INFLUENCE OF TWO-DIMENSIONAL ARTIFICIAL ROUGHNESS ON CONVECTIVE HEAT TRANSFER

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Аннотация—Приведены результаты экспериментального исследования теплоотдачи труб с двухмерной искусственной шероховатостью на их наружной поверхности. Отвод тепла в опытах осуществлялся трансформаторным маслом и водой. Измерены локальные температуры стенки шероховатой трубы и сделана попытка объяснить механизм интенсификации теплообмена под воздействием искусственной шероховатости. Результаты исследования сопоставлены с ранее полученными теоретическими и экспериментальными результатами. Проведено обобщение опытных данных и получено критериальное уравнение, с которым хорошо согласуется большое количество экспериментальных результатов других авторов.

NOMENCLATURE

Nu_0, Nu, Nusselt numbers for the smooth tube and a similar tube with two-dimensional roughness on its surface, respectively. The values are evaluated at the mean temperature of the boundary layer;

Nu_{0f}, Nu_f, the same numbers as above, evaluated at mean temperature of the fluid;

Re, Re_f, Reynolds numbers evaluated at mean temperature of the boundary layer and at mean temperature of the fluid, respectively;

Pr, Pr_f, Pr_w, Prandtl numbers, evaluated at mean temperature of the boundary layer, at mean temperature of the fluid and at mean temperature of the wall, respectively;

ξ_0, ξ, friction factors for smooth tube and a similar rough one, respectively;

U, mean velocity of fluid;

U_s, fluid velocity at the boundary of a laminar sub-layer;

Pr_t, turbulent Prandtl number;

h, height of roughness asperity;

b, width of roughness asperity;

S, pitch of roughness asperities;

H, heat-transfer area, smooth pipe;

φ, ratio of area of rough surface to smooth;

r_1, r_2, internal, external radii of pipe;

λ, thermal conductivity of tube wall material;

t_{w_1}, t_{w_2}, internal and external tube surface temperatures [°C];

q, heat flux over external surface of tube;

α, coefficient of heat transfer;

Q, total heat transmitted by whole tube surface.

$$K_f = \frac{Nu_f}{Pr_f^{0.43}(Pr_f/Pr_w)^{0.25}}$$

$$K_{f_0} = \frac{Nu_{f_0}}{Pr_{f_0}^{0.43}(Pr_{f_0}/Pr_{w_0})^{0.25}}.$$

THE METHOD of artificially roughening the surface of a tube in a turbulent stream in order to increase heat transfer to liquid or to gas is one of the several methods which deserve careful study.

Despite the fact that relatively few systematic investigations of this problem are available,

there is sufficient evidence in the data already to hand that a rational geometric roughness pattern artificially imparted to the wall of a tube improves heat transfer considerably. Indeed it is considered that this method has great practical use [1].

Besides, the method also deserves theoretical interest as the presence of the elements of artificial roughness on the surface may prove useful in revealing the nature of the laminar sub-layer and the character of processes occurring the immediate vicinity of the wall, especially if fluids with different Prandtl numbers are used.

The simplest theoretical relation which allows one to calculate the Nusselt number, Nu, for heat transfer into a turbulent stream within a smooth tube is the following one, obtained by Prandtl on the assumption of the existence of an analogy between momentum and heat transfer.

$$Nu_0 = \frac{\xi}{8} \cdot \frac{Re\,Pr}{1 + U_\delta/U(Pr - 1)} \qquad (1)$$

The derivation of (1) is based on assumptions about the similarity between the velocity and the temperature distributions and about the absence of a transition region between the laminar sub-layer and a turbulent core.

Owing to the difficulty of estimating the thickness of the laminar sub-layer accurately with fluids of $Pr > 1$, theoretical heat-transfer calculations of a turbulent stream are also inadequate although these formulae embody a reasonable physical concept for the laminar sub-layer, the transition layer and turbulent core. In such formulae, moreover, allowance is made for the thermal resistance of each of these elements in contributing to the total resistance of the stream.

The above difficulties relate to heat-transfer calculations for streams in smooth tubes, in which the velocity profile is known with some accuracy. The conditions are far worse in the case of estimating heat transfer through rough walls because the velocity distribution close to rough walls is not known.

Nunner [2] has made the most extensive study of the effect of artificial surface roughness on convection heat transfer. As however the main conclusions in this work are open to question, we will deal with them in some detail.

Nunner considered heat transfer to air moving within a tube roughened on the inside. The velocity distribution was measured at the exit end of the experimental tube. This was done by means of a 1 mm dia. Pitot tube. Temperature distribution of the air stream was also measured. A Pitot tube might be regarded as a rather crude instrument for measuring the velocities close to the wall, for it is known that similar tests have been done with 0·025 mm tube openings and with hot wire anemometers using 0·007 mm dia. wire.

Bearing this in mind one must conclude that Nunner was really measuring the velocity distribution in the turbulent region and not that of the transition zone or the laminar sub-layer. His results therefore do not allow one to evaluate the thickness of the laminar sub-layer in a rough tube.

