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Enhancement of convective heat transfer in rectangular ducts of interrupted surfaces

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Abstract—Results are presented from systematic experimental investigations into the enhancement of heat transfer of 11 interrupted surfaces of one type and of different size formed by a multitude of short rectangular ducts with a constant cross-sectional aspect ratio of 7 : 1 in the absence of burrs and bends on the sharp edges of fins. Investigations were carried out with air in the range of Reynolds numbers from 500 to 10 000 for geometric parameters $l'/d = 0.65$–3.24 and $\delta/d = 0.058$–0.114. Based on these investigations, standard experimental thermohydraulic characteristics of these surfaces are obtained as well as their correlations as functions of *Re* and geometric parameters l'/d and δ/d. New results are obtained on the effect of the geometric parameters l'/d and δ/d of interrupted surfaces on the enhancement of heat transfer in them and also on the experimental determination of the coefficients of overall pressure drop at the entrance and exit from the interrupted surface in a heat exchanger.

IN RECENT years wide usage in engineering practice of the production of the compact plate–fin heat exchanger has been gained by plate–fin interrupted (with short fins) surfaces formed by rectangular ducts (Fig. 1). The analysis of experimental results for such surfaces in actual plate–fin heat exchangers [1–3] has shown that their determining dimensionless geometric parameters were selected only from considerations of providing maximum heat transfer in heat exchangers and technological possibilities for their fabrication. Therefore, in these publications the governing geometric parameters of interrupted surfaces were characterized by the values $l'/d = 1.0$–2.0 for arbitrarily chosen parameters $h/t = 1$–8 and corresponding values of $\delta/d = 0.03$–0.1. The results of these works allow only qualitative conclusions regarding the effect of variation of the parameters l'/d, δ/d and h/t on thermohydraulic characteristics of interrupted surfaces. Moreover, it is important to observe that these results were obtained for interrupted surfaces fabricated by cutting original sheet material. As a result, the quality of sharp edges of fins is unstable and depends both on the type of the cutting tool and time of its operation, and on the mechanical strength and type of the original sheet material (copper, aluminium, stainless steel).

The aforementioned does not make for reliable reproducibility of the thermohydraulic characteristics of traditionally fabricated interrupted surfaces even with the use of the same cutting tools and original sheet materials. Therefore, those who fabricate heat exchangers with interrupted surfaces prefer to rely on

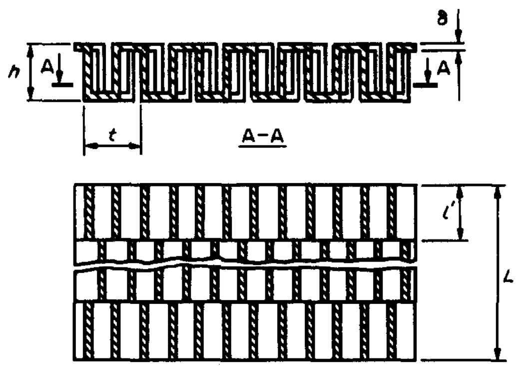

FIG. 1. An interrupted surface formed by rectangular ducts.

their own thermohydraulic relations which are generally not reported. In ref. [4] correlational thermohydraulic relations are presented resulting from the analysis of published experimental data on 22 standard-size actual constructions of interrupted surfaces formed by different rectangular ducts and fabricated traditionally by cutting sheet materials. However, the inadequate accuracy of these relations does not ensure reliable design of heat exchangers.

Recently, attempts have been made [5, 6] to obtain experimental thermohydraulic characteristics of interrupted surfaces, formed by rectangular ducts, by employing the analogy between mass and heat transfer processes. Enlarged composite models of interrupted surfaces were used in which one row of wide side walls [5] or one wide side wall [6] of numerous

NOMENCLATURE

d reduced hydraulic diameter of ducts in surfaces
f_{un} coefficient of air-side unimpeded area of surface in heat exchanger
K_c, K_e irreversible pressure losses at the entrance and exit from the surface in the heat exchanger according to refs. [1, 4]
l' continuous length of a short duct in an interrupted surface
l'/d relative length of short duct
L/d relative air-side depth of the surface
T_w, T_a mean wall temperature of fins and mean-arithmetic air temperature in the surface.

Greek symbols
δ fin thickness
δ/d relative thickness of the duct fin
ξ coefficient of total pressure losses in the surface involving pressure losses in ducts, at the entrance and exit from the surface in the heat exchanger, $(\xi_{en}+\xi_{ex})d/L+\zeta$
λ_T turbulent thermal conductivity of air flow
ε_T turbulent viscosity of air flow
ζ coefficient of pressure losses in surface ducts.

Subscripts
in, sm interrupted and smooth surfaces, respectively.

short rectangular ducts was coated with naphthalene. The results obtained for hydraulic resistance do not raise doubts, while the accuracy of heat transfer results can be called in question, because the greater portion of the heat transfer surface of short rectangular ducts was not covered with naphthalene. Therefore, the concentration gradient of naphthalene vapours in an air flow in the ducts studied was higher than that for the duct fully covered with naphthalene. As a result, the data on the heat transfer of surfaces were found to be distorted with respect to their true values.

In ref. [7], based on the experimental investigation of standard-size enlarged models of interrupted surfaces formed by different rectangular ducts, correlations are derived for the friction factor, which are more accurate than those suggested in ref. [4]. Nevertheless, their accuracy does not meet the demands of the design calculations of heat exchangers.

It should be noted that the enlarged composite models of interrupted surfaces used in refs. [5–7] differ substantially from actual constructions by the cross-section profile of a multitude of short rectangular ducts. Thus, in actual constructions of the surfaces the apexes of two corners of each short rectangular duct cross-section are rounded off to a certain radius, while the other two corners are rounded by the solder meniscus. One of the rectangular duct side walls is covered with solder and has a roughness different from that on the other walls. In enlarged composite models of interrupted surfaces, the cross-section profile of short rectangular ducts has acute corners and the walls of these ducts have identical roughness. Since actual constructions of interrupted surfaces are characterized by small hydraulic diameters of their ducts, $d = 0.5$–3.5 mm, the above-mentioned differences in geometries will significantly influence their thermohydraulic characteristics, with all other geometric and operational parameters being identical.

The present work was undertaken in order to investigate experimentally the effect of change in determining geometric parameters l'/d and δ/d of interrupted surfaces, formed by rectangular ducts with the aspect ratio 7 : 1 (the case of a plane duct with the smallest effect of laminarized angular zones of coolant flow), on their thermohydraulic characteristics and on the thermohydraulic efficiency of the process of heat transfer augmentation realized in interrupted surfaces in the absence of burrs and bends on sharp edges of fins. The other objective of this investigation was to determine experimentally the standard generalizing thermohydraulic characteristics of interrupted surfaces formed by rectangular ducts with an aspect ratio of 7 : 1.

Moreover, the aim of the study was to determine experimentally the overall coefficient of entrance and exit pressure losses of a coolant flow in interrupted surfaces in actual plate–fin heat exchangers, to verify experimentally the common method of their determination suggested in refs. [1, 9] and also to estimate their effect on thermohydraulic characteristics of interrupted surfaces.

The constructions of heat exchangers with interrupted surfaces, which were investigated in the present work, are shown in Fig. 2. The geometric parameters of the interrupted surfaces are set out in Table 1.

All the surfaces were made of a 0.6 mm thick AMgZ/5 aluminium alloy strip. The absence of burrs and bends on sharp edges of the fins of short ducts was controlled with the aid of a BP50–150-type 20-fold magnifying microscope. The hydraulic diameter of the surface ducts was determined as the arithmetic mean value of the hydraulic diameters of ducts 1 and 2 (Fig. 3).

