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QUALITY INFLUENCE IN POST-BURNOUT HEAT TRANSFER

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Abstract—The analysis of many experimental results on post burnout heat transfer, obtained with a steam–water mixture flowing upward through a tubular duct at a pressure of 70 kg/cm^2 and at a specific mass flowrate of 220 g/cm^{2}s, has led the authors to the following main conclusions:

In the post-burnout regime the heat flux φ goes partly to the liquid phase (φ_l) and partly to the vapor phase (φ_v):

$$\varphi = \varphi_l + \varphi_v.$$

The vapor heat flux is related to the wall-to-bulk temperature difference by a heat-transfer coefficient which is a linear function of the quality.

The liquid heat flux is directly proportional to the liquid quality of the mixture.

NOMENCLATURE

C_p, specific heat;
D, diameter of the tubular duct;
G, specific mass flowrate;
h, heat-transfer coefficient;
K, thermal conductivity;
L, heated length of the tubular duct;
p, pressure;
T, temperature;
X, quality of the mixture (vapor mass flowrate/total mass flowrate);
α, void fraction;
φ, heat flux;
λ, latent heat of the vaporization;
μ, dynamic viscosity;
$\Delta\theta_F$, temperature difference between heating wall and saturation point.

Subscripts
BO, burnout or dry-out;
in, inlet conditions;
l, liquid;
s, saturation;
v, vapor;
w, heating wall.

INTRODUCTION

From a literature survey on post burnout heat transfer [1–13] it appears that the phenomenon is very complex and that it is probably impossible to find directly a general heat transfer correlation. It appears also that it is necessary to perform separate researches on the influence of the main parameters. In this line, the present work is obtained from a theoretical analysis of many experimental results obtained in CISE Laboratories of Milano [1].

Post burnout heat transfer is the particular heat-transfer regime downstream of the burn-out or dry-out point at positive qualities. The wall of the heated duct is wetted, upstream of the dry-out point, by a liquid layer which is exhausted at the dry-out point [2]; downstream the post burnout regime begins, characterized by a vapor stream which entrains the liquid phase as dispersed droplets.

The post burnout region extends downstream of the burnout point to the point where the last droplets are evaporated by a vapor stream which has become superheated. The heat transfer has two main components [3]: direct vapor convection and heat flux from the wall to the liquid droplets which, impinging against the wall, partially vaporize thus provoking turbulence within the thermal boundary layer of vapor.

At some point the vapor superheats [10] with a thermodynamic disequilibrium of the mixture. The degree of disequilibrium rises with increasing quality and wall heat flux, and falls with increasing specific mass flowrate, on which the thermal interaction between the two phases depends.

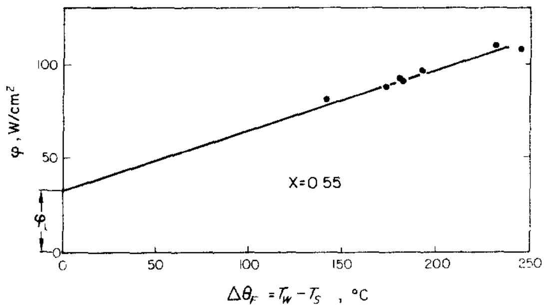
The parameters of the phenomenon are, apart from the nature of the coolant and geometry, the pressure p, the specific mass flowrate G, the quality of the mixture X and the heat flux (if uniform) φ.

A typical two phase flow relates to tubular ducts, uniformly heated, with upward flow (at high Reynolds numbers the gravity influence is almost negligible). Many experiments have been performed in this reference situation in CISE Laboratories, employing water at 70 kg/cm^2 through a tubular test section with an inner diameter (D) of 0·59 cm and a heated length (L) of 480 cm.

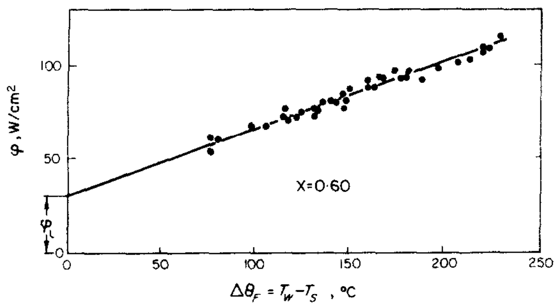
The experimental results have been selected in such a way to keep constant the specific mass flowrate ($G = 220$ g/cm^{2}s) and to gather, as function of the quality X, values of the heat flux φ and of the temperature difference $\Delta\theta_F = T_w - T_s$ between the inner heating wall (cooled) and the saturation point at the corresponding pressure.

