

IMPROVEMENT OF THE METHOD OF CALCULATING HEAT TRANSFER COEFFICIENTS USING GLYCOLS TAKING INTO ACCOUNT SURFACE FORCES OF HEAT CARRIERS

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Abstract. This study compares the classic calculating method of the heat transfer coefficients of the shell-and-tube heat exchanger tubes using the classic Nusselt, Reynolds, and Prandtl similarity numbers with a new method that takes into account the coefficients of surface tension of heat carriers, their transitional, turbulent viscosity and thermal conductivity, as well as the average thickness of the laminar boundary layer (LBL). The classic method shows a better efficiency of water as a heat carrier compared to a 45% aqueous solution of propylene glycol. Instead, the new calculation method shows that a 45% aqueous solution of propylene glycol at the same Reynolds numbers has higher heat transfer coefficients compared to water in the temperature range of 273–353 K. We divided the "live cross-section" of the flow of the liquid coolant into a medium-thick LBL, where the Fourier equation of thermal conductivity is applied, and into its turbulent part, where the equation of thermal conductivity with turbulent thermal conductivity is also applied. A new formula (14) is proposed for calculating the average thickness of the LBL based on the radius of the "live cross-section" of the coolant flow, as well as the Bl_{urb} similarity number obtained by us in previous works. A new formula (15) is also proposed for calculating the heat transfer coefficient, which includes the transitional and turbulent thermal conductivity of the corresponding zones of the flow "live section", as well as the average thickness of the LBL.

Keywords: transitional, turbulent viscosity and thermal conductivity, shell-and-tube heat exchanger, heat transfer coefficient, average thickness of the LBL, surface tension coefficient of the heat carrier.

1. Introduction

1.1. Theoretical Prerequisites for the Emergence of a New Way of Calculating Heat Transfer Processes Taking into Account Surface Forces

For more than a century, there has been a concept of fluid movement in closed spaces (pipes, channels, *etc.*) with the existence of a laminar boundary layer (LBL). Even long before the appearance of the well-known monograph by H. Schlichting,¹ in which the nature of the LBL phenomenon from the standpoint of hydromechanics is described in sufficient detail and comprehensively, so a wide range of literary sources is provided, researchers of fluid hydromechanics paid comprehensive attention to this issue. However, in our opinion, the key issues in this field remain debatable today. One of the biggest paradoxes is that the very existence of LBL is explained by the appearance of large frictional forces at the wall-flow interface, which lead to the suspension of the free movement of the elementary layers of the liquid.

As you know, the force of friction has two main components - mechanical and adhesive. The mechanical component depends mainly on the geometry of the walls of the channel, the pipe, that is, on the roughness of the surface, namely on the size of the micro-uniformities. For this reason, the authors of numerous scientific works derived formulas for calculating friction coefficients for laminar and turbulent flow. According to these formulas, the consumption of the coolant is directly proportional to the coefficient of friction. Everything would seem logical from the point of view of Newton's law, where the force of friction is directly proportional to the speed. The second component is molecular (adhesive), which depends on the nature of the wall material itself, as well as on the nature of the liquid. A fundamental characteristic of the molecular component of the frictional force is the bond energy, or the energy of cohesion, which manifests itself

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on the surface, *e.g.*, of the metal wall of the pipe, due to the surface energy of the metal, and in the liquid due to its surface tension coefficient. In the mentioned monograph,¹ as in other publications on this issue, the coefficients of surface tension of coolants are not taken into account, but only the mechanical component of the friction force is considered.

In 1984–1991, our research showed that under the conditions of the fretting process, *i.e.*, oscillations in microfriction, the molecular adhesive component of the friction force prevails. In addition, since the speeds of bodies movement have become very large (and the forces of friction as well), for example in aviation technology, the surface energy of metal parts has begun to receive primary attention.²

Considering the movement of liquid in closed systems, pipelines, channels, and taking into account the surface energy, it becomes clear that the force of friction, especially in the LBL of the liquid, largely depends on the surface energy of the metal surface, as well as on the surface energy of the liquid coolant. This is exactly the approach we proposed in our work.³ From such a concept, it becomes clear that the very existence of the LBL indicates the action of powerful forces that cause the movement of the wall layers of the liquid to slow down. These, in our opinion, are surface forces. The following arguments can be given in favor of this: 1) in the monomolecular layer adjacent to the metal wall, the speed of liquid movement is close to zero; 2) friction forces are close to zero, according to Newton's law; 3) the force of friction at rest of the liquid is zero. In this case, the question arises, what forces cause such a powerful deformation of the flow for both laminar (L) and turbulent (T) regimes of fluid movement with the formation of the LBL, if they are not frictional forces?