The experimental heat exchangers consisted of two separate plane multi-channel tubes one side of which was finned with an interrupted surface and in the interior of which hot water passed. Plane tubes were

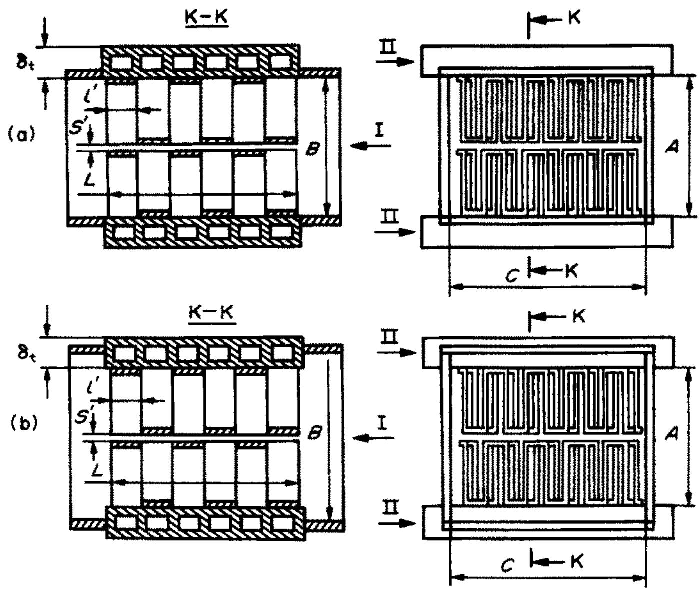

FIG. 2. Construction of a test heat exchanger and means of its installation in the test section of an experimental setup.

Table 1. Geometrical characteristics of rectangular ducts in interrupted surfaces

Geometric parameters	Number of the surface											
	1_{in}	2_{in}	3_{in}	4_{in}	5_{in}	6_{in}	7_{in}	8_{in}	8^*_{in}	9_{in}	10_{in}	11_{sm}
h [mm]	0.1	0.1	0.1	0.1	0.1	0.1	21.6	26.6	26.6	36.6	41.6	0.1
t [mm]	5.0	5.0	5.0	5.0	5.0	5.0	3.6	4.35	4.35	5.8	6.5	5.0
d [mm]	7.72	7.72	7.72	7.72	7.72	7.72	5.27	6.58	6.58	9.12	10.35	7.72
S [mm]	1.8	1.8	1.8	1.8	1.8	1.8	1.2	1.5	1.5	2.1	2.4	1.8
l'/d	0.65	0.97	1.30	1.94	2.77	3.24	1.31	1.29	1.29	1.29	1.30	—
δ/d	0.0777	0.0777	0.0777	0.0777	0.0777	0.0777	0.1138	0.0912	0.0912	0.066	0.058	0.078
$(h-\delta)/(t-\delta)$	6.93	6.93	6.93	6.93	6.93	6.93	7.0	6.93	6.93	6.92	6.95	6.93
L/d	19.43	19.43	19.43	19.43	19.43	19.43	28.8	21.96	21.96	15.53	14.3	19.43
f_{un}	0.87	0.87	0.87	0.87	0.87	0.87	0.82	0.85	0.62	0.89	0.9	0.87
h'/t'	6.93	6.93	6.93	6.93	6.93	6.93	7.0	6.93	6.93	6.92	6.95	6.93
Ω [$m^2\ m^{-3}$]	452	452	452	452	452	452	624	518	377	390	348	452

Note: All the ducts of the interrupted surfaces are made of aluminium alloy AMg tape 0.6 mm thick.

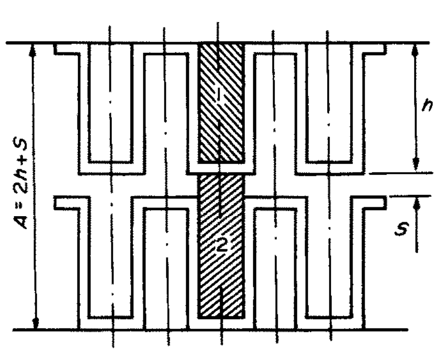

FIG. 3. Scheme for determining the reduced hydraulic diameter of ducts in interrupted surfaces.

made of 20 mm-thick AD-31 aluminium alloy. The interrupted surface was made of separate corrugated strips mounted tightly in succession one after the other and offset by $t/2$. All the surfaces, except for No. 8^*_{in}, were investigated in a heat exchanger which had been mounted in a rectangular test section of an air passage of an experimental setup (Fig. 2(a)). The distance A between the opposite surfaces of plane tubes was equal to the distance B between the opposite walls of the test section, thus allowing experimental determination of air pressure losses in the ducts of the interrupted surfaces.

To determine experimentally the overall pressure losses for air entrance and exit from interrupted surfaces in actual heat exchangers, plane tubes finned with surface No. 8_{in} were positioned in the test section of the air passage as shown in Fig. 2(b). The distance A was smaller than the distance B by the value equal to the plane tube thickness ($A-B = \delta_t = 20$ mm).

Experiments for determining the thermohydraulic characteristics of surfaces were conducted on a water–air setup with a water-heated loop (Fig. 4). The setup was built as recommended in refs. [10, 11]. The pro-

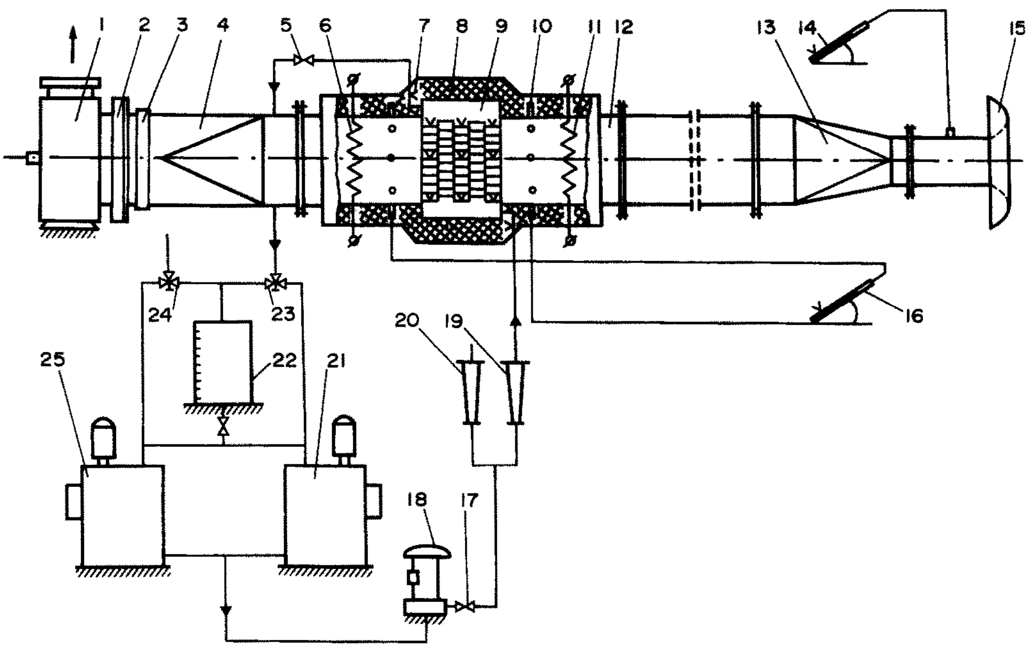

FIG. 4. Schematic diagram of the experimental setup: 1, fan; 2, petal orifice; 3, 4, 13, adapters with the diffusor expansion angle of 10°; 5, 17, water taps; 6, resistance thermometer of mean-integral temperature; 7, thermocouples; 8, thermal insulation; 9, test heat exchanger; 10, static pressure sampling collector; 12, test section; 14, 16, micromanometers; 15, lemniscate flow rate meter; 18, water pump; 19, 20, rotameters; 21, 25, water thermostats; 22, measuring tank; 23, 24, three-pass taps.

cedural distinction consisted in determining the mean-arithmetic temperature of the carrying surface of each plane tube in a heat exchanger (bases of fins) with the aid of nine evenly distributed chromel–coppel micro-thermocouples which virtually did not distort the hydrodynamic structure of the process in the ducts of interrupted surfaces. As a result, the air-side heat transfer coefficients and Nu of the surfaces investigated were determined without using the predicted water-side heat transfer coefficients of the heat exchanger. This excluded additional procedural errors in the measured Nu of the surfaces.

