Modeling of Variable Operating Modes of Water-to-Water Heat Exchangers

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Abstract. Heat exchangers used in heat supply systems operate in conditions of variable flow rates and temperatures of heat carriers. Existing theories for calculating variable modes of heat exchangers are based on the use of dimensionless constant parameters, regardless of their mode of operation. The use of such theories with a large change in temperature or flow rates of heat carriers in the heat exchanger leads to a significant error. Modeling of various variable operating modes of a counter-flow water-to-water heat exchanger with a change in its heat capacity allowed us to establish patterns of change in the parameters of heat exchangers and to construct a technique available for engineering calculations. This task becomes especially relevant when upon transition to new methods of central and local regulation of heat supply systems. Verification of the developed method was carried out on a heat exchanger model in the ANSYS Fluent package. This method is quite accurate; however, it requires the construction of a calculation model and takes a very long time, which does not allow it to be used for operational control of the operation modes of heat supply systems. Comparison of the calculation results showed a rather high accuracy of the developed methodology for different operating modes of heat exchangers.

1. Introduction

Actual heat engineering systems should provide equally comfortable conditions regardless of external temperature differences associated with seasonal and daily weather changes and other circumstances, including technological ones. As a result, such systems must function stably under substantially variable operating conditions. This, in turn, requires the ability to predict, control and manage the operating modes of heat engineering systems. It is clear that to solve this problem experimentally is almost impossible due to the variety of operating modes that are implemented in practice. For this reason, modeling of thermal processes occurring in these systems is actually the only constructive method to solve this problem. This method requires the ability to systematically simulate the operation of heat exchangers under the uncertainty of a number of parameters.

In Russian practice, when calculating the variable modes of water-to-water heat exchangers, the theory, proposed by professor E. Ya. Sokolov [1] is usually used. This theory based on the use of a constant parameter of the heat exchanger \( \Phi_0 = \text{const} \) [2-5]

\[
\Phi_0 = \left( kF \right)_h / \left( W_p W_{hc} \right)^{0.5},
\]  

1
where \((kF)\) – product of heat transfer coefficient under design (installation) conditions by the area of the heat exchanger; \(W_{pc}, W_{hc}\) – estimated (installation) equivalents of primary and secondary heat carrier flow rates \((W = cG, c – \text{heat capacity}, G – \text{flow rate of heat carrier})\).

As well as parameter (1), and also in foreign practice, to evaluate the efficiency of heat transfer processes in variable operating modes the NTU transfer unit number method [6, 7] widely used, which is in good agreement with theory [1].

In particular, for heat carriers with the same mass flow rate \(G\), the number of transport units NTU is found by the formula [8, 9]

\[
NTU = \frac{(kF)}{W_{\text{min}}}. \tag{2}
\]

Using relations (1) and (2), it is implicitly assumed that the heat transfer coefficient and the efficiency of the heat exchanger are practically independent of the temperatures of heat-exchanging media [1, 9, 10]. This assumption is true, however, in a rather narrow range of changes in the temperatures of the heat carriers, therefore, the use of constant parameters was often justly criticized [9, 11, 15].

It was shown in [12] that all quantities determining the change in the heat transfer coefficient can affect the heat exchanger parameter \(\Phi\). This parameter slightly deviates from the setting value, only if both the temperature difference and the flow of heat carriers increase with increasing heat capacity of the heat exchanger. At the same time, the flow rates of primary and secondary heat carriers can vary almost unlimitedly. In heat supply systems a different operating mode is often observed, accompanied by an increase in the temperature head in the heat exchanger with a decrease in its heat power (for the heat exchanger of the second stage of hot water supply). This makes it impossible to use ratios with constant parameters even for calculating a separate heat exchanger.

In addition, in modern heat supply systems, groups of interconnected heat exchangers are often used, with redistribution of heat power under variable operating conditions, which saves heat energy. In a system consisting of at least two heat exchangers, the calculation error associated with the use of the constant parameter \(\Phi_0\) may turn out to be even more significant.

