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Original Research Article

TEMPERATURE PROFILES IN A MICRO-PROCESSOR COOLED BY DIRECT REFRIGERANT EVAPORATION

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Abstract

An analytical solution to the equation for cooling of a unit, in the interior of which heat is generated, is presented. For that reason, a simplified non-stationary model for determination of the temperature distribution within the unit, temperature of the contact between unit and a liquid layer, and the evaporating layer thickness in the function of time, is elaborated. A theoretical analysis of the external cooling of the unit, by considering the phenomenon of the liquid evaporation with the use of the Fourier and Poisson's equations, is given. Both, stationary- and non-stationary description of the cooling are shown. The obtained results of simulation seems to be useful in designing the similar cooling systems. A calculation mode for a cooling systems equipped with the compressor heat pump, as an effective cooling method, is also performed.

Keywords: internal heat sources; cooling; processor; vaprocompressor cycle; evaporation

1. INTRODUCTON

The problem of cooling components with internal heat sources is now one of the classic problems of heat exchange, commonly recognised in literature. This type of phenomena is described by a general heat conduction equation, which is

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widely discussed in scientific literature as evidenced by works [1-3]. One particular problem is determining the conditions on the edge of a cooled object that has internal heat sources that vary versus time. This problem does not receive sufficient attention in scientific literature. Determining the conditions of liquid evaporation on the surfaces of studied units as an effective means of cooling the studied object, including the determination of the evaporating liquid surface thickness, is very important and worthy of attention. In literature, determining the conditions on the edge of the studied object's surface is called an inverse problem [3].

One of the more important problems in contemporary computers is cooling intensively heated electronic units that constitute the computer, and ensuring the temperature does not exceed the prevent temperature. The units used in contemporary computers, primarily integrated circuits, consist of microprocessors whose work efficiency is growing exponentially, leading to an increasingly more intensive generation of heat by the units. This necessitates using increasingly more effective methods of cooling. The cause of heat generation within the components are internal heat sources distributed throughout its space. One of the most effective methods of cooling is the use of the phenomenon of liquid evaporation on the surface of cooled units. In order to induce the phenomenon of evaporation on the surface of cooled units, the use of a compressor heat pump is very convenient. The phenomenon of coolant evaporation takes place in the heat pump's evaporator, which is called the thermodynamic cycle's lower heat source conducted in the heat pump. Designing cooling systems requires the ability to learn the process of heat flow in units submitted to cooling, while paying extra attention to the cooling surface. An example of increased efficiency of computer unit cooling is the improvement of classic ventilation devices in terms of performance [4]. Articles [5,6] present interesting research based on the original method of cooling selected components (MEMS - Micro-Electro-Mechanical Systems). The authors of the works applied a very effective method of cooling based on coolant evaporation on studied units, and a new method of transporting the liquid to a selected location within the computer based on the phenomenon of magnetohydrodynamics. The works are, above all, of an experimental nature.

The issue of magnetic transport for laminar flow of appropriate coolants in pipes was solved theoretically in works [7-9].

Very interesting and important papers are [10-14] which are concentrating on cooling electronic units. The application of two-phase cooling cycles which use micro-evaporation technology have been reported by Marcinichen et al. [11]. These works relate to the cooling systems and are primarily experimental works. The theoretical analysis of the process of microprocessor cooling was conducted in work [15] for cooling via coolant evaporation, while work [16] concentrated on variable load conditions via water stream cooling. Analytical solutions for stationary heat flow through a processor layer with internal heat sources and a layer evaporating liquid, whose thickness was determined a priori, was presented in the aforementioned work [15], which is in the form of a short announcement. There is a noticeable lack of theoretical works in scientific literature about studying the cooling of processors based on the coolant's surface evaporation and solving the full scientific problem, i.e. considering the non-stationary conditions of cooling selected computer components including the determination of evaporating liquid's surface thickness. Not enough attention is paid to temperature distribution within the processor. Processor temperature is an essential parameter determining the work quality of the device and it must not exceed the critical value. Currently, it is hard to acquire data, due to the rapid developments in computer technology, which define the thermophysical properties of the material used in the production of processors; that knowledge is of the utmost importance for a full description of the phenomenon of heat conduction in studied components.

The purpose of this work is to present a theoretical study of heat flow in a component with internal heat sources. Determining temperature distribution within the component, the contact temperature between the component and the liquid layer, and the evaporation layer thickness. There is a need for the creation of appropriate conditions for coolant evaporation on the surfaces of cooled components. One of the methods of delivering the boiling agent to a selected place within the computer and creating conditions in that place to enable coolant evaporation is the use of a compressor refrigeration cycle. The aforementioned technology is used in the work.

