Characteristics of Air Parameters in Hydro-Technical Works

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Abstract: - The paper proposes the study of two important parameters for the hydro-technical works. On one hand, the influence of air humidity over its temperature, resulting thus, into a temperature rise on a portion of the trajectory or on a length unit, and on the other hand the convection coefficient of hydro-technical works according to the air stream and the geometry of the hydro-technical work are calculated.

Key words: - mass transfer coefficient, humidity balance, temperature variation on lineal meter per hydro technical work, hydro-energetically installations, real gases, convection coefficient.

1 Introduction

Comfort, in some mining activities or productive hydro energetic activities rooms, is highly influenced by the air humidity. Humidity influences the physiology of organisms and produces negative effects on technical installations in these rooms.

Air pressure, in these rooms, does not significantly oscillate, being almost the same as the atmospheric pressure outside. When works are being executed at a higher depth, signify changes of pressure occur. Due to the lowered pressure in these rooms, the evaporation of water vapours is possible even at lower temperatures than the one corresponding to the barometric pressure.

Moreover, the natural air stream or the air circulating through the ventilation system is in contact with certain open areas of water or with very humidified walls or walls on to which water a thin layer of water may be flowing. When water vapours reach a temperature below the condensation point they condense on the contacted areas. The condensed quantity is negatively influencing the machineries and the equipment.

In this paper, the influence of air humidity over the temperature of the areas traversed by the air stream and the value of the convective coefficient between air and the objects bounding its flow. The value of this coefficient is very little known, thus we consider that its determination is useful both for those who work in such climate conditions and for the designers of ventilation systems.

2 Calculus Method

Air humidity is calculated by the following known relation:

$$x = 10^{-3} \cdot d = 0,622 \cdot \frac{\varphi \cdot f(t)}{B - \varphi \cdot f(t)} \left[\frac{\text{kg vapour}}{\text{kg dry air}} \right] \quad (1)$$

where:
B - is the barometric pressure;
φ - relative humidity;
f(t) - the pressure of the saturated vapours at temperature t.

But when almost reaching the condensation point, water vapours will no longer respect the Perfect Gas Law, thus they cannot be considered real, leading to the several equations.

For this hypothesis we are going to take into consideration E. Schmidt's equation:

$$v = \frac{R \cdot T}{p} - \frac{b}{\left(\frac{T}{100}\right)^{n_1}} - p^2 \cdot \left[\frac{c}{\left(\frac{T}{100}\right)^{n_2}} - \frac{d}{\left(\frac{T}{100}\right)^{n_3}} \right] \quad (2)$$

where:
v is the specific volume;

T – the absolute temperature of the moist air;
p – the pressure of the moist air;
R – nature constant of the moist air;

The exponents n_1, n_2, n_3 are coefficients determined from specialty literature.

Clapeyron Equation for dry air is the following:

$$p_a \cdot V = m_a \cdot R_a \cdot T \tag{3}$$

where:
m_a is the dried air mass;
R_a – dried air nature constant;
p_a – the partial pressure of the dried air;
V – moist air volume;

The humidity level results from the two equations:

$$x = \frac{p_v}{p_a} \cdot \left\{ 1 + \frac{b \cdot v_v^{-1}}{\left(\frac{T}{100}\right)^{n_1}} + p_v^2 \cdot v_v^{-1} \cdot \left[\frac{c}{\left(\frac{T}{100}\right)^{n_2}} + \frac{d}{\left(\frac{T}{100}\right)^{n_3}} \right] \right\} = \frac{m_v}{m_a} \cdot \frac{R_v}{R_a} \tag{4}$$

where index "v" it refers to water vapours, and index "a" to dry air.

It appears a coefficient that can be calculated knowing the values of a, b, c, d and m from literature and parameters p_v and v_v from diagrams and tables.

$$x = 0.622 \frac{\varphi \cdot f(t)}{B - \varphi \cdot f(t)} \cdot \left\{ 1 + \frac{b \cdot v_v^{-1}}{\left(\frac{T}{100}\right)^{n_1}} + p_v^2 \cdot v_v^{-1} \cdot \left[\frac{c}{\left(\frac{T}{100}\right)^{n_2}} + \frac{d}{\left(\frac{T}{100}\right)^{n_3}} \right] \right\} \tag{5}$$

In relation (5) we neglect the last term of the parentheses, this doesn't influence with relevant errors. So we obtain:

$$x = 0.622 \cdot \frac{\varphi \cdot f(t)}{B - \varphi \cdot f(t)} \cdot \left(1 + \frac{b \cdot v_v^{-1}}{\left(\frac{T}{100}\right)^{n_1}} \right) \tag{6}$$

