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BOILING ON A FINNED TUBE AND A FINNED TUBE BUNDLE

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Abstract—An experimental investigation was carried out in a 151 dm^3 container in R 11, at saturation state of 1 bar and 23.3°C. The heat transfer coefficients measured on a single tube in the container, resembled approximately results for the case that only one tube is heated within an in-line 18 tube bundle.

For a twin tube arrangement, the heat transfer coefficient of the upper tube is up to 80% higher than that of the lower tube. The heat transfer curve of this upper tube represents quite well the heat transfer coefficients of tubes enclosed in the bundle. Only for the bottom and top tubes of the bundle differences do occur. The possibility to obtain tube bundle data, at least to some extent, by much simpler twin-tube or triple-tube experiments would facilitate experimental investigations.

NOMENCLATURE

- A, surface area;
- C, constant;
- d, diameter;
- h, mean heat transfer coefficient;
- k, thermal conductivity;
- L, length;
- $\dot{Q}$, heat flow;
- $\dot{q}$, heat flow density, $\dot{Q}/A$;
- s, tube pitch.

Greek symbol
- ϑ, temperature.

Subscripts
- w, wall;
- o, outside;
- i, inside;
- v, vapour;
- l, liquid;
- n, number of tube, row;
- m, mean.

1. INTRODUCTION

IN MODERN tube bundle evaporators, especially those in refrigeration machinery, finned tubes are widely used. The performance of fins in various shapes for boiling heat transfer has been investigated in [1–7]. The behaviour of single finned tubes and effects of their geometric parameters are presented in [8–10]. In experiments with bundles of finned tubes [11–16] it was shown that convection currents induce a remarkable enhancement in boiling heat transfer. Heat transfer results differ due to tube geometries and bundle geometries. Based on a comparison of experiments of [13, 15, 16], heat transfer correlations were worked out [17–19]. In the recent efforts to produce more efficient heat transfer equipment, special kinds of "enhanced tubes" have been developed with experimental results reported in ref. [20] especially in contributions by Bergles; Yilmaz, Palen and J. Taborek; Stephan and Mitrovic.

For the range of parameters, there still are comparatively few experimental results for finned tube bundles; certainly not enough for widely applicable correlations to be made. The experiments, however, are quite delicate to perform and require a considerable amount of apparatus.

The present aim is to investigate (and provide additional experimental results on) the extent to which the data for single or twin tubes are applicable to bundle heat transfer results. For this purpose, heat transfer coefficients are measured on a single tube, on two tubes positioned vertically above each other, and on an in-line bundle consisting of 6 horizontal rows of 3 tubes each.

2. EXPERIMENTAL SET-UP

The experiments were performed in R 11 ($CFCl_3$, monofluorotrichloromethane) at a pressure of about 1 bar and a saturation temperature around 23°C. The finned heater tubes were made of copper to the dimensions given in Fig. 1. All 18 tubes were produced identically and came from the same production batch.† The mean surface roughness of the tube was $R_p = 0.3$ μm. Each tube could be heated separately with an internal electric heater (3000 W at 220 V). The heated length was 260 ± 6 mm (according to the manufacturer). In order to equalize differences in the electric heater temperature, a copper filler tube was fitted inside the finned tube and then expanded. The electric heater was soldered into the filler tube. The homogeneous soldering contact and the pressure fit was checked by cutting a sample tube on a lathe. The ends of the tubes were insulated with Teflon shrouds.

The tubes were arranged in a rectangular container in three experimental variations as shown in Fig. 2. In

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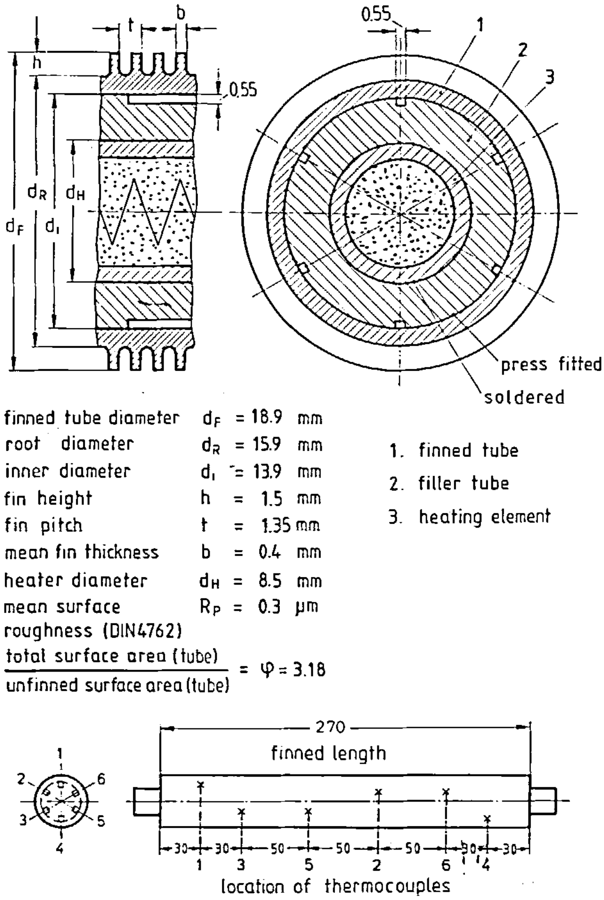

FIG. 1. Finned tube heater geometry.

