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THE HYDRODYNAMICS OF A FREE, LIQUID JET AND THEIR INFLUENCE ON DIRECT CONTACT HEAT TRANSFER—III

DIRECT CONTACT HEATING OF A CYLINDRICAL, FREE FALLING LIQUID JET

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Abstract—It has been proved in the paper that if nozzle geometry causes a change in jet hydrodynamics it also produces a change in heating intensity. It was found that the best effects on heating jets broken by axisymmetrical vibrations were achieved for turbulent jets in the drop-like region. For the description of direct contact heating of turbulent jets the author proposed his own correlations.

INTRODUCTION

Direct contact jet heat exchangers are frequently applied in technology mainly due to the possibility of development of interface, good heat exchange and no negative wall effects which may be important, e.g. for heating of slurries. The most often used apparatuses of this kind are direct contact vapour condensers, thermal deaerators and liquids heaters. On the basis of our own investigations the above mentioned apparatus was proposed to be applied in thermal sterilization of solutions and slurries.

The paper aims at presenting an analysis of vapour condensation on a liquid jet broken by axisymmetrical vibrations on the basis of the conclusions drawn from the investigations of its hydrodynamics. According to the literature our own investigations were mainly carried out for turbulent jets (due to their higher intensity of heat exchange as compared with laminar jets), and for cylindrical nozzles which suffer less from clogging than nozzles with other cross-sections. Finally, for practical reasons (sieve plates) the investigations were carried out for short nozzles ensuring a full cross-section outflow (Iciek 1982).

Dependences presented in the literature describe well the process of direct contact heating of laminar jets (e.g. Hasson *et al.* 1964, Murty & Sastri 1976, and others). More discrepancy can be found in the description of direct contact vapour condensation on a turbulent jet. Figure 1 presents a comparison of results obtained according to the equations developed by particular authors (table 1). From the analysis of figure 1 it is evident that the subject needs further investigation.

EXPERIMENTAL EQUIPMENT

Figure 2 presents the layout of the rig in which direct contact vapour condensation at atmospheric pressure on a single water jet is being investigated. The installation allows continuous measurement of changes in the jet temperature with its length which enables the influence of jet hydrodynamics on direct contact jet heating to be investigated. The basic elements of the apparatus are as follows: direct contact jet heat exchanger (6), jet receiver (5), water vapour generator (18), thermostat (2), pump (15), circulation tank (19), overflow tank (17), excessive vapour condenser (16), jet temperature measuring system (9), thermometers (1, 11, 12, 14), thrust cylinder (3) and interchangeable outlet nozzles (4). Water and vapour circulation systems are closed.

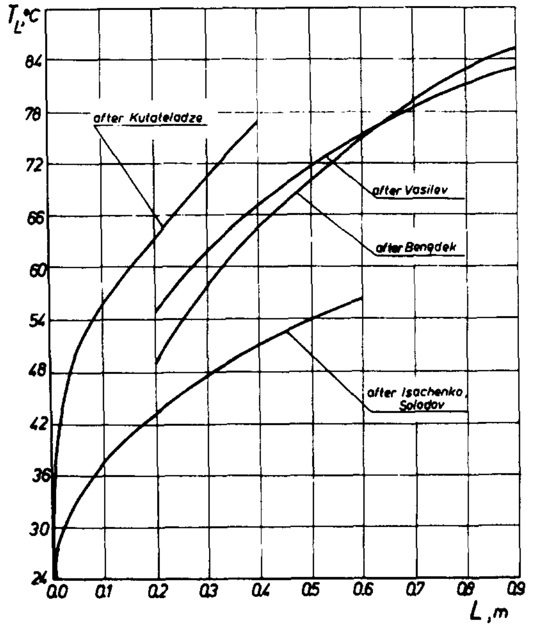

Figure 1. Comparison of results obtained according to dependences presented in table 1.

