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LABORATORY MODELLING OF TANGENTIAL AIR SUPPLY SYSTEM

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Abstract

The international development of air and air conditioning technology has reached a new phase. The reasons for this can be searched in the design of new building structures and the rapid appearance of computer technology. At numerous Universities and research centres in the world efforts are being made so that each element of air technology systems should be measurable even in case of changed parameters. This latter aspect especially applies to one of the highlighted elements of air technology systems, which is the closed space. As it is known the only factor that rates the planned air conditioning technology is the characteristics of the microclimate in the occupied zones of spaces. A long-standing ambition of the designers seems to be accomplished if the spatial distribution of the characteristics of airspeed, temperature and sense of heat can already be defined in the phase of planning.

Our department has joined this international research by undertaking the task of developing a method that supports the design of the occupied zones.

Keywords: Navier-Stokes, tangential air supply system, modelling, k-epsilon.

1. Air Supply System

The impulse of the air led into the space (ventilating air) forces the stationary air in the space to move sensibly and characteristically. This air movement includes partly the air movement resulting from the ventilating air led into the space (primary) and partly the air movement induced by the primary air movement (secondary). These primary and secondary air movements are called the air supply systems by the literature [6], [1], [2].

According to our current knowledge air supply systems are categorised according to the forces generating air movements and air movement directions. According to these aspects the air supply systems can be grouped into two main categories, one is the mixing air system and the other is the displace air system. Mixing air movements are generated by the inertial forces, while the displace air movements are induced by the thermal forces. The above mentioned categorisation can be seen in *Fig. 1*.

This method of categorisation is not autotelic, as a different air supply diffuser belongs to every single air supply system and every single of these can be designed by a different relation. Therefore, the freedom of the designer in deciding what air supply diffuser is to be planned to which airflow is unlimited. His decision will be influenced by – amongst other aspects – the following:

1. What is his aim in respect of the air movement of the space?
2. What is the extent of the heat, humidity and pollution content of the space?

The first mentioned aspect is the direct aim of the air supply systems (e.g.: the elimination or exclusion of the pollution from the occupied zone; release of heat or humidity, keeping the characteristics of warmth index on a pleasant level, etc.). A different air supply system has to be implemented if a relatively high degree of impulse is needed in the occupied zone in order to be able to localise the intensive heat release. In this case jet air supply system can be useful.

The second aspect does not only influence the air movement direction of the ventilating air – what conditions are necessary in order to assure the stable airflow of the occupied zone free of draft and dead zones. The following question should also be considered: which mentioned air supply system would be the solution for the using of the heat generated in the space without the application of a heat recovery?

Air Supply Systems (ASS)

I. Mixing ASS
Driven by impulse
and thermal forces

$\Longrightarrow$ a.) Jet ASS
$\Longrightarrow$ b.) Microclimate ASS
$\Longrightarrow$ **c.) Tangential ASS**
$\Longrightarrow$ d.) d.) Diffuse ASS
$\Longrightarrow$ e.) e.) Extrusive ASS

Displace ASS
Primarly driven by
thermal forces

Fig. 1.

2. International Practice of the Definition of Air Courses

By now it is an indubitable fact that the design of the airflow of the various spaces is not equal with the random selection of air supply diffusers. The air movement within the space primarily depends on the chosen air supply system. Meanwhile, the choice of air supply system depends on the heat load of the space, its architectural characteristics and the aims of ventilation [1]; [2]; [9].

The generated cross effects as a result of the space heat loads can have a determining effect on the air movement of closed areas, which either help or hinder the planned air movement. Therefore, it is useful to define first which air supply systems are to be considered in respect of the given task.

Research dealing with the air movement of closed spaces currently determines three methods for the choice of air supply systems and for the calculation of the generated temperature, speed and concentration fields.

- semi-empirical calculations [6]; [9]; [3]; [13]; [14]
- application of numerical simulations [4]; [12]; [8]
- experiments on three dimensional air models [4]; [8]; [11] [15].

Experiments with the use of three-dimensional models decidedly give the most accurate results in respect of the given task. Compromises and thus negligence can be decreased to the minimum level with the help of this method. Its disadvantage is its high cost demand. The semi-empirical calculation is also popular due to its simplicity. Its disadvantages are its inaccuracy and its not enough elaboration.

The selection of the air supply diffuser to be applied can only be made after the mentioned calculations have been completed, as the type and placement of the air supply diffuser is only the tool of the execution of air supply systems.

Unfortunately, the current practice of planning does not follow the described process yet. Various other aspects (space design, investments costs, etc.) are more highlighted. Moreover, the choice of the air supply diffuser cannot be determined by how it leads the ventilating air into the space free of draft. If the zone being free of draft could be assured by low impulse level of the air led in then the occupied zone is expected to be only partly or not scavenged at all.

