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EXPERIMENTAL INVESTIGATION OF INFLUENCE OF TURBULENCE OF FLOW ON HEAT EXCHANGE DURING MOTION OF AIR IN PIPES

By

I. T. Shvets, Ye. P. Dyban, M. V. Stradomskiy, and E. Ya. Epik

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ABSTRACT: Experiments were conducted on study of heat exchange during motion of air in relatively short pipes in range of Reynolds numbers ($1 \cdot 10^3$ to $5 \cdot 10^4$) with consideration of initial turbulence of flow. Experimental material was a segment of EYalT steel seamless pipe with internal diameter 51mm, wall thickness 1.5mm, and 4,000mm length. The installation used for experiment is shown in Fig. 1. Conditions for experiment are described in detail. Results of experiments were processed to obtain dependence of average and local Nusselt number on Reynolds number at different sections of pipe for different magnitudes of Karman number. English translation; 18 pages.

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U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

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Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

EXPERIMENTAL INVESTIGATION OF INFLUENCE OF TURBULENCE OF FLOW ON HEAT EXCHANGE DURING MOTION OF AIR IN PIPES

I. T. Shvets, Ye. P. Dyban, M. V. Stradomskiy, and E. Ya. Epik

To investigation of heat transfer and friction during motion of liquids in pipes have been dedicated a large number of experimental and theoretical works [1-3].

It has been established that the magnitude of local coefficient of heat transfer monotonically changes along length of pipe, with maximum value in entrance section and minimum constant value in sections located at 30-50 diameters from the entrance. Change of magnitude of average coefficient of heat transfer along length of pipe has analogous character.

Results of numerous investigations made it possible to determine for turbulent and laminar regions of flow in pipes criterial dependences for calculation of heat transfer after the stabilization section. For initial segments of pipe, determination of magnitude of coefficient of heat transfer is produced by means of multiplication of value of α corresponding to a long pipe by a correction factor (larger than unity) depending on distance to the entrance (for local) or on length of pipe (for average heat transfer).

Magnitude of correction factor was determined by all authors by experimental means, as a function of Reynolds number and length of pipe for turbulent region of flow and length of pipe for the laminar region [2]. For Reynolds numbers corresponding to transition from laminar flow to turbulent flow, reliable data on regularity of heat exchange in initial segments of pipe are practically non-existent.

During experimental determination of correction factor for length of pipe, there were usually taken measures to decrease initial disturbances introduced into the flow by local resistances before the entrance into the pipes and also conditions of the entrance. It is in our opinion absolutely impermissible to disregard the indicated circumstances.

Considering what has been presented by us, there were conducted experimental works on the study of heat exchange during motion of air in relatively short pipes in range of Reynolds numbers ($1 \cdot 10^3$ to $5 \cdot 10^4$) with consideration of initial turbulence of flow.

The Experimental Installation and Method of Investigation

A schematic diagram of the installation is shown in Fig. 1. Experiments were conducted on a segment of seamless pipe of steel [EYa1T] (ЭЯ1Т) with inner diameter of 51 mm, thickness of wall 1.5 mm and length 4000 mm (maximum ratio of length to diameter about 80).

Supply of air to experimental section was carried out through system of receivers and a damper with metal grids and cloth filter installed in it. Entrance of air into pipe was realized through a convergent channel, which provides at the output a smooth velocity profile and has an outlet section with diameter equal to inner diameter of pipe. At outlet from convergent channel there was fixed an insert with inner diameter also of 51 mm, which provides possibility of