The coefficient of friction pressure losses in the surfaces is given in this paper only for non-isothermal conditions characterized by the values $T_w/T_{air} = 1.03$–1.06.

In the present investigation the maximum error in taking a heat balance on the water- and air-sides of the heat exchanger did not exceed $(Q_{wat}-Q_{air}):Q_{wat} \times 100\% \leqslant 5\%$. The maximum relative errors in measuring the thermohydraulic characteristics of the surfaces in the confidence interval 0.997 amounted to: $\delta Nu = \pm 9.5\%$; $\delta_\zeta = \pm 4.9\%$; $\delta Re = \pm 3.6\%$.

The approximated experimental results for the thermohydraulic characteristics of the surfaces are presented in Tables 2 and 3.

The results for the effect of the parameter l'/d on the thermohydraulic characteristics of interrupted surfaces Nos. 1_{in}–6_{in} and for smooth surface 11_{sm} are presented in Figs. 5 and 6. The analysis of these results shows that with a decrease of the parameter l'/d in the range $l'/d = 0.65$–19.43 at $Re = idem$ and $\delta/d = 0.0777$ the heat transfer and coefficient ζ increase over the entire range of Re studied. When $l'/d < 1.3$–1.5, the increase of ζ outstrips the increase in heat transfer. This is due to the fact that the wall

Table 2. Results of the approximation of experimental heat transfer characteristics of surfaces

Number of the surface	$Nu = A\,Re^n$		
	Re	A	n
1_{in}	550–8010	0.57	0.48
2_{in}	550–1680	0.5	0.49
	1680–8080	0.44	0.5
3_{in}	550–2930	0.46	0.49
	2930–8310	0.36	0.52
4_{in}	550–3160	0.46	0.48
	3160–8420	0.26	0.55
5_{in}	550–2730	0.44	0.47
	2730–7790	0.19	0.58
6_{in}	550–3550	0.43	0.47
	3550–9710	0.12	0.62
7_{in}	550–5910	0.14	0.64
8_{in}	550–10020	0.31	0.54
8^*_{in}	550–9730	0.28	0.55
9_{in}	550–2810	0.59	0.47
	2810–9400	0.51	0.49
10_{in}	550–2180	0.86	0.42
	2180–9620	0.38	0.52
11_{sm}	550–2340	0.8	0.3
	2340–4380	0.16	0.51
	4380–10050	0.016	0.78

Table 3. Results of the approximation of experimental hydraulic characteristics of heat transfer surfaces

Number of the surface	Re	$\zeta = c\,Re^m$ c	m
1_{in}	550–1260	40.92	−0.68
	1260–2750	5.98	−0.41
	2750–4500	0.87	−0.17
	4500–8010	0.26	−0.029
2_{in}	550–1270	42.22	−0.72
	1270–2640	4.19	−0.40
	2640–4860	0.78	−0.18
	4860–8080	0.23	−0.037
3_{in}	550–1100	69.75	−0.82
	1100–2410	10.82	−0.55
	2410–4720	0.73	−0.20
	4720–8310	0.18	−0.042
4_{in}	550–1130	47.0	−0.79
	1130–2200	12.35	−0.60
	2200–4150	1.2	−0.29
	4150–8420	0.21	−0.084
5_{in}	550–840	46.24	−0.81
	840–2120	12.18	−0.61
	2120–3770	1.87	−0.36
	3770–7790	0.32	−0.15
6_{in}	550–1640	26.79	−0.73
	1640–2920	7.95	−0.56
	2920–5290	0.77	−0.27
	5290–9710	0.24	−0.14
7_{in}	550–990	36.14	−0.72
	990–2070	4.39	−0.41
	2070–2960	0.54	−0.14
	2960–5910	0.35	−0.084
8_{in}	550–1470	26.49	−0.68
	1470–2900	14.74	−0.28
	2900–5070	0.37	−0.11
	5070–10 020	0.18	−0.026
8^*_{in}	550–1440	31.21	−0.70
	1440–3080	13.41	−0.26
	3080–5400	0.37	−0.10
	5400–9730	0.14	−0.008
9_{in}	550–1350	54.87	−0.78
	1350–2970	6.42	−0.49
	2970–5270	0.92	−0.24
	5270–9400	0.17	−0.049
10_{in}	550–1350	71.06	−0.82
	1350–2690	7.59	−0.51
	2690–5140	1.60	−0.31
	5140–9620	0.19	−0.062
11_{sm}	550–2555	34.4	−0.82
	2555–3785	5.66	−0.59
	3785–5600	0.722	−0.34
	5600–10 050	0.0746	−0.077

Note: Here, for surface No. 8^*_{in} the values of the coefficient are presented instead of the results of the approximation of coefficient ζ.

vortices, generated on sharp edges of the duct fins, gradually decay with an increasing l'. This, in turn, causes a corresponding decrease in the magnitudes of flow turbulence characteristics λ_T and ε_T in the wall layer leading to a corresponding decrease in heat transfer augmentation.

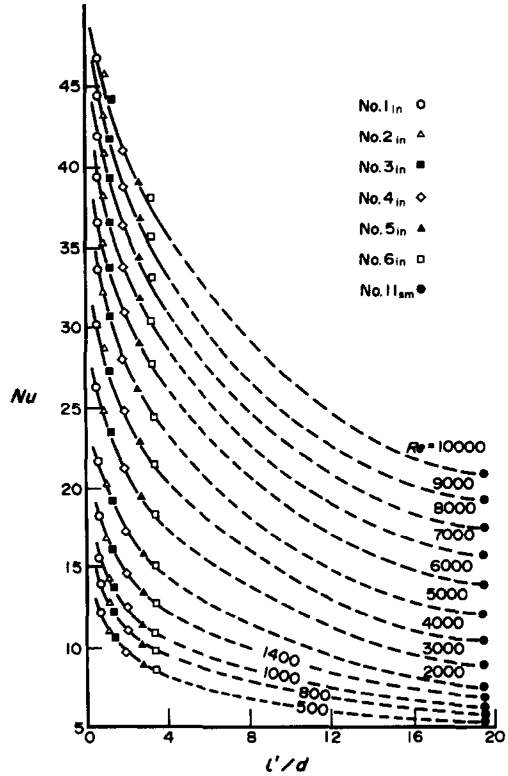

FIG. 5. The function $Nu = f(l'/d)$ at $Re = idem$ and $\delta/d = 0.0777$ for surfaces Nos. 1_{in}–6_{in}, 11_{sm}.