The difficulty in calculating the variable modes of such devices is that the flow rates or temperatures of the heat carriers at the inlet to the heat exchanger or at the outlet from the heat exchanger may not be known. Previously, in our works [13, 14], the main variable operating modes of heat exchangers were determined and a general formula for changing the heat exchanger parameter with varying flow rates and temperatures of the primary and secondary heat carriers was obtained.

In this paper, we present a method for calculating the variable operating modes of heat exchangers based on the use of variable parameters of heat exchangers.

2. Methods for calculating variable operating modes of heat exchangers
The main difficulty in calculating the operating modes of heat exchangers is the failure of input data. The design scheme of the heat exchanger is shown in Fig. 1.

![Figure 1. The scheme of the heat exchanger.](image)

At the beginning of the calculation, the following variables are usually known: the laws of changes in the temperatures of the primary \(t_{p1}\) and secondary \(t_{h2}\) heat carriers at the inlet of the heat exchanger and the equivalent of the flow rate of the secondary medium \(W_h\).
It is necessary to find: the temperatures of the primary \( t_{p2} \) and secondary \( t_{h2} \) heat carriers at the outlet of the heat exchanger and the equivalent of the flow rate of the primary medium \( W_p \).

The operation modes of the heat exchanger can be described by a system of equations:

\[
\begin{align*}
Q &= \varepsilon W_{\text{min}} D_t \\
\varepsilon &= \frac{1}{1 - \exp \left[ -\frac{kF}{W_{\text{min}}} \right]} \left( 1 - W_{\text{min}} \frac{W_{\text{max}}}{W_{\text{min}}} \right) \\
W_p &= \frac{kF^{\frac{1}{2}}}{\Phi} \left( 1 - \frac{kF}{W_{\text{max}}} \right) \varepsilon D_t \quad \text{if} \quad W_h < W_p \\
W_h &= \frac{kF^{\frac{1}{2}}}{\Phi} \left( 1 - \frac{kF}{W_{\text{min}}} \right) \varepsilon D_t \quad \text{if} \quad W_h > W_p \\
t_{h2} &= t_{h1} + \frac{Q}{W_h} \\
t_{p2} &= t_{p1} - \frac{Q}{W_p} \\
\Delta t &= (\delta t_{\text{max}} - \delta t_{\text{min}}) (\ln \delta t_{\text{max}} - \ln \delta t_{\text{min}})^{-1} \\
\delta t_{\text{max}} &= \begin{cases} 
\frac{t_{p2} - t_{h1}}{W_h} & \text{if} \quad W_h < W_p \\
\frac{t_{p1} - t_{h2}}{W_h} & \text{if} \quad W_h > W_p 
\end{cases} \\
\delta t_{\text{min}} &= \begin{cases} 
\frac{t_{p2} - t_{h1}}{W_h} & \text{if} \quad W_h < W_p \\
\frac{t_{p1} - t_{h2}}{W_h} & \text{if} \quad W_h > W_p 
\end{cases} \\
kF &= Q / \Delta t \\
\Phi &= kF \left( \frac{W_p W_h}{W_{\text{min}} \cdot W_{\text{max}}} \right)^{\frac{1}{2}} \\
D_t &= t_{p1} - t_{h1}
\end{align*}
\]

where \( Q \) – thermal power; \( \varepsilon \) – heat exchanger efficiency; \( D_t \) – temperature difference of heat carriers at the inlet to the heat exchanger; \( W_{\text{min}} \) и \( W_{\text{max}} \) – the smaller and the larger of the two equivalents of flow rates; \( \Delta t \) – the average temperature difference of the coolants is the smaller and the larger of the two equivalents of heat carrier flow.

At the same time, in system (3), the desired variables enter the solution in an implicit form. Therefore, the methodology is constructed by the method of successive approximations.

