INFLUENCE OF HYDRODYNAMIC REGIMES OF OIL AND GAS MIXTURES ON THE EFFICIENCY OF HEAT EXCHANGE

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Abstract. The results of experimental studies identify the degree of intensification of the processes of oil and gas mixtures heating in horizontal pipe due to changes in hydrodynamic regimes. So, at oil and gas mixtures heating processes it is desirable to consider their implementation in a turbulent flow, which will help to develop energy-saving technology of thermal preparation of hydrocarbons to the primary distillation in refineries by optimizing the hydrodynamic conditions in tubular heat exchangers.

Key words: physical and chemical properties of oil and gas mixture, similarity criteria, calorific efficiency, heat transfer coefficient, the amount of transferred heat, degree of process intensification, heat exchange efficiency.

Introduction

The problem of rational and efficient use of energy resources is one of the most important. With increasing of energy capacity and output more and more weight and dimensions of heat exchangers use are increasing. For their operation a huge amount of electricity and heat is in use. Improving the efficiency of heat transfer and reducing the weight and dimensions of these devices depends on the rational organization in their hydrodynamic regimes of heat exchanging traffic flows. Horizontal pipe and tubular devices are characterized with the fact that they can organize any hydrodynamic regimes of flow: laminar ($Re < 2\,300$), transitional ($2\,300 < Re < 10\,000$) and turbulent ($Re > 10\,000$), which differ with varying intensity of hydro, thermal and mass transfer processes [6].

At refineries of the Uzbekistan a hydrocarbon feedstock oil-gas mixture is used. The share of gas condensate in the mixture varies in the redistribution of 20 to 80 %.

Objects and methods

To study the heating of hydrocarbons (oil, gas condensate, and mixtures thereof) with vapors of gas condensate and to establish the influence of operating parameters of the efficiency of heat transfer in tubular heat exchangers, we collected the experimental setup the concept of which is shown at Fig. 1. The experimental setup consists of steam generator *15*, the heat exchanger of the "tube in tube" with pipes *8* and *9* are mounted on the supporting stands *16*, meters for metering gas *13* and hydrocarbon feed *4*, the centrifugal pump *2* for pumping the raw material, expenditure feedstock tank *1* and the dipstick *11* collecting the heated feed. The unit is equipped with valves *3* and *12* to control the flow of natural gas and hydrocarbon mercury thermometers *5* and *7* for measuring the temperature of the raw material in the inner tube and the heating medium in the annulus apparatus and gauges *6* for measuring the differential pressure of heating medium in the end portions of the heat exchanger. The heat transfer tube *9* has an inner diameter of 20 mm and effective length 2 000 mm. The housing unit *8* is made of pipes with an inner diameter of 50 mm [4].

To the surface of the inner tube with pitch size 500 mm five pockets with outer sleeve are welded, into which in an oil bath thermometers are laid for temperature measurement control
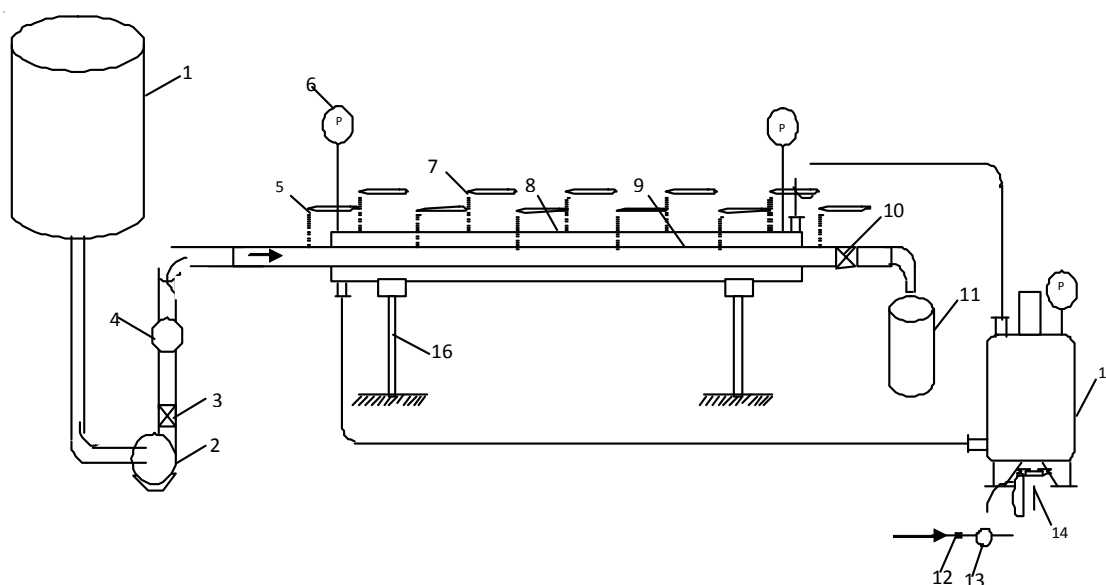

