STUDY CONCERNING THE LOCAL HEAT TRANSFER COEFFICIENTS FOR A TUBE WITH A LOCAL TWISTING OF THE STREAM BY MEANS OF BUCKET WHIRLERS

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Results of an experimental study concerning the local heat transfer coefficients for tubes with bucket whirlers are shown here in the form of criterial equations.

Whirling a stream of heat carrier fluid is one effective method of improving the heat transfer. It has been confirmed in several studies on the subject [1-6] that owing to the high efficiency of this method, by which the heat dissipation per unit surface area can be increased appreciably without an increase in drag losses, it becomes feasible to reduce the size and the weight of a heat exchanger.

The degree of improvement depends on the whirl intensity, on the geometrical and structural characteristics of the whirler, on the Reynolds number, and on the tube length. Most complete information is

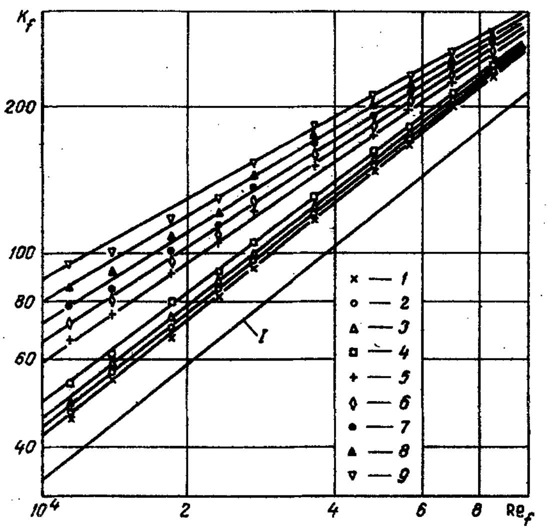

Fig. 1

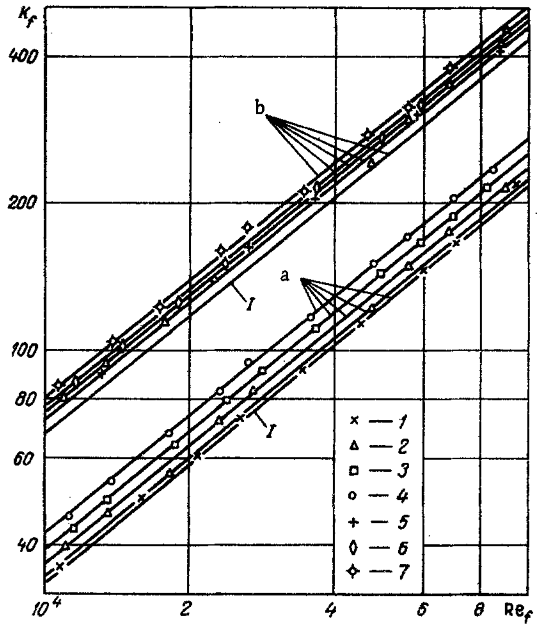

Fig. 2

Fig. 1. Heat transfer rate as a function of the tube length and of the Reynolds number: $l/d = 60$ (1), $z/d = 40$ (2), 20 (3), 10 (4), 5 (5), 4 (6), 3 (7), 2 (8), 1 (9), tube without whirler (I).

Fig. 2. Effect of the bucket whirler geometry on the heat transfer in the tube with $l/d = 60$, (a) $K_f = f(Re_f)$, (b) $2K_f = f(Re_f)$, tube without whirler (I), $\varphi = 30°$ and $n = 0$ (1), $\varphi = 45°$ and $n = 0$ (2), $\varphi = 60°$ and $n = 0$ (3), $\varphi = 75°$ and $n = 0$ (4), $\varphi = 45°$ and $n = -1$ (5), $\varphi = 45°$ and $n = 1$ (6), $\varphi = 45°$ and $n = 3$ (7).

