INVESTIGATION OF THE PARAMETERS OF A VORTEX FLOW INSIDE A RANQUE-HILSCH TUBE

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Experimental data are presented on the parameters of a vortex flow, measured along a Ranque-Hilsch tube in adiabatic and nonadiabatic conditions. Graphs are given showing the variation of the thickness of the external stream, the temperature, and the heat transfer coefficient along the tube.

In references [1–4], dealing with an experimental investigation of the internal parameters of a vortex flow in a Ranque-Hilsch tube, there are no data relating to processes occurring in nonadiabatic conditions. The vortex process, however, is most effective [5] in these very conditions. The present paper, by way of supplement to the data taken along a nozzle section [6], now gives results of measured parameters for a vortex flow along a vortex tube, both in adiabatic and in nonadiabatic conditions.

The measurements of the internal parameters along a conical vortex tube were made by means of radial probes [6]. The probes were positioned by means of a traverse mechanism in special fittings located at five sections (I–V) along the tube, at distances of 90, 190, 290, 490 and 740 mm, respectively, from the nozzle section (Fig. 1a). At these sections, during radial traversing of the probes, measurements were made at each point of the angle φ (indicating the direction of the total velocity vector), the total pressure p_0, the static pressure p, and the stagnation temperature T_0 of the vortex stream. From the values of p_0 and p at each point of the flow the total velocity c was determined, and then, from the angle φ, values of the tangential and axial flow velocity components, c_τ and c_a, were computed. The measurements were carried out with a closely constant initial pressure $p_{00} = 4 \cdot 10^5$ N/m^2, for three values of $\mu = G_x/G = 0.5$, 0.8, and 1. Expansion as far as atmospheric pressure took place.

The direction of the axial velocity component c_a of the vortex stream along the tube (Fig. 1) allows us to determine the boundary which divides the vortex stream into an external component V_e and an internal component V_i, as regards axial motion. In the external stream the vector is directed from sections I to V, and in the internal stream from V to I. The boundary between the external and the internal layers is the surface where the axial velocities are equal to zero ($c_a = 0$). Here reversal of the stream takes place, and an internal layer is formed from the external layer, with a change in the sign of the axial velocity vector.

The thickness δ of the external annular layer of the vortex stream depends on μ. Values of δ, measured at the five sections, are shown in Fig. 1b. The graph shows that the thickness δ of the external layer increases from 5 to 10.5 mm along the tube from sections I to V. The relative quantity δ/r_c varies from 0.263 to 0.578. For $\mu = 1$ the thickness δ is a minimum,

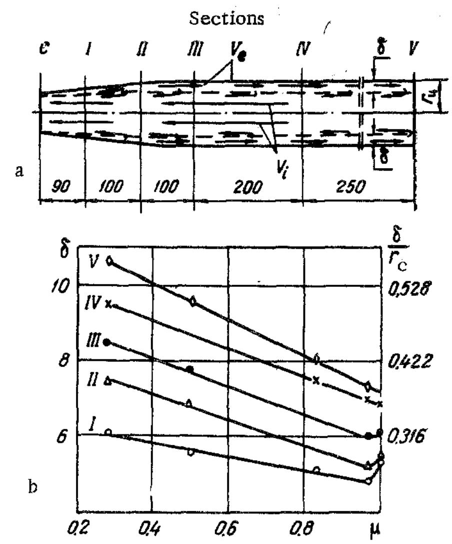

Fig. 1. Schematic of the tube (a) and thickness of the external stream (b).

and for the internal layers V_i a zone is formed with a maximum area of the available cross section. This is due to the fact that, as the internal flow advances from sections V to I, its magnitude increases continuously, which leads to increase of the cross section of the internal stream and to decrease of the thickness of the external stream. The thickness of the external stream at various sections changes to a different degree, depending on the value of μ. Thus, with change of μ from 0.28 to 1, the thickness of the external stream at the I section decreases by a factor of 1.1, while at the V section—by a factor of 1.47. Thus, from examination of the dependence of δ on μ at the different sections, it has been established that at $\mu = 0.3$–1, reversal of the external layers and formation of the internal stream occurs along the whole length of the vortex tube.

The static temperature T and the total temperature T_0 of the external and internal streams in the vortex tube are shown by graphs of the variation of ΔT_{0e}, ΔT_e, and ΔT_i along the tube (Figs. 2, 3).

