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REVIEW

Study of the Influence of Dissipative Effects on the Temperature Stratification in Gas Flows (Review)

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Abstract—We review publications devoted to various types of gasdynamic energy separation. Processes occurring in a vortex tube, ejectors with a negative ejection factor, stratification in gas flows and flows around the walls, etc., are discussed. The data and information on the methods and methodologies allowing one to carry out investigations and/or estimations of the energy separation effect are presented. Particular attention is paid to the effect of gas dynamic temperature stratification.

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CONTENTS

INTRODUCTION

Currently, many examples of temperature separation in gas flows are known. Thermal (temperature) stratification can be caused by various reasons. Sometimes, it occurs as a result of a disbalance between the amount of heat released at the expense of work of friction forces and the amount of heat which can be transferred due to the thermal conductivity at a given temperature. In other cases, temperature separation is initiated by vortex flows or pulsations of pressure or due to the generation of acoustic waves. Some of these phenomena are used in various technical devices when obtaining temperature differences.

VORTEX METHOD OF ENERGY SEPARATION

To date, the best known method of energy separation is the vortex method proposed in 1931 by French metallurgical engineer Georges Joseph Ranque. On December 12, 1931, at 2:41 p.m., he made an application for an invention to the patent office in France and on March 24, 1932, he received the patent FR 743111 for the process of getting two flows with different temperatures from a flow of compressed gas or vapor. A fragment of the first page of this patent is shown in Fig. 1. A year later, Ranque received a complimentary patent FR 43164R on improvements and changes in the design of the vortex tube. In addition to France, applications were also filed in the United Kingdom (patent GB 405781 of February 15, 1934) and the United States (patent US 1952281 of March 27, 1934).

In his work [1] and patents, Ranque tried to declare not only the design features known to him, but also suggested possible ways to develop this product and patented constructional elements required for this purpose (Figs. 2, 3).

In 1933, Ranque made a report at a meeting of the French Physical Society about the discovery of the phenomenon of separation of compressed gas, in which he attempted to develop a theory of the centrifugal vortex effect. As was shown, the effect of energy separation (32 K) obtained in the experiment was about four times higher than stems from the results of theoretical calculations. This communication of Ranque was met with disbelief, while the experimental data were announced to be the result of measurement error, so his effect was forgotten.

The second time, the effect of vortex energy separation was discovered during the Second World War by Robert Hilsh, professor at University of Erlingen, who published a number of works (see, for example, [2, 3]) dedicated to the study and improvement of the vortex effect and Ranque's tube in 1946–1948 (sometimes it is called the Ranque–Hilsh tube). At that time, the adiabatic efficiency of turborefrigerators did not exceed 0.3, their resource and reliability did not meet the consumer needs, and, thus, alternative sources of cooled flows were of great interest.

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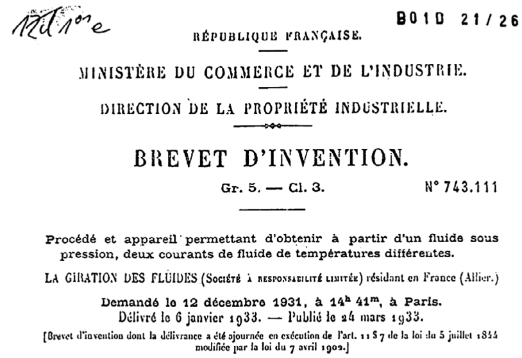

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DIRECTION DE LA PROPRIÉTÉ INDUSTRIELLE.

BREVET D'INVENTION.

Gr. 5. — Cl. 3. N° 743.111

Procédé et appareil permettant d'obtenir à partir d'un fluide sous pression, deux courants de fluide de températures différentes.

LA GIRATION DES FLUIDES (Société à responsabilité limitée) résidant en France (Allier.)

Demandé le 12 décembre 1931, à $14^{h}\ 41^{m}$, à Paris.
Délivré le 6 janvier 1933. — Publié le 24 mars 1933.
[Brevet d'invention dont la délivrance a été ajournée en exécution de l'art. 11 § 7 de la loi du 5 juillet 1844 modifiée par la loi du 7 avril 1902.]

Fig. 1. A fragment of the first page from the patent FR 743111 in France that was obtained by Ranque.

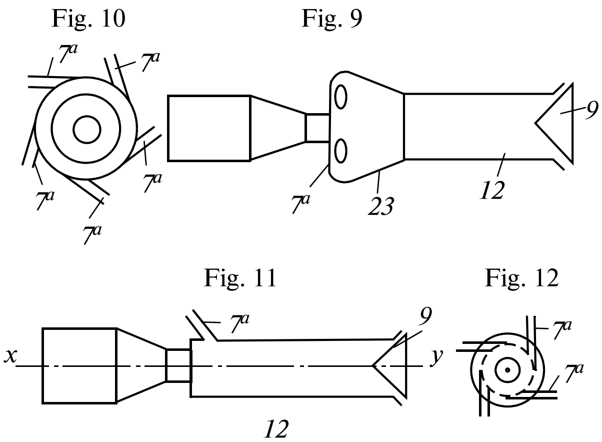

Fig. 2. Multi-input options proposed by Ranque (fragments of the patent FR 743111).

devoted to the vortex effect, and the patent library has hundreds of patents, copyright certificates, etc. However, in spite of the extreme simplicity of the device, and the large amount of accumulated data, there is no common accepted theory of energy separation and mass transfer, with which the calculating of the gas dynamic and physical parameters in a vortex tube is possible.

