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DEVELOPMENT AND EVALUATION OF A NEW SCALING METHOD FOR PREDICTING NON-ISOTHERMAL ROOM VENTILATION FLOWS

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ABSTRACT

Scaling method for predicting room air motion with reduced scale models was investigated theoretically and experimentally with full and 1/4th scale test rooms. The critical Archimedes number, at which the diffuser air jet fell immediately after entering the room, was found to decrease when the room dimensions decreased. A new scaling method was proposed based on the relative deviation of Archimedes number from its critical value.

Preliminary evaluation of the new scaling method was conducted by comparison between the 1/4th scale tests and the corresponding full scale tests, which indicated that the new scaling method predicted well the overall room air flow patterns, distributions of mean velocity, temperature, and levels of turbulence intensity and turbulent kinetic energy in the occupied regions. Ways for improving the scaling method further are also proposed.

KEYWORDS

Ventilation, Air Distribution, Physical Modelling, Scaling Method.

INTRODUCTION

Similitude modelling of room air distribution is a useful tool for studying room air distribution and ventilation effectiveness. Using reduced scale models, a broad range of ventilation conditions can be investigated more conveniently and less expensively, and experimental results can be applied to rooms of different sizes. However, a proper scaling method is needed for the model tests in order to extrapolate the model results to different sized rooms quantitatively as well as qualitatively.

Scaling methods have been developed for isothermal and fully developed room ventilation flows (e.g., Pattie and Milne 1966, Timmons 1984, Baturin 1972 and Anderson and Mehos 1988), but a proper scaling method is still not available for predicting non-isothermal and low turbulence room airflows due to the difficulties involved (Moog 1981, Yao et al. 1986 and Christianson et al. 1988). Most realistic room ventilation flows are non-isothermal and involve low turbulence (Zhang 1990). Developing proper scaling methods for realistic room ventilation flows with internal heat sources and obstructions is one of the research needs in the studies of room air and air contaminant distributions (Int-Hout 1989).

The objectives of the present study was to develop a new scaling method for predicting room ventilation flows and evaluate the method with experimental measurements in an 1/4th scale model room and its full scale prototype.

automatically satisfied.

Reynolds Number: Re_d It represents the ratio of the inertial force to the viscous force. Similarity between the air motion within the model and its prototype requires:

$$\left(\frac{U_d w_d}{\nu}\right)_m = \left(\frac{U_d w_d}{\nu}\right)_p \tag{6}$$

Assuming $(\nu)_m=(\nu)_p$, we have

$$\frac{(U_d)_m}{(U_d)_p} = \frac{(w_d)_p}{(w_d)_m} = \frac{1}{n} \tag{7}$$

where, n is the geometric scale of the model relative to the prototype.

For a reduced scale model, $n<1$. Therefore, the diffuser air velocity in the model will be higher than in its prototype if one conducts model tests based on the diffuser Reynolds number.

Archimedes Number: Ar_{fd} It represents the ratio of inertial force to thermal buoyancy force. Similarity between the motions within the model and prototype requires:

$$\left[\frac{\beta g w_d (T_f - T_d)}{U_d^2}\right]_m = \left[\frac{\beta g w_d (T_f - T_d)}{U_d^2}\right]_p \tag{8}$$

Assuming $(T_f-T_d)_m=(T_f-T_d)_p$, $(\beta)_m=(\beta)_p$ and $(g)_m=(g)_p$, we have

$$\frac{(U_d)_m}{(U_d)_p} = \left[\frac{(w_d)_m}{(w_d)_p}\right]^{1/2} = n^{1/2} \tag{9}$$

Therefore, the diffuser air velocity in a reduced scale model ($n<1$) would be smaller than in the prototype if one conducts model tests based on the Archimedes number.

Froude Number: Fr_d It represents the ratio of inertial force to the gravitational force. Similarity between the motion within the model and prototype requires:

$$\left[\frac{U_d}{(g w_d)^{1/2}}\right]_m = \left[\frac{U_d}{(g w_d)^{1/2}}\right]_p \tag{10}$$

Since $(g)_m=(g)_p$, we have the same relation as Eq. 9.

Prandtl Number: Pr It represents the ratio of the thermal diffusivity to momentum diffusivity (i.e., viscosity). Similarity of air motions within the model and its prototype requires:

$$\left(\frac{\nu}{\alpha}\right)_m = \left(\frac{\nu}{\alpha}\right)_p \tag{11}$$

which can be satisfied by using the same working fluid in the model as in the prototype and maintaining the same testing temperatures.

