# CALCULATION OF HEAT EXCHANGE IN PIPES WITH SPIRAL SHEET INTENSIFIERS

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A method of calculating the heat exchange in pipes with spiral sheet intensifiers is developed on the basis of a superposition of the effects of the separation and swirled motions of streams.

Pipes inside which spiral sheet intensifiers are mounted [1], which provide a considerable reduction in the weight and volume of the apparatus, have found practical application in a number of industrial heat exchangers. The heat exchange in such pipes is calculated on the basis of experimental work [2] in which the region of the operating parameters and geometrical dimensions of the spirals is limited. It is of practical and theoretical interest to develop a theory of heat exchange for such devices which will be applicable for pipes with internal ribbing also. The spiral intensifier consists of a steel ribbon (Fig. 1) with a width $h = 0.2d$, where d is the inner diameter of the pipe, and a thickness $\delta = 1$ mm. This ribbon is given a spiral shape with any pitch H/d with the help of a simple connector with a spiral channel mounted on an ordinary lathe.

A spiral mounted in a round pipe swirls and turbulizes the stream, which leads to an increase in the heat removal (for resistance losses equal to those for a smooth pipe) by 1.4 times [2]. The plane of the spiral in each cross section is oriented at a 90° angle to the pipe surface. The interaction of the stream with the spiral leads to the fact that part of the stream is swirled while the rest is separated from the upper end of the spiral, increasing the overall level of turbulence.

An investigation of the stream structure in such channels, made on an air installation using a thermoanemometer and visually on a water installation, showed that under stabilized conditions the axial velocity field is close to uniform and that the stream moves along the spiral, with the fraction of separated flow increasing with a decrease in the pitch of the spiral. An increased level of turbulence (by 30-50%) in comparison with ordinary rectilinear motion is observed in the investigated stream.

Let us resolve the velocity vector $u_0$ of the stream flowing onto the spiral into two components (Fig. 1): $W_2$ along the direction of the spiral (determines the stream swirling) and $W_1$ perpendicular to the plane of the spiral (provides stream separation). Thus, $W_2 = u_0 \cos\gamma$ and $W_1 = u_0 \sin\gamma$.

The procedure used clearly covers the two limiting cases: for $H/d \to \infty$ (H is the pitch of the spiral) rectilinear nonseparation flow is obtained; for small H/d purely separation flow is obtained. We assume that $W_2$ is the determining effective velocity of the swirled stream in the process of heat exchange, and in accordance with [4] we will determine Nu for the swirled-stream component from the equations for ordinary rectilinear motion with the introduction of the effective velocity into the Reynolds number:

$$\mathrm{Nu}_{s.s} = 0.021\,\mathrm{Re}_{W_2}^{0.8}\mathrm{Pr}^{0.4}. \tag{1}$$

This assumption, valid for moderate swirling, is justified by the agreement of the generalizing velocity profile $u_{tot}/v_x = f(yv_x/\nu)$ with the usual dependence for a round pipe, where $u_{tot} = \sqrt{u_x^2 + u_\tau^2}$.

We note that this correspondence is valid only for the boundary region; there are differences in the core of the stream. Since the main thermal resistance is concentrated in the boundary zone, however, this equivalence principle will be valid for a swirled stream with the introduction of the effective velocity.

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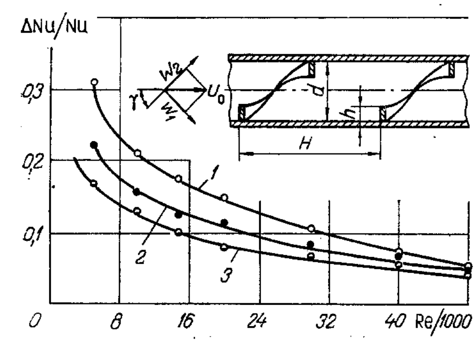


Fig. 1. Correction of heat exchange for vortex mixing in a pipe with sheet spirals: 1) H/d = 3.5; 2) 7; 3) 10.

Thus,

$$Nu_{s.s\ 1} = 0{,}021 \left(\frac{u_0 d}{\nu}\right)^{0,8} Pr^{0,4} \cos\gamma^{0,8}. \tag{2}$$

It was shown in [4, 7] that secondary flows which additionally intensify the heat exchange are observed in the swirled stream in a pipe containing a screw. Because of the pressure drop between the pipe wall and the center of the stream, the fluid slowed in the boundary layer flows along the surface of the screw, from the periphery to the center, providing additional mixing of the boundary layers of fluid with the core of the stream. The increase in heat exchange $Nu_{mix}$ due to this fact was determined analytically in [4] from the condition of mass exchange between the boundary layer and the core of the stream. Since in the case under consideration the spiral is located in one half of the pipe in a cross section, rather than in both halves as for a screw, and a significant part of the spiral is located in the core of the stream, we can assume that the influence of the effect of vortex mixing for the spiral can be allowed for by the correction $\Delta Nu = Nu_{mix}/2$, where $Nu_{mix}$ is calculated from the equations for a screw [4].

