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STUDY OF BURNING STABILITY IN LOW EMISSION GAS TURBINE COMBUSTOR

Introduction: The paper is devoted to investigation of gaseous fuel burning stability in low emission combustor of stationary gas turbine engine (GTE). The mathematical model of unsteady processes in GTE low emission combustor is developed. A methodology of numerical experiment concerning stability of gaseous fuel burning in low emission combustor with using complex of computational fluid dynamics (CFD) is proposed. Theoretical studies of non-stationary processes in a low emission gas turbine combustor were performed using the Large Eddy Simulation (LES) model of turbulence. The results of the model verification confirm its validity for the wide spectrum of fuel nozzle design. The performed 3D calculations allowed defining main pulsating features of a low emission gas turbine combustor. For the entire considered frequency range, a pronounced frequency of 189 Hz is traced, which is caused by combustion processes, as evidenced by the spectra of temperature fluctuations and the mass concentration of fuel. The calculated local mean square pulsations of the static pressure inside the combustion line can reach maximum values of about 11.5 kPa. The obtained results and recommendations can be used for modeling of unsteady processes in low emission combustor and stability improvement of GTE combustor with partially premixed lean fuel-air mixtures.

Keywords: gas turbine, combustion, non-stationary process, fuel-air mixture, stability.

1. Introduction

There is significant success in designing, constructing and adjusting of gas turbine combustors. One of the key issues in development of a low emission combustor is providing operation stability which practical solution requires serious material expenses. Despite the fact that there have been a lot of theoretical and experimental studies and there is significant experience in reducing pressure oscillation the nature of pulsation processes occurrence has not been studied properly [2, 6, 10].

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Oscillatory combustion is not separate chaotic oscillations of pressure, velocity and temperature of the flow which always accompany fuel combustion, but regular oscillations with a high amplitude and specific frequency. They start due to some reasons and then are preserved at the expense of a regular auto-oscillatory process [2].

A gas turbine combustor provides several sources of oscillatory combustion occurrence. The main reason of low frequency pulsations is delay of a chemical oxidation reaction after fuel supply to the combustor which leads to periodic change of fuel supply via nozzles and, thus, to periodic heat release. The reasons for high-frequency pulsation combustion are acoustic oscillations of pressure and other parameters of working medium with additional sources which are connected with the change of the sound velocity at unsteady heat release, turbulent pulsations of the flow and density oscillation. The combustors with lean fuel-air mixture are especially predisposed for oscillatory combustion. There the boundaries of concentration limits of burning may be reached and combustion stability may be disturbed due to the lack of heat supply for fresh mixtures inflammation. The reasons for oscillatory combustion occurrence in low emission combustors may be [8]:

– high increase of the burning time of "lean" mixture;

– proximity to the boundary of the concentration limit of steady combustion;

– reverse influence of the pressure pulsations in the combustor on fuel flow, thus, on the oscillations of the air excess coefficient α and the temperature T in the combustion zone and the heat release.

A significant source of pressure disturbance in the combustion liner is turbulent oscillations. Though this oscillation source exists regardless heat release, it significantly influences the velocity of flame distribution. So, in low emission combustors with lean fuel-air mixtures the turbulent processes may play a significant role in burning stability disturbance.

Unsteady combustion is characterized by various oscillation frequencies: < 20-50 Hz (low frequency), 130-500 Hz (medium frequency), and >1000 Hz (high frequency). Low frequency oscillations in a range of 20-50 Hz are connected with dynamic instability of a GTE as a compressor system. Thus, a simple change of combustion liner design cannot influence the pressure oscillations in this range.

The oscillations in a range of 130-500 Hz are the most dangerous; there can be coincidence of the frequencies of burning oscillations in a combustion liner with the frequency of compressor rotation and coincidence of the self frequency of the combustion liner with the burning oscillation frequency. It can lead to resonance processes and destruction of combustion liners and combustor elements.

The main source of oscillations in this range is heat release inequality. The excited oscillations transform into standing waves inside the combustor at comparability of its length to the one of the combustion liner [9].

Applying to the gas turbine combustor, it is convenient to divide oscillations into low frequency and high frequency ones depending on the correlation of the oscillation time τ_o and the residence time τ_r of the mixture in the working volume. If $\tau_o > \tau_r$, i. e. gas oscillates in the combustor as an integral unit, the oscillations are low frequency. At $\tau_o << \tau_r$ separate masses of the total volume are in significantly different phases of oscillatory motion, and there is distribution of the waves inside the combustor, and the oscillations will be high frequency. At measurement of the parameters with the frequency of f < 500 Hz (τ_o>>2 ms and $\tau_r \approx$ 1 ms) there are obvious low frequency oscillations [2]. In most cases the frequency of the oscillations is defined by acoustic properties of the combustor itself, the properties of air and fuel supply lines and other elements as the oscillated medium are elastic masses of gas. Gas oscillates not only in the area close to the combustion zone, but also in the adjoining air and gas lines [2, 22].

