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Research paper

New Approach for Refined Efficiency Estimation of Air Exchange Organization

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Abstract

In the modern conditions, energy efficiency is one of the most important world problems. One of the important factors influencing the overall energy efficiency of buildings is air distribution in rooms. Literature review shows different options of efficiency estimation of air exchange with significant limitations. Some of them have non-obvious physical meaning. Certain of them do not take into account possible room zoning. Consideration of turbulence intensity in the efficiency estimation was not found. In this work, we propose an approach to estimate the air exchange efficiency in different kind of rooms. It is a relation of minimum room demands and inlet air potentials. Additional definitions are introduced for the parameters and demands estimation. The mechanical energy of the air is used for the estimation, which includes the energy of averaged motion and turbulent pulsation. Special approach is offered for turbulence intensity computation for energy calculations. The example of efficiency estimation of air exchange in a museum room with constant air volume system of air conditioning is solved.

Keywords: *air conditioning; air exchange; energy efficiency; turbulence intensity; ventilation*.

1. Introduction

Efficiency of air distribution in rooms is one of the influencing factors on overall energy efficiency of building. There are different approaches to definition of air distribution efficiency in rooms. They take into account different factors and criteria, but the air distribution development and tightening of energy efficiency requirements cause the necessity of defining an improved energy efficiency indicator for rooms with single and multiple occupied (working) zones. There are different meanings of "zone": a part of a room, a ventilated room, a group of ventilated rooms with the same parameters and requirements, etc. For disambiguation, in this work, "zone" always means a part of a room with the same requirements for air parameters or without any requirements.

2. Literature Review

There are different approaches for estimation of air exchange efficiency. In post-socialist countries, the approach of N. Sorokin, further developed by M. Grimitlin and G. Pozin [1, 2] is most used. It is based on air exchange factor K_L. The air exchange factor K_L is a temperature simplex that connects the temperature [°C] of leaving (exhaust) air t_ℓ, inlet air t_{in} and occupied zone t_{wz}:

$$K_L = \frac{t_\ell - t_{in}}{t_{wz} - t_{in}}. \tag{1}$$

By the same principle, the coefficient of air exchange for enthalpy, moisture content, concentration of gases, etc. was written. M. Grimitlin interprets it as a factor of inlet air usage. Room with amount of air exchange G [kg/h] at isobaric specific heat of air c_p [kJ/(kg·K)] has sensible heat gains $\Delta Q = c_p\, G\, (t_\ell - t_{in})$ [kJ/h]. If we accept sensible thermal assimilation potential of inlet air as $\Delta Q_{in} = c_p\, G\, (t_{wz} - t_{in})$ [kJ/h], the equation (1) will show the efficiency of air exchange $K_L = \Delta Q / \Delta Q_{in}$. For multi-zone rooms, the average temperatures by occupied zones are used in the equation (1).

The amount of air exchange

$$G = \frac{Q}{c_p K_L \left(t_{wz} - t_{in}\right)} \tag{2}$$

may be approximately equal at high factor K_L and low temperature difference and vice versa [2]. Therefore, the factor is more a measure of temperature distribution uniformity rather than air exchange efficiency. In some rooms, such as certain museum halls, the uniformity is normed. Therefore, for such rooms, the factor is a limitation, but not an efficiency indicator. G. Pozin [2] proposed using temperature difference between an occupied zone and inlet air (so-called "working temperature drop") $t_{wz} - t_{in}$ in conjunction with K_L for estimation of air exchange efficiency.

Air parameters distribution in a working zone may be too non-uniform, which cause the necessity for significant increase/decrease of the average temperature for achieving the normative parameter values in the worse point(s). Therefore, the efficiency cannot be determined from simple balance equations.

Equation (1) is used for the vertical distribution of air parameters. There is no wide experience of its use for piston ventilation at the horizontal direction of indoor air movement.

European norms [3] use the same factor ε_v by CO_2 concentration C [mg/m^3]:

$$\varepsilon_v = \frac{C_\ell - C_{in}}{C_{wz} - C_{in}} . \quad (3)$$

It is because the most popular solution in the EU countries is a combination of room cooling for heat gain compensation with ventilation for CO_2 elimination only by external air with minimum treatment. This criterion does not take into account the uneven concentration distribution by zones.

