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Heat and momentum transfer in gas flowing through heated tube equipped with turbulence promoters

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Abstract—The intensification of the heat transfer between a gas stream and the wall of a tube provided with a set of perforated discs used as turbulence promoters, and the accompanying pressure drop, were investigated experimentally. The tube diameter, the geometrical parameters of the sets of the discs, and the gas flow rate, were varied in individual experiments. Appropriate correlations for the flow resistance coefficient of a single disc in a set, equations (26)–(28), and for the Nusselt number for sets of discs, equation (33), were formulated. It was indicated that application of described promoters can reduce the heat transfer surface area to 70% of that in the conventional type exchanger when the gas flow rate, the pressure drop and the amount of exchanging heat will be identical in both apparatuses. The reduction can be considerably larger when larger gas pressure drop in the exchanger with the disc is permissible.

INTRODUCTION

THE MAJORITY of heat transfer equipment used in various branches of industry is made up of conventional tube-in-shell exchangers because of their easy exploitation and their flexibility for operation in broad ranges of temperature and pressure. The installation of baffles within the inter-tubular space brings the partial heat transfer coefficient α at the outside surface of the tubes to the two- to threefold increase. In such a case, equalization of the heat transfer resistances at both sides of the tubes walls, and, in consequence, an increase in the overall heat transfer coefficient, needs enhanced partial heat transfer coefficient α within the tubes. Such an increase should be at least threefold when gas flows both outside and inside the tubes, but it should be more than tenfold when steam or liquid flows within the inter-tubular space and gas flows within the tubes.

The enhancement of heat transfer within tubular space of an exchanger by active methods (vibration, rotation, suction of boundary layer) is strongly limited for technical reasons. Therefore the passive methods have become attractive; e.g. tubes with internal fines or with various types of packings or inserts, tubes with creased walls or with coarse internal surface of the walls.

Within tubes with helically creased walls the heat transfer coefficient increases by up to 50–100% [1, 2], in comparison with ordinary tubes. Larger values of the partial heat transfer coefficient α can be reached in tubes with internal fins, but the technology of their production is complicated and some material limitation in their production arises, generally to copper, aluminum and their alloys [3–5].

Such an increase is about two times larger within tubes with coarse internal surface of their walls [6–12]. However, such tubes are very sensitive to impurities in fluid stream exchanging heat with the tube wall. The fact that their effectiveness in heat transfer enhancement decreases with decrease in Prandtl number, characteristic for the fluid exchanging heat with the tube wall, makes these sort of tubes less attractive for construction of heat exchangers for gases [6, 12].

The heat transfer enhancement in tubes provided with various packings and inserts is in general more pronounced, but the mechanism of the enhancement is more complicated than in both the above mentioned cases in which the increase in the heat transfer surface area and the turbulence in the fluid boundary layer play the decisive roles. The mechanism of the action of various types of inserts depends on their construction. In such cases an increase in local fluid velocity due to the reduction of the free cross-section of the tube, the turbulence in the fluid boundary layer and its separation from the tube wall, and the generation of the circulation zones, are the main factors responsible for heat transfer enhancement. The fluid flow resistance in tubes provided with inserts is in general much larger than in ordinary tubes, or even in tubes with coarse inner surface of their walls.

The largest increase of the partial heat transfer coefficient, up to tenfold of that in ordinary tubes, can be reached in tubes packed with granular beds or with wire-net matrices made of metals with high thermal conductivities [13–15]. However, the increase in the resistance of the fluid flow to the level of 10^3–10^4

NOMENCLATURE

c_p	specific heat of gas [J kg^{-1} K^{-1}]
D	diameter of tube [m]
d	diameter of openings in baffle [m]
e	equivalent of height of baffle [m]
F	surface area for heat transfer [m^2]
G	mass flow rate [kg s^{-1}]
L	length of tube [m]
l	spacing of baffles (distance between neighbor baffles in set) [m]
N	number of tubes
n	number of baffles on length L
Nu	Nusselt number $= \alpha D/\lambda$
Pr	Prandtl number $= c_p\eta/\lambda$
Q	heat [W]
Re	Reynolds number $= Gd/\eta$
$t1$	temperature of gas at inlet of tube [K]
$t2$	temperature of gas at outlet of tube [K]
tw	temperature of tube wall [K]
T	type of baffle
u	velocity of gas, average over tube cross-section [m s^{-1}]
u^+	dynamic velocity of gas $= u(\xi/8)^{1/2}$ [m s^{-1}].

Greek symbols

α	heat transfer coefficient [W m^{-2} K^{-1}]
Δp	static pressure drop [Pa]
Δt	temperature difference, logarithmic average [K]
ε	porosity of baffle (the ratio of the surface area of the openings and that of the tube)
η	viscosity [kg m s^{-1}]
λ	thermal conductivity of gas [W m^{-1} K^{-1}]
ξ	flow resistance coefficient $= 2\Delta p\, D/\rho u^2 L$
ξ_p	flow resistance coefficient for isolated baffle, equation (21)
$\hat{\xi}$	flow resistance coefficient referred to single baffle in set $= 2\Delta p\, \varepsilon^2/\rho u^2 n$
ρ	density of gas [kg m^{-3}]
τ^+	effective shear stress, equation (32) [kg m^{-1} s^{-2}].

Subscripts

o	for ordinary tube
max	maximal.

times that of ordinary tubes, and the sensitivity to mechanical impurities in the fluid stream exchanging heat are the main reasons that this type of apparatus is in general not installed in equipment designed exclusively for intensive heat exchange.