In addition to the above similarity parameters, the boundary conditions need to be maintained similar between a model and its prototype. This includes the equalities of U_i^*, $u_i^* u_j^*$, Θ, $u_i^*\theta$ and P^* between the model and its prototype at the diffuser, exhaust and surfaces of walls, ceiling and floor. These dimensionless parameters are affected by the diffuser characteristics and surface roughness (Zhang et al., 1991).

Difficulties in Scaling for Non-isothermal Room Ventilation Flows

An ideal scaling method would satisfy the complete similarity conditions as represented by Eqs. (4), (6), (8), (10) and (11) and maintain boundary conditions in the model similar to its prototype. However, the restrictions on selecting a proper working fluid for the reduced-scale model room have made it difficult to satisfy the above complete similarity conditions (Moog, 1981). Model studies are therefore usually conducted with some convenient fluids (e.g., air or water), in which the distortion of some parameters is unavoidable. In this case, scaling methods are usually derived so that the model can predict the overall room air flow pattern and the distributions of air velocities and temperatures within the regions in which one is most interested (e.g., the occupied regions).

In the present study, air was used as the working fluid for both the prototype tests and the reduced scale model tests. It is generally more convenient to maintain $(T_r-T_d)_m=(T_r-T_d)_p$ so that the model and its prototype have the same air properties (Baturin 1972). i.e., $(\rho)_m=(\rho)_p$, $(\nu)_m=(\nu)_p$, $(\alpha)_m=(\alpha)_p$ and $(\beta)_m=(\beta)_p$.

Therefore, the equality of Prandtl number (Eq. 11) is satisfied automatically. However, similarity for Reynolds, Archimedes and Froude numbers results in contradictory scaling factors (Eq. 7 versus 9). That is, scaling based on the diffuser Reynolds number would result in higher diffuser air velocity in a reduced scale model than in its prototype (Eq. 7), but scaling based on the Archimedes number and Froude number would result in a lower one (Eq. 9).

The diffuser Reynolds number describes the degree of turbulence generated by the diffuser jet. When the Reynolds number is higher than a threshold, the room flow becomes fully turbulent and no longer depends on the Reynolds number. Therefore, for fully turbulent non-isothermal ventilation flows, the Archimedes number can be used as the scaling parameter as long as the Reynolds number is high enough to achieve fully developed turbulent flows in the model.

However, the entire room flow field under realistic ventilation conditions is generally not fully turbulent even though the diffuser jet region is (Zhang et al. 1990). In this case, the air movement within the room would still be diffuser Reynolds number dependant since the viscous effect can not be neglected. Therefore, a proper scaling method for non-isothermal low turbulence ventilation flow should account for both Archimedes number and Reynolds number similarities.

The Critical Archimedes Number

A horizontally projected diffuser air jet would start to change its direction to downward immediately after entering the room at a certain critical value (Ar_{fdc}) of Archimedes number. This value is different depending on direction from which the A_{fdc} is reached (i.e., whether by increasing the Ar_{fd} from a low value or by decreasing Ar_{fd} from a high value) due to the hysteresis of the air motion (Nielsen 1979 and Zhang 1991). In this paper, Ar_{fdc} is defined as the value of Archimedes number at which the horizontally projected jet falls immediately after entering the room when one gradually increases the Archimedes number (either by decreasing the diffuser air velocity or by increasing the internal heat load). It was found (Zhang 1991) that the critical Archimedes number decreased with the room size (Table 1).

A New Scaling Method

It is generally recognized that the Archimedes number is a more important parameter for similitude model study of non-isothermal ventilation flows, since it determines the trajectory of the diffuser jet, which is a predominant factor in determining the overall air flow pattern within the room (e.g., Baturin 1972 and Christianson et al. 1988). However, the Archimedes number in a reduced scale model test can not be the same as that in the prototype because the critical Archimedes number decreases when the room size decreases as discussed in the last section.

It would be reasonable to assume that if the relative deviation from the critical Archimedes number

TABLE 1
Dimensions of the Test Rooms and the Critical Archimedes Number

Rooms	H (m)	W (m)	w_d (cm)	y_d (cm)	w_e (cm)	y_e (cm)	Ar_{fdc}
Full scale prototype	2.4	18.3	5.08	30.48	20.32	91.44	0.0230
1/4th scale model	0.6	1.35	1.27	7.62	5.08	22.86	0.0127

in a reduced scale model test is the same as that in the prototype, a similar air flow pattern can be produced. In expression,

$$\left(\frac{Ar_{fdc}-Ar_{fd}}{Ar_{fdc}}\right)_m = \left(\frac{Ar_{fdc}-Ar_{fd}}{Ar_{fdc}}\right)_p \tag{12}$$

