Journal of Engineering Physics and Thermophysics, Vol. 82, No. 1, 2009

ESTIMATION OF THE THERMOHYDRAULIC EFFICIENCY OF SWIRLERS AT SMALL REYNOLDS NUMBERS

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UDC 621.036.536.244

To estimate the thermohydraulic efficiency of different types of swirlers of heat exchange in pipes, we have generalized the experimental data, which has made it possible to determine the optimal range of Reynolds numbers and the optimal geometric sizes of swirlers.

Keywords: *transient regime, intensification of heat exchange, twisted tape, wire insert, screw, one-propeller insert, two-propeller insert, thermal efficiency, hydraulic efficiency, energy efficiency.*

Current energy programs call for carrying out an energy-saving policy in all fields at the cost of improving, adopting, and investigating the energy-saving equipment, increasing the efficiency and reliability of its operation, and environment protection.

The development of any heat exchange apparatus (HEA) proceeds from the requirement of a certain power intensity of the processes proceeding in it with the observation of technological, ecological, and other norms and requirements and is reduced, in the final analysis, to the search, for a given time and branch, for a compromise between capital and working costs. For instance, for transport power plants the application of different levels of forced regimes with a weak development of cogeneration is justified. In stationary power plants, preference is commonly given to various nonforced cogeneration regimes.

Comparison of the efficiency of different electric energy producers according to ISO-86 [1] to the new data, mainly for low-power plants, vividly demonstrates the state of the art in heat power engineering (Fig. 1). The relatively low thermodynamic efficiency of the power plants given in Fig. 1 is associated with the attained level of limiting temperatures in the cycle, which largely depends on the heat exchange with the screening structural elements and stimulates the development of methods for its intensification. In some cases, this is done with the aim of increasing the fuel utilization factor in the operating plant (organization of deep cooling of flue gases, recovery of waste heat after thermal engines, and increasing the performance of the heat exchangers in noncalculated regimes). In other cases, a decrease in the specific consumption of materials of products and their capital consumption (development of new heat-exchange and boiler equipment) is planned. Thirdly, it makes it possible to bring the process of electric energy production to a new qualitative level (development of radically new systems for cooling gas-turbine blades and fuel elements of nuclear reactors, and creation of an air-gas heater for a solid-fuel steam-gas plant (SGP) with an open cycle).

Wide use of new hydrodynamic regimes characterized by lower Reynolds numbers ($Re < 10^4$), nondeveloped turbulence, alternating turbulent and laminar regimes, and low heat transfer coefficients requires new data on the heat exchange intensification at low Re. These data are of particular importance for designing and redesigning thermal power plants (TPP), objects of industrial power engineering, and heat supply systems under new condition in connection with the universal change in the operational conditions of the power industry (changeover to maneuver and underloaded regimes) and its restructurization (with the creation of distributed systems, rapid development of the sector of small power plants, and changeover of small power plants to cogeneration regimes). For the existing equipment this means the changeover to noncalculated operating conditions, burn of the heat exchange surfaces, and a decrease in the thermal efficiency (efficiency) of the HEA. Its modernization or the development of new apparatuses calls for the elaboration of methods for intensifying the transfer processes in nonintensive (in general) hydrodynamic regimes. At heat supply enterprises with combined production of thermal and electric energy, it is necessary to maintain a high ef-

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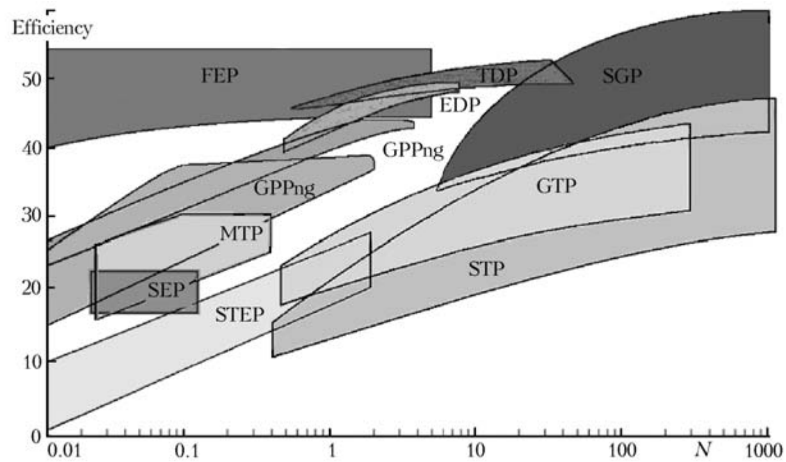

