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Studies on a Pressurized Type Under-floor Air Conditioning System

(Analysis of Governing Factors for Uniform Air Velocity **Profile)**

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Summary

Recently, a lower type free-access floor system which realizes improved ease of walking and less sense of confinement has been attracting attention. However, it is known that the lower the design of the air supply chamber, the larger the non-uniformity of the diffuser air velocity profile and the greater the deterioration in room temperature distribution. In this paper, an analytical model is proposed for predicting the non-uniformity of diffuser air velocity at the design stage. The validity of the analytical model was confirmed by the goodcoincidence of results obtained in scale model experiments. It was found that the limit of the floor height at which a uniform diffuser air velocity is obtained was 50 mm. Moreover, a relation was found whereby the inverse of coefficient of resistance at a diffuser up to a value of 1 is proportional to the maximum non-uniformity. This indicates the possibility that design work aimed at constructing a pressurized type under-floor air conditioning system with a uniform air velocity profile from diffusers may be readily performed from parameters of the coefficient of resistance at a diffuser by utilizing this relation.

1. Introduction

The rapid spread of information processing equipment has occasioned the adoption of free-access floor system in offices, in which can be housed power lines. communication cables and other items, and which lends itself to changes in layout. Recently, there has been much interest in lower type free-access floor systems which provide improved comfort when walking and alleviate the sense of confinement which is a problem with other systems. If the space in lower systems could be used to house air supply chambers for air conditioning system, the construction costs required for an under-floor air conditioning system could be reduced. However, the lower the floor, the less uniform the air velocity profile from diffusers. And, it is known that in a pressurized type under-floor air conditioning system, excessive pressurization of air supply chambers can cause air leakage, so that the thermal environment of the room is degraded as a result.

There have been numerous reports of scale model experiments on pressurized type under-floor air conditioning systems and measurements of thermal environments. However, there have been extremely few studies analysing the air velocity profile.') In this paper, we propose a model which can be used to analyze the non-uniformity in the air velocity profile, and after verifying the

validity of the model through scale model experiments, we studied the floor height lower limit the limits to establishment of analytical equations. In addition. the sensitivity of factors gover non-uniformity **in** diffuser air velocity was analyzed, and a method for adjustment to make velocity profiles more uniform, as well as a simple design technique to determine the diamete air diffuser outlets and the height of air supply chambers, were proposed.

2. Construction of an Analytical Model

The assumptions for the purpose of constructing an analytical model were as follows;
Assumption (1) : The pressure within the chamber is higher than the air pressure in the ro (condition of continuity)
Assumption (2) : An equal diffused air velocity is obtained at all diffuser outlets. (continu uniform division branch model)`
Assumption (3) : The diffused air velocity is proportional to the 1/2 power of the press difference between the chamber and the room. (turbulent flow model)
Assumption (4) :The air velocity in the chamber is distributed only in the length direction. (o diinensional model)
Assumption (5) : A state of thermal uniformity obtains within the chamber. (isothermal model)

2.1 Mass Balance

Physical parameters such as the height of the chamber and the diameter of diffuser outlet, air f and pressure loss are indicated in Figure 1. From the mass balance, we obtain the follo relations;

cl.

U 3: neck f l 1 ter V b.1k,ead RO

P-

U

x=t

Fig. 1 Analytical model for supply chamber and diffusers.

Diffuser aperture area: 1 s=RK.(7r.d',/4).M.N -----

Diffuser aperture ratio: K 7- s/1 (L. W) -----

Ratio of air path cross-section to floor area: 8 (D. W),-'(L. W) -----

Neck air velocity of chamber: U (Q/--3600),/,6 -----

Diffuser air velocity: v= fl -U'l K ----- (51)

2.2 Momentum Balance

Taking the coordinate origin at the bulkhead, we use Equation (6) to approximate the air velo at position x in the chamber length direction.

validity of the model through scale model experiments, we studied the floor lower limit and the limits to establishment of analytical equations. In ad the sensitivity of factors governing non-uniformity in diffuser air velocit analyzed, and a method for adjustment to make air velocity profiles more un as well as a simple design technique to determine the diameter of air d outlets and the height of air supply chambers, were proposed.

2. Construction of an Analytical Model

The assumptions for the purpose of constructing an analytical model w follows;

Assumption (1) : The pressure within the chamber is higher than the air pr in the room. (condition of continuity)

Assumption (2) : An equal diffused air velocity is obtained at all diffuser o (continuous uniform division branch model)[2)]

Assumption (3) : The diffused air velocity is proportional to the 1/2 power pressure difference between the chamber and the room. (turbulent flow mod

Assumption (4) :The air velocity in the chamber is distributed only in the l direction. (one-dimensional model)

Assumption (5) : A state of thermal uniformity obtains within the chamber. (isothermal model)

2.1 Mass Balance

Physical parameters such as the height of the chamber and the diame diffuser outlet, air flow and pressure loss are indicated in Figure 1. Fro mass balance, we obtain the following relations;

Fig. 1 Analytical model for supply chamber and diffusers.

Diffuser aperture area: $\Sigma s = RK \cdot (\pi \cdot d^2 / 4) \cdot M \cdot N$ --

Diffuser aperture ratio: $\kappa = \Sigma s / (L \cdot W)$ --

Ratio of air path cross-section to floor area: $\beta = (D \cdot W) / (L \cdot W)$ --

Neck air velocity of chamber: $U = (Q/3600) / \beta$ --

Diffuser air velocity: $v = \beta \cdot U / \kappa$ --

2.2 Momentum Balance

Taking the coordinate origin at the bulkhead, we use Equation (approximate the air velocity at position x in the chamber length direction.

U,, = x. U/'L ------- (6)

The fluid momentum in the interval from x to x+dx is as in Equation (7).
(R,+dPx)+(P -12). (U,,+dU.)2 = R,+(p //2). U,2+(p /,2). A - (dx,-'DE). U,2
+ (main stream branch loss) ------- (1)

The equivalent diameter of chamber3) and tube friction coefficient,') are computed using Equation and Equation (9). Here **E** *and Re* are the relative roughness and the Reynolds number, respectively.

DE = 1. 3. [(W. D)F).,.' (W+D)210.12.,-, ------ (8)
A = 0.0055.[1+(20000. ------- (9)

The momentum balance between the origin and position x is obtained byintegrating Equation (7) showed as Equation (10).

f dP, = (p,-"2)-(U//L)2 f[A X2.'Di-2x1dx
+ (sum of main stream branch losses) ------- (10)

In Equation (5), the sum of the main stream branch losses can be approximated by zero.` Hence pressure difference within the chamber corresponding to the ition of the nth diffuser outlet is give Equation(l 1).

posi 1 1

P.-Po = [A n.p,-'(3.Di,)-1].(p,-'2).(n.p/L)2.U2 ------- (1 1)

On the other hand, the position of appearance of the minimum pressure is obtained from the condi dP.,-'dn = 0, and becomes X.i. = **ni.p =** 2 D E,/ A . When 0 <X .. i.< L, the position of minimum pres occurs within the chamber; if we suppose that the inth diffuser outlet corresponds to the positio occurrence of the **minimum** pressure, then the pressure difference within the **chamber** correspondin the position of the inth outlet is as iven by Equation (12).

