Use of compressed gas heat utilizers in gas-compressor units

O N Medvedeva, S I Astashev
Dept. of Heat and Gas Supply, Ventilation, Water Supply and Applied Fluid Dynamics, Yuri Gagarin State Technical University of Saratov, 77, Politeknicheskaya street, Saratov 410054, Russia
E-mail: medvedeva-on@mail.ru

Abstract. The study considers the possibility of using the Ranque-Hilsch vortex tube in the construction of a heat exchanger for heating fuel gas to ensure normal operation of the gas-pumping units used to pump gas into the underground gas storages. According to the results of the gas-hydrodynamic and thermal methods for calculating structural elements, a final equation was obtained describing combined operation of the vortex tube and heat exchanger. The calculation method allows taking into account the impact of the main temperature and design parameters. The results of technical and economic calculations and thermal modeling based on the SOLIDWORKS program make it possible to judge the effectiveness of the proposed design.

1. Introduction
To increase the pressure and gas shift, coming from the inlet collector of the compressor stations of the main gas pipeline, gas distribution stations (GDS), underground gas storage facilities (UGS) and the other special process installations, gas-transfer units (GTU) are used. Gas pumping units (for example, 10UGS-01 ‘URAL’), by means of which gas is pumped into the storage facility, consume about 80÷85% of the total gas costs for their own process needs, as the normal process of their operation requires gas heating going to the fuel system of the unit. There are used various types of heaters with a special design to maintain required gas temperature. According to the standards of «Gazprom», gas heaters with an intermediate coolant should be used at a designing stage of new construction, reconstruction and overhaul facilities. Along with all positive issues of existing heaters, they also have serious drawbacks: significant fuel gas consumption for the heater; harmful emissions of nitrous oxide and carbon monoxide into the atmosphere; periodic replacement of the coolant; operational difficulty; depreciation of equipment.

2. Alternative method for heating fuel gas
As an alternative method of heating fuel gas, in our opinion, the heat exchanger using the heat of the compressed gas at the outlet of the supercharger can be considered. Today, design of the GTU does not provide heat utilizers, this heat is removed with the help of a gas air cooler (AC) to the atmosphere. A feature of the proposed design is the use of a vortex tube, designed on the basis of the Ranque-Hilsch tube, for temperature compensation differences by controlling the gas flow (Figure 1).
Most of the streams in engineering are vortex, small particles of the gas or liquid move both translationally and rotate around the instantaneous axis [1÷5]. In technical devices, swirling flows are formed and maintained due to the constructive implementation of the component parts of chambers, channels, input and output elements, etc. The effect of temperature separation of high-speed swirling gas flow into cold flow (+5 ÷ –20 °C and below) in the central part and the hot stream (+40 ÷ +80 °C and above) on the periphery was devised by the French engineer Joseph Ranque, who received a patent for the vortex tube in 1932. Later, German engineer Hilsch experimentally and theoretically substantiated the influence of the inlet pressure and the geometrical parameters of the vortex tube on its performance, presenting some explanation of the process of energy separation, a wide study of which began after the World War II.

In our country, the greatest contribution to the development of studies of the vortex effect was made by Martynovskiy V.S., Merkulov A.P. [6÷9] and the other domestic scientists [10÷14]. Today, there are various designs of the vortex tube: cylindrical dividing, direct-flow, conical, cooled, double-circuit, with a curved geometric shape, and others. The principle of operation of a cylindrical vortex tube is as follows. Compressed gas under the pressure and at ambient temperature is tangentially fed through nozzles into the vortex chamber, expanding to form a rotating vortex that moves along the periphery of the pipe to the hot end. Part of the gas flow is output through the annular slot in the outlet end of the pipe at a temperature exceeding the input values. The rest of the gas flow passes through the central part of the pipe countercurrently to the peripheral one and outputs along the cold end, passing through the diaphragm with a temperature below the ambient temperature. Moreover, the pressure values of the cold and hot gas flows at the outlet from the pipe are lower than the inlet pressure. In direct-flow pipes, cold and hot flows are abstracted from one side of the energy separation chamber. The design of the conical pipe allows to reduce the overall dimensions and increase the cooling effect. Operation principle of the cooling tubes is based on the additional drain by using a cooling jacket or additional external fins. Merkulov A.P. and Piralishvili S.A. proposed to use an additional gas stream, supplied to the axial zone of the flow separation chamber on the side of the choke, – double-flow vortex tubes. The advantages of the vortex tubes include: design simplicity, the absence of moving parts (as a result, high reliability of the design), high speed of the installation to the operating mode and simultaneous implementation of several processes, environmental friendliness.

