UDC 621.56

A. A. Andrizhievski, D. Sc. (Engineering), professor (BSTU);
A. P. Voronitskaya, PhD student (BSTU);
A. G. Sinkevich, PhD (Engineering),
chief researcher (SSO "JIPNR – Sosny", Belarus NAS)

COMPARATIVE ANALYSIS OF TEST RESULTS FOR HYDRODYNAMIC CHARACTERISTICS OF A POWER RANGE OF PLATE HEAT EXCHANGERS FOR HEAT SUPPLY STATIONS

The method of determination of industrial plate-type heat-exchangers collector system hydrodynamic parameters is presented. Calculation dependences on the basis of comparative analysis results of tests of three groups of plate-type heat-exchangers is received. Every group included heat-exchanger module and industrial heat-exchanger. The use of calculation model is reasonable for determination of hydraulic resistance coefficient to determination of pressure of the industrial heat-exchanger collector system.

Introduction. Plate heat exchangers are widely used in industry, as well as in heating and hot water supply of the housing sector and industrial buildings and structures [1, 2].

As a rule, the plate heat exchanger is constructed from corrugated plates that form channels through which heat and warming carrier move in counter-flow. There are numerous experimental facilities and stands in the Joint Institute of Energy and Nuclear Research under the Academy of Sciences of the Republic of Belarus intended for research of heat and mass exchange processes in the elements of power units, as well as for thermal-hydraulic testing of heat exchangers.

Data on pressure difference depending on the water consumption was obtained during tests of ductile heat exchangers which are the basis for establishment of the estimate dependences, allowing to define parameters of hydraulic resistance of tested heats-exchangers.

Experimentally it has been revealed, that at the same consumption in one channel, the general resistance of the industrial heat exchanger is higher, than for the modular one. It is possible to explain the given fact the collector array of distribution of the heat-carrier on channels upon the general resistance is influenced. The collector array-impact evaluation of the industrial heat exchanger on hydrodynamic resistance was carried out on the basis of the analysis of the experimental data received in JIPNR – Sosny National Academy of Sciences of Belarus for three groups of heat exchangers. There were two heat exchangers with single channel form but different number plates in each group. Further the heat exchanger with a small number of channels will call as "modular", and with a large number of channels – "industrial".

The aim of this work is to develop methods for calculating hydrodynamic resistance of collectors of industrial ductile heat exchangers on the basis of the analysis of results of tests both modular and industrial devices. At tests of ductile heat exchangers obtained data on pressure difference depending on the water consumption is a basis for an establishment of the estimate dependences, allowing defining parameters of hydraulic resistance of tested heats-exchangers.

The test object. Three groups of heat exchangers with a surface of plates 0.04; 0.15 and 0.4 m^2 were subject to test.

Small, modular, and industrial heat exchangers with heat transfer surface were tested in each group. The main technical characteristics of the tested heat exchangers are shown in the table.

The table shows that all three heat exchangers are double-pass exchangers.

Modular T1, T3, T5 and industrial T2, T4, T6 heat exchangers have relatively identical flow sections in one channel, equivalent diameters, lengths of the channel, diameters of collectors, these heat exchangers differ in number of plates, and consequently, for surface of heat exchange and flow sections.

Pressure differential calculation of the ductile heat exchanger. Technique of definition of pressure difference of the ductile heat exchanger. Estimations of hydrodynamic parameters for small heat exchangers T1, T3 and T5 have shown that the basic losses of pressure occur in flat channels of the heat exchanger, and the contribution to hydraulic resistance of distributing and collecting collectors and chambers is slight. So for these heat exchangers was made an assumption about determining the effective hydraulic resistance coefficient, which includes components from both the pressure drop due to friction resistance in flat channels and pressure drop at the expense of local resistances [3].

Thus, the effective hydraulic resistance coefficient can be found on the basis of known dependence

$$\Delta p = \xi_{\text{ef}} \frac{Zl}{d_{\text{e}}} \frac{\rho W^2}{2}, \qquad (1)$$

where Zl / d_{e} – this parameter, defined by the design; ρ – density of the fluid, kg/m^3; W – speed of the coolant, m/s. Having expressed the value of

rate of flow through discharge and geometrical parameters, we will receive the equation for calculation of coefficient of hydraulic resistance in a kind

$$\xi_{ef} = \frac{\Delta p}{G_m^2} \frac{2\rho S_p^2}{\left(\frac{Zl}{d_e}\right)}, \quad (2)$$

where G_m – coolant consumption, kg/s.

Having defined from experimental data of value of pressure difference at corresponding values of the consumption on dependence (3), it is possible to find values of effective coefficient of hydraulic resistance.