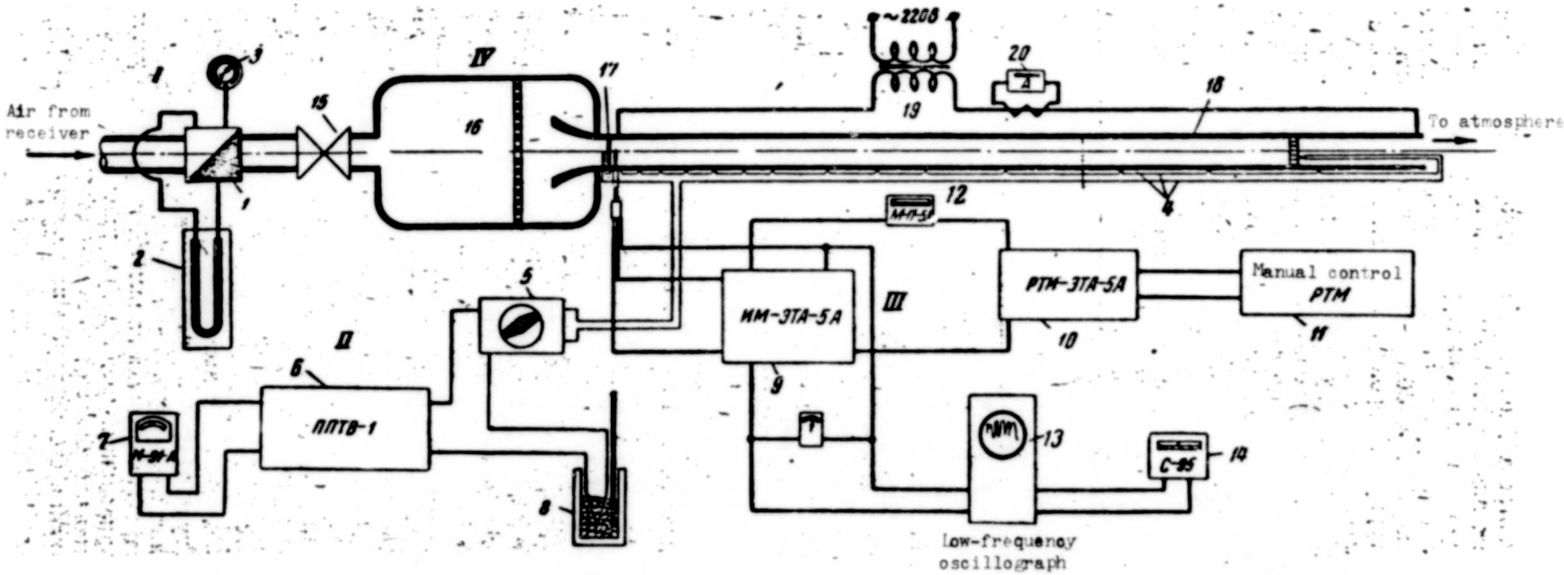

Fig. 1. Schematic diagram of experimental installation for investigation of heat exchange during flow of air in relatively short pipes. I — equipment for measurement of flow rate of air; 1 — double diaphragm; 2 — water differential manometer; 3 — standard spring-type manometer; II — equipment for measurement of temperature; 4 — chromel-copel thermocouples; 5 — switch; 6 — laboratory potentiometer of type [PPTV-1] ППТВ-1; 7 — null galvanometer; 8 — auxiliary thermocouple, measuring temperature of cold junction of basic thermocouples. III — equipment for measurement of velocity pulsations: 9 — measuring bridge [IM ETA-5A] ИМ ЭТА-5А; 10 — current regulator of bridge, [RTM ETA-5A] РТМ ЭТА-5А; 11 — stabilized manual-control rectifier [RTM] РТМ; 12 — milliammeter of type [M-11-5A] М-11-5А 13 — broad-band amplifier with filter; 14 — electrostatic voltmeter [S-95] С-95. IV — experimental installation; 15 — air flow rate regulation valve; 16 — flow damper; 17 — removable turbulence generator; 18 — experimental pipe; 19 — step-down transformer; 20 — ammeter with current transformer.

placement of artificial turbulence generators directly at the entrance to the investigated segment of the pipe.

As turbulence-generating devices there were applied perforated plates with thickness of 3 mm and wire grids (geometric characteristics of which are given in Table 1), which provide, as was established earlier the greatest increase of level of axial pulsations of velocity [4].

Table 1. Geometric Characteristics of Investigated Turbulizers and Magnitude of Radial Components of Pulsation of Velocity Created by Them

Diameter of openings d, mm	Step I, mm	Obstruction coefficient ϕ	Designation of turbulizers	Magnitude of Karman number $Re(1-5)\cdot 10^4$, %	
				Cross section I	Cross section II
10	—	0.0384	1 holes Ø 10	3—4.5	4.2—5.6
10	20.80	0.1806	VII Ø 10 к	3.5—5.8	1.5—3.4
5	10.40	0.1806	VII Ø 5 к	8—9	2.3—3.7
3	6.25	0.1806	VII Ø 3 к	5.6—6.8	2.7—3.6
3	4.92	0.2916	VI Ø 3 к	4—6.4	1.6—3.4
2.3	4.00	0.1806	II from VII	1.5	1.2—1.8
51	—	1.00	Natural turbulization	<0.2	<0.2

Note: 1. Turbulizer No. 1 — diaphragm with one central hole.

2. Turbulizers No. 2-5 — perforated diaphragms with unstaggered holes.

3. Turbulizer No. 7 — insert providing smooth entrance into pipe (natural turbulization of flow).

Experiments were conducted with heating of walls of pipe by alternating electric current with voltage from 1.5 to 6.2 v at current of 400-450 amp.

Feed of current to pipe was carried out through contacts made of thin stainless steel sheet wrapped around the pipe. For preventing of local overheating of contacts, their surface was covered with copper foil of thickness 0.3-0.5 mm. Leakage of current through the flange and contacts was prevented by compensating adjustable Nichrome heaters located on them and fed through autotransformers of type [LATR-1] (ЛАТР-1).