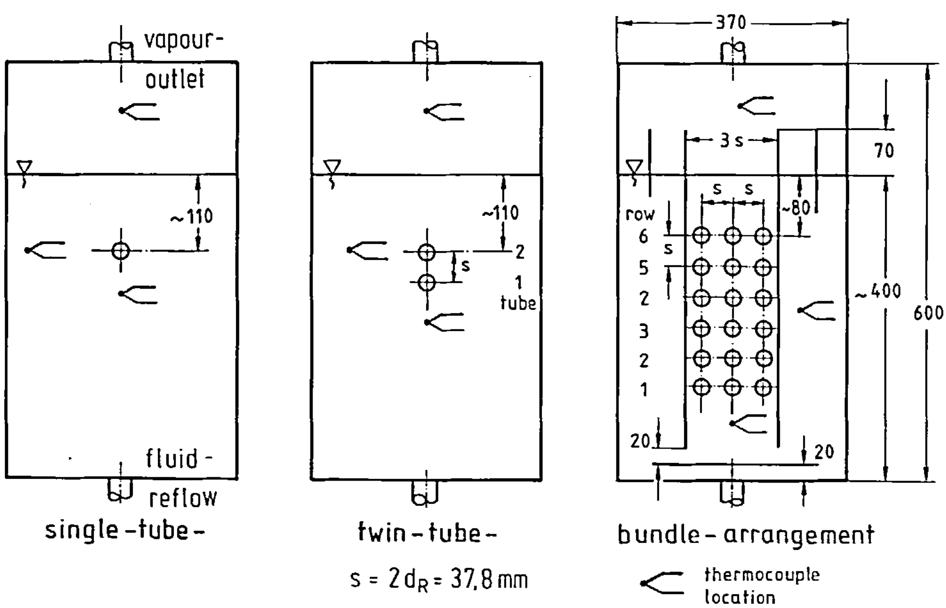

FIG. 2. Finned tube arrangements and thermocouple locations inside the test vessel.

order to prevent lateral flow, and to simulate a fraction of a larger tube bundle, the 18 tubes were placed between glass plates, their upper ends extend out of the fluid by about 70 mm in order to help prevent splashing. Additional plates were inserted for the same reason.

The container was made from stainless steel and measured 680 × 600 × 370 mm. Three sides (the 680 × 600 mm front and back, and one 370 × 600 mm side) were equipped with glass covers to allow observation. The vapour produced in the test vessel was condensed in a separate, controllable condenser and fed back through a thermostatic pre-heater into the test vessel. The entire test apparatus was set up in a temperature-controlled compartment with air temperatures adjusted to the saturation temperature inside the vessel.

3. MEASUREMENTS

3.1. *Temperatures*

Temperatures were measured with coated Philips NiCr–Ni thermocouples (0.5 mm O. diameter). The heater inside wall temperatures were taken at six different points along the heater length and around the circumference as shown in Fig. 1.

The outside wall temperatures (around d_R) were calculated from

$$\vartheta_{wo} = \vartheta_{wi} - \left(\frac{\dot{Q}}{2\pi k L}\right) \ln \frac{d_R}{d_i}. \tag{1}$$

The arithmetic mean of six outside wall temperatures was taken as the representative wall temperature ϑ_w. The difference between ϑ_w and the temperature of the saturated vapour above the liquid, ϑ_v, is

$$\Delta\vartheta = \vartheta_w - \vartheta_v.$$

Fluid and vapour temperatures are measured at the locations shown in Fig. 2. In twin-tube and bundle-arrangements, the thermocouples inside the respective upper tubes were used to determine the temperature of the approaching liquid ϑ_{ln} (n indicates on-flow to tube or row 2,3,...). When these tubes were used for temperature measurement they were not electrically heated, of course. As an example, excess temperatures

$$\Delta\vartheta = \vartheta_l - \vartheta_v$$

and

$$\Delta\vartheta = \vartheta_{12} - \vartheta_v$$

for the twin-tube arrangement are shown in Fig. 3.

3.2. *Pressure*

The saturation pressure is measured above the liquid level with a pressure gauge of class 0.1 within the range 0–1.5 bar.

3.3. *Heating rate*

All tubes could be heated separately. Their heating power was measured individually with a power meter of class 0.1.

4. EXPERIMENTAL PROCEDURE

Special precautions were taken in order to provide reproducible results: the carefully cleaned vessel was flushed several times with R 11, evacuated and filled; all tubes after careful cleaning were placed in the vessel and heated for at least 50 h (at $\dot{q} = 4 \times 10^4$ W m^{-2}); all boiling experiments were performed under decreasing heat flow densities. In all experiments, the saturation pressure was kept at an average of $p_s = 1.000 \pm 0.005$

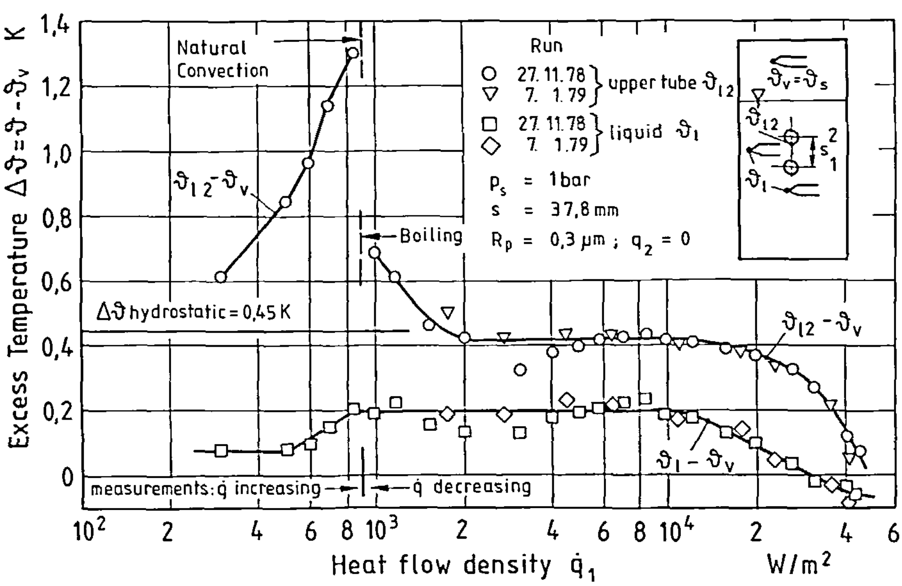

FIG. 3. Temperature differences between liquid and saturated vapour as a function of heat flow density $\dot{q}_1$ of the lower tube.

bar at a saturation temperature of $\vartheta_s = 23.31°C$. The measurements were repeated at intervals of several months or 1 year.

