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HEAT TRANSFER TO SATURATED MIST FLOWING NORMALLY TO A HEATED CYLINDER

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Abstract—A flow of steam/air containing a fraction f (0.5–20% by weight) suspended mist was swept across a roughened heated tube. The average heat-transfer coefficients were characterized by a "dryness parameter", ϕ, defined as;

$$\phi = \frac{q''_{w0}}{Gf h_{fg}}\left(\frac{A_0}{d}\right)$$

(q''_{w0}, heat flux, A_0, area/width ratio, G, mass velocity, and d, diameter). The measured average Nusselt number was independent of moisture content for $\phi \ll 1$, the "fully wet" state, but dependent on the vapor/liquid density ratio and the Reynolds number;

$$\overline{N^w_{Nu}} = 4.08 \times 10^{-8}\left(\frac{\rho_L}{\rho_V}\right) N^2_{Re}.$$

Likewise in the "fully dry" state, $\phi > \sim 1$, characteristic convective heat transfer was observed;

$$\overline{N^D_{Nu}} = 0.250\, N^{0.647}_{Re}\, N^{\frac{1}{3}}_{Pr}.$$

Finally in the "partially wetted" state, $\sim 0.7 > \phi > 0.1$, heat transfer was a strong function of moisture content;

$$\frac{\overline{N_{Nu}} - \overline{N^D_{Nu}}}{\overline{N^w_{Nu}} - \overline{N^D_{Nu}}} = \tanh\left(\frac{0.015}{\phi^2}\right)$$

at a given N_{Re}.

NOMENCLATURE

A_0, surface area of tube per unit width;
d, diameter;
D, diffusivity of steam in air;
f, moisture fraction;
G, mass velocity;
h, h_θ, heat-transfer coefficient, local h at polar angle θ respectively;
h^D, h^w, h measured under dry and wet conditions respectively;
h_{fg}, latent heat of evaporation;
k_V, thermal conductivity of steam–air mixtures;
$\bar{k}^M$, average mass-transfer coefficient;
M_S, molecular weight steam;
N_ϕ, $\dfrac{\overline{N_{Nu}} - \overline{N^D_{Nu}}}{\overline{N^w_{Nu}} - \overline{N^D_{Nu}}}$;
$\overline{N_{Nu}}$, Nusselt number, $\dfrac{\bar{h}d}{k_V}$;
$\overline{N^w_{Nu}}$, $\overline{N^D_{Nu}}$, Nusselt numbers wet and dry respectively;
$\overline{N^M_{Sh}}$, Sherwood number $\bar{k}^M\, d/cD$;
N_{Re}, Gd/μ_V;
N_{Pr}, $(\mu C_p/k)_V$;
N_{Sc}, $\mu_V/\rho_V D$;
p, static pressure;
q''_{w0}, heat flux based on total external surface area;
R_G, universal gas constant;
T_i, T_w, T_∞, absolute temperatures, evaporating interface, local wall and far field respectively;
θ, polar angle;
μ_V, dynamic viscosity of air–steam mixtures;
ρ_L, ρ_V, liquid and air–steam densities;
ϕ, dryout parameter $\left(\dfrac{q''_{w0}}{Gf\,h_{fg}}\right)\left(\dfrac{A_0}{d}\right)$;
$\bar{f}$, $= \dfrac{1}{2\pi}\displaystyle\int_{-\pi}^{\pi} f\,d\theta$.

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INTRODUCTION

THE POTENTIALLY high heat-transfer capability of a mist laden gas vapor or gas has been recognized for many years. Finlay [1–4] in particular has measured the high heat-transfer coefficients obtained by spraying a water mist in an air carrier on to a variety of heat-transfer surfaces. Toda [5] on the other hand, was interested in the fundamental investigation of boiling of liquid films on surfaces where the film was created by a fine mist spray. Naturally in these experiments there is a large sensible component in the evaporation of highly subcooled water as well as latent heat in the enthalpy change experienced by the mist flow. The experiments reported in this work were aimed to evaporate liquid droplets suspended in their own vapor. This mist is flowed across a heated tube. Obviously in a one component system the liquid and vapor phases are both close to the same saturation temperature. Unfortunately if one also wishes to know the weight fraction of liquid in a one component system all the heat exchanges need to carefully be monitored to predict the evaporation or condensation of the liquid phase. Experimentally this is a difficult requirement. To avoid this hazard in these experiments a mixture of air and steam was used as the carrier fluid and a monitored supply of superheated deionized water was sprayed into the test chamber. Evaporation occurred and the net weight fraction of mist was then calculable.

