

INFLUENCE OF PRESSURE ON HEAT TRANSFER FROM HORIZONTAL TUBES TO BOILING REFRIGERANTS

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Influence de la pression sur la transmission de chaleur de tubes horizontaux à des fluides frigorigènes en ébullition

RÉSUMÉ: *On connaît les données expérimentales concernant les coefficients de convection dans le domaine d'ébullition nucléée des fluides frigorigènes pour des pressions supérieures à 1 bar seulement. Pour les applications techniques, cependant, la transmission de chaleur à des pressions nettement inférieures présente un grand intérêt. L'influence de la pression sur le coefficient de convection étant très variable suivant les calculs des différents auteurs, on a mesuré les coefficients de transmission de chaleur pour divers fluides frigorigènes (R 11, R 22, R 113 et R 502) bouillant à l'extérieur de tubes horizontaux, lisses et à ailettes, sous des pressions de saturation variant entre 0,1 et 10 bars. La température de saturation variait entre -15° et $+60^{\circ}\text{C}$.*

Toutes les mesures effectuées sur des tubes lisses et à ailettes montrent une élévation linéaire du coefficient de convection α en fonction de la pression. Les mesures effectuées sur le tube lisse peuvent s'exprimer par la relation: $\alpha \sim 0,14 + 2,2 p/p_k$ (p_k = pression critique) d'après Danilova. Cette relation peut être vérifiée expérimentalement pour des pressions atteignant $p \sim 0,003 p_k$, tandis que la limite inférieure du domaine de validité donnée par Danilova était de $p = 0,1 p_k$. Pour les tubes à ailettes l'influence de la pression sur α peut s'exprimer quantitativement si la relation de Danilova est corrigée par l'introduction du rapport $\varphi = \frac{\text{surface du tube à ailettes}}{\text{surface du même tube sans ailettes}}$ de la manière suivante:

$$\alpha \sim 0,14 + 2,2 (p/p_k) 1/\sqrt{\varphi}$$

In refrigerating systems with flooded shell-and-tube evaporators, where the medium to be cooled is normally water or brine, the refrigerant boils on the outside of horizontal tubes. Heat transfer to the refrigerant is effected by natural convection, partially in the form of a convectational flow without bubble formation, partially with nucleate pool boiling. The design of such flooded evaporators requires the knowledge of heat transfer coefficients of refrigerants when pool-boiling in various system conditions.

Because with nucleate pool boiling both the properties of the liquid and boundary surface phenomena at the heated wall are important, a theoretical description of the heat transfer is difficult even for simple geometrical configurations. Experimental investigations of nucleate pool boiling with

different liquids on plates and plain tubes have shown that the heat transfer depends mainly upon the following parameters [1]—[5].

- 1) heat flux on the heated surface,
- 2) pressure of the liquid at the heated surface,
- 3) properties of the liquid, e. g. heat conduction coefficient, viscosity, and surface tension,
- 4) characteristics of the heated surface, e. g. roughness [6].

This paper is concerned mainly with the influence of pressure on heat transfer at nucleate pool boiling. A theoretical description of their dependence, however, is difficult: Primarily, there is a direct influence of pressure on the superheating, necessary for bubble formation in the liquid, and secondly the material properties are also a function of the boiling temperature, i. e. of the boiling pressure. This may explain why the equations given by different authors have led to widely differing numerical results [5]—[9].

Experimental data on the heat transfer coefficient at nucleate pool boiling for refrigerants are only known at present for pressures greater than one bar. For technical applications the knowledge of the heat transfer coefficient at substantially lower pressures is also of interest. This paper investigates the heat transfer coefficient at nucleate pool boiling for different refrigerants (R-11, R-22, R-113, and R-502) on plain tubes in a wide pressure range. Boiling pressure and temperature were varied from 0.1 to 10 bar and -15°C to $+60^{\circ}\text{C}$, respectively. The experimental data span a wide range of heat fluxes, beginning with evaporation without bubble formation, and extending over the range of nucleate pool boiling to nearly the burn-out point. With the refrigerant R-11, corresponding experiments have been made also with seven finned tubes instead of a plain tube. As is well known, increasing the heat flow per unit of tube length by using fins is of special interest in refrigeration techniques, because with moderate nucleate pool boiling of refrigerants the heat transfer coefficient on the outside of the tubes is substantially lower than for the forced convectational flow of water or brine on the inside.

