ISSN 1561-8323 (Print)
ISSN 2524-2431 (Online)

ТЕХНИЧЕСКИЕ НАУКИ
TECHNICAL SCIENCES

UDC 532.526.5
https://doi.org/10.29235/1561-8323-2018-62-6-756-762

Received 04.10.2018
Поступило в редакцию 04.10.2018

Yulia V. Zhukova[1], Aleksandr M. Terekh[2], Aleksandr I. Rudenko[2]

[1]A. V. Luikov Heat and Mass Transfer Institute of the Academy of Sciences of Belarus, Minsk, Republic of Belarus
[2]Igor Sikorsky National Technical University of Ukraine "Kiev Polytechnic Institute", Kiev, Ukraine

CONVECTIVE HEAT TRANSFER AND DRAG OF TWO SIDE-BY-SIDE TUBES IN THE NARROW CHANNEL AT DIFFERENT REYNOLDS NUMBER

(Communicated by Academician Oleg G. Penyazkov)

Abstract. We studied numerically and experimentally convective heat transfer and drag of two side-by-side tubes in different arrangements in the narrow channel at the Reynolds number ranging from 8000 to 40000 in comparison with circular tubes. The visualization results of the flow structure in the wake behind the arrangements of the investigated tubes are presented. It is shown that the thermal and hydraulic performance of side-by-side drop-shaped tubes is by a factor of 1.2–2 higher than that of circular tubes due to a lower drag of drop-shaped tubes.

Keywords: heat transfer, drag, side-by-side tubes, drop-shared tubes, narrow channel, numerical simulation, flow vizualization, thermal and hydraulic performance

For citation: Zhukova Yu. V., Terekh A. M., Rudenko A. I. Convective heat transfer and drag of two side-by-side tubes in the narrow channel at different Reynolds number. *Doklady Natsional'noi akademii nauk Belarusi = Doklady of the National Academy of Sciences of Belarus*, 2018, vol. 62, no. 6, pp. 756–762. https://doi.org/10.29235/1561-8323-2018-62-6-756-762

Ю. В. Жукова[1], А. М. Терех[2], А. И. Руденко[2]

[1]Институт тепло- и массообмена им. А. В. Лыкова Национальной академии наук Беларуси, Минск, Республика Беларусь
[2]Киевский политехнический институт им. И. Сикорского, Киев, Украина

КОНВЕКТИВНЫЙ ТЕПЛООБМЕН И АЭРОДИНАМИЧЕСКОЕ СОПРОТИВЛЕНИЕ ДВУХ РАСПОЛОЖЕННЫХ БОК О БОК ТРУБ В УЗКОМ КАНАЛЕ ПРИ РАЗЛИЧНЫХ ЧИСЛАХ РЕЙНОЛЬДСА

(Представлено академиком О. Г. Пенязьковым)

Аннотация. Проведено численное и экспериментальное исследование конвективного теплообмена и аэродинамического сопротивления двух расположенных бок о бок каплеобразных труб в различной компоновке, размещенных в узком канале в диапазоне чисел Рейнольдса от 8000 до 40000, в сравнении с трубами круглого сечения. Представлены результаты визуализации структуры течения в следе за компоновками исследуемых труб. Показано, что теплоаэродинамическая эффективность расположенных бок о бок капельных труб в 1,2–2 раза выше, чем у труб круглого сечения за счет более низкого аэродинамического сопротивления каплеобразных труб.

Ключевые слова: теплообмен, аэродинамическое сопротивление, расположение бок о бок труб, каплеобразные трубы, узкий канал, численное моделирование, визуализация потока, теплогидравлисческая эффективность

Для цитирования: Жукова, Ю. В. Конвективный теплообмен и аэродинамическое сопротивление двух расположенных бок о бок труб в узком канале при различных числах Рейнольдса / Ю. В. Жукова, А. М. Терех, А. И. Руденко // Докл. Нац. акад. наук Беларуси. – 2018. – Т. 62, № 6. – С. 756–762. https://doi.org/10.29235/1561-8323-2018-62-6-756-762

Introduction. Since today the use of circular tubes without intensifiers, which can make additional aerodynamic drag, has exhausted itself, it is obvious that for the sake of increase in the performance of heat exchange devices and for the sake of improvement of their weight-and-size indicators, changing the form of a cross section of a heat transfer surface based on a round tube will cause controlled flow to separate, the wake structure to change and, as a result, aerodynamic drag to reduce. To do this, non-circular tubes can find use, for example, these are flat-oval, drop-shaped, elliptic tubes, whose drag is by a factor of 1.5–2 less than that of circular tubes and whose heat transfer characteristics are close and, hence, have a higher thermal and hydraulic efficiency.