Sufficiently detailed reviews of the existing hypotheses explaining the vortex effect by Ranque are given in [4–7]. The most common physical model describing temperature stratification in a vortex tube is the model by Fulton [8]. According to this model, a gas flow entering the vortex tube chamber is twisted according to the law of a free vortex (ωr = const) with a high angular velocity in the center of the vortex. The static temperature varies along the vortex radius and increases towards the periphery, while the braking temperature is distributed uniformly. In the process of subsequent restructuring of the flow from the law of a free vortex (ωr = const) to the rotation at a constant angular velocity ω = const, a heat flux arises due to the gradient of static temperature that is directed from the periphery to the center, as well as the flow of mechanical energy from the axis to the periphery due to the viscosity. This results in the alignment of the thermodynamic gas temperature across the cross section of the tube while the braking temperature increases with increasing radius. Figure 4 shows the picture of the gas flow in the counter-flow vortex tube. While providing transfer of the cold gas from the near-axial region and the heated gas from the near-wall region (flows *3* and *4* in Fig. 4), one can get a temperature stratification in the vortex tube.

According to the Fulton model, to obtain the energy separation effect comparable with the experimental data, a free vortex is transformed into an induced vortex having a greater peripheral velocity as compared to the velocity of the free vortex.

As is usually accepted, the efficiency coefficient in vortex tubes used for cooling and heating is estimated by its temperature η and adiabatic $\mu\eta$ magnitudes. Sometimes, the temperature efficiency coefficient is also called the coefficient of thermal efficiency. It can be written in the form

$$\eta = \frac{\Delta T_X}{\Delta T_S} = \frac{\Delta T_X}{T_0\left[1 - \left(\frac{P_X}{P_0}\right)^{\frac{k-1}{k}}\right]},$$

where ΔT_S is the temperature decrease upon isoentropic cooling in the process of gas expansion from P_0 to P_X, and ΔT_X is the decrease of temperature when cooling the portion of gas in the vortex tube.

However, thermal efficiency coefficient η does not take into consideration the share of cold and hot flows in the consumption. This does not allow one to compare vortex tubes with different consumption ratios. To solve this problem, parameter $\mu = G_X/G_0$ is introduced, which characterizes the ratio of the mass of cold flow to the total mass that is consumed. The efficiency of vortex tubes is estimated by the product $\mu\eta$, which is called the adiabatic efficiency coefficient.

Several works devoted to the experimental and numerical study of this effect have been published [9–20]. However, despite the long history (more than 80 years), the problem of creating a conventionally accepted mathematical model of the vortex tube has not been solved until now.

WAVE AND VORTEX METHODS OF ENERGY SEPARATION

Other methods of energy separation in gas flows are also known. Figure 5 shows the scheme of a working part of the resonance tube by Sprenger [21, 22]. When a flow of air flows around the tube with a closed bottom, the temperature of the surface on the bottom may exceed the initial temperature of braking of the gas by several times under certain conditions. In this case, the gas temperature in the jet flowing out of the tube decreases. This effect is observed when resonance acoustic regimes appear, and the magnitude of the temperature stratification depends considerably on the distance between the nozzle and the open hole of the tube. Figure 6 shows a photograph of the working section made of wood with a cavity that was burned out [22].

During repeated experiments performed in ZAGI, ignition of the resonance tube made of wood occurred many times.

For the first time, this effect was registered by Hartmann [23], but he failed to obtain a considerable effect of energy separation and the works in this direction were stopped. Currently, study of this apparatus is continued [24, 25], and it is often called a Hartmann–Sprenger tube.

An interesting invention was proposed by Emin and Zaritskii [26, 27]. The corresponding scheme is shown in Fig. 7. After a supersonic nozzle *4*, one part of the gas passes through diaphragm *6*, while another part enters camera *3* and leaks out through valve *2*.

During normal operation of this device, the heated gas leaks out of diaphragm *6*, and cold gas leaks out through valve *2* of the antechamber. As follows from the analysis of works [26–29], one can assume that in this apparatus the eddy effects are combined with the wave effects, so with careful selection of the parameters, the process of energy separation may occur in the mixing chamber.

Sometimes in the literature, the ejector with a negative ejection coefficient is called an in-line analogue of the vortex tube. This is likely due to the fact that, in the process of energy separation, of greatest importance is vortex formation in the mixing chamber (or a system of vortexes). A characteristic scheme of the gas flow in the mixing chamber is shown in Fig. 8.

