

# DISPERSED FLOW FILM BOILING OF NITROGEN WITH SWIRL FLOW

**A. E. BERGLES**

Georgia Institute of Technology, Atlanta, Georgia, U.S.A.

**W. D. FULLER**

Massachusetts Institute of Technology, Cambridge, Massachusetts, U.S.A.

and

**S. J. HYNEK**

Stone & Webster Engineering Corporation, Boston, Massachusetts

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**Abstract**—The heat-transfer characteristics of two-phase film boiling of nitrogen in tubes with tape-generated swirl flow are described. The swirl flow substantially augments the heat transfer and minimizes the thermal non-equilibrium, which is responsible for the relatively poor performance of boilers without twisted-tape inserts. Assuming thermal equilibrium, it is possible to formulate a relatively simple superposition correlation for both evaporation and superheat regions of the once-through boiler. The correlation gives good prediction of wall temperatures along the tube for the entire range of test conditions.

## NOMENCLATURE

<p><math>A</math>, surface area based on inside diameter of tube;</p> <p><math>A_c</math>, free flow area;</p> <p><math>a</math>, radial acceleration in swirl flow given by equation (7);</p> <p><math>C</math>, constant defined in equation (7);</p> <p><math>\bar{c}</math>, concentration of liquid, assuming all liquid in core;</p> <p><math>c_p</math>, specific heat at constant pressure;</p> <p><math>D</math>, inside diameter of tube;</p> <p><math>D_h</math>, hydraulic diameter of tube with twisted tape insert [ft in equation (4)];</p> <p><math>F</math>, function given in equation (15);</p> <p><math>G</math>, mass velocity;</p> <p><math>Gr_v</math>, Grashof number, defined in equation (8);</p> <p><math>g</math>, gravitational acceleration;</p> <p><math>h</math>, heat-transfer coefficient;</p> <p><math>h_{fg}</math>, enthalpy of vaporization;</p> <p><math>h_{fg}^*</math>, enthalpy of vaporization corrected for vapor superheat, equation (9);</p>	<p><math>K_1</math>, constant in equation (11);</p> <p><math>k</math>, thermal conductivity;</p> <p><math>L</math>, test-section heated length;</p> <p><math>N_2</math>, surface density of droplets;</p> <p><math>N_3</math>, volume density of droplets;</p> <p><math>Nu_{v,v}</math>, Nusselt number = <math>h_{w,v} D_h / k_v</math>;</p> <p><math>P</math>, pressure;</p> <p><math>\Delta P</math>, pressure difference between axial stations of test section;</p> <p><math>Pr_v</math>, Prandtl number = <math>(c_p \mu / k)_v</math>;</p> <p><math>q''</math>, heat flux;</p> <p><math>Re_v</math>, Reynolds number, defined in equation (6);</p> <p><math>T</math>, temperature;</p> <p><math>\Delta T</math>, temperature difference = <math>T_w - T_b</math>;</p> <p><math>V_w</math>, average axial velocity for two-phase mixture;</p> <p><math>X</math>, flowing equilibrium equality;</p> <p><math>y</math>, tube diameters per 180° tape twist;</p> <p><math>z</math>, axial position along heated length;</p> <p><math>\beta</math>, volumetric coefficient of thermal expansion;</p>
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$\delta$ ,	droplet diameter ;
$\rho$ ,	density ;
$\mu$ ,	dynamic viscosity ;
$\nu$ ,	kinematic viscosity.

#### Subscripts

$b$ ,	bulk fluid condition ;
DO,	dryout condition ;
$f$ ,	vapor properties evaluated at film temperature, $(T_w + T_b)/2$ ;
in,	condition at entrance to heated section ;
$l$ ,	liquid condition ;
sat,	saturation condition ;
$v$ ,	vapor condition ;
$w$ ,	condition at inside wall of tube.

### INTRODUCTION

SYSTEMS involving forced-convection vaporization may operate at least partially in a post dryout or liquid deficient mode. This occurs commonly in once-through steam boilers, in rocket motors which are regeneratively cooled by vaporization of the cryogenic fuel, and in cryogenic vaporizers. Post dryout behavior is also of interest in determining the transient conditions during a boiling water reactor accident. At the high quality conditions normally of interest, the heated surface is nominally dry, and the liquid flows as droplets dispersed in the channel core; thus the regime can be described as dispersed flow film boiling. One of the significant features of established dispersed flow film boiling is the existence of superheat in the vapor well below 100 per cent equilibrium quality, and the presence of liquid droplets downstream of the 100 per cent quality point. Direct evidence of this nonequilibrium situation has been given by Parker and Grosh [1] who observed water droplets in an exit sight section at steam qualities greater than 100 per cent, as well as by Forslund and Rohsenow [2] who made similar observations for nitrogen. The existence of a considerable amount of vapor superheat in the presence of liquid droplets has

been established by Forslund and Rohsenow [2] using a helium tracer technique and by Mueller [3] using a steam temperature probe.