5. EVALUATION OF EXPERIMENTAL DATA

5.1. *Temperature distribution in the liquid*

The temperature in the liquid is not homogeneous. For twin-tubes, Fig. 3 shows the excess temperature in various places of the vessel versus the heat flow density of the lower tube.

For natural convection conditions, the temperature ϑ_l (of the liquid approaching the lower tube 1) is only slightly above the vapour temperature. It increases to values amounting to $\Delta\vartheta = 0.2$ K for most of the boiling region. With fully developed boiling at $\dot{q} = 10^4$ W m^{-2} the excess temperature drops to zero (the deviations into the negative range of Fig. 3 are probably due to subcooled condensate hitting the lower thermocouple).

The temperatures of the liquid approaching tube 2, i.e. ϑ_{12}, are quite different from ϑ_1: in the natural convection regime there is a steep rise, for incipient boiling a decline is followed by a constant excess temperature. This excess temperature agrees very well with the theoretical excess temperature of $\Delta\vartheta = 0.45$ K obtained from the hydrostatic head above tube 2. Again, for fully developed boiling $\Delta\vartheta$ drops to zero.

The temperature distribution within an 18-tube bundle, with the bottom row heated, is shown in Fig. 4. The excess temperatures range from 0.7 to 0.5 and gradually decrease with increasing heat flow density $\dot{q}_1$. Because of mixing and heat losses, as one would expect, the excess temperatures decrease upwards from row to row. The same kind of temperature distributions are obtained when the lower two or three rows are heated. This is shown in Fig. 5 for a heat flow density of $\dot{q}_n = \dot{q} = 10^4$ W m^{-2}. For comparison the distribution of the excess temperature according to the hydrostatic head $\Delta\vartheta_{\text{hydrostatic}}$ is given for the various tube positions. Some agreement with the data measured is reached only around rows 4 and 5.

5.2. *Reference temperature and characteristic difference*

In boiling it is customary to use the difference between the heater wall temperature and the saturation temperature of the fluid as the driving potential for heat transfer. In the tube bundle the theoretical saturation temperature changes from one tube row to the next and does not agree with the actually measured values. These latter values indicate a liquid subcooling around rows 1 and 2 and a superheating around rows 5 and 6.

In natural convection the on-flow temperature is considered characteristic to form the temperature difference for the heat transfer coefficient. In our experiments these on-flow temperatures were measured when the tube hit by the approaching fluid was not heated. If the tube is heated, the increased on-flow might cause changes in the on-flow temperature, so that our data of the unheated case may deviate from such of the heated case. At any rate, approaching-flow temperatures are difficult to measure and control.

The simplest reference temperature to obtain is the vapour temperature above the liquid level: as long as the vapour pressure is kept constant, it does not vary. A comparison between various tube arrangements is quite easy, but specific local results might be camouflaged. It was tried to correlate results on the

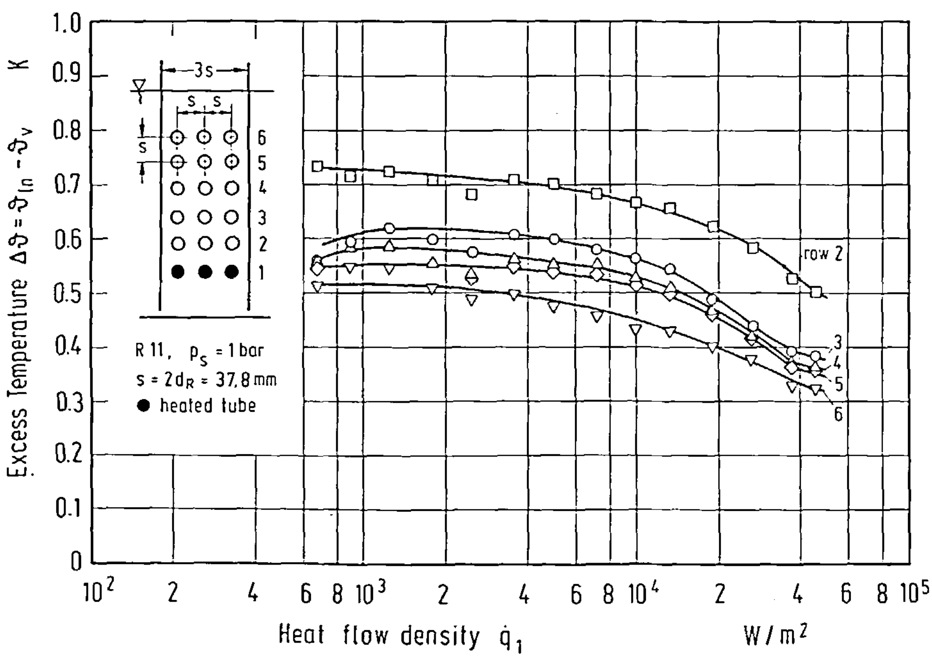

FIG. 4. Excess temperature in the bundle as a function of heat flow density.