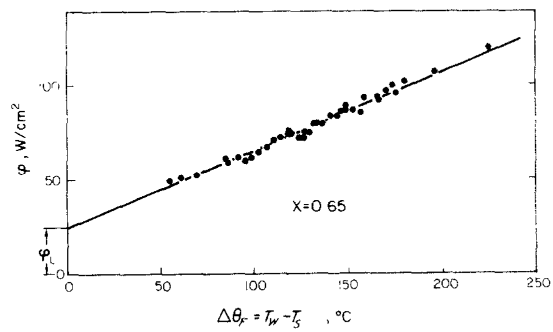
ANALYSIS OF THE EXPERIMENTAL RESULTS

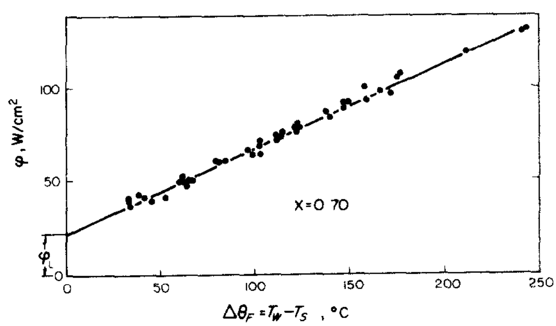
The experimental results are plotted as φ vs. $\Delta\theta_F$ in Figs. 1–7 in which each plot is given for a particular value of the quality X (0·55 $\leqslant X \leqslant$ 1). In each plot the experimental points follow lines which, at the origin of coordinates ($\Delta\theta_F = 0$), correspond to values of $\varphi_l > 0$ and not to $\varphi_l = 0$ as in the case of monophase fluids.

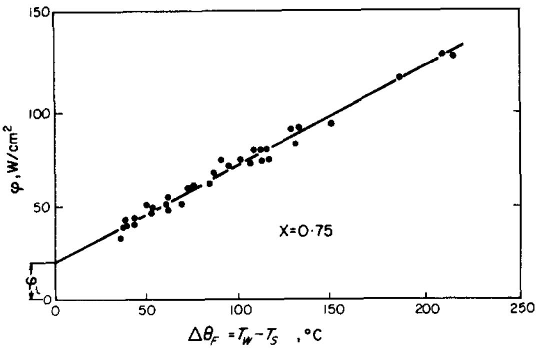


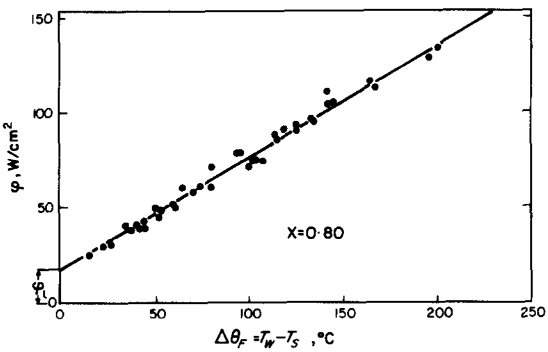
FIGS. 1–7. Heat flux φ vs. wall to saturation temperature drop $\Delta\theta_F$ at constant quality X and flowrate G. $G = 220$ g/cm^{2}s; $p = 70$ bar.

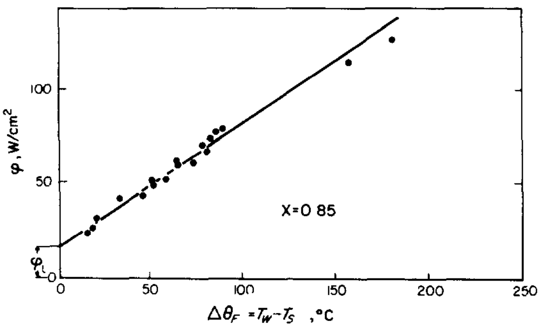












Such a linear trend is represented by the relation (1) with h_v depending on $\Delta\theta_F$:

$$\varphi = \varphi_l + h_v \Delta\theta_F. \tag{1}$$

Considering the many groups of results corresponding to various qualities X (some of which are represented in the Figs. 1–7) it is possible to plot the two parameters of (1), h_v and φ_l, vs. the same quality.

