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Numerical modeling and calculation of heat transfer between heat carriers in heat exchangers

Heating of oil and oil products is widely used to reduce energy loss during transportation. An approach is being developed to determine the effective length of the heat exchanger and the temperature of the cold heat carrier at its outlet in the case of a strong dependence of oil viscosity on temperature. The oil of the Uzen field (Kazakhstan) is considered as a heated heat carrier, and water is considered as a heating component. The method of the average-logarithmic temperature difference, modified for the case of variable viscosity, and methods of computational fluid dynamics are used for calculations. The results of numerical calculations are compared with the data obtained on the basis of a theoretical approach at constant viscosity. When using a theoretical approach with constant or variable viscosity, the heat transfer coefficients to cold and hot heat carriers are found using criterion dependencies. In the case of variable oil viscosity, the transition of the laminar flow regime to the turbulent one is manifested, which has a significant effect on the effective length of the heat exchanger. To solve this problem comprehensively, a mathematical model of hydrodynamics and heat transfer of heat carriers has been developed and multiparametric numerical calculations have been performed using the "Ansys Fluent" software package.

Keywords: power engineering, heat transfer, heat carrier, viscosity, hydrodynamics, oil products, numerical modeling, laminar-turbulent transition.

Introduction

The change in the qualitative state of the raw material base leads to the development and involvement in the operation of oil fields with a high content of paraffins, resins, asphaltenes. The development of such fields requires the use of unconventional methods of oil production and its preparation for transportation. Light oil products (petroleum, kerosene) are easily transported through pipelines at any time of the year and operations with them do not cause any difficulties. Operations with dark oil products (fuel oil, lubricating oils) and crude oil cause significant difficulties due to the fact that dark oil products become more viscous when the air temperature decreases, and their transportation without heating becomes impossible. For pipeline transportation of oil and oil products, an approach based on the regulation of the rheological properties of oil is used, for example, by heating oil with its subsequent transportation through a pipeline with increased thermal insulation (hot oil pumping). In some cases, an increase in the viscosity of oil with a decrease in temperature leads to unacceptable stresses on the walls of the pipe and stops transportation.

Heat exchange processes are carried out in heat exchangers of various types and designs [1-5]. Various heat carriers are used for heating, for example, hot water or water vapor. Energy consumption is one of the important factors that has a significant impact on the design of the heat exchanger [6]. Shell-and-tube heat exchangers are used in the oil and gas industry, which provide good performance characteristics in a wide range of operating conditions, high reliability and low cost. To determine the efficiency of heat exchange processes, final temperatures and required operating parameters of heat carriers, a thermal calculation is carried out.

The composition of oil (in particular, the content of asphaltenes, resins, paraffins) has a significant effect on the dependence of viscosity on temperature [7, 8]. Empirical formulas describing the change in kinematic viscosity depending on temperature have the form of various functions (exponential, polynomial, power, etc.), which are characterized by the presence of coefficients depending on properties of the liquid. Constant coefficients are determined based on the values of the measured kinematic viscosities at experimental points. The generalized Lee–Kessler equation of state is used to calculate the thermodynamic parameters of oil, gas condensates and their fractions [9]. In [10], studies of the dependence of the kinematic viscosity of oil and oil mixtures on temperature were carried out, and existing formulas for calculating the kinematic viscosity of oil in main pipelines were analyzed.

The flow of liquid in the inter-tube space of the heat exchanger is complex and depends on many factors. Numerical simulation of heat transfer in heat exchange devices of various designs is carried out in [11, 12]. The results of numerical calculations are used to find optimal ways to intensify heat transfer processes [13-15]. The obtained results indicate a decrease in the influence of the viscosity of the pumped oil on the hydraulic characteristics of the pipeline when pumping in developed turbulent conditions.

In classical heat exchangers, a bundle of pipes for one heat carrier is placed inside the casing through which another heat carrier moves. In the design of helicoid heat exchangers, profiled tubes and screw profile ribs are used, with the help of which heat exchange conditions are improved. The tubes in such devices have a small diameter and thin walls (about 0.3 mm). In the case when the viscosity depends on temperature, the flow regime in such thin tubes can vary from laminar to turbulent.

