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MATHEMATICAL MODELING OF HEAT TRANSFER THROUGH SINGLE-LAYER AND MULTI-LAYER CYLINDRICAL WALLS IN BUILDINGS AND STRUCTURES

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Abstract: This article presents an analysis of mathematical modeling processes for heat transfer in the walls of buildings and structures under current Uzbekistani conditions. The article also outlines the types of energy sources and the differences in the use of heat transfer processes during the construction of buildings, highlighting their main advantages and disadvantages.

Keywords: Let's consider heat transfer through a homogeneous cylindrical wall (Figure 1). A hot heat transfer fluid with temperature T1 and heat transfer coefficient $\alpha 1$ transfers heat to a cold heat transfer fluid with temperature T2 and heat transfer coefficient $\alpha 2$.

Introduction. The system components include: energy, solar panels, reservoir, building, construction process, system, natural air, conductor, water, battery, supply, collector, and body.

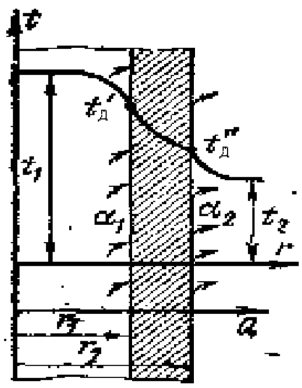

Figure 1

Then, the following three equations can be written for the heat flux:

$$Q = \alpha_1 \pi d_{ich} l (T_1 - T_\partial^1)$$

$$Q = \frac{\pi l}{\frac{1}{2\lambda} \ln \frac{d_{tash}}{d_{ich}}} (T_\partial^I - T_\partial^{II})$$

$$Q = \alpha_2 \pi d_{tash} l (T_\partial^{II} - T_2)$$

Solving these three equations for the temperature differences and then adding them sequentially, we obtain the following:

$$Q = \frac{\pi l (T_1 - T_2)}{\alpha_1 d_{ich} + \frac{1}{2\lambda} \ln \frac{d_{tash}}{d_{ich}} + \frac{1}{\alpha_2 d_{tash}}} \quad (1.37)$$

here

$$k_{ц} = \frac{1}{\frac{1}{\alpha_1 d_{ich}} + \frac{1}{2\lambda}\ln\frac{d_{tash}}{d_{ich}} + \frac{1}{\alpha_2 d_{tash}}} \qquad (1.38)$$

This is referred to as the linear heat transfer coefficient, with units of W/(m·K) [or W/(m·°C)]. The heat flux density through the cylindrical wall is equal to:

$$q_u = \frac{Q}{l} = k_{ц}\pi(T_1 - T_2)$$

The linear heat transfer coefficient is equal to the amount of heat transferred per unit time from the hot heat transfer fluid to the cold heat transfer fluid through a pipe of length 1 m when the temperature difference between them is 1°C. Therefore, equation (1.18) can be rewritten as::

$$Q = k_{ts}\pi l\,(T_1 - T_2) \qquad (1.39)$$

The heat flow through a multi-layered cylindrical wall is equal to:

$$Q = \frac{\pi l(T_1 - T_2)}{\frac{1}{d_1 d_{ich}} + \sum_{i=1}^{i=n}\frac{1}{2\lambda_i}\ln\frac{d_{i+1}}{d_i} + \frac{1}{\alpha_2 d_{tash}}} \qquad (1.40)$$

The heat flux density, taken relative to the inner or outer surfaces, is determined from the following equations:

$$q_{ц1} = \frac{Q}{\pi d_1 l} = \frac{k_{ц}}{d_1}(T_1 - T_2)$$

$$q_{ц2} = \frac{Q}{\pi d_2 l} = \frac{k_{ц}}{d_2}(T_1 - T_2)$$

The quantity that is the reciprocal of the linear heat transfer coefficient is called the linear thermal resistance of heat transfer:

$$R_{ц} = \frac{1}{Q_{ц}} = \frac{1}{\alpha_1 d_{ich}} + \sum_{i=1}^{i=n}\frac{1}{2\lambda_i}\ln\frac{d_{i+1}}{d_i} + \frac{1}{\alpha_2 d_{tash}} \qquad (1.41)$$

here $\frac{1}{\alpha_1 d_{ich}}$ ва $\frac{1}{\alpha_2 d_{tash}}$ - External thermal resistances; $\sum_{i=1}^{i=n}\frac{1}{2\lambda_i}\ln\frac{d_{i+1}}{d_i}$ - The thermal resistance of a multi-layered cylindrical wall; the unit of measurement for R_{ts} is m·K/W (or °C·m/W).

We determine the inner surface temperature from the following formula

:

$$T_{\partial}^{1} = T_1 - \frac{Q}{\alpha_1 d_{uч} \pi l} \tag{1.42}$$

And the outer surface temperature is determined by:

$$T_{\partial}^{II} = T_2 + \frac{Q}{\alpha_2 d_{tash} \pi l} \tag{1.43}$$

Intensification of Heat Transfer. When using heat exchange equipment, it is often necessary to increase the heat flow passing through surfaces of various shapes. As can be seen from the heat transfer equation Q = kSΔT, if the dimensions of the body's surface and the temperatures of the fluids are given, the heat flow depends on the heat transfer coefficient. However, knowing only the value of the heat transfer coefficient is not sufficient to study the heat transfer process. A correct conclusion can be drawn by analyzing the mutual ratio of all thermal resistances, which, as a result, creates the opportunity to significantly change the heat flow.

