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HEAT TRANSFER BETWEEN A JET AND A HELD PLATE NORMAL TO FLOW

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Abstract—The heat transfer between a submerged jet of liquid and a plate held normal to the flow is studied experimentally. Three heat transfer regions varying in relative length of jet h/d are established The rated formulae satisfactorily agreeing with the experimental data of other investigators [1], [2], [3] are recommended.

For the ranges $h/d \leqslant 0{\cdot}5$, $0{\cdot}5 < h/d < 10{\cdot}0$ and $h/d > 10{\cdot}0$ we give rated formulae. The rated formulae obtained are valid for a wide range of variable main parameters, defining heat transfer processes of the jet stream-line of plates normal to the flow. For the ranges $0{\cdot}5 < h/d < 10{\cdot}0$ and $h/d > 10{\cdot}0$ particular rated formulae which include the angle of attack are recommended. The correlation factor for the angle of attack ϵ_ψ is found from one of the figures.

The influence of the walls which limit the flow of jet heat transfer is investigated.

Résumé—La transmission de chaleur entre un jet de liquide submergé et une plaque tenue normalement à l'écoulement est étudiée expérimentalement. Trois régions de transmission de chaleur variant suivant la longueur relative du jet h/d sont établies. Des formules, en accord satisfaisant avec les données expérimentales des autres chercheurs [1], [2], [3], sont recommandées.

Les formules sont données pour les domaines $h/d \leqslant 0{,}5$; $0{,}5 < h/d < 10{,}0$ et $h/d > 10{,}0$. Les formules obtenues sont valables dans un grand domaine des principaux paramètres variables définissant les processus de transmission de chaleur entre le jet et les plaques tenues normalement à l'écoulement. Pour les domaines $0{,}5 < h/d < 10{,}0$ et $h/d > 10{,}0$ des formules particulières sont recommandées en tenant compte de l'angle d'attaque. Le facteur de corrélation pour l'angle d'attaque ϵ_ψ est trouvé à partir d'une des figures.

L'influence, sur la transmission de chaleur, des parois qui limitent le jet est étudiée.

Zusammenfassung—Der Wärmeübergang zwischen einem getauchten Flüssigkeitsstrahl und einer zur Strömung senkrechten Platte wird experimentell untersucht. Drei Wärmeübergangszonen in Abhängigkeit von der relativen Strahllänge h/d lassen sich unterscheiden. Es werden Gleichungen empfohlen, die befriedigend mit den Ergebnissen anderer Autoren [1], [2], [3] übereinstimmen.

Im Bereich $h/d \leqslant 0{,}5$; $0{,}5 < h/d < 10$ und $h/d > 10$ geben wir bestätigte Gleichungen an, die für einen weiten Bereich der Hauptparameter gelten. Im Bereich $0{,}5 < h/d < 10$ und $h/d > 10$ werden teilweise bestätigte Gleichungen empfohlen, die den Einfluss des Anströmwinkels enthalten, der aus einem der Diagramme zu entnehmen ist.

Auch der Einfluss von Begrenzungswänden wird untersucht.

Аннотация—Экспериментально исследован теплообмен между затопленной струей капельной жидкости и пластиной, расположенной нормально потоку.

Установлены три зоны теплообмена, различающиеся по относительной длине струи h/d, и рекомендованы для этих зон расчётные формулы, удовлетворительно согласующиеся с опытными данными других исследователей [1], [2], [3].

В области $h/d \leqslant 0{,}5$ расчётной формулой является формула (4), в области $0{,}5 < h/d < 10{,}0$ — формула (7), в области $h/d > 10{,}0$ — формула (2) или (8). Полученные обобщенные формулы справедливы в широком диапазоне изменения основных параметров, определяющих процесс теплообмена при струйном обтекании поверхностей, расположенных нормально потоку.