In this paper, a mathematical model of a heat exchanger is developed that takes into account the laminar-turbulent transition. For simplicity, a “pipe-in-pipe” heat exchanger circuit with a thin and smooth inner tube is selected. A method for calculating a direct–flow type heat exchanger is given, in which the working fluid in the inner pipeline is oil (cold heat carrier), and in the outer pipe is water (hot heat carrier). Calculations are carried out for the model design of the heat exchanger both using a theoretical approach based on the method of Log-Mean Temperature Difference (LMTD) at constant and variable viscosity, and on the basis of Computational Fluid Dynamics (CFD). The data obtained within the framework of various approaches are compared with each other, which allow us to conclude about the accuracy of each of the approaches and the possibility of their application in practice.

Dependence of oil viscosity on temperature

The Uzen oil and gas field is located in the Mangistau region of Kazakhstan. Oil fields are located at a depth of 0.9–2.4 km. The density of oil is 844-874 kg/m3, viscosity — 3.4–8.15 $mPa \cdot s$, sulfur content — 0.16–2 %, paraffins 16-22 %, resins — 8-20 %.

In the literature, various dependences of viscosity on temperature are used. In the oil industry, the Walter formula is used to calculate the kinematic viscosity depending on temperature [10]

$$\lg[\lg(\nu + 0.8)] = a + b\lg T \,, \tag{1}$$

where a and b are empirical coefficients determined experimentally for a given fluid. The coefficients a and b in formula (1) are from the relations

$$a = \lg[\lg(\nu_1 + 0.8)] - b\lg T$$
$$b = \frac{\lg[\lg(\nu_1 + 0.8)] - \lg[\lg(\nu_2 + 0.8)]}{\lg T_1 - \lg T_2},$$

here ν_1 and ν_2 are the values of the kinematic viscosity of the liquid at temperatures T_1 and T_2.

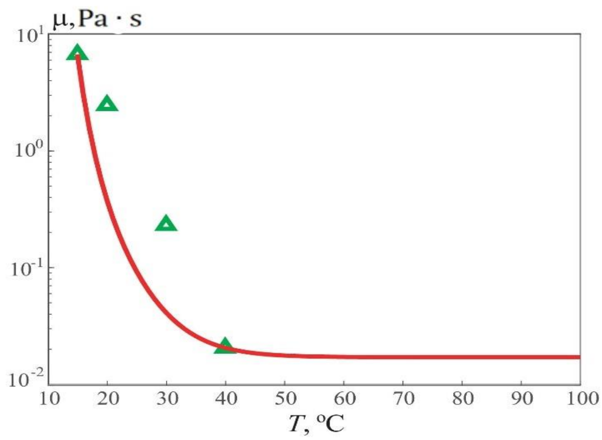

Figure 1. Dependence of the dynamic viscosity of the Uzen oil field on temperature. Triangular icons — experimental data [16], solid line — calculations according to the Walter formula

A comparison of the results of calculations using the Walter formula with experimental values of dynamic viscosity is shown in Figure 1 for the oil of the Uzen field. The temperature varies from 10 to 100 °C. The solid line corresponds to the dependence of viscosity on temperature obtained by the Walter formula (1), and the triangular icons correspond to the results of a physical experiment [16].

Calculation method for constant viscosity

To estimate the heat flows from a hot heat carrier to a cold one, a heat carrier model with a constant viscosity along the length is used, based on the use of an average logarithmic temperature difference. In a recuperative heat exchanger, two liquids with different temperatures move in a space separated by a solid wall (Fig. 2).

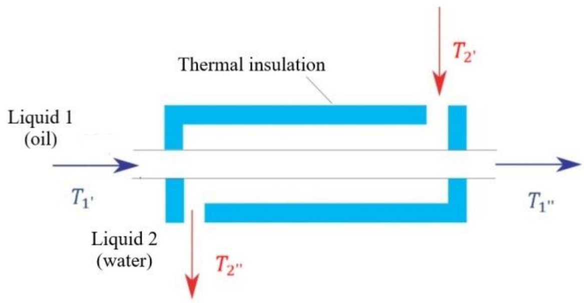

Figure 2. Diagram of a heat exchanger in which heat carriers move in the opposite direction

