INVESTIGATION OF CONICAL VORTEX TUBES

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The results of tests on vortex tubes of new construction are presented.

During the last two decades work has been done in many countries to increase the effectiveness of the vortex tube as a generator of cold gas. The success attained has resulted from purely empirical research. The hypotheses advanced to explain the effect not only fail to point to ways for improvement, but also do not permit assessment of the significance of successes attained, since there has been no definition of what the "ideal" vortex tube might be. The constructional development of the vortex tube has passed through the following basic stages.

Hilsch [1] established optimum relations between the dimensions of the basic elements of the cylindrical vortex tube with circular inlet nozzle: $f = F_c/F = 0.057$, $d/D = 0.37-0.48$, $L/D \approx 50$. The vortex tubes investigated by Hilsch were distinguished by a small value of f (small relative amount of gas supplied) and attained a comparatively high refrigeration capacity $\eta\chi$ even at high pressure ($(\eta\chi)_{max} = 0.22$ at $\pi = 4$ and $(\eta\chi)_{max} = 0.19$ at $\pi = 11$), the thermal efficiency η increasing with increased pressure and reaching a value $\eta_{max} = 0.47$ only at $\pi = 11$.

Merkulov [2] suggested twisting the gas in a plane cochlea defined by an Archimedes spiral, with a rectangular-section nozzle inlet. This improvement of the flow passage allowed an increase in the relative section of the nozzle inlet up to $f = 0.096$ (in the range $2 \leq \pi \leq 6$) and increased the thermal efficiency to $\eta_{max} \approx 0.5$ ($d/D = 0.45$), without reducing the values of $(\eta\chi)_{max}$ ($d/D = 0.55$). Merkulov also noted that the cylindrical vortex tube with a crosspiece in a flow of heated gas has an optimum length $L = 9D$, the operation of the tube sharply deteriorating in the event $L < 9D$, while the $L > 9D$ the influence of the length is small.

Finally, Hendal [3] proposed a vortex tube in the form of a divergent conical nozzle (length ~5D, $\alpha = 2-6°$), followed by a long cylindrical portion. A tube of this construction ($D = 10$ mm, $\alpha = 3.6°$) proved to be more effective by 10% than a cylindrical tube, and with external water cooling, by 15%.

A smooth increase in curvature of the surface of the cochlea and a smooth decrease in the area of the nozzle channel may also increase the efficiency of the tube by several percent [4]. The assertion in [4] that "vortex deceleration" has a considerable influence is unfortunately not borne out by the experimental data. In contrast to the extended form of the comparatively long vortex tube ($L/D > 10$) [1, 2, 3], Metenin [4], in tests on a vortex tube in the form of a conical nozzle with a vane diffuser at its hot end, obtained satisfactory results only with a short tube ($L/D = 3$). The question of optimum length is also very important for an understanding of the vortex tube mechanism.

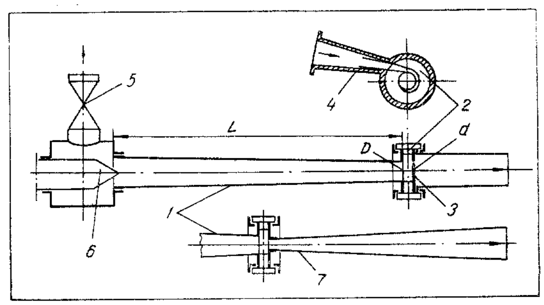

Fig. 1. Circuit of the vortex tubes investigated: 1) conical tube; 2) plane cochlea (Archimedes spiral); 3) diaphragm; 4) nozzle; 5) control valve; 6) cone; 7) diffuser for cooled gas.

The present paper sets out some results of practical value obtained in tests on long conical tubes, performed with the object of explaining the physical principle of the vortex effect.

Experimental setup. The conical tubes ($D = 30$ mm, $\alpha = 2.3°$, $L = 400-840$ mm) were made of stainless steel sheet 0.6 mm thick (Fig. 1). The tubes had a mirror finish, and the deviation of their sections from the circular, measured particularly in the region of the contact weld seam, did not exceed 0.6 mm (≤2%). Compressed gas was supplied through drying nozzle 4, which had a smooth transition from a circular to a square (12 × 12 mm) section, to a plane spiral cochlea 2 (Archimedes spiral), whose section decreased from 12 × 12 mm to an exit section of 6 × 12 mm ($f = 0.10$). Tube conditions were controlled by the valve 5; the cone 6 gave a symmetrical outflow of heated gas. The cooled gas discharged either through the aperture of diaphragm 3 or through the conical diffuser 7 of angle 4° and length about 500 mm, made, like the tube, of stainless steel sheet. All the elements of the vortex tube were covered with thermal insulation. Because of the small mass of the elements of the tube, the "thermal inertia" of the equipment was small, which allowed short term variations of the tube conditions to register during continuous recording of the thermocouple emf's.