USA standards use the indoor air quality IAQ (Indoor Air Quality) mark and energy consumption for the ventilation efficiency comparison [4]. This allows system comparison that provides better or poorer air quality. Often a ventilation or air conditioning system can provide better quality consuming more energy. Not well-designed systems cannot provide good microclimate conditions in complete maintained (working) zone(s) due to very uneven parameter distribution. Therefore, some additional restrictions are necessary for efficiency calculations and comparison.

The USA norm [5] provides an approach for estimations of whole ventilation system based on "zone ventilation efficiency" $E_{VZ} = 1 + X_s - Z_{pz}$, where X_s – "the average outdoor air fraction for the system", which works for multiple rooms, – and Z_{pz} – "the primary outdoor air fraction for the zone". The last value is dependent on "Zone Air Distribution Effectiveness" E_z at different options of "Air Distribution Configuration".

The physical meaning of E_z is an estimation of air share passed from air distributors to target zones. Today, there are too many different air distribution possibilities, such as [6-8], which require high amount of reference data. The data may be regularly widen because of the air distribution development. E_z can be calculated using standard jet calculation procedure minimizing unnecessary air losses during jet development. It is possible to use the author's approach based on the theory of turbulent flows by A. Tkachuk [9-11]. The factor cannot take into account other factors than CO_2 concentration. A jet injects ambient air losing its heat, moisture and other potentials, which cannot be estimated using the factor.

Another method for estimating the efficiency of air exchange is the concept of air age, proposed by N. Brelih [12]. The average age of air at a point in a room is the average time spent in the room by the air molecules that hit the point. Near exhaust devices, the average age of air [hours] is a reciprocal of air change per hour (ACPH or ACH). The efficiency of the air exchange ε_a [%] is defined as the ratio of the minimum possible time for replacing the air in the room τ_n to the actual air change time τ_r, equal to twice the average age of the air in the room τ:

$$\varepsilon_a = 100 \cdot \frac{\tau_n}{\tau_r} = 100 \cdot \frac{\tau_n}{2\tau_r} . \quad (4)$$

Air age is the ability estimation of the ventilation to dilute CO_2 and other gases to threshold limit value. Direct estimation of the influence of ventilation on heat and moisture is impossible. The physical meaning of this efficiency estimation is not obvious.

The detailed analysis of air exchange efficiency was performed in work [13]. Other efficiency indicators shown in the work have the same limitations.

As a result, it is necessary to create a factor for estimating the efficiency of air distribution that takes into account the majority of factors with obvious physical meaning. To do this, it is necessary to determine inlet air potentials and room demands.

3. Goal of the Work

The goal of this research is defining a refined method for efficiency estimation of air exchange organization that takes into account the maximum number of influencing factors and modern energy efficiency requirements.

4. Physical Processes in a Ventilated or Air Conditioned Room

There are two main types of zones:

- **occupied zones** – occupied zones with parameters, defined by norms, technology requirements or a customer desire;
- **zones without requirements**.

Ventilated rooms by number of zones can be classified as:

- **single-zone**, such as museum rooms that have historical painting on walls and ceiling;
- **dual-zone** (working zone and upper zone without any requirements), such as most of one-level residential, office, enterprise, and agricultural (hot-houses) rooms;
- **multi-zonal** (with at least two zones with air parameter requirements), such as museums with parts for exhibition keeping and for people, atrium buildings etc. If a room has only two occupied zones and no zones without requirements, it will be classified as multi-zonal.

Inlet air can have common parameters for all air distributors or different parameters in different groups of air distributors. It contains some potentials of full and sensible heat (enthalpy/temperature difference), moisture content, CO_2 content etc.

The inlet air can be supplied by jets, which mix ambient air and loose the potentials (mixing ventilation). The air can be supplied with very low velocity and potentials directly to the target zones (displacement ventilation).

The potentials of inlet air are used for assimilation or heat supply, moisture, odorants, CO_2 dilution etc. in occupied zones (useful effect) and zones without requirements (useless effect). This shows that only occupied zones may be taken into account for useful effect estimation of air exchange organization.