Fig. 1. Efficiency of modern thermal power plants as a function of electric power (N): GPP — gas-piston plants; GPPpg — mini-TPP-ICE burning producer gas, GPPng — mini-TPP-ICE burning natural gas (NG); GTP — gas-turbine plants burning NG; TDP — two-cycle diesel plants; MTP — microturbine plants; STEP — steam engine plants; STP — steam turbine plants burning NG; SGP — steam gas plants burning NG; SEP, Stirling engine plants; FEP — fuel element plants; FDP — four-cycle diesel plants. Efficiency, %; N, MW.

ficiency of operation of equipment under all operating conditions in order to manage seasonal and technological fluctuations of the thermal load.

In stationary devices of small power plants (a new, in world practice, dynamically developing kind of communal and household power-generation objects of the distributed power system [1]) operating in parallel with the power network, preference is given to optimal nonforced basic regimes with ultimately developed cogeneration and the highest possible installed capacity utilization factor (ICUF). Objects of the base power engineering seek to reach the highest possible efficiency of electric energy production by increasing the thermodynamic efficiency of a cycle (application of combined and hybrid cycles and superhigh parameters).

The chosen object of investigation — the gas-to-water heat exchanger with short wide smooth pipes — is a typical representative of HEAs needing intensification of the heat transfer and optimization of the design with the preservation of the low-head operating conditions. The fields of direct use of the developed design are tubular air heaters of steam boilers of TPPs and production prototypes of gas-water heat generators and low- and medium-power heating hot-water boilers burning gaseous and local fuel. Analogous devices are considered for use as high-efficiency, compact, low-head boilers-recoverers for the operation of gas-turbine, gas-piston, and diesel power plants in the cogeneration regime [2, 3].

Estimation of the Efficiency of the Heat Transfer Intensification Measures. The operation of HEAs in the laminar-turbulent transition region of the flow conditions and nondeveloped turbulence means that the heat transfer processes in them formally do not obey the thermohydraulic Reynolds analogy commonly used in designing heat-exchange apparatuses [4]. It will be recalled that, being some idealization, Reynolds analogy based on the similarity of velocity and temperature fields is realized in its simplest form for zero-gradient flows at Pr = 1 in the absence of internal heat sources. In HEA pipes and channels, where the flow is practically always enforced, the similarity of velocities and temperatures can only be approximate, and, what is more, with the fulfillment of a number of restrictions.

In reality, in flows in pipes and channels there exist mechanisms leading to the violation of these conditions and requiring special consideration in each particular case. Among the best known reasons for such a deviation from the requirements of analogy are [4, 5]: 1) the difference of the Prandtl number from unity; 2) the variability of the thermophysical properties; 3) the dependence of heat transfer on the kind of the heat transfer agent; 4) injection and drawing off of matter on the surface; 5) volume sources and sinks of matter and energy in the flow (due to compression, friction, and proceeding of chemical reactions); 6) the nonisothermality of the boundary conditions; 7) detached flows and swirling flows; 8) laminar and transient regimes.