ENERGY SEPARATION IN TWO-PHASE FLOWS

An interesting problem is the separation of energy in two-phase flows proposed by Stolyarov [30]. The scheme of his method is shown in Fig. 9. Upon increasing the temperature of the two-phase flow in nozzle *5*, the static temperature of the flow decreases. Moreover, the gas temperature diminishes much faster than the temperature of liquid drops moving together with the flow. Due to the process of thermal conductivity, the liquid drops transfer the heat to the surrounding gas; i.e., the heat passes from the liquid phase to the gas phase. The temperature of the gas increases, while the temperature of the liquid drops decreases approaching the static temperature of the gaseous phase. When the flow of liquid droplets encounters an obstacle, it has no time to change the direction of motion and, together with the supersonic gaseous flow, it is deposited onto the surface of separa-

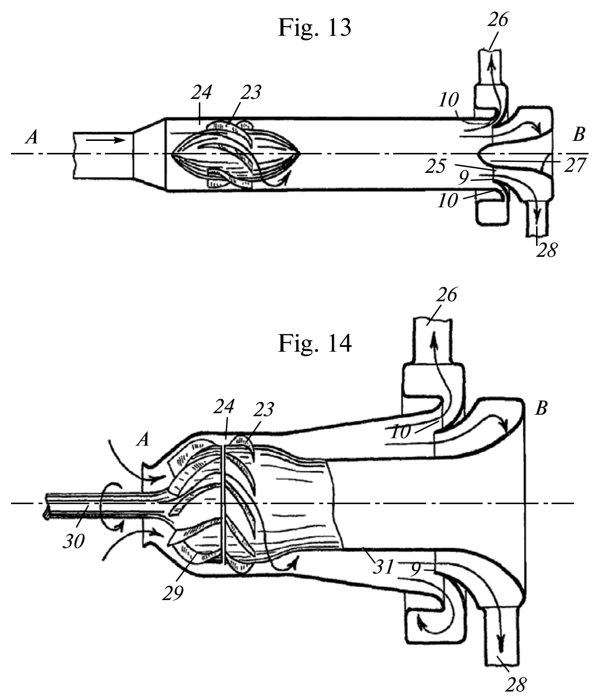

Fig. 3. Mechanical swirler options proposed by Ranque (fragments of the extra patent FR 43164E).

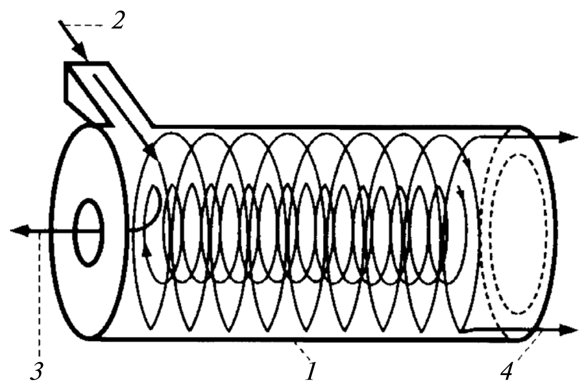

Fig. 4. The picture of the gas flow in a counterflow vortex tube: *1*, vortex tube; *2*, input of compressed gas; *3*, output of cold gas; *4*, output of the heated flow.

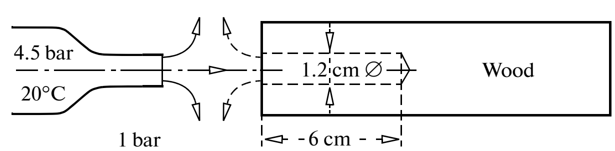

Fig. 5. Scheme of the working part of Sprenger's resonance tube [22].

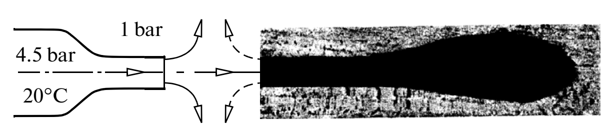

Fig. 6. Photo of the working part of Sprenger's resonance tube (made of wood) after 30 s operating time (the working part was sawn along the axis of the hole) [22].

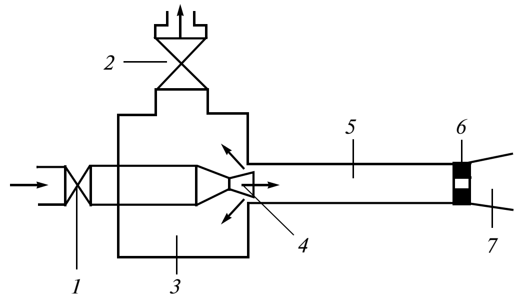

Fig. 7. The ejector with a negative value of the ejection coefficient [26, 27]: *1*, *2*, regulating taps; *3*, antechamber; *4*, supersonic nozzle; *5*, mixing chamber; *6*, diaphragm; *7*, diffuser.

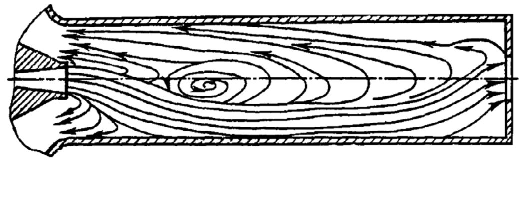

Fig. 8. Scheme of the gas flow in a mixing chamber of the ejector with a negative value of ejection coefficient (right, output of hot gas; left, input of compressed gas and output of cold gas).

