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ROOM VENTILATION ON THE BASE OF NUMERICAL MODELLING OF AIRO- AND THERMODYNAMICS

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SUMMARY

Presented are the mathematical models and the concrete examples of calculations of air distribution, of the determination of regularities of velocity field formation, of temperature and pollutant concentration in ventilated premises taking into account the power of thermo-, dust- and gas excreta, of premises configuration, of the arrangement of inlet and outlet ventilation openings and of other parameters.

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The effectiveness of system of ventilation, conditioning and heating of industrial rooms is increasing under the conditions of industrial intensification. The complication of technological and engineering systems of buildings and industries and the increasing of sanitary requirements make urgent development of air distribution of calculating methods and determination of regularities of velocity field formation, temperature and concentration of pollutents in the ventilated rooms.

At present enough important investigations are carried out abroad in this direction. Numerical models of dynamic of the ventilating streams are stated in the papers of B.Hanel, P.Sholtc, P.V.Nilson, Y.Sakamoto, Y.Matsuo, A.D.Lemyre, M.Villand, F.Bern, L.Davidson and others. However, in this investigations authors consider two-dimensional or izothermal tasks, but they do not consider the main task (distribution of pollutants). Moreover, in this investigations authors analyse the rooms with primitive configuration. M.I.Poz, V.E.Golovichev, G.M.Pozin, V.N.Varapaev, V.E.Perekalsky and other scientists conduct investigations in this direction in the former USSR. However, these models are limited by two-dimensional or izothermal organization and authors consider separate tasks only. Therefore the task of building of mathematical models of ventilation, conditioning and heating of the industrial rooms is highly urgent.

The complex mathematical models of aerodynamics and ventilation of underground rooms and chambers of optinal shape have been developed at Kola Science Centre of the Russian Academy of Sciences / 1, 2, 3/. These models take into account the influence of heat factore and are based on numerical solution of the basic equations (averaged according to the Reynolds'rule) of aero- and thermodynamics. Authors use Bussineck's approximation in two- and three-dimensional organizations and models of turbulence of the first order. Accumulated experience of mathematical modelling allowed to realize three-dimensional model of aero- and thermodynamics and ventilation of industrial rooms of complicated configuration / 4, 5/.

DESCRIPTION OF MODEL

We consider the room of arbitrary form (fig. 1) and introduce rectangular system coordinates x, y, z and use next means:

- Ω - inside space of rooms;
- S_i - solid room boundaries, including surface of ceiling (S_1), surface of floor with equipment (S_2) and side surface of walls (S_3);
- Σ_i - ventilating windows, air conduit, openings for extracted ventilation and other airation apertures ($i = 1,n$);
- n - number of inlets and outlets.

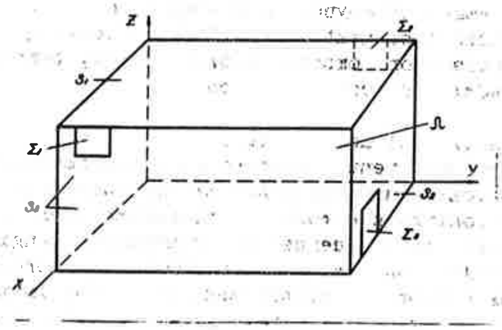

Fig1. The common shape of room.

The basic equations (averaged according to the Reynolds' rule) of aero- and thermodynamics and transfer of admixtures in turbulent atmosphere for description dynamics of air streams dynamic and distribution of polluting admixtures in rooms space are used in next form:

$$\rho\left(\frac{\partial u_i}{\partial t} + u_j\frac{\partial u_i}{\partial x_j}\right) = -\frac{\partial P}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + J_i + \beta_T T\delta_{i3} g \qquad (1)$$

$$\rho\left(\frac{\partial T}{\partial t} + u_i\frac{\partial T}{\partial x_i}\right) = \frac{\partial H_i}{\partial x_i} + J_T + \frac{L}{C_p}\Phi \qquad (2)$$

$$\frac{\partial u_i}{\partial x_i} = 0 \qquad (3)$$

$$\rho\left(\frac{\partial C_k}{\partial t} + (u_i - W_{g_k}\delta_{i3})\frac{\partial C_k}{\partial x_i}\right) = \frac{\partial H_i^*}{\partial x_i} + J_c \qquad (4)$$