EXPERIMENTAL

Steam, water and heated air were blown horizontally across a test heater arranged perpendicularly to the flow in a rectangular cross-section pressure vessel. The test heater was 152 × 25.4 mm. It had a small annular machined roughness which eliminated lateral motion of de-entrained liquid. It was equipped with 4 thermocouples arranged at polar angles 0, $\pi/2$, π and $3\pi/2$ counting from the downstream direction. Flashing occurred and the mist and vapor–gas streams assumed a uniform saturation temperature somewhat lower than the original steam–air mixture. The fraction of water flashed was calculated from an enthalphy balance so that the water impinging on the test heater was known.

To measure a heat-transfer coefficient in the system it is necessary to measure the heat flux based in the external surface area, q''_{w0}, and the temperature difference between the surface of the heater and the free stream. Then a local heat-transfer coefficient may be defined by;

$$h = \frac{q''_{w0}}{(T_w - T_\infty)} \tag{1}$$

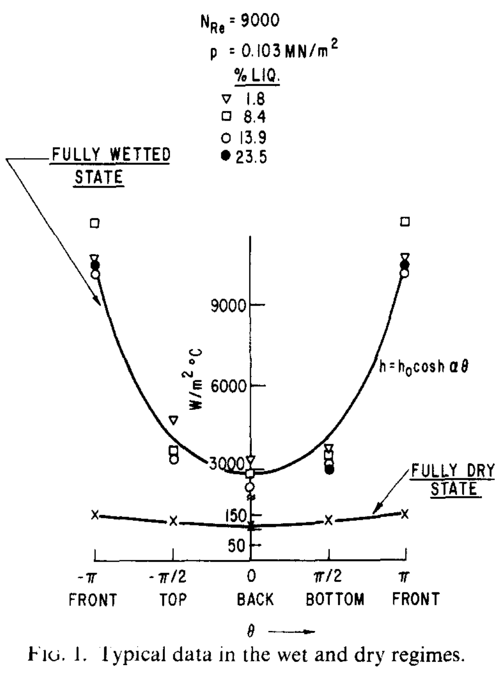

FIG. 1. Typical data in the wet and dry regimes.

where T_w was measured at each of four primary polar positions noted in Fig. 1. The Reynolds number defined in Fig. 1 is based on the mass velocity, the O.D. of the test heater and the viscosity of the vapor–gas mixture.

TREATMENT OF DATA

It was found that the relationship between h and θ can be expressed as a cosh function for both the wet and the dry sets of data (Fig. 1).

Then:

$$h = h_0 \cosh \alpha\theta \tag{2}$$

where

$$\alpha = \frac{1}{\pi}\cosh^{-1}\left(\frac{h_\pi}{h_0}\right).$$

Hence

$$\bar{h} = \frac{h_0}{\alpha\pi}\sinh(\alpha\pi). \tag{3}$$

Hence knowing h_0 and h_π from the experimental data we can calculate the mean heat-transfer coefficient for the tube.

From the raw data local h's and $\bar{h}$'s were calculated. Nusselt numbers were calculated from definition, $\overline{N_{Nu}} = \bar{h}d/k_V$. All data were reduced with respect to k_V in order that one can immediately appreciate the magnitude of the decrease in the heat-transfer resistance due to mist content. Thus, the value of $\bar{h}$ rose from ~ 180 W/m²°C (dry) to ~ 8000 W/m²°C (fully wet) with a proportionate rise in $\overline{N_{Nu}}$ from ~ 110 to ~ 5000. There is, furthermore, an effect of heat flux (Fig. 2) which involves a gradual reduction in Nusselt number from ~ 5000 to < 2000 with increasing heat flux. Actually the range extends down to the fully dry Nusselt numbers of ~ 110 at high enough heat fluxes.

