MODELAGEM MATEMÁTICA DO UM NOVO MÉTODO DE PROTEÇÃO TÉRMICA BASEADA NA INJEÇÃO DE REFRIGERANTES ESPECIAIS

MATHEMATICAL MODELING OF A NEW METHOD OF THERMAL PROTECTION BASED ON THE INJECTION OF SPECIAL COOLANTS

МАТЕМАТИЧЕСКОЕ МОДЕЛИРОВАНИЕ НОВОГО СПОСОБА ТЕПЛОВОЙ ЗАЩИТЫ НА ОСНОВЕ ВДУВА СПЕЦИАЛЬНЫХ ОХЛАДИТЕЛЕЙ

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RESUMO

Um modelo matemático fechado de transferência de massa e calor sob a filtração não-isotérmica através dos poros organizados de refrigerantes com uma forte dependência da viscosidade dinâmica da temperatura com injeção adicional em uma flotação dinâmica gasosa viscosa com a formação de filme líquido e evaporação sob a influência de fluxos de calor aerodinâmicos foi desenvolvida. O objetivo do trabalho foi desenvolver a base física e matemática da proteção térmica com o fornecimento automático de refrigerante tendo uma forte dependência da viscosidade dinâmica da temperatura (de 3 a 5 ordens de grandeza quando a temperatura muda de 200 a 2500°C). Esse sistema de proteção térmica permite operar sem a remoção de massa que preserva a geometria dos elementos estruturais no aquecimento intensivo, o que é muito relevante. Para atingir este objetivo, um modelo matemático do fornecimento automático de refrigerante com filtração através dos poros organizados, injeção em uma camada limite dinâmica de gás de alta temperatura com a formação da película líquida protetora e evaporação foi formulado. Os resultados em relação ao caudal mássico do refrigerante, à taxa de evaporação em massa da película líquida formada e às temperaturas da estrutura, que em todos os casos permanecem abaixo da temperatura de evaporação do refrigerador, foram obtidos.

Palavras-chave: *transferência de massa e calor, filtração, camada limite, pressão parcial de vapor, poros organizados, anisotropia.*

ABSTRACT

A closed mathematical model of heat and mass transfer under the non-isothermal filtration through the organized pores of coolants with a strong dependence of dynamic viscosity on temperature with further injection into a viscous gas-dynamic flotation with the formation of liquid film and evaporation under the influence of aerodynamic heat flows has been developed. The aim of the paper is to develop the physical and mathematical basis of thermal protection with the automatic supply of coolant having a strong dependence of dynamic viscosity on temperature (by 3–5 orders of magnitude when the temperature changes by 200–2500°C). Such thermal protection system allows for operating without the mass removal that preserves the geometry of the structural elements at the intensive heating, which is very relevant. To achieve this goal, a mathematical model of the automatic coolant supply with filtration through the organized pores, injection into a high-temperature gas-dynamic boundary layer with the formation of the protective liquid film and evaporation is formulated. The results with respect to the mass flow rate of the coolant, the mass evaporation rate of the formed liquid film and the temperatures of the structure which in all cases remain below the evaporation temperature of the cooler have been obtained.

Keywords: *heat and mass transfer, filtration, boundary layer, the partial pressure of vapor, organized pores, anisotropy.*

АННОТАЦИЯ

Разработана замкнутая математическая модель тепломассопереноса в условиях неизотермической фильтрации через организованные поры охладителей с сильной зависимостью динамической вязкости от температуры с дальнейшим вдувом в вязкое газодинамическое течение с образованием жидкой пленки и испарением под действием аэродинамических тепловых потоков. Целью работы является разработка физико-математических основ тепловой защиты с автоматической подачей охладителя, имеющего сильную зависимость динамической вязкости от температуры (на 3–5 порядков при изменении температуры на 200–2500°C). Такая система тепловой защиты позволяет функционировать без уноса массы, что сохраняет геометрию элементов конструкции при интенсивном нагреве, что является актуальным. Для достижения поставленной цели сформулирована математическая модель автоматической подачи охладителя с фильтрацией через организованные поры, вдувом в высокотемпературной газодинамический пограничный слой с образованием защитной жидкой пленки и испарением. Получены результаты по массовому расходу охладителя, массовой скорости испарения образовавшейся жидкой пленки и температурам конструкции, которая во всех случаях остается ниже температуры испарения охладителя.

