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FORCED CONVECTION BOILING INSIDE HELICALLY-COILED TUBES

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Abstract—Forced convection boiling heat transfer to water at atmospheric pressure was studied in two helically-coiled tubes. The coils were constructed from 10-ft lengths of 0·625 in O.D. × 0·492 in I.D. Inconel 600 tubing. The helix diameters were 9·86 and 20·5 in, measured from tube axis to tube axis. The heat was generated by direct current resistance heating in the tubing wall. The ranges of conditions investigated were: Water feed rate—77–306 lb/h; Heat flux—19000–81000 Btu/h ft^2; Exit quality—1·4 per cent vapor to 50 deg F superheated steam; non-boiling and subcooled boiling heat-transfer experiments were also performed.

At vapor qualities below 80 per cent, the heat-transfer coefficient was high all around the tube, although it was usually highest on the side furthest from the helix axis. The complete wetting of the tube wall at the higher qualities is attributed to the secondary flow in the vapor core which exerts a drag on the liquid and forces it to flow to the surface closest to the helix axis. The top and bottom portions of the tube surface became dry at about 95 per cent vapor quality, but liquid was apparently present on the tube surface at the points nearest and farthest from the helix axis at over 99 per cent quality.

The circumferential average heat-transfer coefficient and the friction pressure drop were correlated as functions of the Lockhart–Martinelli parameter over the entire range except at low vapor flow rates where a nucleate boiling contribution was present. The results from the two coils did not differ appreciably.

NOMENCLATURE

D, coil diameter, tube center [ft];
d, tube inside diameter [ft];
f, friction factor [dimensionless];
F, Chen's F factor, $(Re/Re_{1c})^{0.8}$ [dimensionless];
G, mass flux [lb h^{-1} ft^{-2}];
g, gravitational acceleration [32·2 ft/s^{-2}];
g_c, conversion factor [32·2 lb lbf^{-1} ft s^{-2}];
h, local heat-transfer coefficient (subscript gives angular orientation in degrees—see Fig. 1) [Btu h^{-1} ft^{-2} °F^{-1}];
$\bar{h}$, circumferential average heat-transfer coefficient [Btu h^{-1} ft^{-2} °F^{-1}];
h_l, heat-transfer coefficient for liquid flowing in a straight tube [Btu h^{-1} ft^{-2} °F^{-1}];
h_{lc}, heat-transfer coefficient for liquid flowing in a coil [Btu h^{-1} ft^{-2} °F^{-1}];
h_{TPF}, heat-transfer coefficient for two-phase flow in a coil [Btu h^{-1} ft^{-2} °F^{-1}];
k_l, thermal conductivity of liquid [Btu h^{-1} ft^{-1} °F^{-1}];
L, length measured along center line of tube [ft];
Nu, Nusselt number for liquid [dimensionless];
Pr, Prandtl number for saturated liquid [dimensionless];

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Q''',	volumetric heat generation rate in coil wall [Btu $h^{-1}ft^{-3}$];
R_1,	volume fraction of tube occupied by liquid;
Re,	Reynolds number based on inside diameter of tube [dimensionless];
T,	temperature [°F];
ΔT,	local temperature difference between wall and liquid [°F];
x,	weight fraction of liquid vaporized;
X_{tt},	Lockhart–Martinelli parameter, equations (3, 4);
ΔP_a,	acceleration pressure drop [lbf ft^{-2}];
$(dP/dL)_g$,	pressure gradient due to friction of vapor phase alone flowing in tube [lbf ft^{-3}];
$(dP/dL)_l$,	pressure gradient due to friction of liquid phase alone flowing in tube [lbf ft^{-3}];
$(dP/dL)_{TPF}$,	pressure gradient due to friction for two-phase flow in tube [lbf ft^{-3}];
μ_g,	vapor viscosity [lb$ft^{-1}h^{-1}$];
μ_l,	liquid viscosity [lb$ft^{-1}h^{-1}$];
ρ_g,	vapor density [lbft^{-3}];
ρ_l,	liquid density [lbft^{-3}];
ϕ_{gtt},	Lockhart–Martinelli parameter, equation (5).

1. INTRODUCTION

BECAUSE of its practical importance, boiling inside conduits under forced circulation has been studied by many investigators. However, few investigations have dealt with boiling in a coil. The aim of the present work was to gain a general understanding of forced convection boiling in helically coiled tubes. The practical incentive for studying this configuration was the possibility of achieving continuous de-entrainment of liquid droplets from the vapor because of the large radial accelerations induced by the helical path. Suppression of the fog flow regime by this process would hopefully keep a stable liquid film on the wall to higher vapor qualities than in a straight conduit and hence delay the onset of departure from nucleate boiling (DNB)*.

Hendricks and Simon [1] investigated heat transfer to subcritical (two-phase), supercritical, and gaseous hydrogen in electrically-heated curved tubes at pressure of 100–600 lb/in^2, heat fluxes of $5{\cdot}2 \times 10^5$–$1{\cdot}8 \times 10^6$ Btu/h ft^2, and radii of curvature of 2–7·5 in. The qualitative results which they obtained were as follows: the heat-transfer coefficient at the 270° position (see Fig. 1) can be as much as three times that at the 90° position for the flow of near-critical hydrogen and two times that for gaseous hydrogen. Their near-critical and two-phase data indicated that the coefficient at the 90°

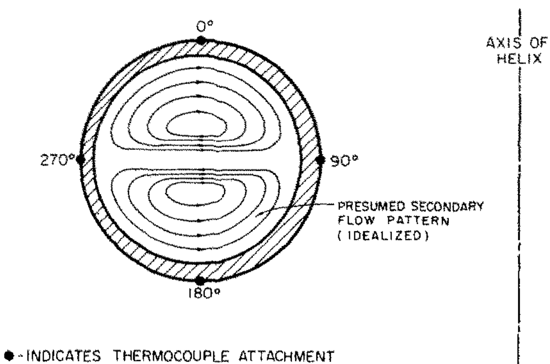

FIG. 1. Diagram of tube identifying thermocouple positions.

* Throughout this paper, DNB will be used as a general term to denote situations under which the local heat-transfer coefficient is sufficiently low that the authors infer that a stable, wetting liquid mass does not exist on the wall at that point. DNB is often a misnomer, since in forced convection, heat transfer may occur through a thin liquid film from a wall above the saturation temperature without bubble formation. However, such related terms as burnout, peak heat flux, critical heat flux, boiling crisis, film breakup, and dry wall boiling have equally inconsistent etymological and technological usage in the literature. In the present study and most of those referenced, visual observation of the heat-transfer surface was not possible. Hence it seems preferable to adopt a single term defined by reference to a quantity obtained by computation from the experimental observations.

position was somewhat lower than that of a straight tube under similar conditions.

