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# Forced convective boiling in horizontal tube bundles

T. H. HWANG and S. C. YAO
Department of Mechanical Engineering, Carnegie-Mellon University, Pittsburgh, Pa 15213, U.S.A.

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**Abstract**—Experimental studies of crossflow boiling on a horizontal tube at various mass fluxes, local flow qualities and geometric arrangements are investigated. Since abundant information is available for the boiling on a single tube in a pool but it is still not clear whether this information may be applicable to tubes in bundles, the present study is therefore performed on three different conditions, namely: (1) a heated tube in a channel; (2) a heated tube in a non-heated, in-line tube bundle; and (3) a heated tube in a heated, in-line tube bundle. The different heat transfer results between a single tube in a channel and a tube in a non-heated bundle, and between a non-heated bundle and a heated bundle are discussed in terms of the different flow field geometry and thermal environment respectively due to the presence of different structures and the heating conditions near the tube. A modified Chen's correlation is established to predict the heat transfer of a single tube in a channel or in a bundle. The correlation is also in good agreement with other data in the literature.

## INTRODUCTION

CROSS-FLOW boiling in horizontal tube bundles is of great importance to a variety of engineering applications, such as evaporators, steam generators, chemical reboilers and many major components in chemical and power plants. Reliable prediction of boiling heat transfer in horizontal tube bundles is one of the important factors for the successful design of those components.

In a typical horizontal steam generator, a steam–water mixture flows by the gravity-induced natural circulation. The flow is normal to the tubes and boiling occurs at the shell side of the tubes. The overall boiling phenomena is very complicated; however, at any position in the bundle the boiling heat transfer on a tube is mainly dependent upon the local flow velocity and the local quality. The study of heat transfer in a particular region of a large bundle can, therefore, be performed in a smaller bundle at a forced convective boiling condition with a preserved velocity and quality environment.

A single tube is the building block of a bundle heat exchanger. Although fundamental understanding and parametric effects have been fairly well established for the boiling heat transfer of a single tube in a channel or in an infinite pool at crossflow conditions [1–7], the results are still far removed from direct application to tube bundles. This is because the two key factors, flow field environment and quality environment effects, have not been simulated and investigated systematically to relate the single tube heat transfer to the heat transfer of a tube in a bundle.

The available information of crossflow boiling heat transfer in horizontal tube bundles is limited to the experience of the reboiler design in chemical processes and in power plants [8–14]. The effect of relative location of heated cylinders on pool boiling heat transfer has been reported by Wallner [8]. It was observed that vapor rising from the lower tube induced circulation and turbulence around the upper tube where a higher heat transfer coefficient is obtained. Three bundles containing 12 tubes each, including two staggered arrays with 1.33 and 1.5 pitch-to-diameter ratios and one in-line array with a pitch-to-diameter ratio of 1.33, were studied in ref. [8]. An astonishing fact was that almost no difference were observed among the average heat transfer coefficient of these three bundles. This is possibly because the bundle is at a natural circulating boiling condition. The average heat transfer coefficient is dependent upon many factors; for example, the power level, pressure drop of bubbly flow, etc. Therefore, the variation of the detailed bundle geometry does not induce significant difference to the integral result of the average bundle heat transfer. No correlation of the data was presented in the paper.

The overall pool boiling heat transfer in 14 HTRI reboiler bundles was investigated in ref. [9]. The results also show that no obvious difference in the overall performance was detected in staggered and in-line array bundles. Therefore, it was concluded that under the condition of highly turbulent boiling in bundles at high heat flux, the tube arrangement was not critical although the pitch-to-diameter ratio was influential.

The fluid flow and heat transfer behavior on the shell side of a reboiler bundle were observed for a 241-tube bundle by Cornwell *et al.* [10, 11]. Examination by high speed photography of the flow between the upper tubes of the horizontal bundle revealed the presence of a turbulent bubbly flow. It was concluded that sliding bubbles are possibly the main contributors to the enhancement of heat transfer that is observed at the upper portion of the bundle. Since the flow is at the natural circulation condition, the effects of flow velocity and bundle geometry were not separable in refs. [10, 11]. The local flow quality and velocity near the tubes

**NOMENCLATURE**

- $d$ tube diameter [mm]
- $F$ two-phase Reynolds number factor
- $G$ mass flux, $\rho_f U$ [kg m$^{-2}$ s$^{-1}$]
- $H$ width of flow channel [mm]
- $\bar{h}$ average heat transfer coefficient [kW m$^{-2}$ °C$^{-1}$]
- $h_{FC}$ single-phase forced convective heat transfer coefficient [kW m$^{-2}$ °C$^{-1}$]
- $h_{FNB}$ forced nucleate boiling heat transfer coefficient [kW m$^{-2}$ °C$^{-1}$]
- $h_{fg}$ latent heat of vaporization [J kg$^{-1}$]
- $h_{PNB}$ pool boiling heat transfer coefficient [kW m$^{-2}$ °C$^{-1}$]
- $h_l$ liquid-only forced convective heat transfer coefficient [kW m$^{-2}$ °C$^{-1}$]
- $k_f$ thermal conductivity of liquid [kW m$^{-1}$ °C$^{-1}$]
- $\overline{Nu}_d$ average Nusselt number based upon tube diameter, $\bar{h}d/k_f$
- $p$ pitch of the bundle [mm]
- $Pr$ Prandtl number, evaluated at $T_f$
- $Pr_w$ Prandtl number, evaluated at $\bar{T}_w$
- $Pr_L$ Prandtl number, evaluated at $T_b$
- $q_w$ heat flux [kW m$^{-2}$]
- $R^*$ Bond number, $(d/2)/[\sigma/g(\rho_f-\rho_g)]^{1/2}$
- $Re_d$ Reynolds number, $Ud/v$
- $S$ suppression factor
- $T_b$ bulk fluid temperature [°C]
- $T_f$ film temperature, $(T_b+\bar{T}_w)/2$ [°C]
- $\bar{T}_w$ average wall temperature [°C]
- $T_{sat}$ saturation temperature [°C]
- $\Delta T$ $\bar{T}_w - T_{sat}$ [°C]
- $U$ flow velocity at minimum flow area [m s$^{-1}$]
- $U_\infty$ freestream velocity [m s$^{-1}$]
- $X_{in}$ inlet flow quality
- $X_{Loc}$ local flow quality
- $Y$ variable, defined in equation (9).

