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Modern Air-Conditioning Systems without Wiring Duct

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Abstract. The calculation involved in the design of integrated heat-hydraulic air-conditioning system in which air is supplied through the raised floor was presented. Calculations were made using the premises of an office building in Yekaterinburg, Russia. Raised floor in the current study was presented in the form of a channel with a concrete wall on one side and chipboard - on the other. It has been successfully established that the loss of pressure during air movement in the raised floor space is minimal, this was suggested as a result of the possible uniformity in air supply to the air distributors. It was also demonstrated that due to transmission losses through the walls of the raised floor, the cooling capacity of the air conditioning system was significantly reduced (up to 15%). Following the results obtained, conclusions were drawn about the main features of such systems.

Keywords: Raised Floors; Displacement Ventilation; Air Distributor.

1. Introduction

The use of air-conditioning systems without ducting in office buildings project have been employed in western countries. This system is gaining prominence in Russia recently, such as the Okhta Center (St. Petersburg), the Moscow-City MIBC, the Waltzing Couple building (Moscow).

The principle of the system relies on the air handling unit supplying air to the ceiling or raised floor space. Air distributors are provided in the raised floor covering, through which air escapes from the raised floor space or suspended ceiling to the room. The structure composition of floor and ceiling are of high density and low air penetration. However, ceilings are more expensive than floors, and such systems are less common. When air is supplied from under the raised floor with a sufficient height of rooms (> 3 m), it becomes possible to use displacement ventilation.

The main distinguishing feature of such systems lies in the lack of channels for air transport from the central unit to the air distributors. The space under the raised floor or behind the suspended ceiling has a sufficiently large cross-sectional area, making air moves in it with relatively low speeds (0.1-0.8 m/s), which results in a minimal pressure loss (0.1 - 0.2 Pa), thereby ensuring uniform air distribution.

2. Subject and methods of research

To estimate the pressure loss, we consider a simplified duct model with a cross section of width $B = 7$ m and height $h = 0.3$ m, channel length $L = 25$ m. The dimensions of the room are taken according to the design of an office building in Yekaterinburg plan of 25 m by 7 m, and the height of the room 4 m. In the upper wall of the duct, which is the floor of the conventional office space, there are holes with air distributors. The number of distributors depends on the number of work places. When calculating, it was assumed that the floor area of the workplace is A_p = 6 m^2. The number of workplaces for the project is n = 30. Specific heat excess for office space with windows oriented to the west and walls of double-glazed windows amount to 90 W/m^2. The air parameters in the room are determined taking into account the air exchange coefficient. Supply air parameters: temperature 18°C, relative humidity 70%; exhaust air parameters: temperature 27°C, relative humidity 40%. In the working area, the air temperature is 23°C; relative humidity of air 50%.

Consumption of air flow is determined by the formula [1]:

$$L = \frac{Q}{c \cdot \rho \cdot (t_s - t_{ex})} \cdot 3600$$

where: Q – excess heat, W;
c – heat capacity of air, equal to 1005 J/(kg °C);
ρ – is the air density equal to 1.2 kg/m^3;
t_s – supply air temperature, °C;
t_{ex} – temperature of exhaust air, °C;

Determine the air flow at 1 workplace:

$$L = \frac{6 \cdot 90}{1005 \cdot 1.2 \cdot (27 - 18)} \cdot 3600 = 180\,\frac{m^3}{h}.$$

We accept that the consumption for 1 workplace equal to 200 m^3/h. The total air flow was 30×200 = 6000 m^3/h. According to the manufacturer, we could determine that for 1 workplace, 2 air distributors with a diameter of 200 mm are needed.

Noise level and air pressure drop are within acceptable limits. In order to estimate the pressure loss using standard aerodynamic calculation techniques, we assume that the channel under the raised floor is a duct with branches. In such case, the air distributors are located 6 in a row, and having a total of 10 rows.

The layout of the air distributors is shown in Figure 1.

