The heat transfer of round impinging jets

V V Lemanov, V V Terekhov and V I Terekhov
Kutateladze Institute of Thermophysics SB RAS, Novosibirsk
E-mail: lemanov@itp.nsc.ru

Abstract. Heat transfer of round impinging jets has been investigated in a wide range of Reynolds numbers (Re=40-21000). A jet flowing out of a profiled nozzle and a long tube at a constant wall temperature and at a distance to the obstacle h/d=1-20 was studied. In experiments at Re<3000 for an outflow from the tube, a noticeable increase in heat transfer by 200-400% is observed, compared with the case when the jet flows out of the nozzle. At Re>3000, the difference in heat transfer for the two variants of jet formation (from the tube and from the nozzle) decreases asymptotically. For laminar and turbulent flow modes, the results of numerical calculations are consistent with the experimental data.

1. Introduction
Impact jets are widely used for cooling devices in various technologies. However, a number of problems concerning the heat exchange in the interaction of the jet with the surface remain unresolved. The complexity of the problem lies in a large set of parameters that affect heat transfer. It is known [1,2] that the heat transfer in the impinging jets depends on the Reynolds (Re) and Prandtl (Pr) numbers, the relative distance from the nozzle to the surface (h/d), the degree of the jet turbulence (Tu), the shape and size of the nozzle and the state of its edge, as well as the shape of the cooled surface. The Reynolds number (Re=Uj/d/ν) is one of the determining parameters, which for round jets is usually defined by the bulk velocity Uj, the nozzle diameter d and the kinematic viscosity ν. In this case, the correlation formula for calculating the heat transfer in the region of the frontal point of the flat obstacle is written in the following form

$$\text{Nu}_0 = C \text{Re}^m \text{Pr}^n \left( \frac{h}{d} \right)^p$$

(1)

where Nu0 = α0d/λ, α0 is the heat transfer coefficient, λ is the heat conductivity coefficient, and the coefficients C, m, n, p vary in the works of different authors [1-2]. To solve the problem of effective cooling of heated surfaces, mainly jets with a large Reynolds number (Re>3000) at small distances to the barrier (h/d<8) were studied. There are relatively few literature sources studying heat transfer at low Reynolds numbers (Re<3000) [3-6]. The dependence (1) leads to a monotonic increase in the Nusselt number with an increase in the Reynolds number (m=0.3-0.8), including at Re<3000, which is confirmed by experimental data [4-5]. In this regard, the aim of this work is an experimental and numerical study of heat transfer in the vicinity of the stagnation point of a round impinging air jet in a wide range of Reynolds numbers (40<Re<21000). In the experimental part of the work, the main attention is paid to heat transfer in the region of Re numbers corresponding to the laminar, transition and initial turbulent flow modes.
2. Experimental and numerical method

2.1. Experimental part
The physical and numerical modelling of the problem is carried out in the paper. A detailed scheme of the experiment with an impinging jet is presented in [6]. Air under pressure is supplied from the compressed air line and is regulated by a fine adjustment valve. Then the gas passes through the flow meter (Bronkhorst company), and through a flexible hose (internal diameter of 15 mm) it enters the working area. As a jet source, we used: a) a tube with an internal diameter \(d = 3.2\) mm and a length \(l = 1\) m and b) a nozzle with an internal diameter \(d=3.2\) mm, a length of 35 mm and with a degree of contraction of 20. The distance from the beginning of the jet to the obstacle was \(h/d=16-20\). The heat exchange section was made in the form of a copper disc with a diameter of 190 mm and a thickness of 50 mm. Heating was carried out using an electric heater; the boundary condition on the wall was close to the condition \(T_w = \text{const}\) \((T_w = 50-60\text{°C})\). Film heat flux sensors (HFS) 4 with a size of 2x2x0.2 mm [7] were glued to the surface of the plate. The temperature of the plate and the jet was measured by thermocouples made of wire with a diameter of 0.21 mm (chromel-copel). Sensors through the amplifier and analog-to-digital converter (ADC L-Card E14-140) were connected to the data acquisition system. This scheme allowed measuring the instantaneous value of the heat flux in the frequency band up to 2 kHz, and on the time series (sample size \(1.2\times10^4\)) calculating the mean \(q\) and the root-mean square \(q'\) values of the heat flux. The mean values of the heat transfer coefficient \(\alpha\) and the root-mean square values of pulsations of heat transfer coefficient \(\alpha'\) were determined by \(q\) and \(q'\), respectively, and by the wall \(T_w\) and the jet \(T_j\) temperature difference. The air parameters (kinematic viscosity and thermal conductivity coefficient) necessary for the calculation of the Reynolds number \(Re=Ud/\nu\) and Nusselt number \(Nu_0 = \alpha_0d/\lambda\) were determined by the flow temperature at the nozzle outlet. In the experiments we measured: instantaneous value of the heat flux, the gas flow through the nozzle or tube, the temperature of the plate and the jet in the initial cross-section, and barometric pressure. The program of experiments also included the flow visualization.