Carver *et al.* [2], conducted boiling experiments with water at 26000 lb/in^2 in electrically-heated helical coils of 16 and 65 in. in radius with tube I.D. of 0·42 in. Their main purpose was to obtain data on DNB in coiled tubes and its comparison with that of a straight vertical tube. Their results indicated that (1) the DNB steam quality was different for different positions around the tube periphery while being constant in a straight vertical tube; (2) a coiled tube has a higher average steam quality at DNB than a straight tube; (3) DNB is more gradual and the fluctuations in temperature are much lower in a coil than in a straight tube; (4) steam quality at DNB for the small coil is higher than the large coil; and (5) increasing the mass velocity increases the steam quality at DNB.

Yudovich [3] experimented with boiling water and n-hexane in a helical coil of 0·5-in O.D. copper tubing with a coil radius of 3·5 in. Heating was effected by immersing the coil in a steam bath. For n-hexane runs and for water runs in the flow range of 5·1–17·5 lb/h, the temperature of the superheated vapor leaving the coil increased with flow rate. At higher water flow rates, giving only partial vaporization of the feed, the average heat flux for the entire coil increased from 4660 Btu/h ft^2 at 34·5 lb/h to 6400 Btu/h ft^2 at 63·0 lb/h and then decreased to 4360 Btu/h ft^2 at 113·5 lb/h.

No attempt was made by the above authors to correlate the local heat-transfer coefficient as a function of their system parameters.

Miropolskiy *et al.* [4] reported wall temperature distributions and heat fluxes at DNB for single and two-phase flow of water and steam in 90° and 360° pipe bends at pressures from 20–295 bar. The key finding is that the heat flux at DNB in bends is less than in straight pipes at low qualities, but the reverse is true at high qualities. The discussion of [4] by Lacey [5] is especially interesting and is consistent with the findings reported in the present paper.

Two of the present authors published a shorter communication [6] on the present work in order to bring early attention to some of the interesting features of boiling in helical coils. In addition to expanding upon the earlier note, the present paper includes new findings in support of some of the speculations made there.

Two-phase, air–water studies were made by Rippel *et al.* [7], who investigated isothermal pressure drop, hold-up and axial mixing in a helical coil of 4 in. in radius made of 0·5-in tubing. Although their pressure drop data could be represented by the Lockhart–Martinelli correlation [8], it was evident that the liquid rate was a parameter. The liquid hold-up fell below the Lockhart–Martinelli correlation. Axial mixing diminished with increasing ratios of gas to liquid rate.

A visual flow study using air–water mixtures in a coiled transparent plastic tube was made by Banerjee *et al.* [9]. For most of their runs, they observed that the liquid film was displaced toward the 90° position whereas one intuitively expects the liquid to be thrown to the outer wall. They term this phenomenon "film inversion" and explain it by a force balance at the gas–liquid interface, assuming a large slip velocity between the phases and neglecting secondary flow effects. A few runs at the high velocity end of their experimental range showed liquid entrainment (as droplets) and deposition effects; these runs did not correlate well with their analysis. We infer (detailed data were not given) that their velocities were generally much lower than ours and on this basis their results and ours are quite consistent.

For recent studies of pressure drop and heat transfer to single-phase flow of fluids in coils, the reader is referred to the works of Ito [10], Seban and McLaughlin [11], Rogers and Mayhew [12], Mori and Nakayama [13, 14]; and Kubair and Kuloor [15].

Koutsky and Adler [16] studied axial mixing in helical coils and noticed progressively stronger peaking of the residence time distribution curve with increasing Reynolds number, indicating increasingly stronger secondary flow effects.

2. EXPERIMENTAL PROGRAM

A. *The heat-transfer studies*

A schematic diagram of the heat-transfer facility is given as Fig. 2.

Two coils with radii of 4·93 and 10·28 in, measured to the tube axis, were studied: each coil was made from a 10-ft length of 0·625-in O.D. cold-drawn seamless tubing with a nominal wall thickness of 0·068 in. Inconel 600 was selected for the coil material because of its high electrical resistivity and high melting range of 2500–2600°F. The coil was mounted with its axis in the vertical direction. Heat was generated in the coil by passing a regulated DC current through it from an AC–DC motor-generator.

The electrodes were made of copper bars and were brazed to the coil. The coil was electrically insulated from the rest of the piping by inserting a short piece of silicone rubber tubing at each end of the coil.

The coils were cold-formed and then stress-relieved in a molten salt bath for three hours at 1400°F. The tube was slightly flattened during bending as evidenced by the measurements of its major and minor diameters. However, no crimping was visible. The dimensions of the coils are listed in Table 1.

Table 1. Coil dimensions

	Small coil	*Large coil*
Coil diameter (in. center-to-center)	9·86	20·57
Straight tube outside diameter (in.)	0·629	0·629
Straight tube inside diameter (in.)	0·492	0·492
Coiled tube major outside diameter (in.)	0·637	0·631
Coiled tube minor outside diameter (in.)	0·619	0·625
Distance between turns (in. center-to-center)	3·9	4·6
Ratio of coil diameter to straight tube inside diameter	20·0	41·8
Heated length of coil (ft axial)	9·35	9·35
Electric resistance of coil (Ω)	0·0382	0·0388

After the thermocouples and the electrodes were connected to the small coil, a box was formed around the coil and filled with Vermiculite. The large coil was insulated with glass wool and then covered with aluminum foil.

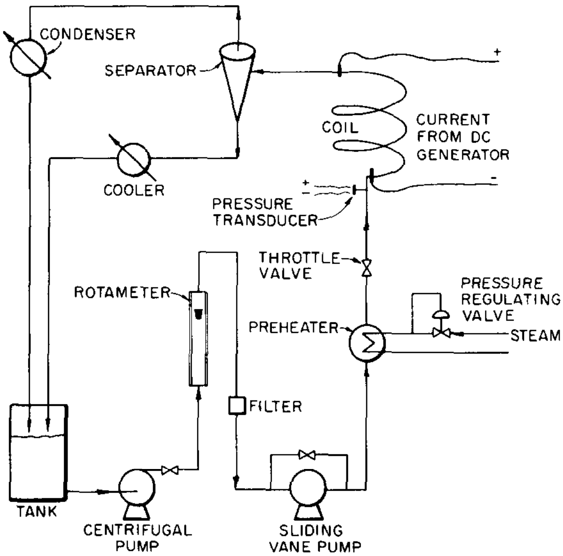

FIG. 2. Schematic diagram of the system.