Quantity of air proceeding into investigated segment was measured with help of double diaphragm fixed before the flow damper. Pressure drop was measured by a U-shaped water differential manometer, and absolute pressure was measured by a laboratory spring-type manometer of class 0.35. Air pressure in pipe was taken equal to atmospheric pressure determined by barometer.

Determination of temperatures of air and pipe wall was carried out by means of chromel-copel thermocouples; emf of thermocouples was measured by null method with the help of potentiometer of type [PPTV-1] (ППТВ-1). Thermocouples measuring temperature of wall of pipe were welded to it and imbedded flush with the surface; thermocouple measuring initial temperature air was installed in flow damper before entrance section of the convergent channel.

Measurement of temperature of air at outlet from working segment was carried out by a perforated copper diaphragm with diameter of 47 mm, thickness of 9 mm, having a large number of holes with diameter of 2 mm. For removal of influence of walls of pipe on temperature of diaphragm, the latter is thoroughly insulated around its outer diameter by a layer about (2 mm) of asbestos insulation impregnated with liquid glass.

The diaphragm, fixed inside pipe perpendicularly to flow and possessing very high coefficient of thermal conductivity takes the average-flow temperature of the incident flow [5], which is measured by a chromel-copel thermocouple calked in it. Displacement of diaphragm along length of pipe was carried out from the outlet section with the help of a set of thin metallic rods connected with each other.

During experiment, at constant flow rate of air coolant and constant current intensity, there was measured temperature of wall of pipe at 17 points along the length and also distribution of temperature of air along length of pipe.

The obtained distribution temperature curves of wall and flow along length of pipe were used for determination of local and mean values of coefficients of heat transfer.

Quantity of heat absorbed by air was determined according to the enthalpy difference between outlet and inlet sections of pipe:

$$Q = Gc_p \cdot \Delta t_B, \tag{1}$$

where G – flow rate of air per hour; Δt_B – heating of air in the given segment.

During calculation of average value of coefficient of heat transfer for the segment, temperature head was determined as difference of mean-integral values of temperatures of wall and air, where integration of distributions of temperature was carried by the trapezoidal rule:

$$\Delta t_{cp} = \frac{\sum_{0}^{n} (t_{n-1}^{ct} + t_{n}^{ct}) \cdot (l_n - l_{n-1}) - \sum_{0}^{n} (t_{n-1}^{B} + t_{n}^{B}) \cdot (l_n - l_{n-1})}{2l_n}. \tag{2}$$

For finding of local values of coefficient of heat transfer, heat-exchange surface was taken equal to one diameter and located uniformly on both sides of the considered section of the pipe.

Temperature head in this case was calculated as the difference between temperatures of wall and air in the given section.

Physical constants of air for Reynolds and Nusselt numbers were determined according to its mean-flow-rate temperature over the section; the characteristic dimension was considered to be the inner diameter of the pipe and the velocity was considered to be velocity in the given section.

In process of carrying out the investigation, there were conducted nine series of experiments (eight with the use of turbulizers and one with natural turbulization of flow), in each of which there were performed 30-35 experiments at different Reynolds numbers. Error in determination of Nusselt number was 5% and in Reynolds number — 2%.

Evaluation of turbulization of flow in given section was carried out according to the meacured distribution of radial component of pulsation of velocity along radius with the help of an electro-thermo-anemometer of type [ETA-5a] (ЭТА-5а). Method of measurement of radial component of pulsation of velocity and results of experiments are presented in article [7].

Results of Experiments

Results of experiments were processed to obtain a dependence of average and local Nusselt number on Reynolds number at different sections of the pipe for different magnitudes of Karman number.

During heating of pipe by transmission of electrical current, change of quanti :y of heat of released in wall along its length is proportional to change of specific electrical resistance of the material and constitutes approximately 2%. Therefore, change of difference $t^{CT} - t^{B}$ along length of pipe qualitatively characterizes changes of local values of coefficient of heat transfer. This made it possible to use for analysis also distribution curves of wall temperature. Consideration of curves of change of wall temperatures and air along the length of the pipe in case of natural turbulization of flow (Karman number in entrance section is of the order of 0.2%) has shown that monotonic change of value of local coefficient of heat transfer occurs only at values of Reynolds number lower than 2000 and higher than 30,000.

