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2018

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ABSTRACT

Two types of gas-dynamic coolers were considered: Rank-Hilsh vortex tubes and resonance tubes, known as devices of Hartmann-Sprenger. These devices have a number of similar physical and structural features: no moving parts, low inertia, environment safety, they can produce cold and heat, generate pressure pulse, and separate gas mixtures. Gas-dynamic devices can be used in the technologies of rare gases obtaining as coolers of reflux condensers. They use pressure drop and don't need additional energy consumption. The temperature drop of mixtures phase equilibrium allows to reduce the concentration of by-products. Due to this, costs of gas concentrates transportation are reduced. Schemes, in which the separation function of the gas-dynamic devices are used, were investigated. An influence of a scale factor in the interval of the energy separation chamber's diameter up to 2 mm was studied.

1. INTRODUCTION

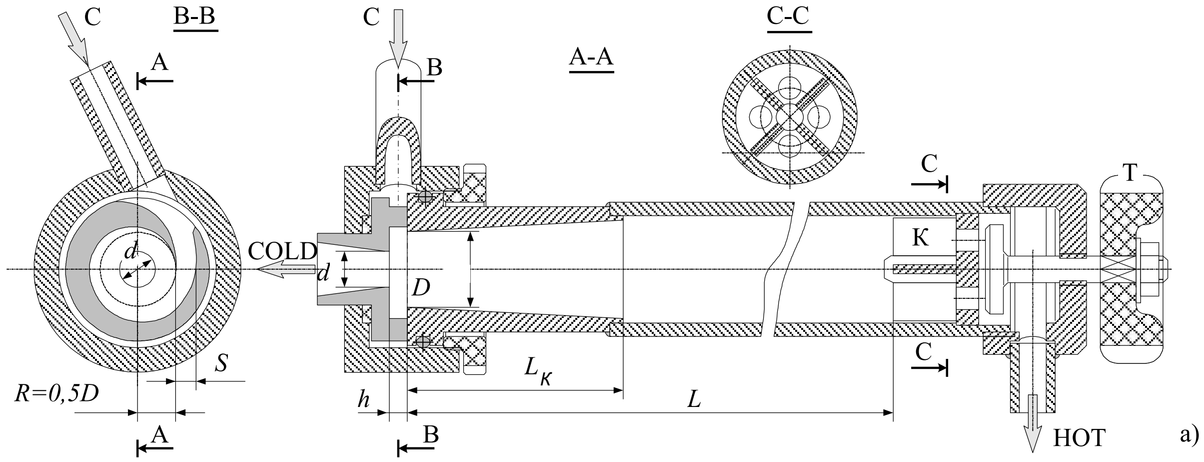
In gas-dynamic devices, the energy of the compressed gas is transformed into the thermal energy and partially discharged to the external medium through the device walls or as an exhausted gas. In this case, the temperature of the main flow at the device outlet decreases. The first samples of such coolers appeared relatively recently. Following the works of J.Rank (1934) and R.Hilsch (1946) the regularities of the vortex effect were studied by M.G. Dubinsky (1954), V.S. Martynovsky (1956) and V.M. Brodyansky (1960).

Along with the development of vortex engineering in the 1950s, was developed another type of gas-dynamic cooler, a resonance tube (Sprenger, 1954). Investigations of the energy separation nature in a pulsating gas column were conducted at the Moscow Bauman State Technical University under the academic supervision of A.M. Arkharov *et al.* (1983). In the same place, A.D. Suslov (1980) studied in detail the cooling processes in pulsating devices with regenerators. In them, the working substance temperature is lowered during the internal adiabatic expansion with the periodic exhaust of gas from the constant volume.