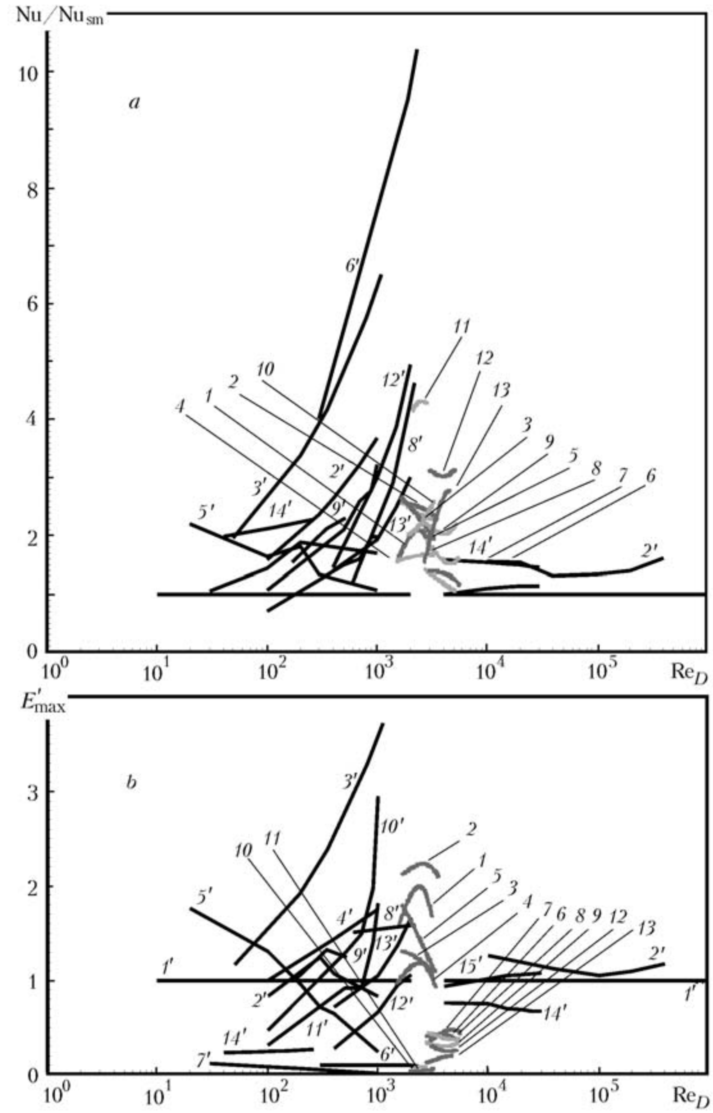

Fig. 2. Values of the thermal (a) and energy (b) efficiency of swirlers according to the data of the authors (1–13, numbering corresponds to Table 1) and the data of [7] (1′–15′).

Successful solutions of most of these problems have been the subject of numerous articles on convective heat transfer, in which theoretical constructions are approximated, as a rule, by criteria dependences in the form of (modified) Reynolds analogy. An analogous method is used to estimate the efficiency of measures taken to intensify the processes of heat and mass transfer in HEAs.

By now a large number of methods for estimating of the efficiency HEAs have been developed [6]. Of these, the most justified, clear, and recognized in the modern literature is the method of energy coefficients developed by academician M. V. Kirpichev in the form of the relative universal efficiency criterion [7]:

$$\overline{E}' = E'/E'_{sm} \, . \tag{1}$$

With the use of criteria similarity equations expression (1) can be rewritten in the form convenient for comparison of empirical data:

TABLE 1. Influence of the Reynolds Number and Geometric Parameters of Intensifiers on the Energy $\overline{E}'$, Thermal Nu/Nu_{sm}, and Hydraulic ξ/ξ_{sm} Efficiency in the Laminar-Turbulent Transition Region