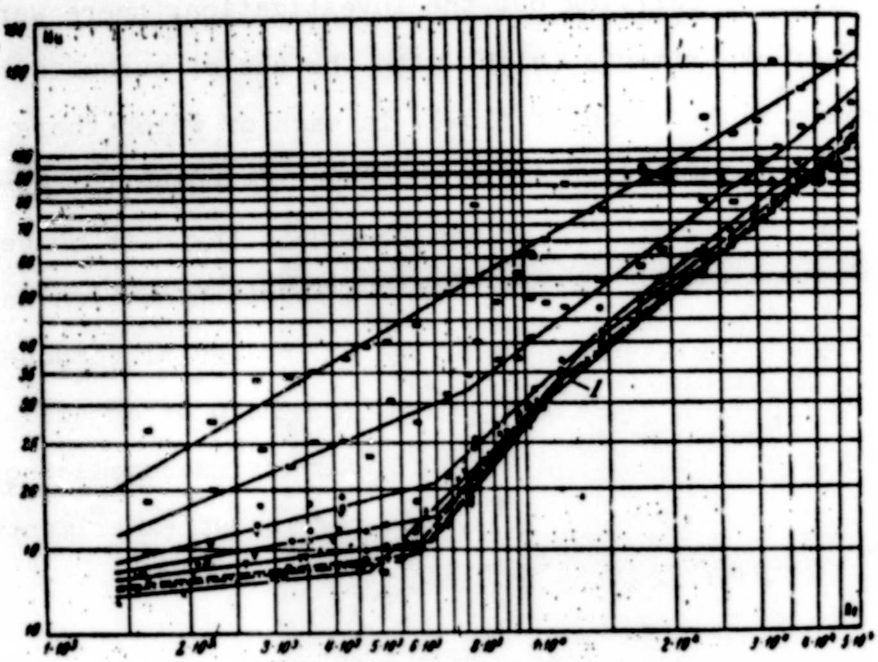

Fig. 2. Local heat transfer during natural turbulization of flow for segments located at the following distances from the entrance section: ▭ — 2d; ■ — 5d; ○ — 10d; △ — 15d; ∨ — 20d; ▼ — 30d; × — 40d; ● — 75d; I — $Nu = 0.018 \cdot Re^{0.80}$.

As processing of experimental data has shown (Fig. 2), during natural turbulization of flow, intensity of local heat transfer for segments from $l/d = 10$ to $l/d = 40$ is described by three different criterial relationships:

a) at Reynolds number lower than 3500, the exponent of Reynolds number is of the order of 0.1;

b) in range of Reynolds numbers from 3500 to 20,000 with exponent of the order of 1.2-1.0;

c) in range of Reynolds numbers higher than 20,000, with exponent close to 0.80.

Near the entrance (segment l/d from 0.5 to 5), exponent of Reynolds numbers is 0.5, and the criterial equation after recalculation acquires form with error not more than 10%, which coincides with equation for heat transfer from a flat plate:

$$Nu = 0,66 Re^{0,5}. \qquad (3)$$

For sections located more than 50 diameters from the entrance, regularities of heat transfer obey the following two equations (Fig. 2):

a) in range of Reynolds numbers over 3500 — equation characteristic for turbulent regime of flow:

$$Nu = 0,018 Re^{0,80}; \qquad (4)$$

b) at value of Reynolds number lower than 3500 — equation characteristic for laminar regime of flow:

$$Nu = 5,6 Re^{0,10}. \qquad (5)$$

It is necessary to note that disappearance of third segment with increase of distance from entrance or Reynolds number corresponds to appearance of a monotonic distribution curve of temperature of wall of pipe.

On the basis of the conducted experiments, it is possible to make the conclusion that the segment of pipe occupied by zone with transient regime of flow has relatively small extent, and at a distance from the entrance of over 50 diameters transition from laminar regime of flow to turbulent regime occurs directly (Fig. 3). The exponent of Reynolds number for initial segments of pipe depends not only on regime of flow but also on distance from the entrance, and in the case of natural turbulization is changed from 0.5 to 0.1 during laminar and from 0.8 to 1.5 during transient regimes.

The indicated character of change of intensity of heat transfer along length of pipe during natural turbulization of flow also renders an influence on average Nusselt numbers (Fig. 4).

In the conducted experiments, lengths of applied pipe (75 diameters) turned out to be insufficient for achievement of transition from laminar flow to turbulent flow: according to Reynolds number, intensity of average heat transfer has three zones. For length of pipe over 40 diameters, in range of Reynolds numbers over $1 \cdot 10^4$, criterial equation for heat transfer has the form $Nu = 0.018 Re^{0.80}$; in the transition zone, the exponent of Reynolds number changes with increase of length of pipe not monotonically: in the beginning it increases (to 1.25) and then starts to decrease (to 0.90).