3 Problem Solving

From the specialty literature we know $b = 0.9172$ and $n_1 = 2.82$, and for $t = 20^0\mathrm{C}$, the searched correction coefficient has the value:

$$1 + \frac{b \cdot v_v^{-1}}{\left(\frac{T}{100}\right)^{n_1}} = 1 + \frac{0.9172 \cdot 57.8^{-1}}{\left(\frac{293}{100}\right)^{2.82}} = 1.01$$

In order to observe the modifications of air humidity on a hydro technical work length $\Delta y = 120\,\mathrm{m}$ with a certain flow, on an area S, perimeter P and radius R_0, measurements have been made on both ends of the hydro technical work and the values have been inserted into table 1. It has been considered that the air is in a state close to saturation.

Temperature variation on lineal meter of air is obtained using the relation:

$$\Delta T = \frac{r}{c_p} \cdot \frac{\Delta x}{\Delta y} = \frac{2500 \cdot 10^3}{1011.6} \cdot \frac{0.00057}{120} = 0.01173 \frac{^0\mathrm{C}}{\mathrm{m}} \tag{7}$$

where: r is the evaporation latent heat.

Table 1 - Air temperature variation

Crt. no.	1
Flow and working geometry	
$\dot{m}$ [kg/s]	29.08
R_0[m]	1.59
S[m^2]	6.5
P[m]	8.19
$c_p\left[\frac{J}{kg \cdot K}\right]$	1011.6
Air humidity balance	
B[mmHg]	725.3
t_1[^{0}C]	16.4
t'_1[^{0}C]	15.58
φ_1[%]	92.3
x_1	0.0113
t_2[^{0}C]	16.0
t'_2[^{0}C]	15.9
φ_2[%]	98.3
x_2	0.0118
Δx [m]	0.00057

The simplifying hypothesis, has been taken onto consideration, where there are no other sources to contribute to the change of temperature.

The convection coefficient is another very important parameter for air contacting open areas of water or moist walls.

A.N. Şcerban considers the criterial equation:

$$Nu = \xi \cdot A \cdot Re^m \tag{8}$$

Be it the case of turbulent ventilation we may consider $\xi = 1$, taking into account Nu and Re:

$$\alpha = 2.44 \cdot \frac{\dot{m}^{0.8} \cdot P}{S} \qquad (9)$$

where:
$\dot{m}$ is the air flow in kg/s;
P – hydro technical work perimeter in meters;
S – cross section of the hydro technical work in m^2.

Using the expression no (9) the values of the convection coefficient have been calculated and inserted into table 2.

Table 2. Convection Coefficient Values

Type of hydro tech-nical work	Flow [kg/s]	Work Peri-meter [m]	Cross Secti-on [m^2]	Convec-tion Coeffi-cient [$W/(m^2 \cdot K)$]
Concrete Well	116 - 490	12.5 – 18.84	12.65 – 28.26	13.75 – 20.85
Double Metallic Grove	47 – 167	7.25 – 11.48	5.2 – 10.0	12.61 – 24.23
Inclined Planes	63 – 150	11.83 – 12.09	8 – 10	12.74 – 21.93
Winze	25 - 92	5.9 – 8.2	2.8 -5.3	14.29 – 25.64

The graph in Figure 1 is based on Table 1 (ANNEX 1)

There is also a mass transfer next to the heat transfer calculated using the relation no. (10):

$$\alpha' = \beta \cdot r \cdot \frac{p_v^{'} - p_v}{t_p - t_a} \qquad (10)$$

where:
β is the evaporating mass coefficient in $\frac{kg}{m^2 \cdot h \cdot mbar}$;
$p_v^{'}$ - partial pressure of water vapours to the walls temperature in mbar;
p_v - partial pressure of water vapours in air in mbar;
t_p – wall's temperature in °C;
t_a – air temperature measured with a dry thermometer in °C.

If $\beta = 0.01$, is accepted in the speciality literature $\varphi > 0.8$, then the average coefficient through mass transfer becomes $\alpha' \cong 3 \frac{W}{m^2 \cdot K}$.

The same result may also be reached by another criterial relation:

$$Nu = 0.46 \cdot Re^{0.57} \qquad (11)$$

The convection coefficient α of the Nusselt criterium depends on the temperature, the physical properties of air and on the flow regime.

Explaining the relation no.(11), for hydro technical works, the following relation has been determined:

$$\alpha = 0.45 \cdot \frac{\lambda_a}{\upsilon_a^{0.58}} \cdot \frac{w^{0.58}}{d^{0.42}} = k(t) \frac{w^{0.58}}{d^{0.42}} \qquad (12)$$

Where k(t) is a term depending only on temperature;
w – air speed in hydro technical works in m/s;
υ_a - relative sliminess of air in m^2/s;
λ_a - conduction coefficient of air, in $\frac{W}{m \cdot K}$.