where

- t - time;
- u_i - the components of averaged velocity vector in the

directions of Decart axes of system coordinaties;
ρ - density;
P and T - averaged pressure and deflection of air temperature from mean value;
C_K - concentration of admixture component with index K ($K=\overline{1,N}$);
N - number admixture components;
g - free fall acceleration;
L - latent evaporation or condensation heat;
Φ - velocity of liquid phase formation;
C_P - heat capacity of the air at $P=$ const;
W_{gK} - velocity of admixture setting with index K;
$\beta_T = T_0^{-1}$ - thermal coefficiant of the volume expansion;
J_α - members characterizing power of air streams, the sources of heat and pollutant ($\alpha = x, y, z, T, C_K$);
τ_{ij} - symmetric Reynolds stress tensor ($i, j = \overline{1,3}$);
H_i, H_i^* - turbulent heat and admixture flows ($i = \overline{1,3}$);
u'_i, T', c' - pulsation components of velocity, temperature and admixture concentration.

Including of the equation of heat transfer (2) in the system (1)-(4) is necessary to calculate the ventilation of rooms, because of the presence of thermal technological sources in industrial rooms, and also for calculation air conditioning and rooms heating. However, along with the fact that in our tasks the influence of air density changing is essential only in terms of floation force / 7/ we use atmosphere dynamics equations with Bussineck's approximation. Such approach limits the value of full derivative velocity vertical component

$$\frac{du_3}{dt} \ll g$$

and describes the process until convective of heat emission component surpasses the radiant, i.e. overheating of heat emission surface relatively temperature of environment no exceeds 20-25°C.

Reynolds' stresses and heat and admixture turbulent streams we write according to Bussinesk's approach by gradients of averaged values / 6/:

$$\tau_{ij} = -\rho\overline{u'_i u'_j} = \rho\nu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}K\delta_{ij},$$

$$H_i = -\rho\overline{u'_i T'} = \rho\Gamma_t^T\frac{\partial T}{\partial x_i}, \quad H_i^* = -\rho\overline{u'_i c'} = \rho\Gamma_t^c\frac{\partial c}{\partial x_i},$$

where
ν_t, Γ_t^T, Γ_t^c - coefficient of turbulent transfer of impulse, heat and admixture.

Values Γ_t^T and Γ_t^c are connected with ν_t by number Prandtl-Shmidt

$$\Gamma_t^T = \Gamma_t^c = \frac{\nu_t}{\sigma_t}$$

Values σ_T depends on space dimension: for three-dimensional problems $\sigma_T = 0.72$ and for two-dimensional - $\sigma_T = 0.64$.

For description of turbulence two models of clossure were used:
- model of undergrid scale Smagorinsky-Dirdorff / 8/;
- ($K-\varepsilon$)-model of turbulence / 6, 9/.

In the first variant for determining of the ν_t the next correlations are used:

$$\nu_t = \nu_\alpha = (\tilde{K}\Delta)^2 \left[\frac{1}{2} \sum_{i=1}^{3} \sum_{j=1}^{3} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)^2 \right]^{1/2} + \Gamma_0 \quad \text{at } Ri > 1$$

$$\nu_t = \nu_\alpha \cdot (1 - Ri)^{1/2} \quad \text{at } Ri \leq 1$$

where

Δ - the grid scale;
$\tilde{K}$ - the nondimensional grid parameter determined experimentally (0.10 - 0.26);
Γ_0 - the value of background coefficient of turbulence;
Ri - Richardson's number.

With realization the ($K-\varepsilon$)-model a turbulent viscosity coefficient is determined as:

$$\nu_t = C_\mu \frac{K^2}{\varepsilon}$$

where

K - kinetic energy of turbulence ($K = \frac{1}{2}\overline{u_i' u_i'}$);
ε - velocity of its dissipation.

Values K and ε are determined from equations (5)-(7):

$$\frac{\partial K}{\partial t} + u_i \frac{\partial K}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\frac{\nu_t}{\sigma_K} \frac{\partial K}{\partial x_i} \right) + \nu_t \cdot G - \varepsilon \qquad (5)$$

$$\frac{\partial \varepsilon}{\partial t} + u_i \frac{\partial \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\frac{\nu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right) + \left(C_{1\varepsilon} \frac{\nu_t \cdot G}{\varepsilon} - C_{2\varepsilon} \right) \frac{\varepsilon^2}{K} \qquad (6)$$

$$G = \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_i} \right) \cdot \frac{\partial u_i}{\partial x_j} \qquad (7)$$

where the following values for constants of model were assumed:

$C_\mu = 0.09$; $\sigma_K = 1.0$; $\sigma_\varepsilon = 1.3$; $C_{1\varepsilon} = 1.44$; $C_{2\varepsilon} = 1.92$.