Nunner's temperature distribution results showed that the temperature profiles for the case of rough and smooth tube were hardly distinguishable. In this connexion it should be observed that the distances from the wall to the measuring stations was of the order of 0·5 mm.

Nunner's measurements showed that with equal Reynolds numbers the velocity profile is more drawn out in the case of the rough tube than with the smooth. It is known that this had already been established by Nikuradse in his work [3] now considered as a classic. Whereas the formula for velocity distribution based on Nikuradse's results is only used for the turbulent region, Nunner extrapolated his results to the region close to the wall and came to the conclusion that the laminar sub-layer was thickened as a result of surface roughness. On the other hand he took into account the possibility of breakdown of the laminar sub-layer on the roughness asperities, and taking into account both these factors came to the conclusion that the laminar sub-layer was the same in both rough and smooth tubes. Nunner then combined the velocity and the temperature profiles on the rough tube and this led him to believe that the effect of surface roughness on heat transfer was similar to the effect of increasing the Prandtl number. In other words Nunner concluded that roughening the surface had the effects of reducing the thermal resistance of the centre part of

the stream, the thermal resistance of the laminar sub-layer remaining unaltered.

To explain this, Nunner assumed that the influence of the wall surface roughness was to cause turbulence at the outer limits of the core (or centre portion) of the stream, this turbulizing of the core not affecting the thermal resistance of the sub-layer, incidentally.

On this assumption Nunner used the momentum–heat analogy and arrived at the following relationship, similar to (1)

$$Nu = \frac{\xi}{8} \cdot \frac{Re\, Pr}{(U_\delta/U)\,[Pr\,(\xi/\xi_0) - 1]} \qquad (2)$$

Having used the empirical formula

$$\frac{U_\delta}{U} = 1\cdot5\, Re^{-1/8}\, Pr^{-1/6}, \qquad (3)$$

Nunner represented the relationship (2) in the following final form

$$Nu = \frac{\xi}{8} \cdot \frac{Re\, Pr}{1 + 1\cdot5\, Re^{-1/8}\, Pr^{-1/6}\,[Pr\,(\xi/\xi_0) - 1]} \qquad (4)$$

$Pr_t = 1$ was assumed in the derivation of both formulae (2) and (1).
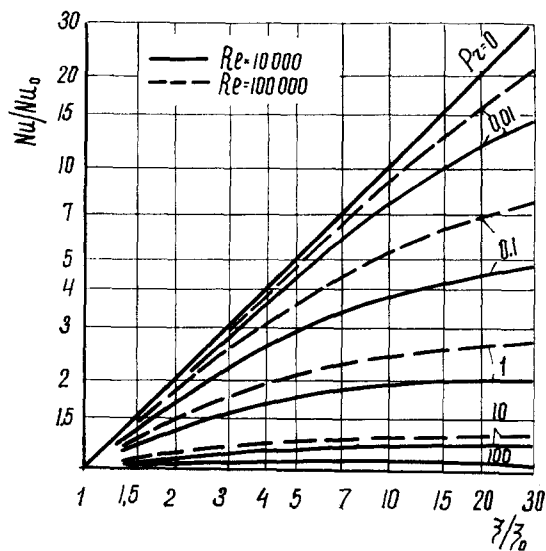
Figure 1 shows results of calculations for

$Re = 10^4$ and $Re = 10^5$ by Nunner using (4) for values of Pr ranging from 100 to zero. Nu/Nu_0 is given as a function of ξ/ξ_0.

These results reveal that hardly any increase in heat transfer takes place in the case where the hydraulic resistance of a tube is enhanced by surface roughness with high Prandtl number fluids. In the case of liquid metal media with $Pr \ll 1$ increase in ξ/ξ_0 should evoke a multifold
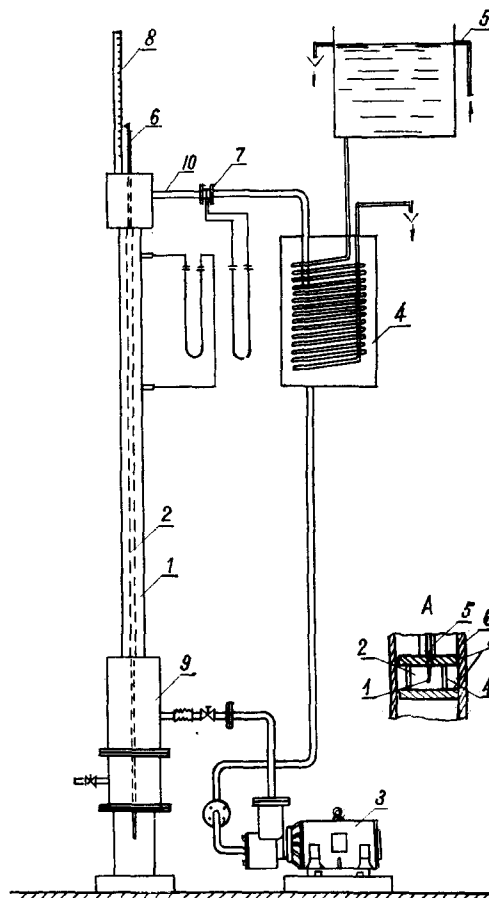

FIG. 2. Schematic diagram of experimental unit:
1—annulus; 2—experimental tube; 3—pump with electric driver; 4—cooler; 5—upper tank; 6—indicator of thermocouple position; 7—orifice; 8—scale; 9—lower tank; 10—draining tube.