The assimilation or supply process consists of two parts:

- influence of the inlet air potential (from inlet air parameters to average parameters in the occupied zones);
- influence of internal room resources on indoor air parameters (from the occupied zones parameters to leaving air parameters).

For supplied energy estimation, only the influence of the inlet air potential may be used.

For unified description, the processes of assimilation or supply of heat, moisture, CO_2 dilution etc. common definitions and formulas may be used, which allow similarity of efficiency criterion, such as for K_L.

Let us use a restriction for the task solution. It will be assumed, that all ventilation and air conditioning systems in a comparison provide required microclimate conditions in all occupied zones. If not, such systems will be rejected. In other case, the task will be incorrect because of unequal conditions.

The following definitions and calculations will be performed primarily for multi-zonal rooms. They can be easily applied for single-zone or dual-zone rooms.

5. Basic Definitions

Each parameter of the air environment X (enthalpy $X = I$ [kJ/kg], moisture content $X = d$ [g/kg], air temperature $X = t$ [°C], gas concentration $X = q$ [g/kg], averaged time of air velocity v [m/s] etc.) has its normative range. With most of X parameters (except the air velocity), there is gain or lack of a certain "substance" ΔS [(kg/h)·(dimension of X)] (correspondingly, full heat $\Delta S = \Delta Q_{hf}$ [kJ/h], moisture $\Delta S = W$ [g/h], sensible heat $\Delta S = \Delta Q / c_p$ [kg·°C/h], mass of the corresponding gas M [g/h] etc.). In these terms, required air exchange can be calculated by the following common equation [kg/h], which unifies formulas in [14]:

$$G_{\Delta S} = G_{wz} + \frac{\Delta S - G_{wz}\left(X_{wz} - X_{in}\right)}{X_\ell - X_{in}} =$$

$$= G_{wz} + \frac{\Delta S - G_{wz}\left(\Delta X_{in} - \Delta X_{wz}\right)}{\Delta X_{in} - \Delta X_{\ell}}, \qquad (5)$$

where G_{wz} is flow rate [kg/h] of local suctions in the working zone; $\Delta X = \bar{X} - X$ is an air (inlet – *in*; working zone – *wz*; leaving – ℓ) potential by the parameter *X*, which is the difference between some basic value $\bar{X}$ (usually inside the room) and its value *X* in inlet or leaving air. The last part of the equation (4) is useful for calculation of air exchange efficiency.

The basic parameter value $\bar{X}$ may be chosen in such a way as to ensure that the potential of the inlet air shows the influence of inlet air only. For this, new definitions are necessary.

Saving air parameter value X_S in a zone is a value of the parameter that corresponds to minimum expenses for inlet air treatment and minimum room demands in the range of normative or other requirements. In heating period, in most of the cases, this is the minimum temperature and moisture content. In cooling period, usually, it is the maximum temperature and the maximum moisture content. At such parameters the demands in rooms will be minimal as well as the cost for treatment of the inlet air.

Effective gain (deficit) of "substance" ΔS_e is the sum of surpluses (deficiencies) of the "substance" in occupied zones at saving air parameters.

Saving average weighted air parameter of a room $\bar{X}_S$ is an average weighted saving air parameter in occupied zones [dimension of *X*]:

$$\bar{X}_S = \frac{\sum_{i=1}^{n} X_{S,i} V_i}{\sum_{i=1}^{n} V_i}. \qquad (6)$$

Mixing of air in occupied zones causes the parameter *X* to be equal to $\bar{X}$. This causes using the parameter value as basic.

Effective (absolute) potential of inlet air by parameter X is the difference (difference module) of the save weighted average air parameter of the room and the parameter of the inlet air [dimension of *X*]:

$$\Delta X_{in,e,j} = \bar{X}_S - X_{in}, \ \Delta X_{in,ea,j} = \left|\bar{X}_S - X_{in}\right|, \qquad (7)$$

where *j* – number of the group of air distributors that supplies the same air parameters. The difference may be negative if there is supply of the corresponding "substance".