Fig. 1. Experimental setup:

1 – the storage container for hydrocarbon raw materials; *2* – centrifugal pump; *3*, *10* and *12* – valve;
4 – counter flow of raw materials; *5* – thermometers for raw materials; *6* – pressure gauges;
7 – thermometers for heating agent; *8* – outer pipe (casing); *9* – inner tube; *11* – dimensional container heated raw materials;
13 – gas flow meter; *14* – burner gas; *15* – steam generator; *16* – support column

of hydrocarbon feedstock flowing through it. A cylindrical gap formed between the pocket layer and the liner is filled with insulating material. Such an arrangement allows the pocket to minimize the influence of temperature heating medium that is streamlined sleeve for temperature measurement accuracy of the hydrocarbon feedstock in the inner tube in the experiments. To the outer tube with 400 mm step six pockets are welded with oil for bookmark thermometers that measure the temperature of the heating agent in the annulus of the machine. To prevent heat losses to the environment the entire outer surface of the pilot plant and the steam generator is coated with glass wool and sealed with aluminum foil.

In experiments gas condensate vapors were used as a coolant, basic parameters of which are given in Table 1. The experiments were performed at a rate of hydrocarbon feedstock in the inner pipe 0.1061; 0.2123; 0.3184; 0.4246; 0.5307; 0.6369; 0.7431 and 0.8492 m/s. The process was organized in countercurrent movement of heat exchanging flows. The limits of speed variation of hydrocarbons provide the establishment of various hydrodynamic modes of their motion in the experimental apparatus.

Using the results of the experiments, the basic physical, chemical and thermal properties of the investigated materials and coolant were studied and then the heat transfer coefficients of the heating medium to the outer wall of the inner tube and α_1 from the inner wall of the pipe to the heated feedstock α_2 along the length L of the device were calculated.

The criterion equation to calculate the heat transfer coefficient α_1 is selected on the basis on the value of the Reynolds number R_e [2; 3]:

$$R_e = \upsilon (D_{in} - d_{ex}) \rho / \mu, \qquad (1)$$

where $\upsilon = 4V / \pi (D^2_{in} - d^2_{ex})$ – average speed of heating medium (steam condensate) in the annulus system, m/

s; V – volumetric flow rate, m³/s; D_{in} – inner diameter of the housing unit, m; d_{ex} – the outer diameter of the inner tube, m; ρ and μ – respectively, the density (kg / m³) and dynamic viscosity ($Pa \cdot s$) of coolant.

Re meaningfully turbulent motion of vapor condensate in the annulus experienced exchanger was set, as the Reynolds number is in the redistribution of 147.683 ÷ 162.985 (see Table 2-5). In this case, the criterion equation to determine the Nusselt number is:

$$Nu = 0023 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot \left(\frac{D_{in}}{d_{ex}} \right)^{0,45}, \qquad (2)$$

where $Pr = (S\mu) / 1$ – Prandtl number; C and 1, respectively – specific heat (kJ / kg) and the coefficient of thermal conductivity (W/m · K) of the coolant.

According to the calculated values of the criteria Nu heat transfer coefficient α_1 from the heating medium to the outer wall of the heat exchange tube was determined [6]:

$$\alpha_1 = \frac{Nu \cdot \lambda}{d_{eq}}, \qquad (3)$$

where d_{eq} – equivalent diameter of the annulus section of the apparatus; $d_{eq} = d_{in} - d_{ex} = 0.025$ m.

