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NUMERICAL SIMULATION OF THERMOACOUSTIC GAS OSCILLATIONS IN A PIPE WITH TOROIDAL HEAT EXCHANGE ELEMENTS

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The paper simulates the modes of a gas flow in a pipe closed at one end and open to the atmosphere at the other end; for this purpose, a numerical solution of the system of Navier – Stokes equations in a two-dimensional axisymmetric formulation has been used. The excitation of oscillations of gas-dynamic functions is caused by the temperature gradient in the pipe section resulting from the contact of the gas with differently heated toroidal elements inside the pipe, their temperature being maintained constant. When the specified gradient reaches the threshold value, a stable thermoacoustic oscillation of the gas column is observed in the pipe. The time dependences of the pressure and the axial component of the gas velocity, as well as the heat flows in the heat exchange unit of the resonator, were calculated. The obtained results were in good agreement with the experimental data.

Keywords: thermoacoustics, Sondhauss effect, mathematical simulation, numerical solution of Navier – Stokes equations

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ЧИСЛЕННОЕ МОДЕЛИРОВАНИЕ ТЕРМОАКУСТИЧЕСКИХ КОЛЕБАНИЙ ГАЗА В ТРУБЕ С ТЕПЛООБМЕННЫМИ ЭЛЕМЕНТАМИ ТОРОИДАЛЬНОЙ ФОРМЫ

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В работе проведено моделирование режимов течения газа в трубе, закрытой с одного конца и открытой в атмосферу другим концом; для этого использовано численное решение системы уравнений Навье – Стокса в двумерной осесимметричной постановке. Возбуждение колебаний газодинамических функций обусловлено градиентом температуры на участке трубы, возникающим в результате контакта газа с разнонагретыми тороидальными элементами внутри трубы, температура которых поддерживается постоянной. По достижении указанным градиентом порогового значения, в трубе наблюдается устойчивое термоакустическое колебание газового столба. Рассчитаны зависимости от времени давления и осевой составляющей скорости газа, а также тепловые потоки в теплообменном блоке резонатора. Полученные результаты хорошо согласуются с экспериментальными данными.

Ключевые слова: термоакустика, эффект Зондхаусса, математическое моделирование, численное решение уравнений Навье – Стокса

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Introduction

Many devices are available for converting thermal energy into mechanical or electrical energy. A particular focus is on the thermoacoustic engine (TAE), since acoustic energy in the form of standing or traveling waves acts as an intermediate link in this transformation. Many research groups have considered the mechanisms generating a sound field and maintaining it at a sufficient level in operation. The engine itself has a simple design, with the option to eliminate moving elements from the working fluid. External heat supply allows using any source of energy, while the motion of the working fluid in a closed system makes it possible to fabricate autonomous structures that can be used on Earth and in space.

The effect when sound is generated in a pipe closed at one end and open into the atmosphere at the other was first described by the German physicist Karl Sondhauss [1]. While the study did not explain why sound appeared, it established the general patterns behind the behavior of such characteristics as the dependence of frequency and amplitude of generated waves on the parameters. Rayleigh [2] made qualitative assessments of the phenomena when oscillations emerge and persist, ultimately laying down the foundations for the entire discipline of thermoacoustics. Expressed concisely, the concept is that if heat is given to the gas at the instant of greatest compression or is taken away from it at the instant of greatest rarefaction, this leads to an increase in acoustic oscillations. An important point in this statement is that the source driving the oscillations is identified as the time dependence of heat addition and its synchronization with pressure oscillations. Putnam and Dennis [3] provided a mathematic proof for this hypothesis, which came to be called the Rayleigh criterion. The formulation of the criterion indicates a phase shift between pressure and heat addition oscillations, whose absolute value should be less than $\pi/2$; oscillations are excited in this case. If the phase shift lies in the range from $\pi/2$ to π, the oscillations are damped.

The first TAE based on a standing wave was created in 1962 by Carter, White and Steele (USA) [4]. A stack acting as a heat exchanger with thermal inertia was placed between the heat supply and heat removal sections in the experiments, similar to regenerative Stirling engines. This improvement made it possible to significantly enhance the thermoacoustic effect in the Sondhauss pipe. A particularly interesting prototype was built at the Los Alamos National Laboratory in the 1990s [5]. An experimental setup was constructed at the same time at the National Center for Physical Acoustics of the University of Mississippi to study a stationary thermoacoustic wave in a wide range of parameters [6]. Further experimental studies were aimed at producing a system consisting of a stack combining an acoustic prime mover and an acoustic refrigerator, united by a common resonator [7]. Changing the rectilinear resonator to a *U*-shape and using a liquid piston in the resonator made it possible to significantly reduce the size of the twin setup [8].

