The Proposal of a Quantitative Evaluation Method on Mixing Loss in the HVAC System Design

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Key words: Energy conservation, Mixing loss, Set-point temperature differences, HVAC system, CFD(Computational Fluid Dynamics)

Abstract

It is a serious subject for energy conservation to prevent the energy loss caused by the mixture of heated and cooled air jets in perimeter and interior zone of a building operated with two kinds of air-conditioning systems simultaneously. The purpose of this paper is to clarify the quantitative and qualitative mechanisms of mixing loss and to propose a evaluation method for it. By using the dynamic heat load calculation, heat extraction load of a typical office building in Busan are calculated. According to the results, numerical simulations based on CFD (Computational Fluid Dynamics) were performed in order to evaluate mixing loss in the physical size of HVAC system. Then, the distributions of air temperature and airflow patterns according to the differences of set-point temperature are analyzed to grasp relations how to influence mixing loss.

Nomenclature

G_h, G_c	: Mass transfer rate [kgf/h]
ML	: Total mixing loss
MLR	: Total mixing loss rate
t_{sP}, t_{rP}	: Temperatures of supplied air and exhausted air in perimeter [℃]
t_{sI}, t_{rI}	: Temperature of supplied air and exhausted air in interior [℃]

Subscripts

h	: heating
c	: cooling
P	: perimeter
I	: interior

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1. Introduction

Mixing loss can be explained as a phenomena occurred at the place where there are heating and cooling simultaneously, and it may not be in such a case of small scale building or no internal heat generation. However, with increasing number of intelligent buildings, the amount of internal heat generation grows, and it makes interior zone cooling even in the winter, whereas it is needed to heat perimeter zone at the same time. In order to solve these problems in designing HVAC of office buildings, it is necessary to establish judgments and methods for mixing loss in heating indoor rooms.

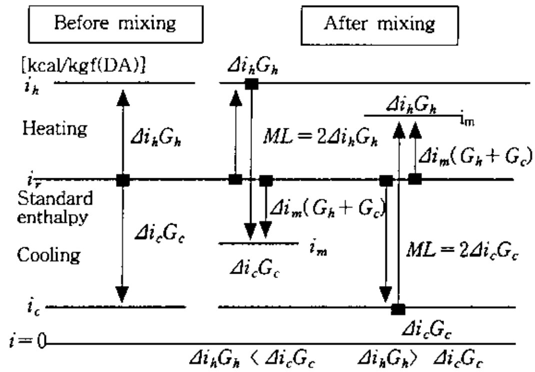
With background stated above, Nakahara[1] made a full-scale experiment on the relation of each factor relevant with mixing loss. However, this analysis method to know the correlations among each factor under constrained conditions requires a laboratory that can be controlled precisely and high-cost apparatus to measure. This experiment, nevertheless, has limitations to show indoor airflow patterns in detail that is needed to know characteristics of mixing loss.

The purpose of this paper is to suggest new evaluation method for quantitatively estimating mixing loss by using CFD (Computational Fluid Dynamics) simulation, and apply for some cases as the difference of set-point temperatures between perimeter and interior zone to verify its possibilities.

2. Evaluation method for mixing loss

Generally, effective heat in HVAC system can be defined as heating or cooling on the basis of the amount of enthalpy a indoor room has. For this standard enthalpy of i_r, the enthalpy and the amount of airflow supplied with respect to heating can be expressed as i_h and G_h, and with respect to cooling can also be expressed as i_c and G_c respectively. In that case, effective heat transfer rate is $\triangle i_h G_h + \triangle i_c G_c$, which can be heat loss as $2\triangle i_h G_h$ or $2\triangle i_c G_c$ only if either heating or cooling is operated.

