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Methods of unstable heat transfer calculation and experimental verification

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Technical calculations of heat transfer are carried out on the basis of equations of the stable process. However, the lack of the more universal methods of calculation of unstable heat transfer regimes does not allow a deeper analysis of the influence of the body's (building's) massiveness on the development of the heat transfer process, the utilized amount of energy, and the possibilities to control the process. The analysis of thermal interaction of a system consisting of several bodies is based on F. M. Camias's impulse theory of conductivity and G. M. Kondratyev's theory of the regular thermal regime. Supposing that in a certain temperature range the physical properties of the body remain stable, in the case of the regular heat transfer regime the temperature in any point increases (decreases) at an equal rate, there exists the sum of points that at a given moment fit to the generalized-average temperature of the object. The temperature regime of this implied surface will actually describe the generalized temperature regime of the whole body. To an extremely thin material surface Newton's heat transmission law can be applied. The following presumptions are accepted for the formulation of the task: a) boundary conditions of the third degree are accepted for heat exchange between the bodies and their environment; b) the sum of heat transmission coefficients by radiation and convection is constant; c) heat flows among the bodies are proportional to the difference of temperatures, and heat exchange is regular; d) each body is treated as a material surface.

Experimental verification of the method has been carried out in a special stand with the use of the thermo-accumulative heating equipment (AEG). A comparison of theoretical and experimental results is also presented.

Key words: regular heat transfer regime, equivalent material isothermic surface, generalized temperature, reduced surface coefficient of heat transfer, heating-cooling rate, undimensional massiveness, Newton's heat transmission law

1. INTRODUCTION

In Lithuania, considerable attention is given to a moderate consumption of fuel energetic resources. Great amounts of heat and fuel are consumed for the heating of buildings of various destination (~40% of burned fuel) as well as for technological equipment in order to heat certain products up to the required temperature.

Most often, when solving the technical tasks of heat transfer, the investigation must be based on the examination of several bodies with different properties that simultaneously participate in the process of heat transfer. In fact, the heating (cooling) of an object inevitably involves the heating (cooling) of the bodies around this object.

Technical calculations of heat transfer are carried out on the basis of equations of a stable process. However, the majority of heat transfer processes that take place in heating equipment, technological processes, in buildings and generally in nature are unstable. The lack of the more universal methods of calculation of unstable heat transfer regimes does not allow a deeper analysis of the influence of the body's (building's) massiveness on the development of the transfer processes, the utilized amount of energy (heat), and the possibilities to control the process. Actually, it is often useful to check which materials are more effective, what are the possibilities of controlling the thermal regimes and heat expenditure, etc.

2. METHODS

Analysis of thermal interaction of a system consisting of several bodies is based on F. M. Camia's impulse theory of conductivity as well as on G. M. Kondratyev's theory of the regular thermal regime.

Supposing that in a certain temperature range the properties of the body remain stable (ρ = const,

c = const, λ = const), in the case of a regular heat transfer regime the temperature in any point increases (decreases) at an equal rate. In any body, including a heterogeneous one, there exists the sum of points that at a given moment fit to the generalized-average temperature of the object. Such sum of isothermic (at a given moment) points forms a closed surface (one or more) in the body, whose location in the body in the case of the given regular thermal regime is fixed. The temperature regime of this implied surface will actually describe the generalized temperature regime of the whole body. For the analysis of the thermal regime of a real body, it would be enough to analyze the generalized temperature θ_a of the body corresponding to that of the isothermic surface, which at any moment is

$$\theta_a = \frac{\sum_{1}^{i=n} V_i \rho_i c_i \theta_i}{\sum_{1}^{i=n} V_i \rho_i c_i}, \qquad (1)$$

where $V_i\rho_i$ is the mass of a separate (heterogeneous) body element, c_i is the heat capacity of the same element, and θ_i is the temperature of the same element at a given moment.

When distinguishing the case of the regular heat exchange among all possible variants of the heat transfer process, the conclusion may be made that the speed of temperature alteration in the case of an equivalent isothermic surface similar to that of each point of the test object is the same.

