Chem. Chem. Technol., 2024,
Vol. 18, No. 3, pp. 409–416

Chemical Technology

THE METHOD OF CALCULATING THE HEAT TRANSFER COEFFICIENT IN THE HELIOSYSTEMS WITH LAMINAR AND TRANSIENT MODES OF HEAT CARRIER FLOW MOVEMENT STRUCTURED INTO PARTS

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https://doi.org/10.23939/chcht18.03.409

Abstract. In this study, a new method of choosing classical empirical equations for calculating heat transfer coefficients in the tubes of a shell-and-tube heat exchanger in the transient mode is proposed. This method is based on the fact that the flow is structured into a laminar boundary layer (LBL) zone and a turbulized part, and the heat transfer coefficient is calculated through the transient and turbulent heat conductivity, as well as the average thickness of the LBL and, accordingly, the average thickness of the rest of the coolant flow. At the same time, the key point of this method is the condition that the transient thermal conductivity of the LBL should be lower than the thermal conductivity of the turbulized part. If this condition is not fulfilled, it is concluded that the corresponding classical empirical equation is not suitable for calculating the heat transfer coefficient. A 45 % aqueous solution of propylene glycol was taken as a model liquid, which can be widely used in solar collectors, in particular with nanofillers. This coolant is interesting because at a constant speed of $V = 0.93$ *m/s*, and the linear size (diameter) of the "live section" of the flow D = 0.021 m in the temperature range of 243–273 K it moves in the laminar mode, in the temperature range of 283–323 K – in transient mode and 333–353 K – in turbulent mode. A new formula is proposed for calculating the coefficient of turbulence of the coolant flow a, the numerical values of which are experimentally found in literary sources only for the air coolant.

Keywords: transient, turbulent viscosity and thermal conductivity, heat transfer coefficient, average thickness of LBL, coefficient of surface tension of the coolant, transient mode.

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1. Introduction

Due to the high prices of fossil energy sources, as well as environmental problems, alternative energy has gained wide popularity, in particular, the use of various solar systems.

Aqueous solutions of glycols are widely used in such systems since the key point in such cases is the non-freezing of the working fluid at sub-zero temperatures, that is, the possibility of their uninterrupted operation in winter. These and other factors impose certain restrictions and features on the operation of such solar systems, the main of which are the following:

– the use of coolants that are resistant to sub-zero winter temperatures, in particular, aqueous solutions of glycols;

– operation of solar systems with natural circulation, and therefore with a low flow rate of the heat carrier, where laminar (*L*) or transient (*Tr*) modes of movement prevail;

– relatively high values of dynamic viscosity coefficients of coolants;

– the possibility of using nanofluid coolants to increase heat transfer coefficients.

These factors create prerequisites for the needs of non-traditional, *i. e.*, non-classical thermal and hydraulic calculation and selection of such solar systems, and therefore the use of specific approaches and laws. At the same time, in modern heat transfer technologies, systems are often found in which the aforementioned surface forces have a significant impact on the heat transfer coefficient. This is the case, for example, when using liquid heat transfer fluids with nanofillers or surfactants, or in microchannels where powerful capillary forces are present.

According to studies by various authors (including ours), classical calculation methods (especially for shell-and-tube and plate heat exchangers) do not work in such cases, that is, they give an incorrect estimate of the heat transfer coefficient. We reviewed the literature on this issue in our previous works[1, 2]. The vast majority of researchers followed the path of modifying classical empirical formulas, both for shell and tube[3–7] and for plate[8–10] heat exchangers, that is, they use classic Nusselt, Reynolds, and Prandtl similarity numbers with various corrections for the fractional and percentage composition of nanofillers.

We proposed an approach that has several significant differences from the classical one. In particular, the effect of surface forces characteristic of the heat carrier is included in the calculation of the heat transfer coefficient (based on dimensional theory). Our proposed method uses, in particular, turbulent physical characteristics of the coolant flow[11–13.] In addition, two parts (the wall layer – LBL) and the rest of the flow are separated in the flow, and for each of them, the heat transfer is calculated according to the formulas, according to the mode in each part of the coolant flow[14]. Another significant difference is that this method is semi-analytical[14].