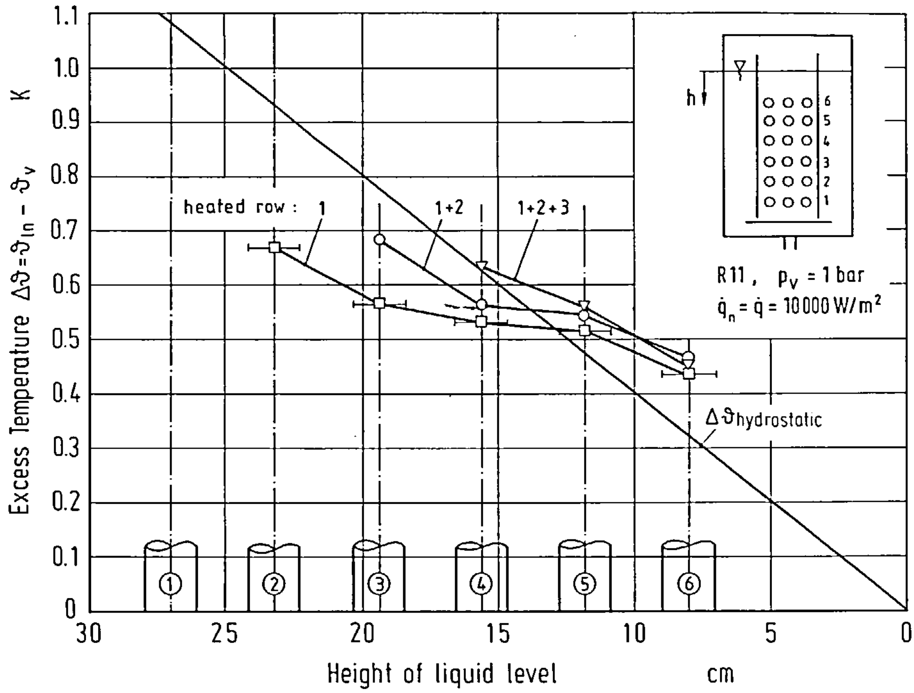

FIG. 5. Excess temperature in the tube bundle.

basis of each of the above mentioned temperatures, $\vartheta_{s\,local}$, ϑ_{ln} and ϑ_v; best results were obtained with ϑ_v. Therefore the characteristic temperature difference is taken as

$$\Delta\vartheta = \vartheta_{wall} - \vartheta_{vapour}.$$

(Any other differences may be obtained from Figs. 3–5.)

5.3. *Heat transfer coefficients and correlations*

From measured data the mean heat transfer coefficient is calculated using

$$h = \dot{Q}/A_{total}(\vartheta_{w,m} - \vartheta_v) \quad (2)$$

with the heat flow $\dot{Q}$, the total transfer area A_{total} of the finned tube, the mean wall temperature $\vartheta_{w,m}$ obtained from six local temperatures (section 3.1), and the vapour temperature ϑ_v. The correlation of heat transfer coefficient and heat flow density is presented as usual in the simple form

$$h = C\dot{q}^n. \quad (3)$$

6. RESULTS

6.1. *Single tube*

In Fig. 6, the heat transfer coefficient is plotted against the heat flow density. Various regions of the boiling curve can be discerned.

(a) The transition region between natural convection and incipient boiling for

$$700 < \dot{q} < 3000 \text{ W m}^{-2}$$

with

$$h = 2.27\,\dot{q}^{0.64}. \quad (4a)$$

(b) A region of moderate nucleate boiling for

$$3000 < \dot{q} < 20\,000 \text{ W m}^{-2}$$

with

$$h = 0.697\,\dot{q}^{0.79}. \quad (4b)$$

(c) The region of fully developed nucleate boiling for

$$\dot{q} > 20\,000 \text{ W m}^{-2}$$

with

$$h = 8.53\,\dot{q}^{0.54}. \quad (4c)$$

Visual observation reveals that all spaces between the fins are filled with vapour in the fully developed boiling regime. In moderate boiling, a number of fin interspaces remains without vapour production. Thus in this latter regime, the effect of convection appears still influential and the activation of new nuclei helps towards a steeper increase in heat transfer than in the fully developed boiling where the off-flow of additional vapour is hampered by the fins.

6.2. *Twin-tubes*

The simplest case of a tube-bundle evaporator can be seen in two tubes positioned one above the other. Results for this case are presented in Fig. 7. The heat transfer curve for the lower tube 1 exhibits little difference to the single tube curve: The single tube curve yields lower heat transfer coefficients in the region of

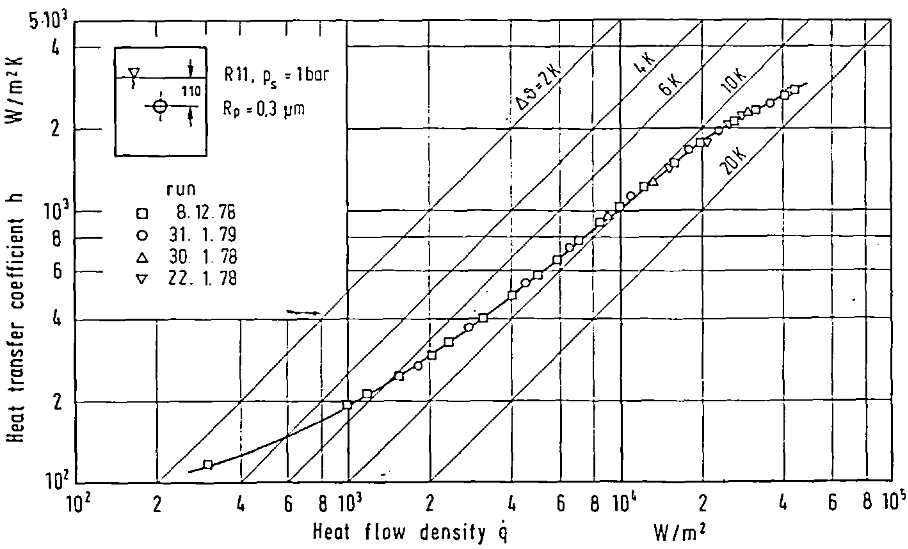

FIG. 6. Heat transfer coefficient on a single finned tube vs heat flow density.