Numerous empirical correlations have been developed for the heat-transfer coefficient in dispersed flow film boiling [4]. Lavery and Rohsenow [5] suggested a more fundamental approach based on a two-step heat-transfer process, from the wall to the superheated vapor and then from the vapor to the liquid droplets. Forslund and Rohsenow [2], Bennett *et al.* [6] and Hynek [7] extended this model and found that experimental data could be reasonably well described with the proper choice of empirical constants. It is clear that dispersed flow film boiling is not a particularly efficient heat-transfer process. With nitrogen, for example, the non-equilibrium effects cause actual qualities to be as much as 50 per cent below equilibrium qualities; furthermore, liquid droplets are still observed at equilibrium qualities of 300 per cent [2]. As a consequence of this, boilers operate at high surface temperatures and must contain a large surface area to effect complete vaporization.

An examination of methods to augment convective heat transfer suggests that swirl flow should be particularly effective in the case of dispersed flow film boiling [8]. Burke and Rawdon [9] presented one set of data for film boiling of nitrogen which indicated that heat-transfer coefficients were substantially increased with tape-generated swirl flow. With the exception of some work with mercury [10, 11], which is normally non-wetting, this exploratory study by Burke and Rawdon appeared to be the only source of information relating to the influence of tape-generated swirl flow on dispersed flow film boiling. Accordingly, a systematic study of this phenomenon was undertaken as a continuation of a project in the M.I.T. Heat Transfer Laboratory concerned with the general problem of film boiling [2, 5, 7]. The present paper describes the correlation developed for the large amount of experimental data obtained in this study.

### CHARACTERISTICS OF DISPERSED FLOW FILM BOILING

The boiling data were obtained with a 0.4 in. i.d., 96 in. long circular Inconel tube which was oriented vertically. Subcooled liquid nitrogen was supplied to the electrically heated test section from a large dewar. Five local pressures and 24 local wall temperatures were measured at equally spaced increments. After making measurements with the standard tube, full-length twisted tapes (Inconel, 0.018 in. thick) were inserted into the tube. The tape twists were  $\gamma = 8.5$  and 4.1, and the maximum estimated clearance was 0.004 in.

Two modes of operation were possible depending on the start-up conditions: Type I operation was achieved by heating the tube before supplying the liquid, while Type II operation involved system cooldown before power was applied. For all tests, the inlet conditions were slightly subcooled and the pressure level in the test section was kept at 20–25 psia. A wide range of flow rates and heat fluxes was covered. The heat flux, which was essentially uniform along the tube due to the favorable resistivity characteristics of the Inconel, was based on the total heat input and the nominal tube inner surface area. This does not consider the heat generated in the tape (about 8 per cent of the total heat input); however, it follows the convention used in most previous swirl flow investigations.

A detailed description of the apparatus, procedure, and data reduction can be found in [7, 12]. Only brief mention of the results will be given here since the data are presented in detail in these reports as well as in [13].

The general characteristics of film boiling with and without twisted tape inserts are shown in Fig. 1. Typical temperature profiles for the two modes of boiling are shown in Fig. 2, where each of the curves represents the 24 wall temperature measurements for that run. Consider first the results for the plain tube initially preheated (Type I). The temperature profile for Run 47 in Fig. 2 is typical for this condition.

Within the first few inches of the tube the flow pattern corresponds to the classical picture of film boiling since there is a vapor annulus at the wall and a liquid core. The subcooling is removed in a short length, however, and the

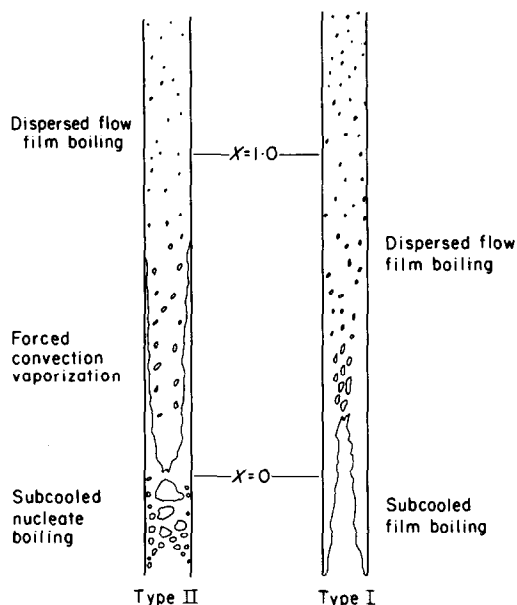


FIG. 1. Schematic representation of the two types of dispersed flow film boiling.

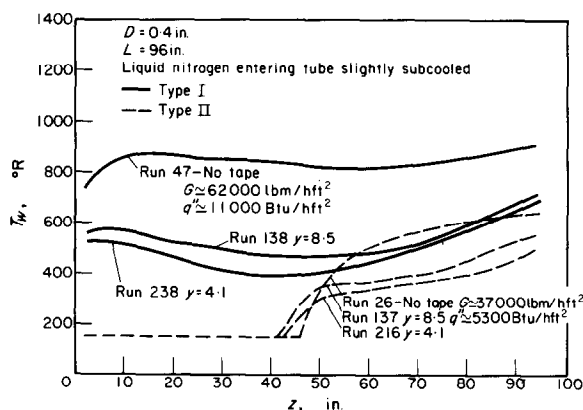


FIG. 2. Comparison of temperature profiles for film boiling in tubes with and without twisted tape inserts.

core flow becomes dispersed (the void fraction is about 80 per cent at a quality of 3 per cent). The wall temperature remains constant or

decreases somewhat in the evaporation region, then increases in the superheat region. When the flow was established before applying power (Type II), the dryout point originated at the tube exit and moved upstream with increasing power. As shown for Run 26 in Fig. 2, the wall temperature is close to the saturation temperature before dryout, as would be expected from the high heat-transfer coefficients for forced convection vaporization. The wall temperature rises rapidly at the dryout point and increases gradually as the superheat region is entered.

