HOT-WIRE ANEMOMETER MEASUREMENT OF MEAN VELOCITY VERY CLOSE TO A WALL

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UDC 532.574

This paper analyzes the errors arising in the use of a hot-wire anemometer for measurements in flow near the wall and associated with proximity to a heat-conducting surface and large velocity gradients.

1. From analysis of available papers on heat transfer from cylinders [1-7] approximate boundaries have been determined as a function of the Reynolds number (Re_w) and the Grashof number (Gr), for regions where heat conduction and free or forced convection are dominant. Figure 1 shows these boundaries and presents recommended theoretical relations for heat transfer for the respective regions. The shaded regions correspond to mixed heat-transfer conditions.

It was established in [5] that in a medium at rest with $Ra_w < 10^{-3}$ heat transfer from a horizontal cylinder is due only to heat conduction, and a stationary "film" of hot air is formed around the wire (the "film" regime). Here the Nusselt number Nu remains constant, and, according to the data of various papers [5, 8, 9, 15], has values in the range 0.34-0.45. For $Ra_w > 10^{-3}$ convective currents appear and Nu increases with increase of Ra_w. This is the transition regime from the film regime to laminar free convection. It occurs for values $Ra_w < 10^2$. However, in some other papers [1,4] there is reported to be a dependence of Nu on Gr even for values $Ra_w < 10^{-3}$.

2. The experimental section used in this present work (Fig. 2) is a tube with semi-circular cross section. In the tube of diameter 74 mm there is a polished insert occupying one-half of the tube cross section. The equivalent channel diameter (d_e) is 45.2 mm, and its length is 2800 mm. A heated plate is set up at the end of the insert. All the measurements were taken at the exit section of the channel, above the plate. The total length of the channel is $60d_e$, and of the heated section it is $4d_e$.

On the inside surface of the heated plate there are thermocouples to measure the wall temperature distribution along and across the plate; the air temperature at the entrance was also measured. The plate was heated by a wire heater located above it, fed with dc current from a stabilized source. Air flow was supplied by a fan and was measured to an accuracy of ±0.5% with a previously calibrated orifice-plate.

The measurements were conducted using tungsten wire sensors of different lengths and diameters 3 and 5 μm. Only simple wires mounted horizontally were used. The measurements were conducted using the 55-m DISA hot-wire anemometer. The probe coordinate was determined by means of a type KM-6 cathetometer to an accuracy of ±0.005 mm, as one-half the distance between the wire and its reflection in the polished plate surface. The probe was moved by the DISA electrical stepping switch with a 0.02-mm step.

3.1. To determine the influence of free convection on heat transfer from wires of 5-μm diameter, experiments were conducted with an overheat value ΔT = 100-200°, which are ordinarily used with hot-wire anemometers. The anemometer wire was introduced in one case from above, relative to the horizontal surface, and in a second case from below. The Rayleigh number, computed for the wire, was less than 10^{-5} for the tests. Comparison of the results of these measurements shows that the influence of free convection is negligibly small for all the overheat values and wire diameters which are ordinarily used in hot wire anemometry. The heat was removed mainly by conduction through the film of hot fluid, whose average thickness one can evaluate approximately by solving the heat-conduction equation for an infinite wire

High-Temperature Institute, Academy of Sciences of the USSR, Moscow. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 36, No. 6, pp. 985-990, June, 1979. Original article submitted July 17, 1978.

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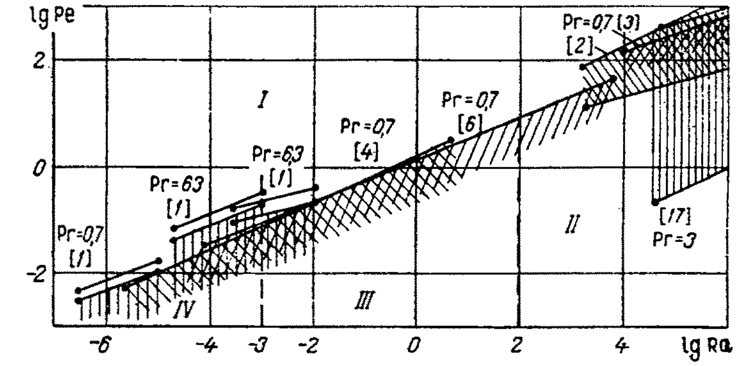


Fig. 1. The free convection, forced convection, and heat-conduction regimes for horizontal cylinders: I) forced convection, $Nu = (0.24 + 0.56Re_w^{0.45})(T_w/T_\infty)^{0.17}$ [4]; II) free convection, $Nu = (0.525 + 0.422Ra^{0.315})(T_w/T_\infty)^{0.154}$ [4]; III) transition regime, $Nu_{max} = 1.18Ra_w^{1/8}$ [5]; IV) heat conduction, Nu = const.

