IMPROVING THE HEAT TRANSFER IN ANNULAR CHANNELS

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Results are shown of a study concerning an improvement of the heat transfer by means of circumferential grooves around an annular channel with a diameter ratio $d_2/d_1 = 1.455$.

Results of an experimental study were shown in [1, 2] concerning an improvement of the heat transfer by means of grooves spaced uniformly at various relative pitches around pipe bundles in a longitudinal stream. The grooves were produced by crimping. This method of improving the heat transfer is applicable also to annular channels. Grooves around the outside surface of the inner pipe improve the heat transfer between this surface and the stream through the annular channel. At the same time, all advantages of this method realizable in the case of closely packed pipe bundles are retained here too.

The authors have studied the heat transfer in an annular air channel with an outside diameter $d_2 = 16$ mm and an inside diameter $d_1 = 11$ mm (Fig. 1). The air in the annular gap was heated by hot air flowing inside the pipe. The heated segment was made 1008 mm long ($l/d_e = 201.6$). We determined the mean heat transfer rate over the entire channel by measuring the air flow rate, both the inlet and the outlet air temperature, and the temperature of the pipe wall with 16 Chromel–Copel thermocouples.

The improvement in the heat transfer was evaluated on eight pipe specimens with different groove depth d/d_1 and groove pitches t/d_1: 1) $t/d_1 = 0.237$ with $d/d_1 = 0.953$, 0.970; 2) $t/d_1 = 0.459$ with $d/d_1 = 0.881$, 0.900, 0.928, 0.960; and 3) $t/d_1 = 0.910$ with $d/d_1 = 0.863$, 0.922.

The basic parameters in these tests were varied within the following ranges of values: inlet air temperature 5-20°C, outlet air temperature 50-120°C, Reynolds number Re_b = 10,000-100,000, air pressure 4.5-12 atm abs., and temperature factor T_w/T_b = 1.15-1.20.

The presence of grooves was disregarded in the determination of the equivalent diameter, the wetted parameter, and the transition section area. The heat transfer coefficient was referred to a smooth pipe. The mean temperature of the stream was defined as the reference temperature and the equivalent diameter $d_e = d_2 - d_1$ was defined as the reference dimension.

The heat transfer data obtained in these tests are shown in Fig. 2. The heat transfer rate is 30-50% higher with grooves than with a smooth channel surface. The ratio Nu/Nu_{sm} increases as d/d_1 and t/d_1

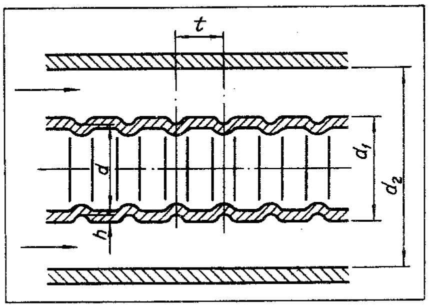

Fig. 1. Longitudinal section through an annular channel with grooves around the inside surface.

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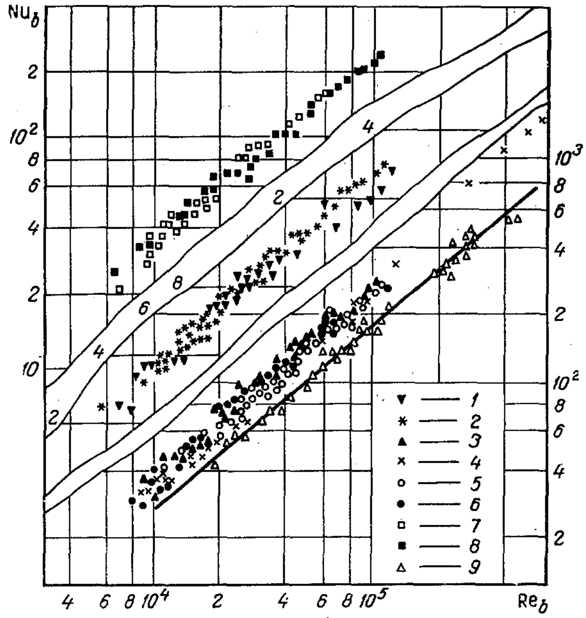