Experiment number	Type of intensifier	Re_D	Intensifier parameters	$\overline{E}'$	Nu/Nu_{sm},	ξ/ξ_{sm}
1	Twisted tape	1565 2755 2918 3089	d_e = 30 mm, s/D = 2.6, δ = 0.33 mm	1.59 1.87 1.84 1.64	1.61 1.94 2.00 1.70	1.01 1.03 1.09 1.04
2	Twisted tape	1630 2950 3230 3560	d_e = 30 mm, s/D = 3.0, δ = 0.33 mm	2.10 2.36 1.94 2.18	2.56 2.53 2.17 2.34	1.25 1.07 1.12 1.07
3	Twisted tape	1660 2710 2950 3220	d_e = 30 mm, s/D = 4.2, δ = 0.8 mm	1.31 1.24 1.01 1.12	2.59 2.39 2.20 2.32	1.98 1.92 2.17 2.07
4	Twisted tape	1470 2780 2990 3350	d_e = 30 mm, s/D = 5.0, δ = 0.8 mm	0.95 1.35 0.78 1.00	1.55 1.76 1.56 1.69	1.62 1.31 2.13 1.68
5	Wire insert	1635 2926 3170 3310	d_e = 30 mm, s/D = 6.8, δ = 0.8 mm	1.81 1.26 1.02 1.14	2.66 2.15 1.84 2.01	1.47 1.70 1.80 1.77
6	Wire insert	2680 4580 4970 5300	D = 50 mm, d_w = 1.0 mm, s = 40 mm, s/D = 0.8	0.40 0.42 0.36 0.37	1.43 1.12 1.06 1.05	3.92 2.68 2.97 2.72
7	Wire insert	2880 4970 5190 5710	D = 50 mm, d_w = 1.5 nn, s = 90 mm, s/D = 1.8	0.35 0.49 0.44 0.44	1.39 1.24 1.27 0.14	3.94 2.51 2.90 2.61
8	Wire insert	2807 4570 4870 5600	D = 50 mm, d_w = 2.5 mm, s = 40 mm, s/D = 0.8	0.45 0.39 0.36 0.39	1.92 1.52 1.53 1.62	4.23 3.92 4.28 4.15
9	Wire insert	2690 4480 4675 5380	D = 50 mm, d_w = 5.0 mm, s = 45 mm, s/D = 0.9	0.34 0.34 0.27 0.35	2.24 2.19 1.92 2.25	6.60 6.46 7.07 6.49
10	Screw	1900 2800 3150 3300	d_e = 20 mm, s = 0.34 mm, l = 170 mm, φ = 38^o	0.04 0.05 0.06 0.06	2.01 2.36 2.45 2.66	48.01 45.61 43.11 42.69
11	Screw	2100 2770 2780 2860	d_e = 22 mm, s = 0.41 mm, l = 500 mm, φ = 36^o	0.07 0.07 0.08 0.07	4.16 4.22 4.36 4.25	58.63 62.11 52.10 60.75
12	One-propeller insert	3000 4300 4420 5180	D = 50 mm, n = 0.3, d_0/d = 0.3, φ_m = 37^o	0.31 0.26 0.23 0.27	3.13 3.12 2.95 3.17	10.08 12.18 12.63 11.53
13	Two-propeller insert	2890 4050 4291 4800	D = 50 mm, n = 0.3 d_0/d = 0.3, φ_m = 37^o	0.12 0.18 0.21 0.20	1.52 2.71 2.68 2.79	12.91 14.88 13.05 13.94

TABLE 2. Maximum and Minimum Values of the Thermal, Hydraulic, and Energy Efficiency of the Tested Intensifiers according to the Experimental Results

Type of intensifier	Nu/Nu_{sm}		ξ/ξ_{sm}		$\overline{E}'_{max}$	
	max	min	max	min	max	min
Twisted tape	2.66	1.55	2.17	1.01	2.36	0.78
Wire insert	2.25	1.05	1.05	2.68	0.49	0.27
Screw	4.36	2.01	62.11	42.69	0.08	0.04
One- and two-propeller insert	3.17	1.52	14.88	10.08	0.31	0.12

$$\overline{E}' = \frac{Nu/Nu_{sm}}{\xi/\xi_{sm}} . \tag{2}$$

In [6, 7], it is suggested to reveal the possibilities of a particular intensifier by means of the criterion $\overline{E}'_{max}$ at each particular number Re. For example, for transverse projections at a fixed number Re_i = idem for a number of constant heights (h/d) of diaphragms at all possible step sizes (s/h) the set of coefficients $\overline{E}'$ is found. From this set the variant $\overline{E}'_{max}(Re_i)$ indicating projection parameters (h/d, s/h) optimal for a given number Re_i is chosen. The dependences of the limiting efficiency $\overline{E}'_{max}(Re)$ for some types of intensifiers are presented in Fig. 2. Comparison of the limiting regime characteristics for different types of intensifiers makes it possible to justifiably choose the optimal design in the given range of Reynolds numbers.

On the basis of such methods the authors have done much work to reveal the limiting themohydrodynamic regimes for different types of intensifiers [7] and summed up a kind of intermediate results on the problem of creating effective heat transfer intensifiers in the channels of power plants. In so doing, the authors noted, in particular, the lack of information in the region of the laminar-turbulent transition in intensified channels and pointed to the urgency of investigating the processes in this region. Comparison of the results of these investigations with our data is given in Fig. 2. Several points are worthy of notice: the quasi-resonant character of the fields of parameters with a maximum in the transition region; the presence of two regimes qualitatively differing in efficiency (>1 and <1), and "successful" filling with our data of the relatively free interval in the range of $Re_D = (2–6)\cdot 10^3$.

Returning to the analysis of the efficiency of thermohydraulic regimes of channels with heat transfer intensifiers, note that from the $\overline{E}'$ value, one can judge the efficiency of a given intensifier. Depending on the $\overline{E}'$ value, several groups are distinguished.