Для областей $0{,}5 < h/d < 10{,}0$ и $h/d > 10{,}0$ рекомендуются расчётные формулы соответственно (7′) и (8′), учитывающие влияние угла атаки ψ. Поправочный множитель на угол атаки ϵ_ψ в формулах (7′) и (8′) берется из рис. 5.

Исследовано влияние стенок, ограничивающих поток, на струйный теплообмен.

INTRODUCTION

For the last few years there has been a heightened interest in heat transfer problems near the drag (stagnation) point of a body placed into a potential flow of liquid or at the jet stream-line.

Among cases at the limit of heat transfer phenomena near the critical point of a body at the jet stream-line is the heat transfer occurring between a jet and a plate held normal to the flow. In this case it is possible to take approximately the whole surface of the plate for a drag point.

Heat transfer from the hot air jet to the plate held normal to flow has been investigated in [1] and [2].

Perry's main task was a study of the influence of convective heat transfer on the total heat transfer in fuel spray furnaces. Basically, with such a statement of the problem attention was directed to the investigation of heat transfer at the jet stream-line of a plate in the narrow range of outflow rates for various angles of attack. Thus the Reynolds number Re_d related to the outflow rate at the nozzle throat and to its diameter varied from 11,000 to 30,000. The second important shortcoming of Perry's experiments lay in the fact that the distance h from the nozzle to the plate was kept constant in all experiments (the distance was equal to $8{\cdot}0d$ where d is the diameter of the nozzle). This circumstance resulted in the fact that Perry's rated formulae do not show the influence of the relative jet length h/d on the intensity of heat transfer, though physically this influence is obvious. Really, at $h/d \to \infty$ the heat transfer coefficient must approach the minimum corresponding to its value for natural convection. The heat transfer coefficient will be at its maximum for the given outflow rate at $h/d \to 0$.

The experiments of Perry were carried out with a nozzle having $d = 16{\cdot}5$ mm while the temperature of air varied from 300° to 600°C. The surface receiving heat had the form of a round plate with a diameter of 344 mm. The rated formula of the heat transfer coefficient for an angle of attack measuring 90° is of the following form:

$$Nu = 0{\cdot}181 \,.\, Re^{0{\cdot}7} \,.\, Pr^{1/3}. \qquad (1)$$

In this formula we took the diameter of a calorimeter (16·5 mm) by which the heat flow at various points of the plate was measured for the required dimension. The physical parameters of air were related to the so-called "boundary layer" mean temperature (half the sum of the temperature of air at the outlet from the nozzle and the temperature of the wall of the calorimeter).

G. G. Thurlow [2] investigated in his experiments the influence of the relative jet length (h/d) on the heat transfer coefficient for a plate normal to the flow. The relative length of the jet varied from $h/d = 10{\cdot}0$ and higher. The Reynolds Re_d number related to the nozzle diameter fluctuated between 22,000 to 60,000 and the air temperature at the nozzle throat varied from $t_n = 50°$ to 200°C. The copper plate measuring 610 × 152 mm served as a heat receiving surface.

The data of Thurlow made it possible to find out that the heat transfer coefficient at $h/d > 10{\cdot}0$ decreased sharply with an increase of the relative jet length, and, as the author pointed out, the influence of natural convection would become predominant at considerable values of $h/d \gg 10{\cdot}0$. In this connection it is interesting to note that one of the approximate formulae of Thurlow had a Grashof criterium of about 1/2. Unfortunately Thurlow gives no indication as to the range of values of Re_d and h/d between which the influence of natural convection should be taken into account.

The concluding formula of Thurlow for an angle of 90° is of the form:

$$Nu_d = C \cdot Re_d^{1/3} \exp\left(-0{\cdot}037\, h/d\right) \qquad (2)$$

where the coefficient C is equal to 1·06 for a nozzle of 1 in. and 0·33 for the nozzle of ½ in., i.e. the coefficient $C \sim d^{3/2}$ where d is expressed in inches.