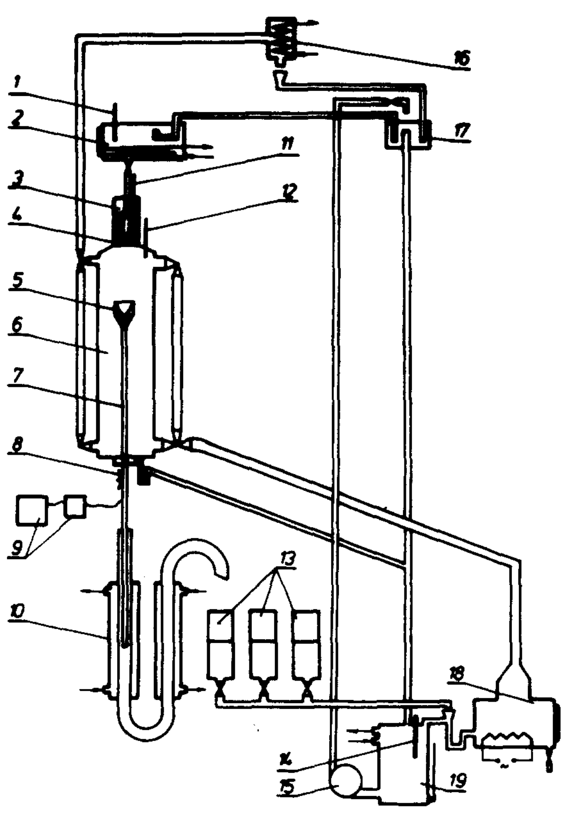

Figure 2. Experimental equipment: 1, 11, 12, 14, thermometers; 2, thermostat; 3, thrust cylinder; 4, outlet nozzle; 5, jet receiver; 6, direct contact heat exchanger; 7, outlet pipe; 8, nonius; 9, mirror galvanometer; 10, cooler; 13, measuring tanks; 15, pump; 16, excessive vapour condenser; 17, overflow tank; 18, water vapour generator; 19, circulation tank.

Table 1. Proposed descriptions of the process of direct contact heating of the cylindrical, free falling turbulent jet

Author	Equation
Kutateladze (1979)	$\lg \frac{T_s - T_o}{T_s - T_L} = 0.16 + 10.1 \left\{ \frac{a \cdot L}{v\, d^2} + \frac{\mathcal{E}^* \cdot v^2}{5\, \varphi^{2.5} g \cdot d} \left[\left(1 + \frac{2 \cdot \varphi^2 g L}{v^2} \right)^{5/4} - 1 \right] \right\}$ where $\mathcal{E}^* = 5 \cdot 10^{-4}$
Isachenko (1977)	for $We_a \leqslant 2.7$ $4St = 0.133 \left(\frac{L}{d} \right)^{-0.41} \cdot Re^{-0.18} \cdot Pr^{-0.05} \cdot K^{0.11} \cdot \exp\left(0.16\, We_a\right)$
Vasilev /according to Kutateladze (1953)	$\lg \frac{T_s - T_o}{T_s - T_L} = 0.029 \left(\frac{g\, d}{v^2} \right)^{0.2} \left(\frac{L}{d} \right)^{0.7}$
Benedek (1976)	$St = 0.00286 \left(\frac{S}{F} \right)^{0.06} \cdot K^{0.084}$

Vapour from the generator (18) flows to the test section (6) where by means of valves, a parallel flow or counter flow can be established. Non-condensed vapour and inert gases are transmitted to the excessive vapour condenser (16). Water at precontrolled temperature is pumped by a pump (15) from the circulation tank (19) to the tank (17) and then it overflows again to the circulation tank (19). Water flows from the tank (17) to the thermostat (2) due to the principle of connected vessels, and then through a valve and distributor it flows into the thrust cylinder (3) from which through the interchangeable nozzle (4) it flows in the form of a jet to the vapour zone of the exchanger (6). The heated water is discharged to the jet receiver (5) and then through the pipe (7) it is transmitted to the cooler (10). Finally, through a system of measuring tanks it is recirculated to the tank (19) or vapour generator.