Each body (fluid) in which a regular unstable heat transfer process occurs might be regarded as a material surface where the whole mass of the body is uniformly concentrated throughout the entire surface. To such an extremely thin surface Newton's heat transmission law could be applied. The heat amount transferred to the equivalent material isothermic surface (EMIS) will remain the same if the methods of calculation would use the surface heat transmission coefficient value equivalent to that of a real object.

In the case of regular heat transmission when a stable heat transfer regime is achieved, the location of the EMIS, the temperature $\bar{\theta}$, and the interior $\bar{h}_i$ as well as the exterior $\bar{h}_e$ surface heat transmission coefficients (Fig. 1) are found with the aid of the following equations:

$$\bar{\theta} = \frac{\rho_1 c_1 \frac{\theta_1 + \theta_2}{2} d_1 + \rho_2 c_2 \frac{\theta_2 + \theta_3}{2} d_2}{\rho_1 c_1 d_1 + \rho_2 c_2 d_2},$$

$$d = \frac{\theta_1 - \bar{\theta}}{\theta_1 - \theta_2} d_1. \qquad (2)$$

The reduced surface heat transmission coefficients with respect to the EMIS wall are:

$$\bar{h}_i = \frac{1}{\frac{1}{h_i} + \frac{d}{\lambda_1}}; \quad \bar{h}_e = \frac{1}{\frac{d_1 - d}{\lambda_1} + \frac{d_2}{\lambda_2} + \frac{1}{h_e}}, \qquad (3)$$

where d_1, d_2 are the thickness of the layer, λ_1, λ_2 are the heat conductivity coefficients of the layer, h_i, h_e are the heat transmission coefficients of the enclosure's interior and exterior surfaces; θ_e is the temperature of surrounding air; θ_1, θ_2, θ_3 is the temperature of the enclosure; $\bar{h}_i$, $\bar{h}_e$ are the thermal resistance coefficients of the EMIS; $\bar{\theta}$ is the temperature of the EMIS; d is the thickness of the EMIS.

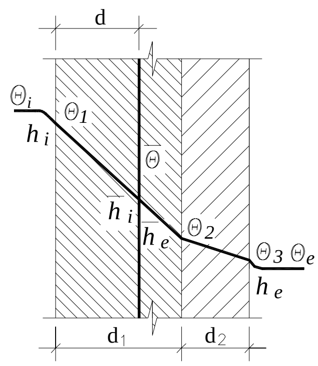


Fig. 1. Scheme of regular heat transfer through flat wall

Thus, in the case of regular heat transfer, Newton's heat transmission law may be applied to any body:

$$V\rho c \frac{d\theta}{d\tau} = \bar{h} A \left(\bar{\theta} - \theta_e\right). \qquad (4)$$

The following presumptions are accepted for the formulation of the task:

1. Boundary conditions of the third kind are accepted for heat exchange between bodies and their environments.

2. The sum of heat transmission (by radiation and convection) coefficients is constant.

3. Heat flows among the bodies are proportional to the difference of temperatures. Heat exchange is regular.

4. Each body is treated as a material surface.

The main object of the task is a generalized heated body (building). The aim of the solution is to determine the temperature regime of interior air (fluid), when the constructive solution of the body (building) as well as temperature fluctuations of the exterior air (fluid) are known and when the heat source of a certain capacity is presented (Fig. 2).

At the regular heat transfer regime, the following equation of heat balance could be used:

$$Nd\tau = V_a \rho_a c_a d\theta_a + dq_n, \tag{5}$$

where N is the power of the heat source, $V_a \rho_a c_a d\theta_a$ is heat accumulated by the body a; dq_n is heat delivered by the same body to other surrounding bodies (fluids), here: $dq_n = h_a A_a(\theta_a - \vartheta)d\tau$, ϑ is the operative temperature of the premise.

