Hartmann–Sprenger Energy Separation Effect for the Quasi-Isothermal Pressure Reduction of Natural Gas: Feasibility Analysis and Numerical Simulation

Artem Belousov 1,*, Vladimir Lushpeev 2, Anton Sokolov 3, Radel Sultanbekov 4,5, Yan Tyan 5, Egor Ovchinnikov 5,6, Aleksei Shvets 5, Vitaliy Bushuev 6 and Shamil Islamov 7,*

1 Predictive Analytics Department, Industrial Digital Platform, 199178 Saint Petersburg, Russia
2 Institute of Earth Sciences, Saint Petersburg State University, 199034 Saint Petersburg, Russia; v.lushpeev@spbu.ru
3 Scientific and Education Center Gazpromneft-Politekh, Peter the Great Saint Petersburg Polytechnic University, 195251 Saint Petersburg, Russia; sokolov_an@spbstu.ru
4 Resource Management Centre, Gazpromneft Marine Bunker, 199106 Saint Petersburg, Russia; radelsultanbekov@mail.ru
5 Department of Oil and Gas Transport and Storage, Saint Petersburg Mining University, 199106 Saint Petersburg, Russia; yan_ukg@mail.ru (Y.T.); egor.owchinnikov@yandex.ru (E.O.); schvetzaleksey@gmail.com (A.S.)
6 School of Economics and Management, National Research University Higher School of Economics, 194100 Saint Petersburg, Russia; bushuevvtly@gmail.com
7 Department of Petroleum Engineering, Saint Petersburg Mining University, 199106 Saint Petersburg, Russia
* Correspondence: belousovartemevg@gmail.com (A.B.); sh.islamoff@gmail.com (S.I.)

Abstract: The present paper provides a brief overview of the existing methods for energy separation and an analysis of the possibility of the practical application of the Hartmann–Sprenger effect to provide quasi-isothermal pressure reduction of natural gas at the facilities within a gas transmission system. The recommendations of external authors are analyzed. A variant of a quasi-isothermal pressure regulator is proposed, which assumes the mixing of flows after energy separation. Using a numerical simulation of gas dynamics, it is demonstrated that the position of the resonators can be determined on the basis of calculations of the structure of the under-expanded jet without taking into account the resonator and, accordingly, without the need for time-consuming calculations of the dynamics of the processes. Based on the results of simulating the gas dynamics of two nozzle–resonator pairs installed in a single flow housing, it is shown that, in order to optimize the regulator length, the width of the passage between the two nearest resonators should be greater than or equal to the sum of diameters of the critical sections of the nozzles. Numerical vibroacoustic analysis demonstrated that the most dangerous part of the resonator is the frequency of its natural oscillations.

Keywords: Hartmann–Sprenger effect; quasi-isothermal reduction; unheated reduction; gas distribution system; natural gas distribution networks; pressure regulator

1. Introduction

1.1. Topicality

The system for the transportation, distribution and consumption of natural gas assumes a step-by-step pressure decrease as it approaches the final consumer. It is impractical to transport natural gas at low pressure due to the increase in hydraulic losses and decrease in the mass of the pumped product. The initial decrease occurs during the transition from the main transport system to the gas distribution system. Further, depending on the type of consumer and their distance from the supply point, additional stepwise pressure reductions can be performed. For example, in the Russian Federation (RF) the following classification [1] is adopted according to the level of working pressure in the gas pipelines...
of the gas distribution system: high 1—from 0.6 to 1.2 MPa; high 2—from 0.3 to 0.6 MPa; medium—from 0.005 to 0.3 MPa; and low—up to 0.005 MPa inclusive. Thus, in the general case, there can be up to four stages of pressure reduction, which can be implemented both within the framework of one gas-reducing object, and within a framework with several such objects, located at a considerable distance from one another. There are more than 350 thousand natural gas pressure reduction points in the Russian Federation. At the same time, there are about 3.8 thousand large gas distribution stations providing the transition from main pipeline transport to gas distribution.

The pressure reduction at the pressure reduction points is performed using valve-type throttle valves, which act as local resistances to the flow movement, dissipating the pressure energy. In addition, according to the Joule–Thomson effect, following a decrease in pressure, the temperature also drops, which can lead to freezing of pipeline fittings and the precipitation of gas hydrates. To maintain the natural gas temperature above the dew point and prevent such negative phenomena, heating is used which maintains the temperature at a level not lower than −10 °C or 0 °C in heaving soils [2]. Heating is performed using an external energy source, most often by burning part of the transported natural gas. About 0.3% of the amount of transported gas is burned and 99.7% of gas that entered at the pressure reduction point reaches consumers. Either heating of the gas flow itself or heating of the area where the pressure regulator is located can be performed. The second option is used for small gas-reducing points.

Thus, there is a double expenditure of energy: first, the energy earlier supplied to the gas flow by the main gas-pumping units is irretrievably lost, and then an additional energy supply is used to restore the temperature.

1.2. Existing Developments

Naturally, the most effective way to solve this problem is to utilize the energy of the gas flow itself by transforming it into heat.

To implement this, it is possible to use expander generators that convert potential pressure energy into mechanical energy, followed by conversion into electricity and heat [3–5]. However, the expander generator system is rather complex and, most importantly, due to the removal of additional mechanical work from the gas flow in the expander, the gas temperature drops more significantly, which, in turn, requires the organization of more substantial heating and the utilization of cold temperatures; for example, to obtain liquefied natural gas [6].

For the purpose of converting pressure energy into thermal energy, it is interesting to use energy separation methods. There is no single understanding of the mechanism of their occurrence and classification [7,8]. It is also worth noting the particularly controversial nature of the term “machineless” energy separation. According to Eckert’s classification, there are two types of machineless separation of the energy in the flow; that is, the spontaneous formation of zones with total temperatures above and below the initial total temperature of a moving flow of matter. The first is associated with vortex flows and the resulting pressure pulsations; the second, with an imbalance between the heat fluxes released as a result of the viscosity of gas and the heat removed due to the heat conduction mechanism.

The most famous mechanisms and methods of energy separation are presented below:

Energy separation during gas flow around various obstacles. For example, the division in the Karman vortex street of the initial flow into three streams, one of which has an enthalpy lower than the initial one, and the other two, higher [9].

In a free flowing stream. Temperature stratification due to vortex formation at the exit of an underexpanded jet from a nozzle with a temperature at the center of the flow that is higher than the temperature at the periphery [10,11].

In a compressible gas stream. Temperature stratification due to the difference between the adiabatic temperature of the channel wall and the flow stagnation temperature near the wall, formed by dissipative processes in the boundary layer [7,12]. It is implemented in
Leontiev pipes [13], or in energy separation devices with the suction of a cold flow through a permeable wall [14].

**Vortex separation.** Separation of a high-speed flow tangentially fed into a Ranque–Hilsch tube into a colder one in the axial zone of the formed single vortex, and a hotter one at its periphery [15]. Another example is the division of a high-speed flow coaxially fed into the Emin–Zaritsky pipe into a colder one in the axial zone of the pipe and a hotter one in vortices formed at the periphery due to the interaction between compression and decompression waves [16–18].