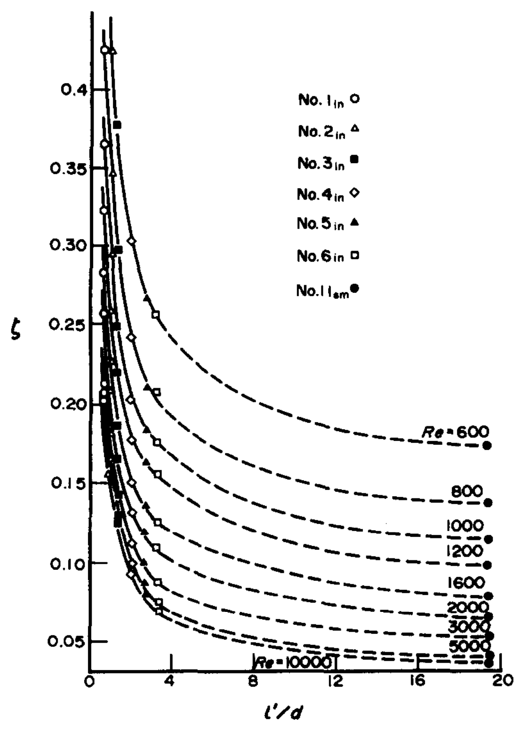

FIG. 6. The function $\zeta = f(l'/d)$ at $Re = idem$ and $\delta/d = 0.0777$ for surfaces Nos. 1_{in}–6_{in}, 11_{sm}.

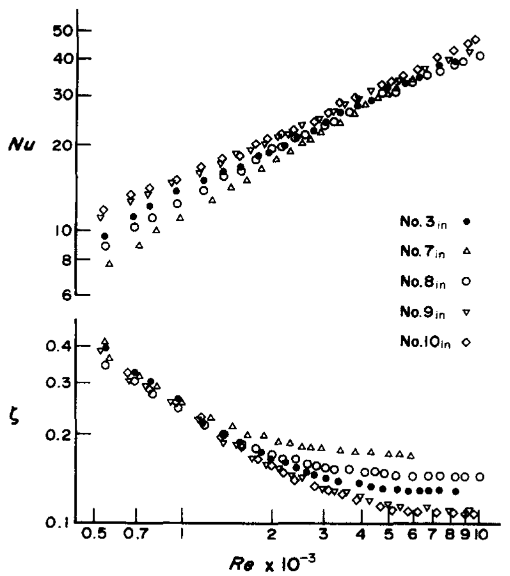

FIG. 7. The functions $Nu = f(Re)$ and $\zeta = f(Re)$ for surfaces Nos. 3_{in}, 7_{in}–10_{in}.

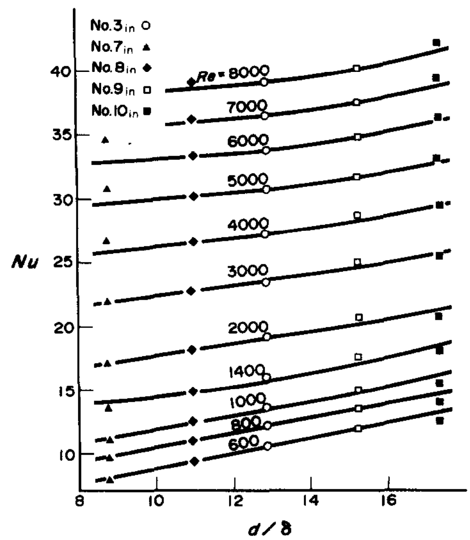

FIG. 8. The function $Nu = f(\delta/d)$ at $Re = idem$ and $l'/d = 1.3$ for surfaces Nos. 3_{in}, 7_{in}–10_{in}.

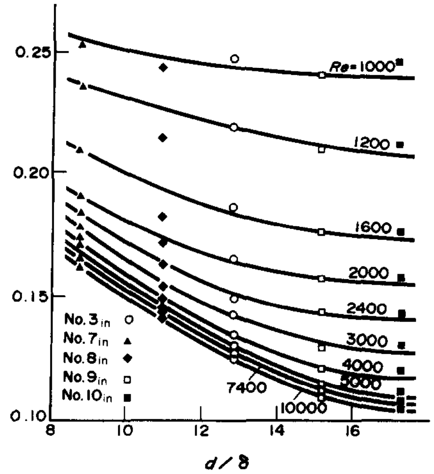

FIG. 9. The function $\zeta = f(Re)$ at $Re = idem$ and $l'/d = 1.3$ for surfaces Nos. 3_{in}, 7_{in}–10_{in}.

These results agree well with similar data of systematic investigations of interrupted surfaces formed by the ducts having the cross-section of an equilateral triangle and presented in refs. [12, 13].

The results of investigations into the effect of a change in the fin thickness on the thermohydraulic characteristics of interrupted surfaces were obtained for surfaces Nos. 3_{in}, 7_{in}, 8_{in}, 9_{in}, 10_{in}. These are presented in Figs. 7–9. The analysis of these results shows that, with a decrease in the fin thickness, the hydraulic resistance decreases, whereas the heat transfer increases in the entire range of Re studied. In this case, the effect of a change in the fin thickness on heat transfer is most vividly manifested in the region $Re < 2000$ and on hydraulic resistance, when $Re > 2000$. This is due to the fact that on the edges of thinner fins smaller vortices are formed which are located closer to the duct wall and which turbulize a less thick flow region. In this case, in the transition flow region, the heat transfer is especially sensitive to the size, location and intensity of eddies that provide an earlier flow turbulization near the wall. The hydraulic resistance is associated with overall losses due to an earlier (with respect to Re) flow turbulization. In the turbulent region ($Re > 2000$) larger vortices, generated on a thicker fin, noticeably increase ζ but do not influence the increase of Nu so significantly, since the main production of turbulence on their upper boundary occurs further away from the wall. Moreover, ζ increases at the expense of losses due to rear eddies behind a thicker fin that turbulizes the flow core in a subsequent short duct and exerts a weak influence on heat transfer augmentation. As a result, with a decreasing fin thickness in the range $\delta/d = 0.058$–0.0777 in the considered standard-size constructions of interrupted surfaces the process of heat transfer augmentation is realized most effectively at moderate hydraulic resistances which are mainly governed by the additional energy supply to the flow of the heat carrier in a narrow wall layer, not influencing substantially the flow core.

With an increasing parameter $\delta/d > 0.0777$, the outer edge of the separation zone recedes from the duct wall, thus increasing the thickness of the recirculation zone in the separation region. As a result, in the immediate vicinity of the wall the values of the turbulent flow parameters λ_T and ε_T decrease in the recirculating separation zone, thus leading to a reduction in the heat transfer enhancement. Moreover, in the region of the reattachment and further development of the boundary layer the values of the turbulent flow parameters λ_T and ε_T are smaller as compared with the case of a thin fin which is characterized by the values $\delta/d \leqslant 0.0777$. This is attributed to a comparative decrease in the kinetic energy of a reattaching boundary layer and in the values of turbulent velocity fluctuations in it in the region of reattachment and, correspondingly, in the region of the subsequent boundary layer development. In this case, generated vortices increase in scale and their propagation along the flow occurs in both the wall layer and flow core regions. The flow core region, occupied by propagating vortices, increases in this case with $\delta/d > 0.0777$. As was shown earlier [12–15], an additional energy, supplied to the flow for the formation and propagation of vortices beyond the limits of the wall layer, does not virtually contribute to heat transfer enhancement and only leads to higher hydraulic resistance.

The results obtained in the present work on the effect of a change in the parameter δ/d on the thermo-hydraulic characteristics of the surfaces investigated agree well with similar results of systematic studies of interrupted surfaces formed by ducts having the cross-section of an equilateral triangle [12, 13] and also with the qualitative conclusions for interrupted surfaces formed by rectangular ducts [2].