Values of the coefficient of heat transfer from the inner wall of the pipe unit to a heated environment, α_2 are determined depending on the mode of flow in a horizontal pipe.

The value of the Reynolds number, which determines the modes of movement of hydrocarbons in a horizontal pipe system, was determined by the well-known formula $Re_{li} = \frac{\upsilon_{in} \cdot d_{ex}}{\nu_{in}}$, where υ_{in} – speed test liquid, m/s; ν_{in} – the kinematic viscosity, mm²/s.

To calculate the Nusselt number in laminar flow in a horizontal pipe the following criterion equation was used [2]:

Table 1

The main parameters of gas condensate

Heating agent	Pressure, P, kPa	Temperature, t, ⁰C	Velocity, ω, m/s	Density at 20 ⁰C temperature, ρ, kg/m³	Kinematic viscosity at 20 ⁰C temperature, ν, 10^{-6} m²/sec
Vapors of gas condensate	250	120	6.8	759	1.03

$$Nu_{li} = 0,17 Re_{li}^{0,33} Pr_{li}^{0,43} Gr_{li}^{0,1} \left(\frac{Pr_{li}}{Pr_{wa}} \right)^{0,25} \cdot \varepsilon_t, \qquad (4)$$

An approximate calculation of the criterion Nu_{li} when forced movement of the fluid flow in the pipe in the transition mode is performed by the following equation:

$$Nu_{li} = 0,008 Re_{li}^{0,9} Pr_{li}^{0,43}. \qquad (5)$$

Criterion equation of heat transfer for turbulent movement of oil in double-tube apparatus is as follows:

$$Nu_{li} = 0,021 Re_{li}^{0,8} Pr_{li}^{0,43} \left(\frac{Pr_{li}}{Pr_{wa}} \right)^{0,25} \cdot \varepsilon_t, \qquad (6)$$

where $Nu_{li} = d_{in} \cdot \alpha_2 / \lambda$ – is a Nusselt number characterizing the similarity of heat transfer processes at the interface between the wall and the flow of the sample liquid; $Pr_{li} = C \cdot \nu \cdot \rho / \lambda$ – is a Prandtl number at an average temperature of fluid flow; $Gr_{li} = g \cdot d^3 / \nu$ – is a Grashof criterion at an average temperature of fluid flow; $Pr_{wa} = C \cdot \nu \cdot \rho / \lambda$ – is a Prandtl number at the average temperature of the pipe wall; $\varepsilon_t = f(L / d)$ – is a coefficient taking into account the effect of the size of the heat exchange tube to increase the heat transfer coefficient α_2.

Coefficient of heat transfer from the wall to the liquid heat exchange tube α_2 is calculated from the following expression:

$$\alpha_2 = \frac{Nu_{li} \cdot \lambda}{d_{in}}. \qquad (7)$$

The value of heat transfer coefficient K for 1 m length of experienced heat exchanger was defined by the equation set out in the source [3]:

$$K = \frac{1}{\frac{1}{\alpha_1 r_{ex}} + \frac{1}{\alpha_2 r_{in}} + \frac{1}{\lambda} 2,3 lg \frac{r_{ex}}{r_{in}} + \frac{1}{r_{pol\,ex}} + \frac{1}{r_{pol\,in}}}, \qquad (8)$$

where r_{ex} и r_{in} – are outer and inner radius of inner tube, m; $r_{pol\,ex}$ and $r_{pol\,in}$ – are external and internal thermal conductivity of contamination of the inner tube, W/(m²K).

To determine the amount of heat transferred from the coolant to heated liquid the basic heat transfer equation was used:

$$Q = KL\Delta t_{av}, \qquad (9)$$

where K is the heat transfer coefficient, W/(m²K); Δt_{av} – the average temperature difference, ⁰C; L – length of horizontal pipe, m.