P.,-Po = [A m.p,-'(3.DF,,) -11.(p,,2).(m.p,-'L)2.U2 ----- (12)

The interval from the coordinate origin to the first diffuser outlet can be approximately given p therefore, the necessary pressure difference between the bulkhead and the room to obtain a des diffuser air velocity is given bs, Equation (13).

po-p, = (1 + (@ oHP,-'2).V2

+ [.@ 2 + A -p,,DE1'(P @,2)-(p,-'2,,L)2.U2 + Pf(V) ---- (13)

At the position of the inth diffuser outlet, Assumption (1) for construction of **an** analytical model obt and the necessary pressure difference between the nith

$$U_x = x \cdot U / L$$ ------

The fluid momentum in the interval from x to x+dx is as in Equation (7).

$$(P_x+dP_x)+(\rho/2)\cdot(U_x+dU_x)^2 = P_x+(\rho/2)\cdot U_x^2+(\rho/2)\cdot\lambda\cdot(dx/D_E)\cdot U_x^2 + \text{(main stream branch loss)}$$ -----

The equivalent diameter of chamber[3] and tube friction coefficient computed using Equation (8) and Equation (9). Here ε, and Re are the re roughness and the Reynolds number, respectively.

$$D_E = 1.3\cdot[(W\cdot D)^5/(W+D)^2]^{0.125}$$ ------

$$\lambda = 0.0055\cdot[1+(20000\cdot\varepsilon/D_E+106/Re)^{1/3}]$$ -----

The momentum balance between the origin and position x is obtain integrating Equation (7) and showed as Equation (10).

$$\int dP_x = (\rho/2)\cdot(U/L)^2 \int[\lambda\, x^2/D_E-2x]dx + \text{(sum of main stream branch losses)}$$ -------

In Equation (5), the sum of the main stream branch losses can be approxi by zero.[2] Hence the pressure difference within the chamber corresponding position of the nth diffuser outlet is given by Equation(11).

$$P_n-P_0 = [\lambda\cdot n\cdot p/(3\cdot D_E)-1]\cdot(\rho/2)\cdot(n\cdot p/L)^2\cdot U^2$$ -------

On the other hand, the position of appearance of the minimum pressu obtained from the condition $dP_n/dn = 0$, and becomes $X_{min} = m\cdot p = 2\cdot D$ When $0 < X_{min} < L$, the position of minimum pressure occurs within the cham we suppose that the mth diffuser outlet corresponds to the position of occurre the minimum pressure, then the pressure difference within the cha corresponding to the position of the mth outlet is as given by Equation (12).

$$P_m-P_0 = [\lambda\cdot m\cdot p/(3\cdot D_E) -1]\cdot(\rho/2)\cdot(m\cdot p/L)^2\cdot U^2$$ -------

The interval from the coordinate origin to the first diffuser outlet ca approximately given p/2, therefore, the necessary pressure difference bet the bulkhead and the room to obtain a desired diffuser air velocity is giv Equation (13).

$$P_0-P_r = (1+\zeta_0)\cdot(\rho/2)\cdot v^2 + [\zeta_2+\lambda\cdot p/D_E]\cdot(\rho/2)\cdot(p/2/L)^2\cdot U^2 + P_f(v)$$ ------

At the position of the mth diffuser outlet, Assumption (1) for construction analytical model obtains, and the necessary pressure difference between the diffuser outlet and the room to obtain a desired air flow velocity is as in Equation (14).

P. - P, = (1 + @ o). (p ,2). V2
+ [1@ 2 +A . p,-'DE - 11'(P -12). (m. p,-'Q2. U2 + Pf (V) ------ (14)

Depending on the position of occurrence of the minimum pressure, the following two types of distribution of pressure differences between diffusers and a room may appear. Pressure differences between the nth diffuser outlet and the room when X.@.i.@IL is o the form shown at the top in Figure 2, and iven by the following 91 equation.

(P.-R) = *(R'-Po)* + W(-P,) ------ (15)

On the other hand, the pressure difference between the nth diffuser outlet **and** the ro when 0 <Xi.< L is as shown at the bottom of Figure 2, and given by Equation (16).

(P.-P,) = (P.-po) + (P()-P") + (P,,, - P,) ------ (16)

PO

PO

Ai rf 1 ow

pl-p@ P@ Bulkhead ('D ==> P,-P, p 0 p Bulkhead

p

Airflow Q~

PO- P,

P..- p

P@-P, P, P,

Neck 1 Neck -P, 1

Distance Distance

Fig. 2 Two typical necessary pressure difference proftles between diffusers an room.

The diffuser air velocity at the nth position may be computed using the following equati appl ing Assumption (3), given the pressure difference between the nth Y1

diffuser and the room as computed using analytical models.

v. = C i). @2. (P. - P,)/,- 0 @ 1/2

In order for the continuity condition to hold, the air velocity at each diffuser must coin with the average of the diffuser air velocity as determined from the analytical model

The constant of proportionality C[) is chosen such that the average value of the compu air velocities coincides with the air velocities at each diffuser obtained from the supply flow.

As the outlet ollifice loss coefficient, the value 2.4, equal to the average for the

diffuser outlet and the room to obtain a desired air flow velocity is as in Eq
(14).

$$P_m - P_r = (1 + \xi_0)\cdot(\rho/2)\cdot v^2 + [\xi_2 + \lambda\cdot p/D_E - 1]\cdot(\rho/2)\cdot(m\cdot p/L)^2\cdot U^2 + P_f(v) \quad \text{------}$$

Depending on the position of occurrence of the minimum pressure, the foll two types of distribution of pressure differences between diffusers and a roor appear. Pressure differences between the nth diffuser outlet and the room $X_{min} \geqq L$ is of the form shown at the top in Figure 2, and given by the foll equation.

$$(P_n - P_r) = (P_n - P_0) + (P_0 - P_r) \quad \text{-----}$$

On the other hand, the pressure difference between the nth diffuser outle the room, when $0 < X_{min} < L$ is as shown at the bottom of Figure 2, and giv Equation (16).