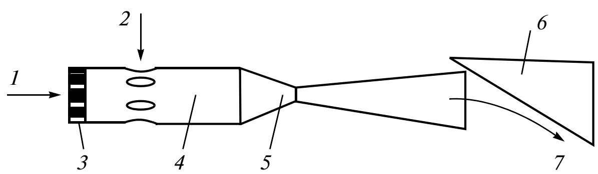

Fig. 9. Scheme of the energy separation device in a two-phase flow [30]: *1*, input of water; *2*, input of gas; *3*, pulverizer; *4*, mixer; *5*, supersonic nozzle; *6*, separator; *7*, output of gas.

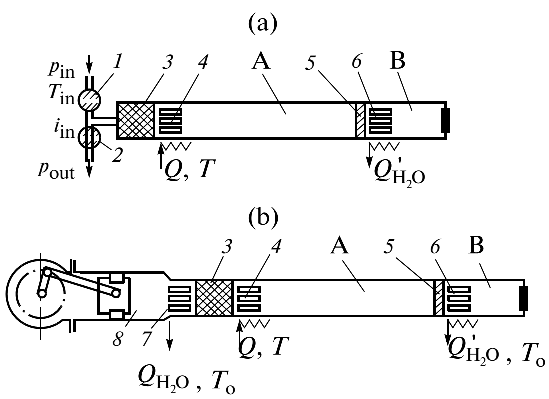

Fig. 10. Improved designs of pulsating tube [36]: (a) with valves; (b) with piston attached; *1*, *2*, input and output valves; *3*, regenerator; *4*, heat exchanger–cooler; *5*, nozzle (throttle hole); *6*, refrigerator; *7*, end refrigerator; *8*, compressor–expander; A, working space of the tube; B, receiving vessel.

tor *6*. As a result, heated gas flows from the output *7* and a film of cooled liquid phase is formed on the surface of separator *6*. The impact of particles of different inertia on the intensity of turbulence of the carrying gas in the tube was studied in [31, 32]. It was shown that stratification of temperature in the two-phase flows may be due to the dynamic and thermal slip of gaseous and dispersed phases.

PULSATION TUBES

In 1963, Gifford and Longsworth [33] proposed a new type of cooler. This simple and robust design was named the "pulsation tube." It provides for the appearance of the temperature gradient when the gas enters the tube and leaks out of it.

This invention proved to be interesting for obtaining low (cryogenic) temperatures. As was shown in [34], in a single-stage pulsation tube, the minimum attainable temperature is 124 K, while in the two-stage system, the minimum temperature is 79 K.

An improved version of a pulsation tube with an extra volume receiver and a tapered device (ventil, nozzle, or a muzzle) was developed at the Bauman Higher Technical School [35] (Fig. 10). In this case, a minimum temperature of 100 K was reached when expanding air in one stage ($p_{in}/p_{out} \approx 5$). When the gas leaks out through nozzle *5* into receiver volume B, it produces additional work that is transformed into heat and removed in the end cooler *7* [36].

In some refrigeration systems with pulsation tubes, resonator chambers (Fig. 11), which produce a large momentum needed to change the direction of the flow, are used [37].

SEPARATION IN GAS FLOWS WHEN FLOWING AROUND BARRIERS

In 1940, Eckert and Weise [38] obtained interesting results on the energy separation when they measured the restoring coefficient *r* of temperature in the case of the cross-flow around the cylinder (Fig. 12).

As approaching rear frontal point, a significant decrease in the temperature restoring coefficient down to negative values was observed. This indicates that the temperature on the surface of a cylinder on the leeward side can be lower than the thermodynamic temperature of the incoming flow.

Parameter *S* and temperature restoring coefficient *r* are defined as

$$S = r - 1 = \frac{T_W^* - T_0^*}{T_0^* - T_0}, \quad r = \frac{T_W^* - T_0}{T_0^* - T_0},$$

where $T_0^* = T_0 + W_0^2/(2C_p)$ is the temperature of braking of the flow, T_W^* is the restoring temperature on the wall, T_0 is the static temperature in the flow of gas,

W_0 is the velocity of the flow, and C_p is the isobar heat capacity.

It is known that the temperature of a thermally isolated plate (the temperature of recovery on the wall) around which the flow of compressible gas flows differs in the general case from the temperature of braking of the flow (see [39]) and, for a gas with a constant heat capacity, it is determined by the formula

$$T_W^* = T_0 + r\frac{W_0^2}{2C_p} = T_0\left(1 + r\frac{k-1}{2}\mathrm{M}^2\right) = T^*\left[1-(1-r)\frac{k-1}{k+1}\lambda^2\right]. \quad (1)$$

In (1), k is the index of adiabate, $\mathrm{M} = W_0/a_s$ is the Mach number, a_s is the local velocity of sound, $\lambda = W_0/a_{cr}$ is the reduced velocity, and a_{cr} is the critical velocity.

