of the agitator Reynolds number ($D_a^2 N\rho/\mu$) up to a value of $2{\cdot}1 \times 10^5$ of this number, corresponding to a speed of 500 rev/min. This value of 0·67 agreed with the same value reported recently by Oldshue and Gretton [4], and Nooruddin and Rao [1] for turbine agitation in agitated vessels.

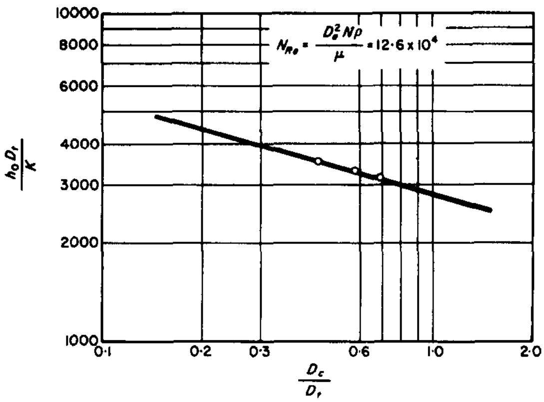

FIG. 3.

CORRELATION

The effect of Prandtl number ($C_p\mu/K$) has been studied by Nooruddin and Rao [1] who found that (h_oD_t/K) varied with 0·33 power of $C_p\mu/K$.

The following correlation incorporating the effect of Prandtl number has been obtained from the treatment of the experimental data

$$\frac{h_o D_t}{K} = 0{\cdot}18 \left(\frac{D_a^2 N\rho}{\mu}\right)^{0\cdot67} \left(\frac{C_p\mu}{K}\right)^{0\cdot33} \left(\frac{d_o}{D_t}\right)^{-0\cdot48} \times \left(\frac{D_c}{D_t}\right)^{-0\cdot27} \left(\frac{H_a}{D_t}\right)^{0\cdot14}. \quad (3)$$

This correlation predicts our experimental results with an average deviation of 1·7 per cent and a maximum deviation of 3·8 per cent; it could be utilized to estimate, h_o, coil outside film heat-transfer coefficients, in agitated vessels, using a 4-flat blade turbine agitator for $D_a = \frac{1}{3}D_t$.

Values of β were correlated as a function of the coil curvature ratio, d_i/D_c, which was varied from 0·0264 to 0·0603 in the present work:

$$\frac{h_{i(\text{coil})}}{h_{i(\text{st. tube})}} = \beta = \left[1 + 3{\cdot}46\left(\frac{d_i}{D_c}\right)\right]. \quad (4)$$

CONCLUSIONS

For heat transfer through coiled tubes in agitated vessels, using flat-blade turbine agitator, the proposed correlation, equation (3), could estimate h_o values as a function of the geometrical parameters of helical coil and also the depth of the agitator for the standard configuration of the vessel for which $D_a = \frac{1}{3}D_t$. Further, the modified Dittus–Boelter equation with the included correction factor, β could be satisfactorily used for predicting the inside film heat-transfer coefficient, h_i, of any helical coil.

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Int. J. Heat Mass Transfer. Vol. 10, pp. 397–401. Pergamon Press Ltd. 1967. Printed in Great Britain

FORCED CONVECTION BOILING INSIDE HELICALLY-COILED TUBES

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(*Received* 14 *June* 1966)

THE AUTHORS are currently conducting a study of forced convection boiling inside electrically heated, helically coiled, tubes. Some of the phenomena appear to be sufficiently unexpected and interesting to call early attention to them by this communication. A more complete presentation of results and analysis will appear later.

The incentive for this study was the possibility of achieving continuous de-entrainment of liquid droplets because of the large radial accelerations induced by the helical path. Continuous separation of phases would make production of high quality or superheated outlet vapor streams easier by eliminating the fog flow regime.