The results of investigations into the effect of sudden contraction and expansion of air flow in the case of a rectangular cross-sectional profile of the entrance and exit from an actual plate–fin heat exchanger on thermohydraulic characteristics of interrupted surfaces were obtained when testing surfaces Nos. 8_{in} and 8_{in}^* and are presented in Fig. 10. The analysis of these results showed that for $f_{un} \geqslant 0.62$ the values of the thermal characteristics of surfaces Nos. 8_{in} and 8_{in}^* are practically the same over the entire range of Re studied (Re = 550–10 000). The coefficient of the overall pressure losses for surface No. 8_{in}^*, when $f_{un} \geqslant 0.62$ and $Re \leqslant 750$, is equal to the coefficient of pressure losses in the ducts of surface No. 8_{in}. In the range of increasing Re, Re = 750–10 000, the coefficient of overall pressure losses in surface No. 8_{in}^* gradually increases as against the coefficient of pressure losses in the ducts of surface No. 8_{in}. At Re = 10 000 the difference in the magnitudes of these coefficients is insignificant and amounts to 5%.

Moreover, the tests carried out with surfaces Nos. 8_{in} and 8_{in}^* made it possible to determine experimentally the coefficient of the overall pressure losses for an air flow at the entrance and exit from ducts of the interrupted surface in an actual plate–fin heat exchanger. Within the knowledge of the present authors, no similar experimental works have been reported. Therefore, it was of practical interest to verify experimentally the accuracy of Kays' method [1, 9] widely applied for determining the coefficients of air flow pressure losses at the entrance and exit from ducts of interrupted surfaces in actual plate–fin heat exchangers.

According to Kays' method, the coefficient of pressure losses at the entrance is defined as

$$\xi_{en} = (K_c + 1 - f_{un}^2) \tag{1}$$

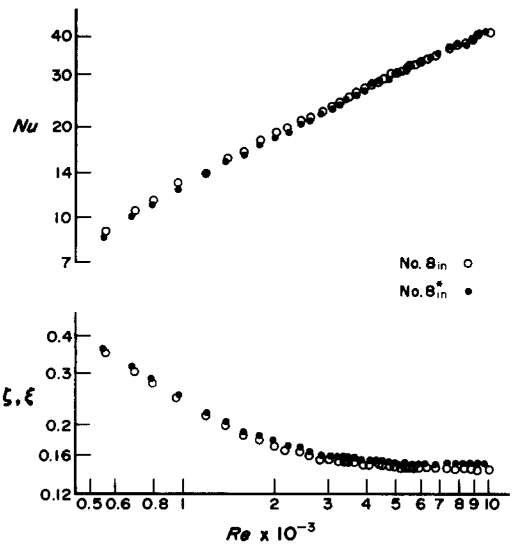

FIG. 10. The functions $Nu = f(Re)$, $\zeta = f(Re)$ and $\xi = f(Re)$ for surfaces Nos. 8_{in} and 8_{in}^* under non-isothermal conditions.

and at the exit it is defined as

$$\xi_{ex} = (1 - f_{un}^2 - K_e) \tag{2}$$

where K_c and K_e are determined from the nomogram in Fig. 5-4 of ref. [1].

Using the measured values of $\zeta = f(Re)$ for surface No. 8_{in} it is possible, with the aid of equations (1) and (2), to calculate the values of $\xi = f(Re)$ for $f_{un} = 0.62$ from the expression

$$\xi = \zeta + (\xi_{en} + \xi_{ex})\frac{d}{L}. \tag{3}$$

By comparing the values of $\xi = f(Re)$ obtained from equation (3) with experimentally measured values of $\xi = f(Re)$ for surface No. 8_{in}^*, it is possible to assess the accuracy of Kays' method. The results of this comparison are presented in Fig. 11. The analysis of these results shows that the values obtained by Kays' method for the coefficients of overall pressure losses at the entrance and exit from an interrupted surface in an actual heat exchanger exceed the measured values on average by a factor of 2.65. As a result, the values of $\xi = f(Re)$ predicted by Kays' method are overestimated by 8.5% by contrast with the corresponding experimental values for surface No. 8_{in}^*. When coefficients of pressure losses for entrance and exit are calculated from relations recommended by Idelchik [17]

$$\xi_{en} = 0.5(1 - f_{un}) \tag{4}$$

$$\xi_{ex} = (1 - f_{un})^2 \tag{5}$$

and are then substituted into equation (3), the predicted and experimental results agree satisfactorily. In this case the predicted values of the coefficient of overall pressure losses for the entrance and exit from an interrupted surface in an actual heat exchanger exceed the corresponding experimental values on average by

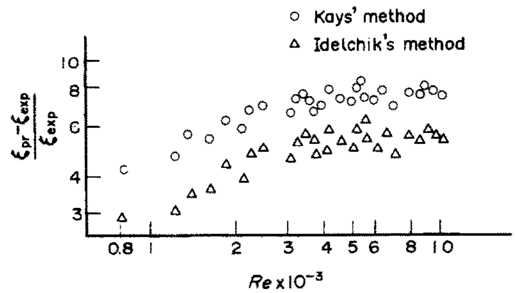

FIG. 11. Comparison between the predicted and experimental values of ζ for surface No. 8^*_{in} under the conditions of isothermal tests.

a factor of 2.1, whereas the predicted values of the coefficient of overall pressure losses exceed the corresponding experimental values for surface No. 8^*_{in} by about 5–6% (Fig. 11).

The above analysis shows that, within the accuracy of the experiment, the prediction methods of Kays and Idelchik, used for determining the coefficient of overall pressure losses for entrance and exit of a coolant from non-circular surface ducts in actual plate–fin heat exchangers, are actually equivalent. However, the preference should be given to the method of Idelchik, since, in contrast to the grapho-analytical method of Kays, the method of Idelchik is analytical and more convenient for heat exchanger design purposes.

The results of the assessment of the thermo-hydraulic efficiency of heat transfer augmentation by interrupted surfaces according to the method of ref. [16] in relation to surfaces Nos. 1_{in}–6_{in} are presented in Fig. 12. The analysis of these results shows that in the range of Reynolds numbers from 500 to 6000 the heat transfer rate in surfaces Nos. 3_{in}–6_{in} increases more rapidly than the hydraulic losses. In this case, when the parameter l'/d increases over the range $l'/d = 1.3$–3.24, the *Re*-based upper limit, corresponding to $(Nu/Nu_{sm})/(\zeta/\zeta_{sm}) \geqslant 1$, increases respectively within the range $Re = 2400$–6000. The greatest heat transfer augmentation, $Nu/Nu_{sm} = 2.6$, at $(Nu/Nu_{sm})/(\zeta/\zeta_{sm}) = 1$ was obtained for surface No. 3_{in}. In surfaces Nos. 1_{in} and 2_{in}, characterized by the values of the parameter $l'/d = 0.65$ and 0.97, the hydraulic resistance increases faster than heat transfer over the entire range of *Re* studied, $Re = 500$–$10\,000$.