Restoring coefficient of temperature shows the part of kinetic energy which is transferred into heat on the wall.

Later in [40], a significant impact of the generation of acoustic waves on energy separation was noted, especially in the range of resonance when separation of the vortexes from the surface of the cylinder occurs.

In 2004, Goldstein and Sanitjia [41] repeated the experiments [38]. The results of temperature restoration on the circumference of the cylinder are shown in Fig. 13.

Goldstein and Seol [42, 43] discovered considerable energy separation in the area of interaction of the flow with a barrier, which is explained by the generation of vortex structures in the area of interaction of the flow with the environment.

An interesting fact is the temperature stratification in a freely flowing flow of gas [42]. Figure 14 shows the results of the experiments performed. Parameter S was defined as $S = \left(T^* - T_0^*\right)/\left(T_0^* - T_0\right)$, where T^* is the current temperature of braking. The graphs show the transverse section of the flow at a distance of two calibers from the cut of the nozzle. The temperature of braking of the flow is equal to that of the ambient medium into which the flow moves out.

The thermodynamic temperature in the flow is lower than the temperature of the ambient medium, and the flow of heat is transferred to the nucleus of the flow due to heat conduction. However, the energy released due to the work of the forces of viscosity is transmitted in the opposite direction. These energy flows have a different intensity, which also varies along the length of the jet. As a result, one may conditionally distinguish the nucleus with a small increase in the temperature (due to the thermal conductivity), an annular area in which the temperature is slightly lower (the energy was spent on accelerating the outer layers of gas), and the annular area that consists of air cap-

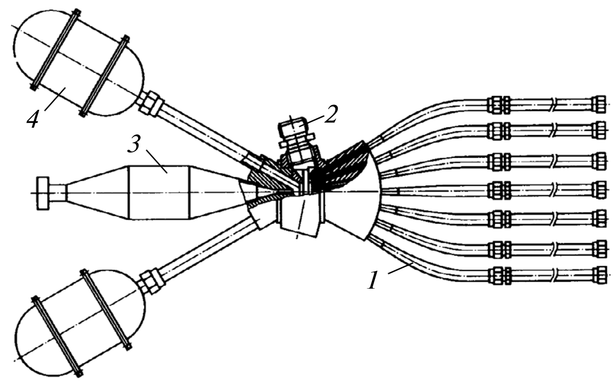

Fig. 11. Structural scheme of the pulsation tube [37]: *1*, receptor tubes; *2*, input transfer tube; *3*, output transfer tube; *4*, resonator chambers.

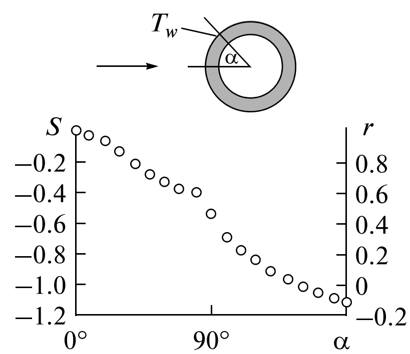

Fig. 12. Parameter S and coefficient r of temperature restoring on the surface of a cylinder around which the flow of gas flows [38].

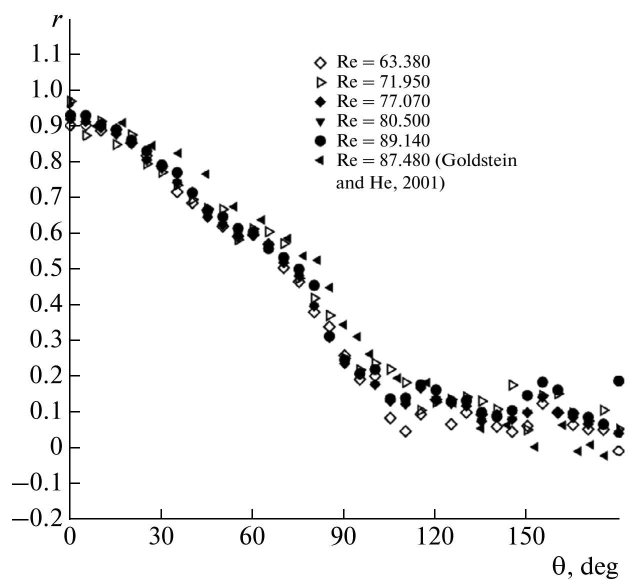

Fig. 13. Variation of the restoring coefficient of temperature over the circumference of the cylinder around which the flow of gas flows [41].

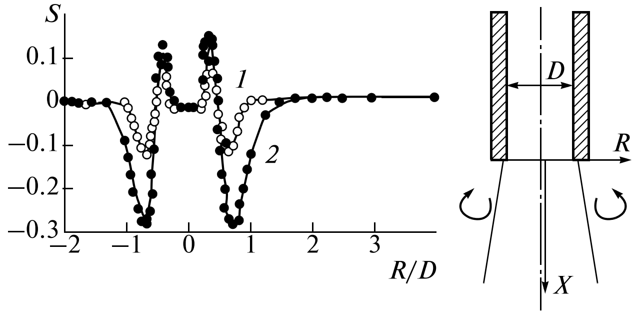

Fig. 14. Stratification in the flow of gas (at the distance $X/D = 2$) [42]: *1*, laminar flow; *2*, turbulent flow.