Distilled water was drawn from a 5 l. glass container by a centrifugal pump and passed through a rotameter and a fine wire-mesh filter. A sliding-vane pump forced the water into the coil after being preheated to the desired temperature. The vapor–liquid mixture leaving the coil was separated in a glass cyclone separator. The vapor stream, after being condensed, and the liquid stream were returned to the feed tank through separate lines.

The outside wall temperature of the coil was measured by 20-gauge iron–constantan thermocouples, cemented to the wall with Sauereisen cement. The thermocouples were electrically insulated from the coil by a thin layer of cement between the thermocouples and the coil. The thermocouples were constructed from rather heavy wire to withstand repeated handling during calibration and experimentation; this precaution subsequently proved unnecessary. To minimize thermal conduction effects, the lead wires were laid along the coil for a distance of about 1·5 in before being brought out through the insulation. The effectiveness of this procedure (and of the coil insulation) was checked by calibrating the thermocouples in situ with saturated steam at 210°F flowing through the coils; all thermocouples for the small coil read between 208·2 and 209·8°F with an average of 209·3°F and all thermocouples for the large coil read between 207·3 and 209·4°F with an average of 208·4°F. The differences between coils were attributed to the difference in insulation procedure; the differences also incorporate any temperature drop across the Sauereisen cement bond. Using these individual calibrations at 210°F, a conduction loss correction was applied to each thermocouple assuming that the correction was proportional to the difference between the coil temperature and ambient. Heat loss from the coil through the insulation, electrodes, etc., was found to be less than 1 per cent of the heat transferred to the fluid.

There were nine longitudinal thermocouple stations at intervals of 1 ft; the first station was 1·10 ft downstream from the electrode. At each station, four thermocouples were installed around the circumference, as shown in Fig. 1. The water temperature at the coil inlet and the vapor temperature after the cyclone separator were also measured by thermocouples. The thermocouples were calibrated in place. A steam bath was used for the reference junction in all the temperature measurements in order to minimize errors arising from the variation of the atmospheric pressure. Considering that the primary interest was in the difference between the wall and the saturation temperature, this procedure was felt satisfactory, although a more conventional method would be to use ice for the reference junction and measure the atmospheric pressure in each run.

The inlet pressure of the system was measured with a mercury manometer and a strain gauge pressure transducer. The pressure transducer output and in some cases the temperature outputs were recorded on an optical oscillograph. After all the temperature measurements were made on the small coil, the thermocouples were removed and pressure taps were drilled at 1, 3, 5, 7 and 9 ft from the inlet. Many of the runs were repeated and the pressures at these points were measured.

All instruments were read and recorded at least twice to assure steady-state conditions. Some runs were repeated after several weeks to see if the results changed due to surface aging; no change was noticed. The water was occasionally analyzed for its oxygen content. Since usually 1·5 h of boiling preceded any temperature measurement and since the total system hold-up was only 6 l., the oxygen content was always under 2 ppm. The system was drained and refilled with distilled water several times prior to the first run and a few times thereafter to minimize the solids content of the water.

Two corrections are necessary in reducing the data to give local heat fluxes and heat-transfer coefficients. First, because the tube is coiled, the wall is thicker and the electric path shorter at the 90° position than at the 270° position, resulting in a substantially larger heat flux generation at the 90° position. Second, peripheral temperature gradients in the tube cause conduction of heat through the wall, reducing the local heat flux at high temperature points and enhancing it in cooler regions. For convenience in data reduction, an approximate analytical treatment of the second effect was made by integrating the Fourier equation with internal generation in cylindrical coordinates subject to the following assumptions:

(a) The curvature of the coil was neglected.
(b) Heat generation and wall thickness were assumed constant during integration, but the correct local values were used in the numerical computation of the inside wall temperatures and the radial heat flux.
(c) Longitudinal conduction was neglected.
(d) Constant thermal conductivity and electrical resistivity were assumed.
(e) The outside of the tube was assumed to be perfectly insulated.
(f) The known temperature profile of the

outside of the tube was represented as a sinusoidal function of position.

The details of the analyses and checks on the validity of the assumptions are given in [17]. The analysis for the second effect is only approximate and leads to some uncertainty (perhaps as much as ± 25 per cent) in the local coefficients at high qualities mainly because of the breakdown of assumption (f). The circumferential average heat-transfer coefficient is much less sensitive.

B. *The visual flow studies*

From the original heat-transfer results, the authors were drawn to some speculations concerning the flow patterns in the coiled tube [6]. To test these speculations, a transparent coiled tube was constructed by wrapping 0·625 in O.D. $\times$ 0·5 in I.D. Tygon tubing around a cylindrical mandrel, giving a coil radius to tube center of 7·125 in. Colored water was pumped through the coil and laboratory air was introduced upstream from the coil.

Tygon is poorly wetted by water, so that small quantities of liquid tend to gather in individual drops rather than as a thin stable film as in Inconel. Within this limitation however, the visual observations confirmed the speculations in [6] and the observations reported by Lacey [5] and diagramed in Figs. 1 and 2 of his discussion. Briefly, it was found that, at conditions corresponding to the experiments reported here, the major portion of the liquid was concentrated in a relatively slow-moving stream on the inside tube surface at the 90° position. The surface of this stream was strongly rippled and liquid was entrained into the air. This liquid was deposited at the 270° position as fine drops, and these drops spiraled back to the 90° position in paths that appeared to be symmetrical about a plane through the 90–270° points. These spiral paths are, in our opinion, the resultant of a longitudinal drag force on the drops by the primary gas flow and a peripheral drag force due to the twin vortices of the secondary flow.

3. RESULTS

Only a few experimental results can be presented and discussed in this paper. Complete tables of experimental and calculated data are given in [17] for thirty-four heat-transfer runs.

One run with each coil was made with water in laminar flow. The circumferential average heat-transfer coefficients from these runs were about three times the asymptotic values predicted from the Seban and McLaughlin [11] laminar flow equation:

$$\frac{hd}{k_l}(Pr)^{-\frac{1}{3}} = 0{\cdot}13\left[\frac{f}{8}(Re)^2\right]^{\frac{1}{3}} \tag{1}$$

which is based on experimental values obtained from laminar flow of liquids having a Prandtl number of 100–657. In the above equation f is the friction factor calculated from White's formula [18]. The large discrepancy is perhaps due to natural convection and the much lower Prandtl number of water, in which range no correlation is available.