In heat transfer through a flat wall, increasing the heat transfer coefficient can be achieved by using a thinner wall, choosing a material with higher thermal conductivity, and increasing the heat transfer coefficient. If the thermal resistance of the wall is small ($\delta/\lambda \approx 0$), then the heat transfer coefficient depends on the heat transfer coefficients α_1 and α_2.

$$k = \frac{1}{\frac{1}{\alpha_1} + \frac{1}{\alpha_2}} = \frac{\alpha_1 \alpha_2}{\alpha_1 + \alpha_2} \tag{1.44}$$

From equation (1.44), it is evident that k is always smaller than the smallest α value. Therefore, to increase k, it is necessary to increase the smallest α. If $\alpha_1 \approx \alpha_2$, then to increase k, it is necessary to increase any α.

If the values of α are large, then k is mainly dependent on the thermal conductivity of the wall. In heat transfer through a cylindrical wall, the thermal resistances $1/(\alpha_1 d_1)$ and $1/(\alpha_2 d_2)$ depend on the heat transfer coefficients and surface dimensions. If α is small, the thermal resistances can be reduced by increasing the corresponding surface areas. The same result can be achieved for a flat wall by adding fins to one side. Let's illustrate these points with some examples.

In a steam boiler, the heat transfer coefficient from furnace gases to the wall is α_1 = 30 W/(m^2·K); from the wall to the boiling water, α_2 = 5000 W/(m^2·K); the thermal conductivity coefficient of the steel wall is λ = 50 W/(m·K); its thickness is δ = 0.02 m. Let's assume the wall is flat. Under these conditions, the heat transfer coefficient k = 29.5 W/(m^2·K), i.e., it is smaller than the smallest α. The heat transfer coefficient k cannot be increased by improving the heat transfer conditions from the wall to the water or by using a thinner wall. k can only be increased by improving the heat transfer from the furnace gases to the wall.

In apparatuses where the heat transfer coefficients α_1 and α_2 are large, the situation is different. For example, in a water condenser, let α_1 = 5000 W/(m^2·K) on the water side and α_2 = 10,000 W/(m^2·K) on the steam side.

If the wall of such a condenser is made of steel with a thickness of 20 mm, k = 1428 W/(m²·K). If a wall with a thickness of 3 mm is used, then k = 2770 W/(m²·K). If the steel is replaced with copper and a wall with a thickness of 1 mm is used, then k = 3400 W/(m²·K).

From the example above, it is clear that at high values of heat transfer coefficients, k mainly depends on the thermal conductivity of the wall. Thus, to intensify heat transfer in apparatuses, it is necessary to try to reduce the largest resistance.

To fully study the heat transfer process in a homogeneous wall, a test problem was considered. A homogeneous wall is given with a temperature of 10 °C on surface 1 and 100 °C on surface 2. It is necessary to consider the unsteady-state heat transfer process through this wall with a thickness of 0.25 m. In the considered process, the wall material should be considered as concrete, fired brick, and unfired brick, and the heat transfer in all three cases should be studied.

To construct the unsteady-state mathematical model of this process, we use the two-dimensional energy equation.

$$\frac{dT}{dt} = \alpha \frac{d^2T}{dx^2} \tag{1.45}$$

The analytical solution of the equation exists and is equal to:

$$T = T_s + (T_o - T_s) erf\left(\frac{x}{2\sqrt{\alpha t}}\right) \tag{1.46}$$

In real devices, due to the large number of boundary conditions, it is usually not possible to use the analytical method. Or, as a result of the imposed conditions, the resulting expression deviates from real, experimental results. Therefore, it is advisable to use numerical methods..

To solve the problem, we used the following difference schemes

1. Explicit difference scheme

$$\frac{T_j^{n+1} - T_j^n}{\Delta t} - \frac{\alpha(T_{j-1}^n - 2T_j^n + T_{j+1}^n)}{\Delta x^2} = 0 \tag{1.47}$$

1. Richardson extrapolation scheme

$$\frac{T_j^{n+1} - T_j^{n-1}}{2\Delta t} - \frac{\alpha(T_{j-1}^n - 2T_j^n + T_{j+1}^n)}{\Delta x^2} = 0 \tag{1.48}$$

3. DuFort-Frankel scheme

$$\frac{T_j^{n+1} - T_j^{n-1}}{2\Delta t} - \frac{\alpha[T_{j-1}^n - (T_j^{n-1} + T_j^{n+1}) + T_{j+1}^n]}{\Delta x^2} = 0 \tag{1.49}$$

4. Explicit finite difference scheme

$$\frac{T_j^{n+1} - T_j^n}{\Delta t} - \frac{\alpha(T_{j-1}^{n+1} - 2T_j^{n+1} + T_{j+1}^{n+1})}{\Delta x^2} = 0 \tag{1.50}$$

5. Crank-Nicolson scheme

$$\frac{T_j^{n+1} - T_j^n}{\Delta t} - \frac{\alpha}{2}\left(\frac{T_{j-1}^n - 2T_j^n + T_{j+1}^n}{\Delta x^2} + \frac{T_{j-1}^{n+1} - 2T_j^{n+1} + T_{j+1}^{n+1}}{\Delta x^2}\right) = 0 \tag{1.51}$$

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CONTENTS

PRIMARY PROCESSING OF COTTON, TEXTILE AND LIGHT INDUSTRY