The effective (absolute) potential can estimate the influence of inlet air on parameters of the internal air in occupied zones. The same principle gives the potential of an outlet air, which indicates the internal influences in the room [dimension of *X*]:

$$\Delta X_{\ell,e,j} = \bar{X}_S - X_{\ell}, \ \Delta X_{\ell,ea,j} = \left|\bar{X}_S - X_{\ell}\right|. \qquad (8)$$

The total effective (absolute) potential $\Delta X_{in,e,j}$ ($\Delta X_{in,ea,j}$) of the inlet air by parameter X is the effective (absolute) potential of the inlet air by the parameter *X*, weighted by the flow rate $G_{in,j}$ [kg/h] of the inlet air [dimension of *X*]:

$$Y_{G,e} = \sum_{j=1}^{m} G_{in,j} \Delta X_{in,e,j}, \ Y_{G,ea} = \sum_{j=1}^{m} G_{in,j} \Delta X_{in,ea,j}. \qquad (9)$$

The average effective (absolute) potential $\Delta X_{in,e,j}$ ($\Delta X_{in,ea,j}$) of the inlet air by parameter X is the effective (absolute) potential of an inlet air by the parameter *X*, the weighted average by flow rate of the inlet air [dimension of *X*]:

$$\Delta X_{in,e,j} = \frac{\sum_{j=1}^{m} G_{in,j} \Delta X_{in,e,j}}{\sum_{j=1}^{m} G_{in,j}} = \frac{Y_{G,e}}{\Sigma G_{in}},$$

$$\Delta X_{in,ea,j} = \frac{\sum_{j=1}^{m} G_{in,j} \Delta X_{in,ea,j}}{\sum_{j=1}^{m} G_{in,j}} = \frac{Y_{G,ea}}{\Sigma G_{in}}, \qquad (10)$$

where ΣG_{in} – total inlet airflow [kg/h].

The same definitions can be introduced for leaving air replacing "*in*" by "ℓ".

The equations (10) are valid for both modes of operation of ventilation and air conditioning:

- mode of compensation of the lack of parameter $\Delta S < 0$;
- mode of assimilation of the gains of parameter $\Delta S > 0$.

6. Efficiency Definition of Air Exchange Organization

Now it is possible to define the efficiency of air exchange organization. It is relation of numerical representation of the useful effect per numerical quantity of the supplied resources in the same dimension. The useful effect is assimilation of gains or compensation of lack of the "substance" ΔS [(kg/h)·(dimension of *X*)]. The corresponding resource is general effective (absolute) potential of the inlet air by the parameter *X*: $\Sigma X_{G,e}$ or $\Sigma X_{G,ea}$ [(kg/h)·(dimension of *X*)].

The efficiency of air exchange organization by "substance" S or by parameter X is the ratio of gains or lack of the "substance" ΔS to the general effective (absolute) potential of the inlet air by the parameter *X:*

$$\varepsilon_{\Delta S} = \frac{\Delta S_e}{Y_{G,e}}, \ \varepsilon_{\Delta S} = \frac{\left|\Delta S_e\right|}{Y_{G,ea}}. \qquad (11)$$

The equation (11) takes into account zoning of internal space. The average effective potentials by the equation (9) include the effective potential of inlet air (7) above the air parameters $\bar{X}_S$ averaged by occupied zones. In general, the equation (11) cannot be easy converted to a parameter simplex because it uses the actual air exchange rate and saving air parameters.

It is easy to calculate this efficiency by the equation (11) for heat (full and sensible), moisture, gas concentration etc. Nevertheless, air velocity and turbulence intensity, which are microclimate parameters [14], cannot be estimated directly. It is possible to take into account the parameters by mechanical air energy.

7. Mechanical Energy as an Air Parameter

Mechanical energy in premises or flows has two forms:

- energy of averaged flow;
- energy of turbulent pulsations (zero in laminar flows).

The energy of averaged flow during the air motion converts into energy of turbulent pulsations, the last one dissipates to sensible heat. The heat in rooms causes very little temperature change, so it is considered as energy losses.

Energy losses of a room are the sum of

- energy losses in the room due to air circulation, resistances of internal surfaces, indoor obstacles etc.;
- energy losses with exhaust air.