low heat flow densities where the heat transfer is convection-controlled. The increased agitation by a recirculatory flow inside of the test vessel may be the cause of the somewhat higher heat transfer coefficient on the lower tube of the twin-tube arrangement. The enhanced heat transfer coefficients from the upper tube 2, clearly exhibit the effect of approaching flow. Higher heat transfer coefficients are observed throughout the moderate boiling region where a convection-controlled regime prevails. Differences in heat transfer coefficients between the lower and the upper tube vanish with fully developed boiling.

6.3. *Tube-bundle*

6.3.1. *Separately heated middle tubes.* Within the 18-tube bundle only the middle tubes in each row were heated here and only one at a time; results of this variation are shown in Fig. 8. For each tube, approximately the same relation $h \sim \dot{q}$ is obtained regardless of the position within the bundle. A closer scrutiny reveals the heat transfer coefficients of tubes 1 and 5 as being higher than those of tube 2, in most cases.

6.3.2. *Simultaneously heated middle tubes.* For the case that all middle tubes are heated simultaneously

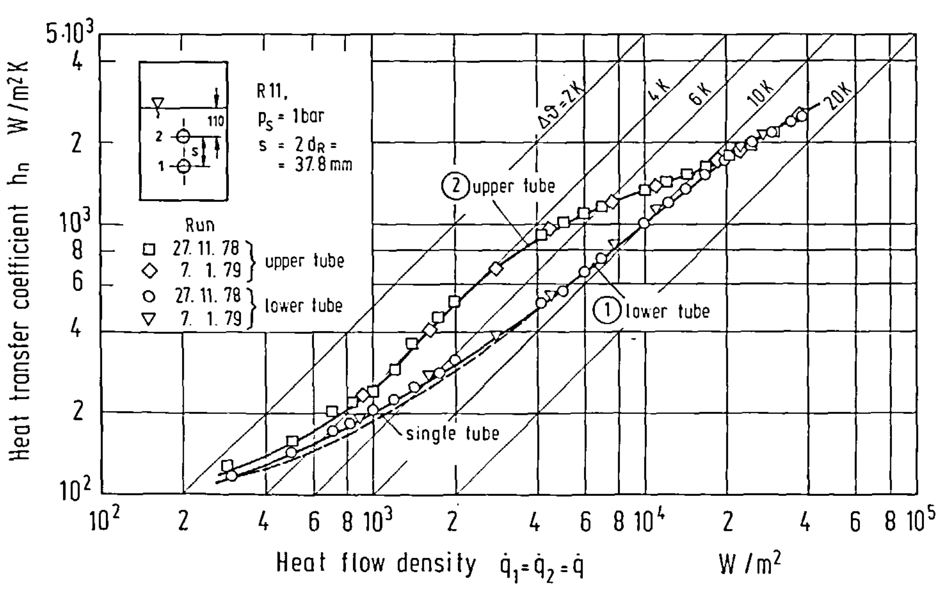

FIG. 7. Heat transfer coefficient h_n as a function of heat flow density for the two tube arrangement ($\dot{q}_1 = \dot{q}_2 = \dot{q}$).

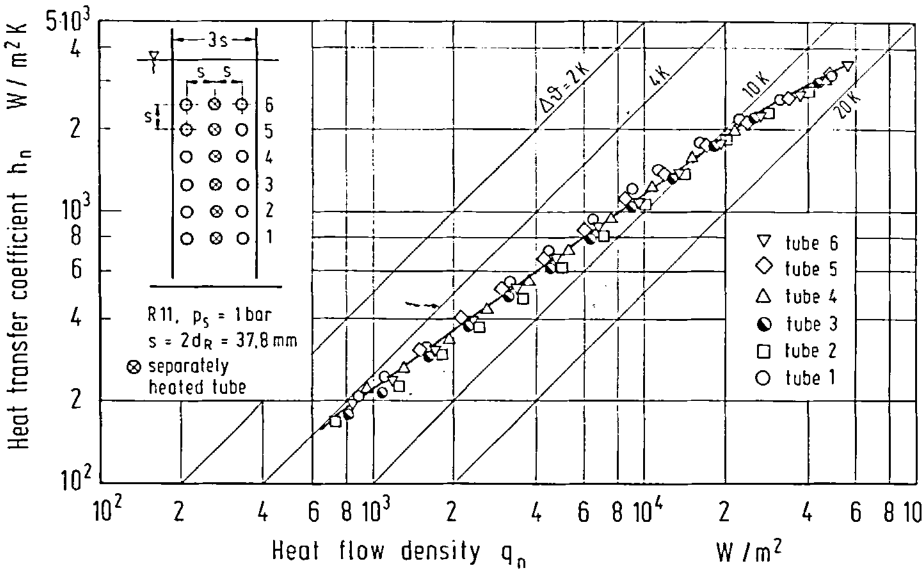

FIG. 8. Heat transfer coefficient h_n for middle tubes heated separately in a bundle arrangement.

with the same heat flow density, the results are shown in Fig. 9. Due to increased convection flows, now the heat transfer is enhanced: the highest heat transfer coefficients are obtained in the top row, the lowest in the bottom row. In the convection-controlled regime, the increase can be 100%; with fully developed boiling the various curves merge. As one would expect, the heat transfer is better than on separately heated tubes.

