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Evaluation of condensation heat transfer in air-cooled condenser by dominant flow criteria

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ABSTRACT

In this paper, a modification to the dominant flow criteria is presented for the study of heat transfer by confined condensation in Air Cooled Condenser (ACC) systems. The new methodology combines in one single procedure the analysis, which with the current methods requires tedious grouping processes. A new proposal reduces the average error by computing 22% in 88.7% of the available samples and includes the shear stress produced by the steam drag when it flows at speeds greater than 40 m/s. New method is also valid for a vapor quality located between 0.9 and steam flows between 3 and 590 kg/(m2/s-1), values for the Reynolds number for the liquid portion between 660 and 58 540 and the Reynolds number for the vapor portion located between 1 320 and 333 120, internal equivalent diameters of the tubes comprised between 7.4 to 49 mm.

1. INTRODUCTION

The totality of the condensation processes found in the application of dry condensation systems (ACC) to power plants are related to condensation on the interior surfaces of horizontal or vertical pipes. The analysis of the heat transfer of condensation inside pipes is complicated by the fact that the speed of the steam and the rapidity of the accumulation of liquid on the walls of the tubes strongly influence it [1-2]. The volume of steam is limited by the walls inside the tubes. A large amount of steam was condensed in the tubes because it was considerable long. The steam flows inside the tube and condenses as it moves along it. The steam flow is oriented and its speed can be very high (more than 100 m/s). The friction at the vapor-condensate interface can therefore be considerable.

If the direction of the steam flow coincides with that of the condensate flowing by gravity. The friction produces an acceleration of the latter, thinning the film and increasing the surface heat transfer coefficient. If the steam flows in the opposite direction to that of the condensate, the film can be decelerated, its thickness increased, and the intensity of surface heat transmission reduced. An increase in the steam speed can cause the drag of the film and its partial separation from the wall. Which produces an increase in the heat transmission, therefore inside the pipes. This can depend on the dynamic effect of the steam on the condensate film [3-4].

This effect manifests itself in different ways, depending on the direction of gravity and friction, which is determined not only by the position of the tube in space, but by the direction of the steam flow, up or down, in inclined or vertical tubes. In the available literature and consulted due to the complexity of the problem in question the process of heat transfer by condensation in ACC systems is divided into four intervals or study areas. Which becomes somewhat cumbersome, especially in multiple systems of panels ACC, because the lengths of the tubes are already appreciable and therefore the simultaneity of two or more zones in the same system is frequent. This drawback is currently a limiting factor in modern power plants, since the output, powers are high and therefore, the heat volumes to be rejected in these also take appreciable values, which requires the combined operation of multiple systems of ACC panels.
At present, there is no single methodology to solve these limitations and errors in the results obtained are lower than those computed with the use of currently available methods (25%). For this reason, the authors are imposed as fundamental task in the present investigation to develop in a compact way a methodology that includes all of these effects in the process of heat transfer by condensation, and that is valid in the four known regions and whose average error is less than 25% [3-4].

For them, available experimental quantities obtained from direct communication with specialists of recognized prestige in the area of action at international level, as well as values reported in works elaborated with intentions similar to the present one.

2. METHODS AND VALIDATION

2.1 Criteria for differentiation of the dominant condensation flow

In a previous work, the authors developed a methodology for obtaining the average heat transfer coefficient for the condensation of water vapor inside ACC systems, considering that for this purpose the steam has a negligible speed. This expression was obtained by the author and his collaborators and is given by [5]:

$$Nu = 0.923\sqrt[4]{d^3\frac{(\rho_L-\rho_V)gsin\phi\left(r_{LV}+\frac{3}{8}Cp_L(T_{Sat}-T_p)\right)}{\nu_L\lambda_L(T_{Sat}-T_p)d}} \quad (1)$$

However, the different operative situations make the criterion of dominant flow inside the ACC tubes variable, which is why it is necessary to determine which is the dominant criterion for later proceeding to the case analysis. A criterion used to solve this problem is the one given by Martinelli-Lockhart [6], it is based on the combination of two dimensionless criteria which are:

Martinelli's parameter:

$$W = \left(\frac{1-x}{x}\right)^{0.9}\sqrt{\frac{\rho_V}{\rho_L}}\left(\frac{\mu_V}{\mu_L}\right)^{-0.1} \quad (2)$$

Dimensionless speed:

$$J = \frac{xG}{\sqrt{g\rho_V(\rho_L-\rho_V)d_I}} \quad (3)$$

The results obtained through the application of relations (2) and (3) make it possible to identify the dominant flow criterion, using the identifiers provided in table 1 for this purpose.

