INVESTIGATION OF THE INTENSIFICATION OF HEAT TRANSFER DURING FLOW OF GASES AND LIQUIDS IN PIPES

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The article presents the results of an experimental investigation of intensification of heat transfer by means of periodically arranged annular diaphragms in pipes with heating and cooling of air, carbon dioxide, helium, water, and water–glycerol mixture in the range Re = 500-2·10^5 and Pr = 0.7-50.

Intensification of heat transfer during turbulent flow of a heat-transfer agent in channels and especially in circular pipes is an urgent problem for many branches of engineering. By means of intensification it is possible either to reduce the size and weight of tubular heat-exchange apparatuses or devices, or to reduce the temperature of the heat-transferring surface in the channels of atomic reactors or other heat-transferring devices with a prescribed heat release.

A common method of intensifying heat transfer is to create turbulence of the flow artificially, which increases the turbulent heat-transfer coefficient A_q. Artificial agitation of the flow is usually attained at the price of an increase of the resistance coefficient.

We can show that a heat-exchange apparatus in which heat-transfer intensification methods are used will be smaller than a smooth-pipe apparatus (for the same heat fluxes) if the inequality

$$\left(\frac{\mathrm{Nu}}{\mathrm{Nu_s}}\right)^2 > \left(\frac{\xi}{\xi_s}\right)_i \left(\frac{\mathrm{Re}}{\mathrm{Re}_s}\right)^{1,2} \frac{(T_w - T_l)_s^2}{(T_w - T_l)^2} \tag{1}$$

is fulfilled. Here the subscript i denotes that the ratio of Nu or ξ is taken for a pipe with intensification and smooth pipe at the same Re numbers. The second and third factors on the right are the ratios of the Re numbers and thermal heads for pipes of the heat-exchange apparatuses being compared.

As an analysis shows [1, 8], inequality (1) can be secured most successfully by agitating only comparatively thin wall layers of the transfer agent. In addition, the method of intensification should satisfy the following requirements: possibility of mass production of pipes with intensification; the assembly of the heat-exchange apparatus from such pipes should be no more complex than from smooth pipes.

The reliability of operation of heat-exchange apparatus from pipes with intensification should be no worse than that of a smooth-pipe apparatus.

The method of intensification in which knurling is used to create small periodically arranged diaphragms on the inside surface and grooves on the outside surface satisfies inequality (1) and all aforementioned requirements.

Mass production according to the technology of the All-Union Scientific Research, Planning, and Design Institute of Metallurgical Machinery is simple and cheap. Knurling is done on lathes with additional equipment.

The grooves on the outside surface intensify heat transfer on the outside of the pipes during longitudinal flow past bundles of pipes [7, 8]. Special tests showed that the fatigue strength of knurled pipes is close to that of smooth.

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TABLE 1. Characteristics of Knurling of Experimental Pipe Specimens

Air, carbon dioxide, helium													
t/D	0,5	0,5	1,0	1,0	1,0	1,0	1,5	2,5	2,5	5	5	7	10
d/D	0,97	0,95	0,97	0,95	0,93	0,9	0,97	0,95	0,9	0,95	0,95	0,95	0,9
Water										Water – glycerol mixture			
t/D	0,496	0,498	0,497	0,523	0,496	1,0	0,992	2	4		0,497	0,523	
d/D	0,983	0,966	0,943	0,922	0,875	0,94	0,912	0,94	0,94		0,943	0,922	

The results of an experimental investigation of heat transfer and hydraulic resistance in knurled pipes for gases and liquids are given below.

