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LOCAL AND AVERAGE HEAT TRANSFER COEFFICIENTS AT AN AIR STREAM IN A TUBE WITH A POINTED INLET

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Abstract—This work adduces experimental data on local and average coefficients of heat transfer from air to the wall of a tube with a pointed inlet in the range of *Re* numbers including a transient region.

Résumé—Ce travail apporte de nouveaux résultats expérimentaux sur les coefficients locaux et moyens de transmission de chaleur, pour l'air, à la paroi d'un tube à entrée effilée, pour un domaine de nombres de Reynolds qui comprend la région de transition.

Zusammenfassung—Diese Arbeit liefert weitere Versuchswerte über den örtlichen und mittleren Wärmeübergangskoeffizient von Luft an die Wand eines Rohres mit scharfkantigem Einlauf in einem Bereich der *Re*-Zahl, der auch den Übergangszustand enthält.

Abstract—В работе приведены экспериментальные данные по локальным и средним коэффициентам теплообмена от воздуха к стенке трубы с острым входом в диапазоне чисел *Re*, включающем переходную область.

THE influence of l/d on heat transfer was discovered by some investigators who suggested that the generalized heat transfer equation should take this influence into account [1, 2].

Other investigators who examined the heat transfer process under the heating conditions of a working media came to the conclusion that the influence of l/d on heat transfer is negligible and that it should not be taken into account by the generalized equation.

The well-known method of calculation [3] for determining the influence of the tube length on heat transfer coefficients by introduction of the factor $(l/d)^{-0 \cdot 054}$ into equations of such a type as $Nu = c.Re^n Pr^m$ contradicts the conditions of the experiment since an increase of l/d gives an infinite decrease of heat transfer.

Experimental data [4] were obtained when water was heated for the single case of an inlet. They are of a particular character and valid for the given conditions of the experiment.

A horizontal, cylindrical steel tube with an experimental part 1236 mm long and an inside diameter 31·66 mm is the main element of the installation for the investigation of heat transfer in a tube (Fig. 1).

The experimental tube was mounted between the reservoir (2) and a feeder of the ventilator. The tube was joined to the reservoir by means of a coupling flange through the feeder (4), the diameter of which is equal to that of the tube. Heat transfer from the coupling flange was avoided with the help both of the heat insulation packing 8 mm thick, and the bolt plugs, which eliminated direct contact of the metallic surfaces.

The room air entered the reservoir, which was divided by a fixed partition into two wells; then while passing the electric heater (3), mounted in the hoisting well of the reservoir, the air was heated, and having passed the drop well, made its way into the experimental tube, out of which it was sucked by the ventilator. The entrance to the tube from the reservoir side is a pointed one and answers the usual entrance conditions.

From the outlet the experimental tube also had a coupling flange with an insulation packing, a straight part $8d$ long, a bend of 90°, and a transition to the vertical part (10) 49·8 mm in diameter, where a double diaphragm for the measuring of air expanse was mounted. A regulating target was placed in front of the feeder.

The experimental part of the tube was placed concentrically into the casing (6), made of a tube 89/80 mm in diameter.

To receive the data on heat distribution along

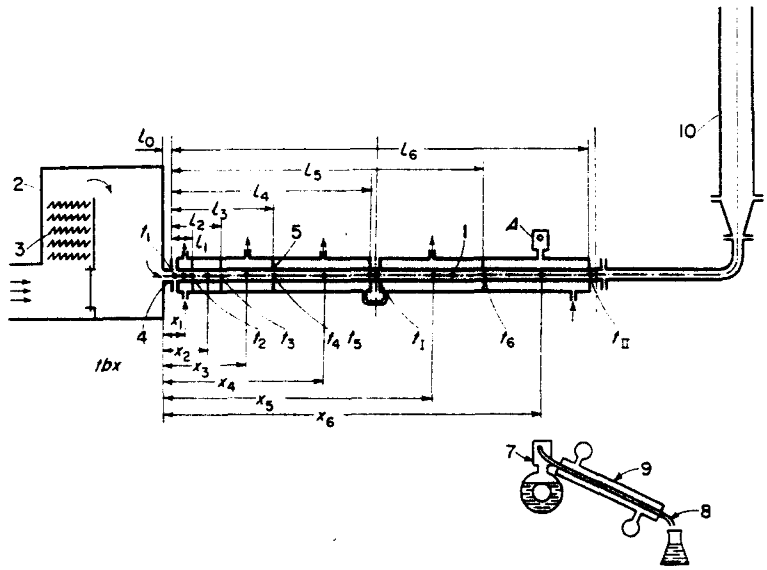

FIG. 1. The scheme of the experimental installation for the investigation of heat transfer in an annular pipe with a pointed inlet.

the length of the experimental tube the annular space between the tube and the casing was divided by transversal partitions (5) into six compartments of different length. All the compartments are joined and they are fed from both ends by the boiling water flowing out of the tank. The operation both of the tank and the casing is based on the principle of communicating vessels.

