Calculation of Room Velocity Using Kinetic Energy Balance

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ABSTRACT

For a long time, scientific research has tried to establish the relationships between jet momentum and room velocities. The final breakthrough is still to come. One approach is to use a kinetic energy balance, which was initially suggested by Elterman (1980).

This paper presents a thorough kinetic energy analysis. Based on the analysis, a new method is developed for calculating the average room velocity. The calculation method is evaluated with the experimental data from laboratory experiments with three different air distribution methods. An excellent correlation is found between the kinetic energy method and the experimental data within the occupied zone for two air distribution methods and two room scales. With the third air distribution method, vertical air supply, a linear correlation is found, but the calculated values are smaller than the measured velocities because the occupied zone is partly within the main jet area.

INTRODUCTION

For a long time, scientific research has tried to establish the relationships between jet momentum and room velocities (Li 1995; Chow et al. 1996; Chow and Wing 1997). However, room air movement is to a great degree three-dimensional, which makes it difficult to describe using momentum that is a vector quantity (Priest 1996). Additionally, room air velocities are significantly influenced by internal heat sources (Hagström et al. 1999).

The importance of the room kinetic energy was initially introduced by Elterman (1980), who stated that volumetric kinetic energy influences the turbulent exchange between the supply air jet and the bulk flow and the convective heat and mass transfer between different zones in the room. The distance from the air inlet to the point where the jet disintegrates and becomes inseparable from the room bulk flow depends upon the volumetric kinetic energy. The disintegration occurs when the room and the jet volumetric kinetic energies are equal.

According to Elterman, the total kinetic energy, E_r, can be generated from three sources: the energy of incoming air, convective jets attenuating in the room, and the energy introduced by moving objects.

Priest (1996) and Zhivov et al. (1996) presented a very straightforward approach. They proposed that it is possible to calculate the average room "dissipation" velocity assuming that the kinetic energy flux introduced into the space is dissipated into heat by the room volume.

$$\frac{e_r}{V_r} = \frac{1}{2}\rho(1 + TI^2)u_r^2 \tag{1}$$

where V is the volume of the room space (m^3), ρ is the density of the room air, (kg/m^3), TI is the average turbulent intensity in the room space (dimensionless), and u_r is the room average dissipation velocity (m/s).

The experimental studies showed good agreement between the predicted (Equation 1) and measured room air velocities in rooms with obstructions, when the reduction of the total room volume by the obstructions was accounted for in the calculations.

Further analysis of the present experiments (Hagström et al. 1999) shows that there exists a linear correlation between the calculated and measured occupied zone average velocities when Equation 1 is used. However, the equation does not result in the same velocity, so an additional correction is

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needed. An example of the calculated room velocities applying Equation 1 as a function of the measured velocity is presented in Figure 1.

In the following, a thorough analysis of the kinetic energy balance is conducted in order to find the exact correlation. A method for calculating the room average velocity is developed and validated, and an algorithm for its application is presented.

ROOM KINETIC ENERGY ANALYSIS

Room Kinetic Energy Balance

Defining a room as a control volume, the conservation of kinetic energy can be expressed as

$$\frac{dE_r}{dt} = e_{input} - e_{output} + e_{sources} - e_{sinks}, \quad (2)$$

where E_r (J) is the room kinetic energy and e (J/s) is the kinetic energy flux influencing the room space. As the kinetic energy balance is applied, the following assumption can be made about the room air motion: any addition into the room kinetic energy will increase the room velocity level without delay. Thus, there is no "kinetic energy capacity," and the room air momentum can be considered as a chain of steady-state conditions. In such conditions,

$$\frac{dE_r}{dt} = 0. \quad (3)$$

The kinetic energy flux introduced can be calculated with the aid of the outlet area, A (m^2), and initial velocity, u, or volume flow rate, q (m^3/s) from the source:

$$e = \frac{1}{2}\rho u^3 A = \frac{1}{2}\rho u^2 q = \frac{1}{2}\rho \frac{q^3}{A^2}. \quad (4)$$

The potential kinetic energy sources and sinks in the room space are listed in Figure 2. It is also important to differentiate the dissipation zone from the main areas of the supply and thermal jets. In the dissipation zone, the velocity level will be a function of the room kinetic energy. In the main areas of the jets, the velocity depends mainly on the source. Different kinetic energy sources and sinks and their importance to the room velocity are analyzed more closely in the following sections.

Kinetic Energy Inputs

Supply Air Jets (e_j). Supply air jets are usually the major source of kinetic energy in the room. They are generated by an external mechanical force (typically a fan) and the kinetic energy flux introduced by the jet can simply be calculated by Equation 4 using the mean velocity and the effective area of the jet opening.

Infiltration Through the Envelope (e_i). Infiltration flow through the envelope is generated in envelope openings, such as apertures (windows, doors, large openings) and cracks in building elements. The driving force in infiltration is the pressure difference over the envelope between the room and its surroundings, which can be created by a temperature difference, wind, or mechanical forces (i.e., pressurization). Thus, the kinetic energy is created by the transformation of potential energy into kinetic energy in the opening. In practice, only major openings can be considered on their own and minor openings have to be estimated as a whole based on the tightness of the envelope.