The force analysis in the LBL conducted by us in the publication³ showed that the Froude number and the inverse Reynolds number are by 2–5 orders of magnitude smaller than the surface criterion and the Euler number in the LBL, which makes it possible to neglect the forces of gravity and friction in these conditions.

A similar situation is observed when using nanofluids as heat carriers. A corresponding force analysis was carried out by us in the literature,⁴ which showed that surface forces also dominate the LBL that occurs around a nanoparticle during its turbulent motion in a mostly liquid medium. Meanwhile, the Reynolds number, which is present in almost all numerical empirical equations of heat transfer and is actually in the maximum degree (0.6–0.8), *i.e.*, it is assumed that the influence of frictional forces in such systems is maximal. A paradox appears. Using a simple comparison of the similarity numbers in LBL,^{3,4} we get that the surface forces dominate, but they are not taken into account anywhere because the classical Nusselt,

Reynolds and Prandtl similarity numbers do not account for them. To date, in such systems, during the movement of liquid in pipelines or channels to determine heat transfer coefficients, classical similarity numbers and the structure of numerical equations look like this (1):

$$(Nu)^l = B \cdot (Re)^X (Pr)^Y. \quad (1)$$

According to the most optimistic estimates, the error in determining the heat transfer coefficient in such numerical empirical Eq. (1) is about (15–20) %. In some cases, this indicator reaches 50%.^{5–8} Such a significant deviation of experimental data from calculated data indicates a certain imperfection of the method of determining heat transfer coefficients or the theoretical correlation dependencies themselves.

Analysis of the dimensionality of physical quantities is one of the main methods of optimization in heat exchange processes, in particular in the plasma environment.⁹ However, this method is very effective under one key condition, when at the initial stage of the analysis, the dominant factors affecting the physical process are correctly selected, and their physicochemical characteristics with dimensions are included in the functional of the corresponding power equation.

Our studies in papers,^{3,4} cited above, show that the dominant factor – the surface energy of the liquid coolant is not included in the power equation, and therefore Eq. (1) is solved empirically, where the power indicators X , Y , and the constant B are determined experimentally. This leads to the fact that when the liquid heat carrier and its thermophysical properties change, especially in the conditions of using nanofluid heat carriers, the structure of equation (1) is preserved, and the unknown power exponents and the constant change significantly. Equations of type (1) lose their universality. Numerical equations for calculating heat transfer using TiO₂ nanoparticles with a classical structure (1) were compactly collected by us in works,^{4,10–12} and a corresponding analysis was also carried out. As one can see, these equations are far from analytical, from universal and have a specifically empirical character.

The heat transfer coefficient h , which is included in the classical Nusselt number, does not analytically describe the heat transfer process, since the empirical Eq. (1) contains three unknown quantities B , X , Y , which are determined experimentally. The disadvantages of this value are described in a number of works, in particular in the literature,¹¹ where heat transfer processes are analyzed from the standpoint of fluid mechanics.

For the calculation of heat exchange equipment using, for example, nanofluid coolants, Eq. (1) loses its relevance. This is due to the fact that exponents at Reynolds and Prandtl numbers, as well as the constant, change. This causes the need for additional expensive

experiments. Equations of type (1) lose their universality. In works,^{4,10} as well as in many other sources, dependences for determining the viscosity and thermal conductivity of nanofluids are given, for example, for the base fluid - water and for the base fluid - ethylene and propylene glycols. These equations are also not universal, as they also require numerous additional experiments for each specific heat carrier.

One of the important aspects of our approach is to consider the movement of coolants not in statics, but in dynamics, that is, using not molecular (static) characteristics of coolants, but dynamic ones. The classic Nusselt, Reynolds, and Prandtl numbers contain molecular thermophysical characteristics in the static state of the heat carrier, that is, they characterize Brownian motion, not L or T regimes, where the transient and turbulent viscosity and thermal conductivity of thermal agents are several orders of magnitude higher. For the first time, an approach to the interpretation of turbulent viscosity as different from static viscosity was given in his writings by the French scientist J. Boussinesq, who proposed to consider turbulent fluid flow as "Newtonian". In addition, he suggested that the behavior of free liquid jets is the result of the joint action of surface and gravitational forces.

1.2. The Main Analytical Regularities Obtained by Us in Previous Works

In a previous work,¹³ with the help of dimensional analysis, we showed that in LBL, taking into account surface forces, the so-called transitional (imaginary) viscosity of the coolant can be calculated from equality (2):

$$\mu_{trans} = \frac{\sigma \cdot \cos \theta_{trans}}{\sqrt{C_p}} = \left[\frac{N \cdot s}{m \cdot m} = Pa \cdot s \right] \quad (2)$$

where σ is coefficient of surface tension of the coolant, N/m; $\cos \theta_{trans}$ is cosine of the angle (surface hydrophilicity); C_p is specific heat capacity of the coolant, J/kg·K.