The degree of intensification (or increasing of the efficiency of heat exchange) with heating of oil and gas mixtures by changing the hydrodynamic regime is determined from the following relations:

$$i_\alpha = \frac{\alpha_2''}{\alpha_2'}; \quad i_K = \frac{K''}{K'}, \quad i_q = \frac{Q''}{Q'}, \qquad (10)$$

where i_α, i_K and i_q – are the degree of intensification of heat transfer and heat transfer coefficients K, increasing the amount of heat transferred Q due to changes in hydrodynamic regimes; α_2', K' и Q' – accordingly, are heat transfer, heat transfer and the quantity of heat transferred in a laminar flow regime coefficients; α_2'', K'' and Q'' – values of factors transfer and heat transfers, and also quantity of transferable heat – with transitional and turbulent motion of oil and gas mixtures in a horizontal pipe.

Results and discussion

The results of studies on determination of the degree of influence of hydrodynamic conditions on the efficiency of heat irradiation and heat transfer coefficients as well as the amount of heat transferred at oil and gas flow moving in a horizontal pipe were summarized and presented in Tables 2-5.

The data in Table 2 indicate that the regimes of the oil in the inner tube of a double-tube heat exchanger do smoothly transit from laminar (at $\upsilon_{ex} = 0.1061 \times 0.4246$ m/s) to the transition mode ($\upsilon_{ex} > 0.5307$ m/s). However, under the conditions of the experiment the turbulent regime was not formed. In laminar flow heat transfer coefficient of the wall to the heated oil α_2 increases from 420 to 681 W/(m²K), the coefficient of heat transfer K increases from 7.3 to 10.9 W/(m²K), and the amount of transferred heat Q varies in the redistribution of 8.7 to 219 watts. In the transient mode the coefficient of heat transfer from a wall to the heated oil α_2 increases from 681 to 1011 W/(m²K), the coefficient of heat transfer – from 10.9 to 14.9 W/(m²K), and the amount of transferred heat varies at the redistribution of 219.5 to 1 162.9 watts.

Table 2

Improving the efficiency of heat exchange with heating oil due to changes in hydrodynamic regimes

Heat transfer agent (vapors of gas condensate)				Heated medium (oil)						Performance of heat transfer		
Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K	Rate of hydrocarbons raw materials, w, m/s	Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K	K, W/m^{2}K	Δt_{av}, ^{0}C	Q, W
$t_н$	$t_к$					$t_н$	$t_к$						
120	76	165138	4122	3495	0.1061	20	66	615	66.23	420	7.3	0.6	8.69
120	72	163872	4073	3457	0.2123	20	63	1136	84.1	516	8.7	1.2	20.96
120	71	163241	4029	3447	0.3184	20	61	1826	94.2	587	9.7	4.7	91.45
120	67	162581	4012	3428	0.4246	20	56	2021	102.4	681	10.9	10	219.5
120	63	196257	3975	3386	0.5307	20	50	2326	125.4	762	11.9	15.8	378.7
120	58	158364	3962	3378	0.6369	20	42	2456	142.7	976	14.4	23.4	677.5
120	54	156435	3921	3354	0.7431	20	34	2531	153.8	1012	14.8	30.5	905.9
120	50	156421	3892	3298	0.8492	20	23	2623	146.2	1011	14.9	39.3	1162.9

Table 3

Improving the efficiency of heat transfer by heating gas condensate due to changes in hydrodynamic regimes

Heat transfer agent (vapors of gas condensate)				Heated medium (oil)						Performance of heat transfer		
Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K	Rate of hydrocarbons raw materials, w, m/s	Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K	K, W/m^{2}K	Δt_{av}, ^{0}C	Q, W
$t_н$	$t_к$					$t_н$	$t_к$						
120	57	157354	3921	3335	0.1061	20	82	3412	69.52	475	8.1	1.75	28.4
120	53	155231	3901	3321	0.2123	20	80	6913	136.8	889	13.5	2.93	79.1
120	49	152123	3756	3301	0.3184	20	77	9152	161.2	1349	18.1	8.21	297.2
120	45	152872	3834	3127	0.4246	20	72	12894	279.3	1958	22.5	13.5	607.5
120	41	151293	3885	3246	0.5307	20	66	14589	326.8	2258	24.7	19.3	953.4
120	36	150251	3812	3241	0.6369	20	58	16492	365.2	2654	26.9	26.9	1447.2
120	32	143284	3735	3184	0.7431	20	50	17294	397.5	2816	27.5	34.0	1870
120	28	147683	3725.3	3112	0.8492	20	39	18657	439.6	3087	28.5	42.8	2439.6