APPARATUS AND TEST PROCEDURE

A schematic representation of the test apparatus is shown in Fig. 1. The refrigerant evaporates on the outside of the test tube which is placed in a horizontal position in the evaporator. The vapour of the refrigerant rises to the condensers; at boiling temperatures below 30°C the cooling medium is trichlorethylene (condenser no. 1), and at temperatures above 30°C the cooling medium is water (condenser no. 2). The condensed vapour returns in free circulation back to the evaporator. The condensers and the evaporator are mounted in a box, the temperature of which is kept equal to the boiling

temperature of the circulating refrigerant. This arrangement reduces to a minimum the heat exchange between the circulating refrigerant and the outside.

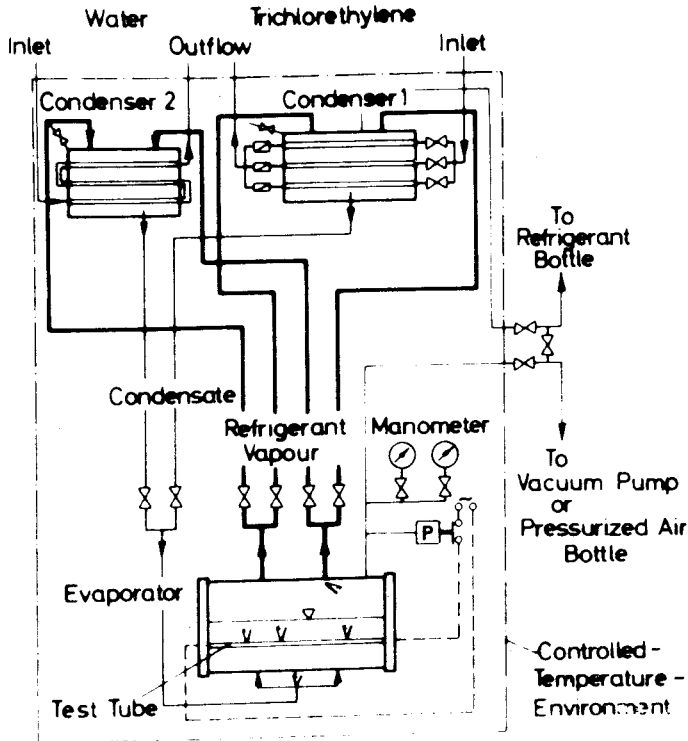


Fig. 1. — Diagram of the test apparatus.

In the tests commercial-type tubes were used; they included a plain copper tube and seven finned copper tubes the fins of which are extruded in spiral form from the base metal itself (see Fig. 2). A resistance-type heating device was soldered concentrically into the test tubes, as shown in Fig. 3.

Stationary conditions within the test apparatus are obtained by accurately controlling the heat carried off in the condenser, combining the following control possibilities:

- 1) extension or reduction of the condenser surface,
- 2) mass control of the cooling medium,
- 3) temperature variation of the cooling medium.

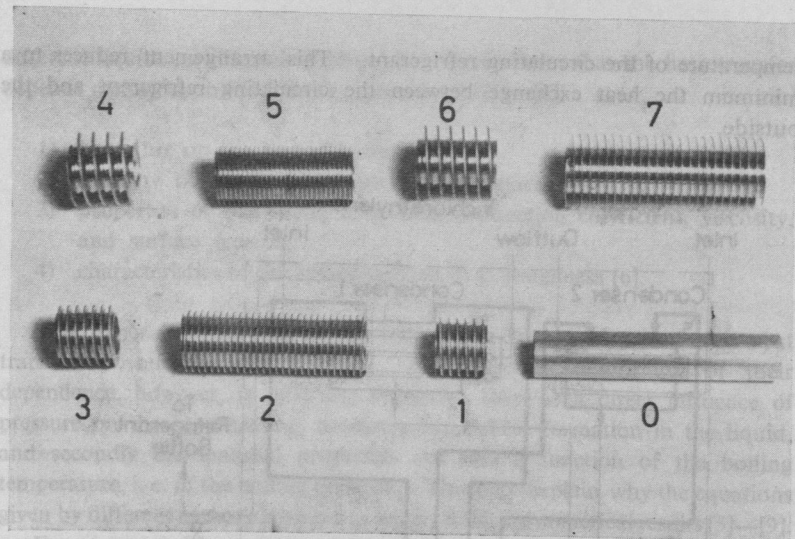


Fig. 2. — Photograph of samples from the plain and the different finned tubes used in the present investigation.