In the same paper,¹³ a new similarity number Bl (3) was obtained:

$$Bl = \frac{\mu \cdot \sqrt{C_p}}{\sigma \cdot \cos \theta_{trans}} = \left[\frac{Pa \cdot s}{Pa \cdot s} = \frac{m/s}{m/s} \right] \quad (3)$$

where Bl is dimensionless number; μ is coefficient of dynamic viscosity of the coolant, kg/m·s.

The Bl number is a dimensionless quantity that shows the ratio of the product of the forces of internal friction and the forces of cohesion in the liquid heat agent to the forces of surface tension.¹⁰ We understand the physical meaning of the Bl number as the ratio of the molecular viscosity of the liquid to its viscosity in the transitional zone of the LBL. At the same time, this is the ratio of the value reflecting the rate of thermal movement of liquid molecules to the value responsible for the rate of

relaxation of these molecules.^{4,10} In our study,¹⁴ we got a new similarity number Bl_{turb} (4):

$$Bl_{turb} = \frac{\mu_{turb} \cdot \sqrt{C_p}}{\sigma \cdot \cos \theta_{trans}} = \left(\frac{\sqrt{C_p}}{V} \right)^{-X} \quad (4)$$

where μ_{turb} is the coefficient of turbulent viscosity of the coolant, kg/m·s; V is speed of movement of the coolant, m/s; $(-X)$ is an indicator of degree.⁴

The Bl_{turb} number differs from the Bl number by the value of the turbulent viscosity, *i.e.*, the ratio of the turbulent viscosity of the coolant to the molecular viscosity or the ratio of the turbulent thermal conductivity to the molecular thermal conductivity is equal to the ratio of the similarity numbers Bl_{turb}/Bl . In addition, the ratio of turbulent viscosity to transitional viscosity in the LBL transitional layer is equal to the Bl_{turb} number.

The turbulent thermal conductivity k_{turb} in the turbulent zone of the coolant flow can be calculated by formula (5) or (6), and the number Bl_{turb} by the formula (7).^{4,14}

$$k_{turb} = k_{trans} \left(\frac{\sqrt{C_p}}{V} \right)^{-X} \quad (5)$$

where $(-X)$ is an indicator of degree:^{4,14}

$$(-X) = \frac{\ln \frac{a \cdot \sqrt{2 Re}}{0,769 \cdot 1 / Bl}}{\ln \frac{\sqrt{C_p}}{V}}$$

Turbulent thermal conductivity k_{turb} can also be calculated using the known ratio (6)

$$k_{turb} = k_{trans} \cdot Bl_{turb} = \mu \cdot a \cdot \sqrt{2 Re} \cdot C_p \quad W / m \cdot K \quad (6)$$

where k_{turb} is coefficient of average turbulent thermal conductivity, W/m·K; k_{trans} is coefficient of average transitional thermal conductivity in LBL, W/m·K; Bl_{turb} is a turbulent dimensionless number; $a \approx (0.05-0.08)$ is an experimental coefficient for air in the middle of the flow, where turbulence is considered free.¹⁵

The Bl_{turb} number can be calculated by formulas (7) or (8).^{4,14}

$$Bl_{turb} = k_{turb} / k_{trans} \quad (7)$$

$$Bl_{turb} = \left(\frac{\sqrt{C_p}}{V} \right)^{-X} \quad (8)$$

The transitional thermal conductivity k_{trans} in the transitional zone of the LBL can be calculated using the formula (9).¹⁰

$$k_{trans} = \sigma \cdot \cos \theta_{trans} \cdot \sqrt{C_p} \left[\frac{N \cdot m}{m \cdot s \cdot K} = \frac{W}{m \cdot K} \right] \quad (9)$$

When analyzing literary sources regarding the correctness of calculating heat transfer processes using classical empirical numerical equations of the type (1), we found the following main problems and questions:

– if we abstract from the question of the use of aqueous solutions of glycols at temperatures below 273 K, then the question of whether water is always the most optimal main heat carrier in terms of its thermophysical properties turned out to be debatable?

– classical similarity numbers, and therefore classical empirical numerical equations of the type (1) contain molecular thermophysical characteristics, in particular viscosity and thermal conductivity in static conditions of the heat carrier, and all heat exchange equipment works in a dynamic mode, where transient and turbulent viscosity and thermal conductivity at several orders of magnitude higher than the molecular (static) ones;

The purpose of the study is to improve the method of calculating heat exchange equipment taking into account surface forces in heat carriers, in particular, taking into account the average thickness of the LBL based on the structuring of the fluid flow into turbulent, transitional and LBL zones. Also, the purpose of the work is to show the peculiarities of calculating heat exchange equipment by classical and proposed methods using glycols and "ice water".

