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HEAT TRANSFER TO MERCURY IN A CIRCULAR TUBE AND ANNULAR CHANNELS WITH SINUSOIDAL HEAT LOAD DISTRIBUTION

V. I. PETROVICHEV

Moscow Institute of Physical Engineering,
Moscow, U.S.S.R.

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Abstract—To calculate the maximum wall temperature for a turbulent mercury flow in circular tubes with $l/d = 44{\cdot}6$ and 67 for a sinusoidal heat load distribution along the axis the following formula was derived

$$\theta_{\max} = \frac{(t_s)_{\max} - t_a}{t_b - t_a} = 0{\cdot}22\, Pe^{0{\cdot}27}$$

at $Pe = 400$–3900, which can be used for the estimation of heat transfer in tubes with sinusoidal heat distribution sources.

The experimental data were obtained on heat transfer to mercury when $q = q_0 \cdot \sin \pi x/l$ in annular channels with $d_m = 5{\cdot}4$ mm and $3{\cdot}5$ mm which can be used for comparative estimation.

The experimental data qualitatively agree with the solution of the present problem on an electric integrator

Résumé—Pour calculer la température maximum de paroi relative à un écoulement turbulent de mercure dans des tubes circulaires pour lesquels $l/d = 44{,}6$ et 67, dans le cas d'une distribution sinusoïdale des charges thermiques sur l'axe, on a établi la formule suivante

$$\theta_{\max} = \frac{(t_s)_{\max} - t_a}{t_b - t_a} = 0{,}22\, Pe^{0{,}27}$$

pour $Pe = 400$–3900. Cette formule peut être utilisée pour le transfert de chaleur dans des tubes avec des apports de chaleur à variation sinusoïdale.

Les résultats expérimentaux obtenus sur le transfert de chaleur avec du mercure pour $q = q_0 \cdot \sin \pi x/l$ dans des conduits annulaires $d_m = 5{,}4$ mm et 3,5 mm peuvent être utilisés à titre de comparaison.

Les résultats expérimentaux concordent bien avec ceux que l'on obtient par le calcul à l'aide des machines électriques.

Zusammenfassung—Zur Berechnung der maximalen Wandtemperatur einer turbulenten Quecksilberströmung im Kreisrohr mit $l/d = 44{,}6$ und 67 bei sinusförmig längs der Achse verteiltem Wärmestrom wird die Gleichung

$$\theta_{\max} = \frac{(t_s)_{\max} - t_a}{t_b - t_a} = 0.22\, Pe^{0{,}27}$$

für Pe 400 bis 3900 abgeleitet.

Versuchswerte über den Wärmeübergang an Quecksilber mit $q = q_0 \cdot \sin \pi x/l$ in ringförmigen Kanälen mit $d_m = 5{,}4$ mm und 3,5 mm können als Vergleich dienen.

Die Versuchswerte stimmen qualitativ mit der Lösung dieses Problems durch den elektrischen Integrator überein.

Abstract—Для вычисления максимальной температуры стенки при турбулентном течении ртути в круглых трубах с $l/d = 44{,}6$ и 67 при синусоидальном распределении тепловой нагрузки вдоль оси получена расчётная формула

$$\theta_{\max} = \frac{(t_s)_{\max} - t_a}{t_b - t_a} = 0{,}22\, Pe^{0{,}27}$$

при $Pe = 400 - 3900$, которой удобно пользоваться для оценочных расчётов теплоотдачи в трубах при синусоидальном тепловыделении.

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Получены экспериментальные данные по теплоотдаче ртути при $q = q_0 \cdot \sin \pi x/l$ в кольцевых каналах d_m = 5,4 mm и 3,5 mm, которые могут быть использованы для сравнительных оценок.

Экспериментальные данные качественно согласуются с расчётом данной задачи на электроинтеграторе [6].