The quantity ΔT_{0e} is the difference between the stagnation temperature T_0 of the external stream and the initial temperature of the compressed gas T_{00}. The quantities ΔT_e and ΔT_i are, respectively, the difference between the static temperatures of the external flow T_e and the internal flow T_i, and the initial temperature of the compressed gas T_{00}. These quan-

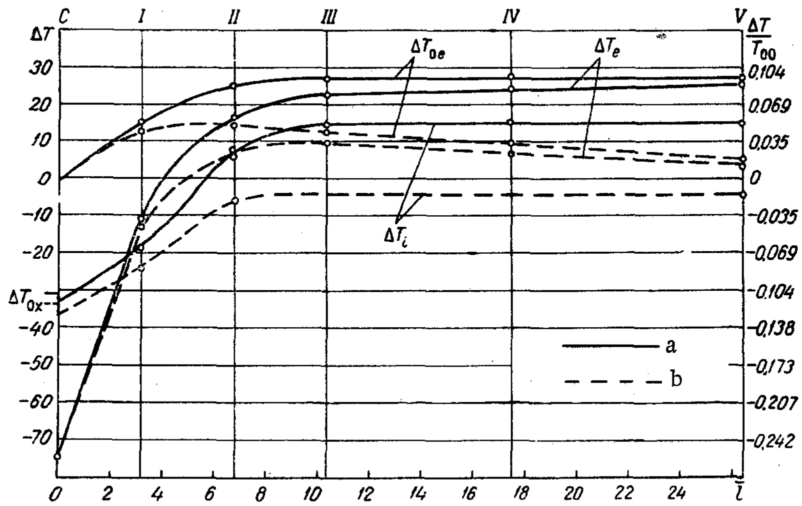

Fig. 2. Variation of stagnation temperature and static temperatures along the tube (a—adiabatic; b—nonadiabatic), for μ = 0.5, $p_{00} = 4 \cdot 10^5$ N/m^2.

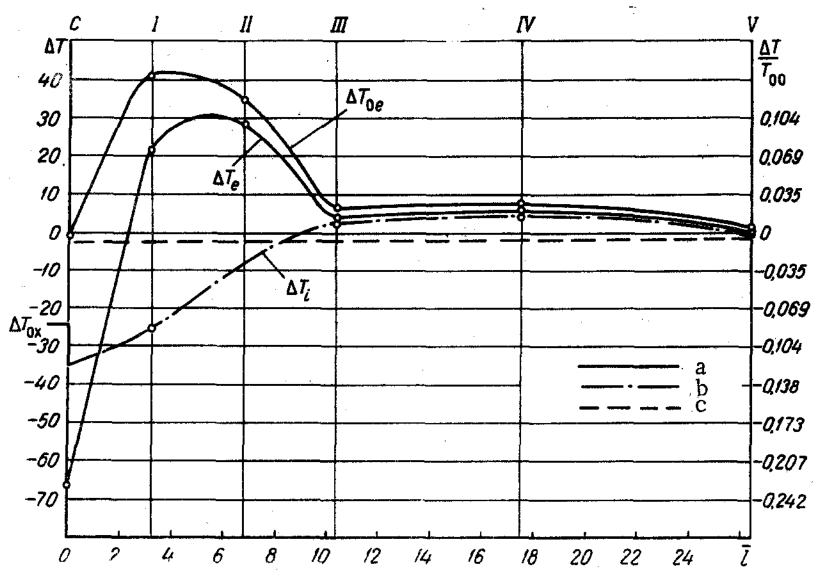

Fig. 3. Variation in stagnation and static temperatures along a nonadiabatic tube for μ = 1, $p_{00} = 4 \cdot 10^5$ N/m^2; a) for the wall layers; b) axial layers; c) cooling water.

tities may be referred to the initial temperature of the compressed gas ($\Delta T_{0e}/T_{00}$, $\Delta T_e/T_{00}$ and $\Delta T_i/T_{00}$).

Figure 2 gives graphs showing the variation of the total temperature of the external stream and of the static temperatures of the internal and external streams as a function of reduced tube length $\bar{l} = l/D_c$. Measurement of the stream temperatures was made both in adiabatic (the vortex tube insulated), and in nonadiabatic conditions [5].

The lowest static temperature is that of the external stream at the nozzle section in the discharge of the gas from the nozzle. The maximum temperature drop $\Delta T_e \simeq -75$ deg is caused by the large velocity which the external stream has upon discharge from the nozzle. As the external stream advances from sections I to V, its static temperature increases, ΔT_e increases correspondingly, and ΔT_e at section V is 25 deg. The increase of static temperature of the external stream is due to transfer of energy to it from the internal stream, which moves upstream and is cooled.