2. Computer Simulation of Thermophysical Parameters of the Heat Transfer in Different Zones of Its Flow

2.1. Theoretical Aspects of the Analytical Approach to Constructing the Equation of Convective Heat Transfer Taking into Account Surface Forces

In this case, we depart from the classical approach to this heat transfer system in the closed space of the channel (pipe). The classical approach involves dividing the flow of the liquid coolant into a two-layer system with the LBL and a turbulent part. Heat from the wall is transferred by heat conduction through the LBL, according to the Fourier's law. The same amount of heat through the turbulent part is transferred by convection according to the Newton's law, where the coefficient of heat transfer (convection) appears. Neglecting heat losses and equating these heats, we obtain the classical equation at the interface of the LBL and the turbulent part (10):

$$k \frac{dt}{d\delta} = h \cdot \Delta t \quad (10)$$

where k is thermal conductivity, W/m·K; h is heat transfer coefficient, W/m²·K; T is temperature, K; δ is average layer thickness, m.

From this equation, using the similarity theory, the classic Nusselt number (11) is obtained:

$$Nu = \frac{h \cdot \delta}{k} \quad (11)$$

Next, classical numerical equations of the type (1) are composed, where the unknown quantities are obtained experimentally.

Instead, we divided the left and right parts of equality (10) into zones L and T of the movement of the heat carrier, using, respectively, the transitional thermal conductivity k_{trans} and the turbulent thermal conductivity k_{turb} of these zones. In the right-hand side of equation (10), we replaced the heat transfer (convection) coefficient h , which, according to the research results of some authors, does not fully reflect the real picture of heat exchange, with the turbulent thermal conductivity of the turbulent part of the flow from Eqs. (5) or (6). In addition, in the left-hand side of equality (10), we introduced the so-called transitional thermal conductivity in the LBL from equality (9). From equation (10) we get (12):

$$k_{trans} \frac{dt}{d\delta} = k_{turb} \frac{\Delta t}{r - d\delta} \quad (12)$$

where r is channel (pipe) radius, m.

Since, by definition, the turbulent thermal conductivity of the k_{turb} flow is the product of the transitional thermal conductivity in the transition layer LBL by the number Bl_{turb} , then, after getting rid of the signs of differentiation, we enter the number Bl_{turb} in the right-hand side of equality.¹⁰ We also neglect temperature changes in the flow of the coolant, which are practically not noticeable. Then equality (12) takes the form (13):

$$\frac{k_{trans}}{\delta_{LBL}} = \frac{k_{turb}}{r - \delta_{LBL}} = \frac{k_{trans} \cdot Bl_{turb}}{r - \delta_{LBL}} \quad (13)$$

where δ_{LBL} is the average thickness of the LBL, m.

From equality (13), the average thickness of the LBL δ_{LBL} can be written as (14):

$$\frac{1}{\delta_{LBL}} = \frac{Bl_{turb}}{r - \delta_{LBL}} \quad \delta_{LBL} = \frac{r}{Bl_{turb} + 1} \quad (14)$$

As a result, we obtain the formula (15) for determining the heat transfer coefficient taking into account the thermal conductivity of the LBL and the turbulent zone of the coolant flow:

$$h_{NEW} \approx \left(\frac{\delta_{LBL}}{k_{trans}} + \frac{r - \delta_{LBL}}{k_{turb}} \right)^{-1} \quad (15)$$

where h_{NEW} is heat transfer coefficient, calculated by the new method, W/m²·K.

2.2. Methodology for Calculating the Heat Transfer Coefficient Taking into Account the Thermophysical Characteristics of the Turbulent and Transitional Zones of Heat Carriers

For further studies, as usual, water was chosen as the reference coolant, as well as a 45% aqueous solution of propylene glycol in the temperature range (273–353) K. At the same time, the linear speed of the heat carrier $V=0.927$ m/s is adopted for the normalized shell-and-tube heat exchanger. The inner diameter of the pipes of the normalized heat exchanger was $D=0.021$ m.

The volumetric consumption of heat carrier, respectively, was $W=0.3208 \cdot 10^{-3}$ m³/s.

