanging surfaces in condensing economizers and related facilities are used to deliver coatings to protect them from the corrosive condensates. By delivering increased corrosion protection, intact thermal conductivity, along with robust mechanical and wear resistance characteristics at low process costs, the lifetime of the investment is increased and the service periods become longer. However, during the past years surface processing has provided tools towards solutions for additional favorable surface features related to wetting phenomena. Coatings to enhance the dropwise condensation over the film-wise condensation, to increase the condensate collection rate and to promote the self-jumping of the condensates are far from rare. These features have a tremendous effect on the heat transfer coefficient and hence on the thermal efficiency of related heat exchange applications. Still, such features are typically not included on the testing protocols for the coatings being developed or demonstrated and they are not among the standard engineering selection criteria. In this work we briefly describe these features and provide preliminary equations for design and evaluation of their importance on the heat transfer coefficient.

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\* Corresponding author. Tel: +30 22280 99688

E-mail address: [vasta@teiste.gr](mailto:vasta@teiste.gr)

## 1. Introduction

Heat recovery from low temperature heat streams may contribute significantly to the energy efficiency of boilers, furnaces etc. Exploiting the latent heat and condensing the flue gases with the use of condensing economizers and pertinent facilities, can offer significant energy savings and substantial greenhouse gas emission reductions [1-3]. One of the main obstacles hindering their widespread usage is related to corrosion issues, since along with water, aggressive acids are also condensing on the surfaces leading to the damage of the heat exchange surfaces and reducing their lifetime [4, 5]. Therefore, appropriate surface processes are required to deliver coatings to protect the structure.

Biased by this obstacle the testing protocols for the quality assurance of the coatings are focused mainly on the corrosion protection characteristics of the coating system [6-8], accompanied by thermal conductivity measurements to ensure that the conductive heat transfer will not be suppressed. In order to ensure the structural integrity endurance upon installation and during service, wear resistance, tribological and associated mechanical properties are also measured.

The flue gas condensation however alone, i.e. even if pure water vapor, introduces additional phenomena that are crucial for the energy efficiency of the condensing economizers, and may not be directly assessed by the above mentioned criteria. These include a series of features related to wetting phenomena [9, 10]. The outmost goal of such processes is to increase the overall Heat Transfer Coefficient (*HTC*), i.e. the proportionality constant between the heat flux and the temperature difference:

$$\dot{q} = U \cdot A \cdot (T_{dew} - T_s) \quad (1)$$

where  $\dot{q}$  is the thermal power (W),  $A$  is the heat transfer surface ( $m^2$ ),  $T_s$  is the surface temperature (K) and  $T_{dew}$  is the ambient dew-point temperature (K). In this context  $U$  is the overall *HTC* expressed in  $W/(m^2K)$  and represents the heat flux per surface area and per temperature difference in the flue gas part, i.e. the heat recovery finally achieved per surface and temperature difference.

The importance of this parameter in the techno-economic evaluation of pertinent investments is briefly described hereafter. Condensing economizers are used to exploit thermal energy that would be otherwise wasted in the environment. The savings that may be attained by such facilities may be estimated, in terms of back-of-the-envelope calculations, from the following product:

$$Savings = HTC \cdot t \cdot P_{power} \quad (2)$$

with  $P_{power}$  being the power price in (€/Wh) that is recovered, and  $t$  being the operation time in (h). Within this context Eq.2 provides a rough estimation of the maximum savings per heat exchange surface, i.e. the scale of the economizer, and temperature difference. The investment cost on the other hand can be approximated, again for back-of-the-envelope calculations, as a function of the heat exchanging surface  $A$ , i.e. the scale of the facility:

$$Investment = f(A) \quad (3)$$

Of course both Savings and Investment cost, depend also on other parameters. However, encountering a *ceteris paribus* analysis, i.e. assuming all other technical parameters remain constant, provides us a basis for preliminary calculations. Using Savings and Investment cost along with assumptions for the cost of capitals, the payback period, the break-even point and pertinent indices can be calculated. It is therefore of primary importance, to increase the *HTC* in order to improve the investment metrics.