**Resonance separation.** Separation of the initial high-speed flow into a heated volume inside a closed cavity and a cold flow outflowing from this cavity after energy is transferred to the heated one, in accordance with the Hartmann–Sprenger resonance effect [19]. In this case, the frequency of pressure pulsations can reach thousands of Hertz.

**Wave separation.** Energy separation based on the Herzberg direct energy exchange [20] with direct contact between high- and low-pressure gases in devices called wave expanders or rotary cryogenerators [21,22].

**Pulsating separation.** Creation of high temperature gradients in a constant volume of gas in the process of its cyclic adiabatic filling and emptying, which are provided by various types of jet and mechanical pulsators, as well as gas distributors. The active gas is cooled as a result of the work of compressing the passive gas inside the cavity and the subsequent expansion when leaving it, whereas the passive gas is heated by compression. Examples of applications are refrigeration units based on Gifford–Longsworth pulse tubes [23], which require heat exchangers and regenerators to function. Most often, the frequency of the pressure pulsations in such systems is relatively low: from a few to several tens of Hertz [24].

Based on the rather high similarity between these methods of energy separation, a certain level of confusion has naturally arisen surrounding their classification and terminology. Some researchers [25,26] argue that the resonance, wave, pulsating and thermoacoustic methods of energy separation should be considered as a single type, grouped according to the wave nature of energy exchange.

By its essence, any energy separation can be applied to produce both cold and heat. However, specific methods of energy separation have features that contribute to their more targeted use for certain needs.

Let us analyze the best-known examples of the application of energy separation, suitable for utilizing pressure energy at reduction points.

The most studied are vortex tubes that implement the Ranque–Hilsch effect, discovered in 1931 by Georges Ranque [15] and popularized in 1946 by Rudolf Hilsch [27,28]. Several methods and devices are known that also use this effect for refrigeration units [29]; the liquefaction of gases [30]; isothermal throttling by mixing “hot” and “cold” streams [31]; and heating the working body of the pressure regulator at the reduction point with a “hot” flow to prevent it from freezing.

An important requirement to ensure that the efficiency of vortex tubes remains high is the requirement for an increased smoothness of their inner surfaces. As such, vortex tubes have a relatively high cooling capacity.

The temperature stratification method patented by Leontiev in 1998 [32] involves the separation of a high-pressure flow into subsonic and supersonic flows with their subsequent heat exchange through the wall, which provides heating to the supersonic part and cooling to the subsonic one. This effect can be used in refrigerating machines; installations for low-temperature separation of natural gas; natural gas reduction units with associated heating of transit gas; and the production of liquefied natural gas (LNG).

The main advantage of this solution is that it preserves the total pressure of the subsonic gas flow at a higher level than other energy separating devices. Such an advantage can be useful for utilizing pressure energy in natural gas main transmission systems or similar ones in which pressure reduction is undesirable. Accordingly, at the reduction points, the advantages of this method can be offset. In addition, a fairly high cooling capacity makes it necessary to supplement the technological scheme of the reducing facility
with installations for its utilization for the production of LNG or gas hydrates, for example, which may not always be economically justified.

The most promising and least studied alternative to the above effects is the Hartmann–Sprenger resonance effect. The first studies in the field of resonant acoustic vibrations were associated with the experiments of the Danish scientist Julius Hartmann, who discovered the aeroacoustic effect at the beginning of the 20th century [33]. Later, in 1954, when using longer resonators \( \frac{l_T}{d} = 34 \), while Hartmann’s tube dimensions were \( \frac{l_T}{d} < 4 \), the thermoacoustic effect was discovered by Herbert Sprenger [19].

The Hartmann–Sprenger energy separation effect is based on the principle of gas-dynamic resonance and thermal energy separation through nonlinear self-oscillating gas pressure fluctuations in a plugged cavity, which arise as a result of a supersonic gas flow around it. Temperature separation occurs due to the cumulative supply of heat to the gas volume that becomes trapped the resonator’s dead end [34], which is formed as a result of the dissipative processes of affecting this volume through periodic shock waves propagating inside the resonator [35,36], as well as due to gas friction against its walls [37]. This ensures the attainment of high temperatures and a low cooling capacity, which is additionally explained by the essence of the flow mixing mechanism [38]. This feature also contributes to the possibility of effectively using such devices for reduction.

It should be noted that high temperatures at the end of the resonator generate a high temperature gradient within the environment, due to which it is possible to provide efficient heat removal and also to implement a refrigeration machine based on this effect [25,34,38]. Interestingly, this effect is often referred to as a negative one, precisely from the point of view of generating high temperatures, for example, in the case of unacceptable heating of the structural elements of a supersonic aircraft. In the modern practice of pipeline transportation of natural gas, there are cases of the risers in the valve units of main gas pipelines being heated when natural gas is transferred through a bypass line [39].

Due to the possibility of reaching high temperatures (up to 1600 K [40]) and the rate of temperature growth (up to 160 K/ms [41]), the resonance effect is widespread in warhead detonation systems [42] and rocket engine fuel ignition systems [43–45]. In addition, it is used to generate electricity based on the utilization of temperature differences created by thermoelectric elements [46]; thermal desorption [47]; and magnetohydrodynamic energy conversion [48].

Despite the obviousness of the possibility of using the effects of energy separation in order to utilize the pressure energy created by natural gas, there are very few real examples of such use. The following factors contribute to the low occurrence:

1. Insufficient knowledge and lack of unified theories about why the above effects arise;
2. A lack of clear methods for designing devices based on these effects;
3. The necessity of such systems functioning in non-stationary conditions within real production processes.

Devices that use the Hartmann–Sprenger effect, in particular, have the same disadvantages. There are various scattered data from the results of experiments and general recommendations for implementing the effect, but there are no structured algorithms and techniques that allow them to be transferred to practical applications.

In particular, in order to use the Hartmann–Sprenger effect as the basis of any gas pressure utilization device, it is necessary to know the answers to the following questions:

1. What temperature and thermal power can be achieved in the resonator and at the outlet of the device under certain configurations?
2. How should we select the parameters and number of nozzle–resonator pairs for a certain flow rate and pressure drop?
3. At what distance from the nozzle should the resonator be located? Including in the process of non-stationary changes in the nozzle’s supply parameters.
4. How close to each other can the nozzle–resonator pairs be installed in order to optimize the dimensions?
5. How much noise and vibration will be caused by such devices and what resources will they need?

2. Essence of the Proposal

Devices based on Hartmann–Sprenger resonant energy separation can be used to utilize the energy from the natural gas’ pressure in the gas distribution system to reduce the drop in gas temperature after reduction.

Of course, the ratio between the heat capacity and cooling capacity of the Hartmann–Sprenger energy separation will be determined by geometric relations similar to those used to describe energy separation in the Ranque–Hilsch vortex tubes [49–51].