Figure 3 shows a scheme of the water distributor. The distributor serves for eliminating disturbances in the thrust tank caused by the axisymmetrical feeding. Figure 4 presents a cross-section of the jet receiver (4). The jet flows through the funnel (3) made of Tufnol resin and through the interchangeable nozzle (5) to the thermocouple finned brass cup (2, 6), and through the Tufnol overflow (1) it flows down in the main pipe (7). According to the jet diameter and outflow rate the funnel nozzles (5) of diameters ranging from 4.0 to 14.0 mm were chosen. In order to avoid additional heating of the jet the funnel nozzle diameter was chosen so as to enable the funnel to operate when its nozzle is fully flooded with no water in the funnel. To ensure good reception of the jet four air ducts (8) were placed in the funnel (3). The section in which the jet is being heated was assumed to be the distance between the funnel bottom to the nozzle outlet. The length of the jet on which vapour condensation takes place was changed by either lowering or raising the jet receiver. The jet length was measured by means of a scale marked on the pipe (7) and by the nonius (8) (figure 1). Thermocouples and a mirror galvanometer were used to obtain an averaged value of jet temperature.

The heat exchanger is a cubicoid 0.15 m wide, 0.20 m deep and 1.05 m high. Its front wall is detachable and after dismantling, the jet hydrodynamics in the apparatus can be observed and the nozzle (5) (figure 4) may be changed.

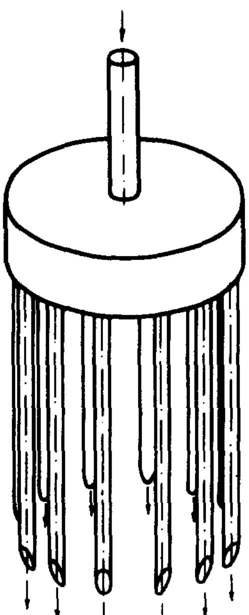

Figure 3.

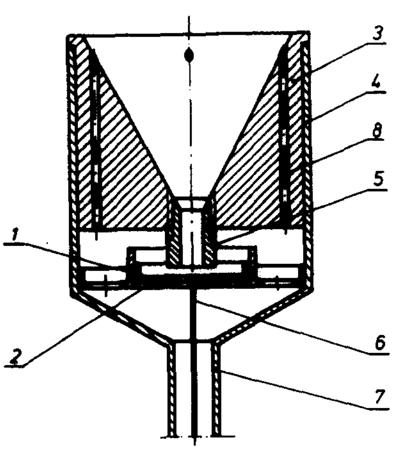

Figure 4.

Figure 3. Water distributor feeding the thrust cylinder.

Figure 4. Jet receiver cross-section: 1, overflow; 2, thermocouple fin; 3, funnel; 4, jet receiver; 5, interchangeable nozzle; 6, thermocouple; 7, outlet pipe; 8, air duct.

The ranges of the investigated variables were a follows: jet nozzle diameters ($2.01 \leq d \leq 7.03$ mm); the shape of the nozzle inlet (sharp edges and conically tapered edges); the nozzle length to diameter ratio ($0.0 < l/d \leq 9.90$); the hydrostatic pressure in the thrust cylinder ($0.020 \leq H \leq 0.300$ m w.g.); the inlet jet temperature ($23.9 \leq T_0 \leq 65.0°$); vapour temperature ($100.5°C \leq T_s \leq 100.8°C$); the jet length ($0.040 \leq L \leq 0.900$ m). All experiments were performed with a constant slight excess of vapour.

RESULTS OF THE MEASUREMENTS

One of the aims of the present work was to show the influence of jet hydrodynamics on direct contact heating. In order to do this our own conclusions concerning the influence of the nozzle geometry on the jet hydrodynamics were applied. Figures 5–7 present examples of the results of measurements illustrating the above mentioned dependences. On the basis of the results obtained it can be stated that the influence of the nozzle geometry is apparent and confirms the conclusions concerned with the jet hydrodynamics, namely a change in the nozzle geometry causes a change in the jet character, which then produces a change in the jet heating intensity.