Ключевые слова: *тепломассоперенос, фильтрация, пограничный слой, парциальное давление паров, организованные поры, анизотропия.*

1. INTRODUCTION

One of the challenging methods of thermal protection of the nose parts of hypersonic aircrafts are the active methods of thermal protection through the injection of gaseous or liquid coolants into the high-temperature gas-dynamic boundary layers since the passive methods of thermal protection function under conditions of mass removal when the geometry of the nose parts changes that results in the significant deterioration of aerodynamic characteristics (Amirgaliyev *et al.*, 2017; Mussabayev *et al.*, 2018; Fitsak *et al.*, 2019).

The main difficulty when organizing the active methods of thermal protection through the injection of coolants is in the technological implementation of these methods and in the automatic adjustment of the supply of coolant. The methods of thermal protection through the injection of coolants were considered in (Polezhaev and Yurevich, 1976; Mugalev, 1965).

This paper proposes and mathematically models a new method of thermal protection based on the filtration through the organized pores of special liquid coolant the dynamic viscosity of which changes by 3-5 orders of magnitude when the temperature changes by 200-2500C. As a result, it is possible to supply the coolant automatically without changing the pressure drop between the tank and the external surface (Alekshev *et al.*, 2017; Kulzhumiyeva and Sartabanov, 2017). If the constant pressure of the coolant in the tank exceeding the possible pressure on the external surface is created, an approximately constant pressure drop is created under the action of which the coolant is filtered

with increased flow rate, if the temperature increases and the viscosity of the coolant decreases sharply and vice versa, with decrease of temperature and increase of viscosity the flow rate decreases. For more rational consumption of the coolant, the filtration openings are arranged at different angles; as a result of this, the body becomes anisotropic with different thermal conductivity in different directions. For anisotropic thermal conductivity, it is possible to specify the works of authors (Formalev and Kolesnik, 2007; Formalev *et al.*, 2009; Formalev and Kolesnik, 2013; Formalev and Kolesnik *et al.*, 2015; Formalev *et al.*, 2015; Formalev *et al.*, 2016a; Formalev *et al.*, 2016b; Formalev *et al.*, 2016c; Formalev, and Kolesnik, 2016; Formalev *et al.*, 2017a; Formalev *et al.*, 2017b; Kolesnik, 2017a; Formalev and Kolesnik, 2017b; Formalev and Kolesnik, 2018; Formalev *et al.*, 2018a; Formalev *et al.*, 2018b; Formalev *et al.*, 2018c) and other specialists (Gidasov and Severina, 2015; Gidasov *et al.*, 2016; Prokofiev *et al.*, 2016; Okonechnikov *et al.*, 2016; Gidasov and Severina, 2017; Lurie *et al.*, 2017; Babaytsev *et al.*, 2017; Bulychev *et al.*, 2018a; Bulychev *et al.*, 2018b; Bulychev *et al.*, 2018c; Bulychev *et al.*, 2018d; Formalev *et al.*, 2018c; Rabinskiy and Tushavina, 2018; Gidasov *et al.*, 2018). The mathematical modeling of the complex problem, namely, the anisotropic thermal conductivity, filtration, injection, formation of a protective liquid film with downstream flow and evaporation is carried out for the first time (Gridina and Andreev, 2017).

2. RESEARCH METHODOLOGY

The thermal protection diagram is shown

in Figure 1. The mathematical model includes the following relations. The balance of thermal energy is recorded on the external film boundary (Equation 1). The one-dimensional stationary heat conduction equation, taking into account the filtration of dropping liquid, is as follows (Equation 2). The condition is set on the internal boundary w_2 (Equation 3), and the continuity of heat flows and temperatures is set at the boundary w_1 between the liquid film and the body (Equations 4, 5). The nonlinear filtration law (Formalev and Kolesnik, 2017a) can be represented as (Equation 6). The continuity equation for incompressible liquid (Equation 7). The dynamic viscosity of liquid depending on temperature (Vargaftik, 1972) is determined using the following Equation 8, where A , B and C are the experimental data for one of the types of coolant. The mass evaporation rate of the coolant proportional to the difference between the pressure of saturated steam $(p_{ev})_w$ and its partial pressure $(p_{ev})_{part}$ in the gas mixture is calculated as follows in Equation 9 (Mugalev, 1965). The relationship of the pressure of saturated vapor $(p_{ev})_w$ of the coolant at the boundary w with a temperature T_w of this boundary is set by the Clapeyron-Clausius Equation 10. The partial vapor pressure $(p_{ev})_{part}$ can be determined from the analogy between heat transfer and diffusion (Equation 11) (Avduevsky, 1981).