SETTING OF INITIAL AND BOUNDARY CONDITIONS AND OF VENTILATING STREAMS.

The set of initial and boundary conditions depends on calculated room and type of problem. Therefore the set is determined specifically with input informations being taken into account at realization of numerical expirements. However, we think that it is possible to make some formalization.

The initial fields of velocity and pressure deviation are set equal to zero due to complex empirical presentation of streams redistribution and then, if necessary, non-stationary process is calculated. At integration equations (5) - (6) and calculation of coefficients turbulent transfer of impulse that

$$K_0 = \mathcal{U}_v^2 \cdot 0.01 \quad , \ \varepsilon_0 = \mathcal{U}_v^3 \cdot 0.001/d_0, \quad \Gamma_0 = \nu_0 \sim \mathcal{U}_v \cdot 0.1,$$

where $\mathcal{U}_v = \{\max(\mathcal{U}_{\Sigma i})\}$, $d_0 = \{\max(d_{\Sigma i})\}\ (i = 1,n)$;

d_Σ - dimensions ventilating windows.

Initial distribution of temperature and concentration along the space are determined as depending on concrete conditions of task.

Conditions of "adhesion" are accepted for air velocity on solid surfaces. We use "soft" condition for temperature

$$\frac{\partial T}{\partial n} = 0,$$

where n - normal to surface

or distribution of temperature over surface. However, such conditions organization requires considerable thickening of grid in front of the surface. When it is impossible, we suggest that we should use "soft" conditions near the wall (they are to be moved on half-step into the region) and describe the velocity in near-wall layer according to universal logarithmical "rule of wall" /10/.

In common case a balance equation is written for each admixture component with calculation experimental coefficients of admixture absorption by the surface, capacity of sources and number of other parametres for making an equation transfer of admixtures as boundary conditions /1, 2/.

The simpliest approach is used for description of pressure on the surfaces

$$\frac{\partial P}{\partial n} = 0.$$

Simple approach is used for turbulent characteristics of ($K - \varepsilon$)-model on solid surfaces:

$$K = 0 \quad \text{and} \quad \varepsilon = 0.$$

To discribe viscous sublayer during calculations with near-surface dense grid, a next approach may be used. This approachs presents "hard" requirements to quickness of computers:

- the first approach is a method of "near-by-wall" functions /10/ according to which a bond is established between velocity values along wall, kinetic energy of turbulence and turbulence dissipation velocity in the nearest-to-surface node of numerical grid with total dynamic velocity;
- the second approach is modified ($K - \varepsilon$)-model at low values of turbulent Reynolds number. Some constants of equati-

on (6) are changed into functions /11/;

- the third approach is method PSL, which allows to calculate the near walls flows more exactly /12/.

Tasking of ventilating streams entering or going out of the room presents strong requirements to initial and boundary conditions, choice of grid and numerical algorithm. The simpliest situation is when the quantity of air entering and coming out in every is known. Numerical algorithm for Dirihle condition ($\mathcal{U} = f_i(\vec{x},t)$ for $\vec{x} \in \Sigma_i$) is examined rather effectively. However, distribution of quantaty of air going out through each opening, is unknown in ventilating windows and openings for a whole number of tasks. We think, that in this case "soft" conditions or more common conditions of the III sort are worth-while to use:

$$\frac{\partial \mathcal{U}_n}{\partial n} + \eta\, \mathcal{U}_n = \xi,$$

where values η and ξ characterize resistances of corresponding ventilating windows or openings. However, the determination of these coefficients and solütion algorithm require further development. We beleive that another version of determinating the velocity field is the one under the condition that pressure at ventilating volume inlet-outlet or difference in pressures at the entrance and discharge of room Σ_i are known.