Infiltration may have importance, for example, in large industrial buildings with openings during wintertime. However, as the analysis is applied to the laboratory experiments in this study, infiltration can be neglected in the kinetic energy analysis.

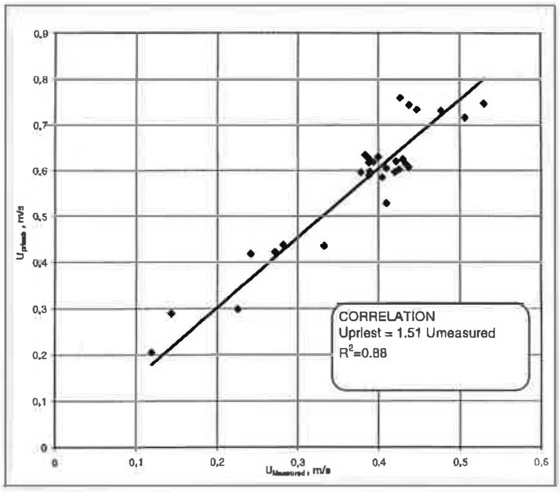

Figure 1 *Horizontal nozzles, correlation of measured and calculated average velocities using Equation 1 (Priest 1996), 56 experiments.*

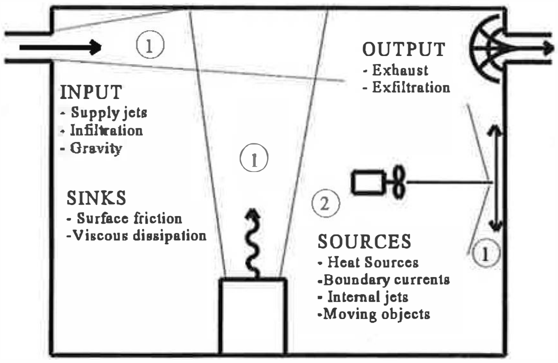

Figure 2 *Kinetic energy components and zones in a room space; (1) the jet (supply or thermal) main zone and (2) the dissipation zone.*

Kinetic Energy Outputs (e_e)

Exhaust Airflow. Exhaust air is drawn from the room at a certain average velocity over the opening area. However, the momentum required to move exhaust air out of the room through the exhaust opening is created upstream by an exhaust fan, not in the room. As a matter of fact, the negative potential created by the fan increases the room momentum by creating movement around the exhaust opening, but the increase is only local and small compared to other movement in the space. Thus, exhaust air openings can be ignored in the kinetic energy analysis.

Exfiltration Through the Envelope. Just as for infiltration, exfiltration through the envelope is also generated in the envelope openings and the driving force is the pressure difference over the envelope between the room and its surroundings. Whether the momentum is exhausted through the opening depends on the formation of the pressure difference at the opening. If the pressure difference is based merely on the static pressure between the room and its surroundings, no momentum or kinetic energy is lost in the room. If the room air has initial velocity against the opening, in other words dynamic pressure, kinetic energy is also exhausted (Equation 4). As the latter case is not common in room flows except in special situations, the exfiltration of the kinetic energy also can be neglected.

Thus, the kinetic energy outputs, e_e, can be ignored from the energy balance.

Kinetic Energy Sources Inside the Room

Heat Sources and Sinks (e_h). The amount of kinetic energy introduced from heat (or cold) sources can be calculated based on the volume flow and the area of the plume at height of the room ceiling (or floor, correspondingly).

The volume flow in a circular plume that is created from a concentrated heat source can be calculated, for example, from the following equation (Mundt 1996):

$$q_h = 0.006\phi_h^{1/3}(\Delta h + h_o)^{5/3}, \tag{5}$$

where ϕ_h (J/s) is the convective heat flux from the source, Δh (m) is the distance from the top of the source to the ceiling (or to the height of the neutral buoyancy, if there is a temperature gradient in the room and the plume dies earlier), and h_o (m) is the distance from the virtual origin of the plume to the top of the heat source. According to Heiselberg (1997), h_o equals twice the hydraulic diameter of the heat source. The convective heat flux from the source is about 70%-90% of the total heat flux for pipes and ducts, 40%-60% for the smaller components, and 30%-50% for large machines and components. The cross-sectional area of the plume can be calculated using the spreading angle of the plume given by Turner (see Burmeister 1993):

$$A_h = \pi\left(\frac{6}{5}0.1(\Delta h + h_o)\right)^2. \tag{6}$$

Thus, the equation for the kinetic energy flux becomes

$$e_h = 5.3\times10^{-5}\rho\phi_h(\Delta h + h_o). \tag{7}$$

Boundary Layer Flow (e_b). Free convection flow is created at warm and cold vertical surfaces. The boundary layer flow is laminar for the dimensionless Grashoff number, Gr < 1×10^9, and fully turbulent for Gr > 1.0 ~1.6×10^{10} (Heiselberg 1997). Equations for the airflow rate and velocity in laminar and turbulent situations can be found in the literature, such as those derived by Andersen (1996).