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TABLE 1. Correction Factor ε_l for the Tube Length, within the Re_f = 10,000-30,000 Range

	$n=0$				$\varphi=45°$		
z/d	$\varphi=30°$	$\varphi=45°$	$\varphi=60°$	$\varphi=75°$	$n=-1$	$n=1$	$n=3$
			$Re_f=10^4$				
1	1,58	1,84	1,92	2,04	1,55	1,66	1,87
2	1,46	1,72	1,72	1,84	1,46	1,52	1,70
3	1,40	1,56	1,60	1,66	1,39	1,43	1,59
4	1,34	1,44	1,48	1,50	1,32	1,36	1,48
5	1,30	1,36	1,40	1,40	1,26	1,30	1,41
10	1,12	1,13	1,16	1,17	1,08	1,10	1,17
20	1,04	1,06	1,06	1,06	1,05	1,06	1,07
40	1,02	1,02	1,02	1,02	1,02	1,02	1,02
60	1,00	1,00	1,00	1,00	1,00	1,00	1,00
			$Re_f=3\cdot10^4$				
1	1,42	1,52	1,56	1,59	1,36	1,44	1,55
2	1,36	1,43	1,46	1,50	1,30	1,37	1,44
3	1,30	1,35	1,37	1,40	1,24	1,31	1,36
4	1,26	1,28	1,30	1,34	1,20	1,25	1,31
5	1,23	1,22	1,24	1,26	1,17	1,20	1,26
10	1,11	1,10	1,11	1,11	1,07	1,08	1,12
20	1,04	1,05	1,06	1,06	1,05	1,05	1,06
40	1,02	1,02	1,02	1,02	1,02	1,02	1,02
60	1,00	1,00	1,00	1,00	1,00	1,00	1,00

available about the heat transfer in tubes with paddle whirlers, with tangential fluid feed, and with bucket whirlers mounted on hubs. Not enough is known about the heat transfer in tubes with hubless and specially profiled buckets.

For an experimental study of local heat transfer coefficients the authors used the gradient method [7, 8].

The local heat transfer coefficient for a tube wall with constant thermal conductivity is defined as

$$\alpha = \frac{\lambda}{\Delta T}\cdot\frac{d\varepsilon(z)}{dz}. \tag{1}$$

Here

$$\Delta T = T_w - T_f;\ \ T_f = T_{f_0} + \frac{2\pi r_1 \lambda \varepsilon(z)}{Gc_p};\ \ \varepsilon(z) = \int_0^z \left(\frac{\partial t}{\partial r}\right)_{r_1} dz.$$

Function $\varepsilon(z)$ is determined from the temperature field in the tube wall, which in turn can be found by solving the differential equation of heat conduction with the temperature distribution on the tube surface known from tests.

Our experimental study of local heat transfer coefficients was made on eight whirler variants with buckets profiled to satisfy the relation ur^n = const. The buckets were mounted at the entrance to a tube

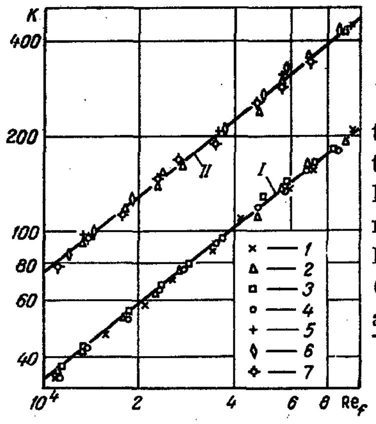

Fig. 3. Universal characteristics of heat transfer in tubes with l/d = 60 and a local stream twisting by means of bucket whirlers: $K_\varphi = f(Re_f)$ (I), $2K_n = f(Re_f)$ (II). Parameters φ and n as in Fig. 2 (1-7).

TABLE 2. Correction Factor ε_l for the Tube Length, at Re_f = 90,000

φ=30–75°, n=–1–3

z/d	1	2	3	4	5	10	20	40	60
ε_l	1,26	1,20	1,17	1,14	1,11	1,06	1,03	1,01	1,00

with a diameter d = 32.5 mm and a relative length of 60d. The geometrical characteristics of the whirler and of the test stand have been described in [9].

The test data were processed on a Ural-2 computer. The local heat transfer coefficients for each whirler were calculated with 8-10 different values of the Reynolds number and 16 values of the relative tube length.

The obtained values of the local heat transfer coefficient for a tube and a whirler with φ = 75°, n = 0 are shown in Fig. 1 at nine different values of z/d. Line I represents the heat transfer in a tube with l/d = 60 and without a whirler.