With $(T_r-T_d)_m=(T_r-T_d)_p$, $(\beta)_m=(\beta)_p$, $(g)_m=(g)_p$ and $(w_d)_m=n(w_d)_p$, we can easily derive:

$$(U_d)_m = \left[\frac{(Ar_{fdc})_p}{(Ar_{fdc})_m}\right]^{1/2} n^{1/2} (U_d)_p \tag{13}$$

It is interesting to note that the value of the scaling factor, $[(Ar_{fdc})_p/(Ar_{fdc})_m]^{1/2}n^{1/2}$, in Eq. (13) is between $n^{1/2}$ and 1/n, which are the scaling factors derived from Archimedes number similarity (Eq. 9) and Reynolds number similarity (Eq. 7), respectively. Therefore, the new scaling equation (Eq. 13) appears to be a compromise between the Archimedes number similarity and the Reynolds number similarity.

EXPERIMENTAL EVALUATION

Experimental Facilities and Procedures

A Room Ventilation Simulator (Christianson et al., 1992)has been developed to study room air and air contaminant distribution under well controlled environmental testing conditions. For the present study, three tests were conducted in an 1/4th scale test room (Figure 2) and its full scale prototype. Test conditions are listed in Table 2, in which the diffuser air velocities in the 1/4th scale room were determined based on Eq. (13). The test rooms had a continuous slot diffuser opening and exhaust, resulting in two dimensional room ventilation flows (Zhang 1991).

The velocities and temperatures within the room were measured with a hot wire probe and a thermocouple probe, respectively. A microcomputer based data acquisition and probe positioning system was developed to collect the data and move the probes automatically (Zhang et al. 1991). Additionally, the temperatures at the diffuser, exhaust of the test rooms and the floor surface were monitored by a separate data logger with thermocouple probes. Room air flow patterns were visualized with titanium tetrachloride ($TiCl_4$) smoke which is neutrally thermal buoyant.

Test Results

<u>Flow Patterns</u> The flow patterns observed in the 1/4th scale model tests were in general similar to those observed in the prototype tests (Figures 3, 4 and 5, respectively). However, the secondary eddy at the upper right corner in the prototype test P4 was not clearly revealed by smoke in the 1/4th scale test M4.

<u>Mean Velocity.</u> The dimensionless mean velocities measured in the 1/4th scale model tests in

critical Archimedes number to reduce such variation.

To extend the application of the proposed scaling method, further researches is needed to determine the effects of room aspect ratio, diffuser location and internal obstruction on the critical Archimedes number, and to develop a functional relationship between the critical Archimedes number and the room scale.

SUMMARY AND CONCLUSIONS

For non-isothermal ventilation flow, the critical Archimedes number at which the incoming air dropped immediately after entering the room was found to decrease with the room size. A new scaling method was proposed based on the equality of the relative deviation of Archimedes number from their critical values between a reduced scale model and its prototype. Comparison between the 1/4th scale model tests based on the new scaling method and the prototype tests indicated that the new scaling method is promising.

The proposed scaling method may be improved by exploring a different temperature differential as a reference for the critical Archimedes number to reduce the dependence of the predicting performance on the internal heat load. More experiments are also needed to determine the effects of room aspect ratio, diffuser location and internal obstruction on the critical Archimedes number, and to develop a functional relationship between the critical Archimedes number and the room scale.

NOMENCLATURE

Variables

Ar_{fd}	= Archimedes number defined as $\dfrac{\beta g w_d (T_f - T_d)}{U_d^2}$;
Fr_d	= Froude number defined as $U_d/(g w_d)^{1/2}$, which represents the ratio of gravitational effect to inertial effect;
g	= Gravitational acceleration rate, m/s^2;
H	= Room height, m;
k	= Turbulent kinetic energy, $(m/s)^2$;
L	= Length of the room (in Z direction), m;
l_d	= Length of the diffuser slot (in Z direction), m;
P	= Thermodynamic pressure, Pa;
P^*	= Dimensionless pressure (ratio between the pressure at a point and P_d);
P_d	= Diffuser pressure, Pa;
Pn	= Pressure number defined as $\dfrac{P_d}{\rho U_d^2}$;
Pr	= Prandtl number defined as ν/α;
P_{ref}	= A reference pressure (e.g., the pressure outside the test room, Pa;
Re	= Reynolds number defined as $\dfrac{U_d w_d}{\nu}$;
T, t	= Mean temperature and fluctuation component, °C;