Its opposite is also in practice when the so-called slot air supply diffusers are installed near to the occupied zones. Every air supply system has its own equipment for leading in the air. One cannot design a displace air supply system with jet blowers or radial diffuser air courses of flow jet ventilation.

3. Modelling of the Tangential Air Supply System

Various researchers [1]; [3]; [6]; [9] have managed to develop a semi-empirical method, with which the applicable air supply system can be determined with relative accuracy. The method basically uses the results of laboratory modelling and determines a similarity criterion with the application of the theory of similarity. Thus it is enough to determine the value of the similarity criterion in order to decide on the type of air supply system to be implemented in the given case.

From the movement equations describing the air movement of the space an Archimedes number [3]; [5] can be expressed that characterises the given air supply system according to the conditions of conspicuity. The aim of the modelling experiments mentioned in the introduction is that the range of the Ar-value of the types of air supply systems can be determined. Thus the semi-empirical calculations can assure better conditions for the choice and design of air supply systems.

With the help of experiments the air movement, temperature and PMV/PPD distribution of the planned spaces in case of different variations of air supply systems can be analysed. Current experiments aim at one type of air supply systems, the tangential air supply systems.

4. Mathematical Modelling of Closed Areas

The air movements of closed areas are described by the differential equations of continuity and Navier-Stokes. The thermo balance of the areas is expressed by the equation of energy, its distribution of concentration is described by the differential equation of material balance, its thermo comfort can be determined on the basis of the PMV and PPD equations [4]. As we are talking about turbulent air conduction, the proportion of the kinetic energy and the dissipation $(k - \varepsilon)$ of the airflow also have to be determined. This is the mathematical model of the closed areas resulting from this system of equations, which includes all the above listed equations [8]. Assuming an incompressible agent the listed equations are formed as follows:

Continuity:

$$\operatorname{div}\,(\rho \cdot u_i) = 0.$$

Equation of movement:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot u_j) = \frac{\partial}{\partial x_i}\left((\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right) - \frac{\partial}{\partial x_j}\left(p + \frac{2}{3} \cdot \rho \cdot k \cdot \delta_{ij}\right) + g_i(\rho_x - \rho).$$

Equation of energy:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot h) = \frac{\partial}{\partial x_i}\left\{\left(\frac{\mu}{\sigma} + \frac{\mu_t}{\sigma_t}\right)\frac{\partial h}{\partial x_i}\right\} + Q.$$

Concentration of pollution:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot C) = \frac{\partial}{\partial x_i}\left\{\left(\frac{\mu}{\sigma_{cl}} + \frac{\mu_t}{\sigma_{ct}}\right)\frac{\partial C}{\partial x_i}\right\} + C_L \cdot \rho.$$

Turbulent viscosity:

$$\mu_t = K \cdot \rho \cdot \frac{k^2}{\varepsilon}.$$

Turbulent kinetic energy:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot k) = \frac{\partial}{\partial x_i}\left\{\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right\}$$
$$-K_4 \cdot \rho \cdot \varepsilon + \mu_t \frac{\partial u_i}{\partial x_j}\left\{\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right\} + F.$$

Dissipation of the turbulent kinetic energy:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot \varepsilon) = \frac{\partial}{\partial x_i}\left\{\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right\}$$
$$-K_2 \cdot \rho \cdot \frac{\varepsilon^2}{k} + K_1 \cdot \mu_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\frac{\varepsilon}{k} + K_3 \cdot F \cdot \frac{\varepsilon}{k},$$

where:

$$F = g_i\left\{\beta \frac{\mu_t}{\sigma_t}\frac{\partial T}{\partial x_i} + \beta_c \frac{\mu_t}{\sigma_{ct}}\frac{\partial C}{\partial x_i}\right\}.$$

5. Design of the Three-dimensional Air Technology Model

We can get the most accurate characteristics of climate technology in respect of a given case if measurements in the built object are possible. We can get similarly accurate results if the object to be analysed – in this case it is the closed space – is executed according to the rules of modelling and then measurements are carried out on this. In such three-dimensional air models cross effects can be observed, the impact of which on the mathematical model is not known, therefore it cannot be calculated.

In order to analyse the desired phenomenon on a small model we have to create the connection between reality and the changes in the model. As it is known the condition of similarity is the identity of the differential equations and the similarity of the conditions of conspicuity. As the aim is to investigate the distributions within the closed space, integrated equations cannot be applied. The processes in the closed space are described by the differential forms of the transport equations (impulse, weight and material transport) so these are the equations that provide a basis for the modelling.

