INCREASING THE RATE OF HEAT TRANSFER IN DUCTS WITH A TURBULENT GAS FLOW

V. K. Migai

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Results are shown of a study concerning the heat transfer and the hydraulic drag in an air stream through ducts of various shapes with wavy rough walls and with boundary-layer breakers.

The problem of increasing the rate of heat transfer in a gas stream through small-diameter pipes which form compact heat transfer surfaces is an essential one. The most practical way of solving this problem is by affecting the inside boundary-layer structure: by a rational design of wavy surface roughness and boundary-layer breakers, in order to turbulize the boundary layer and to reduce its thickness. This must also be technologically feasible. In orifice ducts of heat exchangers it is quite simple to design for wavy roughness. In this case the ducts comprise a sequence of convergers and diffusers, which ensures favorable ratios of heat transfer to energy losses on overcoming the hydraulic drag [1]. Such ducts have been installed, for instance, at the heating surfaces of regenerative air preheaters for boilers as early as in the nineteen twenties [2]. With small hydraulic diameters here, unlike in the ducts of [1], the converger–diffuser systems were not passing a straight stream but a slanted one, i.e., at an angle $\beta \neq 90°$ between the direction of flow and the converger–diffuser junction boundary. Tests have shown that $\beta = 30°$ is optimum. We will present here results of studies pertaining to orifice ducts with $\beta = 30°$ (Fig. 1) (for generality, also data for $\beta = 90°$ and $\beta = 15°$ are shown). In these tests the height of the roughness wave along walls a and b as well as the distance between walls c were varied. The heat transfer and hydraulic drag tests were performed on an apparatus and by a procedure well described in [2]. The mean Nusselt number Nu and the mean drag coefficient ζ were determined at relative duct dimensions corresponding to practically stabilized conditions.

The geometrical dimensions of the test ducts are given in Table 1. The effect of asperities and their height is shown in Fig. 2. Here ζ_0 and Nu_0 correspond to a smooth duct, while the absolute roughness a + b is referred to the wavelength m. In each case the test data on the heat transfer were approximated by the formula

$$\mathrm{Nu} = A\,\mathrm{Re}^{0.8}.$$

The essential test points were obtained by varying the magnitude of a + b and a slight variation of m (m = 26-30 mm). In order to illustrate the effect of parameter m, in Fig. 2 are shown data for ducts No. 15 and 16 with a + b = 2.5 mm but m = 17 and 58 mm respectively. According to Fig. 2, the dimensionless group (a + b)/m almost uniquely generalizes the heat transfer data for a varying a + b and for a varying m. In ducts No. 15 and 16 the angle β was not 30°, as in the other ducts, but 90° (roughness waves perpendicular to the stream) and 15° respectively. In this way, the proposed dimensionless parameter k generalizes data obtained with roughness waves perpendicular to the stream or slanted. Developing this parameter, we have

$$k = \frac{a+b}{m} = \frac{a}{m} + \frac{b}{m} = \frac{1}{2}(\mathrm{tg}\,\alpha_1 + \mathrm{tg}\,\alpha_2),$$

with α_1 and α_2 denoting the slope angles of diffusers and convergers respectively at each wall relative to the duct axis. Consequently, parameter k characterizes the slope angles of the diffuser and the converger