The simplest and most attractive option is mixing the flow around the resonator with the one periodically leaving the hot cavity while ensuring heat transfer between the hot end of the resonator and the flow around it (Figure 1). At first glance at this kind of mechanism, one can only say that there will be the smallest possible drop in temperature in comparison with conventional throttling.

![Diagram of the experimental setup by N.A. Shushin: 1—supersonic nozzle; 2—cylindrical resonator; 3—flow mixing housing.](image)

Figure 1. Diagram of the experimental setup by N.A. Shushin: 1—supersonic nozzle; 2—cylindrical resonator; 3—flow mixing housing.

However, publications are known that show significantly more impressive test results, such as an increase in the temperature of dry air at the outlet of the device relative to its inlet temperature by up to 15 °C [52]. In this case, according to theoretical calculations, the maximum possible temperature of natural gas after mixing the hot and cold flows after the resonator with a pressure drop of 50 can reach 1.4 of the flow’s stagnation temperature. However, there are not many results from tests which mix the streams, which does not allow them to be assessed unambiguously.

This kind of scheme (Figure 1) cannot be used to utilize the natural gas flow energy in the gas distribution system due to its unacceptably long length (about 4 m with a flow rate of 50 thousand m³/h) and the lack of the possibility of regulation in the case of irregularities (seasonal, daily and hourly) in the gas usage by consumers. During peak hours, the amount of gas consumption, and hence the flow through the device, can change by 2 times [53]. In addition, it is worth considering periodic changes and pressure surges both before and after the reduction points.

Based on the above, for the purpose of lowering the pressure of natural gas and maintaining it at a given level, it is proposed to use annular (in this case, pin) adjustable nozzles and resonators with an adjustable position relative to the nozzles installed in several pairs to separate the gas flow coming to the device between them. The flow rate passing through the device is changed by increasing or decreasing the nozzle throat area by moving the central body. Operational positioning of the resonator is performed in accordance with the change in the structure of the underexpanded jet at various pressure drops. The separation of the input flow allows a reduction in the dimensions to be achieved, as well as the uniformity of heat transfer between the hot ends of the resonators and the flows around them to be increased, which will subsequently be combined into a single output flow. In addition, the use of annular nozzle profiles contributes to a reduction in the length of the accelerator. A schematic diagram of such a device is shown in Figure 2.
1. Analysis of scientific sources;

was 70.5 mm long, with a 4.2 mm inner diameter and a wall thickness of 1.9 mm.

Below, we will present the most interesting recommendations obtained as a result of an analytical review of the current literary sources, as well as our own results, which allowed us to assess the possibility of realizing quasi-isothermal reduction using the Hartmann–Sprenger effect.

3. Research Methods

The research was carried out according to the following steps:
1. Analysis of scientific sources;
2. Model development;
3. Adequacy check;
4. Numerical Simulation of Gas Dynamics;
5. Numerical Vibroacoustic Analysis;
6. Results.

3.1. Analysis of the Sources

An analysis of the literature and patent materials was carried out, which describes the application of the Hartmann–Sprenger effect for the purpose of obtaining high temperatures or generating heat.

3.2. Numerical Simulation of Gas Dynamics

To carry out numerical simulation of gas dynamics in ANSYS Fluent 2020 R1, 2021 R2 Academic, a computational model was developed that provided simulation of the self-sustaining resonance Hartmann–Sprenger effect in the same geometry as used in the experimental setup (Figure 3). The calculations were carried out only using a resonator that was 70.5 mm long, with a 4.2 mm inner diameter and a wall thickness of 1.9 mm.

Figure 3. Experimental setup: 1—RS-485/USB interface converter; 2—inlet valve; 3—filter separator; 4—threaded sleeve; 5—pressure sensor; 6—temperature sensor; 7—outlet valve; 8—input module.
For calculations related to supersonic flows, it is preferable to use a density-based solver \cite{54–56}, an explicit calculation method and take into account the gas’ compressibility; although, with relatively small Mach numbers, it is possible to use a pressure-based solver as well \cite{57}.

For some studies, it may be preferable to calculate the gas dynamics of the Hartmann–Sprenger effect process using mathematical simulation \cite{58–61}.

### 3.2.1. Computational Model and Boundary Conditions

The problem was formulated as a plane axisymmetric one. When calculating two nozzle–resonator pairs, the axisymmetric condition was not used. Small design features of the real design (resonator mountings, nozzles, threads) were excluded from the model to reduce mesh complexity and computation time without a loss of resolution in the target areas. In addition, the thermal conductivity through the walls of the resonator, heat exchange with the environment, the temperature dependence of the gas’ heat capacity and the influence of gravitational forces were not taken into account.

The mesh parameters after fitting were as follows. For the case of using a supersonic Laval nozzle, the size of the cells in the nozzle zone, between the nozzle and the resonator and also inside it was \(2 \times 10^{-4}\) m; the size of the first near-wall was \(1 \times 10^{-5}\) m; there were 10 near-wall layers and 40,000 cells altogether. For the case of using a supersonic annular nozzle, the size of the cells in the narrowing zone was \(1 \times 10^{-5}\) m and \(5 \times 10^{-5}\) m in the zones before and after it; the size of the first near-wall layer was \(1 \times 10^{-6}\) m; and there were 70,000 cells altogether. The maximum height of the first cells was calculated based on the quantity \(y^+\), determined from the flow velocity.

The boundary conditions were set as follows: at the inlet, the absolute pressure was 701,325 Pa and the temperature was 27 °C, and at the outlet, the absolute pressure was 101,325 Pa and the temperature was 27 °C. The operating pressure was taken to be zero.

As a working fluid, dry air with the following characteristics was used: isobaric heat capacity \(C_p = 1006.43\) J/(kg·K); thermal conductivity \(\lambda = 0.0242\) W/(m·K); dynamic viscosity coefficient \(\mu = 1.7894 \times 10^{-5}\) kg/(m·s); molecular weight \(M = 28.996\) kg/kmol. The gas density was calculated using the Soave–Redlich–Kwong real gas equation of state.

The Fluent solver settings were a density-based solver; axisymmetric model; energy = ON; viscosity = k-omega SST; method = Explicit Roe-FDS; controls = CFL 0.9.

To accelerate the convergence of the calculation, the “Patch” and two-level FMG (full multi grid) initialization functions were used.

Initially, for each calculation case, a stationary calculation was carried out until the mass flow rates at the input and output of the model were equalized. Further, non-stationary calculations were performed with a time step of \(1 \times 10^{-6}\) s with the same solver settings. One time step included 20 calculation iterations.

### 3.2.2. Confirmation of the Model’s Adequacy

To use this computational model, its adequacy was verified according to the following criteria: the qualitative correspondence of the computational stages of the resonant heating process to the behavior of the real process; the correspondence of the resonance frequency obtained in the numerical simulation to the frequencies calculated from the theoretical dependence.

The stages of the compression and expansion process obtained by calculation are presented in Figure 4. It is clear that the stages qualitatively coincide with the stages of the real Hartmann–Sprenger process in the regurgitant mode.