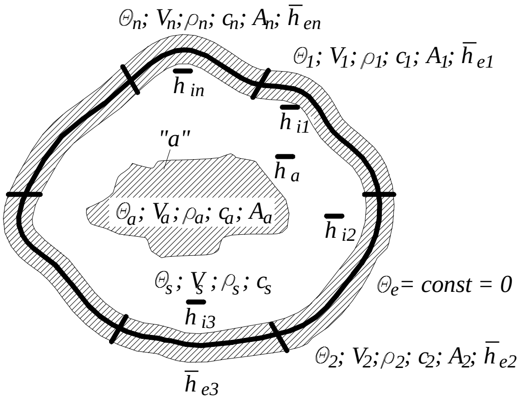


Fig. 2. Generalized model of heat exchange in heterogeneous body

The enclosing construction of the building (parts of the external body) will also accumulate part of the received heat, at the same time the rest part of heat is accumulated by the environment (fluid). Heat exchange may be defined by the following formula:

$$\frac{d\theta_a}{d\tau} + \mu_s \frac{d\vartheta}{d\tau} + \mu_1 \frac{d\theta_1}{d\tau} + \mu_2 \frac{d\theta_2}{d\tau} + \ldots + \mu_n \frac{d\theta_n}{d\tau} + \tag{6}$$
$$+ m_a\theta_a + m_1\theta_1 + m_2\theta_2 + \ldots + m_n\theta_n = k,$$

where μ is the value that expresses the part of the accumulated heat attributed to each body (the building element) in respect of the heat delivering body, when the difference between the temperatures is one degree. It is the thermal inertia index of the body (undimensional "massiveness"),

here: $\mu_1 = \dfrac{V_1\rho_1 c_1}{V_a \rho_a c_a}$,

m_a is the specific cooling rate of the body a, m_1 is the cooling rate of the body a, if the intensity of heat delivery is equal to the intensity of heat delivery of the body studied, k is the heating intensity of the body a, θ is the generalized temperature of the heat source, τ is time,

here: $m_a = \dfrac{\bar{h}_a A_a}{V_a \rho_a c_a}$, $m_1 = \dfrac{\bar{h}_1 A_1}{V_a \rho_a c_a}$, $k = \dfrac{N}{V_a \rho_a c_a}$,

where A_a is the area of EMIS, A_1 is the area of the heat source.

Insertion

A heterogeneous body consists of three main components:

a) an external body that delivers its heat to the environment;

b) interior environment (fluid), interior equipment and contructions covered by the external body;

c) heat source, *i.e.* one of the interior bodies.

The method aims at the mathematical modelling of the heterogeneous body heat transfer process into the mentioned three main components that correspond to the real heat transfer process because of ever-changing temperature in both internal and external fluids. The generalized equation system of the heat balanse of this model is as folows:

$$\begin{cases} \dfrac{d\theta_a}{d\tau} = k - m_a(\theta_a - \vartheta), \\ \dfrac{d\vartheta}{d\tau} = \dfrac{1}{\mu_s}\left[m_a(\theta_a - \vartheta) - \mu_1 m_1(\vartheta - \theta_1)\right], \\ \dfrac{d\theta_1}{d\tau} = m_1(\vartheta - \theta_1) - m_{i1}(\theta_1 - \theta_e), \\ \theta_e = f_1(\tau). \end{cases} \tag{7}$$

In the fourth equation of the equation system (7) the external fluid (*i.e.* the environmental) temperature is an independent time function.

The result of the equation solution is the determination of the generalized temperature of each component:

$$\theta_a = f_a(\tau); \quad \vartheta = f_s(\tau); \quad \theta_1 = f_1(\tau).$$

This equation system may be applied either to any heterogeneous body (object) or to its part when non-stationary heat thansfer occurs in it. The following conditions have been accepted for the analytical solution of this equation system:

1. The heat source power is constant. The heat source is either turned on or turned off depending on a desirable regime.

2. The heat receptivity of the environment S in comparison with the receptivity of the body a is extremely small:

$$\frac{V_s \rho_s c_s}{V_a \rho_a c_a} \to 0.$$

3. The limited conditions of the third kind ($\theta_e = 0$) have been accepted with regard to the entire system.

Thus, the heat regime of the entire system may be expressed by the following heat balance equation system:

$$\begin{cases} \dfrac{d\theta_a}{d\tau} = k - m_a(\theta_a - \theta_1)\,, \\ \dfrac{d\vartheta}{d\tau} = \dfrac{m_a}{\mu}(\theta_a - \psi\theta_1)\,, \\ \theta_e = 0\,, \end{cases} \tag{8}$$

here $\psi = \dfrac{\bar{h}_e A_1}{\bar{h}_i A_a} = \dfrac{\theta_{\max}}{\vartheta_{\max}}$.

