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INVESTIGATION AND OPTIMIZATION OF AIR EXCHANGE IN INDUSTRIAL HALLS VENTILATION

S.P. STEPHANOV
Higher Institute of Mechanical and Electrical Engineering,
1156 Sofia (Bulgaria)

ABSTRACT

The paper presents a method for a more accurate calculation of air exchange in the mechanical ventilation of industrial halls with uniformly distributed heat load. The method has been developed on the basis of finding the relation between the parameters of the air supplied and those of the working zone. A cybernetic approach has been applied based on a thermal and technological experiment with a statistical analysis. The experiment has been carried out by a physical model of a three-body shop built up by geometrical and thermal modelling. Air was supplied into the model by air distributors for uniform supply designed on the basis of an optimization procedure having the criteria: degree of non-uniformity of supply, and aerodynamic resistance. A non-linear optimization problem for minimizing the amount of the supplied air and the energy for its treatment has been defined on the basis of the derived regression dependences. The method suggested is illustrated by a numerical example.

The methods used for determining the parameters of air, supplied for ventilation of industrial halls when heat is emitted, are based on balance equations connected with a preliminary assumption of the ψ coefficient of heat that remains in the working zone. The application of these method ofteh leads to overcomputation of air exchange because of the wide boundaries of variation of this coefficient in the existing literature. On the other hand, its accurate determination is made difficult by the great variety of technological processes, the construction of industrial buildings and the air-exchange schemes.

Hence, the aim of this paper is to suggest a method for a more accurate computation of air exchange when mechanical ventilation of industrial halls is effected by a uniformly distributed heat load.

This method is chiefly based on finding the relation between the working zone parameters and the supplied air parameters. Such a functional dependence implicitly accounts for the heat flow distribution in the working zone and allows for the application of a procedure for minimizing the flow rate of air supplied as well as the energy needed for its treatment.

The multi-dimensional character of heat and mass transfer processes in room ventilation, as well as the complex interaction between the factors defining them, make their analytical investigation considerably difficult. A good opportunity in this respect is offered by the cybernetic approach based on a real physical experiment followed by a statistical analysis.

By applying the theory of geometrical and thermal similarity (ref.3), a physical model was designed of a three-body, technologically equipped shop, and an investigation was made of five air-exchange schemes (see Fig.1) that are commonly used. After

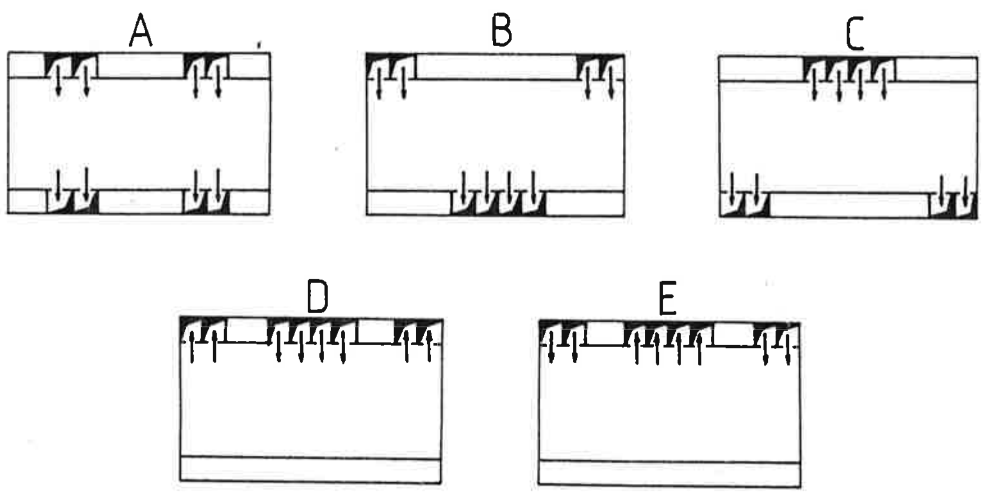

Fig.1. Five commonly used air-exchange schemes.