RESULTS

1. *Fully dry state*

The data may be correlated by (Fig. 3)

$$\overline{N^D_{Nu}} = 0.250\, N_{Re}^{0.647}\, N_{Pr}^{\frac{1}{3}} \tag{4}$$

2. *Fully wetted state*

The fully wet asymptotic data may be correlated as a function of N_{Re} by plotting N_{Re} vs $\overline{N_{Nu}}$ on log–log coordinates as per Fig. 4. To the degree of scatter (primarily determined by the fact that the high $\overline{N^w_{Nu}}$ infers high $\bar{h}^w$ which in turn infers a small measured ΔT)

$$\overline{N^w_{Nu}} = 4.08 \times 10^{-8} \frac{\rho_L}{\rho_V} N_{Re}^2 \tag{5}$$

represents the fully wetted state data.

3. *Partially dry state*

This behavior can be understood by means of a simple energy balance in the power to evaporate the deposited moisture. In a highly idealized sense the tube intersects water at a flow rate of Gfd per unit width of heater. Of this, the amount evaporated is $q''_{w0}A_0/h_{fg}$.

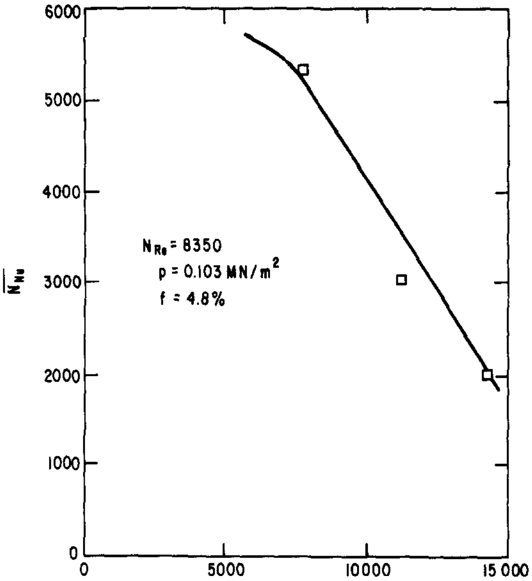

FIG. 2. Onset of dryout.

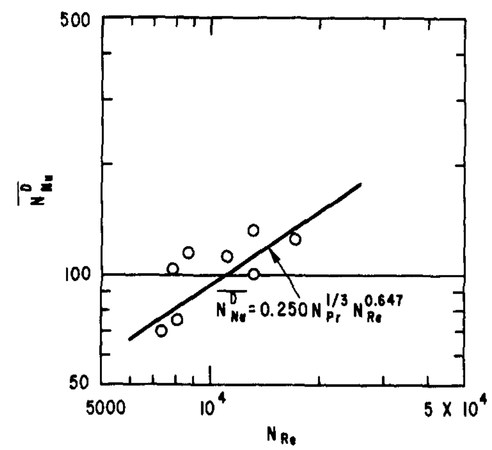

FIG. 3. Correlation of fully dry data.

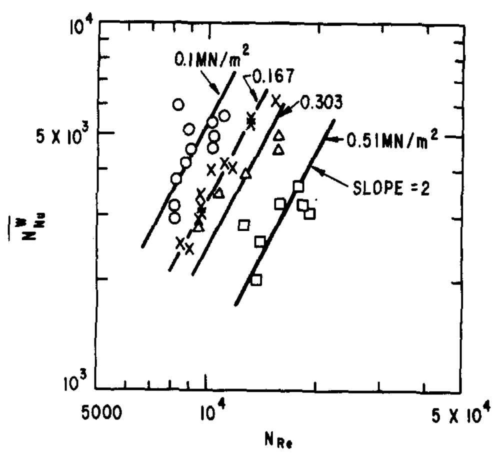

FIG. 4. Correlation of fully wet data.