The first group includes conditionally low-efficiency regimes ($\overline{E}' < 1$) most widely used before the middle of the 20th century. Having often a high thermal efficiency with $Nu/Nu_{sm} > 1$, devices operating under such conditions are characterized by a low hydraulic efficiency $\xi/\xi_{sm} >> 1$. An illustrative example of such specific heat exchangers are cyclone apparatuses (primary furnaces) in which the Nu value increases, due to the swirling, by a factor of 25–30 and the hydraulic resistance increases by two orders of magnitude. Systems with a less clearly defined prevailance of hydraulic losses also exist. For instance, when on the inner surface of the pipe continuous sandy roughness is created (Nikuradze's experiments), the $\overline{E}'$ value appears to be only 15–20% less than unity. As was shown in [8, p. 288], violation of similarity of the boundary conditions in the equations of motion and heat transfer towards the friction forces lowers the efficiency of the surface with sandy roughness at $Re_D = 10^4–10^5$ even at Pr = 1 and grad $P = 0$ to $\overline{E}' \approx 0.86$ at $Nu/Nu_{sm} = 1.57$. In the special literature [4, 6, 9, 10], one can find many examples of a much lower thermohydraulic efficiency ($\overline{E}' << 1$) of intensified surfaces ($Nu/Nu_{sm} > 1$) compared to smooth pipes (Fig. 2).

The second group represents effective thermohydraulic regimes with leading intensification of the heat transfer compared to the increase in hydraulic losses $\overline{E}' > 1$. In many cases, they are of practical interest since they permit energy-effective optimization of the design. Its manifestation is maximum in the transition region with $\overline{E}' \approx 2.5–3$ and insignificant in the region of developed turbulence at $Re_D > 10^5$ or at Reynolds numbers close to zero.

Choice of the Intensification Method. Among the various methods of creating effective thermohydrodynamic regimes is the easiest-to- realize method of flow swirling. As the heat transfer coefficient increases by a factor of 2–2.5* or higher, it produces a much stronger effect on the proceeding of different heat-power processes, which makes it possible to intensify heat flows, decrease the temperature inhomogeneity in the structure, stabilize combustion, and

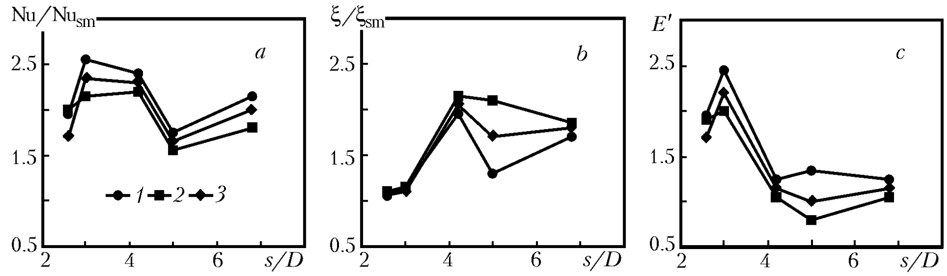

Fig. 3. Influence of the relative step of the twisted tape on the heat exchange (a), the hydraulic resistance (b), and the energy efficiency (c) of the working part: 1) Re = 2800; 2) 3200; 3) 3560.

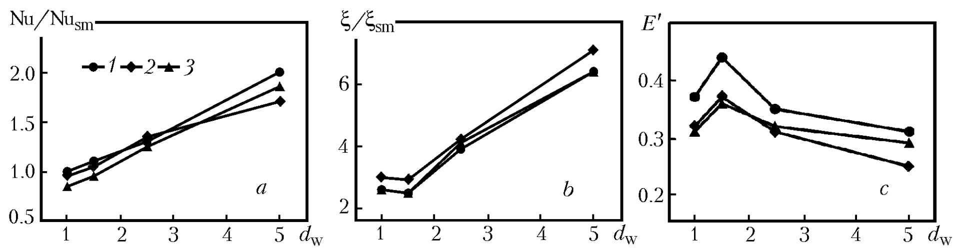

Fig. 4. Influence of the diameter of the wire swirler rod on the heat transfer (a), the hydraulic resistance (b), and the energy efficiency (c) of the working part: 1) Re = 4480; 2) 5000; 3) 5700. d_w, mm.