Based on the data for the pressure change at a point on the damped bottom of the medium resonator in ANSYS, the resonance frequency of 1190 Hz was determined. The theoretical frequency of resonant oscillations \cite{62,63} in regurgitant mode was:

\[
    f_{res} = \frac{c}{Kl_{res}} = 1277 \text{ Hz}
\]
where \( c = 360.32 \, \text{m/s} \) is the local speed of sound in dry air, taken from the average temperature inside the resonator \([54]\); \( K = 4 \) is the shape factor equal to 4 for cylindrical resonators and conventionally equal to 2 for conical resonators \([64,65]\); \( l_{res} = 0.0705 \, \text{m} \) is the length of the inner cavity of the medium resonator.

![Figure 4. Cycle of the Hartmann–Sprenger effect: 1—the gas under pressure moves to the muffled region of the resonator; 2—having reached the muffled cavity of the resonator, the gas is compressed and heated; 3—the compressed gas due to the pressure gradient moves to the open area of the resonator; 4—the gas under pressure leaves the resonator, in which a rarefaction zone is formed, due to which the process repeats cyclically.](image)

The deviation of the frequency value obtained by numerical simulation from the theoretical value was 6.8%. Thus, the computational model can be considered adequate with some assumptions for the qualitative study of the processes under consideration.

3.3. Numerical Vibroacoustic Analysis

The analysis of vibration velocities using the modal superposition method in a prestressed structure was performed using the Steady-State Thermal, Static Structural, Modal and Harmonic Response modules in ANSYS. The thermal loads and the pressure acting on the structure were obtained from the results of the CFD calculation in Ansys Fluent described in Section 3.2.

3.3.1. Computational Model and Boundary Conditions

The computational model consisted of 507,937 tetrahedral SOLID 187 elements and included two bodies: the pressure regulator housing and the medium resonator. The bodies were connected by a bonded contact that provided full adhesion of the objects and included the CONTA 174 and TARGE 170 elements. The Fixed Support boundary condition was applied to the outer mounting surfaces of the pressure regulator mounting rack, as well as at the inlet and outlet connections.

Material parameters such as Young’s modulus, volume modulus of elasticity, shear modulus, thermal conductivity, and the linear thermal expansion coefficient were set depending on the temperature’s effect.

The simulation was carried out in stages. In the first stage, based on the results of 2D CFD calculations, the temperature distribution in the 3D structure of the device was obtained. In the second stage, pressure was applied to the surface of the device structure. In the third stage, the natural frequencies of the stressed structure were obtained in the range from 0 to 2500 Hz. In the fourth stage, the analysis of vibration velocities in the housing and resonator was carried out. To analyze vibrations, it is preferable to determine the vibration...
velocities, since this parameter takes into account the movement of the controlled point and the energy effect from the forces that caused the vibration. The calculation took into account the internal damping coefficient for steel (0.02).

Due to the fact that the CFD calculation was carried out for the purposes of qualitative assessment, the vibroacoustic analysis carried out on its basis serves the same purpose.

3.3.2. Confirmation of the Model Adequacy

According to the numerical model, the frequency of natural vibrations in the medium resonator, in which a sharp increase in vibrations was observed, was 2151.5 Hz (Section 4.3).

The theoretical frequency of natural vibrations of the cylindrical tube equals:

\[ f_f = \frac{c}{2l_{res}} = 2553 \text{ Hz} \]  

(2)

The deviation from the theoretical value is 15.7%, which allows us to consider that this calculation model is adequate.

4. Results

4.1. Results of the Analysis of Sources

Using the Hartmann–Sprenger effect, many researchers have empirically achieved high temperatures. Using air, temperatures of 800…1600 K were achieved [40,45,66,67]; for helium, 1000 K [63,68]; for nitrogen, 870 K [69]; for hydrogen, 811 K [70,71], etc. The maximum temperature is usually reached near the dead-end wall of the resonator and is approximately at the same level for another 10% of the length from the bottom of the resonator. In this case, about 30% of the length from the entrance to the resonator remains cold [40].

Shapiro [35] presented a calculation to determine the maximum temperature rise \( \Delta T \) from the open to the closed end of the resonant tube relative to the average temperature \( T \) depending on the adiabatic exponent \( k \), which equals:

\[ \Delta T / T \cong 2(k + 1). \]  

(3)

At the same time, he made assumptions about the ideality of the gas, the absence of energy losses through the tube, the vibration frequency equal to a quarter of the acoustic wave and that the flow velocity was equal to the speed of sound through the medium. Following this formula, it can be said that the temperature rise for methane is more than 4.5 times; however, it should be noted that, in practice, the temperature rise does not always reach 2.0 times.

4.1.1. General Recommendations

The value of the maximum temperature and the nature of its distribution depend on the composition of the gas, the relationships between the geometry of the devices, the location of the inlet of the resonance tube relative to the compression waves of the flow, the operating mode of the nozzle–resonator pair and the parameters of the gas jet (structure of the compression waves), [45], as well as the materials used for the tube.

The molecular mass of gases influences the speed of sound in the medium and the frequency of oscillations in the resonator. Thus, gases with a lower molecular weight will exhibit faster heating and higher temperatures [71]. Monatomic gases similarly allow higher temperatures and growth rates to be reached in comparison with diatomic ones, since, in accordance with the Rankine–Hugoniot relations, the irreversibility of the shock waves is higher [68]. According to the Eric Brocher formula [72], monatomic gases allow temperatures about three times higher than the ambient temperature to be attained, while diatomic gases allow temperatures two times higher to be attained. According to [41], heating from 300 K to 600 K takes 1.7 ms for helium, 6.9 ms for argon and 49 ms for hydrogen.
The humidity of gases and the direct occurrence of non-equilibrium condensation reduce the temperature at the damped end of the resonator [73].

The nozzle–resonator pair has the following operating modes [69,74]:

1. Instability of the flow. This mode exists only for flows with a subsonic velocity and is characterized by the formation of large periodic vortices at the nozzle exit. In this case, the frequency of the pressure waves inside the resonator is equal to the frequency of the formation of these vortices. In this operating mode, weak shock waves are formed in the resonant tube, which most often do not lead to a significant increase in the gas temperature [75];

2. Regurgitation of the flow. This mode is characterized by periodic ingestion and release of a jet stream by the tube at the tube’s natural resonance frequency. In this case, before entering the resonator, the jet has either a structure with diamond cells or with cylindrical shock waves, depending on the pressure drop across the nozzle. The mode is most effective for relatively long resonators, inside which there is enough space for the superposition of compression waves and the formation of strong shock waves. The temperature reached and the rate of its growth are relatively low;

3. Screech of the flow. This is caused by a relatively high pressure in front of the nozzle and consists of the formation of a normal shock wave (Mach disk) before entering the resonator, oscillating at a higher frequency than in other modes. In this case, weaker shock waves are formed. The mass transfer between the cavity of the resonator and the surrounding flow is reduced, which makes it possible to reach the highest temperatures inside the resonator precisely in this operating mode. The highest efficiency is observed for relatively short resonators [69,74].