For the internal stream ΔT_i is decreased from 15 deg at section V to −34 deg. The internal stream, issuing through a diaphragm, is brought to rest, and the temperature drop decreases to the value $c^2/2c_p$, where c is the velocity of the stream issuing through the diaphragm; c_p is the stream enthalpy. The increase in stagnation temperature of the external stream, occurring because of transfer of energy from the internal stream, is determined by the quantity ΔT_{0e}.

The sharpest variation of temperature, denoting the most intense transfer of energy from internal layers to external, is observed in the initial part of the tube (up to $\bar{l} \simeq 7$). It is characteristic that in the initial section of the tube ($\bar{l} = 0-3$) ΔT_e for the layers nearest the wall is less than ΔT_i for the axial layers. For the remaining part of the tube ($\bar{l} = 3-26$) ΔT_i for the axial layers is less than ΔT_e for the wall layers.

For a nonadiabatic vortex process more favorable flow conditions exist, which lead finally, other conditions being equal, to increase in the thermal efficiency of the vortex tube [5]. All values of the temperatures (ΔT_{0e}, ΔT_e, ΔT_i) for nonadiabatic conditions are lower than the corresponding temperatures for adiabatic conditions.

In the nonadiabatic tube energy is given up by the external stream in the vortex interaction process. The result of this is that, starting from section I, ΔT_{0e} for nonadiabatic conditions has smaller values, and drops to 5 deg towards section V, i.e., it decreases by roughly 22 deg in comparison with the temperature for adiabatic conditions.

The effect of cooling is to give a reduction in the thermodynamic temperatures both for the external and the internal streams. Therefore the internal stream in a nonadiabatic tube reaches the diaphragm with a lower temperature, which in turn leads to reduction in the stagnation temperature of the cold stream. It may be seen from the graph that for $\mu = 0.5$, ΔT_{0x} is decreased by roughly 3 deg when the vortex process takes place in nonadiabatic conditions.

At large values of μ, a larger difference between the values of ΔT_{0x} for adiabatic and nonadiabatic conditions is observed.

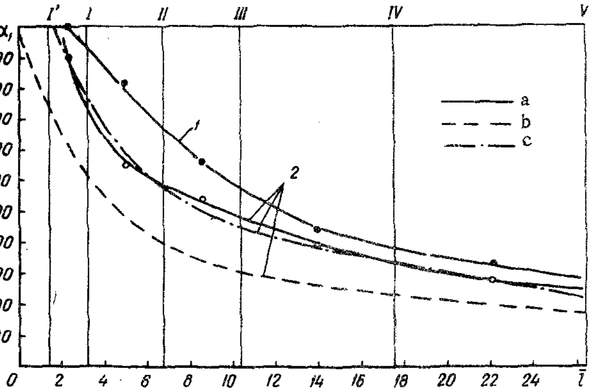

Fig. 4. Variation in the heat transfer coefficient of a vortex stream along a nonadiabatic tube with $\mu = 0.5$ (1) and 0.8 (2); a—from formula (2); b—(1); c—(3).

When the vortex process is carried out in nonadiabatic conditions the possibility exists (in addition to increase in the thermal efficiency when $\mu < 1$) of operating in a qualitatively new regime giving cooling of the total amount of gas brought into the tube ($\mu = 1$). Other conditions being equal, this leads to a considerable increase in the cooling effect of the vortex tube [5].

Figure 3 shows the variation in temperature (ΔT_{0e}, ΔT_e, ΔT_i) of the external and internal streams along the length $\bar{l}$ of a nonadiabatic tube operating with $\mu = 1$. Due to transfer of energy from the external layers to the internal layers, and from them through the wall to the cooling water, at the exit from the vortex tube we find that the whole stream ($\mu = 1$) is cooled by 24 deg. For an inlet pressure of $p_{00} = 6 \cdot 10^5$ N/m², $\Delta T_{0x} \simeq -30$ deg [5].

For $\mu = 1$ there is a sharp increase in the stagnation temperature of the external stream. Even for a cold condition, ΔT_{0e} up to section I reaches 42 deg (at the same section in the adiabatic tube with $\mu = 0.5$; ΔT_0 is 15° C in all). Therefore, with $\bar{l} = 3-5$, the vortex energy transfer effect reaches a maximum strength. From sections I to III, the strength of the vortex effect falls off, and ΔT_{0e} begins to decrease. The variation in static temperatures of the external and internal streams (ΔT_e and ΔT_i) in the range $\bar{l} = 10-26$ has the same character as ΔT_{0e}. In the range between sections I and III an increase in ΔT_e occurs, followed by a sharp fall to $\Delta T_e = -66$ deg. In the range between sections III and C, ΔT_i decreases from 3 to −34 deg.