Classical equations of heat exchange, as well as constant and variable classical complexes for the tube space of a shell-and-tube heat exchanger in an expanded form look like this:

$$Nu = 0,17 \cdot Re^{0,33} \cdot Pr^{0,43}$$

$$h \approx 2,2065 \rho^{0,33} \cdot k^{0,57} \cdot C_p^{0,43} \cdot \mu^{0,1} \text{ W/m}^2\text{K} \quad (16)$$

for laminar mode:

$$Nu = 0,008 \cdot Re^{0,8} \cdot Pr^{0,43}$$

$$h \approx 0,0163 \frac{\rho^{0,8} \cdot k^{0,57} \cdot C_p^{0,43}}{\mu^{0,37}} \text{ W/m}^2\text{K} \quad (17)$$

for the transitional mode

$$Nu = 0,021 \cdot Re^{0,8} \cdot Pr^{0,43}$$

$$h \approx 0,0428 \frac{\rho^{0,8} \cdot k^{0,57} \cdot C_p^{0,43}}{\mu^{0,37}} \text{ W/m}^2\text{K} \quad (18)$$

for the turbulent mode.

In order to obtain the most transparent picture of the influence of all factors on the heat exchange process,

we have proposed variable classical complexes in the form (16), which under the conditions of computer analysis take into account all variable thermophysical characteristics of coolants depending on temperature.

For computer modeling using the new method, we used known ratios derived by us in previous works in the following sequence:

1. Number Bl according to formula (3).
2. Transitional thermal conductivity in the LBL according to formula (9).
3. Reynolds number.
4. Turbulent thermal conductivity from formulas (5) or (6).
5. Turbulent number Bl_{turb} from formulas (7) or (8).
6. Average thickness of LBL according to formulas (14).
7. Heat transfer coefficient according to formula (15).

3. Results and Discussion

The heat exchanger has the following parameters: $d=0.6$ m, $z=4$, $n/z=51.5$; $n=206$, $L=6$ m. The inner diameter of the pipes $D=21 \cdot 10^{-3}$ m. The heat transfer coefficient is calculated according to classical equations (16–18), as well as according to the new relation (15) for water (H₂O) and the 45% aqueous solution of propylene glycol C₃H₆(OH)₂. Thermophysical characteristics of the 45% aqueous solution of propylene glycol are taken from the handbook.¹⁵

At the beginning, the computer modelling of the indicated coolants was carried out at the same speed of movement in the tube space of the shell-and-tube heat exchanger $V=0.927$ m/s or flow rate

$$W=0.3208 \cdot 10^{-3} \text{ m}^3/\text{s} \text{ (Fig. 1).}$$

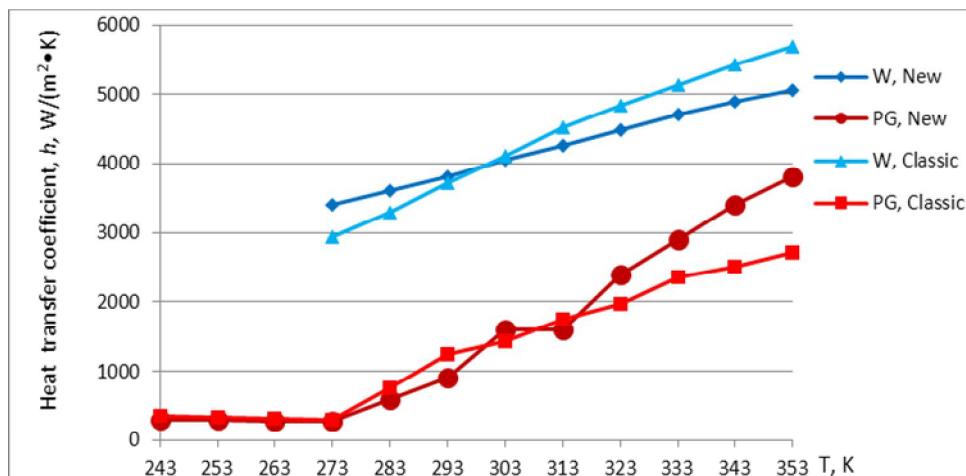


Fig. 1. Dependence of the heat transfer coefficient for water and an aqueous solution of propylene glycol (45%) from equations (15) and (16) on the temperature of the heat carrier (average flow velocity – $V=0.927$ m/s; diameter of the tubes of the shell-and-tube heat exchanger $D=0.021$ m. consumption $W=0.3208 \cdot 10^{-3}$ m³/s)

It can be seen from Fig. 1 that the classical and proposed methods of calculating the heat transfer coefficients correlate well with each other. At temperatures close to 273 K, the proposed method shows slightly higher values of heat transfer coefficients for glycol, and at high temperatures close to 353 K, slightly higher values for water. In the temperature range $T \approx (303-313)$ K, the graphs cross each other, which indicates the identity of the calculations by both methods. At the same costs, "ice water" shows approximately an order of magnitude higher heat transfer coefficients. Such a situation occurs because at $T \approx 273$ K and with the same speed of movement $V = 0.927$ m/s and consumption, "ice water" moves in the T mode, and glycol - in the L mode due to its very high viscosity. That is, in glycol, the amount of heat spreads due to only transitional thermal conductivity