For the values of $h/d > 10{\cdot}0$ formula (2) should be considered as rated for heat transfer at the jet stream-line of a plate held normal to flow. The nozzle diameter d is taken in formula (2) as the required dimension. The physical parameters of air were calculated according to its temperature at the nozzle throat. The flow velocity for Re_d was related to the cross-section of the nozzle.

Thurlow could not manage to determine the exact dependence of the coefficient C on the nozzle diameter (only two nozzles 1 in. and $\frac{1}{2}$ in. in diameter respectively were used for these experiments). Therefore for nozzles with diameters different from those used in Thurlow's experiments one should handle formula (2) with care.

There are practically no experimental data on drop liquids in literature except the four experimental points received on water by E. Schmidt and his co-workers [3]. In the experiments carried out by Schmidt a free water jet 3·9 mm in diameter played on a perpendicularly placed copper plate 134 mm in diameter, heated by condensated vapour. The jet velocity at the outlet of the nozzle was 5·8, 7·0, 8·9 and 10·7 m/sec respectively; the Reynolds number Re_d, related to the nozzle diameter, varied from 27,000 to 47,000. The relative jet length was constant ($h/d \cong 8{\cdot}0$).

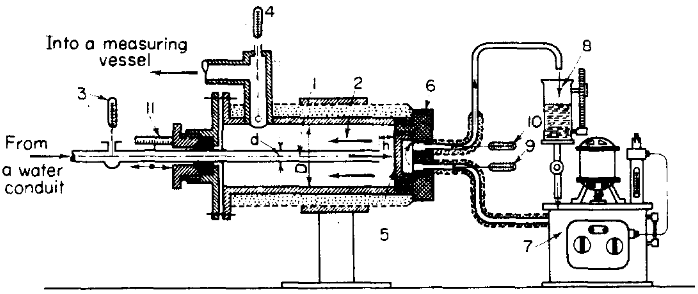

FIG 1. Scheme of the experimental installation.

Taking all this into account we conclude that heat transfer at the jet stream-line at a plate oriented normally to the flow has not been studied adequately. The only formula to be recommended in this case, namely Thurlow's formula (2), is valid only for a narrow range of variation of the main parameters which determine the mode of heat transfer.

In the present work the authors made an experimental study of heat transfer between the submerged jet of a drop liquid (water) and a plate held normal to the flow for a much greater range of Reynolds numbers, relative lengths of the jet and variations in the diameters of nozzles, than in the previous works. Simultaneously the authors sought for a generalization of the experimental data collected by the other investigators.

THE EXPERIMENTAL INSTALLATION AND TESTS

The heat transfer between the submerged jet of a drop liquid (water) and a plate held normal to the flow was studied with the help of the apparatus shown (Fig. 1).

The cooling water was supplied into the glass tube (1) and the outflow was from a tube of a larger diameter filled with water. Water discharge was regulated by a valve system and controlled by the choke washer. The thermometers 3 and 4 with a scale division of 0·1°C measured the temperature of water both at the inlet and at the outlet from the control section 2. A weight method was applied to measure the cooling water discharge.

The control section (2) was made from a metallic tube 54 mm in diameter and 190 mm long. A heated copper calorimeter (5) (the plate mentioned) 48 mm in diameter was placed at one end of this tube and fixed to a thread in a Textolite case (6) thus ensuring a minimum of heat loss both at the periphery and from the front surface of the calorimeter. The ultrathermostat (7) was used to keep the temperature of the heat transfer surface of the calorimeter constant; it was automatically regulated and exact to within 0·1°C. Near 90°C the heating water got through the isolated pipeline into the calorimeter and then into the measuring tank (8). The temperature of the hot water at the inlet and the outlet of the calorimeter respectively was measured by thermometers (9) and (10) with scale divisions up to 0·1°C. Three thermocouples were used to measure the temperature of the heat transfer surface of the calorimeter; one was put in the centre and two others at the periphery. It was known that the thermostat practically ensured a constant temperature of the heat transfer surface.