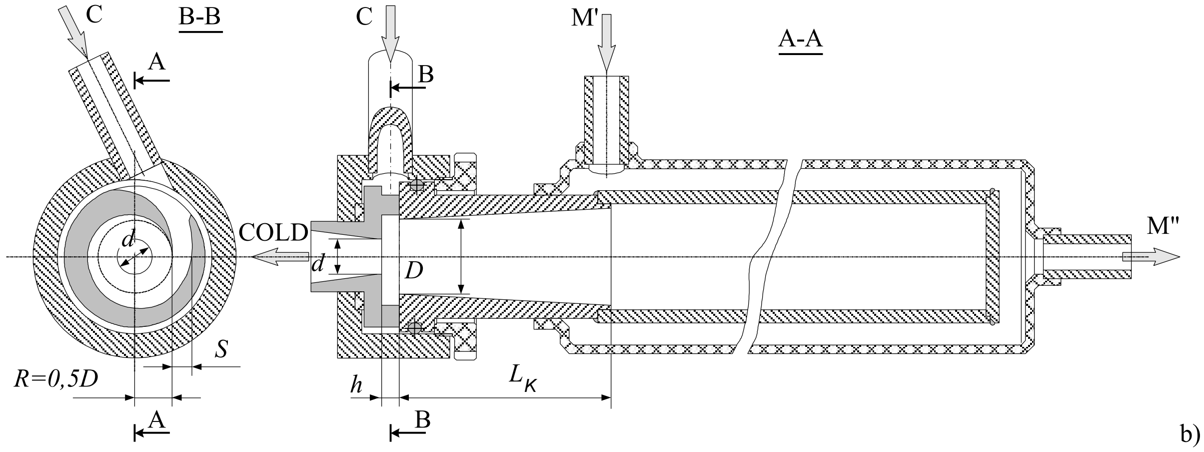
The theoretical and experimental studies made it possible to identify the optimum dimensions of the flowing part of gas-dynamic cooling devices. The first industrial models, which were used for the purposes of conditioning and in other fields of refrigeration technology, were created. Nowadays, machine-free devices are used in heat protective suits, in the repair of power facilities, in the gas industry and in transport.

2. OPERATION PRINCIPLE AND GENERAL FEATURES OF GAS-DYNAMIC DEVICES

Among the variety of cooling processes, vortex and wave energy devices take a special place. They combine important operational advantages: a complete absence of moving parts, low inertia, environmental safety, the possibility to produce cold, heat and vacuum simultaneously. The range of temperatures at which machineless energy separators are used is very wide – from low-temperature steps of helium liquefiers (Simonenko *et al.,* 1992) to fuels in jet engines (+ 700 °C) (Piralishvili and Novikov,1984). The special vortex devices capable to generate low-frequency pressure oscillations and sound waves (Azarov *et al.,* 1982), as well as to extract liquid (solid) fractions from a mixture of substances (Leites *et al.,* 1974, Rachevsky, 2009, Suslov *et al.,*1980) are known. Universality, reliability, and long work-term of gas-dynamic devices allow to predict the growing role of these devices in the newest technologies of the XXI century.
Many varieties of gas-dynamic devices have been created (Desyatov, 1985) depending on the active tasks and fields of application. The typical types of such coolers are shown in Figure 1.
The compressed flow is fed in the vortex tubes energy separation chamber (a and b) through the tangential nozzle inlet C. The gas forms a high-speed flow and is divided into a cold (with temperature $T_{cold}<T_c$) and hot (with temperature $T_h>T_c$) constituents. The cold flow is formed in the axial zone of the chamber and flows out of the vortex tube through the diaphragm located near the B-B nozzle section. The hot flow is formed in the peripheral area of the chamber and moves along its wall in a direction opposite to the direction of the cold flow. The hot gas H ejected of the vortex tube through the throttle valve D. When the cross-section of the valve is varied, the flow rate and temperatures of the cold and hot flows vary. In the case of full opening of the valve D, the vortex tube can switch to ejector mode sucking ambient air into the chamber through the diaphragm ($G_h>G_c$). The full closing of the valve results in the exhaust of the all-entering gas through the vortex tube's diaphragm (G_c=G_{cold}, G_h=0), practically without changing its temperature. To ensure a temperature drop in this mode, it is necessary to remove heat from the vortex tube through the chamber walls (Figure 1-b). Tor this purpose, a cooling jacket or finning are used (Azarov *et al.,* 1984).
Despite the obvious technical differences, the vortex and wave devices have a number of common physical features. The mechanism of temperature separation in the vortex chambers is accompanied by redistribution of energy between rotating layers of gas. The first studies of the structure of a swirling flow showed that the radial distribution of circumferential velocity in the axial and peripheral vortex areas is not the same. The variation of the circumferential velocity along the radius of the vortex tube for the sections located at different distances from the diaphragm plane is shown at Figure 2. In the peripheral area, the variation of the circumferential velocity obeys the law of potential flotation in which v_τ increases in the direction from the wall to the axis (1). In the central part, the circumferential velocity v_τ increases in the opposite direction – from the axis to the chamber wall in proportion to the radius *r* (2). Such law of variation of v_τ is typical of the rotation of solid body parts with a constant angular velocity.
As the gas moves along the chamber, the circumferential velocity and the radial gradient of the static pressure decrease. Gradually, the free vortex begins to propagate to the axis transferring a part of the rotational kinetic energy to the axial flow. In accordance with the hypothesis of A.P. Merkulov (1969), the gas elements are intensely turbulized during the process of transition to the near-axial area. There is an intensive transfer of heat and mass under conditions of a radial drop of static pressure. When the gas element moves from one position to the other, it adiabatically expands (or compresses) in a field with a high-pressure gradient, and its temperature changes. If the temperature of the microvolume is different from the temperature of the other elements located on this radius, then gas in this layer will be heated (cooled) as a result of mixing. In particular, the elements of the gaseous medium perform the cooling cycles. They transfer heat to the peripheral layers, and turbulent pressure pulsations are the source of the mechanical energy of these cycles. The special means are sometimes used to increase pressure fluctuations in vortex chambers (Alekseyev *et al.,* 1984).