2. UNIT WITH INTERNAL HEAT SOURCES COOLING MODEL

The location of the studied component is described with a set of coordinates with the vertical axis y having its beginning at the lower, horizontal coordinate of its surface (Fig. 1). The geometric dimensions of the component describe the height H and frontal horizontal surfaces F. The unit's material has density ρ_C, specific heat c_C and thermal conductivity λ_C. Inside the unit are evenly arranged internal heat sources with volumetric heat load $\dot{q}_v$. Heat flow in unit $\dot{q}$ determines temperature distribution T. By assumption, one side of the component is thermally isolated from the environment $\partial T \partial y^{-1} = 0$, while on the other side the unit is in contact with a layer of evaporating liquid of thickness δ. The liquid has density ρ_L, specific heat c_L and thermal conductivity λ_L. The contact layer between the liquid and the unit is at contact temperature T_C. The phenomenon of liquid evaporation with vapor $\dot{m}_S$, heat of evaporation h_{fg} and temperature of evaporation T_S takes place on the free surface of liquid. A liquid of stream $\dot{m}_l$ flows through the layer of liquid at the time t.

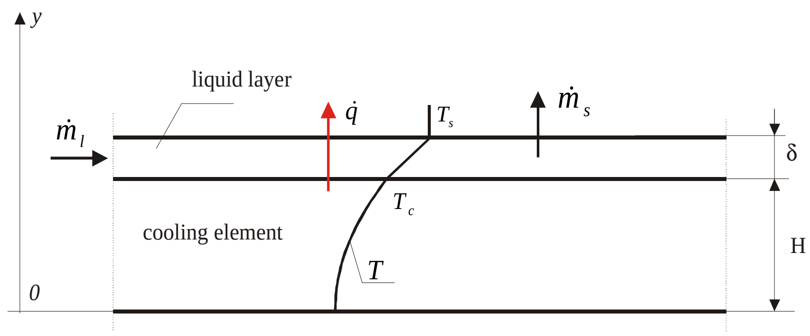

Fig. 1. Element needing cooling of a component with heat source

The process of studied unit cooling can be split into the following special cases:

a. A stream of liquid inflowing to the liquid layer is equal to vapor flow $\dot{m}_L = \dot{m}_S$. It corresponds to the constant thickness of liquid layer $\delta = const.$, $d\delta = 0$;
b. A stream of inflowing liquid is greater than the jet stream $\dot{m}_L > \dot{m}_S$, then $d\delta > 0$;
c. A stream of inflowing liquid is smaller than the jet stream $\dot{m}_L < \dot{m}_S$, then $d\delta < 0$.

Case (a) may be or not a stationary process. If, apart from the equality of liquid and jet streams, the condition of the equality of streams of heat generated within the component $\dot{q}_v(t)HF$ and of heat removed by cooling $\dot{m}_S h_{fg}$ is met, then a stationary process takes place. If the heat stream generated within the unit is greater than the heat stream received, one observes an increase of temperature in time. Otherwise there is a decline in the unit temperature.

The other two cases (b and c) are unconditional non-stationary processes, which might occur differently depending on the relation between the heat generated within the unit and the cooling heat. Notice that non-stationary processes are of a limited duration time unless external cyclic ones changing the direction of heat flow occur.

3. STATIONARY PROCESS OF UNIT COOLING

In a stationary process, liquid and jet streams are equal to each other thermodynamic parameters do not change in time *t*. The solution to the stationary problem is based mainly on designating the temperature field in the studied unit T and liquid layer T_L and the thickness of evaporating liquid layer δ.

1.1. Heat flow in the studied unit

Heat flow in a unit with internal heat sources can be described by the dimensionless Poisson′s equation and dimensionless boundary conditions:

$$\frac{d^2\theta}{d\tilde{y}^2} + \tilde{q}_v = 0 \, , \tag{1}$$

$$\frac{d\theta}{d\tilde{y}} = 0 \text{ for } \tilde{y} = 0 \text{ and } \theta = \theta_c \text{ for } \tilde{y} = 1 \tag{2}$$

where coordinate, unit temperature, contact layer temperature and internal heat source parameter are respectively designated:

$$\tilde{y} = \frac{y}{H}; \;\; \theta = \frac{T - T_s}{T_0 - T_s}, \;\; \theta_c = \frac{T_c - T_s}{T_0 - T_s}; \;\; \tilde{q}_v = \frac{\dot{q}_v H^2}{\lambda_c (T_0 - T_s)} \tag{3}$$

The solution to differential Poisson equation (1) while applying the boundary conditions (2) is equation

$$\theta = \theta_c + \frac{\tilde{q}_v}{2}\left(1 - \tilde{y}^2\right) \tag{4}$$

which presents the stationary parabolic heat distribution within the unit’s layer. It depends on two parameters: temperature of the contact layer and the internal heat source.