The water level in the casing was maintained constant and exceeded the mark of the upper generating line on the outside surface of the experimental part of the tube by 6–7 mm; this ensured a constant submerging of the experimental part of the tube in the casing.

The air was cooled while passing the experimental part of the tube, giving up its heat to the boiling water in the casing. The steam received from the separate parts was drawn off into the refrigerators (9), where it was condensed. In order to measure the condensate, it was drained out of each compartment through separate branches (8) into glass bulbs. The selection of steam from the compartments was carried out from the upper point of steam collectors (7) to avoid the priming of moisture by the steam.

The experimental tube was safely isolated to decrease heat transfer into the external medium.

Fig. 1 shows the tube division into the experimental parts and the geometrical characteristics are given in Table 1. The beginning of the experimental tube is $l_0 = 36$ mm distant from the entrance and therefore it is assumed that $l'_i = l_i + l_0$.

The points for measuring the balanced values which characterize the state of flow at the entrance, at a distance $l = 20d$ and at the outlet with $l = 40d$ are depicted by cross-sections 0–0, I–I and II–II, respectively.

The measurement both of the total pressure and temperature fields at the cross-sections I–I and II–II was made at two reciprocally perpendicular radii with the help of micropipes of total pressure, and with the help of microthermocouples moving by means of a micrometer screw. The temperature of the heated air in the reservoir at the tube entrance was measured by several thin thermocouples and was checked by a suction thermocouple. The measurement of the wall temperature along the tube was made by thin thermocouples (copper–constantan couples) 0·20 mm in diameter, placed at the

Table 1. A diagram of the division of the experimental tube into parts

Designation	Nos. of the experimental tube parts (1)	(2)	(3)	(4)	(5)	(6)
The distance from the entrance to the middle of the experimental section x_i (mm)	70	146	267	493	822	1122
Relative length $\frac{x_i}{d}$	2·21	4·63	8·45	15·6	26·0	35·5
The length of the tube experimental part from the beginning of the tube l_i (mm)	64	155	306	608	936	1236
Relative length $\frac{l_i}{d}$	2·02	4·9	9·67	19·2	29·6	39·1
The length of the experimental part from the entrance l'_i (mm)	100	191	342	644	972	1272
Relative length $\frac{l'_i}{d}$	3·17	6·05	10·8	20·4	30·7	40·3

points of average length in each compartment on the upper generating line of the experimental tube's outer surface. The thermoelectromotive force of all the thermocouples was measured by a potentiometer. The air discharge was registered by a twin diaphragm calibrated for cooled and for heated air.

When the installation was set into operation, the measurements were made after the achievement of a stationary regime after a time interval of 3–4 hr.

The isothermal blowing of the experimental tube with the attached stabilizing part $l = 46d$ long and a diameter equal to that of the experimental tube was carried out before the non-isothermal experiments took place. It was ascertained under these conditions for the experimental tube that the resistance factor obeys the well-known resistance law for smooth tubes which is expressed by the following equation $\zeta = 0{\cdot}316\ Re^{-0{\cdot}25}$. It proves that the experimental tube is technically smooth.

The non-isothermal experiments were carried out without the attached stabilizing part and covered a range of variation of Reynolds numbers from $3{\cdot}9 \times 10^3$ to 19×10^3 including the transition region. The air temperature at the tube entrance was maintained within the limits 340° to 421°C.

The values of average and local coefficients of heat transfer by convection were determined from the heat balance equations for the tube section under consideration according to the amount of heat transferred per hour through the tube element, considered as a result of heat transfer between the air flow and the surface of the tube element [5].

On the other hand, the heat transferred by the air to the part of the tube under consideration is spent on the evaporation of water and on heat transfer into the external medium.

The heat transfer from the experimental tube into the external medium was determined by direct measurements both of the isolation and the room air temperatures. During the whole period of the experiments the temperature of the isolation of part 0–I was 43·3°C and of part I–II it was 43·2°C.

As the casing wall temperature along the length of the tube did not vary and was equal to the temperature of boiling water under atmospheric pressure then we consider the distribution of the heat transfer over separate elements of the tube proportional to their lengths.

All the experimental data on local and average values of heat transfer coefficients were processed in the generalized equations:

$$Nu = \frac{Qd}{F\Delta tx} \quad (1)$$

and

$$Re = \frac{4G}{\pi dg\mu} \quad (2)$$

The local values of heat transfer coefficients are given in Fig. 2 as the dependence $Nu_x = c_1 Re^{n_1}$.