Twisted tape data are shown for comparison with the straight tube data in Fig. 2. As expected, the wall temperatures are markedly reduced with the tape inserts, with the reduction being greater for the tighter tape twist. In broad terms, the increased heat-transfer coefficients with tape generated swirl flow are due to enhancement of the single-phase vapor coefficient and increased rate of droplet evaporation at the wall due to centrifuging of the drops to the wall. This latter effect is particularly important since the amount of vapor superheat required for evaporation is minimized.

expected. With Type II tests, the dryout point also moves upstream as the heat flux is increased. The most significant feature of these data is that the wall temperatures for Type II runs become essentially coincident with the data for the comparable Type I run when the film boiling is well established. This suggests that any swirl flow correlation or analytical formulation should be applicable to either type. As noted in Fig. 3, both types of film boiling are possible at the same mass flux and heat flux, within certain ranges of both variables. As in pool boiling, it is possible to have either film boiling or nucleate boiling at the same heat flux as long as conditions are above the Leidenfrost point.

In general, it has been established that twisted tape inserts are a simple and effective method of augmenting heat transfer in dispersed flow film boiling. The present tests with nitrogen indicate that heat-transfer coefficients can be increased as much as 200 per cent over the straight flow value at comparable conditions. When swirl and straight flow are compared at equal pressure gradient, it was demonstrated that local coefficients can be increased by over 50 per cent. On the basis of equal pressure drop, the length of the boiler can be reduced by over 50 per cent for the same heat-transfer rate. [13]

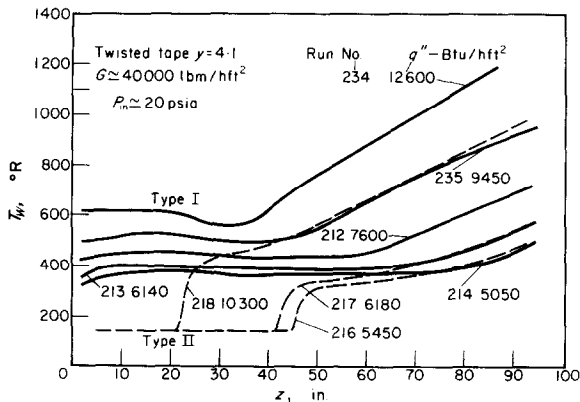


FIG. 3. Influence of heat flux on wall temperatures for Type I and Type II swirl runs.

Figure 3 presents a comparison of typical temperature profiles which illustrate the effect of heat flux at a given mass velocity and tape twist. For the Type I runs, the increasing temperature level with increasing heat flux is to be

## HEAT-TRANSFER CORRELATION

### General assumptions

The heat-transfer process in dispersed flow film boiling can be described in a general way as a superposition of heat transfer from the wall to the vapor and from the wall to the droplets which impinge on the wall. Although the validity of superimposing complex heat-transfer processes, without considering any cross-coupling, can be questioned on fundamental grounds, the concept has been found to be extremely useful in developing correlations for both single-phase and boiling situations. Superposition of vapor and droplet contributions for dispersed flow film boiling has been suggested by Parker and Grosh [1] and

Poppendiek *et al.* [11], and was a basic assumption in the semi-analytical correlation developed by Forslund and Rohsenow [2]. There is precedent, then, for seeking an additive correlation for the present data for dispersed flow film boiling with swirl flow.

The complicating feature of film boiling in axial flow is that the core flow consists of liquid drops and superheated vapor, hence, a simple heat balance is inadequate to predict the local vapor temperature. A complicated computer program is then required to march out a solution for the wall temperature at various positions along the tube [2]. No simple correlation equation has yet been developed for the axial flow case. With swirl flow, however, it appears that the non-equilibrium effects are less important. Since the droplets are centrifuged to the wall, it is reasonable to assume that the vaporization takes place primarily at the wall without the bulk vapor becoming appreciably superheated. The case for a process which is basically at equilibrium rests on both visual and quantitative observations. No appreciable core droplet flow was observed in the exit visual section for  $X > 1$ . Although it was estimated that hydrodynamic considerations would permit some liquid to flow along the twisted tape [7], it is probable that this liquid would be rapidly evaporated by the heat generation in the tape or by heat transfer from slightly superheated vapor. The most compelling evidence for equilibrium, however, is given by Fig. 3 where it is seen that the wall temperature profiles for Type I and Type II runs coincide. As demonstrated in [7], this does not occur with axial flow since the Type I regime builds up non-equilibrium from the tube inlet, whereas the Type II flow must be essentially in equilibrium at the dryout point. The wall temperature for the Type I runs is thus considerably higher than that for the Type II run at the dryout position.