So h_v is represented by the diagram of Fig. 8 in which the representative data follow, within a reasonable margin of uncertainty, straight lines with a negative intersection on the ordinate axis.

Symbolically:

$$h_v = -h_{v_0} + \left(\frac{\partial h_v}{\partial X}\right)_0 . X \tag{2}$$

in which $(\partial h_v/\partial X)_0$ is, as stated, a constant.

The negative term $-h_{v_0}$ has no physical meaning; the relation (2) holds only within the range of post burnout qualities, namely between the critical quality X_{BO} and the point $X = 1$.

In the reference situation ($p = 70$ kg/cm^2; $G = 220$ g/cm^{2}s; etc.) to define the limit-value X_{BO}, it is possible to employ the CISE correlation [14] thus obtaining (for a uniform heat flux φ):

$$X_{BO} = X_{in}\left(1 - \frac{L}{L - 46{\cdot}4}\right) + 0{\cdot}53\frac{L}{L + 46{\cdot}4}$$

being for a thermal balance:

$$\varphi_{BO} = \frac{GD\lambda}{4L}(X_{BO} - X_{in}).$$

Otherwise, employing the A.R.S. correlation [15] one gets:

$$X_{BO} = 1 - \frac{\varphi_{BO}}{384}(\text{if } X_{BO} > 0{\cdot}5)$$

$$X_{BO} = 0{\cdot}692 - \frac{\varphi_{BO}}{1{\cdot}031}(\text{if } X_{BO} < 0{\cdot}5).$$

The experimental data indicate that the burnout quality varies, for different inlet qualities, but it is always greater than 40 per cent over the experimental range considered. Now let us examine the other term of the relation (1), φ_l.

If φ_l is drawn for the various groups of experimental data (Figs. 1–7) vs. the qualities X, one gets the diagram of Fig. 9 in which the representative points have been subdivided according

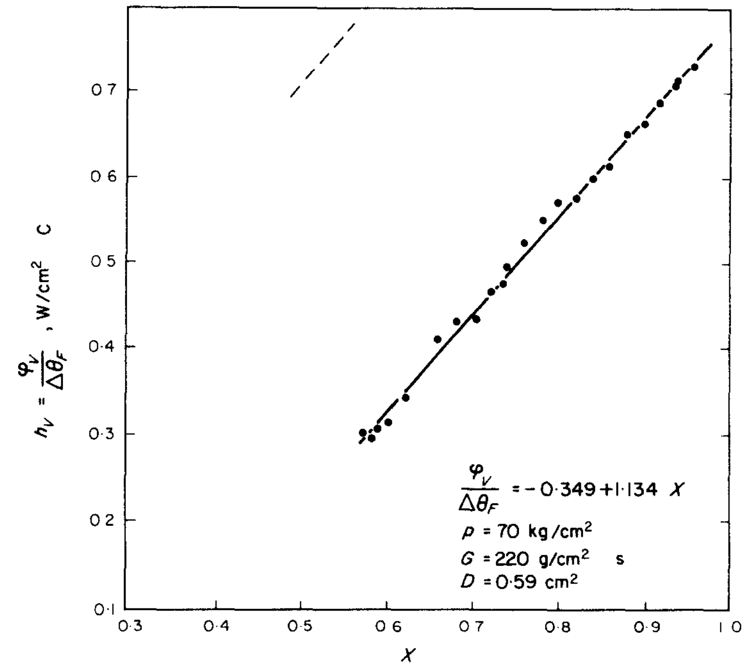


FIG. 8. φ_v vs. quality X; the broken line is referred to the Dittus–Boelter–McAdams correlation for the saturated steam at the same flowrate.

to the heat flux and the original experimental runs.

It is possible to advance the following

All the three groups of representative points have a linear trend vs. X.

With increasing φ, the intercept on the X axis moves to progressively lower values of X.

If X increases, in general the departure from the thermodynamic disequilibrium [10] increases; this means that, in effect, the abscissa X, computed in the equilibrium hypothesis, should be displaced to lesser values, more and more as the quality X approaches 100 per cent. Therefore the best fit lines drawn in Fig. 9 should cross the X-axis near the point $X = 1$.