Following this reasoning the fraction of deposited water, ϕ, which is evaporated is given by;

$$\phi = \left(\frac{q''_{wo}}{Gfh_{fg}}\right)\left(\frac{A_0}{d}\right). \tag{6}$$

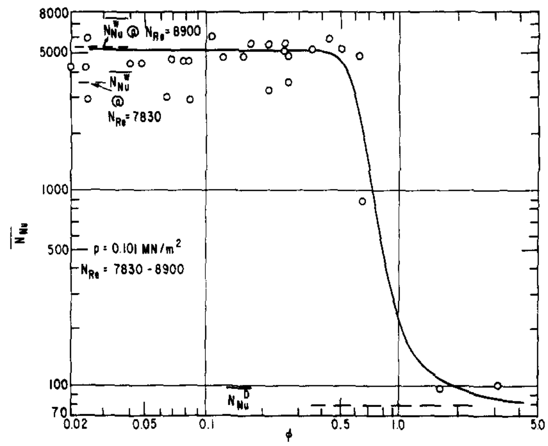

FIG. 5. The dryness parameter as a correlating variable.

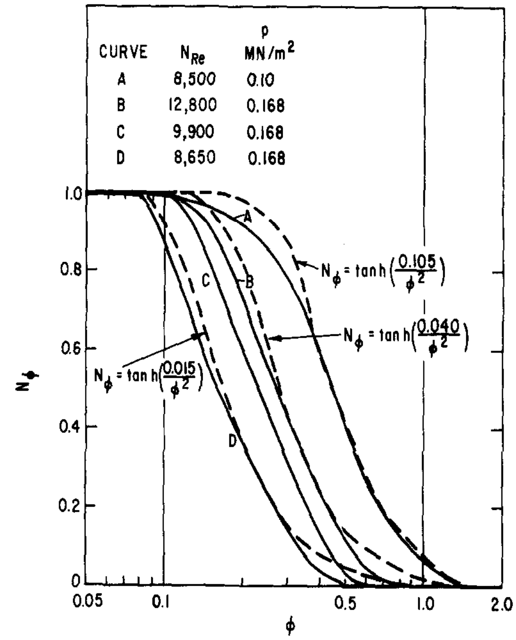

FIG. 6. Correlation of data.

It is postulated that if ϕ is small, i.e. $\phi \ll 1$ the heated tube will behave as if it is fully wet; if ϕ is large, $\phi \geqslant 1$, dryout will occur and a tendency to fully dry conditions will be established.

In view of its significance the quantity ϕ will be called the "dryness parameter". Data for $\overline{N_{Nu}}$ vs ϕ are given as Fig. 5. A clear correlation is achieved with, in this example, a fully wet condition (superscript 'W') for $\phi \leqslant \sim 0.5$, a fully dry condition for $\phi \geqslant \sim 1$ and a partial dryout condition for $0.5 \leqslant \phi \leqslant 1$. The asymptote for $\phi > 1$ is close to that calculated from equation (4) for $\overline{N^D_{Nu}}$.

The form of the solution so far arrived at suggests a canonical correlation scheme of the form;

$$N_\phi(\phi) = \frac{\overline{N_{Nu}} - \overline{N^D_{Nu}}}{\overline{N^W_{Nu}} - \overline{N^D_{Nu}}}. \tag{7}$$

The data are plotted in this fashion in Fig. 6. At a given pressure and Reynolds number the data are suitably represented. The unresolved effects of pressure and of Reynolds number mean that the final correlation represents a smearing of these data. To a good approximation each of the curves shown in Fig. 6 can be represented by a function of the form;

$$N_\phi = \tanh\left(\frac{b}{\phi^2}\right) \tag{8}$$

where b is a parameter of p and N_{Re}. The results are summarized below:

	$(N_\phi)_{\min}$	$(N_\phi)_{av}$	$(N_\phi)_{\max}$
b	0.015	0.040	0.105.