Fig. 2. Heat transfer in annular channel with grooves on internal surface: 1) $d/d_1 = 0.953$; $t/d_1 = 0.227$; 2) respectively 0.970 and 0.227; 3) 0.881 and 0.455; 4) 0.90 and 0.455; 5) 0.928 and 0.455; 6) 0.96 and 0.455; 7) 0.863 and 0.91; 8) 0.922 and 0.91; 9) smooth channel.

increase. The heat transfer rate increases only slightly with a higher Reynolds number as long as Re $<(4\text{-}6)\cdot 10^4$ and does not change with the Reynolds number when the latter is very high. For generalizing the test data, as was done in the case with pipe bundles, we used the ratios of groove depth and groove pitch to equivalent diameter (h/d_e and t/d_e). In our tests $h/d_e = 0.033\text{-}0151$ and $t/d_e = 0.5\text{-}2.0$. These data have been plotted in Fig. 3 for Re = 60,000 as a function of the groove depth h/d_e. Also shown here are data obtained by Kemeny and Cyphers [3] for circumferential semicircular grooves ($h/d_e = 0.0275$, $t/d_e = 0.275$), data obtained by Bauer [4] for triangular grooves ($h/d_e = 0.0697$, $t/d_e = 0.221$), as well as data obtained by Bogdanov, Korshakov, and Utkin [5] for triangular grooves ($h/d_e = 0.158$, $t/d_e = 0.247$). For $h/d_e \leq 0.158$ and $t/d_e = 0.22\text{-}2.0$ the data fit the universal relation (Fig. 3):

$$\frac{Nu}{Nu_{sm}} = 1 + 0.64\left[1 - \exp\left(-35.8\frac{h}{d_e}\right)\right]\left(1 - 0.274\frac{t}{d_e}\right). \tag{1}$$

This relation is valid when Re = 40,000-100,000 and when the ratio of outside to inside diameter $d_2/d_1 = 1.133\text{-}1.455$. At lower values of the Reynolds number the ratio Nu/Nu_{sm} becomes smaller. At Re = 10,000 the ratio Nu/Nu_{sm} is 10-15% smaller than at Re > 40,000.

Data pertaining to the hydraulic drag in grooved annular channels are scarce. For calculating the hydraulic drag in these channels within the same ranges of h/d_e, t/d_e, and d_2/d_1 values, one may – to the first approximation – use the relations in [2] with d_e as the reference dimension and with the hydraulic drag coefficient for a smooth channel determined according to [6]. The discrepancies between test data in [3-5] within the ranges $h/d_e = 0.0275\text{-}0.158$ and $t/d_e = 0.221\text{-}0.247$ and the formulas in [2] do not exceed ±10%.

According to Fig. 3, an improvement in the heat transfer with circumferential grooves depends largely on the groove depth when $h/d_e < 0.05$, but Nu/Nu_{sm} is independent of h/d_e when $h/d_e > 0.1$. Inasmuch as the hydraulic drag increases with higher h/d_e ratios over the entire test range of h/d_e, so the optimum groove depth must lie within $h/d_e = 0.04\text{-}0.08$, i.e., where an appreciable improvement in the heat transfer is attained with an only moderate increase in the hydraulic drag.

A formation of circumferential grooves improves the heat transfer, but also results in the formation of annular diaphragms inside the pipe and this must be taken into account in optimizing the groove design. As has been noted in [7], the improvement in the heat transfer inside a pipe is best with a groove depth $d/D = 0.93\text{-}0.96$ or $h/D = 0.02\text{-}0.035$. Since the dimensionless groove depth is the same on the outside and on the inside of a pipe, hence on the outside too $h/d_1 = 0.02\text{-}0.035$ or $h/d_e = (0.02\text{-}0.035)d_1/d_e = (0.02\text{-}0.035)\cdot 1/(d_2/d_1 - 1)$.

