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NUMERICAL ANALYSIS OF FORCED–CONVECTION HEAT TRANSFER IN RADIOELECTRONIC DEVICE CHANNEL

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A problem of finding the temperature field of printed circuit boards with the sources in the radioelectronic device unit under forced–convection conditions is considered. A heat equation with special type boundary conditions is described. The aim of this study is to increase the velocity of finding the temperature field. The results of numerical experiment are given.

1. Introduction

A forced–convection cooling problem of radioelectronic devices calls for the calculation of printed circuit boards solid temperature fields and air in plane–parallel channels taking into account constructive and thermophysical properties of object investigated. In the classic setting this problem is reduced to solving the heat equation system for solids and energy and motion equations for air flows in conjugated setting (Bastos and Soulnier, 1988; Cadre *et. al.*, 1988; Wang and Saulnier, 1989). The simplified calculation methods of temperature fields based on the simultaneous solving of heat equations for printed circuits and one–dimensional energy equations for air flows and using the correlation dependencies for solving the Navier – Stokes and energy equation systems under given temperature distribution or heat flow density on the solid surfaces have also been studied (Cadre *et. al.*, 1989; Viault *et. al.*, 1989).

A sufficient simple and efficient calculation procedure of forced–convection heat transfer of nonisothermal plates is proposed in (Dorfman, 1982). A calculation of radioelectronic device heat regime, presented in the form of parallel smooth plates with irregular distribution of heat source on surfaces is considered in (Bochkovaja *et. al.*, 1990).

The above mentioned approach to investigation of heat transfer in plane channel is considered in this paper. Numerical experiments and model efficiency estimation are also presented.

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2. Notation

a is a coefficient of temperature conductivity
α is a heat transfer coefficient of isothermal surface
α_ξ, α_x are local heat transfer coefficients
T is a temperature
T_T is a heat carrier temperature
T_e is a environment temperature
ξ, x are coordinates in heat carrier motion direction
y is a coordinate in direction perpendicular to heat carrier motion
i, k are indices
c_1, c_2, g_1, g_2 are coefficients
ν is a coefficient of cinematique viscosity
λ is a coefficient of thermal conductivity
Θ is a overheat
V is a heat carrier motion velocity
f is a function of cold section influence
q is a heat flow density
Re_x is Reynolds number, $\mathrm{Re}_x = \frac{Vx}{\nu}$
Pr is Prandte number, $\mathrm{Pr} = \frac{V}{a}$

3. Forced Convection Heat Transfer Model of Parallel Cannel Boards

Let us consider a simplified problem setting. Let the plate temperature on distance ξ from the edge vary irregulary to some new value T. The heat carrier has the laminar motion with constant velocity. An approximate method is used for solving temperature field problem. The main feature of this method is that it fails to satisfy the differential equations of boundary layer for each individual fluid stream in it. The integral equation is solved by the approximate method and for this reason the main equation can be satisfied only for the mean value in the layer.

Based on Fourier's and Newton's laws we can write the equations as follows (Dorfman, 1982)

$$q = -\lambda \left(\frac{\partial \Theta}{\partial y}\right)_{y=0} = \alpha_x \Theta_\infty , \qquad \alpha_x = \frac{0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{0.5}}{\left(1-\left(\frac{\xi}{x}\right)^{\frac{3}{4}}\right)^{\frac{1}{3}}}, \tag{1}$$

Then we find the law of varying heat flow for random non–isothermal surface. As the heat boundary layer equation is linear, we apply a principle of superposition to it according to which a random heat surface can be obtained as a sum of particular solutions found for a case of surface step–by–step temperature change behind the section.

A solution of this particular problem is given by

$$(T_T - T_e)/(T - T_e) = \Theta(x, y, \xi) \tag{2}$$

At the random temperature distribution $T(x)$ of surface a solution has the following form (Dorfman, 1982)

$$T_T - T_e = \int_o^x \Theta(x, y, \xi)\frac{dT}{d\xi} + \sum_{k=1}^{i} \Theta(x, y, \xi)\Delta T_k \tag{3}$$

In this case the integral refers to the continuous part and a sum refers to the interruptions of temperature distribution curve, ΔT_k is the temperature jump at the point with coordinate ξ_k.