$F = h \cdot s$; $F = \sigma \cdot \dfrac{\pi}{4} \cdot D^2$; (where σ=0,05…0,1); d=(0,5…0,6)$\cdot D$; L_K=3D; $L \geq 9 \cdot D$



$F = \dfrac{\pi}{4} \cdot (d^2 - b^2)$; b=0,75$\cdot d$; $L = \dfrac{a}{4 \cdot f}$; $a = \sqrt{R^* \cdot k \cdot T} = \sqrt{\dfrac{R_0}{M} \cdot k \cdot T}$; f=100…150 *Hz*

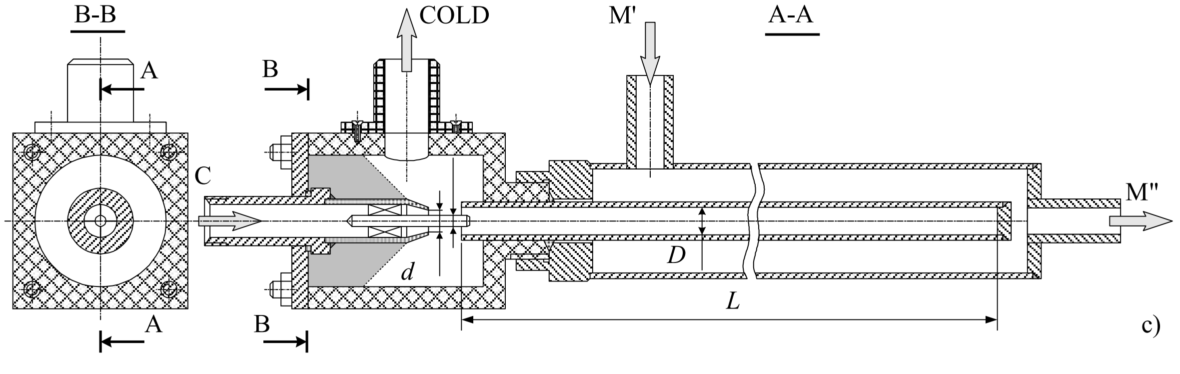
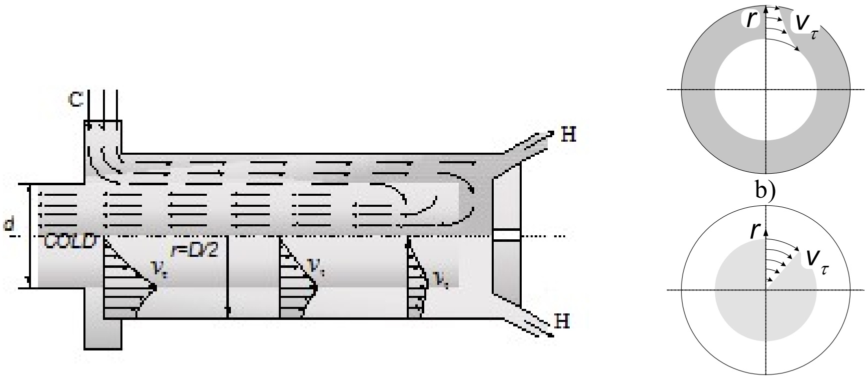


Figure 1: Classical designs of gas-dynamic coolers

a) – uncooled (adiabatic) vortex tube with the exhaust of a hot flow; b) – a vortex tube with a cooling jacket of the energy separation chamber; c) – resonant cooler with sound source of oscillations; C – compressed gas; COLD – cold flow; HOT – hot flow; M′ and M″ inlet and outlet of the cooling medium; K – cross tee; T – hot flow throttle

