Water-evaporative cooling of sealed volumes

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Abstract: The problem of cooling sealed volumes, in particular electronic equipment, with the help of counter-flow evaporative coolers of an indirect principle of operation is discussed in this paper. The equation of the heat balance of a limited volume is given according to its cooling by a closed cooling air flow. The paper presents a mathematical model of heat and mass transfer processes in counter-flow water-evaporative coolers of the indirect principle of operation, which includes parabolic and elliptic differential equations with distributed parameters, and a method for the numerical implementation of this model is also proposed. The results of the study allow concluding that it is possible to significantly reduce the temperature in sealed volumes with the help of inexpensive eco-friendly water-evaporative coolers of the indirect principle of operation.

1. Introduction.

The task of cooling electronic equipment has long been relevant in its design process. Oscillations, and especially a strong temperature increase in analog circuits, can change the thresholds of comparators, the gain of transistors, the nominal values of resistors and capacitors. In digital technology, temperature fluctuations are also dangerous. They can lead to equipment malfunction, thermal breakdown of transistors, decrease in measurement accuracy, etc. A temperature rise in power electronics is the most dangerous, as it can lead to the destruction of the power source and the malfunction of the consuming units.

Cooling systems of sealed volumes can be divided into two main groups: passive and active methods. The first group includes heat removal using convection, heat conduction and radiation. The second one includes forced heat removal through ventilation, thermoelectric coolers and washing liquids [1].

Passive methods and thermoelectric coolers, the operation of which is based on the Peltier principle, are ineffective with sufficiently large heat releases in the cooled volumes. Ventilation should also be treated with some caution, since air contains dust and moisture, which can be deposited on the elements of the cooled volume. Liquid cooling systems of semiconductor devices are quite effective, but inconvenient in case of unsteady conditions, such cooling objects in portable tents, on the road, in the field, etc.

2. Materials and methods
The choice of cooling location of sealed volumes. In recent years, a fairly large number of works by foreign researchers [2-6] have been devoted to issues of indirect water-evaporative cooling. We propose the use of an indirect counter-flow evaporative cooler to lower the temperature in various limited volumes, including electronic equipment blocks. The air flow which is not contacting with the wet surfaces of the cooler plates, through the use of a duct fan, is driven through the cooled object and enters the cooler. We will call this flow “dry” or “hot”. The air supplied from the environment is cooled by evaporating water from the wet surfaces of the plates and cools the dry air stream through heat transfer. We will call this stream “wet” or “cold”. It can be thrown into the external environment, and can be used to cool the room in which the volume to be cooled is located.

![Diagram of a cooler containing a closed loop for cooling a sealed volume.](image)

**Figure 1.** Scheme of cooling of pressured volume

Fig. 1 shows a diagram of a cooler containing a closed loop for cooling a sealed volume. Note that both direct-flow and more efficient counter-flow indirect cooling schemes can be used. The channels of the evaporative block during cooling of the indirect principle are divided into “dry” and “wet” (Fig. 2).

The temperature of the air passing through the “wet” channels, in which the evaporation of water occurs, is denoted by $t$. The temperature of the air passing through the “dry” channels and not changing its moisture content is denoted by $T$. The surface of the plates forming these channels and having a temperature $T_p$ is waterproof (indicated by a dark line).

### 3. Material of the evaporation plates.

The surface material of the plates forming the “wet” channels should have sufficient capillary properties and free volumes have open pores and water resistance. This problem was solved by a group of specialists led by V.K. Duboviy, who created “Composite material for special equipment”, which possess precisely this structure [7-8]. Experimental studies have shown that this material is well suited for the formation of “wet” channels of evaporative nozzles.

The problems of modeling physical processes in indirect water evaporative coolers are discussed in [9-10]. The problems of modeling physical processes in indirect water-evaporative coolers are described in scientific papers [9-10].
4. Discussion

The processes of heat transfer in the “dry” and “wet” channels in the case of a counter-flow are described by parabolic equations:

\[- \rho \cdot V_T(x, y) \cdot C \cdot \frac{\partial T}{\partial x} = \frac{\partial}{\partial y} \left( \lambda(T) \frac{\partial T}{\partial y} \right), \quad x \in (0, L), \ y \in (Hp, Hp + H), \]

(1)

\[\rho \cdot V_t(x, y) \cdot C \cdot \frac{\partial t}{\partial x} = \frac{\partial}{\partial y} \left( \lambda(t) \frac{\partial t}{\partial y} \right), \quad x \in (0, L), \ y \in (-h, 0); \]