The heat exchanger on the basis of the Ranque-Hilsch vortex tube is a construction of a vortex tube, a heat exchanger and a receiver (Figure 2). The vortex tube is connected to the unheated fuel gas tube at the inlet and the tubes of the heat exchanger and the receiver at the outlet. At the «hot» outlet, the vortex tube is equipped with a needle valve, which ensures the variability of the vortex tube operation that allows to use construction in a different range of initial parameters of the heated gas and the gas at the outlet from the supercharger. The flow rate is controlled in such a way that the gas temperature in the receiver is 25÷30°C. The heat exchanger consists of a bundle of pipes with fins, is connected to the «cold» outlet of the vortex tube and has an initially calculated heat transfer coefficient. The heat exchanger is located above the compressed gas air cooler and uses the hot air blown by the fans as an intermediate heat carrier.
Figure 2. Gas flow control scheme: \( G_c \) – flow rate of the cooled stream, \( G_h \) – flow rate of the heated stream, \( \Delta T_h \) – heating effect, \( \Delta T_c \) – cooling effect.

After the receiver, the fuel gas is supplied to the reduction unit to prevent the reducing reinforcement from freezing due to the Joule-Thomson effect. Today, due to the shift complexity of the vortex flow, the presence of significant turbulent pulsations, experimental methods and numerical simulation are the main methods for studying the processes occurring in the vortex tube [6, 15÷18].

3. Calculation method of the vortex pipe

The main differences between the actual processes occurring in the vortex tube are:

- expansion process of the cold gas flow occurs through an irreversible polytrope;
- the cold flow outgoing of the vortex tube through the diaphragm has kinetic energy, resulting from the rotational and axial flow of the flow in the diaphragm, which later turns into heat. As a result, the enthalpy and temperature of the cold gas stream increases. There are no sufficiently accurate methods for analytic description and calculation of the actual process. In a simplified form, operation of a real vortex tube is shown in \( T - S \) diagram in Figure. 3.

During calculation process, the following provisions were taken into account, based on experimental data:

- since at a constant cross section of the diaphragm \( f_d \) the velocity of the gas flow passing through the nozzles is a function of the flow rate, the relative decrease in the cold flow temperature \( \Delta T_{cf} / T_c \) will depend on the fraction of the cold flow \( \mu \);
- in real conditions of operation, compared with ideal conditions, a smaller amount of energy is transferred to the hot gas flow, because the same values \( T_3 = T_d = T_c + \frac{w_2^2 + w_0^2}{2C_p} \), of enthalpy \( i_c \), pressure
ratio of the compressed flow before the pipe and the cold flow after the diaphragm $F_{cf}$; and the fraction $\mu$ of the enthalpy of cold flow after the pipe is much larger than in the ideal pipe;

– the radial distribution of tangential velocities in the nozzle section approximately obeys the law of a quasi-solid vortex: $w_T = \omega \cdot r$, where $w_T$ is the tangential velocity of any point of the stream; $\omega$ – angular velocity, almost constant over the cross section; $r$ is the radius of rotation of the stream.

– braking temperature of the cold gas flow in the diaphragm:

$$T_3 = T_d = T_2 + \frac{w_0^2 + w_T^2}{2c_p},$$

where $w_T$ is the average tangential flow rate in the diaphragm, m·s\(^{-1}\); $w_0$ is the average axial velocity in a diaphragm, m·s\(^{-1}\); $c_p$ is the mass heat capacity of the gas, J·(kg·K)\(^{-1}\).

– for the case when the pressures in the diaphragm and in the section behind the diaphragm are equal at the cold end of the vortex tube, i.e. $p_d = p_{cf}$, the cold flow temperature is equal to the gas deceleration temperature in the diaphragm $T_{cf} = T_d$.

– if the pressure in the diaphragm $p_d$ exceeds the pressure at the cold end of the pipe behind the diaphragm $p_{cf}$, i.e. when $p_d > p_{cf}$, the temperature of the cold flow will differ from the braking temperature of the gas in the diaphragm by the value of the differential drag effect.

$$T_4 = T_{cf} = T_d + \left(\frac{\partial T}{\partial p}\right)_i \cdot \Delta p_{cf},$$

where $\left(\frac{\partial T}{\partial p}\right)_i$ – differential throttling effect; $\Delta p_{cf} = p_d - p_{cf}$ is the pressure difference in the diaphragm and at the cold end of the pipe after the diaphragm.