The equations of the system (8) are the first grade linear differential equations. The solutions of these equations, when the system consists of two main components, *i.e.* heat sources and the surrounding external body, are:

$$\begin{aligned} T &= A\exp(-a\tau) + B\exp(-b\tau), \\ t &= A\exp(-a\tau) + B\exp(-b\tau), \end{aligned} \tag{9}$$

here *A, B, C, D, a* and *b* are the integration constants whose expression is presented in Table.

In spite of the fact that, in general, the analytical solution of the equation system (7) is complicated, these equations were solved with the use of the approximation principle offered by computer technique. However, the drawback of the computer solutions is their incapability to reflect the contents of the integration constants (*i.e.* the dependence on the initial data and on the values of functional extremes). Nevertheless, the possibility to control the extremes of general temperature alteration in the components remains. For instance, the maximal heat source temperature is reached when

$$\frac{d\theta_a}{d\tau} = k - m_a(\theta_a - \vartheta) = 0\,; \quad (\theta_m - \vartheta_m) = \frac{k}{m_a}\,, \tag{10}$$

and the maximum (minimum) of the internal fluid temperature is reached with the satisfaction of the following equation:

$$\vartheta^{\min}_{\max} = \frac{m_a\theta_a + \mu_1 m_1 \vartheta_m}{m_a + \mu_1 m_1}\,, \tag{11}$$

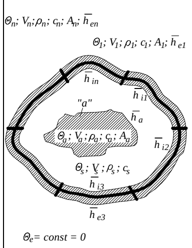
here θ_m and ϑ_m are the values of the generalized temperatures of the heat source and the external body at the moments of the interval fluid temperature extremes.

The external body's general temperature extremes are reached with the satisfaction of the following condition:

$$\vartheta^{\min}_{\max} = \frac{m_1\vartheta_m + m_{1i}\vartheta_m}{m_1 + m_{1i}}\,. \tag{12}$$

Figure 3 shows the heating-cooling process of a heterogeneous body consisting of three components in the initial heat transfer stage. As the general temperature alteration diagrams of the given components demonstrate, the temperature maximums are