$$(P_n - P_r) = (P_n - P_0) + (P_0 - P_m) + (P_m - P_r) \quad \text{------}$$

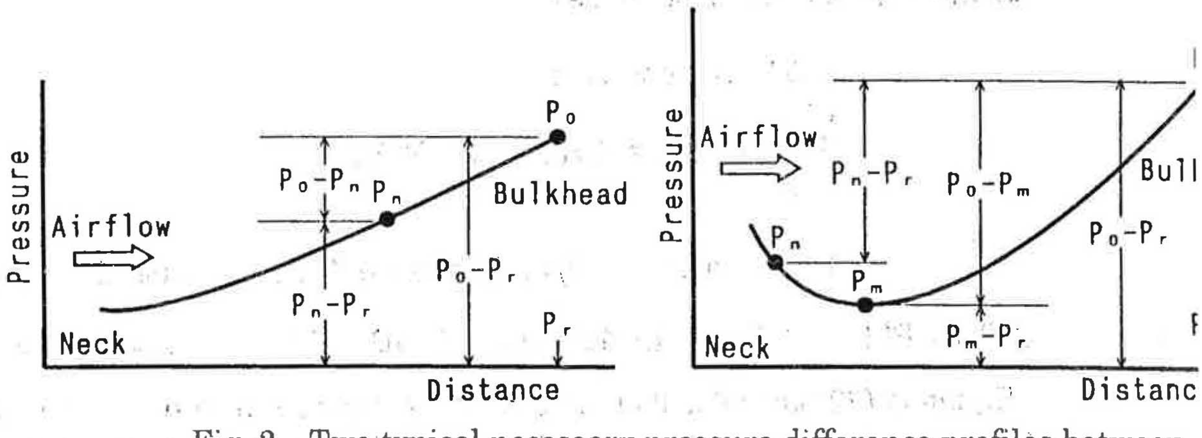

Fig. 2 Two typical necessary pressure difference profiles between diffusers and a room.

The diffuser air velocity at the nth position may be computed using the follo equation applying Assumption (3), given the pressure difference between th diffuser and the room as computed using analytical models.

$$v_n = C_D\cdot\{2\cdot(P_n - P_r)/\rho\}^{1/2} \quad \text{-------}$$

In order for the continuity condition to hold, the air velocity at each dif must coincide with the average of the diffuser air velocity as determined fron analytical model v_n. The constant of proportionality C_D is chosen such tha average value of the computed air velocities coincides with the air velociti each diffuser obtained from the supply air flow.

As the outlet orifice loss coefficient, the value 2.4, equal to the average fo path based on the resistance of the flow passing through the outlet, was employed.,5) As branch loss coefficient, the value 1.1, equal to the average for the path based on the branch l for a uniform-flow tube, was employed.';) The air velocity per diffuser outlet obtained from supply air flow is equivalent to the design air flow assuming that the air flow is divided equu among all diffusers. This air velocity is denoted by v(design); using the design air flow reference, the absolute value of the deviation in the diffuser air velocity is defined to be the n uniformity in the diffuser air velocity. Its maximum value is further defined to be the maxi non-uniformity.

NU. = 1 v.-v(design) 1 -'v(design).100 ------ (18)

The irregularity in the diffuser air velocity was adjusted by adding filters of equal resista uniformly to all diffusers.
The filter resistance is expressed as a function of the diffuser air velocity. Here ider de convenience in using Pf o to denote the filter resistance at a

we cons 1 1
diffuser air velocity vo = 1 [m/s]. Hence the filter resistance for an arbitrary air velocity is follows;

Pf (v) = *Pf* o - (V,,_1V02 ------ (19)

2.3 Range of Validity of Analytical Model

Equation (20) and Equation (21) express the conditions of mass and momentum balance at chamber neck and diffuser outlets.

p D.W.U = p v. TE s --------
(20)
Pi+(p,-'2).U2 = P.+(p,/2).V2+Pf(V) -------
(21)

From Equation (20) and Equation (2 1), we obtain Equation (22).

@P,1 + Pf (v) - P i@ (p -12). U2. @ 1 - (D - W,/' Z s)2@ (p.,2)-U2-11-(,8,/K)2@ ------- (22)

Equation (22) indicates that the sign of the pressure difference between the chamber neck diffuser outlets is reversed at (D. W,-' 7 s) = 1. The condition for air to flow from the neck tow diffuser outlets is Pi @@'@P., + Pf (v)@. The condition (D. W,-' 7_ s) 1 does not sat Assumption (1), and so the condition that the ratio of the air path cross-sectional area at the ne the total area of the diffuser outlets be greater than unity must be met for the analytical **model** to obtain.

3. Analysis of Non-uniformity in Air Flow **Velocities**

We define the unit air volume as the air flow required to accommodate the air conditioning load per unit floor area. Diffuser air flow velocities were analvzed path based on the resistance of the flow passing through the outlet employed.[5] As the branch loss coefficient, the value 1.1, equal to the avera the path based on the branch loss for a uniform-flow tube, was employed.[6] air velocity per diffuser outlet obtained from the supply air flow is equival the design air flow assuming that the air flow is divided equally amo diffusers. This air velocity is denoted by v(design); using the design air fl reference, the absolute value of the deviation in the diffuser air velocity is d to be the non-uniformity in the diffuser air velocity. Its maximum va further defined to be the maximum non-uniformity.

$$NU_n = | v_n - v(design) | / v(design) \cdot 100 \quad \text{------}$$

The irregularity in the diffuser air velocity was adjusted by adding filt equal resistance uniformly to all diffusers.

The filter resistance is expressed as a function of the diffuser air velocity. we consider design convenience in using P_{f0} to denote the filter resistance diffuser air velocity v_0 = 1 [m/s]. Hence the filter resistance for an arbitra velocity is as follows;

$$P_f(v) = P_{f0} \cdot (v/v_0)^2 \quad \text{------}$$

2.3 Range of Validity of Analytical Model

Equation (20) and Equation (21) express the conditions of mass and mome balance at the chamber neck and diffuser outlets.

$$\rho \cdot D \cdot W \cdot U = \rho \cdot v \cdot \Sigma s \quad \text{-------}$$

$$P_l + (\rho/2) \cdot U^2 = P_n + (\rho/2) \cdot v^2 + P_f(v) \quad \text{-------}$$

From Equation (20) and Equation (21), we obtain Equation (22).

$$\begin{aligned} \{P_n + P_f(v) - P_l\} &= (\rho/2) \cdot U^2 \cdot \{1 - (D \cdot W / \Sigma s)^2\} \\ &= (\rho/2) \cdot U^2 \cdot \{1 - (\beta / \kappa)^2\} \quad \text{-------} \end{aligned}$$

Equation (22) indicates that the sign of the pressure difference betwee chamber neck and diffuser outlets is reversed at $(D \cdot W / \Sigma s) = 1$. The cond for air to flow from the neck toward diffuser outlets is $P_l \geqq \{P_n + P_f(v)\}$. condition $(D \cdot W / \Sigma s) \leqq 1$ does not satisfy Assumption (1), and so the cond that the ratio of the air path cross-sectional area at the neck to the total area o diffuser outlets be greater than unity must be met for the analytical mod obtain.

3. Analysis of Non-uniformity in Air Flow Velocities

We define the unit air volume as the air flow required to accommodate th conditioning load per unit floor area. Diffuser air flow velocities were ana for the range of values appearing in Table 1.

Table 1 . Dimensions and the given range of physical parameters.