The optimum groove depth, in terms of improving the heat transfer inside a pipe, is shown in Fig. 4 as a function of d_2/d_1. The optimum range for improving the heat transfer outside the pipe is also shown

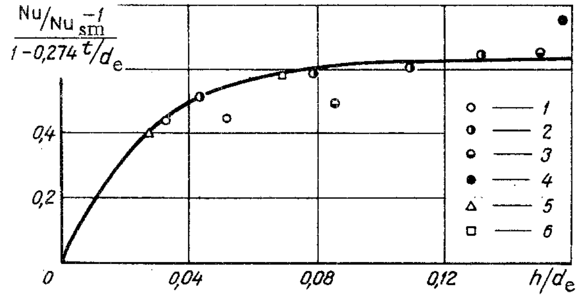

Fig. 3

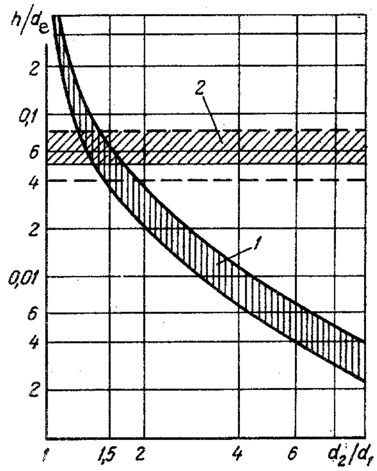

Fig. 4

Fig. 3. Improvement in the heat transfer as a function of the groove depth for annular channels with Re = 60,000: the authors' data for a channel with $d_2/d_1 = 1.455$ and $t/d_e = 0.247$ [5] (4), data for a channel with $d_2/d_1 = 1.38$ and $t/d_e = 0.275$ [3] (5), data for a channel with $d_2/d_1 = 1.287$ and $t/d_e = 0.221$ [4] (6). Solid line represents Eq. (1).

Fig. 4. Ranges of optimum groove depth, as function of the ratio of outside diameter to inside diameter d_2/d_1 in an annular channel: optimum range for improving the heat transfer inside a pipe (1), optimum range for improving the heat transfer in a grooved channel (2).

here. (The optimum groove depth $h/d_e = 0.04$-0.08 from the tests with $d_2/d_1 = 1.133$-1.455 is assumed to extend to the wider range $d_2/d_1 = 1.105$-10.0.) These ranges coincide for $d_2/d_1 = 1.25$-2.0. Namely, for these values of d_2/d_1 the given method of improving the heat transfer is most effective. The best improvement of the heat transfer in an annular channel is attained with grooves shallower than the optimum when the d_2/d_1 ratio is smaller, and with the d/D ratio larger than the optimum when the d_2/d_1 ratio is larger. The choice of the optimum depth will, of course, depend on the specific conditions as, for instance, on the ratio of heat transfer coefficients outside and inside the pipe respectively.

NOTATION

d_1, d_2	are the inside diameter and outside diameter of grooved channel;
d	is the diameter of circumferential groove;
$d_e = d_2 - d_1$	is the equivalent diameter;
h	is the groove depth;
l	is the length of test segment;
t	is the groove pitch.

Subscript

sm	denotes a smooth channel.

LITERATURE CITED

1. G. A. Dreitser and É. K. Kalinin, Inzh. Fiz. Zh., 15, No. 3 (1968).
2. É. K. Kalinin, G. A. Dreitser, and A. K. Kozlov, Inzh. Fiz. Zh., 22, No. 2 (1972).
3. C. A. Kemeny and J. A. Cyphers, Trans. ASME, 83C, No. 2 (1961).
4. H. Brauer, Atomkernenergie [German], No. 4, 152-161 (1961); No. 5, 207-211 (1961).
5. F. F. Bogdanov, A. I. Korshakov, and O. I. Utkin, At. Énerg., 22, No. 6, 428-432 (1967).
6. J. A. Brighton and J. B. Jones, Trans. ASME, 86D, No. 4 (1964).
7. E. K. Kalinin, G. A. Dreitser, S. A. Yarkho, and V. A. Kusminov, "Augmentation of convective heat and mass transfer," Papers ASME, December, 1970.