Lower flow resistances are observed in static mixers which are most effective in heat transfer enhancement for viscous liquids in the range of low Reynolds number, $Re < 0.1$ [16, 17]. Their application for gases is rather inexpedient.

The inserts whirling the fluid stream, and thus promoting turbulence within it, like propellers and spirals of wires or metal bands, also draw the interest of researchers. The spirals of metal bands seem to be attractive because they are easy to produce and they generate comparatively low flow resistance. At optimal conditions they can produce a 2.5-fold increase in the partial heat transfer coefficient accompanied by increase in the flow resistance by a factor of only 10, in comparison with ordinary tubes [18–20]. Their main shortcoming is that their effectiveness in heat transfer enhancement depends on the heat flux direction. When the fluid flowing within the tube is heated by its wall the centrifugal force generated within the fluid stream by the spiral enhances the transportation of the colder (heavier) fluid elements from the center of the tube to its wall. This is the reason that the heat transfer enhancement is larger by 30–50% in heating than in cooling of the fluid flowing within the tube. As the media at higher temperature are in general passed through the tubes, the heat exchangers with tubes provided with spirals will, in such cases, be less effective. The application of the wire spirals for heat transfer enhancement in gases is rather inexpedient because their effectiveness decreases as the Prandtl number, characteristic for the fluid exchanging heat with the tube wall, decreases; the maximal increase in the heat transfer coefficient, in comparison with ordinary tube, is $\alpha_{max}/\alpha_o = 3$ for liquids and $\alpha_{max}/\alpha_o = 1.6$ for gases [21, 22].

The installation of flow reducers like orifices, discs and cones within the tubes has also been considered as the means for heat transfer enhancement. It allows to increase the heat transfer coefficient up to four times at fivefold increases in the flow resistance as compared with ordinary tubes [23].

Among the turbulence promoters described in the literature a set of wire-netting baffles spaced along a tube at certain distance seems to be very effective in heat transfer enhancement [24]. It is possible to increase the heat transfer coefficient up to eight times as compared with the case in ordinary tube. However, the formation of the baffles is rather inconvenient because of their troublesome edging and fitting to the internal surface of the tube wall. Sets of perforated discs, which can be formed easily, seem to be more attractive [25, 26]. The application of this type of baffle for mass transfer enhancement in liquid stream has made it possible to double the Sherwood number for a 100-fold increase in flow resistance.

This work presents the results of investigation of the effectiveness of application of sets of perforated discs for heat transfer enhancement in gas flowing within tubular space of the tube-in-shell exchanger.

EFFECTIVENESS CRITERIA

The investigations of the effectiveness of the heat transfer enhancement in tubes of an exchanger by a set of perforated discs fixed within the tubes as the turbulence promoter have been based on the criterion formulated previously [27, 28]. It gives the information concerning the ratio of the investment costs of the exchanger with enhanced performance and those of the conventional one, when the exploitation costs for both compared exchangers are identical. This is possible when the fluid temperature at the inlet and that at the outlet, as well as the temperature of the tube wall, and, in consequence, the physical properties of the fluid, are identical in the exchangers compared; it means that the following equalities occur:

$$t1_o = t1, \quad t2_o = t2, \quad tw_o = tw, \quad \rho = \rho_o, \quad c_{po} = c_p,$$
$$\eta_o = \eta, \quad \lambda_o = \lambda_o. \tag{1}$$

The mass flow rate of the gas, its pressure drop, and the amount of heat exchanged, are also identical in both apparatuses:

$$G_o = G, \quad \Delta p_o = \Delta p, \quad Q_o = Q. \tag{2}$$

Under such conditions the economy of the investment costs is determined by the ratio of the heat transfer surface areas in the exchanger with turbulence promoters and in the conventional apparatus, F/F_o. However, the exchangers compared may differ by their constructional parameters, as the numbers of tubes (N), their diameters (D), and lengths (L), as well as the Reynolds numbers (Re), in single tube in each exchanger. It means that:

$$D_o \neq D, \quad L_o \neq L, \quad N_o \neq N, \quad Re_o \neq Re. \tag{3}$$

The balance equations for the exchangers compared may be written in the forms:

mass balance: $$\frac{N}{N_o}\frac{D}{D_o}\frac{Re}{Re_o} = 1 \tag{4}$$

momentum balance: $$\left[\frac{D_o}{D}\right]^3 \frac{L}{L_o}\left[\frac{Re}{Re_o}\right]\frac{\xi}{\xi_o} = 1 \tag{5}$$

heat balance: $$\frac{D_o}{D}\frac{F}{F_o}\frac{Nu}{Nu_o} = 1. \tag{6}$$

The solution of the equations (4)–(6) in relation to the constructional parameters is represented by the formulae (7) and (8):

$$\frac{N}{N_o} = \left[\left(\frac{D_o}{D}\right)^5 \frac{Nu_o}{Nu}\frac{\xi}{\xi_o}\right]^{1/3} \tag{7}$$

$$\frac{L}{L_o} = \left[\left(\frac{D}{D_o}\right)^5 \left(\frac{Nu_o}{Nu}\right)^2 \frac{\xi_o}{\xi}\right]^{1/3}. \tag{8}$$