T_f	=	Maximum temperature in room (e.g., on the heated surface), °C;
T_d	=	Diffuser air temperature, °C;
T_e	=	Air temperature at the exhaust, °C;
ΔT	=	T_f-T, °C;
ΔT_{ed}	=	T_e-T_d, °C;
ΔT_{fd}	=	T_f-T_d, °C;
ΔT_{rd}	=	Temperature difference between the diffuser air and the room air, °C;
U, u	=	Mean air velocity and fluctuation component, m/s;
u'	=	Standard deviation of velocity, m/s;
U^*, u^*	=	Dimensionless mean air velocity and fluctuation component based on U_d;
U_d	=	Reference velocity, diffuser velocity at the measurement plane (z=0), m/s;
W	=	Width of the test room (in X direction), m;
w_d	=	Width of the diffuser slot (in Y direction), m;
w_e	=	Slot width of the exhaust, m;
x, y, z	=	Eulerian Cartesian coordinates with the origin at the upper left corner of the 2-D room flow (Figure 5), m;
x^*,y^*,z^*	=	Dimensionless Eulerian Cartesian coordinates based on w_d;
y_d	=	Distance from the ceiling to the diffuser upper edge, m;
y_e	=	Distance from the ceiling to the upper edge of the exhaust, m;
α	=	Thermal diffusion coefficient, m^2/s;
β	=	Thermal expansion coefficient, 1/K;
δ_{i2}	=	Kronecker delta (=1 only when i=2 and =0 otherwise);
Θ, θ	=	Dimensionless mean temperature difference and fluctuation component;
ν	=	Kinematic viscosity, m^2/s;
ρ	=	Air density, kg/m^3;

Subscripts

i,j	=	Indices representing direction of coordinates (i,j=1,2,3, referring to longitudinal, vertical and lateral coordinates);
m	=	denote model;
p	=	denote prototype.

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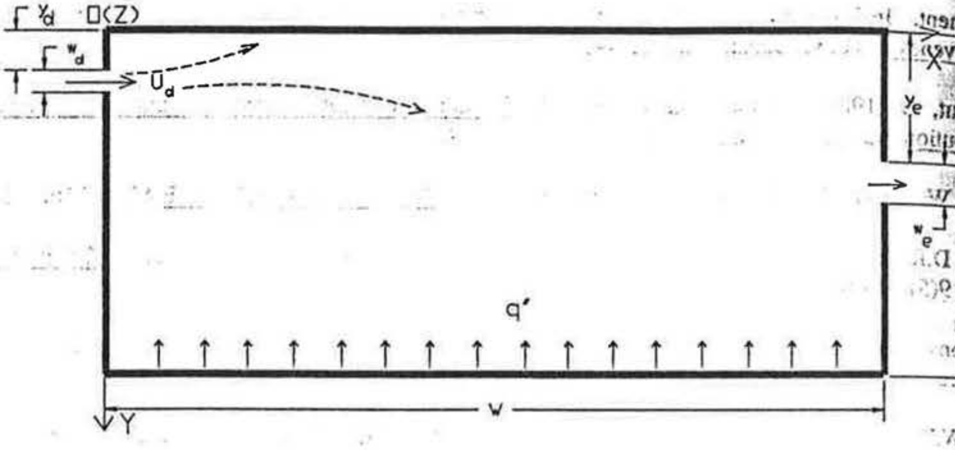


Figure 1 Schematic of the room geometry studied and the definition of coordinates (Z is into the page)