Pulsation modes with low frequencies but with high oscillation amplitudes lead to oscillation of the flow, length, and flame brightness up to flame blow off accompanied by low-tone sounds, may cause changes in the GTE power and even structure destruction.

High frequency oscillations with lower amplitudes the sounds can have adverse consequences also. Pressure oscillations sharply increase average hydraulic resistance of the combustor, controlling even redistribution of the flows in some channels. Together with velocity oscillations they destroy the flow in the near-wall zones (deforming the barrage flows of cooling air) and worsen walls cooling of the combustion liner. Usually at the oscillatory burning modes the walls temperature increases by 100-200 degrees. Under the conditions of high frequency oscillations the cases of destruction of some parts and assemblies are specific: combustion liners, shields and other elements of the combustor as well as turbine blades and the engine in general [22].

The increase of the mixing time of fuel-air mixture leads to the growth of the delay time between fuel supply to the burning device and its combustion in the flame front. The presence of the delay time leads to matching of heat emission with pressure pulsations in the combustor in the phase. Time matching between fuel burning in the flame front and pressure pulsation according to Rayleigh criterion leads to auto-oscillation excitement in the combustor at rather small loss of acoustic energy. It should be noted that several frequencies of auto-oscillations may be excited in the combustor, so at rather high delay time detuning from one frequency leads to excitement of another one. This excitement mechanism is possible only in improperly mixed mixture as at high quality of mixing the generation of acoustic energy in the flame front from pulsations of fuel concentration cannot exceed its dissipation. When the mixture is completely mixed on the burning device outlet, it allows suppressing auto-oscillations caused by oscillations of fuel concentration, but auto-oscillation excitement may be caused by another mechanism [2]. This mechanism of auto-oscillation excitement is connected with a low flame temperature (the flame temperature is lower than 1,800 K). It should be noted that the selection of this temperature is connected with prevention of formation of above-standard concentrations of nitrogen oxides NO_x. The low flame temperature means significant reduction of the velocity of flame distribution or, in other words, increase of the influence of the content of fresh fuel-air mixture on the fuel burning completeness. When the maximum allowed load on the flame front is exceeded, local flame front extinction is possible. The most powerful mechanism which is able to increase the velocity of involvement of fresh fuel-air mixture to the flame front is formation of large-scale eddy structures. On the edge of the burning device large-scale eddy structures are able to capture fresh fuel-air mixture and transit it via the flame front without combustion. After the decomposition of large-scale eddy structures there is fast burning out of fuel-air mixture which is accompanied by high heat release. Coincidence of its heat release in the phase with pressure oscillations leads to auto-oscillations occurrence. Synchronization of separation of large-scale eddy structures from the burner edge with pressure pulsations takes place via the field of acoustic velocity oscillation synchronized with pressure pulsations [11].

In the cases, when a part of fuel is supplied directly to the combustion zone to support the flame of preliminary prepared fuel-air mixture, occurrence of auto-oscillations is possible due to low volume flow rate of this fuel. A low velocity of the fuel jet simplifies capturing over-rich fuel-air mixture with large-scale eddy structures and their transfer via the flame front and the zone of reverse flows to the decomposition zone where rich fuel-air mixture is burned out [11].

The mechanisms of reverse connected caused by large-scale eddy structures are usually hardly controlled, so the main methods of auto-oscillation suppression in low emission combustors are development of passive resonance circuits which absorb acoustic energy and reduction of the mixing time of air and fuel.

Oscillatory combustion may lead to damage of pipe lines, fittings of the front device, cracks in the combustion liners, etc. The resonance oscillations of the combustor elements caused by oscillations may lead to fatigue cracks in combustion liners and turbine blades.

In combustion liners the temperature stresses may be summed with stresses from gas pressure oscillations and reach dangerous values which lead to destructions. Occurrence of local temperature stresses causes warping or cracking of the combustion liner walls [4]. There are the following serious defects of the combustor which occur at oscillatory combustion: scorching of the burning device, destruction of the combustion liner shells, elements of body fittings, packing of combustion liners, cracks and deformations of thermal nature, cracks in welds, burning out of combustion liner walls [1, 10, 17].

The modes of unsteady combustion may occur in various combustor designs working on various fuels. They are accompanied by unpleasant noises, may cause oscillations of separate details and systems (control, adjustment), mechanical and thermal destructions of the assemblies, oscillations of GTE power and engine destruction [2]. Thus, avoiding these modes is an urgent issue.

The main objectives of the present work are theoretical investigations of burning stability in low emission combustor of 25 MW gas turbine engine.