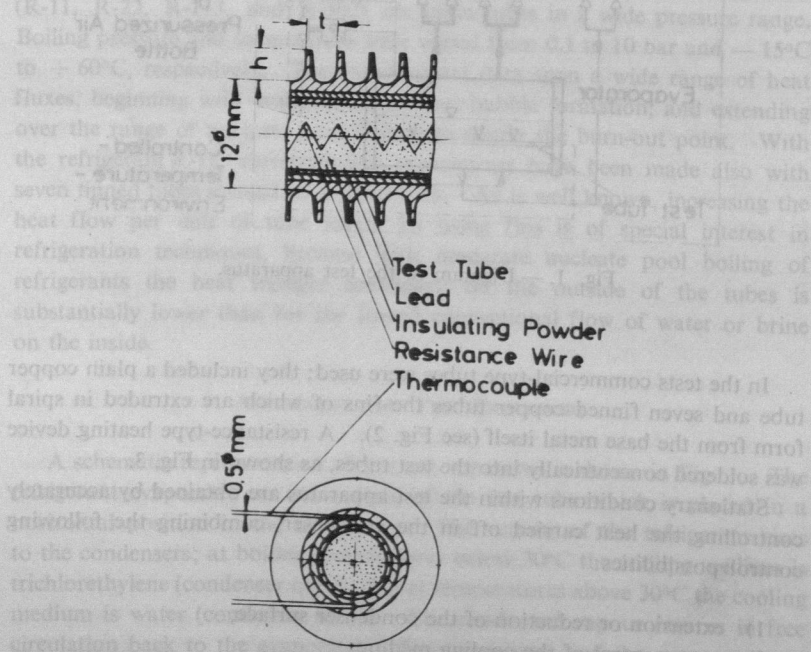


Fig. 3. — Test tube design. Heating device and location of thermocouple.

It is a well-known fact that because of the gas trapped within the crevices of the heated wall, the heat transfer coefficient at nucleate pool boiling at the beginning of the tests is always greater than it is after a certain running time [10]. For this reason no measurement was taken until the apparatus had run for sixty hours and preliminary measurements had shown a fully stabilized value of the heat transfer coefficient [11]. All tests were taken in the direction of decreasing heat fluxes in order to eliminate the phenomenon of hysteresis in the transition between free convectional flow without bubble formation and nucleate pool boiling [11].

To describe the heat transfer from the tube wall to the boiling liquid the heat transfer coefficient α is defined in the following manner:

$$\frac{Q}{A} = q = \alpha (t_w - t_s) \quad (1)$$

where

- Q is the total heat flow from the tube,
 A is the outer surface area of the test tube; in the case of finned tubes this area is extended by the factor φ , defined as

$$\varphi = \frac{\text{outer surface of the finned tube}}{\text{surface of the same tube without the fins}}$$

- t_w is the temperature at the outer surface of the tube, computed from the measured values of the inner wall temperature; in the case of finned tubes t_w is the temperature at the base of the fins,

and

- t_s is the boiling temperature of the liquid corresponding to the given pressure p at the liquid-vapour interface. t_s coincides with the temperature of the bulk liquid, within 0.1°C.

EXPERIMENTAL RESULTS

In Figs. 4 and 5 the heat transfer coefficient α at pool boiling of R-502, R-113, and R-11 on the plain test tube is represented as a function of the heat flux q in a double-logarithmic diagram. To this is added a typical result for the pool boiling of R-11 on finned tubes (c.f. Fig. 5). There are clearly two regions with different slopes of the heat transfer coefficient:

In the region of free convection without bubble formation the heat transfer coefficient increases only slightly with the heat flux. In this range, the increase of α with q is caused, of course, by the density difference between the liquid at the heated wall and at a certain distance from it, increasing with the heat flux.