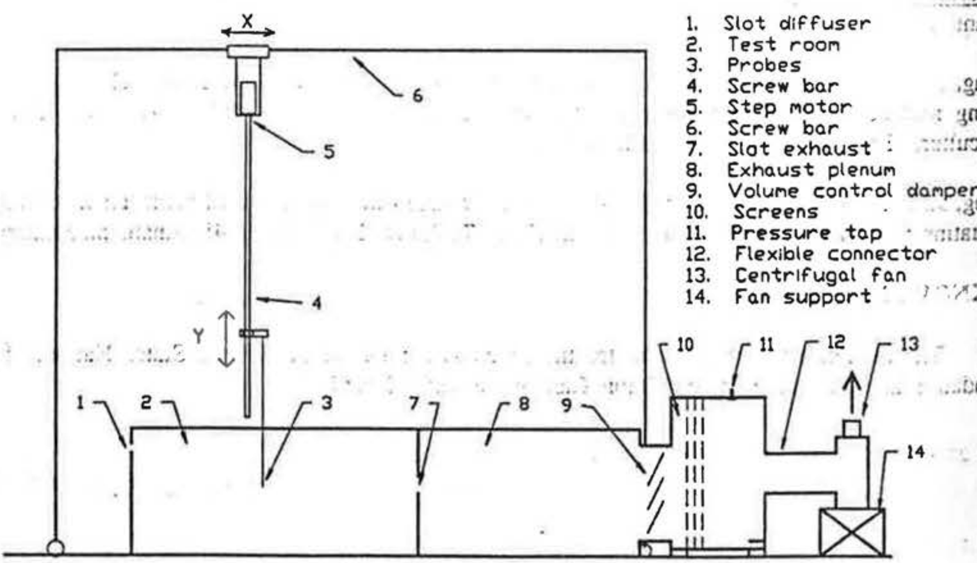
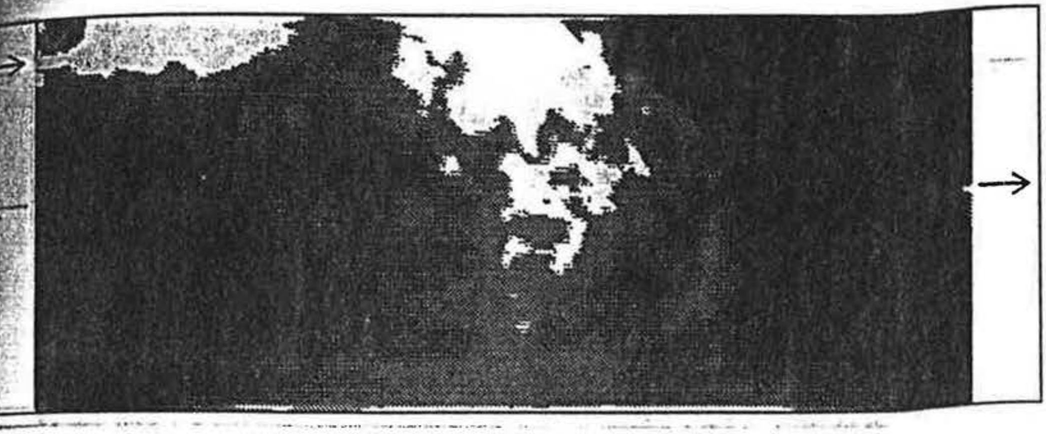
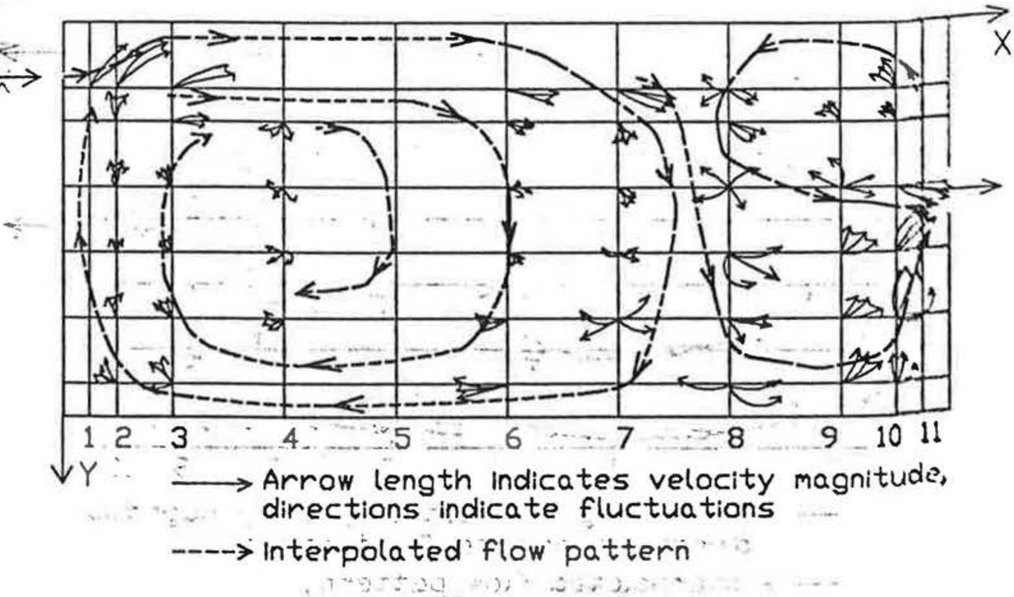


