1. Introduction

The transverse flow around the cylinders covers most of the classical hydrodynamics problems [1]. In practice, the stability of the boundary layer and its transition from laminar to turbulent state and the location of its separation on a convex surface play an important role [2]. A sharp decrease in hydraulic resistance associated with the turbulence of the boundary layer has been discovered in 1912 in experiments with the flow of the ball [3]. Combining cylinders into bundles allows intensifying heat transfer additionally, especially when forming their outer surface with holes or grooves [4, 5]. Pipe bundles are characterized by manufacturability and strength. Their positive qualities are especially important when used in high-pressure equipment. Studies of hydrodynamics and heat transfer in pipe bundles are very relevant for the creation of efficient gas turbines [6].

Heat exchangers with cylindrical tubes, the surface of which is formed by spiral grooves, have long been known. The first patent for this type of device has been issued in Britain in 1887. Now these heat exchangers are manufactured in India, Italy, South Korea and other countries. Intensification of heat exchange in such tubes bundles occurs not only due to turbulence of the coolant, which shifts the flow separation in the stern part of the tube, but also as a result of its asymmetry due to the formation of a transverse velocity component in the vortex zone behind the tube. However, there is little data of heat transfer and hydraulic resistance of tubes with spiral grooves.

2. Literature review and problem statement

In [7], a simulation of laminar mixed convection of heat transfer from a series of three isothermal square cylinders is
presented to study the behavior of fluid flow around these cylinders. Numerical results are presented and discussed for
the range of Reynolds numbers $Re = 10^{-40}$, with a fixed value of the Prandtl number $Pr = 1$ and with a fixed geometric config-
uration. The total coefficient of resistance and the average Nusselt number for each cylinder have been determined.

In [8], a two-dimensional mathematical research has been performed to investigate the effect of the distances
between the gaps for the flow through a series of five rectangular cylinders. The parameters of the gaps between the
cylinders changed systematically. The distances between the gaps significantly influence the hydrodynamic interaction
of the cylinders, which affects the flow structure. The flow behavior has been grouped into five models: symmetric,
rotational, phase and antiphase modulated rejected, phase modulated asynchronous, and phase modulated synchronous.
The Struhal number and the average humidity coefficient have been studied. It has been found that they differ significantly depending on the distance between the gaps. In this work, the dependence of the hydrodynamics parameters for another cylinder cross-sectional shape has not been investigated.

The work [9] is aimed at studying the characteristics of laminar flow and heat transfer of Newtonian fluid for a num-
er of semicircular cylinders, which are placed in a uniform configuration of the transverse flow. The effect of the ratio
between the diameter and the gap and heat transfer have been studied with the value of the Reynolds number of 100
for air as a working fluid. Variations of the measured global quantities have been studied and discussed in detail: coeffi-
cient of resistance, Struhal number, Nusselt number.

In [10], a two-dimensional heat transfer with natural convection of the stationary type from a series of closely spaced isothermally heated cylinders has been investigated for the laminar flow regime. In this study, numerical and statistical simulations were performed to propose the ratio of the average Nusselt number for one row of horizontal cylinders immersed in molten solar salt. The influence of the shape of the cylinder surface on the heat transfer efficiency has not been studied.

In [11], research of the two-dimensional flow of non-Newtonian uncomprssed fluid around isothermal cylinders with a virtual physical model is presented. The verification of the self-made dynamic modeling code has been performed for two specific cases. In the first case, the effect of heat transfer in the flow around the group of three cylinders has been studied by forced and mixed convection. In the second case, different numbers of cylinders have been arranged in a single-row configuration with different distances between them. Flow lines, isotherms, Struhal and Nusselt numbers are presented. The results confirmed the large influence of the distances between the cylinders on the aerodynamic coefficients and on the Nusselt number.

In [12], the results of an experimental researching of the average heat transfer and hydraulic resistance in the trans-
verse flow of a single-row bundle of circular cylinders with spiral ribs on the outer surface are presented. The obtained results can be considered as limiting at very wide grooves.