6.3.3. *All tubes heated.* These results, given in Fig. 10, are similar to those of the simultaneously heated middle tubes with respect to their overall tendency. In detail: in the region of lower heat flow densities where convection has its effects, the heat transfer coefficients are somewhat higher when all tubes are heated. At high heat flow densities, no clear difference in the heat transfer coefficients is observed between the case that all tubes were heated or only middle tubes were heated. It should be noted that heat transfer coefficients were calculated with temperatures measured on each middle tube only. So location effects within a horizontal row were not distinguished.

7. COMPARISON AND CORRELATIONS

As shown in Section 6, the heat transfer coefficients from identical tubes in a bundle vary with tube position and heating arrangement. When the effect of heating

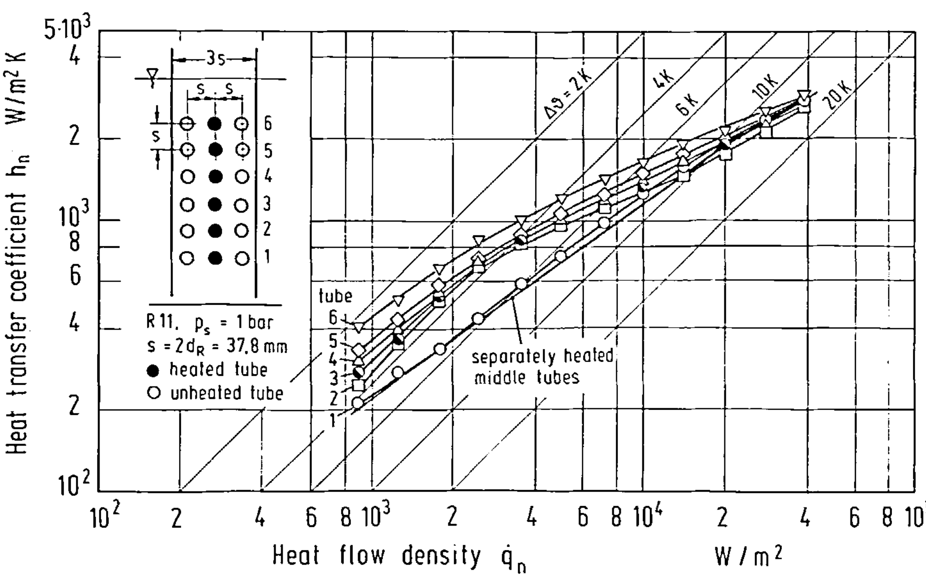

FIG. 9. Heat transfer coefficient h_n for simultaneously heated middle tubes ($\dot{q}_1 = \dot{q}_2 = \ldots \dot{q}_n$; $n = 1,2,\ldots,6$).

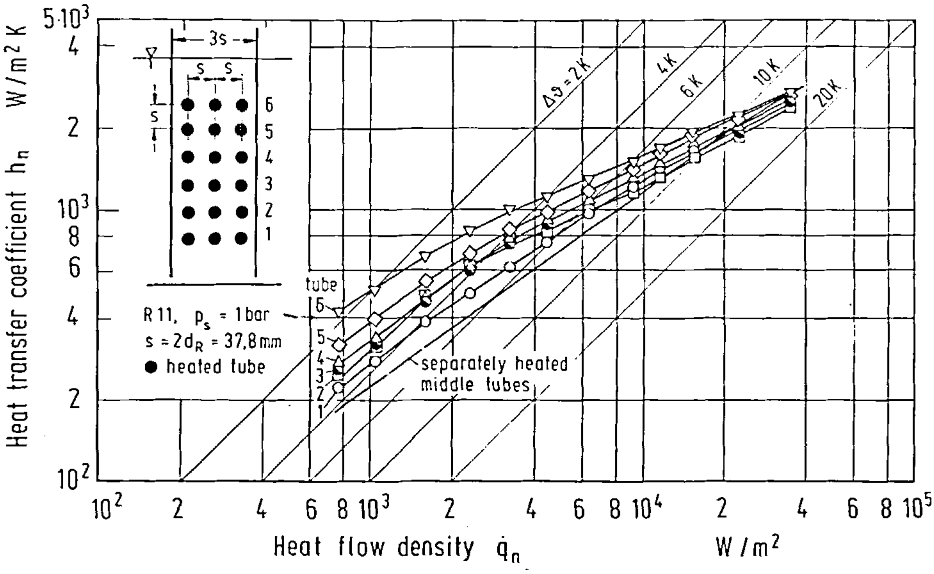

FIG. 10. Heat transfer coefficient h_n for all tubes heated ($\dot{q}_1 = \dot{q}_2 = \ldots \dot{q}_n$).

arrangement is compared for differently located tubes and also comparison is made to the simplest form of a tube bundle (the twin-tubes) the following results are obtained: In Fig. 11, the heat transfer coefficient h_1 on the middle tube 1 (bottom row) is presented as measured in six different heating arrangements. The amazing result is, that this heat transfer coefficient when obtained in a test with only tube 1 heated and 17 dummy tubes around, is about the same as when all 18 tubes are heated. Compared to experiments on twin-tubes, the heat transfer coefficient h_1 fits tolerably between and around the lower and upper twin-tube results. In Fig. 12 the heat transfer coefficient h_2 obtained on the middle tube in row 2 (second from bottom) in five different heater arrangements is given. Here the results split up into an upper and lower curve which both follow quite well the twin-tube results. The upper curve represents test data with other tubes heated in addition to tube 2 and again indicates the effect of enhanced flow agitation. The lower curve represents data for a separately heated tube 2. Results for tubes 3, 4 and 5 are quite similar to those shown in Fig. 12.