The boundary conditions for filtration pressure are set as follows Equations 12, 13. In Equations 1 – 13 $\overline{q_c} = \frac{0.21Q_c}{c_p(T_e - T_w)}$ is the relative combustion heat, Q_c – is the combustion heat; β_{ev} – is the coefficient of injection determined using Equation 14 (Mugalev, 1965). The coefficients of thermal conductivity λ_{yy} and permeability k_{yy} are calculated using the Equations 15, 16 (Formalev and Kolesnik, 2013), where λ_ξ , λ_η and k_ξ , k_η – are the coefficients of thermal conductivity and permeability reduced to the main axes $O\xi$, $O\eta$; φ – is the angle between axes $O\xi$ and Oy .

Herewith, the effective value of the thermal conductivity of the porous body $(\lambda_{yy})_{eff}$ is in accordance with (Polezhaev and Yurevich, 1976).

The Reynold's filtration number Re_L in the nonlinear filtration law Equation 6 is determined by Equation 17 (Formalev *et al.*, 2009) in which Equation 18. The mass evaporation rate can be determined from relation Equation 1 based on the concept of effective enthalpy Equation 20, depending on the unknown temperature T_w . The result is as follows Equation 21. The linear filtration rate is determined in accordance with continuity Equation 7 and filtration law (Equation 6) from which the following is obtained Equation 22. Substituting this pressure into Equation 5, it is possible to calculate the value of the linear filtration rate (Equation 23) using a linear temperature distribution which is determined by substituting the Equation 10 and Equation 11 into Equation 9 (Equation 24). From Equations 5, 7, and Equation 2, it is possible to obtain the implicit temperature distribution in the body (Equation 25), where (Equation 26), where C is the constant from Equation 8.

3. RESULTS AND DISCUSSION:

Figures 2–5 show the results of calculations of the characteristics of the proposed thermal protection method. Glycerin, the dynamic viscosity of which is determined by the approximation expression (8) with constants $A = 0.55 \times 10^{-4}$, $B = 800$, $C = 14.29$, (Vargaftik, 1972), was used as the coolant.

The other characteristics of the coolant and the frame had the following values:

$$\begin{aligned} c_f &= 2.5 \text{ kJ/kgK}; c_p = 1 \text{ kJ/kgK}; \Pi = 0.4, \\ (\lambda_{yy})_k &= 0.168 \text{ kW/mK}; \lambda_f = 0.294 \text{ kW/m} \cdot \text{K}, \\ p_{we} &= 2 \times 10^5 \text{ Pa}; p_{w2} = (2.5 - 5.0) \times 10^5 \text{ Pa}, \\ k &= (0.5 - 3) \times 10^{-14} \text{ m}^2; \\ q_w &= \alpha_w (T_e - T_w) = (4 - 14) \cdot 10^3 \text{ kW/m}^2, \\ Q_{\bar{n}} &= 15540 \text{ kJ/kg } (O_2); Q_{ev}^* = 829 \text{ kJ/kg}, \\ M_{ev} &= 0.0921 \text{ kg/mol}, T_{ev}^* = 563 \text{ K}; T_0 = 300 \text{ K}, \\ R_{ev} &= 0.09022 \text{ kJ/kgK}; \delta_T = 0.003 \text{ m}. \end{aligned}$$

The main characteristics of the described method of thermal protection are the mass flow rate of the coolant $(\rho_f v_f)_w$, the mass evaporation rate $(\rho_f v_f)_{ev}$, the temperature of the external boundary of the liquid film and the external boundary of the body w_1 , the distribution of temperature and filtration pressure across the

wall thickness.