On the basis of the preceding considerations and within a due range of experimental error (the representative points have been deduced through some interpolations) it is possible to summarize that the term φ_l of the equation (1) may be given by the linear relation (see Fig. 9)

$$\varphi_l = \phi_{l\mathrm{BO}}(1 - X). \quad (3)$$

Let us examine now the experimental relations (1)–(3).

The equation (1) shows a split of φ in two terms. The second term is proportional to the temperature difference $\Delta\theta_F = T_w - T_s$, typical 'driving force' of the single phase convection. The two terms on the R.H.S. of equation (1) and this proportionality suggest two different mechanisms for the heat flux: the one due to the droplets impinging against the wall (φ_l) and the other due to the forced convection of the vapor

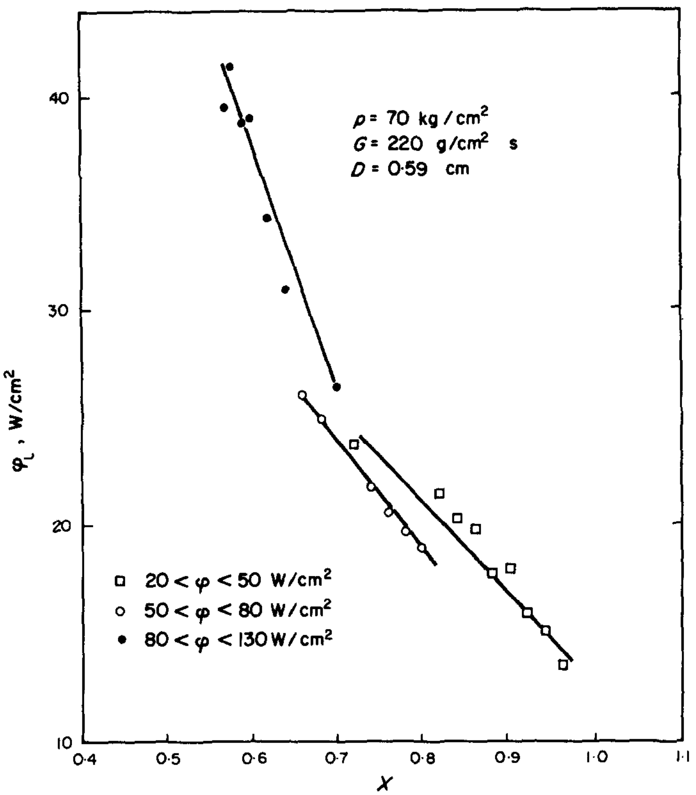


FIG. 9. φ_l vs. quality X.

alone ($h_v \cdot \Delta\theta_F$) measured by a heat-transfer coefficient h_v.

Relation (3) supports this hypothesis showing that the heat flux absorbed by the impinging droplets is directly proportional to the liquid quality $(1 - X)$.

Relation (2) shows the linear dependence of the heat-transfer coefficient h_v on X. The h_v dependence on the first power of X shows a little displacement from the forced-conversion heat-transfer correlations as, for instance, from the well known Dittus–Boelter–McAdams correlation we have:

$$\frac{h_v D}{K_v} = 0{\cdot}023 \left(\frac{GDX}{\alpha \mu_v}\right)^{0\cdot8} \left(\frac{C_{p_v} \mu_v}{K_v}\right)^{0\cdot4} \quad (4)$$

for the hypothesis that the vapour (quality X and void fraction α, with a slip ratio close to one [6]), flows alone within the tubular duct.

In equation (4), being the void fraction α for the given range of qualities a weakly increasing function of X, h_v is a function of a power of X with an exponent $\leqslant 0{\cdot}8$, against the value 1 indicated by the experimental data.

Nevertheless the available data are not sufficient to definitely confirm the difference of the exponents.

The possible increment, in respect of X, may be due either to the droplets which, impinging against the wall, increase the wall turbulence, destroying the thermal boundary layer, or to a non-Newtonian trend of the transverse distribution of the flow.

Velocity measurements performed at CISE (Milano) with a gas–water mixture in annular dispersed flow have evidenced a lesser flattening of the velocity profile in front of a case of single phase gas flow with the same Reynolds numbers. That is, with two-phase-highly-dispersed mixtures the wall boundary layer is higher: growing X, the transversal velocity profile flattens, the boundary layer becomes thinner and the heat transfer coefficient increases.