Physical Parameter	Symbol	Unit	Quantity		
unit supplied air volume	0	m'/m'.h	30	40	50
width of chamber	W	m		4.8	
height of chamber	D	m	0.05	-	0.30
length of chamber	1	m	7.20	-	36.0
effective opening ratio	RK			0.4	
pitch of diffuser	p	m		*1.2*	
diameter of diffuser	d	m	0.10	-	0.30
density of air	p	kg/m'	1.217 at 17T		
outlet loss coefficient	0			2.4	
branch loss coefficient					

100

-w 36. 0 m

--c 80

28.8 m

60

-A- 21. 6 *m*

40

-c 14.4 m

-v- 7. 2 m

20

0 -

A

0 50 100 150 *200* 250

80

-w 36. 0 m 4. 28.8 m 60 **-A-** 21.6 m

E

14.4 m

40 7. 2 m

c

>< *20*

0

0 50 100 150*200* 250

D mm

Fig. 3 Relation between NUsi.,, and Xmin at various chamber heights. (d = 200 = 30)

In general, the effective aperture of a diffuser outlet varies with the outlet diameter, but here take all apertures to be 0.4.7) In consideration of the

'bility of rupture of the caulking in floor seams, the upper limit to chamber possi 1 1

pressurization was set at 50 **Pa.8)**

Figure 3 shows an example of analysis of the maximum non-uniformity resulting when the diffuser o resistance is not adjusted using filters. **When** X.in>_L, the dynamic pressure near the neck area is h and diffuser air flows are lower than the design values. As a result the maximum non-uniformity in for the range of values appearing in Table 1.

Table 1 . Dimensions and the given range of physical parameters.

Physical Parameter	Symbol	Unit	Quantity
unit supplied air volume	Q	$m^3/m^2 \cdot h$	30 40 50
width of chamber	W	m	4.8
height of chamber	D	m	0.05 ～ 0.30
length of chamber	L	m	7.20 ～ 36.0
effective opening ratio	RK	-	0.4
pitch of diffuser	p	m	1.2
diameter of diffuser	d	m	0.10 ～ 0.30
density of air	ρ	kg/m^3	1.217 at 17°C
outlet loss coefficient	ζ_0	-	2.4
branch loss coefficient	ζ_2	-	1.1

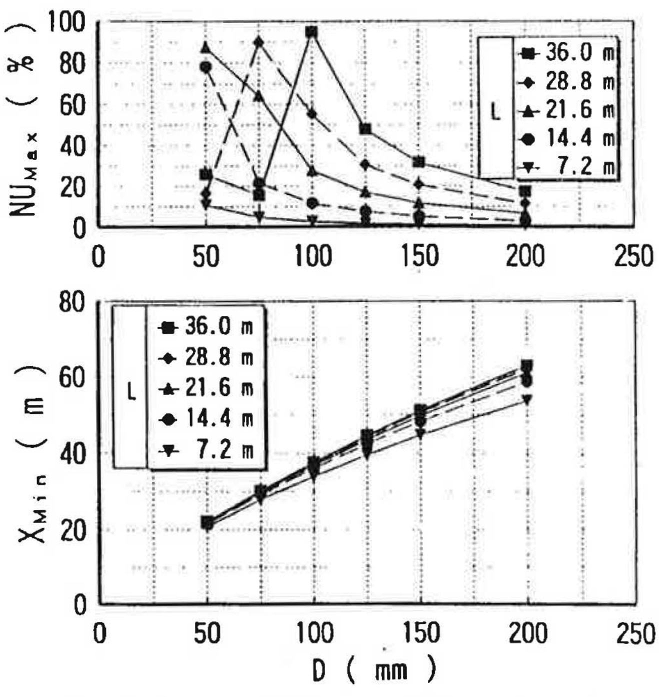

Fig. 3 Relation between NU_{Max} and X_{min} at various chamber heights. (d = 200, Q = 30)

In general, the effective aperture of a diffuser outlet varies with the diameter, but we here take all apertures to be 0.4.[7] In consideration of possibility of rupture of the caulking in floor seams, the upper limit to cha pressurization was set at 50 Pa.[8]

Figure 3 shows an example of analysis of the maximum non-unifo resulting when the diffuser outlet resistance is not adjusted using filters. $X_{min} \geqq L$, the dynamic pressure near the neck area is high, and diffuser air are lower than the design values. As a result the maximum non-uniform diffuser air velocity appears in the neck area. And,.the lower the chamber height. the greater the irregularity. On the other hand, **when 0** <X.i.< L the lionuniformity is maximum at the positi of occurrence of the pressure **minimum.** When the position of occurrence of **minimum** press **is** exceeded the lionuniformity declines, and then increases once again.

m 80

l- 60

30 m31m2. h

--v - 40 m3/m2. h

40

50 m3/m2. h

20

0

0 *20* *40* 60 80

NUm.

80

60

-m- 2.5%

5 %

40

10 %

20

o

M

4@

Cn **0** **50** **100** 150 *200* 250

D mm)

Fig. 4 Relation between necessary static pressure of a blower and chamber height, when NU,@t,@,, was controll

(Up: D = 100, d = 200, L = 36), (Down: d = 200, L = 36, Q = 30)

When filters with the same resistance value are installed on all diffusers, the blower static pressure required to obtain the tar diffuser air velocity is as ill Figure 4. The greater the increase in unit air volume, the higher is the pressure required of the blo Further, the longer the chamber, the higher the pressure that must be generated by the blower. When the chamber height is mm or less, the static pressure required of the blower tends to rise rapidly. In view of the pressure limit imposed on the cham and the rate of increase of the static pressure required of the blower, the lower limit to the floor height is thought to be 50 mm.

4. Analysis of Sensitivity to Factors Governing Non-uniformity

As shown in Equation (23), the ratio of the pressure loss at the diffuser to the dynamic pressure at the neck in the chamber can define as the diffuser resistance coefficient.2)

K o. (p -12). V2 + Pf (v)@,-'(p..,2)U2 -------- (23)

Equation (23) can rewrite as follows from Equation (5) and Equation (19).

diffuser air velocity appears in the neck area. And, the lower the chamber the greater is the irregularity. On the other hand, when 0 <X_{min}< L th uniformity is maximum at the position of occurrence of the pressure min When the position of occurrence of minimum pressure is exceeded th uniformity declines, and then increases once again.

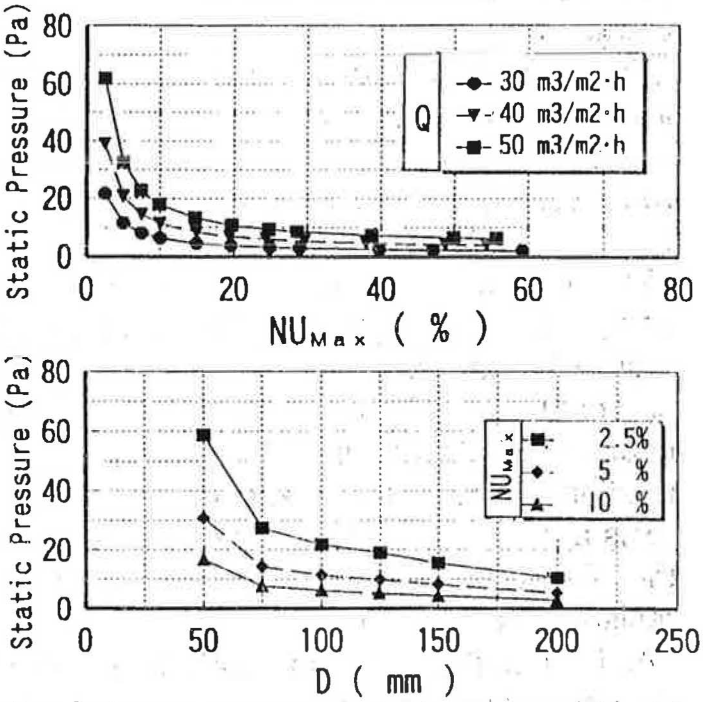

Fig. 4 Relation between necessary static pressure of a blower and chamber height, when NU_{Max} was controlled. (Up: D = 100, d = 200, L = 36), (Down: d = 200, L = 36, Q = 30)