Thermal calculation is reduced to the joint solution of the equations of thermal balance and heat transfer. The heat balance equation has the form [21]

$$Q = G_1 c_{p1}(T_{1'} - T_{1''}) = G_2 c_{p2}(T_{2''} - T_{2'}) > 0\,, \tag{2}$$

here Q is the amount of heat transferred per unit of time from a hot heat carrier to a cold one, G is the mass flow rate of the heat carrier, c_p is the isobaric heat capacity, $T_{'}$ is the inlet temperature, $T_{''}$ is the outlet temperature. Index 1 corresponds to a hot heat carrier, and index 2 corresponds to a cold heat carrier. The heat transfer equation for a heat exchanger is represented as

$$Q = \bar{k} F \Delta\bar{T}\,, \tag{3}$$

where $\bar{k}$ is the average heat transfer coefficient, which is calculated at an average temperature $(T_{1'} + T_{1''})/2$ and $(T_{2'} + T_{2''})/2$, $\Delta\bar{T}$ is the average temperature difference. The average temperature difference is determined by the expression

$$\Delta\bar{T} = \frac{1}{F}\int_0^F \Delta T\, dF\,,$$

where F is the heat exchange surface area.

Using the notation $\Delta T = (T_1 - T_2)$, equations (2) and (3) in differential form will take the form

$$\frac{d(\Delta T)}{\Delta T} = -mk\,dF, \quad m = \left(\frac{1}{G_1 c_{p1}} \pm \frac{1}{G_2 c_{p2}}\right).$$

The plus sign is selected in the case of a direct–flow apparatus, and the minus sign is selected in the case of a counter-flow heat exchanger. The above equation is valid along the direction of movement of the hot heat carrier. Assuming that m and $\bar{k}$ are constant along the length of the apparatus and integrating from 0 to F and from $\Delta T'$ and ΔT, we obtain

$$\Delta T = \Delta T' \exp\left(-m\bar{k}F\right), \tag{4}$$

where $\Delta T'$ is the temperature difference at the inlet of the hot heat carrier. Along the heat exchange surface, the temperature pressure varies exponentially. By averaging the temperature head over the entire heat exchange surface, the average logarithmic temperature head is from the ratio

$$\Delta\overline{T} = \frac{\Delta T'' - \Delta T'}{\ln(\Delta T'' / \Delta T')}.$$

In the constructive calculation of heat exchange devices, the thermal performance Q is determined by equation (2). The heat exchange surface area F is found from the equation

$$F = \frac{Q}{\overline{k}\Delta\overline{T}}.$$

When calculating the heat exchange surface area, the task is reduced to calculating the average heat transfer coefficient and the average logarithmic temperature pressure. The length of the heat exchanger is calculated by the formula

$$L = \frac{F}{\pi n d},$$

where n is the number of inner tubes, d is their hydraulic diameter.

The temperature distributions along the heat exchange surface are expressed by the relations:

– direct-flow circuit

$$T_1(x) = T_{1'} - \Delta T' \frac{1 - \exp\left[-\overline{k} m F(x)\right]}{1 + (G_1 c_{p1})/(G_2 c_{p2})}$$

$$T_2(x) = T_{2'} + \Delta T' \frac{1 - \exp\left[-\overline{k} m F(x)\right]}{1 + (G_2 c_{p2})/(G_1 c_{p1})},$$

here $F(x)$ is the dependence of the heat exchange surface area on the length measured along the path of the hot heat carrier. In the case of a cylindrical surface, the heat exchange area is expressed in terms of length $F(x) = \Pi \cdot x$, where Π is the wetted perimeter of the heat exchange surface.

For the case of thin cylindrical walls, the relations for surface temperatures have the form

$$T_{w1} = \frac{\left(\dfrac{\alpha_1 F_1}{\alpha_2 F_2} + \dfrac{\alpha_1 F_1 \delta_w}{\lambda_w F_a}\right) T_1 + T_2}{1 + \dfrac{\alpha_1 F_1}{\alpha_2 F_2} + \dfrac{\alpha_1 F_1 \delta_w}{\lambda_w F_a}}$$

$$T_{w2} = \frac{\left(\dfrac{\alpha_2 F_2}{\alpha_1 F_1} + \dfrac{\alpha_2 F_2 \delta_w}{\lambda_w F_a}\right) T_2 + T_1}{1 + \dfrac{\alpha_2 F_2}{\alpha_1 F_1} + \dfrac{\alpha_2 F_2 \delta_w}{\lambda_w F_a}},$$

here $F_a = (F_1 + F_2)/2$, F_1 is the heat exchange area on the heat carrier side 1, F_2 is the heat exchange area on the heat carrier side 2, δ_w is the wall thickness, λ_w is the thermal conductivity coefficient of the wall, α is the heat transfer coefficient. The above relations for the wall temperature are implicit and require an iterative solution, since the heat transfer coefficient α depends on temperature.