The distances between the rows are equal and amount to: 25 / (10 + 1) = 2.27 m. A 10-site duct was obtained with air flow rates from 600 to 6000 m^3/h. The area of the side branch of each section is equal to six times the area of the hole with a diameter of 200 mm. In the calculation, it was assumed that the air flow evenly enters the channel from the entire side surface at an average speed v, m / s, determined by [2]:

$$V = \frac{L}{3600 \cdot f},$$

where f – is the cross-sectional area of the channel, m^2, equal to 2.1 m^2.

$$V = \frac{6000}{3600 \times 2.1} = 0{,}794\,\frac{m}{s}.$$

In that case, if air is supplied to the underfloor through the hole at an average speed of 3-5 m/s, a jet will be generated, that eventually fades out at some distance from the hole. In this case, it can be assumed that the flow rate will gradually be aligned with the width of the box, and the accepted calculation model of the flow will be executed starting only from the third section.

Furthermore, the average air flow rate, pressure loss for each section was determined according to [2]. The results of the calculations are shown in Table. 1, it highlighted that the pressure loss in the raised floor space was 0.2 Pa. However, in some areas, the pressure loss is from 0.002 Pa to 0.04 Pa. The pressure loss in the air distributors according to the manufacturer's nomograms is 20 Pa, which is

hundreds of times greater than the pressure loss in the raised floor space. Pressure losses on the branches are also a small fraction - from 0.4 Pa to 0.85 Pa.

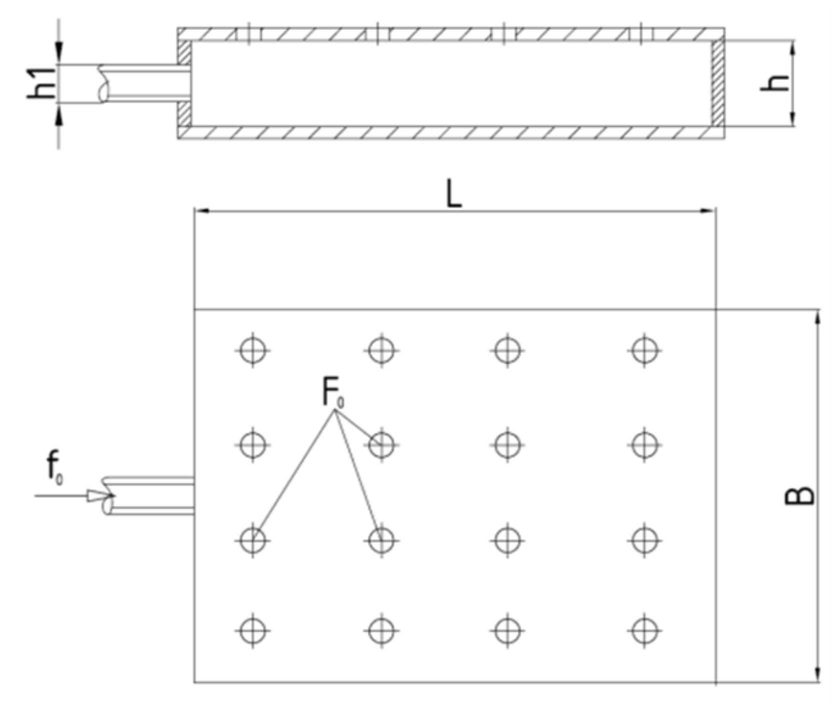

Figure 1. Channel design.

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Therefore, it should be expected that the uniform distribution of air across all air distributors is provided without additional devices, with the help of which pressure losses in networks are connected with ducts.

Table 1. Hydraulic channel calculation (results)

№ study	Air flow, L, m^3/h	Air velocity, v, m/s	Coefficient of local resistance	Pressure loss at the site, Pa	Total pressure loss, Pa	Air temperature, °C
1	600	0.079	0.4	0.002	0.002	21.36
2	1200	0.159	0.1	0.004	0.006	20.79
3	1800	0.238	0.04	0.006	0.013	20.30
4	2400	0.317	0.03	0.01	0.022	19.93
5	3000	0.397	0.02	0.014	0.036	19.58
6	3600	0.476	0.01	0.019	0.055	19.24
7	4200	0.556	0.01	0.025	0.08	18.92
8	4800	0.635	0.01	0.031	0.112	18.60
9	5400	0.714	0.01	0.039	0.15	18.30
10	6000	0.794	0	0.045	0.195	18.00