Two runs with the small coil and one with the large coil were performed with water flowing in the turbulent regime. The ratio of the heat-transfer coefficient of the concave side to that of the convex side was approximately 2·8 for the small coil and 2·2 for the large coil. The circumferential average heat-transfer coefficients for the two coils agree within 5 per cent with Seban and McLaughlin's turbulent flow equation [11]:

$$Nu = 0{\cdot}023\, Re^{0\cdot 8}\, Pr^{0\cdot 4}\left[Re^{1/20}\left(\frac{d}{D}\right)^{1/10}\right] \tag{2}$$

where the liquid properties were evaluated at a film temperature defined as the average of the bulk temperature and the circumferential average temperature of the wall.

The study of heat transfer in two-phase flow constitutes the main part of this investigation. The entire range of steam quality was covered in these experiments.

For illustrative purposes, the calculated results for run 5 and run 20 are given in Figs. 3 and 4

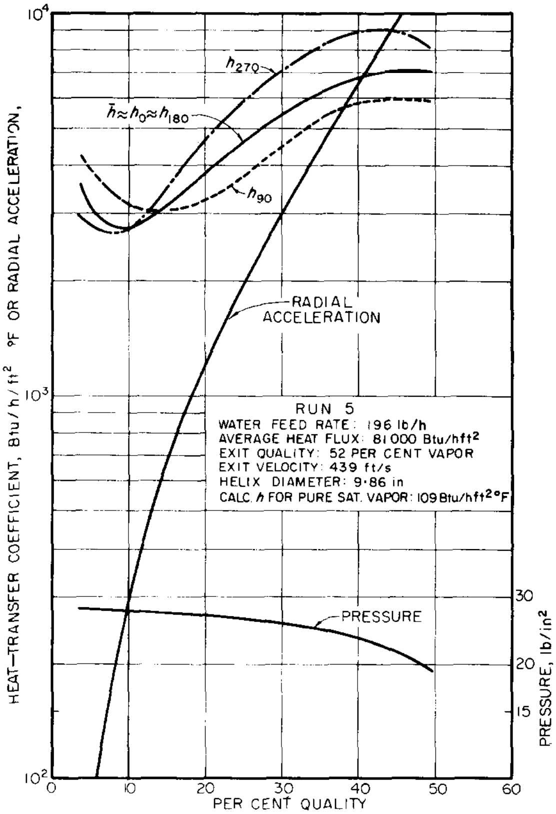

FIG. 3. Typical experimental results with two-phase exit conditions.

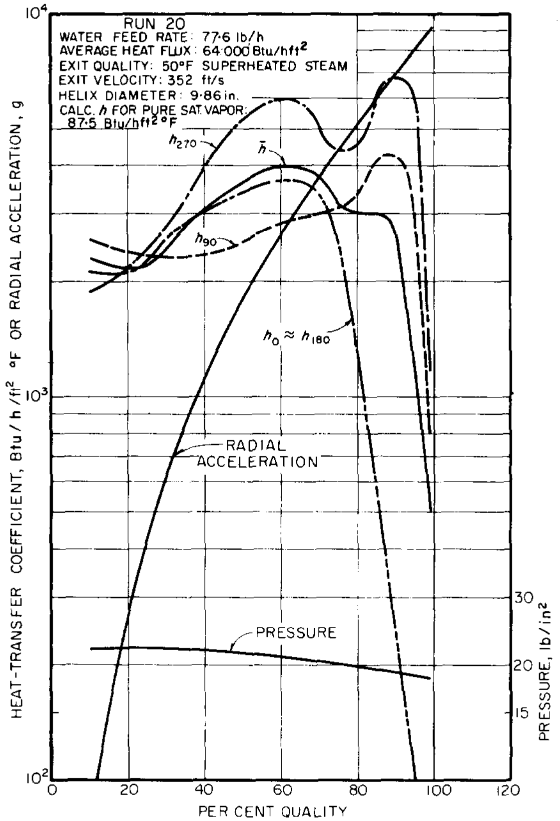

FIG. 4. Typical experimental results with superheated vapor at exit.

respectively. Run 5 was typical of runs with a nearly saturated feed and a two-phase effluent. The abscissa, local quality, is computed assuming thermodynamic equilibrium between the phases; this assumption is supported by measurements of the exit stream temperatures. For run 5, the coefficients at the top and bottom of the tube, h_0 and h_{180}, are nearly identical with $\bar{h}$. For run 20, h_0 and h_{180} are nearly identical.

Run 20 produced 50°F superheated steam at the exit of the heating section, though the last point at which local coefficients could be computed was 3 in upstream at 99·1 per cent quality. The 50°F superheat was confirmed by a thermocouple placed in the exit stream, and there was no evidence of any liquid drops being carried over.

Radial acceleration is computed using the local mean vapor velocity assuming the vapor fills the entire cross-section of the tube. Since the area fraction occupied by liquid is small after a few weight per cent vapor is generated, the calculated vapor velocity should be very close to the actual.

Near the coil inlet, in general, the heat-transfer coefficient was highest for the 90° position and lowest for the 270° position. This is due to the varying extent of nucleate boiling around the circumference. Because of the deformation of the tube in bending, the wall thickness and heat generation Q''' are highest at the 90° position. Therefore, this point has the highest radial heat flux and inside wall temperature. The higher wall temperature gives rise to a large nucleate boiling component at this point compared to the other three. Since the nucleate boiling heat flux is proportional to ΔT^a where a is usually larger than two, the heat-transfer coefficient for this point is higher than the others, even though its ΔT is the highest of the four.

Further from the inlet, the heat-transfer coefficient often went through a minimum. This was particularly true of the 90° position. This behaviour is thought to be a result of the suppressing effect of turbulence on nucleation, causing a drop in the nucleate boiling component while not enhancing the convective component to a comparable extent. The total heat transfer is assumed to consist of a nucleate boiling component superposed (not necessarily in a linearly additive way) on the forced convective heat transfer.

As vaporization proceeds, the vapor volume increases and the remaining liquid is concentrated as a thin film on the wall; nucleate boiling is suppressed and the heat-transfer mechanism becomes predominininantly convective. Wattendorf [19] has shown that in single-phase flow of a fluid in a curved channel of rectangular cross-section, the shear stress on the concave wall (as seen by the fluid) was higher than the convex wall. By analogy to momentum transfer, the resistance to heat transfer is expected to be less at the concave wall than at the convex wall. In the authors' experiments, the heat-transfer coefficient for the 270° position was the highest of all at moderate and high vapor velocities.