Figure 2 Experimental set up for the 1/4th scale tests

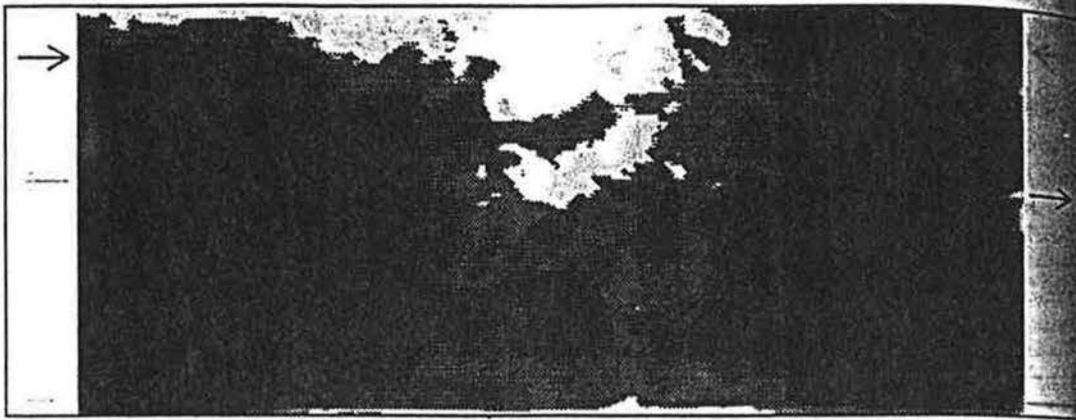


a. Test M4: U_d=1.19 m/s, ΔT_{td}=37.5 °C

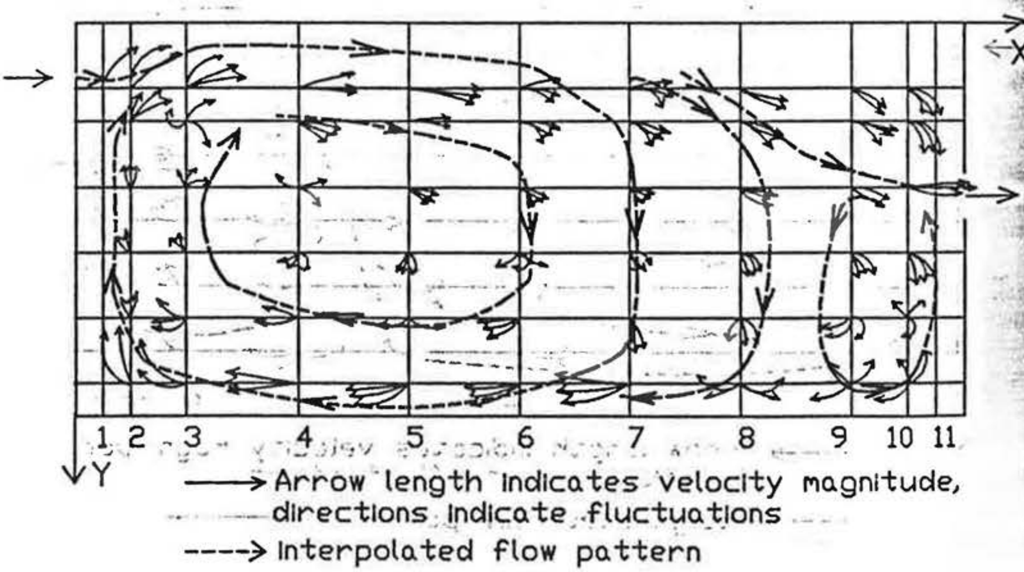


b. Test P4: U_d=1.78 m/s, ΔT_{td}=37.4 °C

Figure 3 Comparison of flow patterns between tests M4 and P4

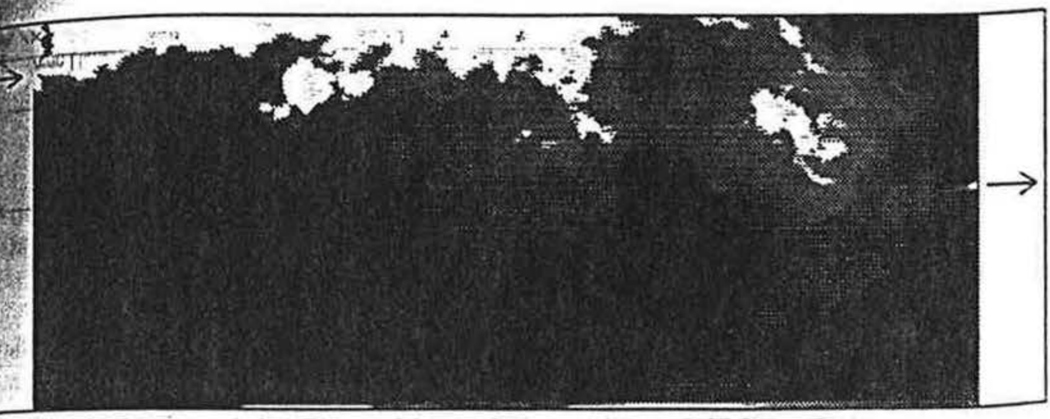


a. Test M5: U_d=1.19 m/s, ΔT_{td}=26.8 °C

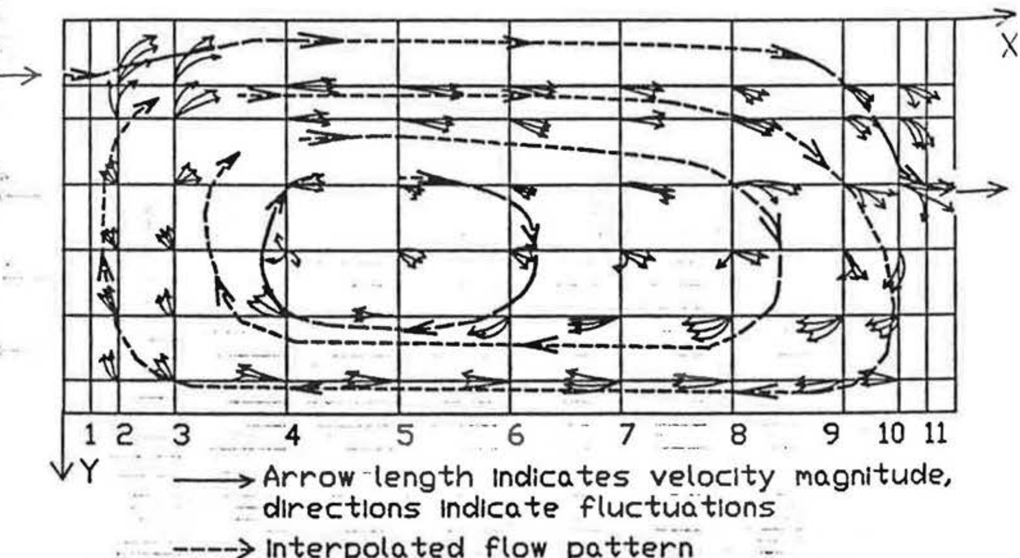


