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## Global analysis and development of a predictive tool of the effect of tube inclination on two-phase heat transfer: boiling, condensation and heated gas-liquid flows

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### Abstract

The influence of gravity on two-phase heat transfer represents a challenge to the design and development of thermal systems in which the tubes can be oriented at different inclination angles. In this regard, the present study is based on a global analysis of the effect of the channel orientation on the two-phase heat transfer coefficient, which was performed by collecting 5408 data points from independent published works that include flow boiling, flow condensation and heated gas-liquid flows at various orientations with respect to the gravity force. A comparison against existing prediction methods for two-phase heat transfer including the tube orientation effect has shown a very poor agreement, and no general tool is available in the literature to determine whether the effect of gravity is negligible or of major importance for specific applications. For these reasons, a new prediction tool is proposed. The Inclination effect  $I_h$  is defined as the maximum expected variation of the heat transfer coefficient when the tube inclination angle is changed, with respect to the predicted value in a horizontal arrangement. Using an unconstrained nonlinear optimization, the Inclination effect is correlated with the most influencing nondimensional groups of parameters for boiling and condensation heat transfer, providing satisfactory agreement with the experimental data ( $MAE=17.6\%$  and  $19.3\%$ , respectively). The  $I_h$  is found to be a decreasing function of the vapor quality and mass flux, and it increases with larger diameters and alongside heat flux in flow boiling. This new tool can be employed in the design process of condensing and evaporating units, since it is able to establish whether the heat transfer coefficient might be affected by the channel orientation or, instead, a typical correlation for non-inclined tubes can be used.

**Keywords:** Heat transfer; Inclination effect; Flow boiling; Flow condensation; Gas-liquid two-phase flow

## Nomenclature

### Roman

$a$	exponents and constants of the correlation	[-]
$c_p$	specific heat at constant pressure	[J/kgK]
$CHF$	critical heat flux	[W/m <sup>2</sup> ]
$d$	internal tube diameter	[m]
$F_\beta$	correlation parameter	[-]
$F_B$	flow pattern factor	[-]
$G$	mass flux	[kg/m <sup>2</sup> s]
$g$	acceleration of gravity	[m/s <sup>2</sup> ]
$h$	heat transfer coefficient	[W/m <sup>2</sup> K]
$I_h$	inclination effect	[%]
$i_{LV}$	latent heat of vaporization	[J/kgK]
$J_V$	non dimensional vapor velocity	[-]
$k$	thermal conductivity	[W/mK]
$MAE$	mean absolute error	[%]
$N$	number of parameters or experiments	[-]
$p$	generic parameter	[-]
$q$	heat flux	[W/m <sup>2</sup> ]
$RMSE$	root mean square error	[-]
$S_{L,eff}$	effective wetted perimeter	[m]
$T$	temperature	[°C]
$u$	velocity	[m/s]
$x$	vapor quality	[-]
$Z$	inclination parameter	[-]

### Greek

$\beta$	tube inclination angle (upward for $\beta > 0^\circ$ )	[°]
$\delta$	thickness	[m]
$\Delta$	variation	[-]
$\mu$	viscosity	[Pa s]
$\rho$	density	[kg/m <sup>3</sup> ]
$\sigma$	surface tension	[N/m]

$Bd$	Bond number	$\frac{g(\rho_L - \rho_V)d^2}{\sigma}$
$Fr$	Froude number	$\frac{u_{SV}}{\sqrt{gd}} \sqrt{\frac{\rho_V}{\rho_L - \rho_V}}$
$Ja$	Jacob number	$\frac{c_p(T_{sat} - T_w)}{i_{LV}}$
$Nu$	Nusselt number	$\frac{hd}{k}$
$Pr$	Prandtl number	$\frac{\mu}{c_p k}$
$Re_L$	liquid Reynolds number	$\frac{G(1-x)d}{\mu_L}$
$We_L$	liquid Weber number	$\frac{[G(1-x)]^2 d}{\rho_L \sigma}$
$X_u$	Martinelli parameter	$\left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_V}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_V}\right)^{0.1}$
$Bo$	Boiling number	$\frac{q}{G i_{LV}}$

### Subscripts

$0^\circ, REF0^\circ$	degree reference value
$boil$	boiling
$cond$	condensation
$exp$	experimental
$h$	hydraulic
$L$	liquid
$LF$	liquid film
$min$	minimum
$max$	maximum
$pred$	predicted
$sat$	saturation
$SL$	superficial liquid
$SV$	superficial vapor
$T$	thermal boundary layer
$u$	hydrodynamic boundary layer
$V$	vapor
$w$	wall

### Non-dimensional numbers

## 1. Introduction

Two-phase heat transfer, including the liquid-vapor phase change process, is widely studied for different kinds of applications. The characterization of the condensation and boiling heat transfer phenomena are of primary importance in the design of refrigeration and air/conditioning plants, and the latent heat during evaporation can also be successfully employed in enhanced cooling systems that require high heat fluxes to be handled [1] [2]. Finally, nonboiling heat transfer in gas-liquid flows is also often encountered in chemical processes, the petroleum industry, air-lift systems, solar collectors, nuclear reactors, etc.

For all these applications, heat exchangers and heat spreader systems might be orientated horizontally, vertically or even at a specific angle due, for instance, to space optimization concepts or technical issues. A- or V- shaped heat exchangers can especially be used in remote areas where the water availability is low and air-cooled systems represent the best options to sustain the cooling of condensers in power and refrigeration plants. Moreover, headers and tubes can also be inclined to vent the condensed liquid out of the condenser in the case of conventional heat exchangers.

The influence of the pipe orientation on the two-phase flow morphology has been investigated by several researchers, highlighting the inclination effect on the void fraction [3] [4], entrainment [5], pressure drop [6] [7] [3] [8] and flow pattern transition criteria [9] [10] [11] [12], for both diabatic and adiabatic flows.

In regard to the heat transfer phenomenon, it has been proven that inclined surfaces may provide a significant change in the heat transfer performance with respect to the classical disposition [13]. However, although two-phase flows have been widely studied in the last decades for a comprehensive range of operating conditions in terms of pipe geometry and fluids, most of the works available are related only to vertical or horizontal tube arrangements, whereas flow boiling, condensation and gas-liquid heat transfer data for inclined surfaces are relatively scarce. As a consequence, while there are numerous correlations which may be employed to predict the two-phase heat transfer coefficient in horizontal and vertical tube orientations, very limited prediction tools take into account the effect of the inclination angle and the available heat transfer coefficients prediction methods were mostly conceived by considering few databases with limited amount of points. Moreover, to the authors' knowledge, there are currently no practical tools able to determine whether the effect of gravity on two-phase heat transfer is significant or negligible for specific applications.

The first objective of the present paper is to provide a critical literature review of the influence of the tube inclination angle on the two-phase heat transfer coefficient, by collecting data from boiling, condensation and heated gas-liquid flows. In the second part of the paper, we provide a useful tool able to state whether the heat transfer coefficient is affected by the tube orientation or, instead, whether the typical prediction methods for horizontal flows may be used. The nondimensional parameter *Inclination effect*  $I_h$  is therefore defined, representing the maximum relative variation of the local heat transfer coefficient with the channel inclination. This parameter is finally correlated to physically-driven nondimensional numbers through a statistic tool, providing useful information for the correct design of condensing and evaporating units conceived to work at a specific inclination angle. The definition of the Inclination effect is limited to boiling and condensation heat transfer, since data for cases of heated air-water are scarce, and the investigated values of the inclination angles are quite limited to nearly- horizontal or nearly- vertical configurations.

## 2. Presentation of the database

The present study relies on the analysis of experimental results available in the literature that deal with two-phase flows in inclined channels. A thorough literature review has been carried out to collect most of the available data for the heat transfer coefficient at multiple tube inclinations, for the three types of two-phase flows of interests (flow boiling, flow condensation and heated air- water flows). Only the experimental works that deal with multiple pipe inclination angles (including studies of both horizontal and vertical configurations only) are considered in this review, to perform a fair comparison of the heat transfer coefficients at different tube orientations by employing the same test facility and exactly the same operating conditions. A total amount of 5408 data points was collected. The three databases, corresponding to each type of two-phase flow, are described in the present section.

### 2.1. Flow boiling database

The research has highlighted 15 studies of flow boiling at multiple channel orientations and different operating conditions in terms of the tube geometry, fluid, saturation temperature, heat and mass flux, vapor quality and inclination angles investigated, from which 2371 heat transfer coefficient data have been collected. In some cases, especially for multi-minichannel geometries, the authors did not directly provide the heat transfer coefficient values, which have been included in this database by calculation from the boiling curves and the available information. Table 1 summarizes the mentioned studies in chronological order, along with the values and ranges of the main operating conditions. Specifically, the inclination angle  $\beta$  has been defined with respect to the horizontal disposition, ranging from  $-90^\circ$  (vertical downward) to  $+90^\circ$  (vertical upward). However, only a few studies focused on the entire possible range of  $\beta$ , and even in these cases, the number of inclination angles employed is very limited. A significant part of the database is instead related to only two or three configurations (exploring vertical upward, vertical downward and horizontal dispositions, at most). Regarding the tube geometry, only six works employ regular surfaces (smooth and circular single channels), whereas in other papers, enhanced heat transfer tubes (corrugated, grooved or even internally finned) are used. The remaining works are related to enhanced cooling systems with particular attention to the CHF phenomenon in rectangular multi-minichannel heat sinks (with hydraulic diameters below 1.0 mm) working in different orientations. Additionally, according to the transition criteria of Kandlikar and Grande [14], the tube diameters cover both the miniscale (1.0 and 1.6 mm) and the macroscale (from 4 up to 10 mm). Refrigerants such as R134a, R245fa and R407C are the typical fluids employed in experiments performed in single tubes, whereas water, HFE-7100 and HC-72 are used in cases of multi-minichannel heat sinks for enhanced cooling purposes (electronic components, for instance). The investigated mass fluxes range from 10 to 1000 kg/m<sup>2</sup>s, and the imposed heat flux typically ranges from 2 to 60 kW/m<sup>2</sup> for conventional geometries and reaches considerably higher values (up to 317 kW/m<sup>2</sup>) for multi-minichannel geometries.

### 2.2. Convective condensation database

For condensing flow, a total amount of 1878 heat transfer coefficient data points coming from 11 independent studies has been collected. Table 2 summarizes the authors and the related operating conditions. Unlike the flow boiling database, there are no multi-minichannel heat sink geometries, and most of the studies are related to smooth circular channels. Among the exceptions, the work of Akhavan et al. [15] is performed with an internally finned tube of 8.92 mm internal diameter, whereas Koheini et al [16] used a corrugated channel with a similar diameter of 8.32 mm. Finally, Del Col et al. [17] presented a comprehensive study of the condensation of R134a and R32 in a square cross-section minichannel having an equivalent internal diameter of 1.18 mm, thus representing the narrowest tube of the condensation database for multiple orientations. Except for Wang et al. [18], in which the working fluid is water condensing at 105 °C in smooth circular tubes having 4 different diameters, all the experiments have been performed using refrigerants (especially R134a) condensing from 29 to 55.4 °C. Similar to the flow boiling database, the investigated inclination angles range from the vertical downward ( $-90^\circ$ ) to the vertical upward ( $+90^\circ$ ) disposition. In some papers (Akhavan et al. [15], Koheini et al. [16], Mohseni et al. [19] and Del Col et al. [17]), the two-phase heat transfer coefficient is obtained using at least 3 upward and downward inclination angles. More detailed analyses were performed by Lips et al. [20], Meyer et al. [21], Olivier et al. [22] and Ewim et al. [23], which shared the same test facility with a circular smooth tube of 8.38 mm ID and included up to 20 values of  $\beta$  for each operating condition. Wang et al. [18] and Mozafari et al. [24] limited their analysis to the upward orientation. The first used 4 inclination angles from 0 to 45°, whereas the latter studied the condensation of isobutane (R600a) in a helicoidally shaped condenser at 0, 30, 60 and 90°, with a saturation temperature of 43 °C. Finally, Xing et al. [25] evaluated the heat transfer performance of vertical upward, vertical downward and horizontal flows for the refrigerant R245fa at 55.4 °C in a macrochannel of 14.81 mm ID, which is also the largest diameter in the present condensation database. The investigated mass fluxes have similar values to those collected for the flow boiling database, ranging from 11.4 to 705.4 kg/m<sup>2</sup>s. The heat flux ranges instead from 6.3 up to 73.3 kW/m<sup>2</sup>. In most cases, these values are not directly provided by the original reference and were evaluated afterwards with the data at disposal from the

secondary cold fluid. When the necessary information is not given in the original reference, the heat flux could not be calculated and is labeled as not available (NA) in Table 2.