$$k_{transPG} = \sigma \cdot \cos \theta_{trans} \cdot \sqrt{C_p} = 47,58 \cdot 10^{-3} \cdot 0,99 \cdot \sqrt{3560} = 2,81 W / m \cdot K$$

throughout the "live section" of the flow. Under these conditions, water is divided into LBL with an average thickness

$$\delta_{LBL} = r / (Bl_{turb} + 1) = 0,0105 / (13,74 + 1) = 0,713 \cdot 10^{-3} m$$

with transitional thermal conductivity

$$k_{transH_2O} = \sigma \cdot \cos \theta_{trans} \cdot \sqrt{C_p} = 75,64 \cdot 10^{-3} \cdot 0,99 \cdot \sqrt{4217} = 4,86 W / m \cdot K$$

and a turbulent part with turbulent heat conduction

$$k_{turb} = k_{trans} \cdot Bl_{turb} = \mu \cdot a \cdot \sqrt{2Re} \cdot C_p = 1,788 \cdot 10^{-3} \cdot 0,06 \cdot \sqrt{2 \cdot 10885} \cdot 4217 = 66,76 W / m \cdot K$$

With $Bl_{turb} = k_{turb} / k_{trans} = 66,76 / 4,86 = 13,74$ (used formulas 7–9, 14). However, water cannot cross the 273 K boundary. An aqueous 45% solution of propylene glycol $C_3H_6(OH)_2$ can work up to (243) K. The thermophysical parameters of this glycol are given in (Table).

As can be seen from Table, both calculation methods are sufficiently correlated with each other for L mode of movement of the coolant. However, if you give the heat transfer medium transit mode of motion, and calculate the heat transfer coefficients according to the classical scheme for the transit mode (Table, column 8), the picture changes significantly, since the heat transfer coefficient decreases with a decrease in temperature in the interval $T \approx (273-303)$ K. In this range, under the transit mode, the heat transfer coefficient significantly depends on the dynamic viscosity, which increases by almost an order of magnitude (Table, column 5). This means that if the heat carrier is given the mode of transitional movement, then a number of problematic issues arises:

- the heat carrier by gravity cannot reach the speed of movement $V = 0.927$ m/s;
- additional energy costs for electric discharge pumps;
- reduction of heat transfer coefficients.

Under these conditions, *i.e.*, under the transitional and turbulent modes of movement of the heat carrier, guided by the classic calculation scheme, an increase in the dynamic viscosity of the heat carrier will inevitably lead to loss of heat transfer coefficients, and such a decision is not appropriate. However, it is known that in the temperature range $T = (323-343)$ K, where nanoparticles are added to aqueous solutions of glycols and at the same time the viscosity of the solutions increases significantly, heat transfer coefficients also increase.¹⁶ This fact cannot be explained based on classical considerations and using classical Eq. (15). We explain this behavior of nanofluids by the fact that classical equations are not sensitive to the action of surface forces, which become dominant in such systems when nanoparticles are used. We made a detailed analysis of this phenomenon in a number of previous works.^{4,14} In this sense, the new method of calculating heat transfer coefficients seems to us to be more appropriate.

Table. The main thermophysical properties and heat transfer coefficients for the L mode of heat carrier movement, calculated according to the classic and new methods for the 45 % aqueous solution of propylene glycol at minus temperatures

Temperature [K]	Density ρ kg/m ³	Heat capacity C_p [kJ/kgK]	Thermal conductivity, K , [W/mK]	Dynamic viscosity μ , [N.s.m ⁻²]	Surface tension σ , [N/m]	Heat transfer coefficient $Re < 2320$ from equality (16) for the L regime (classic)	Heat transfer coefficient $10000 > Re > 2320$ (from equality (17) for Trans mode)	Heat transfer coefficient from equations (14, 15) (new)	Reynolds number Re
243	1066	3.45	0.397	160	54.41	359.4	166.3	301.3	130
253	1062	3.49	0.396	74.3	52.11	333.9	221.4	290.3	278
263	1058	3.52	0.395	31.74	49.81	307.4	304.3	278.6	649
273	1054	3.56	0.395	18.97	47.58	291.9	368.7	267.7	1082

In order to understand the advantages of using glycol over "ice water" under these conditions, *i.e.*, its self-gravity movement under the action of gravity alone (without additional injection by an electric pump), we brought both coolants at the same temperatures to similar Reynolds numbers, recalculating the injection speed of "ice water". The results are presented in Fig. 2.