If the energy losses in the room is significantly greater than the losses with exhaust air (below 10 – 20 % of the total losses) – the ***air exchange is slow***. If both losses are comparable – the ***air ex-***

change is moderate. If the losses with exhaust air prevail over the losses in the room (below 10 – 20 % of the total losses), the ***air exchange is quick***. In the last case, only the mechanical energy of the exhaust air is completely lost in the room.
Mechanical energy gains are the mechanical energy of motion and convective flows E_+ [kJ/h].
Mechanical energy of air in occupied zones with volumes V_i [m^3] and total volume ΣV_i [m^3] of the occupied zones [J]

$$E_m = \int_{\Sigma V_i} \left(\frac{\rho v^2}{2} + \frac{\rho v'^2_E}{2} \right) dV = \int_{\Sigma V_i} \frac{\rho v^2}{2} \left(1 + Tu_E{}^2\right) dV \approx$$

$$\approx \sum_{i=1}^{n} \frac{\rho \overline{v}_i^2}{2} + \frac{\rho \overline{v'}_E^2}{2} = \sum_{i=1}^{n} \frac{\rho \overline{v}_i^2}{2} \left(1 + \overline{Tu}_{E,i}^2\right) V_i \ , \qquad (12)$$

where over-line shows average value in the volume of the zone; ρ – air density [kg/m^3]; v – velocity of averaged air motion [m/s]; v'_E – velocity averaged time of air motion pulsation for energy calculations (averaging rules will be considered in the following chapter) [m/s]; $Tu_E = v'_E / v$ – turbulence intensity for energy calculations. Effective energy gain $E_{m,e}$ is calculated by the equation (12) using minimum permitted velocity(ies) and turbulence intensity. The first variant of direct representation of turbulent component is required for the case of near-to-zero averaged velocity at piston or displacement ventilation in comparison to turbulent pulsations due to internal influences.
The specific mechanical energy of air in mass m [kg] in occupied zones is equal to energy losses per one hour of air exchange [J/h]

$$E_- = K_e E_m = K_e \int_{\Sigma V_i} \frac{\rho v^2}{2} \left(1 + Tu_E{}^2\right) dV \approx$$

$$\approx K_e \sum_{i=1}^{n} \frac{\rho \overline{v}_i^2}{2} \left(1 + \overline{Tu}_{E,i}^2\right) V_i \ , \qquad (13)$$

where K_e – energy change per hour [h^{-1}] or energy change rate (ECPH or ECH). In the case of slow air exchange, this is a characteristic of a room equal to one divided by the time of complete energy damping at the complete stopping of ventilation and all sources of mechanical energy. At fast air exchange, it is (normative) air change per hour (ACPH or ACH) for total room volume V [m^3]: $K_e = K = L / V$ [h^{-1}], where L – air exchange [m^3/h]. In this case [J/h],

$$E_- = L \int_V \frac{\rho v^2}{2} \left(1 + Tu_E{}^2\right) \frac{dV}{V} \approx L \sum_{i=1}^{n} \frac{\rho v_i^2}{2} \left(1 + Tu_{E,i}^2\right) \frac{V_i}{V} \ . \qquad (14)$$

Mechanical energy by its nature differs significantly from heat and moisture. Its assimilation is possible only as an exception and only in the form of turbulent pulsations. In this case, special designed obstacles must mechanically destroy formed airflows. The potential of the inlet air over the energy of the internal air, in general, can be zero (infinite efficiency). Therefore, it is most correct to calculate the efficiency of mechanical energy not by gains, but by the need for energy. The energy of the inlet jets may be accepted by absolute value and not by the potential over the energy of the zones.
In the presence of internal sources of mechanical energy, its need [J/h]

$$\Delta E = E_- - E_+ \ . \qquad (15)$$

At the absence of internal sources of mechanical energy, the energy need is always positive (energy deficit) [J/h]

$$\Delta E = E_- \ . \qquad (16)$$

Amount of air energy after air distributors (without the member in square brackets) and immediately before air distributors with pressure loss Δp_j [Pa] (with the member in square brackets) [J/kg]