When one introduces in equations (7) and (8) the Dittus and Boelter correlations, and the Blasius correlation (9) and (10), respectively:

$$Nu_o(Re_o) = 0.023 Re_o^{0.8}\, Pr^{0.4} \tag{9}$$

$$\xi_o(Re_o) = 0.3164 Re_o^{-0.25} \tag{10}$$

representing the heat transfer rate and the flow resistance coefficient in an ordinary tube, one obtains the final forms [25] of the criteria of the economy of the heat transfer surface area:

$$\frac{F}{F_o} = 6.53 \times 10^{-3}\left(\frac{D}{D_o}\right)^{0.179} Re^{1.231}(\xi(Re))^{0.41} \times (Nu(Re))^{-1.41} \tag{11}$$

$$Re_o = 0.2437 Re^{1.538}\left(\frac{D_o}{D}\right)^{1.026}[\xi(Re)/Nu(Re)]^{0.513}. \tag{12}$$

$\xi(Re)$ and $Nu(Re)$ are the experimentally determined correlations representing the flow resistance coefficient and the Nusselt number characteristic for tube provided with a given heat transfer intensifier investigated.

The minimal value of F/F_o, calculated by equation (11), meets the optimal solution which will ensure maximal economy in the heat transfer surface area for a given type of the turbulence promoter. As the criterion F/F_o is concerned, it should be emphasized that each value of Re characteristic for the exchanger with the turbulence promoter corresponds to different values of Re_o in the conventional exchanger. This means that at various Re in the exchanger with the turbulence promoter its heat transfer surface area is compared with that in conventional apparatus differing in constructional parameters but working at identical exploitation costs; e.g. at identical mass flow rate, pressure drop, amount of heat exchanged, inlet and outlet temperature, and tube wall temperature, respectively.

It is worth while to mention that, in industrial practice, the maximal gas pressure drop in the exchanger is often limited by the parameters of the blower used. Its available compression (Δp_{max}) often significantly exceeds the pressure drop produced due to the gas flow within exchangers of conventional type, and this pressure excess is in general lost. This means that the condition $\Delta p = \Delta p_o$ has not to be always executed and the economy criterion of the heat transfer surface area may be modified to the form:

$$(F/F_o)_{\Delta p = \Delta p_{max}} = (F/F_o)_{\Delta p = \Delta p_o}(\Delta p/\Delta p_o)^{-0.41}. \tag{13}$$

The criterion $(F/F_o)_{\Delta p = \Delta p_{max}}$ expresses the maximal reduction of the heat transfer surface area, possible when the available blower compression Δp_{max} exceeds considerably the pressure drop due to gas flow in exchanger with empty tubes. On the other hand, it indicates to what extent the amount of exchanging heat of a conventional exchanger can be increased by fitting its tubes with a proper set of perforated discs.

The effectiveness of the perforated discs for enhancement of heat transfer between the wall of a tube and the gas stream flowing through it has been estimated from the data collected in numerous experiments. These data were used to estimate the values of the average heat transfer coefficient (α), and the pressure drop measured at the steady-state conditions during gas flow within a tube provided with a set of perforated discs.

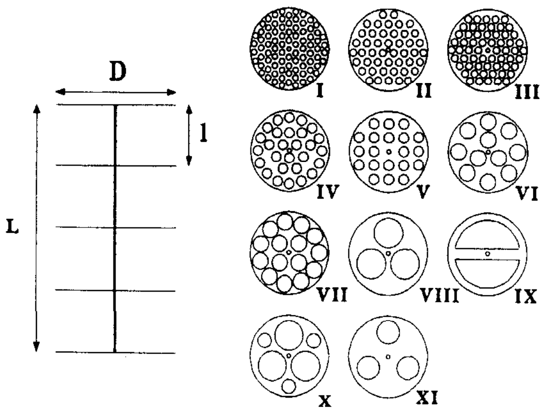

FIG. 2. Geometry of baffles and scheme of their set construction.

EXPERIMENTS

The experiments were performed with test setup shown in Fig. 1. It consisted of a concentric tube heat exchanger (1) heated by steam condensing at atmospheric pressure and supplied from the generator (2) provided with the pressure equalizer (3). The condensate was removed through the hydraulic closure (4). The orifice (5) measured the flow rate of the air stream supplied by the blower (6). The construction of the heat exchanger permitted the use of the inner tubes of different diameter (D) in series of experiments. All the thin-wall tubes of constant length (L) were made of copper and they were outfitted with interchangeable sets of perforated discs presented in Fig. 2. The diameters of the perforated discs were 0.2 mm smaller than the diameter of the tube, and therefore the fin effect was negligible as was indicated by the experiments performed with discs made of materials of various thermal conductivities [25, 28]. Their geometrical parameters are collected in Table 1, and the parameter ranges applied in various series of experiments are shown in Table 2.

The heat transfer coefficient (α), average over the tube length, was evaluated according to the formula:

$$\alpha = \frac{Q}{\Pi D L \Delta t}, \tag{14}$$

in which Q denotes the rate of heat input into the gas stream, calculated from the mass of condensate collected in a certain time interval, and Δt is the logarithmic average temperature difference based on the average temperatures of the gas stream at the inlet, $t1$, and at the outlet, $t2$, of the tube, and on its wall temperature, tw, taken as the temperature of steam condensation at actual atmospheric pressure.

The α-values were then used to calculate the Nusselt number according to its definition:

$$Nu = \frac{\alpha D}{\lambda}. \tag{15}$$

The Reynolds number based on the gas flow rate and the tube diameter:

$$Re = \frac{GD}{\eta} \tag{16}$$

was the variable parameter for the set of perforated discs with given geometrical parameters.