During our substantiation of this approach, several problems arose, which prompted us to deepen the scientific search. In more detail, the way of forming an idea is as follows.

During the analysis of literary sources on this issue, many contradictions are observed. For the calculation of heat exchange equipment, in particular, heat transfer coefficients in the *Tr* mode, the Gnielinski equation[15] (1) is widely used. At the same time, the coefficients of hydraulic friction are recommended to be calculated according to equation (2) for turbulent (*T*) or *Tr* modes, respectively[16–22]

$$Nu = \frac{(0.125 \cdot f)(\mathrm{Re}-1000)\cdot \mathrm{Pr}}{1+12.7\cdot(0.125\cdot f)^{0.5}\cdot(\mathrm{Pr}^{0,66}-1)} \tag{1}$$

$$f = (0.79 \ln Re - 1.64)^{-2} \tag{2}$$

Equation (3)[20] is also widely used to calculate the heat transfer coefficient during the movement of the coolant in the *Tr* mode. In addition, some authors present in equation (3) the power indicator at a Reynolds number of 0.8, *i. e.*, equation (4)[21] is used for the calculation:

$$Nu = 0.008 \cdot Re^{0,9} \cdot Pr^{0,43} \tag{3}$$

$$Nu = 0.008 \cdot Re^{0,8} \cdot Pr^{0,43} \tag{4}$$

Such, at first glance, an insignificant change in the degree of 0.9 or 0.8 leads to a significant change in the heat transfer coefficient. Thus, an example of calculating the heat transfer coefficient for a heat carrier – 45 % aqueous solution of propylene glycol according to the classic method (T = 293 K, V = 0.93 m/s, D = 0.021 m)[14] according to equations (3) and (4), respectively, give completely different values and differ by more than 2 times, for example[14]:

Reynolds number:

$$\mathrm{Re} = \frac{V\cdot D\cdot \rho}{\mu} = \frac{0.93\cdot 0.021\cdot 1044}{6.264\cdot 10^{-3}} = 3255 \quad \mathrm{Tr\ \ regime}$$

The variable complex is calculated according to the classical heat transfer equation for the tube space of a normalized shell-tube heat exchanger from equality (3) and (4) in accordance:

$$h_{(\text{from equality (3)})} \approx 0.011\frac{\rho^{0,9}\cdot k^{0.57}\cdot C_P{}^{0,43}}{\mu^{0.47}} =$$

$$= 0.011\frac{1044^{0,9}\cdot 0.394^{0.57}\cdot 3620^{0,43}}{(6.264\cdot 10^{-3})^{0.47}} = 1240, \left[W/m^2\cdot K\right].$$

$$h_{(\text{from equality (4)})} \approx 0.0163\frac{\rho^{0,8}\cdot k^{0.57}\cdot C_P{}^{0,43}}{\mu^{0.47}} =$$

$$= 0.0163\frac{1044^{0,8}\cdot 0.394^{0.57}\cdot 3620^{0,43}}{(6.264\cdot 10^{-3})^{0.47}} = 552, \left[W/m^2\cdot K\right].$$

Classical empirical formulas for calculating heat transfer coefficients in the *Tr* mode of motion (1), (3), (4) are insensitive to changes in the coefficient of surface tension of the coolant. Therefore, when we use aqueous solutions of glycols as heat carriers, as well as with the addition of nanoparticles, these equations cannot always be used to calculate the heat exchanger, in particular in solar collectors. This happens because the glycols themselves significantly change the coefficient of surface tension of water in the direction of its decrease, and also the nanofilling itself also affects its change. As a result, classical equations of the type (1), (3), (4) mostly do not reflect the real picture of heat transfer when using aqueous solutions of glycols with nanofillers, especially in the transient mode of heat carrier movement. At the same time, researchers need to conduct expensive experiments to find appropriate numerical empirical equations, which mostly take on very complex and cumbersome forms[1, 2].

