The analysis of optimal heat discard parameters in the combined cooling system of radio-electronic equipment

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Abstract. The paper presents the mathematical model of the combined cooling system and calculation methods of optimal heat discard parameters of radio-electronic equipment as compared to standard cooling methods in high-power radiolocation stations. The optimization problem is solved to determine the control actions: the flow rate of cooling air in the air cooling system and the coolant in the liquid cooling system. The total cooling costs are considered as the main function, they consist of the costs of the cooling air and coolant circulation and the costs of cooling the heat carriers themselves. Restrictions are the conditions of air and liquid cooling thermal balance and maintaining the set temperature in the space where the radio-electronic equipment operates. The optimization problem in a limited area is solved by the bitwise search method. Air and circulating fluid cooling is carried out in the devices with free cooling which are located outside the radio-electronic system. The parameters of thermal modes of operation of these devices are given in the form of tables which are processed by means of regression analysis methods to solve the optimization problem. The air cooling system uses heat pipes with radiators which ensure efficient removal and dissipation of heat. Heat removal from radiators occurs in the turbulent mode, as the analysis shows. In conclusion a comparative analysis of different types of cooling systems energy consumption is carried out, depending on the required heat power and outside air temperature.

1. Introduction
Forced air cooling is the most common method of electronic equipment cooling; this is due to low energy consumption, simplicity of design and high reliability in operation. However, the constant power increase of modern electronic equipment leads to the need for the increase of the heat flux rate required for cooling, which can be achieved by using liquid cooling systems or combined systems [1–7]. Despite the fact that the described above systems do exist, the problem of effective cooling is not completely solved [8–14]; this is explained by low organization efficiency of the cooling medium interactions with the elements of the device, the inconsistency of interaction of the subsystems in different operating modes and lack of reliable methods of calculation and simulation.

2. Research Method
The authors of this paper introduce the new design of the combined air-liquid cooling system (Figure 1), which provides an opportunity for effective cooling at different temperatures. The effectiveness of
the system is confirmed by a series of experimental theoretical studies carried out on a manufactured pilot plant. The effect of temperature and heating power on the power consumption has been analyzed during the experiments. The following source data have been used in the calculation of the cooling system, where $Q$ [(kW)] is total heat radiation power; $C_2$ [(J/(kg°K))] is liquid specific heat; $\rho_2$ [kg/cm$^3$] is liquid density; $\Delta t_2$[°C] is temperature difference of the liquid on the refrigerating unit condenser (RU); $C_1$ [J / (kg·K)] is air specific heat; $\rho_1$ [kg/cm$^3$] is air density; $t_{out}$ [°C] is ambient temperature; $t_{in}$ [°C] is air temperature in the area, where the electronic equipment is installed. Moreover, the radiators operational characteristics are presented in the paper.

Figure 1. The general scheme of the combined cooling system of the radio-electronic unit, where 1 is frame; 2 are channels for the coolant; 3 and 4 are unions; 5 is air section shell; 6 are air unit rods with fins; 7 is fan.

3. Results and Discussion
The total expenditures on cooling consist of the power supplied for the coolant and air circulation and the power consumed absorbed by the liquid cooling system and the air-cooling system. Consequently, this can be expressed in the following way:

$$N_5 = N_{cl} + N_{ca} + N_1 + N_2.$$ (1)

Power expenditure for air and coolant circulation is represented by the Equation [2]:

$$N_{ca} = \frac{10^{-3}P_aV_1}{\eta_1} [kW], \quad N_{cl} = \frac{10^{-3}P_2V_2}{\eta_2} [kW],$$ (2)

where $P_1$ is the hydrodynamic thrust of the fan which varies in the range of 60÷100 Pa; $\eta_1$ is the mechanical performance index of the fan which is measured in the range of 0.5÷0.8; $P_2$ is the hydrodynamic thrust of the pump, which varies in the range of 600÷800 kPa; $\eta_2$ is the mechanical performance index of the pump which is measured in the range of 0.7÷0.9.

The power consumption of external cooling systems depends on ambient temperature and cooling capacity, while the compressors have variable performance. Experimental data concerning power
consumption of air and liquid cooling systems (depending on ambient temperature and cooling capacity) are presented in Tables 1 and 2. The power values at temperatures from -20 to 5 °C are 0.050 [kW] and are not given in the tables.

Imagine the correspondence $N_i = N_i(Q, t)$ in the form of surfaces of the 3rd order and planes $N=0.050$, describing the negative temperature conditions:

\[
N_i=0.052 - 0.01 Q + (0.02 Q^3 - 0.013 Q^2 + 0.0017 Q - 0.00014)t + (0.0002 Q -
- 0.000035)t^2 + (0.000052 Q - 0.0000031)t^3; \tag{3}
\]

\[
N_2=0.045 + 0.02 Q - 0.025 Q^3 + (- 0.0011 Q - 0.00067)t + (0.0002 Q -
- 0.000073)t^2 + (0.0000133 Q - 0.00000532)t^3
\]

**Table 1.** Power consumption at heat power 0.5 [kW].

| $t_{out}$ [kW] | $N_1$ Q=0.2 [kW] | Q=0.3 [kW] | Q=0.4 [kW] | Q=0.5 [kW] |
|----------------|-----------------|-------------|-------------|-------------|
| 10             | 0.052           | 0.054       | 0.054       | 0.058       |
| 15             | 0.054           | 0.056       | 0.064       | 0.068       |
| 20             | 0.056           | 0.068       | 0.082       | 0.094       |
| 25             | 0.058           | 0.084       | 0.110       | 0.150       |
| 30             | 0.062           | 0.100       | 0.140       | 0.170       |