The volume flow rate for the laminar flow was determined to be

$$q_{b,l} = 0.024\, h^{3/4}\, \Delta T^{1/4}\, w \tag{8a}$$

and for the turbulent flow,

$$q_{b,t} = 0.021\, h^{6/5}\, \Delta T^{2/5}\, w, \tag{8b}$$

where h (m) is the height of the vertical surface, ΔT (°C) is the temperature difference between the wall and room air, and w (m) is the width of the surface.

The average velocity for the laminar flow is

$$u_{b,l} = 0.05\sqrt{h\Delta T}, \tag{9a}$$

and for the turbulent flow,

$$u_{b,t} = 0.019\sqrt{h\Delta T} \tag{9b}$$

Moving Objects (e_m). Large moving objects or surfaces increase the kinetic energy level in the space. The kinetic energy from mechanical sources can be calculated from the body's drag coefficient (C_d), area (A), velocity (u_m), fraction of movement (t), and the room air density (ρ) as follows:

$$e_m = \frac{1}{2}C_d t\rho A_m u_m^3. \tag{10}$$

Internal jets. Internal jets, like mixing fans or propellers, are located inside the room space. They add kinetic energy to the room space in two ways: the kinetic energy through the air jet and through the plume generated above the possible external electric motor. Equations for such situations were presented above.

Gravity. When the supply air jet has a different temperature than the room air, it has "a kinetic energy potential" due to buoyancy. The kinetic energy flux is relative to the density difference between room air and supply air and to the height of the supply air inlet. An equation for the calculation of the kinetic energy flux due to jet buoyancy was derived by Priest (1996) in his doctoral thesis:

$$e_g = gHq_j|\rho_r - \rho_j| = gHq_j\beta\,|T_r - T_j| \tag{11}$$

where β is the volumetric thermal expansion coefficient, H is the height from the floor or ceiling, and T_j and T_r are the air temperatures in the supply air inlet and in the room. Priest (1996) suggested that the height would equal the room height.

This is valid for cooled ceiling diffusers. However, for wall and descended jets, the height "potential" exists only between the height of the jet inlet and the floor for cooled jets and the ceiling for heated jets. The transformation into kinetic energy is always positive regardless of the direction of the buoyancy.

Kinetic Energy Sinks

In the present approach, the following assumptions are made:

- The bulk room airflow close to the wall boundaries is turbulent with no defined direction.
- The surface friction in the room occurs in a laminar sublayer because the conditions for the development of a turbulent boundary layer do not exist at the wall surfaces.
- The travel distance sufficient for the development of a turbulent boundary layer exists only in the core areas of the wall jet and boundary layer flow.
- Thus, the equations for the laminar boundary are valid for the most of the room surfaces.
- Turbulent stress layers in the room space outside the laminar wall boundary can be characterized by turbulent intensity.

The justification for the assumptions and for the strict distinction between surface friction and turbulent stresses are explained in Appendix A.

Diffusion and Turbulent Viscous Dissipation. It is assumed that the relative intensity of the turbulent fluctuations is described by turbulent intensity, which is a ratio of the velocity standard deviation to the average velocity (see justification in Appendix A):

$$TI = \frac{u_{rms}}{u_r}. \tag{12}$$

The total kinetic energy in a space is the sum of the kinetic energies of the mean motion characterized by the average velocity and turbulence:

$$E_{tot} = \frac{1}{2}\rho u_r^2(1 + TI^2)V_r. \tag{13}$$

Thus, when the turbulence dissipation is excluded, the amount of kinetic energy in the mean motion can be calculated as follows:

$$E_m = \frac{E_{tot}}{(1 + TI^2)}. \tag{14}$$

Surface Friction. Kinetic energy dissipation through surface friction can be calculated by multiplying the total friction force by the average room air velocity:

$$e_f = F_f u_r \tag{15}$$

The equation for the total friction force can be developed from the boundary layer theory. The local viscous wall shear can be calculated from the following equation,

$$\tau_w = c_f \frac{1}{2}\rho u^2, \tag{16}$$

where c_f is the friction coefficient for laminar flow and u is the mainstream air velocity;

$$c_f = \frac{0.664}{\mathrm{Re}_x^{1/2}}, \tag{17}$$

where Re_x is a dimensionless Reynolds number.