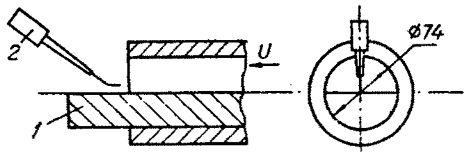


Fig. 2. Experimental section: 1) plate; 2) hot-wire anemometer.

$$\bar{\delta} = \frac{1}{2}\pi\int_0^{2\pi}\delta(\varphi)\,d\varphi = \exp(2/\mathrm{Nu}), \qquad (1)$$

$$\mathrm{Nu} = I^2R_w/\pi l_w\lambda_1(T_w - T_1).$$

The value of $\bar{\delta}$ was determined experimentally as follows. The anemometer filament was brought down to the heat-conducting surface, with no air flow. The boundary of the hot fluid film was considered to be a constant distance from the surface, at the point where the anemometer reading differs by 5% from the readings without the surface present. Thus, for a wire with $l_w/d_w = 1200$ ($d_w = 5$ μm, $\Delta T = 210°$), $Nu_0 = 0.38$ and $\bar{\delta} = 200$. Equation (1) gives $\bar{\delta} = 193$. The distance $y_{min} = \bar{\delta}$ in the region of Re_w numbers where heat transfer by conduction predominates (see Fig. 1) can be interpreted as the minimum distance from the wall at which the wall influence does not appear. For large values of Re heat transfer from the wire is mainly due to forced convection and the value of y_{min} decreases.

3.2. During measurements in the viscous sublayer near the wall one can find two effects which distort the measured results appreciably, depending on the probe construction and the location of the anemometer supports.

For the closest approach to the wall the wire is sometimes welded to the side surface of the supports, not to the end. In our experiments, for measurement of mean velocity, the influence of this structural feature was not apparent in the case where the wire was welded at a distance of no less than 10 μm from the end of the supports, but the values of fluctuating intensity were then always increased.

With the wire supports inclinded at an angle (α) to the flow direction we observed the influence of velocity gradients, associated with shedding of vorticles from the supports. It was established in [10] that one obtains correct readings with a probe with supports directed along the flow. However, for measurements in the immediate vicinity of a surface, investigators are often compelled to use probes with inclined supports, which raises the question of the limiting angle of inclination of the supports to the flow direction. Figure 3 shows the results of velocity measurements for various angles near the wall, in universal coordinates. The support slope angle does not affect the velocity measurement in the region $\eta \geq 15$, when one uses calibrated relations obtained in uniform flows with the same probe position. Experiments show that perturbations from the supports do not reach the anemometer wire for values $\alpha = 0$–$5°$.

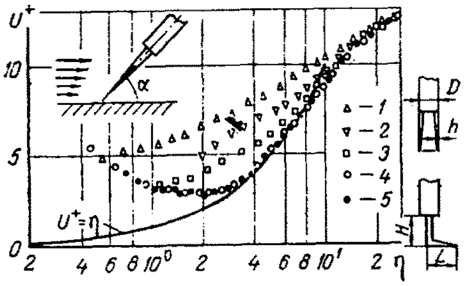


Fig. 3. Influence of orientation of hot-wire anemometer supports on the measurement of velocity near the wall (d_w = 5 μm; Re = 8800): 1) α = 90°; 2) 12; 3) 7; 4) 5; 5) 1.

Sometimes one uses probes with supports which are bent back against the flow. With this type of probe construction one must experimentally determine the length of the convex part of the supports for which perturbations imparted by the sensor body to the probe do not affect the measured results. In our experiments ($(dU/dy)_{max} = 6 \cdot 10^3$ 1/sec; h = 0.3 mm; D = 3.5 mm; H = 6–8 mm) this limiting value is L = 7 mm (Fig. 4).

3.3. Taking into account the results presented in Sec. 3.2, measurements were made of the average velocity fields in air flow near two heat-conducting walls: a copper wall ($\lambda_2/\lambda_1 \approx \infty$) and a Textolite wall ($\lambda_2/\lambda_1 = 11.5$). The information in the literature on the effect of wall material on hot-wire anemometer readings is contradictory. In one reference [11] the influence of the material is definitely reported, and in another reference [12], such an influence was not observed.