As can be seen in table 1, the condensation process of water vapor inside the pipes of an ACC system is extremely complex, since several zones are formed from the steam inlet to the formation of the sub-cooled liquid [7].

It would be convenient to have initially the fundamental characteristics of each flow area as well as the most accepted and widespread expression in the literature for the determination of the average coefficient of heat transfer in it.

Table 1. Validity ranges for the dominant flow condensation criteria in ACC systems

Validity range	Dominant flow criterion
$J > 1.5$; $W < 1$	Annuli
$J \leq 1.5$; $W < 1$	Stratified-wavy
$J \leq 1.5$; $W \geq 1$	Intermittent
$J > 1.5$; $W \geq 1$	Burbles

Stratified-wavyflow: when the steam has a medium or low velocity, the convective heat transfer in the stratified liquid that is stored in the bottom of the tube may not be negligible;secondly, the axial steam flow may interfere in the speed and heat transfer of the film around the tube wall. In the work [8] this zone was studied in detail, being further reported that the stratified-corrugated flow regime is present inside a tube when it is fulfilled that $G < 500\ kg/m^2s$, $V > 0.5m/s$ and $Fr^* < 20$.

The term Fr^* is the modified Froude number, which is given by [9]:

$$Fr^* = A\left(\frac{\mathrm{Re}_L^B}{\sqrt{Ga}}\right)\left(\frac{1+1{,}09W^{0.039}}{W}\right)^{1.5} \quad (4)$$

In Equation (4) the Reynolds number for the liquid state Re_L and the Galileo number $Ga$are determined by the following expressions

$$\mathrm{Re}_L = \frac{G(1-x)d_I}{\mu_L} \quad (5)$$

$$Ga = \frac{\rho_L(\rho_L-\rho_V)gd_I^3}{\mu_L^2} \quad (6)$$

Finally the numerical value of the constants A and B present in the expression (4) are dependent on the Reynolds number for the liquid state Re_L. This dependence is shown in table 2.

Table 2. Values of constants A and B in equation (4)

Applicability range	A	B
$\mathrm{Re}_L \leq 1250$	0.025	1.59
$\mathrm{Re}_L > 1250$	1.26	1.04

The average heat transfer coefficient in this area is dependent on three factors, which are:

1- Heat transfer coefficient for the steam portion

$$Nu_P = \frac{0.23\left(\frac{Gd_i}{\mu_V}\right)^{0.12}}{1+1.11\cdot W^{0.58}}\left(\frac{Ga\cdot \mathrm{Pr}_L(r_{LV})}{Cp_L(T_{Sat}-T_P)}\right)^{0.5} \quad (7)$$

2- Sweeping angle of the steam portion (see figure 1)

$$\left(1-\frac{\theta}{\pi}\right) = \cos^{-1}\left(\frac{2\left[1+\frac{1-x}{x}\left(\frac{\rho_V}{\rho_L}\right)^{2/3}\right]-1}{\pi}\right) \quad (8)$$

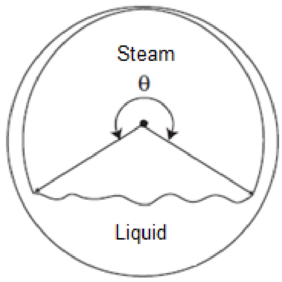

Figure 1. Sweeping angle of the steam portion

3- Heat transfer coefficient for the liquid portion

$$Nu_F = 0.0195\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}_L^{0.4}\sqrt{1.376+\frac{C}{W^D}} \quad (9)$$

Numerical value of the constants C and D present in the expression (9) are shown in table 3.

Table 3. Values of constants C and D in equation (9)

Range	$\left[\frac{(G/\rho_L)^2}{gd_I}\right] \le 0.7$	$\left[\frac{(G/\rho_L)^2}{gd_I}\right] > 0.7$
C	$4.172 + 5.48 \cdot \left[\frac{(G/\rho_L)^2}{gd_I}\right] - 1.564 \left[\frac{(G/\rho_L)^2}{gd_I}\right]^2$	7.242
D	$1.773 - 0.169 \cdot \left[\frac{(G/\rho_L)^2}{gd_I}\right]$	1.655

The results obtained with the use of equations (7), (8) and (9) are combined to obtain the total heat transfer coefficient by the following expression [10]:

$$Nu = Nu_P + \left(1 - \frac{\theta}{\pi}\right) Nu_F \tag{10}$$

Expression (10) is is correlated with 383 experimental available, finding that it is adjusted with an average error of 15% in the 79.4% experimental data available.