some preliminary studies, the following parameters were assumed to be the basic factors of the prototype shop: the parameters of the air supplied, i.e. flow rate L, temperature $\bar{t}_{Bx}$ and mean velocity $\overline{W}_{Bx}$, as well as the heat load of the shop Q_T ; while the temperature field in the working zone, i.e. mean temperature $\bar{t}_{p3}$ and its standard deviation in the working zone $\sigma_{\bar{t}_{p3}}$, as well as the basic parameters of the velocity field, i.e. mean velocity $\overline{W}_{p3}$ and $\sigma_{\overline{W}_{p3}}$, were assumed to be the initial quantities. Independence between factors L and $\overline{W}_{Bx}$ was achieved by varying the aperture width of air distributors for a uniform supply that were used for this model. Thus, by introducing an additional factor, the aperture width b , the necessity for measuring the supply air velocity was avoided; the simplifies the experiment considerably and increases the accuracy during measurements. The independence of the factors, as well as the possibilities of the physical model, allowed an active experiment to be conducted according to scheme b which is a optimum plan on a factor space Z having the following boundaries:

$$Z = \{L,\ Q_T,\ t_{Bx},\ b\} \tag{1}$$

for the model:	for the real shop:
$50 \leq L \leq 150\ m^3/h$	$156250 \leq L \leq 468750\ m^3/h$
$0{,}2 \leq Q_T \leq 1{,}4\ kW$	$625 \leq Q_T \leq 4375\ kW$
$15 \leq \bar{t}_{Bx} \leq 36^oC$	$15 \leq \bar{t}_{Bx} \leq 36^oC$
$4 \leq b \leq 6\ mm$	

The air distributing devices used in the experiment are of the type that permits a minimum variation of the static pressure along the axis and a great variety of geometric profiles of the longitudinal and cross section: rectangular, circular, V-type, etc.

The computation of these air distributors, including the determination of the configuration that is optimal for the given conditions, has been effected by an automated procedure. It is based on a boundary problem for non-linear differential equations of the second order with one boundary condition being known; the problem describes air distribution along the length of the aperture (ref. 4).

For solving the boundary problem, a numerical method (ref. 4) is suggested that includes a modification of "shooting" by a double interpolation, the integration being done according to a combined scheme of the Runge-Kutta and Adams methods. The solution of the boundary problem permits the formulation of integral assessments of the degree of non-uniformity of air flow and the aerodynamic resistance. An optimal assignments algorithm was formulated on this basis, for determining the most suitable configuration and its respective parameters securing an optimal process of air distribution with respect to the degree of non-unifor mity and the energy consumption required for effecting it.

Temperature in the working zone was registered in 36 evenly distributed points by means of a "Thermoair" hot-wire anemometer and a digital thermometer of the TT 4000 type.

Concerning the initial quantities, the statistical processing of experimental data resulted in regression dependences of the following type:

$$\bar{t}_{p3} = A.L^{a_1}.Q_T^{a_2}.\bar{t}_{Bx}^{a_3}.B^{a_4} \qquad (2)$$

$$\bar{W}_{p3} = C.L^{c_1}.Q_T^{c_2}.\bar{t}_{Bx}^{c_3}.B^{c_4} \qquad (3)$$

The significance of the regression coefficients was ascertained after a statistical test by the Student criterion. The goodness of fit of regression polynomials (2) and (3) was tested by Fisher's variance ratio, while the sufficiency of the plan of the experiment with respect to number and rendomization of tests was proved by Cochran criterion.

The $\bar{W}_{Bx}$ factor was included in these equations by using the balance equation

$$B = \frac{L}{n.\ell.\mu.\bar{W}_{Bx}}. \qquad (4)$$

where L = flow rate of air supplied to air distributors; n = number of air distributors; B = width of aperture; ℓ = length of air distributors; and μ = 0,65 is the flow rate coefficient for the accepted type of apertures;

for the model: $0.3 \leq \bar{W}_{Bx} \leq 1.5$ m/s

for the real shop: $1.5 \leq \bar{W}_{Bx} \leq 7.5$ m/s

The derived regression equations allow the formulation of a new method of air exchange computation.

Let us assume that the technological and hygienic requirements determine the following admissible boundaries of air-conditioning parameters in the working zone:

$$\bar{t}_{p3} \in [\bar{t}_{p3,gon}^{min}, \bar{t}_{p3,gon}^{max}] = T \qquad (5)$$

$$\bar{W}_{p3} \in [\bar{W}_{p3,gon}^{min}, \bar{W}_{p3,gon}^{max}] = W \qquad (6)$$

Then, considering a certain zone of air exchange, the supplied ai parameters L, $\bar{t}_{Bx}$ and $\bar{W}_{Bx}$, necessary for satisfying the conditions of (5) and (6), may be sought among the values that minimize the energy consumption needed for air treatment under summer conditions:

$$Q = L . \rho . c \, (t_{BH} - t_{Bx}) \qquad (7)$$

where t_{BH} = temperature of outdoor air; ρ = air density; and c = specific heat capacity.