The idea that an analogy can be drawn between a Stirling engine and a thermoacoustic engine has been further developed, leading to revision of the basic theory describing the generation of stationary waves in a resonator. Ring-shaped resonators with the operating principle based on traveling waves were constructed as a result [9, 10].

Notably, even though the basic principles of thermoacoustic effects are fairly well-understood, with extensive experimental experience accumulated in creating functional devices, the issues of theoretical analysis and mathematical modeling are far from fully resolved. Because the processes occur on different scales, either an extremely simplified formulation is obtained, allowing to describe the object 'as a whole', or the analysis focuses on the unsteady heat transfer mechanism neglecting the processes outside the near-wall layer of the heat exchanger.

Let us consider an experiment with a Sondhauss tube without a regenerator. In a sense, this formulation is more complicated, since the presence of a regenerator in some modes stabilizes the unsteady gas flow. Let us rely on the results of the experimental study [11], which returns us to the basic formulation of the Sondhauss pipe problem.

A plane channel of length L with a square section $d \times d$ was described in [11]. One end of the channel was closed, the other open, and the gas in the channel interacted with the atmosphere. A heater was located inside the channel at a distance σ from the closed end; a cooler was located at a distance l ($l << \sigma$) from the heater. The heater and cooler were grids consisting of an equal number N of cylindrical rods of the same radius a. The heater had a constant temperature T^+, and the cooler T^-. All rods were parallel to each other. It was confirmed experimentally that the threshold at which self-oscillations occurred in the resonator was a function of two parameters: the temperature difference and the distance between the heat exchangers. Another parameter is the location of the heat exchangers relative to the closed end of the pipe.

The main difficulties in theoretical analysis of thermoacoustic self-oscillations, the main difficulties are caused by complex mathematical description of feedback mechanisms accounting for nonlinear properties; besides, nonlinear partial differential equations have to be solved for this purpose. Therefore, linearizing the system of differential equations by some method provides certain insights into the laws of thermoacoustics.

We should note that early studies were based on a one-dimensional approximation in a nonlinear formulation. Friction and heat transfer were represented by closing relations depending on the flow regime. However, this approach turned out to be ineffective for describing periodic processes because it is impossible to correctly reflect the phase shifts of the signal in the closing relations.

The Rott equation was later formulated [12, 13] based on a linearized system of equations for the balance of mass, momentum and energy in the one-dimensional approximation; a software version of the solution to this equation is available: DeltaEC [14] (free access). The range of applicability is determined for this model by the 3% pressure deviation from its baseline value. Many examples assessing the performance of devices using this software are given in literature [15 – 17]. As linear theory was applied in practice, the understanding of thermoacoustic oscillations was supplemented with the knowledge that a critical temperature gradient must be exceeded to excite oscillations if the heat source and sink are separated in space [5].

A different approach was used in [18], based on the system of boundary layer equations. This made it possible to construct a mathematical model based on the behavior of functions towards the wall, gaining new data on possible losses of acoustic energy. Losses are accounted for as an energy balance, which consequently sets not only the threshold level of the temperature gradient when gas oscillations occur, but also the possible parameter values of traveling or standing waves. Taken together, this information made it possible to achieve better agreement between experimental results and theoretical estimates. Monograph [19] describes the specifics of problems related to heat addition during combustion.

However, it is fundamentally impossible to model some effects within the framework of linearized theory. This primarily refers to the evolutionary process of oscillation generation. Linearized theory can only characterize the state of oscillations that are already established.

Turbulence is another important issue. Changing the direction in which the gas moves leads to the fact that the turbulence model should describe the transition phenomena when turbulent flow is generated and vice versa. The appearance and destruction of turbulent structures are characterized by a relaxation time scale, which is not represented in any way in the Rott equation. Notably, the problem of describing transitions within turbulence models has not been solved completely even for the so-called canonical flows on plates, in pipes, or in the wake.