Annuli flow: This type of flow occurs when the speed of the steam is high, so that the gravitational effects can be ignored, while the condensate is deposited in a thin annular layer around the tube wall, without the presence of stratification. A significant part of most condensers operates under the conditions of this flow regime.

Of all the known models, the Chato's equation has the greatest acceptance and use in the ACC systems that operate in the annular zone. This is given by:

$$Nu = 0.023 \, \mathrm{Re}_L^{0.8} \, \mathrm{Pr}_L^{0.4} \left[1 + \frac{2.22}{W^{0.89}}\right]^{0.8} \tag{11}$$

In this zone, the laminar flow models predict low values of the average heat transfer coefficient, so turbulent models should be used in this case. Chato's model (11) is expressed in function of the local number of Nusselt, therefore they must be integrated over the entire length of the tube in function of finding the average coefficient of heat transfer, so that [11-13]:

$$\alpha = \frac{1}{L} \int_0^L \alpha(z) dz \tag{12}$$

A drawback in equation (12) lies in the fact that the dependence of the steam quality xon the axial positionzmust be known. This is usually solved by subdividing the total length into a number of sub-elements of lengthΔz from the beginning of the condensate process, i.e. from inlet to outlet of the tube, using the local coefficient of heat transfer for each sub-element (normally 4 elements are taken for achieve a medium precision). Assuming that the steam quality varies linearly, which unfortunately does not happen in many cases, then the heat transfer coefficient can be determined approximately by taking the steam quality as x=0.5 in the expressions for the determination of the local heat transfer coefficient.

An approximate solution for this problem was proposed by Mishra [14], who reports from a total of 813 measurements made in the laboratory, that the behavior of the variation of the quality of the steam presents a parabolic behavior whenever it is fulfilled in the tube entry that $x < 0.95$. This curve has a average deviation of 15%., being described by the following expression:

$$x'' = 0.43\left(\frac{l}{l_T}\right)^2 - 1.341\left(\frac{l}{l_T}\right)^2 + 0.011 + x \tag{13}$$

Intermittent flow: It occurs during condensation in tubes when the steam velocity is too low, (less than 0.5 m / s) flow can be dominated by gravitational effects, and then forces and stratification of the condensate can occur, that is, the condensate forms a thin film in the wall of the upper portion of the tube and drains by the periphery of this by the effect of gravitational forces towards the bottom of the tube where it joins the axially flowing condensate due to the shear stress of the fluid stream.

In the known literature there are not many works on this type of condensation criterion, and in all cases the most recommended expression is that obtained by Shah, which is given by [15-16]:

$$\overline{\alpha} = 0.728 \frac{\left(\frac{\lambda_L^3 \rho_L (\rho_L - \rho_V) g \cdot (r_{LV})}{\mu_L (T_{Sat} - T_P) d_I}\right)^{1/4}}{\left[1 + \frac{1-x}{x}\left(\frac{\rho_V}{\rho_L}\right)^{2/3}\right]^{0.75}} \tag{14}$$

The expression (14) correlates moderately with available experimental data, since in a total 274 test an average deviation of 28% was found in 80.1% of the samples.

Burbles Flow: This regime of condensate flow appears when inside the tube and most of its content is subcooled liquid. However there are still individual bubbles, which collapse and are controlled both by the inertia of the liquid and by heat transfer. Depending fundamentally on the degree of sub cooling of the liquid. One of the most well-known and recommended expressions for this type of flow is the Jaster-Kosky equation, which is given by [18-20]:

$$Nu = \frac{\alpha d_I}{\lambda_F} = \frac{0.728}{\left[1 + \frac{1-x}{x}\left(\frac{\rho_V}{\rho_L}\right)^{2/3}\right]} \left(\frac{\rho_L (\rho_L - \rho_V) g \left(r_{LV} + \frac{3}{8} Cp_L (T_{Sat} - T_P)\right)}{\lambda_L \mu_L (T_{Sat} - T_P) d_I}\right)^{1/4} \tag{15}$$

Expression (15) correlates moderately with available experimental data, because in 104 tests an average deviation of 35% was found in 78.1% of the samples [21-24].