As a first approximation, some assumptions are made:
- \( \Phi_0 = \text{const} \), then according to the E.Ya. Sokolov’s formula [1] the efficiency of the heat exchanger \( \varepsilon \) does not depend on the temperatures of the coolants;
- \( W_{\text{min}} = W_p \) in order to solve the first equation of the system (3);
- The temperature of the primary fluid at the outlet of the heat exchanger \( t_{p2} \) is equal to the temperature of the secondary fluid \( t_{h1} \) at the inlet, as in a heat exchanger with an infinitely large heating surface;
- The equivalent flow rate of the primary fluid \( W_p \) is set equal to the installation value.
In this case, the system of equations takes the following form:

\[ Q = \varepsilon W_{\text{max}} D_t, \]

\[ \varepsilon = \left( a \frac{W_{\text{min}}}{W_{\text{max}}} + b + \frac{1}{\Phi_0} \sqrt{\frac{W_{\text{min}}}{W_{\text{max}}}} \right)^{-1} \leq 1 \]

\[ W_p = Q / \left( t_{p1} - t_{p2} \right) = Q / \left( t_{p1} - t_{h1} \right) = \frac{Q}{D_t}, \]

\[ t_{h2} = t_{h1} + \frac{Q}{W_h}, \]

\[ D_t = t_{p1} - t_{h1} \tag{4} \]

where \( a = 0.35; b = 0.65 \) – constant coefficients taken according to [1] for the countercurrent flow of heat carriers.

According to the results of calculating the first approximation, the variable equivalent of the primary coolant flow rate \( W_p \) is previously determined, which differs from that adopted at the beginning of the calculation, but the temperature of the primary medium at the outlet of the heat exchanger \( t_{p2} \) is not determined, therefore, in the second approximation, the calculation is also carried out according to the simplified formulas by E.Ya. Sokolov [1].

The system of equations of the second approximation will have the form:

\[ Q = \varepsilon W_{\text{max}} D_t, \]

\[ \varepsilon = \left( a \frac{W_{\text{min}}}{W_{\text{max}}} + b + \frac{1}{\Phi_0} \sqrt{\frac{W_{\text{min}}}{W_{\text{max}}}} \right)^{-1} \leq 1 \]

\[ W_p = y \Phi_0^2 W_h \left[ -1 + \sqrt{1 + i \Phi_0^2 \left( \frac{D W_h}{Q} - j \right)} \right] \geq \frac{Q}{D_t}, \]

\[ t_{h2} = t_{h1} + \frac{Q}{W_h}, \]

\[ t_{p2} = t_{p1} - \frac{Q}{W_p}, \]

\[ \Delta t = (\delta t_{\text{max}} - \delta t_{\text{min}}) \left( \ln \delta t_{\text{max}} - \ln \delta t_{\text{min}} \right)^{-1} \]

\[ \delta t_{\text{max}} = \begin{cases} t_{p2} - t_{h1} & \text{если } W_h < W_p \\ t_{p1} - t_{h2} & \text{если } W_h > W_p \end{cases} \]

\[ \delta t_{\text{min}} = \begin{cases} t_{p1} - t_{h1} & \text{если } W_h < W_p \\ t_{p2} - t_{h2} & \text{если } W_h > W_p \end{cases} \]

\[ D_t = t_{p1} - t_{h1} \tag{5} \]

where \( y = 4b^2, i = 4b, j = a =0.35, \) if \( W_h < W_p \) and \( y = 4a^2; i = 4a; j = b = 0.65, \) if \( W_h > W_p \).

In the third approximation, system (3) is solved, in which the values of temperatures and heat carrier flow rates from the second approximation are used, where they were determined through constant parameters \( \Phi_0 \). Therefore, it is necessary to clarify all the characteristics, for which the system of equations (3) is again sequentially solved. After the fifth approximation, the results diverge by less than 0.02\%, after the sixth approximation, the discrepancy is less than 10^{-6} \%.
3. Heat exchanger simulation in ASYS Fluent

To test the developed method, a heat exchanger model was created in ANSYS Fluent, see Fig. 2. The calculations were performed on a grid with the number of elements is 381968, Fig. 3. The grid was generated by the Sweep method.