Figure 13 presents the estimates of the thermo-hydraulic efficiency of the heat transfer augmentation process for different constructions of heat exchanging surfaces from the data of refs. [12, 13, 18] and of the present study. There, in contrast to surfaces I and II, calculations for surface III were made with the use of the function $\zeta = f(Re)$ for interrupted and smooth surfaces. With regard for the above analysis of the effect of sudden contraction and expansion at entrance and exit from interrupted surfaces in actual heat exchangers, it is possible to conclude that the corresponding curve $\xi/\xi_{sm} = f(l'/d)$ for surface III will lie somewhat below that shown in Fig. 13. The

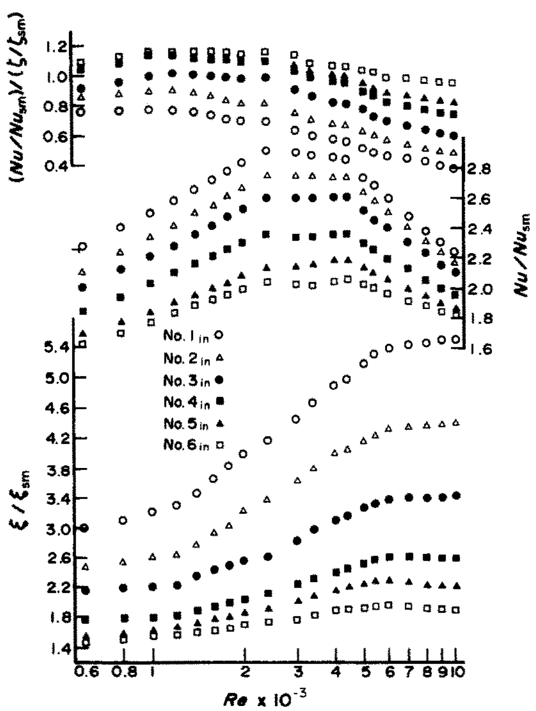

FIG. 12. Relations $\zeta/\zeta_{sm} = f(Re)$, $Nu/Nu_{sm} = f(Re)$ and $(Nu/Nu_{sm})/(\zeta/\zeta_{sm}) = f(Re)$ for surfaces Nos. 1_{in}–6_{in}, 11_{sm} (Nu, ζ, for surfaces Nos. 1_{in}–6_{in}; Nu_{sm}, ζ_{sm}, for surface No. 11_{sm}).

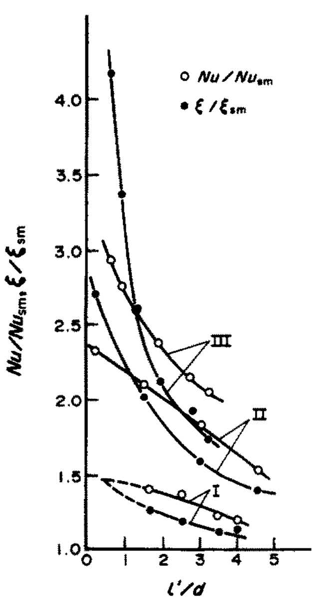

FIG. 13. Relations $Nu/Nu_{sm} = f(l'/d)$ and $\xi/\xi_{sm} = f(l'/d)$ at $Re = idem$ for different plate–fin surfaces (Nu, ξ, for surfaces with vortex generators). (I) Interrupted surfaces formed by triangular ducts [12, 13]: $h/t = 1$; $\delta/d = 0.0287$; $L/d = 20.1$; $Re = 1250$. (II) Surfaces formed by triangular ducts with three-dimensional transversal projections and grooves [18]: $h/t = 1.52$; $d^*/d = 0.797$ (d^*, d, hydraulic diameter in the duct narrowest cross-section and on its smooth-surface portion,, respectively); $L/d = 23.6$; $Re = 1700$. (III) Interrupted surfaces Nos. 1_{in}–6_{in} formed by rectangular channels (present investigation): $h/t = 7$; $\delta/d = 0.0777$; $L/d = 19.4$; $Re = 2400$.

analysis of these results allows the following conclusions to be drawn.

(1) The degree of heat transfer enhancement in ducts of different cross-sections with different types of vortex generators in the heat carrier wall layer depends, in the case of compliance with the condition $Nu/Nu_{sm} \geqslant \xi/\xi_{sm}$, only on the determining type of the cross-section of surface ducts and does not depend on the design of such vortex generators.

(2) When $Nu/Nu_{sm} \geqslant \xi/\xi_{sm}$, the heat transfer rate and corresponding Re increase with an increasing cross-sectional aspect ratio of ducts which is characterized by the parameter h/t. With all other identical conditions, these values of Re in smooth ducts, being compared, decrease with an increase in their length L due to the assembly of turbulent slugs under the conditions of a weakly developed turbulent flow structure.

For these conditions the heat transfer enhancement is characterized by the values of $Nu/Nu_{sm} \leqslant 2.88$ found experimentally (the value $Nu/Nu_{sm} = 2.88$ is obtained for round tubes with smoothly rounded projections and grooves [15]).

(3) When $Nu/Nu_{sm} = \xi/\xi_{sm}$, the maximum heat transfer rates and corresponding Re can be obtained in ducts of different cross-sections. For ducts with the cross-section in the form of an equilateral triangle it was found that $Nu/Nu_{sm} \approx 1.5$ and $Re = 1250$ at $L/d = 20.1$. For ducts with the cross-section in the form of an isosceles triangle at $h/t = 1.5$, $Nu/Nu_{sm} = 2.15$ and $Re = 1700$ for $L/d = 23.6$. For plane rectangular channels with $h/t = 7$ it was found that $Nu/Nu_{sm} = 2.6$ and $Re = 2400$ at $L/d = 19.4$. For round channels $Nu/Nu_{sm} = 2.88$ and $Re = 2000$ at $L/d = 120$.

The above results can be interpreted physically as follows.

Under the conditions of transitional and turbulent modes of flow in non-circular ducts, a portion of their cross-section in the corners is occupied by stable laminarized zones. In these regions the generation and propagation of wall vortices are hindered or impossible at all. Therefore, the enhancement of heat transfer in these regions is also hindered or impossible. As a result, a portion of the perimeter of the non-circular duct cross-section, which encloses the laminarized corner zones, characterizes the portion of the non-circular duct heat transfer surface which does not participate in the process of heat transfer enhancement. The duct with the cross-section in the form of an equilateral triangle has the greatest fractions of the flow cross-section and of the duct perimeter which are occupied by stable laminarized corner zones under the conditions of transitional and turbulent modes of flow. This is accounted for by the fact that all the three corner zones are characterized by the same minimally possible angle of 60°. The narrower the duct, characterized by the parameter h/t, the smaller the fractions of the flow cross-section and of the duct cross-section perimeter which are occupied by stable laminarized corner zones, up to complete disappearance of the latter in a circular duct. An explanation is that triangular ducts with $h/t > 1$ have two corners of above 60° and one corner of below 60°. Thus, in the duct with the cross-section in the form of an isosceles triangle, as against an equilateral triangle, a smaller portion of the flow cross-section is occupied by laminarized corner zones. Moreover, all non-circular ducts of the surfaces are made so that their cross-section angles are rounded on a radius. Therefore, the corner zone of the smallest angle of an isosceles triangular duct will decrease with an increasing parameter h/t, while the cross-section of this duct with rounded corners will approach a rectangle all corner regions of which have angles of 90°. When h/t of such a duct increases, the fraction of the flow cross-section occupied by laminarized corner zones decreases. Therefore, when the spacing between two walls of non-circular ducts decreases, the fraction of their heat transfer surface, participating in heat transfer augmentation, increases and, when $Nu/Nu_{sm} \geqslant \xi/\xi_{sm}$, the ducts with a smaller spacing have higher values of Nu/Nu_{sm} at corresponding values of Re. In non-circular ducts of different cross-sections with vortex generators in the heat carrier flow wall layer, the condition $Nu/Nu_{sm} = \xi/\xi_{sm}$ may develop at different values of Re which increase with a decreasing spacing between the two walls of the ducts. This is attributed to the fact that in corresponding smooth ducts the transitional mode of flow develops at different Re. The reason consists in the different fraction of the flow cross-section occupied, in different non-circular ducts, by stable laminarized corner zones. Therefore, in the case of $Re = idem$, $d = idem$ and $\nu = idem$ and the same mean flow velocity the local maximum velocities in the cross-sections of different non-circular ducts are different. Their greatest values are observed in the duct with the cross-section in the form of an equilateral triangle and the smallest in a plane rectangular duct. Therefore, the transitional mode of flow and the attainment of the maximum value of Nu/Nu_{sm}, provided $Nu/Nu_{sm} = \xi/\xi_{sm}$, are observed earlier, with respect to Re, in a channel with the cross-section in the form of an equilateral triangle, than in a plane rectangular duct.