Then we differentiate this inequality with respect to y and use the definition of influence function: $f(x, \xi) = \alpha_\xi/\alpha$, where α is the heat transfer coefficient of the isotermal surface. The value $f(x, \xi)$ defines the cold section effect on coefficient α_ξ behind the jump. Hence, the heat flow density on the random nonisothermal surface has the following form

$$q = \alpha \left[\int_0^x f(x, \xi)\frac{dT}{d\xi}d\xi + \sum_{k=1}^{i} f(x, \xi_k)\Delta T_k\right] \tag{4}$$

This formula is true at any flow regime. The function depends on the relation of variables x and $\xi : f(\xi/x)$ for the non–gradient streamlined plate. If function stress temperature distribution has no interruptions, then

$$q = \alpha \left[\int_0^x f(x/\xi)\frac{dT}{d\xi}d\xi + T(0)\right] \tag{5}$$

The influence function of the general form can be found in the form

$$f(\xi/x) = \left[1 - \left(\frac{\xi}{x}\right)^{c_1}\right]^{-c_2} \tag{6}$$

After carrying out the integration by parts we obtain

$$q = \alpha \left[T + \sum_{k=1}^{\infty} g_k x^k \frac{d^k T}{dx^k}\right] \tag{7}$$

where $g_k = \frac{(-1)^{k+1}}{k!}\left[k\int_0^1 (1 - \xi)^{k-1} + f(\xi)d\xi - 1\right]$

Formula (7) connects the heat flow density and temperature stress for the non–gradient streamlined nonisothermal surface.

Coefficients g_k, c_1, c_2 are calculated for different heat carriers and different flows (Dorfman, 1982). Therefore the heat flow estimation is obtained for body boundary and heat carrier. The correlation thus determined between the heat flow density and temperature stress makes it possible to reduce the conjugate heat transfer problem to the equivalent problem for the heat conduction equation with boundary condition given in (7).

Using integral formula (5) for computing the residual term we obtain the boundary condition as follows

$$q = \alpha \left(T - T_e + g_1 x \frac{dT}{dx} + g_2 x^2 \frac{d^2T}{dx^2} + \varepsilon(x) \right) \tag{8}$$

where the residual term is defined by the difference of heat flow densities q_1 and q_2 computed through integral (5) and differential (7) forms the main correlation $\varepsilon(x) = (q_1 - q_2)/\alpha$.

It is proved theoretically and experimentally, that when the temperature stress increases in the direction of heat carrier flow, then the heat transfer coefficients are greater, but when it decreases, heat transfer coefficients are less than corresponding heat transfer coefficients for the isotermal surface the temperature stress of which is constant (Dorfman, 1982).

If the temperature stress increases by 1.5–2 times, then the local heat transfer coefficient increases by 20–30%. If the temperature stress decreases by 1.5–2 times, then the local coefficient is 1.5–2 times less at the end of plate then corresponding value of isothermal surface. If the plate is longitudinally streamlined, then the temperature stress may be decreasing as well as increasing.

It follows from the above that when designing radioelectronic devices in order to get the optimal cooling of separate units and blocks it is necessary to receive an increase in temperature stress in the direction of heat carrier flow. In practice, the intensity of heat transfer depends not only on the values of heat transfer coefficients but on the distribution of temperature stress because the inlet and outlet heat flow is determined by the integral of heat transfer coefficients products for corresponding temperature stresses. Hence choosing the distribution of temperature stress that locate the heat sources on the printed circuit board in such a manner that it can ensure for cooling system the performance of a certain condition, as for example the maximum intensity of heat transfer.