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TABLE 1. Duct Dimensions

Item No.	a, mm	b, mm	H, mm	k, mm	M, mm	d_{eq}, mm	N, mm	M, mm	l/d_{eq}	M/d_{eq}	m, mm
1	0	0	0	0	0	8,18	0	4	50	0	0
2	3,04	2,87	—	—	—	12,4	—	4	31,45	—	30
3	3,04	2,87	—	—	—	10,92	—	3	35,7	—	30
4	3,04	2,87	—	—	—	8,86	—	2	44	—	30
5	3,04	2,87	—	—	—	7,44	—	1	52,4	—	30
6	3,05	2,2	—	—	—	10,6	—	3,33	36,8	—	30
7	2,5	2,5	—	—	—	10,15	—	3,33	38,4	—	30
8	2,0	1,9	—	—	—	10,1	—	3,33	38,6	—	30
9	1,5	1,85	—	—	—	9,92	—	3,75	39,4	—	26
10	1,5	1,4	—	—	—	9,23	—	3	42,2	—	30
11	2,4	0	—	—	—	7,35	—	3	45	—	30
12	2,2	—	6	12,5	106	6,15	2	—	63,4	17,23	30
13	2,2	—	6	12,5	50	6,67	2	—	58	7,5	30
14	2,2	—	6	12,5	30	6,72	2	—	58,2	4,5	26
15	2,5	—	—	—	—	8,31	—	3,3	40	—	17
16	2,5	—	—	—	—	8,00	—	3,3	41,3	—	58

Note. M is the groove width, N, the groove depth, see along A in Fig. 1.

generatrices. A larger k (angles α_1 and α_2) implies an increase in turbulence in the diffuser segment and, thus, a higher rate of heat transfer in the following converger. Function $Nu/Nu_0 = f(k)$ is approximated by the following expression:

$$\frac{Nu}{Nu_0} = 1 - 3.2k + 108k^2 - 440k^3 - 400k^4.$$

According to Fig. 2, there is a certain limit to the effect of roughness height. Under the given conditions the heat transfer stabilizes at $a + b = 4.7$ mm ($k > 0.165$). The hydraulic drag continues to increase, however. At $a + b = 4.7$ mm, the unilateral aperture angle of an elementary diffuser formed by corrugation parallel to the stream is $\alpha \sim 9°$. A further increase of a + b causes this angle to increase too.

Angle $\alpha \sim 9°$ is the limiting preseparation angle of a diffuser. At such an angle, as is well known, there occur slight transient separations beneficial to heat transfer. The presence of localized macroseparations makes for a higher resistance, in terms of drag pressure, which in the Reynolds similarity model does not affect the heat transfer and, therefore, does not increase the heat transfer rate. In this way, the angle of elementary diffusers along a rough surface must correspond to the limiting preseparation angle.

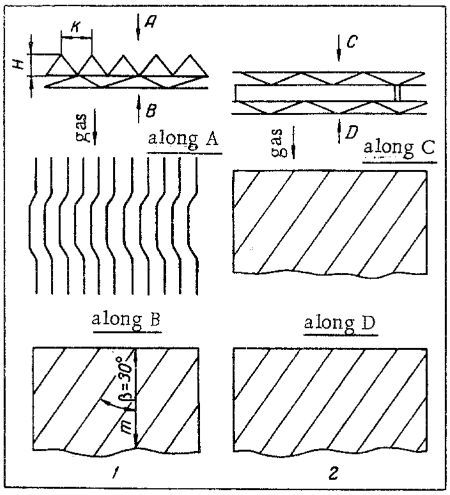

Fig. 1

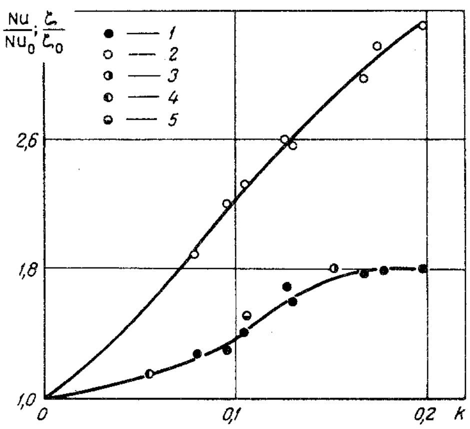

Fig. 2

Fig. 1. Sketches of the test ducts: 1) triangular duct with an offset and a wavy wall; 2) orifice duct with wavy walls.

Fig. 2. Effect of roughness height on the heat transfer and the friction in wavy ducts (Re = 3000): 1) Nu/Nu_0 (ducts No. 2-11, Table 1); 2) ζ/ζ_0 (ducts No. 2-11, Table 1); 3) Nu/Nu_0 (duct No. 15); 4) Nu/Nu_0 (duct No. 16); 5) data in [1].