### 2.3. Gas-liquid flow database

The results of the literature review on heat transfer for nonboiling heated gas-liquid flow in multiple pipe orientations are quite scarce compared to those for condensation and flow boiling. In the present study, 1159 heat transfer coefficient data points have been collected from 6 different sources, as summarized in Table 3. All the works are related to heated air-water flows at atmospheric or nearly-atmospheric pressure at inlet temperatures close to ambient (20-25 °C). Similar to the abovementioned condensation works, for all the air-water data the applied heat flux is not given in the original reference and its value cannot be extrapolated from the information at disposal. Due to the differences in the fields of applications, all the tubes are smooth, single and circular, having diameters considerably higher than those shown in the previous boiling and condensation reviews, so that no heat transfer coefficient data fall into the micro or mini-scale. The works of Bhagwat and Ghajar [26] [27] and Hossainy et al. [28] share the same test facility, with an internal diameter of 12.5 mm, which is the smallest tube in the heated air-water database. In the experiments by Ghajar and Tang [29], the pipe diameter is 27.9 mm, and finally it goes up to 49.2 mm for the papers by Hetsroni et al [30] [31]. The range of orientations investigated is very limited compared to those in the previous databases. Only the work of Hetsroni et al. [31] provides all the possible orientations from the vertical downward (-90°) to the vertical upward (+90°) using small steps of 15°. A comprehensive analysis of the effect of inclination for upward and downward flows is also performed by Bhagwat and Ghajar [26] [27], but it is split into two different studies that do not share exactly the same operating conditions. Hossainy et al. [28] experimented using four different inclination angles for downward flow (-20°, -10°, -5° and 0°), whereas Hetsroni et al. [30] and Ghajar and Tang [29] focused only on nearly vertical upward (85° and 90°) and nearly horizontal (0°, 2°, 5°, and 7°) tube orientations, respectively.

Unlike the flow boiling and condensation tests, the mass flux  $G$  is not directly given by the authors, who usually provide instead either the liquid and vapor Reynolds numbers or the vapor and liquid superficial velocities. The range of mass fluxes given in Table 3 is therefore obtained from subsequent calculations.

### 2.4. Range of experimental conditions in terms of nondimensional parameters

The three databases involve very different types of fluids and channel diameters. To compare the various experimental conditions, nondimensional numbers can be used. For any kind of two-phase flow, the value of the heat transfer coefficient is indeed a direct consequence of the flow pattern occurring in the tube, which can in turn be influenced by the gravity effects and, therefore, by the orientation of the channel [20] [32]. Note that in the case of a boiling flow, the effects of the nucleation phenomenon and capillary forces might also play important roles and should therefore be taken into account [33]. As reported by different researchers [34] [35] [36] [37] [38], the effect of gravity is progressively reduced when the inertia and/or the surface tension become significant. The relative importance of the three contributions is well summarized by the vapor Froude number  $Fr_V$  and the Bond number  $Bd$ , as defined by [36]:

$$Fr_V = \frac{u_{sv}}{\sqrt{gd}} \sqrt{\frac{\rho_v}{\rho_L - \rho_v}} \quad (1)$$

$$Bd = \frac{g(\rho_L - \rho_v)d^2}{\sigma} \quad (2)$$

**Table 1** Effect of the tube inclination on the heat transfer coefficient: flow boiling database

Author	Geometry*	$T_{sat}$ [°C]	$d_h$ [mm]	$G$ [kg/m <sup>2</sup> s]	$q$ [kW/m <sup>2</sup> ]	Fluid	$\beta$ [°]	$x$	Number of data points
Kandlikar and Balasubramanian [39]	M,Rc,R	110	0.332	120	317	Water	-90, 0, 90	[0 0.4]	15
Baba et al. [40]	S,C,R	56	4	40-500	0.5-60	FC72	0, 90	[0 1]	192
Akhavan et al. [41]	S,C,F	-15	8.92	53-136	2.1-5.3	R134a	-90, -60, -30, 0, 30, 60, 90	[0.2 1]	224
Wang et al. [42]	M,Rc,R	61	0.825	100-300	25, 37.5	HFE-7100	-90, -45, 0, 45, 90	[0 1]	125
Akhavan-Behabadi and Esmailpour [43]	S,C,Cr	-15	8.3	46-136	4.56-9.13	R134a	-90, -60, -30, 0, 30, 60, 90	[0.2 1]	217
Kundu et al. [44]	S,C,R	7.5	7	100-300	3.0-6.0	R407C	0, 30, 45, 60, 90	[0.1 0.9]	141
Lee et al. [45]	M,Rc,R	78.6	0.375	180-644	43.0-66.0	FC72	-90,0,90	[0 0.6]	93
Mohseni and Akhavan-Behabadi [46]	S,C,R	-15	8.92	53-136	2.1-5.3	R134a	-90, -60, -30, 0, 30, 60, 90	[0.2 1]	224
Kundu et al. [47]	S,C,R	7.5	7	100-300	3.0-10.0	R134a, R407C	0, 30, 45, 60, 90	[0.1 0.9]	400
Hsu et al. [48]	M,Rc,R	61	0.44	100,200	25,40	HFE-7100	-90, -45, 0, 45, 90	[0 1]	115
Leao et al. [49]	M,Rc,R	30	0.197	300-1000	0-300	R245fa	0,90	[0 1]	56
Cheng et al. [50]	S,C,G	20	10	10.0-30.0	3.0-10.0	R134a	-90,0,90	[0.1 0.8]	273
Gao et al. [51]	M,Rc,R	29	0.745	136-347	15-135	R134a	-90,0	[0 1]	44
Saisorn et al. [52]	S,C,R	31.3	1	250-820	1.0-60	R134a	-90,0,90	[0 1]	54
Layssac et al. [33]	S,C,R	81	1.6	150, 300	13.1	R245fa	-90, -60, -45 -30, -15, 0, 15, 30, 45, 60, 90	[0 1]	198
<b>Overall ranges</b>	<b>see below</b>	<b>-15 - 110</b>	<b>0.197 - 10</b>	<b>10 - 1000</b>	<b>0 - 317</b>	<b>Water, FC72, R134a, HFE-7100, R407C, R245fa</b>	<b>-90 - +90</b>	<b>0 - 1</b>	<b>2371</b>

\*S=single, M=multi; C=circular, Rc=rectangular; F=finned, Cr= corrugated, R=regular, G=grooved

**Table 2** Effect of the tube inclination on the heat transfer coefficient: condensation database

Author	Geometry*	$T_{sat}$ [°C]	$d_h$ [mm]	$G$ [kg/m <sup>2</sup> s]	$q$ [kW/m <sup>2</sup> ]	Fluid	$\beta$ [°]	$x$	Number of data points
Wang et al. [18]	S,C,R	105	1.94;2.80;3.95;4.98	11.4-29.3	NA	Water	0, 17, 34, 45	[1 0]	84
Akhavan et al. [15]	S,C,F	29	8.92	54;81;107	8.7-20.3	R134a	-90, -60, -30, 0, 30, 60, 90	[1 0]	84
Khoeini et al. [16]	S,C,Cr	30	8.32	87-253	8.8-42.2	R134a	-90, -60, -30, 0, 30, 60, 90	[1 0]	112
Lips et al. [20]	S,C,R	40	8.38	200-600	10.2	R134a	wide range between [-90 90]	[0.9 0.1]	158
Mohseni et al. [19]	S,C,R	35	8.38	53-212	NA	R134a	-90, -60, -30, 0, 30, 60, 90	[1 0]	126
Meyer et al. [21]	S,C,R	30-50	8.38	100-400	10.2	R134a	wide range between [-90 90]	[0.9 0.1]	299
Del Col et al. [17]	S,Rc,R	40	1.18	100-390	NA	R134a/R32	-90, -60 -45, -30, -15, 0, 30, 45, 60, 90	[1 0]	462
Mozafari et al. [24]	S,C,Hel	43	8.3	155-255.5	6.3-7.8	R600a	0, 30, 60, 90	[0.8 0.1]	112
Xing et al. [25]	S,C,R	55.4	14.81	191.3-705.4	21.2-79.3	R245fa	-90, 0, 90	[0.6 0.1]	69
Olivier et al. [22]	S,C,R	40	8.38	100-300	10.2	R134a	wide range between [-90 90]	[0.8 0.1]	237
Ewim et al. [23]	S,C,R	40	8.38	50-100	NA	R134a	wide range between [-90 90]	[0.8 0.1]	135
<b>Overall ranges</b>	<b>see below</b>	<b>29 - 105</b>	<b>1.18 - 14.81</b>	<b>11.4 - 705</b>	<b>6.3 - 79.3</b>	<b>Water, R134a, R32, R600a, R245fa</b>	<b>-90 - +90</b>	<b>1 - 0</b>	<b>1878</b>

\*S=single, M=multi; C=circular, Rc=rectangular; F=finned, Cr= corrugated, R=regular, Hel=helicoidal

**Table 3** Effect of the tube inclination on the heat transfer coefficient: heated gas-liquid database

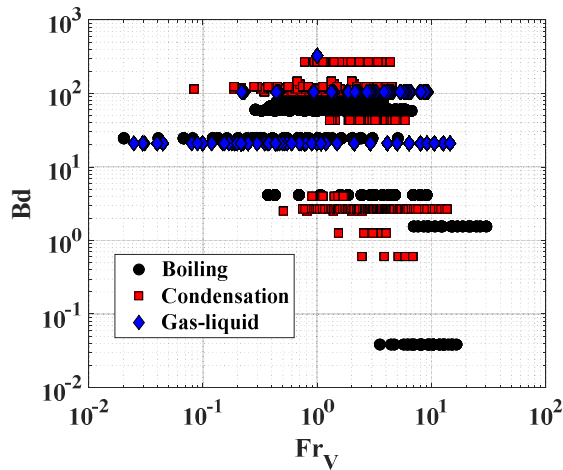
<b>Author</b>	<b>Geometry*</b>	$d_h$ [mm]	$G$ [kg/m <sup>2</sup> s]	<i>Fluid</i>	$\beta$ [°]	<b>Number of data points</b>
Hetsroni et al. [30]	S,C,R	49.2	137-465	Air-water	85, 90	30
Hetsroni et al. [31]	S,C,R	25; 49.2	29-54	Air-water	-90, -75, -60, -45, -30, -15, 0, 15, 30, 45, 60, 75, 90	39
Ghajar and Tang [29]	S,C,R	27.9	32-1080	Air-water	0, 2, 5, 7	440
Hossainy et al. [28]	S,C,R	12.5	210-1491	Air-water	-20, -10, -5, 0	160
Bhagwat and Ghajar [26]	S,C,R	12.5	150-1200	Air-water	0, 30, 60, 90	256
Bhagwat and Ghajar [27]	S,C,R	12.5	162-930	Air-water	-90, -75, -60, -45, -30, -20, -10, -5, 0	234
<b>Overall ranges</b>	<b>S, C, R</b>	<b>12.5 – 49.2</b>	<b>29 - 1491</b>	<b>Air-water</b>	<b>-90 - +90</b>	<b>1159</b>

\*S=single; C=circular; R=regular

The relative importance of inertia with respect to the buoyancy forces can be measured by the Froude number. Higher Froude values lead to a more symmetric flow, whose patterns are then more likely independent of the channel inclination. For low inertia, instead, the effects of the tube orientation can be different according to the channel diameter and the capillary forces due to the fluid surface tension, which can both be measured according to the Bond number: higher diameters and/or low surface tensions may easily lead to flow stratification and a greater impact of the channel orientation on the flow pattern and the heat transfer. Differently, low Bond numbers are usually associated with microchannel geometries, in which stratification may never appear, thus having a lower effect from the inclination.