Based on the analysis of the results obtained in the present work for interrupted surfaces, the following thermohydraulic correlations have been obtained which generalize the results of experiments within the confidence range 0.997, with errors indicated below.

(a) Heat transfer:

for $Re \leqslant Re_{\lim}$

$$Nu = 0.000437(\delta/d)^{-2.6}(l'/d)^{-0.15} \times Re^{2.2(\delta/d)^{0.55}(l'/d)^{-0.02}}$$

$$\delta Nu = \pm 7\% ; \qquad (6)$$

for $Re \geqslant Re_{\mathrm{lim}}$

$$Nu = 0.00723(\delta/d)^{-1.6}(l'/d)^{-0.9}\,Re^{1.2(\delta/d)^{0.34}(l'/d)^{0.15}}$$

$$\delta Nu = \pm 10\% \quad (7)$$

$$Re_{\mathrm{lim}} = 3960(\delta/d)^{0.25}(l'/d)^{0.42}. \quad (8)$$

(b) Coefficient of friction pressure losses:

for $Re \leqslant Re_{\mathrm{lim}}$

$$\zeta = 1.05(\delta/d)^{-1.05}(l'/d)^{-0.217} \times Re^{-0.277(l'/d)^{0.064}(\delta/d)^{-0.285}}$$

$$\delta\zeta = \pm 10\%; \quad (9)$$

for $Re \geqslant Re_{\mathrm{lim}}$

$$\zeta = 0.131(\delta/d)^{-0.44}(l'/d)^{-0.234} \times Re^{-0.0042(\delta/d)^{-1.25}(l'/d)^{0.39}}$$

$$\delta\zeta = \pm 12\% \quad (10)$$

$$Re_{\mathrm{lim}} = 448(\delta/d)^{-0.653}(l'/d)^{0.09}. \quad (11)$$

Correlations (6)–(11) are valid for interrupted surfaces formed by rectangular ducts only in the following ranges of their determining geometric parameters:

$$l'/d = 0.65\text{–}3.24; \quad \delta/d = 0.058\text{–}0.113; \quad h/t = 7:1.$$

To obtain the final thermohydraulic correlations for these surfaces, it is necessary to carry out further experimental investigations for different values of h/t within the range of 1–7.

CONCLUSIONS

(1) A systematic experimental investigation of heat transfer enhancement is carried out in 11 standard-size models of actual constructions of interrupted surfaces formed by short rectangular ducts with the cross-sectional aspect ratio 7 : 1 in the absence of burrs and bends of the sharp edges of fins. The investigation is carried out with air over the ranges $Re = 500$–$10\,000$, $l'/d = 0.65$–3.24 and $\delta/d = 0.058$–0.114. Standard experimental thermohydraulic characteristics of these surfaces are obtained and also their correlations as functions of Re and geometric parameters l'/d and δ/d.

(2) In surfaces Nos. 3_{in}–6_{in} with $l'/d = 1.3$–3.24, $\delta/d = 0.0777$ and $h/t = 7$ over the range $Re = 500$–5500 the heat transfer augmentation is characterized by a higher or identical increase in heat transfer against the increase in the coefficient of the friction pressure losses as compared with a geometrically identical smooth surface. In this case, for surface No. 3_{in} with the geometric parameters $l'/d = 1.3$ and $\delta/d = 0.0777$ at $Re = 2400$ the maximum value of 2.6 is obtained experimentally for Nu/Nu_{sm} at $(Nu/Nu_{\mathrm{sm}})/(\xi/\xi_{\mathrm{sm}}) = 1$.

(3) The process of heat transfer augmentation in interrupted surfaces can be reliably controlled by purposeful variation of the geometric parameters l'/d and δ/d for corresponding values of Re. Thus, when the parameter l'/d decreases over the range $l'/d = 0.65$–3.24 (surfaces Nos. 1_{in}–6_{in}) at $Re = idem$, the heat transfer and the coefficient of pressure losses in ducts increase in the entire range of Re studied. In this case, for $l'/d < 1.3$–1.5 the coefficient of pressure losses in interrupted surfaces increases more rapidly than heat transfer.

When δ/d decreases over the range 0.058–0.114 (surfaces Nos. 3_{in}, 7_{in}–10_{in}) at $Re = idem$, heat transfer increases in the entire range of Re studied, while the coefficient of pressure losses decreases. In this case, the effect of a change in δ/d on heat transfer is most substantial in the range of small values of Re. Conversely, the effect of a change in δ/d on the coefficient of friction pressure losses in the region of low values of Re does not virtually show up and becomes substantial only with increasing Re.

(4) For interrupted surfaces Nos. 8_{in} and 8^*_{in} with the geometric parameters $l'/d = 1.29$, $\delta/d = 0.0912$ and $h/t = 7$, an experimental investigation was made into the effect exerted on heat transfer and hydraulic resistance by the overall heat carrier pressure losses at the entrance and exit in an actual plate–fin heat exchanger. In this case, the coefficient of the unimpeded flow area in an interrupted surface was equal to $f_{\mathrm{un}} = 0.62$. It is found that for $f_{\mathrm{un}} \geqslant 0.62$ the perturbations, introduced into the heat carrier flow at the entrance and exit from a heat exchanger in the entire range of Re studied, virtually do not change the heat transfer characteristics of the surface studied. Under these very conditions the coefficient of total pressure losses for $Re \leqslant 750$ is equal to the coefficient of pressure losses of the heat carriers in the surface studied. When $Re > 750$, perturbations lead to a gradual increase in the coefficient of total pressure losses as compared with the coefficient of pressure losses of the heat carrier, and at $Re = 10\,000$ this increase attains 5%.

(5) A direct experimental verification is made of Kays' and Idelchik's methods commonly used in predicting heat carrier pressure losses for the entrance and exit from a plate–fin heat exchanger. It is found that Kays' method overshoots these losses on average by a factor of 8.5, while Idelchik's method by a factor of 2.1. Over the range of Re studied, this leads to the overestimation of the predicted coefficient of total pressure losses of a heat carrier in surface No. 8^*_{in} by 36% with Kays' method and by 5–6% with Idelchik's method.

The inadequate accuracy with which these methods predict heat carrier pressure losses for entrance and exit from plate–fin heat exchangers requires their refinement or the development of new techniques and their wide experimental verification with the use of different constructions of surfaces.

(6) Based on the earlier and present experimental

investigations into the enhancement of heat transfer in different non-circular ducts with different vortex generators in the heat carrier flow wall layer, it is found that for $Nu/Nu_{sm} \geqslant \xi/\xi_{sm}$ the degree of heat transfer enhancement depends only on determining the type of surface duct cross-section and is independent of the type of vortex generator constructions. When $Nu/Nu_{sm} = \xi/\xi_{sm}$, a decrease in the spacing between two walls of non-circular (from an equilateral triangle to a plane rectangle) ducts of length $L/d \approx 20$ leads to an increase in the heat transfer enhancement and in the corresponding Re over the ranges $Nu/Nu_{sm} = 1.5$–2.6 and $Re = 1250$–2400.