A—sliding thermocouple:

1—thermocouple junction; 2—chamber; 3—washers; 4—connecting piers; 5—thin brass tube; 6—experimental tube.

FIG. 1. $Nu/Nu_0 = f(\xi/\xi_0)$ predicted by equation (4) (from W. Nunner's work [2]).

increase in Nu/Nu_0. It is evident that these results are in full agreement with the assumptions made by Nunner in deriving (4), but they are not confirmed by experimental data published by Fedynsky [4] and Brauer [5] who made fairly detailed studies of heat transfer between rough walls and turbulent water streams.

The great interest in the problem prompted us to investigate the effect of artificial surface roughening on the heat transfer between a wall and a turbulent flowing liquid. The tests were done on the experimental rig depicted in Fig. 2.

Tubes heated by electric current passing through them were located axially within a circular channel of 33 mm internal diameter. The outsides of the tubes were roughened. Heat from the rough surfaces was transmitted to the liquid flowing along the annular channel. A travelling (sliding) thermocouple was fitted to the heated tube to measure its internal surface temperature. The junction of this thermocouple was located in a chamber insulated with teflon (fluorocarbon trans.) and this consisted of two washers tightly fitted to the internal tube surface and connected together by two thin partitions. The distance between the washers was 4 mm. The end of the chamber was fixed to a thin brass tube which allowed the chamber plus thermojunction to be moved up and down, and itself constituted the entry for the leads to the experimental tube. The heat from the outside of the tubes was extracted by means of transformer oil and water.

Tests carried out with transformer oil incorporated four stainless steel tubes with varying degrees of surface roughness. Data on the latter are shown in the Table 1.

The roughness on tubes 1 and 2, with outside dia. 10·2 mm, wall thickness 2·25 mm, was put in by machining on a lathe.

In the case of tubes 3 and 4, outside dia. 10 mm, wall thickness 1·15 mm, "roughness" consisted of 0·68 mm nichrome wire in the form of rings, lead-soldered to the surface. Diagrams of both forms of artificial surface roughness are shown in Fig. 3.

The length of all four experimental tubes was 1642 mm. Wall temperature measurements were made 1100–1200 mm from the entry of the liquid to the annular channel. Because the surface roughening extended over the whole
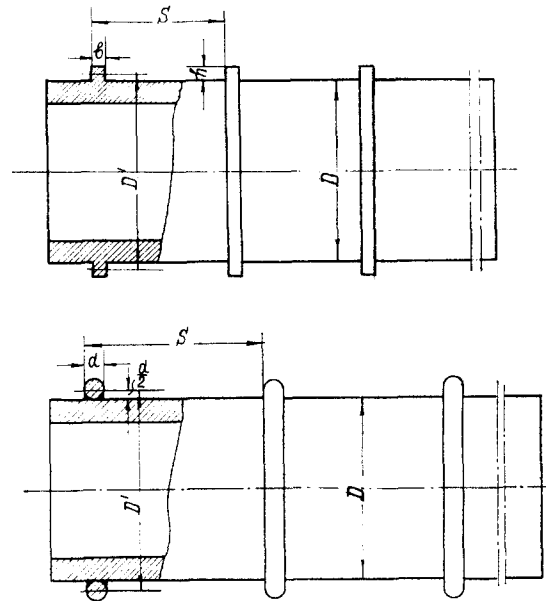

Fig. 3. Profiles of roughness elements.

Table 1

No. of tubes	Height of roughness elements h (mm)	Width of roughness elements b (mm)	Pitch between roughness elements S (mm)	Pitch to height ratio S/h	Ratio of rough surface area to smooth tube surface area φ
1	0·30	0·30	2·45	8·2	1·244
2	0·30	0·30	4·90	16·4	1·222
3	0·68	0·68	20·00	29·5	1·050
4	0·68	0·68	40·00	59·0	1·025

length of the experimental tubes, and the equivalent diameter of the annular channel was about 23 mm, it can be reckoned that the heat-transfer experiments were carried out under fully stabilized thermal and hydrodynamic conditions.

In working out the results, much attention was devoted to the question of correctly defining the tube surface roughness. It can be seen from Fig. 3 that for the ring or annular roughness elements with rectangular cross section, in tubes 1 and 2, the increase in surface area over that of the smooth tube is given by:

$$\varphi = 1 + \frac{2}{S/h} \qquad (5)$$

For tubes 3 and 4, half the ring surface area was added to the smooth tube surface area.

The mean temperature of the external surface which corresponded to the measured internal temperature was calculated from:

$$t_{w_2} = t_{w_1} + \frac{q_r r_1^2}{4\lambda} \left[2 \ln \frac{r_2}{r_1} + 1 - \left(\frac{r_2}{r_1}\right)^2 \right]. \qquad (6)$$

This expression applies to a cylinder with internal source of heat and an adiabatic internal surface [6]. In this formula

$$q_r = \frac{2qr_2}{r_2^2 - r_1^2}$$

the heat flux q was found by dividing the total heat released Q by the product φH.