Table 4

Improving the efficiency of heat transfer by heating a mixture of oil and gas condensate consisting of 20 % oil and 80 % gas condensate, due to changes in hydrodynamic regimes

Heat transfer agent (vapors of gas condensate)				Heated medium (oil)						Performance of heat transfer		
Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K	Rate of hydrocarbons raw materials, w, m/s	Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K	K, W/m^{2}K	Δt_{av}, ^{0}C	Q, W
$t_н$	$t_к$					$t_н$	$t_к$						
120	63	159267	3980	3383	0.1061	20	77	2750	63.05	431	7.5	1.75	26.2
120	59	157814	3950	3357	0.2123	20	75	5600	118.7	811	12.5	2.34	58.5
120	55	156382	3919	3331	0.3184	20	72	8032	167.1	1144	16.2	7.62	246.8
120	51	154968	3890	3306	0.4246	20	67	10240	256.4	1757	21.5	12.9	554.7
120	47	153575	3860	3281	0.5307	20	61	12129	301.0	2066	23.6	18.7	882.6
120	44	152542	3838	3262	0.6369	20	51	13306	340.3	2343	25.2	26.4	1330.6
120	38	150510	3794	3226	0.7431	20	45	14710	376.5	2596	26.5	33.4	1770.2
120	34	149180	3766.3	3201	0.8492	20	34	15231	407.8	2821	27.6	42.3	2334.9

Table 5

Improving the efficiency of heat transfer by heating a mixture of oil and gas condensate consisting of 40 % oil and 60 % gas condensate, due to changes in hydrodynamic regimes

Heat transfer agent (vapors of gas condensate)					Rate of hydrocarbons raw materials, w, m/s	Heated medium (oil)					Performance of heat transfer		
Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K		Change in the temperature of heat carrier, ^{0}C		Re	Nu	α_1, W/m^{2}K	K, W/m^{2}K	Δt_{av}, ^{0}C	Q, W
$t_н$	$t_к$					$t_н$	$t_к$						
120	63	159267	3980	3383	0.1061	20	77	2750	63.05	431	7.5	1.75	26.2
120	59	157814	3950	3357	0.2123	20	75	5600	118.7	811	12.5	2.34	58.5
120	55	156382	3919	3331	0.3184	20	72	8032	167.1	1144	16.2	7.62	246.8
120	51	154968	3890	3306	0.4246	20	67	10240	256.4	1757	21.5	12.9	554.7
120	47	153575	3860	3281	0.5307	20	61	12129	301.0	2066	23.6	18.7	882.6
120	44	152542	3838	3262	0.6369	20	51	13306	340.3	2343	25.2	26.4	1330.6
120	38	150510	3794	3226	0.7431	20	45	14710	376.5	2596	26.5	33.4	1770.2
120	34	149180	3766.3	3201	0.8492	20	34	15231	407.8	2821	27.6	42.3	2334.9

The data in Table 3 shows that the regimes of gas condensate movement in the inner tube of a double-tube apparatus smoothly transit from the transitional regime (at $\upsilon_{ex} = 0.1061 \div 0.3184$ m/s) to turbulent (at $\upsilon_{ex} > 0.4246$ m/s). In the transient mode the coefficient of heat transfer from a wall to the heated oil α_2 increases from 475 to 1349 W/(m^{2}K), the coefficient of heat transfer K – from 8.1 to 18.1 W/(m^{2}K), and the amount of transferred heat Q changes in the redistribution of 28.4 to 297.2 watts. At a turbulent flow the coefficient of heat irradiation from the wall to the heated oil α_2 increases from 1 958 to 3 087 W/(m^{2}K), the coefficient of heat transfer – from 13.5 to 42.8 W/(m^{2}K), and the amount of transferred heat varies within the redistribution of 607.5 to 2 439.6 watts (Table 4).