2. Directions of stability increase in gas turbine combustors

Control of burning processes stability in the combustor may be realized by passive and active methods. The passive methods are the following: installation of anti-oscillation walls, ablating coverings and resonance absorbers in the combustor, using the nozzles of various forms and dimensions, various additives for the fuel and redistribution of fuel and air flows in the combustion liner, change of the velocities of gases flow in the flow part [2, 22]. Despite of the simplicity and low cost of the passive control methods, their main disadvantage is low reliability of suppressing unsteady burning. Moreover, the passive methods are efficient only for some combustor designs and mode operating parameters.

The active methods of oscillations suppression suppose dynamic automatic regulation of fuel flow in the channels during engine operation. These methods suppose additional high-cost systems for the combustor and complicated regulation algorithms.

In many cases, there is an anti-vibration shield along the wall inside the combustors to suppress high-frequency oscillations. It has a form of a fluted perforated structure. The shield is an acoustic resonance absorber adjusted for suppressing oscillations of definite frequencies. At low-frequency vibration combustion the shield efficiency is low, so suppression of low frequency oscillations is performed at experimental adjustment of the combustor and is quite a difficult task. Controlling the vibration combustion is performed by changing acoustic volume, shifting of the maximum heat release zone, changing of fuel distribution in the cross section of the combustor, changing of the form and separation of the stabilizers, changing of the gas flow velocity, etc. [3, 22].

Using a combustion liner of larger dimensions for two-fuel Siemens combustor allowed reducing the boundaries of lean flame blow-off and providing steady combustion at the rate of additional pilot fuel >3% [6].

Burning stability in the ring General Electric combustor is provided by installation of quarter-wave resonators. The front device of the combustor consists of three ring rows of the burners, separated by ring baffles. An external and medium ring zones have 30 burners each; the internal ring zone has 15 burners [6].

A preliminary mixing technology is used in the low emission Rolls Royce combustor with pulsation suppression system. Fuel and air are preliminary mixed with obtaining of homogeneous mixture at the combustor inlet which provides conditions for low emissions burning, however, leads to pulsations growth. They are formed at lean fuel-air mixture inside the combustor when flame oscillations resonate with the frequency of self-oscillations of surrounding components. A special system of passive damping is developed to mitigate pulsations [6].

Each burner of GTE GT-10 developed by ABB consists of two cone shells which form a cone with the height of 400 mm and the baseline diameter of 150 mm. There is nozzle along the axis near its top to which fuel can be supplied liquid or natural gas for diffusion combustion. As a result of flow swirling, there is an eddy zone which provides proper inflammation of fuel-air mixture and stabilization of the flame front in all the operation modes [5].

Additives of high-reaction substances to the combustion zone, for example hydrogen or oxygen, significantly improve combustion stability. This method is attractive due to its simplicity of the design solution, though at its use there is the necessity to intensify spraying and mixture formation in startup modes and there are also problems of the high-reaction gas source. Using of chemically active particles which form in the hydrocarbon fuel oxidation reactions as additives to the main mixture leads to expansion of the combustion stabilization limits and stability [12, 14, 18, 19].

The system of plasma-chemical combustion intensification which is aimed at flame stabilization in GTE combustion devices consists of a plasma- chemical element and its power source [7, 16, 18, 19]. At fuel supply to the plasma air jet there are reactions which define high output of active components (radicals, atoms, intermediate compounds). These components quickly diffuse from the

zone of direct plasma and fuel part contact to the zone of the primary fuel-air mixture and provide intensification of its combustion, decreasing the activation energy, increasing the velocity of fuel combustion in the turbulent flow [7, 18].
Thus, using similar systems based on heat and kinetic affecting on the combustion of fuel-air mixture will allow expanding of the limits of flame distribution of lean fuel-air mixtures and increasing stability of operation processes in low emission combustors.

3. Features of mathematical modeling of unsteady processes in gas turbine combustors

In low emission gas turbine combustors which operate on gaseous fuel, fuel-air mixture burning is characterized by kinetics of chemical reactions and mixture formation physical processes. Simulation of unsteady chemically reacting turbulent flows in the combustors includes subsimulations of mixture formation, combustion of gaseous fuel and turbulent transfer. The main task of combustion numerical simulation in such combustors is joint simulation of gas dynamics and chemical kinetics. At selection of mathematical simulation of a low emission gas turbine combustor the proper turbulence simulations are selected firstly as well as the ones of gaseous fuel and air mixing.
The mathematical simulations are represented as the systems of differential and algebraic equations which solution is agreed with the parameters of the elements of a studied device or a process. The simulation is considered as description of the existing dependences between design and operation combustor parameters and is based on the solution of the equation systems describing diffusion or convective transfer of the reacting mixture components.
Mathematical simulation of a low emission combustor operating on gaseous fuel includes the following equations [13, 15, 20, 21]: continuity, momentum conservation, energy conservation, mixture chemical components transfer, harmful component emission formation and decomposition, etc.
To calculate unsteady combustion processes in low emission gas turbine combustors the FR/ED simulation is used (Finite-Rate/Eddy-Dissipation) – a combination of the simulation considering the finite rates of chemical reactions with the simulation of eddy dissipation. The concentrations of the *i*-th chemical component of the mixture are defined considering rates of the direct and reverse reactions [14, 15]:

$$R_i = M_{\varpi,i} \sum_{r=1}^{N_R} R_{i,r} \,, \tag{1}$$

where:

