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# DETERMINATION OF THE CHARACTERISTICS OF GAS-DUST FLOW OF EXHAUST GASES FROM THE GAS-TURBINE PLANT TO THE RECONSTRUCTED BOILERS OF THE BEREZA STATE DISTRICT POWER STATION

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*Flow of the exhaust gases in intricately shaped gas ducts between the gas-turbine unit and the steam boiler in the steam-gas plant of the Bereza State District Power Station has been investigated. The distribution of the gasdynamic parameters of three-dimensional turbulent flow of exhaust gases in channels of different geometries has been studied. The influence of certain structural elements of the gas duct on the hydrodynamic characteristics of flow has been considered. The amplitude-frequency analysis of the natural oscillations of the gasdynamic parameters in different cross sections of the channel has been performed using fast Fourier transformation.*

The development of a modern economy is characterized by an intense growth in the consumption of energy resources. In developed countries, expenditures on fuel and energy account for over 15% of gross capital investment into the economy. At the same time, at present Belarus spends over two times more primary energy resources per unit of the national income than the USA. According to different estimates, the average efficiency of the use of fuel in mechanical engineering, metallurgy, and the construction-material industry is 20% to 30%. The high power intensity of industrial products leads to a reduction in the competitiveness of domestic manufacturers.

An efficient use of gas fuel, which remains the main type of fuel in the fuel and energy balance of the country at present and in the near future, makes it possible to substantially raise the efficiency of power plants. Whereas the burning of coal and fuel oil produces soot, ash, and sulfides, the use of which outside the boiler unit is difficult, the burning of natural gas yields, as a combustion product, a high-quality heat-transfer agent which can be used as a working medium in gas-turbine and gas-piston engines with subsequent utilization of heat in steam and water-heating boiler plants, and also in drying technologies.

The problem of efficient use of gas is multifaceted. It includes the minimization of loss in its burning due to its basically incomplete combustion, the high dilution of combustion products with excess air, and the high temperature of escaping gases. Of no less importance is the task of reducing heat loss and creating heat utilizers (utilizer-boiler, regenerator, recuperator, etc.). Here appears the need for a stepwise use of natural-gas-combustion products with successive extraction of heat in high-temperature units and then in medium- and low-temperature plants. Investigations have shown that clean natural-gas-combustion products may find application in catalytic reactors with the use of the latent heat of steam condensation (burning one cubic meter of gas produces two cubic meters of steam).

One of the most efficient ways of using natural gas currently is a steam and gas technology (binary cycle of power generation), in which high-temperature waste gases from a gas turbine are used for the production of steam in a special steam boiler and then the steam is used in a traditional scheme of a steam-turbine plant, thus allowing a rise of 15–20% in the efficiency of a power unit. Due to a high "electric" efficiency, modern steam-gas plants allow a substantial economy of fuel, have a high potential for heat supply, and are efficient for technological purposes, ventilation, air conditioning, etc. At the same time, there is a significant reduction in the harmful emissions into the atmosphere.

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One structural element of a steam-gas plant is gas ducts, through which exhaust gases from the gas turbine arrive at the steam boiler. Investigation of the flow of high-temperature combustion products in the channel of a gas duct of complex configuration is of significant interest in terms of determination of the technical characteristics of the gas flow and influence of the structural features of the gas duct on the static and dynamic parameters of the flow. In this work, we give the results of calculations and theoretical studies of hydrodynamic and amplitude-frequency parameters of the flow of exhaust gases in gas ducts. The main purpose is to study the influence of the gas-duct structure on the parameters of the flow and its acoustic characteristics. To this end, systematic calculations of the flow of high-temperature exhaust gases from the gas turbine to the boiler of the steam-turbine plant were carried out. The gas ducts to the steam boilers of the Bereza State District Power Station currently under construction were taken as the basis.

The mathematical model used for analysis is based on solution of a system of gasdynamic equations in a three-dimensional formulation and describes weakly compressible viscous turbulent flow. We have considered flows in channels having characteristic details, in particular, bends, turns, divergences and convergences, junctions connecting cylindrical and rectangular cross sections, etc. The hydrodynamic pattern of flow in typical cases has been obtained. The acoustic characteristics of flow have been studied. The eigenmodes of oscillations of the gas flow in a channel have been found. A relationship between the structural features of a gas duct and the acoustic signal generated by exhaust-gas flow has been obtained.