Equality (5) is widely used to calculate heat transfer coefficients in the turbulent mode F. W. Dittus, L. M. K. Boelter[22]:

$$Nu = 0.021 \cdot Re^{0.8} \cdot Pr^{0.43}. \tag{5}$$

We proposed the concept of consideration of the movement of liquids in pipelines and channels taking into account surface forces[11, 12]. Next, we proposed to calculate the heat exchange equipment taking into account surface forces[12]. In the same work, we derived the number *Bl*, which is the ratio of the product of the coefficient of dynamic viscosity of the liquid coolant by the root of the specific heat capacity to the coefficient of surface tension, taking into account the hydrophilicity of the wetting surface (6)[12].

$$Bl = \frac{\mu \cdot \sqrt{C_P \cdot 1K}}{\sigma \cdot \cos\theta_{trans}}, \quad (6)$$

where Bl – dimensionless number; μ – coefficient of dynamic viscosity of the coolant, kg/m·s; $\cos\theta_{trans}$ – cosine of the angle (Surface hydrophilicity); C_P – specific heat capacity of the coolant, J/kg·K; σ – coefficient of surface tension of the coolant, N/m.

In the following paper[13], we derived the turbulent number Bl_{lturb}, which is the ratio of turbulent viscosity to transient viscosity (or the ratio of the corresponding thermal conductivities) (7):

$$Bl_{turb.} = \frac{\mu_{turb.} \cdot \sqrt{C_P \cdot 1K}}{\sigma \cdot \cos\theta_{trans}}, \quad (7)$$

where Bl_{turb} – dimensionless turbulent number; μ_{turb} – coefficient of turbulent viscosity of the coolant, kg/m·s.

Since the liquid coolant in a non-laminar flow can be conditionally divided into the LBL zone and the turbulized part, we have proposed a method and formula for the semi-empirical calculation of the heat transfer coefficient[14]. (Turbulized flow is interpreted as such, the central part of which is in a transient (Tr) or turbulent (T) mode). The purpose of this study is to substantiate the choice of classical equations for calculating the heat transfer coefficient in the Tr mode based on the approach of structuring the flow of the liquid coolant on the LBL and the turbulized part.

2. Experimental

Since each flow of a liquid coolant in the T mode can be conventionally divided into a LBL zone and a turbulent part, we have proposed a technique and formula for semi-empirical calculation of heat transfer coefficients[14].

It should be assumed that the amount of heat that passes through the LBL is approximately equal to the amount of heat that passes through the turbulent part. It is obvious that the higher the speed of the coolant, the higher the turbulent thermal conductivity of the turbulent part (Fig. 1), and at the same time, the average thickness of LBL[14] becomes smaller (8):

$$\frac{k_{trans}}{\delta_{LBL}} = \frac{k_{turb}}{r - \delta_{LBL}} = \frac{k_{trans} \cdot Bl_{turb}}{r - \delta_{LBL}} \quad (8)$$

At the same time, we derived an analytical dependence for calculating the average thickness of LBL[14] (9), as well as formula (10) for analytical calculation of the heat transfer coefficient taking into account the thermal conductivity of LBL and the turbulent part of the coolant flow[14]:

$$\delta_{LBL} = \frac{r}{Bl_{turb} + 1}, \quad (9)$$

$$h_{NEW} \approx \left(\frac{\delta_{LBL}}{k_{trans}} + \frac{r - \delta_{LBL}}{k_{turb}} \right)^{-1}. \quad (10)$$

where δ_{LBL} is the average thickness of the LBL, m; r – channel (pipe) radius, m; k_{turb} – coefficient of average turbulent thermal conductivity, W/m·K; k_{trans} – coefficient of average transient thermal conductivity in LBL, W/m K; h_{NEW} – heat transfer coefficient, calculated by the new method, W/m^2·K.