In [2], it has been shown that one of the methods to improve the operation of tubular heat exchangers is the use
of cylinders with a rough outer surface. For example, an increase in the relative roughness of the sand type from 0.004
to 0.007 causes a decrease in hydraulic resistance during the air flow by 30%. Using of recesses in the form of ellipsoidal
segments, in particular placed in curved tracks, gives even better results [13]. The roughness of the sand type turbulutes
the boundary layer over the entire surface of the cylinder, while the tape depths, as in [13], can create a non-uniform velocity field, which will provide additional mixing of the flow. Note that in technological terms, the method of heat transfer intensification used in [13] is less technological. An option to overcome technological difficulties may be using spiral grooves.

Thus, the study of heat transfer and hydraulic resistance of a single-row bundle of tubes with intensifiers such as spiral
grooves is appropriate and useful for engineering practice. Moreover, even in the textbook for universities [14] and in
its latest electronic citations, the heat transfer characteristics of the first two rows of the tube bundle are questioned.

3. The aim and objectives of the study

The aim of the study is to determine the hydraulic resistance and heat transfer of cylindrical surfaces with grooves suitable for use in engineering practice, for example, in heat exchangers of gas turbines.

To achieve this goal, the following objectives were set:
- determine the magnitude of the increase in heat transfer of cylindrical heat exchange surfaces with grooves;
- evaluate the impact of using screw grooves on the hydraulic resistance of the proposed heat exchange surfaces;
- experimentally investigate the effects of non-identical arrangement of tubes with grooves in the bundle.

4. Materials and methods of research

The experimental part of the work has been performed on an open wind tunnel, the scheme of which is shown in Fig. 1.
The list of applied devices is presented in Table 1.

![Fig. 1 Scheme of experimental installation: 1 – fan; 2 – conical part of the working area; 3 – control valve; 4 – honeycomb; 5 – input anemometer; 6 – moving Pitot tube (moves along the flow); 7 – wet thermocouple; 8 – dry thermocouple; 9 – piezoceramic sensor; 10 – frequency meter; 11 – analog-to-digital converter; 12 – meter of relative humidity, temperature, speed; 13 – digital voltmeter; 14 – micromanometer; 15 – air velocity and inlet temperature meter; 16 – Pitot tube (moves across the stream); 17 – lower beginning of the groove (its position is the same for all cylinders, except the middle); T1 – mercury thermometer; T3 – ice calorimeter; p1, p2 – static pressure measurements to determine hydraulic resistance]
The experimental setup worked «on suction». Room air (in most experiments with a temperature of 20 °C) was passed through a system of measuring instruments and after washing the bundle of cylinders with grooves, the middle of which was a calorimeter, was released into the environment. In front of the fan, an additional pipe with a valve 3 was installed for the suction of air from the atmosphere in order to regulate the main flow (in the direction of its reduction). The channel walls of the working area were made of organic glass with a surface roughness of not more than $R_z = 1.0 \mu m$.

The total length of the channel is 970 mm.

The outer diameter of all cylinders $D$ was 22 mm, the length of each tube was 220 mm, and the working length, within the cross-section of the channel, $l = 105$ mm. The relative transverse pitch of the cylinders in a single-row bundle was $s_1/D = 1.7$. On the outer surface of the pipe, single-way or double-way screw groove of rectangular cross-section with the depth of 1.8 mm and width of 3.0 mm was made. The investigated cylinder-calorimeter was placed in the middle of a single row (Fig. 1). The parameters of the studied cylinders are given in Table 2.

The view of the studied cylinders is shown in Fig. 2.

### Table 2

| Marking | Spiral step, mm | Number of ways |
|---------|-----------------|---------------|
| SPI 40  | 40              | 1             |
| SPI 20  | 40              | 2             |
| SPI 10  | 20              | 2             |
| Sm      | Smooth cylinder |

The amount of heat removed from the air for the process of moisture condensation has been determined by the equations of heat balance using the displays of anemometers 15 and 12 (Fig. 1), which in addition to speed and temperature measured the relative humidity.

The average values of the measured values, absolute and relative errors are summarized in Table 3. Errors are determined taking into account the recommendations [16]. Comparison of errors in determining the heat transfer coefficients by the temperature difference in the wall and using the calorimetric method shows that the use of calorimetry can reduce the error by more than three times.