In this work we will briefly present some of the most important features, related to the wetting phenomena that are enabled by surfaces processes, towards improving the *HTC*. Based on such analysis the importance for introducing new criteria for coatings used in next generation condensing economizers will be emerged. In Fig. 1 we illustrate the four condensation cases that will be covered in this work. Starting from a fully developed Film-wise condensation (FwC), in which all the heat exchanging surface is covered by a film, and going to Dropwise condensation (DwC) in

which the condensates form droplets that may pin to the surface or may slide down the surface, and finally to self-jumping droplets, in which the condensates coalesce and self-jump off the surface, leaving it dry and susceptible for additional thermal transfer.

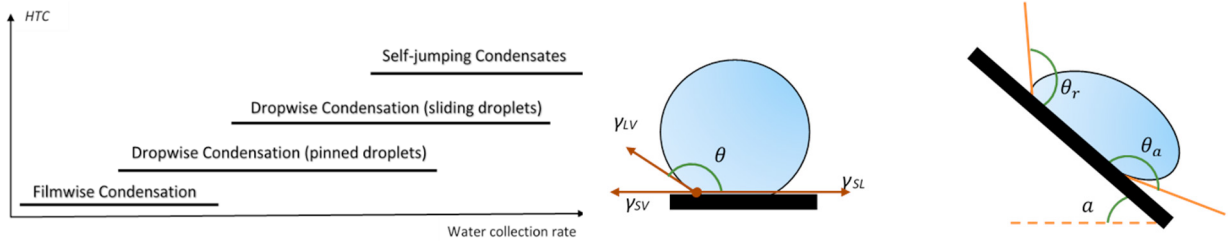


Fig. 1 (a) Different types of condensation that are being studied in this work. *HTC* increases from bottom to top and water collection rate (*WCR*) increases from right to left; (b) Schematic definition of static and dynamic contact angle. Static ( $\theta$ ) contact angle is measured on a quiescent droplet on a surface, Advancing ( $\theta_a$ ) and receding  $\theta_r$  contact angle may be measured on a surface tilted up to the point where the droplet starts to move, i.e. at the critical tilt angle ( $\alpha$ ). Other methods are also available.

## 2. Surfaces with special wettability

There is a great amount of work presented during the past years on surfaces exhibiting special wettability [11, 12]. One of the main property of a surface related to wetting phenomena is the apparent contact angle (*CA*,  $\theta$ ) (see **Error! Reference source not found.a**). For smooth, non-porous, chemically inert and homogeneous surfaces *CA* is given by the Young equation, derived either by a force balance at the triple point, i.e. the point at which liquid (L), solid (S) and vapor (V) phase co-exist, or by energy considerations:

$$\cos \theta = \frac{\gamma_{SV} - \gamma_{SL}}{\gamma_{LV}} \tag{4}$$

namely the *CA* depends on the surface energy ( $\gamma$ ) of the phases implicated. For most common combinations of surface energies *CA* range from ca. 0° to ca. 120°.

The second main property of a surface related to wetting phenomena is the tilt angle ( $\alpha$ ), or sliding angle or roll-off angle (see **Error! Reference source not found.b**). This defines the minimum angle at which a surface has to be inclined in order to leverage the downward movement of a droplet. The lower, the  $\alpha$  the easier is for a droplet to be removed off a surface. Much work has been done lately for the phenomena at the verge of a droplet movement [13, 14]. The governing equation for a droplet to slide, dictates that the gravitational force ( $F_g$ ) shall overcome the force ( $F_\gamma$ ) evolved due to the contact angle hysteresis (*CAH*). This force balance is expressed as follows:

$$F_g \sim F_\gamma \Rightarrow \rho_L g \Omega \sim 2w\gamma(\cos \theta_r - \cos \theta_a) \tag{5}$$

where  $\rho_L$  is the density of the condensate (liquid),  $g$  is the standard gravity,  $\Omega$  is the volume of the droplet,  $2w$  is the projected width of the droplet,  $\gamma$  is the condensate surface tension, and  $\theta_r$  and  $\theta_a$  are the receding and advancing contact angle respectively. The difference  $\cos \theta_r - \cos \theta_a$  is usually referred as contact angle hysteresis (*CAH*) and is a measure of the difficulty of a droplet to slide, or roll down a surface. The origin, methods of measurement and physical explanation of *CAH* have been discussed in the literature extensively [15-17]. *CAH* evolves when some of the prerequisites of the applicability of the Young equation appears; in surfaces where there is surface roughness, chemical inhomogeneity, surface energy anisotropy etc.