$(N_\phi)_{\min}$ indicates the lower bound of N_ϕ, $(N_\phi)_{\max}$ the upper bound and $(N_\phi)_{av}$ the average value for a given ϕ.

DISCUSSION

The recorded heat-transfer data for an evaporating mist can be grouped according to their moisture content in the three regimes fully wet, partially dry and fully dry.

In the fully wet regime a strong variation with N_{Re} was observed but no effect of the moisture content. It is suspected that the thermal resistance to the mist flow was confined to a thin flowing film on the surface of the tube. The general characteristics of the $\overline{N_{Nu}}$ vs ϕ curve (or h vs q''_{w0}) indicates that no boiling nucleation occurred. It is also of interest to ask whether the mass-transfer resistance of the evaporating water into the steam–air mixture contributes an appreciable fraction of the measured thermal resistance and which would not be present if only the vaporizable component were present. It is in fact easy to show that the mass-transfer resistance is negligible in this case. This is most easily achieved by considering the mass-transfer analog to equation (4) to represent the concentration driving forces. Some manipulations then lead to a simple result for the temperature of the evaporating interface, T_i.

$$\frac{T_i - T_\infty}{T_w - T_\infty} < \left(\frac{N_{Sc}}{N_{Pr}}\right)^{\frac{2}{3}} \frac{C_{PV}\, R_G\, T_\infty}{M_S\, h_{fg}^2}. \tag{9}$$

This result is sensibly pressure independent. The maximum relative error in this case in ignoring the difference $T_i - T_\infty$ is about 4% of $T_w - T_\infty$ for the air–steam–water system.

The uncertainty in the relations $N_\phi = N_\phi(\phi)$ remains unsolved. To ensure a conservative result using the results of this work it is recommended that the curve for $(N_\phi)_{\min}$ be adopted.

The fully dry Nusselt numbers appear similar to those presently in the literature for similar flow systems. Finally we note that the Nusselt numbers have all been reduced with respect to k_V whereas in the interpretation given, a liquid film is responsible for the thermal resistance under wet conditions. This has been a point of convenience in order to directly emphasize the magnitude of the effect of water addition. Therefore, in extrapolating these results to fluids other than water, caution is advised since no empirical information is known to exist on how to parametize with respect to this variable.

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TRANSFERT DE CHALEUR DANS UN BROUILLARD SATURE S'ECOULANT NORMALEMENT A UN CYLINDRE CHAUFFE

Résumé—Un écoulement d'air et de vapeur contenant une fraction f (0,5 à 20 pour cent en poids) de brouillard en suspension attaque un tube rugueux chauffé.

Les coefficients moyens de transfert de chaleur sont caractérisés par un "paramètre de séchage" ϕ défini par:

$$\phi = \frac{q''_{w0}}{G f h_{fg}} \left\{\frac{A_0}{d}\right\}$$

(q''_{w0}, flux thermique, A_0 rapport surface-largeur, G vitesse de débit massique et d diamètre).

Le nombre de Nusselt moyen mesuré est indépendant du taux d'humidité pour $\phi \ll 1$, soit l'état "complètement humide", mais dépend du rapport des densités de la vapeur à celle du liquide et du nombre de Reynolds.

$$\overline{N}^w_{Nu} = 4{,}08 \times 10^{-8}\left(\frac{\rho_L}{\rho_V}\right) N^2_{Re}.$$

De même, à l'état "complètement sec", soit $\phi > \sim 1$, on observe un transfert de chaleur convectif caractéristique;

$$\overline{N}^D_{Nu} = 0{,}250\, N_{Re}^{0{,}647} N_{Pr}^{1/3}.$$

Enfin, dans le cas de l'état "partiellement humide", $\sim 0{,}7 > \phi > 0{,}1$, le transfert de chaleur dépend fortement du degré d'humidité.

$$\frac{\bar{N}_{Nu} - \bar{N}^D_{Nu}}{\bar{N}^W_{Nu} - \bar{N}^D_{Nu}} = \tanh\left(\frac{0{,}015}{\phi^2}\right)$$

à un nombre de Reynolds N_{Re} donné.