The Prandtl number was taken as constant:

$$Pr = 0.72. \tag{17}$$

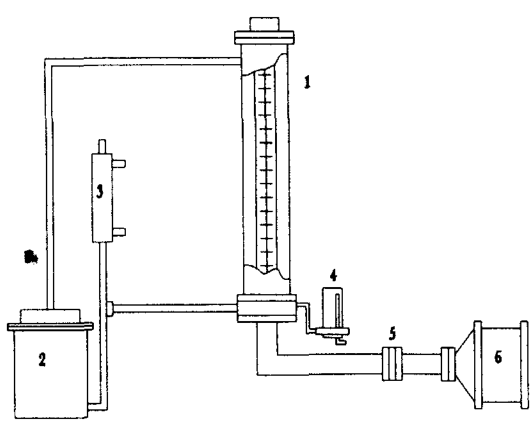

FIG. 1. Scheme of experimental equipment.

Table 1. Geometrical parameters of investigated baffles, thickness = 0.5 mm

Type of baffle T	d/D	ε	Number of openings	Remarks
I	0.0714	0.44	85	
II	0.100	0.47	46	
III	0.100	0.61	60	
IV	0.121	0.42	28	
V	0.143	0.42	20	
VI	0.200	0.45	11	
VII	0.200	0.61	15	
VIII	0.400	0.49	3	
IX	0.550	0.61	2	d = hydraulic diameter
X	0.334	0.61	6	d = hydraulic diameter
XI	0.320	0.3[illegible]	3	

Table 2. Ranges of geometrical, dynamic and thermal parameters investigated

$L = 1300$ mm;	$D = 33.5; 70; 100$ mm
$2.8 \leqslant l \leqslant 433$ mm;	$0.0714 \leqslant d/D \leqslant 0.55$
$1 \leqslant u \leqslant 15$ m s^{-1};	$5000 \leqslant Re \leqslant 50\,000$
$99.1 \leqslant tw \leqslant 101.6$°C	20°C $\leqslant t1 \leqslant$ 50°C; $60 \leqslant t2 \leqslant 98$°C

The flow resistance coefficient ξ for the tube with the insert of perforated discs investigated at constant Re and D was evaluated by the formula:

$$\xi = \frac{2\Delta p\, D}{\rho u^2 L}, \tag{18}$$

where Δp is the pressure drop in the air stream along the insert length, L. An analogical coefficient for a single disc was evaluated by the formula presented in our previous paper [29]:

$$\hat{\xi} = \frac{2\Delta p\, \varepsilon^2}{\rho u^2 n}. \tag{19}$$

The relation between both coefficients has the form:

$$\hat{\xi} = \xi \frac{\varepsilon^2 L}{nD}. \tag{20}$$

RESULTS OF EXPERIMENTS

The graphs in Figs. 3–5 illustrate the effects of the geometrical parameters of the perforated discs on the heat transfer intensification, the flow resistance, and the reduction of the heat transfer surface area. The analysis of these graphs leads to the following conclusions.

The insertion of a set of baffles into a tube intensifies the heat transfer ($Nu/Nu_0 > 2$), independently of their geometry, but only some of the inserts make the reduction ($F/F_0 < 1$) of the heat transfer surface area possible (see Figs. 3 and 4).

The increase in the opening cross-section area (porosity) in the discs with the opening diameter d remaining constant results in a decrease in the heat transfer intensity. However, it promotes the reduction of the heat transfer surface area at lower Re due to the reduction of the flow resistance (Fig. 4).

The insert T-VIII presents the optimal Nu/Nu_0 characteristics. The inserts T-VIII, T-IX and T-X demonstrate comparable reductions of the heat transfer surface area F/F_0 (Fig. 5), and therefore these were investigated thoroughly.

The graphs in Fig. 6 illustrate the effects of spacing (l/d) between the baffles T-VIII, T-IX and T-X, fixed in sets, on the heat transfer intensification Nu/Nu_0, the heat transfer surface area reduction F/F_0, and the flow resistance ξ/ξ_0, at constant D and Re. It can be seen that all the relations $Nu/Nu_0 = f(l/d)$, $F/F_0 = f(l/d)$, and $\xi/\xi_0 = f(l/d)$ are non-monotonic.

The graphs in Fig. 7 illustrate the effects of the spacing in the set of the baffles T-VIII on the heat transfer intensity Nu/Nu_0 and the reduction of heat transfer surface area F/F_0, observed in tubes with different diameters at constant Re. One can conclude that the Reynolds number, defined with the tube diameter and the average fluid velocity, cannot be considered as the proper criterion of the dynamic similarity in tubes with the turbulence promoters investigated.

The graphs in Fig. 8 illustrate the effect of spacing in the sets of baffles T-VIII and T-XI on the flow