It is known from analytical investigations [1–4] that boundary conditions for a turbulent flow have greater effect on the heat transfer from liquids with low Prandtl numbers, i.e. from liquid metals, because of their high "molecular" thermal conductivity.

Having in view the modern development of physical power plants where heat sources are distributed sinusoidally, investigation of the influence on heat transfer of such heat load distribution along the length of a channel is of great interest. One may suppose that the influence of the sources of heat distribution along the length of a channel will also affect the heat transfer from liquid metals to a greater extent in comparison with the transfer from agents with Prandtl numbers of the order $Pr \gg 1$.

The calculation of heat transfer for a sinusoidal heat load distribution along the length of a channel for laminar flow of a heat transfer agent in a circular tube and between parallel planes was made by Dzung [5]. In a previous paper [6] we calculated the heat transfer for a turbulent flow of liquid metal when $Pr = 0{\cdot}025$ in a circular tube for various ratios of l/d. The purpose of this investigation is to check the results given in reference [6] and to perform some more calculations.

EXPERIMENTAL METHODS

The experimental installation for the investigation of heat transfer in mercury for a sinusoidal heat load distribution along the length of a channel represents (as shown in Fig. 1) a circulation contour which consists of a mercury container, a centrifugal pump, a Venturi nozzle, a differential manometer, a water cooler, a regulating valve and an experimental heat exchanger. All the parts of the installation in contact with mercury are made of stainless steel. For each series of

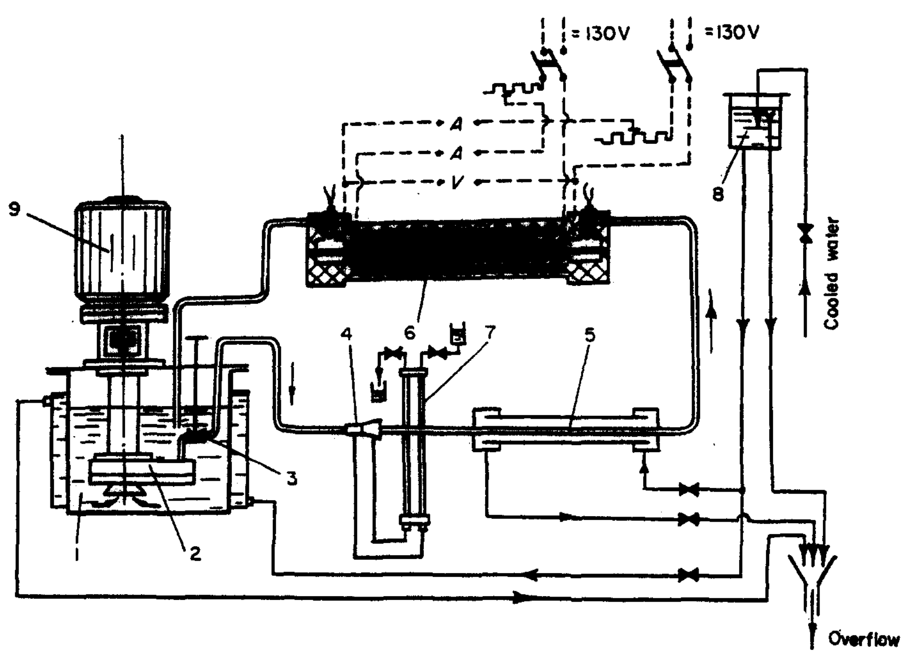

FIG. 1. Experimental installation scheme: (1) mercury container; (2) circulating pump; (3) regulating valve; (4) Venturi nozzle; (5) water cooler; (6) experimental heat exchanger; (7) differential manometer; (8) tank with a constant cooling water level.