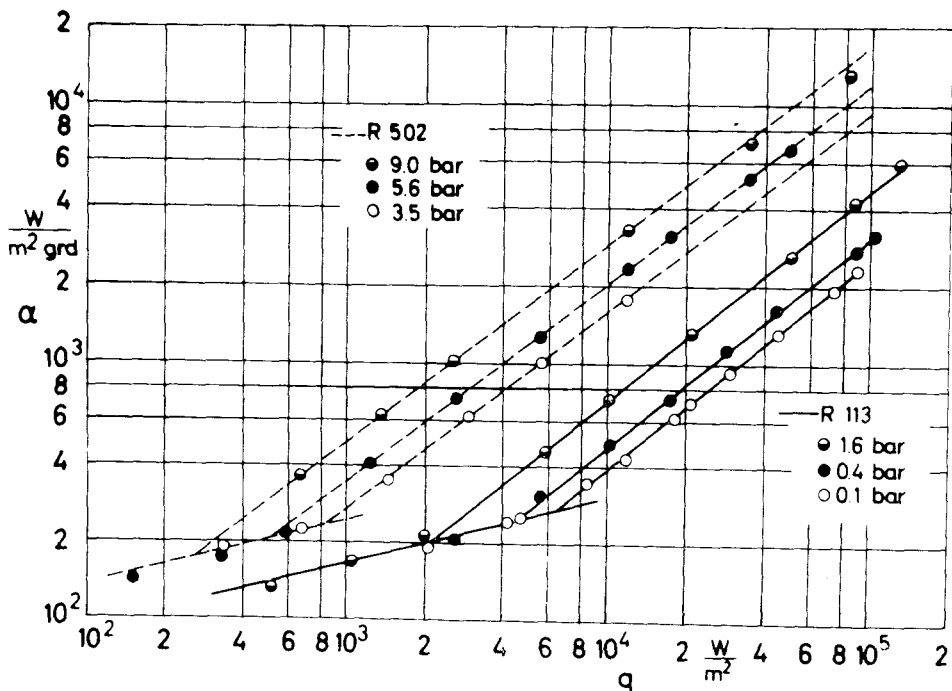


Fig. 4. — Heat transfer coefficient α as a function of heat flux q for pool boiling of refrigerant R-502 and R-113 at various saturation pressures. Plain tube (12 mm ϕ); surface roughness $R_p = 0.4 \mu\text{m}$.

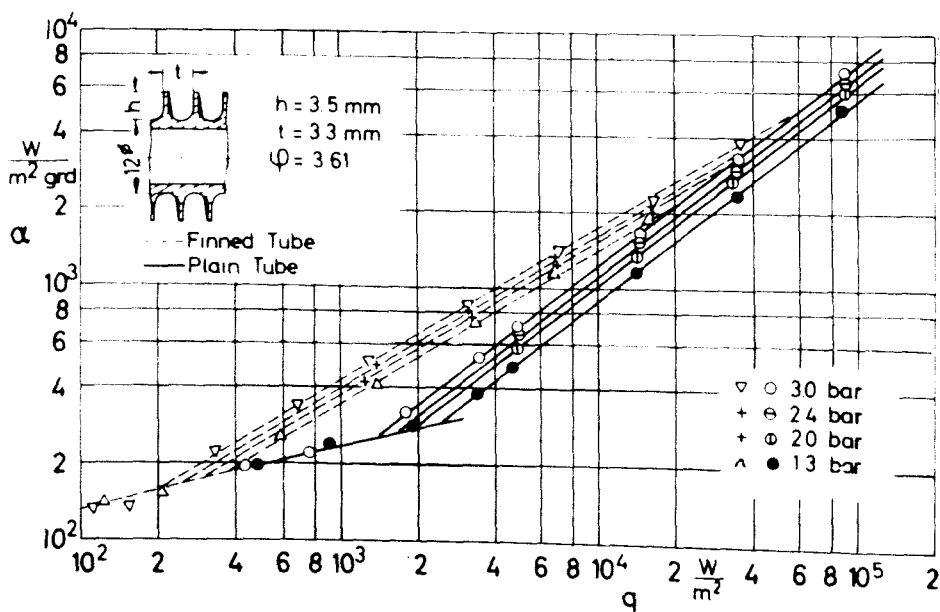


Fig. 5. — Heat transfer coefficient α as a function of heat flux q for pool boiling of refrigerant R-11 on plain tube No. 0 and finned tube No. 6, cf. Fig. 2. Surface roughness $R_p = 0.4 \mu\text{m}$.

In the region of nucleate pool boiling a more pronounced increase of the heat transfer coefficient with the heat flux is observed, due to the mixing effect produced by the bubbles ascending from the heated surface.