A very small number of works have been devoted to the study of drag of non-circular tubes and their banks. The known works in their majority contain the comparison results of single elliptic and circular tubes and are mainly concerned with the study of the influence of the tube cross section on drag and heat transfer [1; 2]. However the important factor influencing drag and heat transfer of tube banks are a tube distance and a distance from tubes to the wall of the working section. Therefore, the objective of the work to be presented herein was to obtain new results on convective heat transfer influenced by the location of two different-configuration side-by-side tubes in the narrow channel at the 0° angle of attack.

This work presents the numerical and experimental results on convective heat transfer and drag of two side-by-side tubes of different cross section in the narrow channel. The visualization results of the flow structure in the wake behind the arrangements of investigated tubes are also given.

Test model and task statement. An object of investigation (Fig. 1) is a pair of side-by-side circular (Fig. 1, *a*) and drop-shaped tubes (Fig. 1, *b*), the latter being considered in three arrangements. In the first arrangement, drop-shaped tubes are in the flow on the side of the large diameter (Fig. 1, *b*); in the second arrangement, tubes are in the flow on the side of the small diameter (Fig. 1, *c*); in the third arrangement (mixed), one of the tubes is in the flow on the side of the large diameter and the other is in the flow on the side of the small diameter (Fig. 1, *d*).

The studies of the influence of both the distance from the channel wall to circular tubes and the tube distance on drag [3; 4] revealed three flow regimes. The first flow regime is realized within the tube distance range of 0.2–0.3D (D is the tube diameter); tubes are streamlined as one bluff body, generating one common vortex street. The second regime is within the tube distance range from 0.2–0.3D to 1.2–1.5D; in this case, the asymmetric regime arises in the wake – the off-center flow separation causes one wide wake and one narrow wake to form behind cylinders. The third regime occurs within the tube distance