Figure 5 presents results obtained for three types of nozzles with all other conditions for heat transfer kept constant. Our previous investigations have indicated that, for flow through either a sharp edged nozzle or a short nozzle with conically tapered inlet (at Re = 4100) a laminar jet occurs, but with a cylindrical sharp edged nozzle, of a length which ensures full cross-section outflow, the jet is turbulent. This is confirmed by the results shown which indicate a greater jet heating intensity for the last type of nozzle than the lower jet heating intensity which has about the same range of values for both of the first two nozzles. Additionally, it is found that in practice the curves obtained for heating the laminar jets are in line, despite the fact that from the point of view of jet hydrodynamics flow through a sharp edged nozzle or a short nozzle with conically tapered inlet differs significantly. For the first type of nozzle a smaller jet diameter and a smaller length of the continuous segment of the jet are obtained due

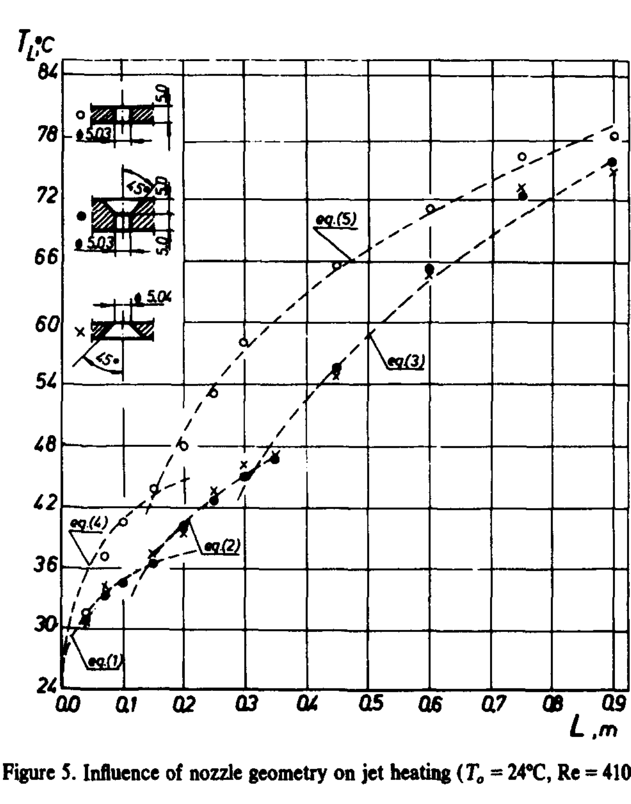

Figure 5. Influence of nozzle geometry on jet heating ($T_o = 24°C$, Re = 4100).

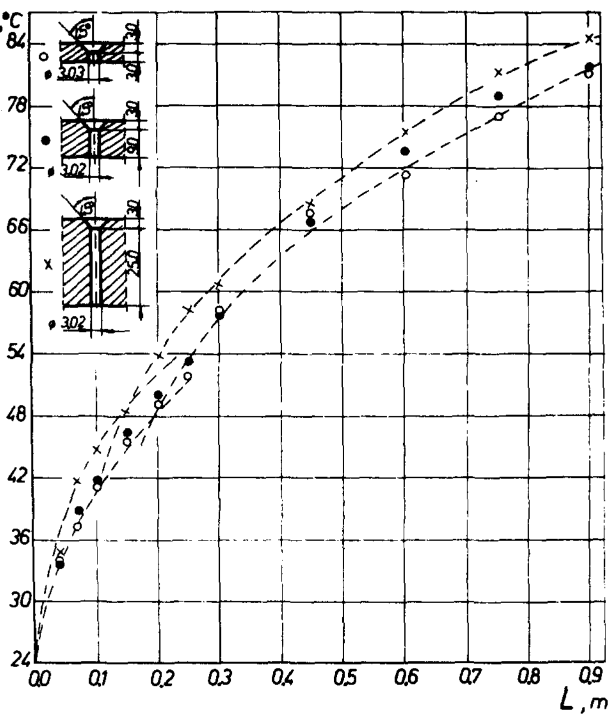

Figure 6. Influence of the nozzle length with conically tapered inlet edges on jet heating (T_o = 24°C, Re = 4200).