The total friction force over the surface element, $\Delta A_s = wx$, in a flat plate is then

$$\Delta F(x) = w\int_0^x \tau_w(x)dx = 0.664 w\rho^{1/2}\nu^{1/2}u^{3/2}x^{1/2}, \tag{18}$$

where x (m) is the characteristic length of the friction. Introducing the area of the surface element, we get

$$\Delta F(x) = 0.664\rho^{1/2}\mu^{1/2}x^{-1/2}\Delta A_s u^{3/2}. \tag{19}$$

The wall function for computational fluid dynamics calculations is typically set by defining the local friction coefficients in each cell limited to the room surface area (dA_w). Applying a similar, though universal, approach, the total friction force against the total room surface can be calculated by using the room average velocity instead of the local velocity and by multiplying with the number of the differential surfaces ($N\,\Delta A_s = A_s$):

$$F_f - 0.664\rho^{1/2}\mu^{1/2}x^{-1/2}A_s u_r^{3/2}. \tag{20}$$

ROOM AVERAGE VELOCITY CALCULATION USING A KINETIC ENERGY BALANCE

Development of Calculation Method

Summarizing the discussion above, the kinetic energy flux through the exhaust outlet is close to zero. If infiltration and exfiltration also can be ignored, all the kinetic energy introduced into the room volume will be dissipated through viscous diffusion and surface friction. From Equations 2 and 20, we get an equation for the surface friction in steady-state conditions:

$$e_f = \frac{e_{input} + e_{sources}}{(1 + TI^2)}. \tag{21}$$

Importing Equations 15 and 20, we can derive an equation for the average room velocity:

$$u_r = \left[\frac{e_{input} + e_{sources}}{0.664(\rho\mu)^{1/2}A_s(1 + TI^2)}x^{1/2}\right]^{2/5}. \tag{22}$$

Knowing the value of the characteristic length for the friction process, x, we are able to calculate the room average velocity. It is derived in the following with the aid of a parameter analysis and the experimental data.

Characteristic Length, *x*. Since Equation 1 shows a linear correlation between the measured and predicted velocities, it can be assumed that most of the necessary physical parameters exist in the equation. By adding μ and A_s from Equation 22, we can assume that the characteristic length is a function of the following variables:

$$x \sim x\,(\rho, \mu, A_s, TI, V_r, u_r). \tag{23}$$

Through the analysis of the experimental data (Hagström et al. 1999) with two different air distribution methods and room sizes, the equation for x takes the following form:

$$x = C_x \nu (1 + TI^2)^2 u_r \left(\frac{V_r}{A_s}\right)^{2/3}, \tag{24}$$

where C_x = 1.40 $m^{11/3}/s^{5/3}$ is an empirical coefficient. The dimensions for the variables are x (m), ν (m^2/s), u_r (m/s), V_r (m^3), and A_s (m^2).

The correlation of the characteristic length calculated from Equation 24 with the measured average velocity is presented in Figure 3.

Room Average Velocity Equation. When the equation for x is placed into Equation 22, we get the equation for the room average velocity:

$$u_r = \left(\frac{C_x^{1/2}}{\rho} \frac{e_{input} + e_{sources}}{0.664 A_s}\right)^{1/2} \left(\frac{V_r}{A_s}\right)^{1/6}, \tag{25}$$

where the ratio $C_x^{1/2}/\rho$ can be ignored in the normal range of room temperatures because the influence on the result is only

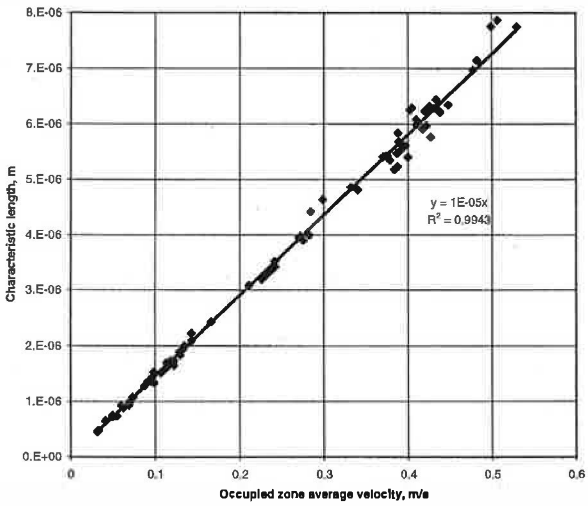

Figure 3 *Predicted characteristic length of friction as a function of the room velocity.*

plus or minus 2%. If we reorganize Equation 25, the equation for the room average velocity gets the form

$$u_r = \left(\frac{1}{0.664} \frac{e_{input} + e_{sources}}{V_r}\right)^{1/2} \left(\frac{V_r}{A_s}\right)^{2/3}. \tag{26}$$

Comparing Equation 26 with Equation 1, we see that the main difference between the two is the additional coefficient needed in Equation 1, which is the ratio of the room volume and the room surface area to the power two over three. This ratio could be described as a kinetic energy intensity factor that changes with room dimensions.

An interesting feature is that the influence of the turbulence, $(1+TI^2)$, disappeared from the numerator. This could be explained by the fact that globally, all the kinetic energy dissipation into thermal energy occurs at the wall boundaries, whereas the net dissipation inside the fluid is equal or close to zero. This makes sense if we consider a fluid cell that is heated through viscous dissipation. The heated fluid expands and forces other fluid cells to move around; thus, the instantaneous heating is transformed back to kinetic energy inside the fluid. This proves that viscous dissipation in a room space is a reversible process.

