UDC 536.24: 541.11
DOI: 10.15587/1729-4061.2020.205656

RESEARCH OF HYDRODYNAMICS AND HEAT TRANSFER DURING THE TRANSVERSE AIR FLOW OF A ROW OF CYLINDERS WITH SCREW GROOVES

A. Khalatov
Doctor of Technical Sciences, Professor,
Academician of NAS of Ukraine, Head of Department*
E-mail: Artem.Khalatov1942@gmail.com
G. Kovalenko
PhD, Senior Researcher*
E-mail: gkliashkova5@gmail.com
M. Muliarchuk
Postgraduate Student
Department of Power Systems Physics
National Technical University of Ukraine
«Igor Sikorsky Kyiv Polytechnic Institute»
Peremohy ave., 37, Kyiv, Ukraine, 03056
E-mail: mariyamul997@gmail.com
*Department of High-Temperature Thermogasdynamics
Institute of Engineering Thermophysics
of National academy of sciences of Ukraine
Mariyi Kapnist str., 2a, Kyiv, Ukraine, 03057

Поперечне обтікання циліндрів – поширене явище в багатьох галузях техніки. Технологічна простота виконання трубчастих конструкцій робить їх привабливими, особливо при застосуванні робочих тіл, які знаходяться під різними величинами тиску. Разом з тим, циліндри відносяться до категорії «погано обтічних» тіл, і існує широка можливість покращення їх гідродинаміки і тепловіддачі. Для кругового циліндра існує діапазон швидкостей, в якому його гідравлічний опір може зменшуватись від деформації поверхні циліндра. Це явище може бути використане для раціонального проектування теплообмінників.

В аеродинамічній трубі відкритого типу були визначені коефіцієнти тепловіддачі та гідравлічні опори однорядних пучків циліндрів з декількома типами спіральних канавок на зовнішній поверхні. Найбільший приріст тепловіддачі (64 %) показав циліндр з найменшим кроком канавки (10 мм), на другому місці опинився екземпляр з порівняно великим кроком – 40 мм.

Застосування найкращої спіральної канавки дозволило зменшити гідравлічний опір на 19 %. Для пояснення ефектів застосовувались візуалізація і комп'ютерне моделювання. Відповідність комп'ютерного моделювання експериментальним результатам визначалась порівнянням середнього коефіцієнту теплообміну (розрахункового і визначеного за допомогою льодового калориметра). В результаті вибрано модель турбулентності RNG_ke, яка забезпечує кращу відповідність моделі експерименту. Комп'ютерне моделювання пояснило фізичну картину обтікання циліндрів зі спіральними канавками, в тому числі їх взаємний вплив при відмінній осьовій орієнтації в пучку.

Показано, що наявність спіральної канавки, яка з одної сторони збільшує тепловіддачу, а з другої сторони зменшує гідравлічний опір, може суттєво збільшити теплогідравлічну ефективність (фактор аналогії Рейнольдса)

Ключові слова: тепловіддача, гідродинаміка, інтенсифікація тепловіддачі, гідравлічний опір, спіральні канавки, аналогія Рейнольдса

Received date 10.05.2020
Accepted date 17.06.2020
Published date 23.06.2020

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 configuration. 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, phase and antiphase modulated asynchronous. 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 number 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: coefficient 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 uncompressed 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 transverse 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 turbulates 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.

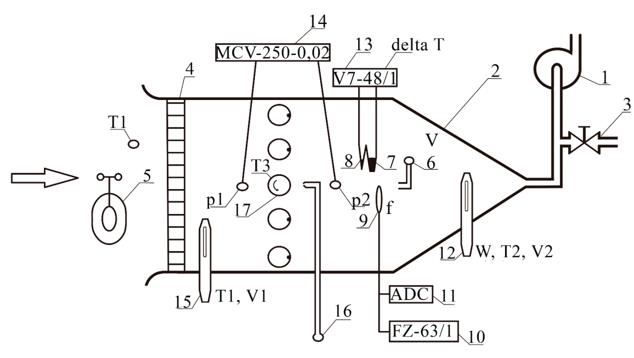

Fig. 1 Scheme of experimental installation:
1 – fan; 2 – conical part of the working area; 3 – control valve; 4 – honeycomb; 5 – input anemometer; 6 – moving Pitot tube (moves along the flow); 7 – wet thermocouple; 8 – dry thermocouple; 9 – piezoceramic sensor; 10 – frequency meter; 11 – analog-to-digital 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

