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BOILING OF LIQUIDS IN A COMPACT PLATE-FIN HEAT EXCHANGER

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Abstract—Prior to this study, no logical design method has existed for boiling heat transfer in compact, plate-fin, heat exchangers. This paper shows that a single-fin analysis can be used to predict the performance. Confirming tests were made with boiling Freon-113 and isopropanol heated by condensing steam, using a commercial compact heat exchanger having an 8·25 × 8·25 × 7·87 cm core which contained 11 200 fins on the boiling side and 23 660 fins on the condensing side. The flow passages were 3·78 × 1·46 mm on the boiling side and 2·39 × 1·43 mm on the condensing side.

NOMENCLATURE

$A_{0,1}, A_{1,3}, A_{5,6}, A_{6,7}$, area normal to conduction path from node 0 to 1, 1 to 3, 5 to 6, 6 to 7;
h, h_1, h_6, local heat-transfer coefficient, for node 1, for node 6;
k, thermal conductivity of solid;
$L_{0,1}, L_{1,3}, L_{5,6}, L_{6,7}$, length of conduction path from node 0 to 1, 1 to 3, 5 to 6, 6 to 7;
ΔP, pressure drop;
Q_T, total heat duty;
S_1, S_6, convective area of node 1, of node 6;
T_L, saturation temperature of the liquid;
$T_0, T_1, T_3, T_5, T_6, T_7$, temperature at node 0, 1, 3, 5, 6, 7;
T_{av}, average of steam inlet and outlet temperatures;
ΔT, temperature difference from metal surface to boiling liquid;
W, steam flow rate.

BACKGROUND

THE CATEGORY of compact heat exchangers usually means those having heat-transfer areas greater than 100 ft^2 per ft^3 of exchanger volume (328 m^2/m^3). They have very small flow passages and large numbers of fins. The most common application is in the aircraft industry, for transferring heat between clean gases and liquids, with no change of phase. The purpose of this communication is to show that they can be used also for boiling and condensation of nonfouling fluids. In particular, a design method for boiling liquids is proposed, and data are presented to show that the method is rational.

EXPERIMENTAL

The small heat exchanger in Fig. 1 is a standard, off-the-shelf, aluminum, compact exchanger intended for aircraft applications. It was supplied by the AiResearch Manufacturing Co. The core was 8·25 cm long in each cross-flow direction and 7·87 cm in the nonflow direction. In the present investigation it was used to boil Freon-113 (CCl_2F–$CClF_2$) and isopropanol ($CH_3CH_2OHCH_3$) at atmospheric pressure with the heat source being steam condensing at absolute pressures between 13 and 563 kN/m^2. With the available steam supply, the maximum heat duty obtained was 40 700 W for Freon-113 and 54 250 W for isopropanol. The corresponding overall temperature drops, steam-to-fluid, were 108 and 69°C. A stainless steel condenser of tube and shell design was used to condense the test vapors. This condenser is the large item in Fig. 1. These two exchangers of equal heat duty had an area ratio of 14 to 1, a volume ratio of 58 to 1, and a weight ratio of 102 to 1, with the small number referring to the compact exchanger in every case. The aluminum compact exchanger sells at about one-third the price of the stainless steel condenser.

Figure 2 indicates the internal geometry of the repeating units in the compact heat exchanger. It contained vertical parallel plates connected by fins. The test fluid flowed upward in a single pass through a double-stack arrangement, and the steam flowed horizontally in two passes through a single-stack arrangement. The fins were very short parallel to the flow direction, and each following fin was staggered to split the flow and enhance the heat transfer. The total number of fins in the assembly was 34 860. They were constructed of ribbons, folded in a rectangular wave with 90 degree bends and brazed at all contacts. The repeating unit containing one fin on the boiling