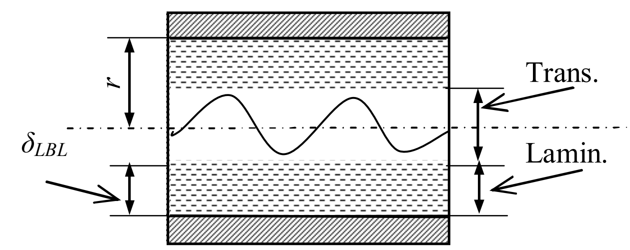

Fig. 1. Scheme of the movement of a liquid heat carrier in transitional modes with the formation of LBL

We analytically calculate the thermal conductivity of LBL k_{trans} on the basis that the turbulent Prandtl number in LBL, which is numerically equal to the number Bl, is equal to unity[12]. Based on the number Bl of formula (6), the transient (apparent) viscosity of the coolant flow in the LBL is equal to (11), and, accordingly, the transient thermal conductivity in the LBL is equal to (12):

$$\mu_{trans} = \frac{\sigma \cdot \cos\theta_{trans}}{\sqrt{C_P \cdot 1K}} = \left[\frac{N \cdot s \cdot \sqrt{1K}}{m \cdot m \cdot \sqrt{1K}} = Pa \cdot s \right] \quad (11)$$

$$k_{trans} = \mu_{trans} \cdot C_P = \sigma \cdot \cos\theta_{trans} \sqrt{C_P} = \left[\frac{N \cdot m}{m \cdot s \cdot K} = \frac{J}{m \cdot s \cdot K} = \frac{W}{m \cdot K} \right] \quad (12)$$

The turbulent thermal conductivity k_{turb} of the turbulent part of the coolant flow can be calculated from equation (13)[12]:

$$k_{turb} = k_{trans} \cdot Bl_{turb} = \mu \cdot a \cdot \sqrt{2\mathrm{Re}} \cdot C_P, \ [W/m \cdot K]; \quad (13)$$

where $a \approx (0.05 - 0.08)$ is an experimental coefficient for air in the middle of the flow, where turbulence is considered free[23]. At the same time, we could not find in the literature an experimentally determined value of the coefficient a for different liquids.

3. Results and Discussion

To simulate the calculation of heat transfer coefficients in the flow of a liquid coolant, we chose a 45 % aqueous solution of propylene glycol, because this coolant can be widely used in solar collectors, in particular with nanofillers. In addition, this coolant is convenient in that at a constant speed of $V = 0.93$ m/s and a linear size (diameter) of the flow $D = 0.021$ m in the temperature range of 243–273 K, it moves L, in the temperature r*ange of 283–323 K – for the Tr mode and 333–353 K –* for the *T* mode. Thermophysical and hydromechanical properties and values of this coolant are presented in Table.

After considering the formula (13), it can be seen that the unknown quantity is the coefficient a, which was experimentally determined in the air environment. It is known that there is a complete analogy between aerodynamic and hydrodynamic relationships. However, for aqueous solutions, the coefficient a has not been determined experimentally in the literature.

Dividing the flow of the liquid coolant conditionally into two zones and deriving the formula for calculating the heat transfer coefficient (9), we theoretically calculated the coefficient a for our model coolant (Table) according to the formula (14), using the known empirical dependencies (1), (3), (4), which are based on a large array of experimental data of many authors. Taking into account the relations (5)–(13) derived by us, we present the equation for the theoretical determination of the coefficient a (14):

$$a \approx \frac{1}{\sqrt{2\,\mathrm{Re}}}\left(\frac{r \cdot h_{classic}}{2 \cdot \mu \cdot C_P} - \frac{1}{Bl}\right), \tag{14}$$

where $h_{classic}$ is the heat transfer coefficient, which can be calculated according to various classical empirical equations of the type (1), (3), (4) and others, W/m^{2}K.

