Modeling of thermal and gas dynamic processes in additive shell and tube heat exchangers

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Abstract. As part of a gas microturbine engine development, a number of new heat exchanger flow channels, which are produced by using additive technologies, are considered. Geometric three-dimensional models of several heat exchanger elements (with internal, external and combined finning – 3 and 6 rows of ribs, spade spiral of three sizes) are created. Also a mathematical modeling of heat carriers in their flow channels is carried out, taking into account heat transfer through the walls. The analysis of calculated thermohydraulic parameters, namely the heat power and pressure losses, is realized. The spade shape of the heat exchanger element allows intensifying heat exchange (comparable to a 3-fin pipe). It has lower hydraulic resistance compared to 6-fin pipes. It has been found that the change in a standard size of spade spiral pipes does not affect their heat exchange characteristics. The disadvantage of pipes with internal fins is a rapid increase of hydraulic resistance during the operation due to a deposition of microturbine exhaust gases soot.

1. Introduction

Heat exchangers are one of the most widely used devices in engineering. They are applied in housing and utilities, automobile industry, aviation, computer technology and many other fields, and modern production cannot manage without these heat transfer devices. A huge field of their application gives an impetus to the development of even more types of heat exchanger designs. Shell-and-tube heat exchangers, "tube-in-tube" heat exchangers, spray-type heat exchangers, coiled heat exchangers, plate heat exchangers have main distribution. Each type and design has its own disadvantages and advantages. High requirements for characteristics of heat exchangers are put forward by the energy sector, where even minor improvement of efficiency is especially important. Gas turbine plants have one of the leading positions in distributed energy. There is a high demand for effective heat exchangers with compact dimensions in this field. They act as heat exchangers in microturbines and save energy in a cycle of combustion, returning heat from the exhaust gases back into the working cycle. Microturbine recuperators have a significant impact on the plants efficiency, improving their designs. Increasing thermohydraulic characteristics of processes will enhance the efficiency of plants by several percent. Therefore, activities of various advanced companies and research institutes are aimed at creating more efficient and compact heat exchangers [2].

Taking into account the latest trends in technology development, modern methods of production and calculating gas dynamic processes which occur in the flow channels of heat exchangers are used. It has become possible to obtain an extremely complex geometric configuration of heat exchangers...
with the development of additive technologies. In this regard, it is necessary to carry out thermal and
gas dynamic calculations in order to optimize geometry of a heat exchanger, which is not limited by
the complexity of manufacturing using the classical method. A shell and tube heat exchanger is the
most suitable for additive manufacturing, because its design has directional elements-pipes. The main
task for achieving the maximum efficiency of a shell and tube heat exchanger is to develop an
effective spatial configuration of hot and cold heat carrier flow channels, a mathematical model for
calculating thermohydraulic parameters of the suggested designs and to select the most effective
design based on modeling and comparison of calculated data.

2. Review of research and calculation methods.
It is necessary to conduct experimental research to compare the gas dynamic and thermophysical
parameters of processes which occur in channels of a shell and tube heat exchanger. However, this is a
rather time-consuming and economically impractical way to determine the geometric configurations of
flow channels with maximum efficiency. In order to avoid the production of many experimental
samples and other difficulties during the testing, mathematical modeling is used. Developing a
mathematical model and conducting a series of computational experiments with subsequent analysis of
trends in parameter changes is the first step in improving the efficiency of a microturbine recuperator.
Based on the obtained calculated data, it is proposed to select the most effective configurations and
compare their characteristics using subsequent experimental studies.

A review of scientific publications on the topic of determining thermohydraulic parameters and
creating additive shell and tube heat exchangers [1–3] has shown that creating the most effective
configuration of a heat exchanger is an urgent task. For example, Fraunhofer IFAM has developed an
additive heat exchanger for a gas microturbine plant, which consists of 18 layers of channels. These
channels have a waveform and they are located as close to each other as possible. This helps to
maximize contact with heat carriers. The research also considers fractal structures that are often found
in nature. Many types of geometric surfaces are proposed, and computational studies are presented to
compare thermohydraulic parameters, but methods for obtaining these surfaces and parameters of their
effectiveness in complete plants relate to closed information.

