The structure of separated flow and heat transfer in a round tube with single detached diaphragm

V I Terekhov and T V Bogatko
Kutateladze Institute of Thermophysics SB RAS, Novosibirsk, Russia

Email: bogatko1@mail.ru

Abstract. Here we present the results of numerical investigation of the flow structure and turbulent heat transfer obtained in a round tube with a diaphragm of height $h$ and some gap $c$ between the tube and the diaphragm. The value of gap between the diaphragm and tube wall was changed within $A = c/h = 0 \div 0.33$. It was defined that an increased distance between the diaphragm and the tube wall modifies the structure of recirculation zone and eliminates the stagnation areas in the region of secondary vortices. The flow regime, when the values of average heat transfer behind the diaphragm with a gap are higher than the same values for the attached rib, was determined. When increasing the gap height from 0 to 0.33, the 30-% increment in the coefficient of thermal enhancement was achieved.

1. Introduction
Enhancement of heat transfer in channels and tubes is one of the most important problems of modern aerohydromechanics and heat transfer. Among the large number of intensification methods, the passive control of flow characteristics by alternating obstacles of various configurations along the tube length is of the most preference. This is mainly caused by the simple design and possibility of practical implementation in various heat exchangers. Introduction of an obstacle into the boundary layer causes significant disturbances that lead to formation of return flows, increase in the turbulence scale, intensification of flow mixing, and, respectively, intensification of convective heat transfer.

The thermohydraulic characteristics of systems with passive intensifiers of heat transfer have been investigated and described in detail in some monographs and reviews [1-5]. In the classical works on enhancement of heat transfer in channels with ribs [6,7], the alternating rectangular obstacles are used as the discrete elements. The authors have determined the limits of heat transfer enhancement and the effect of the rib size and their location on thermohydraulic characteristics.

Since the 60s of the twentieth century, the flow structure and heat and mass transfer behind a flat obstacle and system of such obstacles were studied extensively [1-14]. Peculiarities of the flow around disconnected ribs attracted the attention of researchers much later. The idea of their application for heat transfer intensification is based on formation of the wall jet flow, which violates the stagnation zone directly behind an obstacle. There are a few experimental and calculation papers dealt with this problem [15-23]. All these investigations were performed mainly for the rectangular tubes and detached square obstacles. A series of papers [24-26], studying the development of the flow with disconnected diaphragms in a round tube and periodic destruction of the flow by the rings of a round wire or twisted tape, is an exception. The use of the system of conical nozzles mounted inside the tube is close by the physics of the process, but it has the distinguishing features [27]. The nozzle inserts can be also mounted with some gap from the tube wall that leads to additional turbulization in the wall.
region of the flow. Interchange of connected and disconnected obstacles along the channel provides the additional possibility of changing the integral heat transfer coefficient [28].

The effect of Reynolds number, obstacle sizes and relative height of the clearance on the flow structure and turbulent heat transfer in a rectangular channel with the ribbed wall was studied experimentally in [16-19]. Investigation results show that the thermohydraulic efficiency of this method of heat transfer intensification is higher than application of the attached ribs.

According to the above mentioned, the mechanism of heat transfer enhancement in the channels with detached obstacles is studied insufficiently. This is explained by the multifactor character of complex heat transfer, three-dimensional character of the flow, and features of the vortex flow generation in the rectangular channels with different numbers of ribbed walls in comparison with the round tubes. Moreover, interference of separated flows from the alternating obstacles causes additional difficulties in analyzing [29]. Therefore, it is necessary to study in detail the flow structure under the simpler conditions of the flow around a single two-dimensional obstacle with varying disconnection parameter $A = c/h$, where $c$ is the size of a gap between the diaphragm and the tube wall, and $h$ is the diaphragm height (Fig. 1). At that, it is important to determine the influence of parameter $A$ on the local and integral characteristics of heat transfer and resistance. Analysis of more complex transfer processes in channels with a system of obstacles can be based on these data.

2. Flow chart and calculation technique

The scheme of computational domain is shown in Fig. 1. A flat annular diaphragm is mounted into the cylindrical channel with the diameter of 100 mm and length of 900 mm at the distance of 100 mm from the inlet. Its height was constant $h = 15$ mm, and the size of a gap was varied $c = 0 \div 5$ mm. In general, the tube geometry corresponded to that in [30], where the influence of the single diaphragm shape on friction and heat transfer was studied. The velocity profile at the tube inlet was uniform; air at 10°C was used as the working medium. A change in thermal-physical properties was neglected in calculations. The Reynolds number $Re = 27500$, calculated by the tube diameter and mean mass velocity, was constant.