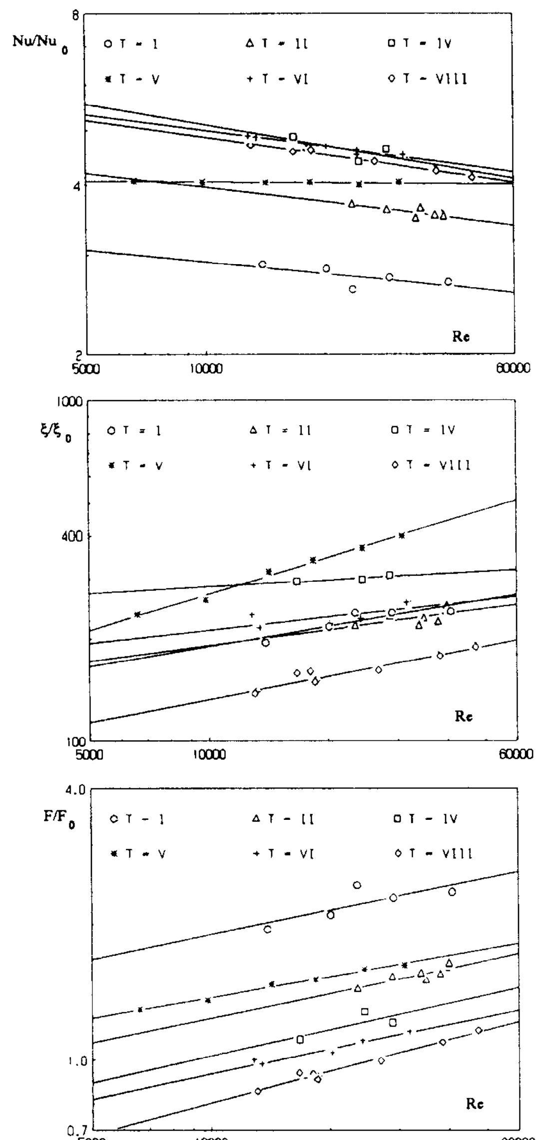

FIG. 3. Effect of the baffle type at constant ε on the relation. $F/F_0 = f(Re)$, $Nu/Nu_0 = f(Re)$ and $\xi/\xi_0 = f(Re)$, $D = 70$ mm, $l = 65$ mm.

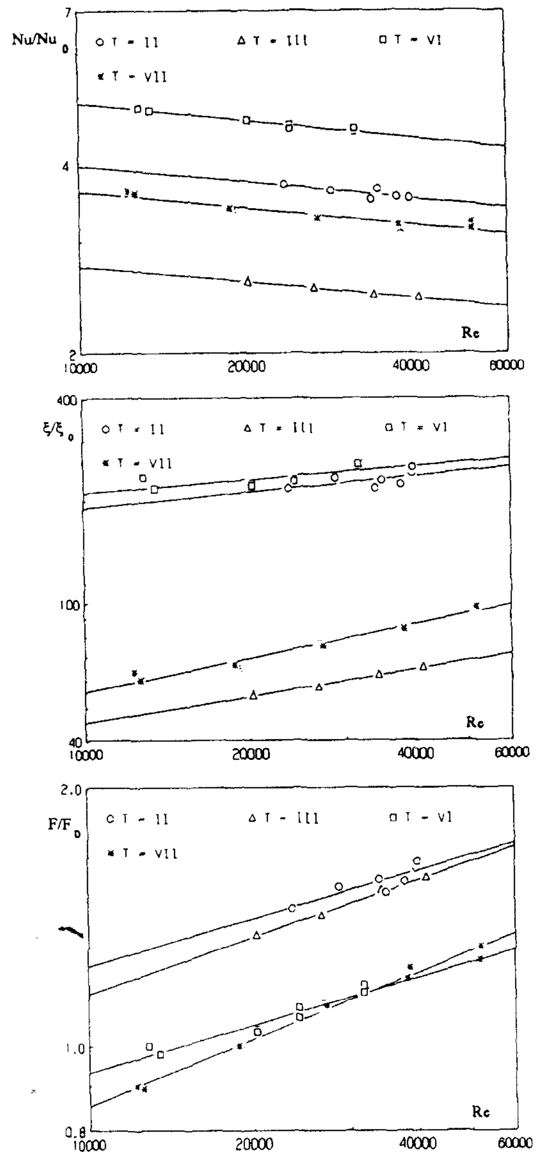

FIG. 4. Effect of baffles porosity at const d on Nu/Nu_0, F/F_0, ξ/ξ_0. $D = 70$ mm, $l = 65$ mm.

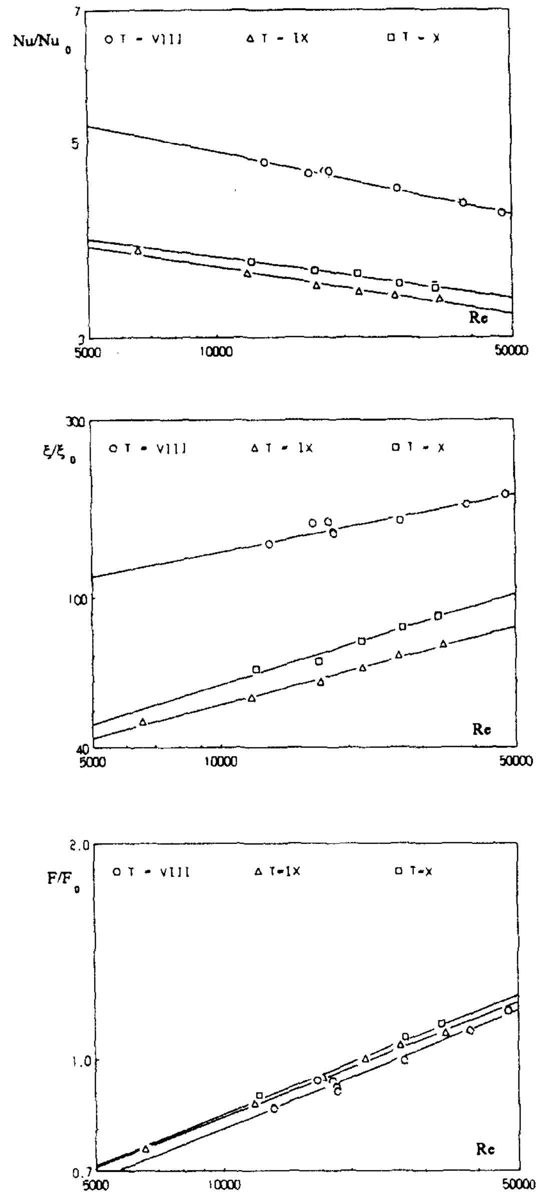

FIG. 5. Comparison of Nu/Nu_0, F/F_0 and ξ/ξ_0 for baffles T-VIII, T-IX, T-X. $D = 70$ mm, $l = 65$ mm.