The heat-transfer coefficient for all four points increases as more vaporization takes place until in the very high quality range (90–95 per cent) the coefficients for the 90° and 270° positions are still quite high while for the other two points they diminish and approach the values of a gas coefficient (see Fig. 3). This is attributed to a liquid deficiency at the 0° and 180° positions. As the vapor quality approaches 100 per cent, liquid deficiency also develops at the 90° and 270° positions and these coefficients drop rapidly also. The predicted dry vapor coefficients for the exit conditions are 109 Btu/h ft^2°F for run 5 and 87·5 for run 20, based on equation (2).

4. PHYSICAL INTERPRETATION OF OBSERVED PHENOMENA

In the present study, the experimental values of the heat-transfer coefficient indicate that a liquid film exists all around the tube periphery, up to high qualities. The stability of this film is attributed to the secondary flow. The secondary flow is caused by the action of the centrifugal force when a fluid flows in a curved path. The fluid from the center of the tube has the highest velocity and is most strongly acted on by the centrifugal force. This force drives the fluid from the center to the outer wall of the tube. This in turn induces a motion from the outer wall around the tube and back to the center (see Fig. 1). The net result is a pair of symmetrical recirculation patterns superposed on the main flow. The details of turbulent secondary flow are very poorly understood.

Regarding heat transfer in two-phase flow in a helix, the following mechanism is postulated: a secondary flow exists in the vapor core of the two phase flow. This secondary flow exerts a drag on the liquid film and causes it to flow from the outer to the inner wall of the tube, replenishing the losses due to evaporation and entrainment. Liquid on the wall is subject to small radial acceleration effects because its axial velocity is small. The vapor, being surrounded by a liquid film, is 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 (the 90° and 270° positions) and the heat-transfer coefficient at the other points decreases to the pure vapor coefficient.

5. CORRELATION OF TWO-PHASE HEAT-TRANSFER DATA

A theoretical analysis of heat transfer in two-phase flow is difficult due to the complex flow patterns, slip between vapor and liquid, the unknown conditions at the vapor–liquid interface, and the phenomena associated with the formation and motion of bubbles. Hence, a satisfactory theoretical treatment of the subject has not yet been offered for any geometry. Several empirical correlations have been proposed for straight tubes. No previous correlation for heat transfer to a two-phase flow in coils is available.

Rohsenow [20] and others have suggested that forced convection heat transfer in subcooled or saturated boiling may be considered to be composed of a nucleate boiling component and a convective component, each of which may be correlated separately. Although this concept has been successfully applied to some problems, in general it is not clear how these two components should be combined, since the two mechanisms affect each other. Agitation suppresses bubble nucleation and growth, while at high nucleate boiling fluxes the effect of forced convection is substantially diminished.

Heat transfer to a two-phase flow in straight tubes has been correlated as a function of the Lockhart–Martinelli parameter by nearly all the authors in this field, some of them adding a nucleate boiling term. The reader is referred to Tong [21] for a summary of the existing correlations.

The Lockhart–Martinelli parameter X evolved from an anlysis and empirical correlation of isothermal two-phase, two-component pressure drop data in straight horizontal tubes [8]. It is defined as:

$$X = \frac{(\mathrm{d}P/\mathrm{d}L)_1}{(\mathrm{d}P/\mathrm{d}L)_g}. \tag{3}$$

If both phases are turbulent, this parameter becomes

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0\cdot 9} \left(\frac{\rho_g}{\rho_1}\right)^{0\cdot 5} \left(\frac{\mu_1}{\mu_g}\right)^{0\cdot 1}. \tag{4}$$

The extension of the Lockhart–Martinelli-type correlation to two-phase heat transfer is based on the analogy between heat and momentum transfer. In two-phase pressure drop correlation, the modulus

$$\phi_{gtt} = \frac{(\mathrm{d}P/\mathrm{d}L)_{TPF}}{(\mathrm{d}P/\mathrm{d}L)_g} \tag{5}$$

is correlated against X; for heat transfer, h_{TPF}/h_1, or a similar ratio, is correlated against X_{tt}.

For the purpose of correlation, only the circumferential average heat-transfer coefficient was considered. Of the four X's associated with the four flow regime combinations, X_{tt} was felt to be suitable even when the liquid Reynolds number was below the critical value. For two-phase flow in a straight tube, the critical Reynolds number was arbitrarily set at 2000 by Lockhart and Martinelli. Since the liquid is not flowing by itself, this number loses the significance that it has in single-phase flow, as the authors readily admit. Furthermore, the liquid is not flowing by itself, this number loses the significance that it has in single-phase flow, as the authors readily admit. Furthermore, in a coil the transition for laminar to turbulent flow takes place smoothly and the difference between the two regimes is less pronounced than in a straight tube. Lastly, the Lockhart–Martinelli curve for the case of viscous liquid–turbulent gas regime does not differ radically from the curve for the case when both phases are turbulent. It should be pointed out that Re for each phase is based on the total cross-sectional area of the tube.

Figures 5 and 6 are the correlations of h_{TPF}/h_{lc} as a function of X_{tt} for the two coils. h_{lc} is the value of the heat-transfer coefficient if the liquid phase alone were flowing in the coil; it was calculated from equation (2).

The physical properties of the liquid evaluated at the saturation temperature were used to calculate h_{lc}. The double enclosed points in each figure indicate data taken when the vapor Reynolds number is below the critical value for the laminar–turbulent transition in the coil.

The solid curves in Figs 5 and 6 are the authors' best estimates of the forced convection correlation curves for the geometries tested. The dashed curve in each figure is Chen's F factor [22] for the forced convection contribution to heat transfer in straight conduits.