$$e_{in} = \frac{1}{\rho} \cdot \left(\frac{\rho v_j^2}{2} \cdot \left(1 + Tu_{E,j}^2\right) + \left[\Delta P_j\right] \right) = \frac{v_j^2}{2} \left(1 + Tu_{E,j}^2\right) + \left[\frac{\Delta P_j}{\rho}\right] , \qquad (17)$$

where v_j and $Tu_{E,j}$ – the velocity and energy turbulence intensity at the beginning of the flow from the air distributor group j.
Efficiency of organization of air exchange by mechanical energy

$$\varepsilon_m = \frac{\Delta E}{\sum_{j=1}^{m} G_{in,j} e_{G,e,j}} = \frac{K_e \sum_{i=1}^{n} \frac{\rho \overline{v}_i^2}{2} \left(1 + \overline{Tu}_{E,i}^2\right) V_i - E_+}{\sum_{j=1}^{m} G_{in,j} \frac{v_j^2}{2} \left(1 + Tu_{E,j}^2\right) + \left[\frac{\Delta P_j}{\rho}\right]} \ . \qquad (18)$$

The equation (18) shows the efficiency raise with decreasing of the inlet velocity v_j. It shows that the most efficient air exchange organization is piston or displacement ventilation with low potentials of inlet air from false floors, walls or ceilings, low velocity panels or other air distributors directly to the occupied zones.

8. Turbulence Intensity Definition for Mechanical Energy Calculations

The instantaneous pulsation velocity [15] $\dot{v}' = \dot{v} - v$ [m/s] is the difference between average velocity v [m/s] and instantaneous velocity $\dot{v}$ [m/s]. For energy calculations, this velocity should be averaged. The law of energy conservation [J/m^3]:

$$\frac{\int_t \frac{\rho \dot{v}^2}{2} dt}{t} = \frac{\rho v^2}{2} + \frac{\rho v'_E{}^2}{2} \ , \qquad (19)$$

where t is a time of averaging [s]; $\int_t \left(\rho \dot{v}^2 / 2\right) dt / t$ – total energy of motion [J/m^3]; $\rho v^2/2$ – energy of the averaged motion [J/m^3]; $\rho v'_E{}^2/2$ – energy of the pulsating motion [J/m^3].
After transformations taking into account that the average (in time) velocity v [m/s] is a constant in time [m/s]:

$$v'_E = \sqrt{\frac{\int_t \left(\dot{v}^2 - v^2\right) dt}{t}} = \sqrt{\overline{\overline{\left(\dot{v}^2 - v^2\right)}}} \ . \qquad (20)$$

The double over-line shows averaging by time.
The turbulence intensity:

$$Tu_E = \frac{v'_E}{v} = \frac{\sqrt{\overline{\overline{\left(\dot{v}^2 - v^2\right)}}}}{v} \ . \qquad (21)$$

By root mean square (RMS) deviation, that is usually used for turbulence measuring, the pulsation velocity [m/s] and turbulence intensity:

$$v'_{RMS} = \sqrt{\frac{\int \left(\dot{v} - v\right)^2 dt}{t}} = \sqrt{\overline{\overline{\left(\dot{v} - v\right)^2}}} \ ; \qquad (22)$$

$$Tu = \frac{v'_{RMS}}{v} = \frac{\sqrt{\overline{\overline{\left(\dot{v} - v\right)^2}}}}{v} \ . \qquad (23)$$

The equations (21) and (23) give different results.

Relative difference of pulsation velocity if the instantaneous and average velocities are close to each other:

$$\frac{v'^{2}_{RMS}-v'^{2}_{E}}{v'^{2}_{E}} \approx \frac{\left(\dot{v}-v\right)^2-\left(\dot{v}^2-v^2\right)}{\dot{v}^2-v^2} = \frac{2v}{\dot{v}+v} \approx 1 . \qquad (24)$$

The equation (20) gives:

$$v'_E \approx \frac{v'_{RMS}}{\sqrt{2}}; \quad Tu_E \approx \frac{Tu_{RMS}}{\sqrt{2}} . \qquad (25)$$

For example, at typical value of $Tu_{RMS} = 40$ % by the equation (25) $Tu_E \approx 28.3$ %. This shows unreasonableness using the turbulence intensity by RMS deviation for mechanical energy calculations.