Based on the definitions of the modes above, to achieve maximum temperatures, high pressure drops across the nozzle should be used, which provide high Mach numbers. Indeed, the amplitude of pressure fluctuations inside the resonator is directly proportional to the pressure before the nozzle [71], whereas the temperature is proportional to the Mach number behind the nozzle [68]. However, it should be kept in mind that, for different media, the values of the optimal pressure drops at which the maximum temperatures are reached will be different [63]. In addition, the values of the pressure drops must correspond to the geometry of the nozzles and the position of the resonator in order to provide the necessary structure of the underexpanded jet in front of its entrance. An excessive increase in the pressure difference can lead not only to a decrease in the heating value [45,63,71,76], but also to a delay in the occurrence of the Hartmann–Sprenger effect [77] until the pressure difference decreases, or to the complete impossibility of its occurrence [63].

Depending on the pressure drop across the nozzle and the degree of deviation from the design \( n = p_n / p_a \), where \( p_n, p_a \) are the pressures in the nozzle outlet and the surrounding space, respectively), the structure of the underexpanded jet [78] and the location of the “instability” zones change to be inside or near [61] the first [62] or the second [75], making it necessary to place the open resonator inlet coaxially with the nozzle to ensure the occurrence of the Hartmann–Sprenger effect. Of course, as the pressure in front of the nozzle increases, the jet structure will “stretch”; correspondingly, the resonator should be installed further from the nozzle, and vice versa [63]. In addition, with an increase in the distance between the nozzle and the resonator, the transition from regurgitant mode to screech mode will also be achieved at higher pressures in front of the nozzle [69]. The specific position of the resonator affects both the value of the maximum temperature and the rate of its growth [77].

For \( n \leq 2 \), a flow structure with compression shocks in the form of diamonds is observed (Figure 5); for \( n = 2 \ldots 4 \), the structure of the first cell will have compression shocks in the form of a barrel with a Mach disk; and with a significant increase to \( n > 4 \), the formation of Mach disks can be repeated many times in the cells. However, at too high values of \( n \), an increase in the thickness and diameter of the first Mach disk can prevent the formation of subsequent ones [79].
until the pressure difference decreases, or to the complete impossibility of its occurrence [63].

Depending on the pressure drop across the nozzle and the degree of deviation from the design ($n = \frac{p_2}{p_1}$, where $p_2$, $p_1$ are the pressures in the nozzle outlet and the surrounding space, respectively), the structure of the underexpanded jet [78] and the location of the “instability” zones change to be inside or near [61] the first [62] or the second [75], making it necessary to place the open resonator inlet coaxially with the nozzle to ensure the occurrence of the Hartmann–Sprenger effect. Of course, as the pressure in front of the nozzle increases, the jet structure will “stretch”; correspondingly, the resonator should be installed further from the nozzle, and vice versa [63]. In addition, with an increase in the distance between the nozzle and the resonator, the transition from regurgitant mode to screech mode will also be achieved at higher pressures in front of the nozzle [69]. The specific position of the resonator affects both the value of the maximum temperature and the rate of its growth [77].

For $n \geq 2$, a flow structure with compression shocks in the form of diamonds is observed (Figure 5); for $n = 2 \ldots 4$, the structure of the first cell will have compression shocks in the form of a barrel with a Mach disk; and with a significant increase to $n \geq 4$, the formation of Mach disks can be repeated many times in the cells. However, at too high values of $n$, an increase in the thickness and diameter of the first Mach disk can prevent the formation of subsequent ones [79].

Figure 5. Structures of underexpanded jet flow at various degrees $n$ of deviation from the design.

The structure of the jet and its zones of “instability” can be determined experimentally using Schlieren photography, but this is rather laborious. Another way is analytical calculation using some model to predict the flow structure [79]. In order to determine the boundaries of the “instability” zone, one should calculate the position of the direct compression shock (Mach disk) for a flow structure with a barrel shock cell structure, or the position of the intersection point of inclined shock waves for a flow structure with compression shocks in the form of diamonds (diamond shock cell structure). The position of the Mach disk can be determined based on various empirical formulas [80] that can be applied depending on the pressure drop across the nozzle and the total pressure (unfortunately, most of them are applicable only for converging nozzles). In the framework of their studies, E. Brocher and C. Maresca [68] concluded that, for the subsonic regime, the optimal ratio of the distance between the converging nozzle and the resonator to the resonator diameter is approximately equal to one; for the supersonic regime (Mach number is 2), it is about two for a diatomic gas and about three for a monatomic one. For nozzles with a diffuser and a jet structure in the form of barrels, the position $x$ of the Mach disk (Figure 5) can be determined using the following relations [81,82]:

\[ x = 0.8L, \]  
\[ L = 1.558\cdot x', \]  
\[ \frac{x'}{r} \sim \sqrt{n(1 + kM^2 \cos a) - 1}, \]
where \( L \) is the length of the first cell; \( M \) is the Mach number in the outlet section of the nozzle; \( r \) is the radius of the outlet section of the nozzle; \( k \) is the adiabatic exponent; and \( a \) is nozzle half-opening angle. The point at which the inclined shock waves for the flow structure intersect with diamond-shaped shocks will be located approximately in the middle of the cell. For nozzles with more complex geometries, numerical methods of calculation should be used.

To achieve significant heating, it is necessary to use relatively longer resonators [68] than those of Hartmann’s “whistles”. A longer cavity allows the heated gas to be retained in the dead-end zone and to intensify the effect of pressure shock waves on it. However, for each family of characteristics of each installation, there exists an optimal resonator length. Above this length, the rate and magnitude of heating decrease due to an increase in the mass of the resonator walls and the heat exchange area. Below the optimal length, the rate and magnitude of heating decrease due to an increase in the transverse nonuniformity in the jet and the intensification of heat transfer between the hot and cold parts of the flow [40]. The range of ratios of the length of the resonator to the diameter of its inlet most often lies in the range of 15...37 [40,55]; however, there are studies of resonators with a relative length which is smaller (7.5 [71]) or larger (about 70 [38]).

4.1.2. Recommendations for Reaching the Maximum Temperature

The first need concerning the selection of parameters that not only ensure the appearance of the Hartmann–Sprenger effect, but also make it possible to achieve its maximum efficiency, is the correct selection of the inlet diameter of the resonator for the transverse dimensions of the underexpanded jet. It is important that the largest part of the flow energy is directed precisely inside the resonator, and not past it. On the other hand, an excessively large area of the inlet cross section of the resonator will contribute to the formation of flow recirculation zones at the inlet [83], and with a further increase, it will contradict the very nature of the effect. The most popular recommendation is to ensure that the ratio of the area of the resonator inlet to the area of the jet in the range of 1.0...2.5 [45,68,71]; there are also recommendations for the value 6.0 [55], although they are likely proposed under certain conditions.