Greek symbols

- $\alpha_m$ modified void fraction
- $v$ kinematic viscosity of liquid phase [m$^2$ s$^{-1}$]
- $\rho_f$ density of liquid phase [kg m$^{-3}$]
- $\rho_g$ density of vapor phase [kg m$^{-3}$]
- $\sigma$ surface tension of liquid phase [N m$^{-1}$].

are also not clearly known. Their proposed correlation appears to overpredict significantly the reboiler data [6].

Some further study of heat transfer in tube bundles was published in ref. [12]. The heat transfer coefficient on tubes was measured at the top row of a 6 × 6 in-line bundle with a pitch-to-diameter ratio of 1.24 in the upflowing Freon-113 at low quality of flow. It was observed that the mean heat transfer coefficient on the top row increased, when flow quality was increased.

Recently, Chan and Shoukri [13] have investigated the boiling heat transfer of a single tube and a tube in bundles (3 × 3 or 3 × 9 in-line bundle) in a liquid pool. The data were presented in terms of the location of the tube in the bundle and the heat flux level of its neighboring tubes. The velocity and quality of the circulating flow in the bundles were not measured. Hence, the results presented in their paper were restricted to the tested bundle geometry and were not general.

A calculational scheme considering several heat transfer mechanisms was proposed by Palen and Yang [14] to predict the overall pool boiling heat transfer rate of large tube bundles. The model was tested and adjusted against 400 HTRI kettle reboiler data points, which included a wide range of pressure, flow properties, tube surface conditions and bundle geometries. Although local heat transfer is deduced from their experiment, assumptions are made to evaluate the local information from the integral results of the whole bundle.

In the investigations mentioned above [1–14], the heat transfer of a single tube and that of a tube in a bundle, have been studied separately. Therefore, it has been very difficult to compare systematically the results of these two problems. Also, in most of the tube bundle studies the bundles were at natural circulation boiling so that the flow and quality conditions were not clearly known. Furthermore, many of the studies are for the overall performance instead of the local behavior in a bundle.

The objectives of the present study are to reveal systematically the fundamental behavior of boiling and to obtain a better prediction of the heat transfer coefficient on a tube in horizontal bundles. Since abundant information is available for the boiling on a single tube in pool but whether this information may be applicable to tubes in bundles is not known, the present study is therefore focused on three different conditions, namely: (1) a heated tube in a channel; (2) a heated tube in a non-heated, in-line tube bundle; and (3) a heated tube in a heated, in-line tube bundle. In this paper, the experimental heat transfer results of a tube in heated or non-heated bundles are reported and compared with those of a single tube in a channel.

The difference between the heat transfer of a single tube in a channel and that in a non-heated bundle is mainly due to the different flow field geometry. And, the difference between the heat transfer of a tube in a non-heated bundle and that in a heated bundle is attributed to the different thermal environment, e.g. local quality distribution. Therefore, proper comparison between the heat transfer results of a single tube in a channel and that in a heated bundle can be related in terms of the effects of fluid flow geometry and quality environment separately. Then, the existing heat transfer knowledge on the single tube heat transfer [1–7] can have a better justification for its implication to the horizontal tube bundles.

## EXPERIMENTAL APPARATUS AND PROCEDURE

A forced convective Freon-113 loop is used for flow boiling and pool boiling experiments in this study. Some details are described in refs. [7, 15]. A simplified sketch of the test bundle is shown in Fig. 1. The test chamber is vertically oriented with Freon-113 flowing against gravity. Glass windows are installed at several locations in the chamber to enable visual observation, lighting and photographing of the flow pattern in the test section. The bundle is a three column, in-line tube bundle with a pitch-to-diameter ratio of 1.5. The test chamber has smooth sidewalls with its distance to the adjacent tubes equal to half the tube-to-tube spacing. The bundle consists of six rows of upstream aluminum rods, which allow the flow to develop fully. The heated tubes are stainless 304 seamless tubing with 19.1 mm O.D., 0.51 mm wall thickness, and 49.4 mm heated length. These thin-wall tubes are heated with direct current which contains 0.6% ripple and the current is measured from the voltage drop over a shunt. Particular cares are taken to polish the heated tube surface before each test.

The instrumented tube is located at the center of the seventh row of the heated bundle. Two J-type stainless-steel-sheathed, ungrounded thermocouples of 0.81 mm diameter are firmly pressed against the inner wall by plate-springs. The thermocouples are also covered with insulating cement to reduce the heat loss through the

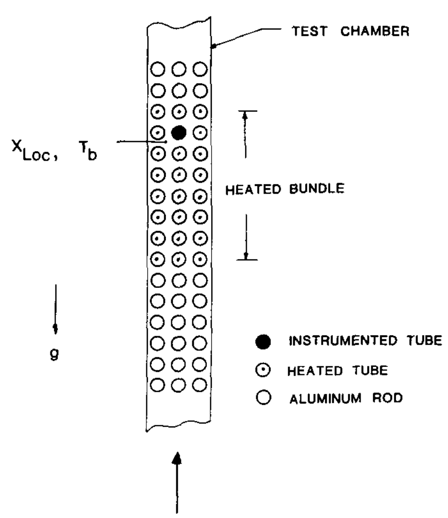


FIG. 1. Schematic of test bundle.

thermocouples and the natural convection inside the tube. The thermocouples are located at the middle of the heating length and can be rotated circumferentially along the inner wall of the tube. The outer surface temperature is calculated by assuming one-dimensional, steady-state heat conduction. The average heat transfer coefficients on a tube are obtained by averaging the local heat transfer coefficients.

The maximum heat loss by natural convection at the inside of the heated tube and the heat loss through the thermocouples are estimated [16] to be less than 1.4% of the power applied to the heated tube. The temperature measurement errors due to the thermocouples' heat sink effect are estimated [16] to be less than 0.14°C. The circumferential conduction is also negligible due to the thin tube wall and low thermal conductivity of stainless steel. The same instrumented tube is used for all the tests to eliminate any systematic error. Detailed error analysis is described in ref. [15].

The inlet flow quality, $X_{\rm in}$, which can be calculated through the energy balance, is produced by preheating the fluids and then throttling through the regulating valve located at the entrance of the test section. The local flow quality near the instrumented tube, $X_{\rm Loc}$, can also be evaluated through the energy balance. All the temperatures in the experiments are recorded by Accurex Autodata Logger (Model Ten/5) with the accuracy of $\pm 0.1$°C, and programmed with an interfacing terminal.