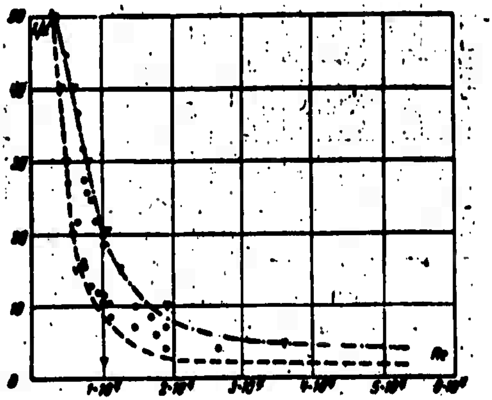

Fig. 3. Influence of Reynolds number on extent of segments with laminar and transient regimes of flow during natural turbulization of flow. — — — end of laminar regime of flow; —·—·— beginning of developed turbulent regime of flow; .—. through extreme points on temperature distribution curves of wall of pipe along its length; ▼-▼ through points of break on curves Nu = f(Re).

From consideration of Fig. 2-4 one may see that with increase of Reynolds number, influence of length of pipe on local and average heat transfer considerably decreases.

Calculations show that at Reynolds number above $1.2 \cdot 10^5$ local heat transfer directly from the entrance will be described by the equation which is characteristic for turbulent regime of flow; for average heat transfer, for a length of pipe of 30 diameters, this phenomenon will occur at Reynolds number of $3.1 \cdot 10^5$, and for a pipe with length of 5 diameters — at Reynolds number $7 \cdot 10^6$.

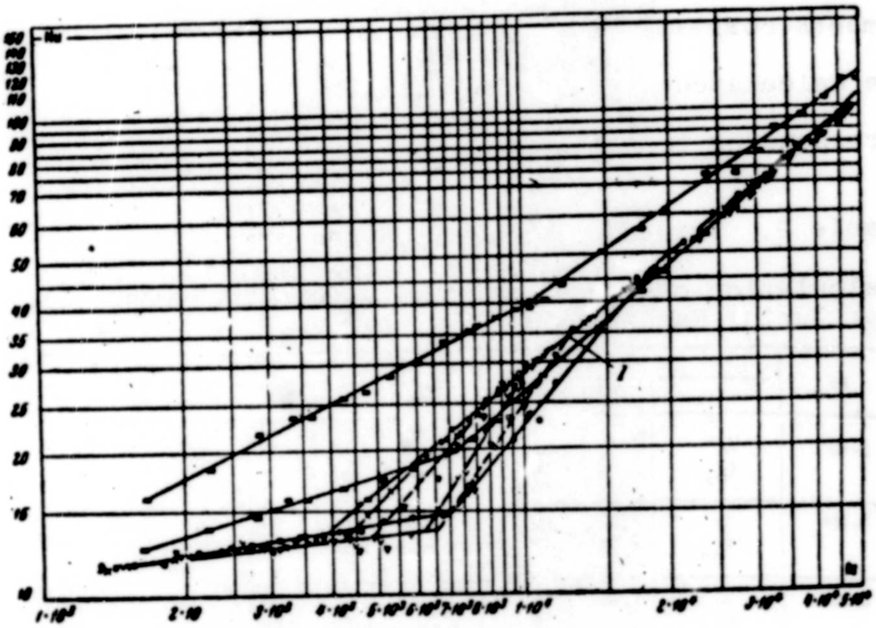

Fig. 4. Average heat transfer during natural turbulization of flow for segments of pipe of length: ▭ — 2d; ■ — 5d; ○ — 10d; ● — 15d; ∨ — 20d; ▽ — 30d; ▼ — 40d; × — 75d; I — $Nu = 0.018Re^{0.80}$.

Analysis of results of experiments made it possible to establish approximate empirical dependence for determination of length of segment of stabilization of local values of Nusselt number in range of Reynolds numbers $1.5 \cdot 10^4$ to $5 \cdot 10^4$:

$$l/d = 25 (Re \cdot 10^{-4})^{-1.3}. \qquad (6)$$

Length of segment of stabilization for mean values of Nusselt number in the same ranges of Reynolds numbers is determined by the equation

$$l/d = 42 (Re \cdot 10^{-4})^{-0.4}. \qquad (7)$$

On the basis of experiments there were determined magnitudes of corrections for consideration of influence of length of pipe on intensity of local and average heat transfer during natural turbulization of flow (Table 2). During determination of magnitude of correction factor, magnitude experimentally determined Nusselt number pertained to the value calculated by means of equation (4) in range of Reynolds numbers corresponding to turbulent and transient regimes, and to the value calculated by means of equation (5) in range of Reynolds numbers corresponding to the laminar regime (Table 2).