#### Development of the correlation

Utilizing superposition with no cross-coupling and assuming the vapor to be at

saturation temperature, the total heat-transfer rate for  $X < 1$  is given as

$$q'' A = h_{w,l} A_l (T_w - T_{sat}) + h_{w,v} A_v (T_w - T_{sat}). \quad (1)$$

Since the void fractions are large above qualities of a few per cent, it is reasonable to assume that  $A_v \approx A$ ; thus the heat flux can be given by

$$q'' \approx \{h_{w,l} (A_l/A) + h_{w,v}\} (T_w - T_{sat}). \quad (2)$$

For  $X > 1$ , the droplet contribution vanishes and the superheated vapor temperature is given by a simple heat balance

$$q'' = h_{w,v} (T_w - T_v). \quad (3)$$

Of the available correlations for swirl flow of single-phase fluids, it is most appropriate to consider the correlation of Thorsen and Landis [14] which has been demonstrated to be valid for gases at high  $\Delta T$ . This correlation which includes curvature, buoyancy, and radial property variation effects, is given in the following form:

$$Nu_v = C Re_v^{0.8} Pr_v^{0.4} (T_b/T_w)^{0.32} \times (1 + 0.25 Gr_v^{0.5}/Re_v) \quad (4)$$

where

$$C = 0.021 [1 + 0.035 \pi^2 / Dy^2 (1 + \pi^2 / 4y^2)].$$

The Reynolds number must be expressed in terms of the average vapor velocity so as to cover both the evaporating and superheat regions. The vapor velocity can be approximated by the throughput velocity based on homogeneous flow

$$V_a = \frac{G}{\rho_l} \left[ X \left( \frac{\rho_l}{\rho_v} - 1 \right) + 1 \right]. \quad (5)$$

The Reynolds number then becomes

$$Re_v = \frac{\rho_v V_a D_h}{\mu_v} = \frac{G D_h}{\mu_v} \left[ X + (1 - X) \frac{\rho_v}{\rho_l} \right]. \quad (6)$$

Since the acceleration for a rotating slug flow model is given by

$$a = (V_a\pi)^2/2Dy^2 \quad (7)$$

the Grashof number can be expressed as

$$Gr_v = \frac{(V_a\pi)^2\beta_{v,f}\Delta TD_h^3}{2Dy^2\nu_v^2} \quad (8)$$

It is appropriate to base the droplet heat transfer on a correlation for droplets on a horizontal hot surface in a gravity field. The correlation of Baumeister *et al.* [15] is given as

$$h_{w,l} = 1.1 \left[ \frac{k_v^3 h_{fg}^* a \rho_l \rho_v}{(T_w - T_{sat}) \mu_v \left( \frac{\pi \delta^3}{6} \right)^{\frac{1}{3}}} \right]^{\frac{1}{4}} \quad (9)$$

where

$$h_{fg}^* = h_{fg} [1.0 + (7/20)c_{pv}(T_w - T_{sat})/h_{fg}]^{-3}$$

This correlation was derived from data at normal gravity ( $a = g$ ). In the present case, the acceleration would be given by equation (7). It is noted that the heat-transfer coefficient for saturated pool film boiling exhibits a somewhat stronger dependence on acceleration ( $h \sim a^{0.336}$ ) [16]; however, this has not been verified for Leidenfrost-type behavior. A further consideration is the effect of relative velocity between the liquid drops and the heater wall. Schoessow and Baumeister [17] have demonstrated that the coefficients are higher for moving drops. The effect of relative velocity was not included in the present correlation, primarily due to the lack of a good estimate of the relative velocity. However, the effect is ultimately included in the final empirical constant given below in equation (15).

The drop size,  $\delta$ , in equation (9) and the surface covered by drops,  $A_l/A$ , in equation (2) must now be accounted for. Assume that  $\delta$  can be interpreted as a mean effective drop size; then

$$A_l/A = N_2\pi\delta^2/4 \quad (10)$$

where  $N_2$  is the drop density at the heated surface.  $N_2$  can be related to the volume density of droplets in the core,  $N_3$ , by

$$N_2 = K_1(N_3)^{\frac{2}{3}} \quad (11)$$

where  $K_1$  is quite probably a function of the radial acceleration induced by the swirl flow.

The core density of droplets is related to the concentration of liquid in the core by

$$N_3 = \frac{\bar{c}}{(\pi/6)\delta^3\rho_l} \quad (12)$$

where  $\bar{c}$  is estimated by again using the homogeneous model

$$\bar{c} = \frac{\rho_l(1-X)}{X(\rho_l/\rho_v - 1) + 1} \quad (13)$$

A final expression for the droplet heat-transfer coefficient is now obtained by combining equations (9)–(13):

$$h_{w,l} \frac{A_l}{A} = 1.1 \left\{ \frac{k_v^3 h_{fg}^* a \rho_l \rho_v}{\Delta T \mu_v (\pi/6)^{\frac{1}{3}}} \right\}^{\frac{1}{4}} \times \left\{ \frac{(1-X)(6/\pi)}{X(\rho_l/\rho_v - 1) + 1} \right\}^{\frac{1}{4}} \frac{\pi}{4} \left\{ \frac{K_1}{\delta^{\frac{2}{3}}} \right\} \quad (14)$$

The inevitable empirical constant appears as the last bracketed term in this equation. As a first approximation, it would be reasonable to assume both the surface density function and the drop size a function of the *ges* induced by the swirl flow:

$$\frac{K_1}{\delta^{\frac{2}{3}}} = F(a/g) \quad (15)$$

The test of this assumption will come from applying the complete superposition model to the experimental data.

