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COMPLEX RESEARCH RESULTS OF THE EVAPORATIVE CONDITIONER FOR DIESEL LOCOMOTIVE CAB

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Summary. A set of theoretical and experimental research of evaporative air conditioner unit has provided a design that can be used on railway rolling stock and that meets the requirements of regulations of the microclimate parameters of the locomotive cab.

Keywords: evaporative conditioner, diesel locomotive, air flow, the metal nozzle.

INTRODUCTION

When providing comfortable sanitary and hygienic indoor microclimate parameters [Pankova, 2000] it allows people to increase the productivity of work and reduce the probability of professional diseases. The results of statistical studies show that the diseases of the locomotive crews' workers depend to a large extent on the unfavourable climatic conditions in the cab of the locomotive, and are reflected in the cardiovascular, musculoskeletal, and nervous systems. Moreover, the absence of favorable climatic conditions in the work of the locomotive driver and the driver's assistant accumulates fatigue, lethargy, and similar phenomena that affect the security of movement in the similar way.

RESEARCH ANALYSIS

To maintain the necessary sanitary and hygienic parameters of microclimate on the modern locomotives [Sorokin, 1996; Sokolov, 1998] the air-conditioning system, which typically consists of a steam compressional air conditioner with hermetic compressor or compressor of packing design and a heating-ventilation unit is used. This scheme has many flaws which at pesent do not allow to solve the problem of conditioning on the railway rolling stock completely:

- taking into account the fact that locomotives use air conditioners that can not work as a heat pump, the additional use of the heating-ventilation unit, which makes the design of the system more complicated and increases its value becomes obvious;

- steam compressional conditioners are structurally complex, expensive and dangerous for environment; during the operation due to vibration and sudden accelerations, the depressurization of the cooling system is possible;

- the use of one-piece scheme of the conditioner and considering its complex structure it makes the layout of the locomotive and the further rational distribution of air masses in the driver's cab more complicated;

- steam compressional conditioners use much energy consumption, resulting in additional operational costs.

Elimination of flaws relevant to the currently used air conditioning system in the locomotive cab is possibly by using of the evaporative cooling water systems. The settings of such type are clean, reliable, and structurally simple. The main advantages that characterize the evaporative coolers are the follows: environmental cleanliness, the use of renewable sources of energy, little energy consumption (comparing to steam compressional air conditioners is 10 ... 15 times lower), absence of non-ferrous metals, simple design and operation [Doroshenko, 1983; Maysotsenko, 1987; № 85.4.14.010, 1986].

The given design uses the following means of air cooling:

- evaporation of water from the surface of the porous nozzle, which is moistened by vertical capillary rise of fluid from the reservoir;

- evaporation of water from the surface of liquid film formed by the forced irrigation of a metal nozzle from the top to the bottom with the subsequent collection of fluid in the reservoir;

- separation of air flow, moving on "wet"and "dry" channels of the nozzles; in the "wet" channels the water film and nozzles are cooled, the air saturated with moisture from the "wet" channel entering the atmosphere; in the "dry" channel the air is cooled and then goes to the cab of the vehicle.

The presented schemes of the air coolers have the following disadvantages:

- the use of porous nozzles results in a reduction of their height taking into account the limits of capillary rise of liquid; the reduction of heat transfer due to the low coefficient of thermal conductivity of the nozzle material, to their pollution resulting in the deterioration of capillary fluid rise and the reduced cooling capability of the device;

- the use of metal nozzles assumes the strict film flow in the "wet" channels, which is structurally difficult to implement provided there are some "dry" channels on the back of the nozzle;

- the use of both types of nozzles, which ensure the air cooling and assumes the realization of the small values of the average coefficient of heat transfer (≈40-60 $Вт/м^2К$) from the wall to the air flow;

- both schemes are one-piece and very complex, which causes the problems with their installation on the locomotive and further air distribution in the driver's cab.

THE AIM AND THE TASK OF THE RESEARCH

The elimination of defects and the adaptation of the evaporative cooler to be used on the railway rolling stock is possible when the following positions are realized:

- during the evaporation it is preferable to use the cooled water which will cool the air directly through recuperative heat exchangers which realize the great values of the heat transfer coefficient;
- to divide the conditioner into two parts: evaporative (with nozzles, fan, water pump) and cooling (with a recuperative heat exchanger and fan);
- to use metal nozzles, which during the rotation come into the lower part, irrigated by the water and further, having passed the compressor's device go out to the top, where they are blown by the air flow;
- to connect the cooling unit of the conditioner with the cooling system of internal combustion engine of the locomotive and use it as a heating unit for locomotive cab in the cold season.