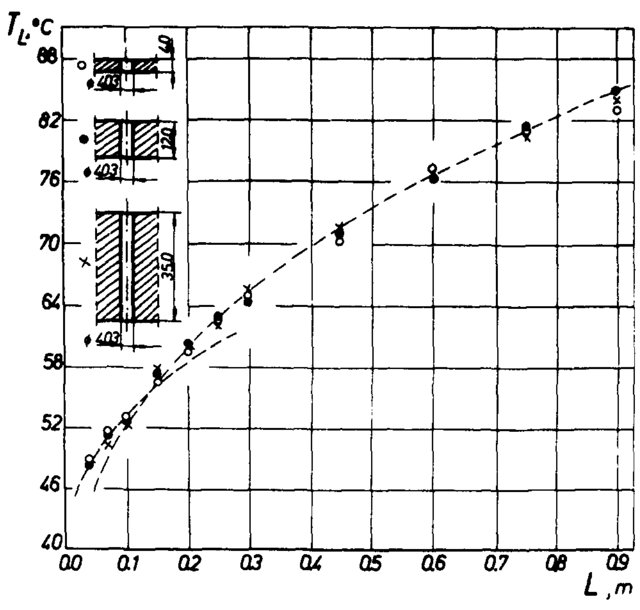

Figure 7. Influence of length of the nozzle with sharp inlet edges on jet heating (T_o = 40°C, Re = 6900).

to contraction. From the point of view of heat exchange these jets behave in the same way because the same heating effects were obtained. Thus, the experimental result confirms the conclusion drawn by Hasson *et al.* (1964) who stated that the degree of laminar jet heating depends solely on the volumetric flow rate (with other conditions of heat transfer kept constant) and not on the jet diameter.

The results concerning laminar jet heating presented in figure 5 allow three ranges to be determined. These ranges correspond to heating of the smooth segment of the jet, rippled segment of the jet and drop-like flow. For particular ranges the following relationships were found:

for smooth segment of the jet

$$T_L = 42.2 + 3.22 \ln L \text{ at } r_c = 0.971 \qquad [1]$$

for rippled segment of the jet

$$T_L = 60.6 + 12.8 \ln L \text{ at } r_c = 0.981 \qquad [2]$$

for drop-like segment of the jet

$$T_L = 79.1 + 28.8 \ln L \text{ at } r_c = 0.981. \qquad [3]$$

Here T_L is the outlet jet temperature, L is the jet length and r_c is a dimensionless correlation coefficient.

From the above equations the lengths of particular segments of the jet were estimated. They were for the smooth segment of the jet about 0.14 m, and for the continuous segments (the break-up length) about 0.32 m. However, previous investigations indicate that for the nozzle with conically tapered inlet ($d = 5.0$ mm, Re = 4100) the breakup length is $L_b = 1.25$ m, and for the sharp edged nozzle it is $L_b = 1.05$ m. It may be supposed that a reduction of the length of the smooth segment and continuous segment of the jet during the direct contact jet heating is caused by vibrations of the installation accompanying the process of condensation. The significant influence of vibrations on the breakup length of the laminar jet has been discussed in the first part of this study.

For the turbulent jet only two ranges corresponding to the continuous segment and the drop-like flow can be distinguished. However, it should be stressed that the separation into two ranges is less clear than for the laminar jet. The following equations describe turbulent jet heating curves in figure 5:

for the continuous segment of the jet

$$T_L = 54.1 + 5.74 \ln L \text{ at } r_c = 0.946 \qquad [4]$$

for the drop-like flow of the jet

$$T_L = 81.3 + 20.1 \ln L \text{ at } r_c = 0.989. \qquad [5]$$

In the above equations the breakup length was estimated as 0.15 m. This value is close to the value of length $L = 0.20$ m obtained in the investigation of the jet hydrodynamics. The results confirm the conclusion presented in the first paper of this series, which showed that for the turbulent jet the influence of external disturbances on the breakup length was not significant.

The results presented in figure 5 illustrate one more important conclusion that for the long jets (in the drop-like flow) the influence of jet hydrodynamics on the heating intensity decreases.

For the jet of small diameter (figure 6) the influence of jet hydrodynamics, and consequently, of the nozzle geometry, on the jet heating intensity is smaller. However, there is a tendency consistent with the previous conclusions stating that for the conically tapered inlet, the jet turbulence increases with the orifice length increase which causes a slightly better heat exchange. In particular cases for the range of the continuous part and the drop-like flow a difference can be found in the character of the influence of the jet length on the heating intensity.

