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STUDY OF A FINNED TWO-CHAMBER VORTEX TUBE

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A two-chamber "module" of a multichamber vortex-type cold-air machine is examined in four operating regimes.

Vortex tubes with internal and external finning of the cold-cooled energy-separating chamber [1, 2] are used in small air-conditioning units for commercial electronics, in devices providing individual and group protection of workers from heat in hot shops, and in other applications. Expansion of the range of use of this equipment is requiring a further improvement in their service characteristics. The foremost needs are to increase the range of temperature control and the discharge of cooled flow and to enable the equipment to be attached to existing air supply system – which are chracterized by a wide range of compressed-air pressures. The use of two-chamber [3] and multichamber [4] vortex units has some potential in this regard.

The simplest two-chamber module of a multichamber vortex-type cold-air machine is a design with identical ($\overline{F}_1 = \overline{F}_2$; regime A) or different ($\overline{F} \neq \overline{F}_2$; regime B) inlet sections in the swirl chambers (Fig. 1). If necessary, only one inlet ($\overline{F}_1$ or $\overline{F}_2$; regime C) can be connected to a compressed-air main. It is also interesting to explore the possibility of producing cold with two inlet connected to the source ($\overline{F}_1$ and $\overline{F}_2$) but with the removal of the cold flow through a single outlet channel 3 (regime D) rather than two, as in regimes A, B, and C.

Figure 2 schematically depicts the operating regimes of the two-chamber module.

The diameter of the swirl chambers of the module we studied was 38 mm; the relative cross-sectional area of the inlets $\overline{F} = 0.08$-0.11, the length of the unfinned initial section 7 (see Fig. 1) was 114 mm, the distance between the nozzle sections of inlets 1 and 2 (the axial length of the module without the outlet channels 3) was 533 mm, and the area of the internal and external finning was 0.21 and 2.94 m^2, respectively.

Tests were conducted with undried compressed air. The temperature in the experiment was measured by thermocouples to within 0.1 K, pressure was measured by manometers with an accuracy of class 0.6, and flow rates were determined by means of Venturi meters.

In regime A, we obtained two cold flows with identical temperatures and flow rates. Figure 3a shows the temperature-energy characteristics of a two-chamber module with cold-flow outlet channels in the form of a pipe (solid curves) or a slitted diffuser (dashed lines).

In regime B, the discharge of the cold flows was redistributed in the following proportion

$$\frac{G_{x1}}{G_{x2}} = \frac{F_1}{F_2}. \quad (1)$$

Thus, a swirl chamber with a large inlet cross section operates at $\mu_1 = G_x/G_c < 1$, while the second chamber studied operates at $\mu_2 > 1$. The temperatures of the cold flows in the left and right outlet channels 3 (see Fig. 1) were different, with the size of the axial hole 6 in the central fin having a decided effect on this difference. The relative diameter of

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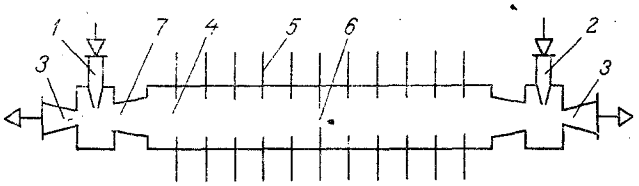


Fig. 1. Diagram of two-chamber module: 1, 2) inlet nozzles for compressed air; 3) outlet channels for the cold flow; 4) energy-separating chamber; 5) internal and external finning; 6) variable axial hole in the central fin; 6) unfinned conical section of the chamber (α = 3°).

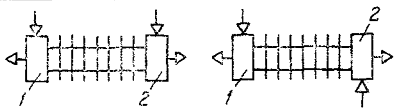
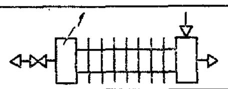
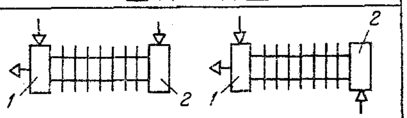
Operating regime	Cross section of nozzle inlets	Operating scheme
A, A'	$F_1 = F_2$	
B	$F_1 \neq F_2$	Unidirectional rotation; Counter rotation
C	$F_1 = 0$	
D, D'	$F_1 = F_2$	 Unidirectional rotation; Counter rotation

Fig. 2. Operating regimes of the two-chamber module.

the hole $\bar{D}_c = D_c/D$ was taken equal to 0.29; 0.45; 0.64; 1.16. Here, the relative diameter of the axial hole in all of the remaining fins $\bar{D}f = Df/D = 1.16$ and the relative cross section of the nozzle inlets $\bar{F}_1 = 0.11$, $\bar{F}_2 = 0.08$, respectively, for the first and second chambers of the module (Fig. 3b-e). It was established that a decrease in $\bar{D}_c$ is accompanied by a difference in the temperatures of the two cold flows, with no change in the energy efficiency of the module. In the case of counter rotation of the vortical flows in the two-chamber module, the difference between the temperatures and velocities of the two cold flows (regime D') is lower than with unidirectional rotation (regime D), i.e., Eq. (1) is not satisfied. This can be attributed to the development of considerable turbulence at the site where the vortices meet and, thus, to an increase in the resistance to the movement of the axial flow through hole 6 (see Fig. 1).