Table. **Equations of unstable heat exchange between two bodies**

Scheme of heat exchange	Transformed differential equation	Solution	Constants in equations	Extremes
	$\frac{dT}{d\tau} = \mu\frac{dt}{d\tau} + \frac{m_a}{\varphi}t = k$; *k = 0* Hot body *a* is brought into spase under consideration	*T* = *A*exp(*-a*τ) + *B*exp (*–b*τ) *t* = *C[*exp(*-a*τ)-exp(*–b*τ)	$A = \frac{\psi + r - \mu}{2r}T_0$; $B = \frac{\mu + r - \psi}{2r}T_0$ $C = \frac{T_0}{r}$; $r = \sqrt{(\psi - \mu)^2 + 4\mu}$ $a = \frac{m_a}{2\mu}(\psi + \mu - r)$; $b = \frac{m_a}{2\mu}(\psi + \mu + r)$	$T_{\max} = T_0$; $\tau_{\max} = \frac{1}{b-a}\ln\frac{b}{a}$ $t_{\max} = C\,(\frac{a}{b})^{\frac{a}{b-a}}(1 - \frac{a}{b})$ $\psi = \frac{T_{\max}}{t_{\max}} = \frac{\bar{\alpha}_i F_i}{\bar{\alpha}_a F_a}$
	$\frac{dT}{d\tau} + \mu\frac{dt}{d\tau} + \frac{m_a}{\varphi}t = k$ $k = \frac{N_a}{V_a\rho_a c_a}$ Body *a* is heated up to limited temperature	*T* = *A* + *B* – [*A*exp(–*a*τ) + + *B*exp(–*b*τ)] *t* = *C* + *D* – [*C*exp(–*a*τ) + + *D*exp(–*b*τ)]	$A = t_{np}\frac{(\psi + r)^2 - \mu^2}{4r}$; $t_{np} = \frac{\mu^2 - (\psi - r)^2}{4r}$ $C = D - t_{np}$ $D = t_{np}\frac{\psi + \mu - r}{2r}$ *a* and *b* are as in variant I	$T_{\max} = T_{np} = \frac{\psi k}{m_a(\psi - 1)}$ $t_{\max} = t_{np} = \frac{T_{np}}{\psi}$ $\tau_{\max} \to \infty$; $\psi = \frac{T_{\max}}{t_{\max}}$
	$\frac{dT}{d\tau} + \mu\frac{dt}{d\tau} + \frac{m_a}{\varphi}t = 0$ Body *a* is getting cool	*T* = *A*exp(–*a*τ) + *B*exp (–*b*τ) *t* = *C*exp(–*a*τ) – *D*exp(–*b*τ)	-------- “ --------	$T_{\max} = T_0$ $t_{\max} = \frac{T_0}{\psi}$
	$\frac{dT}{d\tau} + \mu\frac{dt}{d\tau} + \frac{m_a}{\varphi}t = k$ $\frac{dT}{d\tau} + \mu\frac{dt}{d\tau} + \frac{m_a}{\varphi}t = 0$ Body *a* is heated up to temperature T, then both bodies are cooled	Heating: $T_{\lim}$ – [A^1exp(–*a*τ) + B^1exp (–*b*τ)] $t_{\lim}$ – [C^1exp(–*a*τ) + D^1exp (–*b*τ)] Cooling: *T* = *A*exp(–*a*τ) + *B*exp (–*b*τ) *t* = *C*exp(–*a*τ) + *D*exp (–*b*τ)	Heating as II variant. Cooling: $A = \frac{T_p - (\psi - \mu) - 2\mu t_p}{2r}$ $B = \frac{T_p(\mu + r - \psi)t_p}{2r}$ $D = \frac{2T_p - (\psi - \mu + r)t_p}{2r}$ *a* and *b* are as in variant I	$T_{\max} = T_p$ $\tau_{\max} = \left(\frac{aC}{bD}\right)^{\frac{a}{b-a}} C(1 - \frac{a}{b})$ $T_m = \left(\frac{aC}{bD}\right)^{\frac{a}{b-a}}(A - \frac{aC}{bD}B)$ $\tau_{\max} = \frac{1}{b-a}\ln\frac{bD}{aC}$

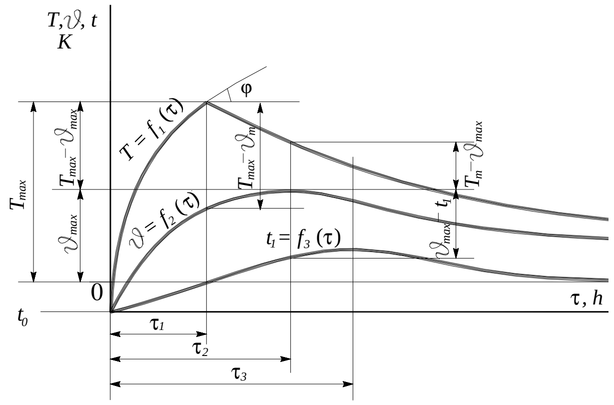


Fig. 3. The heating-cooling process of a heterogeneous body consisting of three components in the initial stage of heat transfer

reached not simultaneously, *i.e.* the retardation of the heating-cooling process of the components with regard to the heat source is obvious. During the solution of the practical tasks of the interval fluid temperature regulation (control), the periods of the component temperature extreme are especially important for the production of heat regime control microprocessors to ensure an optimal energetic effect.

A more detailed analysis of the obtained solutions was carried out when heat interaction occurred between two components of the heterogeneous body. The integral form of the equation expression is useful, since it clearly demonstrates the dependence of the integration constants on the initial data and the interdependence of all the constants.

As is presented in Table, heat interaction between the components of a heterogeneous body is complicated. Since the heating-cooling of the bodies follows the general laws of the saturation process and thus the body temperature alteration is expressed by the exponent function, the heat interaction of the components of a heterogeneous body is expressed as the summing result of the potential heating-cooling process of separate components of the body. Despite the heat changes that occur inside a heterogeneous body, the general temperature dynamics of each component is expressed by the familiar heating cooling laws. In the case of the previously described components of the two heterogeneous bodies:

$$a\tau = \frac{1}{2\mu}(\psi + \mu - r)K_n F_0\ , \qquad b\tau = \frac{1}{2\mu}(\psi + \mu + r)K_n F_0\ , \tag{13}$$

here ψ, μ and r nameless constants are determined on the basis of the initial and limited conditions of heat transfer; K_n is Kondratyev's criterion, F_0 is the Fourier criterion.