The analysis revealed that in the case of tubes 1 and 2 this method of determining the mean temperature of a rough wall gave the most accurate results, whilst in the case of tubes 3 and 4 the error in heat-transfer coefficient with transformer oil amounted to some 3 per cent. In experiments with water, carried out with tubes 3 and 4 only, it was hard to estimate the errors in heat transfer, but in any case this did not exceed some 10 to 12 per cent.

Before the tests were carried out with the rough tubes, heat-transfer tests were done on the rig in Fig. 2 using smooth tubes in turbulent streams of transformer oil and water. Logarithmic plots of these results showed that they were satisfactorily expressed by Mikheev's criterial equation [7]

$$Nu_0 = 0{\cdot}021 \, Re_f^{0{\cdot}80} \, Pr_f^{0{\cdot}43} \, (Pr_f/Pr_w)^{0{\cdot}25} \qquad (7)$$

The tests on the rough tubes then followed these.

The mean heat-transfer coefficients were determined from the expression

$$a = \frac{Q}{(t_{w_2} - t_f) \, \varphi H}. \qquad (8)$$

Figure 4 shows experimental results expressed in terms of relations between K_f and Re_f.

Some of Brauer's [5] and Fedynsky's [4] results are also shown. It should be mentioned that in plotting these results allowance was made for increase in tube surface area due to surface roughness.

The results on Fig. 5 show that the effect of artificial surface roughening is to increase the heat transfer with transformer oil ($Pr \simeq 80$) to a greater degree than with water. Moreover the increase in heat transfer in both cases, water and oil, exceeds that estimated by equation (4).

Both Fedynsky's tests and those of Brauer clearly demonstrate that the increase in heat-transfer rate due to two dimensional surface roughening depends on the ratio S/h and maximizes at a certain value of this ratio. Use is made of this relationship in plotting results with water in Fig. 4. The effect of ratio S/h on the heat-transfer rate for various degrees of roughness is shown in Fig. 5.

The curves in Fig. 4 are constructed from data obtained from the rig, Fig. 2, Fedynsky's [4], Brauer's [5] and Nunner's [2] tests. These are all expressed in terms of the relation $K_f/K_{f_0} = f(S/h)$.

The curve which represents this relation for water and for air $Pr = 0{\cdot}72$ and $9{\cdot}0$ clearly shows that K_f/K_{f_0} has a definite maximum within the interval $S/h = 12$ to 14, attaining then a value $K_f/K_{f_0} = 2{\cdot}4$.

Transformer oil, with $Pr = 80$ behaves in a similar way. Although there is insufficient experimental data to hand to establish the maximum value of S/h accurately in this case, the general character of the curves in Fig. 5 gives some basis for assuming that transformer oil gives a maximum K_f/K_{f_0} also within $S/h = 12$ to 14, and is of the order $K_f/K_{f_0} = 2{\cdot}6$ to $2{\cdot}7$.

It is evident from Fig. 5 that Brauer's tests corresponding to $S/h = 9{\cdot}6$ give K_f/K_{f_0} much less than those corresponding to the curve
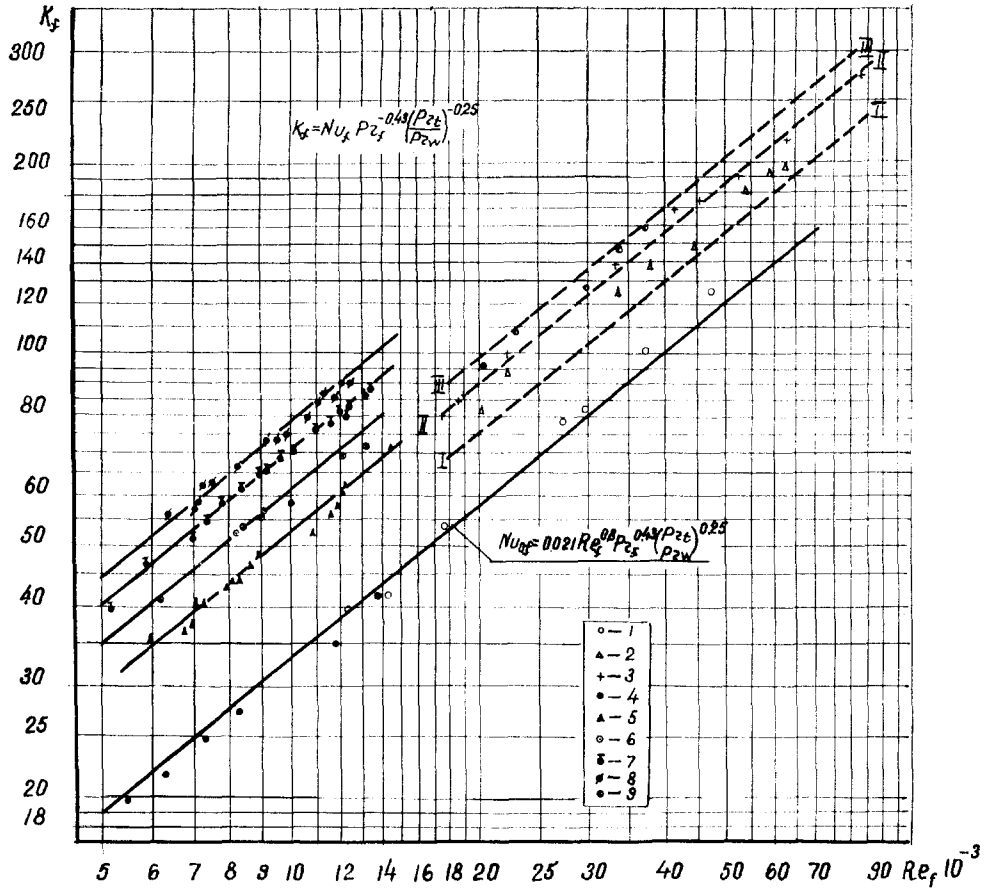
FIG. 4. Plot of K_f versus Re_f for water and transformer oil.