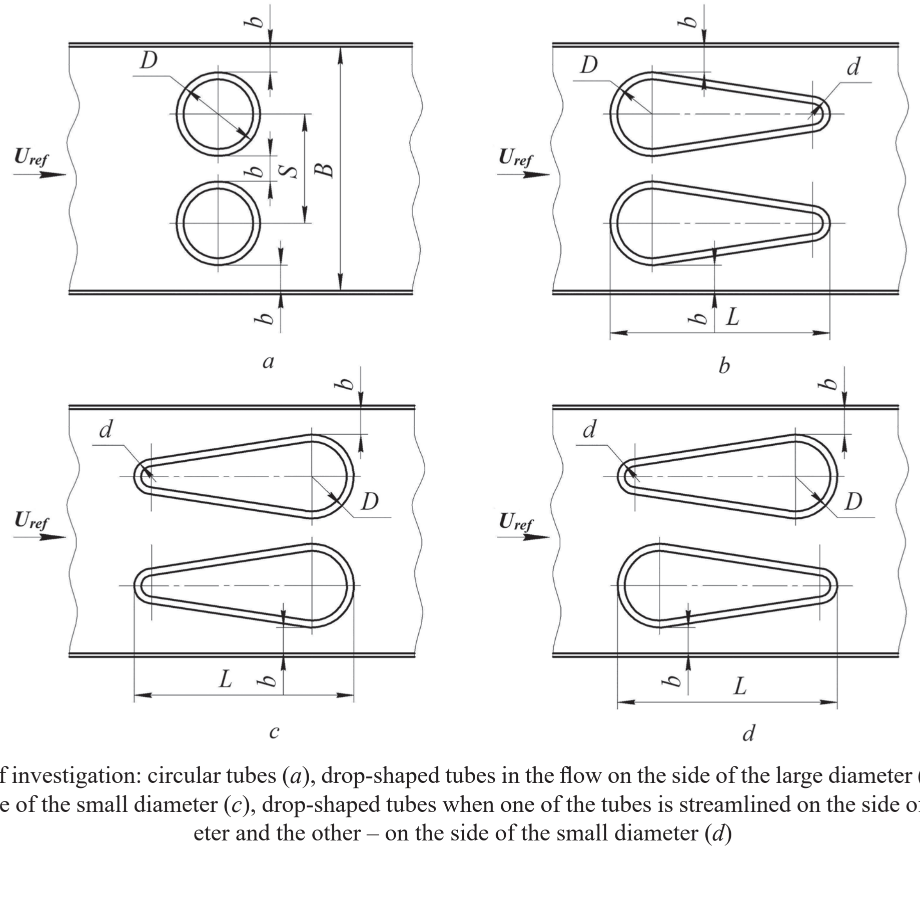
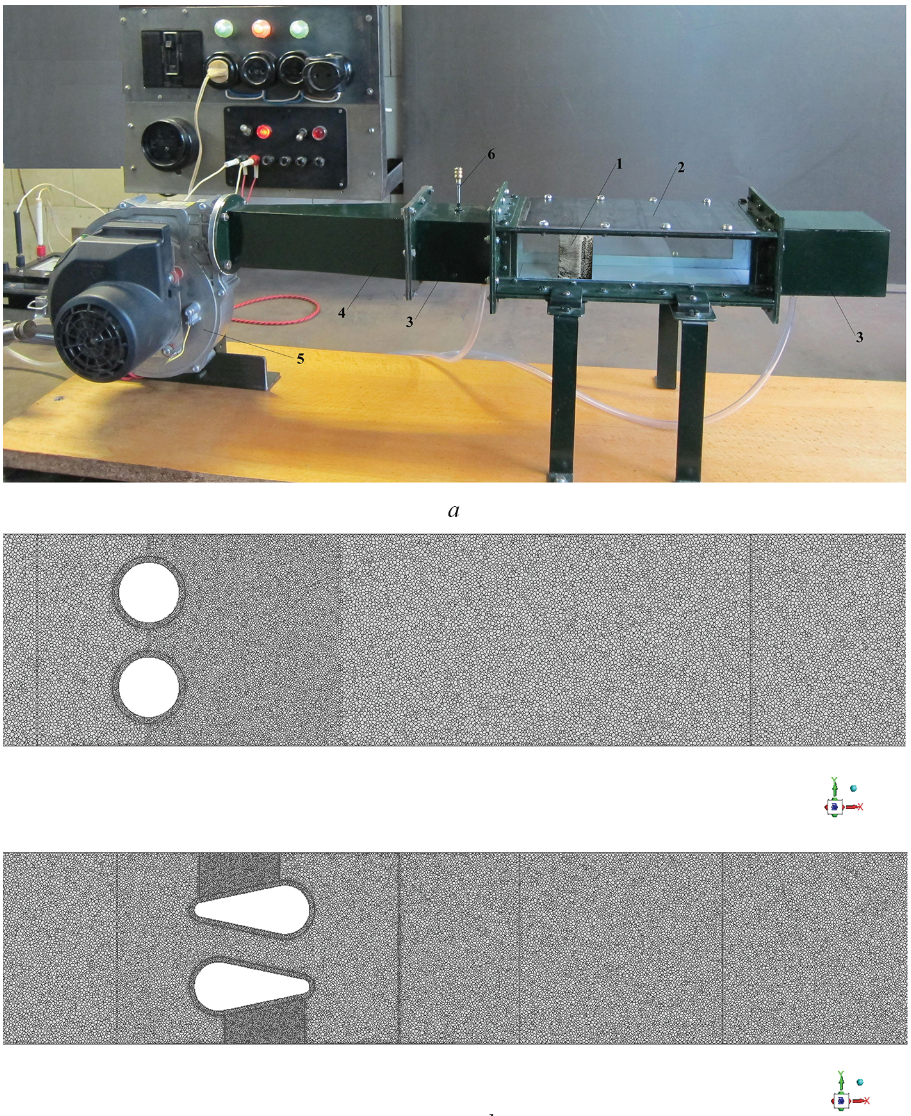