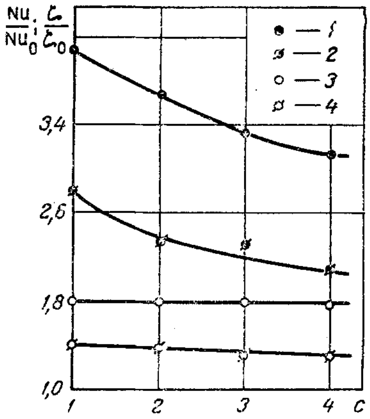

Fig. 3

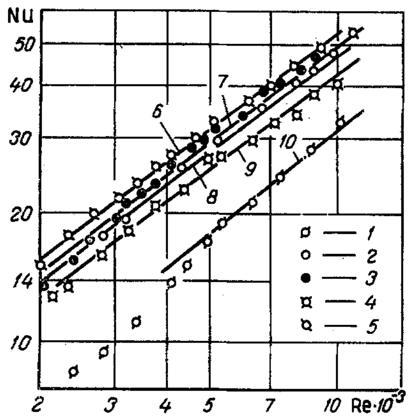

Fig. 4

Fig. 3. Effect of the height of an orifice duct with wavy walls on the heat transfer and on the hydraulic drag (Re = 3000): 1) ζ/ζ_0 with a + b = 5.9; 2) ζ/ζ_0 with a + b = 2.9; 3) Nu/Nu_0 with a + b = 5.9; 4) Nu/Nu_0 with a + b = 2.9.

Fig. 4. Heat transfer and hydraulic drag in rough ducts of composite shapes: 1) duct No. 1 (Table 1); 2) duct No. 13; 3) duct No. 14; 4) duct No. 12; 5) duct No. 2; 6) $Nu = 0.0347Re^{0.8}$; 7) $Nu = 0.0312Re^{0.8}$; 8) $Nu = 0.03Re^{0.8}$; 9) $Nu = 0.0278Re^{0.8}$; 10) $Nu = 0.02Re^{0.8}$.

The effect of the distance between walls on the heat transfer rate and on the hydraulic drag is shown in Fig. 3. It is noteworthy that a decrease of c from 4 to 1 mm has almost no effect on the heat transfer but increases the hydraulic drag. Consequently, changing the value of parameter $(a + b)/c$ (relative roughness) by changing c will not affect the heat transfer, but changing it by increasing a + b will affect the heat transfer appreciably. Therefore, the relative roughness is not a generalizing parameter as far as heat transfer processes are concerned.

The heat transfer processes at each wall are in this case autonomously determined by the stream (boundary layer) characteristics there and do not affect one another.

Within our test range, therefore, the principle of independent heat transfer applies to the boundary layer at each wall, which in the case of such narrow ducts has not been obvious.

The test results from [1] for a duct with c = 16.8 mm and m = 80 mm (equal diffuser and converger segments) have also been plotted in Fig. 2. These data agree fairly well with the universal relation. At c > 16.8 mm, however, the effect of parameter c begins to set in (the heat transfer rate increases, with all other conditions unchanged) and that universal relation ceases to be valid.

It also does not account for unequal diffuser and converger segments.

It is to be noted that ducts with c $\gtrsim$ 16 mm do not merit consideration for compact heat exchangers.

The independence principle makes it possible to analyze the heat transfer in ducts of composite sections as, for example, ducts combining rough walls with boundary-layer breakers.

As boundary-layer breakers one commonly uses individual roughness elements in the shape of transverse baffles—protrusions. A boundary layer can be broken up by very simple technical means, namely by offsetting the duct (sharp bending) (Fig. 1). In this case the boundary layer will also be restored. In air regenerators one uses triangular ducts with two walls made up of offset elements and the third wall wavy, similar to the one analyzed earlier. Such ducts are manufactured by staggering flat wavy sheets and sheets crimped triangularly with an offset. The results for three versions of such ducts with different distances between the offset elements are shown in Fig. 4. For calculating the heat transfer in such a composite duct, we will use the established principle of independent boundary layers.