Special wetting phenomena appear combining special surface roughness, inhomogeneity and anisotropy. Enhanced hydrophobicity can be reached for example by introducing surface roughness to an inherently hydrophobic material as dictated by the Wenzel model:

$$\cos \theta_w = r_f \cos \theta \quad (6)$$

with  $r_f$  being the roughness factor, i.e. the ratio of the actual surface area over the projected surface area. In this case the surface exhibits higher  $CA$  but remains still fully wetted. Superhydrophobicity (SH), an extreme case of wettability, is defined as the state at which a surface exhibits high  $CA$  along with very low  $CAH$ , and is attained with high aspect ratio surface morphology and dual scale roughness [18]. In this state only a part of the surface is wetted by the liquid phase, while a part of the actual surface remains dry, forming gas pockets in between the liquid and the solid interface. Cassie-Baxter model is usually implemented to predict the  $CA$  in this case:

$$\cos \theta_{CB} = r_f f \cos \theta + f - 1 \quad (7)$$

where,  $r_f$  is the roughness factor of the wetted surface, and  $f$  is the fraction of the projected area of the solid surface.

By manipulating the surface features such as its chemistry, texture and anisotropy,  $CA$  and  $CAH$  may be precisely engineered. Going from highly hydrophobic to SH surfaces, the  $CA$  is increased, the  $CAH$  is decreased, the dry surface area is increased, and  $a$  is decreased; the latter facilitating easier droplets removal. A series of various surface processes enabling such features, have been emerged during the last years [19-27].

Another type of surfaces exhibiting interesting wettability is the lubricant infused porous surfaces [28, 29]. In this case a porous scaffold is infused with a lubricant, thus rendering the surface highly slippery to a numerous fluids and towards various applications [30].

The applicability of surfaces with special wettability on heat and mass transfer applications has been thoroughly reviewed recently [9, 31]. In all cases for such surfaces the stability remains still an open issue [32-34], especially under extreme temperature and pressure conditions. For the case of SH coatings collapse of the gas pockets at the solid-liquid interface, along with mechanical damage of the surface topography are the most important factors leading to wettability deterioration. For the case of lubricant infused porous surfaces, the deterioration comes mainly from drying of the lubricant.

Another interesting issue emerging from wetting phenomena and affecting thermal transport phenomena is the switching between wetting states actively and dynamically. Of primary importance is the switching between states in order to leverage the actuation of a pinned droplet to a mobile droplet that may be removed off a surface. Various actuation tools have been implemented to this direction including electricity, magnetism, thermal energy, gas pressure, optics and vibrations [35-45].

Designing surfaces with special wettability may rev up the removal of condensates from the surface and thus enhance the thermal efficiency of the heat recovery facilitates as will be discussed hereafter.

### 3. Film-wise vs Dropwise condensation

The first feature for a coating in order to increase  $HTC$  is to promote the dropwise condensation (DwC) over the film-wise condensation (FwC) [46-48]. This is leveraged by two parameters: (i) the surface chemistry and (ii) the surface roughness, both of which affect  $CA$ . DwC can be attained by most surfaces that are not highly hydrophilic, i.e. with  $CA < 10^\circ$ . Highly hydrophilicity is typically attained by surfaces with polar groups. Even in this case and because of the high surface energy these surfaces may absorb hydrocarbon compounds [49], resulting in  $CA$  increase. Therefore DwC, where condensate is in the form of droplets co-existing with bare surface areas is the most common morphology of condensation. In Fig. 2 we present snapshots taken during condensation on bare and untreated plexiglass surface at Relative Humidity (RH) equal to 90% and surface temperature equal to 4 °C. Even though the  $CA$  of bare and untreated plexiglass is ca. 70°, i.e. lower than 90°, DwC is recorded throughout the experiment. Of course, DwC can be typically be attained on hydrophobic surfaces, i.e. surfaces with  $CA$  higher than 90°.