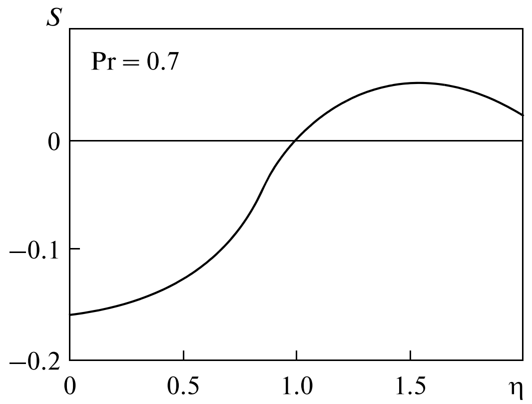

Fig. 15. Variation of parameter S in the boundary layer of a compressible gas on a thin plate [44].

tured by the jet which transfers additional energy to the air due to the viscosity forces.

For a turbulent flow, this effect is expressed more strongly than for laminar flow (Fig. 14).

In other known methods of energy separation, such a simple explanation of the reasons for the energy separation cannot be found.

SEPARATION IN THE FLOWS OF COMPRESSIBLE GAS

For the first time, a particular distribution diagram of the braking temperature in the boundary layer of compressible gas (air) in the flow around a thin plate was observed by Eckert and Drewitz [44] (Fig. 15). They used parameter S and a relative transverse coordinate of the form

$$S = \frac{T^* - T_0^*}{T_0^* - T_0}, \quad \eta = \frac{y}{2x}\sqrt{\mathrm{Re}},$$

where x, y are coordinates in the longitudinal and transverse directions relative to the plate; Re is the Reynolds number, which is determined by the parameters of the incoming flow and coordinate x. On a thermally isolated wall, $S = r - 1$.

Such a curved graph (epure) of the braking (restoring) temperature for the gas flow in the boundary layer of the wall is caused by the dependence on the Prandtl number which influences the form and curvature of the graph.

$$\mathrm{Pr} = \frac{\mu C_p}{\lambda},$$

where μ and λ are the kinematic viscosity and thermal conductivity coefficients.

As was shown in [45], at $\mathrm{Pr} \neq 1$ the braking temperature varies over the thickness of the boundary layer and the unevenness of this dependence is greater when the Prandtl number differs from unity more considerably. Figure 16 shows the dependence of the dimensionless temperature $\Theta = (T^* - T_0)\big/\left(T_0^* - T_0\right)$ on $\eta = y/\delta$ when the flow of compressible gas flows around a thermally isolated plate. In this case, y is the transverse coordinate relative to the flow, δ is the thickness of the boundary layer; $T^* = T_W^*$ and $\Theta = r$ on the thermally isolated wall.

As we can see from Fig. 16, at $\mathrm{Pr} > 1$ the dimensionless braking temperature near the wall is higher than in the other part of flow (the greater the Prandtl number, the greater the difference). This is due to the fact that near the wall the amount of heat released increases with increasing viscosity, whereas the outflow of heat from the boundary layer into the gas flow decreases upon a decrease in the heat conductivity (at the given temperature). Both of these effects lead to an increase in the gas temperature near the wall.

At $\mathrm{Pr} < 1$, the dimensionless braking temperature near the wall is lower than in the remaining flow. In both cases, the gas temperature in the boundary layer changes from the temperature of the thermally isolated plate up to the braking temperature in the main flow.

At $\mathrm{Pr} = 1$, the braking temperature is constant over the whole thickness of the boundary layer.

As was shown in [46, 47], for the Prandtl numbers in the range of 0.6–2.0, the restoring coefficient of temperature for the laminar boundary layer is described well by the expression

$$r = \sqrt{\mathrm{Pr}}. \tag{2}$$

Johnson & Rubesin [48] obtained an approximation to the exact solution of the equations of the laminar boundary layer of a compressible gas in the case of a flat plate. As follows from their work, the restoring coefficient of temperature is described well by Eq. (2) for the Prandtl numbers of 0.72–1.2, Mach numbers from 0 to 10, and index n in the dependences of viscosity and thermal conductivity vs. temperature in the range of 0.5–1.25 [39].

Analytical studies in the case of a subsonic flow were experimentally confirmed by Hilton [49] (measurement of the adiabatic temperature of the surface of a thin plate) and Eckert and Weise [38] in the cases where the flow flows around a flat plate or has a symmetry along the cylindrical probe.

Analyzing the impact of different factors on the restoring coefficient of temperature in the laminar flow mode was carried out in [50]. As was shown in this work, the Mach and Reynolds numbers, as well as the longitudinal gradient of pressure weakly affects the restoring coefficient of temperature which is equal to 0.85 ± 0.012 in a wide range of parameters. This agrees well with Eq. (2), according to which restoring coefficient r lies in the range of 0.83–0.85 at $\mathrm{Pr} = 0.69$–0.72.