*Experimental procedure*

Three different conditions are investigated in the experiment, namely: (1) a single tube in a channel; (2) a tube in a non-heated bundle; and (3) a tube in a heated bundle. In the second case, only the instrumented tube is heated and all the other tubes are non-heated. In the third case, all the heated tubes can be heated uniformly at the same heat flux. However, in most of the experiments, two separate heating zones at different heat fluxes are maintained. The power to the first six rows of the heated tubes at the upstream of the instrumented tube are supplied from a welder (up to 40 V, 300 A). The seventh and eighth rows of heated tubes use a separate power supply (up to 100 V, 1500 A). The heat flux on the instrumented tube can, therefore, be varied without affecting the local flow quality approaching this tube.

At the beginning of each experiment, the Freon-113 loop is degassed for about 30 min. Then the steady-state convective condition is established; this takes approximately 1 hr. The tube is then heated by direct current. During the experiment, data are taken 3 min after a steady state is reached. The range of local flow quality is between 0 to 0.143. For subcooled boiling studies the local subcooling is fixed at 6°C. The mass flux varies between 0 and 817 kg m$^{-2}$ s$^{-1}$. The pool boiling experiments are conducted at a flow velocity of less than 1 cm s$^{-1}$ ($G < 12$ kg m$^{-2}$ s$^{-1}$). The test section pressure is maintained at atmospheric pressure. Details of the experimental procedure are available in ref. [15].

## RESULTS AND DISCUSSION

This section contains three parts: (A) heat transfer behavior; (B) overall comparison of heat transfer data; (C) correlation of data. Only typical data are presented here, the detailed data are available in ref. [15]. Although only one bundle geometry is considered, the parametric effects are expected to be general for similar conditions.

(A) *Heat transfer behavior*

*A single tube in a channel.* In the following discussion, the single-phase, forced convection heat transfer and nucleate boiling heat transfer on a single tube in a channel with various channel widths are examined. As a result, the present single tube results ($d/H = 0.22$) are similar to those in infinite pool at extreme cases: single-phase, forced convection and fully developed boiling.

*Single-phase, forced convection.* The single-phase, forced convection experiments are performed at both highly subcooled (40°C) and slightly subcooled (6°C) conditions. The present results are in good agreement with the empirical correlation [1]

$$\overline{Nu}_{\mathrm{d}} = 0.21\, Re_{\mathrm{d}}^{0.62}\, Pr_{\mathrm{L}}^{0.38} \left(\frac{Pr_{\mathrm{L}}}{Pr_{\mathrm{w}}}\right)^{0.25} \tag{1}$$

for $1000 \leqslant Re_{\mathrm{d}} \leqslant 2\times10^5$, where the blockage-corrected Reynolds number, $Re_{\mathrm{d}}$, is based upon the flow velocity at the narrowest cross section which can be related to the freestream velocity as

$$\frac{U}{U_\infty} = \frac{H}{H-d}. \tag{2}$$

Further examination of equation (1) reveals that the average Nusselt number is insensitive to the channel blockage ratio at a given Reynolds number ($Re_{\mathrm{d}}$). This insensitivity is due to the choice of reference velocity in which the $d/H$ effects is implicit.

*Nucleate boiling.* It is observed that at nucleate boiling the surface temperature profile is rather uniform and it becomes more uniform as the heat flux increases. However, the highest temperature is always detected at the top center of the heated tube (the rear stagnation point). This is because when the two-phase mixture passes the 90° position, the vapor tends to move into the wake region, and the liquid tends to continue in a tangential direction upward. It is, therefore, difficult for the top center portion of the tube to be reached by incoming liquid. The critical heat flux might also start at the top center portion. The trend of CHF data has been observed by Mckee and Bell [2], Yao Hwang [19].

● *Effect of mass flux.* The average heat transfer coefficients at slightly subcooled boiling conditions are shown in Fig. 2 as a function of wall heat flux. This is a typical way the heat transfer in horizontal tube bundles is presented [8]. The results show the expected behavior of velocity effect. At low heat flux, the heat transfer is enhanced significantly by the increasing of flow velocity. When the heat flux is sufficiently high, boiling is fully developed. No effect of velocity on

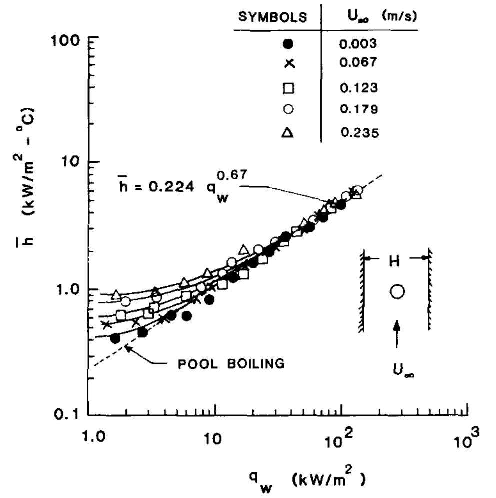


FIG. 2. Variation of average heat transfer coefficient with flow velocity at slightly subcooled condition.

boiling heat transfer is observed. Generally, at low velocity flow the fully developed boiling curve can be assumed to coincide with the extrapolation of the pool boiling curve. In present experiments, the fully developed boiling can be described as

$$\bar{h} = 0.224 q_{\mathrm{w}}^{0.67}. \tag{3}$$

This is consistent with the observation of Dul'kin *et al.* [4] for the nucleate pool boiling of a tube in an infinite pool. Present pool boiling data of Freon-113 on a single tube are also compared with some other pool boiling data [1, 6] and correlations [4, 8] for various fluids and channel blockage ratios. As shown in Fig. 3, when the channel blockage ratio is less than 0.6 it does not affect significantly the pool boiling heat transfer coefficient

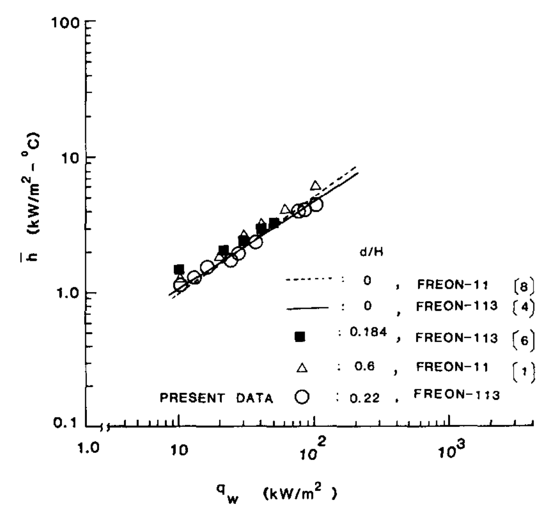


FIG. 3. Comparison of single-tube, pool boiling data and correlations at various channel blockage ratios.