Table 2. Values of Correction Factors ε_l; $\varepsilon_{l_{cp}}$ for Calculation of Local and Mean Values of Nusselt Number During Natural Turbulization of Flow

Re		l/d = 2	5	10	15	20	30	40	50	75
$2\cdot10^3$	ε_l	1,49	1,11	1,02	1,00	1,00	1,00	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	2,06	1,56	1,26	1,18	1,13	1,08	1,08	1,05	1,02
$3\cdot10^3$	ε_l	1,76	1,22	1,03	1,00	1,00	1,00	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	2,50	1,77	1,35	1,23	1,15	1,08	1,08	1,05	1,02
$5\cdot10^3$	ε_l	1,73	1,09	0,84	0,80	0,80	0,82	0,95	1,00	1,00
	$\varepsilon_{l_{cp}}$	2,55	1,36	1,18	1,03	0,93	0,87	0,86	0,84	0,86
$7\cdot10^3$	ε_l	1,56	0,93	0,68	0,71	0,77	0,94	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	2,36	1,46	1,04	0,90	0,82	0,83	0,87	0,90	0,93
$1\cdot10^4$	ε_l	1,40	0,92	0,77	0,89	1,00	1,00	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	2,18	1,39	1,06	0,99	0,95	0,94	0,96	0,98	1,00
$1,5\cdot10^4$	ε_l	1,33	0,94	0,91	1,00	1,00	1,00	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	1,99	1,36	1,14	1,07	1,05	1,01	1,00	1,00	1,00
$2\cdot10^4$	ε_l	1,28	0,96	1,00	1,00	1,00	1,00	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	1,86	1,34	1,13	1,06	1,04	1,01	1,00	1,00	1,00
$3\cdot10^4$	ε_l	1,21	0,99	1,00	1,00	1,00	1,00	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	1,70	1,31	1,12	1,05	1,02	1,00	1,00	1,00	1,00
$5\cdot10^4$	ε_l	1,14	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
	$\varepsilon_{l_{cp}}$	1,51	1,28	1,11	1,04	1,01	1,00	1,00	1,00	1,00

With such processing of data, influence of change of physical properties of liquids along length of pipe and conditions of experiment on value of correction factor decreases. In particular it turns out to be possible to compare experimental data obtained under different conditions of the entrance.*

Increase of level of initial turbulization of flow considerably changes both the magnitudes of coefficients of heat transfer in initial segments of the pipe and the character of their distribution along the length.

With increase of Karman number in entrance section to 4% of magnitude of local values, Nusselt numbers increase by 7-8 times: section occupied by transient regime of flow vanishes, and in sections located more than 20 diameters from the entrance, starting at Reynolds number of 2800, local heat transfer is described by equation (4), which is characteristic for a well-developed turbulent regime of flow (Fig. 5).

Due to growth of intensity of heat exchange in inlet sections, length of segment of stabilization for mean values of Nusselt number is increased. At Reynolds numbers of the order of $5 \cdot 10^4$, correction factor for length of pipe $\varepsilon_{l_{cp}}$ becomes equal to unity in approximately 200 diameters. In an artificially turbulized flow, for all investigated lengths of pipe in the region of Reynolds numbers $4 \cdot 10^3 - 1 \cdot 10^4$, the exponent of Reynolds number turns out to be lower than 0.80, and not higher, as for a naturally turbulized flow.

Monotonic change of the exponent of Reynolds number during change of its magnitude is also characteristic for artificially turbulized flow (Fig. 6).

For the most effective of the investigated turbulizers (No. 1 according to Table 1), local heat transfer at inlet sections was increased by 5-15 times (Table 3) with increase of average Nusselt

*During natural turbulization of flow experimental data obtained by us coincided well qualitatively with results of investigation [6] conducted on a viscous liquid (oil).