Figure 2 shows the distribution of the mass flow rate of the coolant the dynamic viscosity of which significantly depends on the temperature at the change of permeability and heat flow q_{w2} at the internal boundary $w2$. It is clear that with the increase of permeability, k_{yy} the flow rate of the coolant in accordance with Equation 23 increases nonlinearly. This figure also shows that the mass flow rate of the coolant depends nonlinearly both on the heat flow q_{w2} at the boundary $w2$.

Figure 3 shows the dependence of the mass flow rate of the coolant on the angle φ that determines the position of the mutually perpendicular axes $O\xi$ and $O\eta$. The decrease of the flow rate with an increase of angle φ is non-linear both with respect to variable φ and parameter q_{w2} .

Figure 4 shows the mass evaporation rates of the coolant, taking into account the heat flow density supplied to q_w and discharged into the coolant q_{w2} . These distributions are linear both with respect to variable q_w and variable q_{w2} .

It is interesting to note that the temperature of the external boundary w (Fig. 5) is not related to the magnitude of the heat flow supplied q_w remaining close to the boiling point T^* of the coolant but slightly below it, as shown in Fig. 5.

Designations: $\alpha, T, Q, \lambda, c, \rho$ – are, respectively, the heat-transfer coefficient, temperature, heat of physical and chemical transformations, thermal conductivity, thermal capacity, density; $\lambda_\xi, \lambda_\eta, O\xi, O\eta$ – are the main coefficients and main axes of the thermal conductivity tensor; $\overline{q_c}$ – is the relative vapor combustion heat; v – is the velocity; k – is the permeability; Π – is the porosity; μ – is the dynamic viscosity; p – is the pressure; δ_T, δ – are the body and film thicknesses; q – is the heat flow density; R – is the gas constant; M_{ev}, M – are the molar masses of vapor and mixture of the vapor and air; L – is the length of cooling channel; Re_L – is the Reynold's filtration number;

k_1 – is the Boltzmann's constant, $k_1 = 1.38 \times 10^{-23} \text{ J/K}$; m – is the mass of vapor molecule.

Indices: $w, w1, w2$ – are, respectively, the external boundary of the film, the external and internal boundaries of the body; f – is the coolant; ev – is the steam, injection; c – is the combustion; eff – is effective; ϵ is the injection; $*$ is the evaporation; e – is the dynamic gas environment; 0 is the initial value.

4. CONCLUSIONS:

The physical and mathematical basis of a new method of thermal protection of high-speed aircraft based on the automatic flow rate and injection into the gas-dynamic boundary layer with the formation of a liquid film with evaporation have been developed. The novelty is in the selection of coolant with the strong dependence of its dynamic viscosity on temperature (by 3–5 orders of magnitude when heated by 200–250°C). A situation arises in which the constant pressure drop between the tank with coolant and the external gas-dynamic flow rate increases the flow rate of the coolant with the increase of heat flows and temperature in the structure reducing the viscosity and consequently increasing the flow rate of the coolant. Conversely, with a decrease of heat flows, the flow rate decreases. It is possible to obtain the thermal protection system with automatic coolant supply at which the temperature of structure does not exceed the evaporation temperature of the coolant, that is, unlike the passive thermal protection systems with mass removal, the proposed thermal protection preserves the geometrical characteristics of the nose parts of high-speed aircraft.

The results obtained demonstrate that the flow rate of the coolant substantially depends on the permeability of the porous medium approximately in a linear fashion. The effect of coolant flow rate on the orientation angle of the organized pores with respect to the vertical axis was investigated; herewith, the flow rate nonlinearly decreases with the increase of this angle. It is demonstrated that the mass rate of evaporation of the formed coolant film increases linearly with the increase of heat flows to the body. The body temperature increases linearly through the body thickness and is constant at the external boundary equal to the evaporation

temperature of the coolant.