Table. Thermophysical properties of a 45% aqueous solution of propylene glycol, crystallization temperature (273 K); ($V = 0.93$ m/s, $D = 0.021$ *m*)*, **, ***

Temperature, [*T*,*K*]	Density, [Kg/m^3]	Heat capacity C_p, [*J*/*kg*·*K*]	Thermal conductivity, *k*, [*W*/*m*·*K*]	Dynamic viscosity $\mu \cdot 10^{-3}$, [$N{\cdot}s/m^2$]	Surface tension $\sigma \cdot 10^{-3}$, [*N*/*m*]	Reynolds number. Mode, *Re*	Prandtl Number, *Pr*	Bilonoga number, *Bl* from eq. (6)	Bilonoga number turbulent, Bl_{turb} from eq. (7)	*a*, according to the ratio (14) using of equations (3 or 4)	*a* according to the ratio (14) using of equation (1)	k_{trans}, [*W*/*m*·*K*] from equation (12)	k_{turb}, [*W*/*m*·*K*] from equation (13)	$h_{classic}$, from equations (3 or 4)
243	1066	3.45	0.397	160	54.41	130 (L)	1390	172.7	0.00	0.00	0.00	3.16	0.00	359.4
253	1062	3.49	0.396	74.3	52.11	278 (L)	654.8	84.23	0.00	0.00	0.00	3.05	0.00	333.9
263	1058	3.52	0.395	31.74	49.81	649 (L)	282.8	37.8	0.00	0.00	0.00	2.93	0.00	307.4
273	1054	3.56	0.395	18.97	47.58	1082 (L)	170.9	24.03	0.00	0.00	0.00	2.81	0.00	291.9
293	1044	3.62	0.394	6.264	45.36	3245 (Tr)	57.5	8.39	**1.40**	0.0021 0.0001	0.0013	**2.72****	**3.82**** 0.2	**1240***** 552
303	1033	3.66	0.3935	4.621	43.15	4352 (Tr)	42.98	6.54	**1.90**	**0.0031*** 0.0004	**0.0025***	**2.58****	**4.91**** 0.636	**1420***** 614
313	1030	3.69	0.393	2.978	40.96	6732 (Tr)	27.96	4.46	**2.73**	**0.00527*** 0.00105	0.0051*	**2.46****	**6.72**** 1.338	**1749***** 725
323	1024	3.73	0.3925	2.301	38.76	8663 (Tr)	21.87	3.71	**3.47**	**0.0071*** 0.00165	**0.0072***	**2.31****	**8.02**** 1.864	**1972***** 796
333	1015	3.76	0.392	1.624	36.57	12166 (T)	15.58	2.75	**4.39**	0.0107	0.0111	2.32	10.19	2365
343	1007	3.74	0.3915	1.362	34.36	14393 (T)	13.01	2.45	**5.32**	0.0107	0.0137	2.08	11.06	2503
353	999	3.82	0.391	1.10	32.14	17679 (T)	10.75	2.14	**6.50**	0.0128	0.0172	1.97	12.80	2812

* Variants of calculation of coefficient a, in which the results of calculations according to formulas (1) and (3) are the closest.

** Values of turbulent thermal conductivity that satisfy the condition $k_{turb} \geq k_{trans}$, *i. e.*, calculated according to formulas (12) and (13), and in formula (13) the coefficient *a* is calculated according to classical relations, respectively (3) and (4).

*** Values of the heat transfer coefficient, calculated according to Equation (3).

We have achieved that the cross-section of the flow is conventionally divided into 2 LBL zones and a turbulized part. Obviously, the average turbulent thermal conductivity of the transient part should exceed the average thermal conductivity of the LBL, since flow turbulence always intensifies heat transfer. However, the result obtained from Equation (4) shows the opposite (Table) and (Fig. 2) – the turbulized part has a lower thermal conductivity than the average thermal conductivity of the LBL. Therefore, the classical Eq. (4), which is based on the fact that the flow is not divided into parts, is incorrect. In addition to the above, Eq. (14) becomes sensitive to changes in the various thermophysical properties of coolants, in particular to the use of glycol solutions (with or without nanoparticles).