Table 1

List of used devices

No.	Marking (Fig. 1)	Name	Factory designation	
1	5	Cup anemometer U2	GOST 6576–74	No. 21137
2	15	Input anemometer	Digital Anemometer	Model ST8021
3	p1	Micromanometer	MCV-2500-0,02	2500 Pa, cl. 0,02
4	T3	Ice calorimeter	–	–
5	16	Pitot tube	d=2 mm	–
6	p2	Micromanometer	MKB-2500-0,02	2500 Pa, cl. 0,02
7	13	Digital voltmeter	S68003	–
8	8	Dry thermocouple	THC – 2076	2000 No. 506
9	7	Wet thermocouple	THC – 2076	2000 No. 507
10	6	Pitot tube	d=2 mm	
11	11	ADC	TRITON 1255	NPCP TEREKS
12	10	Frequency meter	FZ – 63/1	9301099
13	12	Output anemometer	Digital Anemometer	Model ST8021
14	–	Stopwatch	–	–

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 μm. 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

Cylinder parameters

Marking	Spiral step, mm	Number of ways
SPI_40	40	1
SPI_20_2	40	2
SPI_10	20	2
Sm	Smooth cylinder	

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

Due to the complex shape of the calorimeter surface, the method of melting ice has been used to determine the average heat transfer [15]. When using this method, moisture from the air may appear on the surface of the cylinder-calorimeter, especially in the area of increased flow velocity between the cylinder-calorimeter and the adjacent cylinder. Therefore, when processing the experimental data, the processing technique considered in [15] was used, which allows to more accurately determine the average heat transfer.

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

Measurement errors

Parameter designation	Average value	Dimensionality	Absolute error, Δz	Relative error, ε, %
T	180	sec	0.05	$2.78 \cdot 10^{-2}$
Δ	0.002	m	0.00001	$5.0 \cdot 10^{-1}$
S	$2.426 \cdot 10^{-3}$	m	0.00005	2.06
F	$6.283 \cdot 10^{-3}$	m^2	$8.0 \cdot 10^{-6}$	$9.26 \cdot 10^{-2}$
V_p	$1.248 \cdot 10^{-5}$	m^3	$88.36 \cdot 10^{-9}$	$7.08 \cdot 10^{-1}$
ρ_w	999.9	kg/m^3	0.05	$5.00 \cdot 10^{-3}$
ρ_m	7900	kg/m^3	50	$6.33 \cdot 10^{-1}$
t_i	20.0	°C	0.05	$2.5 \cdot 10^{-1}$
t_f	–12.90	°C	0.05	$3.88 \cdot 10^{-1}$
Δt	–12.9	°C	0.05	$3.88 \cdot 10^{-1}$
Δt_m	4.0	°C	0.4	10.0
Q	11323	J	39.89	$8.27 \cdot 10^{-1}$
R	334000	J/kg	500	$1.50 \cdot 10^{-1}$
c_m	500	J/(kg K)	0.5	$1.0 \cdot 10^{-1}$
Q	30.8	W	0.0432	3.42·
q_T	28.28	W	0.0399	3.36·
Λ	0.55	W/(m·K)	0.005	0.909
Λ_{wall}	16.0	W/(m·K)	0.5	3.12
A	160.0	W/(m²·K)	6.96	4.26·
α_T	160.0	W/(m²·K)	15.66	10.55

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

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_kε, LRR and SSG were compared. The RNG_kε 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.