**Physical Properties of Exhaust Gases of the Gas-Turbine Plant.** The physical parameters of combustion products and the flow rate of the gas in a gas duct depend on the operating regime of a gas-turbine plant [1]. The pressure and temperature of the gases at exit from the gas-turbine engine and their flow rate vary with generator-terminal output and a number of other characteristics. The range of variation in the gas temperature at exit from the gas-turbine engine lies within 610–770 K. The flow rate of the gas $G_g$ at exit from the gas-turbine engine varies from 50 to 110 kg/sec. Numerical modeling of flow in gas ducts was carried out for the following regime of rated power of the gas-turbine plant: the flow rate of the gas on the section of the gas duct of the gas-turbine engine was 87.7 kg/sec and the temperature of the gas in the gas-turbine engine was 766 K. The excess pressure at entry into the gas-duct channels was 0.8 atm (absolute pressure 180 kPa). The thermophysical parameters of combustion products for the design operating regime of the gas-turbine plant have the following values: $G_g$ = 87.7 kg/sec, $G_g/S$ = 44.7 kg/(sec·m$^2$), $u_1$ = 56.6 m/sec, $T$ = 766 K, $P$ = 180,000 Pa, $\rho$ = 0.79 kg/m$^3$, $\mu$ = 27.893, $c$ = 551 m/sec, $\gamma$ = 1.324, $\eta$ = $3.64 \cdot 10^{-5}$ Pa·sec, $\lambda$ = 0.0586 W/(m·K), and $c_p$ = 1218 J/(kg·K). The component composition of exhaust gases is determined by the ratio of natural gas to air. A 15% excess air is assumed to ensure the complete combustion of natural gas; it is believed that the latter is entirely methane. In this case, the volume fractions of the components in combustion products are as follows: nitrogen 0.716, steam 0.166, carbon dioxide 0.084, argon 0.009, and oxygen 0.025. The calculations show that this composition is constant in these temperature and pressure intervals.

**Equations of Motion of Exhaust Gases.** Simple evaluations show that the velocity of sound in combustion products is $c = (\gamma RT/\mu)^{1/2} \approx 550$ m/sec under typical conditions, which is more than one order of magnitude higher than the mass velocity of the gas flow $u$ = 45 m/sec. Thus, we may assume, with a high degree of accuracy, that the gas is incompressible (or weakly compressible) under the present conditions. To clear up the question of what kind of gas flow in the channel — laminar or turbulent — we have, we evaluate the Re number. It is common knowledge that flow turns out to be laminar for low Re values and turbulent for high ones [2]. In tubes, laminar-to-turbulent transition occurs for Re $\sim 10^3$ [3]. The Reynolds number is related to the parameters of flow by the relation Re = $\rho u D/\eta$. It follows that we have Re $\sim 10^6$ in order of magnitude, i.e., flow is turbulent under the present conditions. In describing turbulent flows of a weakly compressible fluid [4], use is made of the continuity equation

$$\mathrm{div}\,(\rho \mathbf{u}) = 0\,, \tag{1}$$

the Navier–Stokes equation

$$\frac{\partial \mathbf{u}}{\partial t} + (\mathbf{u}\nabla)\,\mathbf{u} = -\frac{1}{\rho}\nabla P + \frac{1}{\rho}\nabla\left[(\eta + \eta_t)\,(\nabla \mathbf{u} + (\nabla \mathbf{u})^{\mathrm{tr}})\right] \tag{2}$$

and the energy equation (here it is written for the enthalpy $H = E + P/\rho$)

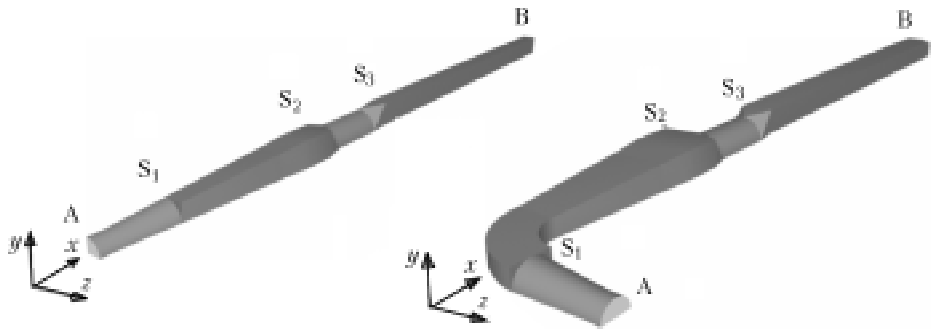


Fig. 1. Geometry of a straight channel and a channel with a bend.

