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PREDICTION OF BOILING HEAT TRANSFER DUTY IN A COMPACT PLATE–FIN HEAT EXCHANGER USING THE IMPROVED LOCAL ASSUMPTION

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Abstract—A previous study presented a design method for compact plate–fin heat exchangers having flow passages with frequent interruptions. The local assumption was applied whereby the local boiling heat transfer coefficient was assumed to be fixed by the local metal-to-liquid ΔT. The present study extends the local assumption to include the effect of velocity on the local heat transfer coefficient. Confirmation of the new method is provided by a comparison of prediction and data for Freon-113 in an actual heat exchanger.

NOMENCLATURE

a,	arbitrary coefficient in equation (3);
h,	boiling heat transfer coefficient [kW/m^{2}K];
Nu,	Nusselt number for liquid [dimensionless];
Pr,	Prandtl number for liquid [dimensionless];
q,	heat flux [kW/m^2];
q_1, q_2,	heat flux at velocity V_1, V_2;
Q_T,	total heat duty [kW];
Re,	Reynolds number for liquid [dimensionless];
ΔT,	temperature difference from metal surface to boiling liquid [K];
ΔT_w,	base plate temperature minus boiling liquid temperature [K];
V,	fluid velocity [m/s];
V_1, V_2,	arbitrary fluid velocity [m/s];
V_{in},	inlet liquid velocity [m/s];
χ,	fluid quality based on mass [dimensionless].

BACKGROUND

A NEW METHOD was proposed in 1975 for predicting the heat transfer to a boiling liquid in a compact plate–fin heat exchanger [1]. This involved the use of the "local assumption" which had been shown previously to be successful for single fins [2–14], transverse fins outside tubes [15], and coated transverse fins [16]. Tests made with Freon-113 and with isopropyl in the compact heat exchanger were very encouraging. In particular, extremely high heat duties per unit volume, up to 101,000 kW/m^3, were achieved with the alcohol, and up to 76,000 kW/m^3 with Freon-113. We believe these are the largest heat duties per unit volume ever reported in this Journal, for convective or phase-change processes. These are impressive numbers and imply that compact heat exchangers are desirable for some boiling heat transfer applications. However, in the previous study it was found that the measured heat duties tended to be larger than the predicted duties. The present analysis was carried out to discover the reason for the too-good performance and to show how to design with improved accuracy.

BRIEF DESCRIPTION OF FIRST MODEL

One needs to know the geometry of the test heat exchanger in order to understand the method of predicting the boiling heat duty. A photograph of the exchanger and two sketches of the internal layout were published previously [1]. The aluminum compact plate–fin heat exchanger, having a core 8.25 × 8.25 × 7.87 cm, was supplied by AiResearch Manufacturing Co. and contained 22 parallel plates. It was tested as a thermosyphon reboiler. The test fluid flowed upward in a single pass through a double-stack arrangement. Steam flowed horizontally in two passes through a single-stack arrangement. Figure 1 indicates one plate and the fins which connect it to the next, parallel plate (not shown). All metal–metal contact surfaces were brazed. The fins in Fig. 1 were 0.101 mm thick, 3.88 mm high (plate-to-plate), and 4.67 mm long in the flow direction. The clearance between fins, 1.46 mm, was large enough to permit easy escape of bubbles [15, 17].

The flow path of the boiling liquid was not straight or smooth. At any cross-section in the exchanger, perpendicular to the flow of boiling liquid, the test fluid was in a set of 672 parallel flow channels each 3.78 × 1.46 mm in cross-section. The flow was split by the next row of staggered fins downstream every 4.67 mm. Well developed flow profiles and well developed boundary layers never built up. It appears hopeless to apply any of the many published 2-phase flow correlations which are intended for long, smooth, uninterrupted flow paths.

The first mathematical model [1] was the simplest. This Model I involved the assumption of one typical fin on the boiling side, acting independently of neighboring fins. The total heat duty was the one-fin duty

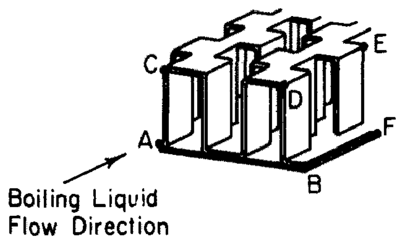

FIG. 1. Sketch of one plate and arrangement of the fin rows in the compact heat exchanger.