At heating of oil and gas condensate mixture (20 % oil + 80 % gas) in a horizontal pipe, the driving modes seamlessly move from transitional (with $\upsilon_{ex} = 0.1061 \div 0.3184$ m/s) to turbulent (at $\upsilon_{ex} > 0.4246$ m/s). In the transient mode the coefficient of heat irradiation from the heated wall to the sample liquid α_2 increases from 431 up to 1.144 W/(m^{2}K), the coefficient of heat transfer K – from 7.5 to 16.2 W/(m^{2}K), and the quantity of heat transferred Q changes in the redistribution of 26.2 to 246.8 watts. At a turbulent flow the coefficient of heat irradiation from the heated wall to fluid α_2 does increase

from 1 757 to 2 821 W/(m^{2}K), the coefficient of heat transfer – from 21.5 to 27.6 W/(m^{2}K), and the amount of transferred heat varies within the range of 554.7 to 2 334.9 watts.

The data in Table 5 shows that the regimes of hydrocarbon feedstock (a mixture of oil and gas condensate 40 % oil + 60 % gas) in the inner tube of a double-tube unit smoothly transit from moving (at $\upsilon_{ex} = 0.1061$ m/s) to the turbulent regime (when $\upsilon_{ex} > 0.2123$ m/s). In the transient mode the coefficient of heat irradiation from the heated wall to the sample liquid α_2 is 481 W/(m^{2}K), the coefficient of heat transfer K – 8.2 W/(m^{2}K), and the amount of transferred heat Q has reached 19.2 W. In the turbulent flow the heat irradiation coefficient from the heated wall to the feedstock α_2 increases from 704 to 2 553 W/(m^{2}K), the coefficient of heat transfer – from 11.2 to 26.3 W/(m^{2}K), and the amount of transferred heat varies in a range of 52.4-2 224.9 watts.

From the analysis of the experimental data presented in Tables 2-5, it follows that the efficiency of heat transfer by heating the oil-gas condensate mixtures depends on their flow regimes. For example, improved efficiency of heat transfer of oil and gas condensate by heating a mixture consisting of 40 % oil and 60 % of gas condensate (i.e., increasing of α_2, K, Q values) due to changes in hydrodynamic conditions are illustrated in the diagrams (see Fig. 2-4).
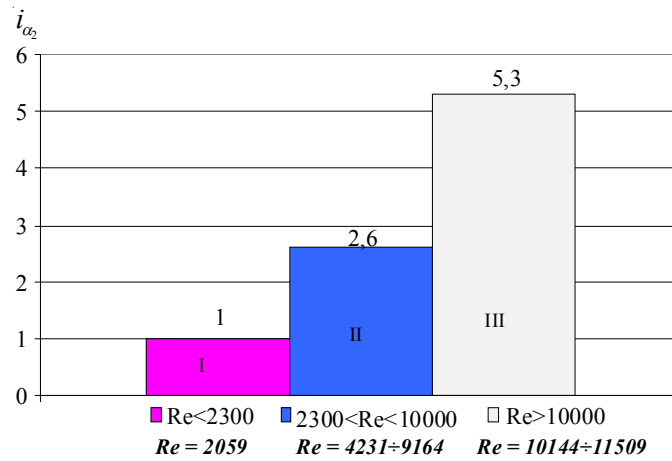
Fig. 2. Increase of the heat irradiation coefficient α_2 at heating oil and gas condensate mixture
with a change in hydrodynamic conditions:

I – laminar; *II* – transitional regime; *III* – turbulent regime
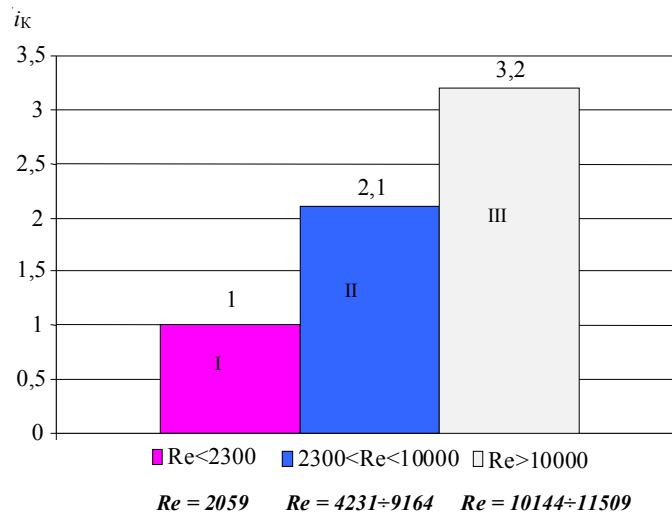