In the low quality range, the high values of

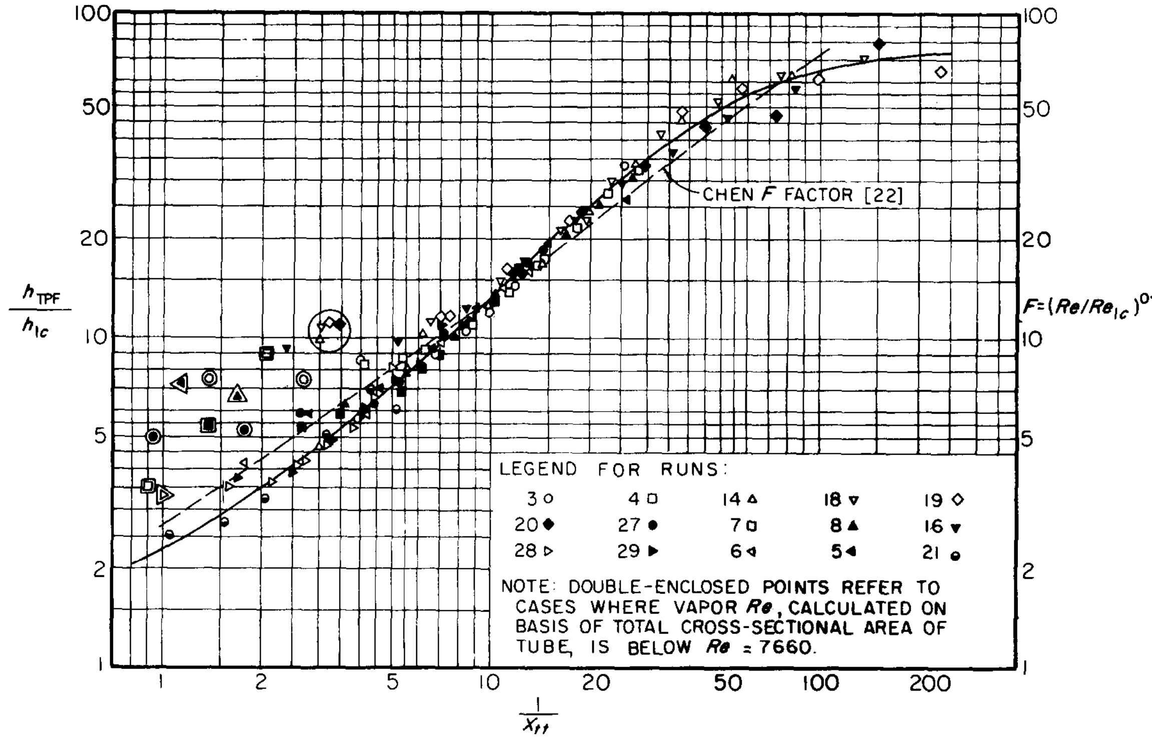

FIG. 5. Correlation of data of heat transfer in two-phase flow, small coil.

h_{TPF}/h_{lc} indicate the presence of nucleate boiling. An improved correlation may be obtained by including a nucleate boiling component which would be progressively suppressed as the liquid Re or the vapor quality is increased, as successfully applied by Chen [22] in the correlation of two-phase heat transfer in straight tubes. It has been noted [25] that at low values of $1/X_{tt}$, the points for each run approach an asymptotic value of h_{TPF}/h_{lc} characteristic of heat flux and inlet liquid velocity for that run. This information could be used to construct a nucleate boiling contribution correlation for the coils, but in view of the success of Chen's correlation noted later there appears to be no great value in doing so.

In the medium quality range, the correlation is good. However, the scatter in the data increases as the quality increases. This is mainly due to the inherent difficulty of obtaining sufficiently accurate data in the very high quality range. For instance, 1 per cent error in the input power or liquid flow rate causes 18 per cent error in the value of X_{tt} when the quality is 95 per cent and corresponding higher errors at higher qualities. Also, in this quality range the temperature variation around the tube is relatively large and four temperature readings are not sufficient to describe accurately the temperature profile around the tube.

The data for all flow rates fall along the same line, and no distinction can be made between a laminar and a turbulent liquid phase. Even though the calculated values of h_{lc} are based on turbulent flow of a liquid in a coil, it was found that equation (2) was still a satisfactory non-dimensionalizing parameter for all the liquid Reynolds number encountered, some of which were close to 1000.

In equation (2), the effect of coil diameter

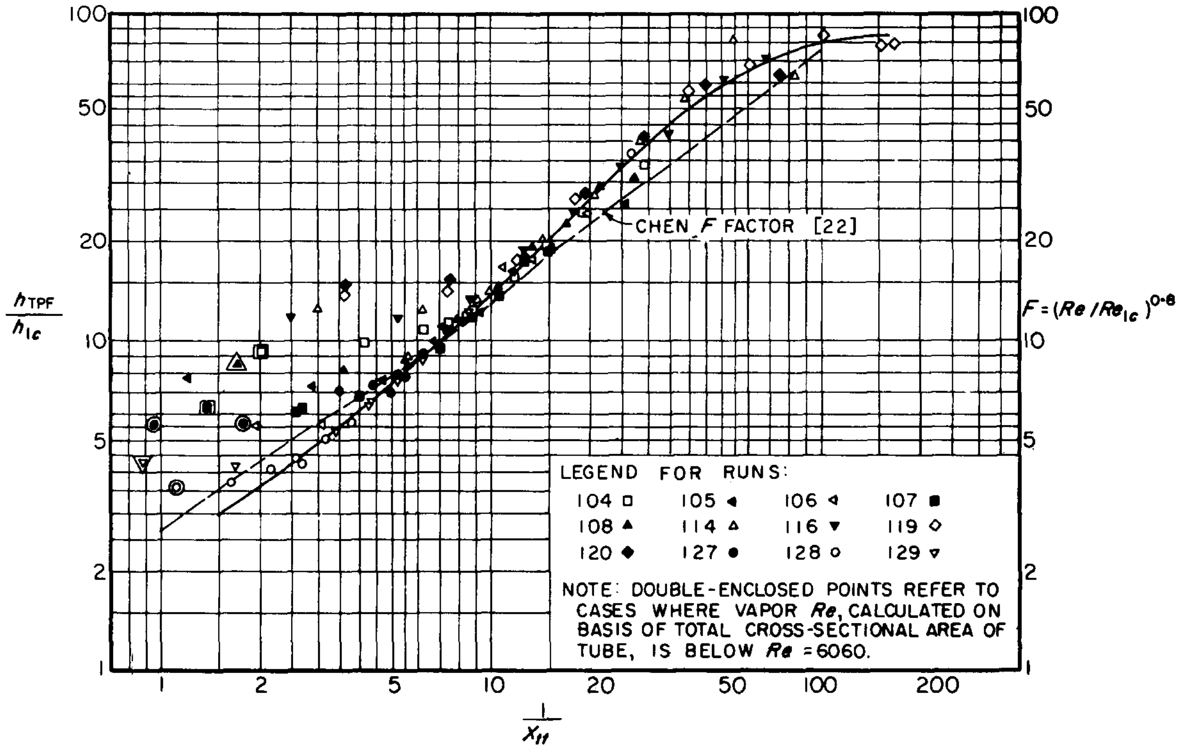

FIG. 6. Correlation of data of heat transfer in two-phase flow, large coil.

appears in the form of $(d/D)^{0.1}$, the value of which is only 7·6 per cent greater for the small coil compared to the large coil. In view of the fact that the results of the two coils give essentially identical correlations, it may be stated that, in the range of coil diameters investigated, the effect of the ratio of tube to coil diameter is not very strong and perhaps not basically different from its effect in single-phase turbulent flow.