multiplied by 11,200 (the number of fins on the boiling side). The duty for the one fin was found by a 1-dim. conduction technique in the metal fin combined with the local assumption for the boiling heat transfer coefficient. Twenty-six nodes were assigned to the fin. For each node a heat balance was written. The local heat transfer coefficient h was taken to be determined only by the metal-to-liquid ΔT. Note that h is not uniform over the fin surface. The local h was taken from pool-boiling curves of h vs. ΔT previously published [2] for boiling outside 6.35 mm dia. horizontal copper tubes. A computer solution was carried out. One boundary condition was that the plate separating the two sides of the double-stack arrangement was an adiabatic boundary, by reasons of symmetry. Thus the plane containing CDE in Fig. 1 is adiabatic. On the opposite side of ABF steam condenses in finned passages, not shown in the sketch. The steam heat transfer coefficient is large, and its estimation is important as discussed previously [1]. Therefore, the plane ABF is assumed to be isothermal.

The first computational model was tested in the laboratory with isopropanol and with Freon-113. Figure 2 shows the predicted and experimental results for Freon-113. In this graph, ΔT_w is the temperature of the plate ABF (Fig. 1) minus the liquid saturation temperature. In general, Model I is too conservative. The actual heat duties are larger than expected. In fact, the measured maximum was 42% greater than the predicted maximum.

VELOCITY EFFECT

Original Model I required knowledge of the h vs. ΔT function to use with the local assumption. The function was assumed to be that obtained experimentally with a liquid pool having a velocity of essentially zero (pool boiling). No other recourse was reasonable at that time; complete boiling curves with forced-flow conditions were not available. The true condition inside the compact heat exchanger involved flow around fins.

To remedy the lack of information, flow–boiling curves were determined experimentally [18] using Freon-113 flowing at velocities up to 6.8 m/s across a 6.5 mm dia. horizontal tube with steam condensing inside. These experimental curves are reproduced as smooth lines in Figs. 3 and 4 and are labelled 0, 2.4, 4.0 and 6.8 m/s. In our practice a large family of interpolations and extrapolations were added to the graphs. Two of these added lines are shown in Figs. 3 and 4 for velocities of 1 and 10 m/s. The particular methods chosen for the interpolations and extrapolations are not crucial to the main point of this paper. The principal rules used were deliberately simple. They are roughly explained below. More details are available [19].

At very low ΔT values, forced flow without boiling exists, and h is constant as seen in Fig. 4. This region was fitted to the Fand–Keswani correlation [20],

$$Nu = (0.255 + 0.699\ Re^{0.5})\ Pr^{0.29}. \qquad (1)$$

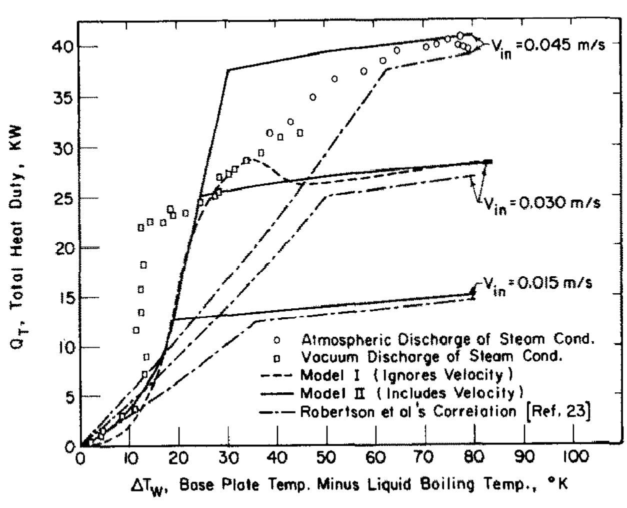

FIG. 2. Comparison of experimental data for boiling Freon-113 with design Models I and II. Model I used local assumption involving ΔT only. Model II involves the local velocity in addition to the local ΔT.

