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CONTROL OF ENERGY CONSUMPTION ON VENTILATION OF COOLING DEVICES OF THE SHAFT TYPE

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Consideration is given to a system of forced air cooling made in the form of a shaft, one of the walls of which is a finned cooler, through which air is blown by a fan unit. The functional dependence of the number of revolutions of the fan motor on the cooling air temperature and other parameters of the cooling system is obtained in order to ensure the thermal regimes of heat-releasing elements.

Cooling systems based on a shaft with ventilation through it of air or another medium are widely used in engineering; therefore, justified specification of parameters for the system can strongly affect its efficiency.

Let ventilation be performed by a unit of centrifugal fans mounted on the electric motor shaft. Energy consumption on the motor largely determined consumption on the cooling system. In this connection we will consider the following problem: how the number of revolutions of the motor depends on the cooling air temperature and other parameters of the cooling system in order that the heating of modules mounted on the cooler not be larger than the maximum permissible value.

Calculation of the cooling system consists of two parts: a ventilation calculation and a thermal one. The basis of the ventilation calculation is composed of the equation

$$\Delta P = P\,, \tag{1}$$

where P is the nominal pressure of the fan; ΔP is the hydraulic resistance of the shaft, $\mathrm{kgf/m^2}$.

In [1] the external aerodynamic characteristic of a centrifugal fan is given:

$$P^* = 0.5\,(1 - r^{*2}) + \tilde{k}\,(0.5 \pm Q^* \cos\beta) - r^{*2} Q^{*2}\,,$$

where $P^* = P/\rho u_2^2$ is the relative pressure; $Q^* = Q/\pi D_2 b_2 u_2$ is the relative consumption; $r^* = R_1/R_2$ is the relative radius; $u_2 = R_2\omega = 2\pi n_0 R_2$ is the translational velocity at the wheel diameter; n_0 is the number of revolutions per second; R_2 is the outer diameter of the wheel; b_2 is the width of a wheel blade.

The value of the coefficient of static pressure recovery $\tilde{k} < 1$ is determined by the design of the section that the air from the fan wheel enters. The sign in the second term is determined by how the wheel blades are bent relative to the direction of rotation of the fan wheel.

Hydraulic losses are determined using the formula

$$\Delta P = \xi \rho V^2/2\,, \tag{2}$$

where ξ is the coefficient of hydraulic resistance of the shaft; V is the velocity of the incoming air flow, m/sec.

We will conduct further consideration without regard for the second and the third terms in P^*; such an approximation is possible for $r^* \ll 1$, small $\tilde{k}$, and the flow rates of the air.

Let us assume that the number of fans (wheels) on the motor shaft is arbitrary and is equal to k_{fan}; then the velocity V is related to the air velocity V_b formed by the consumption of one fan by the relation

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$$V = k_{\text{fan}} V_b. \tag{3}$$

Equality (1) is represented as

$$0.5 \rho u_2^2 (1 - r^{*2}) = 0.5 \rho \xi V_b^2 k_{\text{fan}}^2. \tag{4}$$

In [2] the coefficient of resistance of the shaft is given by the expression

$$\xi = \left(\xi_{\text{n}} + \lambda \frac{L}{d_r} \right) \left(\frac{F_1}{F_{\text{fl}}} \right)^2 + \Delta\xi_t ,$$

where ξ_{n} is the coefficient of local resistance in a sudden narrowing and widening of the air flow; λ is the coefficient of friction; $\Delta\xi_t$ is the coefficient of thermal expansion:

$$\xi_{\text{n}} = 1.5 (1 - F_{\text{fl}}/F_1)^2 , \quad \Delta\xi_t = \left(1.7 + \lambda \frac{L}{d_r} \right) \left(\frac{F_1}{F_{\text{fl}}} \right)^2 \overline{T} ,$$

$$\overline{T} = (T_{\text{outl}} - T_{\text{inl}})/T_{\text{inl}} , \quad \lambda = 0.77 \, \text{Re}^{-1/3} , \quad 10^3 \le \text{Re} \le 2.5 \cdot 10^4 .$$

We will consider two limiting cases of ξ values: the first case when the coefficient of hydraulic resistance of friction prevails over the other two and the second case when the main contribution to ξ is made by ξ_{n}.