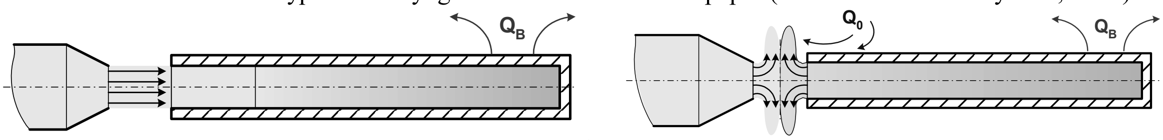


$$v_\tau \cdot r = const\,, \quad (1)$$

$$\omega = \frac{v_\tau}{r} = const\,, \quad (2)$$

Figure 2: Scheme of gas flow in the vortex chamber and diagrams of the tangential velocity component a); the law of variation of the radial velocity in free axial vortex, peripheral *b*); and forced axial vortices *c*)

The work of wave coolers is also based on thermal nonequilibrium in the pulsating gas layer. According to (Arkharov *et al.* 1983), a cycle in a resonant tube is conditionally divided into two phases (Figure 3): during the first phase (a), the jet enters the tube and compresses the gas contained therein. During the second phase, the working medium is expanding into the outer space (b). In this case, a deviation of the gas jet at the nozzle outlet in the transverse direction is observed. There is a "pumping" of heat energy in the end of the resonance chamber distant from the nozzle cross-section and cooling of the exhausted jet. A more detailed thermodynamic model of energy conversion in a resonance-type wave cryogenerator is shown in the paper (Bondarenko and Losyakov, 2006).



a) b)

Figure 3: Operation phases of the resonant tube a) compression; b) expansion

Despite apparent differences, the same principle of thermal nonequilibrium of the pulsating gas is the operational basis of the considered devices. In vortex tubes, this process is densely "packed" along the radius of the vortex chamber, and turbulent phenomena are its source. In resonance and pulsation coolers, the elastic oscillations of the gaseous medium column are generated by mechanical or jet (acoustic) means. The technical solutions in Figure 1-b and 1-c are the closest in physics. The devices that implement the vortex and resonance principle of flow cooling can have identical operational features. They can be distinguished from each other only by indirect signs and external manifestations, which accompany the operation, for example, by the temperature distribution along the chamber walls or by frequency acoustic parameters (Azarov *et al.,* 1982).

3. MACHINELESS COOLERS IN CRYOGENIC ENGINEERING

The method of separation of solid and liquid fractions in a rotating flow is known for more than a century. It should be recalled that the discovery of the temperature vortex effect in 1931 was the result of the study of cyclone separators (Rank, 1934). The problem of separating condensate directly in the vortex tube is of immediate interest (Azarov *et al.,* 1986), since the formation of the solid and liquid phases often distorts the geometry of the flowing part and adversely affects the energy efficiency. It is possible to combine both functions in one device: cooling, and separation, with the right schematic and constructive design of the vortex chamber (Lavrenchenko *et al*., 1989).

The inclusion of machineless devices in the air separation plant scheme allows to reduce the expansion in a high-pressure turbo-expander. Due to the additional stage, the initial pressure at the inlet of the expansion machine decreases from 20 to 8 MPa. Gas-dynamic losses in the flow part of the turbo-expander are diminished; dynamic loads and the optimum rotor speed value are reduced (Shadrina, 1984). Along with this, the geometric dimensions of the flow part increasing and the positive effect of the scale factor is observing. Figure 4 shows the options for inclusion of gas-dynamic cryogenerators into a high-pressure air separation system.