b. Test P5: U_d=1.78 m/s, ΔT_{td}=26.6 C

Figure 4 Comparison of flow patterns between tests M5 and P5



a. Test M6: U_d=1.19 m/s, ΔT_{td}=16.8 °C



b. Test P6: U_d=1.78 m/s, ΔT_{td}=16.4 °C

Figure 5 Comparison of flow patterns between tests M6 and P6

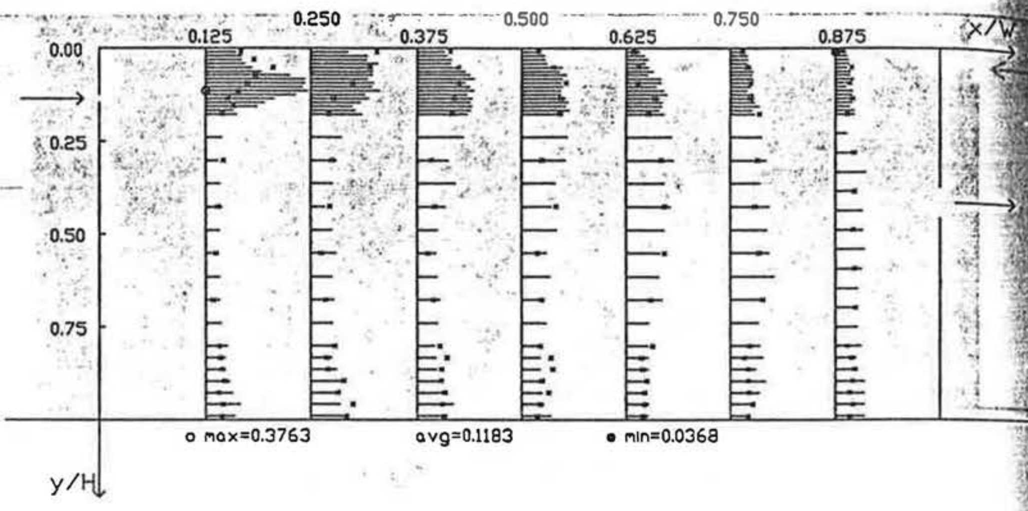


Figure 6 Spatial distribution of mean velocities
(—: 1/4th scale model test M4, ×: prototype test P4)