For a single-layer cylindrical wall, the average heat transfer coefficient is calculated as follows

$$\overline{k} = \left(\frac{1}{\overline{\alpha_1} d_1} + \frac{1}{2\lambda_w} \ln\frac{d_2}{d_1} + \frac{1}{\overline{\alpha_2} d_2}\right)^{-1},$$

where $\overline{\alpha_1}$ is the average heat transfer coefficient to the cold heat carrier, $\overline{\alpha_2}$ is the average heat transfer coefficient to the hot heat carrier.

The Nusselt number depends on the flow mode (laminar or turbulent) and the heat exchange mode (heating or cooling). The average heat transfer coefficient is expressed in terms of the average length of the Nusselt number $\mathrm{Nu} = \bar{\alpha} d_g / \lambda$, where $d_g = 4F_g / \Pi$ is the effective hydraulic diameter, F_g is the area of the passage section of the channel, Π is the wetted perimeter, λ is the thermal conductivity of the liquid. During the flow in the pipe or during the longitudinal flow around the bundles of pipes, the Nusselt number is calculated using a semi-empirical dependence of the form

$$\mathrm{Nu} = \mathrm{Nu}\left(\mathrm{Re}, \mathrm{Pr}, \mathrm{Pr}_w, L / d_g\right),$$

where Pr is the Prandtl number of the liquid, Pr_w is the Prandtl number of the liquid calculated from the wall temperature. Similarity numbers are calculated from the average temperature of the heat carrier. The Reynolds number is determined by the ratio $\mathrm{Re} = \rho V d_g / \mu$, where V is the characteristic flow velocity, ρ is the density, and μ is the dynamic viscosity.

Calculation method for variable viscosity

In the case of a strong dependence of viscosity on temperature, the heat exchanger is divided into elementary sections along the length. At each site, an assumption is made about a small change in viscosity.

With a weak dependence of the viscosity of the heat carrier on the temperature, the average Reynolds number is based on the average temperature of the heat carrier. Such an assumption does not introduce significant errors in the calculation, since it practically does not affect the flow regime. In the case of a strong dependence of viscosity on temperature, as the heat carrier heats up, the flow mode changes from laminar to developed turbulent. In this case, the local heat transfer coefficient $\alpha(x)$ is calculated, and the relations for the local Nusselt number $Nu_x = \alpha(x) d_g / \lambda$, calculated from local similarity numbers are used

$$\mathrm{Nu}_x = \mathrm{Nu}\left(\mathrm{Re}_x, \mathrm{Pr}_x, \mathrm{Pr}_w, x / d_g\right).$$

The average value of the heat transfer coefficient is found by the formula

$$\bar{\alpha} = \frac{1}{L}\int_0^L \alpha(x) dx .$$

Local Nusselt numbers in laminar and turbulent flow regimes are found using the relations given in [17, 18]. To calculate the local Nusselt number in the laminar flow regime in the pipe, the ratio [18] is used

$$\mathrm{Nu}_x = 4.36\left(1 + 0.032\frac{d}{x}\mathrm{Re}_x \mathrm{Pr}_x^{5/6}\right)^{2/5}\left(\frac{\mathrm{Pr}_x}{\mathrm{Pr}_{x,w}}\right).$$

The above ratio is valid at $0.7 < \mathrm{Pr} < 103$. The expression for calculating the local Nusselt number for the turbulent flow regime in a pipe with an additional correction for the change in the Prandtl number has the form [17]

$$\mathrm{Nu}_x = 0.022\mathrm{Re}_x^{0.8}\mathrm{Pr}_x^{0.43}\left(\frac{\mathrm{Pr}_x}{\mathrm{Pr}_{x,w}}\right)^{0.25}\varepsilon_l ,$$

where

$$\varepsilon_l = \begin{cases} 1, \ x/d \geq 15, \\ \dfrac{1.38}{\left(x/d\right)^{0.12}}, \ x/d < 15 \end{cases}.$$

For the annular channel in the turbulent flow regime, the ratio is used as for the flow in the pipe, but with its equivalent hydraulic diameter.