In the papers [1, 3] there are some of the methods of determining the dependencies of resistance coefficient of the Reynolds number, which can be used to determine this parameter modular heat exchangers based on test conditions.

For heat exchanger T1 is obtained:

$$\xi_p = \begin{cases} 16.17 Re^{-0.2}, & Re \le 1{,}800, \\ 7.61 Re^{-0.1}, & 1{,}800 < Re < 4{,}500. \end{cases} \quad (3)$$

For T3-heat exchanger –

$$\xi_p = 7.15 Re^{-0.135}. \quad (4)$$

For T5-heat exchanger –

$$\xi_p = 4.44 Re^{-0.165}. \quad (5)$$

The hydrodynamic characteristics of industrial heat exchangers (T2, T4, and T6) testing have showed that the use of computational dependencies (3)–(5) for this type of exchangers has not led to positive results, although the respective pairs of heat exchangers T1 and T2, T3 and T4, T5 and T6 channel shapes and plates are stay the same.

For scale industrial heat exchangers is it reasonable to consider the full pressure difference as two components:

$$\Delta P = \Delta P_p + \Delta P_c, \quad (6)$$

where ΔP_p – friction head loss in flat channels between parallel plates; ΔP_c – local losses, including losses in seeds, building reservoirs and transitional cells of heat exchangers.

From experimental data it is possible to receive values of coefficient of hydrodynamic resistance of collector array:

$$\xi_c = \frac{2\Delta P_c}{\rho W^2}. \quad (7)$$

On the other hand, the flow along collectors is accompanied by constant change of the consumption on length, connected with outflow of heat carrier in parallel channels for a distributing collector and inflow from channels for a collecting reservoir.

Processes of hydrodynamic flow in reservoirs and transitional cells of plate heat exchangers in some approximation can be put in line for channels with the exhaustion through porous wall, where the cross-flow exhaustion of weight has a significant influence on the coefficient of friction.

Technical characteristics of the tested heat exchangers

Name of parameter	Parameter value					
Identification of heat exchanger	T1	T2	T3	T4	T5	T6
Plate size, m^2	0.04	0.04	0.15	0.15	0.4	0.4
Heat exchanger surface F, m^2	1.64	5	6.75	36.15	8.4	130
Number of plates, PCs.	41	125	45	241	21	325
The number of moves, PCs.	2	2	2	2	2	2
Equivalent diameter d_e, mm	4.31	4.31	4.94	4.94	4.915	4.915
The channel length, m	0.267	0.267	0.564	0.564	1.055	1.055
The number of channels in one move	10	31	11	60	5	81
The cross section of one channel, mm^2	375	375	744.8	744.8	1,044	1,044
Orifice $S_p \cdot 10^3$, m^2	3.75	11.625	8.1928	44.688	5.22	84.564
Diameter of the collector, m	0.032	0.032	0.08	0.08	0.1	0.1
Manifold orifice $S_c \cdot 10^4$, m^2	8.0425	8.0425	50.265	50.265	78.54	78.54
The ratio of diameters of d_c / d_e	7.425	7.425	16.194	16.194	20.346	20.346
Do a walk-through sections S_p / S_c	4.66	14.455	1.63	8.89	0.665	10.767
The number of processing modes received at tests	50	31	33	32	34	23

The characteristic of the influence of the exhaustion weight on the coefficient of friction is the following:

$$\xi = \xi_n \left(1 + 17.5 Re^{0.25} \frac{W_r}{W_a} \right), \quad (8)$$

where ξ_n – coefficient of friction if there is no exhaustion; W_r, W_a – radial and axial components of velocity, respectively.

In the formula (8) influence of suction on coefficient of hydraulic resistance is shown through the parameter characterizing a cross flow stream of weight.

This is applied to the collector effect

$$\frac{W_r}{W_a} = \frac{W_p}{W_c} = \frac{S_c}{S_p},$$

where S_c, S_p – internal cross section and flat channels respectively.

Then we obtain

$$\xi = \xi_n \left(1 + A \cdot Re^{0.25} \frac{S_c}{S_p} \right). \quad (9)$$

The results of the tests. Use of expression (9) for definition of coefficient of hydraulic resistance of collectors at processing of experimental data is preferable as in this case the coefficient of resistance of collectors depends also on Reynolds's number, and from constructive elements of the heat exchanger (the relation of flow areas). Characteristic (9) will be used in the analysis of the results of hydrodynamic testing of plate heat exchangers.