RESULTS AND THEIR ANALYSIS

Mathematical modeling of the presented processes for selecting the rational parameters of structural and regime parameters is given in the work [Lutsenko, 2011].

When considering the cooling surface of the direct evaporative cooler, the thermal balance equation [Neduzhyj, 1981; Ysachenko, 1981] can be presented as:

$$q_a = -q_w = q_\beta - q_\alpha, \tag{1}$$

where: q_a, q_w - the heat flow density in the air and to the water film on the nozzle respectively; q_α, q_β - the heat flow density defined by the heat transfer from the air to the water film and formed by evaporation of the water film.

Solving (1) is possible by using the analogy between hydrodynamic, thermal and diffusive boundary layer of the air when producing the distance from the liquid film and determining the coefficient of heat transfer, flow steam mass density and others.

The distribution of the relative velocity along the hydrodynamic boundary layer [Shlikhting, 1974] on the flat plate was defined as $u/U_\infty = f'(\eta_1)$, where u - the current speed in the boundary layer at the distance from the surface; U_∞ - the air flow rate outside the boundary layer and approximated by the following polynomial:

$$f'(\eta_1) = -3{,}57 \cdot 10^{-2}\eta_1^2 + 0{,}377\eta_1\ ; \tag{2}$$

where: $\eta_1 = 5y/\delta$ - dimensionless boundary layer coordinate; $\delta = 5 \cdot \sqrt{\nu x/U_\infty}$ - boundary layer thickness; x – distance from the starting edge of the plate to the given point; ν - kinematic air viscosity.

With the flow mode in the channel between the nozzles, we finally get the equation for the flow outside the boundary layer:

$$U_\infty = \frac{G_\Sigma}{\rho b\left(H - 0{,}7\delta\right)}; \quad U_{\max} = \frac{G_\Sigma}{0{,}65\rho bH}. \tag{3}$$

Considering the thermal boundary layer, the temperature distribution will be defined as $\frac{\vartheta}{\vartheta_\infty} = \frac{t - t_w}{t_\infty - t_w} = f'(\eta_2)$, where the distribution function on the basis of previously obtained distribution of the relative speed (2)

With the temperature gradient at the nozzle surface, the coefficient of heat transfer is defined as:

$$\alpha = \frac{\lambda}{\vartheta_\infty}\left(\frac{\partial\vartheta}{\partial y}\right)_{y=0} = \frac{1{,}89\lambda}{k}, \tag{4}$$

Based on the analogy of speed and concentration profiles of the water steam at the border layer, the density of mass flow of steam in the boundary layer is defined as

$$j_n = 1{,}89\frac{D\left(p_н - p_\infty\right)}{\delta R_n T}. \tag{5}$$

where $p_н$ - saturated steam pressure near the surfaces of the water film; p_∞ - steam pressure outside the boundaries of the boundary layer in the channel between the nozzles.

The given analysis of the processes occuring in the channel between the nozzles between the air flow and the basic platform of the nozzle allow to solve the equation (1) and determine the change of the air flow parameters.

Based on the equations of the density of heat flow going through the nozzle, taking into account the heat taken away from it and the liquid film at a time, the temperature difference of on the surface of liquid film contacting with air when changing the thermal state of the "nozzle - liquid – air" is defined as:

$$\Delta t_w = \frac{q_w\Delta\tau - (\psi - \psi')\left(\chi_н\delta_w + \chi_w\left(\delta_н + 2\delta_w\right)/2\right)}{\chi_н + \chi_w}, \tag{6}$$

where: $\psi' = \left(t'_н - t'_w\right)/\left(\delta_н + 2\delta_w\right)$, $\psi = \left(t_н - t_w\right)/\left(\delta_н + 2\delta_w\right)$, $\chi_н = c_н\rho_н\,\delta_н/2$, $\chi_w = c_w\rho_w\,\delta_w/2$ - equation coefficients.

With the heat taken away from the air at a time we can specify the current temperature change in the thermal boundary layer

$$\Delta t_\infty = \frac{2q_\alpha\Delta\tau - 0{,}74k\Delta t_w c_p\rho_a}{\left(H - 0{,}74k\right)c_p\rho_a}. \tag{7}$$

Let's define the specific steam mass located between the wall of the nozzle and the middle of the channel between the nozzles:

$$m_{n_F} = \frac{0{,}622}{R_a}\left[\frac{\delta}{5}\int_0^5\frac{p_н - \left(p_н - p_\infty\right)\left(A_1\eta_1^2 + B_1\eta_1\right)}{T_w + \left(T_\infty - T_w\right)\left(A_2\eta_1^2 + B_2\eta_1\right)}d\eta_1 + \frac{p_\infty}{T_\infty}\left(\frac{H}{2} - \delta\right)\right]. \tag{8}$$