The experimental device for investigating intensification on gases was an open circuit. The gas (carbon dioxide, air, helium) was delivered from high-pressure cylinders through a filter, drier, and reducer to the flow-rate measuring section and then successively to the heater and cooler. The device for water and water–glycerol mixture also operated as an open circuit with free overflow of the liquid into a tank from where it was pumped to the measuring section, heater, cooler, and final cooler. The investigations were carried out with heating under conditions q_W = const = 10^3-$2 \cdot 10^5$ W/m^2 and cooling in stainless steep pipes of diameter 10.6/9.6 mm; the length of the heated sections was 110 diameters. The wall temperature was measured by Chromel–Alumel and Chromel–Constantan thermocouples welded at a rate of 20 per section, in combination with semiautomatic potentiometers p–2/1. The temperature of the flow at the entrance and exit was measured by laboratory thermometers in the mixing chambers. In addition, the temperature of the gas flow was measured by a longitudinal thermocouple whose junction, by means of a special device, could be moved along the axis and radius of the pipe in any cross section. The pressure drop in the experimental section was measured by micromanometers, water differential manometers, mercury differential manometers, and standard manometers. The flow rate of the transfer agent was measured by normal diaphragms and by the volumetric method. Table 1 presents the characteristics of the experimental specimens investigated on gases and liquids.

The experimental investigation of heat-transfer intensification was preceded by experiments on heat transfer and hydraulic resistance in smooth pipes, the object of which was to convince us of the reliability of the method of measuring and treating the experimental data and to obtain standard calculation dependences. The experimental data on smooth pipes are generalized by the following dimensionless dependences.

1. Viscosity regime, heating of water [2]:

a) local heat transfer

$$\mathrm{Nu}_x (\mathrm{Pr}_W/\mathrm{Pr}_l)_x^{1/3} = 4.36 X^{-0.5} \cdot 10^{-18X}, \tag{2}$$

where $X = (x/d)/\mathrm{Re}_{en}$. Here all quantities with the subscript x are referred to the temperature of the liquid (and Pr_W to T_W) in cross section x, and Re_{en} is referred to the temperature of the liquid at the entrance;

b) average heat transfer

$$\mathrm{Nu}_s = 1.64 (\mathrm{Pe}\, d/L)^{1/3} (\mu_l/\mu_W)^{1/3}; \tag{3}$$

c) average coefficient of hydraulic resistance

$$\xi_s = \frac{64}{\mathrm{Re}} (\mu_W/\mu_l)^{0.14}. \tag{4}$$

In cases b) and c) the average temperature over the pipe length is taken as the characteristic temperature.

2. Transition region, $2400 < \mathrm{Re} < 10^4$, heating of liquid:

a) average heat transfer

$$\mathrm{Nu}_s = 0.11 (\mathrm{Re}^{2/3} - 125) \mathrm{Pr}^{0.445}; \tag{5}$$

the characteristic temperature is the average temperature over the pipe length;

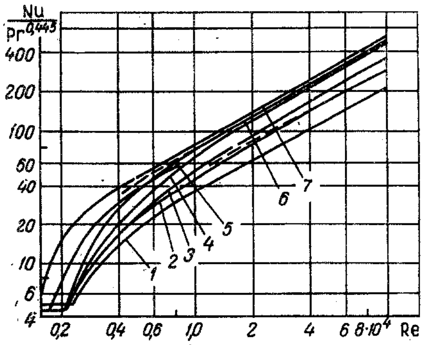

Fig. 1

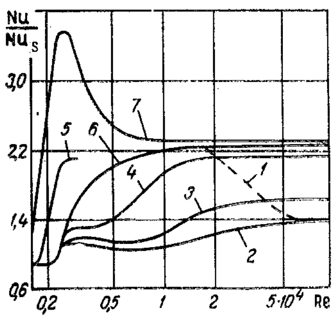

Fig. 2

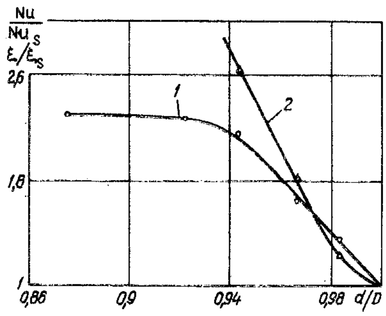

Fig. 3

Fig. 1. Effect of Re number on average heat-transfer coefficient in pipes: 1) smooth pipe; 2) d/D = 0.983, t/D = 0.496; 3) 0.966, 0.498; 4) 0.943, 0.497; 5) 0.912, 0.991; 6) 0.922, 0.523; 7) 0.875, 0.496. Solid lines: water; dashed lines: water–glycerol, d/D = 0.92 and 0.966.