Fig. 1. Object of investigation: circular tubes (*a*), drop-shaped tubes in the flow on the side of the large diameter (*b*), drop-shaped tubes on the side of the small diameter (*c*), drop-shaped tubes when one of the tubes is streamlined on the side of the large diameter and the other – on the side of the small diameter (*d*)

range of 1.2–1.5D; within this regime, tubes are streamlined as two independent bluff bodies and behind them two independent vortex streets with the same vortex shedding frequency are formed. The present work studied the second flow regime for all arrangements

The channel blockage was $k_q = 0.69$ and in the case of mixed arrangement, $k_q = 0.54$. The drop-shaped tube profile length L (Fig. 1, *b*–*d*) was 51 m, the large diameter $D = 24$ mm, and the small diameter $d = 10$ mm. For all-type tubes, the tube center distance S and the gap b between the tubes and the working section walls (with the exception of drop-shaped tubes of mixed arrangement were taken constant and equal to 31.3 and 7.3 mm, respectively, keeping a constant distance between the tube walls and the distance between the investigated tube walls and the working section walls.

Experiments on flow around two side-by side tubes of different cross section in the narrow channel were performed in a closed-circuit wind tunnel with a rectangular working section of width $B = 70$ mm and height $A = 60$ mm (Fig. 2, *a*). The flow-through area was represented by a working section, comprising an investigated tube, two straight channels in front of and behind the working section, and a diffuser connected with one end to a fan outlet connection and with the other – to a straight channel. A maximum capacity of the fan was 0.085 m^3/s. This made it possible to achieve a flow velocity up to 20 m/s in the free area of the working section.



a

b

Fig. 2. Experimental set-up (*a*): *1* – place of mounting objects of investigation; *2* – working section; *3* – straight channel; *4* – diffuser; *5* – fan; *6* – injector; computational grid fragments (*b*)

To level a velocity field and to reduce a turbulence degree at the working section inlet, the honeycomb with 1.2 × 1.2 mm cells was placed between the diffuser (4) and the straight channel (3). It served to break up large vortices and to form a system of small fast decaying vortices. The turbulence degree Tu behind the grid was determined by the Roach formula [5]

$$\mathrm{Tu} \sim 0.8(x / d_{wr})^{-\frac{5}{7}},$$

where $x \approx 180$ mm is the distance from the grid to the investigated tubes; $d_{wr} = 0.3$ mm is the wire diameter. The calculation results of the flow turbulence degree at the working section inlet showed that it is close to 1%.

To visualize the structure of flow around two side-by-side tubes of different cross section, flow features at a different location of tubes to each other and processes occurring in the near wake in transverse flow around them, a device to inject a soot-kerosene suspension into the flow was mounted in the straight channel. The methods to visualize flow by means of a soot-kerosene suspension were detailed in [6].

For numerical simulation to be made, the three-dimensional statement of the task was set. Geometric sizes of tubes were selected from the condition of matching with experiment. Computational domain sizes were 0.06 m × 0.07 m × 0.447 m. Computation was made on an unstructured 3D grid composed of polyhedral elements. In the 3D grid, we selected an area with a structured grid around a tube to describe the boundary layer. Fig. 2, *b* illustrates the computational domain fragments. The total power of the grid was about 10 mln cells.

We solved the steady Reynolds-averaged Navier–Stokes equations, the continuity equation and the energy equation [7]. The Reynolds equations were closed using Menter's κ-ω shear stress transport model in standard statement [7].

An incoming flow velocity U_{ref}, an incoming flow temperature T_{ref}, a turbulence level Tu and a hydrodynamic diameter d_{hyd} matched with experiment conditions were assigned at the computational domain inlet. A constant heat flux was predetermined at the inner surface of tubes; outflow boundary conditions – at the exterior boundary; channel walls were assumed to be smooth and thermally insulated, no-slip conditions were implemented at the walls.

The verification of the computational algorithm using different experimental data is described elsewhere in [8–11]. There, it was shown that the error of numerical simulation results does not exceed 12 % in comparison to experiment.

Results and discussion. Figs. 3–4 demonstrate the numerical and experimental results. The similarity numbers – the Reynolds number $\mathrm{Re}_D = \dfrac{U_{\mathrm{ref}} D}{\nu}$, the Euler number $\mathrm{Eu} = \dfrac{\Delta p}{\rho_{\mathrm{ref}} U_{\mathrm{ref}}^2}$ and the Nusselt number $\mathrm{Nu}_D = \dfrac{\alpha D}{\lambda}$ were formulated using an air density ρ_{ref}, a flow velocity U_{ref} at the inlet of the working section of the channel and a large diameter of a drop-shaped tube equal to a circular tube diameter D as defining values.