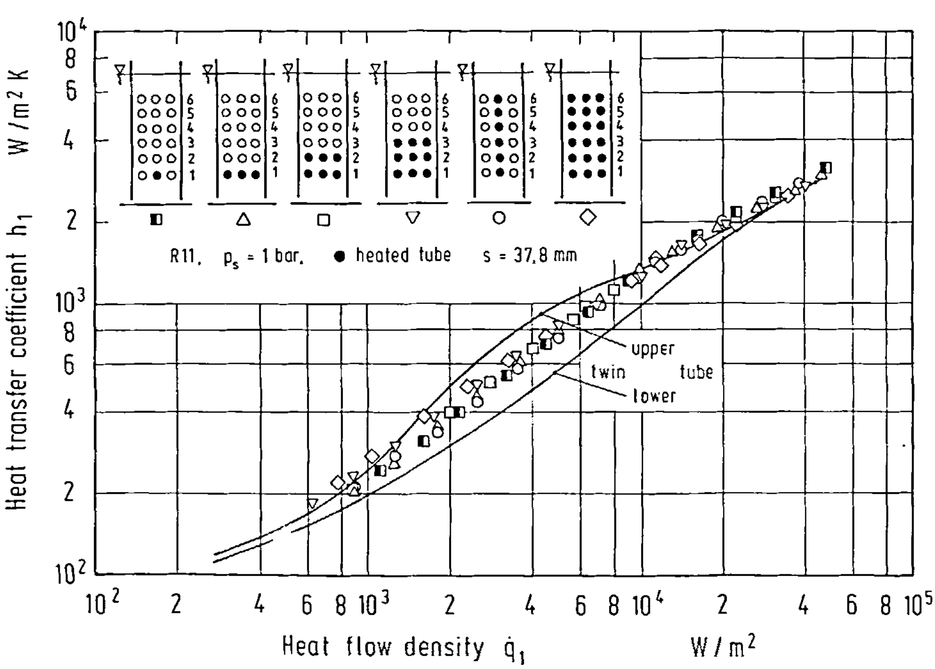

FIG. 11. Heat transfer coefficient h_1 for different heating arrangements ($\dot{q}_1 = \ldots = \dot{q}_n$).

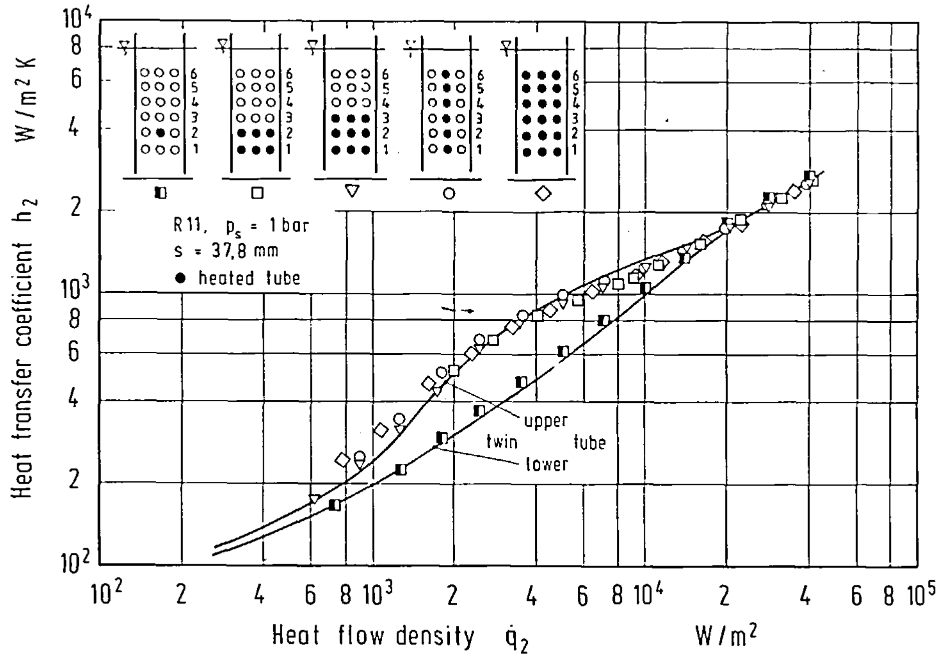

FIG. 12. Heat transfer coefficient h_2 for different heating arrangements ($\dot{q}_2 = \ldots \dot{q}_n$).

In Fig. 13 the heat transfer coefficient h_6 is presented as measured on the middle tube 6 in the top row. In this case only the results of the separately heated tube 6 are somewhat comparable to the lower twin-tube results. The bundle results differ considerably. If this is attributed to a lack of flow hindrance above the top row, it could be concluded that such a boundary effect would disappear with a seventh row above the sixth.