When filters with the same resistance value are installed on all diffuser blower static pressure required to obtain the target diffuser air velocity is Figure 4. The greater the increase in unit air volume, the higher is the pre required of the blower. Further, the longer the chamber, the higher the pre that must be generated by the blower. When the chamber height is 100 r less, the static pressure required of the blower tends to rise rapidly. In view pressure limit imposed on the chamber and the rate of increase of the pressure required of the blower, the lower limit to the floor height is thought 50 mm.

4. Analysis of Sensitivity to Factors Governing Non-uniformity

As shown in Equation (23), the ratio of the pressure loss at the diffuser t dynamic pressure at the neck in the chamber can define as the diffuser resis coefficient.[2)]

$$K = \{\zeta_0 \cdot (\rho/2) \cdot v^2 + P_f(v)\} / (\rho/2) U^2 \quad \text{-------}$$

Equation (23) can rewrite as follows from Equation (5) and Equation (19).

+ Pf o,-'(p @,2)/,"V(2@ - (,8 -'K)2

· **+ Pf o,-'(p @,2),,'V(2@. [(4,-'(;rr .** RK)@. (D,-'L). **(p,-'d)212 ----** (24)

As Equation (24) indicates, the diffuser resistance coefficient is unrelated to the 't air volume or to the chamber width. Factors governing the diffuser

uni 1

resistance coefficient include the chamber height, chamber length, diffuser outlet **pitch and** diameter, effective aperture, and diffuser filter resistance. Of these. the factors to which the resistance coefficient **is** most sensitive are the diffuser pitch and diameter.

The effect of factors governing the diffuser resistance coefficient on air flo-,s,, irregularity was studied. Filters were used to regulate the non-uniformity of the diffuser air flow; the relation between the maximum non-uniformity and the reciprocal of the diffuser resistance coefficient for various unit air volumes appears in Figure 5.

The chamber height tends to have a different effect at 75 mm and less compared with heights of 100 mm and above. This is because, as indicated in Figure 3, the **minimum** pressure position occurs within the chamber when the chamber height is 80 mm or less; the pressure distribution within the chamber changes at this height. If we suppose that for K>- 1 there is a proportional relation between the maximum non-uniformity and the reciprocal of the diffuser resistance coefficient. then the factors governing the maximum non-uniformity of the diffuser air velocity will be identical to the factors comprising the diffuser resistance coefficient.

As indicated by Equation (24), the diffuser resistance coefficient is unrelated to the unit air volume and the chamber width; but from Figure 5, the effect of the unit air volume is less pronounced than that of the other factors. Among the factors to which the maximum non-uniformity **is** highly sensitive, the diffuser outlet diameter, effective aperture and chamber length are given quantities at design time; hence the diffuser pitch, chamber height, and filter resistance remain to be selected dulling designing.

80

30 m3/m2. h

60 *40 m31m2.* h

-k 50 m3/m2.h

40

20

W-A-A

0

2 3 4 K

Fig. 5 Relation between NUrj.. and K-' at various unit air volumes.
(D = 100, d = 200, L = 36)

$$K = \{\zeta_0 + P_{f0}/(\rho/2)/v_0^2\}\cdot(\beta/\kappa)^2$$
$$= \{\zeta_0 + P_{f0}/(\rho/2)/v_0^2\}\cdot[\{4/(\pi\cdot RK)\}\cdot(D/L)\cdot(p/d)^2]^2 \quad \cdot\cdot$$

As Equation (24) indicates, the diffuser resistance coefficient is unrelated unit air volume or to the chamber width. Factors governing the d resistance coefficient include the chamber height, chamber length, diffuser pitch and diameter, effective aperture, and diffuser filter resistance. Of the factors to which the resistance coefficient is most sensitive are the d pitch and diameter.

The effect of factors governing the diffuser resistance coefficient on ai irregularity was studied. Filters were used to regulate the non-uniformity diffuser air flow; the relation between the maximum non-uniformity ar reciprocal of the diffuser resistance coefficient for various unit air vo appears in Figure 5.

The chamber height tends to have a different effect at 75 mm and less com with heights of 100 mm and above. This is because, as indicated in Figure minimum pressure position occurs within the chamber when the chamber l is 80 mm or less; the pressure distribution within the chamber changes a height. If we suppose that for K≧1 there is a proportional relation betwe maximum non-uniformity and the reciprocal of the diffuser resistance coeff then the factors governing the maximum non-uniformity of the diffus velocity will be identical to the factors comprising the diffuser resis coefficient.

As indicated by Equation (24), the diffuser resistance coefficient is unrela the unit air volume and the chamber width; but from Figure 5, the effect unit air volume is less pronounced than that of the other factors. Amon factors to which the maximum non-uniformity is highly sensitive, the di outlet diameter, effective aperture and chamber length are given quantit design time; hence the diffuser pitch, chamber height, and filter resistance re to be selected during designing.

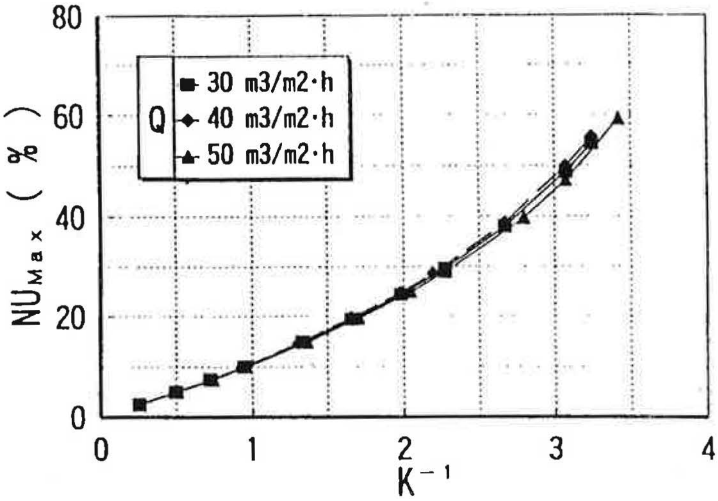

Fig. 5 Relation between NU_{Max} and K^{-1} at various unit air volumes.
(D = 100, d = 200, L = 36)

If this proportional relation is used for K@-> 1, then by specifying the maximum non-uniformity it becomes easy to choose a diffuser pitch, chamber height, and filter resistance values to adjust the non-uniformity in the air velocity proffle.