#### APPLICATION OF CORRELATION EQUATION TO DATA

##### Pure vapor runs

The first test of equation (4) was made by

applying it to data for gaseous nitrogen at cryogenic temperatures. These tests were carried out once the liquid in the supply dewar was

different when data for a wide variety of tape twists are examined.

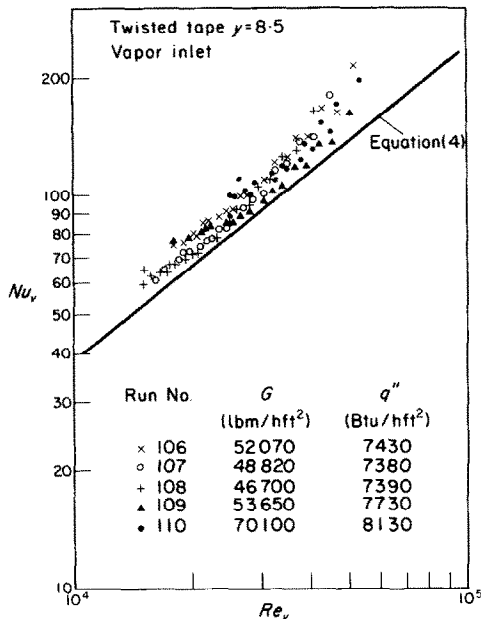


FIG. 4. Correlation of data for swirl flow of single-phase vapor.

expended; the pressurizing nitrogen flowed through the cold dewar and dewar delivery line and reached the test section at a temperature only slightly above saturation. Since this method precluded long running times, only several runs were made for the  $y = 8.5$  tape. The results shown in Fig. 4 indicate that the fully developed heat-transfer coefficients are about 10 per cent above the values predicted by equation (4). Data for the entrance region (points lying to the right) are considerably higher as would be expected. This suggests that equation (4) is quite adequate for the present test conditions. It should be pointed out, however, that other values of the basic constant (0.021) and temperature ratio exponent (0.32) are frequently reported in the literature for correlation of data for gases. Furthermore, the curvature or tape twist effect appearing in the factor  $C$  may be slightly

#### Superheat data

For  $X > 1$ , equation (3) applies and the derived heat-transfer coefficients should be quite well predicted by equation (4). The data for a tape twist of  $y = 8.5$ , shown in Fig. 5, are in very good agreement with equation (4). As shown in Fig. 6, the data for  $y = 4.1$  lie on the average about 10 per cent below the correlation. This might be due to inadequate specification of the tape twist effect. For both tape twists there is some indication of a heat flux effect, with higher heat-transfer coefficients at higher heat fluxes. This is in line with the results for straight flow [2], and suggests that there may be some non-equilibrium still present with the swirl flow. However, the effect is not large enough or conclusive enough to warrant relaxing the assumption of equilibrium.

In general, the pure vapor runs and the data for the superheat region indicate that equation (4) gives an adequate representation of the swirl flow of nitrogen vapor; furthermore the swirl flow appears to promote equilibrium flow conditions. The scatter plot for all data in the superheat region (Fig. 7) further demonstrates the validity of this approach, since over 80 per cent of the wall temperatures are predicted to within 10 per cent. The pure vapor correlation can thus be kept in its recommended form, in keeping with the spirit of the superposition model. In addition, since the complexities of non-equilibrium do not need to be introduced for the superheat region, it is probable that the equilibrium assumption will be equally valid for the vaporization region.

#### Evaporation region

The droplet heat-transfer term in the superposition model has been formulated so that only one empirical function remains to be determined. This function,  $F \sim (a/g)$ , can be backed-out from the data for  $X > 1$  by subtracting the convective heat flux from the total heat