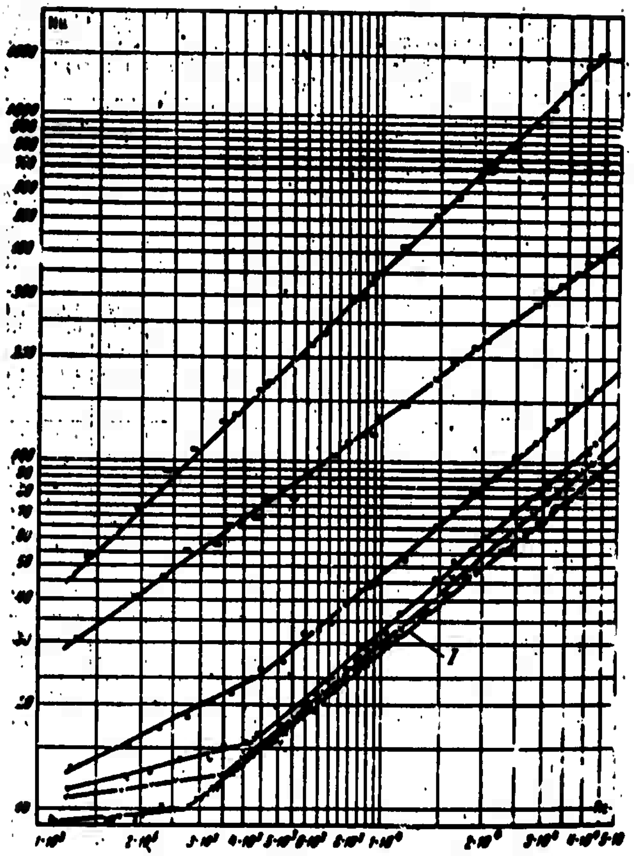

Fig. 5. Local heat transfer during artificial turbulization of flow (turbulizer No. 1, holes Ø 10) for segments located the following distances from the inlet section: ▭ — 2d; ■ — 5d; ○ — 10d; ● — 15d; ∨ — 20d; ∇ — 30d; ▼ — 40d; × — 75d; I — $Nu = 0.018Re^{0.80}$.

number for a pipe with length of 75 diameters by 20-30%. With decrease of level of turbulization of flow, intensification of heat exchange on initial segments of the pipe changes, and for turbulizer No. 6 (wire grid) values of correction factors approach magnitudes characteristic for naturally turbulized flows.

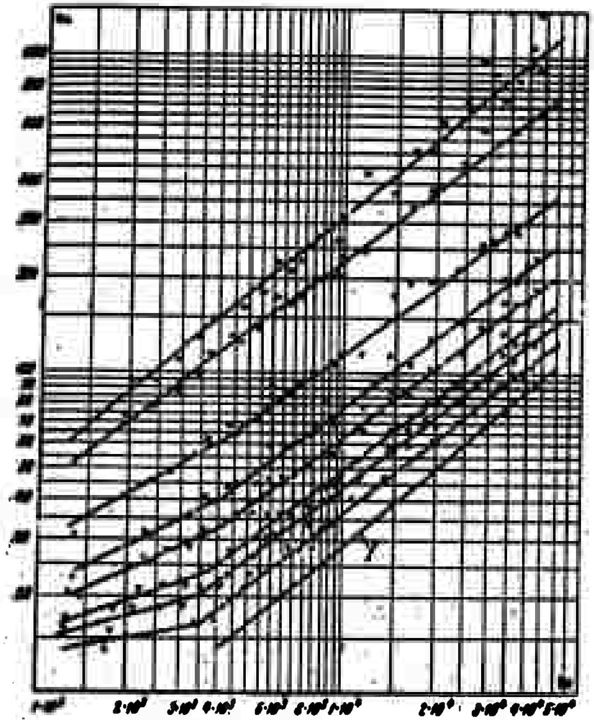

Fig. 6. Average heat transfer during artificial turbulization of flow (turbulizer No. 1, holes Ø 10) for segments of pipe with the following lengths: ▭ — 2d; ■ — 5d; ○ — 10d; ● — 15d; ∨ — 20d; ▽ — 30d; ▼ — 40d; × — 75d; I — $Nu = 0.018Re^{0.80}$.

In the process of carrying out the work, it was not possible to establish a quantitative relation between level of turbilization of flow (Karman number) and intensification of heat exchange in the pipe.

As experiments showed, in cross sections of pipe located more than 30 diameters from the inlet, distribution of radial components of pulsation of velocity turns out to be identical both during natural and during artificial turbulization. As it was shown earlier, at approximately the same distance there is stabilized the level of local heat transfer.

In cross sections located closer than 30 diameters from the inlet character of distribution of radial component of pulsation along the radius turns out to be different for different turbulizers, and there is absent qualitative agreement between mangitude of Karman number and intensity of local heat transfer.

The conducted experiments have shown that frequency of pulsations of velocity created by different turbulizers turns out to be essentially different due to the fact that apparently it was not possible to obtain generalized dependences for calculation taking into account the level of turbulization of flow on heat exchange during flow of air in pipes.

Table 3. Values of correction factors ε_l and $\varepsilon_{l_{cp}}$ for Calculation of Local and Mean Values of Nusselt Numbers During Intense Artificial Turbulization of Flow (Turbulizer No. 1, Holes Ø 10)