Comparing the results found here with the heat-transfer correlations of two-phase flow in straight tubes, it is seen that Chen's F curve is within 15 per cent of the estimated best forced convection curve through the data. Such agreement is about as good as the data in each case and usually good for a two-phase heat-transfer correlation. When Chen's additive nucleate boiling correlation is included, the predicted coefficients are about 15 per cent high at low qualities and up to 15 per cent low at high qualities, assuming that the tube surface is completely wetted. At very high qualities where the tube is not completely wetted, Chen's correlation, as would be expected, predicts coefficients as much as several times too high.

Dengler and Addoms' correlation [23] predicts values which are, on the average, only 10 per cent less than the authors' experimental values in the low and medium quality ranges. However, at qualities above 70 per cent it predicts values which are 30–50 per cent too high until at 90 per cent quality its predicted values are several times the authors' measured values. It should be noted that the highest vapor quality in the data used in Dengler and Addoms' correlation, as well as Chen's, was 71 per cent.

A significant feature of the helix is that it delays the transition from a wetted wall to a dry wall condition compared to a straight conduit. In the experiments of Dengler and Addoms [23] on boiling water in a 1-in I.D. steam heated straight vertical tube, this transition occurred at about 80 per cent vapor quality when the flow rate was 240 lb/h. In the experiments of Woods [24] on boiling water in a standard 1 in, steam-heated, horizontal copper pipe the transition occurred at vapor qualities of 65–85 per cent.

In both coils, the transition appears to occur at 85–90 per cent vapor quality at the 0 and 180° positions, while occurring at 98–100 per cent at the 90 and 270° positions. It should be emphasized that these values apply only to the conditions of these two tests. Nevertheless, they reveal this useful feature of the helix.

6. CORRELATION OF TWO-PHASE PRESSURE DROP DATA FOR THE SMALL COIL

Pressure profiles along the tube were obtained only for the small diameter coil. The measured values of the pressure drop were the sum of the friction pressure drop, the acceleration pressure drop, and the hydrostatic pressure effect. Based on the data of Rippel *et al.* [7], it was concluded that the hydrostatic effect was usually less than 1 per cent of the total pressure drop.

In order to calculate the acceleration pressure drop, the liquid was assumed to flow in an annular film. The volume fraction of the tube filled with liquid, R_1, was estimated from the data in [7]. Then the acceleration pressure drop was calculated from the momentum balance:

$$\Delta P_a = \frac{G^2}{g_c \rho_l}\left[\frac{(1-x)^2}{R_1} + \frac{x^2}{1-R_1}\frac{\rho_l}{\rho_g} - 1\right]. \quad (6)$$

The remaining pressure effect was due to friction; the friction pressure drop was plotted against coil length and the slope of the curve $(dP/dL)_{TPF}$, determined graphically. Then $(dP/dL)_g$ for the coil was determined from the work of Ito [10] and ϕ_{gtt} calculated from equation (5). The resulting correlation is shown in Fig. 7. While there is considerable scatter and the range of variables is not great, it does appear that the Lockhart–Martinelli correlation can be used for estimating pressure drop in two-phase flow in coils.

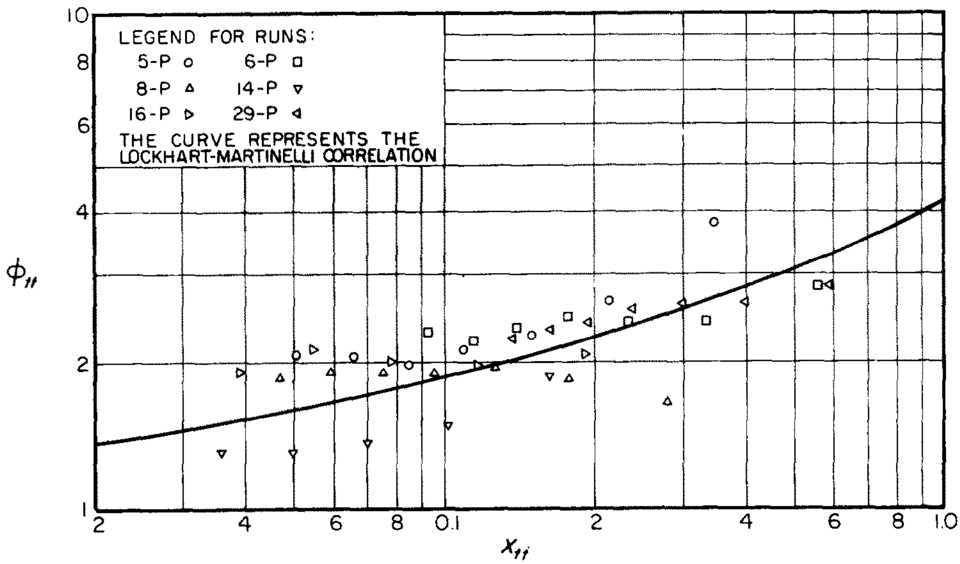

FIG. 7. Pressure drop correlation for small coil.

7. CONCLUSIONS

The heat-transfer coefficient in boiling two-phase flow in helically-coiled tubes is generally quite high on all sides of the tube, although it is usually highest on the outer side.

Over most of the quality region, the prevailing heat-transfer mode is convection, but a nucleate boiling component is present at low qualities.

At about 80 per cent quality, the heat-transfer coefficient at the top and bottom of the tube decreases gradually until at about 90 per cent quality it approaches the value for a gas. The 90 and 270° positions remain wetted until almost all the liquid is vaporized. It is postulated that the secondary flow is responsible for distributing the liquid around the tube.

Chen's correlation [22] predicts the local average heat-transfer coefficient within about ± 15 per cent over the range tested, as long as the wall is completely wetted.