When giving $\int_0^5\frac{p_н - \left(p_н - p_\infty\right)\left(A_1\eta_1^2 + B_1\eta_1\right)}{T_w + \left(T_\infty - T_w\right)\left(A_2\eta_1^2 + B_2\eta_1\right)}d\eta_1 = p_н F_1 - \left(p_н - p_\infty\right)F_2$ we get:

$$F_1 = \frac{1}{A_2\left(T_\infty - T_w\right)\left(q-s\right)} \ln\left|\frac{\left(q-5\right)s}{\left(s-5\right)q}\right|, \tag{9}$$

$$F_2 = \frac{1}{A_2\left(T_\infty - T_w\right)}\left(A_1\chi_1 + B_1\chi_2\right), \tag{10}$$

$$\chi_1 = \frac{1}{q-s}\left[5\left(q-s\right) + q^2 \ln\left|\frac{q-5}{q}\right| - s^2 \ln\left|\frac{s-5}{s}\right|\right], \tag{11}$$

$$\chi_2 = \frac{1}{q-s}\left[q \ln\left|\frac{q-5}{q}\right| - s \ln\left|\frac{s-5}{s}\right|\right]. \tag{12}$$

where: s, q – the root of quadratic equation $T_w + \left(T_\infty - T_w\right)\left(A_2\eta_1^2 + B_2\eta_1\right) = 0$.

Using similar methods of mathematical and physical picture of the coolant one can get some dependences, describing the processes of heat exchange in the rrigated part of the evaporation unit, such as:

- the speed of the fluid in the core of the flow in the channelbetwen the nozzles

$$U_\infty = G_\infty\left[\rho b\left(H - \delta_1\right)\right]^{-1}; \tag{13}$$

- displacement thickness of the turbulent boundary layer:

$$\delta_1 = 4{,}6 \cdot 10^{-2} x \left(\frac{\nu}{U_\infty x}\right)^{\frac{1}{5}}; \tag{14}$$

- local heat transfer coefficient:

$$\alpha' = \frac{\lambda}{\vartheta_\infty}\left(\frac{\partial \vartheta}{\partial y}\right)_{y=0} = \frac{\lambda}{k_1}, \tag{15}$$

The definition of speed and air and water pressure in curvilinear flow is defined by the following equations:

$$u = \exp\left(C_1 - \ln R\right), \tag{16}$$

where: $C_1 = \ln\left(u_m R_m\right)$ - integration constant; $u_m = G_\Sigma\left(\rho H R_m\right)^{-1}$ - the average speed in the channel between the nozzles; R_m - radius of the middle of the current lines.

$$p = C_2 - 0{,}5\rho \exp\left[2\left(C_1 - \ln R\right)\right], \tag{17}$$

where: $C_2 = p_{atm} + 0{,}5\rho \exp\left[2\left(C_1 - \ln R_m\right)\right]$ - integration constant; p_{atm} – atmospheric pressure at the entrance of air flow in the curvilinear plot.

Since the calculation of the process of heat and mass exchange at evaporative cooling was performed numerically [Karimberdieva S., 1983, Patankar S.V., 1984, Peyre R., Teylor T.D., 1986], the nozzle and the direction and heat carriers were covered by polar nets, and when constructing them the minimum radius of the air flow lines was determined from the dependence:

$$R^a{}_{\min} = R_н \Big/ \left[2\cos\left(\alpha/2\right)\right] \tag{18}$$

where: $R_н$ - nozzle radius; α - angle of the coverage of the irrigated nozzle surface.

The maximum radius of the current water lines was defined as:

$$R^w{}_{\max} = R_н tg\,\alpha/2\,. \tag{19}$$

Polar coordinates of the net knot are defined as:

- in the air

$$\begin{cases} R^a{}_i = R^a{}_{\min} + i\Delta R; \\ \phi^a{}_j = \phi^a{}_{j\max} - j\Delta\phi, \end{cases} \tag{20}$$

- in the water

$$\begin{cases} R^w{}_i = R^w{}_{\max} - i\Delta R; \\ \phi^w{}_j = \phi^w{}_{j\max} - j\Delta\phi, \end{cases} \tag{21}$$

where: ΔR, $\Delta\phi$ - the step of the change of radius and angle of the net model; $\phi^a{}_{j\max}$ - maximum angle of the net opening to air at the current R_i; $\phi^w{}_{j\max}$ - maximum angle of the net opening for water at the current R_i; i, j – net knot indices.