5. Experimental Validation of Analytical Model

Using a scale model of the air supply chamber, the validity of the analytical model for determining non-uniformity in diffuser air velocity profiles was studied. 5.1 Experimental Apparatus and Experimental Conditions
Due to constraints on the size of the site for experiments, the validity of the analytical model was examined using a 1: 5 size scale model of the air supply chamber. The similarity of air flow in an actual system and in the scale model depends on agreement of the respective Reynolds numbers.

= 1 ----- (25)

Diff. Pressure Gauge mper

Anemometer

Meter

Chamber(scale model) Fan

Distributor

Fig. 6 Schematic diagram of experimental apparatus for scale model test.

However, the air flow velocities and pressures in a 1: 5 scale model are increased to 5 times and 25 times their values respectively in an actual system, which is not practical from the standpoint of chamber pressure resistance. The air flow is isothermal, and the Reynolds number at actual diffuser outlets is on the order of 10,000. In other words, because the air flow is well-developed turbulent flow, the conditions for self-similar flow obtain. The average Reynolds number for diffuser air flow in the model is of order 5,000, so that self-similar flow conditions obtain here as well. Hence the similarity conditions are relaxed, and for the model air flow a scale of unity was adopted.9)
Figure 6 is a schematic diagram of the model apparatus; the conditions of the model experiments appear in Table 2. The diameter of the model diffuser outlets was set at 20 mm, in consideration of an effective aperture of 0.4. After confirming that there were no leaks at Joints and points of connection with perforated plates in the model, an Annubar Werential-pressure flow meter was used to measure the supplied air flow. Pressure differences were measured using a Benz manometer with a precision of 4- 1 Pa. Air flow velocities were measured

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$$(Re/Re_M) = (d\cdot v/\nu)/(d_M\cdot v_M/\nu) = 1 \quad ----$$

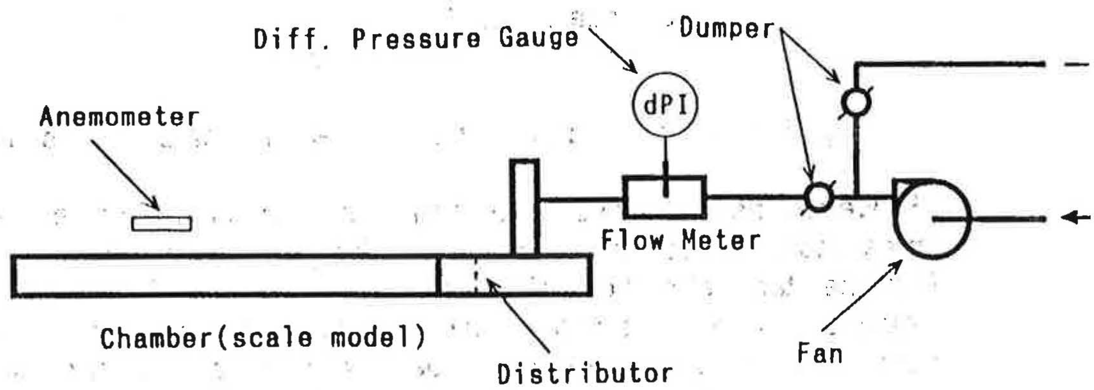

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Table 2. Dimensions of experimental equipment and conditions at 1:5 scale.

Physical Parameter	Symbol	Unit	Quantities(real scale)
width of chamber	WM	m	1.2 (6.0)
diameter of diffuser	d,	mm	20 (200)
effective opening ratio	RG		1 (0.4)
-@e@ight ---of chamber	L		15(75) 30(150
length of chamber	LM	m	*2(10)* 3 (15) 4 (20)
unit supplied air volume	QM	m^3/ml.h	30 50
pitch of diffuser	PM	mm	400 300

Perforated plates were installed near the chamber neck, and by rectifying the supplied air, velocity component in the chamber width direction was eliminated in so far as possible. Th velocity distribution in the chamber width direction was measured near the neck, and straightening effect of the perforated plates was confirmed. Non-uniformity in the air velo profile in the chamber length direction were computed from the air flow velocities measu along the center line of the chamber.

5.2 Comparison of Analysis and Experimental Results

The air flow velocity immediately above diffuser outlets is made unstable by the inductive ef of 'et. Air flow velocities were measured at positions at which

J 1

stable air flow measurement was possible, namely above the central line through the diff outlets, but removed a distance of 10 mm from a diffuser outlet plane. However, because of inductive effect of jet flow, the measured result differs from the true value of the air velo Therefore, we took the air velocity per diffuser as computed from the supplied air flow to be design air velocity, and computed correction coefficients for measurement positions from ratio of the average of the measured diffuser air velocities to the design air velocities; u these, the measured air velocities were converted into air velocities directly above the diff outlet using the following equation.

[Air Velocity Directly above Diffuser]
-'[Measured Air Velocity],-'[Measurement Position Correction Factor] ---- (26)

Because calculated values include a measurement error of $\pm$ 0.3 m/s, measurement posi correction factors were similarly used for conversion into measurement errors directly ab diffusers.

The velocity distribution in the chamber width direction at the neck appears in Figure 7. The air flow velocities were computed as averages of five measurements at each of the measurement positions. Due to the straightening effect of the perforated plates, within the range of the experimental conditions,

using multiple point hot-wire anemometers (KANOMAX model
Measurement errors over the measurement range determined using a wind
for calibration were within ±0.3 m/s.

Table 2. Dimensions of experimental equipment and conditions at 1:5 s

Physical Parameter	Symbol	Unit	Quantities(real scale)	
width of chamber	W_M	m	1.2 (6.0)	
diameter of diffuser	d_M	mm	20 (200)	
effective opening ratio	RK_M	-	1 (0.4)	
height of chamber	D_M	mm	15(75)	30(150)
length of chamber	L_M	m	2(10) 3 (15) 4 (20)	
unit supplied air volume	Q_M	$m^3/m^2 \cdot h$	30	50
pitch of diffuser	P_M	mm	400	300

Perforated plates were installed near the chamber neck, and by rectifyir
supplied air, the velocity component in the chamber width direction
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effect of the perforated plates, within the range of the experimental condit

deviations from average values were in all cases within 4- 10 %. This confirmed validity of Assumption (4).

4

3

2.h 2.h

>

Aye.

0

0 0.20.4 0.6 0.8 1 1.2

Distance m)

Fig. 7 Vertical diffuser air velocity profile on the neck in the scale and the effect a distributor by perforated panels. (1)m = 30,Lm = 4,W-,@i = 1.2)

The experimental results as converted into air velocities directly above diffusers, and air velocity profile in the chamber length direction as computed using the analytical mo appear in Figure 8. The experimental conditions were in all cases such that ((D.W,,' T >I, satisfying this condition on the validity of the analytical model.