A special attention should be paid to the results obtained at the outflow of liquid through the cylindrical sharp edged short nozzles. In the previous papers it was proved that for this type of orifice the phenomenon of hysteresis may take place which is connected with various types of outflow at any disturbances accelerating the change in the outflow type. As a result no reproducible results of measurements are obtained and deviations of the jet from the nozzle axis can occur which may result in erroneous readings (the jet does not get into the funnel) and for multi-jet exchangers the jets can be mixed. Such conditions are disadvantageous for heat exchange. The above conclusions do not agree with those given by Isachenko *et al.* (1976) who stated that the jets outflowing from the sharp-edged nozzle with the value of $l/d \cong 0.5$ could create conditions for particularly intensive heat exchange in direct contact heat exchanger.

On the basis of our own investigations it was admitted that in practice cylindrical nozzles with sharp inlet edges and the length ensuring a full cross-section outflow should be applied. To determine the proper nozzle length the conclusions from the second paper in this series were used. For such orifices their length, as proved earlier, has no effect on the jet hydrodynamics. This conclusion is confirmed by our own investigations (figure 7) concerning the direct contact heating of jets flowing through the nozzles with sharp inlet edges and of different lengths.

To find quantitative relationships describing the direct contact heating of the turbulent jet broken by axisymmetrical vibrations 39 sets of measurements were carried out. The following variables were used: the diameter and length of the aperture, the rate and temperature of the outflowing jet. The ranges of changes are given in the section "Experimental Equipment". In each set of experiments the jet temperature was measured on 11 heights. The results are correlated separately for the continuous part of the jet and for the drop-like flow. The division of experimental points is carried out using [5] from the first paper of the series. All points at which

$$L \leqslant 11.5 \cdot \mathrm{We}_j^{0.31} \qquad [6]$$

belong to the continuous segment of the jet.

The experimental data are correlated using the least squares method according to the algorithm given by Volk (1973).

In the region of continuous jet heating the experimental data were correlated by the equation given by Isachenko & Solodov (1972):

$$\ln \frac{T_s - T_o}{T_s - T_L} = f(L/d, \mathrm{Pr}, \mathrm{K}, \exp(\mathrm{We}_a), \mathrm{Re}). \qquad [7]$$

Here T_s is the vapor temp, T_o is the inlet jet temp., Pr is the Prandtl number, K is the Kutateladze number, We is the Weber number and Re is the Reynolds number. A power equation with a multidimensional correlation coefficient equal to only $R = 0.682$ was obtained.

The significance of particular variables was investigated using the test t and calculating one-dimensional correlation coefficient r_c. The results of the tests are presented in table 2.

The correlation significance was estimated for the significance level $\alpha_s = 0.95$. The value of the test t for this significance level was 1.96 according to Volk (1973).

As is shown, the only significant parameter is L/d and among other parameters the highest test values are achieved by $\exp(\mathrm{We}_a)$, although they are also lower than those statistically justified. Therefore, for the region of the continuous jet the results were correlated according to

Table 2. The values of test t and one-dimensional correlation coefficient r for the variables from [7]

Variable	Value of test t	Correlation coefficient
L/d	11.8	0.652
Pr	0.623	0.101
K	0.643	0.102
exp(We$_a$)	1.1	0.296
Re	0.356	0.194

the equation presented by Vasilev (according to Kutateladze 1953)

$$\lg \frac{T_s - T_o}{T_s - T_L} = f(L/d, \mathrm{Fr}). \qquad [8]$$

The statistical estimates are shown in table 3. The multidimensional correlation coefficient is $R = 0.974$.

Assuming a possible influence of the Weber number on the relationship, the correlation of the form:

$$\lg \frac{T_s - T_o}{T_s - T_L} = f(L/d, \mathrm{We}_a, \mathrm{Fr}) \qquad [9]$$

was checked and the multiple correlation coefficient $R = 0.976$ was obtained. The statistical estimates are presented in table 4. As can be seen, all variables are significant from the statistical point of view although partial correlation coefficients for We_a and Fr numbers are very low and standard deviations for the exponents are of the same order as these exponents.

Since it was stated that there is a strong dependence between the variables We_a and Fr (the correlation coefficient $r_c = 0.78$) the relationship of the form:

$$\lg \frac{T_s - T_o}{T_s - T_L} = f(L/d, \mathrm{We}_a \cdot \mathrm{Fr}) \qquad [10]$$

was checked out and the multiple correlation coefficient $R = 0.975$ was calculated. The statistical estimates are given in table 5.