$$\frac{\partial(\rho H)}{\partial t} + \nabla(\rho \mathbf{u} H) = \nabla\left[\left(\frac{\lambda}{c_p} + \eta_t\right)\nabla H\right]. \tag{3}$$

The turbulent-fluid model is based on the standard $k$–$\varepsilon$ model in which the turbulent viscosity $\eta_t$ is expressed by the quantities $k$ and $\varepsilon$ in the following manner:

$$\eta_t = C_\mu \rho \frac{k^2}{\varepsilon}, \tag{4}$$

and equations for the kinetic energy of turbulent pulsations $k$ and the rate of its dissipation $\varepsilon$ have the form

$$\frac{\partial(\rho k)}{\partial t} + \nabla(\rho \mathbf{u} k) = \nabla\left[\left(\eta + \frac{\eta_t}{\sigma_k}\right)\nabla k\right] + G - \rho\varepsilon, \tag{5}$$

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \nabla(\rho \mathbf{u} \varepsilon) = \nabla\left[\left(\eta + \frac{\eta_t}{\sigma_\varepsilon}\right)\nabla \varepsilon\right] + \frac{\varepsilon}{k}(C_1 G - C_2 \rho\varepsilon). \tag{6}$$

Here we obtain

$$G = \eta_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right). \tag{7}$$

The values of the parameters in the $k$–$\varepsilon$ model are as follows [4]: $\sigma_k = 1$, $\sigma_\varepsilon = 1.3$, $C_\mu = 0.09$, $C_1 = 1.44$, and $C_2 = 1.92$. The values of the velocity $\mathbf{u}$, pressure $P$, and enthalpy $H$ and their derivatives in Eqs. (1)–(6) are average values obtained in averaging over time (by turbulent pulsations) [5].

Boundary conditions at the channel inlet are specified by the values of the density $\rho_1$, the pressure $P_1$, and the velocity of the inflowing gas $u_1$; the velocity is assumed to be directed along the gas-duct axis. Sometimes, it is convenient to prescribe the gas flow rate $G_g$, which is related to the velocity: $G_g = \rho_1 u_1 S$. Free-outflow conditions are realized at the channel outlet: the derivative of the axial velocity component, which is normal to the outflow surface, is assumed to be zero ($\partial u_z/\partial z = 0$) and the pressure is prescribed. On the lateral channel walls, the velocity component normal to them is equal to zero (nonflow condition). The tangential component of the velocity vector, which is directed along the wall, is zero (adhesion condition) and varies logarithmically in its vicinity. For the turbulent energy at the channel inlet we have $k = (bu_1)^2/2$, where $b$ is the semiempirical constant taking on values of $b < 0.03$, $0.03 < b < 0.05$, and $0.05 < c < 0.1$ for the low, moderate, and high turbulization respectively depending on the degree of turbulization of the flow. The rate of turbulent energy dissipation is prescribed in terms of the turbulence scale $L'$ in the following form: $\varepsilon = C_\mu k^{3/2} L'$, where $L' = b' D_1$, and the semiempirical parameter $b'$ takes on values of $b' < 0.03$, $0.03 < b' < 0.1$, and $0.1 < b' < 0.2$ with the same conditions. Zero flows for $k$ and $\varepsilon$ are prescribed at the