Particularly, when  $Bd$  is lower than 1, the tube can be considered as a microchannel [42], and the effect of gravity should be overcome by the importance of the surface tension. In cases of  $Bd > 60$ , instead, the experiments can be considered to be macroscale [53], for which the relative importance of the capillary forces can be neglected. For intermediate values, a progressive change of the flow properties is expected. Regarding the vapor Froude number  $Fr_V$ , there is not a definite threshold between gravity- and inertia-driven phenomena. Cioncolini and Thome [36] found that for  $Fr_V > 10$  the difference between the top and bottom liquid film thicknesses during annular flow becomes negligible. Later, the same authors [38] found that the two-phase friction factor in annular flow was not dependent on gravity approximately when  $Fr_V > 100$ . Finally, Layssac et al. [34] and Lillo et al. [37] showed inertia-driven fluid characteristics and heat transfer for lower vapor Froude numbers (around  $Fr_V > 4$ ).

As a first analysis, Figure 1 provides the overall variations of the  $Bd$  and  $Fr_V$  dimensionless numbers for the collected heat transfer coefficient data points for the three types of heat transfer processes. The differences between the Froude and Bond number ranges for the three different configurations can be explained by the various applications of these three types of flow. For instance, the greater diameters used in experiments for condensation heat transfer lead to higher Bond numbers and a smaller range of Froude numbers than do those used for boiling heat transfer. For gas-liquid flows, the variation of the Bond number is very limited and includes only high values corresponding to macrochannels. The important characteristics of spreading and good distributions of all data points should enable performing a robust analysis of the effects of inertia and capillary and gravitational forces on the two-phase heat transfer.



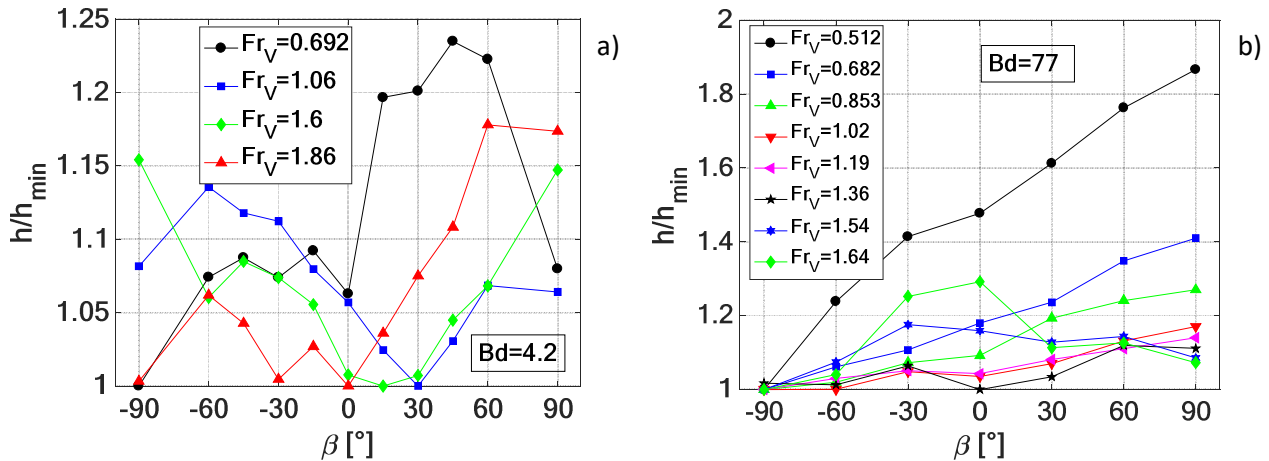
**Figure 1** Vapor Froude and Bond numbers overall ranges for the collected databases

### 3. Qualitative analysis of the effect of inclination on the heat transfer coefficient

To identify the main trends of the effect of the inclination on the heat transfer coefficient, a qualitative analysis is performed in the present section. Its goal is to identify the main phenomena affecting the heat transfer coefficient and to highlight the similarities and difference of behavior between the three types of two-phase flow analyzed in the present study.

### 3.1. Effect of inclination on flow boiling

To highlight the inclination effect during flow boiling, examples of experimental results are extracted from the corresponding database and presented in Figure 2. The evolution of the nondimensional heat transfer coefficient (which is the ratio of the heat transfer coefficient to the minimum value found by the authors for those operating conditions) is presented as a function of the inclination angle for vapor Froude and Bond numbers. The example extracted from the work of Layssac et al. [33] (Figure 2a) corresponds to a small Bond number that is, however, high enough to not consider the channel as being microscale. There is neither a clear effect of the  $Fr_V$  nor a particular recurrent optimum angle of inclination that maximizes the heat transfer performance. The variation of the heat transfer coefficient with the tube orientation is, in any case, limited, reaching a maximum of 24% for the lowest vapor Froude number of 0.69. Under this condition, the inclination angle that maximizes the heat transfer coefficient is  $\beta=45^\circ$ , whereas the lowest value is obtained at  $\beta=-90^\circ$  (vertical downward). More defined trends are observed in the work by Mohseni et al. [46] (Figure 2b), in which the  $Bd$  number is considerably higher. For the investigated operating conditions, the maximum variation of the heat transfer coefficient is higher for small  $Fr_V$  numbers and reaches a maximum of nearly 90%, even if this difference decreases rapidly with increasing vapor Froude numbers. Moreover, almost all the experiments share the same heat transfer coefficient increasing trends upon changing the inclination angles from vertical downward ( $-90^\circ$ ) to vertical upward ( $+90^\circ$ ), with the latter representing the channel orientation that maximizes the heat transfer performance. Similar trends with a clear decreasing inclination effect relative to the vapor Froude number have also been found by other authors [41] [43], even if the threshold  $Fr_V$  value for inclination dependency is different according to each paper and no general outcomes can be extrapolated.



**Figure 2** Evolution of the heat transfer coefficient for boiling experiments as a function of the inclination angle, for different vapor Froude numbers.  $h_{min}$  represents the minimum heat transfer coefficient among the investigated orientations. (a) Layssac et al. [33],  $Bd = 4.2$ ; (b) Mohseni et al. [46],  $Bd = 77$

A global analysis of the database is performed by determining the inclination angles that lead to the minimum and maximum heat transfer. In the present study,  $\beta_{min}$  and  $\beta_{max}$  are thus defined as the tube orientations at which the heat transfer coefficient respectively reaches the maximum and minimum values among the inclination angles investigated by each author. Figure 3 presents the  $\beta_{min}$  and  $\beta_{max}$  combinations for the flow boiling experiments. The size of each disk is related to the frequency of the corresponding combination, whereas the color bar reflects the effect of the inclination in terms of maximum-over-minimum heat transfer coefficient ratio over the range of investigated orientations. For instance, the biggest light blue disk indicates that 59% of the flow boiling studies have reported  $\beta=0^\circ$  as the orientation that minimizes the heat transfer coefficient and  $\beta=+90^\circ$  as the angle that maximizes the heat transfer performance. The frequencies of  $\beta_{min}$  and  $\beta_{max}$  corresponding to low heat transfer coefficient relative variations ( $h_{max}/h_{min} < 1.20$ ) are not considered for the presented data, since very low differences are not significant and could be included in the uncertainty range of the experimental data. In addition, data points coming from

multi-minichannel geometries are excluded from the present analysis, as the investigated orientations are very limited for these geometries (often only vertical and only horizontal). Moreover, the effect of gravity in multiple channels could be influenced or even mistaken for possible mal-distribution phenomena and back-flows, rather than as occurring in a single tube. From Figure 3, the maximum heat transfer coefficient is most often obtained with a vertical upward disposition, whereas the horizontal and vertical downward placements tend to penalize the heat transfer performance for boiling experiments. However, the  $0^\circ/90^\circ$   $\beta_{min}/\beta_{max}$  combination, though it is the most frequently encountered (see the biggest disk), produces low differences in the heat transfer coefficients (less than 35%). A more significant and recurring spread is obtained instead with the combination of  $-90^\circ/90$ . However, under some conditions, the inclination effect is limited ( $h_{max}/h_{min} < 1.3$ ), so there is a need for a better understanding of the phenomena affecting this parameter.

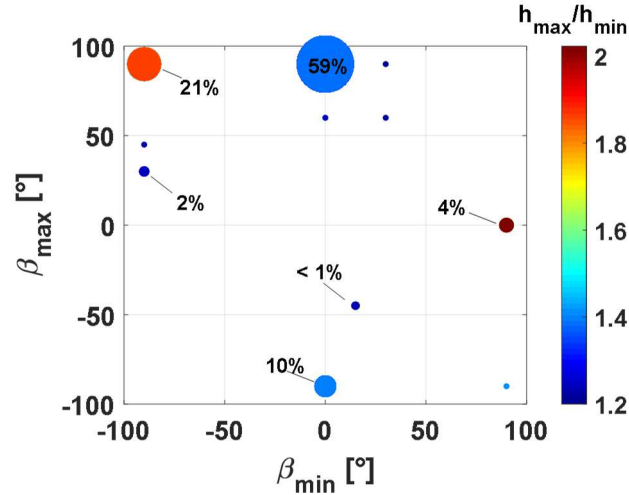
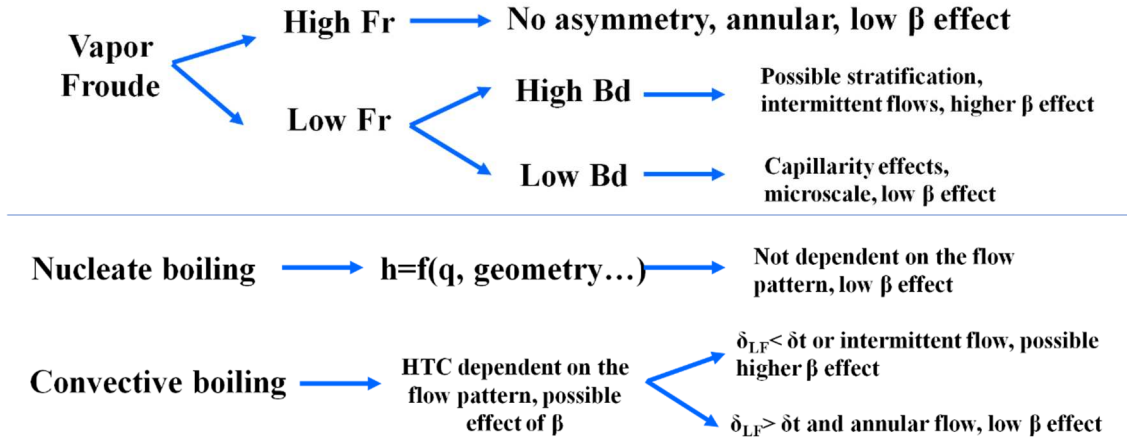


Figure 3 Combinations of  $\beta_{min}$  and  $\beta_{max}$  for the flow boiling experiments. The size of the disks refers to the corresponding combination frequency and the color bar indicates the spread of the heat transfer coefficients with varying inclination angles. The results having a heat transfer coefficient difference lower than 20% have not been included.