1.2. Heat flow in the liquid layer

Stationary heat flow in a liquid layer of thickness *h*, temperatures T_C and T_S on the layer’s borders, from which it is received by the evaporating liquid of heat streams evaporation $\dot{q}_{vap}$ on one edge, while on the second one a heat stream from the cooled component is delivered. The problem of heat flowing through a liquid layer is described in a dimensionless form in the differential Fourier equation

$$\frac{d^2\theta_L}{d\tilde{y}^2} = 0 \tag{5}$$

For which the boundary conditions are

$$\theta_L = \theta_c \text{ for } \tilde{y} = 1 \text{ and } \theta_L = 0 \text{ for } \tilde{y} = 1 + \tilde{\delta} \tag{6}$$

Additional dimensionless coordinates were introduced into equations (5) and (6)

$$\tilde{\delta} = \frac{\delta}{H}, \quad \theta_L = \frac{T_L - T_S}{T_O - T_S} \tag{7}$$

The solution to the differential equation (5) while applying the boundary conditions (6) is described with a linear distribution of temperature in the liquid layer

$$\theta_L = -\frac{\theta_C}{\tilde{\delta}}\tilde{y} + \frac{1+\tilde{\delta}}{\tilde{\delta}}\theta_C \tag{8}$$

In the contact layer, in addition to temperature continuity $(\theta = \theta_L = \theta_C)$ there is also an equality of heat streams

$$\tilde{\lambda}\frac{\partial \theta}{\partial \tilde{y}}\bigg|_{\tilde{y}=1} = \frac{\partial \theta_L}{\partial \tilde{y}}\bigg|_{\tilde{y}=1} \tag{9}$$

where $\tilde{\lambda} = \lambda_C / \lambda_L$ means the relation of the thermal conductivity coefficient of the material of which the cooled component is built and that of the coolant.
From the condition of heat stream continuity (9) with the application of equations describing temperature fields (4) and (8), one receives a simple relation illustrating the relation between the evaporating substance's thickness $\tilde{\delta}$ and the temperature θ_C

$$\theta_C = \tilde{\delta}\tilde{\lambda}\tilde{q}_v \tag{10}$$

The relation presented above leads to the conclusion that contact temperature increases with the increase in liquid layer's thickness. Temperature distribution in the studied component for exemplary contact temperatures θ_C and the corresponding thickness values of evaporating substances $\tilde{\delta}$ are presented in Figure 2.

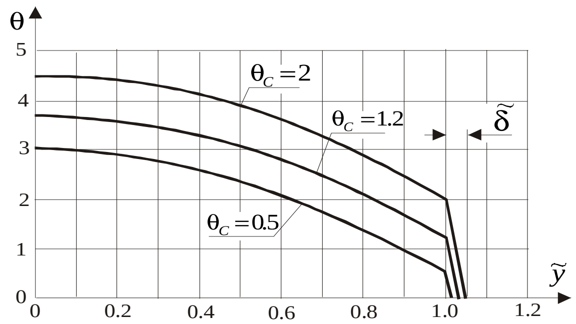

Fig. 2. Temperature distribution in a unit with heat source for $\tilde{q}_v = 5$ and $\tilde{\lambda} = 8$

The maximum temperature in the studied unit is $\theta_m = \theta_c + \tilde{q}_v\, 2^{-1}$ and it is present on the edge of adiabatic surface $\tilde{y} = 0$, which, in order to allow the device to work properly, must not exceed the acceptable value.
In stationary conditions, the heat stream conducted via the liquid layer is equal to the heat stream rejected via the evaporating liquid. This relation is described with equation

$$-\left.\frac{\partial \theta_L}{\partial \tilde{y}}\right|_{\tilde{y}=1+\tilde{h}} = \tilde{q}_{vap} \tag{11}$$

in which a dimensionless evaporation parameter was designated using a complex expression

$$\tilde{q}_{vap} = \frac{\dot{m}_s h_{fg} H}{\lambda_L (T_0 - T_S) F}$$

where: $\dot{m}_s$ is evaporating liquid stream h_{fg} is liquid evaporation heat, F is frontal surface of the studied component.
By using equations (8) and (11), an equation can be formed determining the cooling agent's vapour flow

$$\dot{m}_S = \frac{\dot{q}_v FH}{h_{gf}} \tag{12}$$

4. NON-STATIONARY COOLING OF THE UNIT WITH INTERNAL HEAT SOURCES

The problem of cooling a component with internal heat sources is, as already stated, one of the classic problems of heat exchange, for which a solution is commonly known in literature. The cooling problem considered in this work is complex and concerns the entirety of the circuit: the cooled component and the evaporating liquid surface. That is why the solution is presented in two stages in order to show a full image of the problem: a non-stationary cooling process with the constant contact temperature and a non-stationary cooling process with the varying contact temperature.