The arrangement of circular tubes (Fig. 3, *a*) exhibits the largest drag, while the mixed arrangement of drop-shaped tubes (Fig. 3, *a*) – the smallest drag. This can be attributed to the case when the channel is not heavily blocked ($k_q = 0.54$) in comparison to other arrangements. In terms of heat transfer, circular tubes continue to be most efficient; all the considered arrangements of drop-shaped tubes are very close in total heat transfer (Fig. 3, *b*). However the thermal and hydraulic performance $\xi = \dfrac{(\mathrm{Nu}_{\mathrm{drop}} / \mathrm{Nu}_{\mathrm{cir}})}{(\mathrm{Eu}_{\mathrm{drop}} / \mathrm{Eu}_{\mathrm{cir}})}$ of drop-shaped tubes (Fig. 3, *c*) in any arrangement exceeds that of circular tubes.

Comparing the arrangements of drop-shaped tubes by drag, it can be emphasized that a bank of drop-shaped tubes in the flow on the side of the small diameter (Fig. 3, *a*) has the largest drag. This can be explained as follows: when drop-shaped tubes are in the flow on the side of the small diameter, the flow encounters an obstacle – a drop-shaped tube – having a small rounding radius. Further, moving along the drop-shaped tube the flow enters a convergent channel formed by the drop-shaped tube wall, the velocity increasing and the pressure decreasing. In the rear part of drop-shaped tubes the flow is separated and the rear part of drop-shaped tubes is streamlined in the same manner as in the case