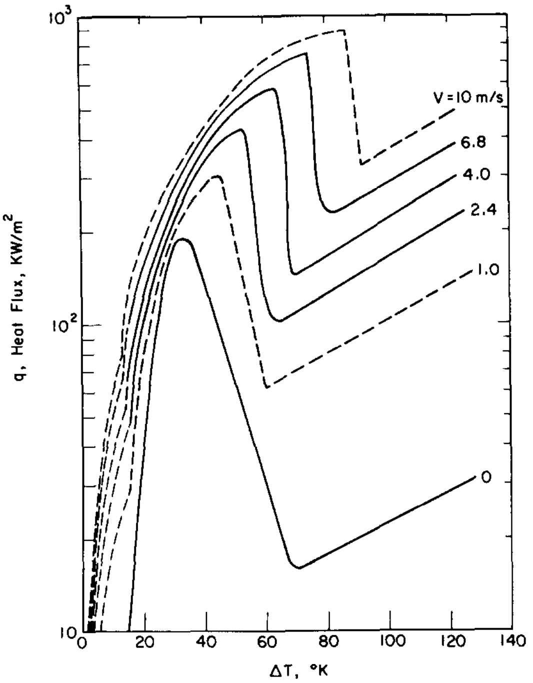

FIG. 3. Isothermal boiling curves used with the local assumption. Solid lines are experimental values for Freon-113 outside a steam-heated 6.5 mm dia. horizontal copper tube at atmospheric pressure [18].

For pool boiling at very low ΔT, the h is so small as to be immaterial, nevertheless a common formula [21] using Nu proportional to the $\frac{1}{4}$ power of the Grashof number times the Prandtl number was employed.

Nucleate boiling was assumed to start at the ΔT represented by the intersection of the forced-convection curves and the nucleate-boiling curves. Interpolations for the heat flux during nucleate boiling were defined by equation (2) for an arbitrary velocity V between two measured velocities V_1 and V_2

$$\frac{q_2 - q}{q_2 - q_1} = \frac{\sqrt{V_2} - \sqrt{V}}{\sqrt{V_2} - \sqrt{V_1}}. \qquad (2)$$

The nucleate boiling curve of q vs. ΔT was assumed to continue up to the locus for the peak fluxes (burn out) established in [18].

Film boiling was assumed to obey equation (3) below, taken from [18]

$$h = a\ V^{0.56}. \qquad (3)$$

This relationship was used for interpolation and extrapolation at given ΔT. The value of a varies slightly from one ΔT to another. The film boiling curve of q vs. ΔT was assumed to extend from high ΔT's down to the locus for the minimum heat flux (Leidenfrost point) established in [18].

The transition-boiling curve of q vs. ΔT was established by assuming a straight line on Fig. 3 between the maximum and minimum heat fluxes.

With velocity included as an important parameter, it was possible to recompute the expected performance of the compact heat exchanger. This Model II procedure definitely is an improvement. It is described in the following. First, the heat duty is calculated for a single fin in the first row of fins using the boiling heat transfer coefficient corresponding to the saturated liquid inlet velocity and to the local ΔT. This is a computer calculation with the fin divided into 39 nodes [19]. The velocity is the same at every node, but the ΔT is not. In other words, the h varies over the fin. The heat duty for the one fin is then multiplied by the number of fins in the row to get the heat duty for the row. This is used to compute the amount of vapor formed at the first row of fins.

The fluid, of quality somewhat higher than zero, now passes to the second row of fins. The liquid–vapor mixture is assumed to be homogeneous, and its velocity is greater in accordance with the known volume increase due to phase change at the first row. For a single fin in the second row of fins, the heat duty is calculated by the same procedure. This fin also has 39 nodes, and the same new velocity exists at every node. The ΔT is different at each node. Again a computer is used to select the proper distribution of h over the fin corresponding to the local ΔT and the new velocity. The heat duty for the second fin is then used to compute the amount of vapor formed at the second row of fins. The increase in fluid volume will result in a new, higher fluid velocity at the next row of fins.

The procedure is repeated from fin-row to fin-row until all 16 rows have been traversed. It is assumed that all 672 fins which are at one particular distance from the inlet are exactly alike. These are here referred to as a single fin row. The sum of the heat duties for all the rows is the predicted duty for the entire exchanger.

Figure 5 illustrates the change in fluid quality from row to row as the 16 rows are traversed. This

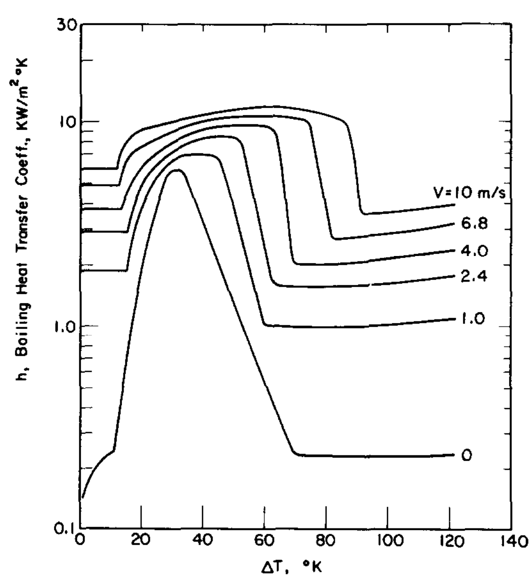

FIG. 4. Boiling heat transfer coefficients for isothermal surfaces. These were computed from Fig. 3.