The coefficient *a* (Table) was calculated according to classical equations (3 and 4) for the specified model coolant in the temperature range of 243–353 K. Turbulent thermal conductivities were also calculated according to equation (13), and heat transfer coefficients according to equation (10). These values for the indicated temperature range were calculated using previously obtained ratios (5), (7), (9)[14] (Table). In addition, the specified values were calculated for different values of the coefficient *a*, using the Gnielinsky equality (1).

The results of the calculations indicate that equality (4) gives incorrect results for determining the turbulent thermal conductivity of the transient part of the coolant flow, as well as the heat transfer coefficient (Table). Table shows that the heat transfer coefficients calculated according to equations (3) and (4) differ by more than 2 times.

Based on the results shown in Table and Fig. 2, it is concluded that equality (3) is a priority for calculations of the coefficient *a* and the heat transfer coefficient during the movement of coolants with a transient mode. The same calculation was performed using Gnielinsky's equality (1) (Table) and (Fig. 2).

Fig. 2 shows that all points of the graphs that lie below the transient thermal conductivity line are incorrectly calculated by the corresponding equations. This means that equation (4) cannot be used at all during the *Tr* mode of the coolant movement (Fig. 2). Without dividing the coolant flow into the LBL zone and the turbulized part, the mentioned shortcomings of equalities (1), (3) and (4) are not visible. At the beginning of the emergence of the *Tr* mode, i. e., in the temperature range of 283–293 K, the use of equalities (1) and (2) is also incorrect. This fact is discussed in detail in works[16–19] and is associated with a significant change in the hydraulic friction coefficient (Darcy) during the transition from the *L* to the *Tr* mode. In this small range, it is advisable to use equation (3) to determine the coefficient *a* according to formula (13). The heat transfer coefficients calculated by the Gnielinsky equation (1) and equation (4) are significantly different (Table).

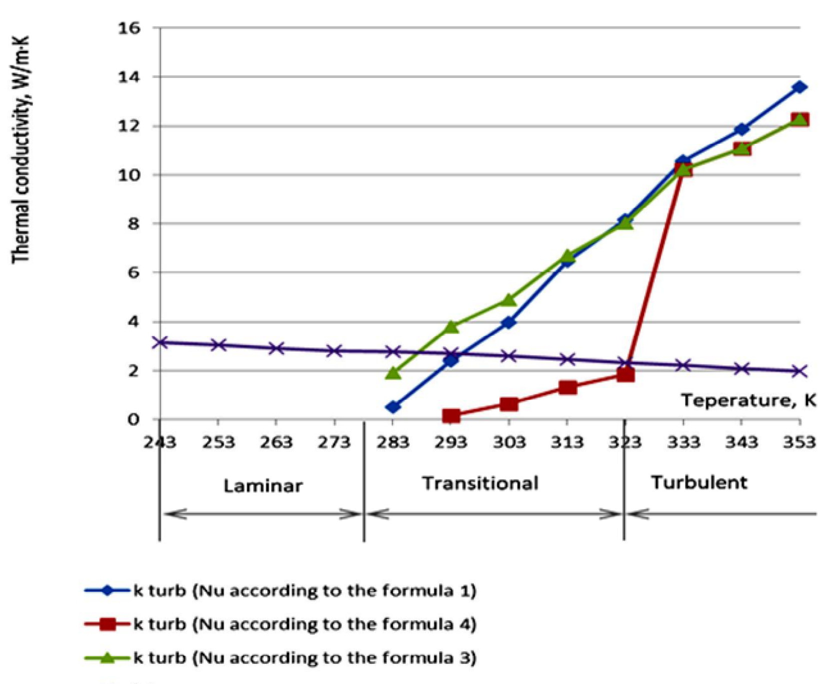

Fig. 2. Changes in the transient (according to the formula (12)) and turbulent (according to the formula (13)) thermal conductivities of different flow zones of the model coolant under the influence of temperature in different modes (temperature, K)