With such a formulation, a non-linear optimization problem can be defined, with the efficiency function

$$\min Q \sim \min \, (L \, (t_{BH} - \bar{t}_{Bx})) \qquad (8)$$

and non-linear parametric constraints defining the space of the admissible values:

$$\Gamma = Z \cap T \cap W \qquad (9)$$

The solution of this problem was sought by the linearization of the parametric constraints, keeping the non-linearity of the efficiency function. The linearization of the non-linear envelope Γ^* of the Γ space was done step-by step by taking the logarithms of the parametric constraints for Z , T and W , as follows:

Space Z is represented by its projection on a plane having the coordinates $\ln \bar{t}_{bx}$ and $\ln L$:

$$\left| \begin{array}{l} \ln \bar{t}_{Bx} = \ln \bar{t}_{Bx}^{min} = const \\ \ln \bar{t}_{Bx} = \ln \bar{t}_{Bx}^{max} = const \end{array} \right. \qquad (10)$$

$$\left| \begin{array}{l} \ln L = \ln L^{min} = const \\ \ln L = \ln L^{max} = const \end{array} \right. \qquad (11)$$

Linearization of corresponding envelopes T^* and W^* is effected by taking the logarithms of equations (2) and (3) respectively, with the values of $\bar{t}_{p3}$, $\overline{W}_{p3}$, and Q_T being known, and the value of $\overline{W}_{Bx}$ being assumed from its respective interval of variation:

$$\begin{array}{l} \ln \bar{t}_{Bx} = A_1 + A_1^* \ln L \\ \ln \bar{t}_{Bx} = A_2 + A_2^* \ln L \end{array} \qquad (12)$$

$$\begin{array}{l} \ln t_{Bx} = C_1 + C_1^* \ln L \\ \ln t_{Bx} = C_2 + C_2^* \ln L \end{array} \qquad (13)$$

By taking the logarithms of equation (8), the efficiency function assumes the form which is convenient for the subsequent optimization procedure:

$$\min \Phi = \min (\ln Q) \sim \min \{ \ln L + \ln (t_{BH} - e^{\ln \bar{t}_{Bx}}) \} \qquad (14)$$

Such a linearization allows for the optimization problem to be defined in the from of

$$
\begin{aligned}
&\min_{\ln L,\ \ln \bar{t}_{Bx}} \Phi \\
&\qquad \ln \bar{t}_{Bx}^{min} \leq \ln \bar{t}_{Bx} \leq \ln \bar{t}_{Bx}^{max} \\
&\qquad \ln \bar{t}_{Bx} \leq A_1 + A_1^* \ln L \\
&\qquad \ln \bar{t}_{Bx} \geq A_2 + A_2 \ln L \\
&\qquad \ln \bar{t}_{Bx} \leq C_1 + C_1^* \ln L \\
&\qquad \ln \bar{t}_{Bx} \geq C_2 + C_2^* \ln L \\
&\qquad \ln L^{min} \leq \ln L \leq \ln L^{max} \\
&\qquad \bar{W}_{Bx} = const
\end{aligned}
\tag{15}
$$

There are two alternatives for solving the defined optimization problem (15): by varying the velocity of air supply $\bar{W}_{Bx}$, or by varying the width of the aperture δ.

From the point of view of finding the final result, the two approaches are identical, i.e. the optimum parameters of supplied air are defined: flow rate L_{OPT}, temperature $\bar{t}_{Bx}$, and velocity $\bar{W}_{Bx}$.

In order to facilitate the computational procedure, the approach of varying the width of the aperture δ was applied in this case. An automated optimization procedure was formulated, including a gradient method of seeking the extremum. To illustrate the problem, here follows an example:

Define the air exchange for summer conditions in an industrial hall with a uniformly distributed heat load Q_T = 1250 kW, if the standard requirements for the air-conditioning parameters in the working zone are: $\bar{t}_{p3}$ = 27 - 29°C and $\bar{W}_{p3}$ = 0.5 + 0.6 m/s, with an outdoor temperature of 32°C. The air-exchange scheme C (see Fig. 1) was assumed. The coefficients of regression equations (2) and (3) concerning the model, according to scheme C, are presented in Table 1.