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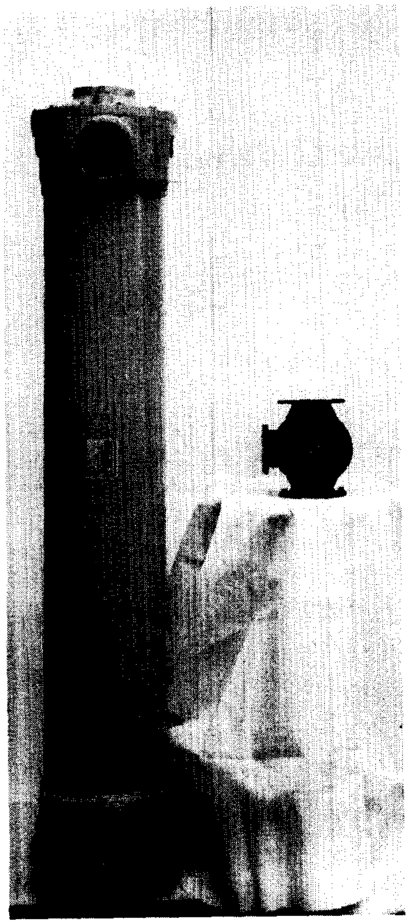

FIG. 1. Comparison of compact heat exchanger (right) used as a boiler and a conventional tube-and-shell condenser (left) of equal heat duties.

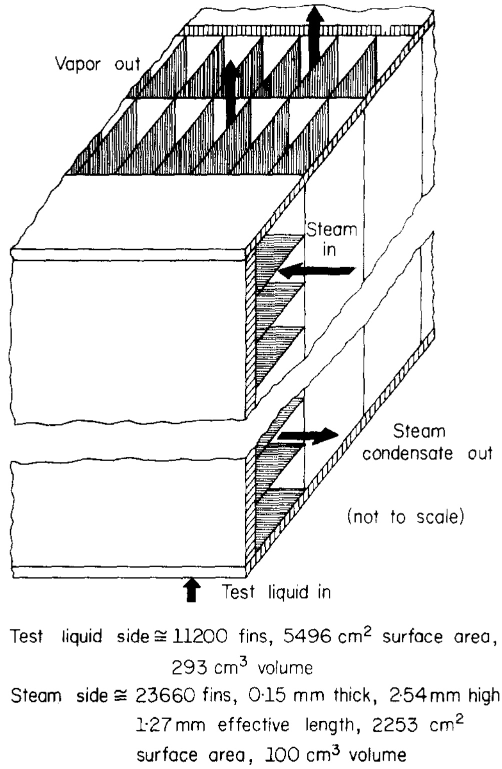

FIG. 2. Overall specifications and cutaway view of double-stack array on boiling side and single-stack array on steam side.

side is shown in Fig. 3. The pertinent dimensions are shown. The test fluid was in channels measuring 3·78 × 1·46 mm, interrupted downstream every 4·67 mm by the next row of fins.

For the tests, the compact heat exchanger was attached in thermosyphon fashion to the side of a stainless steel tank containing about 130 l of test liquid. The steam was condensed and metered for most runs. The steam pressures and temperature, in and out, were always measured. The vapors formed from the test liquid were condensed, metered and returned to the boiler. In addition to these two methods of measuring the heat duty, the heat absorbed by the condenser water was measured by flow rate and temperature rise. Usually the three heat duties agreed within a few per cent. The test liquids were essentially at their normal boiling points of 47·7°C for Freon-113 and 82·3°C for isopropanol, as shown by thermocouples in the boiler. Complete details of the equipment and procedure are available [1,2].

It is believed that only saturated test liquid (no vapor) entered the exchanger. The exit fluid was a liquid–vapor mixture of unknown quality. If the exit fluid had 100 per cent quality, the range of exit velocities in the flow passage was from 0·04 to 11 m/s for Freon-113 and from 0·04 to 9·7 m/s for isopropanol. The true velocities were greater.

The entering steam was saturated or near saturation. The inlet steam velocities in the flow passages ranged from 1·3 to 10·7 m/s for Freon-113 and from 0·6 to 16·5 m/s for isopropanol. The steam condensate exited from the flow passages at velocities less than 0·04 m/s.