Whereas in the region of free convection without bubble formation the heat transfer coefficient is independent of pressure (within the experimental limits of error), in the region of nucleate pool boiling the heat transfer coefficient increases markedly with increasing pressure, i.e. the necessary temperature difference for the transfer of a certain heat flux decreases with increasing pressure. This is mainly due to the fact that the excess pressure necessary for the formation of a stable bubble can be achieved with smaller temperature differences at higher absolute pressures because the slope of the vapour-pressure curve is steeper at higher pressures. In addition it seems to be important that, at constant heat flux and increasing pressure considerably more vapour bubbles are formed while the average bubble diameter decreases markedly (see Fig. 6a and b).

A similar relationship between the heat transfer coefficient, heat flux, and pressure is found also with finned tubes, as is seen in Fig. 5; a closer investigation, however, reveals the following differences between the plain and finned tube:

- a) At the same pressure and the same heat flux, the heat transfer coefficient related to the total surface area is substantially higher for the finned tubes than for the plain ones. This is observed especially in the region of moderate nucleate pool boiling, important for refrigeration techniques. The effect may be explained qualitatively by an additional convectional heat transfer caused by the bubbles rising along the fin surface. In this connection, it has to be considered that, because the surface has been extended, the heat flow per unit length of the finned tube is greater than with the plain tube, making for a stronger bubble formation per unit volume and a correspondingly better mixing of the liquid between the fins.
- b) The relative increase of the heat transfer coefficient with increasing pressure and increasing heat flux is smaller for finned tubes than for plain ones. The decrease of the relative pressure dependence of α might be understood by an additional convectional heat transfer, which depends but slightly on the pressure. The reduced increase of the heat transfer coefficient with the heat flux, on the other hand, can be explained by the fact that the fin efficiency decreases with increasing heat flux and that the gap between the fins is more and more filled with vapour (see Fig. 7a and b).

The representation of the experimental data in the double-logarithmic diagrams (Fig. 4 and 5) shows that the following analytical description of the heat transfer coefficient is possible:

$$\alpha = c q^n F(p) \quad (2)$$

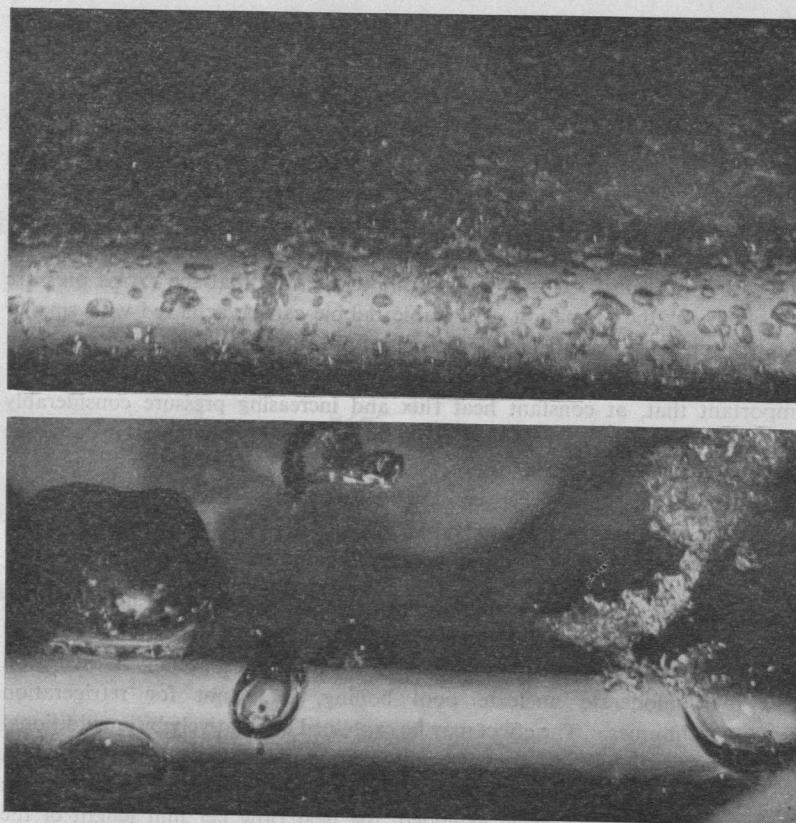


Fig. 6. — Bubble formation in boiling R-113 for different pressures at a constant heat flux ($q = 1.2 \times 10^4 \text{ W/m}^2$, plain tube)

a) $p = 1.6 \text{ bar}$.

b) $p = 0.1 \text{ bar}$

In this relation the value of c is dependent upon the system liquid-heating surface, e.g. c contains, among other parameters, the influence of the surface roughness [6]. On the other hand, the exponent n for the plain tube proved to be almost constant; it lies between 0.77 and 0.81 for all the refrigerants investigated (see Figs. 4 and 5). For the case of finned tubes both c and n depend upon the geometry of the fins [11].