REFERENCES

1. W. M. Kays and A. L. London, *Compact Heat Exchangers*, 2nd Edn. McGraw-Hill, New York (1964).
2. R. K. Shah and A. L. London, Offset rectangular plate–fin surfaces—heat transfer and flow friction characteristics, *ASME J. Engng Pwr* **90**, 218–228 (1968).
3. I. N. Zhuravlyova, Investigation of heat transfer and hydraulic resistance in plate–fin heat exchangers, Author's Thesis (Cand. Sci.), MIKhM, Moscow (1968).
4. A. R. Wieting, Empirical correlation for heat transfer and flow friction characteristics of rectangular offset-fin plate–fin heat exchangers, *ASME J. Heat Transfer* **97**, 488–490 (1975).
5. N. Cur and E. M. Sparrow, Measurement of developing and fully developed heat transfer coefficient along a periodically interrupted surface, *ASME J. Heat Transfer* **101**, 211–216 (1979).
6. E. M. Sparrow and A. Hajiloo, Measurements of heat transfer and pressure drop for an array of staggered plates aligned parallel to an air flow, *Trans. ASME J. Heat Transfer* **102**, 426–432 (1980).
7. R. L. Webb and H. M. Yoshi, A friction factor correlation for the offset strip-fin matrix, *Heat Transfer 1982, Proc. 7th Int. Conf.*, München, 6–10 September 1982, Vol. 6, pp. 257–262 (1982).
8. A. F. Savostin and A. M. Tikhonov, Investigation of the characteristics of heat transmitting elements with short plate fins, *J. Engng Phys.* **21**(4), 713–721 (1971).
9. W. M. Kays, Loss coefficients for abrupt changes in flow cross section with low Reynolds number flow in single and multiple tube systems, *Trans. ASMS* **72**(8), 1067–1074 (1950).
10. V. S. Barinov, V. G. Voronik, E. V. Dubrovsky, A. M. Kosheleva, N. I. Martynova and V. A. Zolotov, A thermal wind tunnel for testing liquid–gas heat exchangers, *Bull. Izobretenii* No. 5, Inventor's Certificate 414510, p. 142 (1974).
11. G. I. Voronin and E. V. Dubrovsky, A resistance thermometer used to measure the mean-integral temperature in the gauging cross-section, *Energomashinostroenie* No. 1, 29–31 (1972).
12. G. I. Voronin and E. V. Dubrovsky, Experimental investigation of heat transfer enhancement in ducts of heat transfer surfaces by means of the effect of periodically generated boundary vortex in a wall layer. In *Deep Cold and Conditioning* (*Transactions of the Moscow Higher Engineering College*) No. 132, pp. 233–255 (1969).
13. G. I. Voronin and E. V. Dubrovsky, *Effective Heat Exchangers*. Mashinostroyeniye, Moscow (1973).
14. A. I. Mikhailov, V. V. Borisov and E. K. Kalinin, *Gas-turbine Closed-cycle Plants*. Izd. AN SSSR, Moscow (1962).
15. E. K. Kalinin, G. A. Dreitser and S. A. Yarkho, *Heat Transfer Augmentation in Channels*. Mashinostroyeniye, Moscow (1972).
16. E. V. Dubrovsky, The thermohydraulic efficiencies of heat transfer surfaces, *Izv. Akad. Nauk SSSR, Energet. Transp.* No. 6, 118–128 (1977).
17. I. E. Idelchik, *Handbook of Hydraulic Resistance*. Mashinostroyeniye, Moscow (1975).
18. E. V. Dubrovsky, Enhancement of convective heat transfer in plate–fin heat exchanging surfaces, *Izv. Akad. Nauk SSSR, Energet. Transp.* No. 6, 116–127 (1978).

ACCROISSEMENT DU TRANSFERT THERMIQUE CONVECTIF DANS DES CANAUX RECTANGULAIRES AVEC SURFACES INTERROMPUES

Résumé—On présente des résultats expérimentaux sur l'augmentation du transfert thermique de 11 surfaces interrompues d'un même type et de taille différente, formées par une multitude de canaux rectangulaires courts avec un rapport de forme constant 7 : 1 en l'absence de courbure sur le bord mince des ailettes. Des recherches sont faites pour l'air avec des nombres de Reynolds allant de 500 à 10 000 et des paramètres géométriques $l'/d = 0{,}65$–$3{,}24$ et $\delta/d = 0{,}058$–$0{,}114$. A partir de cette étude, les caractéristiques thermohydrauliques de ces surfaces sont obtenues empiriquement, ainsi que leurs corrélations en fonction du nombre de Reynolds et des paramètres l'/d et δ/d. De nouveaux résultats sont obtenus quant à l'effet de ces paramètres sur l'accroissement du transfert de chaleur et sur la détermination expérimentale des coefficients de perte de charge à l'entrée et à la sortie de la surface interrompue dans un échangeur thermique.

VERBESSERUNG DES KONVEKTIVEN WÄRMEÜBERGANGS IN RECHTECKKANÄLEN MIT UNTERBROCHENER OBERFLÄCHENKONTUR

Zusammenfassung—Die Ergebnisse einer systematischen experimentellen Untersuchung zur Steigerung der Wärmeübertragung an 11 unterbrochenen Oberflächenelementen desselben Typs, jedoch unterschiedlicher Größe, werden vorgestellt. Der Kanal setzt sich aus einer Vielzahl von rechteckigen Kanalelementen mit konstantem Strömungsquerschnitt (Seiten-Verhältnis 7 : 1) zusammen. Die Fügeflächen der Kanalelemente sind scharfkantig und haben weder einen Grat noch sind sie abgerundet. Für die geometrischen Parameter $l'/d = 0{,}65$–$3{,}24$ und $\delta/d = 0{,}058$–$0{,}114$ wurden Untersuchungen mit Luft bei Reynolds-Zahlen von 500 bis 10 000 durchgeführt. Aufgrund dieser Untersuchungen wurde das thermohydraulische Verhalten dieser Oberflächen ermittelt und als Funktion der Reynolds-Zahl sowie der geometrischen Verhältnisse l'/d und δ/d korreliert. Es ergaben sich neue Erkenntnisse über den Einfluß der geometrischen Parameter l'/d und δ/d auf die Verbesserung des Wärmeübergangs an unterbrochenen Oberflächen und außerdem auf den Gesamtdruckabfall zwischen Eintritt und Austritt eines Wärmetauschers mit unterbrochenen Oberflächen.

ИНТЕНСИФИКАЦИЯ КОНВЕКТИВНОГО ТЕПЛООБМЕНА В ПРЯМОУГОЛЬНЫХ КАНАЛАХ РАССЕЧЕННЫХ ПОВЕРХНОСТЕЙ

Аннотация—В работе представлены результаты систематического экспериментального исследования интенсификации теплообмена в 11 типоразмерах меделей реальных конструкций рассеченных поверхностей, образованных множеством коротких прямоугольных каналов с постоянным значением отношения сторон их сечения 7 : 1, при отсутствии заусенцев и загибов на острых кромках ребер. Исследование проведено на воздухе в диапазоне значений чисел $Re = 500$–$10\,000$ и геометрических параметров $l'/d = 0{,}65$–$3{,}24$ и $\delta/d = 0{,}058$–$0{,}114$. На основе исследований получены эталонные экспериментальные теплогидравлические характеристики этих поверхностей, а также их корреляции в функции числа Re и геометрических параметров δ/d и l'/d. Получены новые результаты по влиянию геометрических параметров l'/d и δ/d рассеченных поверхностей на интенсификацию теплообмена в них. Выявлено, что процесс интенсификации теплообмена в рассеченных поверхностях надежно управляем путем целенаправленного изменения его геометрических параметров l'/d и δ/d для соответствующих значений числа Re.