The feasibility of single pass production of superheated

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vapor from subcooled liquid in a steam-heated coil had already been established by Yudovich [1]. Local flow and heat-transfer mechanisms could not be inferred from that study because the local heat flux and quality as a function of length were unknown. The present authors used a coil as a resistance across a direct-current welder. The local heat flux is thus known, and local heat-transfer coefficients can be computed from surface temperatures. We expected that the coefficient would be much lower on the side of the tube towards the axis of the coil since the outward radial acceleration would be expected to remove the liquid from this surface. Hence, we used Inconel 600 tubing to withstand the high temperatures and temperature gradients expected.

Two coils were constructed from 10-ft lengths of $\frac{5}{8}$-in o.d. × 0·492 in i.d. Inconel 600 tubing; the heated length of each was 9·35 ft. The mean helix diameters were 9·86 in and 20·5 in. Thermocouples were cemented to the outside tube surface at four positions around the tube and at 1-ft intervals axially (Fig. 1). Pressure taps were located at the 12 o'clock positions. Heat fluxes up to 81 000 Btu/h ft^2 to the stream were achieved. Distilled water was circulated by a sliding vane pump. Flow rates ranged up to 306 lb/h and inlet pressures up to 28 psia. Runs were made in the single phase, subcooled boiling, and quality boiling regimes with a few runs producing pure vapor with superheats up to 50 degF at the exit.

The data have been reduced to local values of the heat-transfer coefficients. Corrections were made for non-uniform heat generation due to distortion of the tube during coiling and for circumferential wall conduction effects.

Results of two typical runs on the 9·86-in dia. coil are summarized in Figs. 2 and 3. The abscissa (local quality) has been calculated from an enthalpy balance assuming thermodynamic equilibrium between the liquid and vapor. Ordinarily this is a poor assumption for high quality streams; direct contact between vapor and wall usually causes superheating even though numerous liquid droplets are suspended in the vapor. For the present study, visual observation of the outlet streams for the saturated and superheated vapor runs showed no evidence of entrained droplets. Our current understanding of the mechanism (see below) also suggests that the equilibrium assumption is valid, at least for qualities below 90–95 per cent. Heat-transfer coefficients at the 3, 9, and 12 o'clock positions (h_3, h_9, and h_{12}) are also shown as a function of local quality. An average coefficient obtained by averaging the four local coefficients at each longitudinal position is also shown. (For Fig. 2, h_{av} is nearly identical to h_{12}.) For comparison, the predicted dry vapor coefficients for the exit conditions are 109 Btu/h ft^2 degF for Run 5 and 87·5 for Run 20, based on the correlation of Seban and McLaughlin [2].

The key observation from Figs. 2 and 3 is that h_3 and h_9 are quite comparable over the entire range, contrary to our expectations. The coefficients that do deteriorate at high qualities are h_6 and h_{12}. We conclude that a liquid film is maintained around the entire periphery of the tube to very high qualities; when the film does break down because of insufficient liquid, it does so first at the 6 and 12 o'clock positions.

We tentatively advance the following mechanism: it is well-established that a secondary flow exists in single-phase flow in helical coils (as shown in Fig. 1) [3]. We postulate that this secondary flow exists in the vapor core of the two-phase flow. This secondary flow causes liquid flow from the outer to the inner wall of the tube, constantly replenishing the losses due to evaporation and entrainment. Liquid on the wall is subject to small radial acceleration effects