Dirihle's conditions are used for description of temperature and admixture concentration at inlet ventilating openings

$$T = T_o, \quad C_K = C_{KO} \qquad \text{for } \vec{x} \in \Sigma_{ini}.$$

We think that free "soft" conditions of the II sort are worth-while to use at outlets:

$$\frac{\partial T}{\partial n} = 0, \quad \frac{\partial C_K}{\partial n} = 0 \qquad \text{for } \vec{x} \in \Sigma_{outi}.$$

Free conditions of the II sort are used for turbulent characteristics of ($K-\varepsilon$)-model in ventilating openings as, a rule. Tasking of Dirihle's conditions in inlets is possible to increase effectiveness of ventilating system

$$K = K_1, \quad \varepsilon = \varepsilon_1 \qquad \text{for } \vec{x} \in \Sigma_{ini}.$$

However, task of various kinds of ventilating streams is sufficiently serious question, and it demands individual examination.

We suppose that for conical streams it is, perhaps necessary to use dense grid near air streams and tasking of geometry of cone at sufficient resources of computer (fig. 2,a). However, this variant is very difficult. We think, that for most practical questions we can to use tasking of velocity vectors parameters, determined analytically, and directed along cone (fig. 2,b).

For tasking fan-streams we need to take into account the changed inlet opening angle and its turbulization degree. Modelling of the stream formation by means of directing spades is difficult because initial dimensions of stream are equal to space step of finite-difference grid. The-

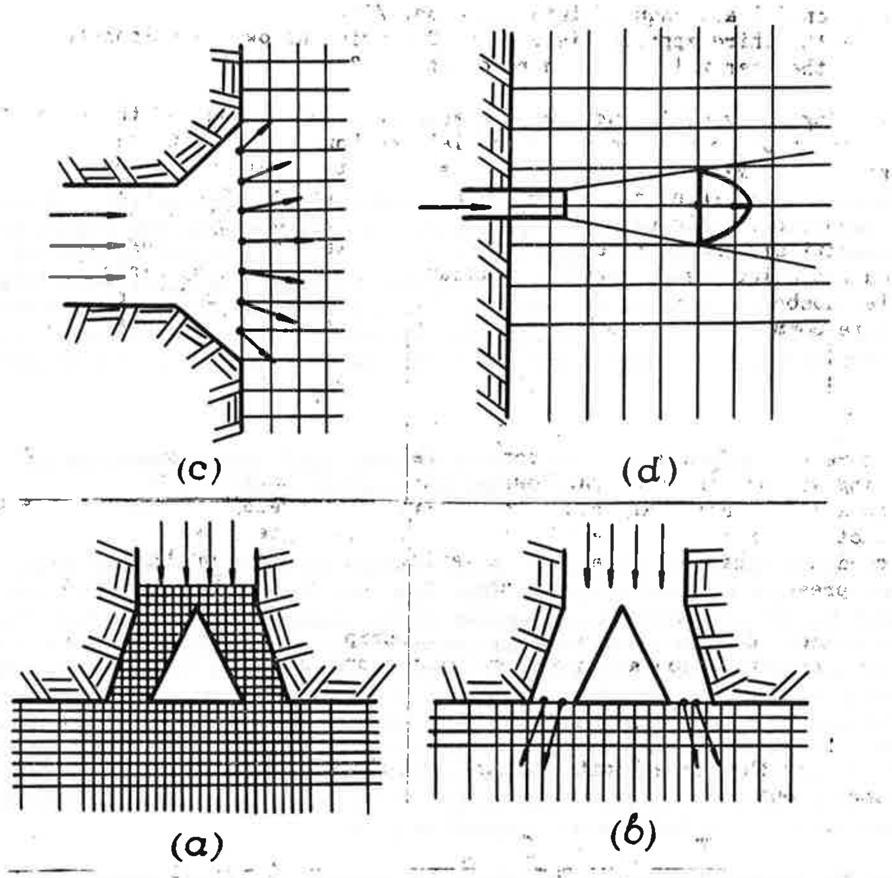

Fig. 2. Possible tasking of ventilating streams: (a) conical stream on small calculation grid taking into account conic geometry; (b) conical stream on more large grid; (c) fan stream; (d) dimensions source are smaller dimensions finite-difference grid.

refore, in this case it is necessary to predict tasking of entering stream in some grid nodes using Dirihle's condition with calculation of different velocity components, with the angle of opening of stream and quantity of entering air being taken into account (fig. 2,c).