In Fig. 1 experimental data on pressure difference depending on the consumption in one channel for two ductile heat exchangers T1 and T2 are given, that having the same form of channels, formed by identical plates by a surface 0.04 m^2, but differing in number of channels for passing of heat-carrier. In the first heat exchanger the number of channels in one pass equals to 10, in the second to 31.

The data obtained is well approximated by the parabola of the 2-th order.

For modular heat exchanger T1 curve *1* has the form

$$\Delta P_{T1} = -0.42857 + 49.05G1 + 1339.865G1^2. \quad (10)$$

For the industrial heat exchanger T2 curve *2* is described by the equation

$$\Delta P_{T2} = -1.32523 + 35.58G1 + 4227.545G1^2. \quad (11)$$

Similar approximations of experimental data are executed for the second and third groups of heat exchangers and received corresponding multinomial dependences.

For heat exchangers T3 and T4:

1) modular version

$$\Delta P_{T3} = -3.629 + 52.1495G1 + 414.1822G1^2; \quad (12)$$

2) industrial version

$$\Delta P_{T4} = 1.7198 - 6.8794G1 + 702.7809G1^2. \quad (13)$$

For T5 and T6 heat exchangers:

1) modular version

$$\Delta P_{T5} = -1.4764 + 31.483G1 + 191.932G1^2; \quad (14)$$

2) industrial version

$$\Delta P_{T6} = -0.1502 + 14.5395G1 + 311.9953G1^2. \quad (15)$$

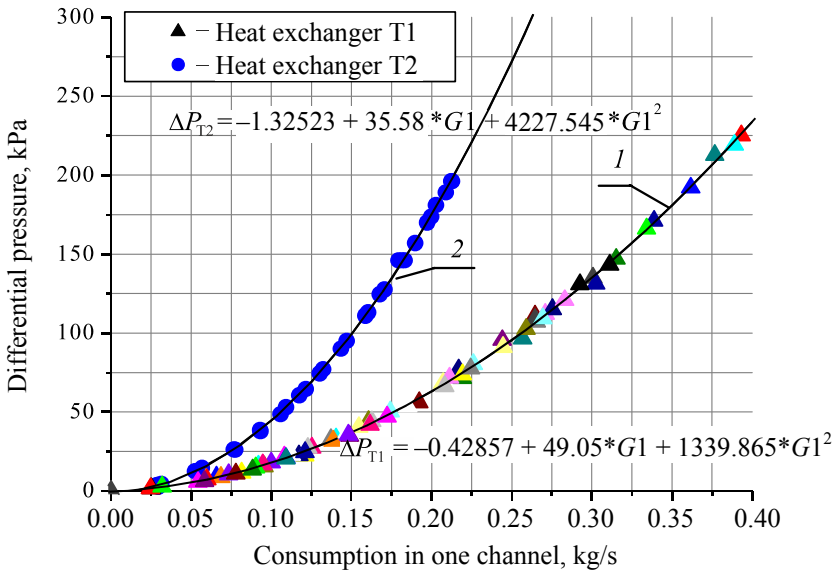

Fig. 1. Dependences on the consumption in one channel for two ductile heat exchangers T1 (a curve *1*) and T2 (a curve *2*)

Comparison of pressure changes in industrial and modular options allows you to select component of the differential pressure caused by the collector.

The approximation process data as generic dependencies (9) has been applied to the synthesis of data on hydraulic resistance coefficient reservoir system with different values of Reynolds number of the collector.

Thus, for the calculation of coefficient of hydraulic resistance of the collector system of heat exchanger T2 a correlation is offered

$$\xi_{K2} = 1.539\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right), \qquad (16)$$

hydraulic resistance coefficient for collectors of heat exchanger T4 –

$$\xi_{K4} = 0.733\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right). \qquad (17)$$

A similar dependence for determination of hydraulic resistance coefficient in flat channels for heat exchanger T6 has the form

$$\xi_{K6} = 0.517\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right). \qquad (18)$$

Introduction of the parameter considering the correlation of diameters (d_c / d_e), has allowed to reduce difference in coefficients of formulas (16)–(18). As a result the following expressions are received:

– for heat exchanger T2

$$\xi_{K2} = 11.43\frac{d_e}{d_c}\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right);$$

– for heat exchanger T4

$$\xi_{K4} = 11.87\frac{d_e}{d_c}\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right);$$

– for heat exchanger T6

$$\xi_{K6} = 10.52\frac{d_e}{d_c}\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right). \qquad (21)$$

Averaging the numeric coefficient in expressions (19) to (21) the number of experienced regimes we will get a single generalized dependence:

$$\xi_c = 11.28\frac{d_e}{d_c}\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right) \qquad (22)$$

to determine the coefficient of hydraulic resistance reservoir of plate heat exchangers.