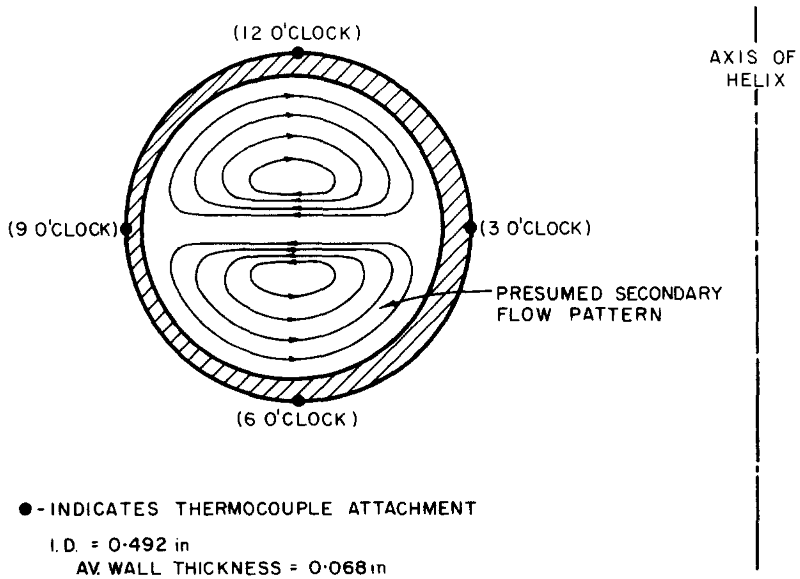

FIG. 1. Diagram of tube identifying thermocouple positions.

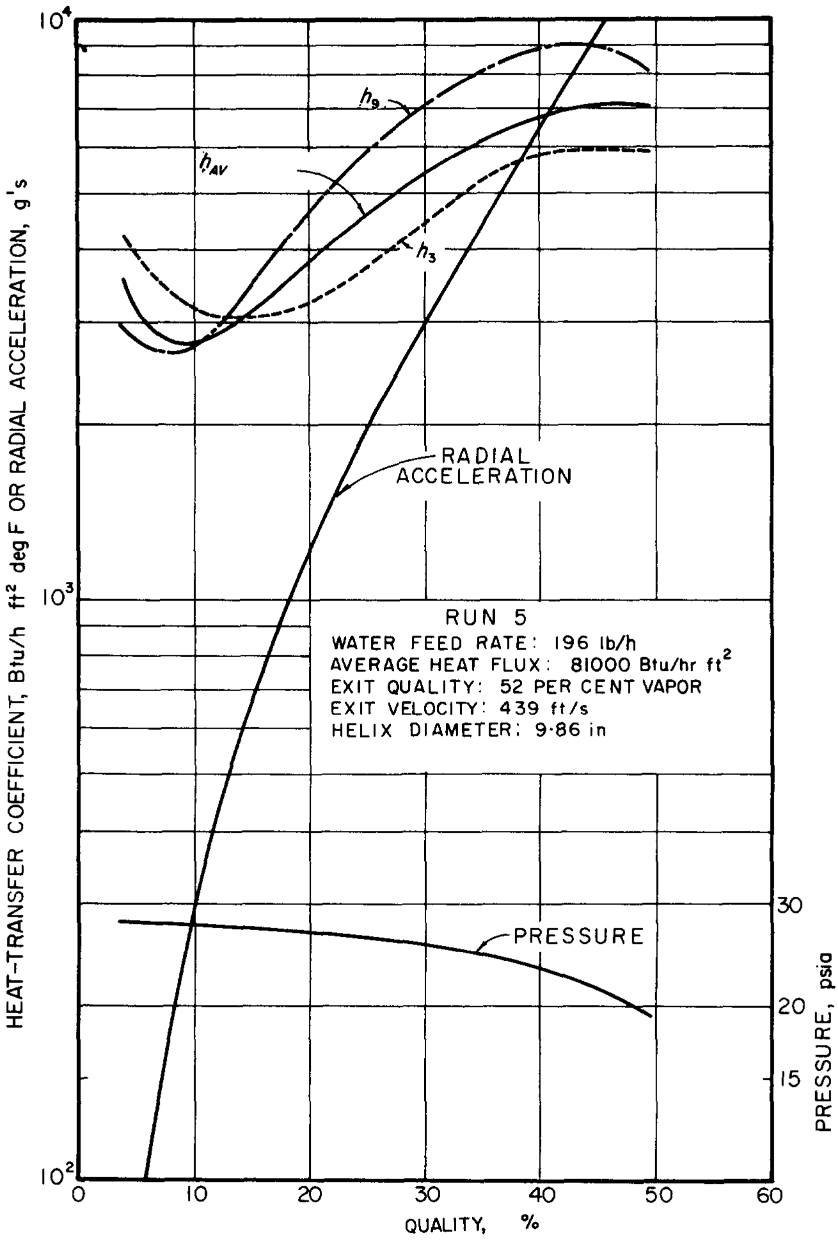

FIG. 2. Typical experimental results with two-phase exit conditions.