Analysis of the experimental results made it possible to establish mathematical relative quantity $\Delta T_{x2}/\Delta T_{x1}$ and the ratio $\bar{F}_2/\bar{F}_1$:

$$\Delta T_{x2}/\Delta T_{x1} = 3\bar{F}_2/\bar{F}_1, \tag{2}$$

as well as with the relative diameter of the axial hold $\bar{D}_c$:

$$\Delta T_{x2}/\Delta T_{x1} = 3{,}8\bar{D}_c + 0{,}5. \tag{3}$$

Equation (2) is valid at $1.0 \leq \bar{F}_2/\bar{F}_1 \leq 1.5$ and $\bar{D}_c = 1.16$, while Eq. (3) is valid at $0.29 \leq \bar{D}_c \leq 1.16$ and $\bar{F}_2/\bar{F}_1 = 1.5$.

It was established that it is possible to additionally reduce the temperature of the cold flow when the module is operated in regime A – with removal of some of the heated peripheral layers of gas from the middle of the energy-separating chamber – or in regime C, i.e., with $\mu < 1$. In regime C, the maximum of exergetic efficiency [5] is attained with operation on undried air at $\mu \approx 0.7$ (Fig. 3f), while the maximum value of the energy efficiency index is reached at μ = 0.6-1.0.

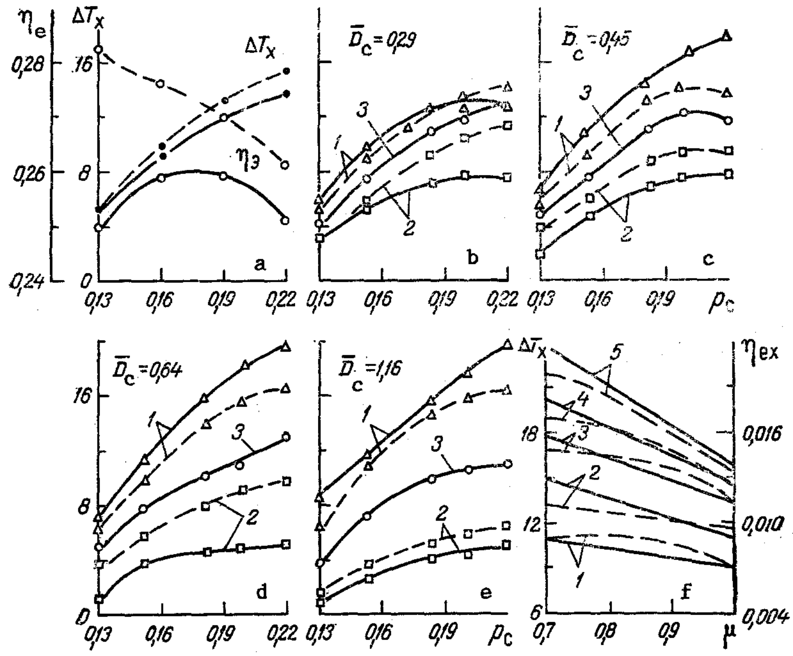


Fig. 3. Characteristics of a two-chamber module: a) dependence of the cooling effect ΔT_x and the energy efficiency coefficient η_e on p_c; b-e) dependence of the cooling effect ΔT_x on p_c, $\bar{D}_c$. and the direction of rotation of the vortical flows (solid lines show data for unidirectional flow, dashed lines show data for counter flow): 1) first chamber of module; 2) second; 3) average value of ΔT_x; f) dependence of the cooling effect ΔT_x (solid curves) on the exergetic efficiency η_{ex} (dashed lines) on μ at π: 1) 1.15; 2) 1.18; 3) 1.20; 4) 1.22; 5) 1.26. ΔT_x, K; p_c, MPa.

NOTATION

D, diameter of the vortex tube; D_c, diameter of the axial hole in the fin; F, cross-sectional area of the inlet nozzle; p_c, p_x, total pressures of the compressed air and cold flow; $\bar{F} = 4F/\pi D^2$, relative cross sectional area of the nozzle inlet; $\pi = p_c/p_x$, pressure ratio; $\Delta T_x = T_c = T_x$, cooling effect; T_c, T_x, stagnation temperature of the compressed air and cold flow; k_e, k_{in}, finning coefficients of the external and internal surfaces of the energy-separating chamber; G_c, G_x, mass flow rates of the compressed air and cold flow; $\mu = G_x/G_c$, relative discharge of the cold flow; $\eta_e = \mu\Delta T_x/\Delta T_s$, energy efficiency coefficient; ΔT_s, cooling effect with isentropic expansion from the pressure p_s to the pressure p_x; η_{ex}, exergetic efficiency of the vortex-type cold-air machine [5].

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