2.2. Numerical part
The numerical part of the work is based on the solution of the averaged Navier–Stokes and energy equations supplemented by the k-\(\omega\) turbulence model [8]. The equations were solved in two-dimensional axisymmetric formulation. The system of equations of motion, energy and turbulence model was solved by the control volume method on a structured combined grid. The approximation was carried out for: convective terms by the QUICK scheme and diffusion terms by the central difference scheme; and interpolation of the values of variables from the centers of control volumes on the face was realized using the method [9]. In the paper, the iterative semi-implicit SIMPLEC algorithm was used. The solutions converge if the residual values for all variables are less than \(10^{-6}\). The geometry of the computational domain corresponded to the experiment presented in this paper. The boundary conditions were set as follows: at the inlet to the computational domain a velocity profile corresponding to the fully developed flow in the pipe was set with the temperature taken as a constant. On a solid wall conditions of no-slip and constancy of temperature were set. The walls of the nozzle were assumed to be heat-insulated. All other boundaries corresponded to conditions of constant pressure. The thermophysical properties of the gas were assumed to be constant and corresponding to room temperature. A computational grid was chosen on the basis of a series of calculations on ever smaller grids until the independence of the solution (within 0.1% of the Nusselt number) from the grid size. The same procedure was realized for all Reynolds numbers and distances from the nozzle to the obstacles, studied in the present work. In subsequent calculations, the distance from the walls to the first node did not exceed one (in the coordinates of the wall law); in the viscous sublayer there were at least several nodes.
3. Results

3.1. Experimental part

At the first stage, test experiments were carried out to measure the local heat transfer for the round impinging jet flowing out of the profiled nozzle of the DISA company (d=8.9 mm, contraction ratio of 25). The distance from the beginning of the jet to the obstacle was h/d = 10-20. Data processing has shown that the local Nusselt number in the stagnation point Nu₀ corresponds to dependence (1) with coefficients C=5.25 m=0.5, n=0.33 p=0.77. In a wide range of Reynolds numbers, a monotonic dependence of the Nusselt number on the Reynolds number of the jet is observed. At the second stage, experiments were carried out for the impinging air jet, flowing out of the tube and nozzle (the diameter of the initial section of the jet d=3.2 mm, the distance to the plate h/d = 20). Fig.1a demonstrates experimental data on the mean value of the heat transfer coefficient in the stagnation point. As can be seen from the figure, for the outflow from the nozzle, there is a monotonic dependence of the average value of the heat transfer coefficient α₀ on the Reynolds number of the jet in the range of its values Re=200-20000. At the same time, for the outflow from the tube the behavior of α₀ is significantly different. There are three characteristic zones. In the first zone (Re<3300) there was an increase in α₀, in the second zone (Re=3300-3500) there was a noticeable decrease in α₀, and in the third zone (Re>3500) there was again an increase in the parameter α₀. Thus, in the region of low Reynolds numbers (Re<4000), a non-monotone change in the mean value of the heat transfer coefficient α₀ was found.

![Graphs](a) (b)

**Figure 1.** The dependence of the heat transfer coefficient (a) and the degree of turbulence (b) on the Reynolds number: 1 – nozzle, 2 – tube.

For the two ways of forming the impact jets the heat flux pulsations are significantly different. In the case of outflow from the nozzle the level of pulsations of heat transfer coefficient is low and approximately constant over a wide range of Reynolds numbers. At the same time, for the outflow from the tube at Re=3320, there is a pronounced maximum, which correlates with the maximum of the average heat transfer coefficient α₀, observed in Fig. 1a. To determine the reasons for this behavior of the heat transfer coefficient, the level of turbulence on the axis in the initial section of the jet was measured. Such data are presented in Fig.1b. As one can see, for the jet outflow from the nozzle, the level of velocity fluctuations is low and in the range of Re=200-20000 it does not exceed Tu<1%. At the same time, the behavior of turbulence is significantly different for the jet outflow from the tube. There are three characteristic zones. For the first zone (Re<3300) there is a slight decrease in velocity pulsations (Tu<1%), for the second zone (Re=3300-3500) there is an extremum of velocity pulsations
at Re=3320, which is 12%, and for the third zone (Re>3500) there is a monotonic decrease in velocity pulsations (Tu<2%).