R_i – rate of forming the *i*-th component as a result of a chemical reaction,

$M_{\varpi,i}$ – molecular weight of the *i*-th component,

$R_{i,r}$ – Arrhenius molar rate of formation / decomposition of the *i*-th component in the reaction *r*.

The rate constant for the direct reaction *r* is calculated using the Arrhenius law:

$$k_{f,r} = A_r T^{\beta_r} e^{-E_r/RT} \,, \tag{2}$$

where:

A_r – pre-exponential factor,

β_r – temperature exponent,

E_r – activation energy for the reaction *r*,

R – universal gas constant.

If the reaction is reversible, then the reverse rate constant is found from the following expression:

$$k_{b,r} = \frac{k_{f,r}}{K_r} \,, \tag{3}$$

where:

K_r – equilibrium constant for reaction *r*.

Molar concentration of the formed and decomposed *i*-th components in the reaction *r* is defined via the formula:

$$R_{i,r} = \Gamma\left(v''_{i,r} - v'_{i,r}\right) \cdot \left(k_{f,r} \prod_{j=1}^{N_r} \left[C_{j,r}\right]^{\eta'_{j,r}} - k_{b,r} \prod_{j=1}^{N_r} \left[C_{j,r}\right]^{\eta''_{j,r}} \right), \tag{4}$$

where:

N_r – amount of chemical components in the chemical reaction *r*,

$C_{j,r}$ – molar concentration of the *j*-th reagent and product in the reaction *r*,

$\eta'_{j,r}$ – direct reaction exponent for the *j*-th reagent and the reaction product *r*,

$\eta''_{j,r}$ – reverse reaction exponent for the *j*-th reagent and the reaction product *r*,

Γ – effect of third substances on the reaction rate.

A special feature of the burning processes in the turbulent lean flames in the combustion liners is their instability and short lifetime. Consequently, for their precise simulation there is need to use adequate chemical mechanisms of fuel oxidation at using theoretical calculations of an aerodynamic structure of the flow in gas turbine combustors.
In this study general three-stage methane burning mechanism is used. This mechanism [22] is applicable for the flames in the range of pressure change from 0.1 to 4.0 MPa and air excess coefficients 0.7-1.7. Reaction rate coefficients are shown in Table 1.

Table 1

Rate constants for three-stage mechanism of methane oxidation

Reaction	A	E, J/mole	β	Reaction Order			
$CH_4 + 1{,}5O_2 \rightarrow CO + 2H_2O$	$4.64 \cdot 10^9$	$1.17 \cdot 10^8$	-0.062	CH_4	0.5	O_2	1.066
$CO + 0{,}5O_2 \rightarrow CO_2$	$3.97 \cdot 10^{11}$	$7.68 \cdot 10^7$	0.215	O_2	1.756	CO	1.258
$CO_2 \rightarrow CO + 0{,}5O_2$	$6.02 \cdot 10^5$	$1.31 \cdot 10^8$	-0.108	CO_2	1.357		

The selection of hydrodynamic turbulence model has a significant impact on the predictive properties. Kolmogorov was the first to introduce an equation combination for turbulence kinetic energy k and specific energy dissipation velocity ω. Wilcox worked actively at k-ω simulations as a tool to calculate turbulent flows. He developed a number of simulations; the simplest one is so called high-Reynolds Wilcox simulation [23]. According to modern concepts turbulent flows have eddies which scale and lifetime change in a wide range. The dimensions of the largest eddies are comparable to with proper geometry dimensions of the flow. The reason for low-scale eddies is dissipation of turbulent kinetic energy.
At using the LES turbulent model large-scale eddies are simulated directly, and low-scale eddies are simulated using one of semi-empirical turbulence models. Direct solution of only large-scale eddies allows using thinner computational grids and larger time intervals in LES models in comparison to the Direct Numerical Simulation (DNS). Nevertheless, these computational grids shall have a larger number of elements than the grids for simple turbulence models. Apart from that, calculation via LES models shall be performed for quire a large time interval in order to obtain steady average data for the flow. However, for very swirled eddies in low emission gas turbine combustors using such model (espe-