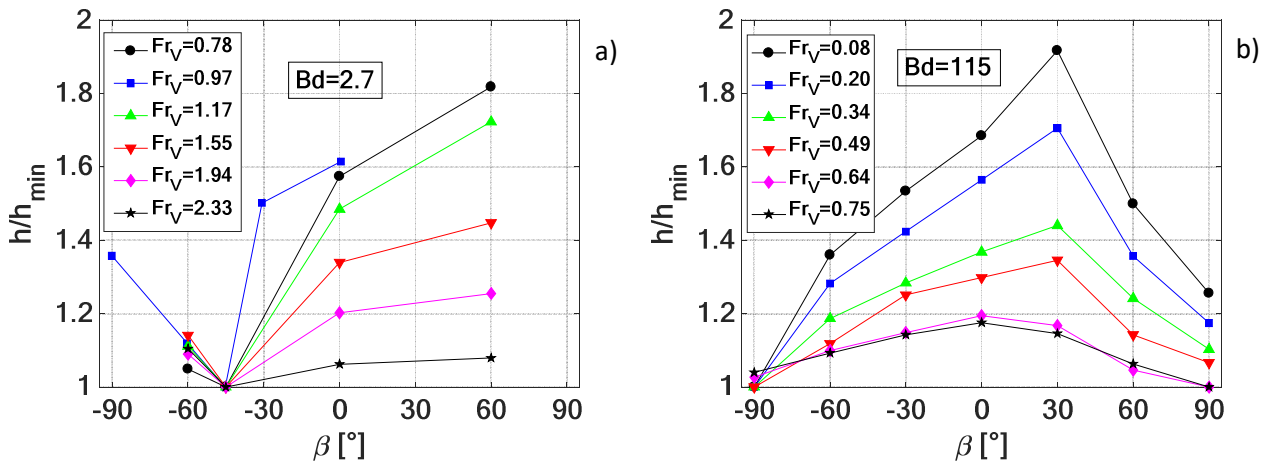
The behavior of the inclination effect on heat transfer during flow boiling can be explained by the fact that in the case of an evaporating flow, the heat transfer mechanism is regulated by both nucleation and convection phenomena. It has already been stated [32] [33] that the inclination angle poorly affects the flow patterns in the case of bubbly or slug flows, and thus, the heat transfer should not be influenced by the tube orientation when nucleate boiling is dominant. For convectively driven heat transfer, instead, the heat transfer coefficient is strongly dependent on the flow pattern (mostly annular flow) and may be easily affected by a change of the tube orientation. In this case, the height of the liquid film thickness relative to the thermal boundary layer is an important parameter. When the boundary layer is greater than the actual liquid film thickness ( $\delta_T > \delta_{LF}$ ), the thermal behavior is then affected by the liquid film, which in turns may vary with changes of the tube orientation. In the other case, when  $\delta_T < \delta_{LF}$ , variations of the liquid film thickness may not influence the heat transfer behavior [54]. The results in Figure 3 can thus be explained by a partial dry-out of the tube occurring when the liquid film thickness tends to be too small, which unfavorably affects the heat transfer coefficient. A vertical upward position often leads to churn, or unstable flows, which provide a very good wetting of the tube wall while breaking the thermal boundary involved in the flows. To summarize the inclination effect on flow boiling, Figure 4 presents a qualitative scheme of the possible key factors affecting its importance. Note that this analysis is only qualitative and needs to be supported by a quantitative analysis to develop a predictive tool that could help in the design of industrial systems.



**Figure 4** Qualitative scheme including the key factors stating whether flow boiling could be affected by gravity effects

### 3.2. Effect of inclination on convective condensation

The inclination effect on convective condensation is different from that on convective boiling. Figure 5 shows examples of the variation of the heat transfer coefficient (referring to the minimum value obtained) with the inclination angle for two different Bond numbers and for various Froude numbers. Although the experiments had a narrow channel ( $Bd=2.7$ ), the data from Del Col et al. [17] (Figure 5a) presents a non-negligible effect of gravity, with a heat transfer coefficient variation of up to 80% upon changing the tube orientation. The increase of the Froude number lowers this effect up to a threshold of approximately  $Fr_V = 2$ , in which the heat transfer coefficient differences are included within a 20% range. In most experiments, the inclination angle that maximizes the heat transfer coefficient is  $60^\circ$  (even if the trends forecast a possible further increase for the vertical upward configuration, which has not been investigated), whereas the minimum heat transfer coefficient is found for  $\beta = -45^\circ$ . Different trends are found by Mohseni et al. [19] (see Figure 5b), in which the tube diameter is larger ( $Bd=115$ ) and the effect of gravity is significant up to a vapor Froude threshold value of approximately 0.5. The inclination angle of  $30^\circ$  gives the maximum heat transfer performance, which is penalized for both vertical upward and downward configurations. Other authors in the present database [18] [24] [23] [22], though they found a strong variation of the heat transfer coefficient with the channel inclination, do not show a vapor Froude-dependent behavior.



**Figure 5** Evolution of the heat transfer coefficient for flow condensation experiments as a function of the inclination angle for different vapor Froude numbers.  $h_{min}$  represents the minimum heat transfer coefficient among the investigated orientations. (a) Del Col [17],  $Bd = 2.7$ ; (b) Mohseni et al. [19],  $Bd = 115$

Figure 6 presents the combinations of  $\beta_{min}$  and  $\beta_{max}$  that respectively minimize and maximize the heat transfer coefficient in the condensation database. The most recurring and also significant values of  $\beta_{max}$  are either negative or positive and are quite restricted within a close range including the horizontal disposition

(between  $-20^\circ$  and  $30^\circ$ ), whereas the values of  $\beta_{min}$  are distributed throughout the whole range of orientations, with higher frequencies of the vertical downward disposition. The most significant  $\beta_{min}/\beta_{max}$  combination is  $-90^\circ/0^\circ$ , while other recurring combinations at which  $h_{max}/h_{min}$  is higher than 2.0 also include  $-30^\circ/30^\circ$  and  $-90^\circ/-20^\circ$ . One can note that the range of the inclination effect is higher for convective condensation ( $h_{max}/h_{min}$  up to 3.2) than for convective boiling ( $h_{max}/h_{min}$  up to 2.0), which means that this type of flow is more sensitive to the inclination angle.

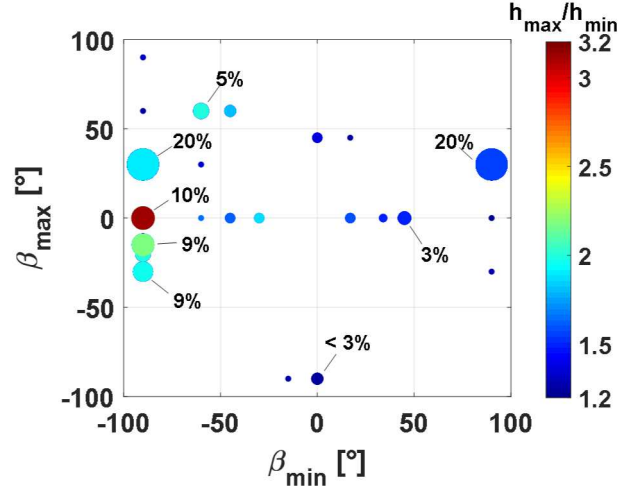


Figure 6 Combinations of  $\beta_{min}$  and  $\beta_{max}$  in the flow condensation experiments. The size of the disks refers to the corresponding combination frequency, and the color bar indicates the spread of the heat transfer coefficients with varying inclination angles. The results having a heat transfer coefficient difference lower than 20% have not been included.

The trends that are observed can be explained by the fact that contrarily to flow boiling, heat transfer during flow condensation is governed by convective phenomena, as the main thermal resistance is mainly induced by heat transfer conduction in the liquid film. In this regard, Figure 7 shows the maximum-over-minimum heat transfer coefficient ratio (obtained under the same operating conditions by changing only the tube inclination) as a function of the liquid film-over-thermal boundary layer for the condensation database. All the analyzed data points are colored differently according to the vapor quality. The liquid film thickness  $\delta_{LF}$  and the hydrodynamic boundary layer are obtained by the method of Hulburt and Newell [54], whereas the thermal boundary layer is calculated by using the Reynolds analogy of heat transfer:

$$\delta_T = \frac{\delta_u}{Pr_L^{1/3}} \quad (3)$$

As expected, the effect of inclination becomes progressively lower when the liquid film thickness increases with respect to the thermal boundary layer. Greater differences are found for  $\delta_{LF} \cong \delta_T$  (with heat transfer coefficients varying by more than 300% with different channel inclinations) and for low vapor quality values. When  $x > 0.6$ , instead, most of the differences are lower than 20% and that can be included in the uncertainty range of the collected data.

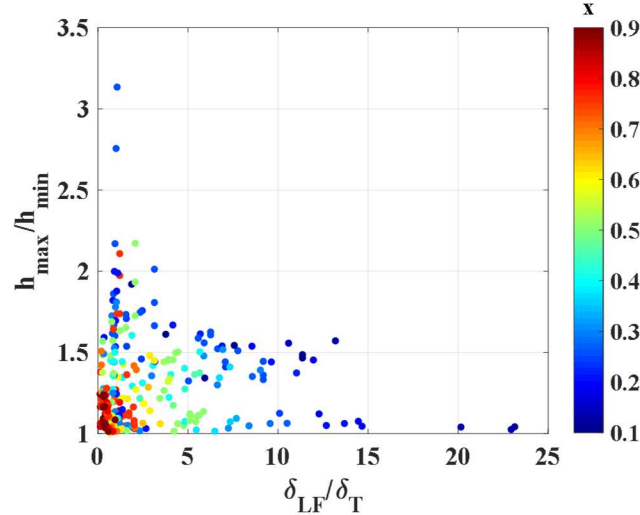


Figure 7 Maximum-over-minimum heat transfer coefficient ratio as a function of the liquid film thickness-over-thermal boundary layer ratio for different vapor qualities. All the data belong to the condensation database.

In conclusion, condensing flows are mostly affected when thin liquid films are involved and when they are themselves affected by the inclination angles. This explains why the maximum heat transfer coefficient can be found for inclination angles ranging from  $-15^\circ$  to  $-30^\circ$ , which correspond to angles at which the liquid films are the smallest (small liquid hold-up and stratified flows). For upward flows, the liquid hold-up increases, and churn flows are often encountered. Inclination angles ranging from  $30$  to  $60^\circ$  can thus both lead to a minimum heat transfer coefficient if the liquid hold-up (and, thus, the liquid film thickness) increases without changing the flow regime or to a maximum heat transfer coefficient if the churn flow breaks the thermal boundary layer. In any case, vertical flows (upward or downward) almost never correspond to an optimal configuration, as the symmetry of the flow pattern prevents the creation of very thin liquid films.

### 3.3. Inclination effect on heated air-water flows

The comparison of the behavior of these three types of flows should enable the identification of the main phenomena affected by the inclination angle. The relative evolution of the heat transfer coefficient with the inclination angle, taking into account the vapor and liquid Froude numbers, is shown in Figure 8 for two different Bond numbers for data belonging to the heated gas-liquid database. Unlike flow boiling and condensation, for the air-water experiments, the liquid and gas superficial velocities can be independently imposed. The diagrams therefore consider this further degree of freedom by using the liquid Froude number, which is defined as:

$$Fr_L = \frac{u_{SL}}{\sqrt{gd}} \quad (4)$$

The works found for heated air-water are very limited, and the range of inclinations investigated is not very wide, further complicating the overall analysis. The study by Bhagwat et al. [27] (see Figure 8a) was performed with a fixed Bond number equal to 21 and negative inclination angles from horizontal to vertical downward. In this work, there is a clear dependency of the liquid Froude number on the effect of inclination, which displays a parable-shaped curve with changing angles but gives the maximum heat transfer coefficient (up to approximately three times the minimum value) when a vertical downward configuration is chosen. Upon changing the vapor Froude number, instead, no particular differences are found. The same authors [26] also studied the effect of inclination for positive angles, but unfortunately, the two mentioned studies cannot be analyzed together since the operating conditions investigated in the original papers are different. From the work of Ghajar and Tang [29] (see Figure 8b) with fixed  $Bd=105$  and  $Fr_v=0.22$ , there is still an evident influence of the liquid Froude number on the effect of inclination, which may change the heat transfer

coefficient by up to 2.5 times. Among the different configurations ( $0^\circ$ ,  $2^\circ$ ,  $5^\circ$  and  $7^\circ$ ), the  $7^\circ$  inclination angle always returns the highest heat transfer coefficients.