As was shown by Ackerman [51], for a turbulent boundary layer at the Prandtl number from 0.5 to 2 and constant parameters of the flow, the restoring coefficient of temperature is determined by the equation

$$r = \sqrt[3]{\mathrm{Pr}}. \tag{3}$$

Seban [52] considered the case of a developed turbulent flow and obtained the expression relating the restoring coefficient with the Prandtl and Reynolds numbers

$$r = 1 - \left(4.71 - 4.11\frac{5\,\mathrm{Pr} - 7\,\mathrm{Pr}}{5\,\mathrm{Pr} + 12} - 0.601\,\mathrm{Pr}\right)\mathrm{Re}^{-0.2},$$

where Re is the Reynolds number determined by the parameters of external flow and the distance from the origin of the streamlined surface.

A simpler formula was obtained by Shirokov [53]:

$$r = 1 - 4.55(1 - \mathrm{Pr})\,\mathrm{Re}^{-0.2}.$$

In their works, Seban and Shirokov used the velocity distribution law in the boundary layer that was experimentally obtained by Nikuradze for turbulent flows in the tubes.

The authors of [38] obtained the value $r = 0.89$ for the restoring coefficient of temperature on a flat plate and a cylindrical probe in the case of a developed flow in a subsonic flow. In [54], the values $r = 0.888$ and $r = 0.891$ were obtained for the cone and paraboloid of revolution at $M = 2$.

Stein and Scherrer [55] studied the restoring coefficient for a cone with an angle of 10° and a combined body consisting of a cone with an angle of 40° and a cylinder. In the former case, they obtained $r = 0.882 \pm 0.08$ at $M = 1.97$ and $M = 3.77$, while in the latter case (for a combined body), they obtained $r = 0.885 \pm 0.011$ at $M = 3.1$ and $M = 3.77$, respectively.

In the case of a combined body consisting of a cone with an angle of 12° and a cylinder, the authors of [56] also studied the restoring coefficient by varying the Mach number from 3.0 to 6.3 and the attack angles up to 45°. In the case of a turbulent air flow, they obtained the value $r = 0.883$ at an attack angle of 0°.

As was shown by the analysis of the impact of different factors on the restoring coefficient of temperature r that was carried out for turbulent flows of compressible gas moving around smooth surfaces [50], almost all of the known dependences lead to a value close to (3) for the air flow at $\mathrm{Pr} = 0.69$–0.72.

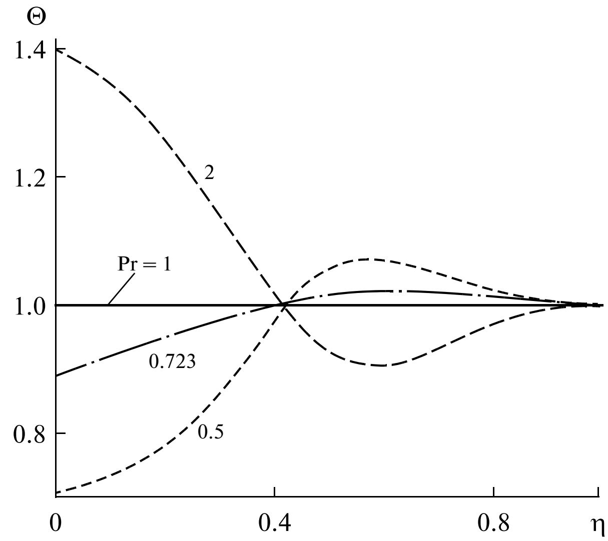

Fig. 16. Distribution of the dimensionless braking temperature for the flow of compressible gas flowing around a thermally isolated wall (for different values of the Prandtl number) [45].

A considerable deviation from Eq. (3) is only observed at a very low values of Prandtl's number [57–63]. One more situation which is not described by Eq. (3) occurs when the gas (air) flows around surfaces with a regular relief [67–74].

GAS DYNAMIC TEMPERATURE STRATIFICATION

It is known that the intensity of heat exchange between a gas and a streamlined surface is proportional to the temperature difference between the flow and the wall. In the case of a compressible gas flow, this value is equal to the difference between the equilibrium temperature of the streamlined surface (the temperature of restoring which acquires a thermally isolated wall with movement around it by the flow of compressible gas without radiative heat exchange) and the temperature of the wall.

A method for temperature stratification in a supersonic gas flow that is based on this peculiarity was proposed by Leontiev in 1996. He also proposed a device to implement this method, which is called the "Leontiev tube" [75, 76]. The principal scheme of this device is shown in Fig. 17. In contrast to the Ranque tube, the Leontiev tube was quickly recognized all over the world. At the 47th International Exhibition of Investment, Research, and New Technologies in Brussels (Eureka-98), the invention proposed by Leontiev received a silver medal with a diploma.