The distance from the jet to the heat transfer surface was measured and fixed by the nonius (11). The hydrodynamic preparation of the jet was made on a length of $80d$.

Depending on the nozzle diameter d, the experiments were divided into six series. The diameters of the nozzles used in the experiments were 2·5, 6·4, 10·7, 21·3, 30·0 and 36·6 mm respectively. Both the outflow rate w_0 and the distance h from the nozzle to the calorimeter (the jet length) varied within this wide range for each series of experiments. The variation range of outflow rate was from 0·014 to 5 m/sec, the Reynolds number $Re_d = (w_0 \cdot d/\nu)$ referred to the nozzle diameter d varied from 50 to 31,000, respectively, where w_0 is the outflow rate at the nozzle throat. The relative distance from the nozzle to the plate also differed considerably.

It should be noted that 5 to 8 points were taken off in each series of experiments according to the main parameter values (the outflow rate and the jet length). Thus every series included 30 to 50 experiments and about 200 experimental points were received in sum.

The position both of the nozzle and the plate was under strict control during the experiment; the jet axis was always normal to the plate (calorimeter) and concurred with the centre of it. This was achieved by a special centering arrangement.

EXPERIMENTAL RESULTS AND DISCUSSION

The analysis of the experiments and equations which serve to describe the heat transfer process at the jet stream-line of a plate normal to the flow makes it possible to conclude that generally this type of heat transfer is determined by the outflow rate w_0, the relative distance from the nozzle to the plate h/d and by the physical characteristics of the liquid. In connection with this the criterion equation describing the given type of heat transfer will be of the following form:

$$Nu = f(Re, Pr, h/d) \tag{3}$$

where the velocity w_0 for the Reynolds number is related to the nozzle diameter. The physical parameters of the liquid were determined by its temperature at the throat of the nozzle.

The treatment of the experimental data according to equation (3) showed that there exist three heat transfer regions depending on the relative jet length h/d. The critical values of the relative jet length are $(h/d)_1^{cr} = 0{\cdot}5$ and $(h/d)_2^{cr} = 10{\cdot}0$.

The rated formula of heat transfer for the values $h/d \leqslant 0{\cdot}5$ is of the form

$$Nu_h = 0{\cdot}55 \cdot \sqrt{(Re_h)} \,.\, Pr^{1/3} \tag{4}$$

where the index h indicates that the distance from the nozzle to the plate is of the required dimension.

Equation (4) is valid for the whole range of the outflow rates investigated.

For small values of h/d the jet practically has no expansion and the recovering flow must not have an appreciable influence on the washout of the jet. Approximately we deal here with the case of heat transfer near the drag (critical) point of a body streamlined by the potential flow of incompressible liquid.

A comparison of our data at $h/d \leqslant 0{\cdot}5$ with the theoretical and experimental works [4–7] on heat transfer near the drag point shows that in both cases there exists not only a qualitative agreement but also a quantitative one. Thus according to the theoretical solutions given by Motulevich [4], Drake [7] and others, the heat transfer coefficient near the drag point of a body streamlined by a potential flow is proportional to the flow rate by 0·5 degree and this takes place in our case. Further, according to Goldstain [8] the rated formula for the case of heat transfer near the critical point will be of the following form:

$$Nu = 0{\cdot}57\, Re^{0{\cdot}5} \cdot Pr^{0{\cdot}371}. \tag{5}$$

Ulsamer [9] generalized the experimental data on heat transfer at the transversal stream-line of bodies by drop liquids and got the equation:

$$Nu = 0{\cdot}6 \cdot \sqrt{(Re)} \cdot Pr^{0{\cdot}31} \tag{6}$$

for the numbers $Re = 50 + 10{,}000$ and $Pr = 6 + 1240$.

Thus formula (4) practically agrees with equations (5) and (6) derived for the transversal stream-line of bodies by the potential flow. The only difference lies in the fact that at a jet stream-line the heat transfer coefficient decreases with the increase of jet length h by 0·5 degree.