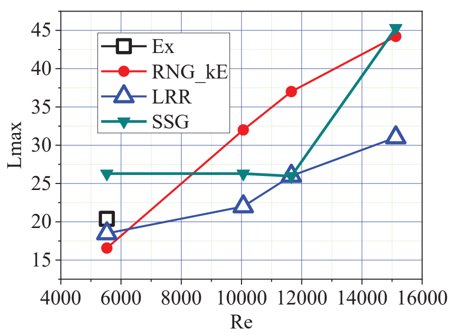

Fig. 3. Dependence of the length of the vortex zone on the middle cylinder on the Reynolds number for different models of turbulence: Ex – experiment (photographing a stagnant zone filled with soap bubbles). At low flow velocities, the model RNG_kε shows the smallest error (10.2 %)

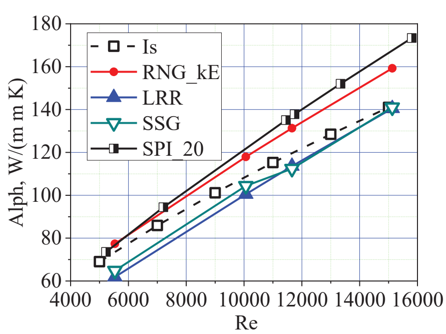

Fig. 4. Average heat transfer coefficient of the central cylinder depending on the Reynolds number for different turbulence models (computer calculation) RNG_kε, LRR, SSG and experimental data for smooth cylinders: Is – with [14] and SPI_20 – the authors' experiments with cylinders with grooves (groove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

In computer simulations, the smallest difference with the experiment was observed in the model RNG_kε (3.4 % at the maximum Reynolds number Re=15804).

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].

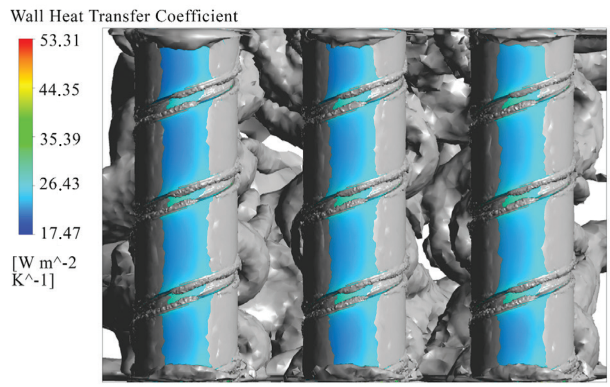

Fig. 5. View of the bundle of pipes from the flowing side

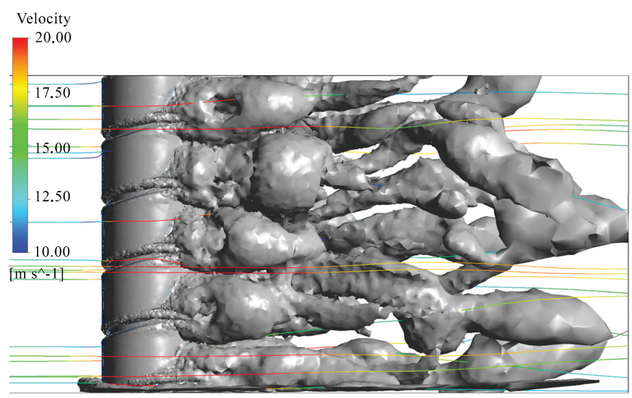

Fig. 6. System of vortices generated behind a cylinder with a groove (computer calculation) [17]

In the narrowest place between the cylinders, thin pointed vortices come from the grooves, along which the coupling vortices move, the size of which is near to the axial distance between the grooves. Different velocities of rotation of vortices determine at least two frequencies, and taking into account the second harmonic and four frequencies of oscillations in the flow after passing the area of the bundle.

Fig. 7 shows a field of velocity components oriented along the axis of the tray in an area, located at a distance 21 mm from the rear generating cylinders of the bundle. The structure of vortices becomes clearer when simultaneously analyzing the heat transfer coefficients on the surfaces of the cylinders and flow lines shown in Fig. 8. The streams, which moved along the grooves on different sides of the cylinder, collide and break on the smooth rear surface. The larger the step of the groove, the larger the distance between neighbor streams and the larger the surface covered by the vortex, generated on the rear surface of the cylinder.