1—smooth tube (water); 2—tube No. 4 (water); 3—tube No. 3 (water); 4—smooth tube (transformer oil); 6—tube No. 3 (transformer oil); 7—tube No. 1 (transformer oil); 8—tube No. 2 (transformer oil); 9—Fedynsky's experimental points [4] for $S/h = 6\cdot7$ (water); I–I—Brauer's experimental curve [5] for $S/h = 69\cdot3$ (water); II–II—Brauer's experimental curve [5] for $S/h = 34\cdot3$ (water); III–III—Brauer's experimental curve [5] for $S/h = 9\cdot4$ (water).

$K_f/K_{f_0} = f(S/h)$ for water and air. This is because Brauer's method of measuring wall temperature led to higher mean wall temperatures at low values of S/h and high heat fluxes. Fedynsky's results are much more reliable at low values of S/h. Nunner's results are more reliable too in this region. These results having been obtained with copper and brass tubes, steam heated.

The satisfactory agreement between Fedynsky, Brauer and Nunner in test results requires some comment. Nunner's experimental results give the impression that the increase in heat transfer with roughness for the case of air with $S/h = 10$ is higher than with water and even with transformer oil. However, if one familiarizes oneself with the method Nunner employs in correlating his results, it appears that the increase in heat-transfer rate of the experiments is of the same order as those with water. It should be mentioned that this and other questions are dealt with in greater detail in reference 8.

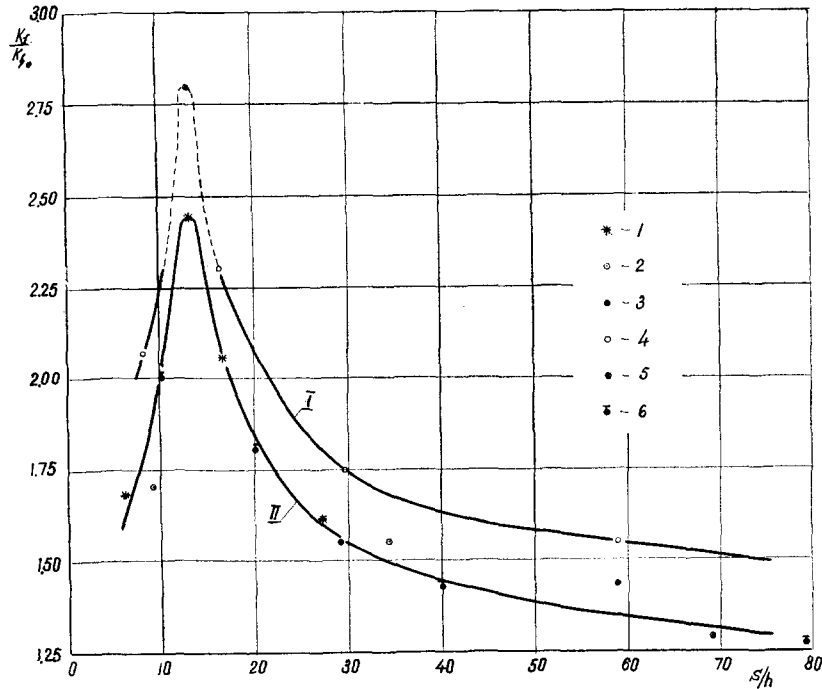The disagreement between Nunner's [2 theory and experiment mentioned above can,

FIG. 5. Plot of K_f/K_{f_0} versus S/h. I—curve for transformer oil; II—curve for water and air.
1—Fedynsky's data [4] (water); 2—Brauer's data [5] (water); 3—data obtained on the unit shown in Fig. 2
(water); 4—the same for transformer oil; 5—value of K_f/K_{f_0} for transformer oil calculated from equation (13);
6—Nunner's data [2] (air).

we think, be explained by the results of the observations in connexion with the wall temperature measurement along tube 4 of the experiments described in this paper. This experiment was done with surface roughness represented by nichrome wire rings, 0·68 mm dia., 40 mm pitch. When these measurements were taken, heat was abstracted by water and by transformer oil. In order to obtain a more accurate picture of the temperature distribution along the tube, the heat-transfer rates were stepped up considerably; the temperatures of the internal surface were fixed whilst the thermocouple was displaced every 5 mm.