Figure 4 summarizes the test data. The range of heat duties is from 163 to 54 200 W, and the mean value of the in-and-out steam temperatures varied from 50 to 150°C. Tests made with steam hotter than 100°C are denoted as Series B in the graphs. Series A denotes tests made with the steam system connected to a vacuum so that the steam temperature was below 100°C. For Series B, the steam flow as well as its pressure drop in the exchanger was measured. These are shown in Fig. 4. For many commercial considerations this ΔP of roughly 2 lb/in^2 for steam is not large. This shows that steam

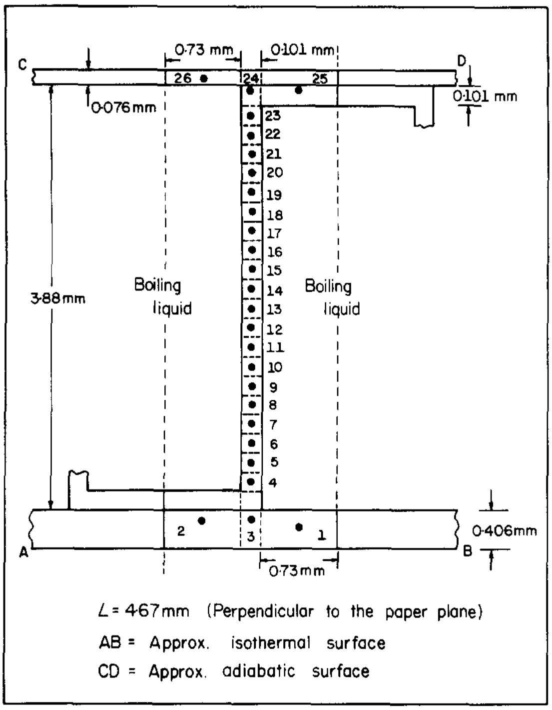

FIG. 3. Specifications for one symmetry unit on the boiling side showing the 26-node distribution for mathematical analysis

can be condensed with satisfactory performance in flow passages as small as 2·39 × 1·49 mm, interrupted downstream every 3·2 mm by the next row of fins. The ΔP on the boiling side was not measured. From the known liquid level in the boiler and the known position of the compact heat exchanger, it was estimated that the boiling side pressure drop was less than 3 kN/m^2 (about 0·5 lb/in^2). The large heat duties and the reasonable pressure drops obtained in the compact heat exchanger indicate that such equipment is attractive for boiling service, provided some means of prediction is available. Of course, with flow passages as small as used herein, the usage must be restricted to nonfouling liquids such as Freon, cryogenic liquids, and water of high purity.

FIRST MODEL ANALYSIS

The first mathematical model is the simplest. It involves the assumption of a typical fin on the boiling side, acting independently of neighboring fins. The total heat duty of the entire exchanger is the one-fin duty multiplied by 11 200 (the number of fins on the boiling side). Prior studies [5,6] of fin spacing in boiling Freon-113 have shown that fins do act independently when the clearance is at least 1·6 mm; interference resulted when the clearance was 0·8 mm. The present exchanger had a clearance of 1·46 mm.

Figure 3 illustrates the single fin and associated plates. The boundary CD separates the two sides of the double stack arrangement on the boiling side. It is assumed to be adiabatic by reason of symmetry. Boundary AB is attached to the fins on the steam side. It is assumed to be isothermal because of the large heat-transfer coefficient on the steam side. In other words, the fins on the steam side are ignored for the heat-transfer calculations of this model. The film of condensed steam contacts surface AB, and an important part of the problem is to estimate the heat-transfer coefficient for this condensate film.

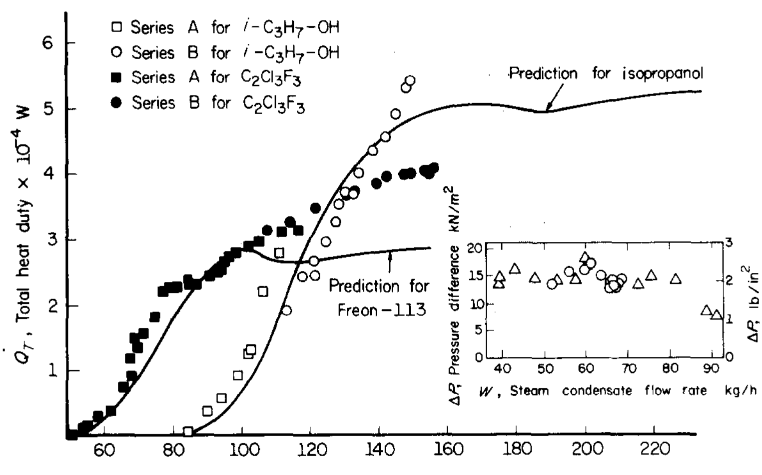

FIG. 4. Boiling heat-transfer prediction and measured performance of the compact heat exchanger. A constant steam film coefficient was used here for calculating the wall temperatures used for the predictions.