Thus, when generating heat exchange surfaces, asymmetric patterns should be preferred. If the positions of the cylinders with grooves applied on their surface are identical (position 17 in Fig. 1), vortices from neighbour cylinders can interfere with each other (Fig. 4) with small steps of their placement in the bundle. This may be an advantage in terms of heat transfer in cylindrical surfaces with single-way grooves over similar surfaces with double-way grooves.

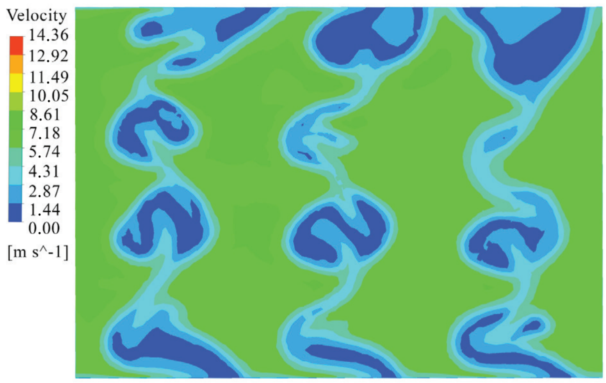

Fig. 7. Field of flow velocities in the area, located at a distance of 21 mm from the rear generating cylinders of the bundle (computer calculation)

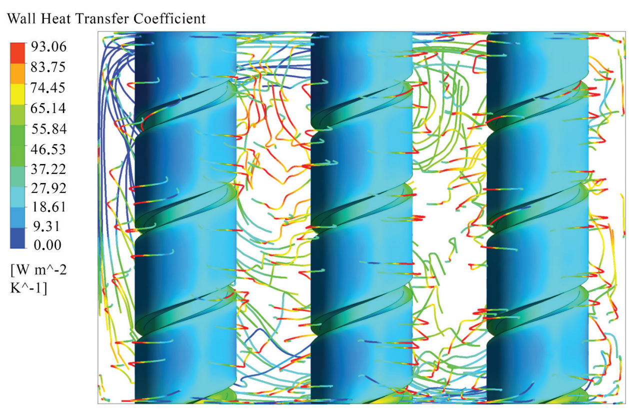

Fig. 8. Heat transfer coefficients on the rear surfaces of the cylinders and the flow line

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].

$$Nu = 0.26\,\mathrm{Re}^{0.65}\,\mathrm{Pr}^{0.33}\left(\mathrm{Pr}/\mathrm{Pr}_w\right)^{0.25}. \quad (1)$$

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.

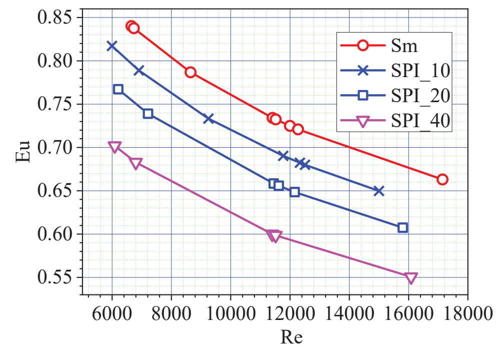

Fig. 9. Dependence of the Euler number on the Reynolds number [18]

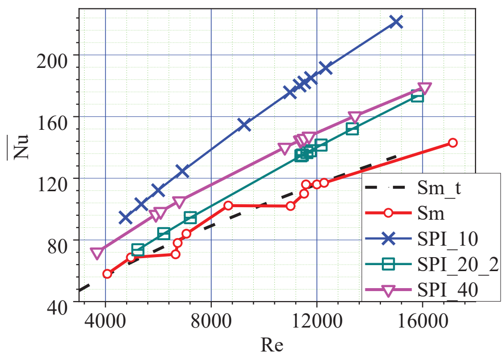

Fig. 10. Dependence of the Nuselt number on the Reynolds number. Sm_t – results calculated by equation (1) [19]