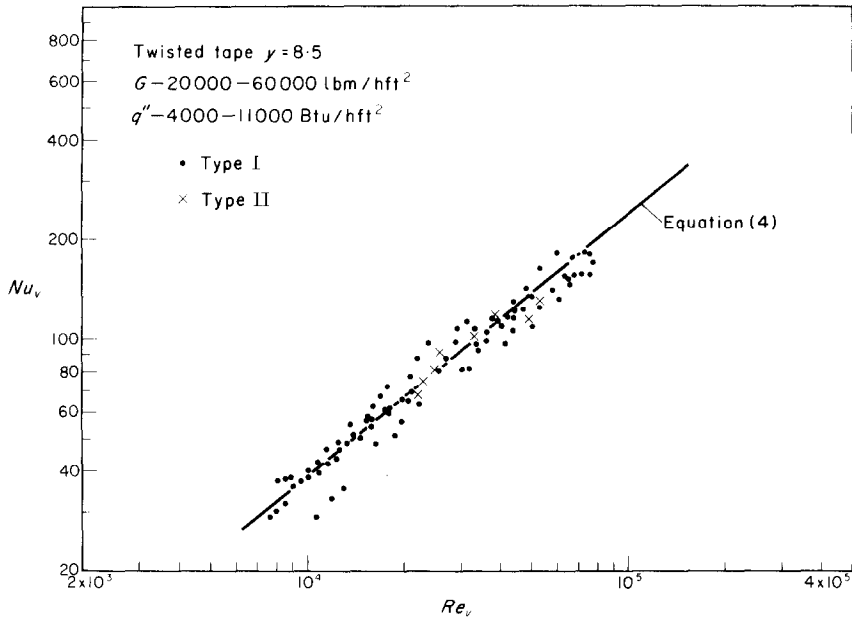


FIG. 5. Comparison of swirl data for superheat region with correlation for single phase vapor;  $\gamma = 8.5$ .

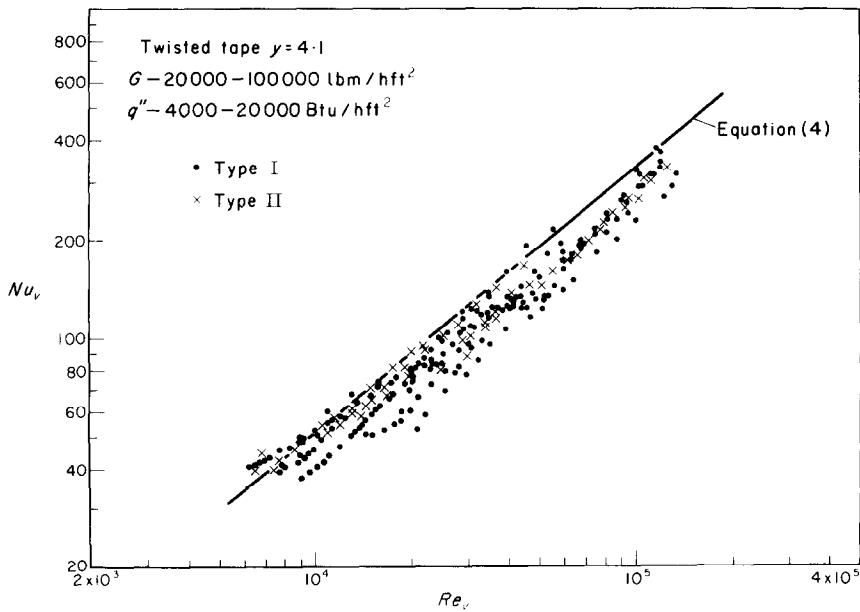


FIG. 6. Comparison of swirl data for superheat region with correlation for single-phase vapor;  $\gamma = 4.1$ .



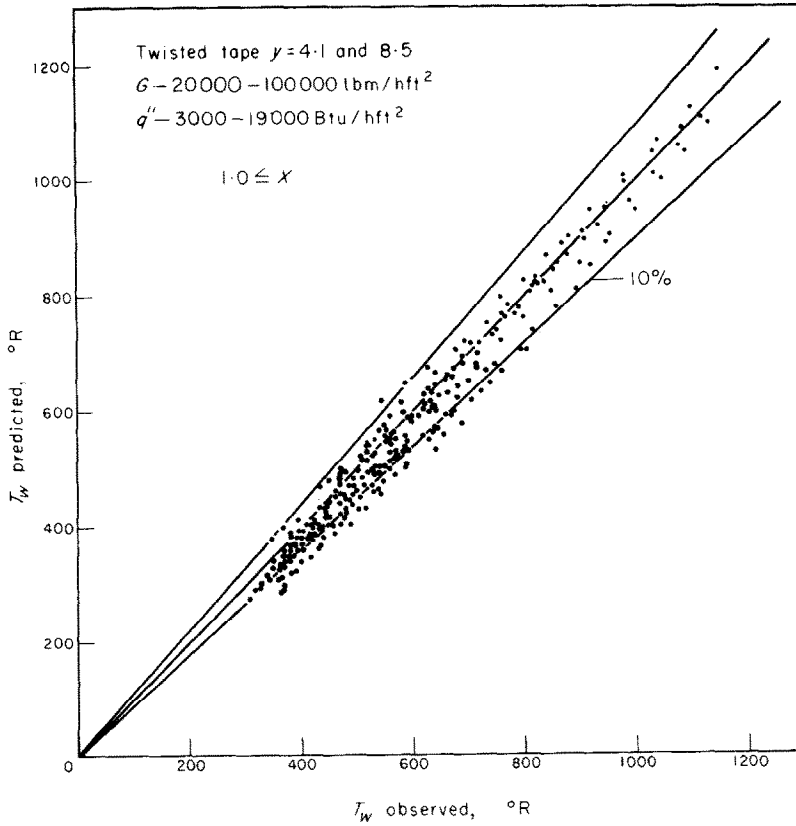


FIG. 7. General comparison of predicted and observed wall temperatures in superheat region.