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Figure 1. Thermal protection diagram: 1 – coolant, 2 – porous body, 3 – cooling channels, 4 – liquid film, 5 – coolant vapor

Figure 2. Dependence of the mass flow rate of the coolant on permeability at 1- $q_{w2} = 5 \times 10^3 \text{ kW/m}^2$, 2- $q_{w2} = 6 \times 10^3 \text{ kW/m}^2$, 3- $q_{w2} = 7 \times 10^3 \text{ kW/m}^2$, 4- $q_{w2} = 8 \times 10^3 \text{ kW/m}^2$, 5- $q_{w2} = 9 \times 10^3 \text{ kW/m}^2$

Figure 3. Dependence of the mass flow rate of the coolant on the angle of orientation of the main axis $O\xi$ at 1- $q_{w2} = 5 \times 10^3 \text{ kW/m}^2$, 2- $q_{w2} = 6 \times 10^3 \text{ kW/m}^2$, 3- $q_{w2} = 7 \times 10^3 \text{ kW/m}^2$, 4- $q_{w2} = 8 \times 10^3 \text{ kW/m}^2$, 5- $q_{w2} = 9 \times 10^3 \text{ kW/m}^2$

Figure 4. Dependence of mass evaporation rate on the heat flow density q_w at 1 – $q_{w2} = 5 \times 10^3 \text{ kWt/m}^2$, 2 – $q_{w2} = 6 \times 10^3 \text{ kWt/m}^2$, 3 – $q_{w2} = 7 \times 10^3 \text{ kWt/m}^2$, 4 – $q_{w2} = 8 \times 10^3 \text{ kWt/m}^2$, 5 – $q_{w2} = 9 \times 10^3 \text{ kWt/m}^2$

Figure 5. Dependence of temperature with respect to the body thickness at 1 – $q_{w2} = 5 \times 10^3 \text{ kWt/m}^2$, 2 – $q_{w2} = 6 \times 10^3 \text{ kWt/m}^2$, 3 – $q_{w2} = 7 \times 10^3 \text{ kWt/m}^2$, 4 – $q_{w2} = 8 \times 10^3 \text{ kWt/m}^2$, 5 – $q_{w2} = 9 \times 10^3 \text{ kWt/m}^2$

$$\alpha_w(T_e - T_w)(1 + \bar{q}_c) - (\lambda_{yy})_{eff} \frac{dT}{dy} \Big|_w \equiv (\rho_f v_f)_{ev} \left[Q_{ev}^* + \beta_{ev}(T_e - T_w)(1 + \bar{q}_c) + \int_{T_0}^{T_w} c_f dT \right], \quad y = \delta_T + \delta. \quad (\text{Eq. 1})$$

$$(\lambda_{yy})_{eff} \frac{d^2 T}{dy^2} - \Pi(c\rho)_f v_f \frac{dT}{dy} = 0, \quad 0 < y < \delta_T \quad (\text{Eq. 2})$$

$$(\lambda_{yy})_{eff} \frac{dT}{dy} \Big|_{w2} = q_{w2}, \quad y = 0, \quad (\text{Eq. 3})$$

$$\lambda_f \frac{dT}{dy} \Big|_{w1} = (\lambda_{yy})_{eff} \frac{dT}{dy} \Big|_{w1} \quad (\text{Eq. 4})$$

$$T_f \Big|_{w1} = T \Big|_{w1}, \quad y = \delta_T. \quad (\text{Eq. 5})$$

$$v_f = - \frac{k_{yy}}{\mu_f(T)(1 + \Pi \text{Re}_L)} \frac{dp_f}{dy}, \quad 0 < y < \delta_T. \quad (\text{Eq. 6})$$

$$\frac{dv_f}{dy} = 0 \quad (v_f(y) = \text{const}), \quad 0 < y < \delta_T. \quad (\text{Eq. 7})$$

$$\mu_f^{-1}(T) = \exp[-A(T - B)^2 + C], \quad (\text{Eq. 8})$$

$$(\rho_f v_f)_{ev} = \sqrt{\frac{m}{2\pi k_1 T_w}} [(p_{ev})_w - (p_{ev})_{part}], \quad y = \delta_T + \delta \quad (\text{Eq. 9})$$

$$\ln \frac{(p_{ev})_w}{(p_w)_e} = \frac{Q_{ev}^*}{R_{ev}} \left(\frac{1}{T_{ev}^*} - \frac{1}{T_w} \right), \quad y = \delta_T + \delta. \quad (\text{Eq. 10})$$

