Studying the efficiency of cooling and resistance of ribbed tubular elements

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Abstract The cooling of compressed medium when passing through the set of ribbed tubular elements is numerically investigated. Numerical experiments are carried out with the model of viscous heat conductive turbulent compressed gas by the control volume method with application of different numerical schemes and algorithms. The process of convective heat exchange between the cold ribbed surface and the hot gas flow is studied as well as the heat conductance processes in tubular elements. After processing the obtained results recommendations are given on the arrangement of the set of ribbed tubular elements to provide the assigned temperature characteristics of the cooling system; and the engineering technique of evaluating the thermohydraulic parameters of the cooling system is developed.

1. Introduction
Ribbed tubular elements are commonly used in various power plants equipped with convective thermal control devices [1, 2]. In most cases they are made as tubular bunches or sets characterized by a high ratio of heat transfer, resistivity to mechanical actions and loads, and ability to withstand the increased pressure and abrupt changes of operational conditions. The layout scheme of such sets depends on parameters of heat exchange devices [2]: the achieved difference of temperatures, the necessary level of flow turbulence, and the value of hydrodynamic resistance.

Improving the performance of the thermal control equipment is the urgent task for the power industry. The applied layout of the tubular bunch should provide the necessary temperature reduction of the operating medium during its passing through the tubular bunch; it can be achieved by increasing the number of ribbed tubes and/or decreasing the distance between them. However, installation of additional cooling elements and their proximity will inevitably lead to the increase in hydraulic resistance. The optimal layout should provide the required cooling of the flow at minimum allowable losses.

The most progressive approach to analyze the parameters of various layouts when designing new heat exchange devices and improving the existing ones is the numerical simulation based on equations of motion and heat transfer.

This work presents the results of three-dimensional analyses of the turbulent flow and heat exchange in sets of ribbed tubes arranged across the air flow motion. The influence of operating parameters of the cooling system (consumed mass and thermophysical characteristics of the cooled gas flow) and the layout of cooling elements on the efficiency of this cooling system is studied.
The present research analyses of viscous, compressible and heat-conducting gas flow are based on computational gas dynamics methods. Libraries of the public integrable platform OpenFoam [3] for solution of medium mechanics problems and additionally the public integrable platform Salome [4] are used for computational modeling.

2. Mathematical simulation
The conjugate problem of heat exchange between the flow of the viscous heat conductive gas and the bunch of ribbed tubular elements was considered (Figure 1). The tubular bunch was made as two rows of aluminum tubes with the diameter d arranged checkerwise. Ribbing of tubes was ring-shaped and transverse; ribs were similar and uniformly arranged along the carrying tube with the pitch 0.16d, the height of ribs was 0.44d and the thickness of ribs was 0.04d. The distance between tubes (from the centre to the centre) along the cooled flow motion was 4d. In the transverse direction the cooling elements were arranged at the same distance from each other and from lateral walls; the distance between tubes varied depending on their number n. Dimensions of the computational area along the flow motion was chosen by the condition of minimizing the influence of the input and output boundaries on the tube flow around.

![Figure 1. Computational area.](image)

The equation system was solved for mathematical simulation of the viscous compressed gas motion; and it included the additional condition equation:

\[
\frac{\partial p}{\partial t} + \frac{\partial pu_j}{\partial x_j} = 0 \quad (1)
\]

\[
\frac{\partial pu_i}{\partial t} + \frac{\partial pu_i u_j}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu \delta_{ij} \right) + F_i \quad (2)
\]

\[
\frac{\partial pE}{\partial t} + \frac{\partial pE u_j}{\partial x_j} = -\frac{\partial p u_j}{\partial x_j} + \frac{\partial u_j}{\partial x_j} \left( \frac{\partial u_i}{\partial x_i} + \frac{\partial q_i}{\partial x_i} \right) + F_j \mu_j \quad (3)
\]

\[
\frac{\partial T}{\partial t} = \lambda \nabla^2 T \quad (4)
\]
\[ p = \rho RT \] (5)

The following designations were used in the above formulas (1)-(4):
- \( \rho \) – density;
- \( u_i \) - velocity components \( u \);
- \( p \) – pressure;
- \( \mu \) - dynamic-viscosity coefficient;
- \( F_i \) – external volume force;
- \( E = C_i T + 0.5u_i^2 \) – total specific energy;
- \( H = E + p / \rho = C_i T + 0.5u_i^2 = h + 0.5u_i^2 \) – total specific enthalpy;
- \( \tau_{ij} = 2\mu\delta_{ij} - \frac{2}{3}\mu\frac{\partial u_i}{\partial x_j} \) – viscous stress tensor;
- \( S_{i,j} = \frac{1}{2}\left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \) - strain velocity tensor;
- \( q_j \) - heat flux;
- \( \lambda \) - heat-conduction factor;
- \( T \) – temperature;
- \( R = 287 \text{ J} / (\text{kg} \cdot \text{K}) \) – specific gas constant.

Coefficients of molecular viscosity heat \( \mu \) and conductivity \( \lambda \) depended on the temperature in accordance with [5].