The heat balance equation for the elementary section in the direction of movement of the hot heat carrier is written as follows

$$\begin{aligned} &\frac{dQ_1}{dx} = G_1 c_{p1}(T_1)\frac{dT_1}{dx}, \frac{dQ_1}{dx} \le 0, \\ &\frac{dQ_2}{dx} = \pm G_2 c_{p2}(T_2)\frac{dT_2}{dx}, \frac{dQ_2}{dx} \ge 0, \quad (5) \\ &\frac{dQ_1}{dx} + \frac{dQ_2}{dx} = 0. \end{aligned}$$

Here dQ_1 is the loss of the amount of heat by the hot heat carrier, dQ_2 is the amount of heat acquired by the cold heat carrier, G is the mass flow of the heat carrier, c_p is the heat capacity, dT is the temperature change. The plus sign corresponds to a direct–flow circuit, and the minus sign corresponds to a counter-flow circuit. The heat transfer equation for an elementary section takes the form

$$\begin{aligned} &\frac{dQ_1}{dx} = k(T_1,T_2)(T_2 - T_1)\frac{dF}{dx}, \frac{dQ_1}{dx} \le 0, \\ &\frac{dQ_2}{dx} = k(T_1,T_2)(T_1 - T_2)\frac{dF}{dx}, \frac{dQ_2}{dx} \ge 0. \end{aligned} \quad (6)$$

Here k is the local heat transfer coefficient, dF/dx is the change in the heat exchange area, which remains constant for a heat exchanger made of straight pipes.

From equations (5) and (6) follows a closed system of equations with respect to the temperatures of heat carriers

$$\begin{aligned} &\frac{dT_1}{dx} = \frac{k(T_1,T_2)}{G_1 c_{p1}(T_1)}(T_2 - T_1)\frac{dF}{dx} \\ &\frac{dT_2}{dx} = \pm\frac{k(T_1,T_2)}{G_2 c_{p2}(T_2)}(T_1 - T_2)\frac{dF}{dx} \end{aligned} \quad (7)$$

Since $dF/dx = const$ and is known, the system of equations (7) is a system of ordinary differential equations with a nonlinear right-hand side. In the case of a direct-flow circuit (plus sign), the Cauchy problem is posed for system (7), and for a counter-current circuit (minus sign), the boundary value problem is solved. In this case, the integration is carried out up to the length L, which is unknown in advance.

The system of equations (7) is solved by the finite difference method on the interval $x \in [0, L]$. To stabilize the iterative process during the linearization of the system, the method of lower relaxation is used. The length of the integration interval L is unknown in advance. Newton's method is used to determine it. The local heat transfer coefficient is found using local heat transfer coefficients.

$$k = \left(\frac{1}{\alpha_1 d_1} + \frac{1}{2\lambda_w}\ln\frac{d_2}{d_1} + \frac{1}{\alpha_2 d_2}\right)^{-1},$$

where α_1 is the local heat transfer coefficient from the cold heat carrier to the wall, α_2 is the local heat transfer coefficient from the side of the hot heat carrier, λ_w is the thermal conductivity coefficient of the tube material.

Numerical simulation of heat transfer

The results of thermal calculations are compared with the data obtained by computational fluid dynamics methods. Oil is considered a Newtonian liquid with a constant density. Calculations are carried out using numerical solutions of Reynolds–Averaged Navier-Stokes equations (Reynolds–Averaged Navier-Stokes, RANS) for a viscous incompressible fluid closed using a turbulence model that takes into account the laminar-turbulent transition.

The SST $k-\omega$ turbulence model is designed for an effective combination of a reliable and accurate $k-\omega$ model in the wall region and a $k-\varepsilon$ model in free flow [19, 20]. To switch between models, a special function is used, which takes a single value in the wall area (the standard $k-\omega$ model is used) and a zero value away from the wall (the $k-\varepsilon$ model is used).

The model taking into account the laminar-turbulent transition (Local-Correlation Transition Model, $\gamma-\mathrm{Re}_{\theta t}$ transition model) is based on a combination of the SST equations of the $k-\omega$ turbulence model with two additional transfer equations for the intermittency parameter γ and the critical Reynolds number $\gamma-\mathrm{Re}_{\theta t}$, constructed from the thickness of the momentum loss [21, 22]. To simplify the model, the equation for $\gamma-\mathrm{Re}_{\theta t}$ is not considered, and in the equation for the intermittency parameter, an assumption is made about the smallness of convective terms [23].This approach leads to algebraic relations for finding the intermittency parameter.