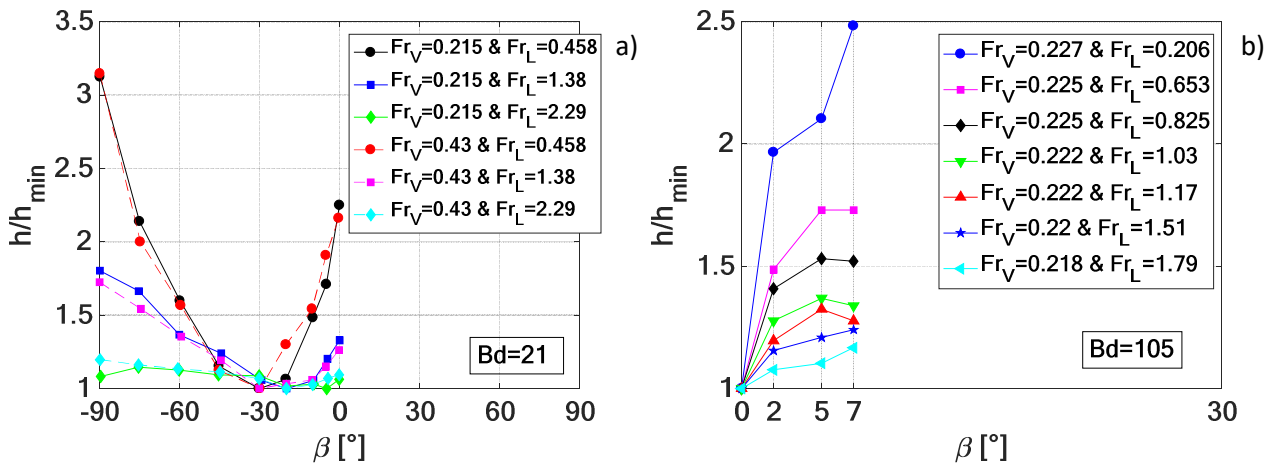


Figure 8 Evolution of the heat transfer coefficient for air-water experiments as a function of the inclination angle for different vapor and liquid Froude numbers.  $h_{min}$  represents the minimum heat transfer coefficient among the investigated orientations. (a) Bhagwat et al. [27],  $Bd = 21$ ; (b) Ghajar and Tang [29],  $Bd = 105$

The summary of the combinations and frequencies of the worst and best inclination angles is shown in Figure 9. The horizontal and  $-30^\circ$  orientations exhibit the minimum heat transfer coefficient for most of the cases in which they have been used in the experiments. Smaller frequencies are found for vertical upward and vertical downward dispositions (with the latter having a higher magnitude of the effect of orientation), whereas the inclination angles corresponding to the highest heat transfer coefficients are more uniformly distributed. The most significant  $\beta_{min}/\beta_{max}$  combination in terms of heat transfer coefficient variation and frequency are found for  $0^\circ/7^\circ$  and  $0^\circ/90^\circ$ . In the first case, the ratio between maximum and minimum heat transfer coefficient can reach 10, which is considerably higher than the values observed for boiling and condensation two-phase flows.

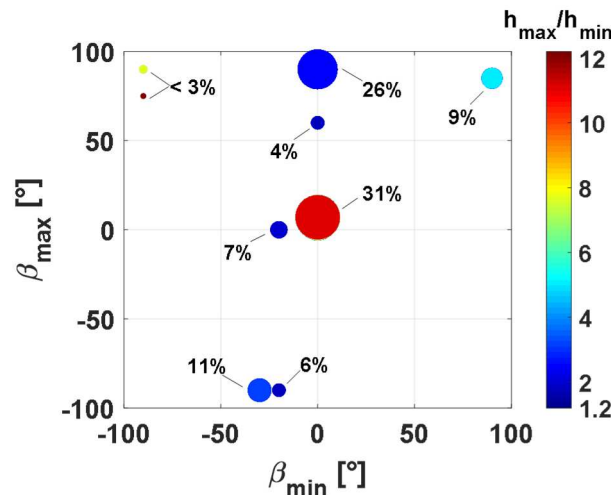


Figure 9 Combinations of  $\beta_{min}$  and  $\beta_{max}$  for gas-liquid experiments. The size of the disks refers to the corresponding combination frequency, and the color bar indicates the spread of the heat transfer coefficients with varying inclination angles. The results having a heat transfer coefficient difference lower than 20% have not been included.

Keeping in mind the weakness of the air-water flow database, the comparison between Figure 6 and Figure 9 is interesting, as the differences between condensing and air-water flows mainly rely on the additional convective thermal resistance that occurs in air-water flows between the liquid and gas phases and on the behavior of the small bubbles which tend to collapse quickly in condensing flows, contrarily to the case in air-water flows. If vertical flows are rarely the best configuration for condensing flows, this is not the case

for air-water flows: vertical air-water flows induce more small bubbles, which may break the thermal boundary layers and thus enhance the heat transfer.

In conclusion, a global summary of the present qualitative analysis is shown in Table 4. The effect of the inclination angle on the heat transfer coefficient in terms of the maximum ratio  $h/h_{min}$  is not the same for the three heat transfer mechanisms. First, in the case of boiling and condensation, the relative variation is somehow restrained (<320%) compared to the greater ratios observed in the heated air-water studies. Then, the most recurring orientations that minimize ( $\beta_{min}$ ) and maximize ( $\beta_{max}$ ) the heat transfer coefficients are also different according to the three databases. Finally, even if a non-negligible effect of the Froude and Bond numbers on the gravity influence is observed, their threshold values are not unique for the three heat transfer mechanisms and are seen to change from one author to another, implying that single investigations of these two numbers are not sufficient to disclose the overall physics. As a consequence, even if the present qualitative analysis enables highlighting the main trends of the inclination effect on the heat transfer and highlighting the similarities and differences between flow boiling, convective condensation and air-water flows, a deeper quantitative analysis seeking all the parameters involved is required for the development of a tool that captures the effect of the inclination angle on two-phase heat transfer flows.

**Table 4** Summary of the effect of the inclination angle for the entire database

	Maximum value of $h/h_{min}$	Most recurrent $\beta_{min}$	Most recurrent $\beta_{max}$	Most significant and recurrent $\beta_{max}/\beta_{min}$ combination
Flow boiling	2.0	-90° and 0°	+90°	+90°/0°
Convective condensation	3.2	-90°	+30°	+45°/-90°
Heated air-water	12.2	0°	+85°	+85°/0° and +7°/0°

#### 4. Quantitative analysis of the effect of inclination on the heat transfer coefficient

A quantitative analysis of the database is proposed in the present section. It is mainly based on a statistical approach and is thus complementary to the qualitative analysis in the previous section, which was based on a phenomenological approach. In the first step, the existing correlations dealing with the inclination effect on heat transfer are studied, and new predictive tools are then proposed.

##### 4.1. Assessment of existing correlations for inclined tubes

Few researchers have tried to quantify the effect of the tube orientation on the two-phase heat transfer coefficient by including the inclination angle in their prediction methods. For flow boiling heat transfer, Akhavan-Behabadi and Esmailpour [43] provided a simple expression of the two-phase Nusselt number, which was developed for their own database with a corrugated tube of 8.3 mm ID:

$$Nu = 0.00379 \cdot Re_L^{1.02} \cdot F_\beta^{0.11} \cdot \left( \frac{Pr_L}{X_{tt}} \right)^{0.92} \quad (5)$$

The parameter  $F_\beta$  takes into account the inclination angle and is obtained according to the experimental vapor quality:

$$\begin{aligned} F_\beta &= 1 + 0.25 \cdot (1+x)^{0.6} \cdot \sin(\beta) & \text{for } x \leq 0.7 \\ F_\beta &= 1 - 0.6x^{0.97} \cdot \cos(\beta - 10^\circ) & \text{for } x > 0.7 \end{aligned} \quad (6)$$

The calculated heat transfer coefficients from Akhavan-Behabadi and Esmailpour [43] are compared with the boiling database in Figure 10a. This prediction method largely underestimates the collected points, especially when the heat transfer coefficient is high and in the vertical upward disposition, and it works reasonably well

only with the original authors' database or under similar operating conditions. By defining the mean absolute error (MAE) as:

$$MAE = \frac{1}{N} \sum \frac{|h_{pred} - h_{exp}|}{h_{exp}} \cdot 100 \quad (7)$$

the overall calculated MAE for this correlation is 69.9%.

For flow condensation heat transfer, Adelaja et al. [55] modified the original Cavallini et al. [56] correlation developed for horizontal smooth tubes. The authors categorized the flow in the gravity and wall temperature difference-independent region (for annular, annular-wavy, intermittent and mist flows, in which the nondimensional gas velocity  $J_V$  is higher than a threshold value  $J_{V,T}$ ) and in the gravity and wall temperature difference-dependent regime (for smooth stratified and stratified wavy flows, in which  $J_V < J_{V,T}$ ). In the first case, the heat transfer coefficient expression was found to be a function of the Dittus-Boelter [57] liquid heat transfer coefficient and was not correlated with the inclination angle  $\beta$ :

$$h = h_L \left( 1 + 0.8247 X_{tt}^{-0.2245} Ja^{-0.23063} \left( \frac{J_V}{J_{V,T}} \right)^{-0.20727} \right) \quad (8)$$

In the second case, the authors used two different expressions of the heat transfer coefficient according to the inclination angle range:

$$h = h_L \left( 1 + 0.0422 X_{tt}^{-0.2056} Ja^{-0.5672} \left( \frac{J_V}{J_{V,T}} \right)^{-0.505} Bd^{0.0316} (3 + \cos(\beta))^{1.3492} \right) \quad \text{for } -90^\circ \leq \beta < -30^\circ$$

$$h = h_L \left( 1 + 0.5191 X_{tt}^{-0.3153} Ja^{-0.3214} \left( \frac{J_V}{J_{V,T}} \right)^{-0.6393} (\cos(\beta) + \sin(\beta))^{-0.1506} \right) \quad \text{for } -30^\circ \leq \beta < +90^\circ \quad (9)$$

The Jakob number represents the wall subcooling as shown in Eq. (10):

$$Ja = \frac{c_{p,L} (T_{sat} - T_w)}{i_{LV}} \quad (10)$$

The nondimensional vapor velocity is analogous to the vapor Froude number:

$$J_V = \frac{xG}{[gd\rho_V(\rho_L - \rho_V)]^{0.5}} \quad (11)$$

and its transition value was defined as follows, as a function of only the Martinelli parameter:

$$J_{V,T} = \left[ \left( \frac{7.5}{(4.3X_{tt}^{1.111} + 1)} \right)^{-3} + 2.4^{-3} \right]^{-1/3} \quad (12)$$

The comparison of the Adelaja et al. [55] prediction method against the present condensation database is shown in Figure 10b. Most of the data are underestimated, regardless of the value of the inclination angle, providing an overall MAE of 60.1%.

A similar approach has been followed by Shah [58], who proposed a suite of correlations for condensation heat transfer in conventional and micro-scale geometries in all orientations, by using a linear interpolation between the heat transfer coefficients evaluated at  $0^\circ$  and at  $-90^\circ$ . Due to the length and complexity of the

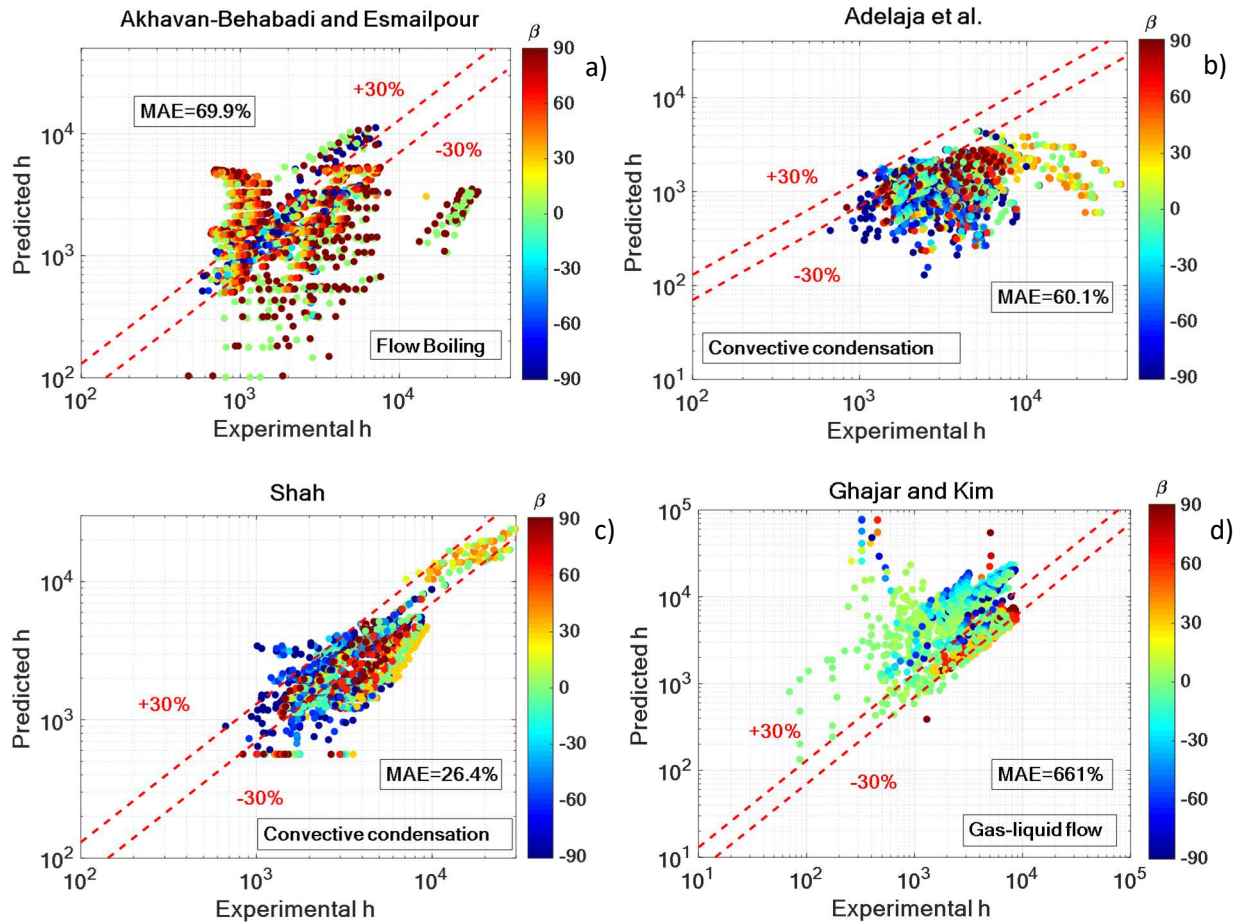
mathematical expression, the details of the equations are available in the reference mentioned. The comparison of the correlation by Shah [58] against the condensation database is shown in Figure 10c. The prediction method works reasonably well for high heat transfer coefficients at positive inclination angles, whereas it fails at lower heat transfer efficiencies for all orientations; however it provides an overall MAE of 26.4%.

