Investigating the hydraulic resistance in the flow part elements of pneumatic systems and heat exchangers

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Abstract. The article deals with the development of shell-and-tube heat exchangers for the needs of power engineering, based on additive technologies, in particular, selective laser sintering technology with new configurations of heat exchange surfaces. The role of heat exchangers in microturbines, the most common units of power plants of small distributed power generation, is considered. To intensify heat transfer and increase the efficiency of microturbines, it is proposed to use various configurations of flow channels of shell-and-tube heat exchangers made on the basis of additive technologies. Mathematical modeling and experimental study of a gas medium flow in the tubes of a heat exchanger are carried out. The dependences of the coefficient of hydraulic resistance between the surface of inlet and outlet of tubes of various configurations on the Reynolds number are obtained. The results of the experiment allow us to conclude that the resistance of spiral-shaped tubes is slightly higher than the resistance of tubes with three ribs.

1. Introduction
Of all the branches of human economic activity, energy has a huge impact on the life of the population [1]. Energy provides functioning of such facilities as: industrial enterprises, medical facilities, housing estates, transport, etc. The future development of the energy industry in Russia is more often associated with distributed small energy, which is based on the use of microturbines.

The absolute electrical efficiency of modern steam turbine power units based on microturbines is approximately 38%. It is known that the combined generation of electric energy and heat has a higher efficiency. Therefore, the highly efficient use of the primary energy source to produce two forms of useful energy – thermal and electric – is a promising direction in the development of energy. The heat power industry of Russia actively uses cogeneration technologies.

An increase in microturbine efficiency may be achieved by heat recovery. This process is largely dependent on the quality of the heat exchangers. The main task of heat exchangers in gas turbine engines (GTE) is to heat the air for the fuel mixture [2].

The solution to this problem can be achieved by increasing the heat transfer surface, which, in turn, increases the dimensions and mass of the heat exchanger. And this is due to increased costs for the production of heat exchanger. Therefore, the most rational way to solve this heat transfer problem is to intensify the heat transfer.

At the moment, various methods of intensifying convective heat transfer between different coolants have been proposed and studied [3–8]: the use of vortex generators, increasing the roughness and
increasing the heat transfer surface due to fins, the use of flow swirling with spiral ribs, worm devices, swirlers, etc. Often being highly efficient, it turns out to use combined methods of intensification [3–5]. Therefore, the intensification of heat transfer is associated with the solution of the following tasks [6]:

1. Ensuring the efficiency of heat transfer by reducing the temperature difference at the exits from the heat exchanger for hot and cold coolants (Fig. 1).
2. The increase in the cross-sectional area on the hot side in order to use a wide range of fuels (to prevent soot settling in the heat exchanger channels).
3. Ensuring uniform temperature at the outlet of the heat exchanger in the coolant.
4. The increase in heat transfer area on the cold side is relatively hot, since the density of the cold coolant is greater than the hot one.

![Figure 1. Schematic diagram of a shell-and-tube heat exchanger.](image)

The following basic types of heat exchange devices are singled out for constructive execution: plate, rotary and shell-and-tube. The research presented in this paper is aimed at studying the effectiveness of heat exchange of casing heat exchangers made using additive technologies, in particular, Selective Laser Sintering (SLS). The limitation of this method of manufacturing is the requirement to tilt the surface of the part to a horizontal plane of at least 50 degrees. If this condition is not met, one has to introduce additional technological support that needs to be removed in post-processing. Another feature of the method of laser caking is that it is necessary to remove metal powder from the hard-to-reach cavities of the manufactured part.

Despite all the difficulties of production using additive technologies, indisputable advantages allow creating heat exchanger (HE) with completely new configurations of flow channels. Wide possibilities for creating geometry require serious research and creation of a scientific base in order to identify the best configurations of flow channels. At the moment, the shell-and-tube type HE is the most suitable for implementation using additive technologies. Since this design consists of arrays of similar channels, the object of study is one channel, the parameters of which are considered separately.

2. Experimental research
A testing rig is developed to assess thermal and hydraulic characteristics of working media flow inside the test samples (prototypes of additive shell-and-tube heat exchangers) (Fig. 2).
Figure 2. Scheme of testing rig of shell-and-tube heat exchanger.

The test setup consists of a flow inducer (fan or compressor) 1, providing air flow at the required speed in the tubes, pressure regulator 2, heater 3, confusor 4 of the experimental model of the heat exchanger 5, a container for reducing the flow rate 6, a constricting device for measuring gas flow 7, systems for supplying coolant TX1 and TX2, pressure sensors P1-P3 and temperature sensors T1-T3. H. t. is the cold coolant. The experimental prototype of the heat exchanger consists of 61 fixed tubes. There are 3 types of such tubes: a hollow tube, a tube with 3 internal ribs and a spiral tube.

Modifications of the test rig are developed and performed using high-precision pressure transducers, systems for recording, collecting and processing information to assess the hydraulic resistance of the channels. In the experiment, several configurations of heat exchangers' tubes are considered; their models are shown in Fig. 3. For each channel configuration, series of experiments are carried out, data is collected on the corresponding pressure drop between the inlet and outlet of a flow from a tube for various air flow rates and inlet temperatures. Based on the experimental data on the pressure drop and the calculation of a hydraulic resistance coefficient \( \xi \) based on the Darcy-Weisbach formula, a dependence of \( \xi \) on the Reynolds number is constructed.