Since for a pipe the main thermal resistance is concentrated at a distance $y < h$ from the wall, the correction $\Delta Nu$ is approximately independent of h in the range of practical interest.

The dependence $\Delta Nu/Nu_{s.s\ 1} = f(H/d, Re)$, calculated from the data of [4], is shown in Fig. 1. Finally,

$$Nu_{s.s} = Nu_{s.s\ 1} + \Delta Nu. \tag{3}$$

To determine the component of heat exchange connected with stream separation, we draw upon an analysis based on the solution of the balance equation for the pulsation energy. The following expression for the Stanton number was obtained in [3] for the case of the turbulent kinetic energy near the wall:

$$St = \frac{\frac{\sqrt{k}}{u_0}}{80}\left(\frac{1.75}{Pr + 8.5} + 1\right)\left(0.37\,Pr^{0.608} + 0.000315\,Pr^{1.75} + \frac{1}{17.8}\ln\frac{\frac{\sqrt{k_0}}{u_0}Re}{160}\right)^{-1}. \tag{4}$$

On the basis of the jet model in [3] we obtained an expression for the maximum turbulent energy near the wall for transverse obstacles of small heights $y < \delta_B$ ($\delta_B$ is the thickness of the buffer region of the boundary layer). In the case under consideration the width of the ribbon exceeds the thickness of the buffer region, so a special determination of the quantity $\sqrt{k_1}/u_0$ is necessary, where $k_1$ is the maximum kinetic energy in the attachment region. To determine $\sqrt{k_1}/u_0$ let us consider the following flow scheme. We will assume that a sheet obstacle set up in a plane is flowed over by a turbulent stream with a developed velocity profile

$$u/u_{max} = (y/\delta)^{1/n}.$$

Here $\delta = R_0$. On the basis of a jet scheme it was shown in [3] that $\sqrt{k_1}/u_0 = cu_1/u_0$. We assume that when $y = h = \delta_l$ ($\delta_l$ is the thickness of the viscous sublayer), the influence of the sheet on the stream is negligibly small and $\sqrt{k_1}/u_0$ in this case corresponds to the value of $\sqrt{k_1}/u_0$ for a smooth plate or pipe, i.e., $\sqrt{k_1}/u_0 = 2v_x/u_0$ [5]. Determining the constant c from this condition, we have

$$\frac{\sqrt{k_1}}{u_0} = \frac{2v_x}{u_0}\left(\frac{h}{\delta}\right)^{1/n}. \tag{5}$$

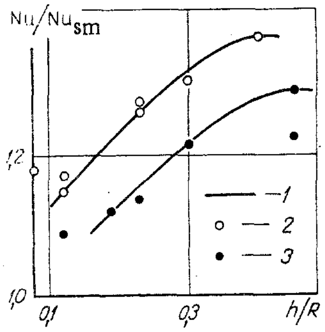


Fig. 2

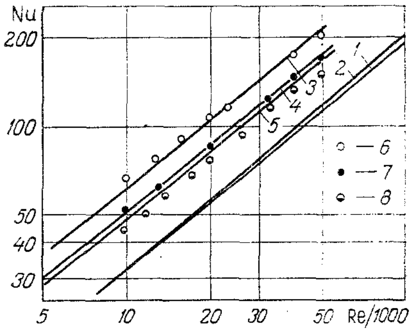


Fig. 3

Fig. 2. Relative increase in average heat transfer on a section of a plate behind a rectangular roughness element (projection) as a function of its height in application to a pipe: 1) calculation by Eqs. (4) and (8); experimental data of [6]; 2) H/d = 3.5; 3) 7.

Fig. 3. Heat exchange in pipes with spiral sheet intensifiers (Pr = 0.72). Smooth pipe: 1) [4]; 2) $Nu = 0.02Re^{0.8}$. Pipe with spirals, h/d = 0.2. Calculated data: 3) H/d = 3.5; 4) 7; 5) 10; experimental data of [2]: 6) H/d = 3.5; 7) 7; 8) 10.