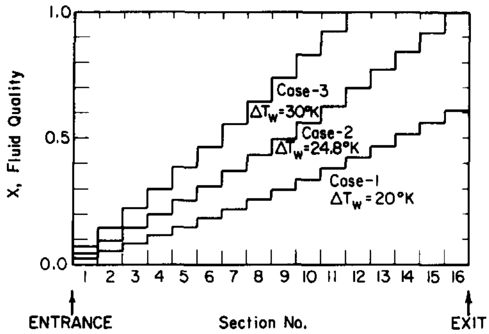

Fig. 5. Illustrative change in fluid quality from fin-row to fin-row. Inlet fluid is saturated liquid Freon-113 with a velocity of 0.03 m/s.

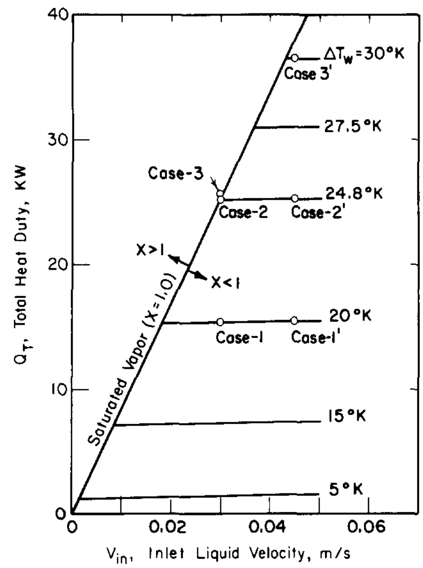

Fig. 6. Computed heat duty of exchanger for various base plate to liquid ΔT's as a function of the inlet liquid velocity.

graph is for saturated liquid entering the first row of fins at 0.03 m/s. Three cases are shown with different ΔT_w, the temperature difference from the base plate to the boiling liquid. Case 1 is for $\Delta T_w = 20$ K, and the exit fluid is partially vaporized. Case 2 for $\Delta T_w = 24.8$ K results in the exit fluid being saturated vapor. Case 3 for $\Delta T_w = 30$ K results in the fluid becoming completely vaporized at the 12th row and slightly superheated thereafter. The velocity of the exit fluid for these three cases is 3.7, 6.1 and 6.2 m/s. The heat duty for the exchanger for the three cases is 15.3, 25.1 and 25.6 kW.

The effect of a change in velocity is very different for the three illustrative cases. For Case 3, a significant increase in inlet velocity causes a significant increase in heat duty. This occurs because liquid becomes available in fin-rows 12–16 which were dry previously. For Cases 1 and 2 the increase in heat duty is much smaller, because all fin rows were in contact with liquid already at the first velocity. Figure 6 illustrates this effect in a graph of total heat duty vs. inlet velocity with ΔT_w being a parameter. Cases 1', 2', and 3' show the result if the inlet velocity is increased by 50%, with ΔT_w unchanged. The change in heat duty is trivial from Case 1 to 1' and from 2 to 2'. However from Case 3 to 3' there is a 42% increase in the heat duty. Thus the inlet velocity is a weak variable provided all fins in the heat exchanger have a supply of liquid. The inlet velocity is a strong variable if some of the fins are starved for liquid and the exit fluid is all superheated vapor. If the latter case occurred in industry, a pump might prove of advantage.

Figure 2 illustrates the predicted velocity effect according to the Model II calculations described herein. The total heat duty is shown as a function of ΔT_w for three liquid inlet velocities: 0.015, 0.030 and 0.045 m/s. Recall that with a thermosyphon reboiler the inlet velocity is uncontrolled in the sense that it is a dependent variable fixed by the geometry, the physical properties, and the temperature difference. In particular, V_{in} changes as ΔT_w changes.