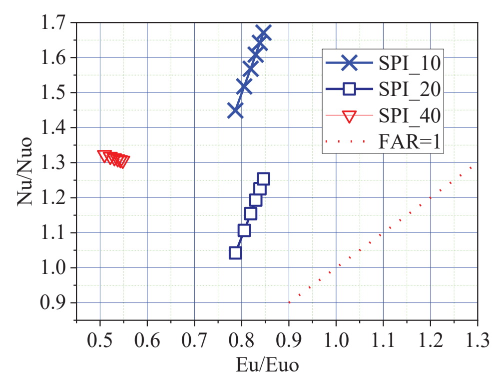

Fig. 11. Reynolds analogy: SPI_10 – groove step of 10 mm; SPI_20 – groove step of 20 mm; SPI_40 – groove step of 40 mm; FAR=1 smooth surface

The dependences of the relative increase in heat transfer Nu/Nu_0 on the relative increase in the hydraulic resistance 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).

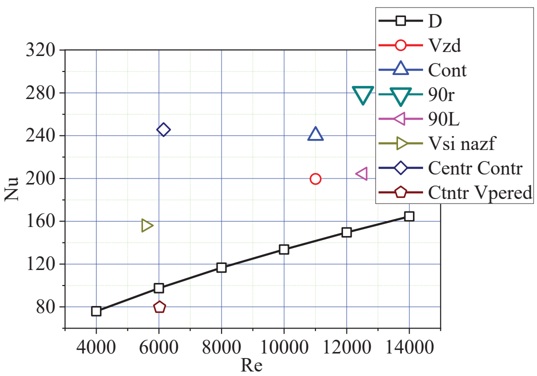

Fig. 12. Reaction of the average heat transfer of the central cylinder SPI_40 bundle to its azimuthal position among neighbour cylinders: D – calculation results for [19]; Vzd – all cylinders with the bottom beginning of the groove (Fig. 1 position 17) turned back; Cont – all cylinders with the bottom 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; Vsinazf – 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; CtntrVpered – all cylinders with the bottom beginning of the groove are turned towards the flow

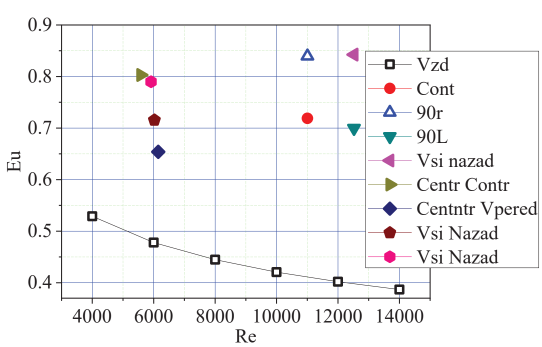

Fig. 13. Increase in the hydraulic resistance of the bundle as a result of ignoring the identity of the placement of the grooves on the cylinder

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 (CentntrVpered). 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 turbulization. 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 %.