The experimental data on heat transfer at the jet stream-line of a plate by a drop liquid which is normal to the flow for $h/d \leqslant 0{\cdot}5$ have been plotted in Fig. 2. Equation (4) is represented in Fig. 2 by a solid line. Root mean-square scatter of the experimental data relative to the curve 4 is $\pm$ 12 per cent.

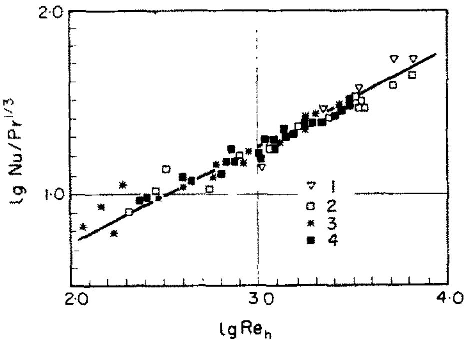

FIG. 2. The range $h/d \leqslant 0{\cdot}5$.
1—nozzle diameter = 10·7 mm.
2—nozzle diameter = 21·3 mm.
3—nozzle diameter = 30·0 mm.
4—nozzle diameter = 36·6 mm.

In the range of $0{\cdot}5 < h/d < 10{\cdot}0$ the data on heat transfer at the jet stream-line of a plate held normal to the flow by the drop liquid is summed up approximately by the equation:

$$Nu_d = C\,,\, Re_d^{0{\cdot}64} \cdot Pr^{1/3} \exp\,(-0{\cdot}037\, h/d) \quad (7)$$

where the index d in Nusselt and Reynolds criteria indicates that the nozzle diameter d is of the required dimension. The coefficient C in formula (7) depends on the nozzle diameter and is equal to

$$C = 0{\cdot}034 \cdot d^{0{\cdot}9},$$

where d is in mm.

Figure 3 represents the experimental data relating to the region $0{\cdot}5 < h/d < 10{\cdot}0$. The experimental points of Perry (21 points) received in the air with the nozzle $d = 16{\cdot}5$ mm for $h/d = 8{\cdot}0$ and at an angle of 90°, and the data represented by Schmidt and his co-workers [3] obtained for water with a nozzle $d = 3{\cdot}9$ mm at $h/d = 8{\cdot}0$ are also plotted in Fig. 3. The range of variation of Reynolds numbers in the experiments carried

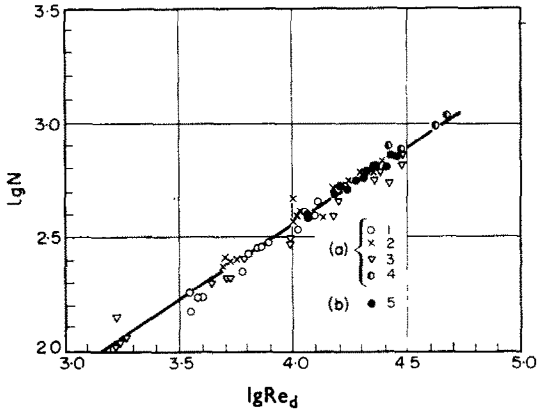

FIG. 3. The range $0{\cdot}5 < h/d < 10{\cdot}0$. The dependence diagram

$$N = \frac{Nu_d \exp 0{\cdot}037\,(h/d)}{c\, Pr^{1/3}}$$

water:
1—nozzle diameter = 2·5 mm.
2—nozzle diameter = 6·4 mm.
3—nozzle diameter = 10·7 mm.
4—nozzle diameter = 3·9 mm—Schmidt's experiments [3].

air: 5—nozzle diameter = 16·5 mm—Perry's experiments [1].

out by Perry and Schmidt was from 11,000 to 30,000 and from 27,000 to 50,000, respectively. As it can be seen from Fig. 3 the experimental data by Perry and Schmidt agree well with our equation (7). The root mean-scatter of the experimental data in reference to curve 7 does not exceed ± 8 per cent.