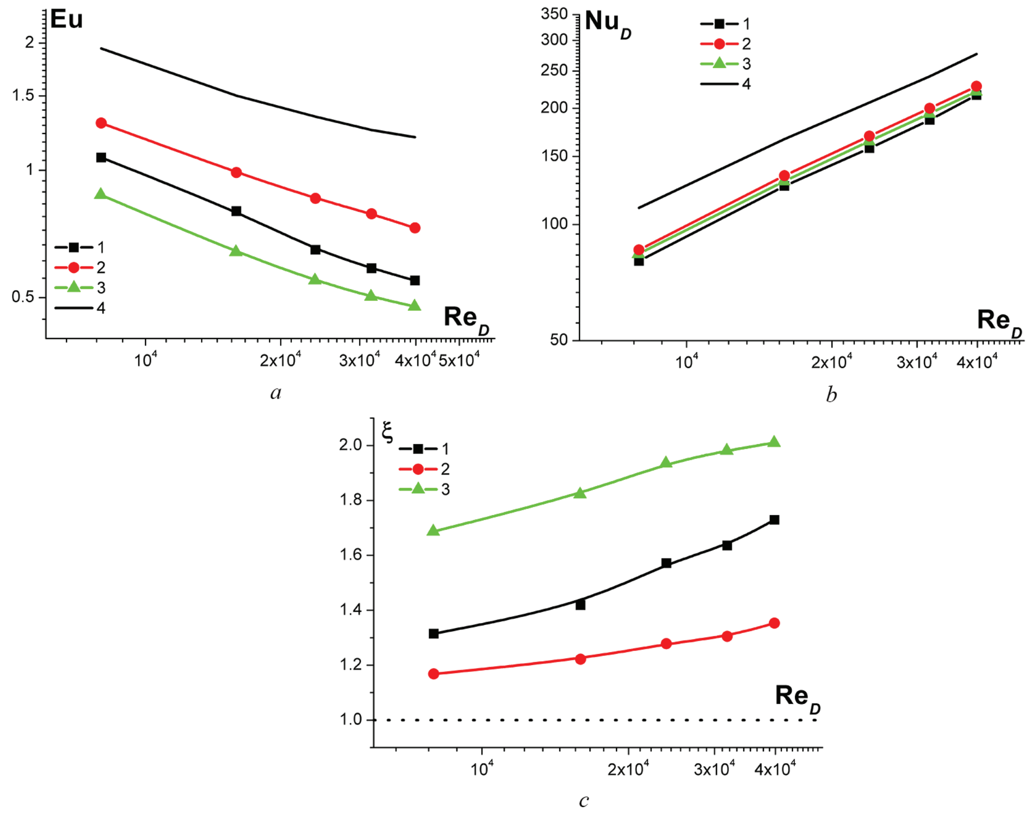


Fig. 3. Hydraulic losses (*a*), total heat transfer (*b*), thermal and hydraulic efficiency (*c*) of the investigated tubes: *1* – pair of side-by-side drop-shaped tubes streamlined on the side of the large diameter; *2* – pair of side-by-side drop-shaped tubes streamlined on the side of the small diameter; *3* – mixed arrangement; *4* – pair of side-by-side circular tubes

of circular tubes. Owing to this, behind the drop-shaped tube streamlined on the side of the small diameter, a wide recirculation zone (Fig. 4, *a*) is formed. All the above facts show that drag grows. The experimental data for flow structure visualization (Fig. 4, *a*) support the fact that asymmetric structures appear in the wake behind drop-shaped tubes.

In flow around a pair of drop-shaped tubes on the side of the large diameter, the frontal part of each of the tubes is streamlined as in the case of circular tubes. The flow then enters the divergent channel – a peculiar diffuser formed by the wall of the both tubes, the pressure increasing and the velocity decreasing. As mentioned in [12], diffuser losses can be up to 30 % of all hydraulic losses. Moreover, the flow can be separated in the diffuser and separation zones can be formed near the edges of the diffuser. Thus, beyond the diffuser, the flow is injected into the near wake and separation is formed in the rear part of drop-shaped tubes (Fig. 4, *b*). Since separation is formed over the section of small-diameter tubes, the separation zone width is not large and the jet injection from the diffuser into the near wake decreases drag. It should be noted that for some velocity regimes, two separation zones can merge into one common zone downstream (Fig. 4, *b*) at Re_D = 15940. This can be attributed to the fact that in diffusers, even at small angles of expansion the flow at some distance from the inlet can be separated at the wall [12]. This circumstance can affect the recirculation zone behind the pair of drop-shaped tubes.

In the mixed arrangement, the asymmetric near wake is formed (Fig. 4, *c*) first due to the asymmetric location of the drop-shaped walls relative to the channel asymmetry and second due to the formation of a wide wake behind the drop-shaped tube streamlined on the side of the small diameter. The formed wide wake suppresses the narrow wake that had formed behind the drop-shaped tube streamlined on the side of the large diameter. The wake asymmetry causes the flow to separate at the channel wall.

As mentioned above [3; 4], flow around a pair of circular tubes occurs simultaneously with the asymmetric wake formation (Fig. 4, *d*). It should be noted that a difference in heat transfer of the pair of the considered circular tubes can achieve up to 7 %.