In previous studies, M.I. Grimitlin [3] provided the recommended ratios of the dimensions of the distribution chamber, the holes and the inlet nozzle, which ensures uniform distribution of air flow through the holes:

$$F_0/f_0 \leq 7; \qquad L/B = 0.5 - 1; \qquad L/h \leq 5; \quad h_1/h > 0.6; \qquad F_0/F < 0.09,$$

where: F_0 – hole area, m^2;
h_1, f_0 – height, m, and the cross-sectional area of the inlet pipe, m^2, respectively;
F – the surface area of the raised floor, m^2;
L, B, h – length, width and height of the underground space, m.

For the adopted model of the duct (Fig. 1), we determined the specified ratios values.

We get $F_0 = 60\ \pi\ (0.22/4) = 1.885\ m^2$, hence the area of the inlet should be no more than $1.885/7 = 0.27\ m^2$. Accordingly, the hole width is $b_0 = 0.27/0.3 = 0.9$ m, the average discharge velocity in the hole is $V = 6000/3600 / 0.27 = 6.17$ m/s; $F = 175\ m^2$. The ratio $L/B = 25/7 = 3.6$ is more than the recommended 3.6 times. The ratio $L / h = 25 / 0.3 = 83.3$ is 16.7 times the maximum. Based on the ratio $h_1/h > 0.6$, the nozzle size should be at least $h_1 = 0.6 \times 0.3 = 0.18$ m. The ratio $F_0/F = 1.885/175 = 0.011$. Based on the results of the analysis, we could conclude that a uniform distribution of air is not provided for all air distributors.

For systems with raised floors, air distributors with highly efficient air mixing should be employed to minimize the volume of the zone with unacceptably high air jet parameters. Air distributors are elements of the floor, so they have a design possessing sufficient mechanical strength for the perception of the design loads on the floor. Some foreign manufacturers offer air distributors which in the design mode of operation limits the dimensions of such a zone within (conditionally) cylinder with a diameter of 30-40 cm and a height of 10 cm.

Manufacturers offer a variety of diffusers in different appearance and form. In the raised floor plates, manufacturer provides openings for air distributors, on order. With discomfort associated with excessive jet parameters which could be experienced in almost any office space, the air distributor can be moved a greater distance from the person by changing the location of the raised floor plates with air distributors. This is also relevant for a free office layout, highlighting the possibility of arranging a personal supply air distributor for each workplace.

When using the air supply from under the raised floor, the basic rules for designing displacement ventilation must be applied [4, 5]. The flow rate of incoming fresh air is not less than the flow rate in convective jets from heat sources in a given room, fresh air consumption - at least person per person [1]. This effect is described theoretically, and the conditions of its existence will be flooding the lower area of the room prepared with relatively cool fresh air. Accordingly, during convective heat transfer, an upward convective air flow arises near the surface of a person, which is fed by fresh air from the lower zone of the room. This will decrease the required amount of outside air while maintaining the same air quality in the human breathing zone and consequently save resources on the treatment of outside air. In this case, the air is removed from the upper zone of the room, where a "heat cushion" will be observed - air with a high temperature and degree of pollution. The need for wiring exhaust air ducts may rely on the room size. Due to the increased temperature difference between the expelled and supply air, air exchange will also decrease due to the apparent warmth. When calculating the air flow, the limitation of the temperature difference over the height of the working area should be considered [6]. There is no simple, unambiguous method for determining the air temperature gradient over the height of a room when air is supplied from the floor. The question of overcooling of the lower area of the room and the possible health consequences remain unclear.

Overlap heat transfer can also result in deterimental effect on the adjustability and controllability of the system, peculiarly on air distributors located in the most distant places from the unit. During condition of low speeds of air movement, the temperature around the concrete slab will be determined by the calculation of [7], using the temperature of the active heat exchange layer of the slab itself.