In Fig. 8a the ratio α/q for the measurements on the plain tube is represented in a double-logarithmic diagram as a function of the normalized

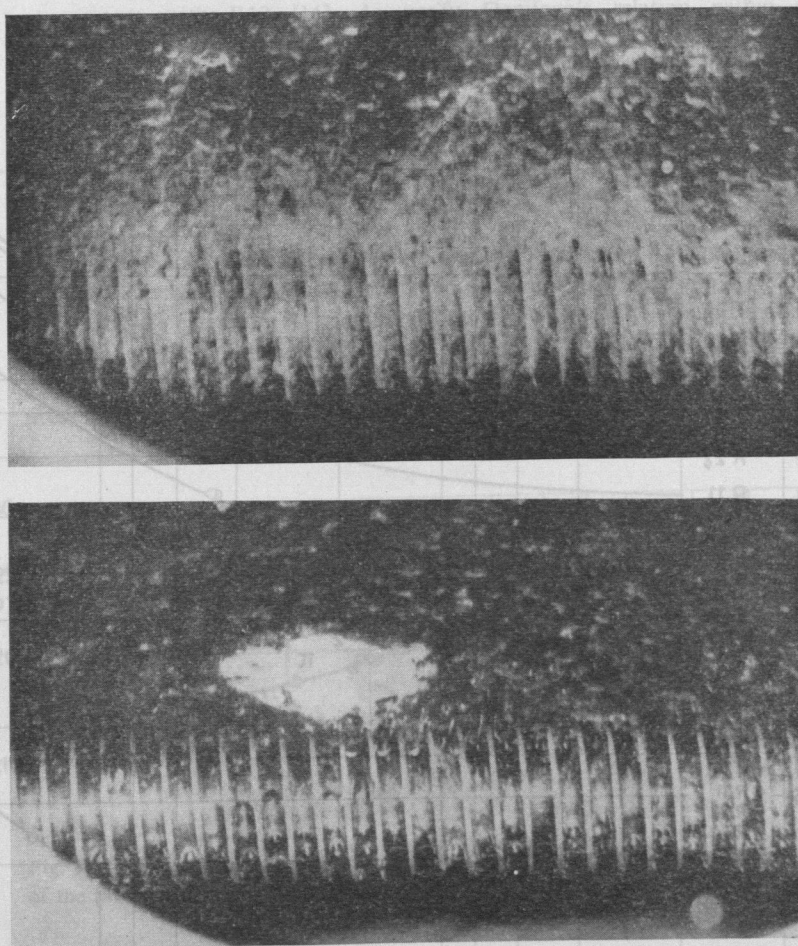


Fig. 7. — Bubble formation in R-11, boiling on a finned tube (No. 6, c.f. Fig. 2) for different heat fluxes at constant pressure ($p = 1.3$ bar)

$$a) \quad q = 4.0 \times 10^4 \frac{W}{m^2}$$

$$b) \quad q = 9.0 \times 10^3 \frac{W}{m^2}$$

pressure $\pi = p/p_{cr}$ (p_{cr} = pressure at the critical point). The curves show the following pressure dependence of the heat transfer coefficient, reported by Danilova [12] and based on similarity considerations by Borishanskij [13]:

$$F(\pi) = \frac{\alpha}{c q^n} = 0.14 + 2.2 \cdot \pi \quad (3)$$

Whereas this linear relation between heat transfer coefficient and pressure was established by Danilova only for the range of $0.1 \leq \pi \leq 0.5$, the experiments with R-113, reported here, show that the relation holds also for substantially lower values of the normalized pressure, namely for $\pi \geq 0.003$.