3. Aims and objects of the research.
The aim of the research is to develop mathematical models and carry out computational research of
heat carrier flows and heat exchange in various configurations of heat exchanger elements for
microturbines, as well as to determine their thermal and hydraulic characteristics.

The choice of geometric configurations of the channels takes into account technological
requirements for additive production of an entire recuperator. For this purpose, the cylindrical surfaces
of the tubes are transformed into polyhedron. A space segment of a shell-and-tube heat exchanger flow
part, where one tube is set (the object of research), is considered. Taking into account current trends,
including trends which are considered in scientific publications, it is important to analyze the
following designs of the main heat exchange element: a tube without fins (v. 1); a tube with six
internal fins (v. 2); a tube with six external fins (v. 3); a tube with six external and internal fins (v. 4);
a tube with three internal fins (v. 5); a tube with three external fins (v. 6); a tube with three external
and internal fins (v. 7); a small-sized spade tube: coolant flow velocity increases in the segment
outside the tube (V. 8); a large-sized spade tube (V. 9); a medium-sized spade tube (V. 10).

Figure 1 shows three-dimensional models of researched tubes, whose half-axial sections allow us to
see the internal geometric configuration of the heat exchange surfaces.
4. Mathematical modeling
The considered objects are transformed into a number of computational domains, containing internal and external flow channels (a segment of a heat exchanger containing a single tube): liquid or gas domain, as well as the solid body domain – the heat transfer wall with or without a fin. Figure 2 presents a computational domain for tube v.1. Also characteristic surfaces are presented: $S_1$ is the surface of the working medium hot heat carrier inlet; $S_2$ – surface of the working medium hot heat carrier outlet; $S_3$ – the surface of the working medium cold heat carrier inlet; $S_4$ – the surface of the working medium cold heat carrier outlet; $S_5, S_6$ - the symmetry surface. Surfaces, which are not shown, are the boundary walls of a computational domain.

Figure 2. Computational domain of the v.1 object.

Computational domains for other research objects are considered in the same way. The mathematical model is based on a classical system of differential equations containing the equations of motion, continuity and energy of the working medium flow, supplemented by the equation of an ideal gas state, as well as the equation of thermal conductivity for the solid body domain. The equations are superimposed with assumptions about the Newtonian fluid, the isotropy of a solid, the presence of perfectly smooth boundary walls, and also the standard SST model of turbulence is used. The boundary conditions of the model are: the given flow rates and temperatures of working media on input surfaces $S_1$ and $S_3$ and pressures on output surfaces $S_2$ and $S_4$ equal to 101.3 kPa.

The complex of calculations based on the described mathematical model supplemented with boundary conditions is performed using the finite element method. In figure 2, the grid division of a computational domain of the research object No.1 is presented. A hexagonal grid with a boundary layer of 4 elements is used mainly. The calculation model of the object v. 1 is a reference model. It uses a dense grid of finite elements. It allows achieving high accuracy of the obtained parameters.

The most important stage in creating workable mathematical models is verifying their adequacy by comparing the calculated data with experimental data that will be obtained in further research works. Modeling of heat exchange between water as a cold coolant with an inlet temperature of $T_w=15^\circ C$ and air as a hot coolant (inside the tubes) at temperatures $T_1=40^\circ C$, $55^\circ C$ and $70^\circ C$ in object v. 1 has shown that the external tube walls are completely thermostated with water, and the temperature drop is no more than 0.6°C at a water flow rate of 1 kg/s. Therefore, it is assumed to use water in the test rig.
[5]. This water is moved in the flow part with a specified flow rate as a cold coolant with the aim of thermostating external walls. Due to the calculated data obtained for the objects of study v. 2...v. 10, modeling of external water flow is not applied. The temperature of tube external surface equal to the temperature of running water \( T_W = 15\,^\circ\text{C} \) is assigned. Parameters of the flow of air, being a hot coolant with inlet temperatures \( T_1 = 40\,^\circ\text{C}, 55\,^\circ\text{C} \) and \( 70\,^\circ\text{C} \), as well as different flow rates \( G = 0.03; 0.06; 0.09; 0.12; 0.15 \, \text{g/s} \) are calculated (according to the range of average speeds available in the flow parts of microturbine recuperators). The outlet parameters are the values of outlet air temperature and the pressure drop between the inlet and outlet surfaces of the working medium. The value of the \( y^+ \) criterion is controlled, which value is about 1. It is acceptable for describing the wall boundary layer by the SST model.