For the conditions of a plate or a pipe the quantity $(y/\delta)^{1/n}$ depends little on the Reynolds number Re. The velocity field at the crest of a transversely mounted plate and behind it is self-similar. Therefore, the quantity $(y/\delta)^{1/n}$ was taken for the conditions of $Re = 10^4$, since this is the most popular region for industrial conditions, and the value $Re = 10^4$ also characterizes the limit of stability loss, i.e., the start of the turbulent region. For these conditions we have

$$\left(\frac{h}{\delta}\right)^{1/6.3} = \left(\frac{h}{d}\right)^{1/6.3}\left(\frac{Re^{7/8}}{57.7}\right)^{1/6.3} = 1.89\left(\frac{h}{d}\right)^{1/6.3} \tag{6}$$

In Eq. (6) we took $\delta v_x/\nu = 11.4$, i.e., a two-layer scheme of turbulent flow is being considered. Thus

$$\sqrt{k_1}/u_0 = 3.78\left(\frac{h}{d}\right)^{1/6.3}. \tag{7}$$

In the region behind the obstacle the kinetic energy distribution is determined by the expression [3]

$$\frac{\sqrt{k}/u_0 - \sqrt{k_0}/u_0}{\sqrt{k_1}/u_0 - \sqrt{k_0}/u_0} = \Phi(x/h),$$

where k, $k_1$, and $k_0$ are the kinetic energy at any point behind the obstacle, at the attachment point, and in a smooth pipe, respectively. Using the expression for Φ from [3], we can determine the average kinetic energy for any distance x/d behind the obstacle, while in our case x/d = H/d:

$$\left(\frac{\sqrt{k}}{u_0}\right)_{sep} = \frac{1}{\bar{x}}\int\limits_0^{\bar{x}}\left[\Phi\left(\frac{\sqrt{k_1}}{u_0} - \frac{\sqrt{k_0}}{u_0}\right) + \frac{\sqrt{k_0}}{u_0}\right]d\bar{x}. \tag{8}$$

According to [3], $\Phi = \frac{1}{6}\bar{x}/h$ in the region of $6 > \bar{x} > 0$, $\Phi = 1$ for $9 > \bar{x} > 6$, and $\Phi = \exp(-[\bar{x} - 9]/16.65)$ for $\bar{x} > 9$.

This analysis is valid only for $h/R < h/R|_{lim}$. When $h/R > h/R|_{lim}$ an increase in heat exchange does not occur with an increase in h/R, since it was shown in [3] that layers with a negligible thermal resistance are turbulized in this case.

According to [3], $y/R|_{lim} = 0.365/\sqrt{Pr}$. For Pr = 0.72, $y/R|_{lim} = 0.432$. In accordance with the theory developed, $St = const = St|_{y/r=0.432}$ for $y/R > y/R|_{lim}$. Using Eqs. (4) and (8) we can determine $Nu_{sep}$ for the corresponding velocities $W_1$.

A comparison of the results of a calculation by Eqs. (4) and (8) with the experiments of V. M. Buznik [6] is shown in Fig. 2. As seen from the graphs, the theory provides close agreement of the calculated and experimental data.

We determine the heat exchange in pipes with spiral sheet swirlers by the principle of superposition of the swirling and separation effects:

$$Nu = Nu_{s.s} + Nu_{sep}. \tag{9}$$

The quantity $Nu_{s.s}$ is found from (2), while the quantity $Nu_{sep}$ is determined from (8) with the corresponding Re determined from the components $W_1$ and $W_2$. The results of the calculations and a comparison with the experimental data of [2] are shown in Fig. 3, the data of which indicate the close agreement of the theory under consideration and experiment and approbate the calculation procedure.

## NOTATION

$u_0$, average flow-rate velocity in pipe or outside boundary layer for a plate; $u_x$, axial velocity component of spiral motion; u, velocity in boundary layer; $u_1$, velocity at end of projection; $u_{max}$, velocity at pipe axis; $W_2$, velocity component parallel to plane of spiral; $W_1$, velocity component perpendicular to plane of spiral; v, tangential component of spiral motion; $v_x = \sqrt{\tau_w/\rho}$, dynamic velocity; x, y, coordinates; $\gamma$, angle between direction of spiral and generatrix of pipe; $\nu$, coefficient of kinematic viscosity; d, $R_0$, inner diameter and radius of pipe; H, pitch of spiral; h, height of spiral; $\delta$, thickness of boundary layer; Nu, Re, Pr, St, Nusselt, Reynolds, Prandtl, and Stanton numbers.

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