From the data obtained it follows that the maximum heat transfer corresponds to the laminar-turbulent transition (LTT) in a long tube. For this mode it is established that LTT is realized according to the intermittency scenario, that is, when areas with laminar flow alternate with areas with turbulent flow (turbulent spots - puff) [10]. This process is illustrated in Fig.2 by the example of propagation a free flow of hydrocarbons (d=3 mm, Re=2850). Visualization was performed using the Hilbert method [10]. According to Fig.2a, at the initial section of the jet there is an extended laminar region, at the end of which a rather sharp transition to turbulence is observed. In Fig.2b (the next frame) near the beginning of the jet there is a turbulent spot with a length of about 10-12 tube diameters. In the near-field region of the jet, one may observe an intermittent character of the instantaneous velocity change over time, which is associated with the presence of turbulent spots - puff.

3.2. Numerical part

The computational studies were carried out in a wide range of Reynolds numbers of the jet Re = 40-2·10^4 and distances between the nozzle and the obstacle h/d = 2-20. In this case, the diameter of the nozzle was unchanged d = 3.2 mm as in the experiments. Data on heat exchange both in the stagnation point and for an arbitrary radius of the heated plate have been obtained. Some results of the calculations are shown in Fig. 3-4.

![Figure 2. The formation of turbulent spots in a free jet of hydrocarbon flowing out of the long tube: (a) – no puff, (b) – with puff.](image)
Analysis of heat transfer at the stagnation point on the plate is shown in Fig. 3. For small Reynolds numbers (Re < 400), the Nu₀ in Fig. 3a behaves noticeably differently than in the area of higher values (Re > 400). In the first case, there is a monotonic decrease in the intensity of heat transfer in the frontal point as the nozzle moves away from the plate. With an increase in the number of Re, the Nu₀ criterion changes slightly at first, and at h/d > 6 it reaches a maximum, and the heat exchange gradually decreases. Apparently, within the framework of the presented model, the value of Re ≈ 400 is a threshold. The flow at lesser Re in the considered conditions is laminar both in the jet part and in the zone of development of the wall boundary layer from the center to the periphery. For higher Re numbers in the impact jet, a laminar-turbulent transition occurs, which leads to a different character of the heat transfer change at the stagnation point depending on the distance h/d. The difference between the modes can be seen in Fig. 3b, where the calculation data is presented as a relationship Nu₀/Re₀.5 = f (h/d) at varying Reynolds number of the jet. This method of processing data on heat exchange in the frontal point of the plate is common in the theory of turbulent impact jets [1-3]. As seen in Fig. 3b the data are generalized only for turbulent flow regimes, which, as was established above, are achieved under our conditions at Re > 400. For the laminar flow data are not generalized, since the patterns of change do not comply with the power dependence with m = 0.5, and all the calculated data are stratified by the Reynolds number.

3.3. Comparison of experimental and calculated data
A comparison of experimental and calculated data is shown in Fig. 4. The graph shows three heat transfer regions in the stagnation point: Re < 400 - laminar, 400 < Re < 2000 – transient and Re > 2000 – turbulent. These boundaries are conditional and depend on the distance between the nozzle and the barrier. In the laminar and turbulent regions, the experimental and calculated data correlate both qualitatively and quantitatively. In the transition region, however, there is no agreement between the calculation and the experiment. The experimental data give a higher value of heat transfer, although there is a qualitative similarity in the behavior of the data in Fig. 4. The reasons for the differences are to be determined by more detailed experimental and numerical studies. Another interesting result is a large difference of the experimental data for the impact velocity profile at the nozzle exit compared to the developed profile in the pipe. This data are also shown in figure 4. A possible reason for this heat transfer behavior is an earlier laminar-turbulent transition in jets flowing out of the profiled nozzle. This is confirmed by the data over the length of the section with an extended transition zone for the initial Poiseuille profile [11-12].
Figure 4. Comparison of experimental and calculated data on heat transfer at the stagnation point of an round impinging jet. Lines – numerical studies, points – experimental investigations (h/d=20): 1 – pipe jet, 2 – nozzle jet.

Conclusions
Experimental and numerical studies have shown that the local heat transfer in the impinging jet is significantly dependent not only on the traditional similarity criteria (Re, Pr, h/d, Tu), but also on the method of the jet formation (outflow from the nozzle, pipe, orifice). This is indicated both by the measured average and pulsation characteristics, and the results of calculations. The authors' experiments [11-12] on the range of axisymmetric submerged jets also demonstrate a strong influence of the initial velocity distribution. A good agreement between the calculation and experiment under laminar and turbulent flow regimes has been obtained. In the area of laminar-turbulent transition, such agreement has not been obtained, which requires further and more detailed study.

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