TABLE 1

	$\bar{t}_{p3}$		$\bar{W}_{Bx}$
A	8.495362	C	0.0221373332
a_1	-0.087333076	c_1	0.636870768
a_2	0.295512402	c_2	0.0234487513
a_3	0.6234553	c_3	-0.0155169904
a_4	-0.0422349668	c_4	-0.588437193

The computational procedure reduced the real problem into a respective problem concerning the model; it solves the problem for the model and transforms the final results into a respective solution concerning the real shop.

The solving of the problem was started with $\mathcal{B}$ = 5 mm. The graphic presentation of the initial solution is shown in Fig. 2.

Parameters of the characteristic points:

p.1	L = 52.47 m^3/h	$\bar{t}_{Bx}$ = 19.15°C	Φ = 6.51
p.2	L = 52.63 m^3/h	$\bar{t}_{Bx}$ = 21.49°C	Φ = 6.31
p.3	L = 70.14 m^3/h	$\bar{t}_{Bx}$ = 22.37°C	Φ = 6.52

After an automatic variation of the width $\mathcal{B}$ in the direction minimizing the efficiency function Φ , the final solution was obtained, the graphic representation of which is shown in Fig. 3.

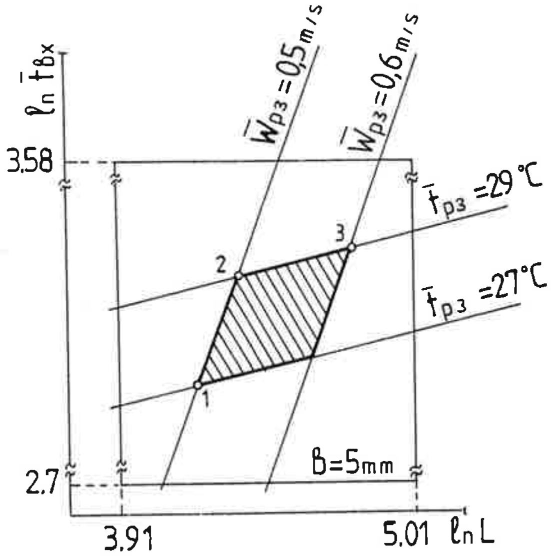

Fig. 2

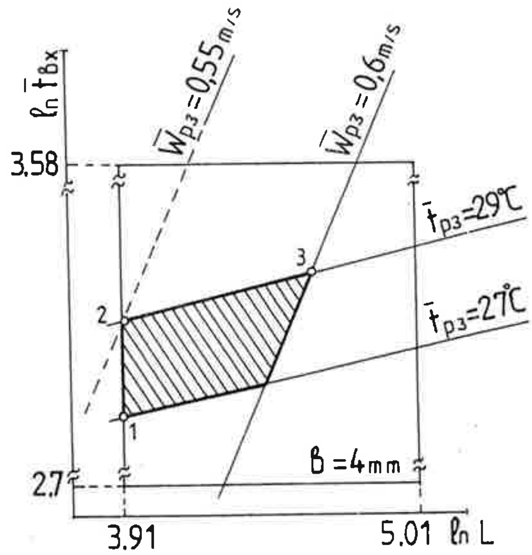

Fig. 3

Parameters:

Final width of aperture = 4 mm

p.1	L = 50 m^3/h	$\bar{t}_{Bx}$ = 18.73°C	Φ = 6.497
p.2	L = 50 m^3/h	$\bar{t}_{Bx}$ = 21.01°C	Φ = 6.309
p.3	L = 57 m^3/h	$\bar{t}_{Bx}$ = 21.41°C	Φ = 6.4

Point 2 is the optimum point.

Parameters of the optimum point for the real shop:
flow rate $L = 156250\ m^3/h$
velocity $\overline{W}_{Bx} = 2.43\ m/s$
supply air temperature $\overline{t}_{Bx} = 21^{\circ}C$
temperature in the working zone $\overline{t}_{p3} = 29^{\circ}C$
mean velocity in the working zone $\overline{W}_{p3} = 0.55\ m/s$

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