The state of the gas flow at the hot end of the vortex tube before the annular gap (gas deceleration temperature $T_k$ and pressure $p_k$) is shown in figure 2 point 5. The temperature of the hot gas flow after the annular slot will differ from the braking temperature to this slot by the value of the throttling effect:

$$T_h = T_h = T_k + \left(\frac{\partial T}{\partial p}\right)_i \cdot \Delta p_h,$$

where $\Delta p_h = p_h - p_h$ is the pressure drop in the annular gap at the hot end of the vortex tube.

The temperature of the hot gas flow after the annular gap $T_h$ is determined from the condition of the energy balance. With low gas pressure after the vortex tube, i.e. for the case $p_{cf} \approx p_a \approx 0.1$ MPa due to insignificance, the throttle effect on the cold and hot ends of the vortex tube can be disregarded, then: $T_{cf} = T_d$; $T_h = T_k$. Using expressions to determine the gas flow rate through the nozzle, we can determine the critical mass flow rate of the cold flow $\mu^*$, which will correspond to the transition of the diaphragm from the subcritical to the critical mode [19]. Consumption of the compressed gas through the nozzles of the vortex tube at the supercritical degree of expansion of the gas flow in them:

$$G_c = \frac{k \cdot F \cdot p_c \cdot f_s}{a^*},$$

In the transient mode, the axial flow velocity in the diaphragm $w_0 = a_{cf}$ and in the diaphragm pressure $p_a$ is set equal to the critical pressure of the cold flow. Then the flow of cold flow through the diaphragm during the transitional mode:

$$G_{cf} = \mu^* \cdot G_c = \frac{k \cdot p_{cf} \cdot f_s}{a_{cf}^*},$$
where \( k \) is the adiabatic index.

From the joint solution of equations (4) and (5) follows:

\[
\mu^* = \frac{a_{c*}}{a_{f*}} \left( \frac{f_d}{f_c} \right) \left( \frac{F_{cc}}{F^*} \right),
\]

where \( a_{c*} \) and \( a_{f*} \) are critical velocities of compressed and cold gas flows; \( f_d, f_c \) – cross sections of the diaphragm and nozzles of the vortex tube; \( F_{cc} = \frac{p_{c*}}{p_{c}}, \ p_{c}, p_{c*} \) – pressures of compressed flow before the pipe and cold flow after the diaphragm, respectively; \( F^* \) is the critical pressure ratio.

The value of the mass flow rate fraction \( \mu \) when operating a vortex tube can be changed by adjusting the width of the annular gap, for example, using a valve [20]. We can compensate for the resulting instability of the temperatures supplied to the heat exchanger using the inverse relationship of the ratio of the gas flow rate per cold outlet of the vortex tube to the flow rate attributable to the hot exit:

\[
\frac{G_{cf}}{G_{h}} = \frac{\Delta T_h}{\Delta T_{cf}},
\]

where \( \frac{\Delta T_h}{\Delta T_{cf}} \) is the ratio of the cooling effect to the heating effect.

To determine the main operating parameters of the installation [21, 22], a vortex tube was calculated using the characteristics \( \Delta T_{cf} = f(\mu) \) and \( \Delta T_h = f(\mu) \) with the following initial parameters: pipe diameter \( d_p = 280 \text{mm} \); the diameter of the cold flow diaphragm \( d_d = 120 \text{mm} \); nozzle diameter \( d_c = 100 \text{mm} \). The vortex tube operates on compressed natural gas (methane) of the following parameters: pressure \( p_c = 5 \text{MPa} \); gas temperature \( T_c = 288 \text{K} \). The pressure of the cold gas flow \( p_{cf} = 3 \text{MPa} \).

The adiabatic index is \( k = 1.33 \). The results of the calculation are presented in table 1.

### Table 1. The results of the calculation.

| Parameter                        | Formula                  | Parameter value |
|----------------------------------|--------------------------|-----------------|
| Nozzle flow area (m²)            | \( f_c = 0.25 \pi d_c^2 \) | 0.00785         |
| Aperture area (m²)               | \( f_d = 0.25 \pi d_d^2 \) | 0.0113          |
| Cross sectional area of the vortex tube (m²) | \( f_b = 0.25 \pi d_d^2 \) | 0.0615          |
| Geometric parameters of the vortex tube | \( f_d \cdot f_c^{-1} \)                   | 1.44            |
|                                  | \( f_c \cdot f_p^{-1} \)                   | 0.184           |
| Relative cold flow pressure      | \( F_{cc} = p_{cf} \cdot p_c^{-1} \)    | 0.6             |
| Preliminary value of the ratio of critical velocities | \( a_{c*} = \left( \frac{T_c}{T_{cf}} \right)^{1/2} \) | 0.05            |
| of compressed and cold flows     |                          |                 |
| Preliminary value of the critical fraction of cold flow | \( \mu^* \) | 0.92            |
|                                 |                          | \( F_{c*} = 0.98 \) at \( \lambda = 0.2 \) |