Additional speculation could be made about the turbulent kinetic energy. Since the dissipation is completely defined by mean air motion, the turbulent intensity does not have any physical importance from the energy point of view. As a matter of fact, the whole room flow outside the jet core area is to some degree a turbulent, fluctuating motion, and the mean velocity describes its magnitude. The turbulent intensity simply describes the scale of the fluctuation. Appplying a similar conclusion to the supply air jet boundary, one can ignore turbulence intensity and just use the average velocity at the outlet to calculate imported kinetic energy. Thus, ignoring the turbulence intensity from the equations presented earlier, one can modify them as follows:

From Equation 21,

$$e_f = e_{input} + e_{sources}, \tag{27}$$

and from Equation 24 for the characteristic length for the wall friction,

$$x = C_x \nu u_r \left(\frac{V_r}{A_s}\right)^{2/3}. \tag{28}$$

Design Algorithm—A Practical Application of the Method

The application of the proposed calculation method is a very straightforward process and little input information is needed. The design algorithm consists of three steps:

1. A collection of the input data from external and internal kinetic energy sources and room dimensions.
2. Calculation of the kinetic energy supplied to the room, the room volume, and the area of the room surfaces.
3. Calculation of the resulting room average velocity.

The numerical results for two validated cases are shown in Table 1 (for example 1,concentrated air supply, and example 2, horizontal air supply, see also Figure 4). The only kinetic energy sources that were present in the laboratory studies were the supply air jet and heat sources. The examples demonstrate two different types of situations. In example 1, the kinetic energy flux of supply air is 96% of the total, whereas in example 2, the kinetic energy flux from thermal forces and the supply air jet are almost equal.

TABLE 1
Numerical Examples of the Application of the Proposed Calculation Method

Design Setup	Data	Example 1	Example 2	Equation Used
Input Data	**Jet**			
	q_o (m^3/s)	0.204	0.081	
	u_o (m/s)	10.6	2.33	
	H (m)	2.1	1.2	
	ΔT (°C)	20.8	12.3	
	Heat Source			
	ϕ_h (W)*	1475	750	
	Δh (m)	1.7	1.7	
	h_o (m)	3	3	
	Room			
	Height (m)	2.4	2.4	
	Length (m)	7.2	7.2	
	Width (m)	3.6	3.6	
	Obstructions			
	Height ratio	30%	30%	
	Area ratio	30%	30%	
Calculation of the Parameters	**Kinetic Energy Supply**			
	e_j (W)	13.72	0.26	4
	e_g (W)	0.15	0.04	11
	e_h (W)	0.44	0.23	7
	Room			
	V_r (m^3)	56.6	56.6	
	A_s (m^2)	108.6	108.6	
Room Average Dissipation Velocity	U_r, predicted	0.40	0.075	26
	U_r, measured	0.39	0.070	

* The convective heat flux was estimated as 50% of the total input.

VALIDATION OF THE CALCULATION METHOD WITH EXPERIMENTAL DATA

Experimental Data

The experimental data used for validation were collected in full- and reduced-scale laboratory experiments (Hagström et al. 1999). The reduced-scale experiments were conducted in a university's room ventilation simulator (RVS). The size of the test room was 7.2 m by 3.6 m by 2.4 m. During the experiments the influence of different parameters on the occupied zone conditions of an industrial hall were studied in reduced-scale experiments. The parameters studied were the air diffusion method, the air change rate, the level of the room obstruction area and its height, and the cooling load. The experimental setup and the results are reported in detail in Hagström et al. 1999). Three different air diffusion methods were used and the schemes are presented in Figure 4.

Case 1. Horizontal, "concentrated" (or high sidewall) air supply attached to the ceiling air supply with the occupied zone ventilated by a reverse flow.

Case 2. Horizontal (mid-sidewall) air supply above the occupied zone.

Case 3. Air supply with vertical jets projected downward.

Case 4. Full-scale experiments with the air distribution method 1 were conducted in the laboratory of Halton Oy in Finland during the summer 1998. The dimensions of the laboratory hall were 25 m by 12 m by 8m.

Validation

The calculated room velocity from Equation 26 covers the whole room space, excluding the main areas of supply and convective jets. The occupied zone velocity is a part of the room volume and will fulfill the same equation given that it is not in the jet area. Thus, it can be used for validation of the method presented. More importantly, the method can also be utilized in practice for evaluation of the occupied zone average velocity. The results of the validation are presented in Figures 5 through 8, where the predicted room average velocities are drawn as a function of the measured occupied zone average velocities.

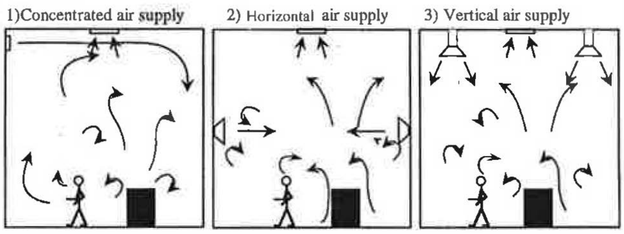

Figure 4 *Studied air distribution methods (Hagström 1998).*

Case 1, Concentrated Air Supply. The results show an excellent correlation with high reliability between the calculated and measured velocities of 56 experiments (Figure 5). The slope of the correlation line is 1.0 and the correlation coefficient $R^2 = 0.91$. The average error of the prediction is 3 mm/s.