2.2 Experimental validation of a single model for condensation in ACC systems

As was shown in the previous section, the study of the process of heat transfer by condensation in ACC systems becomes complex due to the number of elements to be considered, as well as the high number of expressions involved in the study.

It would be reasonable to have a single expression that allows to evaluate the heat transfer coefficient in any of the zones and whose results are close to the precision environment obtained with the use of the current methods, which were not developed for the exclusive use in ACC systems. Main reason for the failure in many cases of the methods available today.

Access to an appreciable group of available experimental data is taken as a starting point for the present study. The correlation of the available experimental quantities allows having a unique function for the evaluation of condensation heat transfer coefficient, which responds to the following expression [25-29]:

$$for\ 0.9 \le x \le 0.95 \rightarrow \alpha_T = C_1 \frac{\cdot\left[1-0{,}03\cdot(P_{back}-5)^{0{,}71}\right]\cdot m^{0{,}8}}{0{,}19\cdot d^{1{,}8}} \tag{16}$$

$$for\ 0.95 < x \le 1 \rightarrow \alpha_T = C_2 \frac{\left[1-0.028\cdot(P_{back}-5)^{0{,}71}\right]\cdot m^{0{,}8}}{0.19\cdot d^{1{,}8}} \tag{17}$$

where:

$$for\ 0.9 \le x \le 0.95 \rightarrow C_1 = 0.25Ln(x)+1.026 \tag{18}$$

$$for\ 0.95 < x \le 1 \rightarrow C_2 = -0.8Ln(x)+0.972 \tag{19}$$

The experimental data used in the generalization and development of the expression (16) and (17) are provided in table 4. Figure 2 shows the correlation of experimental data. In the y-axis is plotted the decimal logarithm of the quotient between film coefficients calculated with the use of equations (16) and (17) while the Shah parameter is plotted along the x-axis, which is given by

$$Z = \left(\frac{1-x}{x}\right)^{0.8} \mathrm{Pr}_L^{0.4} \tag{20}$$

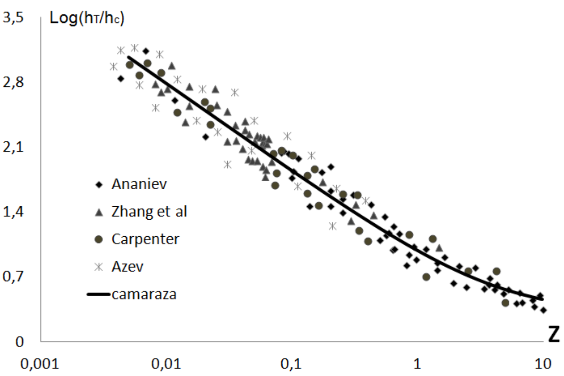

Figure 2. Comparison of condensing data in horizontal tubes with the proposed correlation, Equation (20)

3. TRAWL ANALYSIS CAUSED BY HIGH VAPOR VELOCITIES.

The expressions given above are valid only if the steam drag is negligible or insignificant. This assumption is appropriate in the ACC when the steam velocity does not exceed 50 m/s, but already for higher speeds of the steam flow the effect of the drag on the liquid film cannot be ignored.

To take into account the effect of the drag on the liquid film, it is necessary to include the influence of shear stress on the surface of the liquid, that is:

$$\tau_S = \frac{C_W \rho_V V_E^2}{2} \tag{21}$$

Two apparently appropriate expressions could be the Blasius solutions for the laminar boundary layer and the boundary layer turbulent. However there is an additional problem, and that is that these two expressions are only valid for a waterproof wall, while the surface the liquid has a normal component of speed due to condensation. In many texts of fluid mechanics it would be said that in this case there is presence of suction on the surface. In typical condensation problems the suction speed of the suction is relatively large and causes the thickness of the boundary layer to become almost constant very close to the initial edge, and the steam velocity is, in essence, only a function of V_E

This can be partially solved if the model elaborated by Couette for laminar flow is used, it can be used to determine the shear forces acting on a boundary layer subject to a strong suction effort (see figure 3).