Thus, the best statistical estimates are obtained for the correlation

$$\lg \frac{T_s - T_o}{T_s - T_L} = f(L/d, \mathrm{We}_a \cdot \mathrm{Fr}) \qquad [11]$$

in the form:

$$\lg \frac{T_s - T_o}{T_s - T_L} = 0.0107(L/d)^{0.73}(\mathrm{We}_a \cdot \mathrm{Fr})^{-0.07}. \qquad [12]$$

Table 3. The results of statistical estimation of the relationships presented in [8] for the continuous part of the jet

Variable	Value of test t	Partial correlation coefficient	Value of power coefficient	Standard deviation for the exponent
L/d	60.4	0.967	0.723	0.012
Fr	7.8	0.06	-0.103	0.013

Table 4. The results of statistical estimation of relationships presented in [9] for the continuous part of the jet

Variable	Value of test t	Partial correlation coefficient	Value of power coefficient	Standard deviation for the exponent
L/d	61.8	0.967	0.726	0.0112
We_a	3.1	0.063	-0.08	0.026
Fr	2.7	0.06	-0.05	0.020

Table 5. The results of statistical estimation of the relationships presented in [10] for the continuous part of the jet

Variable	Value of test t	Partial correlation coefficient	Value of power coefficient	Standard deviation for the exponent
L/d	61.9	0.968	0.726	0.012
$We_a \cdot Fr$	8.5	0.65	-0.066	0.0078

However, taking into account the correlations presented already in the literature and considering slight differences in the estimates the obtained results are described by the following equation:

$$\lg \frac{T_s - T_0}{T_s - T_L} = 0.015(L/d)^{0.72}\,\mathrm{Fr}^{-0.10} \qquad [13]$$

In the range of the drop-like flow, the experimental data are correlated using the well known relationship[8] which gives the statistical estimates presented in table 6. The multidimensional correlation coefficient R is 0.993.

As follows from the above estimations the equation

$$\lg \frac{T_s - T_0}{T_s - T_L} = 0.0152(L/d)^{0.78}\cdot \mathrm{Fr}^{-0.18} \qquad [14]$$

correlates very well the obtained experimental data. In our own investigations the value of Froude number ranges from 10.5 to 104. To illustrate the obtained results figures 8 and 9 present several measurement sets in the whole range of the investigation of the jet temperature changes. Figures 8 and 9 show also our own correlations marked by a full line as compared with those obtained by Vasilev (according to Kutateladze 1953) marked by a broken line.

CONCLUDING REMARKS

The following conclusions were drawn from the above experiments:

(1) The jet hydrodynamics have a significant influence on its direct contact heating, thus the nozzle geometry is of great importance (see conclusions from Iciek 1982).

Table 6. The results of statistical estimation of the relationships presented in [8] for the drop-like flow

Variable	Value of test t	Partial correlation coefficient	Value of power coefficient	Standard deviation for the exponent
L/d	131.7	0.949	0.780	0.006
Fr	38.9	0.015	-0.178	0.005

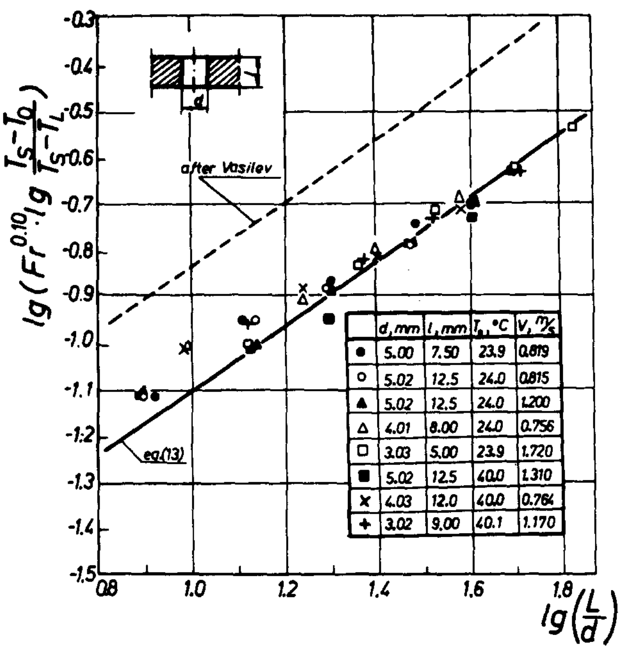

Figure 8. Turbulent jet temperature change vs jet lengths (direct heating in the region of continuous jet part).