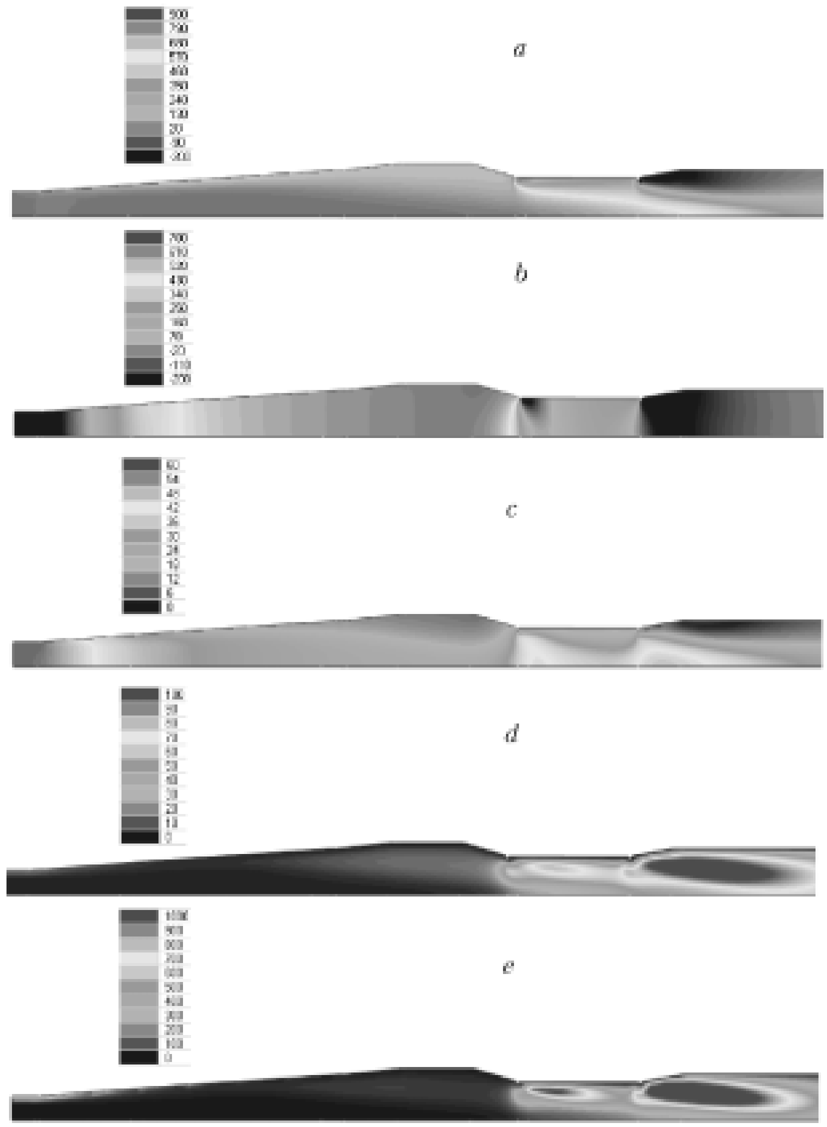


Fig. 2. Distribution of the stationary parameters of flow in a straight channel: a) total pressure $\Delta P_f$; b) gaskinetic pressure $\Delta P$; c) velocity modulus $u$; d) turbulent energy $k$; e) turbulent dissipation $\varepsilon$.

tube outlet. On the lateral channel walls, we compute $k$ and $\varepsilon$ values from the velocity $u_*$ in the vicinity of the walls from the expressions $k = u_*^2/\sqrt{C_\mu}$ and $\varepsilon = u_*^3/(\kappa D)$, where $\kappa \approx 1$ is the dimensionless factor.

**Calculation Results.** Equations (1)–(6) describing flow of a viscous turbulent gas in the gas-duct channel were solved numerically using the FlowVision program complex [4]. The calculations were performed for a straight tube and tube with a rotation of the flow by $90^{\circ}$. The channel geometry is given in Fig. 1; the shape and dimensions of the initial portion of the gas duct correspond to the gas ducts of the Bereza State District Power Station. The initial-portion length is 50.3 m in the first case and 55 m in the second case (along the axis of symmetry). The step of the computational grid along the channel axis is no larger than 0.1 m. We used the conditions of symmetry of flow, which enabled us to calculate the flow in one-quarter of the tube cross section in the first variant and in half the cross section in the second variant. The roughness of the walls was taken to be 1 μm. The total-pressure signal for subsequent Fourier analysis was recorded in several cross sections: at entry (A), at exit (B), and in the cross sections of junction of the channels of rectangular and circular shapes — $S_1$, $S_2$, and $S_3$.