An analysis of the curves in Fig. 1 has shown that whirling a stream with φ = 75° and n = 0 buckets produces a 25% higher heat transfer coefficient in a tube with a relative length of l/d = 60. At a shorter z/d and a higher Reynolds number the heat transfer coefficient along the tube increases at a slower rate.

Data on the local heat transfer coefficients for a tube with l/d = 60 and sets of whirler buckets: 1) n = 0 and φ = 30°, 45°, 60°, 75°, 2) φ = 45° and n = 0, −1, 1, 3 are shown in Fig. 2. According to the graphs, the use of bucket whirlers yields 25% higher local heat transfer coefficients. Most improvement in the heat transfer coefficient is achieved by twisting the stream with φ = 75° and n = 0 buckets. Twisting the stream with φ = 30° and n = 0 buckets produces almost no improvement of the heat transfer in a long tube. As n is increased, the heat transfer improves: by 8-9% with n = 3 and φ = 45°.

In order to generalize the data on local heat transfer coefficients in a twisted stream, we applied the theory of similarity. The effect of twisting was accounted for by introducing into the criterial equation a geometric factor dependent on φ and n.

The results pertaining to local heat transfer coefficients with n = 0 and φ = 30-75° in tubes with a relative length 60d and for a Reynolds number ranging from 10,000 to 90,000 have been generalized by a criterial equation which applies to straight tubes without whirler, with a correction accounting for the effect of stream twisting on the heat transfer:

$$\mathrm{Nu}_f = 0.022\,\mathrm{Re}_f^{0.8}\mathrm{Pr}_f^{0.43}\left(\frac{\mathrm{Pr}_f}{\mathrm{Pr}_w}\right)^{0.25}(1+0.1485\varphi^{*2.02}). \tag{2}$$

This equation corresponds to line $K_\varphi = f(\mathrm{Re}_f)$ in Fig. 3. Here

$$K_\varphi = \frac{K_f}{1+0.1485\varphi^{*2.02}}, \tag{3}$$

$$K_f = \frac{\mathrm{Nu}_f}{\mathrm{Pr}_f^{0.43}\left(\frac{\mathrm{Pr}_f}{\mathrm{Pr}_w}\right)^{0.25}}. \tag{4}$$

The maximum deviation of test points from Eq. (2) does not exceed 6%.

For bucket whirlers with φ = 45° and n = 0, −1, 1, 3 the criterial equation for the local heat transfer coefficients in tubes with a relative length 60d is

$$\mathrm{Nu}_f = 0.024\,\mathrm{Re}_f^{0.8}\mathrm{Pr}_f^{0.43}\left(\frac{\mathrm{Pr}_f}{\mathrm{Pr}_w}\right)^{0.25}(1+0.027n) \tag{5}$$

for the 10,000-90,000 range of the Reynolds number.

This equation corresponds to line $2K_n = f(\mathrm{Re}_f)$ in Fig. 3. Here

$$K_n = \frac{K_f}{1 + 0.027n} . \qquad (6)$$

The maximum deviation of test points from Eq. (5) does not exceed 6%.

At $z/d < 60$ the heat transfer coefficients calculated by formulas (2) and (5) must be multiplied by a correction factor ε_l, whose numerical values shown in Tables 1 and 2 have been obtained for all our whirlers by a proper evaluation of test data.

At $Re_f = 90{,}000$ this correction factor for the tube length is almost the same for all the whirlers studied here (Table 2).

NOTATION

c_p is the specific heat in a constant-pressure process, J/kg · °C;
d is the tube diameter, m;
G is the mass rate of fluid flow, kg/sec;
l is the tube length, m;
Nu is the Nusselt number;
n is the exponent of power-law;
Pr is the Prandtl number;
r is the radial coordinate, m;
r_1 is the inside tube radius, m;
Re is the Reynolds number;
T is the temperature, °K;
u is the circumferential velocity, m/sec;
z is the longitudinal coordinate, m;
α is the heat transfer coefficient, W/m^2 · °C;
ε_l is the correction factor to heat transfer coefficient, accounting for the tube length;
λ is the thermal conductivity, W/m · °C;
φ is the angle between velocity vector and tube axis, deg (°);
φ^* is the angle, radians.

Subscripts

0 refers to tube exit;
f refers to heat carrier fluid;
w refers to tube wall.

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