We will calculate the mode with subcritical outflow through the diaphragm. For all \( \mu < \mu^* \), the relative pressure of the flow in front of the diaphragm is \( F_{cd} = \Pi_{c*} = 0.6 \); \( \lambda_{cd} = \lambda_{c*} = 0.92 \); \( q_{cd} = 0.98 \). Given different values of \( \mu \), we define:

\[
\frac{\Delta T_{cf}}{T_c} = \frac{k - 1}{k + 1} \left( \frac{\phi^2 - 0.445 \frac{f_d}{f_r} \frac{\mu^2}{q_{cd}^2} \frac{f_{c*}^2}{f_d^2}}{\varphi^2} \right),
\]

where \( \varphi = 0.75 \) is the velocity coefficient.
Similarly, we determine the parameters for $\mu=0.1; 0.2; 0.3; 0.4; 0.5; 0.6; 0.7; 0.8; 0.9; 1$. Then we will calculate the modes at critical and supercritical outflow of flow through the diaphragm. Thus, temperature change of the cold stream at the specified critical mode will be $\Delta T_{cf}=3.168$ K.

Therefore, the temperature of the cold flow at the outlet from the diaphragm at the given initial parameters: $T_{cf}=T_c-\Delta T_{cf}=284.832$ K. We determine the temperature effect of the vortex tube ($\Delta T_{cf}$) of the vortex tube at different values of the input pressure $p_c$ of the fuel gas before heating. To carry out the calculation, we set the initial parameters: the proportion of cold flow is $\mu=0.6$; temperature of compressed gas $T_c=288$ K; pressure of compressed gas $p_c=3.5; 4; 4.5; 5; 5.5; 6$ MPa; cold flow pressure $p_{cf}=3$ MPa. The relative pressure of the cold stream at $p_c=3.5$ MPa will be: $F_{ccf}=0.875$. The value of the temperature effect by the formula (10): $\frac{\Delta T_{cf}}{T_c}=0.115$. The remaining characteristics of the vortex tube at a pressure of $p_c=3.5$ MPa, including at other pressures, are given in table 2.

| Parameter | $\mu$ |
|-----------|-------|
| $F_{ccf}$ | 0.6   |
| $\lambda_{ccf}$ | 0.92  |
| $q_{ccf}$ | 0.98  |
| $\Delta T_{cf}/T_c$ | 0.057 |
| $\Delta T_{cf}$ | 16.64 |
| $\Delta T_{cf}$ | 0     |
| $T_{cf}$ | 271.35 |
| $T_{h}$ | 288   |
| $T_{cf}$ | -1.649|
| $T_{h}$ | 15    |

The exergonic efficiency of the vortex tube will be equal to: $\eta=86\%$. In cases where the hot stream is not used: $\eta=51\%$. The size of the heat exchange surface $F$, m$^2$, is determined from the basic equation of heat transfer: $F=2859.6$ m$^2$. According to the obtained values, we select a heat exchanger and nozzles. The content of the constructive calculation of the heat exchanger depends on the features of the chosen construction, i.e., by the choice of heat exchange surface: tubular, lamellar, spiral, etc. Based on the heat exchange area and taking into account the initial parameters (48 pipes with a length of 20m, $d=25$ are taken), we calculate the finning area: $F_f=F-F_p=1392$ m$^2$.

After the vortex tube and heat exchanger, both streams (heated and cooled, which was heated in the heat exchanger) flow into the fuel gas receiver, where heat and mass transfer exchange. As an equation describing heat and mass transfer in the receiver, we take the ideal gas equation of state.