In a number of cases release of air is conducted across the system of small openings. Dimensions of opening are less than he dimensions of finite-difference grid. We suppose, that in this case calculation of initial part of stream development is necessary to execute based of analytical relationship between streams development /13,14/, and then to continue the calculation using the model with fictitious source, as shown in fig.2,d. This method allows to calculate exactly the field of flow in the ventilating zone far from the ventilating channel, but it does not describe the flow structure near the chanel. The quantity of air, entering the room, is controlled being dependent on tasking the stream under boundary condition or as a source member of equation (1) (if it is placed inside the room).

REALIZATION OF MODEL

Method of splitting on aero- and thermodynamics equations into physical processes, presented in the papers /2, 15/, was used as the algorithm of numerical solution of equations (1)-(6) with corresponding initial and boundary conditions. We made a programme package "Numerical modelling of chamber aerodynamics and ventilation processes at blasting operatoins and exploitation of diesil-engine equipment" based on the above said. This package was adapted for the PC/AT computers and US-type computers.

EXAMPLES OF CALCULATION OF SOME AERO- AND THERMODYNAMICS AND ROOMS VENTILATION TASKS

1. The velocity of the field is calculated for sufficently big room, like rectangular parallelepiped. The room has one inlet under the ceiling and two outlets which are disposed on the opposite walls near the floor. A variant is considered when 50 cbm/s of air through the opening 6 x 4.5 m, and angles of inclination of resulting vector of velocity form 330° with X-axis, 90° with Y-axis and 240° with Z-axis. 20 cbm/s of air come through the left outlet and 30 cbm/s through the rigth one, in this case. The field of velocity calculated for this conditions, is presented in fig. 3, where the left column presents sections which are parallel to frontal projection plane and in the rigth column are the section which are parallel to horizontal projection plane.

2. We considered the room which is similar to that represented in fig. 3, but it has one additional factor. A thermal source with 10 sq.m area is placed on the room floor strictly in the centre, with 5°C overheating as compared with the temperature of environment. This source (capacity~3000 W) forms rather large vertical air stream, which considerably changes the room aerodynamic (fig. 4). As numerical experiments show, 1°C overheating practically does not cause any changes in the structure of the velocity field.

3. We can achieve a more detailed picture of the velocity field by using dense grid for plane models. The fields of velocity in 40 x 20 m rooms with and without calculation of thermal influence are presented in fig. 5,a and 5,b, respectively. 6 cbm/s of air enter through the inlet which is placed under the ceiling of the room. And 3 cbm/s of it come out through the outlets placed on the opposite sides of the room near the floor. A source with overheating of 5°C is placed strictly in the centre of the room. The air is heated in the working zone of room along the stream and vertical flows are formed over a more considerable area than the area of the source (fig. 5,a). In this picture izolines of temperature deflexion are presented in 12.5 minutes after the start of source operation. The field of velocity in this room, without calculation of thermal influence, is presented in fig. 5,b.