4.1. Non-stationary cooling process with constant heat sources and constant contact temperature.

Non-stationary heat-flow in the unit is described by a heat conduction equation

$$K\frac{\partial \theta}{\partial \tau} = \tilde{q}_v + \frac{\partial^2 \theta}{\partial \tilde{y}^2} \tag{13}$$

in which new dimensionless parameters are introduced: time and coolant evaporation parameter

$$\tau = K\frac{a_C t}{H^2}, \quad K = \frac{h_{fg}}{c_L\left(T_o - T_S\right)} \tag{14}$$

By following a similar example from literature [Staniszewski, 1980], the temperature, due to the linearity of the equation (13), can be described as a sum of the temperatures for stationary and non-stationary distribution

$$\theta = \theta_1(\tilde{y}) + \theta_2(\tilde{y}, \tau) \tag{15}$$

By substituting the aforementioned expression into heat conduction equation (13), the task resolves itself into to solving two separate differential equations:

$$\frac{\partial^2 \theta_1}{\partial \tilde{y}^2} = -\tilde{q}_v \quad \text{and} \quad K\frac{\partial \theta_2}{\partial \tau} = \frac{\partial^2 \theta_2}{\partial \tilde{y}^2} \tag{16a,16b}$$

By solving those two equations at a constant efficiency of internal heat sources $\tilde{q}_v$ and introducing equation (15), and by meeting boundary conditions (formulated differently than in article [2], due to the assumption that one of the partitions is adiabatic):

$$\tau = 0 \text{ and } o \le \tilde{y} \le 1, \; \theta_2(0) = 1 - \theta_C - \frac{\tilde{q}_v}{2}\left(1 - \tilde{y}^2\right), \; \theta(0) = 1 \tag{17a}$$

$$\tau > 0 \text{ and } \tilde{y} = 0, \; \frac{\partial \theta_2}{\partial \tilde{y}} = 0, \frac{\partial \theta}{\partial \tilde{y}} = 0 \tag{17b}$$

$$\tau > 0 \text{ and } \tilde{y} = 1, \; \theta_2 = 0, \theta = \theta_C \tag{17c}$$

an analytical solution is reached

$$\theta = \theta_C + \frac{\dot{q}_v}{2}\left(1 - \tilde{y}^2\right) - \frac{4}{\pi}\sum_{n=1}^{\infty}\frac{1}{n}\exp\left(-\frac{\pi^2 n^2}{4K}\tau\right)\left[1 - \theta_C - \tilde{q}_v\left(1 - \frac{4}{\pi^2 n^2}\right)\right]\cos\frac{\pi\, n\, \tilde{y}}{2} \tag{18}$$

where constants are respectively equal $n = 1, 3, 5, ...$

As one can see, equation (18) presents a convergent series and for time $\tau \to \infty$ the equation describes a stationary temperature distribution within the unit.

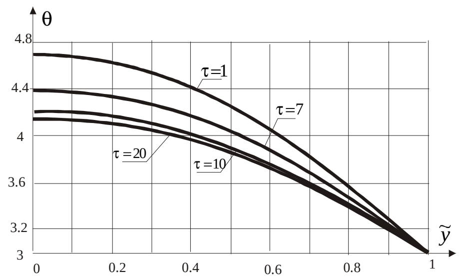

Fig. 3. Temp. distribution within a cooled component for $\theta_C = 3$, $\tilde{q}_v = 2.29$, $K = 6$

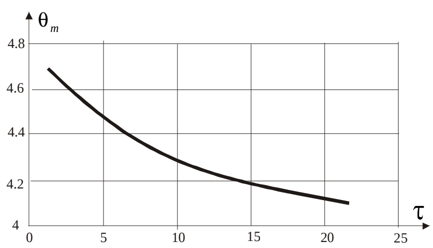

Fig. 4. Maximum time-dependent temperature for $\theta_C = 3$, $\tilde{q}_v = 2.29$, $K = 6$

Temperature distribution (18) depending on location and time in graphic form is presented in Figure 3. Figure 4 presents the dependence of maximum temperature with time. The solution was reached at constant contact temperature for time $\tau > 0$ and constant efficiency of internal heat sources $\tilde{q}_v = const.$

4.2. Non-stationary cooling process with heat sources efficiency and varying contact temperature.

In the solutions presented above, the contact temperature θ_c is a constant and has a set value. In practical terms it is a value that is usually unknown and varying in time. Furthermore, the value of internal sources is dependent on time $\tilde{q}_v = \tilde{q}_v(\tau)$. Heat conduction in the studied component and the liquid layer is simplified and described with time-dependent mean temperatures in order to solve the problem.