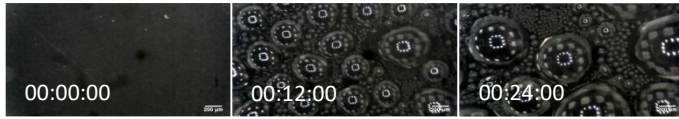


Fig. 2 Condensation monitoring on a clean flat, bare and untreated plexiglass surface (Chamber at 24 °C and RH: 90%, Plexiglass surface at 4 °C).

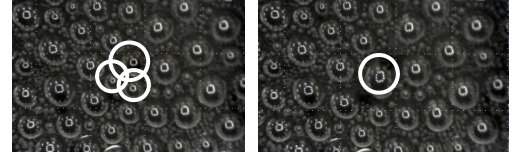


Fig. 3 Coalescence of three droplets into one on a Plexiglass surface during condensation.

With FwC a film of condensates builds up, thus forming up a thermal barrier, by adding additional thermal resistance equal to  $1/k_c$ , with  $k_c$  being the thermal conductivity of the condensate film. For this case, i.e. FwC, the analysis provided first by Nusselt is typically employed, as a first approximation to predict the *HTC* [50]:

$$HTC = 0.943 \left[ \frac{\rho_L (\rho_L - \rho_V) g h_{fg} k_c^3}{\mu_L L (T_{dew} - T_s)} \right]^{1/4} \quad (8)$$

with  $\rho_L$  and  $\rho_V$  being the mass density of the condensate (liquid) and vapor ( $\text{kg/m}^3$ ) respectively,  $g$  the standard gravity,  $h_{fg}$  the latent heat of evaporation ( $\text{kJ kg}^{-1}$ ),  $k_c$  the thermal conductivity of the condensate (liquid) ( $\text{W/mK}$ ), and  $L$  length of the surface from the point on which the condensation takes place.

All cases with the exception of fully developed FwC fall in the case of DwC, in which a fraction of the surface area remains uncovered and condensates form droplets. As will be shown below the mobility of droplets greatly affect the *HTC*, and thus DwC may be further analyzed in subcategories.

In general, with DwC *HTC* is higher compared to FwC because a fraction of area spontaneously dewets and thus is susceptible for new condensation and thermal transfer. The volume and distribution, in static and dynamic mode, of the droplets, both of which depending on the wetting phenomena of the surface, plays a key role in the *HTC* finally attained. For this reason the calculation of the *HTC* for DwC is not straight forward.

Kim S and Kim K J were the first to analyze explicitly the heat transfer through one droplet and take into account the *CA* and the distribution of the condensates, in order to predict the *HTC* [51]. For a single droplet with radius  $r$ , on a surface on which exhibits  $CA=\theta$ , the heat transfer rate is expressed as:

$$q_d = \Delta T \pi r^2 \left( 1 - \frac{r_{min}}{r} \right) \left( \frac{\delta}{k_{coat} \sin^2 \theta} + \frac{r\theta}{4k_c \sin \theta} + \frac{1}{2h_i (1 - \cos \theta)} \right)^{-1} \quad (9)$$

The minimum viable drop size,  $r_{min} = 2T_{dew}\gamma/Th_{fg}\rho_L$  with  $\Delta T = T_{dew} - T_s$ . In this expression  $\delta$  and  $k_{coat}$  is the thickness and thermal conductivity of the hydrophobic coating, and  $h_i$  is the interfacial heat transfer coefficient, usually from  $0.83 \text{ MW/m}^2 \text{ K}$  to  $15.7 \text{ MW/m}^2 \text{ K}$ , for a pressure range of  $0.01 \text{ atm}$  to  $1.0 \text{ atm}$ . Equation 9 has been modified [52] to account also for the surface morphology geometrical characteristics and for a pillared surface morphology becomes:

$$q_d = \pi r^2 \left( \Delta T - \frac{2T_{dew}\gamma}{rh_s\rho_L} \right) \left( \frac{1}{2h_i (1 - \cos \theta)} + \frac{r\theta}{4k_c \sin \theta} + \frac{1}{k_{coat} \sin^2 \theta} \left( \frac{k_p \varphi}{\delta k_p + h_p k_{coat}} + \frac{k_c (1 - \varphi)}{\delta k_c + h_p k_{coat}} \right) \right)^{-1} \quad (10)$$

where,  $k_p$  is the thermal conductivity and  $h_p$  the height of the pillars respectively and  $\varphi$  is the solid fraction of the surface wetted by the liquid. For cylindrical pillars  $= \pi D_p^2/4l_p^2$ , where  $D_p$  is the diameter of the pillars and  $l_p$  is the center-to-center spacing of pillars. Working with surfaces with surface roughness, the geometrical characteristics of the surface shall be measured, e.g. with Scanning Electron Microscopy (SEM) in order to use this equation.

The heat transfer of a single droplet,  $q_d$ , can be also calculated by measuring the droplet growth rate, i.e. by monitoring the radius,  $q_d$ , and the  $CA$ ,  $\theta$  and recording them vs. time [52]:

$$q_d = \dot{m}_w h_{fg} = \rho_L h_{fg} \frac{d\Omega}{dt} = \frac{\pi}{3} \rho_L h_{fg} \frac{d}{dt} \left[ (1 - \cos \theta)^2 (2 + \cos \theta) r^3 \right] \quad (11)$$

There are two options for a droplet during growth. The first is to coalescent with another droplet and either slide down the surface or jump off the surface (analyzed in Section 4). The second is, if none of the previous takes place, to grow as much as the gravity forces overbalance the adhesion forces and the droplet eventually slides down the surface individually.

In Fig. 3 we demonstrate the coalescence of three droplets, which form one bigger droplet. In this particular case this droplet does not slide down the surface. Following the analysis presented in the previous section, the heat energy removed from a droplet up until the coalescence shall be predicted, using the following expression:

$$Q_d = \int_{r_{\min}}^{l_c/2} q_d dt = \frac{1}{3} \pi \rho_L h_{fg} (1 - \cos \theta)^2 (2 + \cos \theta) \left( \frac{l_c}{2} \right)^2 \quad (12)$$

where  $l_c$  is the coalescence length, i.e. the max diameter of the droplet before coalescence and removal. This parameter may be measured experimentally during condensation experiments. The heat transfer rate for an individual droplet may be then approximated as:  $\dot{q}_d = Q_d/\tau$ , with  $\tau$  being the time needed for coalescence, i.e. at  $r = l_c/2$ .

The growth of an individual droplet, however, cannot continue perpetually. With the increase of the volume the mass is also increased leading to increase of the gravitational force exerted on the droplet. The force balance in eq. 5 dictates that the higher the  $CAH$  the most difficult is for a droplet to slide down. Based on eq. 9 and eq. 10, a surface covered by small drops provides a higher heat transfer and therefore condensation performance than the same surface covered by large drops. This is the reason why superhydrophobic surfaces, i.e. surfaces with high  $CA$  (typically higher than  $150^\circ$ ) and very low  $CAH$  (typically lower than  $5^\circ$ ), yield higher  $HTC$ . Therefore, even though DwC is a desirable feature drops forming upon condensation that are pinned to the surface and cannot be easily removed from the heat exchanging surface do not leverage a significant increase in  $HTC$ . Ideally the condensates shall instantly be removed from the surface in order for additional condensation to take place. In this context, the design criteria for coatings used in condensing economizers, shall target for solutions that not only will enhance the DwC, but also the easy removal of the condensates from the surface. This feature can be quantified by the water collection rate ( $WCR$ ) namely the rate of condensates that are being removed from the surface. The force balance at the verge of movement of a condensate (see Fig. 4) is given in Eq.5. Using superhydrophobic surfaces small  $CAH$  can be attained, and therefore a smaller  $\tau$  can be attained. Alternatively and in order to increase the  $WCR$  other approaches may include installation at high inclination angles [53].