From the results gained on separately heated finned tubes within the bundle arrangement (Fig. 8) the following correlation was obtained:

$$h_{\text{single}} = 1.331\ \dot{q}^{0.735}. \tag{5}$$

For the mean heat transfer coefficient of the entire bundle arrangement (Fig. 10) the experimental results

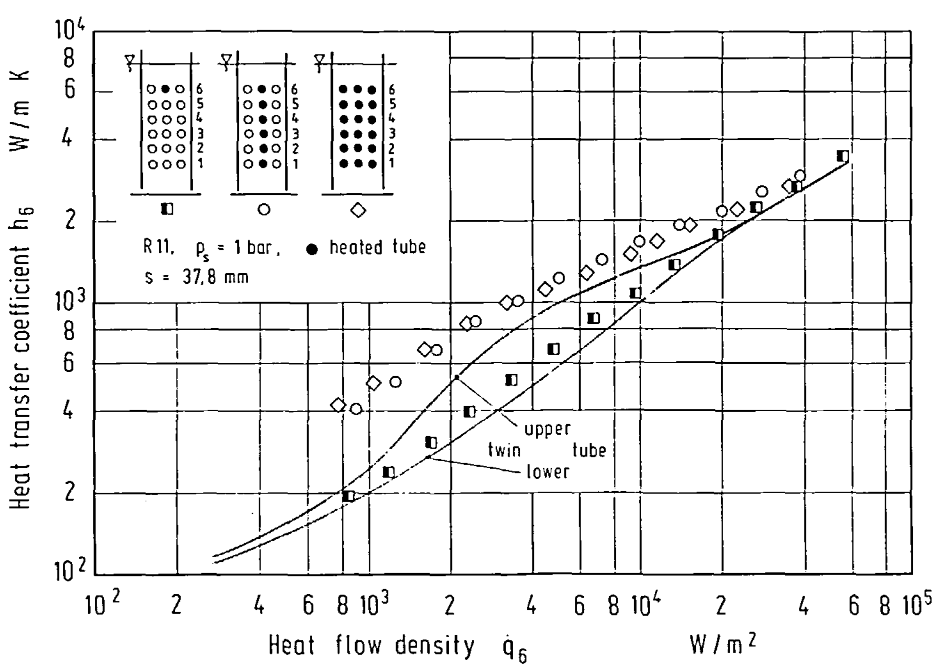

FIG. 13. Heat transfer coefficient h_6 for different heating arrangements ($\dot{q}_6 = \ldots \dot{q}_n$).

can be correlated by

$$h_{\text{bundle}} = 10.0\,\dot{q}^{0.53}. \tag{6}$$

Both correlations hold for $10^3 < \dot{q} < 40 \times 10^3$ W m^{-2}.

8. CONCLUSION

The result that heat transfer coefficients of finned tubes in a bundle do agree tolerably well with those obtained by a simple twin-tube arrangement is promising as experimental expense is decreased considerably. It must be said, however, that this result is so far only proven for an in-line tube arrangement with one quadratic pitch of $s = 2d_R$.

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EBULLITION COMMENCANTE SUR UN TUBE AILETE ET UNE GRAPPE DE TUBES AILETES

Résumé—Une étude expérimentale porte sur un réservoir de 151 dm^3 contenant du R-11, à l'état de saturation de 1 bar et 23,3°C. Les coefficients de transfert thermique mesurés sur un tube unique dans le réservoir ressemblent approximativement au cas où un seul tube est chauffé dans un faisceau en ligne de 18 tubes.

Pour un arrangement à deux tubes, le coefficient de transfert thermique sur le tube supérieur est jusqu'à 80% plus grand que celui du tube inférieur. La courbe de transfert thermique pour ce tube supérieur représente bien les coefficients de transfert des tubes à l'intérieur de la grappe. Seules apparaissent des différences pour les tubes à la base et au sommet de la grappe. La possibilité d'obtenir des données sur la grappe du tube—jusqu'à une certaine limite—par des expériences plus simples sur deux ou trois tubes, devrait faciliter les recherches expérimentales.

SIEDEN AN EINEM RIPPENROHR UND EINEM RIPPENROHRBÜNDEL

Zusammenfassung—Versuche wurden in einem Behälter von 151 dm^3 Inhalt in R 11 beim Sättigungszustand (1 bar; 23,3°C) durchgeführt. Die an einem Einzelrohr gemessenen Wärmeübergangskoeffizienten folgen angenähert jenen, die für ein separat beheiztes Rohr in einem Bündel aus 18 fluchtend angeordneten Rohren erhalten wurden.

Bei einer Zwei-Rohr Anordnung liegt der Wärmeübergangskoeffizient des oberen Rohres bis zu 80% über dem des unteren Rohres. Der Verlauf des Wärmeübergangskoeffizienten dieses oberen Rohres gibt gut den Verlauf der Koeffizienten für Rohre innerhalb des beheizten Bündels wieder. Nur für die untersten und obersten Rohre zeigen sich Unterschiede. Die Möglichkeit, Ergebnisse für Rohrbündel—zumindest angenähert—durch Versuche mit viel einfacheren Zwei- oder Drei-Rohr Anordnungen zu erhalten, würde experimentelle Untersuchungen sehr erleichtern.

ВОЗНИКНОВЕНИЕ КИПЕНИЯ НА ОРЕБРЕННОЙ ТРУБЕ И ПУЧКЕ ОРЕБРЕННЫХ ТРУБ

Аннотация—Проведено экспериментальное исследование кипения в сосуде объемом 151 дм^3, заполненном насыщенным фреоном-11, при давлении 1 бар и температуре 23,3°C. Значения коэффициента теплопереноса, полученные для одной помещенной в сосуд трубы, примерно совпадали с результатами измерений в том случае, когда нагревалась только одна труба внутри коридорного пучка из 18 труб. Для системы из двух труб коэффициент теплопереноса верхней трубы примерно на 80% выше, чем для нижней. Кривая теплопереноса для верхней трубы довольно хорошо описывает данные по коэффициентам теплопереноса для труб в пучке. Различия наблюдаются только для нижней и верхней труб пучка. Возможность получения хотя бы приближенных данных для пучка труб на основе простых экспериментов с системами из двух и трех труб будет способствовать более успешному проведению дальнейших экспериментальных исследований.