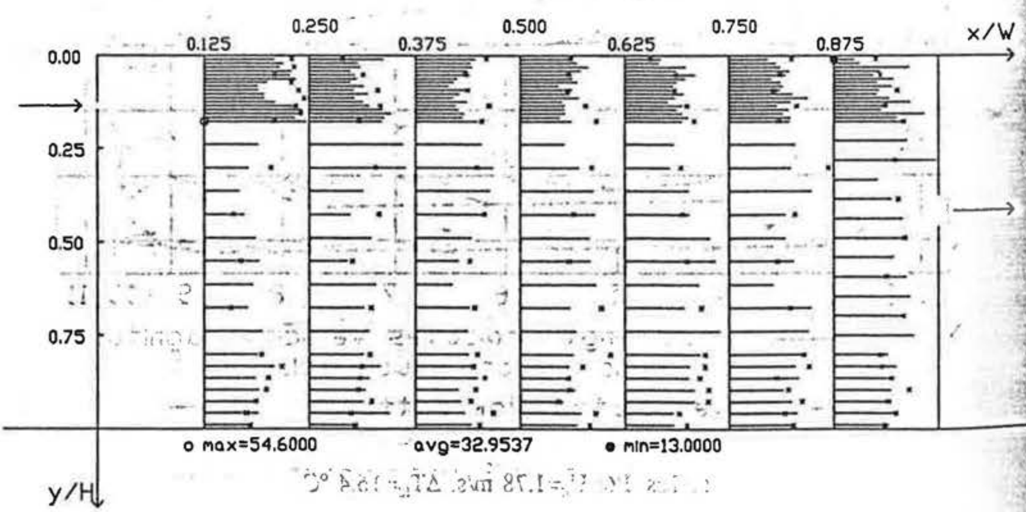


Figure 7 Spatial distribution of turbulence intensities (100u′/U)
(—: 1/4th scale model test M4, ×: prototype test P4)

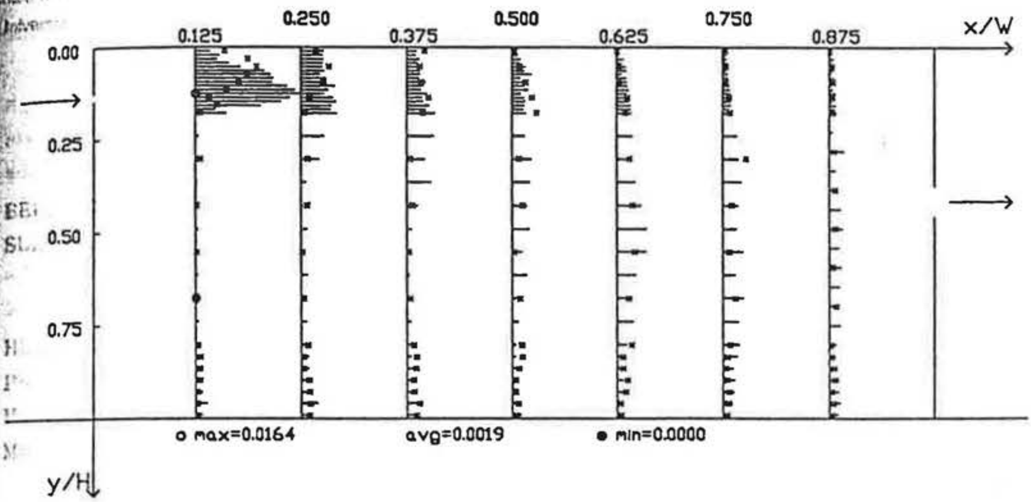


Figure 8 Spatial distribution of turbulent kinetic energy ($k/0.5Ud^2$)
(—: 1/4th scale model test M4, ×: prototype test P4)

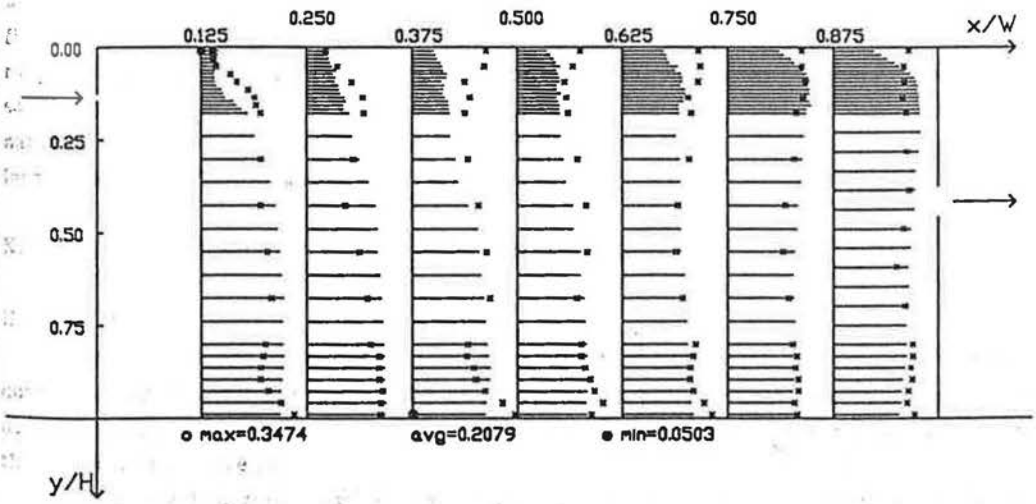


Figure 9 Spatial distribution of mean temperatures [$(T-T_d)/\Delta T_{fd}$]
(—: 1/4th scale model test M4, ×: prototype test P4)