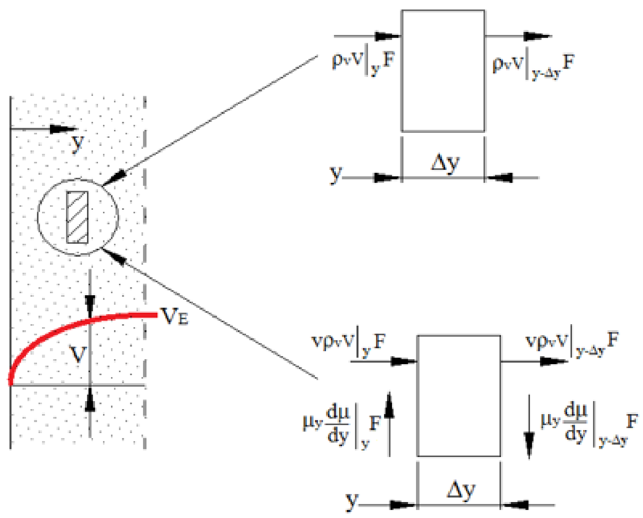

Figure 3. Model problem and elementary volumes employed

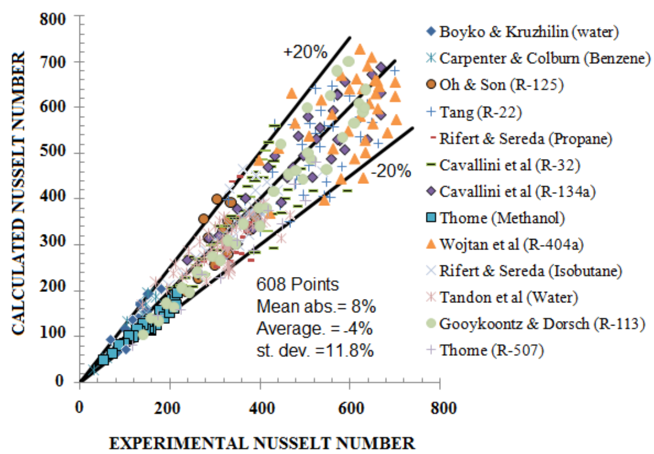

Figure 4. Application of the model to vertical and inclined tubes data reported by several authors

After an appreciable group of mathematical transformations, we arrive at a conclusive expression that allows us to determine the coefficient of transfer of lime by condensation in air-cooled systems when the effect of the steam drag on the surface of the liquid is taken into account. This expression is described by [11]:

$$\alpha = \left[\frac{\lambda_L^2 V_E}{8 \nu_L x} \left\{ 1 + \left(\frac{16\,\mathrm{Pr}_L}{Ja} \frac{gx}{V_E^2} \right)^{0,5} \right\} \right]^{0,5} \tag{22}$$

Therefore, when the steam velocity in an ACC system is higher than 40 m/s, it is necessary to establish a product between the results obtained by using equation (16) and (17) with Equation (22), to obtain the heat transfer coefficient [31-34].

Experimental Nu_E and calculated Nusselt numbers Nu_T, obtained by means of the present model, are compared in the following diagrams. Figures 4 show the results relative to the vertical and inclined tubes, while in the figure 5 show horizontal tubes.

Table 4. Summary of the experimental quantities used

Source	Number of experimental data	Fluid	Diameter (mm)	G (kg/m^{2}s)	x	Re_L	Re_V	p_R	Deviation [%]
Jakob *et al.* (1932)	31	Water	40.0	24 48	0.96 0.88	3427 6854	79438 158870	0.0046	13.7 11.4
Al-Shmmari *et al.* (2004)	9	Water	28.2	3	0.97 0.9	173	8210	0.0008	12.1 9.7
Khun *et al.* (1997)	11	Water	47.5	10	0.94 0.9	2554	32642	0.023	16.2 -6.1
Borishankiy *et al.* (1976)	34	Water	10.0 19.3	12 590	0.92	763 58540	8284 333120	0.036 0.308	12.7 -1.3
Lee and Kim (2008)	15	Water	12.0	27 45	0.98 0.95	1183 1944	27421 45071	0.0046	16.9 8.1
Gooykoontz *et al.* (1967)	26	Water	7.4	131 264	0.99 0.9	3827 6567	78853 167186	0.002 0.0062	1.8 2.5
Gooykoontz *et al.* (1967)	19	Water	15.9	22 74	0.99 0.91	660 2300	1320 4560	0.005 0.017	17.6 8.4
Blageti - Slunder (1978)	21	Water	30.0	4 69	0.99 0.94	408 7474	9173 252428	0.0046	22.9 -0.3
Annaniev (1961)	63	Water	8.0	38 160	0.99 0.91	10254324	21158 89085	0.031 0.004	25.3 19.4
Carpenter (1948)	12	Water	11.6	16 140	0.97 0.95	692 5934	15686.134474	0.0046	21.2 12.8
Annaniev (1961)	68	Water	8.0	38 160	0.99 0.91	1025 4324	21158 89085	0.051 0.004	25.3 19.5
Varma (1977)	20	Water	49.0	12	0.95 0.89	1808	54415	0.0023	6.2 1.5
Gooykoontz *et al* (1967)	20	Water	15.9	20 74	0.99 0.9	660 2800	1320 4960	0.005 0.017	17.4 8.1
TOTAL	349		7.4 49.0	3 590	0.99 0.88	660 58540	1320 333120	0.0008 0.031	16.6 7.5