The results of our measurements are shown in Fig. 5 (open points). Figure 5 also shows the results of Qka's measurements [13] near a steel plate in air. It can be seen that the wall material does affect the measured results. The maximum influence comes from the copper plate, and its influence exists out to η = 5. If one assumes that the law $U^+ = \eta$ is valid in the region up to η = 6, it is possible to determine the friction stress at the wall (τ_w) from the velocity profile measurements (at points in the range $5 < \eta < 6$ and $U^+ = 0$ for $\eta = 0$). The values of dynamic velocity (u_τ) thus obtained are in good agreement with those calculated from d_e. In the logarithmic law region the data are described well by the relation

$$U^+ = 5.75 \lg \eta + 5.2.$$

4. The literature contains a large number of correlations for calculating heat transfer from a cylinder washed by a transverse flow. Practically all these relations have the form

$$\mathrm{Nu} = A + B\,\mathrm{Pr}^m \mathrm{Re}_w^n.$$

In a number of correlations the exponent n = 0.5, and in others n = 0.45 or a value close to it. For the case of interest to us (heat transfer from fine wires at low values of Re_w) this difference in the value of n is insignficant, and therefore we shall take n = 0.5. In addition, we take m = n.

The experimental data on heat transfer from a hot-wire anemometer at various distances from the wall can be described by the following interpolation relation, valid for Λ > 1:

$$\mathrm{Nu} = \mathrm{Nu}_{0,\infty} + \frac{\mathrm{Nu}_0 - \mathrm{Nu}_{0,\infty}}{(1 + 0.065\,B\,\mathrm{Re}_w)^2}\,\frac{\Lambda - 1}{\Lambda + 1} + 0.56\sqrt{\mathrm{Re}_w}, \tag{2}$$

where Nu_0 and $\mathrm{Nu}_{0,\infty}$ are the values of Nu for $\mathrm{Re}_w = 0$;

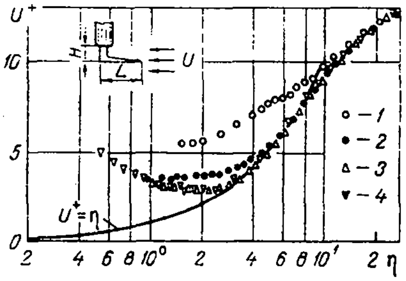
$$\mathrm{Nu}_0 - \mathrm{Nu}_{0,\infty} = \frac{\Lambda - 1}{(\sqrt{B} + 2 \cdot 10^{-4} B^2)(\Lambda + 1)}$$

is a correlation determined from analysis of the experimental data for $\mathrm{Re}_w = 0$.

For $B \to \infty$ Eq. (2) becomes

$$\mathrm{Nu} = \mathrm{Nu}_{0,\infty} + 0.56\sqrt{\mathrm{Re}_w}, \tag{3}$$

which satisfactorily describes heat transfer from the wire in zero-gradient flow.



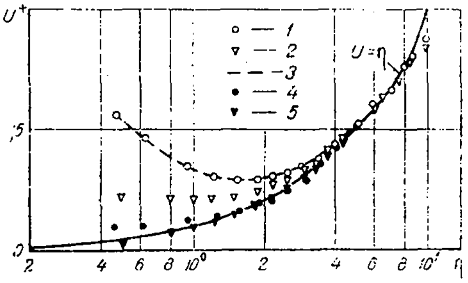


Fig. 4 Fig. 5

Fig. 4. The influence of length L on velocity measurement near the wall (d_w = 5 μm; Re = 8800): 1) L = 4 mm; 2) 6; 3) 7; 4) >7.

Fig. 5. Influence of wall material on velocity measurement (1,2,3) and corrected velocity (4,5): 1,4) copper plate; 2,5) Textolite plate; 3) steel plate [13].

We shall take Eq. (3) as a calibration relation for the wire. In calculating the anemometer readings from this relation for measurements near the wall, one evidently obtains a certain "fictitious" value of velocity (U_{meas}), differing from the true value because of the influence of wall convection.