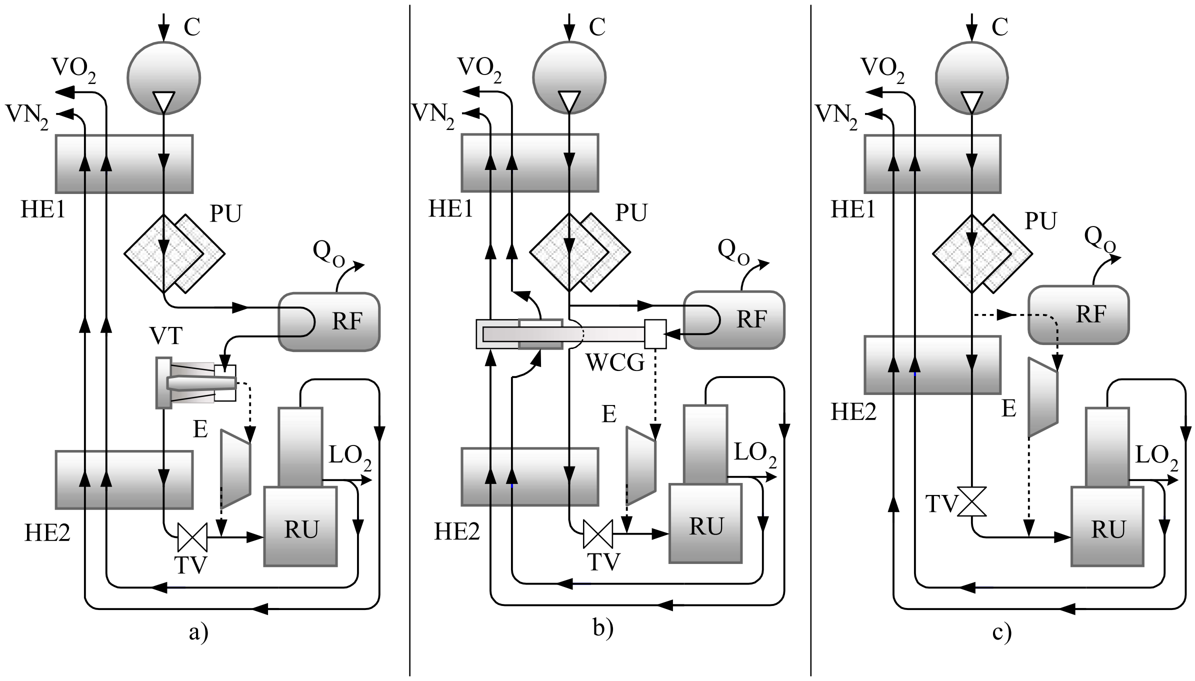


Figure 4: Vortex a) and wave b) coolers included in the high-pressure ASU KzhKAzh -0,25; c) basic version

The helium technique (Figure 5) is an important field of application of machineless devices. In addition to vortex separators (Figure 5-a) and ejectors, gas-dynamic coolers are used. In the low-temperature installation (Arkharov *et al.* 1984), the kinetic energy of the expanding cryogenic agent is converted into the energy of the acoustic oscillations (Figure 5-b). It is removed as heat in the heat exchanger HE3 due to the reverse flow and additionally in HE2 into which the flow from the expander E is introduced.

The modernized scheme CGU-250/4,5 (Arkharov *et al*., 2002, Bondarenko *et al.,* 2008) is shown in Figure 5-c. In it, in parallel with the throttle valve TV1, the wave cryogenerator WCG is switched. With the DV1 closed, the heat released in the resonance tube is diverted to the expander flow E1-E2. Due to the additional cooling of the direct flow part in the T6 heat exchanger, an isothermal throttle effect is increased at the inlet to the HE7-TV2 stage. In turn, this facilitates the increase of the liquefier productivity in comparison with the regular operation mode.