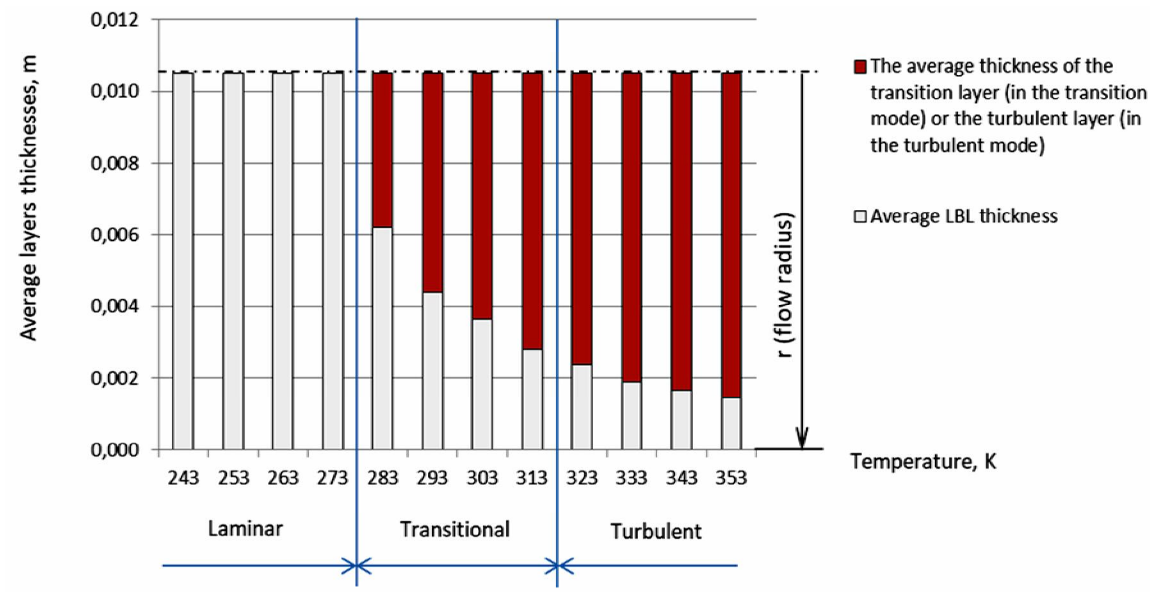

Fig. 3. Changes in the average thicknesses of the LBL and the turbulized part of the flow, calculated according to formula (9) depending on the model coolant temperature (45 % aqueous solution of propylene glycol, crystallization temperature 273 K, V = 0.93 m/s, D = 0.021 m)

The variation of the average thickness of the LBL and, accordingly, the turbulent part of the coolant flow of 45 % aqueous propylene glycol solution in the temperature range of 243–353 K, when it moves with the corresponding parameters (V = 0.93 m/s, D = 0.021 m) is shown in Fig. 3.

In the temperature range of 243–273 K, the flow is laminar and the L mode extends over the entire “live radius” of flow. In the temperature range of 283–313 K, the flow moves in a *Tr* mode, namely, at a temperature of 283 K, approximately 2/3 of the “living radius” of the flow is occupied by the laminar part, and 1/3 – by the transient part (Fig. 3). At a temperature of 293 K, a little less than half of the “living radius” is occupied by the laminar part. Further, as the coolant temperature increases from 293 K to 353 K, the average thickness of the LBL gradually decreases (Fig. 3).

Based on equality (10), for the *L* mode of movement of the heat carrier, the coefficient a is zero, that is, the right-hand side of the equation becomes zero and the heat transfer coefficient is calculated only from the left part of equation (10).

This calculation according to the new method, using the formulas for transient viscosity and thermal conductivity (11) and (12). Is numerically very close to the classical one.