WÄRMEÜBERGANG AN QUER ANGESTRÖMTE, BEHEIZTE ZYLINDER BEI GESÄTTIGTER NEBELSTRÖMUNG

Zusammenfassung—Angerauhte, beheizte Rohre wurden von einem Dampf-Luft-Gemisch mit einem Flüssigkeitsanteil f (0,5–20 Gewichtsprozent) umströmt. Der mittlere Wärmeübergangskoeffizient wurde charakterisiert durch einen "Trockenheitsparameter", ϕ, definiert als

$$\phi = \frac{q''_{w0}}{Gfh_{fg}} \left\{\frac{A_0}{d}\right\}$$

(q''_{w0} Wärmestromdichte, A_0 Fläche/Breite-Verhältnis, G Massengeschwindigkeit und d Durchmesser).

Die gemessene mittlere Nusselt-Zahl war unabhängig vom Dampfgehalt für $\phi \ll 1$, den "völlig nassen" Zustand, aber abhängig vom Dampf/Flüssigkeits-Verhältnis und der Reynolds-Zahl:

$$\bar{N}^w_{Nu} = 4{,}08 \times 10^{-8} \left(\frac{\rho_L}{\rho_v}\right) N^2_{Re}.$$

Entsprechend wurde im "völlig trockenen" Zustand, $\phi > \sim 1$, der charakteristische konvektive Wärmeübergang beobachtet:

$$\bar{N}^D_{Nu} = 0{,}250 N^{0{,}647}_{Re} N^{1/3}_{Pr}.$$

Im "teilweise nassen" Zustand schließlich, $\sim 0{,}7 > \phi > 0{,}1$ erwies sich der Wärmeübergang als eine deutliche Funktion des Dampfgehalts bei einer gegebenen Reynolds-Zahl

$$\frac{\bar{N}_{Nu} - \bar{N}^D_{Nu}}{\bar{N}^w_{Nu} - \bar{N}^D_{Nu}} = \tanh\left(\frac{0{,}015}{\phi^2}\right).$$

ПЕРЕНОС ТЕПЛА К ПОТОКУ НАСЫЩЕННОГО ТУМАНА, НАПРАВЛЕННОГО ПО НОРМАЛИ К НАГРЕВАЕМОМУ ЦИЛИНДРУ

Аннотация — Исследуется обтекание шероховатой нагреваемой трубы потоком пара/воздуха, содержащего часть суспензированного тумана f (0,5–20% веса). Осредненные коэффициенты теплообмена характеризуются «параметром сухости» ϕ:

$$\phi = \frac{q''_{w0}}{Gfh_{fg}} \left[\frac{A_0}{d}\right],$$

q''_{w0} — тепловой поток, A_0 — отношение площади к ширине, G — массовая скорость и d — диаметр.

Измеренное осредненное число Нуссельта не зависит от влагосодержания («полностью влажного состояния») при $\phi \ll 1$, а зависит от отношения плотностей жидкости и пара и числа Рейнольдса

$$\bar{N}^w_{Nu} = 4{,}08 \times 10^{-8} \left(\frac{\rho_L}{\rho_v}\right) N^3_{Re}.$$

Кроме того, в случае «полностью сухого» состояния ($\phi > \sim 1$), наблюдалась картина типичного конвективного теплообмена

$$\bar{N}^D_{Nu} = 0{,}250 N_{Re}{}^{0{,}647} N_{Pr}{}^{1/3}.$$

И, наконец, в «частично влажном» состоянии ($\sim 0{,}7 > \phi > 0{,}1$) перенос тепла является функцией влагосодержания при заданном N_{Re}:

$$\frac{\bar{N}_{Nu} - \bar{N}^D_{Nu}}{\bar{N}^w_{Nu} - \bar{N}^D_{Nu}} = \tanh\left(\frac{0{,}015}{\phi^2}\right).$$