In Fig. 8 is drawn, for comparison, also the plot of the relation (4), which supplies, evidently, values of h_v bigger than the experimental values.

CONCLUSIONS

The analysis of the experimental data obtained from this investigation of the influence on heat transfer of the quality in post burnout-two phase-mixtures has led us to the following conclusions, which are limited to the range of the experimental data:

(a) In post burnout heat transfer there is a direct heat flux to the liquid (φ_l) and a direct heat flux to the vapour (φ_v):

$$\varphi = \varphi_l + \varphi_v.$$

(b) The heat flux to the vapour (φ_v) is directly proportional, through a heat-transfer coefficient h_v, to the temperature-difference $\Delta\theta_F$ between the heating wall and the saturation point: $\varphi_v = h_v \cdot \Delta\theta_F$.
Furthermore, the heat transfer coefficient h_v is a linear function of the quality X: $h_v = A + BX$ with A = negative constant, B = positive constant.

(c) The heat flux to the liquid phase (φ_1) is directly proportional to the liquid quality: $\varphi_l = C(1 - X)$ with C = positive constant.

The conclusions (a), (b) and (c) outline in a simple physical model the influence of quality on post burnout heat transfer.

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INFLUENCE DE LA QUALITÉ DE VAPEUR DANS LE TRANSFERT THERMIQUE EN POST-DÉSSÈCHEMENT

Résumé—L'analyse de nombreux résultats expérimentaux sur le transfert de chaleur dans le cas du post-dessèchement obtenus avec un mélange eau-vapeur s'élèvant à l'intérieur d'un conduit tabulaire à une pression de 70 kg/cm^2 et à un débit massique égal à 220 g/cm^2.s, a conduit les auteurs aux conclusions principales suivants:

—dans le régime de post-dessèchement, le flux de chaleur se répartit dans la phase liquide (φL) et la phase vapeur (φv):

$$\varphi = \varphi L + \varphi v$$

—le flux thermique de la vapeur est relié à la différence entre température de paroi et température moyenne par un coefficient de transfert thermique fonction linéaire de la qualité.

—le flux thermique du liquide est directement proportionnel à la qualité en liquide du mélange.

EINFLUSS DES DAMPFGEHALTES BEIM NACH-BURNOUT-WÄRMEÜBERGANG

Zusammenfassung—Die Auswertung zahlreicher Versuchsergebnisse über den Nach-Burnout-Wärmeübergang, die mit einem Wasser-Dampf-Gemisch gewonnen worden sind, das in einem senkrechten Rohr nach oben strömte, bei einem Druck von 70 kg/cm^2 und einem spezifischen Massenstrom von 220 g/cm^2 sec, hat zu folgenden Schlüssen geführt:

Beim Nach-Burnout-Regime geht der Wärmefluss φ teils an die flüssige Phase (φ_l) und teils an die dampfförmige Phase (φ_V):

$$\varphi = \varphi_l + \varphi_V$$

Der Dampf-Wärmefluss ist auf die Wandflüssigkeitstemperaturdifferenz bezogen durch einen Wärmeübergangskoeffizienten, der eine lineare Funktion des Dampfgehaltes ist.

Der Flüssigkeits-Wärmestrom ist direkt proportional zu dem Flüssigkeitsgehalt des Gemisches.

ВЛИЯНИЕ СТЕПЕНИ НЕДОГРЕВА НА ТЕПЛООБМЕН В ЗАКРИТИЧЕСКОЙ ОБЛАСТИ.

Аннотация—Анализ многочисленных экспериментальных данных по теплообмену в закритической области, полученных для восходящего потока пароводяной смеси в круглой трубе при давлении 70 кг/см^{2}и удельном массовом расходе 220 г/см^2 сек, позволил авторам сделать следующие основные выводы:

—В закритическом режиме часть теплового потока φ передается жидкой фазе (φ_l), часть (φ_v):

$$\varphi = \varphi_l + \varphi_v$$

—тепловой поток к пару связан разностью температуры у стенки и в объеме коэффициентом теплообмена, который является линейной функцией степени недогрева;

—тепловой поток к жидкости прямо пропорционален степени недогрева жидкости в смеси.