Thus formula (7) can be recommended as a rated one for heat transfer between the jet and the plate oriented normally to the flow for the following variations in the range of the determining criteria: $0{\cdot}5 < h/d < 10{\cdot}0$; Re_d from 1600 to 50,000 and Pr from 0·7 to 10·0.

Since the data of Perry for an angle of 90° agrees well with our equation (7) it may be expected that the experimental data of Perry for other angles will also be given approximately by equation (7). Fig. 4 in fact presents the experimental data of Perry for angles of 15°, 30°, 45°, 60° and 75° calculated according to equation (7). For comparison Fig. 4 also shows our generalized curve which we got for heat transfer between the jet and the plate normal to the flow (angle $\psi = 90°$). It is possible to count the influence of the angle ψ and the correction coefficient ϵ_ψ the values of which are given in Fig. 5.

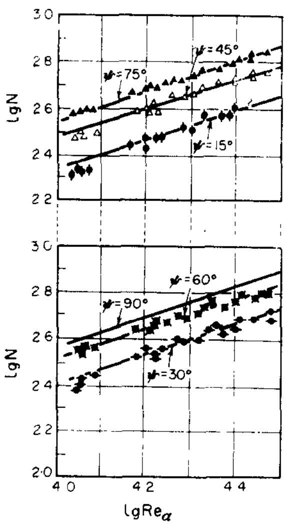

FIG. 4. The range $0{\cdot}5 < h/d < 10{\cdot}0$. The influence of angle of attack upon the jet heat transfer. The curve $\psi = 90°$ was plotted according to formula (7).

$$N = \frac{Nu_a \exp 0{\cdot}037\,(h/d)}{c\,Pr^{1/3}}$$

Equation (7) taking into account the influence of the angle at the jet heat transfer will be of the form:

$$Nu_a = C \cdot \epsilon_\psi \cdot Re_d^{0{\cdot}64} \cdot Pr^{1/3} \exp\left(-0{\cdot}037\,h/d\right) \quad (7')$$

However, one must handle this formula with care since the values of the coefficients ϵ_ψ were obtained for a constant diameter of the nozzle and only for a single value of $h/d \cong 8{\cdot}0$.

A comparison of equation (7) with formula (2) of Thurlow, which is only valid for $h/d > 10{\cdot}0$, shows that in the whole range of h/d starting from $h/d > 0{\cdot}5$ there exists a general regularity in the variation of the heat transfer coefficient depending on the relative jet length h/d and the heat transfer coefficient α decreases proportionally to $\exp(-0{\cdot}037\,h/d)$:

$$\alpha \sim \exp\left(-0{\cdot}037\,h/d\right).$$

At $0{\cdot}5 < h/d < 10{\cdot}0$ the heat transfer coefficient largely depends on the outflow rate w_0. In the above mentioned range the criterion Nu_d is proportional to Re_d to the power 0·64:

$$Nu_d \sim Re_d^{0{\cdot}64}$$

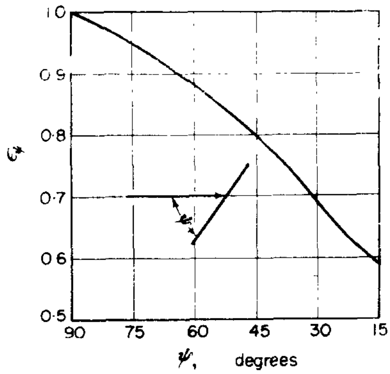

FIG. 5. The correlation coefficient ϵ_ψ taking into account the angle of attack in formulae (7′) and (8′).

In the range of $h/d > 10{\cdot}0$ the dependence of the heat transfer coefficient on the outflow rate greatly decreases and, as stated by Thurlow [2], the criterion Nu_d becomes proportional to Re_d raised to the power 1/3:

$$Nu_d \sim Re_d^{1/3}$$

And in this range the influence of natural convection becomes more perceptible for a larger increase in h/d.