9. Air Velocity Saving and Turbulence Intensity Value for Mechanical Energy Calculations

The most problematic is the definition of minimum (saving) velocity in rooms. Building norms, such as [16], require only maximum velocity dependent on turbulence intensity. Formally, minimum normative velocity can be zero, which means no air exchange at least in some internal volumes. The minimum velocity avoiding air blanketing was not actual for researches in past because properly designed air exchange organization at mixing ventilation always achieves it. At piston or displacement ventilation, extremal low velocity can be provided. Very low velocity can be achieved at air supply using all floor or ceiling surface (correspondingly, from bottom to top or from top to bottom). For a room of size $a \times b \times h$ [m], air change per hour K [h^{-1}] at air velocity v [m/s], the flow rate will be $L = K\,a\,b\,h$ [m^3/h]. The velocity is [m/s]

$$v = \frac{L}{3600ab} = \frac{Kabh}{3600ab} = \frac{Kh}{3600} .$$

At $K = 10\ h^{-1}$ and $h = 10.8$ m, $v = 0.03$ m/s ≈ 0. It is less or comparable to the uncertainty of velocity measures by any modern anemometer. At lower air change or high, the velocity will be lower than threshold of sensitivity of all measuring devices. This short example says about possibility of infinite velocity minimization using piston or displacement ventilation. The velocity in a room will be uncontrolled. It will be defined only by internal convective flows above heat sources or movement of people or mechanisms. Therefore, using piston or displacement ventilation, the minimum air velocity is necessary. It can be found from the fig. 5.3.3A in the USA standard [17]. The minimum velocity in comfort zone is 0.15 m/s and in “still air comfort zone” – 0.1 m/s. Saving turbulence intensity can be accepted by minimum turbulence intensity [14] – $Tu = 10$ %. For energy calculations

$$Tu_E \approx \frac{10}{\sqrt{2}} = 5\sqrt{2} = 7.07 \ \%.$$

10. Example

Let us consider a museum room with

- area $A = 50\ m^2$;
- high $h = 5$ m;
- volume $V = 250\ m^3$;
- normative air change per hour $K = 6\ h^{-1}$;
- full heat gains (to working zone) $\Delta Q_{hf} = 1000$ W;
- humidity gain (to working zone) $\Delta W = 750$ g/h);
- atmospheric pressure $P = 101325$ Pa.

The room has the following zones:

- occupied zones:
  - working zone WZ, usually occupied by visitors and staff (area $A_{wz} = 45\ m^2$; high $h_{wz} = 2$ m; volume $V_{wz} = 90\ m^3$; temperature $t_{wz} = 20...24$ °C, relative humidity $\varphi_{wz} = 25...60$ %, $v \geq 0.1$ m/s);
  - keeping zone KZ with exhibition (area $A_{kz} = 5\ m^2$; high $h_{wz} = 3$ m; volume $V_{kz} = 15\ m^3$; temperature $t_{wz} = 17...19$ °C, relative humidity $\varphi_{wz} = 45...55$ %, $v \geq 0.1$ m/s);
- zone without requirements:
  - top zone ℓ without any exhibition or people (no normative parameters).

The task is to check three options of constant air volume (CAV) air conditioning:

- displacement ventilation with inlet velocity into both zones 0.2 m/s, $Tu = 10$ %;
- mixing ventilation with inlet velocity of 2 m/s, $Tu = 20$ % (grates);
- mixing ventilation with inlet velocity of 5 m/s, $Tu = 50$ % (air diffusers with twirlers).

The turbulence intensity is accepted only for the example. It may be in developer’s datasheets. Minimum mechanical energy gains to the room may occur at absence of people, which can be accepted as the critical conditions for energy calculations.

Solution:

1. The air exchange is accepted as quick.
2. Calculation of mechanical energy demand in the room by the equation (13) and (14) at minimum turbulence intensity 10 %:

$$\Delta E = 6 \cdot \left(\frac{1.2 \cdot 0.1^2}{2} \cdot \left(1 + \left(\frac{0.1}{\sqrt{2}} \right)^2 \right) \cdot 45 + \right.$$

$$\left. + \frac{1.2 \cdot 0.1^2}{2} \cdot \left(1 + \left(\frac{0.1}{\sqrt{2}} \right)^2 \right) \cdot 15 \right) = 2.171 \text{ J/h}.$$

The room has low mechanical energy demand.