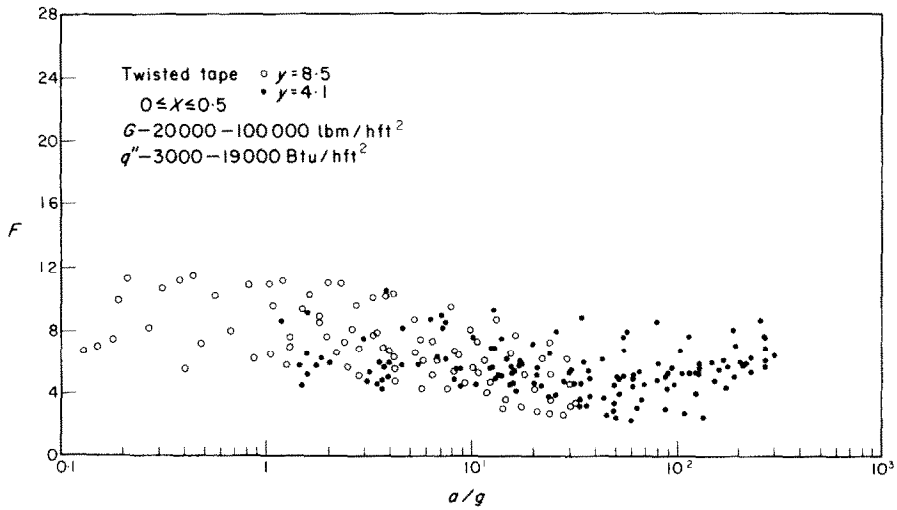


FIG. 8. The function  $F(a/g)$  derived from experimental data.

flux. This operation was performed for the entire range of data, with the result shown in Fig. 8. In order to avoid introduction of undue scatter to subtraction of numbers almost equal in magnitude, only the data for  $X \leq 0.5$  were used. This is not a problem since the drops contribute little to the heat transfer at high qualities.

At first glance, the case for a simple function does not appear to be very promising since there is a rather large scatter in the data. Although the trends were not entirely consistent from run to run, there was some evidence that  $F$  at a given gee value was higher for higher heat fluxes and

lower velocities, particularly at lower  $a/g$ . These trends may be related to drop size variation which is not accounted for by acceleration. At high heat flux the *axial* acceleration of the flow is greater and smaller droplets are expected in the core on the basis of critical Weber number calculations [2]. In addition to this core effect, any streamers or agglomerated drops on the wall would be more readily broken up at a hotter surface. Since drop size is included in  $F$ , this could explain the observed increase in  $F$  with heat flux. On the other hand, with higher mass fluxes, the residence time is reduced and

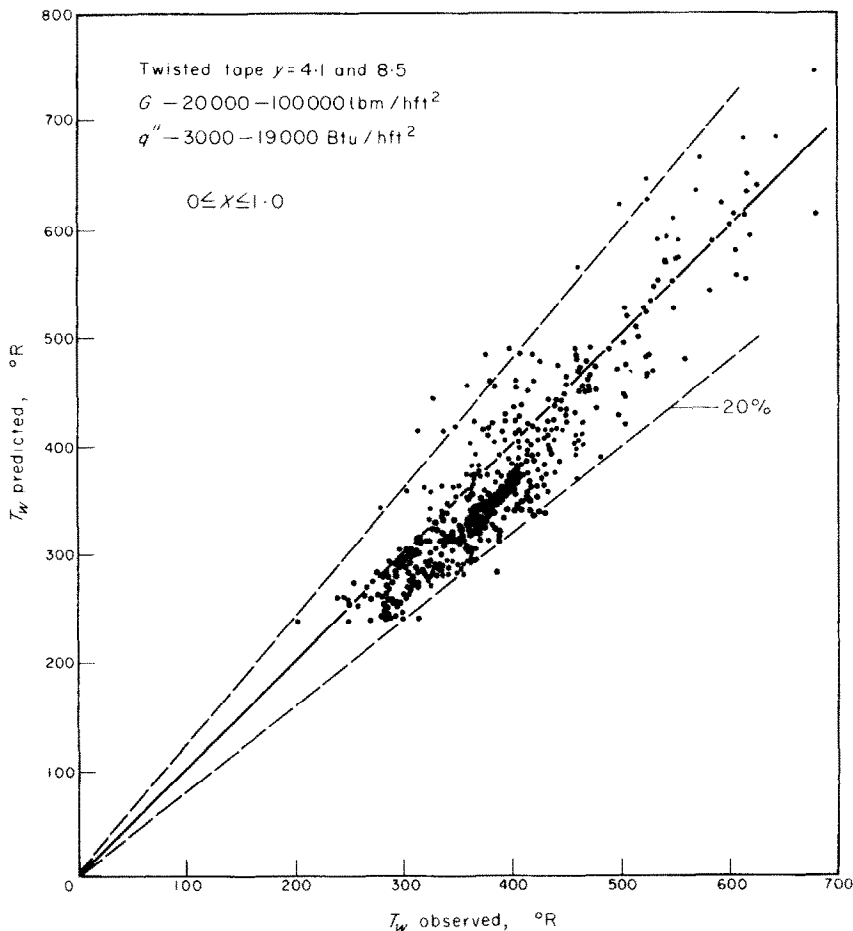


FIG. 9. General comparison of predicted and observed wall temperatures in vaporization region ( $F = 7.0$ ).

there would be less time for the drops to shatter ;  $F$  could then decrease with increasing  $G$ . These factors point out the difficulty of assuming a mean effective drop size, or a drop size which is simply dependent on the radical acceleration.