Finally, the third aspect are different oscillation modes emerging for gas-dynamic functions. In the general case, they are not described by the simplest dependences in the form of harmonic functions, which are the solution to linearized equations. In particular, the problem on oscillatory flow around a cylinder is constructed by two regime parameters: the Keulegan – Carpenter number (the ratio of the hydrodynamic to the geometric scale that is the cylinder diameter) and the Stokes number (the estimate of the boundary layer width as a fraction of the geometric scale). According to the generally accepted classification, up to six zones with completely different flow structures are identified on the regime map in the vicinity of the cylinder [20]. A tandem of two cylinders complements the picture with another parameter, i.e., the distance between the cylinders. This generates an interference pattern, which can be accompanied by both an increase and a decrease in the friction factor of the pair.

Many of these problems can be solved by mathematical modeling via numerical solution of a system of differential equations. Various approaches to numerical study of the problem were considered in a number of works, for example, in [21 – 27].

Let us consider some of these studies in more detail. It was confirmed in [21] that the self-oscillation mode could be obtained by numerically solving a system of Navier – Stokes equations in a two-dimensional formulation. A particular assumption made by the authors is that plates of zero thickness are used as heaters in the calculations. The results of numerical integration in [22] simulate the operation of some of the engine elements. The problem is solved in a two-dimensional formulation, examining heat transfer on a plate in oscillatory flow caused by external excitation. The performance characteristics of a high-frequency (300 Hz) engine with flat heat exchangers were calculated in [24] in a two-dimensional formulation assuming a turbulent flow regime. The mathematical model was closed by the (k-ε) turbulence model. The temperature was set on the channel wall and on the stack connecting the heat exchangers. The authors investigated the influence from the lengths of the resonator and the stack connecting the flat heat exchangers, comparing the data with linear theory. The results of numerical simulation revealed an oscillatory mode of gas-dynamic functions with decreasing amplitude. The LS-FLOW solver, developed by the Japan Aerospace eXploration Agency for analysis of three-dimensional compressible flows within the Navier – Stokes system of equations, was used in [26]. The computations were carried out on an unstructured grid in a two-dimensional formulation. It was found that a short trigger pulse injected from the open end of the computational domain was required to induce oscillations in the system. However, a deviation of the numerical solution from linear theory was observed.

To summarize, mathematical modeling of thermoacoustic self-excitation in the given system by numerically integrating the system of Navier – Stokes differential equations imposes stricter requirements on all aspects of the computational process. Constructing the solution to the problem by the control volume method involves tailoring the size of the domain where the desired functions are defined, the shapes of the discretization elements of the computational domain, the method for approximating the function at the element boundary, the time integration scheme. The computations in all of the above studies were performed in a two-dimensional formulation without accounting for the interaction with the pipe wall. For this reason, this work contains the solution of a two-dimensional problem including the interaction with the pipe wall.

Problem statement and computational aspects

The solution of the problem from [11] requires three coordinate directions. If a channel with a square cross-section is replaced with a cylindrical pipe, and the heat exchanging rods with toroidal elements, then, assuming that the gas-dynamic functions are homogeneous in the circumferential direction, axial symmetry appears in the problem formulation, making it possible to reduce the number of spatial coordinates to 2. This geometric modification does not fundamentally affect the excitation of thermoacoustic oscillations under consideration, allowing to consider not only unsteady heat

transfer on the surface of heat exchangers but also the interaction of the gas with the pipe wall, which is commonly excluded from balance relations in the two-dimensional formulation.

The system of Navier – Stokes equations for the problem formulated in an axisymmetric setting consists of four partial differential equations:

mass balance,

energy balance,

balance of momentum written as projections on two axes, z and r.

The system should be supplemented with two equations of state:

thermodynamic (in the form of the Mendeleev – Clapeyron equation),

caloric, establishing the relationship between internal energy and temperature.

A diagram of the computational domain and explanations for the problem formulation are shown in Fig. 1, the values of the parameters used in the calculations are summarized in Table.

Parameters of stagnant gas values at the open end of the pipe: P^0 = 0.1 MPa, T^0 = 300 K. The temperature of the cooler surface T = 300K. The given quantitative values of the problem parameters correspond to the sizes selected in [11]. Judging by the results of that study for the selected distance between the heat exchangers, the self-excitation mode should occur at the temperature T^+ = 850 K and higher at the heater surface. Thus, it can be established through a series of calculations with varying heater temperature that the mathematical model can reproduce the important conclusions of the experimental study.

The boundary conditions at the permeable boundary of the computational domain are imposed in a simplified formulation. Firstly, the boundary of the computational domain coincides with the exit section of the pipe, which excludes the influence from the so-called synthetic jet arising outside the resonator on the solution [28]. Secondly, when gas enters the computational domain, the steady-state values of gas-dynamic functions are obtained by isentropic formulas; the value P^0 is used for gas outflow.