### Table 3

| Parameter designation | Average value | Dimensionality | Absolute error, $\Delta$ | Relative error, $\varepsilon$, % |
|-----------------------|---------------|----------------|--------------------------|---------------------------------|
| $T$                   | 180 sec       | sec            | 0.05                     | 2.78·10$^{-2}$                  |
| $\Delta$              | 0.002 m       | m              | 0.00001                  | 5.0·10$^{-1}$                   |
| $S$                   | 2.426·10$^{-3}$ m | m          | 0.00005                  | 2.06                            |
| $F$                   | 6.283·10$^{-3}$ m | m²        | 8.0·10$^{-4}$            | 9.26·10$^{-2}$                  |
| $V_T$                 | 1.248·10$^{-5}$ m³ | m³       | 88.36·10$^{-9}$          | 7.08·10$^{-1}$                  |
| $\rho_w$              | 999.9 kg/m³  | kg/m³         | 0.05                     | 5.0·10$^{-3}$                   |
| $\rho_m$              | 7900 kg/m³   | kg/m³         | 50                       | 6.33·10$^{-1}$                  |
| $t_i$                 | 20.0 °C       | °C            | 0.05                     | 2.5·10$^{-1}$                   |
| $t_f$                 | −12.9 °C      | °C            | 0.05                     | 3.88·10$^{-1}$                  |
| $\Delta t$            | −12.9 °C      | °C            | 0.05                     | 3.88·10$^{-1}$                  |
| $\Delta m_e$          | 4.0 °C        | °C            | 0.4                      | 10.0                           |
| $Q$                   | 11323 J       | J             | 39.89                    | 8.27·10$^{-1}$                  |
| $R$                   | 334000 J/kg   | J/kg          | 500                      | 1.50·10$^{-1}$                  |
| $c_v$                 | 500 J/(kg K)  | J/(kg K)      | 0.5                      | 1.0·10$^{-1}$                   |
| $Q$                   | 30.8 W        | W             | 0.0432                   | 3.42                           |
| $q_T$                 | 28.28 W       | W             | 0.0399                   | 3.36                           |
| $\Lambda$             | 0.55 W/(m-K)  | W/(m-K)       | 0.005                    | 0.909                          |
| $\Lambda_{cal}$       | 16.0 W/(m-K)  | W/(m-K)       | 0.5                      | 3.12                           |
| $A$                   | 160.0 W/(m²-K)| W/(m²-K)      | 6.96                     | 4.26                           |
| $\alpha_T$            | 160.0 W/(m²-K)| W/(m²-K)     | 15.66                    | 10.55                          |

Thus, the error in determining the basic values complies with the requirements of the thermophysical experiment.

*Fig. 2. Investigated heat transfer surfaces: 1 — cylinder with a smooth surface; cylinders with spiral grooves; 2 — $S = 10$ mm; 3 — $S = 20$ mm; 4 — $S = 40$ mm*
5. Results of the research of hydraulic resistance and heat transfer during air flow of a row of cylinders with grooves

5.1. Visualization of the stagnant zone behind the cylinders and comparison with computer calculation

Due to the small size of the grooves, the influence of sensors (Pitot tubes, thermocouples) on the results of thin flow elements measurements could be unacceptably significant. Therefore, mathematical modeling has been used, the reliability of which was checked not only by the balances of the amount of heat, but also by the size of the vortex zone behind the middle cylinder in the beam. The method of numerical modeling was used for the research, using the ANSYS CFX software package. The results of calculations with turbulence models RNG_\(k\varepsilon\), LRR and SSG were compared. The RNG_\(k\varepsilon\) model solves 2 additional equations for the kinetic energy of turbulence and dissipation. The LRR model solves the Reynolds averaged Navier-Stokes equation. The Reynolds SSG model is based on the equation of kinetic energy fluctuations. The range of Reynolds numbers for hydrodynamics and heat transfer was 6000<\(Re\)<16000.

The results of verification of turbulence models are presented in Fig. 3.