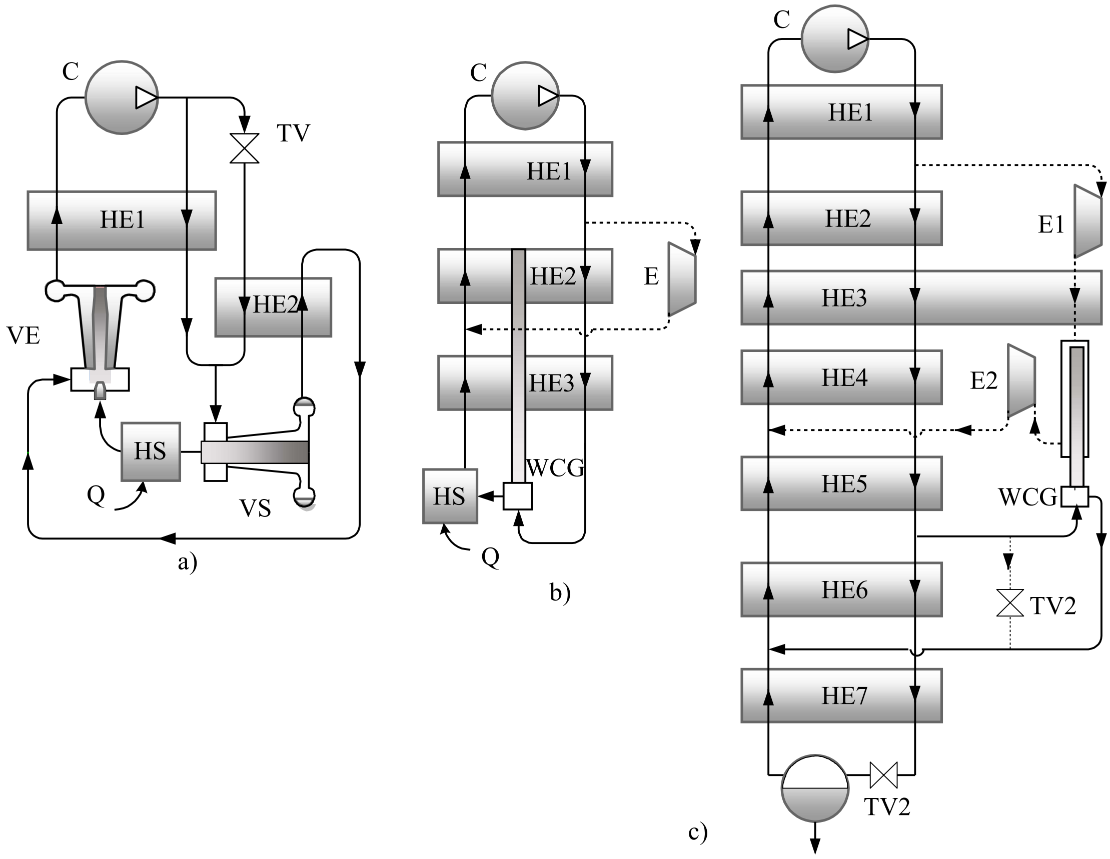


Figure 5: Machineless devices as part of cryogenic helium units
a) the VS vortex separator and the VE vortex ejector in the installation on the helium-freon mixture;
b) WCG low-temperature wave generator of acoustic energy in the refrigerator circuit, with the expander E; c) inclusion of WCG in the scheme of KGU-250/4,5 liquefier;
C – compressors; TV – throttles; HE – heat exchangers; HS – heat source.

4. IMPACT OF THE SCALE FACTOR ON THE GAS-DYNAMIC COOLERS EFFICIENCY

Using machineless devices in cryogenics and in the technologies of rare gases obtaining, the unfavorable impact of the scale factor is observed. This is a consequence of relatively low costs of processed products, low temperatures, and high pressures. Each of these parameters under these conditions leads to a reduction in the nozzle input cross-section F_C. This parameter is the basic design factor of gas-dynamic devices, since the main dimensions of the flow part are related to this parameter (Figure 1).

Supercritical flow regimes are typical for the machineless cryogenerators used in the plant for obtaining rare gases. They are dictated by significant disposable pressure differences in processing cycles. When the ideal gas is discharged, the critical pressure ratio ε_C is determined from the adiabatic exponent k, at discharging of the ideal gas

$$\varepsilon_C = \frac{P_1}{P_2} = \left(\frac{k+1}{2}\right)^{\frac{k}{k-1}}, \tag{3}$$

where P_1 and P_2 are the pressures of the compressed and cold flows, respectively.