Case 2, Horizontal Air Supply. The validation with 30 experiments results in a correlation curve with the slope 1.0 as well, but the R^2 value is much lower, 0.41, than in case 1 because of the disintegration at the lowest velocities (Figure 6). This, however, is more likely a result of the measurement accuracy at such a low velocity, below 0.07 m/s, than of the calculation method. The average error of the prediction was still very small, 3 mm/s.

Case 3, Vertical Air Supply. The validation of 27 experiments shows a clear correlation as well, but the calculated values are only 54% of the measured ones, as can be seen from Figure 7. The reason for this result is dealt with in the discussion below.

Case 4, Full Scale. The validation of the 15 full-scale experiments with air distribution method 1 proves that the calculation method is applicable to various room sizes as well (see Figure 8). The slope of the correlation line is 1.0 and the R^2 is 0.87. The average error of the prediction is 3 mm/s.

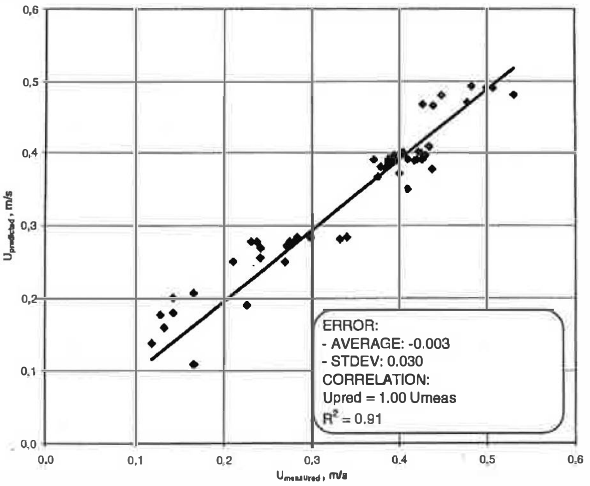

Figure 5 *Reduced scale, concentrated jets, correlation of measured and predicted occupied zone average velocities, 56 experiments.*

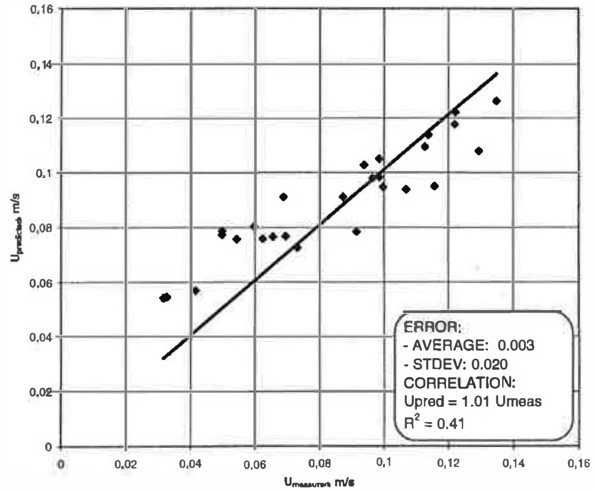

Figure 6 *Horizontal jets, correlation of measured and predicted occupied zone average velocities, 30 experiments.*

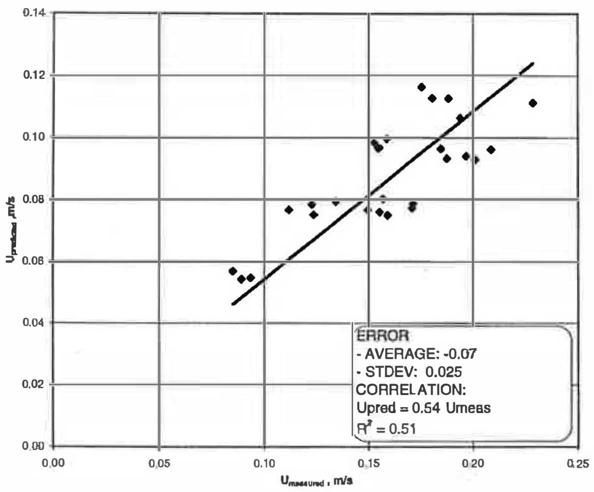

Figure 7 *Vertical jets, correlation of measured and predicted occupied zone average velocities, 27 experiments.*