As shown in Fig. 3, the fin and walls are represented by 26 nodes, and a simple one-dimensional conduction technique was used. Twenty-six heat balances were written, one for each node. To illustrate, for node 1 the balance is equation (1); for node 6, it is equation (2).

$$\frac{kA_{0,1}}{L_{0,1}}(T_0-T_1) = \frac{kA_{1,3}}{L_{1,3}}(T_1-T_3) + h_1S_1(T_1-T_L) \qquad (1)$$

$$\frac{kA_{5,6}}{L_{5,6}}(T_5-T_6) = \frac{kA_{6,7}}{L_{6,7}}(T_6-T_7) + h_6S_6(T_6-T_L). \qquad (2)$$

The local heat-transfer coefficient, h, on the boiling surface was not constant. The "local assumption," which has resulted in success for other cases of boiling on fins [3–17], was used here. Namely, the functionality between the local h and the local ΔT is identical with that for large isothermal surfaces at the same ΔT. Here ΔT means the temperature difference between the solid surface and the surrounding liquid. The particular functionalities available for this study are shown in Fig. 6. These curves were obtained with liquids in pool boiling outside 6·35 mm dia. horizontal, steam-heated copper tubes [3].

A temperature T_0 was selected for the surface AB. Then an iterative technique was used on an IBM-1800 digital computer to solve for the 26 temperatures to within 0·006°C. The heat rejected to the boiling liquid was summed for the 26 nodes to give the heat duty per fin. Figure 5 shows the resulting predictions for the total heat duty of the exchanger as a function of the temperature elevation of the wall at AB. Actually, nodes 1, 2, and 3 were nearly at the same temperature as the boundary AB over most of the range of Fig. 5. The curves show a main branch (left side of the graph), then a long plateau region with practically no change in heat duty with change in ΔT, and then a film boiling region characterized by low heat duties and high ΔT.

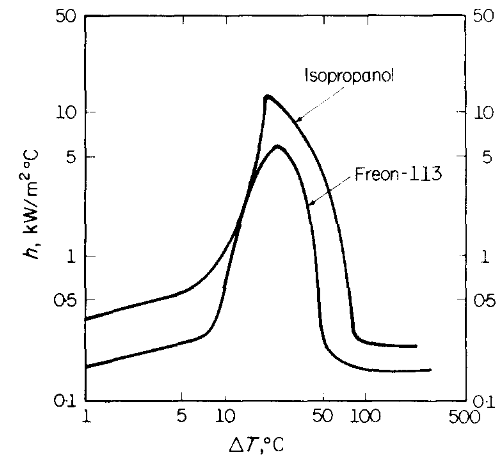

FIG. 6. Boiling curves used for prediction of fin performance. These were obtained for liquids outside a steam-heated 6·35 mm dia. horizontal copper tube at atmospheric pressure [3].

The wall temperature could not be measured in the commercial exchanger. It was computed by subtracting the temperature drop across the steam-condensate film from the steam temperature. This steam film ΔT was found by use of Nusselt's equation. The condensation occurred in rectangular passages, between fins. An equivalent diameter was calculated equal to four times the cross sectional area divided by the *heated* perimeter (not the entire wetted perimeter). This diameter was 4·78 mm, the flow path was 3·2 mm from one row of fins