The temperature coefficient k(t) has values between the interval [1]

$$k(t) = 6.986 \div 7.1812 \frac{J}{m^{2.16} \cdot s^{0.42} \cdot K}$$

The graph in Figure 2 (ANNEX 2) has been drawn taking into consideration, the physical properties of air in the temperature interval, the speed of air comprised between $0.2 \div 8\,m/s$, and the Tables 1 and 2.

The following notes have been made in Figures 1 and 2:
GDM – double metallic gallery/grove;
GDZ – double walled gallery/grove;

GDB – double concrete gallery/grove;
GSM – simple metallic gallery/grove;
GSZ – simple walled gallery/grove;
PB – concrete well;
PLDM – double metallic inclined plane.

4 Conclusions

1. The resulting Images 1 and 2 correlate the parameters needed in order to obtain the thermal comfort stage for the hydro technical works in tunnels or in underground warehouses.
2. Image 2 offers the data regarding the convective coefficient depending on the hydro technical work geometry and on the mass flow. These two are of utmost importance when designing the ventilation system.

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ANNEX 1

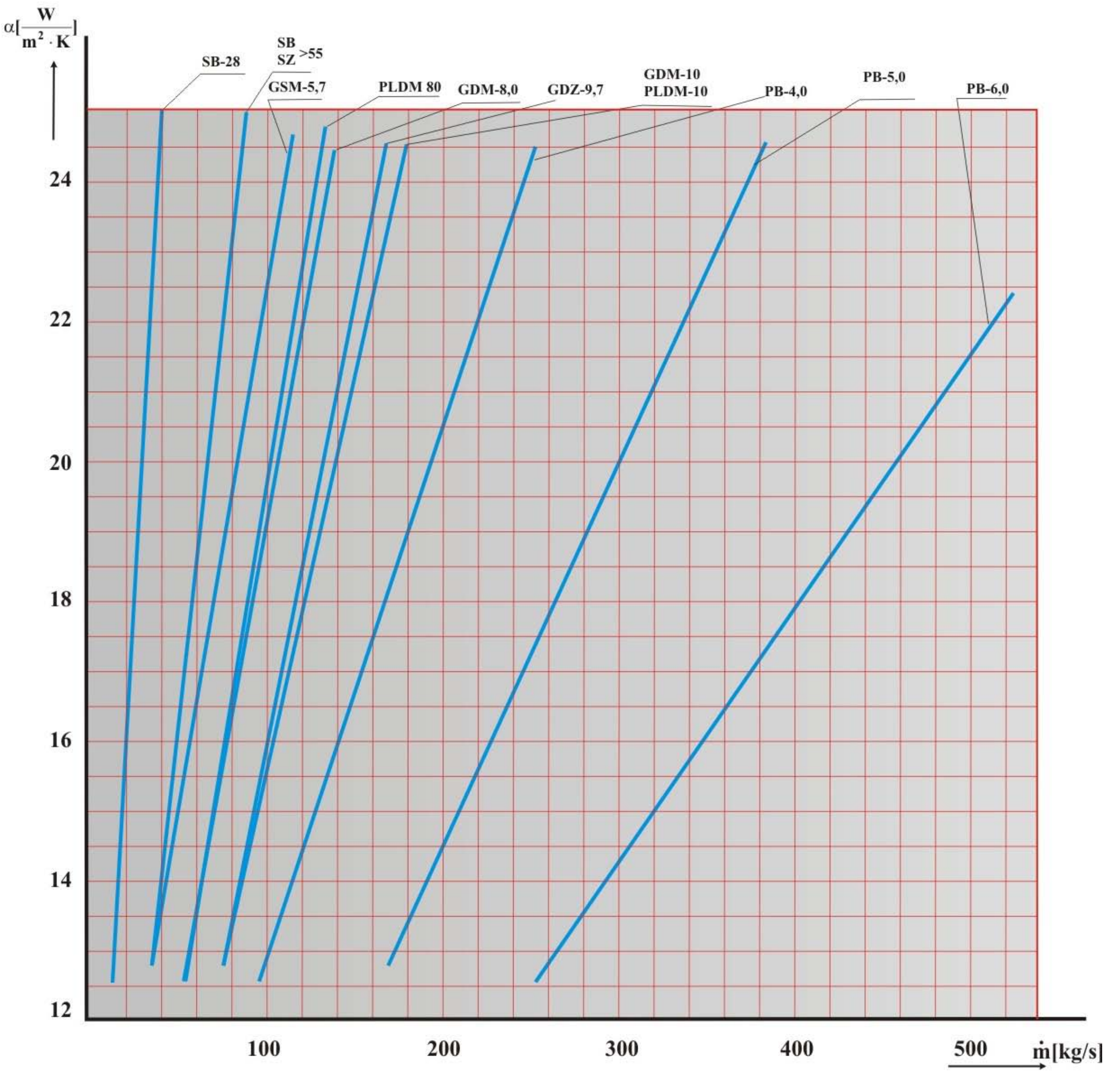

Fig.1 Conduction Coefficient Dependency on the mass flow for different types of galleries/groves