![Figure 2. Heat exchanger plate geometry.](image1)

![Figure 3. Calculation grid.](image2)

According to the ANSYS Fluent help system, the calculation of heat transfer in heat exchangers, when there is not enough information to calculate the logarithmic temperature difference, uses the NTU-efficiency method. The main calculation formulas are as follows.

Equations of conservation of mass and momentum

\[
\nabla \vec{w} = 0 \ ;
\]

\[
\frac{\partial \vec{w}}{\partial t} + \frac{1}{2} \nabla (\vec{w} \cdot \vec{w}) + (\nabla \times \vec{w}) \times \vec{w} + 2 \vec{\Omega} \times \vec{w} + \vec{\Omega} \times (\vec{\Omega} \times \vec{r}) = \frac{1}{\rho} \nabla P + \vec{F} - \vec{\nu} \nabla \times (\nabla \times \vec{w}) .
\]

Nusselt number

\[
\text{Nu} = \alpha l / \lambda ,
\]

where \( \nu \) – kinematic viscosity, \( m^2/s \); \( \vec{\Omega} \) – angular velocity, rad./s; \( P \) – medium pressure, Pa; \( \vec{r} \) – radius vector; \( \vec{F} \) – force per unit mass; \( \alpha \) – heat transfer coefficient, \( W/(m^2-K) \); \( l \) – characteristic length, m; \( \lambda \) – thermal conductivity of the medium, \( W/(m-K) \).

Heat exchanger efficiency

\[
\varepsilon = \frac{1 - \exp\left[-\left(1 - \frac{W_{\text{min}}}{W_{\text{max}}} \right)NTU\right]}{1 - \frac{W_{\text{min}}}{W_{\text{max}}} \exp\left[-\left(1 - \frac{W_{\text{min}}}{W_{\text{max}}} \right)NTU\right]} .
\]
Fluid temperatures are determined from the expression

\[ e = \frac{W_p}{W_h} \frac{t_{h2} - t_{h1}}{t_{p1} - t_{p2}}, \]

(10)

4. Results and discussion

In order to test the developed method, various modes of operation of a countercurrent water-to-water heat exchanger were calculated.

The laws of changing the temperatures of the primary \( t_{p1} \) and secondary \( t_{h1} \) heat carriers at the inlet to the heat exchanger and the speeds of the coolants \( w_p \) and \( w_h \) were set. Calculated: the temperature at the outlet of the primary \( t_{p2} \) and secondary \( t_{h2} \) coolants.

Some results are shown in Fig. 4 and 5. In the figures, the “M” index indicates the results of the calculation according to the developed method, the index “A” indicates the results of the calculation in the ANSYS Fluent package. The calculation results are close to coincide, which shows a fairly high accuracy of the developed methodology.

![Figure 4](image1.png)

**Figure 4.** The temperature of the primary medium rises with increasing heat capacity of the heat exchanger.

![Figure 5](image2.png)

**Figure 5.** The temperature of the primary coolant decreases with increasing heat capacity of the heat exchanger.

5. Conclusion

The main difficulty in calculating the operating modes of heat exchangers is the lack of input data. Therefore, existing calculation methods, including those implemented in large software systems, for example, ANSYS, are based on the application of the NTU method. To perform calculations in ANSYS, a grid is built, within each cell of which NTU is considered constant. The calculation is performed by
the iteration method and is time consuming. When changing the dimensions or configuration of the heat exchanger, the calculation model must be re-created, which also requires a lot of time. In addition, the main disadvantages of this calculation are:

- the impossibility of setting arbitrary input data, for example, the temperature of the primary medium at the inlet to the heat exchanger, and the secondary at the outlet;
- the calculation is carried out only at specified temperatures and flow rates of the heat carrier at the inlet (in real conditions, one of the flow rates may not be known);
- there are restrictions on the flow of heat carriers for a given type of heat exchanger.

The developed method does not require the construction of a calculation model; the calculations are performed quickly enough, which allows you to use it to control the operating modes of heat supply systems in the on-line mode.

6. References

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