Fig. 2. Effect of Re number on effectiveness of heat-transfer intensification: 1) boundary of Re* numbers; 2-7) see Fig. 1.

Fig. 3. Effect of height of agitators on effectiveness of heat-transfer intensification for step d/D = 0.5 and Re = 50,000 (water): 1) Nu/Nu_s; 2) ξ/ξ_s.

b) average coefficient of hydraulic resistance, the data are given in Table 2.

3. Turbulent region:

a) average heat transfer with heating of gases (T_w/T_l = 1-1.6)

$$Nu_s = 0.0207\, Re^{0.8}\, Pr^{0.43}; \tag{6}$$

b) average heat transfer with cooling of gases (T_w/T_l = 0.7-1)

$$Nu_s = 0.0192\, Re^{0.8}\, Pr^{0.43}; \tag{7}$$

c) average heat transfer with heating and cooling of liquids and gases Pr = 0.7-50 and Re = 10^4-10^5

$$Nu_s = 0.0216\, Re^{0.8}\, Pr^{0.445}\psi, \tag{8}$$

where $\psi = 1$ for heating of liquids; $\psi = (T_w/T_l)^{-0.55}$ for heating of gases; $\psi = 1.27-0.27\,(T_w/T_l)$ for cooling of gases. The exponent of the Pr number is determined experimentally for liquids. The characteristic temperature in cases a), b), c) is the average temperature of the liquid over the pipe length;

d) average coefficient of hydraulic resistance

$$\xi_s = \frac{0.316}{Re^{0.254}}(\mu_w/\mu_l)^n, \tag{9}$$

where n = 0.14 for heating of gases, n = 0.2 for cooling of gases, and n = 1/3 for heating of liquids.

These dimensionless dependences generalize the experimental data on heat transfer with a scatter of ±6.5% in the viscosity regime, ±10% in the transition, ±5% in the turbulent, and ±4-6% with respect to resistance.

Results of Investigating Heat Transfer and Hydraulic Resistance in Knurled Pipes. Figures 1-3 present some results of investigating heat-transfer intensification. We see from Fig. 2 that we can distinguish three characteristic regions of Reynolds numbers in which the effect of heat-transfer intensification is governed by different regularities [5].

1. The region of subcritical Reynolds numbers, $Re < Re_{cr}$, is characterized by smaller values of the heat-transfer coefficients in knurled pipes in comparison with a smooth pipe, i.e., $Nu \le Nu_s$. The reason lies in the fact that, in the case of laminar flow, stagnant zones are created between the projections of the diaphragms which are an additional thermal resistance of the little-moving

liquid layer. We note that the critical Reynolds number in knurled pipes decreases with increase of the height of the agitators [4] and at $d/D = 0.875$, $Re_{cr} = 1580$. Thus this region of Reynolds numbers is of no practical interest.

2. The region of transition and weakly developed turbulence, $Re_{cr} < Re < Re^*$, is characterized by a dependence of the effect of heat-transfer intensification Nu/Nu_s on Re numbers as a consequence of the earlier transition of laminar flow to turbulent in knurled pipes. The upper limit of this region (it is shown in Fig. 2 by the dashed line) Re* corresponds to those values of the Reynolds numbers beginning with which the effect of heat-transfer intensification Nu/Nu_s for a given height of the agitators ceases to depend on the Re numbers, i.e., $Nu/Nu_s > 1 = \mathrm{const}$. It was shown in [5] that stabilization of the ratio Nu/Nu_s is achieved at those Reynolds numbers at which the height of the agitators becomes equal to the thickness of the wall layer in which 99% of the total thermal head between the pipe wall and axis is produced. The values of Re* found analytically agree satisfactorily with the experimental data and are described by the following relation:

$$Re^* = \frac{3150}{(1-d/D)^{1.14}\,Pr^{0.57}}. \tag{10}$$

The region being considered is of great interest with respect to heat-transfer intensification. The maximum increase (3.5 times) of heat transfer with a liquid is obtained precisely in it owing to the earlier transition to turbulent flow. Here also occur the best relations between Nu/Nu_s and ξ/ξ_s at high absolute values of these quantities. Thus, for a pipe with $d/D = 0.91$ and $t/D = 1.0$, $Nu/Nu_s = 2.05$ and $\xi/\xi_s = 1.95$. With an increase of the Pr number (at $Pr > 10$ for water–glycerol) Re* decreases and the effect of intensification increases, which is seen clearly in Fig. 1. The character of the dependences $Nu/Nu_s = f(Re)$ and $\xi/\xi_s = f_1(Re)$ in this region is rather complex, and the experimental data cannot be generalized by simple empirical dependences. Therefore, calculation recommendations are given in Table 2. We see from Table 2 that in this region it is expedient to use comparatively high agitators, $d/D \le 0.92$, since they promote an earlier transition and agitate thicker wall layers. We should note that the transition region is characterized by intermittence of the flow with pronounced fluctuations of the heat-transfer coefficient [3]. The use of pipes with artificial agitators reduces the region of Re numbers where intermittence appears, shifts it toward smaller Re number [4], and, with consideration of the effect of heat transfer obtained, permits broadening the range of Reynolds numbers with high and stable values of the heat-transfer coefficients to $Re = 2000$.

3. The region of developed turbulent flow, $Re > Re^*$, as already noted, is characterized by the fact that the effect of intensification $Nu/Nu_s > 1 = \mathrm{const}$ and does not depend on the Re and Pr numbers. In this region the wall layers of the liquid in which the main thermal resistance of the flow is concentrated are equal to and less than the heights of the agitators. Therefore, the best effects of heat-transfer intensification are attained at small heights of the agitators and their comparatively frequent arrangement. It follows from Fig. 3 that for small heights of the agitators we can obtain $Nu/Nu_s > \xi/\xi_s$ at absolute values of these ratios of about 1.5. A further increase of the height of the agitators leads to agitation of the layers of the flow, which as it is have a high degree of turbulence. Therefore, heat transfer increases weakly (and not at all for liquids), whereas the coefficient of hydraulic resistance continues to increase intensely.

The experimental data in the investigated ranges of variation of knurling are generalized by the following empirical dependences.

Heat transfer of gases (heating and cooling) with an accuracy of ±12%:

$$\frac{Nu}{Nu_s} = \left(1 + \frac{\lg Re_w - 4{,}6}{7{,}45}\right)\left(\frac{1{,}14 - 0{,}28\sqrt{1-d/D}}{1{,}14}\right)\exp\left[\frac{9(1-d/D)}{(t/D)^{0{,}58}}\right], \tag{11}$$

where Nu_s is taken according to (8).

Heat transfer of liquids (heating) for $t/D = 0.5$ and $d/D \ge 0.94$ with an accuracy of ±10%:

$$\frac{Nu}{Nu_s} = [100(1-d/D)]^{0{,}445}, \tag{12}$$

where Nu_s is taken according to (8).

TABLE 2. Results of Investigating the Intensification of Heat Transfer on Liquids for Some Experimental Pipes