In the first case

$$\xi \simeq \lambda \frac{L}{d_r} \left(\frac{F_1}{F_{\text{fl}}} \right)^2 ,$$

which we will represent as

$$\xi \simeq V_b^{-1/3} C_0 ,$$

$$C_0 = 0.77 \frac{L}{d_r} \left(\frac{F_1}{F_{\text{fl}}} \right)^2 \sqrt[3]{\left(\frac{\nu}{k_{\text{fan}} d_r} \right)} . \tag{5}$$

Using (5) and representing u_2 in terms of n_0 from (4), we obtain

$$V_b = \overline{n}_0^{1.2} \kappa_1 , \quad \kappa_1 = [4\pi^2 (R_2^2 - R_1^2)/C_0]^{3/5} . \tag{6}$$

In the second limiting case we similarly have

$$V_b = \overline{n}_0 \kappa , \quad \kappa = 2\pi \, [\xi^{-1} (R_2^2 - R_1^2])^{1/2} , \tag{7}$$

where $\overline{n}_0 = n_0/k_{\text{fan}}$. We write relations (6) and (7) as

$$V_b = \overline{n}_0^t \kappa^* . \tag{8}$$

The following component of the considered model of the cooling process in the shaft is an expression for the thermal resistance of the cooler, which has the form of a rectangular finned wall whose thickness δ is much smaller than the remaining dimensions and whose area of contact with one source of heating is located through the center of the cooler [3]:

$$R_T = (\hat{b} + \hat{a}/n^{\gamma})\, D\,, \quad D = (4\lambda\delta m^2 k)^{-1}\,, \quad n = H^2 k_r \alpha_n/\lambda\delta\,,$$
$$k_r = [\tilde{n}\, S_{\text{fin}}\, \eta + (F - \tilde{n}\, S_b)]/F\,, \tag{9}$$

where $\tilde{n}$ is the number of fins on the wall surface; S_{fin} is the area of a fin; S_δ is the surface between the fins; k_r is the coefficient of finning; $m = a/A = b/B$; $k = A/B$; η is the fin efficiency, which depends on α_n. When use is made of domestic fans, having a small capacity, and of aluminum shapes for production of coolers with a fin height ≤ 0.06 m, the fin efficiency is no less than 0.9. Therefore we assume hereafter that $\eta \simeq 0.9$.

The coefficients $\hat{a}$, $\hat{b}$, and γ are functions of the parameters k and m, calculated for the following values of the coefficients: $k = 0.217$, 0.53, 0.8, 1; $m = 2$, 0.4, 0.6. Expression (9) is obtained by numerical solution of a two-dimensional heat-conduction equation for a plate with the condition that the heat flux from its ends is absent and for the case of geometric similarity of the rectangular base of the cooler and the heating source.

We will generalize formula (9). Let some number n_m of identical elements be given, which have equal heat losses Q. Let a representation as the product $n_m = n_x n_y$ be possible. As a cooler common for all elements it is proposed to use a rectangular finned wall with the dimensions $A_1 \times B_1$ such that $A_1 = An_x$, $B_1 = Bn_y$. Let us conventionally divide the plate into n_m of equal areas having the dimensions A, B, which are geometrically similar to the heat removal surface of the elements. We will arrange the elements through the center of each area with the observance of geometric similarity; then by virtue of equal heat losses in each modulue and with the chosen arrangement of the elements it is to be expected that there will be no mutual heating of the elements, i.e., the heat fluxes on the boundaries of each area are negligibly small, and hence, $R_{T,m}$ of each area can be determined by (9). In this case the equality

$$R_T = R_{T,m}/n_m\,.$$

To determine the heat-transfer coefficient of finned heat exchangers in a laminar flow, the following formula is recommended in [4]:

$$\alpha_n = \hat{C} V_0^{0.4}\,, \quad \hat{C} = 1.35\, \text{Pr}_f^{0.33} \left(\frac{\text{Pr}_f}{\text{Pr}_w}\right)^{0.25} \frac{\lambda_f}{(L\nu)^{0.4}\, \Delta^{0.2}}\,,$$

whence

$$\alpha_n = C V_b^{0.4}\,, \quad C = \hat{C} \left(k_{\text{fan}} \frac{F_1}{F_{\text{fl}}}\right)^{0.4}. \tag{10}$$

Eliminating α_n in (9) and then using (8), it is not difficult to obtain

$$R_T = D\,(\hat{b} + \tilde{F}/\bar{n}_0^{0.4\gamma t})\,, \quad \tilde{F} = \hat{a}\,(\tilde{D} C \kappa^{*0.4})^{-\gamma}\,, \quad \tilde{D} = H^2 k_r/\lambda\delta\,. \tag{11}$$

Solving (11) with respect to n_0, it can be written that

$$\bar{n}_0 = \left[\frac{\varepsilon}{(\bar{\varepsilon} - T_{\text{air}})}\right]^{\frac{1}{0.4t\gamma}}, \quad \varepsilon = QD\tilde{F}\,, \quad \bar{\varepsilon} = T_{\max} - \hat{b}QD\,. \tag{12}$$

Since the coefficients γ in (9), given in [3], are of the order of unity, the expression $(0.4t\gamma)^{-1} \simeq 2$. Consequently, the dependence $n_0 = f(T_{\text{air}})$ has the form of a quadratic hyperbola with a singular point at $T_{\text{air}} = \bar{\varepsilon}$.