Re		l/d								
		3	5	10	15	20	30	40	50	70
$2\cdot10^4$	ε_l $\varepsilon_{l_{cp}}$	5,98 7,42	3,43 5,82	1,36 3,46	1,07 2,36	0,97 2.01	0,82 1,63	0,82 1,45	0,82 1,36	0,82 1.24
$3\cdot10^4$	ε_l $\varepsilon_{l_{cp}}$	8,57 9,83	4,43 7,53	1,59 4,18	1,14 2,78	0,98 2,30	0,87 1,80	0,87 1,56	0,87 1,44	0,87 1,27
$5\cdot10^4$	ε_l $\varepsilon_{l_{cp}}$	10,81 11,21	4,89 8,34	1,62 4,31	1,11 2,82	1,03 2,32	1,00 1,83	1,00 1,60	1,00 1,46	1,00 1,29
$7\cdot10^4$	ε_l $\varepsilon_{l_{cp}}$	11,53 11,20	4,78 8,16	1,64 4,19	1,14 2.77	1,04 2,27	1,00 1.79	1,00 1,57	1,00 1,44	1,00 1.27
$1\cdot10^5$	ε_l $\varepsilon_{l_{cp}}$	12,34 11,17	4,66 7,97	1,66 4,06	1,16 2,71	1,06 2,21	1,00 1.75	1,00 1,54	1,00 1,41	1,00 1,25
$1,5\cdot10^5$	ε_l $\varepsilon_{l_{cp}}$	13,32 11,14	4,54 7,76	1,69 3,91	1,20 2,64	1,08 2,16	1,00 1,71	1,00 1,50	1,00 1,40	1,00 1,23
$2\cdot10^5$	ε_l $\varepsilon_{l_{cp}}$	14,08 11,13	4,44 7,62	1,71 3,82	1,22 2,59	1,09 2,11	1,00 1,67	1,00 1,47	1,00 1,36	1,00 1,22
$5\cdot10^5$	ε_l $\varepsilon_{l_{cp}}$	16,75 10,79	4,17 7,18	1,82 3,52	1,30 2,45	1,13 1,99	1,00 1,58	1,00 1,39	1,00 1,29	1,00 1,17

The conducted investigation showed that use of Karman number is insufficient to obtain total characteristic of degree of turbulence of flow; it is necessary, apparently, during experiments, to measure also a magnitude proportional to the mixing length or dimension of artificial vortex, for example, frequency of pulsations.

Conclusions

1. During naturally turbulized flow in a pipe, in a short (less than 5 diameters) segment, regularities of heat transfer coincide with those observed during flow around a plate.

In the range of Reynolds numbers from 2500 to 45,000, after the short segment there is a zone with length of 50 diameters, occupied with a transient regime of flow, after which there is a region with turbulent regime of flow.

2. With increase of initial perturbations, intensity of local and average heat transfer is increased, and length of pipe after which heat transfer is stabilized also increases. Certain of the turbulizing devices investigated in this work provided increase of initial turbulence to 8-9% (as compared to 0.2% with natural turbulization), which caused increase of local values of Nusselt numbers at inlet sections by 14-16 times, and of average values by 10-12 times; thus, in spite of fast damping of introduced perturbations along length of the pipe, average heat transfer is intensified at very considerable distances (on the order of 100-200 diameters).

3. Experiments have shown that initial turbulization of flow renders a very significant influehce on intensity of heat exchange during flow of air in short pipes and must be taken into account during determination of corresponding correction factors.

Obtained experimental data permit us to make a conclusion concerning the fact that installation at the inlet into pipes of artificial turbulizers is one of the ways of intensification of heat exchange.

4. In order to obtain generalized relationships describing law of heat transfer in pipes, taking into account initial turbulization of the flow, it is necessary, in the first place to find the parameters which most fully characterize artificial turbulization. Use of Karman number as such a parameter (ratio of mean-square magnitude of radial component of velocity pulsation to its average-flow-rate value) is, as results of the investigation showed evidently insufficient.

Literature

1. Contemporary state of hydro-aerodynamics of a viscous fluid. IL, 1948.

2. M. A. Mikheyev. Fundamentals of heat transfer. State power engineering publishing house, 1959.

3. G. Greber, S. Erk, and U. Grigul'. Fundamentals of the study of heat exchange. IL, 1960.

4. I. T. Shvets, Ye. P. Dyban, G. F. Selyavin, and M. V. Stradomskiy. Experimental investigation of influence of initial perturbations on development of turbulent regime of flow during motion of air in pipes. News of higher educational institutions, "Power engineering," No. 8, 1960.

5. Nunner Wolfgang, Wärmeübergang und Druckabfall in rauhen Röhren, VDI-Forschungsheft 455, Ausgabe B, Bd 22, 1956.

6. B. S. Petukhov and Ye. A. Krasnoshchekov. Heat exchange in the initial section of a pipe with displaced boundary layer. Collection "Heat transfer and modeling." Publishing House of the Sciences of the USSR, 1959.

7. I. T. Shvets, Ye. P. Dyban, M. V. Stradomskiy, and E. Ya. Epik. Investigation of radial component of pulsation of velocity during turbulent flow of air in relatively short pipes with different level of initial perturbations. IFZh, 1961, No. 9.

Institute of Heat-Power Engineering
of Academy of Sciences of Ukrainian
SSR. Kiev State University.