d/D	t/D	Re	1580	2000	2510	3160	3980	5000	6300	7950	10000	12600	15800	20000	25100	31600	39800	50000	63000
0,983	0,496	A	0,955	1,07	1,340	1,193	1,110	1,097	1,097	1,097	1,111	1,191	1,248	1,260	1,289	1,308	1,346	1,356	1,366
		B	1,0	1,07	1,117	1,080	1,097	1,121	1,135	1,150	1,170	1,178	1,189	1,192	1,204	1,216	1,227	1,230	1,236
0,966	0,498	A	0,94	1,0	1,340	1,193	1,134	1,134	1,161	1,198	1,278	1,352	1,432	1,518	1,607	1,628	1,689	1,645	1,645
		B	1,0	1,0	1,225	1,217	1,270	1,324	1,365	1,412	1,475	1,530	1,591	1,650	1,730	1,768	1,807	1,841	1,880
0,943	0,497	A	1,0	1,07	1,403	1,340	1,367	1,430	1,550	1,651	1,868	1,978	2,097	2,167	2,167	2,167	2,167	2,167	2,167
		B	1,047	1,07	1,439	1,412	1,531	1,719	1,834	1,965	2,097	2,208	2,320	2,380	2,420	2,505	2,580	2,630	2,700
0,922	0,523	A	1,03	1,0	1,500	1,883	1,995	2,080	2,120	2,120	2,170	2,200	2,260	2,270	2,270	2,270	2,270	2,270	2,270
		B	1,12	1,15	1,830	2,140	2,420	2,750	3,00	3,260	3,480	3,650	3,820	4,00	4,180	4,400	4,620	4,820	5,05
0,875*	0,496	A	1,0	2,88	3,52	2,93	2,60	2,425	2,33	2,27	2,27	2,28	2,285	2,300	2,300	2,300	2,300	2,300	2,300
		B	1,62	2,85	1,55	4,50	4,95	5,43	5,78	6,11	6,52	6,94	7,34	7,80	8,26	8,78	9,32	9,84	10,45
0,912	0,991	A	0,94	1,41	2,06	2,05	1,985	1,931	1,896	1,853	1,853	1,833	1,796	1,750	1,695	1,645	1,591	1,534	1,400
		B	1,26	1,48	2,013	1,95	2,09	2,275	2,418	2,550	2,700	2,820	2,960	3,070	3,155	3,240	3,320	3,390	3,460
Smooth		ξ_s	0,0405	0,032	0,034	0,39	0,038	0,0358	0,0342	0,0334	0,0298	0,288	0,0271	0,0257	0,0240	0,228	0,0215	0,0220	0,0190

Note. $A = Nu/Nu_s$; $B = \xi/\xi_s$

The hydraulic resistance of gases and liquids with an accuracy of ±10%:

$$\frac{\xi}{\xi_s} = \left[1 + \frac{100(\lg \mathrm{Re} - 4{,}6)(1 - d/D)^{1,65}}{\exp(t/D)^{0,3}} \right] \exp\left[\frac{25(1 - d/D)^{1,32}}{(t/D)^{0,75}} \right], \tag{13}$$

where ξ_s is taken according to (9). Equation (13) is valid in the entire range of knurlings investigated for gases and for liquids when $d/D \geq 0.94$ and $t/D = 0.5$-4.

The effect of the temperature factor on the coefficient of hydraulic resistance was not specially taken into account for gases. However, it was established that with an increase of the height of the diaphragms the effect of the temperature factor decreases, which could be expected since the temperature gradient near the wall becomes smaller. The effect of temperature on the coefficient of hydraulic resistance under conditions of artificial agitation of the flow was investigated in greater detail on liquid in [6]. It was found that

$$\xi = \xi_0 (\mu_s / \mu_l)^m, \tag{14}$$

where

$$m = 1/3\,(d/D)^{26,4} \tag{15}$$

when $t/d = 0.5$ and

$$m = 1/3 \cdot 10^{-\frac{0,369}{t/D}(1,217 + \lg t/D)} \tag{16}$$

when $d/D = 0.94$.

As already noted, one of the advantages of the proposed method of intensifying heat transfer is the simultaneous intensification on both side of the heat-transferring surface. Comparative results of calculating gas–water coolers made of smooth pipes and pipes with various knurlings are presented in [7]. The comparison showed that the use of heat-transfer intensification by means of annular diaphragms and grooves reduces the weight and volume of the heat-transferring part of heat-exchange apparatuses by about 1.5 times with unchanged hydraulic resistances.

NOTATION

D is the inside diameter of smooth pipe;
d is the inside diameter of diaphragm;
t is the spacing of diaphragms;
T is the temperature.

Subscripts

s denotes smooth;
0 denotes the isothermal flow;
w denotes the wall;
l denotes the liquid.

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