Due to the importance of the jet structure for various gases, there are optimal degrees \( n \) of deviation from the design (for example, when using nitrogen and a sonic nozzle—4.5 [71], 6.0 [69]), which can be controlled by changing the ratio of the cross-sectional area for the gas exit from the cavity between the nozzle and the resonator and the area of the critical section of the supersonic nozzle (for example, for oxygen or air—4.0...8.0 [84,85], 5.5...6.0 [86]).

An obvious way to increase the temperature inside the resonator is to use thermal insulation [34,41] and to reduce the heat transfer surface area [70], as well to use materials with low thermal conductivity [65,71]. However, such methods are suitable only when the magnitude of cooling needed for the second “cold” energy separation flow does not matter.

A common way to increase temperature and regulate heat transfer is to use resonators of various shapes. The most popular for maximizing temperature are resonators of the following shapes: conical with a cone angle of 3\(^\circ\)...4\(^\circ\) [54,67,71]; cylindrical with a transition to a cone [45]; stepped [40,41,54,87]; “resonator in a resonator” [38]; and those with expansion at the end of a resonator with a narrow neck [86]. It is noted that stepped resonators have the highest rate of temperature rise, although conical resonators make it possible to reach similar temperatures. In addition, stepped and conical ones at relatively low oscillation frequencies make it possible to achieve a heating efficiency that is similar to those reached by cylindrical ones at relatively high frequencies [40]. This is achieved by smoothing the flow parameters with respect to the length of the stepped or conical resonators due to a change in their cross-sectional area, which makes it possible to increase the heating efficiency at low-frequency oscillations. In general, of course, to ensure a more uniform heating of the resonant tube along the entire length, oscillations should be maintained at higher frequencies than the natural frequency of the oscillations [88]. The expansion at the end of the resonators makes it possible to retain the heated part of the gas.
by creating additional resistances in the narrow neck to the removal of the hot gas and the formation of a stagnant zone. Taking into account the practical application, it is important to note the different levels of complexity in the design of resonators of various shapes (cylindrical is the simplest), especially if their length is significant. The roughness of the inner surface of the resonators is also important; for example, for resonators of short length, a higher roughness of the inner surface allows increasing the maximum temperature [68].

More extraordinary ways of increasing the temperature are the use of various additional elements, either generating nonstationarity, contributing to the stable existence of the Hartmann–Sprenger effect or “focusing” shock waves. At subsonic speeds, an increase in efficiency can be achieved by crossing the exit section of the nozzle with a thin wire (the Mach number is 0.73) [89] or using elements [90] that reduce the stability of the jet near its surface. For all modes of operation of the nozzle–resonator pair, various pins can be used, which are installed either along the axis of the resonator [34,74] or along the axis of the nozzle [41,91]. Installing a pin with a length of about half (56%) of the effective length of the internal cavity of the resonator increases the heating rate by focusing the compression shocks in the resonator by narrowing its effective cross-section area and increasing the amplitude of oscillations inside the resonator [74]. However, it should be kept in mind that such a pin allows the resonator to function in regurgitant mode in a wider pressure range. Installing the pin as the central body of the nozzle contributes to the creation of a local deficit of kinetic energy in the center of the jet, which reduces the resistance of the incoming flow to empty into the tube and thereby increases the amplitude of pressure fluctuations inside it [91]. In general, the use of annular nozzles, due to their autocontrol property in the abnormal overexpansion mode, allows the range of pressure drops for the stable functioning of the Hartmann–Sprenger effect to be expanded [41].

4.1.3. Practical Recommendations

For the purposes of practical applications, important aspects of reducing the pressure of natural gas include the ability to work with incompletely purified gas; ways to optimize dimensions; the noise and vibrations created during operation; and the reliability of the device.

The hole at the bottom of the resonator can serve either to remove the hot part of the gas [34,65], to measure the temperature at the bottom of the resonator in the experiment [69,71] or to prevent clogging of the tube with condensate and mechanical impurities contained in the gas [92]. During the experiments performed by Kawahashi and Suzuki [34], the ratio of the hole area to the cross-sectional area of the resonator that was equal to 0.12 turned out to be the most effective.

The calculation of the minimum area of the hole at the bottom of the resonator to prevent clogging of its cavity with condensate can be determined [92] on the basis of the mass flow rate of moisture entering the resonator:

\[ m = h \cdot q \frac{S_r}{S_f}, \]  

(7)

where \( h \) is the absolute humidity of the gas; \( q \) is the volumetric gas flow rate through the device; \( S_r \) and \( S_f \) are the cross-section areas of the resonator and the flow mixing housing of the device (Figure 1), respectively.

The minimum area of the hole at the bottom of the resonator to prevent stagnation of condensate, taking into account the alternation in the magnitude of the amplitude of pressure fluctuations inside the resonator relative to the average pressure in the flow, can be expressed by the following formula:

\[ S_{min} = \frac{2m}{\mu \sqrt{2\rho \Delta p}}, \]  

(8)
where \( \mu \) is the flow rate coefficient of the flow through the hole; \( g \) is the acceleration of gravity; \( \rho \) is the density of the condensate inside the resonator; and \( \Delta p \) is the limiting amplitude of pressure fluctuations in the cavity of the resonator [93], calculated by the formula:

\[
\Delta p = 2kM p_f, \tag{9}
\]

where \( M \) is the Mach number and \( p_f \) is the average pressure in the flow.

To optimize the dimensions at high flow rates, the possibility of splitting the flow into several nozzle–resonator pairs is obvious; however, we cannot find any current studies that have set this very goal. There are studies of installations using several Hartmann “whistles”. There are also studies of installations with two resonators, but one accelerating device [94].

When using the Hartmann–Sprenger effect, the question of the influence of acoustic impact and vibrations on the surrounding systems and structures is important. The noise pressure level at a certain resonant frequency can reach 100 db and higher [67, 74, 95]. The noise level is reduced when the resonant tube is operated at higher frequencies [88]. In this case, the peaks in the sound spectrum will refer either to the natural oscillation frequency of the resonator (lower frequencies), or to the regurgitant mode, screech mode, or their intermediate mode [74]. Since sound is mechanical oscillations, its presence cannot but be accompanied by vibrations, which can be significant depending on the field of application [40, 45]. Taking into account the fact that various types of technological equipment often operate under dynamic loading and vibrations, an increase in the level of such an impact upon the integration of the Hartmann–Sprenger effect can negatively affect their reliability and operational safety.

As for the impact of high temperatures and pressure surges, these are factors that significantly affect the duration of the accident-free operation of devices. It is important to note here that, even within the framework of relatively short-term experiments, researchers observed failures of resonant tubes [45, 55, 67, 71]. Resonators must simultaneously be strong, impenetrable, have high thermal conductivity and a wall structure that does not absorb pressure fluctuations [71]. For example, when using the Hartmann–Sprenger effect for ignition systems where not only high temperatures exist, but also corrosion with oxygen oxidation can occur, more resistant materials should be used; for example, an alloy of cobalt and chromium (CoCr powder) [57], zirconium or refractory PPFE polymers [54, 65].