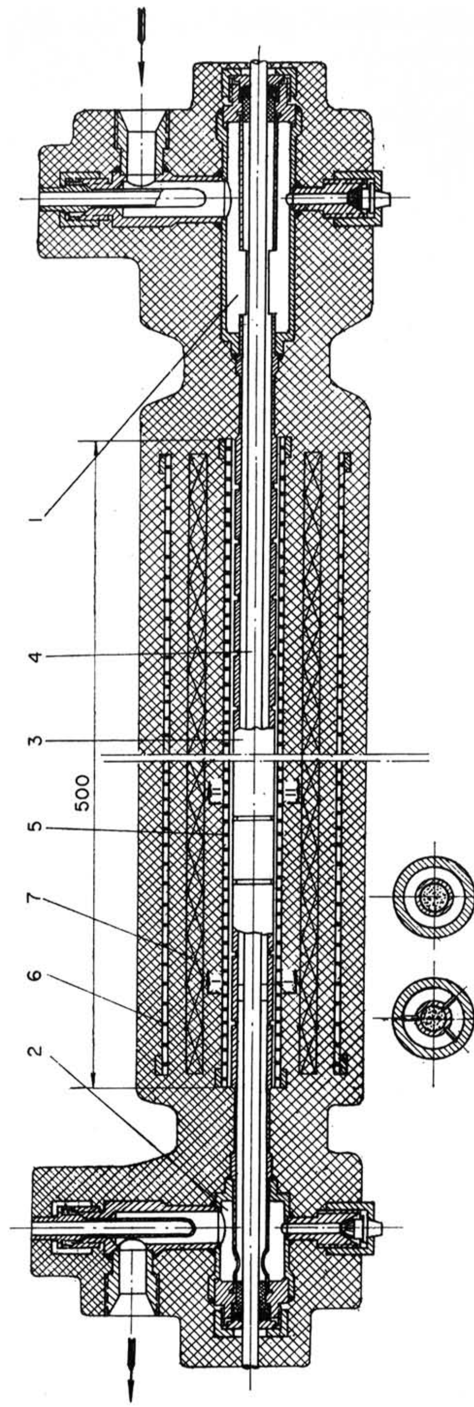

FIG. 2. Experimental heat exchanger $N2$ (1) inlet chamber; (2) outlet chamber; (3) outside tube; (4) displacer; (5) electric heater; (6) compensating electric heater; (7) heat measuring apparatus.

experiments the mercury was filtered and purified by nitric acid, alkali and alcohol before being poured into the installation. Visual observations of the mercury surface in the container during the experiments pointed to the absence of any contamination.

The tests were carried out with two different experimental heat exchangers. The experimental heat exchanger $N1$ was made in the form of a steel tube 600 mm length, with an outside diameter of 14 mm and an inside diameter of 9 mm. The tube has inlet and outlet chambers for mercury at each end, from the cross-cut ends of which it is possible to clean the inside of the tube surface and visual observation of its state can be carried out. The tube was divided into twelve parts by circular grooves to decrease axial heat losses, which could happen for low Pe numbers. An electric heater was wound over the outside tube surface and it provided for heating according to the sinusoidal law.

Control of the winding accuracy was performed by measuring its resistance along the length by a Wheatstone bridge to within 0·01 Ω for a total resistance of the heater equal to 20 Ω. A compensating heater also wound according to the sinusoidal law, was mounted above the main heater to compensate for heat losses. Three sections of heat measuring apparatus were mounted between the heaters to control the compensation of heat losses.

The electric heaters are fed by a direct current of 130 V from a constant current source. The stability of mercury cooling is possible due to the tank with cooling water at a constant level. The wall temperature is measured by twelve copper–constantan thermocouples placed in the middle of each part of the tube (twelve in all). Mercury intake temperature and the temperature at which mercury leaves the heat exchanger are also measured by copper–constantan thermocouples placed in the wells of the chambers.