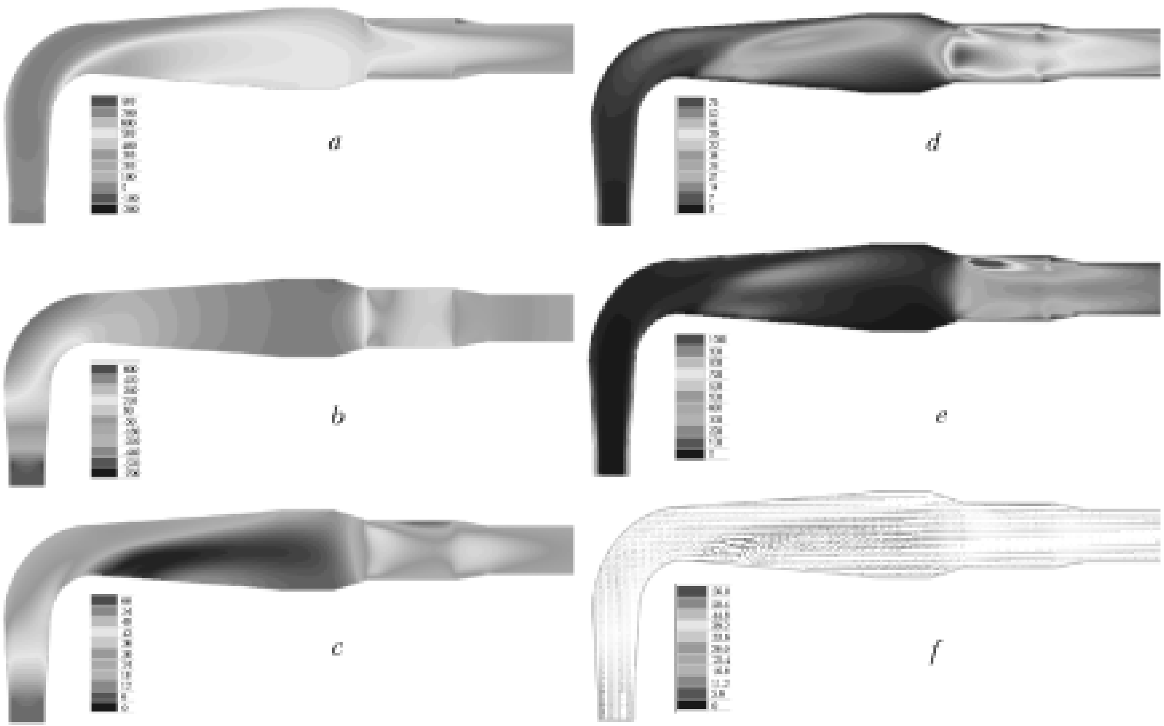

Fig. 3. Distribution of the characteristics of flow in a channel with a bend: a) total pressure $\Delta P_f$; b) gaskinetic pressure $\Delta P$; c) velocity modulus $u$; d) turbulent energy $k$; e) turbulent dissipation $\varepsilon$; f) vector of the velocity $\mathbf{u}$.

Let us consider briefly the calculation results. The total pressure in the case of isentropic flow of a weakly compressible (incompressible) fluid can be represented in the form of the sum $P_f = P + \rho u^2/2$. It is convenient to analyze the excess of the total pressure over a certain reference value $P_{ref}$, which is equal to $1.8 \cdot 10^5$ Pa in this case and corresponds to the pressure at entry into the gas-duct channel. Thus, below, by the total pressure we mean the difference $\Delta P_f = P_f - P_{ref}$ and by the gaskinetic pressure the quantity $\Delta P = P - P_{ref}$. In the stationary regime of flow, the total inlet pressure was $\Delta P_f$ = 819 Pa and the pressure was $\Delta P$ = –451 Pa in the case of the straight tube. In the case of the bent tube, the total pressure was $\Delta P_f$ = 708 Pa and the pressure was $\Delta P$ = –563 Pa. The distribution of the parameters of flow in the straight channel in the stationary regime is shown in Fig. 2 (the fields are given in the cross-sectional plane $xy$ passing through the axis of symmetry (see Fig. 1)). It can be seen that in the initial (divergent) portion of the gas duct, the flow velocity drops, whereas the pressure grows. The gas flow is retarded, ensuring a constant flow rate of the gas in different cross sections of the channel. In the regions of junction $S_3$ and $S_4$, the flow velocity sharply increases. A considerable turbulization of the gas flow is observed — turbulent energy and dissipation grow. Also, a fairly high inhomogeneity of flow in the neighborhood of these junctions is noteworthy.