$$\frac{(p_{ev})_{part}}{(p_w)_e} = \left[1 + \frac{M_{ev}}{M_e} \left(\frac{Q_{ev}^* + c_f(T_w - T_0)}{I_e - I_w} \right) \right]^{-1}, \quad y = \delta_T + \delta. \quad (\text{Eq. 11})$$

$$p_f(0) = p_{w2} = \text{const}, \quad y = 0. \quad (\text{Eq. 12})$$

$$p_f(\delta_T) = p_{w1} = (p_w)_e, \quad y = \delta_T. \quad (\text{Eq. 13})$$

$$\beta_{ev} = 0.56 \left(\frac{29}{M_{ev}} \right) \left(\frac{T_w}{T_e} \right)^{-0.03} \quad (\text{Eq. 14})$$

$$\lambda_{yy} = \lambda_\xi \sin^2 \varphi + \lambda_\eta \cos^2 \varphi, \quad (\text{Eq. 15})$$

$$k_{yy} = k_\xi \sin^2 \varphi + k_\eta \cos^2 \varphi \quad (\text{Eq. 16})$$

$$\text{Re}_L = \frac{\rho_f(v_f)_m \sqrt{k_{yy}}}{\Pi^{3/2}(\mu_f)_m}, \quad (\text{Eq. 17})$$

$$(v_f)_m = \frac{k_{yy}}{(\mu_f)_m} \frac{p_{w2} - p_{w1}}{L}, \quad (\text{Eq. 18})$$

$$(\mu_f)_m = \frac{1}{T_{w1} - T_{w2}} \int_{T_{w2}}^{T_{w1}} \mu_f(T) dT. \quad (\text{Eq. 19})$$

$$(\rho_f v_f)_{ev} = \frac{\alpha_w (T_e - T_w + 0.21 Q_c / c_p) - (\lambda_{yy})_{eff} (T_w - T_{w2}) / \delta_T}{Q_{ev}^* + \beta_{ev} [c_{pw} (T_e - T_w) + 0.21 Q_c] + c_f (T_w - T_0)}, \quad (\text{Eq. 20})$$

$$\begin{aligned} \sqrt{\frac{m}{2\pi k_1 T_w}} (p_w)_e \left\{ \exp \left[\frac{Q_{ev}^*}{R_{ev}} \left(\frac{1}{T_{ev}^*} - \frac{1}{T_w} \right) \right] - \left[1 + \frac{M_{ev} Q_{ev}^* + c_f (T_w - T_{Haq})}{M_e I_e - I_w} \right]^{-1} \right\} = \\ = \frac{\alpha_w (T_e - T_w + 0.21 Q_c / c_p) - (\lambda_{yy})_{eff} (T_w - T_{w2}) / \delta_T}{Q_{ev}^* + \beta_{ev} [c_{pw} (T_e - T_w) + 0.21 Q_c] + c_f (T_w - T_0)}. \end{aligned} \quad (\text{Eq. 21})$$

$$\frac{p_f(y) - p_{w2}}{p_{w1} - p_{w2}} = \frac{\int_0^y \mu_f(T(y)) dy}{\int_0^{\delta_T} \mu_f(T(y)) dy}. \quad (\text{Eq. 22})$$

$$v_f = \frac{k_{yy} (p_{w2} - p_{w1})}{(1 + \Pi \text{Re}_L) \int_0^{\delta_T} \mu_f(T(y)) dy}, \quad (\text{Eq. 23})$$

$$T(y) = T_{w1} - \frac{q_{w2}}{\lambda_{eff}} (\delta_T - y). \quad (\text{Eq. 24})$$

$$T(y) = T_{w1} - \frac{q_{w2}}{(\lambda_{yy})_{eff}} \int_y^{\delta_T} \exp \left\{ -D \int_0^\eta \exp [-A(T(\xi) - B)^2] d\xi \right\} d\eta \quad (\text{Eq. 25})$$

$$D = \frac{\Pi(c\rho)_f k_{yy}}{(\lambda_{yy})_{eff} \delta_T} (p_{w1} - p_{w2}) \exp(C), \quad (\text{Eq. 26})$$