The electromotive force of the thermocouples is measured by a potentiometer. The maximum heat unbalance between the electrical power supplied to the experimental heat exchanger and mercury heating does not exceed 3–4 per cent. The experimental heat exchanger $N2$ has the same length of 600 mm and is made of the same material, but as shown in Fig. 2, it was divided into twenty-four parts and differs from the first heat exchanger in having a tube of larger diameter (d_i = 13·5 mm and d_e = 17·5 mm, l/d = 44·6). The experimental heat exchanger inlet and outlet chambers were designed in such a way, that it is possible to insert displacers with an outside diameter of 8·1 mm and 10 mm into the tube, forming an annular clearance with the ratio d_2/d_1 = 1·68 and 1·35 for the mercury flow through it. The displacers are put into the experimental heat exchanger in jackets at both sides. This prevents deformations when thermal expansion occurs and allows the displacers to be regulated in order to measure the influence of possible eccentricity. Three narrow stops are placed into one section of the displacers at an angle of 120° to avoid eccentricity. The winding of the heater and the whole measuring scheme is the same as with the experimental heat exchanger $N1$. Thus, the heat exchanger $N2$ allowed heat transfer to mercury for two geometries to be measured: in a circular tube with l/d = 44·6 and in annular clearances with the ratios l/d = 110 and 170.

The experiments were performed at an average mercury temperature of 40° to 60°C and at a maximum heat flow at the centre of the tube 50×10^3 kcal/m^2 hr.

RESULTS

Experiments were held with the same experimental heat exchangers and the same thermocouples on determination of heat transfer to mercury in a circular tube with boundary conditions q_s = const. to control the correctness of the work of the installation. In this case a new electric heater with a constant space between the turns of the wire was wound over the tube. The experimental data for the heat exchanger $N1$ are exact to ±5 per cent and coincide with those given by Johnson [8] on heat transfer from mercury with q_s = const. For the experimental heat exchanger $N2$ the values are somewhat higher due to different intake conditions.

If we have a sinusoidal heat load distribution along the length of the channel, then the value of the maximum wall temperature is of particular interest. The results are therefore given in the form of a relationship between the maximum dimensionless wall temperature and the number Pe. The distribution of the wall temperature along the channel was found for each experiment. An average liquid temperature distribution along the length may be found from the heat balance equation. Thus, we can find the local temperature differences and calculate the local dimensionless heat transfer coefficient Nu_x with the help of the temperature distribution throughout the wall and in the liquid. It must be emphasized that the temperature throughout the wall in the maximum region has a gently sloping form. An exact definition of the position of this maximum is therefore not quite easy and strictly we may only speak of the value of the maximum.

The results of heat transfer measurements to turbulent mercury flow in a circular tube with sinusoidal heat load distribution along the length of the tube are shown in Fig. 3 in the form of the dependence

$$\frac{(t_s)_{\max} - t_a}{t_b - t_a} = f(Pe)$$

for l/d = 44·6 and 67. Consequently from the examination of this figure all the experimental

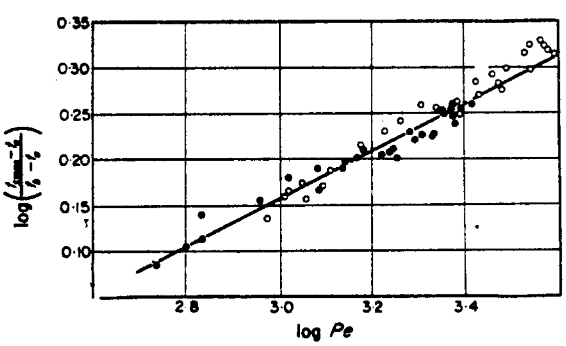

FIG. 3. Heat transfer to mercury in the circular tube at $q = q_0 \cdot \sin \pi x/l$; ○ l/d = 44·6; ● l/d = 67.

points within these co-ordinates may be represented with an exactness equal to ±7 per cent by a single dependence

$$\theta_{\max} = \frac{(t_s)_{\max} - t_a}{t_b - t_a} = 0{\cdot}22\, Pe^{0{\cdot}27}$$