It is interesting to note that if we replace $Re_d^{0{\cdot}64}$ by $Re_d^{1/3}$ and take $Pr = 1$ in formula (7) then the coefficient C will be $\sim d^{1/3}$. According to Thurlow [2] the coefficient C in formula (2) is proportional to $d^{1{\cdot}5}$ i.e. at $h/d > 10{\cdot}0$ and on substitution of $Re_d^{1/3}$ for $Re_d^{0{\cdot}64}$ formula (7) practically agrees with Thurlow's formula (2).

Thus, we may suppose that for heat transfer at the jet stream-line of a plate held normal to the flow there exist three zones which differ in h/d: the first zone corresponds to $h/d \leqslant 0{\cdot}5$, the second zone to $0{\cdot}5 < h/d < 10{\cdot}0$, the third zone to $h/d > 10{\cdot}0$. The rated formula for heat transfer in the first zone is given by equation (4), in the second zone by equation (7) and in the third zone it is Thurlow's formula (2). In the last case, for materials which differ in their properties from air and for diameters of the nozzle other than those used in the experiments of Thurlow ($\frac{1}{2}$ in. and 1 in.) it is possible to suggest from tentative calculations the formula obtained on the basis of equation (7) and extrapolating for $h/d > 10{\cdot}0$:

$$Nu_d = C_1\,(Re_d \cdot Pr)^{1\cdot 3} \exp\,(-0{\cdot}037\; h/d); \quad (8)$$

for angles ψ different than 90°:

$$Nu_d = C_1 \cdot \epsilon_\psi \cdot (Re_d \cdot Pr)^{1/3} \exp\,(-0{\cdot}037\; h/d) \quad (8')$$

where $C_1 = 0{\cdot}034 \cdot d^{1\cdot 3}$ and d is the nozzle diameter in mm.

In their research the authors also showed the influence of the wall restricting flow on heat transfer at the jet stream-line of a vertical plate. From $D/d_{ext} < 4{\cdot}0$ this influence becomes noticeable (where D is the diameter of a vessel into which the outflow occurs and d_{ext} is the external diameter of a nozzle).

The experimental data referring to the heat transfer at the jet stream-line of a vertical plate under the conditions for the restricted volume ($D/d_{ext} < 4{\cdot}0$) were plotted in Fig. 6. The experimental data are satisfactorily described by the formula:

$$Nu_d = 3{\cdot}68 \cdot Re_d^{0\cdot 38} \cdot Pr^{1/3} \exp\,(-0{\cdot}037\; h/d) \cdot (D/d_{ext})^{-0\cdot 5} \quad (9)$$

which is valid for $h/d > 0{\cdot}5$ and

$$Re_d = 300 + 10{,}000.$$

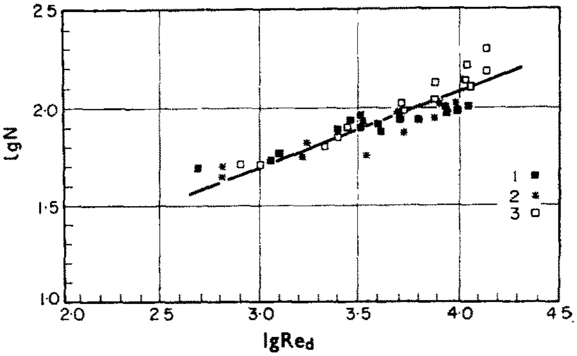

FIG. 6. The jet heat transfer under the conditions of a restricted volume. $h/d > 0{\cdot}5$.

$$N = \frac{Nu_d \exp 0{\cdot}037\,(h/d)}{Pr^{1/3}\,(D/d_{ext})^{-0\cdot 5}}$$

1—nozzle diameter = 36·6 mm.
2—nozzle diameter = 30·0 mm.
3—nozzle diameter = 21·3 mm.

A straight line was plotted according to formula (9).

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