The degree of flow turbulence at the inlet calculated by formula $Tu = 0.16 \cdot Re^{\frac{1}{8}}$ [31], was $Tu = 5.8\%$. The atmospheric pressure was set at the outlet boundary downstream. The thermal boundary conditions on the tube wall corresponded to $q_w = \text{const}$. The diaphragm surface was not heated.

The model of incompressible fluid based on the system of stationary Navier-Stokes and energy equations, Reynolds averaged numerical simulation (RANS) are used for calculations. The universal calculation complex FLUENT is the main tool for the study. The statement of the problem is two-dimensional; the flow is stationary and axisymmetrical. The k-ω SST model of turbulence is chosen for calculations [32,33] as the most appropriate for studying the turbulent separated flows [34]. The previous studies of the authors [30,35,36] prove also that for the models, implemented in the given set, the results obtained with application of the mentioned model match best the physics of the given flow type.

The computational domain was discretized by the grid with tetragonal cells, whose total number varied depending on geometrical sizes of the separated area. The calculation grid was nonuniform; thickening was performed equally towards all solid surfaces, such as tube walls and diaphragm. The computational domain included 120 000 – 140 000 nodes, 150-175 nodes along the vertical and 800 nodes along the horizon. The thickening coefficients were 1.08 and 1.064 along and across, respectively, and this ensured gradual decreasing the cell size. The wall functions were not used. This grid resolution was chosen optimal after the preliminary adaptive calculations; it guarantees location of at least 10 nodes in a viscous sublayer. In calculation, we have used the difference scheme of the
second accuracy order with application of the control volume method for discretizing the initial equations.

3. Calculation results

Figure 2 illustrates the flow structure near the disconnected rib. The longitudinal velocity field is presented by the colour; the streamlines are shown by the vectors.

The flow pattern for the attached rib, \( c=0 \), is shown in Fig. 2a. During liquid motion, the flow near the obstacle is forced considerably to the tube axis, and this leads to a significant velocity increase in the flow core. There are the extended recirculation zone and secondary vortex, typical of such a zone, formed behind the rib. The rotation of this secondary vortex is opposite to the main one, and the rotation speed is relatively small; all these cause formation of the stagnation zone. If there is a heat flux on the tube wall, this zone is characterized by high temperature and low heat transfer coefficient. In terms of engineering applications, this zone is not the most efficient one.

The flow pattern starts changing, when a gap between the tube wall and diaphragm is formed. The wall jet brings significant changes in the structure of the recirculation region. The area of reverse flows, represented by a pair of counter-rotating vortices, is formed behind a rib like behind a poorly streamlined body. At that, the velocity in the flow core decreases significantly.

In the transitional regime a pair of vortices just starts forming and separation bubble reduces in size and moves downstream. This flow can be observed in Fig. 2-c-e at a gap between the tube wall and diaphragm of 1-2 mm. At \( c = 3 \) mm, the above picture is observed, when the pair of equal vortices replaces the primary and secondary vortices in the recirculation zone.

Distribution of pressure and surface friction coefficients over the tube wall (Fig.3 and Fig.4) also demonstrates a significant decrease in the recirculation zone. At that, the reverse flow area along the tube wall degenerates significantly for the high gap values \( c/h = 0.2 - 0.33 \). Actually, two vortices directed oppositely to the center of recirculation zone stay behind the rib.

Data on static pressure distribution behind an obstacle allow us to analyze the wall region, which greatly affects the friction, heat transfer and structure of the separated flow in general. The effect of the gap height \( c \) on distribution of the pressure coefficient along the tube wall is demonstrated in Fig. 3. The pressure coefficient was defined as \( Cp = 2 (p_i - p_o) / \rho u_o^2 \), where \( p_i \) is the static pressure on the wall in the considered cross-section of the tube, \( p_o \) is the pressure at the tube outlet and \( u_o \) is superficial velocity. The value of \( x/h = 0 \) in these figures and then corresponds to the coordinate of the back edge of an obstacles.

Distribution of the pressure coefficient for case \( c/h = 0 \) (Fig. 3) is the classic case of a poorly streamlined obstacle. Flow separation occurs at the front top edge; as a result, the pressure drops sharply and then recovers slowly at large distances from the obstacle. The clearance size has a significant impact on distribution of pressure coefficient. For the attached rib \( c/h = 0 \) and up to \( c/h = 0.13 \), the pressure on the wall reduces sharply from the site of obstacle mounting. This is caused by
local acceleration of the flow and the beginning of its separation. Moreover, the maximal value of rarefaction is reached at the flow around the attached diaphragm, and the pressure recovers slower.