4. The room of more complex shape is presented in fig. 6,a and 6,b:

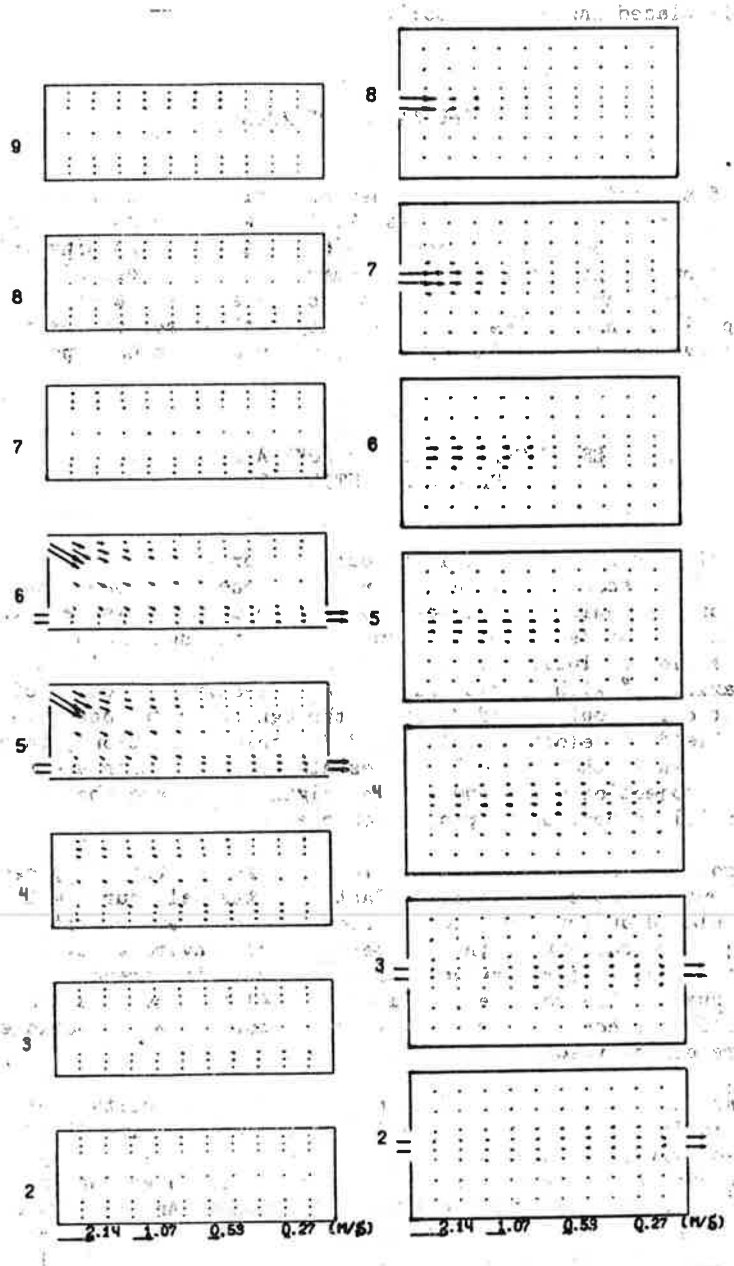

Fig. 3. The field of velocity in room (one inlet and two outlets).