The stationary mode of heat exchange between the air in the channel and the environment must considered, and the surface density of the heat flux is determined by the formula.

$$q = \frac{t_{in} - t_{ext}}{R_{\text{int}} + \sum_{i=1}^{n} R_i + R_{ext}},$$

where q – is the surface density of the heat flux through the walls of the channel, W;
t_{in}– channel air temperature, °C;
t_{ext} – is the air temperature in the room, °C;
R_{int} – resistance to heat transfer on the inner surface of the channel wall, (m^2•°C)/W;
$\sum_{i=1}^{n} R_i$ – total thermal resistance to conductive heat transfer of the multilayer wall, (m^2•°C)/W;
R_{ext} – resistance to heat transfer on the outer surface of the channel wall, (m^2•°C)/W.

Thermal resistance to heat transfer on the surface of the inner and outer surfaces of the wall, and conductive heat transfer in the channel wall are determined using the formulas, respectively:

$$R_{\text{int}} = \frac{1}{\alpha_{\text{int}}};\quad R_{ext} = \frac{1}{\alpha_{ext}};\quad R_i = \frac{\delta_i}{\lambda_i};$$

where α_{int}, α_{ext} – heat transfer coefficients on the inner and outer surface of the channel wall, respectively W/(m^2•°C);
λ_i – coefficient of thermal conductivity of the material of the i-th layer of the wall, W/(m•°C);
δ_i – the thickness of the i-th layer of the channel wall, m.

The determination of the heat transfer coefficient from air to the walls inside the channel α_{int} depends on the flow regime for each section [4]. The main characterization criterion of the flow regime, is the Reynolds number Re.

$$Re = \frac{wl}{\nu},$$

where w – average flow rate of air obtained from aerodynamic calculation;
v – coefficient of kinematic air viscosity, m^2/c;
l – characteristic channel size, m.

The heat transfer coefficient on the inner surface of the channel is determined using the criterial equation

$$Nu = \frac{\alpha_{\text{int}} l}{\lambda},$$

where Nu – Nusselt number,
λ – air thermal conductivity coefficient, W / (m•°C).

Heat transfer coefficient on the outer surface of the channel α_{ext} to the air of the room was determined by the formula:

$$\alpha_{ext} = \alpha_{con} + \alpha_{rad},$$

where $\alpha_{con} + \alpha_{rad}$ – coefficients of radiant and convective heat transfer on the outer surface of the channel, respectively, W/(m^2•°C), defined by [8].

In accordance to the terms of the applied calculation methods, it was assumed that all the walls of the channel are identical and homogeneous, and the surrounding environment has an equal temperature distribution on all sides.

In the considered example, the upper wall of the channel consists of a chipboard 60 mm thick, the lower wall of concrete 250 mm thick, ambient temperature near the upper wall (raised floor coverings) of 21°C, and an ambient temperature near the bottom wall (under the concrete slab) of 27°C. In order to evaluate the combined effect of various conditions, calculations of heat transfer in channels with different composition walls and ambient temperature were performed. The calculation was performed on a computer during the iterations using an algorithm developed in the MS Excel program.

Calculations showed that the heat flux through the concrete wall amounted to 3730 W, through the wall of chipboard – 1040 W. For the selected office space of 175 m^2 with the specific heat excess of 90 W / m^2, the cooling capacity of the air conditioning system will be calculated at 15 750 W. Consequently, the transmission loss of cold will be 15%. In addition, it follows from the calculations that the temperature of the supplied air in the channel will be increased by almost 3°C.

3. Conclusion

From the result obtained from the calculation of aerodynamic of the canal-raised floor model, it was revealed that unlike duct networks, uniform air supply to all air distributors should be provided without installing additional devices to link pressure losses. Additionally, it was shown that the uniform distribution of air when it is supplied through the raised floor is significantly affected by the ratio of the dimensions of the raised floor, the height and cross-sectional area of the inlet pipe and the area of the air outlets. Furthermore, from the calculation of heat transfer between the air in the channel and the environment, it could be established that without isolated floors, and the use of underground distribution of air according to the principle of displacing ventilation, there will be significant transmission losses of cold and an increase in the temperature of the air supplied.

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