(2)

mass transfer in the “wet” channel is also described by the parabolic equation:

\[V_t(x, y) \frac{\partial W}{\partial x} = \frac{\partial}{\partial y} \left( D(t) \frac{\partial W}{\partial y} \right), \quad x \in (0, L), \ y \in (-h, 0). \]

(3)

The law of temperature distribution in the plate is given by the Laplace equation:

\[\frac{\partial^2 T_p}{\partial x^2} + \frac{\partial^2 T_p}{\partial y^2} = 0, \quad x \in (0, L), \ y \in (0, Hp). \]

(4)

The inlet conditions to the cooler in counter-flow are:

\[t_{|x=0} = t_{in}, \varphi_{x=0} = \varphi_{in}, \quad y \in (-h, 0), \]

(5)
\[ T\big|_{x=L} = T_{in}, \quad y \in (Hp, Hp+H). \] 

(6)

By symmetry, on the axes of the channels:

\[ \frac{\partial T}{\partial y}\big|_{y=Hp+H} = 0, \quad x \in (0, L), \quad \frac{\partial t}{\partial y}\big|_{y=-h} = 0, \quad x \in (0, L), \quad \frac{\partial W}{\partial y}\big|_{y=-h} = 0, \quad x \in (0, L). \] 

(7)

At the ends of the plates we set the impermeability conditions:

\[ \frac{\partial Tp}{\partial x}\big|_{x=0} = 0, \quad y \in (0, Hp), \quad \frac{\partial Tp}{\partial x}\big|_{x=L} = 0, \quad y \in (0, Hp) \] 

(8)

At the boundary between the channels and the plate, equal temperatures and heat transfers are set:

\[ T\big|_{y=Hp} = Tp\big|_{y=Hp}, \quad x \in (0, L), \quad t\big|_{y=0} = Tp\big|_{y=0}, \quad x \in (0, L), \]  

\[ \lambda(T) \frac{\partial T}{\partial y} = \lambda_{pl}(Tp) \frac{\partial Tp}{\partial y}, \quad y = Hp, \quad x \in (0, L). \] 

(9)

On the evaporation surface:

\[ \varepsilon R(t) D \frac{\partial W}{\partial y} = \lambda_{pl}(Tp) \frac{\partial Tp}{\partial y} - \lambda(t) \frac{\partial t}{\partial y}, \quad y = 0, \quad x \in (0, L). \] 

(10)

The diffusion coefficient, \( m^2/s \), is taken equal to:

\[ D = 10^{-5} \cdot 2.16 \cdot (1 + t/273)^{1.8}. \] 

(11)

And finally, the vapor density on the saturation line, \( \text{kg/m}^3 \), is determined by the formula:

\[ w_a(t) = \left(0.0004212 t^3 + 0.00183 t^2 + 0.4195 t + 4.727\right) \cdot 10^{-3}, \] 

(12)

obtained by approximating tabular data. Here \( W \) is the vapor density, \( \text{kg/m}^3 \), \( \lambda_a \), \( \rho \), \( C \) are the plate thermal conductivity, \( \text{W/m} \cdot ^\circ\text{C} \), air density, \( \text{kg/m}^3 \) and specific heat, \( \text{J/kg} \cdot ^\circ\text{C} \), \( R(t) = (2500.6 - 2.372r) \cdot 10^3 \) is the specific heat of vaporization, \( \text{J/kg} \), \( \varepsilon \) – experimentally determined multiplying factor considering the additional energy additive during evaporation from porous surfaces.

5. Results

Since this mathematical model includes equations of elliptic and parabolic types, the temperature distribution must be sought in an integral block, dividing the cross section of the plate, as well as half the cross section of the “wet” and “dry” channels, with a rectangular grid on which groups of finite-difference analogues of the equation model proposed above are compiled.
The resulting system of linear algebraic equations is solved with the values of the saturated vapor density and diffusion coefficient calculated at the outdoor temperature. After that, on the “wet” surface of the plate, the value of the density of saturated steam is adjusted depending on the temperature values obtained there. In addition, the values of the diffusion coefficient at the grid nodes are also calculated from the temperature values determined there, and the system is solved again. The iterative process ends when the temperatures in the previous and present iterations at the outlet of the cooler differ by less than 0.1 °C.