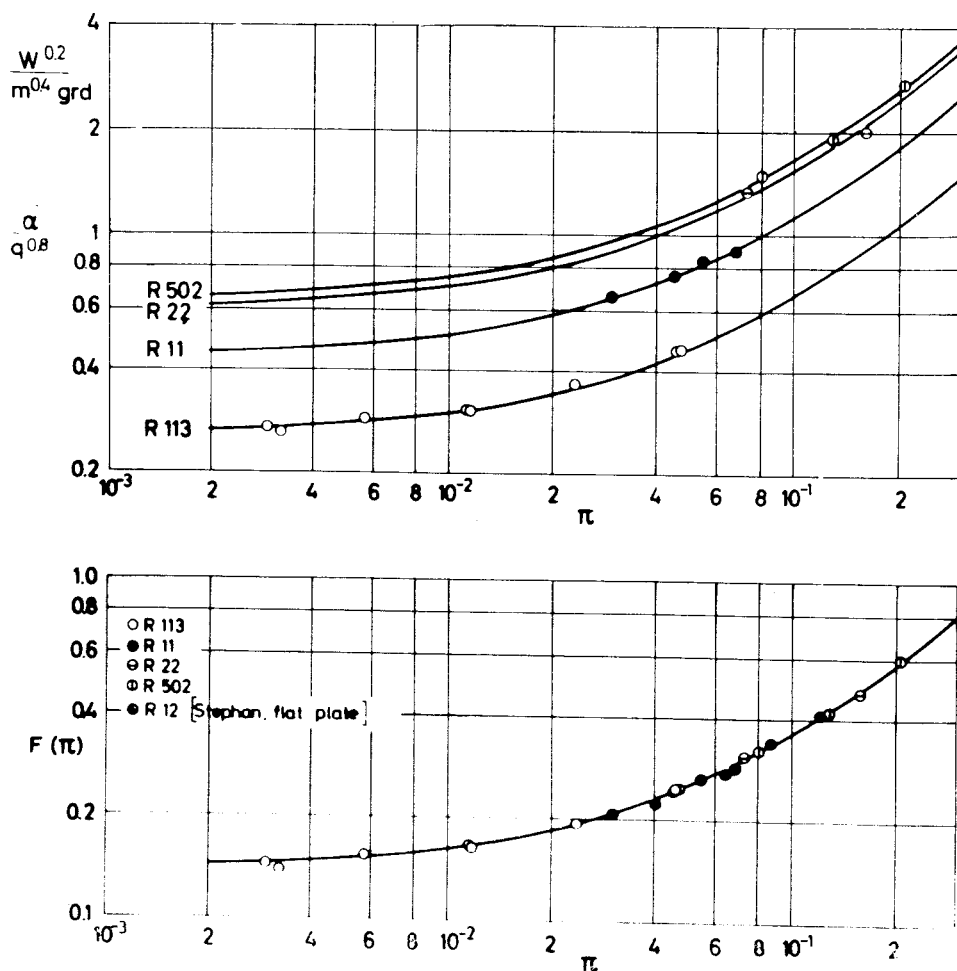


Fig. 8. — Measured values of the ratios $\alpha/q^{0.8}$ and $\alpha/c q^{0.8} = F(\pi)$ as a function of normalized pressure π for nucleate pool boiling of R-11, R-12, R-22, R-113, and R-502 on plain tubes. The full curves give $F(\pi)$ according to Danilova [12].

In Fig. 8b the function $F(\pi)$ is plotted against π for all the refrigerants investigated. All experimental results, including both the data for the azeotropic refrigerant R-502 (48.8% R-22, 51.2% R-115), and for Stephan's

experiments with R12 [14], do verify Danilova's relation within an experimental limit of error of $\pm 5\%$.

In the case of finned tubes, where a description of the pressure dependence of the heat transfer coefficient has been lacking until now, the following extension of Danilova's equation is suggested, using the parameter φ (c. f. eq. (1)):

$$F(\pi, \varphi) = 0.14 + 2.2 \cdot \pi / \sqrt{\varphi} \quad (4)$$

In Fig. 9 this relation is represented as a full curve and compared with the experimental data from all finned tubes investigated here, with the value of the surface extension φ varying between 2.3 and 4.9. It is found that the maximum deviation of the experimental data is $\pm 3\%$ within the investigated pressure range of 1.3 to 3.0 bar. An examination of this relation for a greater pressure range is currently underway.