The results of these measurements are shown on Figs. 6 and 7. It is evident from the graphs that no quantitative conclusions can be drawn from them, but they do give us an insight into one important conclusion about the nature of the phenomena.

As might have been anticipated, because of

the thickening of the walls at the rings and the corresponding increase in heat transfer rate, the wall temperatures here are minimal. However, if this circumstance was the only reason for the lowering of temperatures at the rings, the maximum temperature of the wall should coincide with a point midway between the rings. In actual fact, these points are in all cases displaced from the midway position in the direction of motion of the fluid.

Reasons for this cannot be attributed to the temperature increase of the fluid, which in the case of water nowhere exceeded some 6°C and in the case of oil, 0·7°C—so that over the inter-ring distances this would be negligibly small.

It should be considered therefore that the only reason for the displacement of the maximum temperature points from the midway positions must be the gradual increase in thermal resistance along the heating surface and a sharp drop in this resistance close to a ring. Noticeable
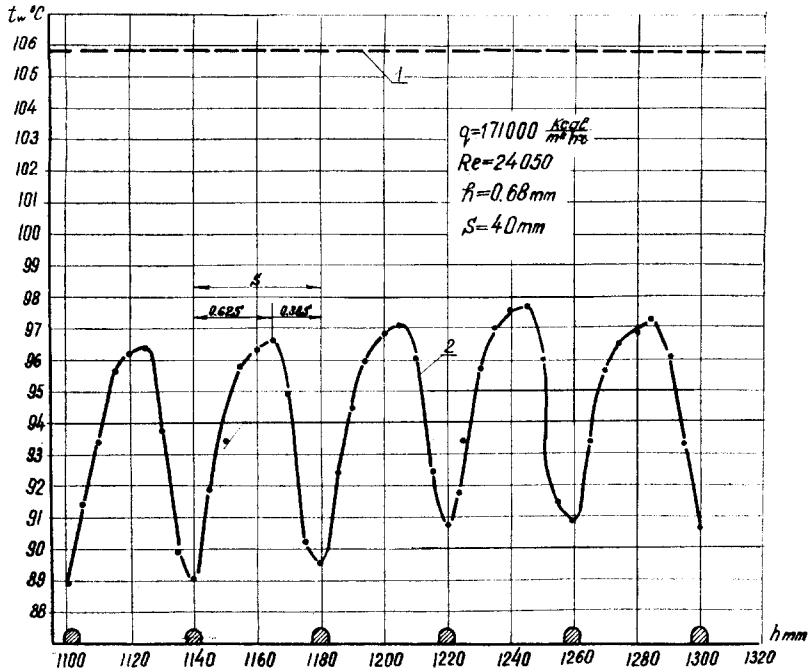
FIG. 6. Local wall temperatures of tube No. 4 (water as heat-transfer medium).

I—value of t_{w2} at $Re_f = 24 \times 10^3$ and $q = 171 \times 10^3$ kcal/m² h calculated from equation (7) for smooth tube.
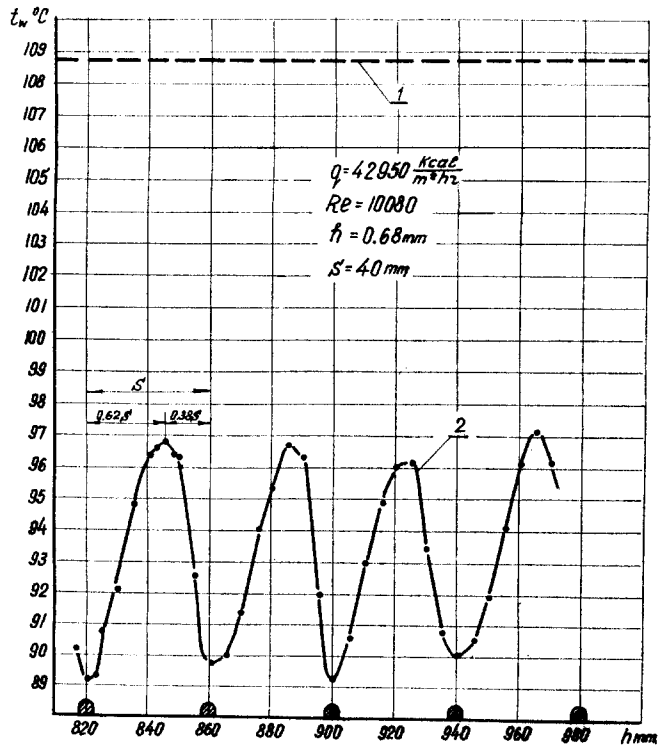
FIG. 7. Local wall temperatures of tube No. 4 (transformer oil as heat-transfer medium).