As shown in Fig. 1, which is sketched on the relation of heat transfer rate in mixing loss, effective heat transfer rate after mixing both of heating and cooling can be expressed as $\triangle i_m(G_h+G_c)$. Thus, the difference of heat transfer rate by mixing loss means the amount of mixing loss in HVAC system. Although mixing loss can be occurred on dual duct system or three-pipe system of FCU (fan coil unit)



Enthalpy after mixing	$i_m=(i_h G_h+i_c G_c)/(G_h+G_c)$
Effective enthalpy before mixing	$\Delta i_m=[\Delta i_h G_h-\Delta i_c G_c-\Delta i_c G_c]/(G_h+G_c)$
Effective heat transfer rate before mixing	$q_t=\Delta i_h G_h+i_c G_c$
Effective heat transfer rate after mixing	$q_e=\Delta i_m(G_h+G_c) =[\Delta i_h G_h-\Delta i_c G_c]$
Total mixing loss	$ML=q_t-q_e$ $=2\Delta i_h G_h(\text{if } \Delta i_h G_h<\Delta i_c G_c)$ $=2\Delta i_c G_c(\text{if } \Delta i_h G_h>\Delta i_c G_c)$

Fig. 1 Mixing loss expression.

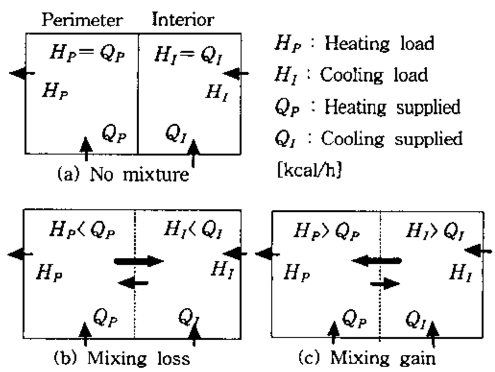


Fig. 2 Diagram for mixing loss.

system, this study is focused on only mixing loss of an indoor room as shown in Fig 2.

In the case that heating in perimeter and cooling in interior are operated without heat loss simultaneously, the amount of supplied heat transfer rate in perimeter is equal to heating load as shown in Fig. 2 (a). Fig. 2 (b) as the case of mixing loss shows that heated air to supply in perimeter is mixed with cooled air in the interior zone before satisfying heating load occurred at perimeter. On the contrary to Fig. 2 (b), mixing gain can reduce heat transfer rate supplied in perimeter as shown in Fig. 2 (c).

From the assumption stated above, the amount of mixing loss can be defined as heat transfer rate that is more supplied than a total load occurred at a room. Likewise, mixing gain can be defined as heat transfer rate that is less supplied than load needed at room.

If *mixing gain assumed* as a loss with a minus sign (−), it leads to define a total mixing loss as

$$\begin{aligned} ML &= (Q_P + Q_I) - (H_P + H_I) \\ &= (Q_P - H_P) + (Q_I - H_I) \\ &= ML_P + ML_I \end{aligned} \quad (1)$$

For applying this theoretical concept for general evaluation on mixing loss, we can also define it as MLR (Mixing Loss Rate), a way of representing mixing loss with respect to extraction load.

$$MLR = ML/(H_P + H_I) \times 100\,[\%] \quad (2)$$

3. Quantitative evaluation of mixing loss using CFD

3.1 Simulation model

To verify the evaluation method of mixing loss proposed on this paper, CFD simulation is performed on an office building in Busan.

The office room simulated is placed on the middle floor of the building. From this condition, it is possible to assume that the temperature of room can be affected only by window-facing wall, and that other walls such as ceiling, floor, and internal walls are completely adiabatic.

3.2 Case description and mathematical model

Before simulating the office model, we cal-

Table 1 Turbulence model for Viollet form of Standard $\kappa-\varepsilon$ model.