4.1.4. Summary Recommendations

If we combine the data from various scientific papers, then we can give the following generalized recommendations that allow the best results to be achieved:

1. A monatomic gas with a minimum molecular weight should be used as a working medium, if the medium is not specified initially;
2. “Screech” mode should be used for the operation of the nozzle–resonator pair;
3. Pressure drops equal to four or more should be used;
4. The location of the resonator inlet should be selected depending on the specific structure of the flow and the position of the “instability” zone;
5. The ratio of the resonator length and the diameter of its inlet section should be in the range of 15...37;
6. The use of stepped or conical resonators with a cone angle of 3°...4°;
7. The need to remove unwanted clusters at the bottom of the resonator through the hole should be taken into account;
8. The use of annular nozzles designed for the operating pressure drop;
9. The ratio of the areas of the resonator inlet and the critical section should be in the range of 1.0...2.5;
10. The ratio of the area of the flow section for the gas outlet from the cavity between the nozzle and the resonator and the area of the critical section of the supersonic nozzle should be at the level of 4.0...8.0;
11. The use of resistant materials such as cobalt–chromium alloys, zirconium or refractory polymers is required.

4.2. Results of Numerical Simulation of Gas Dynamics

4.2.1. Numerical Simulation of the Structures of Underexpanded Jets

Naturally, to achieve supersonic speeds, it is necessary to design supersonic nozzles according to the gas flow parameters, such as pressure drop, temperature drop, and mass flow rate. However, in the process of actual operation, the parameters of technological installations may change. Naturally, when the backpressure or flow rate changes, the nozzle leaves the design mode, and the compression shock cascade changes its position.

Changes in the jet structure at different pressure drops (from 200 kPa to 900 kPa) for the Laval supersonic nozzle are shown in Figure 6. It can be seen that, for the pressure drop under consideration and at a given backpressure, the jet structure has diamond-shaped cells. The cell lengths, and hence the distance of the nozzle from the “instability” zones, are directly proportional to the pressure drop across the nozzle, which must be taken into account when positioning the resonator inlet relative to the nozzle.

![Figure 6](image-url)

**Figure 6.** The position of compression shocks in an underexpanded free jet past the Laval nozzle at various pressure drops across the nozzle: 1—200 kPa; 2—300 kPa; 3—400 kPa; 4—500 kPa; 5—600 kPa; 6—700 kPa; 7—800 kPa; 8—900 kPa.
The boundaries of the instability zone and the maximum values of the width of the first cell, excluding the jet mixing zone, are shown in Figure 7. The lengths of the “instability” zones are directly proportional to the pressure difference, which reduces the requirements for the positioning accuracy of the resonator inlet. However, the rate of increase in the instability zone length is significant only in the range of pressure drops up to the calculated one, further the elongation is insignificant. Satisfactory convergence was not achieved with the formula in Section 4.1.1 due to its empirical nature.

![Figure 7](image_url)

**Figure 7.** The boundaries of the “instability” zones and the width of the jet cell after the Laval nozzle, depending on the pressure drop across it.

The cell width is also directly proportional to the pressure drop, which must be taken into account when determining the resonator inlet area. It can be seen in Figure 6 that, at differences of 200 kPa and 300 kPa, the cell does not have a sufficiently formed structure; accordingly, its width is not shown in Figure 7.

Since the use of an annular nozzle is supposed to provide the device with a flow control function, when modeling its operation, it was not the pressure drops that changed, but the mass flow through the device at one-third of the total design flow rate, two-thirds of the total and the total flow rate (0.0090375 kg/s) (Figure 8). The geometry of the central body of the annular nozzle is a modified and simplified (linearized) version of the one calculated using the standard calculation for similar conditions in which the Laval nozzle operates.

![Figure 8](image_url)

**Figure 8.** The position of compression shocks in an underexpanded free jet after the pin nozzle depending on the flow through the nozzle: 1—one-third of the total design flow rate; 2—two-thirds of the total; 3—the total flow rate.
It can be seen from Figure 8 that, at one-third and two-thirds of the total calculated flow rate, the first cell in the jet is located directly on the central body, which does not allow the resonator inlet to be positioned relative to it. Thus, the positioning of the resonator was performed for the second cell of the annular nozzle. When the nozzle is running at full flow rate, one can observe a small Mach disk after the first nozzle and subsequent cells with a diamond structure with clear zones of “instability”. In addition, detached flows are observed.

Figure 9 shows that the ranges of the boundaries of the zones of instability are directly proportional to the mass flow rate through the nozzle. However, the cell width changes less significantly than in the case of the Laval nozzle (16.9% versus 58.8%).

Figure 9. The boundaries of the second “instability” zones and the width of the second jet cells after the annular nozzle, depending on the mass flow through the nozzle: one-third of the total design flow rate and two-thirds of the total rate and total flow rate.

In general, according to Figures 7 and 9, it can be seen that the “instability” zones have sufficient lengths for successful positioning of the resonator inlets.

Such stationary calculations for determining the boundaries of instability zones are relatively simple, which allows them to be used in the design of devices based on the Hartmann–Sprenger effect. However, consideration should be given to changing the location of compression shocks when resonant tubes are installed in the path of the jet.

4.2.2. Numerical Simulation of the Structures of Underexpanded Jets When Installing Resonators

The position of the compression shocks of an underexpanded jet when a plugged cavity is placed in its path periodically changes due to the cycle of gas outflow from the cavity and its filling. To visualize them, a series of simulations of the process dynamics were carried out.

Figure 10 shows two phases of the process (filling and emptying the resonator) and the corresponding positions of compression shocks when the middle resonator is installed at a distance of 6…9 mm from the exit edge of the nozzle \((l/d = 2…3)\) at a calculated pressure drop of 600 kPa.

For a distance of 10 mm, it can be seen from Figure 10 that the resonator inlet is located much further than the first zone of jet instability; however, the Hartmann–Sprenger effect exists at this position.

In comparison with the data on the boundaries of the “instability” zone calculated without installing a resonator (Figure 6), in the case of installing a resonator (Figure 10), the average values of the boundaries of the “instability” zone changed by less than 4%. The cell width changed by less than 5%. The values were calculated based on the geometric dimensions of the simulated device. The graphs show horizontal distance scales in mm from the nozzle exit edge.
When simulating the pin nozzle, it was determined that it is optimal to install the resonator in the zone of instability of the second barrel, and not the first, otherwise the removal of the gas exhausted in the resonator is hampered, and the Hartmann–Sprenger effect does not occur. The simulation was carried out at the total flow rate through the nozzle in the range of distances from 8 mm to 12 mm ($l/d = 2.67 \ldots 4.00$).

In comparison with the data on the boundaries of the second “instability” zone after the annular nozzle, calculated without installing a resonator (Figure 8), in the case of

**Figure 10.** The position of compression shocks in the phases of filling and emptying the resonator at different distances between the Laval nozzle and the resonator: 1—6 mm; 2—7 mm; 3—8 mm; 4—9 mm; 5—10 mm.
installing a resonator (Figure 11), the average values of the boundaries of the second “instability” zone changed by less than 5%. The width of the second cell changed by less than 1%. Such a small change in width may be due to the fact that this is the second cell, and the entire flow is less constrained as a whole.