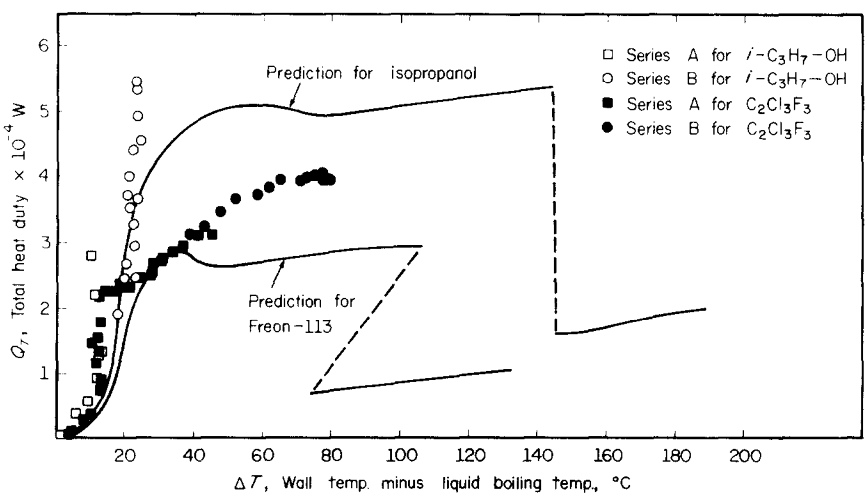

FIG. 5. Prediction vs experimental data based on wall temperature (at AB in Fig. 3). The steam film coefficients used with the experimental measurements here were calculated from the Nusselt equation.

to the next offset row, and the pass length was 8·25 cm before the 180 degree turn. These are unusual conditions for the use of an equation intended for condensation outside straight round tubes. A trial-and-error computation was needed to achieve equal heat transfer on the steam side and the boiling side. For the present tests, Nusselt's formula gave a range of condensing steam film coefficients from 13 200 to 60 000 W/m^2 °K. These were computed at the mean pressure between steam inlet and outlet. Figure 5 shows the experimental data, with the wall temperature computed via Nusselt's formula. Agreement is excellent at a heat duty of about 25 000 W. At this duty the steam coefficient from Nusselt's equation is 16 300 W/m^2 °K for the tests with Freon-113 and 17 020 for the isopropanol tests. At the maximum possible duty which could be achieved with the laboratory steam at full pressure, the boiling duty for Freon-113 was 42 per cent above the predicted value. The measured maximum for isopropanol was 8 per cent higher than predicted.

Whereas in Fig. 5 the abscissa was "as-is" for the predicted curves, it was "corrected" by Nusselt's formula for the experimental data. In Fig. 4, the abscissa is "as-is" for the experimental data, but it is "corrected" for the predicted curves. Here a constant condensing film coefficient of 16 400 W/m^2 °K for the steam side was assumed for the Freon-113 tests. A constant steam coefficient of 18 700 was used for the isopropanol tests. These are arbitrary values, chosen to give good agreement between tests and predictions, particularly in the range of heat duties near 25 000 W. The agreement is good over a wider range of abscissa values in Fig. 4 than it is in Fig. 5. This indicates that Nusselt's equation does not give reliable values for the steam film in the small finned passages, and it is better to use a single representative coefficient for the steam side. For the present geometry, an average steam-side heat-transfer coefficient of 17 500 W/m^2 °K is recommended.

DISCUSSION

Figure 4 shows encouraging agreement between data and predictions for the main branch of the curves up to the predicted peak duty for each liquid. However, the overshoot of the data beyond these heat duties requires explanation. The mathematical model is undoubtedly oversimplified. Some improvement could be achieved by using more nodes per fin, by computing the performance of additional fins (such as at the entrance and exit conditions) and using some integrating technique, and by seeking for ways better to predict the steam film coefficient. However, the most likely deficiency lies in the choice of the local h vs ΔT, Fig. 6. No other data were available; therefore, these pool boiling curves for 6·35 mm dia. were used. They are deficient, because they definitely omit velocity effects. In addition it is questionable whether they fit adequately for geometries as small as the fins considered here.

The effect of velocity on the maximum boiling duty for Freon-113 at a pressure of 483 kN/m^2 has been reported [18]. At a mass velocity of 407 kg/s m^2 across a bundle of 10 mm dia. tubes, the burn-out heat duty was double the duty observed during pool boiling. For the present tests, the mass velocity for Freon-113 at its highest heat duty was 72 kg/s m^2, calculated on the basis of 100 per cent quality. If the velocity effect is similar for these two different geometries, it would cause an increase of 18 per cent in the heat duty. Although precision is impossible in such a calculation, the numbers suggest that the velocity effect is significant. No data are known for evaluating the velocity effect for isopropanol.