When applying this method for the investigation of the non-stationary heat transfer process, the integration constants for a concrete building have been determined and a computer programme was worked out to solve the equation system. According to the formed algorythm, the equation system for buildings is as follows:

$$\begin{cases} \dfrac{d\theta_a}{d\tau} = 8.89 - 0.435\left(\theta_a - \vartheta\right), \\ \dfrac{d\vartheta}{d\tau} = 0.595\left(\theta_a - \vartheta\right) - 0.396\left(\vartheta - \theta_1\right), \\ \dfrac{d\theta_1}{d\tau} = 0.218\left(\vartheta - \theta_1\right) - 0.1\left(\theta_1 - \theta_e\right), \\ \theta_e = 2.5\sin\dfrac{\pi}{12}\tau. \end{cases} \tag{14}$$

The following factors in the system of the non-stationary heat transfer balance equations have been considered:

1. The strength of the heat source, its thermal massiveness, the intensity of heat delivery.
2. Thermal massiveness of interior air and of the building's interior constructions and equipment, the intensity of heat transfer.
3. Thermal resistance of the building's external enclosures, thermal massiveness, the intensity of heat transfer with the air outside.

3. EXPERIMENTAL VERIFICATION OF THE METHOD

The electro-accumulative heating equipment was used for the experimental verification of the method.

The aim of the experiment was to determine the analytical and experimental strength $P = f(\tau)$ that the equipment delivers to the environment during the charging (heating) and discharging (cooling) cycle.

The analytical solution of the task is carried out by determining the values of the constants and with $\Delta\theta_i = f(\tau)$ expression to form a diagram of heat delivery; here $\Delta\theta_i$ is the law of the alteration of temperature differences between the surface of the equipment and the environmental air.

The following system of equations has been compiled made for theoretical investigation:

$$\begin{cases} \dfrac{d\theta_a}{d\tau} = k - m_a\left(\theta_a - \vartheta\right), \\ \dfrac{d\vartheta}{d\tau} = \dfrac{m_a}{\mu}\left(\theta_a - \psi\vartheta\right), \end{cases} \tag{15}$$

were ψ – the ratio of the maximal temperatures of the charge and the surface of the equipment.

The desired law of heat delivery is:

$$P = \bar{h}_i A(\theta_a - \vartheta) = \bar{h}_i A\Delta\theta_i, \quad (16)$$

here $\bar{h}_i$ is the transformed heat delivery coefficient *EMIP* from the equipment; A is the area of the equipment.

The heat delivery curve is expressed by the following equations:

a) at the moment of charging (heating):

$$P_p = \bar{h}_i A\{C - D - [C\exp(-a\tau) - D\exp(-b\tau)]\}, \quad (17)$$

b) at the moment of discharging (cooling)

$$P_p = \bar{h}_i A\{C_a - D_a - [C_a\exp(-a\tau) - D\exp(-b\tau)]\}. \quad (18)$$

A calorimeter of special purpose was constructed and mounted for the experiment. The calorimeter is a closed chamber in which the accumulative heating equipment was located. Its mode of operation is based on the determination of the heat balance. A constant volume of air is sent into the calorimeter, and the curves of temperature changes are determined as follows:

$$\Delta\theta = \vartheta - \theta_e = f(\tau) \text{ and } \Delta\theta = f(P). \quad (19)$$

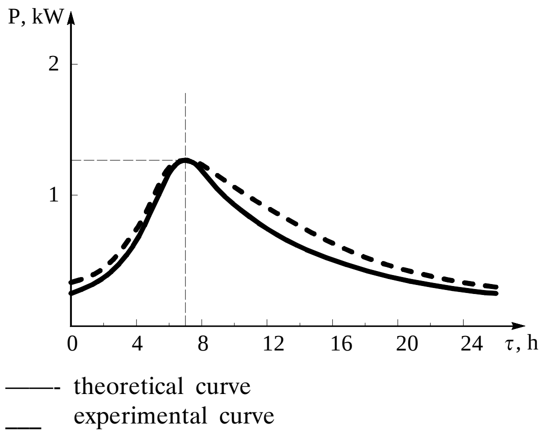


Fig. 4. Theoretical and experimental curves of heat delivery

With these dependencies the main curve of the experiment, $P = f(\tau)$, is found, which describes the capacity of the equipment during any loading-unloading day and night cycle.