The global heat balance for the studied unit and the liquid layer can be described with equation

$$\frac{d}{dt}\left[m_C c_c\left(\overline{T}(t) - T_0\right) + \rho_L c_L F \delta\left(\overline{T}_L(t) - T_0\right)\right] = \dot{q}_v(t) HF - \dot{m}_S h_{fg} \tag{19}$$

Where $\overline{T}(t)$ and $\overline{T}_L(t)$ are mean temperatures of the unit and of the liquid layer in terms of time.

The terms on the left side of the equation (19) designate respectively: the first one: changes of heat capacity in the unit; the second one: changes in heat capacity in the coolant layer. On the right side of the equation is a difference between heat streams generated in the unit and cooling heat generated by liquid evaporation.

By treating the non-stationary process of element and liquid layer cooling as sets of stationary temperature distributions for different times, or, in other words, using a variation of temperatures and the parameter of internal heat source efficiency depending on time based on equations (4) and (8)

$$\theta(\tau) = \theta_c(\tau) + \frac{\tilde{q}_v(\tau)}{2}\left(1 - \tilde{y}^2\right) \text{ and } \theta_L(\tau) = \theta_C(\tau)\left(-\frac{\tilde{y}}{\tilde{\delta}} + \frac{1+\tilde{\delta}}{\tilde{\delta}}\right) \tag{20}$$

time-dependent mean temperatures of the unit and liquid layer depending on time, introduced in equation (19), are expressed by equations

$$\overline{T}(t) = \frac{\int_o^H T(t)dy}{H}, \quad \overline{T}_L(t) = \frac{\int_H^{H+h} T_L(t)dy}{h} \tag{21}$$

which, by using equations (4) describing temperature fields in the unit and the liquid layer in any given moment t are:

$$\overline{T}=\frac{\int_o^H T(t)dy}{H}=T_S+(T_0-T_S)\int_0^1\theta(\tau)d\tilde{y}=T_S+(T_0-T_S)\left[\theta_C(\tau)+\frac{\tilde{q}_v(\tau)}{3}\right] \tag{22a}$$

$$\overline{T}_L=\frac{\int_H^{H+h} T_L(t)dy}{h}=T_S+(T_0-T_S)\frac{\theta_C(\tau)}{2} \tag{22b}$$

The rate of change of the unit's mean temperature in time is equal

$$\frac{d\overline{T}}{dt}=\frac{Ka_C}{H^2}(T_0-T_S)\left[\frac{d\theta_C(\tau)}{d\tau}+\frac{1}{3}\frac{d\tilde{q}_v(\tau)}{d\tau}\right] \tag{23}$$

The aforementioned equation (23) proves that the rate of changes of the unit's mean temperature in time depends on the rate of temperature changes in the contact layer $\theta_C(\tau)$ and internal heat sources efficiency parameter $\tilde{q}_v(\tau)$. Equation (23) becomes significantly more simple if one assumes the invariability of internal heat sources $\tilde{q}_v=const$, as then the unit's mean temperature rate of change is dependent only on temperature change in the contact layer. By using expression (22), the equation for unit and liquid layer heat balance (19) comes down to a heat balance equation in a dimensionless form

$$\frac{d}{d\tau}\left[\theta_C(\tau)+\frac{\tilde{q}_v(\tau)}{3}+\frac{\tilde{a}}{\tilde{\lambda}}\tilde{\delta}\left(\frac{\theta_C(\tau)}{2}-1\right)\right]=\frac{1}{K}\left[\tilde{q}_v(\tau)-\frac{\tilde{q}_{vap}}{\tilde{\lambda}}\right] \tag{24}$$

Differential equation (24) should be solved applying the initial conditions

$$\theta_c=\theta_{C0}\ \text{and}\ \tilde{q}_v(\tau)=\tilde{q}_v(0)\ \text{for}\ \tau=0 \tag{25}$$