The effect of size of the heater element on the maximum boiling duty has been studied by a few workers. For example, Ornatskiy [19] used sub-cooled water flowing at 10^4 kg/s m^2 inside tubes of various diameters. He found that the maximum heat flux was 56 per cent larger for a 2·13 mm tube compared to a 6·35 mm tube. The larger size is the outside diameter of the tubes used to establish the h vs ΔT functionality in Fig. 6. The smaller size is the equivalent inside diameter of the flow passages in the present compact heat exchanger. It is not known how to make a fair comparison for different heater sizes when the geometrical configurations are as different as these, but the evidence does imply an effect of size.

CONCLUSIONS

The encouraging aspect must be emphasized. A one-fin design technique gives surprisingly good results when applied to an assembly of thousands of fins. Commercial, compact heat exchangers can be used successfully for boiling clean, nonfouling liquids. The heat duties are large on an area basis and truly impressive on a volume basis. For the present tests with isopropanol, the best values were 99 kW/m^2 of boiling-side area and 74×10^3 kW/m^3 of total exchanger volume (31 300 Btu/h ft^2 and $7{\cdot}22 \times 10^6$ Btu/h ft^3). Higher values can be obtained by optimization of the fin dimensions.

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EBULLITION DE LIQUIDES DANS UN ECHANGEUR DE CHALEUR COMPACT A AILETTES PLANES

Résumé—Avant cette étude, il n'existait pas de méthode logique dans la conception des échangeurs de chaleur compacts à ailettes planes. Cet article montre que l'analyse d'une ailette unique peut être utilisée pour prévoir les performances. Des essais ont été conduits pour du Fréon 113 et de l'isopropanol, chauffés à l'ébullition par de la vapeur en condensation, en utilisant un échangeur compact commercialisé, de 8,25 × 8,25 × 7,87 cm, contenant 11.200 ailettes côté ébullition et 23.660 ailettes côté condensation. Les sections de passage sont 3,78 × 1,46 mm, côté ébullition et 2,39 × 1,43 mm du côté condensation.

DAS SIEDEN VON FLÜSSIGKEITEN IN KOMPAKTWÄRMEAUSTAUSCHERN MIT PLATTENRIPPEN

Zusammenfassung—Bislang lag keine physikalisch begründete Berechnungsmethode für den Wärmeübergang beim Sieden in Kompaktwärmeaustauschern mit Plattenrippen vor. Diese Arbeit zeigt, daß man bei der Berechnung von den Verhältnissen an einer einzelnen Rippen ausgehen kann. Dies wurde durch Versuche mit verdampfendem Freon-113 und Isopropanol bestätigt, wobei die Wärmezufuhr durch kondensierenden Dampf erfolgte. Die Versuche wurden mit einem handelsüblichen Kompaktwärmeaustauscher (8,25 × 8,25 × 7,87 cm) durchgeführt, der auf der Verdampfungsseite 11 200 Rippen und auf der Kondensatseite 23 660 Rippen aufwies. Der freie Strömungsquerschnitt betrug 3,78 × 1,46 mm auf der Verdampfungsseite und 2,39 × 1,43 mm auf der Kondensatseite.

КИПЕНИЕ ЖИДКОСТЕЙ В КОМПАКТНОМ ПЛАСТИНЧАТОМ ОРЕБРЕННОМ ТЕПЛООБМЕННИКЕ

Аннотация — До сих пор не существовало последовательного метода расчета компактных пластинчатых теплообменников с кипящим теплоносителем. Данная работа показывает, что анализ, проведенный для одного ребра, применим для расчета рабочего режима всей установки. Для проверки были проведены эксперименты по кипению Фреона-113 и изопропанола, нагреваемыми конденсирующимся водным паром в компактном теплообменнике промышленного типа с внутренними размерами 8,25 × 8,25 × 7,87 см и 11 200 ребрами на стороне кипения и 23 600 ребрами на стороне конденсации.