The working gas was nitrogen (except tests d in Fig. 2A), circulating in a closed system, filled with pure dry gas from a tank containing evaporating liquid nitrogen. Measurement of the differences ΔT_1

and ΔT_2 was done with the aid of calibrated copper-constantan thermocouples of diameter 0.2 mm, whose junctions, enclosed in a small stainless steel tube (1 × 0.1 mm) with apertures to let the gas in and out, were located at low-velocity points in the gas stream and in the chamber near the cone 6. A battery of two thermocouples was used to measure difference ΔT_1. The thermocouple emf's were recorded with an EPP-09 automatic potentiometer (10 mV, class 0.5).

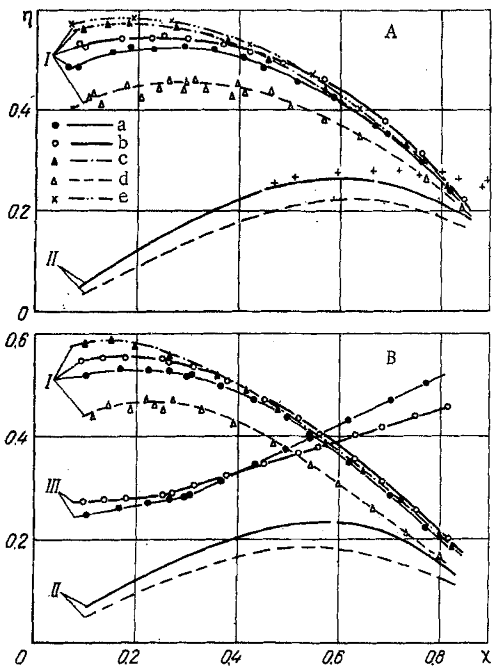

Fig. 2. Characteristics of vortex tubes with $\pi = 4$ (A) and 8 (B): a, b, c, e) conical; d) cylindrical (D = 30 mm); a) with L/D = 13 (diaphragm d = 15 mm); b) 28 diffuser d = = 17 mm); c) 13 (diffuser d = 15 mm); d) 13 (diaphragm d = 15 mm); e) for tube c, the working gas was helium, $\pi = 3.4$; I—values of η; II—ηx; III—p_2/p_0.

In measuring the temperature fields at the surface and inside the tube, the emf's of 12 thermocouples were recorded simultaneously by two six-point potentiometers. The errors in calibration and measurement did not exceed 1% of the temperature difference being measured.

The measurements of mass flow rate of gas with the aid of conventional diaphragms were used mainly to verify the heat balance of the tube. Values of χ were calculated according to the measured differences ΔT_1 and ΔT_2, using the condition of zero thermal losses, since the discrepancy in heat balance did not exceed 10%.

Results and discussion. In testing the cylindrical tube (D = 30 mm, L/D = 13) it was noted that considerably improved results were observed if the vortex tube conditions were controlled with the help of valve 5, and the cone 6 was mounted in some optimum position (curve e in Fig. 2A and B). But in that case the tube regime was unstable, and the good results observed could not always be reproduced; the introduction into the tube of a straightening spider actually worsened the situation somewhat.

To improve the operation of the vortex tube it was expedient to replace the cylindrical tube by a high-performance diffuser channel of sufficient length and small divergence angle. For this purpose a long conical tube was prepared (D = 30 mm, L = 840 mm, $\alpha =$ = 2.3°); the results of investigating this tube are shown in Figs. 2 and 3. The flow regime in the conical tube proved to be more stable, and depended only slightly on the position of the cone 6.

Because of the great length of the tube, its small wall thickness (0.6 mm), and the low thermal conductivity of stainless steel (0.15 W/cm degree), the heat flux along the walls of such a tube may be considered negligibly small in comparison with the heat transfer between the wall and the gas flowing inside. Therefore the temperature of the wall insulated on the outside must be close to the stagnation temperature of the gas directly at the wall. Figure 3 shows the distribution along the tube length of the wall and the stagnation temperature of the gas at the tube axis for various values of χ (the temperatures of all the points at each value of χ are noted simultaneously). It is interesting to note that at small values of χ the distribution of gas stagnation temperature on the tube axis is similar to the temperature distribution on the tube surface, while the gas temperature at the wall with L/D ≈ 13 exceeds the mean temperature of the heated gas emerging from the tube. The temperature distribution obtained permits certain conclusions to be drawn regarding the extent of the "active" zone of the tube. Since the stagnation temperature of the peripheral gas layers in the insulated tube may be exceeded only by receipt of energy from the cooled flow, it is natural to suppose that the active zone corresponds to the region in which the peripheral temperature is exceeded. As may be seen from Fig. 3, this region, which occupies the entire tube at small χ, decreases with increase of χ, but even at $\chi = 0.90$, its length is not less than 13D. In order to clarify the question as to whether the extra length would prove to have a harmful influence on the operation of the tube, tests were also made on a conical tube of length L = 13 D, whose characteristics, other conditions being equal, turned out to be practically the same as those of the long tube.