(although the blockage ratio has been observed as a more influential factor at forced convective boiling condition [6]).

● *Effect of local quality.* It has been known that two-phase oncoming flow generally results in an increased heat transfer coefficient relative to the single-phase flow at the same mass flux. However, it should be pointed out that, as shown in Fig. 4, the increasing of heat tramsfer is due to two effects: (1) the agitation of vapor bubbles and the possible thin film evaporation at the tube surface [24]; and (2) the increasing of two-phase mixture velocity due to the increased quality at the same mass flux (so-called quality-induced flow velocity effect. In other words, for a fixed flow condition, increasing the quality decreases the mixture density, thus—to maintain a constant mass flux—the flow velocity must increase).

The contribution of the quality-induced flow velocity effect on the heat transfer can be examined by crossplotting some data from Fig. 2 to Fig. 4 using the homogeneous flow model. For example, the heat transfer of two-phase flow at 132 kg m$^{-2}$ s$^{-1}$ mass flux and 0.0075 quality is shown as the circular symbols in Fig. 4. The corresponding heat transfer of single-phase flow at the same linear velocity (and, of course, higher mass flux) can be deduced from Fig. 2 by interpolation. This single-phase result is shown in the form of the dashed line in Fig. 4 for comparison. The two cases have the same linear velocity; however, the higher heat transfer of the two-phase condition is likely due to the presence of vapor phase which causes the further agitation of the fluid and the thin film evaporation on the surface. In Fig. 4, the curve of subcooled convection (solid square symbols) is also shown which has the same mass flux as that of the two-phase condition (circular symbols). Therefore, it is at a lower linear velocity than that of the dashed line with the difference coming from the effect of the linear velocity.

The degree to which the heat transfer is increased at

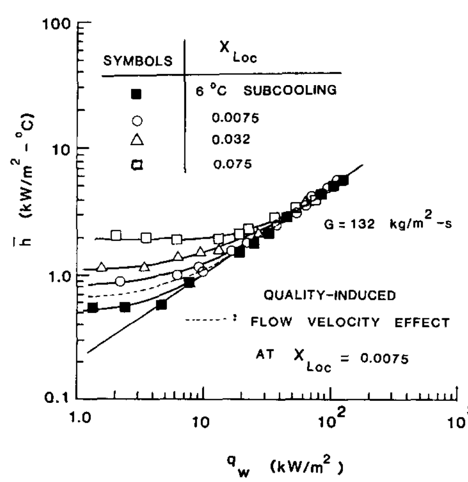


FIG. 4. Variation of the single-tube average heat transfer coefficient with local flow quality.

increasing flow quality is dependent upon the heat flux. At low heat flux, the upstream vapor bubbles impinge on the heated tube and result in a significant heat transfer enhancement. As the heat flux increases, bubbles generated from tube surface become massive and the upstream bubbles have less chance to impinge on the tube. Therefore, the enhancement decreases. At fully developed boiling, the upstream vapor bubble has no observed effect on the heat transfer.

*A heated tube in a non-heated bundle: Single-phase, forced convection.* The single-phase, forced convection experiments are performed at subcooled conditions and over a range of Reynolds number, $Re_d$, between 5300 and 22,000. Typical local heat transfer coefficients around the tube shows the similar trend as for a single tube in a channel. The minimum heat transfer occurs usually in the wake region.

Various single-phase heat transfer correlations are available for a tube in a heated bundle as discussed in ref. [17]. However, no heat transfer correlation exists for a tube in a non-heated bundle. Present data are, therefore, compared with the general correlation of Hwang and Yao [18] which is proposed for a heated bundle at $4000 < Re_d < 2 \times 10^5$

$$\overline{Nu}_d = 0.366 Re_d^{0.6}\, Pr^{1/3}. \tag{4}$$

The experimental results of the non-heated bundle are generally higher than the prediction of equation (4) by about 12%. The difference is possibly due to the fact that in a heated bundle the thermal boundary layer of the upstream tube will attach to the downstream tube, resulting in a reduction of the heat transfer on the downstream tubes.

*Nucleate boiling.* When the heat flux is increased to near the critical heat flux (CHF), the increase of the wall temperature at the upstream stagnation point is much faster than those at other portion of the tube. Thus, the highest temperature is detected at the upstream stagnation point when a premature CHF is observed. This is possibly because the effect of local quality distribution gives a higher quality at the upstream stagnation point due to the wake of the lower tube in the in-line bundle [15, 19].

The effect of mass flux on the nucleate boiling heat transfer at the slightly subcooled condition is similar to that of a single tube in a channel. For a given heat flux, the higher the mass flux, the higher the boiling heat transfer coefficient. When the heat flux is sufficiently high, boiling is fully developed and no effect of mass flux is observed. In present experiments, the fully developed boiling can be described as

$$\bar{h} = 0.16 q_w^{0.77}. \tag{5}$$

The present pool boiling data are obtained at a flow velocity of less than 1 cm$^{-1}$ s ($G < 12$ kg m$^{-2}$ s$^{-1}$). The effect of local quality on the heat transfer is shown in Fig. 5. Similar to that observed on a single tube in a channel, the flow of higher quality results in a higher heat transfer coefficients at a same mass flux. Similar to that of a single tube in a channel, the degree to which the

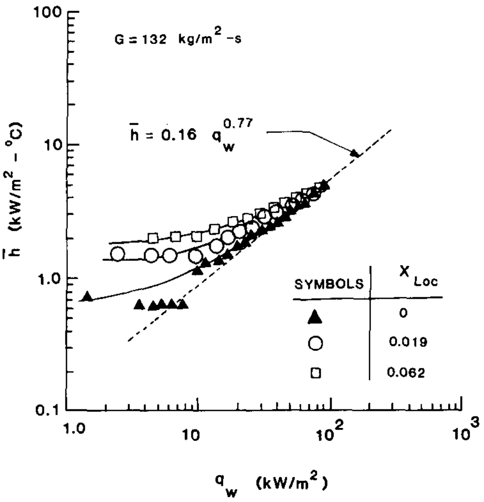


FIG. 5. Variation of average heat transfer coefficient with local flow quality in non-heated bundles.