cially in cold flow conditions when requirements for computational resources are lower) is grounded [24].
The main difference of LES from DNS is in concept of filtering for LES. During filtering those vortices are separated which scale is lower than the filtration criterion or a computational grid cell size. Thus, the filtered variable is defined by the expression [22, 24]:

$$\overline{\phi}(x) = \int_D \phi(x')G(x,x')dx' , \tag{5}$$

where:
D – calculated area,
G – filtration function that determines the scale of vortices.
The implicit method of finite element discretization suggests the following filtering operation:

$$\overline{\phi}(x) = \frac{1}{V}\int_v \phi(x')dx', \quad x' \in v , \tag{6}$$

where:
V – calculated cell volume,
$G(x,x')\begin{cases} 1/V, & x' \in v \\ 0, & x' \end{cases}$ – filtration function.
The Navier-Stokes equations obtained in this way are:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho \overline{u}_i) = 0 , \quad \frac{\partial}{\partial t}(\rho \overline{u}_i) + \frac{\partial}{\partial x_j}(\rho \overline{u}_i \overline{u}_j) = \frac{\partial}{\partial x_j}\left(\mu \frac{\partial \sigma_{ij}}{\partial x_j}\right) - \frac{\partial p}{\partial x_i} - \frac{\partial \tau_{ij}}{\partial x_j} , \tag{7}$$

where:
ρ – mass density,
$\overline{u}$ – local velocity vector,
p – static pressure,
μ – molecular viscosity coefficient,
σ_{ij} – pressure tensor from molecular viscosity,
τ_{ij} – subgrid tension scale.
As a result of the performed preliminary calculations of unsteady processes in low emission gas turbine combustors using various turbulence models it was defined that the LES approach to simulate the processes of unsteady combustion

provides more precise prediction of the levels of temperatures, pressure and concentrations of reagents [22].

4. Investigation of unsteady processes in a combustor

To perform three-dimension calculations of unsteady processes in fuel combusting devices a three dimension model of a 1/16 part of the inner space of a gas turbine combustor was developed (Fig. 1).

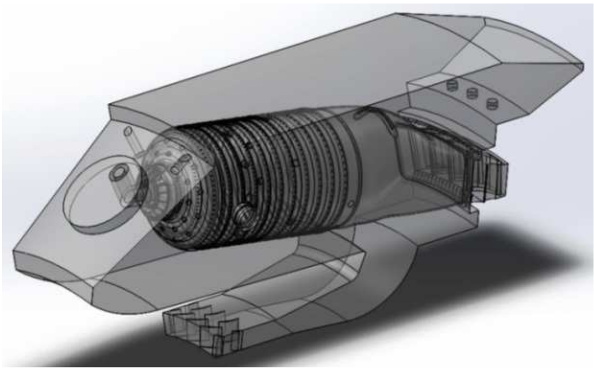


Fig. 1. Geometry combustor model

The calculated model was developed by removal of the material of the details of the combustion liner and accompanying elements from the inner space of the body. The combustion liner (Fig. 2) consists of four basic elements: a nozzle 1, a swirler 2, a combustion liner 3 and an exit diffuser 4. Apart from that, the directing outlet blades of the last compressor stage 6 and high-pressure turbine nozzle blades 5 were additionally installed in the model. The presence of these elements is explained by their potential influence on pulsation parameters of a gas turbine combustor.

A grid model of 1/16 part of the combustor consists of 15.8 million of tetrahedral cells. Such large size of the grid model is required as at using an unsteady LES model one of its filters is a cell size. Consequently, turbulent eddies, which size is smaller than the cell grid size will not be considered.

Simulation of unsteady processes in a gas turbine combustor was performed via two stages:

- steady calculation of aerodynamic structure of the flows in a combustor as first approximation via the definition of acoustic parameters;

- unsteady calculation of pulsation parameters of a combustor using the LES turbulence model.

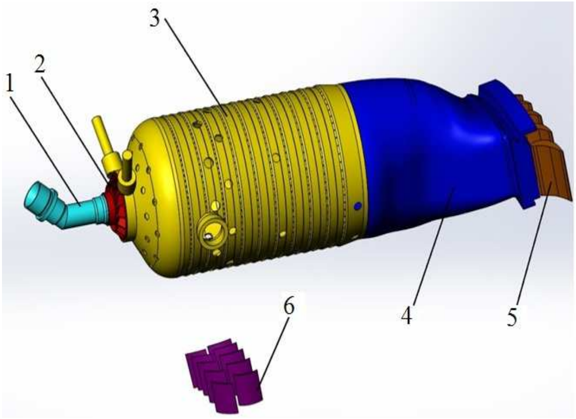
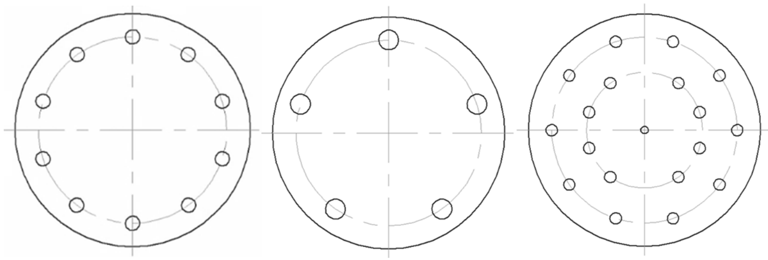