Equation	Governing equations
Continuity	$\frac{\partial U_j}{\partial x_j} = 0$
Momentum	$\frac{\partial U_i}{\partial t} + \frac{\partial U_i U_j}{\partial x_j} = -\frac{\partial}{\partial x_i}\left(\frac{P}{\rho} + \frac{2}{3}k\right) + \frac{\partial}{\partial x_j}\left\{(v+v_t)\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\right\} - g_i\beta\Theta$
Energy	$\frac{\partial \Theta}{\partial t} + \frac{\partial U_j \Theta}{\partial x_j} = \frac{\partial}{\partial x_j}\left\{\left(v + \frac{v_t}{\sigma_\theta}\right)\frac{\partial \Theta}{\partial x_j}\right\}$
Turbulence energy(κ)	$\frac{\partial k}{\partial t} + \frac{\partial U_j k}{\partial x_j} = \frac{\partial}{\partial x_j}\left\{\left(v + \frac{v_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right\} + P_k + G_k - \varepsilon$
Energy dissipation(ε)	$\frac{\partial \varepsilon}{\partial t} + \frac{\partial U_j \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left\{\left(v + \frac{v_t}{\sigma_t}\right)\frac{\partial \varepsilon}{\partial x_j}\right\} + \frac{\varepsilon}{k}(C_{1\varepsilon}P_k - C_{3\varepsilon}G_k) - C_{2\varepsilon}\frac{\varepsilon^2}{k}$
where, $v_t = C_D\frac{k^2}{\varepsilon}$, $P_k = v_t\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j}$, $G_k = g_i\beta\frac{v_t}{\sigma_\theta}\frac{\partial \Theta}{\partial x_i}$	
$C_D = 0.09$, $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $C_{3\varepsilon} = 1.44(G_{k>0})$, $0(G_k<0)$	
$\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$, $\sigma_\theta = 0.7$, $\sigma_C = 0.7$	

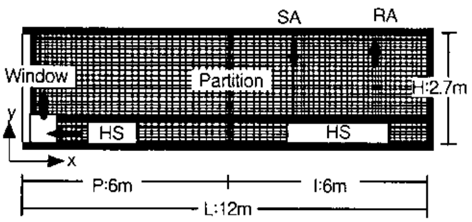


Fig. 3 Geometry for CFD simulation.

culate the outdoor condition for each term of January and February from standard weather data on Busan. From these weather data, a dynamic heat load calculation code, HASP/ACLD/9801, is used to calculate both heating load in perimeter zone and cooling load in interior zone.

By the results of heat load calculation show that heating load in perimeter zone is 1224.2 kcal/h, and cooling load in interior zone is 1935.3 kcal/h. From the results, we can determine the temperature and the amount of airflow at supply diffuser that will be used to solve the airflow and the temperature by CFD.

A CFD program, PHOENICS, is used to solve the three cases as the set-point temperatures mentioned in Table 2. The program discretizes the space into non-uniform computational grid with 90×50, and the discretized equations are solved with the SIMPLE algorithm. The turbulence model is the viollet standard $\kappa-\varepsilon$ model and the scheme used in the numerical simulation is hybrid. The boundary condition used is shown in Table 2.

Table 2 Boundary conditions.

		Perimeter	Interior
HVAC system		FCU	CAV
Set-point temperature	case A	22℃	20℃
	case B	20℃	20℃
	case C	20℃	22℃
Temperature of air supplied	case A	23.5℃	18.4℃
	case B	21.5℃	18.4℃
	case C	21.5℃	20.4℃
Velocity of air supplied		1.1 m/s	1.2 m/s
Heat source		100 w/m³	400 w/m³
Temperature of the inner surface of wall(window)		4.7℃	
Interior wall		free slip	free slip

3.3 Evaluation of mixing loss

Assuming that there is a wall completely insulated on the boundary space between perimeter and interior zone, we can calculate the airflow and the temperature distribution in steady state by means of CFD and it leads to know the amount of heat transfer rate to supply before mixing by using following equations.

$$Q_P = 0.29 \times V \times (t_{sP} - t_{rP}) \quad (3)$$

$$Q_I = 0.29 \times V \times (t_{sI} - t_{rI}) \quad (4)$$

For evaluating heat transfer rate to supply after mixing, the amount of heat to extract after mixing can also be evaluated by the same evaluation method under the same condition without the wall insulated between them, and using Eq. (1) it possible to obtain the amount of mixing loss.

4. Results and discussion

4.1 Airflow pattern and air temperature distribution

Figs. 4 to 9 show the indoor airflows and the temperature distributions obtained according to the set-point temperature of interior zone.