decrease the drift of heating surfaces. Active investigations on these lines** point to its importance and high demand and to the practical significance of the results obtained. It should be noted that along with the new methods of vortex intensification with $\overline{E}' > 1$ [10] by "building-in" self-organizing swirling jets in the turbulent flow (the so-called tornado technologies), the classical variants of creating vortex structures by forming a vortex*** in the flow on a periodic transverse roughness of certain (optimal) parameters continue to be developed. Exciting detached vortex motion with jet effects in the near-wall region, periodic roughness elements are the main reason for the deviation towards higher values of the thermohydrodynamic efficiency $\overline{E}' > 1$ [6], which finds a proper calculation-theoretical justification [10, 11]. However, the multivariance of the system, the poor ability to simulate phenomena, as well as the difficulties of diagnosing small-scale phenomena and the fragmentary character of investigations, with understandable, in general, tendencies, require an individual approach in solving concrete problems.****

The efficiency $\overline{E}'$ of such an intensifier depends on the step s/D and the kind of liquid. When air is passed in pipes with small knurling steps (s/h = 10–15, angle of attack $\varphi \geq 50$–60°), the value of the energy efficiency is close (with some excess) to the value at transverse roughness [12]. Stretched spirals and water streams produce a more clearly defined swirling effect, and the thermohydrodynamic efficiency will decrease ($\overline{E}' < 1$).

Investigation of the Efficiency of Swirlers in the Laminar-Turbulent Transition Region. We carried out investigations with four types of intensifiers under nonisothermal conditions by the method described in [13]. The ther-

*As a rare exception to the general tendency ($Nu/Nu_{sm} > 1$), we can mention the situation with the condensation on knurling pipes investigated in [9], as well as in [10].

**Which is evidenced, in particular, by the topics of the recent conferences on heat transfer (Russian national conference on heat transfer-1, Russian national conference on heat transfer-3).

***Variants of heat transfer intensification by the swirling proper — the creation of helical motion of the whole mass of the liquid or a part of it with different forms of secondary flows (reverse, cross flows) in a cyclone, a screw, a helical coil, etc. are usually low-efficiency ones ($\overline{E}' < 1$).

****The discrepancy between the experimental data for the heat transfer and hydraulic resistance obtained by different authors sometimes reaches 100% and more [4, 6].

mal and hydrodynamical results were published in [13, 14] and are presented in Table 1. Preliminary choice of optimal geometric sizes of intensifiers was carried out by the standard method of a planned experiment. The data obtained reflect well the general picture of the thermohydraylic efficiency of intensifiers complementing the already known values in the little-studied range of Reynolds numbers Re_D = 2000–6000 (Fig. 2).

Table 2 and Figures 3 and 4 show the dependences of the thermohydraulic efficiency of the tested swirlers obtained in experimental data processing. The highest energy efficiency ($\overline{E}'_{max}$ = 1.94–2.36) at a moderate thermal efficiency (Nu/Nu_{sm} = 2.17–2.56) was obtained for thin twisted tapes at large angles of attack $\varphi > 500$ (δ = 0.33 mm, s/D = 3). The thermal efficiency of the wire inserts is lower throughout the range of rod diameters d_w at a high and rapidly increasing hydraulic resistance. Propeller inserts and screws give a maximum increment of the heat transfer Nu/Nu_{sm} = 3.17–4.36) and an order of magnitude higher increment of the hydraulic resistance (ξ/ξ_{sm} = 4.88–62.10).

It is expedient to use the considered methods of heat transfer (flow swirling) intensification in low-power power plants since they are simple, cheap, and permit improving the technical-and-economic indices of heat exchangers.

NOTATION

D, internal diameter of the pipe; d_w, diameter of the rod of the wire swirler; d_e, equivalent diameter; d_0, internal rod diameter; E', energy efficiency; $\overline{E}'$, relative universal energy efficiency criterion; h, height of transverse roughness projections; l, screw length; Nu, Nusselt number; n, exponent; P, fluid pressure; Pr, Prandtl number; Re, Reynolds number; s, step; δ, thickness of the tape swirler tape; ξ, hydraulic resistance coefficient; φ, angle of inclination of the screw. Subscripts: max, maximum; sm, smooth pipe; e, equivalent; w, wire; m, medium.

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