5

Expe r i ment

4.5 - - - --------------- -Calculation(uwer)

Calculation(lower)

> 4

0

0 2 3 4

5

In 4.5

P=

> 4

Fig. 8 Comparison with the ex-

o lr

periment and calculation on longitudinal diffuser

01 2 3 4

Distance m

air velocity profile from
the neck to the bulkhead at Dm = 30, Qxi = 30. (up: Lm = 2, down: L
4)

The experimental and analytical results reveal an increase in air velocity due to

deviations from average values were in all cases within ± 10 %. This con[...]
the validity of Assumption (4).

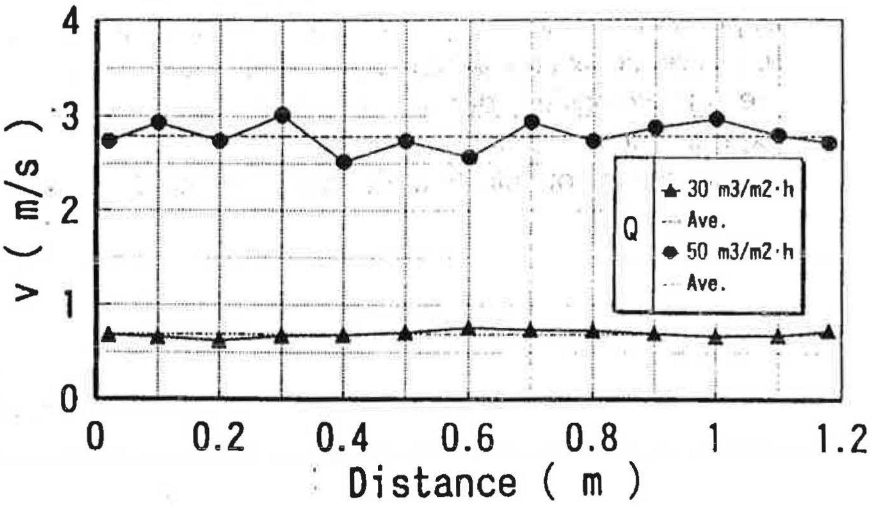

Fig. 7 Vertical diffuser air velocity profile on the neck in the scale and th[...] effect as a distributor by perforated panels. ($D_M = 30, L_M = 4, W_M =$ [...]

The experimental results as converted into air velocities directly above diff[...] and the air velocity profile in the chamber length direction as computed usi[...] analytical model, appear in Figure 8. The experimental conditions were [...] cases such that $((D \cdot W / \Sigma s) > 1$, satisfying this condition on the validity [...] analytical model.

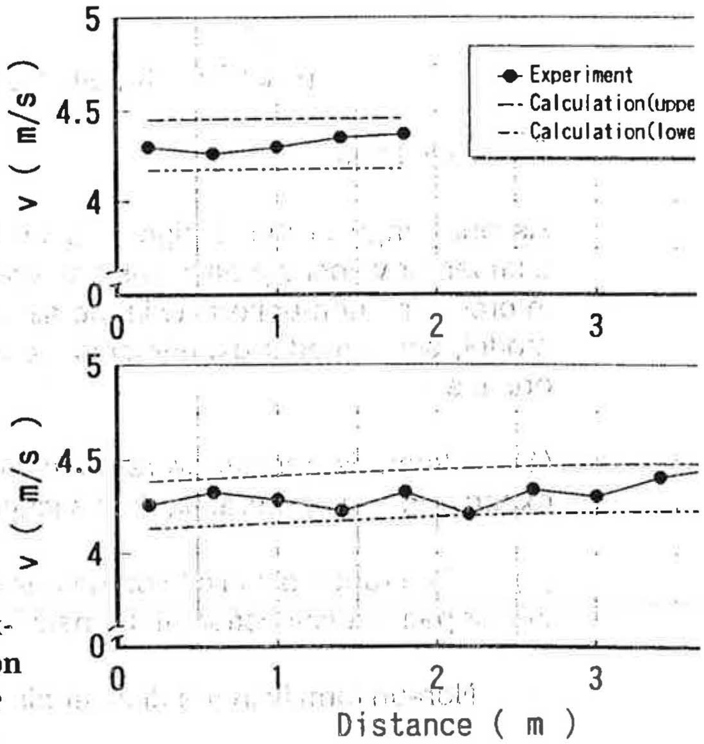

Fig. 8 Comparison with the experiment and calculation on longitudinal diffuser air velocity profile from the neck to the bulkhead at $D_M = 30$, $Q_M = 30$. (up: $L_M = 2$, down: $L_M =$ [...]

The experimental and analytical results reveal an increase in air velocity d[...]

reacquisition of static pressure from the neck area to the bulkhead. The range of measurement error was set for analysis results, but in any event the results are within the range of measurement error, thus confirming the validity of the analytical model. In order to verify experimentally the conditions for validity of the analytical model, the chamber height of the model was changed from 15 to 10 mm, the pitch of the diffusers was reduced from 300 to 200 mm, and air velocities were measured under these new conditions which deviated from the range for which the analytical model applies. Experimental and analytical results obtained for (D. W,-' 1 s) = 0.319 appear in Figure 9. No reacquisition of static pressure is observed from the neck to the bulkhead, and the air velocities decline uniformly. Moreover, the air velocity profiles of the experimental and analytical results are reversed. These experiments thus confirmed the conditions of validity of the analytical model.

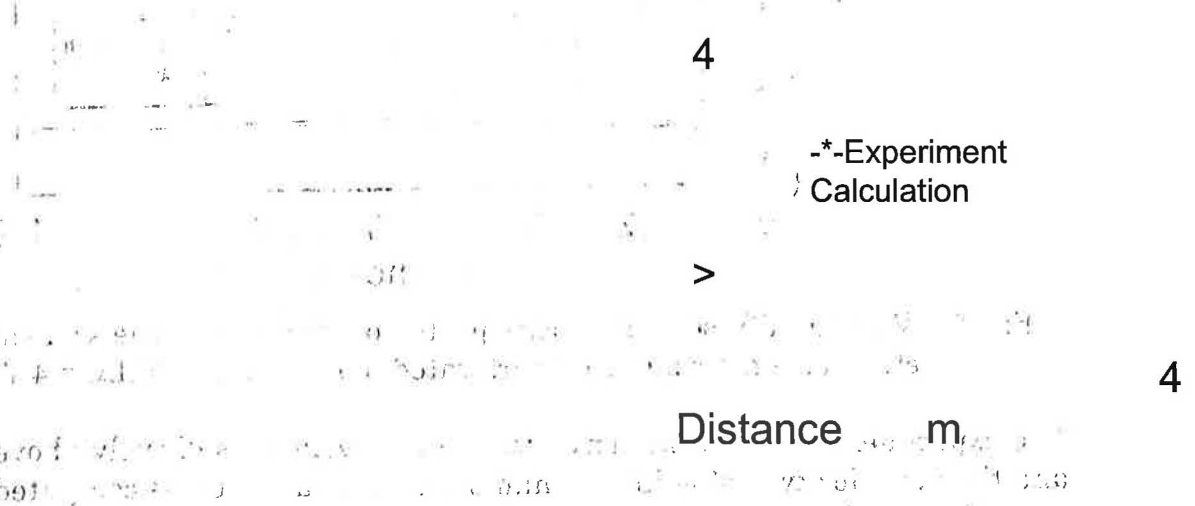

Fig. 9 Comparison with the experiment and the calculation result on lon 'tudinal diffuser air velocity profile at (D.W,,,' 2 s) = 0.319.