It was shown in a series of tests that the conical diffuser 7 had an appreciable influence on the operation of the vortex tube. With identical diameters of diaphragm and diffuser inlet aperture (d/D = 0.54), the diffuser considerably increases the thermal efficiency in the region of small values of χ ($\chi < 0.3$; compare curves *a* and c in Figs. 2A and B). For large diameter (d = 17 mm, d/D = 0.565), the diffuser also increases the refrigeration capacity of the tube (b in Fig. 2A and B).

In order to explain the influence of the isentropic exponent $k = C_p/C_v$, one tube was tested both in nitrogen (k = 1.40) and in helium k = 1.67 (compare curves c and e in Fig. 2A). With helium, only somewhat higher values of η were obtained, which undoubtedly indicates the expediency of reducing the experimental data in the form of the equation

$$\Delta T_1 = \eta \Delta T_s = \eta \left[1 - \left(\frac{1}{\pi} \right)^{\frac{k-1}{k}} \right] T_0, \qquad (1)$$

where the function η may be considered independent of the isentropic exponent.

In one test the tube (b, Fig. 2A) was cooled on the outside with water at a temperature close to that of the gas supplied (~18° C). The water cooling allowed maximum relative refrigeration capacity to be obtained $\eta\chi^{\max} = 0.28$ with $\chi = 0.8$ (the lower points on Fig. 2A), but further increase of χ led to considerable deterioration in the operation of the tube.

Measurements of pressure p_2 of the heated stream ahead of valve 5 showed that the dependence of p_2 on χ is close to linear, and changes slope noticeably at some value χ', which decreases with increase of π ($\chi' \approx 0.25$ with $\pi = 8$, Fig. 2B). The more stable operation of conical tubes allowed observation of small temperature discontinuities at these same values of χ. The phenomenon noted points to the fact that in the vortex tube there exist at least two different hydrodynamic flow regimes, with discontinuous transition from one to the other, and the characteristics of the vortex tube do not necessarily have the form of smooth curves.

The results obtained allow the following conclusions to be drawn:

1. The conical vortex tubes investigated for thermal efficiency and refrigeration capacity surpassed the best cylindrical tubes by 20–25%.

2. The length of vortex tubes of the construction described should not be less than 13 D.

3. Passing the cooled flow through a conical diffuser ($\alpha = 4°$) increases the thermal efficiency at small values of χ ($d/D = 0.5$–0.52) and increases the refrigeration capacity of the tube ($d/D = 0.57$).

4. Change of the isentropic exponent $k = C_p/C_v$ from 1.40 to 1.67 (nitrogen and helium) has very little influence on the thermal efficiency of the process in the tube.

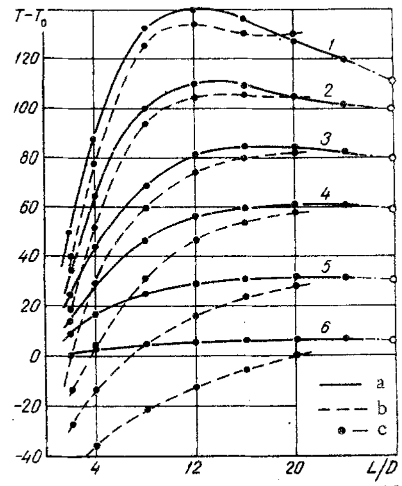

Fig. 3. Distribution of temperature $T-T_0$ (degrees) along the tube for various values of χ (1—0.90; 2—0.77; 3—0.69; 4—0.58; 5—0.36; 6—0.11): a) temperature of the tube surface; b) stagnation temperature of the gas on the tube axis; c) points corresponding to the value of ΔT_2 at the indicated χ.

NOTATION

G—total mass flow of gas through the tube; G_1—mass flow of cooled gas; $\chi = G_1/G$—fraction of cold flow; T_0, T_1, T_2—absolute stagnation temperatures of gas at tube inlet, in cooled flow, and in heated flow, respectively; t_s—absolute thermodynamic temperature of gas at end of reversible adiabatic expansion; $\Delta T_1 = T_0 - T_1$; $\Delta T_2 = T_2 - T_0$; $\Delta T_s = T_0 - t_s$; $\eta = \Delta T_1/\Delta T_s$—thermal efficiency of process in vortex tube; $\eta\chi$—relative refrigeration capacity of vortex tube; p_0, p_1, p_2—stagnation pressure of gas at tube inlet, in cooled flow, and in heated flow, respectively; $\pi = p_0/p_1$; $k = C_p/C_v$; L—tube length; D—least tube diameter near nozzle inlet; d—diameter of diaphragm or of diffuser inlet aperture; α—cone vertex angle; $F = \pi D^2/4$; F_c—area of smallest section of nozzle; $f = F_c/F$.

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