Finally, Ghajar and Kim [59] proposed a heat transfer model for heated gas-liquid flow, taking into account the effects of the tube orientation with the inclination parameter  $Z$  and the flow pattern factor  $F_p$ :

$$Z = 1 + \frac{gd(\rho_L - \rho_V)\sin(\beta)}{\rho_L u_{SL}^2} \quad (13)$$

$$F_p = \left( \frac{S_{L,eff}}{\pi d} \right)^2 \quad (14)$$

The first parameter represents the relative force acting on the liquid phase in the flow direction due to the momentum and gravity forces, while the flow pattern factor tries to capture the realistic shape of the gas-liquid interface and the effective wetted perimeter. Further details on the mathematical expression are available in the original reference [59]. This correlation is then tested against our heated air-water database in Figure 10d, in which it is clear that most of the points are strongly overestimated, especially for horizontal and downward flow orientations. A better agreement is instead found for positive channel inclinations, for which all the data are included in a  $\pm 30\%$  error band. However, the overall calculated MAE is very high (661%).



**Figure 10** Comparison between experimental and predicted heat transfer coefficients for the whole range of tube inclinations. (a) Correlation of Akhavan-Behabadi and Esmailpour [43] for boiling heat transfer; (b) Correlation of

Adelaja et al. [55] for condensation heat transfer; (c) Correlation of Shah [58] for condensation heat transfer; (d) Correlation of Ghajar and Kim [59] for gas-liquid flow

#### 4.2. Inclination effect $I_h$ : definition and development of a practical tool

The assessment of the existing correlations has shown a very poor agreement with the experimental data, and this is mostly due to the shortage of experimental points available in open literature. The limited database, in fact, does not allow the development of a solid prediction method for the evaluation of the two-phase heat transfer coefficient as a function of the experimental conditions including the inclination angle. In this regard, the aim of the present work is to provide a simple tool to establish whether the heat transfer coefficient might be affected by the channel orientation or whether a typical correlation for horizontal tubes can be used. The analysis is limited to boiling and condensation heat transfer, since the amount of data in the case of heated air-water is scarce and the investigated values of the inclination angles are quite limited to nearly horizontal or nearly vertical configurations.

The inclination effect  $I_h$  is defined herein as the expected maximum relative heat transfer coefficient variation brought about by changing the channel orientation. The reference state is represented by the heat transfer coefficient evaluated through a correlation for horizontal two-phase flow. The use of a prediction method helps in obtaining an impartial reference for all the data points.

$$I_h = \frac{\Delta h_{\max}}{h_{0^\circ, REF}} \quad (15)$$

Among the numerous available correlations for the two-phase heat transfer coefficient in the horizontal position, the prediction method of Liu and Winterton [60] and Shah [61] are chosen for boiling and condensing flows, respectively. Both correlations have the advantage of being easy to implement and recognized [62] as benchmark methods in scientific literature on this topic.

The tool development has been carried out using the approach of Layssac et al. [34] for the prediction of the symmetry parameter. Specifically, all the dimensional parameters playing a role in the gravity influence on two-phase heat transfer can be taken into account:

$$I_h = f(G, d, x, (1-x), q, \rho_L, \rho_V, \mu_L, \mu_V, \sigma, k_L) \quad (16)$$

To be physically consistent, all the flow parameters need to be grouped together as dimensionless numbers in the final form of the correlation. According to the physical analysis performed in the section 3 of the present paper, all nondimensional groups considered are displayed in Eq. (17):

$$I_h = f(Bd, Fr_V, Bo, \frac{1-x}{x}, X_{tt}, \frac{\delta_{LF}}{\delta_T}, Re_L, We_L) \quad (17)$$

As mentioned, the Bond number  $Bd$  gathers the diameter effects ( $d$ ) and the capillary forces ( $\sigma$ ), whereas the vapor Froude number  $Fr_V$  takes into account the vapor inertia and the buoyancy forces, respectively, through the use of the vapor mass flux ( $Gx$ ) and an additional term ( $gd$ ). The heat flux ( $q$ ) for boiling heat transfer is considered in the Boiling number  $Bo$ . For condensation heat transfer, the heat flux is not considered as a parameter, since not all the original sources shared this information. Moreover, most of the data for condensation heat transfer fall within the  $\Delta T$  (wall subcooling)-independent regime according to the classification given by Cavallini et al. [56], and therefore, the effect of heat flux should not be significant. The effects of the vapor quality  $x$  and the thermophysical properties are taken into account in the group  $\frac{1-x}{x}$  and in the Martinelli parameter  $X_{tt}$ . The ratio of the liquid film thickness to the thermal boundary layer  $\delta_{LF}/\delta_T$  calculated in the model by Hulburt and Newell [54] accounts for the inertia ( $G$ ), vapor quality ( $x$ ) and liquid thermal conductivity ( $k_L$ ), among other physical properties. Finally, the liquid Reynolds number  $Re_L$

and the liquid Weber number  $We_L$  are also used to group together the liquid inertia ( $G(1-x)$ ) and viscosity forces ( $\mu_L$ ) and the liquid inertia and capillary forces ( $\sigma$ ), respectively.

$$Re_L = \frac{G(1-x)d}{\mu_L} \quad (18)$$

$$We_L = \frac{[G(1-x)]^2 d}{\rho_L \sigma} \quad (19)$$

The parameters are then shaped into a function that is not upper-confined, taking the following form:

$$I_h = a_0 \prod_i p_i^{a_i} \quad (20)$$

where  $p_i$  represents the generic nondimensional parameter among the  $N$  considered, while  $a_1, \dots, a_N$  are the corresponding exponents and  $a_0$  is a constant. The optimization algorithm is based on the multidimensional unconstrained nonlinear minimization and has been carried out with the MATLAB software. The function to be minimized is chosen as the root mean square error (RMSE):

$$RMSE = \sqrt{\frac{1}{N} \sum (I_{h,exp} - I_{h,pred})^2} \quad (21)$$

The algorithm starts by finding the initial constant  $a_0$  and the sets of exponents  $a_1, \dots, a_N$  that minimize the RMSE. Then, a sensitivity analysis of the exponent values is performed to reveal and keep only the independent parameters that affect the minimization function. As a matter of fact, the RMSE can be more or less influenced by each exponent variation during the research of its minimum value. Due to the limited diversification of the database, some of the parameters are not independent of one another and can therefore be excluded. Further details and the complete process of the optimization algorithm for the boiling and condensation databases are provided in the Appendix section.

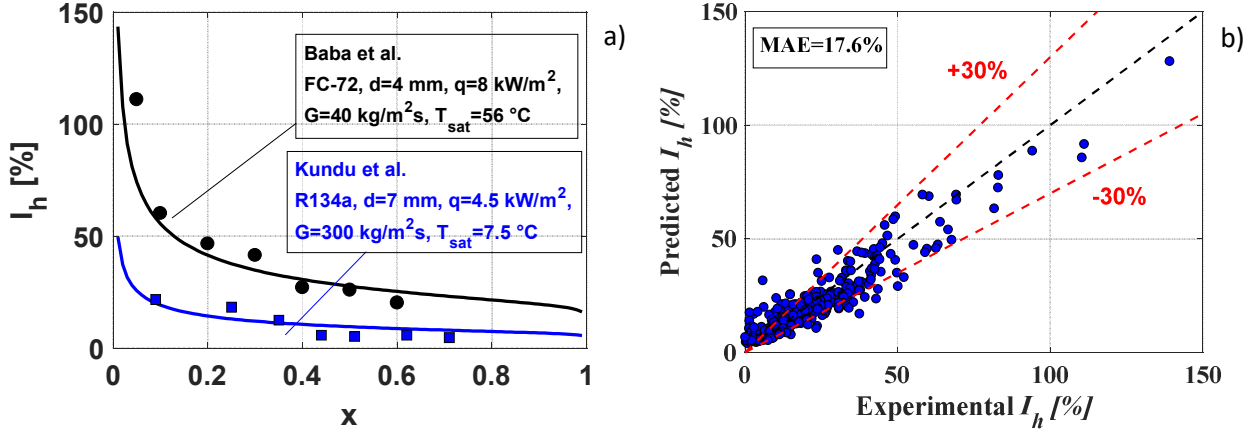
For boiling heat transfer, the RMSE is found to be mostly influenced by only the vapor Froude number, the Boiling number and the liquid-to-vapor mass quality ratio, with the *Inclination effect*  $I_h$  assuming the final form of Eq. (22), which returns a RMSE of 10.8%:

$$I_{h,boil} = 124 \cdot Fr_v^{-0.35} \cdot Bo^{0.23} \cdot \left(\frac{1-x}{x}\right)^{0.064} \quad (22)$$

As expected, the negative exponent of the Froude vapor number indicates a lower inclination effect with increasing flow velocity and with decreasing tube diameter. Surprisingly, the boiling database has led to a positive exponent of the Boiling number, showing that higher heat fluxes exhibit a more emphasized effect of the inclination angle on heat transfer during evaporation. However, the influence of gravity on nucleation in flow boiling has not been fully established yet, and no further outcomes are extrapolated at this point. More experimental data would be of great use for in achieving better understanding of this phenomena. Finally, the effect of the liquid-to-vapor quality ratio is limited and represented by a positive exponent of 0.064. Surprisingly, the Bond number does not seem to have a major influence on the condition of the database, which tends to show that the inclination effect is mainly governed by convective and boiling heat transfer, successfully characterized by the vapor Froude and Boiling numbers only.

The accuracy of the present tool for boiling heat transfer is shown in Figure 11, where the model is first compared to several data points taken from the two evaporating heat transfer studies of Baba et al. [40] and Kundu et al. [47]. The two data sets were chosen to show the accuracy of the developed correlations for different fluids and at explicitly dissimilar operating conditions. The first trend represents the coolant FC-72 used in a single tube of 4 mm ID with a very low mass velocity of 40 kg/m<sup>2</sup>s and a heat flux of 8 kW/m<sup>2</sup>. The latter refers to R134a in a wider channel of 7 mm, with an imposed heat flux of 4.5 kW/m<sup>2</sup> and a higher mass

velocity of 300 kg/m<sup>2</sup>s. In both cases, the agreement with the model that replicates the experimental trends and values is satisfactory. The comparison between the experimental and predicted  $I_h$  values for the entire flow boiling database is shown in Figure 11b. By excluding the points for which the experimental  $I_h$  is lower than 20%, almost all the database is predicted within an error band of  $\pm 30\%$ .