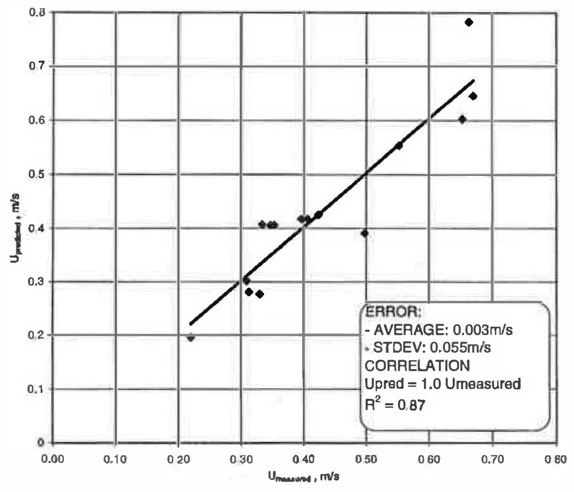

Figure 8 *Full scale, concentrated jets, correlation of measured and predicted occupied zone average velocities, 15 experiments.*

DISCUSSION

Using the proposed method, a designer can estimate the average room velocity level and the acceptability of the conditions in the ventilated room.

When using computational fluid dynamics codes, this simple method can be utilized also to evaluate whether the computation has managed to reproduce skin friction and the proper velocity level without conducting experimental work.

The validity of the method presented is limited to the dissipation zone, which is the room volume outside the main areas of jets and thermal currents. In those areas, the velocity distribution is still governed by the kinetic energy sources (see Figure 2).

In cases 1, 2, and 4, the occupied zone is completely within the dissipation zone, whereas with the vertical air supply (case 3), the occupied zone is partly within the main zone of the vertical supply jets (see Figure 9). Thus, the surplus kinetic energy introduced directly by the supply air jet into the occupied zone increases the zonal velocity above the room average. As a result, the measured occupied zone average velocities are twice the calculated room average velocities, although a linear correlation is found. Further work is needed to asses the influence of this remaining "excess" kinetic energy directed into the specific room zone.

CONCLUSIONS

A simple method for the calculation of the room average velocity has been developed. The method is based on the kinetic energy balance of the room space. Following the design algorithm presented here, a designer can easily estimate the average velocity level within a ventilated room.

The calculation method was compared to data collected from laboratory experiments using three different air distribution methods and two room scales. The calculation method shows excellent correlation with the experimental data from the occupied zone with two air distribution methods and different room scales. With the third air distribution method, vertical air supply, a linear correlation was also found. However, the calculated values were smaller than the measured velocities because the occupied zone was partly within the main jet area.

The dissipation of the kinetic energy inside the room space occurs through skin friction at the laminar wall boundaries, while the viscous dissipation within the air volume can be ignored. The characteristic length for the wall friction was solved experimentally and is a function of the room average velocity and the kinetic energy intensity.

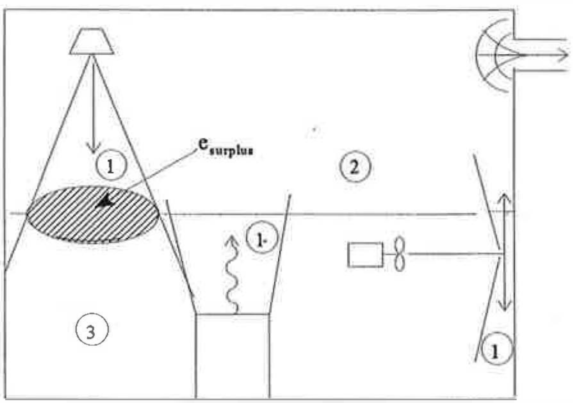

Figure 9 *Case 3, the vertical air supply. The occupied zone was partly within the main zone of the vertical supply jet: (1) is the jet (supply or thermal) main zone, (2) is the dissipation zone, and (3) is the occupied zone.*

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NOMENCLATURE

A	= area, m^2
b	= surface width, m
c_f	= friction coefficient
C	= coefficient
e	= kinetic energy flux, J/s
E	= kinetic energy, J
F	= total friction force, N
g	= specific gravity, kg/m^3
h	= height, m
l	= turbulent mixing length, m
N	= number of surface elements
q	= air volume flow rate, m^3/s
t	= fraction of movement
T	= temperature, °C
TI	= turbulence intensity
u	= air velocity, m/s
V	= volume, m^3
w	= surface width, m
x	= characteristic length, m
δ	= boundary layer thickness, m
μ	= air viscosity, kg/m^2
ν	= air kinematic viscosity, m^2/s
τ	= shear stress, kg/ms^2
ρ	= air density, kg/m^3
Δ	= difference of temperature, °C; height, m
ϕ	= convective heat flux, J/s

Subscripts

d = drag
f = friction
g = gravity
b = boundary layer
h = heat source
i = infiltration

Input

j = air jet
k = kinetic
l = laminar
m = moving object
o = initial (height)

Output

r = room
s = surface

Sources

s = sinks
t = turbulent
tot = total
w = wall
x = related to characteristic length

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APPENDIX A

Justification for Assumptions Made Regarding Kinetic Energy Sinks

Nature of the Room Turbulence. Traditionally, rigid wall boundaries are divided into two basic cases: flow over a flat plate and flow in a pipe. Hinze (1975) described the essential difference between the two groups as follows: in the first, the domain where the wall turbulence occurs increases along the body in the downstream direction, whereas in the second, the domain of wall turbulence remains restricted to the space bounded by the rigid walls. In room flow calculations, the flat plate boundary theory is usually applied. If we consider the above distinction by Hinze, the statement "turbulence remains restricted to the space bounded by the rigid walls" is valid in room airflows, where the flow is restricted in all three dimensions. From this perspective, it would be desirable to study room flow as a third basic type of boundary instead of fitting it into one of those presently considered. So far, the nature of the room airflow is not completely specified. Airflows in a room may be laminar unsteady, locally turbulent, transitional, or fully turbulent. However, very seldom is room airflow laminar (Chen and Jiang 1992).