Separation behind the attached rib leads to more intensive recirculation flow; as a result, the pressure in it is much lower than that behind the separated obstacle. In this case, the length of the zone of pressure relaxation for the separated ribs decreases with an increase in the clearance height \( c/h \).

These features of parameter \( c/h \) also affect the behaviour of the surface friction coefficient. Calculated distributions of this coefficient are shown in Fig. 4. The largest zone of reverse flows, where the friction is negative, is observed at flow separation behind the attached obstacle (\( c/h=0 \)). Respectively, the value of the friction coefficient for the attached obstacle recovers slower.

According to distributions of the friction coefficient (Fig. 4) the structure of separation zone behind the detached rib with large gaps \( c/h = 0.2 \) and \( 0.33 \) differs, and the reverse flows are not formed along the tube wall. That is why in Fig. 5, which shows the relative length of recirculation region, the value in these two cases is zero (\( X_R/h=0 \)). The attachment point was determined as the coordinate of zero surface friction on the wall.

Calculation results in Fig. 5 clearly show the effect of the gap height on the size of recirculation zone. It can be seen that the length of separation bubble is inversely proportional to the height of a gap between the diaphragm and tube wall. It is known that the flow turbulence is one of the factors of heat transfer enhancement; therefore, the analysis of behaviour of the turbulence degree leads to the deeper understanding of the reasons for heat transfer enhancement or reduction. When flowing around the obstacles with sharp edges, as in our case, the flow is turbulized intensively due to the additional vortex formation. This is illustrated by data in Fig. 6, where variation of the turbulent kinetic energy behind the obstacle along the tube is shown. The field of turbulent kinetic energy (Figure 6) demonstrates flow evolution and removal of the stagnation zone from the rib downstream. The classical pattern of the flow around an attached rib (Fig. 6 a) is as follows: the highest values of turbulent pulsations are concentrated in the region of interaction of the shear layer with recirculation
zone, and in the area of the secondary vortex, their minimal values are observed. When a clearance between the diaphragm and tube wall is added (Fig. 6 b-d), the stagnation zone moves downstream, and the maximal value of turbulent pulsations are reduced more than by the factor of 4. As a result, when the scale of the wall jet increases, the structure of the recirculation zone changes, and the area of increased turbulent pulsations reduces dramatically.

**Figure 6.** The stream lines and the field of turbulent kinetic energy around the diaphragm. 

a) A=0; b) A=0.03; c) A=0.07; d) A=0.1; e) A=0.13; f) A=0.2; g) A=0.33.

**Figure 7.** The stream lines and temperature field around the diaphragm. 

a) A=0, attached diaphragm; b) A=0.03; c) A=0.07; d) A=0.1; e) A=0.13; f) A=0.2; g) A=0.33.

The temperature fields at the flow around a rib, removed at the different distances from the wall, are shown in Fig.7. The stagnation zone with increased temperatures is clearly seen there (Fig. 7 a). It is obvious that this zone shifts downstream (Fig. 7 b - e) and degenerates gradually (Fig. 7 f - g) with increasing gap between the diaphragm and tube wall.

Distribution of local Nusselt number along the tube is shown in Fig. 8. In the separation region behind the obstacle, heat transfer intensity increases drastically, and at the point of flow attachment, the heat flux reaches its maximum. Then, with flow motion in the zone of its relaxation heat transfer

**Figure 8.** The distribution of the local heat transfer coefficient on the tube wall behind the obstacle.

**Figure 9.** The relative value of the maximum heat transfer coefficient.
intensity decreases and restores gradually to its value at the stabilized flow. At that, the relaxation zone
with an increased level of heat transfer has a large length, exceeding usually, x/D> 30 calibers.

In general, the development scenario of local heat transfer is maintained in range c/h = 0 ÷ 0.13.
The main differences in the case of c/h = 0.2 and 0.33 are observed in a vicinity of the obstacle and at
that heat transfer in the recirculation zone becomes less intense. Data in Fig. 8 serve as the basis for
choosing the optimal parameter c/h from the viewpoint of heat transfer enhancement and the distance
between the obstacles, where the regime of maximal heat transfer can be achieved.

Dependence of the maximal Nusselt number for different values of the tested parameter is shown in
Fig. 9. Here, the calculated data on heat transfer are presented as Nu/Nu0 ratio, where the value
\( Nu_0 = 0.022 \cdot Re_0^{0.8} \cdot Pr_0^{0.6} \) describes heat transfer in the stabilized region of a round tube without an
obstacle [37]. As it can be seen, the maximal Nusselt number is 1.35 ÷ 2.5 times higher than its value
without an obstacle. Moreover, the relative height of the gap has a significant effect on \( Nu_{max} \) and its
highest value is observed at the flow around an attached obstacle, and the least value is observed for
an obstacle with maximal c/h = 0.33.