**Table 2.** Power consumption at heat power 0.7 [kW].

| $t_{out}$ [kW] | $N_2$ Q=0.4 [kW] | Q=0.5 [kW] | Q=0.6 [kW] | Q=0.7 [kW] |
|----------------|-----------------|-------------|-------------|-------------|
| 10             | 0.054           | 0.058       | 0.060       | 0.062       |
| 15             | 0.058           | 0.060       | 0.070       | 0.074       |
| 20             | 0.060           | 0.070       | 0.088       | 0.106       |
| 25             | 0.064           | 0.086       | 0.120       | 0.150       |
| 30             | 0.070           | 0.110       | 0.170       | 0.220       |

The graphic presentation of the data $N_i = N_i(Q, t)$ is displayed in Figures 2 and 3.
We will search for the minimum power consumption defined by Equation (1) in a limited area. One of the equations included in the constraint system is the heat balance equation which is presented in the following way:

\[ Q = Q_1 + Q_2 , \]  
\[ Q_2 = C_2 \cdot \rho_2 \cdot V_2 \cdot \Delta t_2. \]

The magnitude \( Q_1 \) is calculated by a well-known method [3]. The air velocity in the air-cooling system is determined by the Equation

\[ w = V_1 / S_k, \]

where \( S_k \) is the radiator(s) heat pipes free area. The following characteristics are used in the calculations, where \( \mu \) is length \([\text{m}]\); \( S_{cros}, S_{long}, S_{cros.r} \) are cross-sectional area of the air-duct and the longitudinal and cross-sectional area of the radiator \([\text{m}^2]\); \( Z \) is number of radiator fins.

The area of effective cross section is defined by the Equation:

\[ S_k = S_{long} - S_{cros.r}. \]

Having calculated \( Q_1 \) we then count the air temperature differential on the condenser \( \Delta t_1 = Q_1 (C_1 \rho_1 V_1)^{-1} \). The average air temperature in the area is defined by the Equation \( t_{a} = t_{a, in} + \Delta t_a / 2 \), where \( t_{a, in} \) is air temperature on entering the area. Hence, the condition limiting the problem space is presented as follows:

\[ t_{a} = t_{out}. \]

The optimization problem (1) with the constraints (4) and (6) solved by the gradient descent method [4] in the Maxima system at the given power of heat emission: \( Q = 0.7 \) kW. Therein it was taken into account that the magnitude \( Q_1 \) is actually a function of the air consumption \( V_1 \). The calculations included the values of the model parameters: \( C_2 = 4187; \rho_t = 1000 \) kg/m\(^3\); \( \Delta t_1 = 5^\circ\text{C}; C_1 = 1005 \) J/[kg\(^\circ\text{C}\)]; \( \rho_1 = 1.26 \) kg/m\(^3\); \( t_{a,in} = 20^\circ\text{C}; t_{a,cond} = 22^\circ\text{C}; P_2 = 60 \) kPa; \( \eta_{out} = 0.7; P_1 = 80 \) Pa; \( \eta_a = 0.7; Z = 10; \lambda_m = 2.44 \times 10^2 \) [W/(m\(^2\)^\circ\text{C})]; \( \lambda_1 = 0.026 \) [W/(m\(^2\)^\circ\text{C})]; \( S_{cros} = 0.0285 \) m\(^2\); \( L = 0.397 \) m; \( S_{cros,r} = 0.0093 \) m\(^2\); \( S_{long} = 0.12 \) m\(^2\); \( d = 0.015 \) m.

Figure 4 displays the dependencies \( Q_1 = Q_1(V_1) \) at three values of the parameter \( S_{cros} \). The upper curve corresponds to \( S_{cros} = 0.0195 \) m\(^2\), the middle \( S_{cros} = 0.0295 \) m\(^2\) and the lower \( S_{cros} = 0.0395 \) m\(^2\). Figure
5 shows the lines of equal quantities of the discarded heat at $V_1, V_2$ different for three values of $Q$. The upper graph corresponds to $Q=900$ W, the intermediate $Q=800$ W, the lower $Q=700$ W.

![Figure 4](image1.png)  
**Figure 4.** Rated dependencies $Q_1 = Q_1(V_1)$.

![Figure 5](image2.png)  
**Figure 5.** Lines of equal amounts of discarded heat.

Figures 6 and 7 reflect the dependences $N_s = N_s(V_1, V_2)$ at $t=30^\circ$C (Figure 6), $t=10^\circ$C (Figure 7). The range of values $N_s=0$ which lie in the horizontal plane correspond to those values $V_1, V_2$ which are incompatible with the heat balance condition.

![Figure 6](image3.png)  
**Figure 6.** Rated dependencies $N_s = N_s(V_1, V_2)$ at $t=30^\circ$C.

![Figure 7](image4.png)  
**Figure 7.** Rated dependencies $N_s = N_s(V_1, V_2)$ at $t=10^\circ$C.

In consequence of solving the optimization problem (1) in a limited area (6) the values of the control parameters $V_1, V_2$, were obtained which ensure the minimum power spent on cooling the radio-electronic unit $N_s$:

\[
\begin{align*}
  t &= -10^\circ$C, V_1 = 150 m^3/h, V_2=7.66 \times 10^3 m^3/h, N_s=70.62 W \\
  t &= 10^\circ$C, V_1 = 250 m^3/h, V_2=7.63 \times 10^2 m^3/h, N_s=63.53 W \\
  t &= 30^\circ$C, V_1 = 250 m^3/h, V_2= 1.11 m^3/h, N_s=214.12 W
\end{align*}
\]

4. Conclusion
A mathematical model of optimal heat discard in the air-liquid cooling system of radio-electronic equipment has been developed. The model makes it possible to define the parameters of air and liquid cooling systems for radio-electronic equipment with heat emission power 0.8 kW.
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