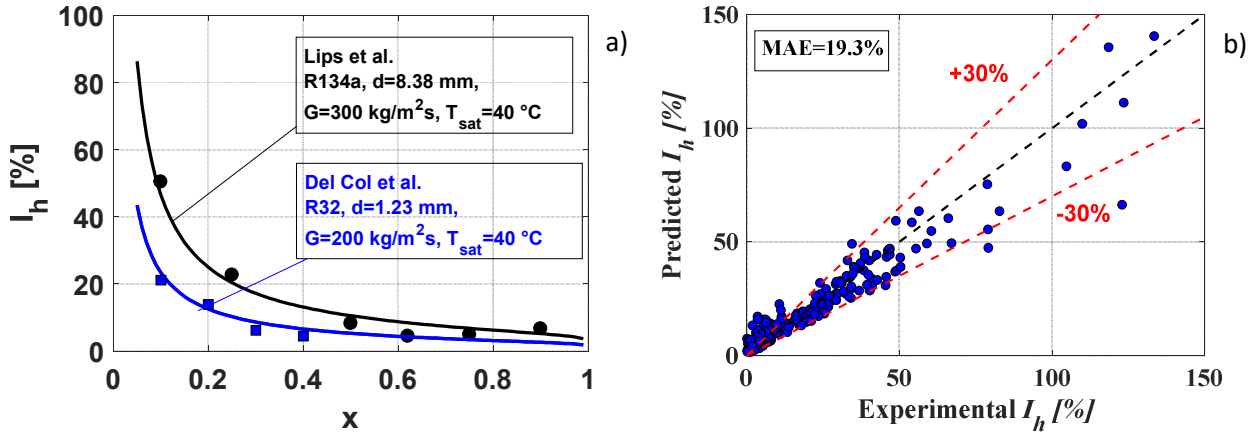
**Figure 11** (a) Validation of the  $I_h$  model with the boiling database for two different studies. (b) Comparison with the entire flow boiling database

The same procedure, of which details are given in the Appendix section, has been followed for the condensation heat transfer, returning the expression of the *Inclination effect*  $I_h$  as shown in Eq. (23):

$$I_{h,cond} = 10.1 \cdot Fr_V^{-0.76} \cdot Bd^{0.060} \cdot \left( \frac{1-x}{x} \right)^{0.37} X_{tt}^{-0.28} \quad (23)$$

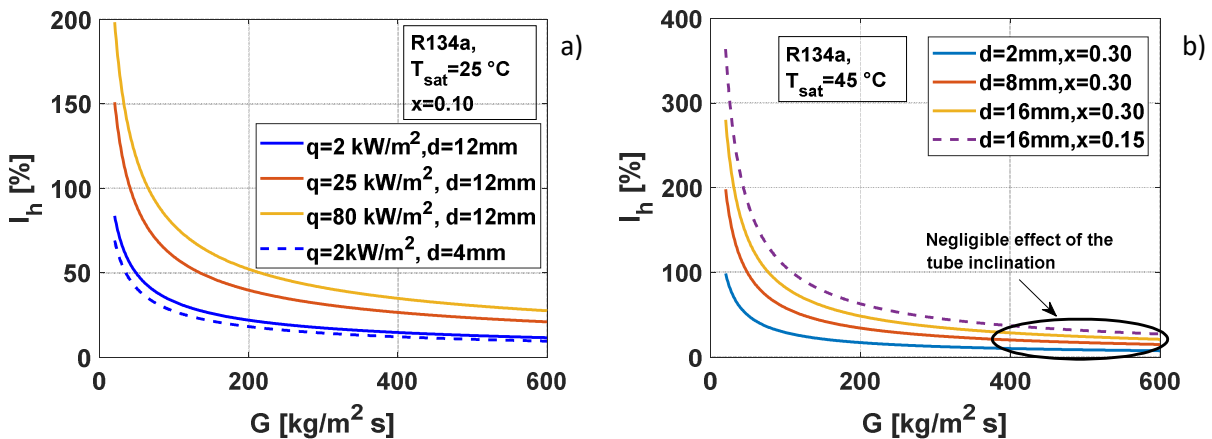
Similar to the case for the boiling tool, the flow inertia decreases the influence of the tube orientation on the heat transfer coefficient by means of the negative exponent of the vapor Froude number. The global effect of the tube diameter (included in the vapor Froude and the Bond numbers) reflects in an overall exponent of 0.50. The vapor quality and the remaining thermodynamic properties provide their influence through the liquid-to-vapor quality ratio and the Martinelli parameter. The ratio of the liquid film thickness and the thermal boundary layer surprisingly does not appear in this correlation, but this parameter may depend on the other parameters involved in Eq. (23).

The accuracy of this tool for convective condensation is shown in Figure 12a-b. The predicted Inclination effects are first compared with the experimental values obtained from the studies of Lips et al. [20] (with the refrigerant R134a at 40 °C in a regular channel of 8.38 mm ID and with a mass flux of 300 kg/m<sup>2</sup>s) and that of Del Col et al. [17] (with R32 at 40 °C in a square 1.23 mm minichannel at a mass flux of 200 kg/m<sup>2</sup>s). In both studies, the new model returns a fair agreement with the experimental trends and data. The comparison with the entire condensation database is shown in Figure 12b. Almost all the experimental data with a significant Inclination effect (>20%) are well predicted falling into an error range of  $\pm 30\%$ .



**Figure 12** (a) Validation of the  $I_h$  model with the condensation database for two different studies. (b) Comparison with the entire convective condensation database

According to the new models, the influence of the operating parameters on the Inclination effect  $I_h$  is shown in Figure 13a-b for boiling heat transfer and convective condensation, respectively. For boiling heat transfer (Figure 13a), the refrigerant R134a at a saturation temperature of 25 °C and a vapor quality of  $x=0.10$  is considered. With the increase of the mass flux, the effect of the orientation decreases rapidly: at  $G=600$  kg/m<sup>2</sup>s,  $I_h$  reaches 20%, meaning that the heat transfer coefficient might be easily evaluated by the Liu and Winterton [60] prediction method for horizontal tubes without facing significant errors. A reduced influence of the tube diameter (from 4 to 12 mm) is found, and for the boiling contribution, a higher heat flux leads to a gradual increase of the Inclination effect. Figure 13b shows the influence of the mass velocity, tube diameter and vapor quality on the condensation  $I_h$  when using the refrigerant R134a at a saturation temperature of 45 °C. Under these conditions, the effect of the tube orientation decreases with the mass velocity and becomes unimportant at  $G < 500$  kg/m<sup>2</sup>s. Larger channels and lower vapor qualities instead lead to an increase of the Inclination effect. For  $G=20$  kg/m<sup>2</sup>s,  $d=8.0$  mm and a vapor quality of 0.30, the heat transfer coefficient in an inclined tube might be up to 200% different from the value evaluated in the horizontal position with the Shah [61] correlation, causing considerable errors for design purposes.



**Figure 13** Predicted trends of the *Inclination effect*  $I_h$  for boiling heat transfer and convective condensation. (a) Evolution of the boiling  $I_h$  with the mass flux, channel diameter and imposed heat flux for the refrigerant R134a at 25 °C and a vapor quality of 0.10. (b) Evolution of the convective condensation  $I_h$  with the mass flux, channel diameter and vapor quality for the refrigerant R134a at 45 °C.

## 5. Conclusions and future perspectives

A global analysis of the effect of the tube inclination on two-phase heat transfer for boiling, condensation and heated gas-liquid flow has been performed in the present study. A total amount of 5408 data points from

several studies experimenting with at least two different channel inclinations were collected and analyzed. The main outcomes of this work are summarized as follows:

- There is not a clear specific inclination angle that enhances or penalizes the two-phase heat transfer coefficient, the trends of which relative to the tube orientation can be either affected by the vapor Froude number or insensitive to changes of the gravity vector. Statistically, the channel orientations minimizing/maximizing the heat transfer coefficients are  $0^\circ/90^\circ$ ,  $-90^\circ/30^\circ$  and  $0^\circ/85^\circ$  for boiling, condensation and gas-liquid flow, respectively.
- The existing heat transfer coefficient correlations for two-phase flow taking into account the effect of the channel orientation show very poor agreement with the experimental data due to the shortage of experimental points available in open literature. For boiling heat transfer, the prediction method of Akhavan-Behabadi and Esmailpour [43] returns an MAE of 69.9%, and for condensing flows, the correlations of Adelaja et al. [55] and of Shah [58] provide overall MAE values of 60.1% and 26.4%, respectively. Finally, the Ghajar and Kim [59] method for heated air-water works reasonably only for positive inclination angles and returns an MAE of 661%.
- The parameter *Inclination effect*  $I_h$  has been defined as the expected maximum relative heat transfer coefficient variation produced by changing the channel orientation. This parameter forecasts the maximum spread of the heat transfer coefficient when using an inclined tube with respect to the predicted value for a horizontal arrangement. The correlations used as references are those of Liu and Winterton [60] for boiling and Shah [61] for condensation. Due to the small range of investigated angles and the limited database at disposal, the Inclination effect has not been proposed for gas-liquid flows. This tool can be employed to provide useful information for the correct design of condensing and evaporating units conceived to work at a specific inclination angle.
- Using an unconstrained nonlinear optimization statistical tool, the Inclination effect is correlated with the most influencing nondimensional groups. For both boiling and condensation heat transfer, the  $I_h$  is found to be a decreasing function of the vapor quality and mass flux, whereas it increases with larger tubes. The RMSE values of the two correlations with the corresponding databases are respectively 10.8% and 15.1%. A higher heat flux in flow boiling also leads to a higher  $I_h$ . These correlations can be very useful tools when designing a system. Even if they are not able to predict the heat transfer coefficient for each inclination angle (which is, however, also not satisfactorily provided by existing methods), they enable quantification of the maximum error that is obtained using a correlation developed for horizontal flow in an inclined configuration.

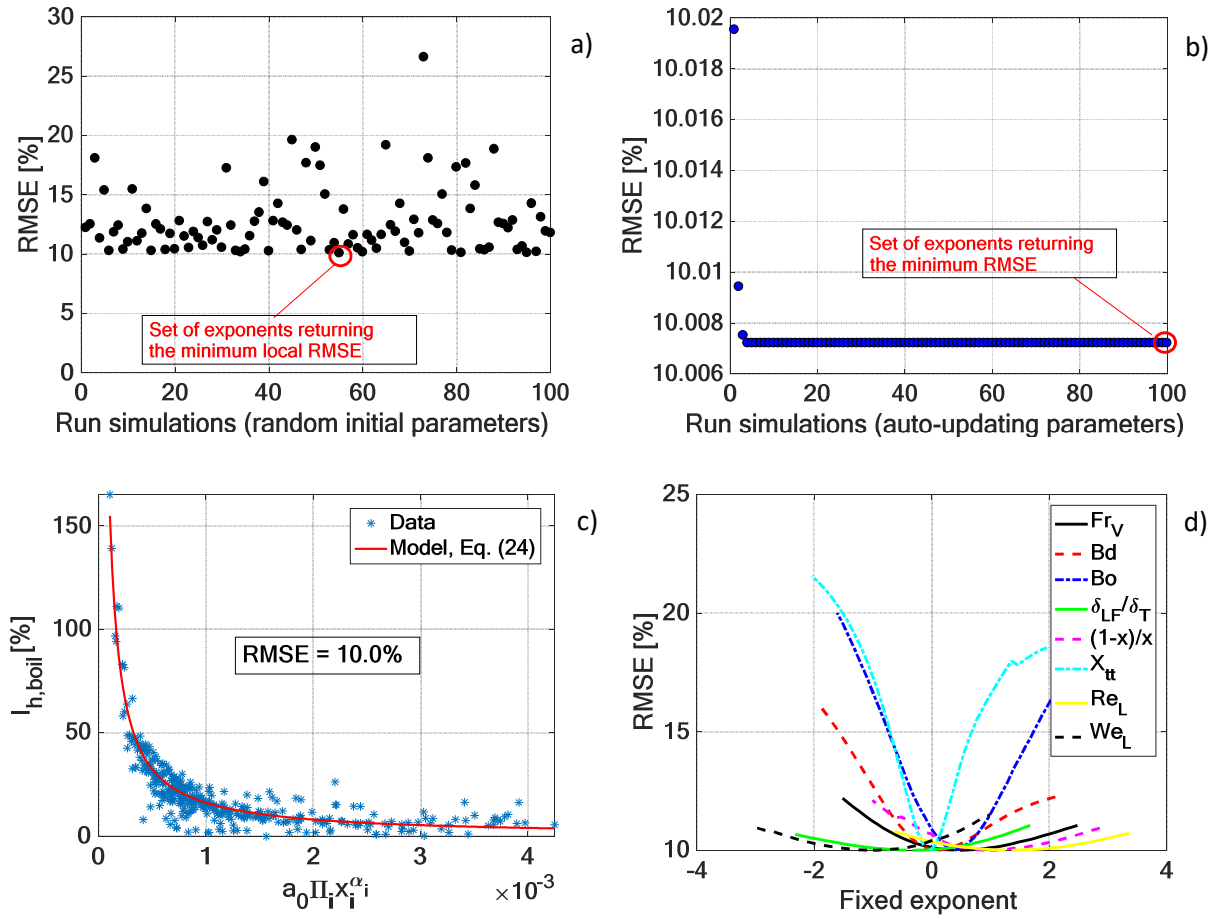
## Appendix

Due to the complexity of the database and the numerous parameters included in the correlation, the RMSE minimization procedure follows an unconstrained nonlinear optimization and is undertaken in different steps:

1. The exponents  $a_i$  and the initial constant  $a_0$  are set randomly between 0 and 1. With these values, 100 simulations are run and the set of coefficients that leads to the local minimum is acquired (see the circled red point in Figure 14a for boiling heat transfer).
2. A new set of 100 simulations is run, starting from the set of coefficients taken previously and auto-updating the results for the upcoming simulation. The final set of coefficients is taken from the last run, which always returns the minimum value of the RMSE. For all cases presented, the minimum RMSE is reached quite soon, after a couple of simulations performed in stage 2 (see Figure 14b). For this reason, there is no need to increase the number of runs. At the end of the second step, the final values of the constants  $a_0, a_1, \dots, a_N$  are determined. For the boiling heat transfer analysis, the complete expression is given in Eq. (24), and the comparison with the experimental  $I_h$  is shown in Figure 14c, with the minimum RMSE of 10.0%.

$$I_{h,boil} = 0.0173 \cdot Fr_v^{0.48} Bd^{0.11} Bo^{0.38} \left( \frac{\delta_{LF}}{\delta_T} \right)^{-0.33} \left( \frac{1-x}{x} \right)^{0.98} X''^{-0.038} Re_L^{1.3} We_L^{-0.99} \quad (24)$$

3. A sensitivity analysis of the quality of the optimization relative to each exponent is performed. As stated by Layssac et al. [34], the calculated RMSE can be more or less affected by variation of each exponent during the research of the minimum. Therefore, the third step consists of running the first two steps of the algorithm by imposing a given value of the exponent to be tested, leaving the other free to change. The evolution of the RMSE as a function of the tested exponent reflects the impact of this exponent on the global optimization. If the RMSE is strongly affected by the fixed exponent, the corresponding physical parameter has a strong influence on the  $I_h$ , and its effect cannot be compensated by the other parameters. In contrast, if the RMSE is only a weak function of the fixed exponent, either the corresponding parameter has a low effect on  $I_h$ , or its effect is compensated by the other parameters. The first sensitivity analysis for boiling heat transfer  $I_h$  is shown in Figure 14d, in which each exponent is fixed within a range of  $[-2 +2]$  of its original value.



**Figure 14** Steps of the RMSE minimization procedure. (a) Step 1: simulations with random initial parameters; (b) Step 2: simulations with auto-updating parameters; (c) Comparison of the initial model containing the complete set of exponents with the experimental data; (d) Step 3: Sensitivity analysis of the exponents performed with the initial set of parameters for boiling heat transfer

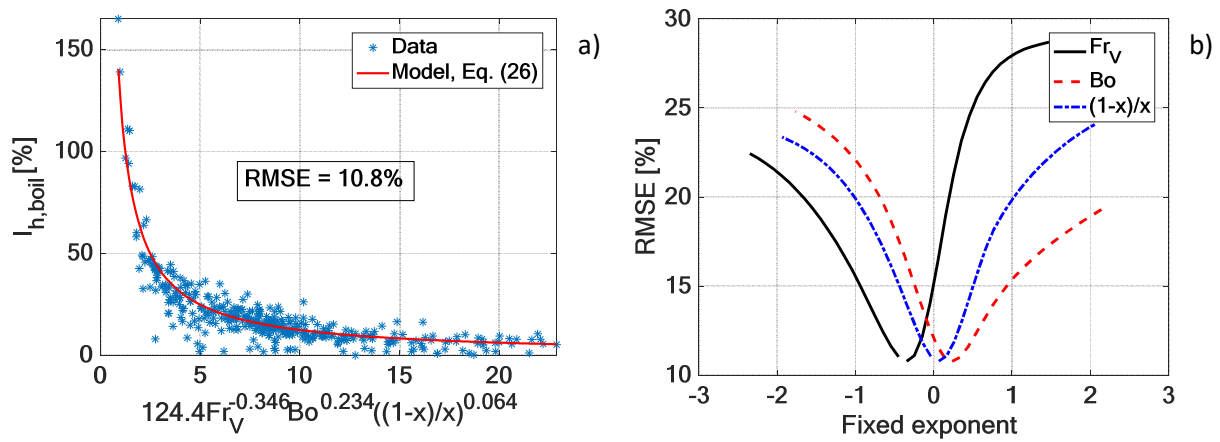
The weak parameters corresponding to low fixed exponent curve slopes are then progressively erased and the algorithm is restarted until only the most influencing parameters survive. As an example, for the  $I_h$  in flow boiling, as shown in Figure 14d, the  $Re_L$ ,  $We_L$ ,  $\delta_{LF}/\delta_T$ , and  $Fr_V$  are dependent upon one another. After deleting  $Re_L$ ,  $We_L$  and  $\delta_{LF}/\delta_T$ , (inertia, viscosity and capillary forces are still included in the remaining parameters), the RMSE is not significantly higher (equal to 10.77%) and the new expression is shown in Eq. (25):

$$I_{h,boil} = 97.7 \cdot Fr_V^{-0.37} Bd^{-0.013} Bo^{0.20} \left(\frac{1-x}{x}\right)^{0.12} X_{tt}^{-0.017} \quad (25)$$

The values of the exponents for  $Bd$  and the Martinelli parameter are almost negligible and are neglected in the final form of the boiling  $I_h$  expression, as shown in Eq. (26):

$$I_{h,boil} = 124 \cdot Fr_V^{-0.35} Bo^{0.23} \left(\frac{1-x}{x}\right)^{0.064} \quad (26)$$

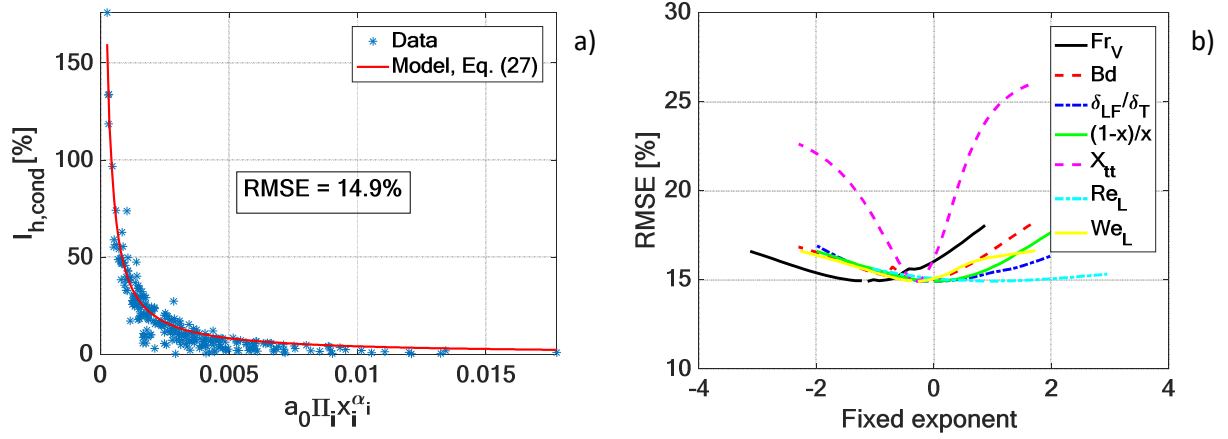
In this case, the RMSE is 10.8%, as shown in Figure 15a (exhibiting only a slight increase compared to the first expression containing all 8 parameters), and the sensitivity analysis (see Figure 15b) reveals that the three remaining nondimensional numbers present a similar and significant impact on the RMSE.



**Figure 15** Final form of the Inclination effect  $I_h$  for boiling heat transfer. (a) RMSE value and comparison with data; (b) Sensitivity analysis of the remaining exponents

The same approach has been used for condensation heat transfer, for which the optimization algorithm started with the same nondimensional parameters, with the exception of the Boiling number. By following the abovementioned procedure, the initial form of the inclination effect for condensation with all the considered parameters is shown in Eq. (27), whereas Figure 16a-b provides the initial comparison with the experimental database (RMSE = 14.9%) and the sensitivity analysis for all the exponents.

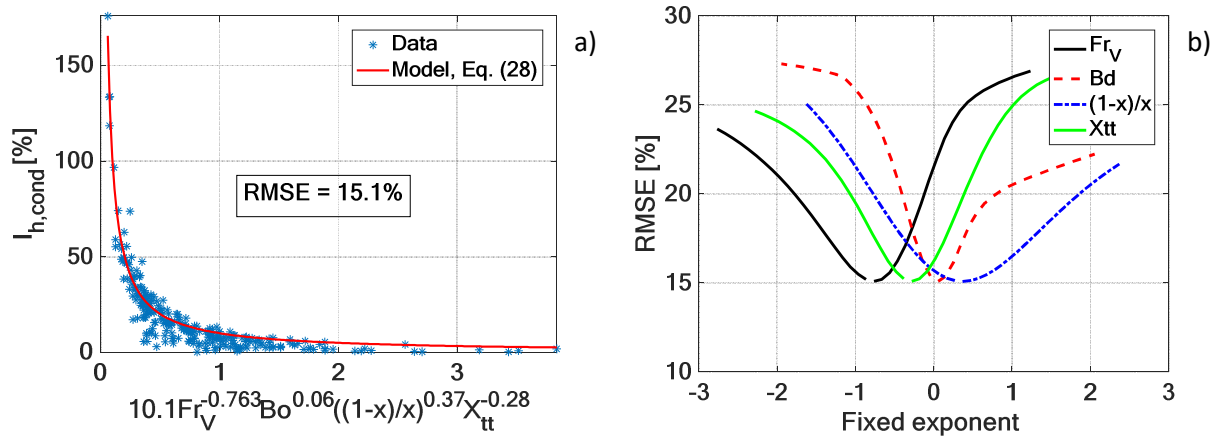
$$I_{h,cond} = 0.0416 \cdot Fr_V^{-1.12} Bd^{-0.30} \left(\frac{\delta_{LF}}{\delta_T}\right)^{0.015} \left(\frac{1-x}{x}\right)^{0.01} X_{tt}^{-0.30} Re_L^{0.937} We_L^{-0.264} \quad (27)$$



**Figure 16** (a) Comparison of the initial model containing the complete set of exponents with the experimental data for condensation heat transfer; (b) Sensitivity analysis of the exponents performed with the initial set of parameters for convective condensation

By erasing the weak parameters from the initial expression, the final form of the  $I_h$  for convective condensation is as shown in Eq. (28). The remaining four parameters ( $Fr_V$ ,  $Bd$ ,  $(1-x)/x$  and Martinelli parameter  $X_{tt}$ ) have a similar and non-negligible slopes (Figure 17b) in the sensitivity analysis, thus having the same importance in the evaluation of the Inclination effect. With respect to the initial form of the correlation, the RMSE becomes only slightly higher (15.1%), with a satisfactory agreement (Figure 17a).

$$I_{h,cond} = 10.1 \cdot Fr_V^{-0.76} Bd^{0.060} \left( \frac{1-x}{x} \right)^{0.37} X_{tt}^{-0.28} \quad (28)$$



**Figure 17** Final form of the Inclination effect  $I_h$  for condensation heat transfer. (a) RMSE value and comparison with data; (b) Sensitivity analysis of the remaining exponents

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