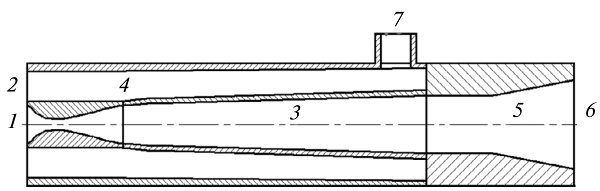

Fig. 17. The principal scheme of the device for gasdynamic stratification: *1*, entrance to supersonic channel; *2*, entrance to subsonic channel, *3*, supersonic channel; *4*, subsonic channel; *5*, diffuser; *6*, exit from supersonic channel; *7*, exit from subsonic channel.

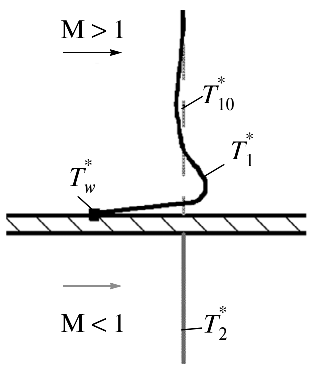

Fig. 18. Schematics of the gas dynamic stratification of temperature in the case Pr < 1.

Currently, several dozen articles have been published; in addition, reports at conferences of different levels, several patents [77–81], and six candidate's dissertations are related to it [82–87].

Figure 18 shows the principal scheme of gas dynamic temperature stratification. The two gaseous flows are separated by a wall. Shown on the top and bottom sides of the figure are supersonic and subsonic flows. Initially, the braking temperatures of the flows are assumed to be equal. A diagram corresponding to the braking temperature T_1^* is redistributed from the side of the supersonic flow. In this case, the mean integral braking temperature T_{10}^* in the boundary layer is constant, and the temperature difference appears (at $T_{10}^* = T_2^*$),

$$\Delta T = T_2^* - T_W^* = T^* - T^*\left[1-(1-r)\frac{k-1}{k+1}\lambda^2\right] = T^*(1-r)\frac{k-1}{k+1}\lambda^2. \quad (4)$$

One can see from (4) that an increase in the reduced velocity of a supersonic flow leads to a decrease in the restoring temperature and growth in the temperature difference between subsonic and supersonic flows of gas. However, for a flow in the channel with a given braking pressure at the input, growth in the velocity of a supersonic stream leads to a decrease in the thermal conductivity coefficient from the side of the supersonic flow. In other words, one can assume that an optimal velocity of the supersonic flow exists, at which the amount of heat transferred from the subsonic to the supersonic flow takes the maximum value.

The case in which flow an ideal gas flows around impermeable and permeable thin plates and the limiting estimates for a possible increase in the braking temperature of the flows were discussed in [88, 89].

A method for calculating temperature stratification based on the one-dimensional gas dynamic equations was proposed in [90, 91]. The numerical verification of this method was performed in [92, 93]. Further on, modification of this method and repeated verification were carried out on the basis of experimental data corresponding to the flows of natural gas and water [94, 95].

As was shown in [96], the strongest influence on the effectiveness of operation of the device for obtaining temperature stratification in the gas flows exerts the restoring coefficient of temperature. In the general case, it depends on the physical properties of the flow and its regime and the geometric shape and features of the surface around which the flow passes. When the gas flows around an impermeable body of a given shape, the restoring coefficient of temperature is a function of the five variables $r = f(\mathrm{Pr}, \mathrm{Pr}_T, k, \mathrm{Re}, \mathrm{M})$.

In [97], the authors studied the combined influence of the Prandtl number of the working body and reduced velocities in the case of subsonic and supersonic flows for impermeable and permeable thin plates. Furthermore, they considered the impact of the same parameters and the ratio of mass flow rates for supersonic and subsonic gas flows in the case of coaxial cylindrical channels.

In [98], the authors showed that for the gases with the Prandtl number $\mathrm{Pr} \approx 0.7$, a stronger temperature stratification effect is possible only upon diminishing the restoring coefficient of temperature and increasing the coefficient of thermal conductivity.

The influence of gas injection (exhaustion) was considered and the limiting estimations for the attainable level of separation were obtained in [99–105]. In [106–113], the authors analyzed the formation of a second phase and its impact on the effectiveness of energy separation. The influence of other factors on the effectiveness of energy separation was studied in [114–117].

Moreover, the cycles for heaters and refrigerators where the effect of gas dynamic separation is used are represented in [45, 96, 118–121].

CONCLUSIONS

The existing methods of gasdynamic energy separation are analyzed (vortex tube, ejector, separation in a gas flow and at flow around the walls, etc.). It has been shown that despite the nearly hundred year history and the relatively widespread usage of some methods of energy separation proposed, we still do not have a unified theory for explaining the corresponding processes. The empirical data and information on the most famous and popular models and methods allowing us to carry out the study and/or assessment of the effect of energy separation are given. It is shown that investigating dissipative effects allows one to draw an analogy between the different types of energy separation, creating a basis for understanding the physical nature of these processes.

Special attention was paid to the effect of gas dynamic temperature stratification proposed by Leontiev, which is based on using a difference between the amount of heat obtained from the work performed by the frictional forces and the amount of heat that can be transferred due to thermal conduction at a given temperature.

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