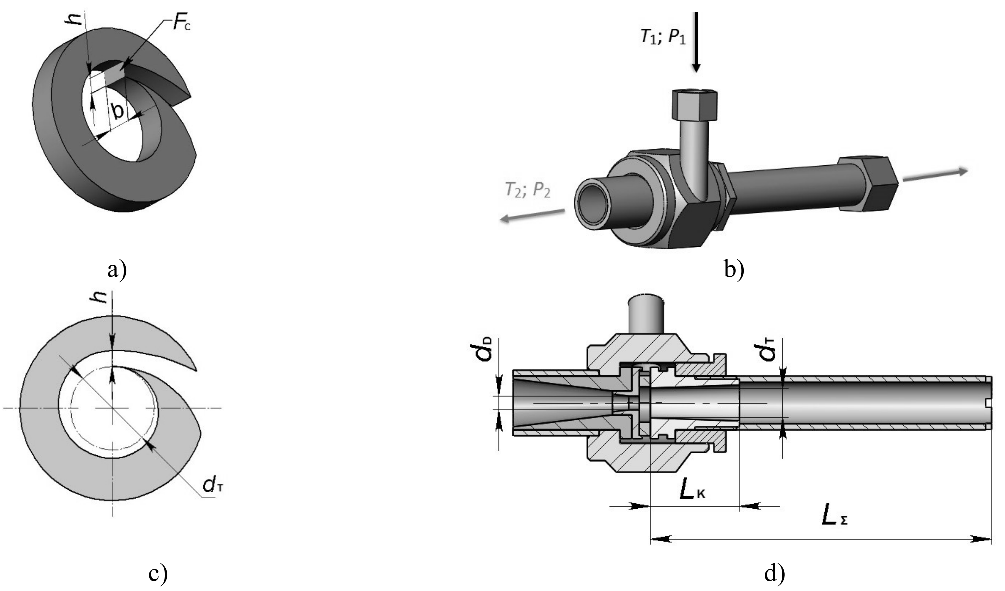


Figure 6: The main dimensions of the flowing part of the vortex tube

The value of k for inert gases (He, Ne, Ar, Kr, and Xe) is in a narrow interval of 1.67±0.01. In accordance with (3) $\pi_C =$ 2.05…2.06. The pressure ratios that we have in the technologies for obtaining rare gases are several times higher than the critical level. For this case, $\varepsilon > \varepsilon_{KP}$ the cold flow pressure P_2 does not influence the gas flow conditions and there is a well-defined relationship between the nozzle cross-section F and the mass flow rate G[kg/s], (Merkulov, 1969, Bondarenko *et al.,*2008)

$$F_C = \frac{G \cdot \sqrt{R^* \cdot T_1}}{P_1 \cdot \sqrt{\frac{2 \cdot k}{k+1}\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}}}, \text{ mm}^2 \tag{4}$$

where T_1 [K] and P_1 [Pa] – temperature and pressure of the compressed gas before the nozzle; $R^* = \frac{R_0}{M}$ – the gas constant of the working substance [J/(kg K)]; R_0 = 8 314 J/(kmol K) – absolute gas constant; M – molecular weight.

The presence of the mass flow rate G [kg/s] in the formula (4) complicates its use in engineering calculations. More convenient and simple is the version of the ratio which is based on the volumetric flow rate V_0 expressed in normal m^3/h. One can see from the Mendeleev-Clapeyron equation that

$$G = V_0 \cdot \rho_0 = V_0 \cdot \frac{P_0}{R^* \cdot T_0}, \tag{5}$$

where P_0 = 0.1013 MPa and T_0 = 293 K – the parameters characterizing the accepted normal conditions.

Taking (5) into account, we will transform (4)

$$F_C = \frac{V_0 \cdot P_0 \cdot \sqrt{T_1}}{3600 \cdot P_1 \cdot T_0 \cdot \sqrt{R^* \cdot \frac{2 \cdot k}{k+1}\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}}}, \text{ mm}^2. \tag{6}$$

When a cold flow pressure close to the ambient level $P_1 \to P_0$, $P_1/P_0 \approx P_1/P_2 = \varepsilon$. Then

$$F_C = \frac{0,948 \cdot V_0 \cdot \sqrt{T_1}}{\varepsilon \cdot \sqrt{R^* \cdot \frac{2 \cdot k}{k+1}\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}}}, \text{ mm}^2. \quad (7)$$

Figure 7 shows the dependence of the basic dimensions of vortex devices in accordance with formula (7) for $V_0 = 40$ norm.m^3/h and $T_1 = 78$ K. As follows from the diagrams, the diameters of the vortex tubes for these operating conditions are in the range $d_T = 2…6$ mm.

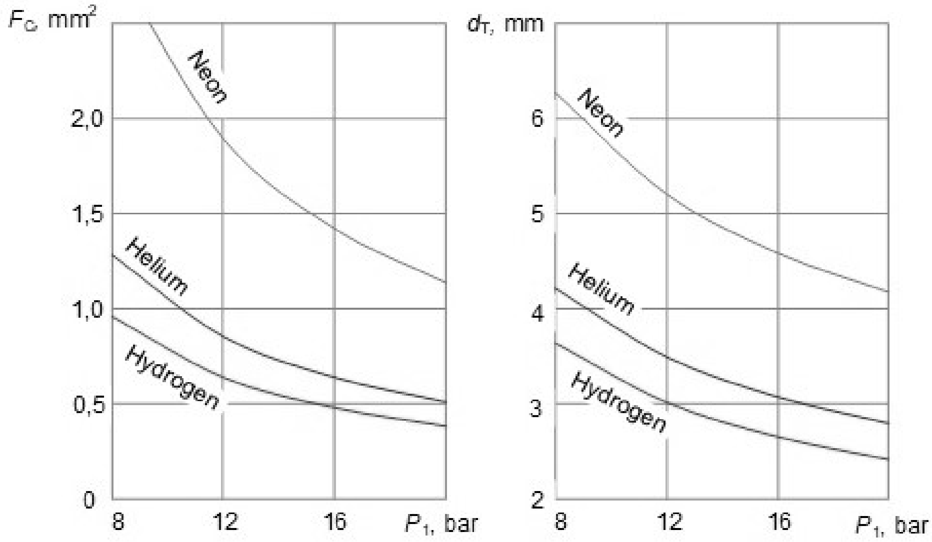


Figure 7: The area of the critical section of the nozzle inlet (F_C) and the diameter of the vortex tube (d_T) depending on the initial pressure (P_1). The flow rate through the nozzle is equal to $V_0 = 40$ norm.m^3/h. The inlet temperature: $T_1 = 78$ K