91

(D,@@i = 10, pm = 200, Lm = 4, Qm = 48, Wm = 1.2)

6. Conclusions

As one stage in the design of a pressurized under-floor air-conditioning system, the diffuser air velocity profile was analyzed, and an analytical model was proposed for use in predicting non-uniformity in the air velocity profile. In order to verify the validity of the model, scale model experiments were performed, and the following conclusions were obtained.

(1) Analysis results were in good coincidence With values obtained in model experiments, corroborating the validity of the analytical model.

(2) The range of validity of the analytical model is that range of parameters for which the air path cross-section at the neck is greater than the total area of all diffuser outlets.

(3) Non-uniformity in the diffuser air velocity profile for chamber heights of 100

reacquisition of static pressure from the neck area to the bulkhead. The ra measurement error was set for analysis results, but in any event the resu within the range of measurement error, thus confirming the validity analytical model. In order to verify experimentally the conditions for vali the analytical model, the chamber height of the model was changed from 1 mm, the pitch of the diffusers was reduced from 300 to 200 mm, and air vel were measured under these new conditions which deviated from the ran which the analytical model applies. Experimental and analytical r obtained for (D·W／Σ s) = 0.319 appear in Figure 9. No reacquisition of pressure is observed from the neck to the bulkhead, and the air velocities d uniformly. Moreover, the air velocity profiles of the experimental and ana results are reversed. These experiments thus confirmed the conditions of v of the analytical model.

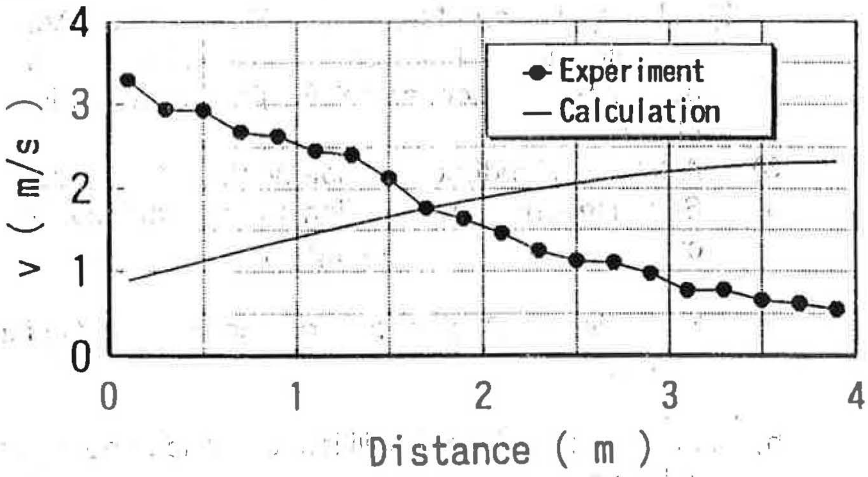

Fig. 9 Comparison with the experiment and the calculation result on longitudinal diffuser air velocity profile at (D·W／Σ s) = 0.319 (D_M = 10, p_M = 200, L_M = 4, Q_M = 48, W_M = 1.2)

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(3) Non-uniformity in the diffuser air velocity profile for chamber heights of

mm to 300 mm, for which there are performance records for numerous pressurized under-floor air-conditioning systems, were approximately 10 % or less. The lower limit for the chamber height was 50 mm.

(4) As a method of regulating non-uniformity in diffuser air velocity emerging as the floor height is lowered, installation of filters with same resistance value on all diffuser outlets is effective. This method **is** more energy-efficient as a means of regulating non-uniformity **in** air flow velocity than is the conventional method of installing dampers at floor difFusers and using the damper apertures to regulate air flow.

(5) It was shown that for diffuser resistance coefficients of unity or greater, the resistance coefficient is proportionally related to the maximum non-uniformity of the diffuser air velocity. Using this relation, the pressurized diffuser pitch, chamber height and filter resistance values can easily be chosen so as to adjust non-uniformity in the air flow velocity profile.

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Symbolls

D	Height of chamber [m 1	
DE	Equivalent diameter of chamber	m
d	Diameter of diffuser outlet [m	
K	Resistance coefficient of diffuser	
L	Length of chamber	m

m Number of diffusers in length direction

N	Number of diffusers in width direction
p	Pressure [Pa 1
Pf (v)	Filter resistance [Pa
p	Pitch of diffuser outletm
Q	Unit air volume [M3,,,M2.h]
RK	E5ective aperture of diffuser outlet
8	Area of diffuser [M21
u	Air speed at neck [m,,,,s 1
NU	Non-uniformity in supplied air velocity profile %
NUm..	Maximum non-uniformity [%
v	Diffused air velocity [m,,s
W	Width of Chamber [m 1
Xrnin	Position of appearance of minimum pressure m
16	:Diffuser aperture ratio [- 1
K	: Ratio of air path cross-section to floor area
A	:Tube friction coefficient [- 1
0	: Outlet orifice loss coefficient
2	:Branch loss coefficient [-
v	Air dynamic viscosity coefficient [M2,,,'S]
p	Air density [kg,-IM31

Subscripts

1	Neck
m	Scale model

m	inth number of diffuser outlet at which minimum pressure appears
·	nth number of diffuser outlet
·	Bulkhead
·	Room
·	Arbitrary distance

M	: Number of diffusers in length direction	[−]
N	: Number of diffusers in width direction	[−]
P	: Pressure	[Pa]
$P_f(v)$	: Filter resistance	[Pa]
p	: Pitch of diffuser outlet	[m]
Q	: Unit air volume	[$m^3/m^2 \cdot h$]
RK	: Effective aperture of diffuser outlet	[−]
s	: Area of diffuser	[m^2]
U	: Air speed at neck	[m/s]
NU	: Non-uniformity in supplied air velocity profile	[%]
NU_{Max}	: Maximum non-uniformity	[%]
v	: Diffused air velocity	[m/s]
w	: Width of Chamber	[m]
X_{min}	: Position of appearance of minimum pressure	[m]
β	: Diffuser aperture ratio	[−]
κ	: Ratio of air path cross-section to floor area	[−]
λ	: Tube friction coefficient	[−]
ζ_0	: Outlet orifice loss coefficient	[−]
ζ_2	: Branch loss coefficient	[−]
ν	: Air dynamic viscosity coefficient	[m^2/s]
ρ	: Air density	[kg/m^3]

Subscripts

1	: Neck
M	: Scale model
m	: mth number of diffuser outlet at which minimum pressure appear
n	: nth number of diffuser outlet
0	: Bulkhead
r	: Room
x	: Arbitrary distance