3. Calculation of air exchange (abridged). Normative air exchange is $L_{min} = K \cdot V = 6 \cdot 250 = 1500\ m^3$/h or $G_{min} = \rho\, L_{min} = 1.2 \cdot 1500 = 1800$ kg/h. The corresponding air enthalpy difference by the equation (4) is $I_\ell - I_{in} = 3.6\ Q / (\rho\, L_{min}) = 3.6 \cdot 1000 / (1.2 \cdot 1500) = 2$ kJ/kg. The relation $\Delta I / \Delta d = 3.6\ \Delta Q_{hf} / \Delta W = 3.6 \cdot 1000 / 750 = 4.8$ kJ/g. By the I-d diagram, this corresponds to $t_\ell - t_{in} < 1$ °C and $d_\ell - d_{in} < 0.5$ g/kg (near to temperature and moisture equilibrium). Therefore, the normative air exchange is enough. In this case, cooled air can be distributed to the keeping zone (with more strict requirements). After that, it may ventilate the working zone.

4. Calculation of the mechanical energy of inlet air. Let us assume that all air distributors have the same parameters. By the equation (15), the energy (without the resistance of air distribution) for:

I. Displacement ventilation

$$e_{in} = \frac{0.2^2}{2} \cdot \left(1 + \left(\frac{0.1}{\sqrt{2}} \right)^2 \right) = 0.0201 \ \text{J/kg};$$

II. Mixing ventilation by grates

$$e_{in} = \frac{2^2}{2} \cdot \left(1 + \left(\frac{0.2}{\sqrt{2}} \right)^2 \right) = 2.04 \ \text{J/kg};$$

III. Mixing ventilation by air distributors with twirlers

$$e_{in} = \frac{5^2}{2} \cdot \left(1 + \left(\frac{0.5}{\sqrt{2}}\right)^2\right) = 14.1 \text{ J/kg.}$$

By the equation (16), the efficiency by mechanical energy for:
I. Displacement ventilation

$$\varepsilon_m = \frac{2.171}{1800 \cdot 0.02} = 0.06 \,;$$

II. Mixing ventilation by grates

$$\varepsilon_m = \frac{2.171}{1800 \cdot 2.04} = 0.00059 \,;$$

III. Mixing ventilation with twirlers

$$\varepsilon_m = \frac{2.171}{1800 \cdot 14.1} = 0.000086 \,.$$

The results show that all ventilation options give a lot of non-necessary energy. Only displacement ventilation minimizes supply of unnecessary mechanical energy to the room. Other options give more than 100 times less efficiency. The worse solution is mixing ventilation with twirlers – about 700 times poorer efficiency than the displacement ventilation. Such a large difference in efficiency is due to the nature of mechanical energy, which is proportional to the square of the air velocity. Therefore, even reducing the air speed by two times leads to in four-time efficiency increase.
To increase the efficiency, it is recommended to use the variable air volume (VAV) system, which can properly react on changeable actual air demand of the room because of different number of visitors, variable external conditions etc. For such kind of ventilation and air conditioning, the efficiency should be calculated for characteristic modes and averaged by probability of the modes.

11. Conclusion

Proposed method for estimation of efficiency of air exchange organization has obvious physical meaning – relation between demands and supplied potential. It takes into account air velocity and turbulence intensity. Therefore, it gives relevant results of the estimation. For correct calculation of air exchange efficiency in multi-zone rooms, only the processes in occupied zones at saving values of normative air parameters may be taken into account. The most effective ventilation principle is displacement ventilation, which can provide the comfort conditions using minimum inlet air energy. The most of rooms have no significant demand of mechanical energy. Therefore, the most of mechanical energy from ventilation system supplied to rooms is not necessary and used for jet decay. Variable air volume (VAV) air conditioning systems can provide additional increase of energy efficiency.

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