Fig. 2. Geometry model of the combustion liner and blade devices of the compressor and turbine

For numeric calculations three design schemes of the nozzles for gaseous fuel supply to a combustor were selected: 1 – nozzle with 10 holes with a diameter of 1.85 mm (Fig. 3, **a**), 2 - nozzle with 5 holes with a diameter of 2.6 mm (Fig. 3, **b**), 3 - nozzle with 18 holes with a diameter of 1.5 mm and one hole with a diameter of 1 mm (Fig. 3, **c**).
These design schemes allow variation of the velocities of gaseous fuel flow from the nozzle channels, change of fuel distribution in the combustor front device section which defines various quality of its mixing with an oxidant and leads to the change of pulsation parameters of a fuel combusting device.



a **b** **c**

Fig. 3. Nozzle design schemes

For intermediate sections, solid walls and the points of supposed sensor installation the pulsations of static pressure were fixed via numeric calculations which are the sources of acoustic oscillations and may be measured at experiments performing (Fig. 4).
All the variant calculations of unsteady processes were performed with a time interval of 0.001 s during approximately 1 s after obtaining statistically steady solution.

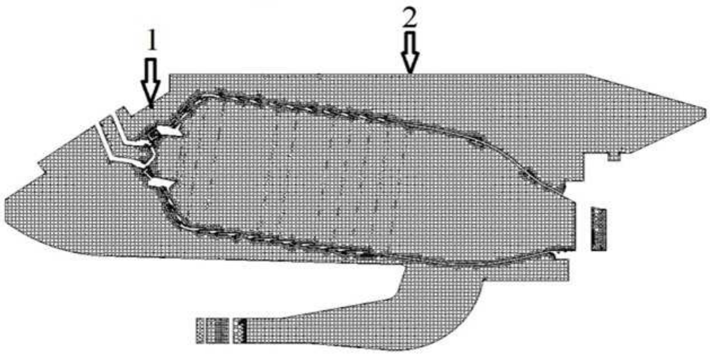
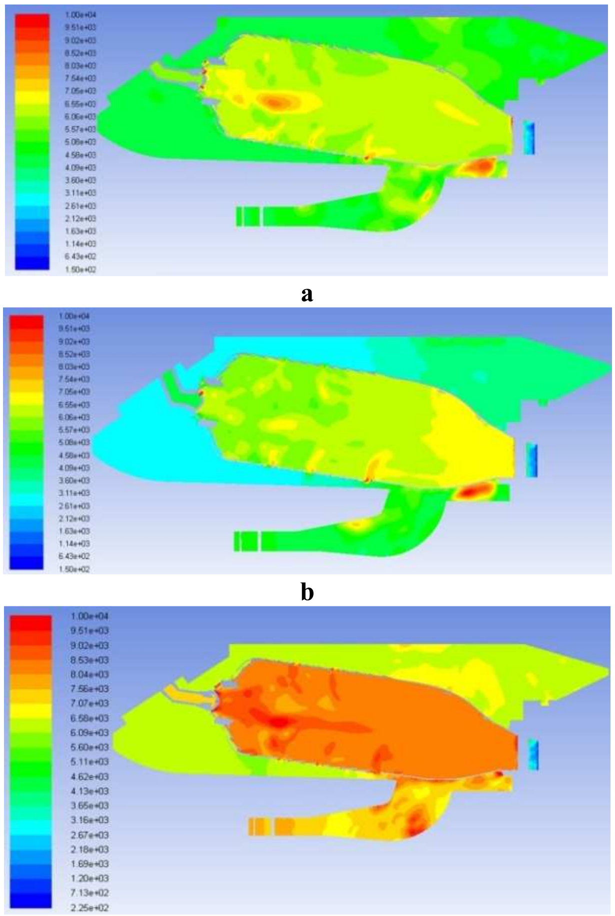