As an example, in fig. 4, the solution of this system is visualized with the following parameters: temperature at the inlet of “cold” air –25 °C, its relative humidity –40%, temperature at the inlet of “hot” air –40 °C, flow velocity of “cold” air –2.7 m/s, “hot” air flow velocity –4 m/s, plate length –0.4 m, temperature at the exit of “cold” air –25.80 °C, temperature at the exit of “hot” air –190 °C. Note that the temperature of the “hot” air at the inlet to the cooler \( T_{\text{outlet}} \) is the temperature at the outlet of the cooled volume \( t_{\text{volume}} \), which, in turn, depends on the heat divisions in it. The equation of heat balance in volume has the form:

\[
C\rho G T_{\text{out}} + Q + \sum k_i F_i (t_n - t_{\text{rev}}) - C\rho G t_{\text{rev}} = 0. \tag{13}
\]

Here \( \sum k_i F_i \) - heat transfer coefficients and the area of the \( i \)-part of the walls of the cooled volume, \( G \) is the flow rate of “hot” air. \( T_{\text{outlet}} \) is the temperature at the outlet of the cooler, which is the temperature at the entrance to the volume. \( t \) is the environment temperature equal to the temperature at the entrance to the “wet” channels. \( Q \) is the heat input in the volume.

From equation (1) the temperature in the volume is taken:

\[
t_{\text{out}} = \frac{C\rho G T_{\text{out}} + Q + \sum k_i F_i t_n}{\sum k_i F_i + C\rho G}. \tag{14}
\]

As a result, the temperature of the “hot” air at the inlet to the cooler depends on the temperature of the “hot” air at the outlet of the cooler. This circumstance leads to the necessity of organizing an external iterative cycle. A certain initial temperature of “hot” air at the inlet to the cooler is \( T_{\text{inlet}} = t_{\text{volume}} \), and the above model of heat and mass transfer in the cooler is implemented, from which the temperature at the outlet of the cooler is determined, which is the temperature at the inlet to the volume. The concept of “residual” is introduced as the absolute value of the difference between the obtained and the initial values of \( t_{\text{volume}} \).
Then \( t_{\text{inlet}} \) is adjusted, and the heat and mass transfer model is implemented again. Calculations are stopped when the value of the residual is less than 0.1 °C. Figure 5 shows the temperature distribution in the cooler of the same geometric dimensions after the last iteration of the proposed algorithm with the following parameters: outdoor temperature \(-300\) °C; relative humidity \(-40\%\); dry air flow rate \(400\) m\(^3\)/h, \%; wet air flow rate \(-270\) m\(^3\)/h; heat input in the facility \(-2000\) W; temperature in the facility \(-37.50\) °C. We note that with conventional ventilation with the same air flow rate, the temperature in the volume would reach 45.30 °C.

### 6. Conclusion

The results of the study allow concluding that it is possible to reduce the temperature in sealed volumes significantly, particularly, in electronic equipment with the help of inexpensive environmentally friendly water-evaporative coolers of an indirect principle of operation. In addition, the proposed mathematical model describing the processes of heat and mass transfer in these coolers and the method of its numerical implementation can be used for cooling units, which apply other cooling principles, as well as for direct-flow and counter-flow plate heat exchangers.

### References

[1] Bocock G 2010 Some aspects of forced air cooling of power supplies *Power Electronics* **5** 80-81

[2] Fakhrabadi F Kowsary F 2016 Optimal design of a regenerative heat and mass exchanger for indirect evaporative cooling *Applied Thermal Engineering* **102** 1384-1394.

[3] Moshari S Heidarinejad G 2015 Numerical study of regenerative evaporative coolers for subwet bulb cooling with cross-and counter-flow configuration *Applied Thermal Engineering* **89** 669-683.

[4] Hasan A 2012 Going below the wet-bulb temperature by indirect evaporative cooling: analysis using a modified \( \varepsilon \)-NTU method *Applied energy* **89** 1 237-245.

[5] Zhao X Li J M Riffat S B 2008 Numerical study of a novel counter-flow heat and mass exchanger for dew point evaporative cooling *Applied Thermal Engineering* **28** 14 1942-1951.

[6] Chengqin R Hongxing Y 2006 An analytical model for the heat and mass transfer processes in indirect evaporative cooling with parallel/counter flow configurations *International Journal of heat and mass transfer* **49** 3 617-627.
[7] Duboviy V K 2006 *Paper-like composite materials based on mineral fibers* Synopsis of Doctor of Technical Science thesis: 05.21.03. St. Petersburg

[8] WO2017086833A1 RF 2017 Duboviy V K *Paper-like composite material based on mineral fibers using aluminum salts and a polyvinyl acetate emulsion (PVAE) as a binder*

[9] Shatskiy V P 2012 Modeling of physical processes in plate water-evaporating air conditioners of the indirect principle of operation *Scientific Herald of Voronezh State University of Architecture and Civil Engineering. Construction and Architecture. 2(26) 29-34.*

[10] Shatskiy V P 2013 Modeling the operation of plate water-vapor coolers of the indirect principle of operation *Forestry Engineering Journal, Voronezh, VSUFT 4(12) 160-166.*