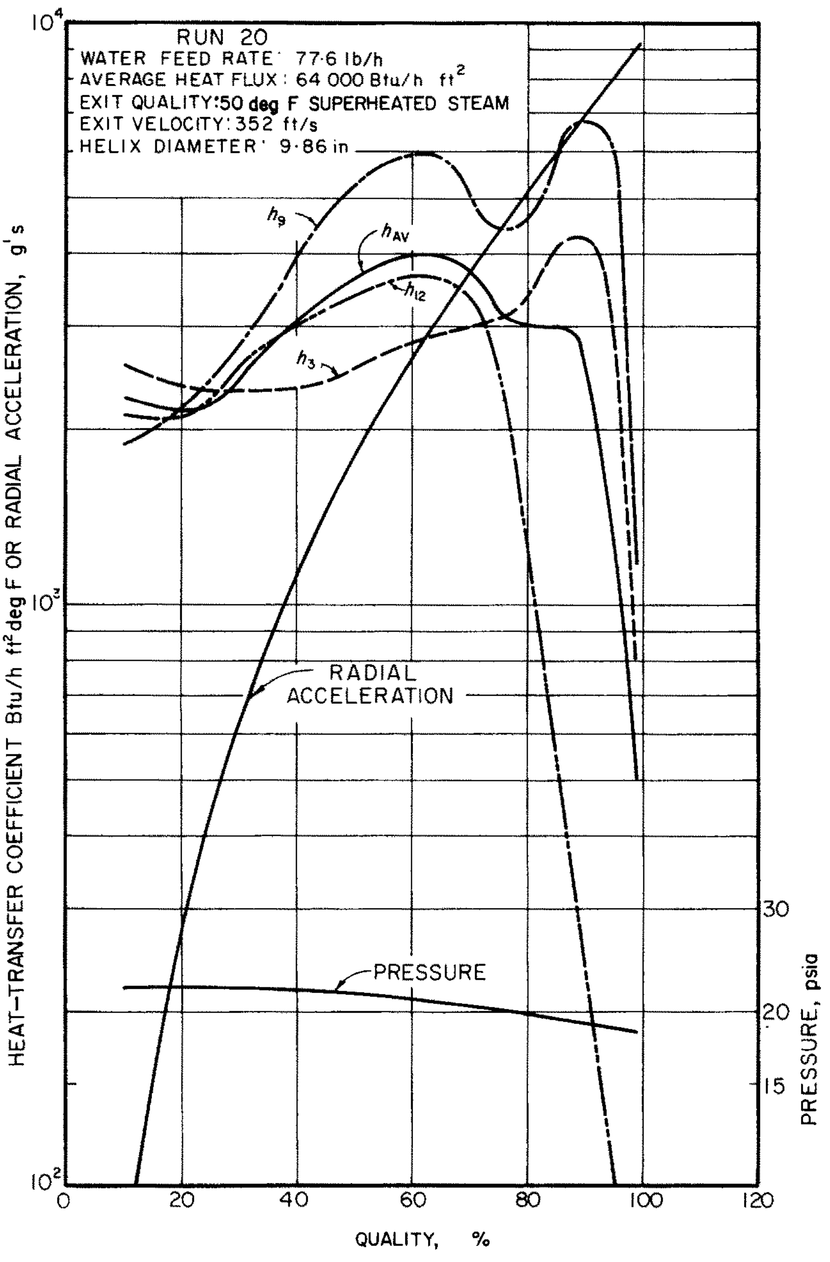

FIG. 3. Typical experimental results with superheated vapor at exit.

because its axial velocity is very small. The vapor, being surrounded by a liquid film, is isothermal at its saturation temperature. At high qualities, when the continuous liquid film finally breaks, the remaining liquid is concentrated at the stagnation points of the secondary flow (3 and 9 o'clock) and the heat-transfer coefficient at other points decreases to the pure vapor coefficient. If this model is valid, we have the somewhat unusual situation of a phenomenon (the radial acceleration, in this case) effective only for that portion of the process where it is helpful and ineffective where it would be damaging.