Figure 2 shows that for ΔT_w in the region of 10 K, good agreement exists between the measured and the predicted heat duties for liquid inlet velocities of roughly 0.015–0.045 m/s. For ΔT_w around 30 K, the inlet velocity would need to be about 0.030–0.045 m/s to produce the measured heat duty. For ΔT_w around 65 K, the inlet velocity would need to be about 0.045 m/s to produce the correct heat duty. This last corresponds to an exit velocity of 9.5 m/s and a quality of 1.0. The other cases correspond to a quality of 1.0 or less and to exit velocities smaller than 9.5 m/s. All of these inlet and outlet velocities are practical for and typical of reboilers.

The model II predictions in Fig. 2 show that the experimental heat duties can be explained better than before. Panitsidis *et al.* [1] used a design method which was too conservative in that the local boiling heat transfer coefficients were assumed to be independent of the fluid velocity. When the velocity effect is included, the highest observed heat duties become quite rational.

ROBERTSON'S BOILING HEAT TRANSFER CORRELATION

Recently [22, 23] reported experimental results with boiling liquid nitrogen and Freon-11. The flow was vertically upward through an electrically heated rectangular channel with an internal finning similar to that in Fig. 1. They developed a correlation. The boiling heat transfer coefficient was assumed to be independent of ΔT but to be a function of the local mass quality and the Reynolds number at the inlet. The wall-to-bulk temperature difference was quite small, less than 2K. In the present paper, Robertson's correlation [23] was used with Freon-113 to estimate the performance of the compact heat exchanger described in this paper. The computation procedures were the same as those of Model II except that the heat transfer coefficient was assumed to be uniform over a fin, and the concept of fin efficiency was used. Extrapolations of Robertson's data were necessary when $Re < 200$; ($V_{in} < 0.03$ m/s). Three predictions by this method with $V_{in} = 0.015$, 0.030 and 0.045 m/s are shown in Fig. 2, labelled 'Robertson'.

At low ΔT_w (< 10 K), the predictions by the two methods are close. At intermediate ΔT (~ 25 K), Robertson's predictions are only about half the measured values. For Robertson's correlation to give a good fit in this region, the inlet velocity would have to be about 1.0 m/s, which is far in excess of the velocities actually used. This value also is far out of the velocity range of the thermosyphon reboiler described in the present study. Furthermore, this V_{in} causes large discrepancies in the heat duty at both lower and higher ΔT's The superiority of Model II appears in this intermediate ΔT region. At some high ΔT the predictions by Robertson's method and Method II come close again as expected. The inlet liquid flow will be all evaporated ($\chi = 1.0$) eventually as ΔT increases. The difference in heat duty will arise only because of the different small amount of sensible heat received by the superheated vapor. Since Model II predicts an earlier dryout with the same V_{in}, the total heat duty of the compact heat exchanger is a little higher.

It is not possible to test Robertson's data with the new method described in this paper until such time as complete flow boiling curves for Freon-11 or liquid nitrogen become available.

OTHER FACTORS

Three other factors which might account for a too-conservative prediction of the heat duty were identified during this study. Each is discussed briefly below.

Arrangement of nodes for computer attack. The original Model I used 26 nodes in one fin. The detailed arrangement is given in [1]. Later models were tried [24] using from 10 to 40 nodes/fin. The original 26-node scheme was found to be adequate. Three models were tried in which 2-dim. conduction inside the fin was accounted for. These produced insignificant changes in the heat duty. This is expected intuitively, because the fin has a slenderness ratio of 38 : 1 and the pitch of the fins is about 15 times the fin thickness. It is concluded that a 1-dim. model with 26 nodes is satisfactory, as used with the Model I calculations. Nevertheless 39 nodes were used throughout the Model II calculations for this paper.

Estimation of the steam-side conductance. The original computational Model I assumed that the fins on the steam side could be ignored and that a constant heat transfer coefficient for the condensing steam was applicable for each steam pressure [1]. Subsequently another model was studied [24] using as the geometric unit a 3-dim. unit cell containing one fin on the boiling side and the associated fins on the condensing side. The condensing side fins were 0.15 mm thick, 2.54 mm high and 1.27 mm long in the flow direction. The nodal arrangement for computer study had 48 nodes on the boiling side and 30 nodes on the condensing side. The results showed a small improvement in the predicted heat duties, but enough to justify the added computational effort.