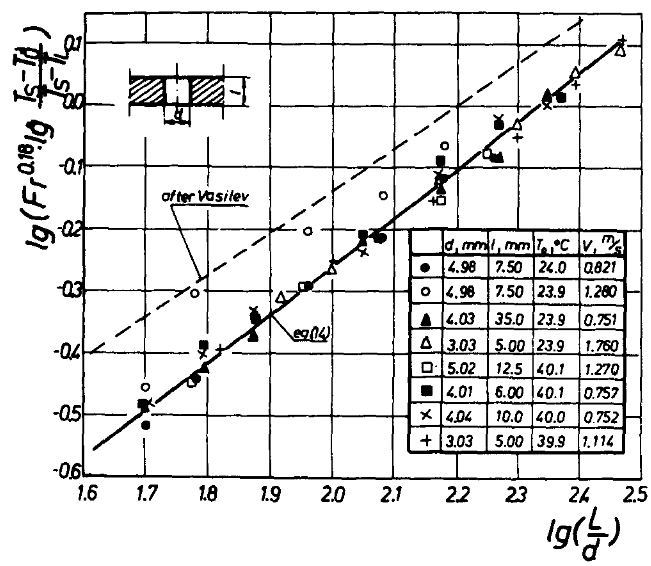

Figure 9. Turbulent jet temperature change vs jet lengths (direct heating in the region of drop-like jet part).

(2) In the character of changes in the jet temperature with the jet length some ranges can be distinguished which are connected with the jet hydrodynamics.

(3) As the distance from the nozzle outlet increases and the jet diameter decreases the effect of jet hydrodynamics diminishes.

(4) Short nozzles with sharp inlet edges are not advisable (see conclusions from Iciek 1982) because of a possible outflow of the jet which can be deviated incidentally from the nozzle axis.

(5) At direct contact heating of the turbulent jet higher intensity of heat transfer is obtained than at heating of the laminar jet. Therefore, it is advisable to apply long nozzles with sharp

inlet edges (see conclusions from Iciek 1982) which ensure that the flow occupies the whole nozzle cross-section and the turbulent jet is obtained at low values of the Reynolds number.

(6) To describe the direct contact heating of the turbulent jet broken by axisymmetrical vibrations, [13] is proposed for the continuous part of the jet ($L \leq 11.5 \cdot \mathrm{We}_j^{0.31}$) and [14] for the drop-like flow ($L > 11.5 \cdot \mathrm{We}_j^{0.31}$).

NOTATION

a thermal diffusivity, m^2/s
c_p specific heat of the liquid, J/kg K
d nozzle diameter, m
F the total surface of water jet, m^2
g acceleration due to gravity, m^2/s
H height of hydrostatic head, m
k thermal conductivity of the liquid, W/mK
l orifice length, m
L_b jet breakup length, m
L jet length, m
r heat of evaporation, J/kg
R, r_c correlation coefficients, dimensionless
S the total cross-section of water jet at the outflow, m^2
T_L outlet jet temperature, °C
T_0 inlet jet temperature, °C
T_s vapour temperature, °C
v jet exit velocity, m/s
α heat transfer coefficient, W/m^2K
ϵ^* experimental coefficient after Kutateladze (1974), dimensionless
η dynamic viscosity of jet, Pa s
ρ density, kg/m^3
σ surface tension, N/m
φ coefficient of discharge, dimensionless

Dimensionless numbers

Fr Froude number, v^2/dg
K Kutateladze number, $r/c_p(T_s - T_0)$
Pr Prandtl number, $c_p\eta/k$
Re Reynolds number, $vd\rho_j/\eta$
St Stanton number, $\alpha/\rho_j c_p v$
We Weber number, $\rho v^2 d/\sigma$

Indices

a ambient medium
j jet

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