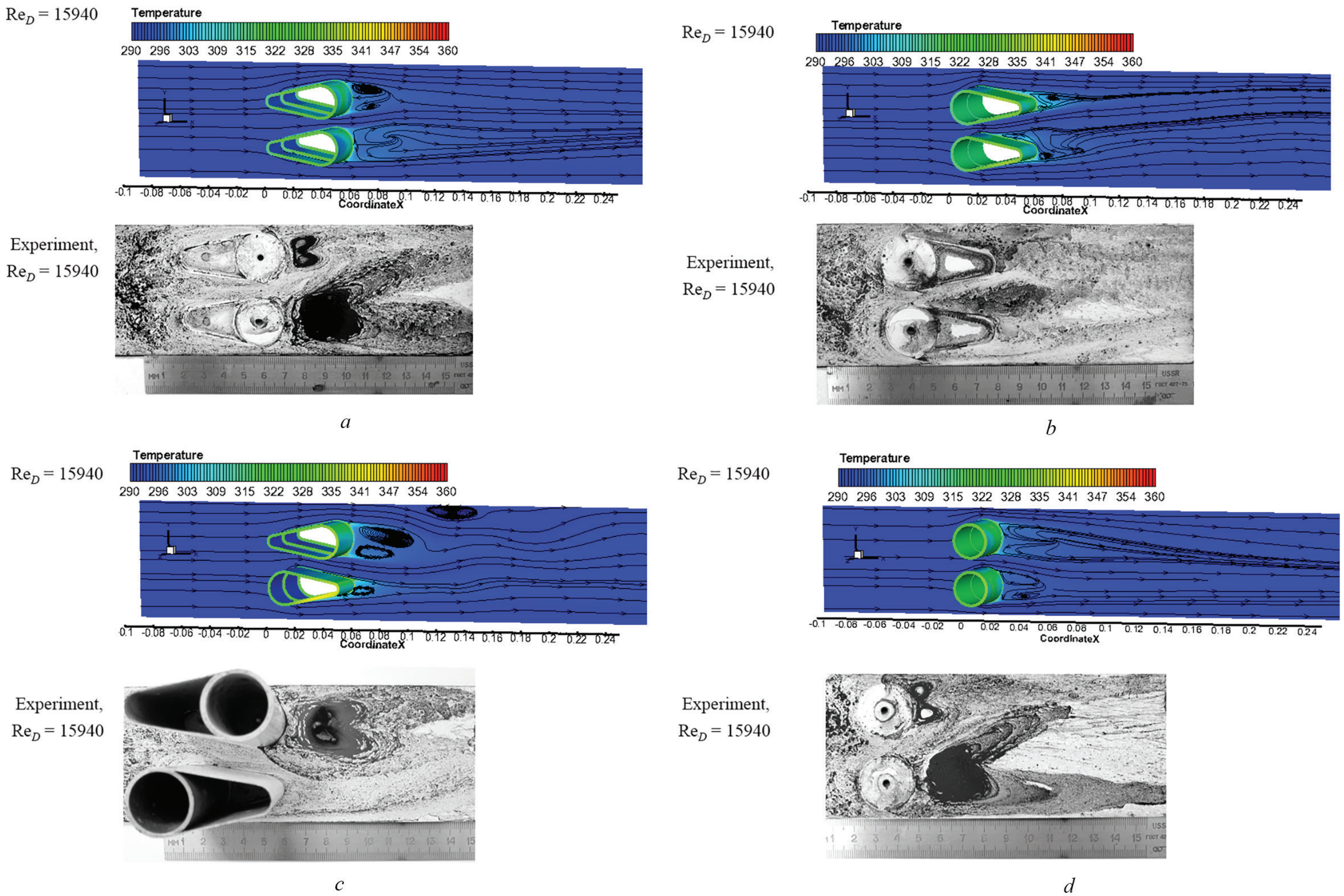


Fig. 4. Structure of flow around a pair of side-by-side tubes depending on the incoming flow velocity: pair of side-by-side drop-shaped tubes streamlined on the side of the large diameter (*a*); pair of side-by-side drop-shaped tubes streamlined on the side of the small diameter (*b*); mixed arrangement (*c*); pair of side-by-side circular tubes (*d*)

Conclusion. We studied numerically and experimentally convective heat transfer and drag of two side-by-side tubes in different arrangements in the narrow channel at the Reynolds number ranging from 8000 to 40000 in comparison with circular tubes. The visualization results of the flow structure in the wake behind the arrangements of the investigated tubes are presented. It is shown that the thermal and hydraulic performance of side-by-side drop-shaped tubes is by a factor of 1.2–2 higher than that of circular tubes due to a lower drag of drop-shaped tubes.

Acknowledgement. The work has been done under financial support of the Belarusian Republican Foundation for Fundamental Research on the Project No F18R-035

References

1. Hasan Ala. Thermal-hydraulic performance of oval tubes in a cross-flow of air. *Heat and Mass Transfer*, 2005, vol. 41, no. 8, pp. 724–733. https://doi.org/10.1007/s00231-004-0612-7

2. Terukazu O., Nishiyama H., Taoka Y. Heat transfer and flow around an elliptic cylinder. *International Journal of Heat and Mass Transfer*, 1984, vol. 27, no. 10, pp. 1771–1779. https://doi.org/10.1016/0017-9310(84)90159-5

3. Alam Md. M., Moriya M., Sakamoto H. Aerodynamic characteristics of two side-by-side circular cylinders and application of wavelet analysis on the switching phenomenon. *Journal of Fluids and Structures*, 2003, vol. 18, no. 3–4, pp. 325–346. https://doi.org/10.1016/j.jfluidstructs.2003.07.005