I—value of t_{w2} at $Re_f = 10^4$ and $q = 42\,950$ kcal/m² h calculated from equation (7) for smooth tube.

changes in thermal resistances when using media such as water, and especially transformer oil, can only be attributed to local changes in the thickness of the laminar sub-layer.

The effect of the surface roughness protrusions on the thickness of the laminar sub-layer seems to us to be according to expectation; the elements of surface roughness, as a rule, jut out from the laminar sub-layer and the protrusions reach the transition layer, and even touch the core of the flow close to its outer boundary. One should assume that the laminar layer breaks up at the surface protrusions for it becomes subject to the turbulent part of the stream. Thus, the laminar sub-layer should build up gradually behind each element of surface roughness until it reaches the next one.

In addition to reducing the thickness of the laminar sub-layer, each element of surface roughness, as assumed by Nunner, generates vortices in the core of the stream or in transition region. It is more than likely that the vortices themselves affect the laminar sub-layer, helping to diminish its thermal resistance.

It should be observed that Fedynsky, in his work on heat transfer from rough surfaces [4], definitely drew attention to the upsetting of the laminar sub-layer by roughness protrusions. Other authors [9] have also made similar observations.

If we assume that this is a correct interpretation of the mechanism of the effect of artificial surface roughening in convective heat transfer, it is to be expected that very great improvements in heat-transfer rate should accrue with roughened surfaces when Pr exceeds unity and attains greater values.

Several other characteristic features of the process also become quite understandable if we assume this picture to be the correct one. In particular, the effect of S/h on the increase in heat-transfer rate and also the existence of an optimum value of this ratio. From the data on Fig. 5 an excessive approach between the surface roughness protrusions should lead to a decrease in heat-transfer rate because of stagnation of the fluid between the protrusions, and also because of the inferior conditions of diffusivity of vortices in the direction of the laminar sub-layer. By similar reasoning, excessive increase in

distances between protrusions, and hence the establishment of normal laminar sub-layer thickness should lead to a weakening of the effects of the vortex generation. Finally, keeping the heat-transfer rate at a certain level, increase in protrusion height should lead to a concomitant increase in inter-protrusion distance, for in this way favourable conditions for vortex diffusion into the laminar sub-layer are created.

In Fedynsky's work [4] tests are done with tubes incorporating rectangular and triangular shaped surface protrusions. It is found that the increase in heat-transfer rate is the same for both shapes. Similar conclusions can be made from Nunner's work [2]. One can consider that these experimental facts do not contradict the above picture of the mechanism of the phenomena.

If we combine the experimental data obtained from the rig (Fig. 2) with those of Fedynsky [4] and Brauer [5] we can conclude that the width of the annular gap hardly affects heat-transfer rate, that is, if its dimensions exceed the height of the surface protrusions by several times. If we assume that artificial surface roughening affects mainly the conditions in the laminar sub-layer, whose thickness is negligible in comparison with the annular gap width, it is to be expected that the thickness of the latter hardly has any effect.

If, finally, the principle reason for increase in heat-transfer rate with surface roughness is the decrease in thermal resistance of the laminar sub-layer, one would not expect the effect to be so marked in the case of liquid metals as might be deduced from (4).

Evidently it should be possible to describe the effect of roughness in increasing heat-transfer rate by the Fourier–Ostrogradsky and Navier–Stokes equations. The test results shown on Fig. 5 give some grounds for expecting the boundary conditions to include the parameter S/h as well as those connected with a smooth surface, and this should be the basic geometric characteristic of a rough surface.

It follows therefore that these equations in criterial form should be expressed as follows

$$Nu_f = f\left(Re_f, Pr_f, \frac{S}{h}\right). \qquad (9)$$
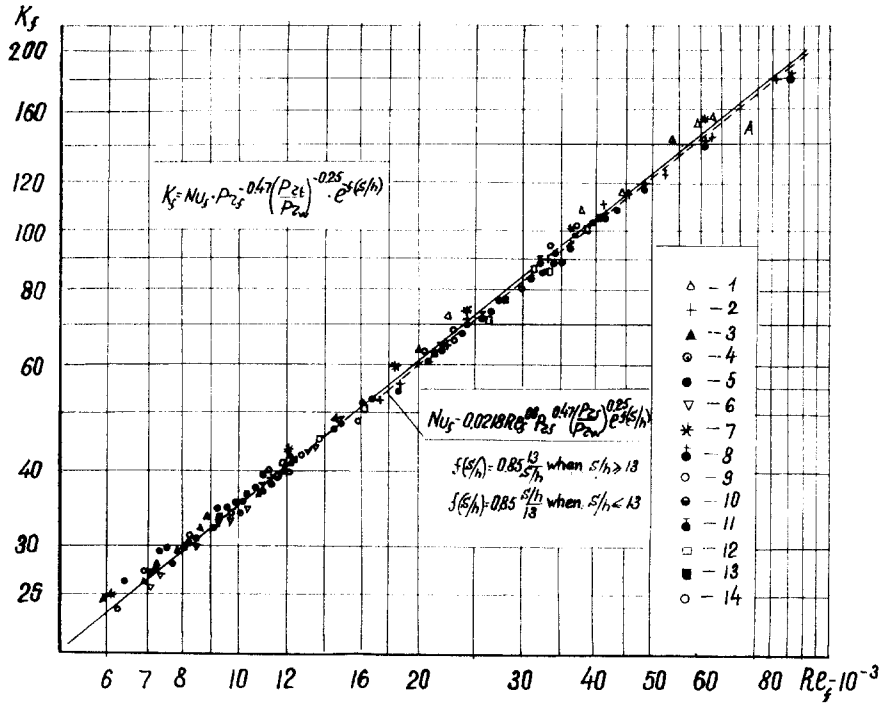
The form of the curves shown in Fig. 5

Fig. 8. Correlation of experimental data.