The data of the closed area to be designed are as follows:

Function	office
Size A^*B^*C	6*8*3
Number of workplaces	8
Air supply system	tangential
Climate parameters	$t = 24\,°\mathrm{C}$, $\varphi = 50\%$
Volume of ventilation	5–15 l/h

We have regarded the workplaces as concentrated sources of heat:

Sitting office worker	120 W
Computer, monitor	250 W
Lighting	30 W
Total	400 W

Because of the glass surfaces on one side of the office a relatively high heat load can be observed. This is replaced by a heat source of convective ventilation in the model as a distributing heat load.

We would expect that the air conducted in through a blow-out slot in the ceiling to the front of the glass surface would scavenge the occupied zone and take up the heat load of the area in such a way that a designing air condition is shaped there. In respect of the stationary cases this process is described by the Navier-Stokes equation as well as the energy and weight balance.

$$\sum_{j=1}^{3}\left(\rho u_j \frac{\partial u_i}{\partial x_j}\right) + \frac{\partial p}{\partial x_i} = \sum_{j=1}^{3} \eta \frac{\partial^2 u_i}{\partial x_j^2} \qquad (i = 1, 2),$$

$$\sum_{j=1}^{3}\left(\rho_0 u_j \frac{\partial u_3}{\partial x_j}\right) + \frac{\partial p}{\partial x_3} = \sum_{j=1}^{3} \eta \frac{\partial^2 u_3}{\partial x_j^2} + \rho_0 g (1 - \beta \Delta T),$$

$$\dot{Q} - \rho \dot{V} \Delta h = 0,$$

$$\operatorname{div}(\rho u) = 0.$$

From the mathematical model we get the criteria of similarity according to the known method:

$$\mathrm{Ar} = g \frac{\beta \Delta T x}{u^2},$$

$$\mathrm{Re} = \frac{x u}{v},$$

where the following conditions are given:

1. there is no impulse transport on the wall surfaces, except for the leading in and out of the air;

2. from the viewpoint of heat technology the wall systems are perfectly insulated, except for the window surfaces;
3. after the starting of the air conditioning system a stationary situation is created;
4. there is no source of weight in the area;
5. we do not aim at the modelling of the speed fluctuation characterising the turbulent flow.

In the course of the model studies the Re-value is increased over the critical level, which ensures the self-modelling range as regards to the Re-value. Therefore the case is the following:

$$\begin{aligned} \mathrm{Re} &> \mathrm{Re}_{\mathrm{critical}}, \\ \mathrm{Ar} &\neq f(\mathrm{Re}). \end{aligned}$$

Considering the conditions described above the connection between reality and the model is created:

$$\mathrm{Ar}_{\mathrm{real}} = \mathrm{Ar}_{\mathrm{model}}.$$

Substituting the Archimedes criterion of similarity, then sorting the equation and applying the linear transformation, the equation of conditions is formed:

$$\begin{aligned} c_u^2 &= c_g c_\beta c_{\Delta T} c_r, \\ c_V &= c_u c_r^2, \\ c_Q &= c_\rho c_u c_r^2 c_C c_{\Delta T}. \end{aligned}$$

The coefficients in the equation of conditions stand for nine physical characteristics (u, g, β, ΔT, c_r, c_V, c_Q, c_p, c_C). Meanwhile there are three equations of conditions available, therefore the degree of freedom of designing is 6. However, when we consider that we primarily want to apply a working agent of air and we do not want to change the field of gravity, there are 4 additional characteristics available. Thus for the planning of the model the geometric and thermal scales are provided the two optional parameters to be chosen.

Considering the characteristics of measurement technology the coefficients of the linear transformation are as follows:

c_r	$c_{\Delta T}$	c_u	c_V	c_Q
0.1	4	0.65	0.0065	0.026

This means that the speed and temperature fields of the closed space can be measured with an acceptable accuracy in the small model of a geometric proportion of 1 : 10 ($c_r = 0.1$). Certainly, the transformation between the model and reality is assured by the coefficients included in the table. The ventilation units serving the model – the ventilator, the surface heater, the calorifier, the filter and the measurement stage – are designed according to this.

Signs

Ar	Archimedes number
Re	Reynolds number
g	acceleration of gravity
h	enthalpy
k	kinetic energy
p	pressure
u	velocity
C	median concentration of the pollution in the air
K	constant
Q	quantity of heat per unit volume
$\dot{Q}$	heat power
T	temperature
$\dot{V}$	air volume
δ	Kronecker delta
ε	dissipation of kinetic energy
η	dynamic viscosity
μ	viscosity
υ	kinetic viscosity
ρ	density
σ	factor

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cl	in laminar flow
ct	in turbulent flow
g	gravity
i	direction
j	direction
r	position
t	factor
u	velocity
C	concentration
Q	heat power
V	air volume
ρ	density
ΔT	temperature difference

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