4. Alam Md. M., Zhou Y. Flow around two side-by-side closely spaced circular cylinders. *Journal of Fluids and Structures*, 2007, vol. 23, no. 5, pp. 799–805. https://doi.org/10.1016/j.jfluidstructs.2006.12.002

5. Roach P. E. The generation of nearly isotropic turbulence by means of grids. *International Journal of Heat and Fluid Flow*, 1987, vol. 8, no. 2, pp. 82–92. https://doi.org/10.1016/0142-727x(87)90001-4

6. Rudenko A. I., Tereh A. M., Semenyako A. V., Nishchik A. P., Baranyuk A. V. The method of the gas flow visualizing on the bodie's surface of different shapes. *Eastern-European Journal of Enterprise Technologies*, 2011, vol. 1, no. 9(49), pp. 51–55 (in Russian).

7. Ansys Fluent. Pensylvania, Ansys Inc., 2011. 11 p. Available at: http//www.ansys.com (Accessed 2 December 2018).

8. Zhukova Yu. V., Terekh A. M., Semenyako A. V. Numerical simulation of unsteady cross-flow past an oval cylinder at various Reynolds numbers. *Doklady Natsional'noi akademii nauk Belarusi = Doklady of the National Academy of Sciences of Belarus*, 2011, vol. 55, no. 1, pp. 102–107 (in Russian).

9. Terekh M., Rudenko A. I., Zhukova Y. V. Aerodynamic drag to flows about single drop-like tubes and visualizaton of these flows. *Journal of Engineering Physics and Thermophysics*, 2013, vol. 86, no. 2, pp. 378–384. https://doi.org/10.1007/s10891-013-0844-9

10. Kondratyuk V. A., Semenyako A. V., Terekh A. M., Rudenko A. I., Zhukova Yu. V. Heat transfer and aerodynamics of single oval-flat tubes. *Sovremennaya nauka: issledovaniya, idei, rezultaty, tekhnologii = Modern Science: Studies, Ideas, Technologies*, 2013, no. 1 (12), pp. 3–6

11. Skrypnik A. N., Shchelchkov A. V., Popov I. A., Ryzhkov D. V., Sverchkov S. A., Zhukova Yu. V., Chornyi A. D., Zubkov N. N. Thermohydraulic Efficiency of Tubes with Internal Spiral Finning. *Journal of Engineering Physics and Thermal Physics*, 2018, vol. 91, no. 1, pp. 52–63. https://doi.org/10.1007/s10891-018-1718-y

12. Povkh I. L. *Aerodynamic experiment in mechanical engineering*. Leningrad, 1974. 480 p. (in Russian).

Information about the authors

Zhukova Yulia Vladimirovna – Ph. D. (Physics and Matematics), Leading researcher. A. V. Luikov Heat and Mass Transfer Institute of the National Academy of Sciences of Belarus (15, P. Brovka Str., 220072, Minsk, Republic of Belarus). E-mail: julia_zhukova@rambler.ru.

Terekh Aleksandr Mikhailovich – Ph. D. (Engineering), Senior researcher. Igor Sikorsky National Technical University of Ukraine "Kiev Polytechnic Institute" (37, Pobedy Ave., 03056, Kiev, Ukraine). E-mail: teram1957@meta.ua.

Rudenko Aleksandr Igorevich – Ph. B. (Engineering), Associate professor. Igor Sikorsky National Technical University of Ukraine "Kiev Polytechnic Institute" (37, Pobedy Ave., 03056, Kiev, Ukraine). E-mail: a_rudenko@ukr.net.

Информация об авторах

Жукова Юлия Владимировна – канд. физ.-мат. наук, вед. науч. сотрудник. Институт тепло- и массообмена им. А. В. Лыкова НАН Беларуси (ул. П. Бровки, 15, 220072, Минск, Республика Беларусь). E-mail: julia_zhukova@rambler.ru.

Терех Александр Михайлович – канд. техн. наук, ст. науч. сотрудник. Киевский политехнический институт им. Игоря Сикорского (пр. Победы, 37, 03056, Киев, Украина). E-mail: teram1957@meta.ua.

Руденко Александр Игоревич – канд. техн. наук, доцент. Киевский политехнический институт им. Игоря Сикорского (пр. Победы, 37, 03056, Киев, Украина). E-mail: a_rudenko@ukr.net.