1—water, tube No. 4; 2—water, tube No. 3; 3—transformer oil, tube No. 4; 4—the same, tube No. 3; 5—the same, tube No. 2; 6—the same, tube No. 1; 7—Nunner's data 2, air, $S/h = 10.2$; 8—the same, $S/h = 20.4$; 9—Fedynsky's data [4], triangular profile, $S/h = 20.0$; 10—the same, rectangular profile $S/h = 13.3$; 11—the same, rectangular profile, $S/h = 16.7$; 12—the same, triangular profile, $S/h = 13.3$; 13—the same, triangular profile, $S/h = 30.0$; 14—the same, rectangular profile $S/h = 6.7$; curve A—Brauer's data [5] $S/h = 67.7$ and $S/h = 34.3$.

allows one to find the form of relationships which take into account the effect of the geometric surface roughness characteristic; Equation (9), then, is written as follows

$$Nu_f = Re_f^n \, Pr_f^m \, \exp\left[f\left(\frac{S}{h}\right)\right] \qquad (10)$$

Existing experimental data on the mean coefficient of heat transfer for artificially roughened surfaces in turbulent streams of air, water and transformer oil have contributed to the following tangible form of the relationship (10):

$$Nu_f = 0.0218 \, Re_f^{0.8} \, Pr_f^{0.47} \left(\frac{Pr_f}{Pr_w}\right)^{0.25} \exp\left[f\left(\frac{S}{h}\right)\right] \qquad (11)$$

where:

$$f\left(\frac{S}{h}\right) = 0.85 \frac{(S/h)_{opt}}{S/h} \quad \text{when} \quad \frac{S}{h} \geqslant \left(\frac{S}{h}\right)_{opt}$$

and

$$f\left(\frac{S}{h}\right) = 0.85 \frac{S/h}{(S/h)_{opt}} \quad \text{when} \quad \frac{S}{h} \leqslant \left(\frac{S}{h}\right)_{opt}$$

Equation (11) may be considered valid over the range $Re = 6 \times 10^3$ to 90×10^3 and within the Pr range 1 to 80.

According to the curve shown in Fig. 5, the optimum value of S/h can be taken as 13.

Figure 8 shows that, within the experimental error, equation (11) adequately expresses the data of the present experiments with water and transformer oils, Fedynsky's water data,

Nunner's air data and Brauer's water data for high values of S/h. It is considered therefore that expression (11) is adequate for engineering calculations.

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Abstract—Results are presented of experimental investigation of heat transfer from tubes with two-dimensional artificial roughness elements on their outer surfaces. Removal of heat was accomplished by means of transformer oil and water. The local temperatures of rough surfaces were measured and an attempt made to reveal the mechanism of increase in heat-transfer rate due to two-dimensional roughness. The results are compared with previously existing results, both theoretical and experimental. The experimental data are correlated and an empirical equation is obtained. Experimental data of other authors show good agreement with this equation.

Résumé—On a présenté les résultats des recherches expérimentales du transport de chaleur à partir de tubes avec des éléments de rugosité artificielle bidimensionnelle sur leurs surfaces extérieures. L'enlèvement de chaleur était obtenu au moyen d'huile de transformateur et d'eau. On avait mesuré les températures locales des surfaces rugueuses et on avait fait une tentative de dévoiler le mécanisme d'augmentation de la vitesse de transport dechaleur due à la rugosité bidimensionnelle. Les données expérimentales sont corrélées et on a obtenu une équation empirique. Les données expérimentales d'autres auteurs montrent un bon accord avec cette equation.

Zusammenfassung—Die Ergebnisse einer experimentellen Untersuchung des Wärmeüberganges an Rohren mit zweidimensionalen, künstlichen Rauhigkeitselementen an der äusseren Oberfläche werden angegeben. Die Rohre wurden von Transformatorenöl und Wasser bespült. Mit den gemessenen örtlichen Temperaturen der rauhen Oberflächen wurde versucht, den Mechanismus der Wärmeübergangserhöhung infolge der zweidimensionalen Rauhigkeit aufzuklären. Die Ergebnisse werden mit früher veröffentlichten theoretischen und experimentellen Werten verglichen. Aus den korrelierten Versuchsergebnissen liess sich eine empirische Gleichung ableiten. Die Versuchsergebnisse anderer Autoren stimmen mit dieser Gleichung gut überein.