The side wall of the pipe is divided into three zones to maintain the thermodynamic balance in the system. The pipe wall temperatures in the sections to the left and to the right of the heat exchanger are T^+ and T^-, respectively. The condition of thermal insulation is imposed between these zones, on the pipe wall section (1.0 cm long), centered relative to the location of the heat exchangers. An identical boundary condition is imposed at the end of the pipe.

The no-slip condition is imposed on the surface of the toroidal elements and on the pipe walls.

The results in this study were obtained using the ANSYS FLUENT 2021R1 package, allowing to solve two-dimensional problems in the axisymmetric formulation. The pressure-based solver in the implicit formulation with the coupled algorithm was used to construct the solution. Reconstructions with third-order accuracy (Third-Order MUSCL) were used to produce the values of functions on the control surfaces. The time integration was performed by an implicit scheme with second-order

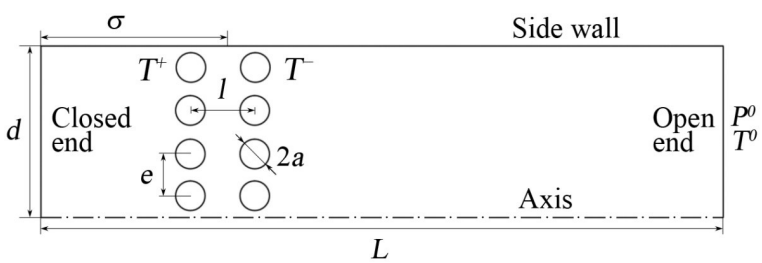

Fig. 1. Schematic of computational domain (geometric proportions are not reproduced):
Side wall of the pipe; Closed & Open ends of the pipe; Axis of symmetry;
T^+, T^- are the temperatures of the heater and the cooler; see also table

Table

Computational parameters of the problem and their values

Parameter	Notation	Unit	Value
Length of computational domain	L	m	1.0
Radius of computational domain	d	cm	3.2
Radius of cylindrical rods	a	mm	1.0
Distance from grid to closed wall of pipe	σ	cm	20
Distance between grid rows	l	mm	6.0
Number of elements in grid	N	–	4
Grid spacing	e	mm	4.0
Gas pressure	P^0	MPa	0.1
Temperature of gas of cooler surface	 T^0 T	K	300

Notes. 1. The values given correspond to the selected dimensions from [11]. 2. The grid spacing is the distance between the central lines of the torus generators. 3. The compressed medium is air.

accuracy. Discretization was carried out on an unstructured mesh consisting of quadrangular elements. The mesh provided clustering to all impermeable boundaries of the computational domain. This made it possible to resolve the structure of the temperature and dynamic boundary layer on the pipe wall and on the surfaces of the heat exchangers.

The results in the study (see below) were obtained on a mesh containing about 40,000 elements, while the time integration step was $\Delta t = 0.1$ ms.

Computational results and discussion

Initial data. The computations were carried out for the initial state of the gas, when its pressure in the pipe was P^0, and its velocity was equal to zero. The stepwise dependence on the axial coordinate was taken as the initial state of the temperature. It was assumed that the temperature to the left and right of the gratings coincides with the temperature of the heat exchangers. The temperature in the section where the thermal insulation condition was imposed on the pipe wall was determined by the half-sum of the selected temperature levels. The gas density was taken in accordance with the ideal gas equation of state. This local state, which is unsteady with respect to temperature between the heat exchangers and the air around them, is essentially sufficient to excite natural oscillations of the gas column in the system.

The excitation of oscillations starts as a temperature field is generated in the vicinity of the heat exchangers. The process is accompanied by pressure gradients appearing, along with a nonzero value of the velocity vector. Pressure and velocity waves are characterized by small amplitude and natural oscillation frequency. This state of gas (oscillations with small amplitude) can last rather long and count more than one hundred oscillation periods. Both an unsteady solution and a steady heat transfer regime can be established depending on the magnitude of the temperature gradient. Let us give the results for four temperatures of the heater (K): 600, 800, 900 and 1200.

Fig. 2 shows the time dependences of the mean cross-sectional velocity at the open end of the pipe and the magnitude of the pressure deviation from the initial value at its closed end at different temperatures of the heater.

Judging from the data presented, the arising oscillations of gas-dynamic functions are completely damped in a time interval of less than 1 s at a heater temperature $T^+ = 600$ K.