In turbulent motion, the total shear stress can be described as a sum of a molecular part and a turbulent part. At some distance within the boundary, this can be expressed with the following equation,

$$\tau = \mu\frac{\partial u}{\partial y} + \mu'\frac{\partial u}{\partial y} = \rho(\nu + \varepsilon)\frac{\partial u}{\partial y}, \tag{A-1}$$

where u is the mainstream velocity in the surface direction at a distance of y from the surface, μ and μ' are the molecular and turbulent viscosities, ν is the kinematic viscosity, and ε is called the eddy diffusivity or turbulent kinematic viscosity. According to Prandtl's mixing length theory, eddy diffusivity is

$$\varepsilon = \rho l^2 \frac{\partial u}{\partial y}, \tag{A-2}$$

where $l = Cy$ is the mixing length. From the equation it is clear that the importance of the eddy viscosity decreases closer to the wall. On the other hand, further away from the wall the molecular effects may be ignored.

Laminar Boundary Sublayer. The assumption of the molecular surface friction is supported by the findings in computational fluid dynamics. The accurate solution of the boundary layer equations for turbulent flows using low Reynolds number models, which evaluate the turbulent viscosity at all points within the flow, require that grid points be located within the laminar sublayer (see Figure A-1) (Anderson et al. 1984). It is also found that when using finite difference equations, the condition that is sufficient for the cell Reynolds number at the wall boundary to provide stable and correct simulation of the viscous effect at the wall boundary is $Re_x \leq 2$. (Fletcher 1988; Anderson et al. 1984).

The laminar boundary layer thickness at the point where the velocity at the edge of the layer is 99% of the mainstream velocity is calculated from Schlichting (1968),

$$\frac{\delta}{x} = \frac{5}{\sqrt{Re_x}} = \frac{5}{\sqrt{\frac{u_\infty x}{\nu}}} \tag{A-3}$$

where δ is a laminar boundary thickness in the y direction and x is the characteristic length for the friction. The requirement for $Re_x \leq 2$ means that both the characteristic length and the boundary layer thickness are close to the magnitude of the kinematic viscosity within the range of velocities in the room space.

Surface Roughness. A universal turbulent velocity profile can be described with the aid of dimensionless velocity and distance:

$$u \qquad , \tag{A-4}$$

where $v^* = (\tau_w/\rho)^{1/2}$ is called the friction velocity and τ_w is the wall shear stress. The laminar sublayer is specified as $y^+ < 5$ and $y = \delta$ (Schlichting 1968).

A necessary condition for the surface to be hydraulically smooth is (Schlichting 1968)

$$, \tag{A-5}$$

where k (m) is the surface roughness. The data on the surface roughness of the room interior surfaces are limited. If we estimate that the scale of the surface roughness for finished and painted wall surfaces is in the order of 10^{-6} m, we can conclude that within the range of velocities in the room space, the surfaces can be considered as hydrodynamically smooth. An exception to this would probably be thick carpeting.

Diffusion and Turbulent Viscous Dissipation. The process of the diffusion and dissipation of a mean airflow is characterized by Hinze (1975) as follows: "Due to the interaction between mean motion and turbulent motion, energy is extracted from the mean motion through work of deformation by the turbulent stresses, converted into turbulence energy which ultimately is converted through work of deformation by the viscous stresses in the turbulent motion into heat."

Turbulence is generated in two ways, at the wall boundaries and as "free turbulence" at shear layers of free jet and wake flows. The friction at the walls has already been discussed.

Viscous shear stresses have an important role in the transition of the mean kinetic energy into turbulent energy, but the importance of the viscous dissipation between fluid particles on the total kinetic energy balance and its magnitude, compared to dissipation due to friction on walls, is not thoroughly understood. Hinze (1975) concluded that the local production of turbulence energy from the mean motion is not equal to the local viscous dissipation in general because turbulence at a point is dependent on upstream conditions and on turbulent conditions outside it.

However, this assumption might be valid globally. Thus, the magnitude of the viscous dissipation could be expressed by the means of average turbulent kinetic energy in the space. Assuming that the relative intensity of the turbulent fluctuations is described by turbulence intensity, which is a ratio of the velocity standard deviation to the average velocity, the total kinetic energy in a space is a sum of the kinetic energies of the mean motion characterized by the average velocity and turbulence:

$$. \tag{A-6}$$