resistance coefficient $\bar{\xi}$ evaluated according to equation (19) as related to the single baffle. There are also marked the values of ξ_p estimated according to the relation [30]:

$$\xi_p = (1+0.707\sqrt{(1-\varepsilon)}-\varepsilon)^2 \tag{21}$$

for turbulent flow through single isolated perforated baffle with arbitrary shape and diameter of the openings within the range of the baffle porosity:

$$0.01 < \varepsilon < 0.9. \tag{22}$$

The graphs in Fig. 8 show that for $l/d \gg 1$ the baffle spacing does not affect the value of $\bar{\xi}$ which approaches the value, ξ_p, characteristic for the isolated single baffle. This indicates that when the neighbor baffles are spaced along the tube at comparatively large distance each of them generates the fluid flow field similar to that generated by an isolated baffle. Each baffle produces the same flow field deformation which is responsible for intensification of the heat transfer between the tube wall and the fluid stream. Hence, the pressure drop for this range of the spacing $l/d \gg 1$ is proportional to the baffle number n:

$$\Delta p \propto n. \tag{23}$$

One can conclude, from equation (19), that the flow resistance coefficient $\bar{\xi}$ approaches a constant value for the baffle spacing of $l/d \gg 1$. This is confirmed by the experimentally obtained data illustrated in Fig. 8.

When the distances between the adjacent baffles are small, $l/d \ll 1$, the structure of fluid flow field may not change along the tube. For instance, when the insert

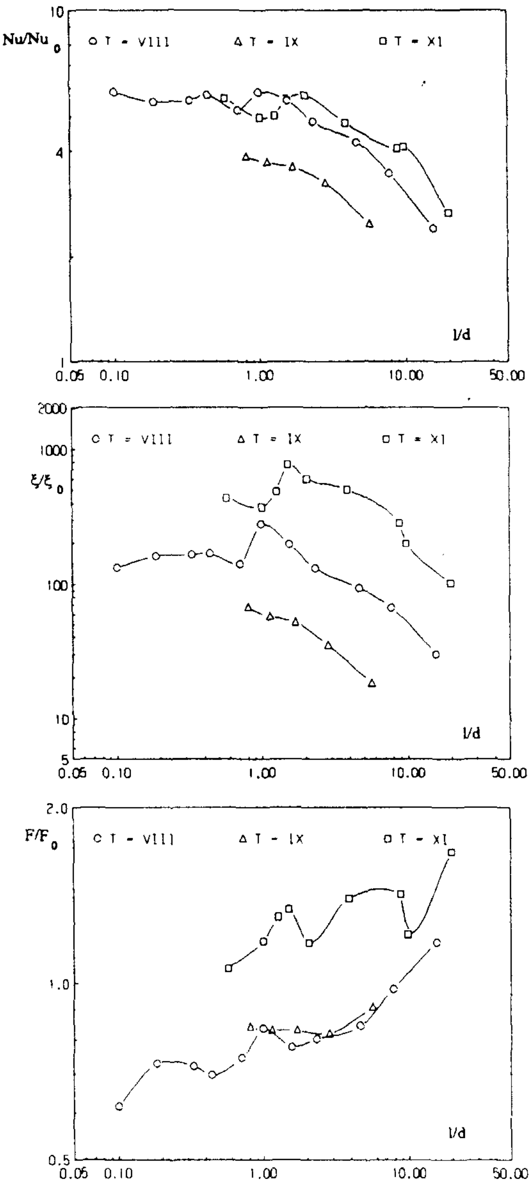

FIG. 6. Effect of baffle spacing in tube on F/F_0, Nu/Nu_0 and ξ/ξ_0 for different baffle types. $D = 70$ mm, $Re = 10\,000$.