We will make the following assumptions: 1) the boundary layer at each wall adds its independent contribution to the heat transfer processes, and 2) the boundary layers are fully restored behind an offset element (δ_0 = 0, δ denoting the boundary-layer thickness). The heat transfer in the initial smooth segments was studied both theoretically and experimentally.

The correction for the effect of relative duct length (tests performed by I. T. Alad'ev in [3]) will be approximated by the following power function:

$$Nu = Nu_0 \frac{1.69}{\left(\frac{x}{d}\right)^{0.143}} = 0.023\, Re^{0.8}\, Pr^{0.4} \frac{1.69}{\left(\frac{x}{d}\right)^{0.143}}. \tag{1}$$

This formula is valid for $Re \sim 5{,}000$-$50{,}000$ with Nu_0 representing a stabilized heat transfer in a circular pipe. Using the equivalent diameter as the characteristic dimension will generalize these data to cover ducts of noncircular sections with turbulent flow.

Using the rule of prorating the total heat transfer to each duct side, we have

$$Nu_{total} = Nu_{w.d} \frac{1}{3} + Nu_{s.d} \frac{2}{3}. \tag{2}$$

Here $Nu_{w.d.}$ is the Nusselt number for a wavy duct and $Nu_{s.d.}$ is the Nusselt number for a smooth duct.

For duct No. 12 ($Pr = 0.7$) we have

$$0.0278\, Re^{0.8} = Nu_{w.d} \frac{1}{3} + \frac{2}{3}\, 0.02\, Re^{0.8} \cdot 1.1, \tag{3}$$

wherefrom $Nu_{w.d.} = 0.0393 Re^{0.8}$ or 12% higher than the Nusselt number for a flat wavy duct (duct No. 2, $Nu = 0.0347 Re^{0.8}$). Some increase in the Nusselt number can be attributed to eddies in the duct bends.

For ducts No. 14 and 13 we have from (3)

$$Nu = \frac{1}{3} \cdot 0.0393\, Re^{0.8} + \frac{2}{3} \cdot 0.0272\, Re^{0.8} = 0.0312\, Re^{0.8}, \tag{4}$$

$$Nu = \frac{1}{3} \cdot 0.0393\, Re^{0.8} + \frac{2}{3} \cdot 0.0253\, Re^{0.8} = 0.03\, Re^{0.8}. \tag{5}$$

A comparison with the test data in Fig. 4 indicates a discrepancy of only 7%. With the eddy effects in the duct bends disregarded and with the heat transfer at the wavy wall determined from the test data for a flat duct, the error here will not exceed 12%. In this way, by the principle of independent boundary layers it is possible to calculate, within an acceptable accuracy, the heat transfer in composite intensifying ducts.

Coefficient r in the Reynolds similarity relation $St = \xi/r$ has a very high value for the ducts studied here ($r = 13$-14), which is indicative of a similarity unbalanced toward momentum transfer. More favorable values $r < 8$ (lower than for a smooth duct) were obtained for several ducts in [1]. This low value of r was obtained, however, for larger ducts (the minimum c in [1] was 16.8 mm, i.e., four times larger than in our ducts). Considering that an increase in c leads to a sharp decrease in hydraulic drag (Fig. 3) and only a slight change in heat transfer, we must conclude that also for our ducts the value of r at large values of c is rather low.

We note, at the same time, that under consideration for modern compact heat exchangers are ducts with small hydraulic diameters ($d_{equ} \sim 5$-10 mm). Of considerable practical interest is the optimum converger–diffuser ratio revealed in [1]. Inasmuch as the pressure loss and the heat transfer are not proportional to the velocity, the governing criterion in evaluating a surface is not parameter r in the Reynolds similarity relation but the energy characteristic instead. Calculations show that there is 40-50% more heat transferred in this case than in smooth ducts with the same power loss per surface area to overcome hydraulic drag.

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