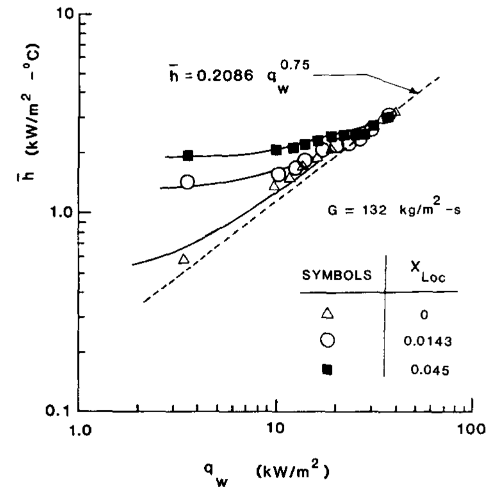


FIG. 6. Variation of average heat transfer coefficient with local flow quality in heated bundles.

heat transfer is increased is also dependent upon the heat flux.

*A heated tube in a heated bundle.* As discussed previously, two separate heating zones (non-uniform heating zones in the bundle) are considered in this section. The heat flux to the first six rows of upstream heated tubes is set at 3.5 kW m$^{-2}$. Although no boiling inceptions are observed on the upstream heated tubes during the tests, the local flow quality is produced by the throttling process through the regulating valve at the entrance of test section. The seventh and eighth rows of heated tubes are heated by the direct current at various powers. The pool boiling data are obtained at a flow velocity less than 1 cm s$^{-1}$ ($G < 12$ kg m$^{-2}$ s$^{-1}$). The experimental results of a tube in a uniformly heated (one zone heating) bundle will be discussed in a later section.

In order to avoid the occurrence of CHF on a non-instrumented tube, the nucleate boiling data are not obtained at high heat flux. The data indicate the same trend to the mass flux variation as occurs on a tube in a non-heated bundle. In the present experiment, the fully developed boiling curve can be expressed as

$$\bar{h} = 0.2086 q_w^{0.75}. \tag{6}$$

The effect of local flow quality on the boiling heat transfer is illustrated in Fig. 6 and is similar to that of a non-heated bundle. The higher the local flow quality, the higher the heat transfer at a same mass flux. When the heat flux increases further, the effect of flow quality diminished and the fully developed boiling is established.

(B) *Overall comparison of the heat transfer data*

It is interesting to compare the heat transfer data of three previously described cases at the same mass flux and local quality or subcooling conditions. Figure 7 shows the comparisons at the same local subcooling.

• *Single-phase, forced convection.* In general, the single-phase, forced convection heat transfer coefficient of a single tube in a channel is the lowest, but that of a tube in a non-heated bundle is the highest.

It has been well known that the single-phase heat transfer from a tube is determined by its position in the bundle. In most cases [17], the heat transfer from tubes in the first row is considerably lower than in inner rows for $Re_d > 3000$. This is due to the increase of flow turbulence leading to an increase of heat transfer of the inner tube. Furthermore, the heat transfer from a single tube in a channel is similar to that of the first row tubes. Hence, the present single-tube heat transfer is lower than that of the inner tubes in a heated bundle. Comparison of equations (1) and (4) at the same

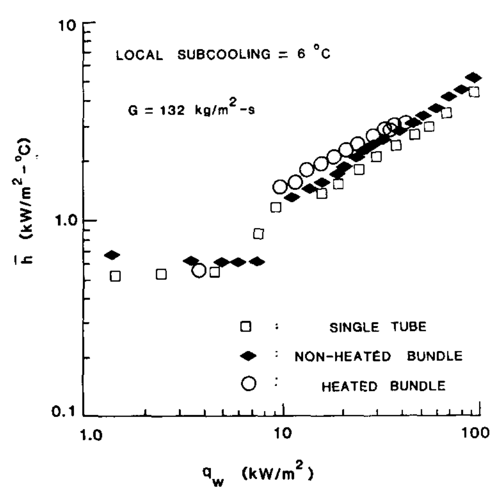


FIG. 7. Comparison of the heat transfer performance of a tube in a channel, in a non-heated bundle and in a heated bundle at subcooled condition.

Reynolds number ($Re_d$) also supports the present data comparison.

As for the difference of single-phase heat transfer between a tube in a non-heated bundle and in a heated bundle, the different heating condition is the main contribution since the turbulence intensities are at the same level in both cases. As shown in Fig. 7, the non-heated bundle has a slightly higher single-phase heat transfer coefficient than in the heated bundle. This is because the upstream thermal boundary layer attaches on the downstream tube and results in a reduction of the heat transfer on the instrumented tube as compared with that in a non-heated bundle.

● *Nucleate boiling*. In all cases, the boiling heat transfer of a single tube in a channel is the lowest, but that of a heated tube in a heated bundle is the highest. The heated and non-heated bundle have almost the same heat transfer coefficient when the heat flux is very high.

The flow regimes near the tube and at the inlet of the test section are observed and compared with the flow regime maps in refs. [20, 23]. It is consistently at the bubbly/slug transition regime in the present study. The schematics of the observed boiling phenomena are shown in Fig. 8. The different boiling heat transfer performance among those three cases can be explained as follows:

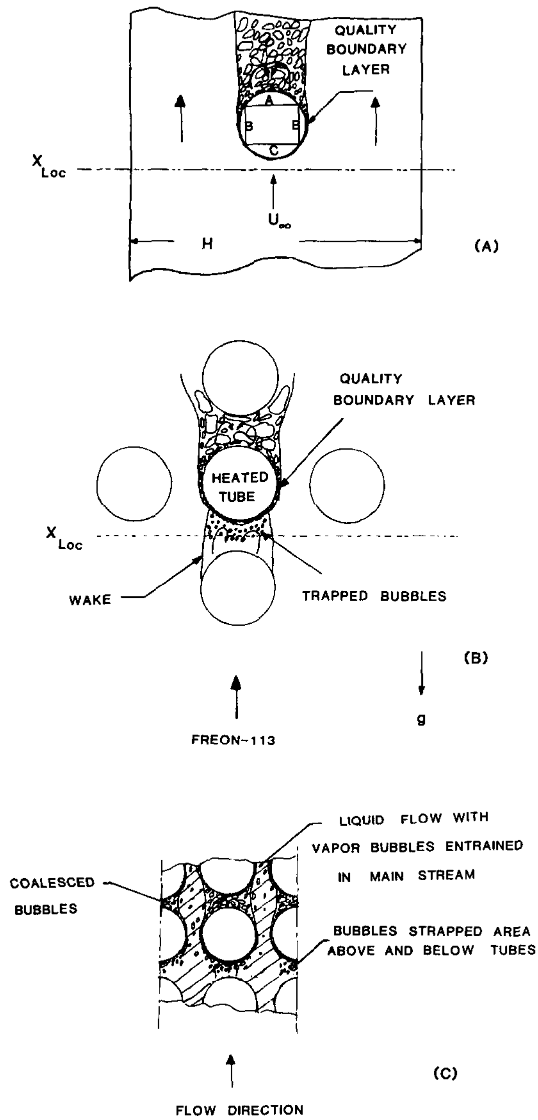


FIG. 8. Schematics of boiling phenomena: (A) a tube in a channel, (B) a tube in a non-heated bundle, (C) a tube in a heated bundle.

1. First, the difference between the nucleate boiling heat transfer of a single tube in a channel and of a heated tube in a non-heated bundle is basically due to difference of flow field geometry near the heated tube. In all cases, the heat transfer in a non-heated bundle is higher than that in a channel. For the sake of discussion, the flow field near the heated tube can be roughly distinguished into three regions: rear (region A), sides (region B) and front (region C) as shown in Fig. 8(a).

   At the top portion of the tube, region A, the boiling is similar to the pool boiling of a horizontal plate facing upward with some circulating flow near the plate. Small individual bubbles are generated from region A and leave tube surface. The heat transfer at region A of a single tube in a channel is almost the same as that in a bundle. The downstream tube does not affect significantly the heat transfer at region A. At the bottom of the tube, region C, boiling occurs. These bubbles eventually slide into the downstream wake region or entrain into the main stream. In a non-heated bundle, some of the bubbles coalesce and circulate in the region C and result in a higher local quality and a higher local heat transfer coefficient as compared with the case of a single tube in a channel. On the left and right sides of the tube, region B, the boiling is similar to that on a vertical wall with Freon-113 flows upward along the wall. 'Quality boundary layers' are formed with its thickness related to the amount of heat flux. The typical quality boundary-layer thickness is about 1.5 mm at a heat flux of 20 kW $m^{-2}$. These boundary layers effectively decrease the flow cross section in the gap, therefore, causing a higher 'effective flow velocity' in the gap. The effect of quality boundary layer is more significant in the bundle than in the channel because the effective flow area is smaller in a tube bundle (0.5*d*) than in a channel (1.75*d*). It is the combined effects of the quality boundary layers in the region B and the quality distribution in the region C that enhance the average heat transfer coefficient of a tube in a non-heated bundle as compared with that of a single tube in a channel at the same flow conditions.
2. The difference between the boiling heat transfer of a heated tube in a non-heated bundle and in a heated bundle is basically attributed to the different thermal environment near the instrumented tube because the geometry of the flow fields are the same.

Conversely to the trend observed in single-phase, forced convection, the boiling heat transfer of a tube in a heated bundle is slightly higher than that in a non-heated bundle. Although the mass flux and average flow quality are the same in both cases, the distribution of the flow quality near region C is slightly different. In a heated bundle, the upstream tube is heated providing a higher local flow quality near the front portion of the next tube and resulting in a slightly higher heat transfer at region C than that in a non-heated bundle.

The adjacent heated tubes also enhance the heat transfer in a heated bundle. Due to the existence of quality boundary layers on the adjacent tubes, the effective flow velocity is even higher than that in a non-heated bundle. Thus, results in a higher heat transfer at region B. As a result, the average heat transfer coefficient of a heated tube in a heated bundle is always slightly higher than that in a non-heated bundle.

Since in an actual vapor generator the tubes are heated at about the same heat flux in a local region of a large bundle, the uniformly heated bundle is also investigated in this study. The mass flux effect in the uniformly heated bundle is presented in Fig. 9. The boiling curves ($\bar{h}$ vs $q_w$) in Fig. 9 are different from all the previously discussed cases. In an uniformly heated bundle, the local flow quality near the instrumented tube increases as heat flux increases although the inlet flow quality is kept the same. It is this combined effect of mass flux and heat flux-induced local flow quality that made the difference among all the curves at intermediate heat flux in Fig. 9. No fully developed boiling is established because the range of heat flux is kept low enough to avoid the possible burnout. The effect of the inlet flow quality on the boiling heat transfer is also investigated. Similar to the previously discussed experiments, two-phase oncoming flow results in an increase of heat transfer coefficient at a same mass flux. The local flow quality at the instrumented tube can be obtained through the energy balance, and the deduced data based upon the local flow quality are replotted in Fig. 10. In this figure, the fully developed boiling result of the non-uniformly heated bundle, equation (6), is also included as a reference. Not many data are available in Fig. 10, however, the data indicate the similar trend of local flow quality effect as the previously discussed cases.

To avoid the confusion, the data of uniformly heated bundle are not replotted in Fig. 7. However, the results of the uniformly heated bundle and that of non-uniformly heated bundle are also compared at the same mass flux and local flow quality conditions. In an uniformly heated bundle boiling occurs on the top portion of the upstream tube. Although a slightly different heat transfer at region C may be expected, the comparisons of data in Fig. 6 and Fig. 10 show no significant difference of average heat transfer.

(C) *Correlation of data*

For engineering application, a correlation is desirable. The widely used Chen correlation [21] developed for in-tube, forced convective boiling by heat/momentum transfer analogy can be modified to predict the flow boiling heat transfer across a bundle. The boiling heat transfer of a horizontal tube in a non-heated bundle has been studied in ref. [24] for down-flow conditions. A correlation has been proposed in a form similar to that of Chen's correlation; however, the heat transfer results are presented in terms of the two-phase pressure drop which is not very easily available in general.