The same dependence for the channels with an annular section (l/d = 110 and 170) is depicted in Fig. 4. An equivalent hydraulic diameter

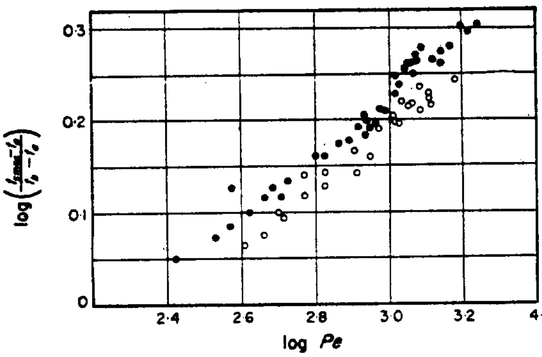

FIG. 4. Heat transfer to mercury in the annular channels at $q = q_0 \cdot \sin \pi x/l$; ○ d_m = 5·4 mm, l/d = 110; ● d_m = 3·5 mm, l/d = 170.

$d_m = d_2 - d_1$ is taken here for a determinative value in numbers Pe. So one can draw the conclusion that $\theta_{\max}$ in annular channels at $q = q_0 \cdot \sin \pi x/l$ in a first approximation will be uniform for both annular clearances (d_m = 5·4 mm and 3·5 mm). Thus we can see that for annular clearances at $q = q_0 \cdot \sin \pi x/l$ besides the ratio l/d, the equivalent diameter d_m influences the heat transfer. Strictly speaking, the equivalent diameter cannot be applied to the calculation of heat transfer from a liquid metal in an annular clearance.

Fig. 5 gives a comparison of the local number Nu_x for a circular tube of the present investigation with the solution of the problem on an electric integrator [6]. The solution was derived for two approximations of a turbulent thermal diffusivity coefficient in previous works [1, 2, 7].

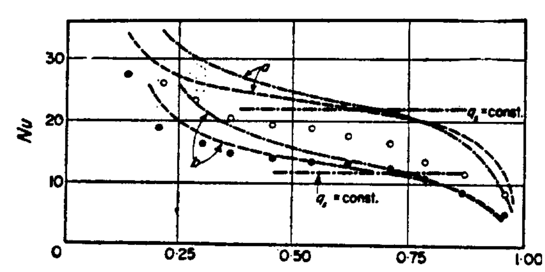

FIG. 5. Comparison of local mercury heat transfer in circular tubes for $q = q_0 \cdot \sin \pi x/l$ with the solution on an electric integrator [6] for $Re = 100 \times 10^3$, Pr = 0·025, (a) with $a_T = \nu_T$, (b) a_T according to [7], ○ l/d = 44·6, Pe = 2480; ● l/d = 67, Pe = 2440; - - - for l/d = 67; —·— for l/d = 44·6.

Fig. 5 shows that the data of the experimental investigation are qualitatively in agreement with the solution and are placed between two extreme limits of approximation a_T. The experimental data for $l/d = 44{\cdot}6$ are somewhat higher in comparison with the calculated values and it can be explained, perhaps, by the different intake condition of the experimental heat exchanger $N2$, which differs from that of the heat exchanger $N1$, which has a rounded entrance.

While winding the electric heater sinusoidally the space between the turns of the wire and the edges was made comparatively larger and it essentially influences the accuracy of the determination of local values along the length of a channel. But the heat flow values and the wall temperatures decrease at the edges and these regions are therefore of less interest for the investigation. We are interested more in the region of a maximum heat flow and a maximum wall temperature (i.e. the regions at the centre of the tube) where the space between the turns of the wire is of the order of thickness of the wall tube and the error due to it does not exceed 3 per cent. The estimation of heat expansion over the tube wall gives evidence of the fact that even at Peclet numbers greater than 100 the error due to expansion does not exceed 0·1 per cent.

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