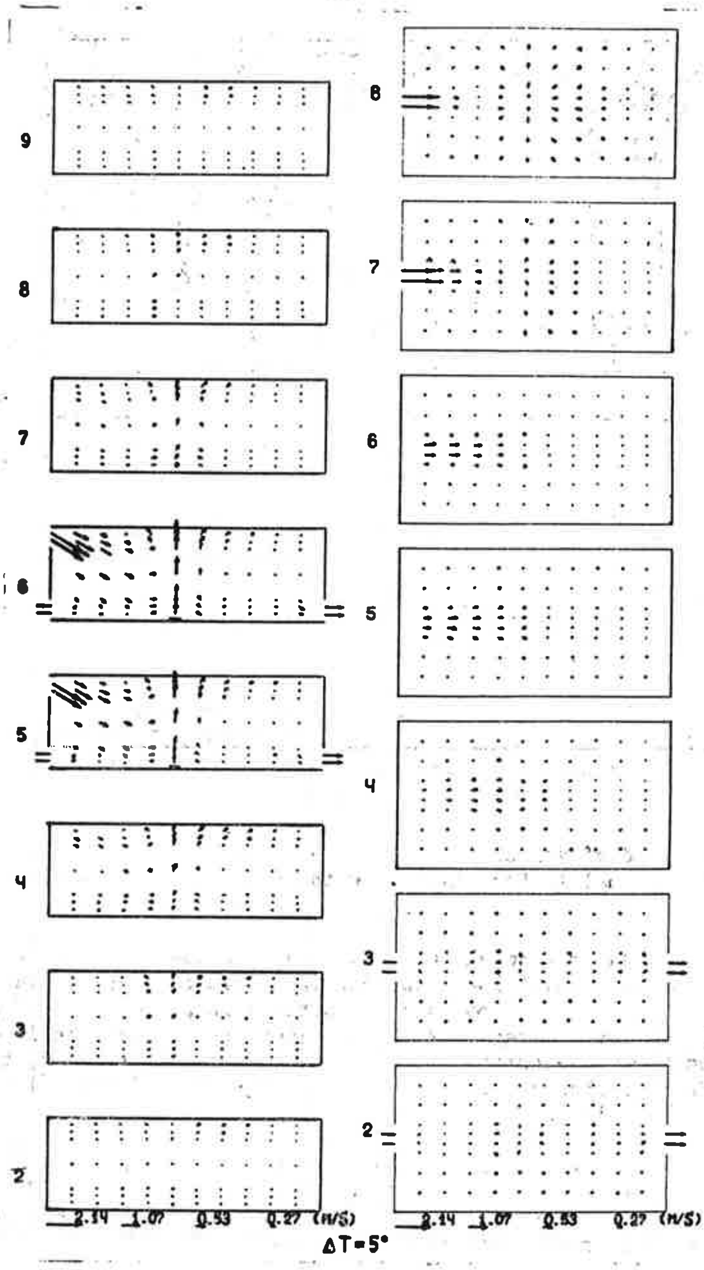

Fig. 4. The field of velocity in room taking into account thermal factor (one inlet and two outlets).

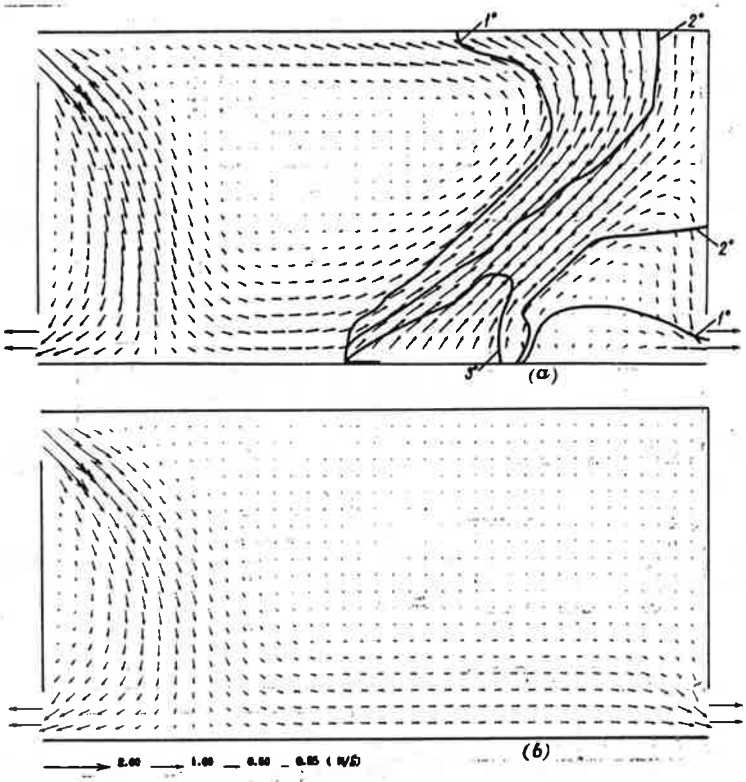

Fig. 5. The field of velocity and izolines of temperature in room taking into account thermal factor (a) and without it (b).

10 cbm/s of air enter through the left upper and lower inlets and it is removed through the right lower outlet. The field of velocity in the room with chosen initial and boundary conditions is presented in fig. 6,a. Four sources of heat placed on the room floor cause considerable changes in the structure of velocity field (fig. 6,b). We want to note, that the most considerable changes of the of velocity field take place in the zones, where the horizontal velocity of the air stream is small.

5. The dynamics of passive admixture distrubution is shown in fig. 7,a, with two units of diesil-engine equipments operating during 400 s (position of equipments is marked by markers in picture of velocity field). 8 and 12 cbm/s of air enter through the inlets (from left and upper), respectively, and it comes out through the outlet on the rigth. The admixture begins to enter in the working zone of the room and to create a situation with exceeding the limited values during a long time. The "instrument" allows to predict a situation, in which a pollution of working zone, described above, may be prevented. We placed an exhaust ventilation device with 6 cbm/s capacity in the zone of one of the pollution source. In this case aerodynamics of room is changed considerably (fig. 7,b) and admixture is localized in the zone of exhaust fan operation.