The results of calculations for objects v. 3 and v. 6; v. 2 and v. 4; v. 5 and v. 7, respectively, show the same results, because the influence of external edges is not considered at this stage, they are thermostated with water. Further, these results will not be duplicated in the shown diagrams.

5. **Evaluation of thermohydraulic parameters**

Based on the results of numerical modeling using the finite element method, the values of density \( \rho \), air velocity \( u_{ave} \), outlet air temperature \( T_2 \), and inlet pressure \( p_1 \) are obtained. Considering these primary data, a pressure drop \( \Delta \rho \), an average temperature drop \( \Delta T \) (1), the coefficient of hydraulic resistance \( \xi \) (an evaluation is based on the Darcy-Weisbach equation) (2), the Reynolds number \( \text{Re} \), heat power \( Q \) (3) and the thermal efficiency \( \eta \) (4) are calculated [4].

\[
\Delta T = \frac{(T_2 - T_1)}{\ln \left( \frac{T_W - T_1}{T - T_2} \right)} \quad (1)
\]

\[
\xi = \frac{2 \cdot (p_1 - p_2)}{\rho \cdot u_{ave}^2} \quad (2)
\]

\[
Q = G \cdot c_p \cdot (T_1 - T_2) \quad (3)
\]

\[
\eta = \frac{c_p \cdot (T_1 - T_2)}{c_{pw} \cdot (T_1 - T_W)} \quad (4)
\]

6. **Results**

Diagrams for the considered heat exchanger elements based on the obtained calculation results are created. By means of the thermohydraulic parameters of these research objects are estimated. Evaluation diagrams of the thermal efficiency \( \eta(\text{Re}) \) and hydraulic losses \( \xi(\text{Re}) \) are presented in a figure 3.
Figure 3. Calculated dependences of thermal efficiency on the Reynolds number (a) and the coefficient of hydraulic resistance on the gas flow (b).

The dependence of heat power on the pressure loss \(Q(\Delta p)\) is used for combined estimation of thermohydraulic characteristics. It is shown in a figure 4 for an inlet air temperature of 55°C.

Figure 4. The dependence of heat power \(Q\) on the pressure loss \(\Delta p\) for an inlet air temperature of 55°C.
Conclusions
Using the developed mathematical model of coupled heat exchange, the thermohydraulic characteristics $\eta(Re)$, $\xi(Re)$, $Q(\Delta p)$ have been calculated for different types of tubes of additive shell and tube heat exchanger in gas turbine unit. The heated air passage through the internal channel of research objects of ten configurations, which are thermostated by a cold heat carrier, has been modeled. For each of tube variants v. 1...v. 10, the calculation was performed for the inlet to the channel at air temperatures $T_1=40^\circ$C, $55^\circ$C, $70^\circ$C and air flow $G=0,03; 0,06; 0,09; 0,12; 0,15$ g/s.

Comparison of data has shown that the spade shape of a heat exchanger element allows intensifying the heat exchange (comparable to a 3-fin tube). In addition, it has a lower hydraulic resistance compared to 6-fin tubes. It has been found that the change of the spade spiral tubes standard size does not affect their heat exchange characteristics. The disadvantage of pipes with internal fins can be a rapid increase in hydraulic resistance during operation due to the deposition of microturbine exhaust gas soot.

The calculated research shows a high potential of using additive technologies to improve the efficiency of microturbine recuperators. In particular, the use of spiral spade tube shapes is recommended, because they have high thermal and hydraulic characteristics and appear to be less susceptible to contamination.

References
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