Figure 4 shows a comparison of the experimental curve with the theoretical one.

Since, according to the accepted experimental method, heat accumulated by the calorimeter is not estimated, a theoretical diagram of heat delivery is a little higher than the experimental one.

4. CONCLUSIONS

1. The method allows for a detailed analysis of the unstable thermal regime in buildings and other objects. With the use of the proposed calculation program it is possible to find the optimal rates of thermal resistance of buildings as well as the rates of massiveness, to determine the optimal thermal capacity of the heating source and to prepare the algorithm of the heating control systems (for one building or similar building groups).

2. Analysis of the equation solutions shows that from the energetic point of view light materials (with a small specific c and a low a heat conductivity coefficient λ) are more economical in the building's enclosing constructions.

3. In many cases, with the use of this method, it is possible to avoid expensive and extensive experiments meant for the determination of the thermal regime of an object.

4. The results of experimental verification precisely agree with the theoretical calculations.

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References

1. Богусловский В. Н. Строительная теплофизика. Москва: Вычшая школа, 1982. 416 с.
2. Camia F. M. Theory of impulse thermal conductance (Traite determocinètique impulsionelle). Dunod. Paris, 1967. 267 p.
3. Gosele K., Schūle W. Noise, heat, moisture (Schall, Warme, Feuchte). Wiesbaden, Berlin, 1989. 283 p.
4. Grober G., Erk C., Grigull U. Background for heat transfer (Grundgesetze der Warmeubertragung). Springer Verlag/Ggttingen/ Heidelberg, 1963. 436 p.
5. Кондратьев Г. М. Регулярный тепловой режим. Москва: Госстрой СССР, 1954. 408 с.
6. Кузьмин М. П. Электрическое моделирование нестационарных процессов теплообмена. Москва: Энергия, 1974. 416 с.
7. Rudolph G. und Autorenkollektiv. Energetics in construction (Baustellen Energetik). Berlin, 1989. 158 p.
8. Фокин К. Ф. Строительная теплотехника ограждающих частей зданий. Москва: Стройиздат, 1973. 288 с.
9. Juodvalkis J., Blaževičius E., Vipartas R. A. Nestacionarinių šilumos mainų pastatuose balanso analizė. Statyba – Civil Engineering. 2000. T. VI. Nr. 1. P. 32–38.

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NESTACIONARIŲ ŠILUMOS MAINŲ SKAIČIAVIMO METODIKA IR EKSPERIMENTINIS PATIKRINIMAS

S a n t r a u k a

Techniniai šilumos mainų skaičiavimai atliekami stacionarinio proceso lygčių pagrindu, tačiau dauguma šilumos mainų procesų yra nestacionariniai. Dėl universalesnių nestacionarinio šilumos perdavimo režimų skaičiavimo metodų stokos plačiau nenagrinėjama kūnų masyvumo įtaka tiriamų šilumos perdavimo procesų eigai, sunaudojamos energijos kiekiui, procesų reguliavimo galimybėms.