The solution to differential equation (24) applying conditions (25), in which the contact layer temperature $\theta_C(\tau)$ and corresponding liquid layer thickness $\tilde{h}(\tau)$ expressed by equation (10) are unknown, is presented as foolows

$$\theta_C{}^2(\tau)-\theta_{C0}{}^2+2\frac{\tilde{\lambda}^2\tilde{q}_v(\tau)-\tilde{a}}{\tilde{a}}(\theta_C(\tau)-\theta_{C0})-\frac{2\tilde{\lambda}^2\tilde{q}_v(\tau)}{\tilde{a}K}\int_0^\tau\left(\tilde{q}_v(\tau)-\frac{\tilde{q}_{vap}}{\tilde{\lambda}}\right)d\tau$$
$$+\frac{2}{3}\frac{\tilde{\lambda}^2\tilde{q}_v(\tau)}{\tilde{a}}[\tilde{q}_v(\tau)-\tilde{q}_v(0)]=0. \tag{26}$$

Contact temperature $\theta_C(\tau)$, which is the solution to quadratic equation (26), is defined by equation

$$\theta_C(\tau)=-\frac{\tilde{\lambda}^2\tilde{q}_v(\tau)-\tilde{a}}{\tilde{a}}+\sqrt{\left(\frac{\tilde{\lambda}^2\tilde{q}_v(\tau)-\tilde{a}}{\tilde{a}}\right)^2-\Lambda(\tau)} \tag{27}$$

Here expression $\Lambda(\tau)$ dependent on time equals

$$\Lambda(\tau)=-\theta_{C0}{}^2-2\frac{\tilde{\lambda}^2\tilde{q}_v(\tau)-\tilde{a}}{\tilde{a}}\theta_{C0}-\frac{2\tilde{\lambda}^2\tilde{q}_v(\tau)}{\tilde{a}K}\int_0^\tau\left(\tilde{q}_v(\tau)-\frac{\tilde{q}_{vap}}{\tilde{\lambda}}\right)d\tau + \frac{2}{3}\frac{\tilde{\lambda}^2\tilde{q}_v(\tau)}{\tilde{a}}[\tilde{q}_v(\tau)-\tilde{q}_v(0)] \tag{28}$$

For an exceptional case when the initial condition for contact temperature $\theta_{C0}=1$ and with an additional assumption of heat source's constancy $\tilde{q}_v=\tilde{q}(0)=const$, the dependence of temperature in the contact layer and of evaporation liquid thickness on time is presented by equations

$$\theta_C(\tau)=-\frac{\tilde{\lambda}^2\tilde{q}_v-\tilde{a}}{\tilde{a}}+\sqrt{\left(\frac{\tilde{\lambda}^2\tilde{q}_v}{\tilde{a}}\right)^2+\frac{2\tilde{\lambda}^2\tilde{q}_v}{\tilde{a}K}\left[\left(\tilde{q}_v-\frac{\tilde{q}_{vap}}{\tilde{\lambda}}\right)\tau\right]} \tag{29}$$

$$\tilde{\delta}(\tau)=-\frac{\tilde{\lambda}^2\tilde{q}_v-\tilde{a}}{\tilde{\lambda}\tilde{q}_v\tilde{a}}+\frac{1}{\tilde{\lambda}\tilde{q}_v}\sqrt{\left(\frac{\tilde{\lambda}^2\tilde{q}_v}{\tilde{a}}\right)^2+\frac{2\tilde{\lambda}^2\tilde{q}_v}{\tilde{a}K}\left[\left(\tilde{q}_v-\frac{\tilde{q}_{vap}}{\tilde{\lambda}}\right)\tau\right]} \tag{30}$$

Equations (27, 28) allow for a theoretical description and study of various conditions of cooling elements with internal heat sources. Generated heat $\tilde{q}_v$ in cooled units, as exemplified by a computer processor, is a derivative of the kind of the device's work.

Functions $\tilde{q}_v$ are usually irregular functions of time and their shape depends on the workload of studied computer units. The efficiency of internal heat sources $\tilde{q}_v(\tau)$ in the processor in actual working conditions of a computer changes in time. The form of this function is difficult to formulate. One may attempt an approximation of actual heat sources with a function featuring cyclic changes in time

$$\tilde{q}_v(\tau)=\tilde{q}_{v0}+\sum_{n=1}^{n}\tilde{\tilde{q}}_{vn}\sin(\omega_n\tau) \tag{31}$$

where $\tilde{q}_{v0}$, $\tilde{q}_{vn}$, ω_n are constants resulting from the nature of processor's workload.