The original Chen correlation should not be expected to fit the present data well, since the effective two-phase Reynolds number factor ($F$) of ref. [21] is

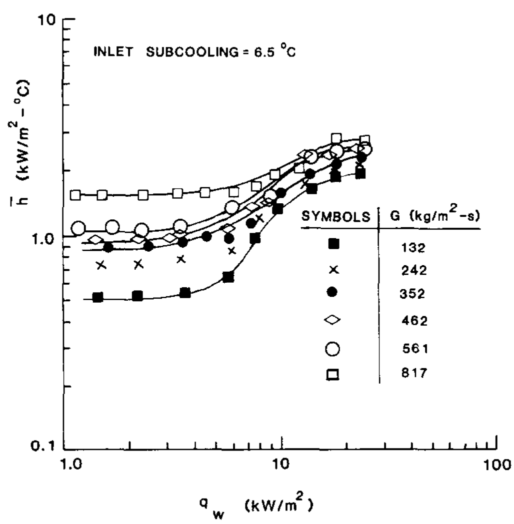


FIG. 9. Variation of average heat transfer coefficient with mass flux in an uniformly heated bundle.

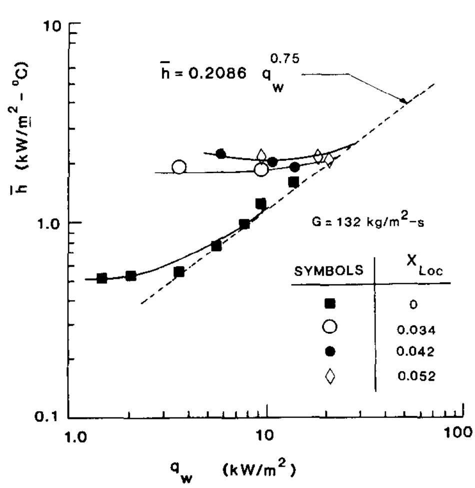


FIG. 10. Variation of average heat transfer coefficient with local flow quality in an uniformly heated bundle.

based upon the ratio of the two-phase pressure gradient to the liquid-only pressure gradient in a straight tube. The present geometry of crossflow over a bundle is considerably different from the in-tube flow, since the pressure loss in bundles contains form drag in addition to the frictional drag. However, the general form of Chen's correlation

$$h_{\mathrm{FNB}} = Sh_{\mathrm{PNB}} + Fh_{\mathrm{l}} \tag{7}$$

is still appropriate if the present geometry is considered. The pool boiling heat transfer coefficient, $h_{\mathrm{PNB}}$, of the bundle geometry can be obtained from equation (5) and equation (6) for non-heated and heated bundles, respectively. The factor $S$, which represents the suppression of the nucleate boiling contribution, developed by Bennett *et al.* [22] for both in-tube and shell-side crossflow forced convective boiling by leaving the geometry influence implicitly in the convective coefficient ($Fh_{\mathrm{l}}$) can be used.

$$S = \frac{K_{\mathrm{f}}}{Fh_{\mathrm{l}}Y}\left[1-\exp\left(-Fh_{\mathrm{l}}Y/K_{\mathrm{f}}\right)\right] \tag{8}$$

where

$$Y = 0.0205\left(\frac{d}{R^*}\right). \tag{9}$$

In equation (7), the liquid-only heat transfer coefficient, $h_{\mathrm{l}}$, can be expressed in terms of the single-phase, forced convective heat transfer coefficient of homogeneous flow in the present bundle

$$h_{\mathrm{l}} = h_{\mathrm{FC}}(1-X_{\mathrm{Loc}})^{0.6}. \tag{10}$$

Thus, for the present geometry, only an appropriate $F$ factor needs to be developed.

Recently, Polley *et al.* [12] studied the boiling heat transfer in an in-line bundle and used a similar heat transfer correlation as equation (7) by assuming the suppression factor equal to one. As a result, their correlation overpredicted the contribution of boiling heat transfer. Since the geometry has been considered in the pool boiling heat transfer coefficient and single-phase, forced convective heat transfer coefficient, therefore, the $F$ factor of [12] may be used with little dependency to the slightly difference of geometry. The $F$ factor is expressed as

$$F = \left(\frac{1}{1-\alpha_{\mathrm{m}}}\right)^{0.744}, \tag{11}$$

and

$$\alpha_{\mathrm{m}} = \frac{0.833X_{\mathrm{Loc}}}{X_{\mathrm{Loc}}+(1-X_{\mathrm{Loc}})(\rho_{\mathrm{g}}/\rho_{\mathrm{f}})} \tag{12}$$

where $\alpha_{\mathrm{m}}$ is the modified void fraction, which is different from the homogeneous flow model by a constant factor as suggested in ref. [12].

Using equations (11) and (8) for the $F$ and $S$ factors, together with equations (4) and (5) or (6), the bundle data are predicted within $\pm 20\%$ of the modified Chen's correlation as shown in Fig. 11. No apparent bias is observed for these two sets of data of heated and non-heated bundle. The data of Polly [12] is also shown with $h_{\mathrm{PNB}}$ taken from their data base.

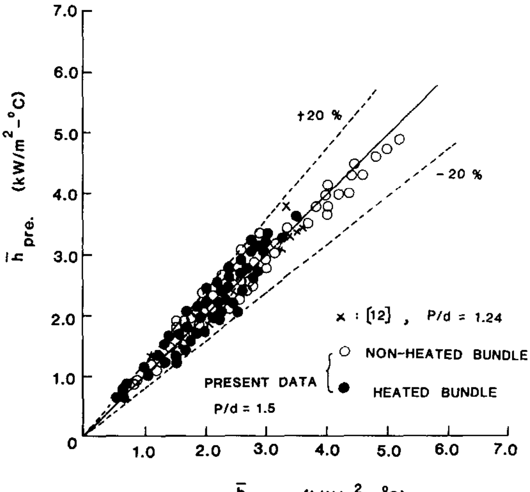


FIG. 11. Comparison of bundle data with the predictions of equations (7), (8) and (11).

To test the generality of equation (7), the single-tube data are also compared with the prediction of equation (7) in Fig. 12. The pool boiling heat transfer coefficient and the single-phase, convective coefficient are obtained from equations (3) and (1), respectively.

Some single-tube boiling data of [1, 6] are also included in Fig. 12. Good agreement between the data and the equation (7) is also observed. This is because the geometry effect has been considered in the heat transfer coefficient of pool boiling and the single-phase forced convection in equation (7). The proposed correlation, equations (7), (8) and (11), therefore, can be widely

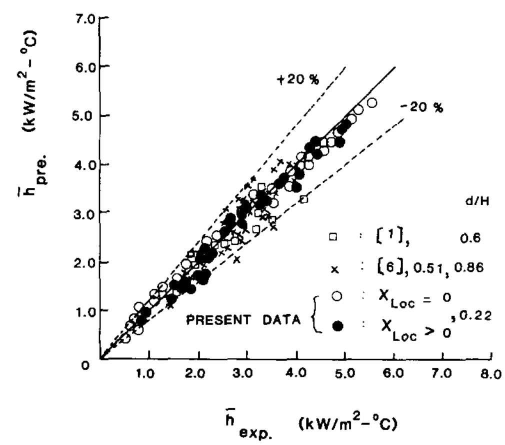


FIG. 12. Comparison of single-tube data with the predictions of equations (7), (8) and (11).

applied to conditions of different geometry of cross flow over tubes in bundles.