Fig. 3. Increase of the value of heat transfer coefficient *K* by heating a mixture of oil and gas condensate
at the hydrodynamic regimes change:

I – laminar regime; *II* – transitional regime; *III* – turbulent regime
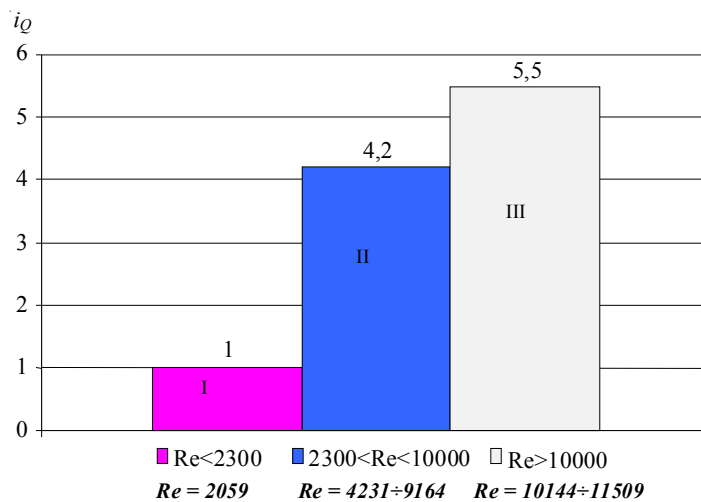

Fig. 4. Increase of the number of values to overeat heat *Q* at heating of oil and gas condensate mixture
(40 % oil and 60 % gas) with a change in hydrodynamic regimes

Conclusion

Based on these studies it can be noted that the hydrodynamic regimes of oil and gas mixtures in the horizontal pipe strongly influence the efficiency of heat exchange during heating. If the efficiency of heat transfer in laminar regime could be taken as one, the transient mode of the heat transfer coefficient from the inner wall to the oil and gas condensate mixture α_2 would be increased twice, while at the turbulent regime α_2 value would be increased by 5.3 times. Thus, the heat transfer coefficient K increases accordingly in 2.1 and 3.2 times, and the amount Q of heat transferred increases significantly with the change of hydrodynamic conditions. If at the transition from laminar to transitional regime, the Q value increases in 4.2 times, then at the turbulent regime it increases in 5.5 times.

The positive effect of liquid hydrocarbons heating in the horizontal pipe is explained as follows. At the core of the flow the heat transfer is carried out simultaneously with conduction and convection effects. The mechanism of heat transfer in the core of the flow in the turbulent motion of the medium is characterized by intensive mixing due to turbulent fluctuations. As approaching the tube wall heat transfer rate decreases, since near the wall the thermal boundary layer is formed, where increasingly important thermal conductivity is becoming. With the development of turbulence boundary layer becomes so thin that convection begins to exert a dominant influence on heat transfer. Whereby the heat irradiation coefficient is increased at turbulent regime of oil and gas flows in a horizontal pipe more then 5 times in comparison with laminar flow of the liquid.

So, at oil and gas mixtures heating processes it is desirable to consider their implementation in a turbulent flow, which will help to develop energy-saving technology of thermal preparation of hydrocarbons to the primary distillation in refineries by optimizing the hydrodynamic conditions in tubular heat exchangers.

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ВЛИЯНИЕ ГИДРОДИНАМИЧЕСКИХ РЕЖИМОВ НЕФТЯНЫХ И ГАЗОВЫХ СМЕСЕЙ НА ЭФФЕКТИВНОСТЬ ТЕПЛООБМЕНА

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Аннотация. В работе приведены результаты экспериментальных исследований для определения степени интенсификации процессов нефтяных и газовых смесей в горизонтальной трубе отопления вследствие изменения гидродинамических режимов. Показано, что для развития энергосберегающих технологий при термической подготовке углеводородного сырья к первичной обработке целесообразно учитывать турбулентность нефтяных и газовых смесей.

Ключевые слова: физико-химические свойства нефтяной и газовой смеси, критерии подобия, тепловой КПД, коэффициент теплопередачи, количество передаваемой тепловой смеси, степени интенсификации процесса, эффективность теплообмена.