When keeping with the airflow pattern on the left-hand side of Figs. 5, 7, and 9 without a wall between perimeter and interior zone, the heated air from the fan coil installed in perimeter zone rises on the ceiling. Then, a part of the air is mixed with the cooled air from

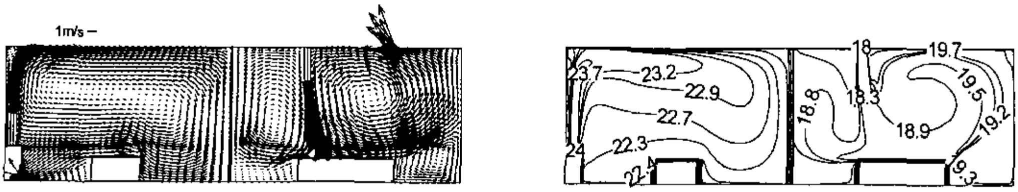


Fig. 4 Airflow pattern and temperature distribution in case of simulating with internal partition(Case A).

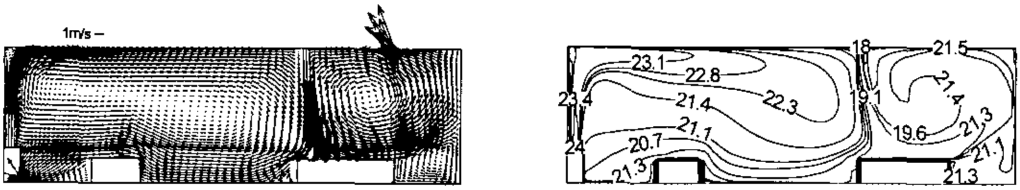


Fig. 5 Airflow pattern and temperature distribution in case of simulating without internal partition(Case A).

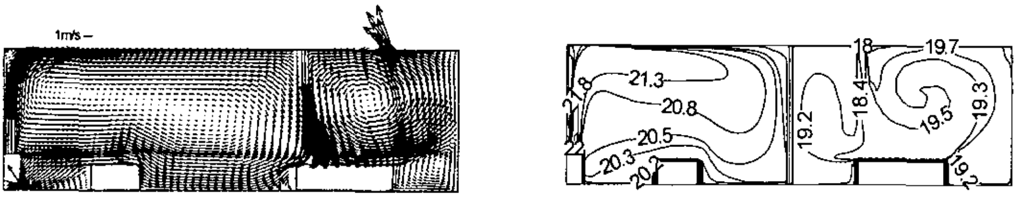


Fig. 6 Airflow pattern and temperature distribution in case of simulating with internal partition(Case B).

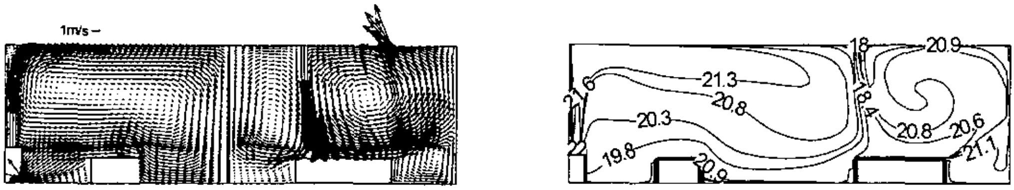


Fig. 7 Airflow pattern and temperature distribution in case of simulating without internal partition(Case B).

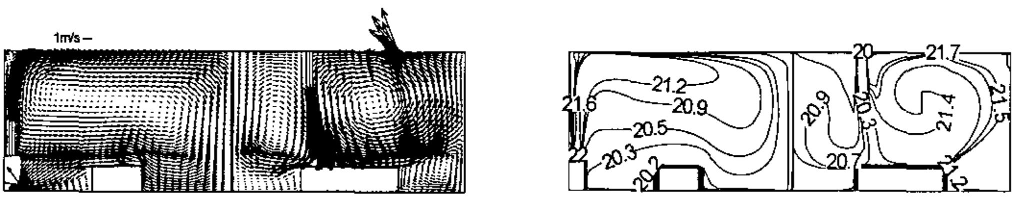


Fig. 8 Airflow pattern and temperature distribution in case of simulating with internal partition(Case C).