To discretize the basic equations, the finite volume method on unstructured grids and the SIMPLE method are used [24]. The discretization of inviscid flows is carried out using the MUSCL scheme (Monotonic Upstream Schemes for Conservation Laws, monotonic counter–flow circuit for conservation laws), and viscous flows — a centered scheme of the 2nd order of accuracy. The MUSCL scheme makes it possible to increase the order of approximation by spatial variables without losing the monotony of the solution, satisfies the TVD (Total Variation Diminishing) condition and is a combination of centered finite differences of the 2nd order and a dissipative term, for switching between which a flow limiter built on the basis of characteristic variables serves. The geometric multigrid method is used to solve the system of difference equations [25].

The calculations use a grid consisting of 19461 cells, of which 500×24 cells are placed in an area filled with oil, 500×5 cells in an area made of steel, and 500×13 cells in an area filled with water. The mesh cells are thickened near the walls of the pipe so that $y+<2$, where $y+$ is a dimensionless wall coordinate.

Calculation results and discussion

The diagram of the direct-flow type heat exchanger is shown in Figure 3 (dimensions are given in millimeters). The *h* index corresponds to the hot medium (water), the *c* index corresponds to the cold medium (oil). The *i* and *o* indexes refer to the input and output cross-sections.

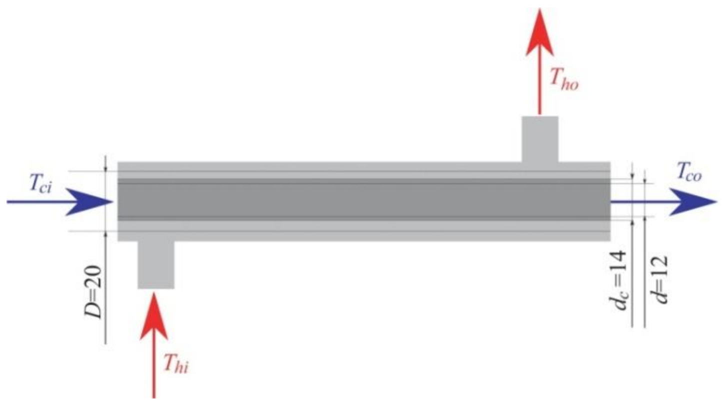

Figure 3. Diagram of a direct-flow type heat exchanger

To calculate the effective length of the heat exchanger and the temperature of the hot heat carrier at the outlet, the parameters given in Table 1 are set (the input and output temperatures of the cold heat carrier, the input temperature of the hot heat carrier, the speeds of both heat carriers, the flow rates of the hot and cold heat carrier, the geometric characteristics of the inner and outer tubes, as well as the physical properties of the tube material).

Table 1

Input data for thermal calculation

Parameter	Unit of measurement	Symbol	Quantity
Number of tubes	piece	n	1
Tube wall thickness	mm	δ_w	1
Inner diameter of the tubes	mm	d	12
Outer diameter of the tubes	mm	d_c	14
Inner diameter of the shell	mm	D	20
Temperature of the hot heat carrier at the inlet	K	T_{hi}	423
Consumption of hot heat carrier	кg/s	G_h	0.6386
The speed of the hot heat carrier	m/s	v_h	4
The temperature of the cold heat carrier at the inlet	K	T_{ci}	303
The temperature of the cold heat carrier at the outlet	K	T_{co}	328
Consumption of cold heat carrier	кg/s	G_c	0.3814
The speed of the cold heat carrier	m/s	v_c	4

To determine the temperature of the hot heat carrier at the outlet and the corresponding effective length of the heat exchanger, the heat balance equations are used. Solving the heat balance equations by the finite difference method, we obtain the distributions of the average mass temperatures of the heat carriers given in Table 2.

Table 2

Calculation results

Heat carrier model	With constant viscosity	With variable viscosity
Length, m	5.28	4.26
The temperature of the hot heat carrier at the outlet, K	416	416

The obtained results are compared with the numerical simulation data. Figure 4 shows the distribution of the average mass temperature of oil (cold heat carrier) along the length of the heat exchanger, obtained using the finite difference method and based on numerical modeling.