The obtained expression (12) enables us to solve a wide range of problems involving systems of cooling based on shaft ventilation.

We will consider an example. Let it be necessary to organize cooling for twelve 2M2-40 modules, having the following characteristics: the dimensions 0.066×0.035 m, $R_{T,h} = 0.75$ K/W, $T_{\max,\text{des}} = 125°$C. The cooler for

TABLE 1

T_{air}, °C	Q, W				
	10	20	30	33	40
0	60	318	861	1106	1878
10	84	384	1080	1386	2400
20	96	480	1368	1788	3120
40	162	1140	2514	3372	6420
60	300	1680	6018	8567	19500

TABLE 2

T_{air}, °C	Q, W				
	10	20	30	33	40
0	0.09	0.67	2.2	3	5.6
10	0.135	0.83	2.9	3.9	7.53
20	0.16	1.09	3.8	5.3	10.3
30	0.3	3.1	8	11.3	24.5
4060	0.62	4.9	22.7	34.7	93

all the modules is produced from an aluminum shape, the wall thickness is 0.006 m, the fin height is 0.025 m, the finning step is 0.01 m, the fin thickness at the base is 0.003 m. The cooler dimensions are 0.24 × 0.33 m. The modules are mounted as described above through the center of 0.058 × 0.11 areas. The cooling is organized by a unit of two fans mounted on top of the unit. The characteristics of the fan wheel are: $R_1 = 0.008$ m, $R_2 = 0.048$, $b_2 = 0.02$ m; it is necessary to calculate the required number of revolutions of the motor in relation to the temperature differences of the air supplied to the shaft when heat losses vary in each module.

We will estimate the coefficient of hydraulic resistance for a number of air velocities according to (6). The calculations yield: $F_{fl} = 4.5 \cdot 10^{-3}$ m^2, $F_l = 7.44 \cdot 10^{-3}$ m^2, $d_r = 1.057 \cdot 10^{-2}$ m, $\xi_n = 0.234$. At $V_0 \simeq 12$ m/sec, $\lambda L/d_r \simeq 1.257$.

The estimate of the thermal expansion coefficient with losses in one module of ~40 W is $\Delta\xi_t \simeq 0.2$. Thus, the main contribution to the hydraulic resistance is made by friction losses.

Further calculations yield: $C \approx 13.6$; $C_0 \approx 6.23$; $\kappa^* = 0.078$; $D = 1.164$; $\tilde{D} = 0.014$; since $\kappa = 0.53$, $m = 0.6$, from the tables in [3] $\hat{a} = 0.508$, $\hat{b} = 0.132$, $\gamma = 1.034$ then $\tilde{F} = 7.4$; $\varepsilon = 8.616$; since $T_{max,cool} = T_{max,des} - Q(R_{T,h} + R_{T,k})$ and assuming $R_{T,k} \simeq 0.05$ K/W, from (12) we obtain

$$n_0 = 2\left[\frac{8.616Q}{125 - 0.95Q - T_{air}}\right]^2.$$

The calculated values of n_0 (rev/min) for different values of Q and T_{air} are given in Table 1.

Using relation (8), we calculate the corresponding velocity (m/sec) in the shaft, which ensures the required operating regime of modules for the specified range of Q and T_{air} (see Table 2).

NOTATION

$R_T = (T_{max} - T_{air})/Q$, thermal resistance of the cooler, K/W; T_{max}, maximum temperature of the cooler, K; T_{air}, average temperature of the ambient air, K; T_{inl}, T_{outl}, air temperature at the inlet and outlet of the shaft, K; F_{fl}, flow cross section of the shaft, m^2; F_l, total area of the shaft (without the radiator), m^2; Π, air-washed perimeter of the shaft, m; $d_r = 4F_{fl}/\Pi$, hydraulic diameter of the shaft, m; L, vertical dimension of the shaft (the cooler), m; Δ, distances between fins, m; Re, Re_L, Re_Δ, Reynolds number with the determining dimensions d_r, L, Δ, respectively; λ, λ_f, coefficients of thermal conductivity for the cooler and the cooling medium, W/(m·K); ν, kinematic viscosity factor of the cooling medium, m^2/sec; α_n, average heat-transfer coefficient on the cooler surface, W/(m^2·K); A, B, cooler dimensions for one module, m; a, b, dimensions of the heat removal region of a module, m.

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