It can be seen from Fig. 2 that at the same Reynolds numbers, propylene glycol behaves better than water in the temperature range $T \approx (273-353)$ K. Calculations according to both methods differ maximally in the transitional regime of heat carrier movement (Fig. 2), when the classical equations significantly depend on the molecular viscosity of the heat carrier (15), and the new method provides for a combination of transient and turbulent thermal conductivities of individual zones in this range. The exception is the range $T \approx 273-303$ K, where both methods are maximally correlated. This happens

because the classical equations for the L regime depend very little on the molecular viscosity, where this value is to the power of 0.1. The proposed equality (8), on the other hand, depends not on the molecular viscosity, but on the (imaginary) transitional viscosity (2) in the LBL. Transitional (apparent) viscosity is a function of the coefficient of surface tension of the heat carrier and specific heat capacity (2). It is obvious that the transitional thermal conductivity of the coolant (8) is determined by the product of transitional viscosity and specific heat capacity.

In addition, the heat transfer coefficients calculated according to the new scheme practically coincide in the temperature range $T \approx 303-313$ K. This fact is a confirmation that in the process of calculating heat exchangers, the average logarithmic temperature difference rarely exceeds this range. Therefore, for aqueous solutions, the viscosity of which does not differ significantly from the viscosity of water, the classical calculation fully satisfies practitioners.

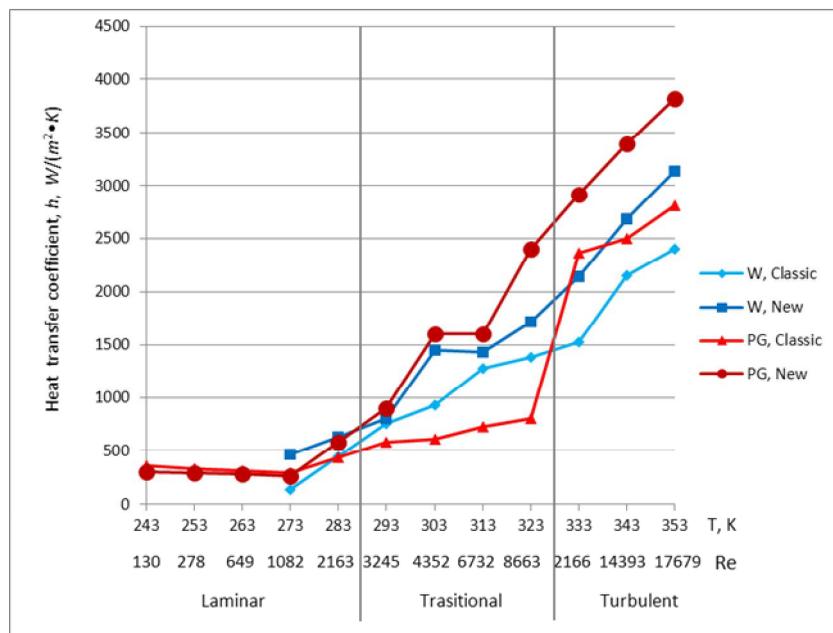


Fig. 2. Dependence of the heat transfer coefficient for water and an aqueous solution of propylene glycol (45%), according to equations (15 and 16), on the temperature and Reynolds numbers of the heat carriers (diameter of the tubes of the shell-and-tube heat exchanger $D = 0.021$ m)

It can be seen from Fig. 2 that in the temperature range $T \approx 313-353$ K at similar Reynolds numbers, the heat transfer coefficients calculated by classical equations (16) are significantly lower than those calculated by the new method using equation (15). This happens because classical equalities and numbers are insensitive to surface forces in a fluid flow. For the classical calculation, there is a characteristic jump in the heat transfer coefficient for glycols at a temperature of $T \approx 328$ K. At this temperature, the transitional mode changes to T mode for glycol. Accordingly,

the calculation equation (16) changes from transitional to turbulent regime. Such a noticeable jump is not observed under the new method (Fig. 2).

The classic Nusselt, Reynolds, and Prandtl similarity numbers do not contain surface characteristics. At relatively low temperatures in the flow of the 45% aqueous solution of propylene glycol, transient thermal conductivity prevails due to high viscosity. At 273 K, the flow moves in L mode, there is no turbulent heat conduction in it. At higher temperatures of about 283 K, light

turbulence begins in the central part of the pipe, and closer to the wall is the L-zone. Roughly, for half the radius of the "live section" of the flow adjacent to the pipe wall, the heat transfer coefficient is calculated according to the left part in brackets of equality (15), and for the second half - according to the right part in brackets (15). The result is the total heat transfer coefficient of these zones (15). As the temperature increases, the T-zone increases, and the L-zone decreases, which can be seen from formula (14) for determining the average thickness of the LBL zone. The result of this turbulence is an increase in the average thickness of the T-zone due to a decrease in the average thickness of the L-zone. At the same time as the temperature increases, the transitional thermal conductivity decreases, and the turbulent one increases. The graphs of the heat transfer coefficient, which were calculated according to the new method in the temperature range ($T \approx 303\text{--}313\text{ K}$), pass through the extremum caused by the turbulence of the flow when the temperature rises. Up to this range, the turbulence is still very insignificant and occupies up to 2/3 of the radius of the "live section" of the flow. Transitional thermal conductivity "wins" here. In the temperature range ($T \approx 303\text{--}313\text{ K}$), the transitional thermal conductivity decreases approximately in proportion to the growth of turbulent thermal conductivity. Above this range, the turbulent heat conduction "wins".