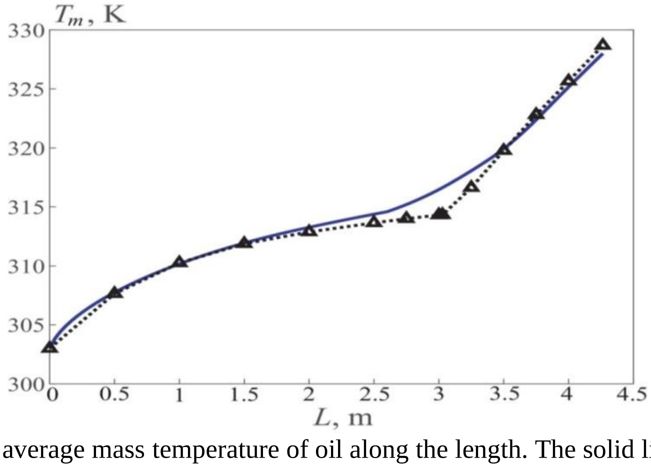

Figure 4. Distribution of the average mass temperature of oil along the length. The solid line corresponds to the results obtained on the basis of the theoretical approach, and the dotted line with triangular icons corresponds to the results of numerical calculations.

The average mass temperature of oil increases along the length due to heating from a heat source (hot heat carrier). The results of analytical and numerical calculations are in good agreement with each other.
The distributions of the average mass temperature of water (hot heat carrier) along the length, obtained on the basis of analytical and numerical calculations, are shown in Figure 5.

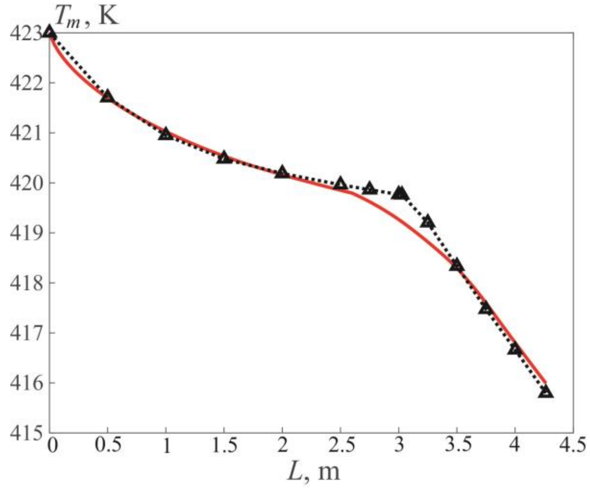

Figure 5. Distribution of the average mass temperature of water along the length. The solid line corresponds to the results obtained on the basis of the theoretical approach, and the dotted line with triangular icons corresponds to the results of numerical calculations.

Compared with the results shown in Figure 4, the average mass temperature of water decreases along the length of the pipe due to heat transfer from it to the cold heat carrier. It should also be noted that the results of calculations obtained on the basis of different approaches are well coordinated.

From the results shown in Figures 4 and 5, it is possible to notice characteristic changes in the curvature of the lines at a distance of about 2.5 m from the input section, where in both cases there are sharp changes in temperature gradients with corresponding signs. This transition occurs at the distance where the laminar flow regime turns into a turbulent one. Figure 6 shows a graph of the temperature distribution along the inner wall of the tube from the oil side. It can be seen that at the entrance the oil is cold and the wall cools down strongly from the oil side, then it warms up, heat exchange by thermal conductivity prevails here, then a laminar-turbulent transition occurs and heat transfer increases significantly and the wall temperature decreases.

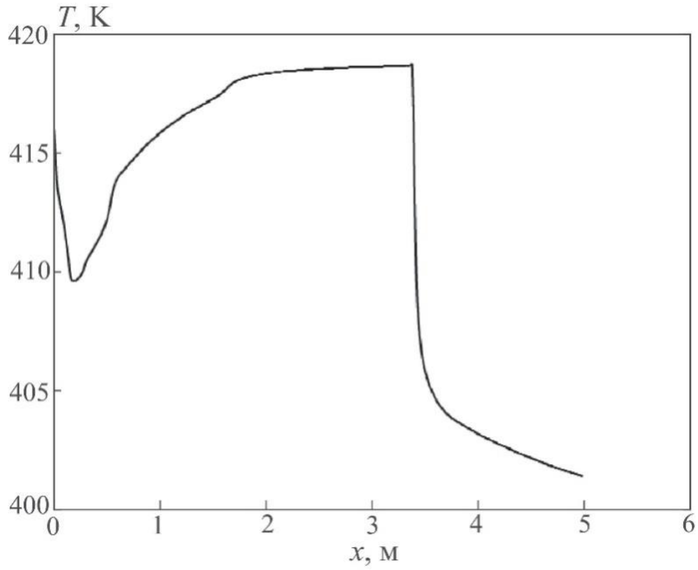

Figure 6. Temperature distribution along the tube wall from the oil side

Conclusion

Reducing the viscosity of oil by heating it is one of the ways to increase the energy efficiency of the process of pumping high-viscosity oil during production and transportation. Numerical modeling allows solving a number of issues related to increasing the efficiency of heat transfer, which remains one of the most important in the design of heat exchange devices in the oil and gas industry.