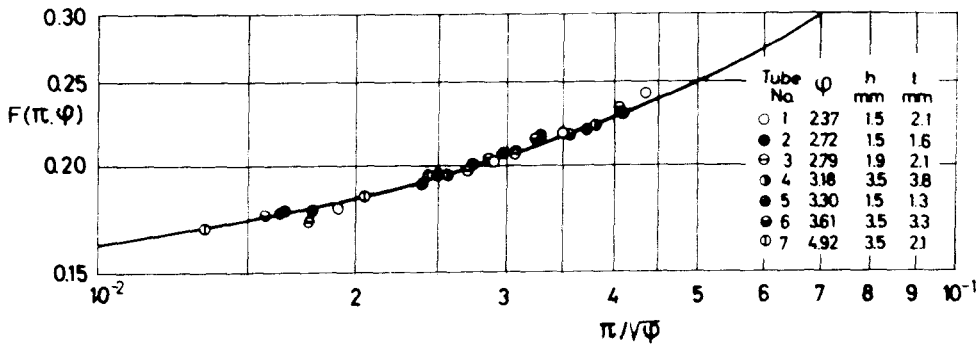


Fig. 9. — Comparison of $F(\pi, \varphi)$ according to eq. (4) with the experimental results of the pressure dependence of α for nucleate pool boiling of R-11 on finned tubes.

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REFERENCES

- [1] W. FRITZ: *Chemie-Ingenieur-Technik* 35, 1963, 753/764
- [2] H. BRAUER: *Chemie-Ingenieur-Technik* 35, 1963, 764/774.
- [3] K. STEPHAN: *Chemie-Ingenieur-Technik*, 35, 1963, 775/784.

- [4] W. M. ROHSENOW: «Modern Developments in Heat Transfer», Academic Press, New York, 1963, W. Ibele ed.
- [5] H. G. HIRSCHBERG: *Kältetechnik* 18, 1966, 155/160; c. f. also «Kältemittel», C. F. Müller Verlag, Karlsruhe, 1966.
- [6] K. STEPHAN: «Abhandlungen des Deutschen Kältetechnischen Vereins», Nr. 18, C. F. Müller Verlag, Karlsruhe (1964).
- [7] T. H. INSINGER, H. BLISS: *Trans. A. I. Ch. E.* 36, 1940, 491/516.
- [8] CH. F. BONILLA, CH. W. PERRY: *Trans. A. I. Ch. E.* 37, 1941, 685/705.
- [9] MOSTINSKI: *Teploenergetika* 10, 1963, N. 4, 66/71.
- [10] M. JAKOB, W. LINKE: *Forschung* 4, 1933, 75/81.
- [11] D. GORENFLO: *Dissertation*, Techn. Hochschule Karlsruhe 1966.
- [12] G. N. DANILOVA: *Kholodilnaya Tekhnika*, 1965, N. 2, 36/42.
- [13] V. M. BORISHANSKIY, A. P. KOZYREV: *Ing. Fiz. Journal* 5, 1962, N. 12, 3/8.
- [14] K. STEPHAN: *Kältetechnik* 16, 1964, 162/166.

DISCUSSION

J. VAN MALE (The Netherlands): What is the dimension of the quantity c in equation (3) on paper 2.41?

D. GORENFLO: The dimension of the quantity c in equation (3) has to be chosen in such a way that the function $F(\pi)$ becomes dimensionless, i.e. dimension of $c = (W/m^2)^{1-n} \cdot (1/grad)$.

J. McINROY (U. K.): I am particularly interested in the finned tubing used for the experiments; I would like to know:

- 1) The type of copper used, i. e. commercial grade or oxygen-free.
- 2) The dimensions of the tubing and fins.
- 3) The method of manufacture.
- 4) The manufacturer, if this is permissible.

D. GORENFLO: The tubes I used in the experiments were commercial-type tubes, and hence the copper used was commercial grade copper.

2) The most important dimensions of the tubing and fins are given in Fig. 9 of the paper. For further information see reference [11].

3) The fins are extruded in spiral form from the base metal of plain copper tubes.

4) The tubes were a gift from Wieland-Werke A. G., Ulm/Donau and R.u.G. Schmöle, Metallwerke, Menden/Sauerland.

R. V. DUNKLE (Australia): I would like to ask Mr Gorenflo why he did not consider the fin efficiency in his parameter φ ? Surely one must consider the effect of fin geometry and thermal conductivity?

D. GORENFLO: The fin efficiency and the fin geometry were certainly considered, but not in connection with the pressure-dependence of the heat transfer coefficient. For the copper tubes used the fin efficiency which was always better than 90 % (with only one exception), had no influence on the pressure-dependence of the heat transfer coefficient, within the experimental limit of error.