Distribution of the local heat transfer coefficient on the tube wall (Fig. 8) shows clearly that at gap
height c/h = 0.2, heat transfer intensity reduces significantly. As compared with a smooth tube, heat
transfer increases by 2.5 times for c = 0, and then it decreases sharply to unity at an increase in the
scale of the wall jet. In this case, the coordinate of maximal heat transfer coefficient also moves
downstream monotonically (Fig. 10).

When choosing the optimal shape of an obstacle for the purposes of heat transfer enhancement, the
integral heat transfer coefficient, total hydraulic losses and thermohydraulic efficiency are the
determining parameters. The last parameter represents the ratio of heat transfer gain to the power
increase, spent to overcome the increased hydraulic losses. In this paper, the average heat transfer
coefficient was calculated by integrating the local distributions in the area, which covers the distance
of 5 heights to the obstacle, zone directly under the obstacle, and distance of 25 heights behind the
obstacle (−5 ≤ x/h ≤ 25).

\[
Nu_w = \frac{1}{L} \int_{-5}^{25} Nu \cdot d \cdot x / h .
\]

The results of calculation of integral Nusselt number Nu_w are shown in Fig. 11 for different values
of the studied parameters. Data are presented in the relative form, where Nu0 is heat transfer in a
smooth tube along the stabilized region.

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For the average values of heat transfer in the specified region (Fig. 11), the maximum is observed at $c/h=0.03$. This is caused by the fact that at the gap height of 0.5 mm, the size of recirculation zone remains almost the same as in the case of the attached rib, but the appeared wall jet provides the increased heat removal in the region under the diaphragm. The data for the hydraulic friction coefficient reduced to this value for a smooth tube are presented in Fig. 12.

An increase in the scale of wall jet affects investigation results significantly. It is worth noting that with an increase in gap height $c$ from 0 to 5 mm, the area of the obstacle surface obstruction changes by the factor of 1.6. A decrease in hydraulic friction associated with the above fact also contributes to the overall result. Finally, we see the three-fold decrease in resistance with the tested parameter varied from 0 to 0.33.

The coefficient of hydraulic losses was calculated by the difference of total pressures in the same cross-sections as the integral $Nu_{sr}$:

$$
\xi = \frac{D}{L} \left( \left[ (\rho v_r^2/2 + p) r dr \right]_{c/h=5} - \left[ (\rho v_r^2/2 + p) r dr \right]_{c/h=25} \right).$$

Data on the coefficient of thermal enhancement for different tested parameters $c/h$ are shown in Fig. 13. According to these data, if a gap between the diaphragm and tube wall increases from 0 to 5 mm, thermal enhancement factor increases by 30%. Basically, this occurs due to a significant reduction in the coefficient of hydraulic losses (Fig. 12) since the relative value of $Nu_{sr}/Nu_0$ in this region of tube is reduced only by 20%. It should be marked out that the studied obstacle gives the high values of hydraulic resistance and cannot be used directly as a heat transfer intensifier. Nevertheless, the comparative analysis and identified flow and heat transfer features for the obstacles with different values of $c/h$ can be useful when selecting and optimizing the surfaces with enhanced heat transfer.
**Conclusion**

The turbulent structure and heat transfer behind a flat diaphragm in a round tube were studied numerically with variation of a gap between the tube wall and diaphragm \( A = c/h = 0 \div 0.33 \).

It is demonstrated that a wall jet generated in the slot leads to such a strong change in the separated flow behind a diaphragm that there is no repeated attachment of the flow for the large scale of the jet \( (A > 0.2) \). Due to the breakdown of recirculation zone, the level of kinetic energy decreases and it is entrained by the jet.

It is determined that with an increase in the distance between the diaphragm and tube wall, the maximal heat transfer is reduced significantly.

The integral characteristics show that for \( c/h = 0.033 \) and \( 0.067 \) heat transfer becomes more intensive due to the increased heat removal from the tube wall under the diaphragm.

The pressure losses decrease by three times at a change in the size of the wall jet from \( c/h = 0 \) to 0.33. At that, thermal enhancement coefficient increases by 30%.

**Acknowledgments**

The work was financially supported by the Russian Foundation for Basic Research (project 18-58-00011) and the state contract with IT SB RAS (AAAA-17-117030310010-9). The computational resources are provided by Siberian Supercomputer Center SB RAS.

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