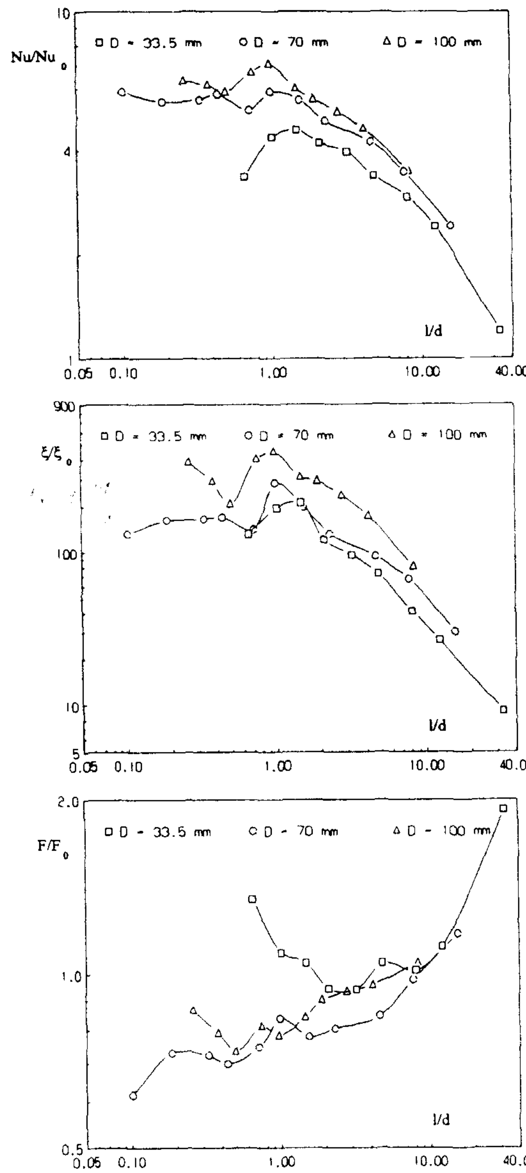

FIG. 7. Effect of baffle spacing T-VIII in tubes with different diameters on Nu/Nu_0, F/F_0 and ξ/ξ_0. $Re = 10\,000$.

T-VIII with three symmetrically displaced openings is mounted in a tube and gas is passed through it, the overall stream is broken into three parallel streams, each with cross-section practically unchanged along the tube. It means that the gas pressure drop will not depend on the distance between the neighbor baffles:

$$\Delta p = \text{const.} \quad \text{for } l/d \ll 1. \tag{24}$$

It results from equation (19) that in such a case the flow resistance coefficient for single baffle is inversely proportional to the number of discs:

$$\xi \propto 1/n. \tag{25}$$

Hence, ξ increases linearly with the increase in spacing, l/d, that is confirmed by experimental results illustrated in Fig. 8.

For a certain "critical" spacing, $(l/d)_{cr}$, depending on the baffle geometry, a rapid increase in ξ value is observed which may be called "flow resistance crisis". For the inserts of baffles T-VIII and T-XI such a "crisis" can be observed in a comparatively narrow range of spacing l/d:

$$0.7 < l/d < 2. \tag{26}$$

CORRELATION OF EXPERIMENTAL RESULTS

According to the results of the experiments performed, one may image a physical model of the gas flow within a tube provided with a set of perforated baffles spaced along it periodically. Such a model is illustrated in Fig. 9.

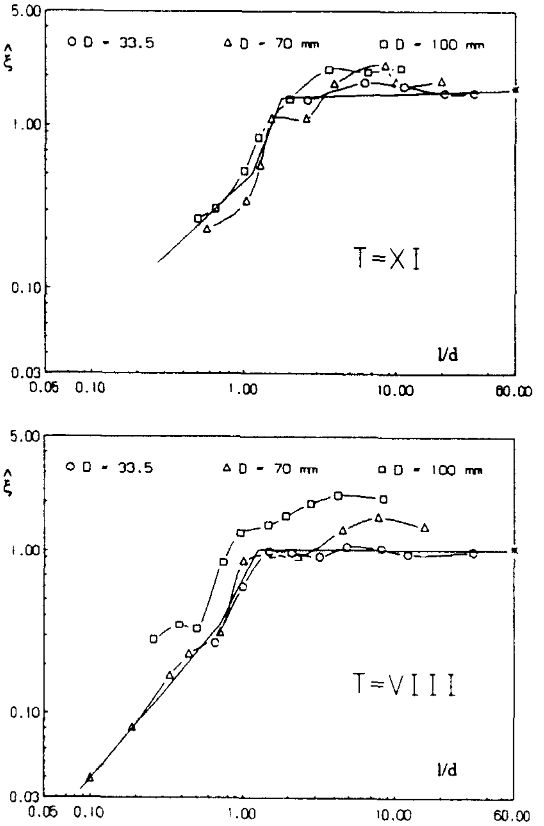

FIG. 8. Effect of baffle spacing in tubes with different diameters on the flow resistance coefficient for single baffle $\hat{\xi}$ calculated according to equation (33). $Re = 10\,000$.

When the distance between the adjacent baffles is large (Fig. 9a) each baffle causes independently the separation of the gas boundary layer from the tube wall. The superposition of the flow resistances produced by individual baffles on the overall resistances produced by individual baffles created the overall resistance observed in the system. This is the case when $\Delta p \propto n$ and $\hat{\xi} = \text{const}$.

When the distances between the adjacent baffles is small (Fig. 9c) the overall gas stream consists of a set of parallel streams in number equal to the number of the baffle openings and interacting with each other rather poorly.

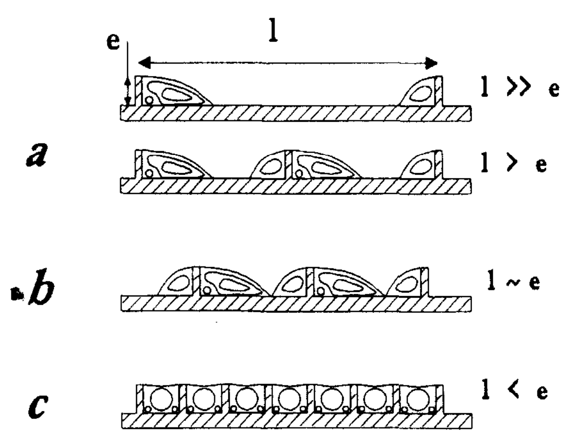

FIG. 9. Hypothetical image of flow within tube with periodically spaced baffles.