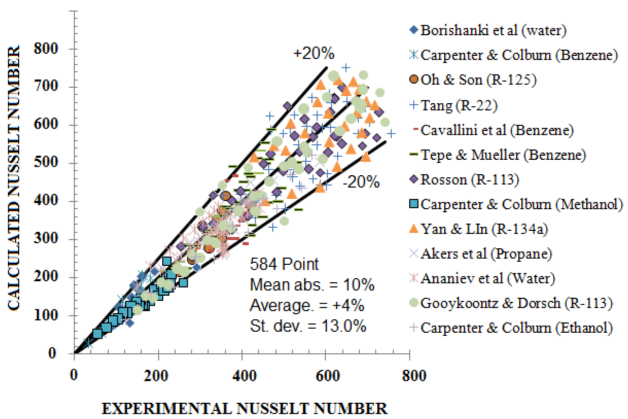

Figure 5. Application of the model to horizontal tubes data reported by several authors

4. CONCLUSIONS

A new model has been developed that unites in a unitary procedure the tedious established procedures for the determination of the average coefficient of heat transfer by means of the dominant flow criterion techniques. The new proposal includes the effect of steam drag when it exceeds the critical speed inside the ACC, and the results obtained with its use compute an average error of 22% in 88.7% of the available samples. The results obtained agree with the initial criterion that supported the investigation, considering that the objectives of the same were fulfilled. The new method is also valid for a steam quality located between 0.9 and one, for steam flows between 3 and 590 kg/(m^2.s^{-1}), values for the Reynolds number for the liquid portion between 660 and 58 540 and the Reynolds number for the vapor portion located between 1 320 and 333 120, internal equivalent diameters of the tubes comprised between 7.4 to 49 mm.

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NOMENCLATURE

G	Mass flux, kg. $m^{-2}.s^{-1}$
m	Steam rate, kg.s-1
P_{back}	Steam pressure, kPa
Cp	Specific heat, J. kg^{-1}. K^{-1}
d	Inner equivalent tube diameter, m
g	gravitational acceleration, $m.s^{-2}$
Re	Reynolds number
Nu	Nusselt number
Ga	Galileo Number
Ja	Jakob Number
Pr	Prandtl number
Pr_L	Prandtl number for single-phase
P	Fluid pressure, kg. $m^{-1}.s^{-2}$
x	Steam quality
(r_{LV})	Latent heat of vaporization, J. kg^{-1}. K^{-1}
Tsat	Saturation temperature, ^0C
T_P	Wall temperature, ^0C
N	Numbers of experimental points.
V	Steam speed, $m.s^{-1}$

Greek symbols

α	two-phase heat transfer coefficient, $kg.m^{-2}.s^{-3}.K^{-1}$
$\overline{\alpha}$	Mean heat transfer coefficient, $kg.m^{-2}.s^{-3}.K^{-1}$
δ	Thickness film, m
α_T	Two-phase heat transfer coefficient, $kg.m^{-2}.s^{-3}.K^{-1}$
θmed	Inscript pipe angle,
μ	Dynamic viscosity, kg. $m^{-1}.s^{-1}$
μ_L	Liquid dynamic viscosity, kg. $m^{-1}.s^{-1}$
μ_V	Steam dynamic viscosity, kg. $m^{-1}.s^{-1}$
ρ_L	Density of liquid, $kg.m^{-3}$
ρ_V	Density of vapor, $kg.m^{-3}$
λ	Fluid thermal conductivity, $W.m^{-1}$. K^{-1}
ν_L	Liquid kinematic viscosity, $m^2.s^{-1}$
ΔT	Temperature difference across the condensate film, K

Subscripts

Eq.	Equation