Thermal and numerical calculations were carried out to determine the length of the heat exchanger and the temperature of the cold heat carrier in the outlet section in the case of constant and variable oil viscosity. With variable viscosity of oil, the transition from laminar to turbulent mode is manifested, while with the analytical method of calculation for constant viscosity, this effect is not taken into account. The obtained results show that a model with a constant viscosity leads to an underestimation of the length of the heat exchanger by about 20 % compared to calculations that take into account the dependence of oil viscosity on temperature.

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Д.Е. Курманова, Н.Ж. Джайчибеков, А.Г. Карпенко, К.Н. Волков

Жылуалмасу аппараттарындағы жылутасымалдағыштар арасындағы жылуалмасуды сандық модельдеу және есептеу

Тасымалдау кезінде энергия шығынын азайту үшін мұнай мен мұнай өнімдерін жылыту кеңінен қолданылады. Мұнай тұтқырлығы температураға қатты тәуелді болған жағдайда жылуалмасу аппаратының тиімді ұзындығын және оның шығысындағы суық жылутасымалдағыштың температурасын анықтау тәсілі әзірленуде. Жылытылатын жылутасымалдағыш ретінде Өзен кен орнының мұнайы (Қазақстан), ал қыздырғыш компонент ретінде — су қарастырылады. Есептеулер үшін ауыспалы тұтқырлық жағдайында модификацияланған орташа логарифмдік температура айырмашылығының және есептеу сұйықтығының динамикасы әдістері қолданылады. Сандық есептеулердің нәтижелері тұрақты тұтқырлық кезінде теориялық тәсіл негізінде алынған мәліметтермен салыстырылады. Тұрақты немесе айнымалы тұтқырлығы бар теориялық тәсілді пайдаланған кезде суық және ыстық салқындатқыштарға жылу беру коэффициенттері критерийлік тәуелділіктерді пайдалана отырып анықталады. Мұнайдың ауыспалы тұтқырлығы жағдайында ламинарлық ток режимінің турбуленттілікке ауысуы байқалады, бұл жылуалмасу аппаратының тиімді ұзындығына айтарлықтай әсер етеді. Бұл мәселені кешенді шешу үшін гидродинамика мен жылутасымалдаушылардың жылуалмасуының математикалық моделі жасалған және «Ansys Fluent» бағдарламалық кешенінің көмегімен көп параметрлік сандық есептеулер жүргізілді.

Кілт сөздер: энергетика, жылуалмасу, жылутасымалдағыш, тұтқырлық, гидродинамика, мұнайөнімдері, сандық модельдеу, ламинарлы-турбулентті ауысу.

Д.Е. Курманова, Н.Ж. Джайчибеков, А.Г. Карпенко, К.Н. Волков

Численное моделирование и расчет теплообмена между теплоносителями в теплообменных аппаратах

Подогрев нефти и нефтепродуктов широко применяется для уменьшения энергопотерь при транспортировке. Разрабатывается подход к определению эффективной длины теплообменного аппарата и температуры холодного теплоносителя на его выходе в случае сильной зависимости вязкости нефти от температуры. В качестве нагреваемого теплоносителя рассматривается нефть Узеньского месторождения (Казахстан), а в качестве нагревающего компонента — вода. Для расчетов используются методы среднелогарифмической разницы температур, модифицированные для случая переменной вязкости, и вычислительной гидродинамики. Результаты численных расчетов сравниваются с данными, полученными на основе теоретического подхода при постоянной вязкости. При использовании теоретического подхода при постоянной или переменной вязкости коэффициенты теплоотдачи к холодному и горячему теплоносителям находятся с помощью критериальных зависимостей. В случае переменной вязкости нефти проявляется переход ламинарного режима течения в турбулентный, который оказывает существенное влияние на эффективную длину теплообменного аппарата. Для комплексного решения данной задачи разработана математическая модель гидродинамики и теплообмена теплоносителей и проведены многопараметрические численные расчеты с помощью программного комплекса «Ansys Fluent».

Ключевые слова: энергетика, теплообмен, теплоноситель, вязкость, гидродинамика, нефтепродукты, численное моделирование, ламинарно-турбулентный переход.

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