Fig. 4. Location of the points of sound receivers

Processing of the results of unsteady calculations was performed as follows. To assess the pulsation values the time-averaged and mean square deviations were built for the velocity, static pressure and the velocity for the calculated time interval and their statistical analysis was performed.
The results of unsteady calculations show that the areas of maximum pressure and velocity pulsations formation in the combustor volume are in general similar to the areas defined at steady calculations. They are located: a) inside the combustion liner near the holes of secondary air; b) in the area of flow exist from the swirler; c) near the 3rd-5th shells of the combustion liner; d) in the inlet diffuser which supplies air from the compressor to the combustor.
For the variant of the nozzle with 10 holes (Fig. 5, **a**) the calculated mean square level of pulsations in inter pipe space near the secondary air holes (the place of sensor No.2 installation) is 5.2 kPa, in the primary combustion zone an average value is 6.7 kPa and a maximum value is 8.3 kPa, on the outlet of the combustion liner an average value is 6.4 kPa and a maximum value is 9.6 kPa.
For the variant of the nozzle with 5 holes (Fig. 5, **b**) the calculated mean square level of pulsations in inter pipe space near the secondary air holes is 3.6 kPa, in the primary combustion zone an average value is 5 kPa and a maximum value is

6.55 kPa, on the outlet of the combustion liner an average value is 6.8 kPa and a maximum value is 7 kPa.

For the variant of the nozzle with 19 holes (Fig. 5, **c**) the calculated mean square level of pulsations in inter pipe space near the secondary air holes is 11 kPa, in the primary combustion zone an average value is 9.4 kPa and a maximum value is 11.7 kPa, on the outlet of the combustion liner an average value is 9.5 kPa and a maximum value is 11.5 kPa.



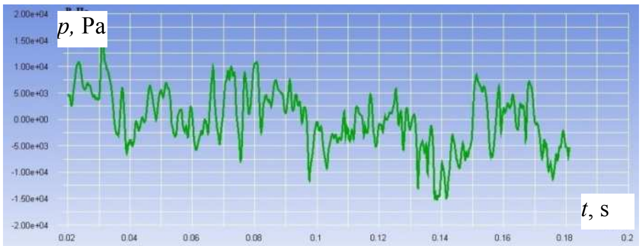
a

b

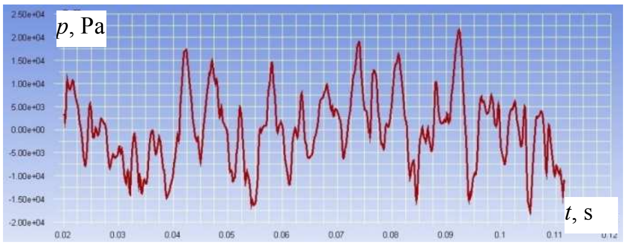
c

Fig. 5. Level of mean square pulsations of static pressure in the combustion liner: **a** – nozzle with 10 holes; **b** –nozzle with 5 holes; **c** – nozzle with 19 holes

Figure 6 shows pulsations of static pressure in the section of the top hole of secondary air supply. Further, Fourier transformation is used and the diagrams of spectrum power of the signal to frequency spectrum are built (Fig. 7).



a



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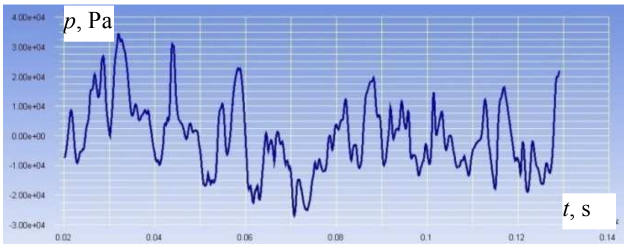
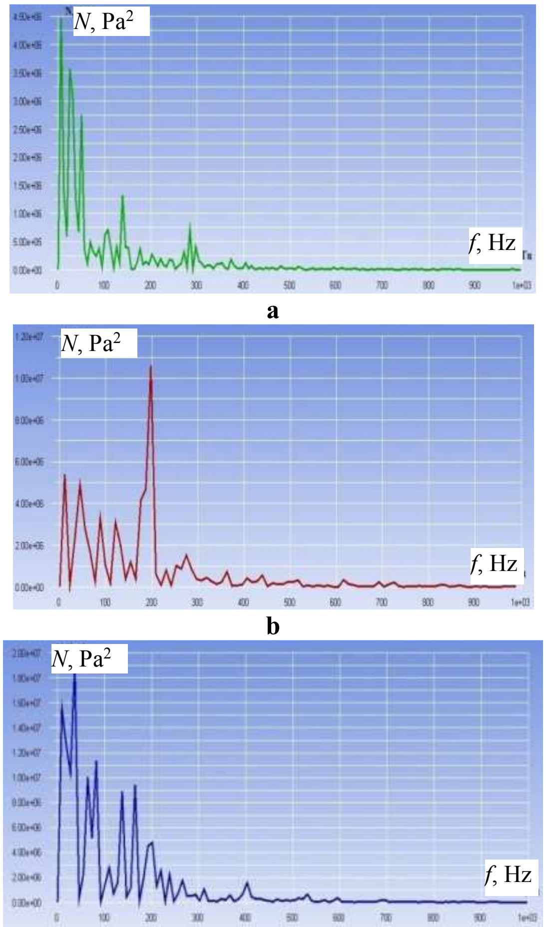


c

Fig. 6. Pulsations of static pressure in the section of secondary air holes in the combustion liner: **a** – nozzle with 10 holes; **b** –nozzle with 5 holes; **c** – nozzle with 19 holes