![Fig. 3](image)

**Fig. 3.** Dependence of the length of the vortex zone on the middle cylinder on the Reynolds number for different models of turbulence: Ex — experiment (photographing a stagnant zone filled with soap bubbles). At low flow velocities, the model RNG_\(k\varepsilon\) shows the smallest error (10.2 %)

Visualization of the vortex zone behind the cylinder was performed using laser-illuminated soap bubbles.

The length of the vortex zone behind the cylinder with a groove pitch of 20 mm is 42 % less than the length of the vortex zone behind the smooth cylinder. Fig. 5 (computer simulation) shows that vortex cords appear in the spaces between the tubes in the area of the groove passage, which intensify the heat exchange on the smooth sections of the adjacent tubes. The interaction of vortices generated in the grooves and on the body of the pipes is shown in Fig. 6 [17].

![Fig. 4](image)

**Fig. 4.** Average heat transfer coefficient of the central cylinder depending on the Reynolds number for different turbulence models (computer calculation) RNG_\(k\varepsilon\), LRR, SSG and experimental data for smooth cylinders: Is — with [14] and SPI_20 — the authors’ experiments with cylinders with grooves (groove pitch 20 mm)

In computer simulations, the smallest difference with the experiment was observed in the model RNG_\(k\varepsilon\) (3.4 % at the maximum Reynolds number \(Re=15804\)).
5.2. Heat transfer and hydraulic resistance in single-row bundles of cylinders with spiral grooves

The results of experiments with different steps of the grooves are shown in Fig. 9, 10 [18]. The hydraulic resistance characteristics of the bundle are shown in Fig. 9.

The surface of the cylinders with a groove step of 40 mm (single-way groove) has the greatest asymmetry and, as a result, the lowest resistance. The average heat transfer coefficient for smooth cylinders was compared with the results given in [19].

\[
Nu = 0.26Re^{0.65}Pr^{0.32} \left( Pr/Pr_a \right)^{0.25}.
\]  

(1)

The amendment was taken into account for the first row [14].

Similar results characterize heat transfer. In Fig. 10, cylinders SPI_40 (with single-way groove) show better heat dissipation than SPI_20_2 (with double-single groove), although the actual heat transfer area is larger in the last.

Similar conclusions can be made by the Reynolds analogy (Fig. 11), based on the calculated results of the controlled experiments presented in Fig. 9, 10.

The dependences of the relative increase in heat transfer \( \frac{Nu}{Nu_0} \) on the relative increase in the hydraulic resistance \( \frac{Eu}{Eu_0} \) due to the formation of grooves are shown in Fig. 11, obtained in the same velocity range (4000 ≤ Re ≤ 16000).

5.3. Influence of non-identical axial arrangement of cylinders in a bundle on hydraulic resistance and heat transfer

The reason for the increase in heat transfer for cylinders with «frequent» grooves (for grooves with small steps and double ways) can be considered a simple increase in heat transfer surface. At high velocity, the relative percentage of heat transfer increase is almost equal to the percentage of the actual increase heat transfer area (lines SPI_20 and SPI_10 in Fig. 11).

The big step and single-way execution of a groove make this dependence opposite. The groove with a step of 40 mm on
the SPI_40 cylinder with an increase in the heat exchange surface only by 10% (Fig. 11) provided the growth of heat transfer by 30%, and the significant reductions in hydraulic resistance.

The relative position of the bundle tubes due to the existence of hydrodynamic traces of the grooves can affect both heat transfer and hydraulic resistance.

In the initial version, all the cylinders in the bundle were in the same position, so that the inputs of the spiral grooves were at identical points (for example, on the front generators). This option was taken as a starting point for comparison. The rotation of the cylinder-calorimeter around its axis caused a change in heat transfer and to a lesser value of hydraulic resistance. (Fig. 12, 13).