**Figure 11.** The position of compression shocks in the phases of filling and emptying the resonator at different distances between the pin nozzle and the resonator: 1—8 mm; 2—9 mm; 3—10 mm; 4—11 mm; 5—12 mm.

Based on the simulation (Figures 10 and 11), it was determined that the position of the resonator has practically no effect on the average position of the shock wave of the underexpanded jet in the regurgitation regime. Thus, for each pair of flow rates and...
pressure drop, the ranges of optimal resonator inlet positions can be determined based on stationary calculations of jet structures without resonators.

4.2.3. Simulation of a Block Installation of Two Nozzle-Resonator Pairs

To achieve the Hartmann–Sprenger effect, longer resonators should be used than in the case of the Hartmann whistle. Their length depends on the geometric parameters of the nozzle, which in turn are calculated depending on the mass flow rate of the gas. Thus, if one nozzle–resonator pair is used for the purpose of reducing high flow rates, then its length will be significant. This fact complicates the arrangement of technological equipment in designing gas-reducing facilities and the modernization of existing reduction points, since it requires a change in the technological scheme.

The way out of this situation is to use several nozzle–resonator pairs to reduce the mass flow through each of them (at 50 thousand m$^3$/h, the length can be reduced from 4.0 m to 0.5 m). In this case, one should take into account the mutual influence of nearby flows on each other, since a situation of mutual damping is possible.

To analyze the possibility of the occurrence of the Hartmann–Sprenger effect, a numerical simulation (Figure 12) of gas-dynamic processes of two nozzle–resonator pairs was carried out. Pairs were installed at different distances $l'$ from each other: from 0 mm to 6 mm ($l'/d = 0 \ldots 2$).

Figure 12. Temperature contours when installing two nozzle—resonator pairs in a single flow housing at different distances from each other: 1—0 mm; 2—2 mm; 3—3 mm; 4—4 mm; 5—5 mm; 6—6 mm.

The behavior of the jets when two nozzle–resonator pairs are installed is similar to the behavior of the jet when only one pair is in operation. While in the case of one pair, when the resonator was empty, the jet was compressed in the axial direction; then, in the case of two pairs, when emptying the jet, the jets deviated from the device axis in the transverse direction. Figure 12 shows the times at the end of the compression phases when the temperatures were at their maximum.
It can be seen that, when the distance between the pairs is from 0 mm to 5 mm \((l'/d = 0 \ldots 1.67)\), no effect is observed due to excessive resistance to the passage of gas between the two resonators. The resistance creates additional counterpressure for the oncoming flows and makes them deviate from the trajectories that provide the necessary compression of the gas inside the resonators and its subsequent ejection into the internal cavity of the device surrounding the resonators. At 6 mm \((l'/d = 2.0)\), there is a strong heating in the dead-end cavity of the resonator, as well as the distribution of flow parameters between the two resonators, similar to that between the resonators and the device housing. In this case, the minimum area required for the passage between the resonators is provided for a uniform gas removal device over the entire area of the passage section and the removal of any undesirable effects on the oncoming flows. With an increase in the distance between the resonators, the effect will be identical. Thus, it can be assumed that the minimum distance between the two nearest pairs is greater than or equal to the sum of the diameters of the critical sections of the nozzles.

4.3. Results of Numerical Vibroacoustic Analysis

The values of natural frequencies in the range from 0 Hz to 2500 Hz are, respectively, 1—853.45 Hz; 2—865.67 Hz; 3—894.40 Hz; 4—990.15 Hz; 5—1592.50 Hz; 6—1926.70 Hz; 7—2151.50 Hz; 8—2248.00 Hz; 9—2318.90 Hz.

Figure 13 shows the forms of natural frequencies of the resonator close in value to the frequency of the resonance effect of the Hartmann–Sprenger effect. The maximum deformation values were 0.8 mm on the resonator and 0.28 mm on the body of the structure.

Figure 14 shows the obtained vibration velocity curves. The vibroacoustic analysis showed the occurrence of resonant vibrations at the natural frequency of the resonator. Based on the results obtained, the following conclusions can be made: a particularly dangerous frequency for this particular design is the frequency of 865.67 Hz; at the resonator’s natural frequency of 2151.5 Hz, a resonant vibration is observed with an increase in vibration velocities in the transverse direction of the resonator by 20 dB.
5. Discussion

An analysis of the literary sources and patent materials showed that the most important factors for the effective functioning of devices based on the Hartmann–Sprenger effect are high pressure drops on the nozzles, which are calculated directly for each one; the location of the resonator inlet and the determination of the inlet size depending on the specific structure of the underexpanded jet; the use of resonators with a variable cross-sectional area; and ensuring sufficient exhaust gas evacuation. The remaining recommendations for intensifying the effect are not universal. From a practical point of view, to ensure reliable operation, the most important points are the choice of materials that are resistant to significant cyclic loads, as well as taking into account the need to remove unwanted accumulations at the bottom of the resonator.

Based on the simulation, it was determined that the resonator positions for each flow rate and pressure drop pair during operation in the regurgitation mode can be determined from a series of calculations for the underexpanded jet structures. The dependencies of the required position of the resonator, in turn, can be used in the design of mechanisms for regulating the position of the resonators under non-stationary operating conditions of the pressure regulator. The operation of various resonators is possible with a fairly wide range of distances to the nozzle, which makes it possible to not impose strict requirements on the accuracy of the control mechanism. To operate the annular nozzle as a working body of the pressure regulator in a wide range of flow rates, it is necessary to perform a special profiling of the central body, which is a separate line of research.

Simulation of two nozzle–resonator pairs confirmed the possibility of installing several pairs in a single flow housing without disrupting the operation of each of them, which allows the dimensions of pressure regulators to be optimized.

The vibroacoustic analysis showed the occurrence of resonant vibrations at the natural frequency of the resonator. Noise and vibration effects, of course, must be taken into account when using devices based on the Hartmann–Sprenger effect, but their magnitudes are not very large and will not create significant obstacles for practical application.

6. Conclusions

The results of the conducted studies suggest that the implementation of a quasi-isothermal natural gas pressure regulator based on the Hartmann–Sprenger effect is possible. Determining the required positions of the resonator at various flow rates through the nozzle and various pressure drops is not difficult. The increased accuracy of the resonator-positioning mechanism is not required due to the fairly wide range of distances from the nozzle to the resonator that is allowable. Compact dimensions can be achieved by dividing the total flow into several nozzle–resonator pairs. In addition, noise and vibrations do not go beyond their usual values for such technological equipment.

However, to determine the quantitative parameters of the efficiency of such a reduction in the conditions of large gas-reducing facilities, it is necessary to conduct additional studies with ranges of flow rates and pressure drops that are close to real ones, which will be carried out in the future.

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