Initial equations were averaged according to Favre [6]. Values were given as a sum \( \bar{\theta} = \rho \bar{\theta} / \bar{\rho} \), where \( \bar{\rho}, \bar{\theta} \) were the averaged parameters according to the Reynolds procedure. Thus, \( \bar{\theta}^* \) included both turbulent and density fluctuations. The Navier-Stocks averaged system was as follows:

\[ \frac{\partial \bar{\rho}}{\partial t} + \frac{\partial \bar{\rho}u_i}{\partial x_i} = 0 \] (6)

\[ \frac{\partial \bar{p}u_i}{\partial t} + \frac{\partial \bar{p}u_j u_i}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial \bar{\rho}}{\partial x_j} \left( \bar{u}_j + \bar{u}_i \right) + \bar{F}_i \] (7)

\[ \frac{\partial \bar{\rho}E}{\partial t} + \frac{\partial \bar{\rho}E u_i}{\partial x_j} = -\frac{\partial \bar{p}u_i}{\partial x_j} + \frac{\partial \bar{\rho}}{\partial x_j} \left[ \bar{u}_j \left( \bar{u}_j + \bar{u}_i \right) \right] + \frac{\partial \bar{\lambda}}{\partial x_j} \left( \bar{q}_j + \bar{q}_i \right) + \bar{F}_j \bar{u}_i \] (8)

\[ \bar{\rho} = \bar{\rho}R\bar{T} \] (9)

The designations in formulas (6–9) were specified according to [7, 8].

The total temperature and average velocity of the oncoming flow were assigned at the input of the computational area. The atmospheric pressure was assigned at the output. The conjugation condition at the solid/gas boundary was set [9].

Discretization of the computational area was made by means of hexahedral and tetrahedral elements with their total number of 4.3 mln elements including prismatic cells for near-wall flows.

The stationary problem was solved by the relaxation method (mass conservation law) based on compliance with the condition: \( \text{RMS} < 10^{-6} \). Discretization of basic equations was made by the finite volume method with account of Rhie-Chow correction. The counter-flow scheme of the second degree of accuracy was applied in order to discretize non-viscous flows and the central scheme of the second degree
of accuracy was applied for viscous ones. The difference equation system was solved algebraically by the multi-grid method with the conjugate gradient method applied for rapid convergence.

3. Calculation results

As a result of the series of computational experiments, the flow structure as fluid flow lines and the fields of gas dynamic and thermophysical values have been obtained. Figures 2 and 3 show the calculation results for the version of cooling the air flow with the temperature of 330 K and velocity of 8.5 m/s during its passing through the set of 5 tubular elements with the temperature of inner surface of tubes equal to 273 K.

The structure of gas flow (Fig. 2) is characterized by formation of detached areas with the shape and dimension specified by the scheme of the cooling element arrangement. Reverse flows with vortex pairs are generated behind tubular elements in flow separation zones. The arrangement of vortexes behind the central cooling elements of the first row is close to symmetrical, while the symmetry is broken behind the side elements and vortex zones are shifted to walls by the flow. Detached areas are more elongated and asymmetrical behind the tubes of the second row.

![Figure 2](image-url)  
**Figure 2.** Fluid flow lines in the longitudinal section at the oncoming flow velocity of 8.5 m/s.

Figure 3 shows the temperature and pressure distribution along the transverse plane of the flow. The local temperature ranges from 319 to 335 °K, and local pressure – from 101306 to 101417 Pa. The obtained fields of pressures and temperatures completely agree with the flow structure; the maximum temperature drop is registered at the low-pressure zone behind the ribbed tubular elements. In this case, the temperature drop for the air flow is ~5.7 K on average, with the pressure loss of ~37 Pa.

![Figure 3](image-url)  
**Figure 3.** Distribution of temperature (a) and pressure (b) at the oncoming flow velocity of 8.5 m/s.

Similar calculations were made for other cooling modes. The studied range of the input flow temperature variation was 240…340 K, the temperature on the inner surface of aluminum tubes was
conditionally taken to be constant along the whole length of the tube and varied from 210 K to 280 K, and the velocity at the input varied from 3.7 to 8.2 m/s. The number of cooling elements varied from 1 to 12. The flow structure was constant at variation of consumed mass and thermophysical parameters of the oncoming gas flow and similar to the behavior of the flow in the considered version (Figures 2, 3).

In order to verify data, calculations were simultaneously made by means of various computational algorithms and schemes of different degrees of accuracy. Divergence of the obtained results did not exceed 8% for the pressure and 3% for the temperature.

Analysis of the obtained results allowed evaluating the efficiency of gas flow cooling by one tube and proving the sufficiency of two rows of ribbed tubes for gas flow cooling. The optimal arrangement of ribbed tubular elements was determined in order to achieve the maximum intensity of cooling at minimum pressure losses; recommendations on the cooling system layout were made on its basis.

In accordance with summarized computational data, the engineering technique is developed to evaluate thermohydraulic parameters of the cooling system on the basis of ribbed tubular elements. This technique involves dimensionless equations relevant for this type of structures:

– to evaluate the dimensionless hydraulic resistance coefficient:

$$\xi = 0.44 \cdot \text{Re}^{0.16} \cdot \text{n}^{0.19}$$

(10)

– to evaluate the Nusselt number at the output section in accordance with the temperature difference at entering the tube bunch and leaving it:

$$\text{Nu} = 2.4 \cdot 10^{-11} \cdot \text{Re}_d^3 \cdot \text{Pr}^{-0.2}$$

(11)

were \(n\) – is number of ribbed tubular elements and the Reynolds and Prandtl numbers vary in the range: \(\text{Re}_d = 2.5 \cdot 10^3 \div 12.5 \cdot 10^4; \text{Pr} = 0.6 \div 0.85\).

Conclusions

Conjugate problem of heat exchange in the device for cooling the flow of heat conductive compressed air has been solved in this work. Based on the analysis of the obtained results the arrangement of tubular elements has been optimized and the serviceability of the chosen cooling system has been proved for the assigned operation conditions. The set of obtained dimensionless equations provides for express evaluation of the efficiency and serviceability of the considered type of cooling systems with the required number of tubular ribbed elements.

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