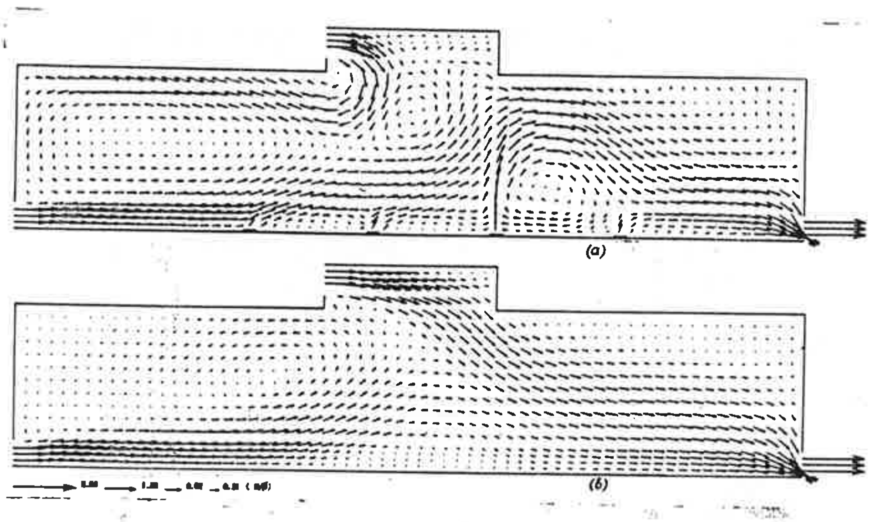

Fig. 6. The field of velocity in room complicated configuration taking into account thermal factor (a) and without it (b).

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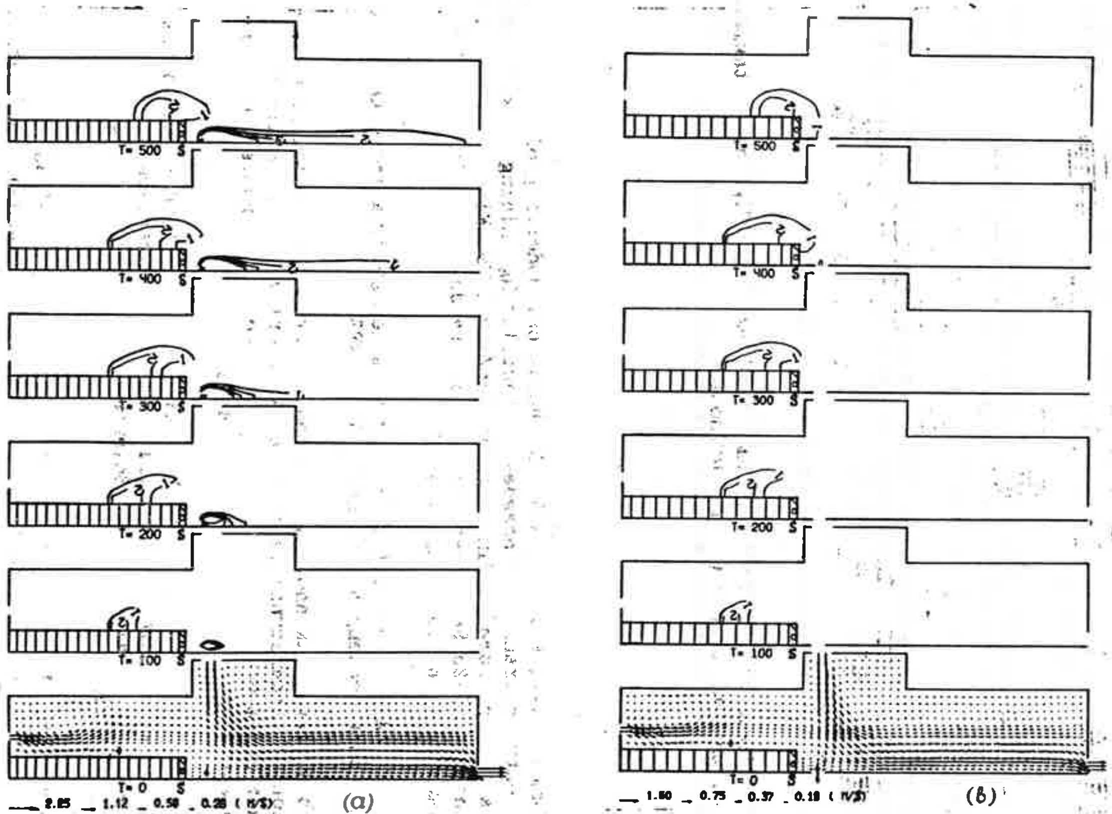

Fig.7. Ventilation dynamics processes at gazosecretion in room: (a) two inlets and one outlet; (b) two inlets and two outlets.

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