Based on the developed model and calculation program, numerous experiments with the influence of cooling efficiency of mode and design parameters of the evaporation unit have been carried out and the results are presented as approximating dependencies:

$$\bar{Q}_{xp} = 688,48 + 69,83z_1 + 7,67z_2 - 3,57z_1^2 - 10,87z_2^2 + 8,98z_1z_2\,, \tag{22}$$

$$\bar{Q}_{xp} = 1722,8 - 185,6x_1 + 885,5x_2 + 330,1x_3 + 43,2x_1^2 + 10,9x_2^2 - \\ -251,1x_3^2 - 40,8x_1x_2 - 10,1x_1x_3 + 76,4x_1x_2 \tag{23}$$

where: z_1 – the air speed when entering the channel, м/с; z_2 – the speed of the nozzle rotation, хв$^{-1}$; x_1 – the nozzle radius, мм; x_2 – nozzle thickness, мм; x_3 – the distance between the nozzles.

The analysis and the calculations based on the developed mathematical model allowed to identify the main design parameters that affect the energy, mass and size characteristics and identify their rational measures: the rotating nozzle diameter, which determines the surface area of heat and mass exchange $D_н$=200…400 мм; the nozzle thickness, which characterizes the possibility of accumulation and transfer of heat (cold) $\delta_н$=1,5…2,5 мм; the distance between nozzles, which determines the flow of heat carriers H=7…8 мм.

The experimental studies on stand models [Idelchik I., 1975, Bagan I.P., 1989, Gerschenko O.A., 1984] of the air conditioner of the evaporative cooling and its elements resulted in obtaining its power, aero-and hydrodynamic characteristics depending on the mode characteristics of the heat flow and heat carriers and environmental parameters [Mohyla V.I., Lutsenko O.A., 2011].

CONCLUSIONS

The results of the research present the following:

- the increase of the air flow speed in the channel between the nozzles to the values of 8 m / s allows to increase the cooling capacity ranging from 1900 to 4000 W without any deterioration in the work of the device, the great values of the air speed corresponding to the great values of the nozzle rotation and consequently on the contrary;

- when the rotation frequency of the nozzle is 100 and 80 rpm and consequently the air speed in the channel between the nozzles is 10 and 16 m/s and more, one may notice the drop removing of the liquid phase, with the smaller values of the nozzle rotation and in the range of the air speed, the moisture removing hasn't been observed;

- the realization of the maximum cooling capacity of the unit, which amounted to 4200 W when the removing of the liquid phase is absent provides the rotation speed of 1980 rpm and the air flow speed of 12 m/s, which allows to define these modes as rational;

- aerodynamic resistance of the air path for these parameters is 340 Pa;

- the cost of mechanical power to the nozzle drive when changing the rotation speed from 40 to 100 rpm amounted to the value in the range from 25 to 100 W.

Considering the experimental studies and the obtained results [Reho, 1987] we may state thefollows:

- there are rational parameters of the air flow speed in the channel between nozzles and the nozzle rotation speed corresponds to 12 m /s and 80 rpm, which provides the maximum cooling capacity of the evaporative unit of the conditioner; these modes do not allow any drop removing of the water environment, which provides the best possible technical and economic parameters of the device of this type;

-the optimum water flow rate was defined ($6 \cdot 10^{-3}$ м^3/s) in the irrigated part of the evaporative unit of the conditioner, which provides the maximum cooling capacity and consequently the heat balance between the energy processes of evaporative cooling in the air part of the nozzle and the processes of heat and mass trasfer in the irrigated part of the nozzle;

- the change of the water temperature entering the evaporator unit of the conditioner makes the proportional impact on its cooling capacity, which contributes to the flexible regulatory characteristics when changing the parameters of microclimate in the locomotive cab;

The use of the results obtained allows to realize the cooling efficiency of the conditioner to the values of 2580 W, which provides the temperature of +26 °C in the locomotive driver's cab 2TE116 at the environmental temperature +45 ^{o}C and the relative humidity 90%.

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РЕЗУЛЬТАТЫ КОМПЛЕКСНЫХ ИССЛЕДОВАНИЙ ИСПАРИТЕЛЬНОГО КОНДИЦИОНЕРА ДЛЯ КАБИНЫ МАШИНИСТА ТЕПЛОВОЗА

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Проведений комплекс теоретичних и експериментальних исследований испартельного кондиционера обеспечил получение конструкции устройства, которая может использоваться на подвижном составе железных дорог и отвечает требованиям нормативных документов относительно параметров микроклимата кабины машиниста локомотива.

Ключевые слова: испарительный кондиционер, тепловоз, поток воздуха, металлическая насадка.