While work still needs to be done to resolve more clearly the processes in the evaporating region, there is merit to picking a simple function  $F$  and completing the correlation. The most straightforward approach is to set  $F$  equal to a constant since there is no particular dependence on the acceleration. This may be due to compensating effects of distribution and drop size. At higher  $a/g$ , more of the available liquid should be at the wall; however, there would then be more chance of agglomeration. The larger drops or rivulets would have a lower heat-transfer coefficient. In any event, several constant values of  $F$  were chosen and used in equation (3) to obtain a prediction of the wall temperature. For  $F = 7$  it was found that 90 per cent of the data were correlated to within 20 per cent as shown in Fig. 9. The agreement is good enough to recommend the superposition correlation for design.

#### Prediction of complete temperature profile

By way of summary, it is appropriate to compare typical measured temperature profiles

with the predictions of equation (2). As shown in Fig. 10, the general shape of the profile is rather well given by the correlation. The magnitude of the temperature is also quite well predicted for these runs, and in general, as shown in Figs. 7 and 9.

#### CONCLUSIONS

Visual and quantitative observations indicate that the large departures from thermal equilibrium, characteristic of nitrogen in dispersed flow film boiling in empty tubes, are not present with swirl flow. Assuming thermal equilibrium, the boiler can be simply divided into evaporation and superheat regions. The heat transfer in the evaporation section can be described by a superposition of vapor swirl flow and droplet film boiling, while the superheat section can be treated as swirl flow of vapor. Conventional correlations can be utilized for both regions. The additive correlation gives a rather good prediction of wall temperatures along the tube for the entire range of test conditions.

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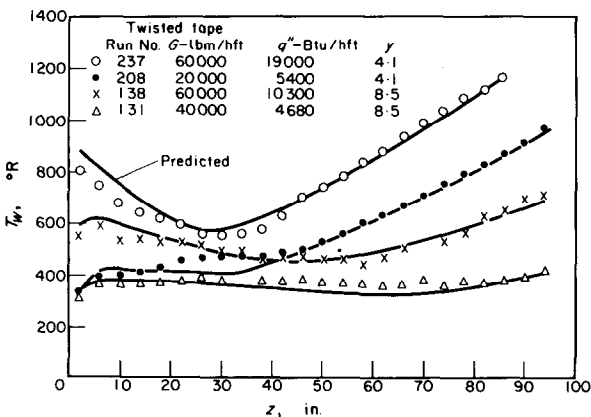


FIG. 10. Comparison of predicted and observed wall temperatures for typical runs.

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### EBULLITION EN FILM D'AZOTE AVEC DISPERSION PAR UN ÉCOULEMENT TOURBILLONNAIRE

**Résumé**—On décrit les caractéristiques de transfert thermique dans l'ébullition en film d'azote dans des tubes munis de rubans générateurs d'écoulement tourbillonnaire. L'écoulement tourbillonnaire augmente notablement le transfert thermique et diminue le non-équilibre thermique qui est responsable de la performance relativement faible des bouilleurs sans introduction de rubans hélicoïdaux. En supposant un équilibre thermique, il est possible de formuler une corrélation de superposition relativement simple pour à la fois les régions d'évaporation et de surchauffe du bouilleur. La corrélation donne une bonne prévision des températures de la paroi le long du tube pour le domaine entier des conditions d'essai.

### FILMSIEDEN VON STICKSTOFF MIT WIRBELSTRÖMUNG

**Zusammenfassung**—Die Wärmeübertragungsvorgänge beim Zwei-Phasen-Filmsieden von Stickstoff in Röhren mit Drallströmung werden beschrieben. Die Drallströmung vergrößert den Wärmeaustausch wesentlich und verringert das thermische Nicht-Gleichgewicht, das verantwortlich ist für die relativ schwache Leistung von Boilern ohne Spiralband-Einsätze. Unter der Annahme thermischen Gleichgewichts ist es möglich, eine ziemlich einfache Superpositions-Beziehung für das Verdampfungs- und Überhitzungsgebiet von Durchlaufverdampfern zu formulieren. Die Beziehung bietet eine gute Voraussage der Wandtemperatur entlang des Rohres für den ganzen Bereich der Versuchsbedingungen.

### ДИСПЕРГИРОВАННОЕ ПЛЕНОЧНОЕ КИПЕНИЕ ЗАВИХРЕННОГО ТЕЧЕНИЯ АЗОТА

**Аннотация**—Описываются теплообменные характеристики двухфазного пленочного кипения азота при течении в трубе, завихренным турбулизатором. Завихренный поток существенно улучшает теплообмен и сводит к минимуму тепловое неравновесие, приводящее к сравнительно плохой работе бойлеров без турбулизирующих вставок в трубе. Принимая существование теплового равновесия, можно сформулировать относительно простую обобщающую корреляцию, применимую как для области испарения, так и для области перегрева пара в прямоточном бойлере. Корреляция дает хорошее предсказание температуры стенки вдоль трубы во всем диапазоне условий опытов.