By equating Eqs. (2) and (3), we obtain a relation between the true and the measured velocities for gases near the wall

$$U_{true}=\left[\sqrt{U_{meas}}-\frac{1.26(\Lambda-1)^2}{\sqrt{y/\nu}\,(1+2\cdot 10^{-4}B^{3/2})(1+0.065B\,\mathrm{Re}_{w,\,true})^2(\Lambda+1)^2}\right]^2, \quad (4)$$

or, in universal coordinates,

$$U^+_{true}=\left[\sqrt{U^+_{meas}}-\frac{1.26(\Lambda-1)^2}{\sqrt{\eta}\,(1+5.6\cdot 10^{-4}(\eta/\Delta\eta_w)^{2/3}(1+0.13\eta U^+_{true})^2(\Lambda+1)^2}\right]^2. \quad (5)$$

The values of U^+_{true} can be found graphically from the measured U^+_{meas}, or by interpolation, taking U^+_{true} = 0 as the first approximation.

5. In conclusion, we note that for measurements of average velocity in an isothermal flow near the wall one can recommend:

1) that the sensor support should be located at an angle of no more than 5° to the flow direction;

2) when one uses probes with supports convex toward the flow, the length of the convex part of the supports should be a value for which measured results show no influence of perturbations from the supports;

3) one should attach the wire to the end of the supports or to the side surface (but only for measurements of mean velocity) at a distance of no less than 10 μm from the end;

4) to find the true value of velocity in measurements near a heat-conducting surface one must use Eqs. (4) or (5) to correct the readings of a sensor, calibrated in a zero-gradient flow.

The proposed corrections for anemometer readings near a wall may be used to correct the measurements of velocity for various liquids near a surface of any material with the condition $\Lambda > 1$.

NOTATION

a, wire radius; Ω, thickness of the liquid "film"; $d_w = 2a$, wire diameter; l_w, wire length; R_w, wire resistance; I, current flowing through the anemometer wire; b, distance of the wire from the wall; T, temperature; U, average velocity; λ, c_p, thermal conductivity and specific heat of the fluid; ρ, density of the fluid; ν, kinematic viscosity of the fluid; q, specific heat flux; τ_w, friction stress at the wall; $u_\tau = \sqrt{\tau_w/\rho}$, dynamic velocity; $B = b/a$; $\eta = yu_\tau/\nu$; $\Lambda \doteq \lambda_2/\lambda_1$; $U^+ = U/u_\tau$; $b = \Omega/a$; $\mathrm{Nu} = qd_w/\lambda_1\,(T_w - T_1)$, Nusselt number; $\mathrm{Re} = U_{av}d_w/\nu$, Reynolds number; Pr, Prandtl number; Pe = Pr Re, Peclet number; $\mathrm{Gr} = g\beta d_w^3[(T_w - T_1)/\nu^2]$, Grashof number; Ra = Pr Gr, Rayleigh number. Subscripts: w, anemometer wire; 1, fluid; 2, wall; ∞, far from the wire.

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HEAT EXCHANGE IN CRYOSTAT THROATS WITH COAXIAL EXTENSION ELEMENTS

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UDC 536.25

The heat exchange in cryostat throats with coaxial extension elements is investigated theoretically and experimentally. It is shown that the efficiency of the coaxial element falls sharply when its length is increased.

In helium cryostats with wide throats intended for cooling relatively large objects, the main part of the heat influx to the liquid helium is heat from the walls of the throat and along the gas colume inside it. The heat influx, and, consequently, the loss of helium by evaporation, can be reduced by increasing the thermal resistance of the walls of the throat. For large-diameter cryostats it was suggested in [1] that throats made in the form of coaxial cylinders should be used. It was experimentally established [1] that forced cooling of both walls of the coaxial element using a cylindrical baffle (Fig. 1, 9) lowered into its end gap and hermetically attached to the thermal lid of the cryostat, does not lead to a reduction in the heat influx.

The temperature profile, taking into account the heat exchange by the gas in the gap between the two walls (Fig. 1, I), can be described by the following two equations:

$$\frac{d^2t_1}{dx^2} - \mu^2 |t_1 - t_2| = 0, \quad (1)$$

$$\frac{d^2t_2}{dx^2} + \mu^2 |t_1 - t_2| = 0. \quad (2)$$

Since the temperature profiles of both walls of the coaxial element are symmetrical $|t_1 = -t_2|$, the system of equations can be reduced to the single equation

$$\frac{d^2t_1}{dx^2} - 2\mu^2 t_1 = 0. \quad (3)$$

For the boundary conditions

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 36, No. 6, pp. 991-996, June, 1979. Original article submitted April 24, 1978.

0022-0841/79/3606-0648$07.50