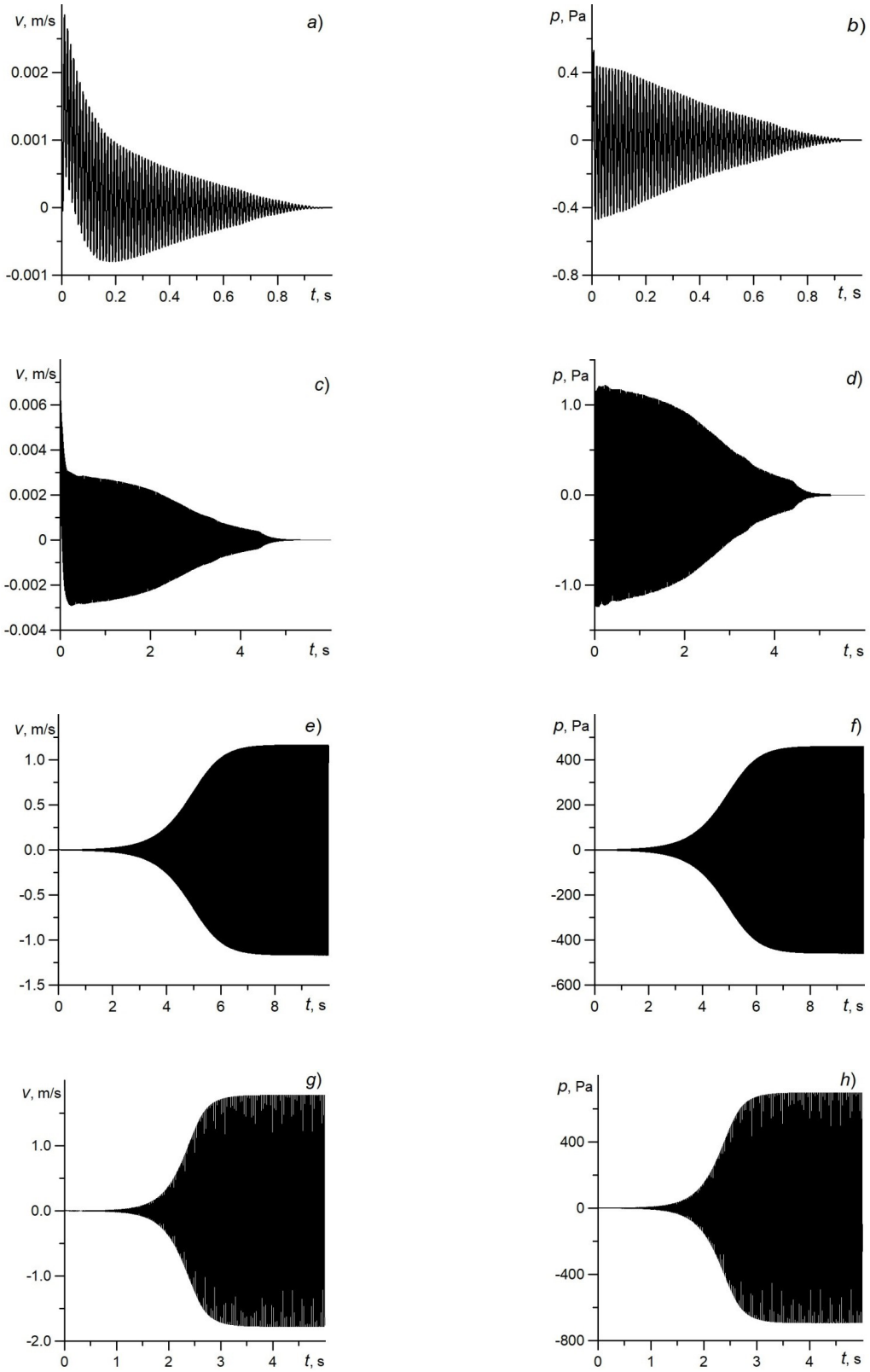

Fig. 2. Time dependences of axial velocity component, averaged over the section, at the open end of the pipe (*a*, *c*, *e*, *g*) and pressure deviation from the initial level at the closed end of the pipe (*b*, *d*, *f*, *h*) at heater temperature T^{+} = 600 K (*a*, *b*), 800 K (*c*, *d*), 900 K (*e*, *f*) and 1200 K (*g*, *h*)

The damping scenario of natural oscillations undergoes changes at a heater temperature $T^+ = 800$ K, namely, the process proceeds much more slowly. For the given initial state of the functions, the transition to steady state occurs in a time interval exceeding 4 s. A possible explanation for this behavior of the functions is that the selected combination of problem parameters produces a regime that is outside the boundaries of oscillation self-excitation. The heater temperature does not precisely coincide with the temperature at which the self-oscillations of gas-dynamic functions are observed in the experiment: this can be related to the change in the channel shape and simplified description given to the behavior of the functions at the open end of the pipe.