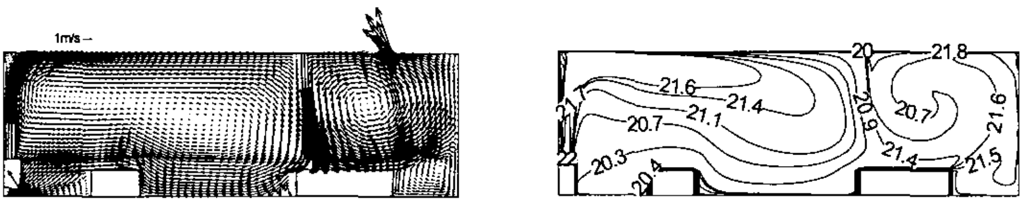


Fig. 9 Airflow pattern and temperature distribution in case of simulating without internal partition(Case C).

the exhaust diffuser of interior zone, and some part of the air is circulated again toward the fan coil diffuser near the window.

In the case that there is a wall dissecting perimeter and interior zone, we can obtain the result similar with those calculated above, although the size of the airflow circulation is smaller.

In a similar way, comparing them according to the difference of set-point temperature, airflow patterns are almost identical although there is only a little difference among airflow patterns on the place where the supply air of perimeter zone is mixed with the supply air of interior zone.

When keeping with the temperature distribution on the right-side of Fig. 5, 7, and 9 without an internal wall, it is possible to recognize the difference of the temperature distribution. It can be clear evidence that substantial mixing loss can be occurred in the case that heating and cooling are operated simultaneously.

Through these numerical simulations, mixing loss seems to be mainly dependent on the set-point temperatures, especially the difference of them in both perimeter and interior zone.

As the set-point temperature in perimeter zone is higher, the amount of mixing loss increases. However, as shown case (c) in Figs 8 and 9, it is possible to reduce mixing loss in case that the temperature of perimeter zone is higher than that of interior zone.

4.2 Evaluation of mixing loss

From a global analysis of the temperature distribution of Figs. 4 to 9 and the use of Eqs. 1 to 4 heat transfer rate after and before mixing, *ML*, and *MLR* are calculated as shown in Table 3 to 5.

The values of *ML* and *MLR* show there is a increase in mixing loss under condition that the temperature in perimeter zone is higher than that in interior zone.

Table 3 Mixing loss for Case A

		Heat transfer rate	*ML*	*MLR*
Before mixing	H_P	987.5 kcal/h	2858.8 kcal/h	97.8%
	H_I	1935.3 kcal/h		
After mixing	Q_P	2273.8 kcal/h		
	Q_I	3507.8 kcal/h		

Table 4 Mixing loss for Case B

		Heat transfer rate	*ML*	*MLR*
Before mixing	H_P	1223.3 kcal/h	1263.8 kcal/h	40%
	H_I	1935.3 kcal/h		
After mixing	Q_P	1515.9 kcal/h		
	Q_I	2906.5 kcal/h		

Table 5 Mixing loss for Case C

		Heat transfer rate	*ML*	*MLR*
Before mixing	H_P	1223.3 kcal/h	−6.5 kcal/h	−0.2%
	H_I	1758.6 kcal/h		
After mixing	Q_P	1171.4 kcal/h		
	Q_I	1804 kcal/h		

In case of setting the set-point temperature higher than that of interior zone to shield cooling radiation and cold draft from the window contacted directly with outdoor air, it leads to generate circulative airflow by the difference of the pressure, and increases mixing loss by circulating the heated air in perimeter zone toward interior zone and by circulating the cooled air in interior zone toward perimeter zone.

To reduce mixing loss it is necessary to minimize the difference of set-point temperatures between the perimeter and interior zone or set the temperature in interior zone higher than that in perimeter.

Comparisons between the results of the full-scale experiment performed by Nakahara[1] and those calculated by numerical simulations show good agreement and confirm that the method suggested in this paper can be used for the evaluation of mixing loss.

5. Conclusion

In this paper, we suggest new evaluation method for quantitatively estimating mixing loss by using CFD simulation, and evaluate for some cases as the difference of set-point temperatures between perimeter and interior zone in order to verify its application.

The results obtained by using the new evaluation method applied in this paper is in good agreement with the full-scale experimental results performed by Nakahara.(1) It means that this new method may be used as an evaluation method for mixing loss. In addition, using this method it is possible to establish efficient preventive specifications for mixing loss.

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