Figure 3. Configurations of researched heat exchanger channels.

For further estimation of hydraulic resistance on the basis of a computational experiment, mathematical modeling of a gas flow process in the channels of additive shell-and-tube heat exchangers is carried out.

3. Modeling of gas-dynamic processes
A mathematical model of the flow of gas in the heat exchanger tube has been developed, the calculation scheme of which is presented in Figure 4.
The mathematical model is based on a system of equations (1–3):

**Navier-Stokes equation in the tensor form (taking into account the turbulent stresses):**

\[
\begin{align*}
\frac{\partial \rho v_i}{\partial \tau} &+ \rho v_j \frac{\partial v_i}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \frac{\partial v_i}{\partial x_j} - \rho v'_i v'_j \right] \\
\frac{\partial \rho}{\partial \tau} &+ \frac{\partial \rho v_i}{\partial x_i} = 0,
\end{align*}
\]  

(1)

where \( \rho \) is the liquid density, \( \tau \) is the time, \( p \) is the pressure, \( v_i, v_j \) are the velocity projection on coordinate axes, \( \mu \) is the coefficient of dynamic viscosity; \( i, j \) are the space axes, \( \rho v'_i v'_j \) are the turbulent stresses.

**Energy equation:**

\[
\rho c_p \left( \frac{\partial T}{\partial \tau} + \bar{v} \cdot \nabla T \right) = \lambda \Delta T + q_v - \frac{\partial p}{\partial \tau} + \mu \cdot \Phi ,
\]  

(2)

where \( \mu \cdot \Phi \) is the frictional energy, \( T \) is the temperature, \( v \) is the flow velocity, \( \lambda \) is the thermal conductivity coefficient, \( c_p \) is the isobaric thermal capacitance of working substance, and \( q_v \) is the power of internal heat sources.

**Perfect gas equation**

\[
\frac{p}{\rho} = R \cdot T ,
\]  

(3)

where \( R \) is the specific gas constant.

The main assumptions of the mathematical model are: gas is perfect; the working environment is subject to Newton's law of viscosity; and SST model of turbulence is used.

Boundary conditions of the mathematical model: \( V_1 \) [m/s] is the gas speed on the surface of S1, the average static gas pressure on the S2 surface is equal to 1 atm and temperature on the surface S3 is set.

The problem is solved by using the numerical method of the end elements in a stationary setting using a certified package of Ansys CFX programs. To do this, the calculation area is converted into a grid of elements (tetrahedrons and prismatic layer). Grid options: 520,000 elements, base tetrahedron with the size of 0.3 mm, and prismatic layer of 10 elements with a total thickness of 0.05 mm.

The convergence control of the calculation is carried out by analyzing the graphs of inlet pressures on the S1 and outlet temperature on the S2 and as well as the graph of "inconsistencies". When a stabilization of pressure and temperature is reached, the calculation is completed when the value of 10^-4 of closure error is reached. As a result, for different speed modes at gas flow rates from 2 to 14 g/s, distributions of gas pressures and velocities are obtained in the flow parts. One of the results of the
numerical study, the pressure distribution in the axial section of the computational domain, is shown in Figure 5.

![Pressure Distribution](image)

**Figure 5.** Local distribution of gas pressure in flow lines at gas flow rate of 0.23 g/s.

4. **Analysis of obtained data**

Based on the calculated and experimental data, the dependences of the hydraulic resistance coefficient on the Reynolds number are constructed (Fig. 6).

![Comparison of Experimental and Calculated ξ (Re)](image)

**Figure 6.** Comparison of experimental and calculated ξ (Re).
The obtained results indicate a satisfactory convergence of calculating results of gas parameters using a mathematical model with experimental values. The convergence of the results along the spiral channel has a significant difference, the obtained values of a pressure drop using mathematical modeling are higher than the experimental values more than 70% in individual points. This may be due to the method of calculating average velocity and the resolution of a computational grid in a boundary layer and in the places of flow mixing, since the spiral configuration provides a highly turbulent flow. However, the error in describing the hydraulic resistance coefficient for the remaining channels is small and amounts to 5–15%, which gives reasons for using the developed mathematical model to calculate characteristics of similar channels of heat exchangers and elements of pneumatic systems.

Analyzing the dependences $\xi$ (Re) obtained in this work and the data on a thermal efficiency of channels from the previous works of the authors [5], we can conclude, that the use of a spiral channel from the point of view of hydraulic resistance at the Reynolds numbers up to 1200 is justified. In this range of Re, a channel with 3 internal ribs has similar values in terms of hydraulic resistance, but underperforms in terms of heat transfer characteristics. At high flow rates, the spiral channel has a hydraulic resistance more than 1.5 times higher than a channel with 3 internal ribs and can only be used to enhance thermal interaction.

Conclusions
Experimental and mathematical modeling of gas flow through channels of shell-and-tube additive heat exchangers of various configurations has been carried out in this work. Comparison of mathematical and experimental methods shows satisfactory convergence of the data. Based on the study of the thermal efficiency of various HE channels [5] and taking into account the dependences of hydraulic resistances obtained in this work, it may be concluded that the spiral-shaped channels are most efficient at flow regimes with Re <1200.

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