The threshold values of the temperature gradient are exceeded at a heater temperature $T^+ = 900$ K, therefore, a continuous oscillatory gas flow evolves. Steady flow is established in the pipe in several stages. First, a time interval exists with small-amplitude oscillations, when the signal magnitude increases by a linear law; secondly, there is an 'avalanche-like' resonant mode of increasing the oscillation amplitude, when there is positive feedback between the change in pressure and the specific heat flux in the heat exchangers; thirdly, the transition to steady non-stationary oscillation mode uis observed. The oscillation amplitude of gas-dynamic functions (third stage) is stabilized due to two factors. The first is the work of the frictional forces of the gas against the pipe wall performed during oscillatory flow, and the second is the radiation that occurs the open end of the pipe, generating a synthetic jet.

The flow regime in the resonator at heater temperature $T^+ = 1200$K is characterized by a shorter transition time to steady oscillations. The amplitude of velocity oscillations at the open end of the pipe and the pressure deviation from the initial value at its closed end is higher than for the case $T^+ = 900$ K. The natural frequency of the gas column is 87.0 ± 0.7 Hz up to the selected time integration step.

Let us focus on the behavior of functions over time for one oscillatory period in steady state. Fig. 3 shows the dependence of the specific heat flux averaged over the entire surface of the grid and the pressure deviation from the initial state for one oscillatory period for the cases $T^+ = 900$ and 1200 K.

The zero value of the pressure deviation from the initial one in the heat exchangers is taken as the beginning of the cycle. The behavior of the specific heat flux function indicates that the maximum heat addition is synchronized with the maximum pressure increase across the heater. In turn, the maximum heat removal in the cooling grid is also in the vicinity of the minimum pressure. Thus, a situation that falls under the Rayleigh criterion happens with both heat exchangers. Notably, the sig-

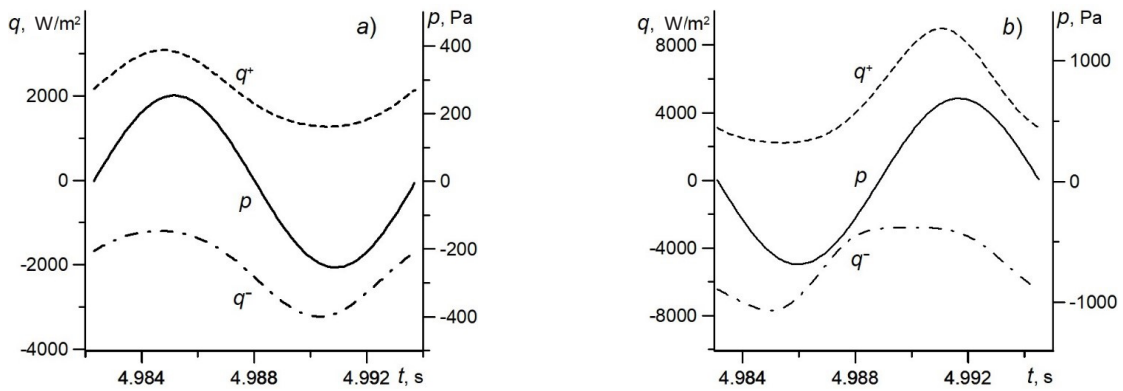

Fig. 3. Time dependences of pressures p (solid line), specific heat fluxes q^+ (dashed line) and q^- (dash-dotted line) for the oscillatory period in the heater and cooler, respectively, for cases $T^+ = 900$ K (a) and 1200 K (b)

nals increase in amplitude and lose their sinusoidal shape with the distance from the parameter values setting the excitation boundary of thermoacoustic oscillations.

Conclusion

The solution to the problem on self-excitation of oscillations in a structure reproducing the Sondhauss effect has been obtained by numerical integration of the system of differential Navier – Stokes equations for compressible gas. Depending on the temperature difference in the heat exchangers, the mode established is either oscillation damping, or a stable unsteady mode of oscillatory flow of a gas column in a pipe closed at one end. The result of mathematical modeling is in good agreement with the experimental data.

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