The analysis of the aforementioned equations suggests, that in a non-stationary process of processor cooling (or heating), both contact temperature and evaporating liquid layer thickness are dependent on time. If, on the other hand, the intensiveness of processor heating and cooling are equal $\tilde{q}_v=\tilde{q}_{vap}/\tilde{\lambda}$, then the non-stationary process changes into a stationary state. If $\tilde{q}_v > \tilde{q}_{vap}/\tilde{\lambda}$ the heating process takes place and the processor temperature increases, while in an opposite case $\tilde{q}_v < \tilde{q}_{vap}/\tilde{\lambda}$ the processor is cooled and its temperature decreases.

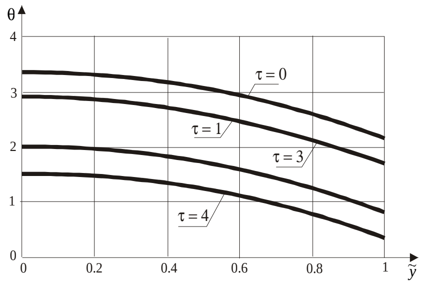

Fig. 5. Time-dependent temp. distribution for constants $\tilde{q}_v = 2.29$ and $\theta_{C0} = 2.2$

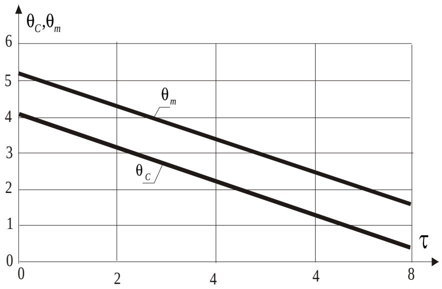

Fig. 6. Time-dependent contact temp. and maximum temp. for constant $\tilde{q}_v = 2.29$

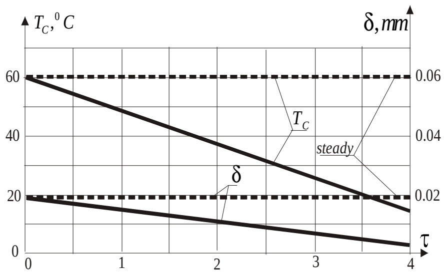

Fig. 7. Time-dependent contact temp. and liquid layer thickness for constants $\tilde{q}_v = 2.29$ and $\theta_{C0} = 2.2$

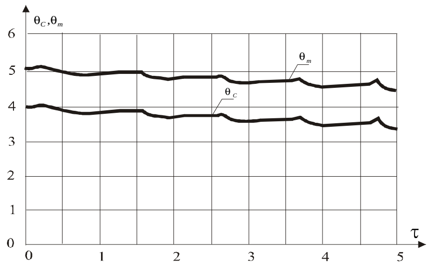

Fig. 8. Contact temp. and maximum temp. dependent on time for variable $\tilde{q}_v = 2.29 + 2.2\sin(6\tau)$

The results, based on numerical calculations, are presented in Figures 5, 6, 7, and 8. By comparing charts presented in Figures 3 and 5 we see the differences in temperature distribution in relation to time. This results from differences in problem presentation. In the first case (Fig. 3) the contact temperature is by definition constant, while the stationary distribution within the element is reached in an infinite amount of time. In practice, time is finite and, as seen in Fig. 3 is approximately $\tau \approx 20$. In the second case (Fig. 5) contact temperature decreases with time (Fig. 6, 7) and stationary state depends on the relation between heat generated in the unit and cooling heat. If the heat streams are equal, then temperature distribution does not change with time (see dashed line in Fig. 7). The second case resulting from the cooling model presented in this work is more real. Figure 8 presents the course of cooling with varying and simulated internal heat sources.

5. CONCLUSIONS

This work presents a solution for a simplified. A new, quasi-stationary solution for internal heat sources cooling model based upon a classic stationary heat exchange that takes place in a circuit featuring an component with internal heat sources and an evaporating liquid layer. The essence of the proposed model is based upon creating a set of time-dependent stationary states of heat distribution variables. The theoretical model thus created allows for a realistic theoretical description of the phenomenon, i.e. to make contact temperature between the studied component and the evaporating liquid layer and internal heat sources time-dependent. The work's result is the establishment of contact temperature and a liquid layer thickness as a function of time.

Existing classic theoretical solutions known in literature with differently set boundary conditions assume time-independent heat sources and a constant temperature at the edge of the area investigated. In terms of cooling, the contact temperature is drastically reduced in the initial moment and treated later as constant value.

Knowledge of the dependency of the contact temperature and the maximum temperature and the thickness evaporating of the liquid layer on time of the component, is very important in heat output. These parameters in this paper are found.