Figure 3 shows stationary flow in the channel with a rotation of the flow (the distributions of the quantities in the plane $xz$ of symmetry of the problems are presented in this case). As is clear from this figure, pressure substantially grows with the turn of the flow as a result of its retardation on the exterior wall. After the turn, we observe a strong inhomogeneity of the gas flow over the tube cross section: the gas flow is mainly localized at the exterior channel wall (it is at the top in the figure). This inhomogeneity of the flow is observed up to the rectangular channel cross section changing to a circular one. A vortex formation with relatively low flow velocities develops at the interior channel wall (it is lower in the figure). It is seen from a comparison of Figs. 2 and 3 that in the straight channel, the bulk of the gas moves along the axis of symmetry, whereas in the channel with a bend, the flow is localized along the wall on which it is retarded with the turn of flow up to the rectangular channel cross section changing to a circular one.

**Natural Oscillations of the Gas Flow in the Gas-Duct Channel.** An amplitude-frequency analysis of oscillations was carried out using a fast Fourier transformation (FFT analysis) [6] of the time dependence of the total pressure of the gas in different cross sections of the tube; the pressure was averaged over the cross-sectional area. The

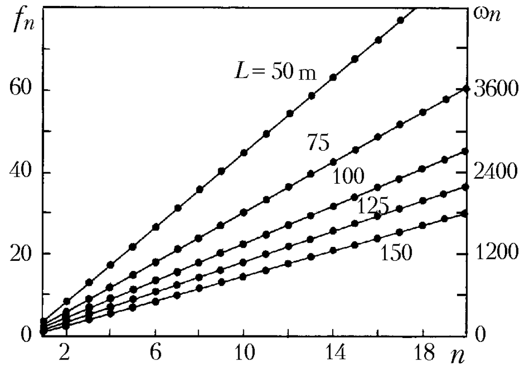


Fig. 4. Frequency spectrum of oscillations of the gas flow. $f_n$, Hz; $\omega_n$, min$^{-1}$.

calculations were carried out with a constant time step. The number of time layers (points of recording of the signal) for different variants was selected so that the error in determining the eigenfrequency of oscillations was no higher than $\pm 0.1$ Hz. The spectrum of natural oscillations was excited by different methods. We considered the flow reaching the stationary regime after discontinuity decay: a stationary flow of an inflowing gas was prescribed at the gas-duct inlet, and the gas in the channel was quiescent with a uniform distribution of all gasdynamic parameters. Furthermore, periodic disturbances caused by the fluctuations of the flow rate of the gas at entry into the channel were imposed on the stationary flow in the gas duct. Their amplitude was no larger than 5% of the flow rate, and the frequency varied within wide limits. The analysis carried out has shown that the spectrum of natural oscillations of channel flows obeys the law

$$f_n = \left(n - \frac{1}{2}\right)\frac{c}{2\alpha L}, \quad n = 1, 2, 3 \ldots,$$

where $\alpha \approx 1.18$ is the correction factor which differentiates the case in question from that of oscillations of the gas in a tube with one open end and the other closed end [7]. In actual practice, the gas freely flows out of the tube (the outlet boundary condition is that of free outflow) and we have an effective increase of 1.18 times in the characteristic length ($L_{\text{eff}} = 1.18L$). Variation of the driving frequency (inlet mass velocity) in a wide range has shown the stability of the spectrum of natural oscillations. The frequency of natural oscillations as a function of the harmonic No. for different lengths of the gas duct is presented in Fig. 4. It follows from the figure that natural oscillations with a large $n$ for the characteristic gas-duct length $L \approx 100$ m fall in the range of frequencies $\omega$ = 1500–3000 rpm (this range is the most hazardous from the viewpoint of the occurrence of resonance phenomena associated with the operation of a gas-turbine plant). An analysis shows that their amplitudes are several orders of magnitude lower than the amplitudes of low-frequency oscillations. The structural details of the gas duct, in particular, the presence of divergences, convergences, junctions between cross sections of different profile, and rotations, lead to a slight deformation of the spectrum of natural oscillations in the region of high harmonics. Their amplitude is virtually constant. It is significant that the spectrum of natural longitudinal oscillations of the gas-duct structure lies in the kilocycle range because of the high velocity of sound in metal ($c_m \approx 5$ km/sec), and numerous heavy supports and fixing arms shift it to the high-frequency region even more. The frequencies of transverse oscillations of the gas duct lie in the range of tens and hundreds of kilocycles. Sonic waves with such a short wavelength (less than 1 cm) decay very fast owing to the processes of viscosity and heat conduction. Thus, the resonance interaction between the oscillations of the gas flow and the natural oscillations of the gas duct is impossible.