ANNEX 2

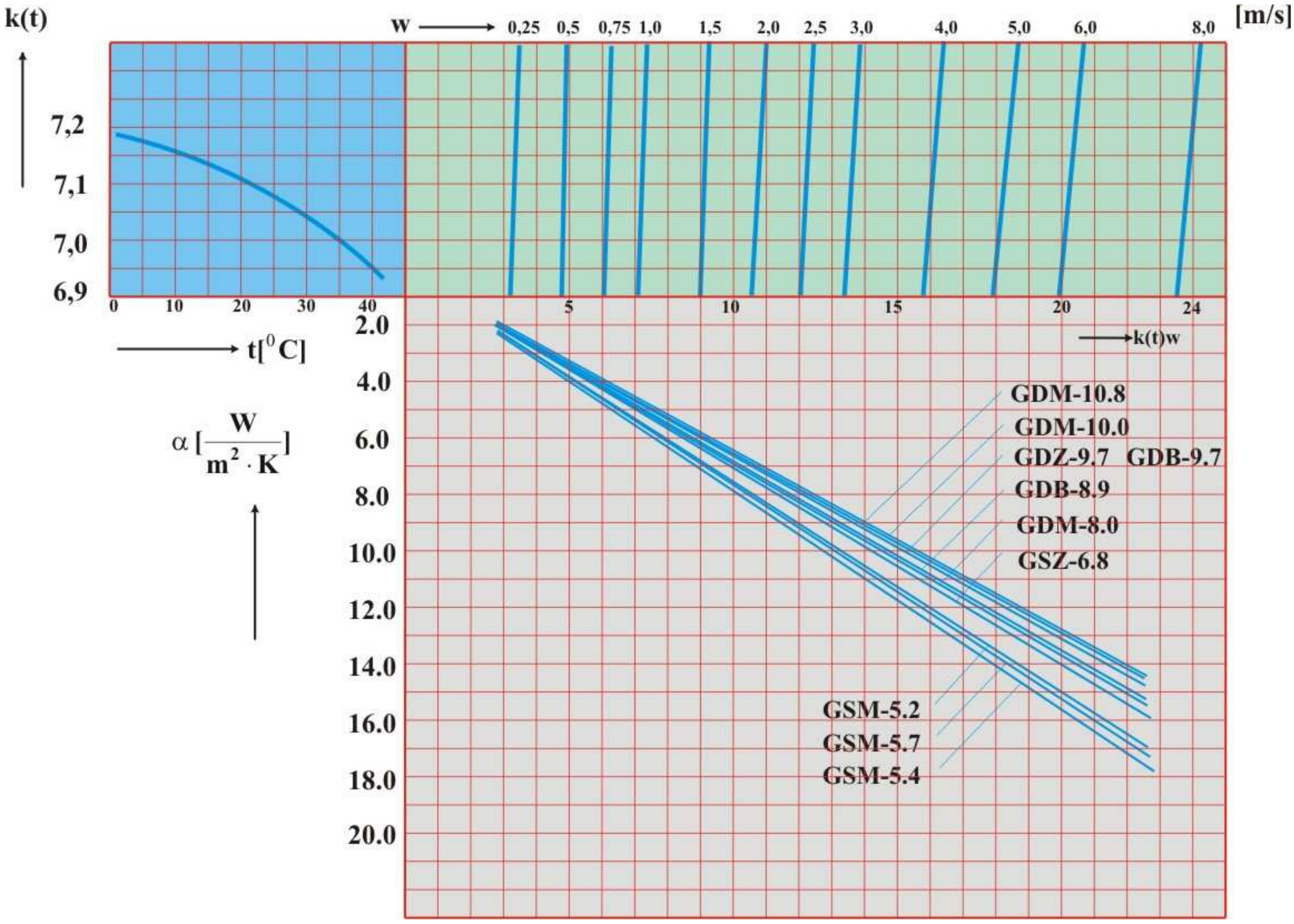

Fig.2 The convective coefficient depending on the geometry of the hydro technical work and on the mass flow

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