The separating plate ABF of Fig. 1 was found to be near isothermal as originally assumed. The reason for this is that the condensation heat transfer coefficient for steam is about 17kW/m^{2}K. Over most of the boiling-side fin, the heat transfer coefficient for Freon-113 is generally much smaller as can be seen in Fig. 4. The separating plate will not be isothermal when the boiling liquid and the condensing vapor have similar heat transfer coefficients. Such a case can be computed, but the effort needed is far greater (particularly for a cross-flow exchanger) than when the condensing coefficient prevails. For the steam/Freon system used in the present study, the 3-dim. model is not needed.

Effect of heater size on boiling curve. The boiling curves displayed in Figs. 3 and 4 were obtained for isothermal surfaces which were the outer area of horizontal copper tubes of 6.5 mm dia. The curves were assumed to apply to the compact heat exchanger having hot fins with rectangular frontal dimensions of 3.88×0.101 mm and a length of 4.67 mm in the flow direction. The only direct way to test the equivalence (or non-equivalence) of the boiling heat transfer coefficients for surfaces as different as these would be to carry out an investigation of h vs. ΔT with forced flow for an isothermal ribbon 0.101 mm thick. Keep in mind that the entire boiling curve including transition boiling and film boiling, is needed. Such complete data do not exist at present. An alternative approach would be to use steam-heated tubes with diameters less than 6.5 mm to discover whether the size effect is important.

CONCLUSIONS

(1) For a compact plate–fin heat exchanger having flow passages which are split numerous times per centimetre along the flow direction, the heat duty may be predicted by an improved technique.

(2) For each fin, the total assumption is made: the boiling heat transfer coefficient varies over the fin surface, and its local value is determined by the local ΔT and local velocity. The boiling 2-phase mixture is assumed homogeneous.

(3) The technique is shown to be successful for boiling Freon-113 in a steam-heated compact exchanger used as a thermosyphon reboiler.

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PREVISION DU TRANSFERT THERMIQUE PAR EBULLITION DANS UN ECHANGEUR DE CHALEUR A AILETTES PLANES, A L'AIDE D'UNE HYPOTHESE LOCALE AMELIOREE

Résumé—Une étude antérieure a présenté une méthode dans le cas des échangeurs de chaleur compacts à ailettes planes avec des passages de fluide ayant de fréquentes interruptions. L'hypothèse locale était que le coefficient de transfert était fixé par le ΔT local entre le métal et le liquide. L'étude actuelle élargie l'hypothèse en incluant l'effet de la vitesse sur le coefficient de transfert thermique local. Une confirmation de la nouvelle méthode est fournie par une comparaison du calcul et des données obtenues avec du Freon 113 dans un échangeur de chaleur.

BERECHNUNG DES WÄRMEÜBERGANGS BEI VERDAMPFUNG IN EINEM KOMPAKTEN LAMELLEN-WÄRMEAUSTAUSCHER UNTER VERWENDUNG DER VERFEINERTEN ÖRTLICHEN BETRACHTUNGSWEISE

Zusammenfassung—In einer vorausgegangenen Studie wurde ein Auslegungsverfahren für kompakte Lamellen-Wärmeaustauscher mit häufig unterbrochenen Strömungskanälen beschrieben. Hierbei wurde die örtliche Betrachtungsweise angewandt, in welcher angenommen wird, daß der örtliche Verdampfungs-Wärmeübergangskoeffizient durch die örtliche Temperaturdifferenz ΔT zwischen Metall und Flüssigkeit festgelegt wird. Die vorliegende Studie erweitert die örtliche Betrachtungsweise um den Einfluß der Geschwindigkeit auf den örtlichen Wärmeübergangskoeffizienten. Die Bestätigung des neuen Verfahrens liefert der Vergleich von Rechnung und Messungen an einem ausgeführten Wärmeaustauscher mit dem Kältemittel R 113.

РАСЧЁТ ТЕПЛОВОЙ НАГРУЗКИ ПРИ КИПЕНИИ ДЛЯ КОМПАКТНОГО ПЛАСТИНЧАТО-РЁБЕРНОГО ТЕПЛООБМЕННИКА

Аннотация — В предыдущей работе авторов был предложен метод расчёта компактных пластинчато-рёберных теплообменников при запирании каналов, основанный на использовании допущения, в соответствии с которым предполагается, что локальный коэффициент теплообмена при кипении определяется локальной разностью температур металла и жидкости, ΔT. В предлагаемом исследовании метод несколько модифицирован за счёт дополнительного учёта влияния скорости потока на локальный коэффициент теплообмена. Проверка метода проведена на основе сравнения расчётных значений с экспериментальными данными, полученными при использовании в теплообменнике фреона-113.