The application of a vapor compression refrigeration cycle for execution of evaporation process on the external surface of cooled unit seems to be well justified. The power demand of a compression refrigeration cycle's drive is negligible. An important part of the refrigeration cycle cooling circuit is the thermodynamic agent's evaporation temperature, whose value must not be too low in order to avoid vapour dropping out from the surrounding air onto the surfaces of the computer elements.

Provided that the compression refrigeration cycle works in invariable conditions, (i.e. both the evaporation temperature and the cooled element's received heat stream are constant), the temperature gradient on the edge of cooled component, the contact temperature and the thickness of evaporating liquid layer will be constant.

2. NOMENCLATURE

a	heat diffusivity $[\ m^2\ s^{-1}\ ]$
$\tilde{a}$	dimensionless heat diffusivity $= a_L a_C^{-1}$ $[\ -\ ]$
c	specific heat at constant pressure $[\ J\ kg^{-1}\ K^{-1}\ ]$
F	frontal surface of processor $[\ m^2]$
H	height of the component $[m]$

h_{fg} heat of evaporation [$J\,kg^{-1}$]
K evaporation parameter $= h_{fg}\, c_L^{-1}(T_0 - T_S)^{-1}$ [—]
$\dot{m}$ stream of mass [$kg\, s^{-1}$]
P power [W]
$\dot{q}$ heat flux per area [$W\, m^{-2}$]
$\dot{q}_v$ source of heat (volumetric heat load) [$W\, m^{-3}$]
$\tilde{q}_v$ dimensionless source of heat $= \dot{q}_v\, H^2 \lambda_C^{-1} (T_0 - T_S)^{-1}$ [—]
$\tilde{q}_{vap}$ dimensionless parameter of evaporation $= \dot{m}_s h_{fg} H \lambda_L^{-1} (T_0 - T_S)^{-1} F^{-1}$ [—]
t time [s]
T temperature [^{0} C]
T_0 reference temperature (may be the initial temperature) [^{0} C]
T_S evaporating temperature [^{0} C]
T_L liquid temperature [^{0} C]
T_C contact temperature [^{0} C]
$\overline{T}$ mean temperature [^{0} C]
T_{CON} condensation temperature [^{0} C]
T_m maximum temperature [^{0} C]
y coordinate [m]
$\tilde{y}$ dimensionless coordinate $= y H^{-1}$ [—]

Greek letters

δ height of the liquid laver [m]
$\tilde{\delta}$ dimensionless height of the liquid layer $= \delta\, H^{-1}$ [—]
θ dimensionless temperature, $= (T - T_S)(T_0 - T_S)^{-1}$ [—]
θ_C dimensionless contact temperature, $= (T_C - T_S)(T_0 - T_S)^{-1}$ [—]
θ_L dimensionless temperature, $= (T_L - T_S)(T_0 - T_S)^{-1}$ [—]
θ_m dimensionless maximum temperature, $= (T_m - T_S)(T_0 - T_S)^{-1}$ [—]
λ heat conductivity [$W\, m^{-1} K^{-1}$]
$\tilde{\lambda}$ dimensionless conductivity, $= \lambda_C \lambda_L^{-1}$ [—]
ρ density [$kg\, m^{-3}$]
τ dimensionless time, $= K a_C\, t\, H^{-2}$ [—]

Subscripts

C component or contact layer
L liquid
S steam (vapor)

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ANALITYCZNE ROZWIĄZANIE ROZKŁADU TEMPERATURY WEWNĄTRZ MIKROPROCESORA CHŁODZONEGO PARUJĄCYM CZYNNIKIEM CHŁODNICZYM

S t r e s z c z e n i e

W pracy przedstawiono analitycznie rozwiązanie problemu chłodzenia elementu, wewnątrz którego wydzielane jest ciepło spowodowane jego pracą. Zaproponowano uproszczony, niestacjonarny model teoretyczny, opisujący zjawisko, dzięki czemu określono rozkład temperatury wewnątrz elementu, temperaturę kontaktu między elementem a warstwą cieczy czynnika chłodniczego oraz grubość warstwy parowania w funkcji czasu. Problem chłodzenia rozwiązano teoretycznie wykorzystując równania Fouriera i Poissone'a przy spełnieniu odpowiednich warunków brzegowych. Rozwiązano dwa przypadki chłodzenia: stacjonarny i niestacjonarny. Otrzymane rozwiązania analityczne wydają się być przydatne w projektowaniu podobnych systemów chłodzenia. Wykonano również obliczenia dla układów chłodzenia wyposażonych w pompę ciepła, kompresor, jako skuteczny sposób chłodzenia.

Słowa kluczowe: wewnętrzne źródła ciepła, grubość warstwy cieczy chłodzącej, odparowywanie czynnika chłodniczego

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