Gas temperature at the outlet from the heat exchanger:

$$T_{he}=T_c - \frac{k-1}{k+1} \lambda_{gd} \left( \frac{\varphi^2 - 0.445 f_d}{f_p} - \frac{\mu^2 f_d^2}{q_{cd} f_p^2} \right) T_c + \frac{G_{c1} \left( t_1 - t_1'' \right) \left( t_2'' - t_2 \right)}{G_{c2} \left( t_2'' - t_2 \right)},$$

where $t_1', t_1''$ is the temperature of the compressed gas before AC and after AC respectively; $t_2', t_2''$, $t_2'''$ is the air temperature before AC, between AC and heat exchanger and after heat exchanger respectively; $P_c, G_c, T_c$ – pressure, weight flow and gas temperature at the entrance to the vortex tube, respectively; $P_d, G_d, T_d$ – pressure, weight flow and gas temperature at the exit from the vortex tube, respectively; $k$ is the adiabatic index; $\lambda_{gd}$ – thermal conductivity of gas; $\varphi$ – coefficient of speed; $f_d, f_p, f_c$ is the cross-sectional area of the diaphragm, tube and nozzle, respectively; $\mu$ is the proportion of cold flow (opening degree of the throttle valve); $q_{cd}$ – relative axial mass flow rate of gas in the diaphragm; $c_1$ is the heat capacity of the gas at $t_1'^{av}$, $c_2$ is the heat capacity of the gas at $t_3'^{av}$. 


The criterion for the efficiency of creating and introducing new automation tools is the coefficient of the overall economic efficiency of the capital investments $Ef = 1.04$ and payback period $T = 0.95$ years.

4. Conclusion

The payback period of the capital investments in the proposed design of the heat exchange apparatus will be up to a year, which indicates its economic efficiency. Thus, the solution of the presented scientific and technical problem, aimed at improving the efficiency of thermal and hydrodynamic processes in jet-vortex recuperators for heat engineering units with active hydrogasdynamic regimes, leads to a significant useful increase in the efficiency of convective heat exchange on the air side of metal tubular recuperators and to reduce their size. The conducted researches allow us to conclude that the installation is fully justified and can be used in the gas supply system.

References

[1] Landau L D and Lifshits E M 1986 *Hydrodynamics* (Moscow: Science) p 736
[2] Ranque G J 1933 Expériences sur la détente giratoire avec productions simultanées d'un échappement d'air chaud et d'un échappement d'air froid *J. de Physique et de Radium* 7(4) 112–115
[3] Hilsch R and Naturforschung Z F 1946 Die Expansion von Gasen im Zentrifugalfeld als Kälteprozeß *I* 1 208–213
[4] The Use of the Expansion of Gases in a Centrifugal Field as Cooling Process 1947 *The Review of Scientific Instruments* 18(2) 108–113
[5] Westley R 1954 *A bibliography and survey of the vortex tube* (Cranfield: College of Aeronautics) p 38
[6] Suslov A D, Ivanov S V and Murashkin A V 1985 *Vortex apparatuses* (Moscow: Machine building) p 256
[7] Berman S S 1962 *Calculation of heat exchangers* (Moscow-Leningrad: Gosenergoizdat) 240 p
[8] Betlinsky V Yu, Zhidkov M A and Ovchinnikov V P 2008 Two-flow adjustable vortex tubes in industrial installations for gas cleaning and drying *Gas industry* 1 72–75
[9] Bath O P 2012 *Basic design and heat calculation of heat exchangers* (SPb: ITMO) p 42
[10] Martynovskiy V S 1955 *Heat pumps* (Moscow: Gos-energoizdat) p 192
[11] Martynovskiy V S 1979 *Cycles, circuits and characteristics of thermotransformers* (Moscow: Energy) p 288
[12] Merkulov A P 1969 *The vortex effect and its application in technology* (Moscow: Mashinostroenie) p 182
[13] Merkulov A P 1997 *Vortex effect and its application in engineering* (Samara: Optima) p 344
[14] Ryabov A P, Gusev A P and Zhidkov M A 2007 Three-flow vortex tubes in the oil and gas industry (analytical review) *Oil and Gas Technologies* 1 2–7
[15] Gupta A, Lilly D, Seared N 1987 *Swirling Streams* (Moscow: World) p 588
[16] Orlov A A 2009 *Gas-phase separation methods* (Tomsk: TPU) p 286
[17] Goldshtik M A and Shtern V N 1989 *Viscous currents with paradoxical properties* (Novosibirsk: Science) p 336
[18] Gutsol A F 1997 *Ranque effect* *Successes physical individuals* 167(6) 665–687
[19] Eckert E R G and Drake R M 1987 *Analysis of Heat and Mass Transfer* (New York: McGraw-Hill) p 806
[20] Balmer R T 1988 *Pressure-Driven Ranque-Hilsch Temperature Separation in Liquids* *Journal of Fluids Engineering* 110 161–164
[21] Balmer R T 2011 *Modern Engineering Thermodynamics* (Academic Press) p 801
[22] Alekseenko S V, Kuibin P A and Okulov V L 2007 *Theory of Concentrated Vortices: an Introduction* (Berlin, New York: Springer) p 488