Mathematical problems dealing with the optimal heat transfer regimes are reduced to the search of functions $T(x)$ or $q(x)$ corresponding to extreme values of integral (5) or quantities depending on them, for example, total heat flow, determined as integral on the surface from the local heat flow density. The solution of such problems in a general from is connected with great difficulties.

It is better to use equalities (5) and (7) or to solve corresponding conjugate problems for comparising different concrete variants and choosing a better one. Though, not all possibilities of optimization are used for that, nevertheless all the results obtained are rather interesting and useful (Dorfman, 1988).

4. Numerical Experiments

Numerical experiments have been carried out on IBM PC/AT(XT). The method of finite differences is used for model discretization. A channel formed by thermal insulation parallel plates is chosen as an object of investigation. The channel cross–section is presented in Figure 1. The thermal sources are located in the horizontal parallel channel. The motion of heat carrier is marked by the arrows.

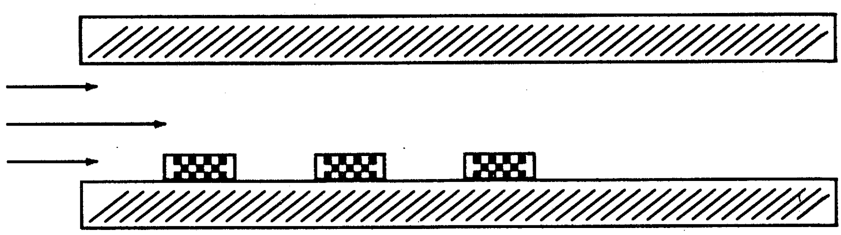

Fig. 1. The channel with heat sources.

The temperature distribution on the wall with the sources is obtained by numerical modelling of the heat transfer process in the channel for a case when Re=600. The curve of temperature distribution in section lining perpendicularly to the plane of middle channel is presented in Figure 2.

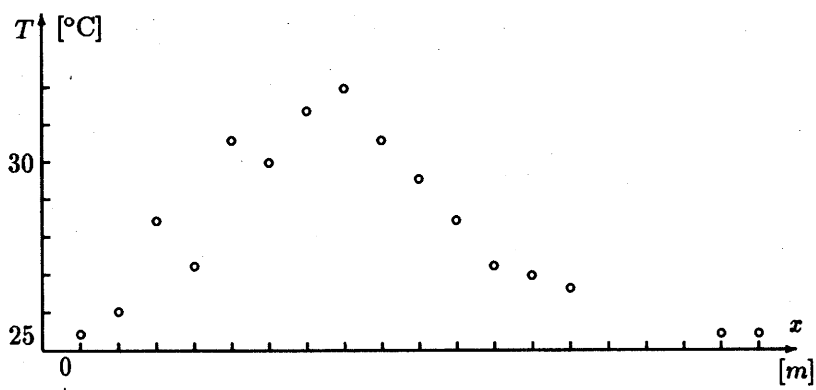

Fig. 2. The curve of temperature distribution.

The comparison of results with the data obtained by using a model comparising the heat equation system for channel walls and motion and energy equations for heat carrier with the conditions of conjugation on the boundary separating two environments shows that the error is 7% and is no more than 2 °C.

The computing time of the problem on IBM PC/AT(XT) depends dimensionally on the finite–difference grid and is several minutes. Consequently the model suggested can be used in radioelectronic devices for which the rate of obtaining a solution is of special importance as a result of the multiplicity and multivariants of computations perfomed.

5. Conclusions

The approximate solution of heat transfer forced–convection problem of plate was obtained in equivalent conjugate setting, allowing us to reduce it to heat equation for solid with the special type boundary conditions taking into account the heat carrier motion and nonisothermal condition of solid. The solution was supposed to be laminar behaviour of heat carrier motion but under the correction of coefficients it may be expanded for a case of the random rate distribution of heat carrier as well as for the turbulent condition. The numerical analysis of model was carried out for heat carrier motion in the channel with the sources on walls.

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