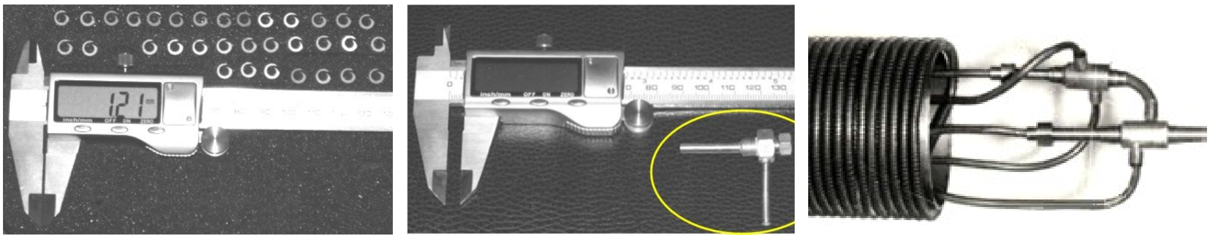
Transition to cryogenic fields of application is almost always accompanied by miniaturization of devices (Figure 7). It is known that reducing the dimensions of gas-dynamic coolers leads to decreasing in their efficiency. The cold flow temperature T_{20} is typical for a vortex tube with a larger diameter d_{T0} , and the cold flow temperature will be higher $T_2 > T_{20}$ for a smaller vortex tube $d_T < d_{T0}$. This regularity is expressed by the relation

$$T_2 - T_{20} = T_1 \cdot m \cdot (d_{T0} - d_T) \cdot \left(1 - \varepsilon^{\frac{1-k}{k}}\right) \quad (8)$$

where ε – pressure ratio, k – adiabatic exponent; T_1 = initial gas temperature; m — empirical coefficient.
In the range $d_T = 40…10$ mm scale influence factor is $m = 0.005$. In the transition to small vortex chambers $d_T = \rightarrow 4$ mm, the value decreases to $m = 0.008$ (Kuzmin, *et al.,* 1984). Small-scale vortex tubes. The diameter range $d_T < 3$ mm has not been studied, even for "high-temperature" vortex devices. The creation and examination of such devices for cryogenic temperatures have a number of design and operational obstacles. The requirements for the accuracy of machine tools are increasing. Especially when making a nozzle scroll (Figure 6-a, c). It is required to ensure a minimum surface roughness of the flow section to reduce boundary effects. Consider the influence of heat transfer. It is necessary to reduce the cross-section of the elements contacting with the cooled gas and use materials with less thermal conductivity.
Another feature is typical for most vortex cryogenic devices. Regardless of the temperature regime, efficient operation of gas-dynamic coolers is provided in the range of pressure ratios $\varepsilon < 2 \cdot \varepsilon_C$. The range of pressure typical for the cryogenic field of application substantially exceeds these conditions. Therefore, low-temperature coolers assume a stepwise switching of vortex tubes. In this case, the preferred ratio of pressures in individual stages is expressed by the known relation

$$\varepsilon_i = \sqrt[T]{\varepsilon_\Sigma} \qquad (9)$$

where ε_Σ – the available ratio of the pressures of the stepwise cooler; T is the number of steps; ε_i is the ratio of pressures in a separate stage.
In accordance with (9), in the case of a successive expansion of the flow in the cascade of vortex tubes, the second step, and subsequent steps have larger dimensions than the first stage. Accordingly, their effectiveness also increases because of the decrease in the negative impact of the scale factor.
The structures of small-scale vortex tubes and their individual elements are shown in Figure 8.



a) b) c)

Figure 8: a) a series of spiral nozzle inlets F_C = 0.34 mm^2; b) vortex tube d_T = 2,2 mm; c) cascade of vortex tubes d_{T1} = 3.5 mm and d_{T2} = 5.0 mm.

5. CONCLUSION

Cryogenics is one of the promising fields of application of vortex and resonance devices. Gas-dynamic coolers are inferior in efficiency to expanders. However, in overpressure conditions, this unsatisfactory feature is mitigated by a number of design and operational advantages of such devices. It can be assumed that when using the plants with a flow rate of the processed product of more than 40 norm.m^3/h the efficiency of gas-dynamic coolers can be acceptable for industrial applications.

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