Between the two extreme cases of fluid flow possible in the system considered there is an intermediate region, in which at a certain "critical" spacing, l_{cr}, between the baffles the flow character changes (Fig. 9b). The recirculation zones produced by adjacent baffles start to influence each other, and this range of spacing generates the so-called "flow resistance crisis". It depends on the baffle equivalent height, e, and, in consequence, on the baffle type T.

The flow resistance coefficient $\hat{\xi}$ is the magnitude which properly characterizes the flow resistance in the system considered. It depends on the baffle spacing, l/d, the Reynolds number, Re, and the baffle type, T, and this relation is represented by different formulae valid for the sets of the baffles T-VIII and T-XI:

$$\hat{\xi} = A(l/d)\,Re^{0.1} \tag{27}$$

$$\hat{\xi} = A(l/d)^{2.7}\,Re^{0.1} \tag{28}$$

$$\hat{\xi} = \xi_p[1 - A(l/d)^{-1.4}]. \tag{29}$$

The values of constant A in equations (27)–(29) and the ranges of their application are given in Table 3. These formulae correlate also the results of experiments performed with the sets of the baffle types T-V, T-IX and T-X. They do not involve the tube diameter, and this means that the characterization of the flow resistance by means of the coefficient $\hat{\xi}$ enables one to keep the dynamic similarity while scaling-up the tube-in-shell heat exchanger with tubes provided with the perforated baffles of types investigated.

The correlations proposed involve the ratio l/d, although more appropriate seems to be the use of the ratio l/e, with e as the "equivalent height" of the perforated baffle. However, the attempts at estimation of this magnitude according to the experimental results available have failed, and the correlation of the spacing responsible for the "flow resistance crisis" with the geometrical parameters of the baffle has been impossible.

Figure 10 illustrates the correlations of the results of experimental investigations of the effectiveness of five different types of turbulence promoters, installed in a tube, in the enhancement of the heat exchange between the tube wall and the gas flowing within the tube. The modified Reynolds number defined by the dynamic gas velocity is the independent variable:

$$Re^+ = \frac{u^+ D \rho}{\eta} \tag{30}$$

in which

$$u^+ = (\tau^+/\rho)^{1/2} = u(\xi/8)^{1/2} \tag{31}$$

and

Table 3. Correlations for ξ, equations (27–29)

Flow character	Equation No.	T	A	Range of l/d
Resistance independent of baffle number	(27)	VIII	0.17	$0 < l/d \leqslant 0.7$
		XI		$0 < l/d \leqslant 1$
Resistance crisis	(28)	VIII	0.32	$0.7 < l/d < 1.0$
		XI		$1.0 < l/d < 1.5$
Additive resistance	(29)	VIII	0.25	$l/d > 1.0$
		XI		$l/d > 1.5$

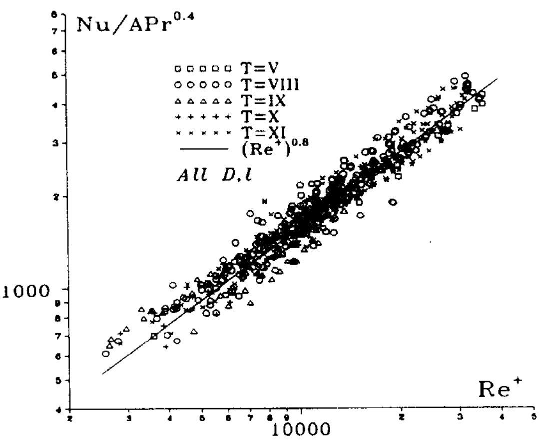

FIG. 10. Relation between heat transfer intensity and Reynolds number defined by the gas dynamic velocity, for different baffles.

$$\tau^+ = \frac{\Delta P D}{4L}. \quad (32)$$

Each set of the results can be correlated with a general formula:

$$Nu\,Pr^{-0.4} = C(Re^+)^{0.8} \quad (33)$$

in which the only fitting parameter C depends on the type of baffle forming the turbulence promoter, and is independent of the tube diameter and the baffle spacing; its characteristic values for the baffles investigated are collected in Table 4.

CONCLUSIONS

The analysis of the heat transfer effectiveness criteria F/F_0 and Nu/Nu_0, evaluated according to experimentally collected data, has revealed that the turbulence promoters made of the baffles T-VIII fixed in a tube is the most effective in the enhancement of the heat transfer between its wall and the gas flowing through it. Strong intensification of the heat transfer at $Nu/Nu_0 \approx 7$ is mainly due to the large velocity gradients and the high level of turbulence in the vicinity of the tube wall [28].

It has been found that the influence of gas velocity on the flow resistance coefficient ξ, and on the heat transfer intensification Nu/Nu_0, is insignificant.

The dependence of the flow resistance coefficient ξ, and the Nusselt number Nu, on the baffle spacing is non-monotonic. For the spacing close to the opening diameter, when $l/d \approx 1$, the flow resistance decreases as the number of baffles increases. The intensification of heat transfer between the tube wall and the gas flowing within the tube provided with a set of perforated baffles depends only on the baffle type and on the modified Reynolds number defined with the dynamic gas velocity.

The application of the turbulence promoters within the tubes of a tube-in-shell heat exchanger for gases makes possible a remarkable reduction of heat transfer surface area to 70% of that in conventional apparatus at the same exploitation conditions. More significant reduction will be possible when an excess in compression of the blower provided for supplying the gas into the tubes is available.

Table 4. Values of constant C in equation (33)

Type of baffle, T	V	VIII	IX	X	XI
C	0.095	0.116	0.131	0.134	0.093

Standard deviation $\pm 14\%$.

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