Nagrinėjant kelių kūnų sistemos šiluminės sąveikos klausimus, remiamasi F. M. Camia impulsine šilumos laidumo ir G. M. Kondratjevo reguliaraus šilumos režimo teorijomis. Tarus, kad tam tikrame temperatūrų intervale nagrinėjamo kūno fizinės savybės išlieka pastovios, tuomet reguliaraus šilumos mainų režimo atveju temperatūra bet kuriame taške kyla (krinta) vienodu tempu. Bet kuriame kūne egzistuoja visuma taškų, kurie tam tikru momentu atitinka apibendrintą – vidutinę nagrinėjamo kūno temperatūrą. Tokia izoterminių taškų visuma kūne sudaro uždarą vientisą paviršių, kurio vieta kūne nagrinėjamo reguliaraus šilumos režimo atveju yra fiksuota. Šio įsivaizduojamo ekvivalentinio materialinio izoterminio paviršiaus temperatūros režimas ir vaizduos viso kūno apibendrintą temperatūros režimą. Tokiam be galo plonam materialiam paviršiui galima taikyti Niutono šilumos atidavimo dėsnį. Šilumos kiekis, perduodamas per ekvivalentinį materialų izoterminį paviršių, bus toks pat, jei šilumos perdavimo koeficientas bus ekvivalentiškas realios sienelės perdavimo koeficientui šio paviršiaus atžvilgiu. Uždavinio formulavimui numatomos šios sąlygos: a) kūnų ir aplinkos šilumos mainams – trečios rūšies ribinės sąlygos; b) šilumos atidavimo (spinduliavimu ir konvekcija) koeficientų suma yra pastovi; c) šilumos srautai proporcingi temperatūrų skirtumui. Šilumos mainai yra reguliarūs; d) kiekvienas kūnas traktuojamas kaip materialus paviršius.

Uždavinio formulavimas aprašytas (6) priklausomybe.

Eksperimentiniam metodo patikrinimui buvo įrengtas stendas. Teorinių ir eksperimentinių rezultatų palyginimas parodytas 4 pav.

Raktažodžiai: reguliarus šilumos režimas, ekvivalentinis materialus izoterminis paviršius, apibendrinta temperatūra, redukuotas šilumos atidavimo koeficientas, šilimoaušimo tempas, bevardis masyvumas, Niutono šilumos atidavimo dėsnis, šilumos atidavimo koeficientas

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МЕТОДИКА РАСЧЕТА И ЭКСПЕРИМЕНТАЛЬНАЯ ПРОВЕРКА НЕСТАЦИОНАРНОГО ТЕПЛООБМЕНА

Р е з ю м е

Как правило, технические расчеты теплообменных процессов производятся на основании балансовых уравнений стационарного процесса, однако в реальных условиях большинство теплообменных процессов являются нестационарными. Недостаток универсальных методов расчета нестационарных режимов теплообмена не позволяет глубже оценить влияние массивности тел на прохождение процесса теплообмена, расход энергии, возможности регулирования процессов.

При рассмотрении теплового взаимодействия нескольких тел за основу были приняты теория импульсной теплопроводности тел Ф. М. Камья и теория регулярного теплового режима Г. М. Кондратьева.

Если допустить, что в определенном интервале температур физические свойства тел остаются без изменений, то в случае регулярного режима теплообмена температура в любой точке тела изменяется с одинаковым темпом. В любом твердом теле существует совокупность точек с одинаковой температурой, соответствующей обобщенной усредненной температуре тела.

Такая совокупность изотермических точек в теле образует закрытую непрерывную поверхность, место которой в случае регулярного режима теплообмена в теле остается неизменным. Температурный режим этой мнимой эквивалентной изотермической материальной поверхности и будет характеризовать обобщенное термическое состояние всего тела. Для такой бесконечно тонкой материальной поверхности применим и закон теплоотдачи Ньютона.

Количество тепла, передаваемое через эквивалентную материальную изотермическую поверхность, будет одинаковым, если коэффициент теплопередачи будет эквивалентом коэффициенту теплопередачи реальной стенки по отношению к этой поверхности.

Для формулировки задачи приняты следующие условия: а) для теплообмена между телами и окружающей средой – краевые условия третьего рода; б) сумма коэффициентов теплоотдачи (конвекцией и излучением) остается постоянной; в) тепловые потоки пропорциональны перепаду температур, теплообменный процесс – регулярный; г) каждое тело представляется как материальная поверхность.

Для экспериментальной проверки метода был оборудован стенд. Сравнение теоретических и экспериментальных результатов приводится на рис. 4.

Ключевые слова: регулярный режим теплообмена, эквивалентная материальная изотермическая поверхность, обобщенная температура, преобразованный коэффициент теплоотдачи, режим нагрева – остывания, безразмерная массивность, закон теплоотдачи Ньютона, коэффициент теплопередачи