The heat transfer process in the nonadiabatic vortex tube plays a decisive role in energy removal. On the basis of the experimental data obtained for fields of velocity and temperature in the adiabatic and nonadiabatic tubes, values were determined for the coefficient α of heat transfer via the vortex flow as a function of tube length $\bar{l}$.

The coefficient of heat transfer α_1 from the vortex flow to the tube wall was determined in two ways.

Using the first method a value was determined from the test data, using the MPI (Moscow Power Institute) formula for the case for a flat plate washed by a turbulent stream of diatomic gas [7]:

$$\mathrm{Nu} = 0.032\,\mathrm{Re}^{0.8} \quad (10^5 < \mathrm{Re} < 2\cdot 10^6). \qquad (1)$$

The Re number was determined from the experimental values of velocity, measured in the nonadiabatic tube. The physical parameters were referred to the stagnation temperature of the oncoming stream, measured at the tube wall at each section (from I to V).

The characteristic dimension in the hydrodynamic and thermal parameters (Re, Nu) was taken to be the circumference of the tube at the given section ($L = 2\pi r$). Values of α_1, obtained from formula (1), are shown by the broken lines in the graph of $\alpha = f(\bar{l})$ (Fig. 4).

Using a second method, values of α_1 were determined from the heat flux q passing through the tube surface included between sections I–V. From the experimental data on behavior of the stagnation temperature in the adiabatic and nonadiabatic vortex tubes, calculations were made for the heat fluxes passing from the wall layers of the gas through the conducting surfaces enclosed between sections I–V. From the amount of heat q transmitted by the gas through each elementary annular surface F_i, and from the mean temperature difference between the gas and the cooling water $\Delta\bar{\tau}$, the mean heat transmission coefficient for each surface element was calculated:

$$\bar{K}_i = \frac{q_i}{F_i\,\Delta\bar{\tau}} \quad \mathrm{W/m^2\cdot{}^\circ C}. \qquad (2)$$

From the mean value K_i for a known mean value of the coefficient of heat transfer to the cooling liquid α_2, and the calculated thermal resistance of the tube wall, a mean coefficient of heat transfer with respect to the air α_1 was calculated for each section of the surface. Values of α_1 obtained in this way are shown by full lines on the graph of $\alpha_1 = f(\bar{l})$ (Fig. 4).

Comparison of the values of α_1 obtained by the two methods shows that over the whole length of the tube α_1 calculated by the thermal method gives larger values than those calculated by formula (1), the divergence reaching as high as 40–45%. This is due to the fact that formula (1) does not take into account special features of heat transfer in turbulent vortex streams washing a curved surface.

Calculation of the heat transfer coefficient for a vortex turbulent stream washing a curved surface may be made from the equation

$$\mathrm{Nu} = 0.052\,\mathrm{Pr}^{0.333}\,\mathrm{Re}^{0.8}. \qquad (3)$$

Values of α_1 calculated from this formula are shown in Fig. 4.

It may be seen from the graph that the largest value of α_1 is observed in the initial cooled section of the vortex tube. Thus, between sections I' and I, values of α_1 are roughly 1000–1100 $\mathrm{W/m^2\cdot{}^\circ C}$, at the section I–II they are about 700–900, and at the section III–V roughly 300–400. Such a great difference between the values of α_1 has the result that the largest amount of energy in the form of heat is transferred at the initial section of the vortex tube. In the section I–II, whose surface is 19% of the total tube surface, up to 40–50% of the total amount of heat removed by the water is transferred. For $\mu = 1$ the mean value of α_1 averaged over the tube length is 450–550 $\mathrm{W/m^2\cdot{}^\circ C}$.

NOTATION

G is the total quantity of air; G_x is the quantity of cold air; $\mu = G_x/G$ is the fraction of cold air; φ is the angle made by the velocity vector; c_τ is the tangential velocity; c_a is the axial velocity; δ is the thickness of the external stream: r_c is the radius of the cylindrical part of the tube; T_{00} is the temperature of the compressed gas upstream of the tube; T_{0e} is the stagnation temperature of the external stream; T_{0x} is the stagnation temperature of the cold stream; T_e and T_i are the static temperatures of the external and internal stream, respectively; $\Delta T_{0e} = T_{0e} - T_{00}$; $\Delta T_{0x} = T_{0x} - T_{00}$; $\Delta T_e = T_e - T_{00}$; $\Delta T_i = T_i - T_{00}$; l is the tube length; D_n is the tube diameter in the nozzle section; $\bar{l} = l/D_n$ is the reduced length; α_1 and α_2 are the coefficients of heat transfer from the vortex stream and to the cooling liquid.

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