![Graph showing heat transfer and hydraulic resistance](image)

**Fig. 12. Reaction of the average heat transfer of the central cylinder SPI_40 bundle to its azimuthal position among neighbour cylinders: D — calculation results for [19]; Vzd — all cylinders with the bottom beginning of the groove (Fig. 1 position 17) turned back; Cont — all cylinders with the bottom beginning of the groove turned back, one central — forward; 90r — all cylinders with the bottom beginning of the groove turned back, one central — 90° to the right; 90L — all cylinders with the bottom beginning of the groove turned back, one central — 90° to the left; Vsinazad — all cylinders with the bottom beginning of the groove are turned towards the flow; CentrContr — all cylinders with the bottom beginning of the groove are turned towards the flow, one central — along the flow; CentrVpered — all cylinders with the bottom beginning of the groove are turned towards the flow.**

![Graph showing hydraulic resistance](image)

**Fig. 13. Increase in the hydraulic resistance of the bundle as a result of ignoring the identity of the placement of the grooves on the cylinder.**

Fig. 11 shows that the rotation of the tubes relative to their axis slightly increases the vorticity of the flow, although there are few such experiments (CentrVpered). If the rotation occurred in the same position in all tubes, for example (Vzd) and (Vsinazad), the difference in the heat transfer characteristics is also small. Even an additional 90°left turn (90L) does not cause significant changes in the situation. The most significant turns were 90° (90R) and 180° (CentrContr).

For increasing the heat transfer coefficient, it is necessary to «pay» pressure losses (Fig. 13). The largest change in the pressure drop across the bundle occurred after turning the middle cylinder to the right by 90° (90R) and the entire row by 180°(Vsinazad). The received recommendations concern bundles of tubes with a «moderate» frequency of elements placement. In this case, bundles with a transverse step of 1.73 tubes were investigated.

6. Discussion of the results of the study of heat transfer and hydraulic resistance during air flow of a number of cylinders with grooves

Intensification of heat transfer on the external surface of the cylinders using spiral grooves exceeds the mechanical increase in the heat transfer area, created by the grooves.

To some extent, the twist of the groove with a large step has advantages in terms of heat transfer over spirals with a small step (Fig. 10). This is due to the positive effect of the asymmetry flow of the cylinder with a spiral groove caused by the difference in the degree of turbulence of the flow on the right and left parts of the cylinder surface within one step of the spiral.

The groove located closer to the narrowest section of the channel in the cylinder bundle causes greater turbulence of the flow and thus provides the flow with greater stability. The separation of the flow from the cylindrical surface is delayed and there is a transverse component of the velocity, which reduces the stagnation zone behind the cylinder and helps to reduce hydraulic resistance. Therefore, two parallel grooves work worse than the same, but spaced at the end of the cylinder diameter (Fig. 10).

The disadvantage and at the same time the limitation of this study are the relatively short cylinders on which the experiments were performed. To expand the range of Reynolds numbers, it is desirable to have a compressor with a higher flow rate. In the future, it is possible to use an air dehumidifier at the entrance to the experimental place. A more powerful computer will allow you to build a more detailed three-dimensional model of the studied phenomenon.

7. Conclusions

1. The formation of spiral grooves on the external surface of cylindrical tubes does not cause significant technological difficulties. It has been experimentally shown that shallow grooves do not create danger for the strength of tubes, but make effective turbulentization. In the zone of transient modes, which is often used in practice (3000<Re<16000), the increase in heat transfer from 30 to 70 % was achieved. The increase in the heat transfer surface by grooves did not exceed 12 %.

2. As a result of the difference of the places of the beginning of turbulence of a flowing stream on the right and left part of the tube caused by the existence of a spiral groove, there was an additional cross component of velocity, which promoted the reduction of a stagnant zone behind the cylinder. This phenomenon caused a decrease in the hydraulic resistance of the bundle by 16–12 %, which in turn led to an increase in the Reynolds analogy from 10 % to 65 %.
3. Rotation of the central calorimeter with the largest pitch of the groove (40 mm) by 90° relative to the axis caused an increase in the heat transfer coefficient from 60 % to 35 %. For comparison, the case was chosen when all the cylinders in the bundle occupied identical positions. The greater influence of the relative position was noticeable in the range of lower velocities. Rotation of the central cylinder by 180° affected the heat transfer approximately twice less. At low flow rates, there were cases of reduction of heat transfer by 25 %. Ignoring the identity of the location of the grooves on the cylinders caused an increase in hydraulic resistance by an average of 40 %.

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