## CONCLUSIONS

1. The thermophysical properties of exhaust gases under typical conditions of operation of a gas-turbine plant have been determined. They may be considered as being constant throughout the gas-duct length in view of the relatively slight change in the temperature.

2. The distributions of the gasdynamic parameters (velocity, pressure, density, and turbulent energy and its dissipation) of three-dimensional stationary turbulent flow of exhaust gases in channels of different geometries have been studied. Knowledge of these characteristics makes it possible to determine the hydraulic resistance of different elements of the gas duct [8], i.e., the difference of the cross section-average pressures at entry and exit from a given unit, which ensures a time-constant flow rate of the gas.

3. The gasdynamic features of flow, determined by different elements of the gas ducts (structural joints, turns, change in the cross-sectional profile of the channel, and its other details) have been studied. It has been shown that a sharp inhomogeneity of the flow over the channel cross section occurs on these transition portions. Here we observe a significant increase in the transverse velocity component; vortices and stagnation zones can be formed, too. Flow turbulization leading to a growth in hydraulic resistance because of the dissipation of turbulent energy is sharply enhanced at these sites. We particularly emphasize that considerable transverse pressure gradients occur in these regions of the gas duct.

4. The frequency spectrum of natural oscillations of the gasdynamic parameters in the gas-duct channel has been found. The frequencies of the oscillations correspond to the frequencies of natural oscillations of the gas in a tube open at one end and closed at the other with a correction factor $\alpha = 1.18$. An analysis of the spectrum and amplitudes of natural oscillations of the flow in straight and bent gas-duct channels shows that the bend of the gas duct scarcely affects these characteristics.

5. A comparative analysis of the gas flow in the straight channel and in the channel with a bend shows that the most significant difference is that in the latter case we have a fairly large transverse load on the gas duct due to the turn of the gas flow. This load depends on the velocity of the gas and the steepness of the bend. As the calculations show, under typical conditions, static loads attain values of $\geq 120$ kgf/m$^2$ with a bend of 90$^{\circ}$. For the characteristic transverse dimensions, the force acting on the gas-duct channel is more than 1000 kgf. Heavy supports and reliable fixing of the gas ducts to them are necessary for neutralizing such actions.

## NOTATION

$c$, velocity of sound, m/sec; $c_p$, specific heat at constant pressure, J/(kg·K); $D$, diameter of the gas-duct channel, m; $D_1$, inlet diameter of the gas duct, m; $E$, specific internal energy, J/kg; $f_n$, frequency of natural oscillations of the gas in the channel, Hz; $G_g$, flow rate of the gas, kg/sec; $H$, enthalpy, J/kg; $k$, turbulent energy, J/kg; $L$, gas-duct length, m; $L'$, turbulence scale, m; $n$, No. of oscillation mode; $P$, pressure, Pa; $P_f$, total pressure, Pa; $P_{ref}$, reference pressure, Pa; $\Delta P_f$, excess of the total pressure over its reference value, Pa; $\Delta P$, excess of the gaskinetic pressure over the reference pressure, Pa; $R$, universal gas constant; Re, Reynolds number; $S$, area of the entrance cross section of the gas duct, m$^2$; $T$, temperature, K; $t$, time, sec; $x$, $y$, $z$, coordinates, m; $\mathbf{u}$, velocity of the gas flow, m/sec; $\gamma$, adiabatic exponent; $\varepsilon$, rate of dissipation of turbulent energy, J/(kg·sec); $\mu$, molecular weight; $\eta$, coefficient of viscosity, Pa·sec; $\eta_t$, coefficient of turbulent viscosity, Pa·sec; $\lambda$, thermal conductivity, W/(m·K); $\rho$, density, kg/m$^3$; $\sigma_k$, $\sigma_\varepsilon$, $C_\mu$, $C_1$, $C_2$, and $\kappa$, dimensionless parameters of the $k$–$\varepsilon$ turbulence model; $\omega$, rotational frequency, rpm. Subscripts: g, gas; eff, efficiency; f, total (full); ref, reference; tr, transposition; t, turbulent; m, metal.

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