The proposed method of calculating heat transfer coefficients can be used for such conditions as filter drying, where powerful surface forces act in the volume of bulk material[23], as well as in modern solar collectors with low speeds of movement of heat carriers, in heat exchangers in food, chemical, and other technologies the use of various surfactants[24], *etc*.

4. Conclusions

1. The approach proposed by us allows not only to calculate the heat transfer coefficients according to the new method but also establishes the limits of the correct use of classical formulas.

2. After dividing the “live cross-section” of the coolant flow conditionally into LBL and turbulent (transient) it was found that the calculation of the heat transfer coefficient for the *Tr* mode according to Eq. (4) is incorrect, since under such conditions the turbulent thermal conductivity of the transient zone is less than the thermal conductivity of the LBL.

3. An equation for calculating the turbulence coefficient a is proposed, which contains the heat transfer coefficient calculated according to classical equations, for example (1), (3) or others.

4. It was found that the classic Gnielinsky equation (1), which is used to calculate heat transfer coefficients, including those for the *Tr* mode, gives an erroneous result at the initial stage of the *Tr* mode, i.e. in the range of Reynolds numbers $4000 \geq Re \geq 2320$. In this range, it is recommended to apply equality (3).

5. Heat transfer coefficients calculated from equation (10) using classical equations (3) and (4) differ by more than 2 times (Table). This also confirms the correctness of the application of relation (3).

6. The theoretical calculation of the turbulence coefficient a for liquid-phase coolants from equation (14) makes it possible to make corrections for the use of glycol solutions with nanofillers since this equation contains the number *Bl*, which is sensitive to changes in the surface characteristics of liquids.

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Received: September 28, 2023 / Revised: April 11, 2024 / Accepted: May 17, 2024

МЕТОДИКА РОЗРАХУНКУ КОЕФІЦІЄНТА ТЕПЛОПЕРЕДАЧІ В ГЕЛІОСИСТЕМАХ З ЛАМІНАРНИМ І ПЕРЕХІДНИМ РЕЖИМАМИ РУХУ ПОТОКУ ТЕПЛОНОСІЯ, СТРУКТУРОВАНОГО НА ЧАСТИНИ

Анотація. *У цій роботі запропоновано новий метод вибору класичних емпіричних рівнянь для розрахунку коефіцієнтів тепловіддачі в трубах кожухотрубного теплообмінника в перехідному режимі. Цей метод ґрунтується на тому, що потік структурований на зону ламінарного примежового шару (ЛПШ) і турбулізовану частину, а коефіцієнт тепловіддачі розраховано через перехідну і турбулентну теплопровідність, а також середню товщину ЛПШ і, відповідно, середню товщину решти потоку теплоносія. Ключовим моментом цього методу є умова, що перехідна теплопровідність ЛПШ повинна бути нижчою за теплопровідність турбулізованої частини. Якщо ця умова не виконується, є підстави для висновку, що відповідне класичне емпіричне рівняння є некоректним для розрахунку коефіцієнта тепловіддачі. Як модельну рідину взяли 45 % водний розчин пропіленгліколю, який можна широко використовувати у сонячних колекторах, зокрема із нанонаповнювачами. Цей теплоносій цікавий тим, що за постійної швидкості V = 0,93 м/с і лінійного розміру (діаметра) "живого перерізу" потоку D = 0,021 м в інтервалі температур 243–273 К він рухається у ламінарному режимі, в інтервалі температур 283–323 К – у перехідному і 333–353 К – у турбулентному. Запропоновано нову формулу для розрахунку коефіцієнта турбулізації потоку теплоносія a, експериментальні числові значення якого знайдено в літературних джерелах лише для повітряного теплоносія.*

Ключові слова: *перехідна та турбулентна в'язкість і теплопровідність, коефіцієнт тепловіддачі, середня товщина ЛПШ, коефіцієнт поверхневого натягу теплоносія, перехідний режим.*