References

1. Migay, V. K., Firsova, E. V. (1986). Teploobmen i gidravlicheskoe soprotivlenie puchkov trub. Leningrad: Nauka, 176.
2. Schlichting, H., Gersten, K. (2017). Boundary-Layer Theory. Springer. doi: https://doi.org/10.1007/978-3-662-52919-5
3. Eiffel, G. (1912). Sur la resistance des spheres dans l'air en movement. Comptes Rendus, 155, 1597.
4. Kovalenko, G. V., Khalatov, A. A. (2003). Fluid Flow and Heat Transfer Features at a Cross-Flow of Dimpled Tubes in a Confined Space. Volume 5: Turbo Expo 2003, Parts A and B. doi: https://doi.org/10.1115/gt2003-38155
5. Kovalenko, G. V., Meyris, A. Zh. (2013). Comparison of different methods of heft transfer enhancement on the cylindrical surfaces different methods of hes. Eastern-European Journal of Enterprise Technologies, 3 (12 (63)), 58–60. Available at: http://journals.uran.ua/eejet/article/view/14882/12686
6. Spitsyn, V. E., Botsula, A. L., Solomonyuk, D. N., Chobenko, V. N. (2008). Vysokoeffektivnaya gazoturbinnaya ustanovka dlya GPA. Vestnik Natsional'nogo tehnichnskogo universiteta «KhPI», 34, 8–11.
7. Laidoudi, H., Blissag, B., Bouzit, M. (2018). Numerical Investigation of Mixed Convection Heat Transfer in Steady Flow Past a Row of Three Square Cylinders. Defect and Diffusion Forum, 389, 164–175. doi: https://doi.org/10.4028/www.scientific.net/ddf.389.164
8. Islam, S. U., Rahman, H., Zhou, C. Y. (2016). Effect of gap spacings on flow past row of rectangular cylinders with aspect ratio 1.5. Ocean Engineering, 119, 1–15. doi: https://doi.org/10.1016/j.oceaneng.2016.04.022
9. Parthasarathy, N., Dhiman, A., Sarkar, S. (2017). Flow and heat transfer over a row of multiple semi-circular cylinders: selection of optimum number of cylinders and effects of gap ratios. The European Physical Journal Plus, 132 (12). doi: https://doi.org/10.1140/epjp/i2017-11791-2
10. Razzaghpanah, Z., Sarunac, N., Javanshir, A., Khalesi, J. (2019). Natural convection heat transfer from a horizontal row of finite number of heated circular cylinders immersed in molten solar salt. Journal of Energy Storage, 22, 176–187. doi: https://doi.org/10.1016/j.est.2019.02.008
11. Lima E Silva, A. L. F., Lima E Silva, S. M. M. (2014). Convection heat transfer around a single row of cylinders. Computational Thermal Sciences: An International Journal, 6 (6), 477–492. doi: https://doi.org/10.1615/computthermalscien.2014010552
12. Sukhotskii, A. B., Sidorik, G. S. (2017). Experimental study of heat transfer of a single-row bundle of finned tubes in mixed convection of air. ENERGETIKA. Proceedings of CIS higher education institutions and power engineering associations, 60 (4), 352–366. doi: https://doi.org/10.21122/1029-7448-2017-60-4-352-366
13. Vicente, P. G., García, A., Viedma, A. (2002). Heat transfer and pressure drop for low Reynolds turbulent flow in helically dimpled tubes. International Journal of Heat and Mass Transfer, 45 (3), 543–553. doi: https://doi.org/10.1016/s0017-9310(01)00170-3
14. Isachenko, V. P., Osipova, V. A., Sukomel, A. S. (1975). Teploperedacha. Moscow: Energiya, 488.
15. Kovalenko, G. V., Halatov, A. A. (2008). Adaptation of the ice calorimeters for investigation of the heat transfer of surfaces formed by dimples. Promyshlennaya teplotehnika, 30 (2), 5–12.
16. Lavrenova, D. L., Khlystov, V. M. (2016). Osnovy metrolohiyi ta elektrychnykh vymiriuvan. Kyiv, 123.
17. Khalatov, A. A., Kovalenko, G. V., Meiris, A. Z. (2018). Heat Transfer in Air Flow Across a Single-Row Bundle of Tubes With Spiral Grooves. Journal of Engineering Physics and Thermophysics, 91 (1), 64–71. doi: https://doi.org/10.1007/s10891-018-1719-x
18. Muliarchuk, M. A., Khalatov, A. A., Kovalenko, H. V. (2019). Hidrodynamika ta teploviddacha odnoriadnoho puchka tsylindriv z hvyntovymy kanavkamy na zovnishniy poverkhni pry poperechnomu obtikanni povitriam. XVII Vseukrainska naukovo-praktychna konferentsiya studentiv, aspirantiv ta molodykh vchenykh «Teoretychni i prykladni problemy fizyky, matematyky ta informatyky». Kyiv: KPI im. Ihoria Sikorskoho, 82–85.
19. Uong, H. (1979). Osnovnye formuly i dannye po teploobmenu dlya inzhenerov. Moscow: Nauka, 216.