Generally, the experiments reported here were under the condition of high vapor velocity and radial acceleration. The extension of the present results to high pressures where the vapor velocity and radial acceleration are substantially lower is not recommended and remains an area of future research. Also, the important phenomenon of departure from nucleate boiling was not studied.

The helical coil should be considered in applications where the total liquid stream is to be vaporized. The coil is a suitable device for boiling in the absence of gravity. It also has attractive features as a chemical reactor where large volumes of vapor react with a liquid accompanied by a high heat of reaction, such as chlorination reactions. It provides for efficient contact and excellent heat transfer. The secondary flow destroys the radial concentration gradient, giving behavior similar to a plug flow reactor.

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Résumé—Le transport de chaleur dans l'eau à pression atmosphérique a été étudié au cours de l'ébullition par convection forcée dans deux tubes enroulés en hélices. Les serpentins ont été construits à partir de tubes en Inconel 600 de 3,05 m de long avec un diamètre intérieur de 12,5 mm et un diamètre extérieur de 15,9 mm. Les diamètres des hélices étaient égaux à 251 mm et à 520 mm mesurés entre les axes du tube. La chaleur était produite par le chauffage à l'aide d'un courant continu d'une résistance placée dans la paroi du tube. Les gammes des conditions étudiées étaient:

Débit massique de l'eau: 9,71 à 38,6 g/s
Flux de chaleur: 60 à 256 kW/m^2
Qualité à la sortie: 1,4 pour cent de vapeur avec une surchauffe de 10°C de la vapeur; des expériences de transport de chaleur sans ébullition et avec ébullition sous-refroidie ont également été effectuées.

Pour des qualités de vapeur au-dessous de 80 pour cent, le coefficient de transport de chaleur tout autour du tube était élevé, bien qu'abituellement plus élevé sur le côté le plus éloigné de l'axe de l'hélice. Le mouillage complet de la paroi du tube pour les qualités les plus élevées est attribué à l'écoulement secondaire dans le noyau de vapeur qui exerce une traînée sur le liquide et le force à s'écouler sur la surface la plus proche de l'axe de l'hélice.

Les portions supérieure et inférieure de la surface du tube sont devenues seches pour une qualité de vapeur d'environ 95 pour cent, mais le liquide était apparemment présent sur la surface du tube aux points les plus voisins et les plus éloignés de l'axe de l'hélice pour une qulité supérieure à 99 pour cent.

Le coefficient de transport de chaleur moyen le long de la circonférence et la chute de pression due au frottement ont été corrélés en fonction du paramètre de Lockhart–Martinelli dans toute la gamme, sauf aux faibles débits de vapeur lorsqu'il y avait une partie de l'ébullition sous forme nucleée. Les résultats des deux serpentins ne différaient pas de façon appréciable.

Zusammenfassung—Der Wärmeübergang beim Sieden bei Zwangskonvektion an Wasser von Atmosphärendruck wurde in zwei schraubenförmig aufgewickelten Rohren untersucht. Die Wicklungen wurden aus Inconel 600–Rohrmaterial von 3,05 m Länge, 15,9 mm Aussendurchmesser und 12,5 mm Innendurchmesser hergestellt.

Die Schraubendurchmesser betrugen 250 mm und 510 mm. Die Wärme wurde durch direkte Widerstandsheizung in der Rohrwand erzeugt. Die Bereiche der untersuchten Grössen betrugen:

Wasserdurchsatzmenge:	35–139 kg/h
Wärmestromdichte:	6–25,6 W/cm^2
Ausgangsqualität:	1,4 prozent Dampfgehalt bis um 27,8 grd überhitzter Dampf, ausserdem wurden Wärmeübergangsversuche bei Konvektion und bei unterkühltem Sieden durchgeführt.

Bei einem Dampfanteil von weniger als 80 prozent war der Wärmeübergangskoeffizient im Rohr ringsherum hoch, obwohl er gewöhnlich auf der seite am höchsten war, die am weitesten von der Schraubenachse entfernt lag. Die vollständige Benetzung der Rohrwand bei höheren Dampfgehalten wird der Sekundärströmung im Dampfkern zugeschrieben, die auf die Flüssigkeit einen Widerstand ausübt und sie zwingt, an der zur Schraubenachse gerichteten Fläche entlangzuströmen. Die obersten und die untersten Teile der Rohroberfläche wurden bei etwa 95 Prozent Dampfgehalt trocken, aber an den Stellen, die der Schraubenachse am nächsten und am entferntesten lagen, haftete Flüssigkeit bei einem Dampfgehalt von über 99 Prozent der Rohrwand an.

Die über den Umfang gemittelte Wärmeübergangszahl und der Reibungsdruckabfall wurden im gesamten Bereich ausser bei niedrigen Dampfstromraten, wo Blasensiedeverteilung auftrat, als Funktionen der Lockhart–Martinelli-Parameter dargestellt. Die Ergebnisse der beiden Wicklungen unterschieden sich nicht wesentlich.

Аннотация—Исследуется перенос тепла в трубах со спиральными змеевиками при кипении воды в условиях вынужденной конвекции при атмосферном давлении. Спирали изготовлены из 600 инконелевых трубок длиной 10 футов с внутренним диаметром 0,492″ и внешним диаметром $\frac{5}{8}$″. Шаг между осями трубок составлял 9,86″ и 20,5″. Тепло генерируется нагревателем постоянного тока, вмонтированном в стенку системы трубок. Условия эксперимента следующие: скорость подачи воды: 77–306 фунтов/час; тепловой поток: 1900–81000 БТЕ/час.футов^2; степень сухости на входе: 1,4% пара перегретого до 50°F; также проводились эксперименты по теплообмену при недогретом кипении и при отсутствии кипения.

При степени сухости меньше 80% коэффициент теплообмена был большим по всей трубе, хотя он обычно возрастает с удалением от оси спирали. Полное смачивание стенки трубы при больших степенях сухости возникает за счет вторичного течения в ядре пара, который влияет на жидкость и вносит её в поток поближе к оси спирали. Верхние и нижние участки на поверхности трубы высушиваются приблизительно до степени сухости 95%, но при этом, очевидно, жидкость имеется на поверхности трубы в точках самых близких от оси спирали и самых удаленных от неё. Средний коэффициент теплообмена по периметру и потеря давления от трения выражаются в зависимости от параметра Локхарта–Мартинелли во всем диапазоне, за исключением случая малых скоростей потока пара при пузырьковом кипении. Результаты, полученные для двух спиралей, незначительно отличаются друг от друга.