For the variant of the design scheme of the nozzle with 10 holes the calculated peak is in the frequency of 10 Hz with amplitude of 2,100 Pa and in the frequency of 140 Hz with amplitude of 1,100 Pa. For the variant of the design scheme of the nozzle with 5 holes the calculated peak is in the frequency of 195 Hz with amplitude of 3,255 Pa. For the variant of the design scheme of the nozzle with 19 holes the calculated peak is in the frequency of 45 Hz with amplitude of 4,400 Pa and in the frequency of 170 Hz with amplitude of 3,100 Pa.



a b c

Fig. 7. Spectrum power of the signal of static pressure in the section of secondary air holes in the combustion liner:
a – nozzle with 10 holes; **b** –nozzle with 5 holes; **c** – nozzle with 19 holes

Figure 8 shows the comparison of the experimental and calculated values of pressure pulsations in the combustor with various nozzle designs. The difference of experimental and calculated values at using a 10-hole nozzle was 3.7 %, using a 5-hole nozzle was 10%, and using a 19-hole nozzle was 11.5%.

The represented results show the satisfying correlation of the experimental and theoretical data and an opportunity to use the developed mathematical simulation of unsteady processes for parameter calculations of low emission gas turbine combustors.

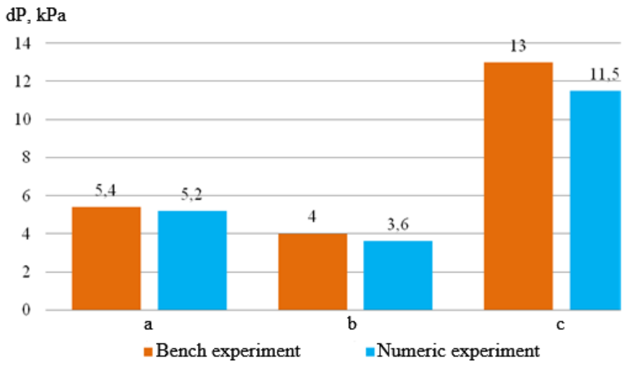


Fig. 8. Comparison of experimental and calculated values of pressure pulsations in a combustor with various nozzle design schemes:
a – nozzle with 10 holes; **b** –nozzle with 5 holes; **c** – nozzle with 19 holes

The variant of the nozzle design scheme with 18 holes with a diameter of 1.5 mm and one hole with a diameter of 1 mm shows maximum levels of dynamic pressure in the combustor, so using this variant of the nozzle is not reasonable. The variant of the nozzle design scheme with 5 holes with a diameter of 2.6 mm provides minimum levels of dynamic pressure in a combustor and can be recommended for further experimental and industrial operation. It can be explained as follows: one of the important factors which define the hold time of the mixture in the combustion zone is the velocity of fuel flow. For this design variant the velocity of the fuel flow from the nozzle channel is minimum, so the residence time of the fuel-air mixture in the chemical reaction zone is the biggest, and, thus, there be more effective flame stabilization.

5. Summary

1. Theoretical investigations of pulsation characteristics of a low emission gas turbine combustor were carried out. They allowed to determine that the maximum amplitude pressure pulsations are observed: a) in the central vortex inside the combustion liner, b) in the area of the secondary air inlets, c) inside the combustion liner: in the region of the 3-5 shells, d) on the walls

of the inlet diffuser, e) in the area of the outflow of fuel in the swirler channels.

2. For the entire considered frequency range, a pronounced frequency of 189 Hz is traced, which is caused by combustion processes, as evidenced by the spectra of temperature fluctuations and the mass concentration of fuel.
3. The amplitude of static pressure pulsations depends on the monitoring points and frequency. These amplitude values were obtained by decomposing in a Fourier series the mass flow averaged static pressure in the cross section of the combustion liner. The calculated local mean square pulsations of the static pressure inside the combustion liner reach maximum values of about 11.5 kPa.
4. The variant of the nozzle design scheme with 5 holes with a diameter of 2.6 mm provides minimum levels of dynamic pressure 3.6 kPa in a combustor and can be recommended for industrial operation.
5. It should be mentioned that the results of the performed numeric experiments using volume mathematical models in the combustors operating on gaseous fuel adequately reflect physical and chemical processes of unsteady combustion and can be recommended for upgrading geometry and mode parameters of low emission combustors.

6. References

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