Based on the analyses presented above, the  $HTC$  and hence the energy efficiency strongly correlates to the rate of condensate formation ( $\dot{m}_w$ ); the higher the  $WCR$  the higher  $\dot{m}_w$ . In the equation below the latent contribution of the overall  $HTC$  is provided:

$$HTC = \dot{m}_w h_{fg} (T_{dew} - T_s)^{-1} \quad (13)$$

where  $\dot{m}_w$  is the  $WCR$  ( $\text{kg s}^{-1}$ ),  $h_{fg}$  is the latent heat of evaporation ( $\text{kJ kg}^{-1}$ ),  $T_s$  is the substrate temperature (K) and  $T_{dew}$  is the ambient dew-point temperature (K).

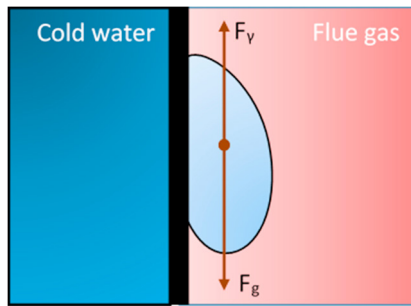


Fig. 4 Forces acting on a condensate droplet.

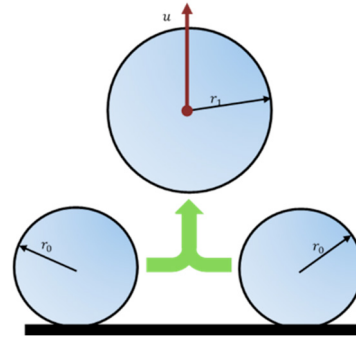


Fig. 5 Two droplets, with radius  $r_0$  on a surface coalescing to a droplet with radius  $r_1$  and jumping with a takeoff velocity of  $u$ .

#### 4. Self-jumping condensates

Since the first observation of self-jumping droplets upon condensation [54] there has been a significant research effort in understanding and optimizing such phenomena. The goal here is to decrease the maximum droplet size of the condensate which enables self-jumping and maximize their take-off velocity after coalescence (see Fig. 5). This will empower higher *HTC* values [55] and increased energy efficiency.

The first approach in treating this problem is to assume two droplets with radius equal to  $r_0$  coalescing into a droplet with radius of  $r_1$  and self-jumping with a takeoff velocity equal to  $u$  [56]. In this context  $r_1 = 2^{1/3} r_0$  and the volume of the final droplet is  $\Omega = (8/3) \pi r_0^3$ . Upon coalescence the surface energy of the system is reduced by:  $\Delta E_\gamma = 4\gamma\pi r_0^2(2 - 2^{2/3})$ . The takeoff velocity  $u$  may be deducted assuming energy transfer from the surface energy, scaling as  $\gamma r^2$  in kinetic energy, scaling as  $\rho_L r^3 u^2$ . The takeoff velocity is therefore:

$$u^* = \sqrt{\gamma(\rho_L r_0)^{-1}} \quad (14)$$

Typically, the coalescence of droplets before self-jumping takes place at lower  $r$  compared to previous cases.

#### 5. Conclusions

Wetting phenomena are crucial in heat transfer processes exploiting the latent heat from condensation. Targeting to high energy efficiency of condensing economizers and related facilities new design criteria should be introduced, on top of corrosion protection, thermal conductivity and wear resistance. Coatings exhibiting high contact angles, low contact angle hysteresis, low tilt angle, and high water collection rate are expected to increase the overall heat transfer coefficient and improve the investment indices, making the installation of such facilities more profitable. In order to assess the enhancement of heat transfer coefficient, droplet morphology parameters, such as condensate diameter distribution, time for coalescence are essential to be recorded. In addition to coatings wetting properties design, experimental setups enabling the monitoring of phenomena during condensation are needed to be included in the testing protocols of coatings in condensing economizers.

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