## CONCLUSIONS

This paper reported the first systematic experimental investigation of the crossflow boiling heat transfer on a horizontal tube at various mass fluxes, local heat fluxes and geometric arrangements that are commonly encountered in horizontal vapor generators. To relate the well known results of single-tube boiling in a pool to the practical problem of a tube in a heated bundle, three different kinds of experiments are performed, namely: (1) a heated tube in a channel; (2) a heated tube in a non-heated, in-line tube bundle; and (3) a heated tube in a heated, in-line tube bundle. Freon-113 is the heat transfer medium in all of the experiments. Based on the investigation of this study, the following conclusions are reached:

1. For a tube in a channel, the channel blockage ratio $(d/H)$ has less effect on the heat transfer of the tube at single-phase flow conditions or at pool boiling conditions as indicated in equation (1) and Fig. 3. However, the channel blockage effect is significant at forced convective boiling conditions. The effects of mass flux and local flow quality on the boiling heat transfer coefficient are similar in those three cases. The higher the mass flux or local flow quality, the higher the heat transfer coefficient at a same heat flux.
2. The differences of the heat transfer behavior among the above mentioned various combination of geometry and fluid flow condition have been examined. Generally speaking, the single-phase convection heat transfer coefficient of a single tube in a channel is the lowest, but that of a tube in a non-heated bundle is the highest. This is because the increase of the turbulence due to the presence of the upstream tubes, as compared with a single tube in a channel. Furthermore, in a tube bundle the thermal boundary layer of the upstream tube attached on the downstream tube and results in a reduction of heat transfer as compared with that in a non-heated bundle. Different from the trend observed in single-phase convection, the boiling heat transfer of a single tube in a channel is the lowest, but that of a heated tube in a heated bundle is the highest. The major reasons determining the boiling heat transfer enhancement on a horizontal tube are likely the local flow quality at upstream stagnation portion and the quality boundary layers at two sides of the tube.
3. The modified Chen's correlation, equations (7), (8) and (11), can be applied to predict the crossflow heat transfer from a heated tube in a channel or in a bundle. Reasonable agreements are obtained for the predictions of the proposed correlation within $\pm 20\%$.
4. This paper offers an interesting systematic approach which may be valuable in reducing the amount of testing needed to develop adequate information of flow boiling in tube bundles.

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## EBULLITION EN CONVECTION FORCEE DANS DES FAISCEAUX DE TUBES HORIZONTAUX

**Résumé**—On analyse les études expérimentales de l'ébullition en attaque transversale d'une tube horizontal pour différents débits surfaciques, différentes qualités locales de l'écoulement et plusieurs arrangements géométriques. Il existe une abondante information sur ce sujet mais l'application au cas des grappes de tubes n'est pas claire. La présente étude est conduite pour trois conditions différentes, à savoir : (1) un tube chauffé dans un canal, (2) un tube chauffé dans un faisceau en ligne et non chauffé, (3) un tube chauffé dans un faisceau chauffé en ligne. Les différents transferts thermiques entre un tube unique dans un canal et un tube dans un faisceau non chauffé, comme entre un faisceau non chauffé et un faisceau chauffé, sont discutés à partir des géométries différentes du champ d'écoulement et des environnements thermiques du fait de la présence de structures différentes et des conditions de chauffage proches du tube. On établit une corrélation modifiée de Chen pour prédire le transfert thermique d'un tube unique dans un canal ou dans un faisceau. La corrélation est aussi en bon accord avec d'autres données de la bibliographie.

## STRÖMUNGSSIEDEN IN HORIZONTALEN ROHRBÜNDELN

**Zusammenfassung**—Experimentelle Untersuchungen zum Sieden an einem querangeströmten Rohr wurden für verschiedene Massenstromdichten, örtliche Dampfgehalte und geometrische Anordnungen untersucht. Da genügend Information zum Behältersieden an einem Einzelrohr vorliegt, aber noch nicht klar ist, wie diese Information auf Rohrbündel anzuwenden ist, wurde die vorliegende Studie bei drei unterschiedlichen Bedingungen durchgeführt: (1) ein beheiztes Rohr im Kanal, (2) ein beheiztes Rohr in einem nicht beheizten fluchtenden Rohrbündel, (3) ein beheiztes Rohr in einem beheizten fluchtenden Rohrbündel. Die verschiedenen Ergebnisse des Wärmeübergangs zwischen einem Einzelrohr im Kanal und einem Rohr in einem nicht beheizten Bündel und zwischen einem nicht beheizten Bündel und einem beheizten Bündel werden in Bezug auf die abweichende Geometrie der Strömungsfelder und der thermischen Umgebung bzw. der unterschiedlichen Strukturen und Heizbedingungen nahe dem Rohr diskutiert. Eine modifizierte Chen-Gleichung wurde aufgestellt, um den Wärmeübergang eines Einzelrohres im Kanal oder in einem Bündel vorauszuberechnen. Die Korrelation stimmt ebenfalls gut mit anderen Daten aus der Literatur überein.

## ВЫНУЖДЕННОКОНВЕКТИВНОЕ КИПЕНИЕ В ПУЧКЕ ГОРИЗОНТАЛЬНЫХ ТРУБ

**Аннотация**—Экспериментально изучается кипение при поперечном обтекании горизонтальной трубы при различных значениях потока массы, локальных свойствах течения и геометрии системы. Поскольку имеются обширные данные по кипению на одной трубе в большом объеме жидкости, но не ясно можно ли перенести их на пучки труб, исследования проводились для трех случаев: нагретая труба в канале (1), нагретая труба в ненагретом (2) и нагретом (3) коридорных пучках труб. Результаты по теплообмену между трубой в канале и ненагретым пучком и между ненагретым и нагретым пучками труб рассмотрены для различных геометрий поля течения и тепловых условий. Получено модифицированное соотношение Чена для расчета теплообмена трубы в канале или пучке, которое согласуется с известными данными.