The local heat transfer coefficients vary in the range of the Reynolds numbers investigated at the air flow in a tube with a pointed inlet, and they have a sharp fall with the increase of the relative length to $x/d = 5$ at the beginning and then, while proceeding away from the entrance

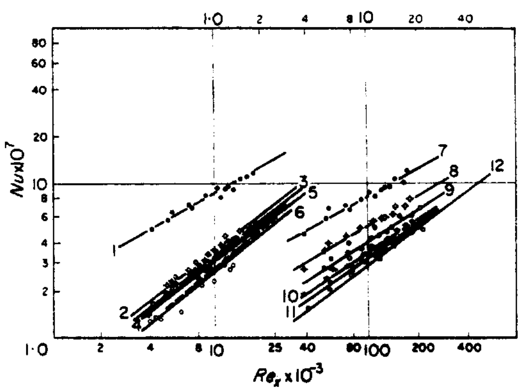

FIG. 2. The local and average values of the heat transfer coefficients for a tube with a pointed inlet without inside insertions.

they vary very slightly at the approach to a definite value for the cross-section which is $x/d \geqslant 40$ distant from the entrance. It is possible to determine local values of heat transfer coefficients for the elements of the tube situated at a distance more than $40d$ from the following equation

$$Nu = 0{\cdot}0145\, Re_f^{0{\cdot}83}\, Pr_f^{0{\cdot}4} \qquad (3)$$

The local values for cross-sections situated from the entrance at a distance less than $40d$ can be determined in accordance with the formula

$$Nu_x = 0{\cdot}0145\, Re_f^{0{\cdot}83}\, Pr_f^{0{\cdot}4} \left(1 + \kappa_x \frac{d}{x}\right) \qquad (4)$$

where κ_x for $x/d \geqslant 5$ are given in Table 2.

In Fig. 2 the experimental data for average values of heat transfer coefficients are given by the equation $Nu = c\, Re^n$.

From the experimental results obtained we can see that the average values of the heat transfer coefficients vary in the investigated range of the Reynolds numbers up to a cross-section which is situated at a distance of $40d$ from the beginning of the experimental tube. Thus with an increase of Re this distance gradually decreases. Beginning with a tube length equal to $l \geqslant 40$ the average values of heat transfer coefficients become practically constant and satisfy an equation of the following type

$$Nu = 0{\cdot}023\, Re_f^{0{\cdot}8}\, Pr_f^{0{\cdot}4} \qquad (5)$$

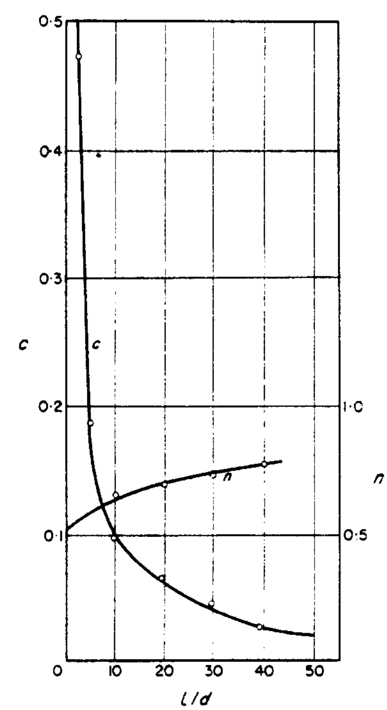

FIG. 3. The dependence of the c and n coefficients for the average values of Nusselt criterium on the relation l/d.

Table 2. The meaning of the coefficient κ_x at various Reynolds numbers

$Re \times 10^{-3}$	4	6	8	10	12	14	16	18	20
κ_x	2·22	1·93	1·89	1·7	1·6	1·55	1·39	1·37	1·34

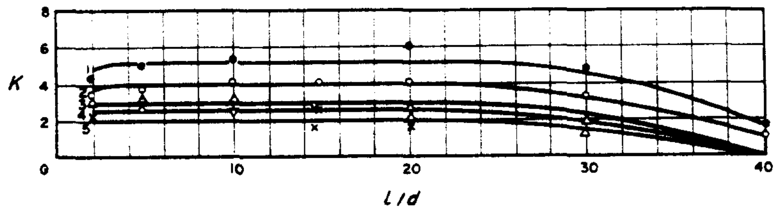

FIG. 4. The dependence of the coefficient κ at a pointed intake of gas on the relation l/d for various values of Reynolds number: (1) 4×10^3; (2) 8×10^3; (3) 12×10^3; (4) 16×10^3; (5) 20×10^3.

which gives a satisfactory approximation for the present experimental data and for the theoretical formula which is valid at $0{\cdot}6 < Pr_f < 100$ [6].

The equation

$$Nu = 0{\cdot}0202\, Re^{0{\cdot}8} \qquad (6)$$

was obtained for air. It is quite possible to use the numerical values of coefficients c and n for short tubes with the relative length of $l/d \leqslant 40d$ given in Fig. 3.

The influence of the tube length on the average heat transfer coefficient for tubes of $l/d \leqslant 40d$ can be accounted for by the principal heat transfer equation by an introduction of the error factor κ which is a function of the number Re (see Fig. 4). And the criterium for the average values of heat transfer coefficients will have the following form:

$$Nu = 0{\cdot}023\, Re_f^{0{\cdot}8}\, Pr_f^{0{\cdot}4} \left(1 + \kappa \frac{d}{l}\right) \qquad (7)$$

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