The classic choice of heat exchange equipment is based on the fact that the average temperatures of heat carriers, which differ significantly at the entrance to the heat exchanger and at the exit, are first calculated. In addition, at the stage of selecting a normalized heat exchanger, the average logarithmic temperature difference (the driving force of the heat exchange process) is calculated, where these temperatures are averaged again. As a result, the calculated temperature practically does not exceed $T \approx (303\text{--}313)\text{ K}$. In this range, the classical calculation and the one proposed by us are very similar, and at temperatures $T \approx 308\text{ K}$ they are identical (Fig. 1). In addition, when choosing a heat exchanger, the thermal resistance of contaminants is taken into account in the overall heat transfer coefficient, which is about half of the total thermal resistance of the system.¹⁴ Therefore, according to the calculation of the overall heat transfer coefficient by the classical method, using the heat transfer coefficients of both heat carriers, as well as the specific resistance of the dividing wall with impurities on both sides, its correctness and accuracy in precession heat exchangers (in computer equipment for cooling microboards, or for heating in solar systems), especially for the use of nanofluids, wants to be better. We are sure that this method of calculating heat transfer coefficients can be used in combination with mass transfer processes, in particular, during filtration drying, where a powerful field of surface forces acts in the capillaries of the volumetric mass of granulated or crushed material.¹⁷

4. Conclusions

1. It is shown that the classical numerical equations containing the Nusselt, Reynolds, and Prandtl numbers are not sensitive to changes in the surface characteristics, especially in the LBL zone of liquid coolants.

2. A semi-analytical approach to the solution of the heat exchange equation is proposed, taking into account the transitional and turbulent thermal conductivity of the "live cross section" of the coolant flow, as well as the average thickness of the heat exchanger.

3. The calculation according to the proposed method showed that the 45% aqueous solution of propylene glycol at the temperatures of 273–353 K and similar Reynolds numbers has higher heat transfer coefficients compared to water, while the classical calculation shows the opposite.

4. Proposed equality (15) for the semi-analytical calculation of heat transfer coefficients in liquids taking into account the thermal conductivity coefficients of the transitional and turbulent zones of the "live cross-section" of the flow and the average thickness of the LBL.

5. A new analytical formula (14) is proposed for determining the average thickness of the LBL, which contains the geometric radius of the "live section" of the coolant flow, as well as the Bl_{urb} similarity number we obtained earlier.

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ОСОБЛИВОСТІ РОЗРАХУНКУ КОЕФІЦІЄНТІВ ТЕПЛОПЕРЕДАЧІ ЗА ВИКОРИСТАННЯ ГЛІКОЛІВ З УРАХУВАННЯМ ПОВЕРХНЕВИХ СИЛ ТЕПЛОНОСІЯ

Анотація. У цьому дослідженні порівнювали класичний метод розрахунку коефіцієнтів тепловіддачі трубного простору кожухотрубного теплообмінника за класичними числами подібності Нуссельта, Рейнольдса і Прандтля з новим методом, який враховує коефіцієнти поверхневого натягу теплоносіїв, їхні перехідні, турбулентні в'язкість і теплопровідність, а також середню товщину ламінарного прилежового шару (ЛППШ). Класичний метод показує кращу ефективність води як теплоносія в порівнянні з 45% водним розчином пропіленгліколю. Натомість нова методика розрахунку показує, що 45% водний розчин пропіленгліколю має вищі коефіцієнти тепловіддачі порівняно з водою в діапазоні температур (273...353) К. «Живий переріз» потоку рідинного теплоносія ми розділили на ЛППШ середньої товщини, де застосовується рівняння теплопровідності Фур'є, і на його турбулентну частину, де також застосовується рівняння теплопровідності з турбулентною теплопровідністю. Запропоновано нову формулу для розрахунку середньої товщини ЛППШ на основі радіуса «живого перерізу» потоку теплоносія, а також числа подібності Bl_{turb} , отриманого нами в попередніх роботах.

Ключові слова: перехідна, турбулентна в'язкість і теплопровідність; кожухотрубний теплообмінник; коефіцієнт тепловіддачі; середня товщина ЛППШ; коефіцієнт поверхневого натягу теплоносія.