The above mentioned dependence is received for two-pass heat exchangers. When entering parameter Z which characterizes the number of passes of the heat exchanger the given dependence will

$$\xi_c = 5.64Z\frac{d_e}{d_c}\left(1 + \frac{S_c}{S_p} Re_c^{0.25}\right). \qquad (23)$$

After getting the dependencies to determine the differential pressure in the reservoir system, heat exchanger resistance calculation method of industrial heat exchangers should contain the following stages:

1) according to the original data coefficient of resistance ξ_p and pressure drop in plate pack ΔP_p is set;

2) according to the formula (23) the coefficient of resistance ξ_c and (7) hydraulic resistance reservoir system ΔP_c is calculated

3) total resistance of industrial heat exchanger is set according to the dependcy (6).

This algorithm was used for comparative analysis of calculated and experimental data for industrial heat exchangers, T2, T4, and T6.

To illustrate the adequacy of the proposed calculation method in Fig. 2, the results of the comparison of the calculated values of the differential pressure with experienced data for heat exchanger T4 are presented

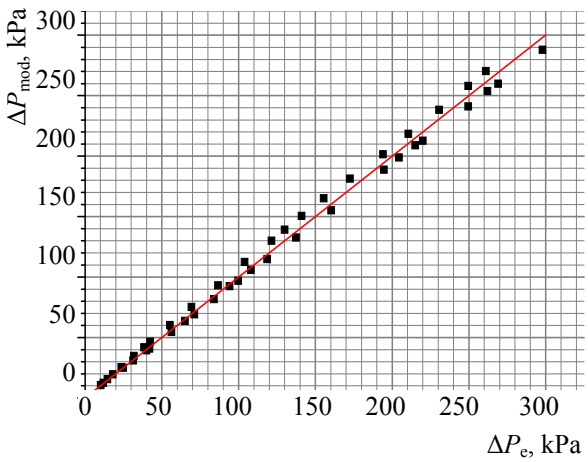

Fig. 2. Comparison of the estimated results with experimental values of the hydraulic resistance of heat exchanger T4

The analysis shows that for the heat exchangers T2, T4, T6 average deviation between the calculated and experimental data is respectively 3.2; 4.4; 3.6%.

Taking into consideration that tested ductile heat exchangers cover a wide range of changes of geometrical parameters, the obtained data show quite satisfactory conformity between the calcu-

lated and experimental data, and therefore, the dependence (23) can be recommended at definition of resistance of collectors in industrial ductile heats-exchangers.

Conclusion. According to the results of tests of three groups of plate heat exchangers which are significantly differ by its geometrical parameters, the suggested method of calculating of total resistance of industrial heat exchangers is regarded as pressure in the plates, and the resistance of the reservoir system.

The given technique is built on the basis of a generalizing correlation for definition of hydraulic resistance coefficient of collector array of industrial heat exchangers in a kind

$$\xi_c = 5.64Z\frac{d_e}{d_c}\left(1+\frac{S_c}{S_p}Re_c^{0.25}\right).$$

The offered is tested on a series of experimental data and can be recommended for use in the following range of geometric and hydrodynamic parameters:

– diameter of the collector d_c – 32–100 mm;
– parameter d_c / d_e – 7.425–20.346;
– paramenter S_p / S_c – 10.5–15.0;
– Reynolds number – $1.3 \cdot 10^5 < Re < 10^6$.

References

1. Отчет о НИР «Комплексные экспериментальные исследования на теплофизических стендах элементов систем локального теплоснабжения». Инв. № 212 / Ю. В. Клименков [и др.]. – Минск: ОИЭЯИ – Сосны НАН Беларуси, 2004. – 187 с.

2. Клименков, Ю. В. Теплогидравлические испытания теплообменных модулей для систем тепло- и водоснабжения / Ю. В. Клименков, В. А. Некрашевич, А. Е. Синкевич // Энергоэффективность. – 2003. – № 12. – С. 15–17.

3. Клименков, Ю. В. Теплогидравлические испытания теплообменных модулей для систем тепло- и водоснабжения / Ю. В. Клименков, В. А. Некрашевич, А. Е. Синкевич // Энергоэффективность. – 2004. – № 1. – С. 12–15.

4. Кинни, Д. Турбулентное течение, тепло- и массообмен в трубе с поверхностным отсосом / Д. Кинни, С. Спэрроу // Теплопередача. – 1970. – № 2. – С. 121.

Received 02.03.2012