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Int. J. Heat Mass Transfer. Vol. 10, pp. 401–402. Pergamon Press Ltd. 1967. Printed in Great Britain

ON THE RADIATION SLIP BETWEEN ABSORBING–EMITTING REGIONS WITH HEAT SOURCES

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(*Received* 10 *January* 1966 *and in revised form* 20 *September* 1966)

Some practical systems in which regions with sharply differing properties and energy sources are exchanging radiation are being extensively studied. Examples are "seeded" layers with flow and the various gaseous-core nuclear propulsion systems. Examination of these situations show that a discontinuity in temperature may exist in certain cases at the boundary between the regions. This is analogous in some ways to the often-noted temperature jump at the boundary of a gas abutted by a solid surface [1–3]. Such slips occur only where radiation is considered independent of conduction or where very low densities are present in the gas.

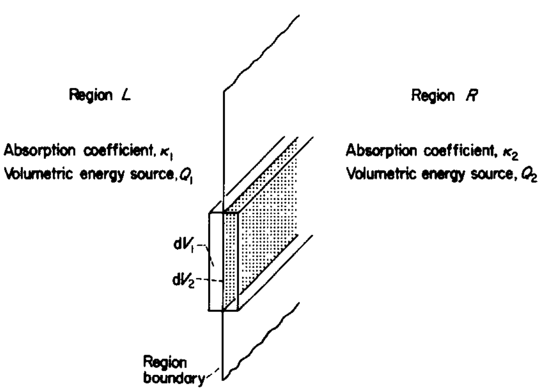

Fig. 1. Semi-infinite regions R and L.

The magnitude of the emissive power slip at a gas–gas interface will now be derived for a simple system.

Consider two semi-infinite regions as shown in Fig. 1. Each region has a given absorption coefficient κ and volumetric heat source strength Q. For simplicity, let Q be a different constant value throughout each region. Then, for an infinitesimal volume element dV_1 wholly within region L, the energy emitted must be equal to the radiation absorbed by the element plus the energy generated within it, or

$$4\int_{\lambda=0}^{\infty}\kappa_{\lambda,1}e_{\lambda,1}\,dV_1\,d\lambda = \int_{\lambda=0}^{\infty}\kappa_{\lambda,1}\Big[\int_{V_L}4\kappa_{\lambda,e}e_{\lambda,e}F_{e-dV_1}\exp\Big(-\int_0^r\kappa_{\lambda,r}dr\Big)dV_e + \int_{V_R}4\kappa_{\lambda,e}e_{\lambda,e}F_{e-dV_1}\exp\Big(-\int_0^r\kappa_{\lambda,r}\,dr\Big)dV_e\Big]\,d\lambda dr + Q_1\,dV_1 \quad (1)$$

where e_λ is the local black spectral emissive power of the medium, F_{e-dV_1} is a shape factor between the emitting element dV_e and the absorbing element dV_1, and r is the radial position in spherical coordinates. The part of the equation within brackets represents the radiation incident on dV_1 in the wavelength interval $d\lambda$. Multiplying by the absorption coefficient $\kappa_{\lambda,1}$ and the mean path length through dV_1, denoted by dr, and then integrating over λ gives the total energy absorbed by dV_1. If $\kappa_{\lambda,1}$ is taken to be independent of wavelength and is a constant within the region, then equation (1) becomes

$$e_1\,dV_1 = A + \frac{Q_1\,dV_1}{4\kappa_1} \quad (2)$$