#6194

PREDICTING ROOM VENTILATION FLOW WITH REDUCED SCALE MODELS

J.S. Zhang
Building Performance laboratory, IRC
National Research Council of Canada
Ottawa, Ontario, Canada

L.L. Christianson and B. Zhang
Bioenvironmental Engineering Research Laboratory
University of Illinois at Champaign-Urbana
IL, U.S.A.

SUMMARY

Scaling methods for predicting various room ventilation flows including isothermal, non-isothermal, fully developed and non-fully developed turbulence flows are reviewed and discussed based on the governing flow equations. When air is used as the working fluid in a reduced scale model for predicting non-isothermal and low turbulence ventilation flow, scaling based on the relative deviation of Archimedes number from its critical value was found to be promising.

In 1/4 scale model test room, neutrally thermal buoyant helium bubbles were used to visualize the airflow patterns and determine the critical Archimedes number at which the room airflow changed its recirculation direction. Dependence of the critical Archimedes number on the location of diffuser opening, diffuser opening size, the location of internal obstructions and the internal heat load was analyzed based on the experimental data.

PREDICTING ROOM VENTILATION FLOW WITH REDUCED SCALE MODELS

J.S. Zhang
Building Performance Laboratory, IRC
National Research Council of Canada
Ottawa, Ontario, Canada

L.L. Christianson and B. Zhang
Bioenvironmental Engineering Research Laboratory
University of Illinois at Champaign-Urbana
IL, U.S.A.

INTRODUCTION

Reduced scale models are useful tools for studying room air movement and ventilation effectiveness. With reduced scale models, a broad range of ventilation conditions can be investigated conveniently and less expensively. The experimental results from reduced scale models can also be applied to rooms of different sizes with proper scaling methods.

Reduced scale models have been successfully applied to predict isothermal and fully developed turbulence flows [1][2]. However, difficulties were encountered when the reduced scale models were applied to non-isothermal and non-fully developed turbulent room air flows[3][4][5][6] and a proper scaling method for this type of ventilation flow is still to be developed.

In most cases, air movement in realistically ventilated rooms involves non-isothermal and non-fully developed turbulence flows. Developing scaling methods for realistic room ventilation flows with internal heat sources and obstructions is one of the current research needs in the study of room air and air contaminant distribution [7]. The objective of the present study is to develop an approximate and practical scaling method for predicting airflow pattern and distribution of air velocities and temperatures in rooms under non-isothermal and non-fully turbulent ventilation conditions.

SCALING METHODS

Governing Equations

The distribution of mean air velocities ($U_i^*=U_i/U_d$), temperature ($\theta=T/\Delta T_{td}$), pressure ($P^*=P/(P_{ref}-P_d)$) and turbulence stress in a ventilated room are governed by the following equations [6] with Cartesian tensor notation (assuming steady mean flow, incompressible, constant viscosity, and no internal volumetric mass or volumetric heat production)[1]:

Continuity equation:

$$\frac{\partial U^*_i}{\partial x^*_i} = 0 \tag{1}$$

Momentum equation:

$$U^*_j \frac{\partial U^*_i}{\partial x^*_j} = -Pn\frac{\partial P^*}{\partial x^*_i} + \frac{1}{Re_d}\frac{\partial^2 U^*_i}{\partial x^*_j \partial x^*_j} + \frac{\partial}{\partial x^*_j}(-\overline{u^*_i u^*_j}) - Ar_d \Theta \delta_{i2} + \frac{1}{Fr_d^2}\delta_{i2} \tag{2}$$

Energy equation:

$$U^*_i \frac{\partial \Theta}{\partial x^*_i} = \frac{1}{Pr}\frac{1}{Re_d}\frac{\partial^2 \Theta}{\partial x^*_i \partial x^*_i} + \frac{\partial}{\partial x^*_i}(-\overline{u^*_i \theta}) \tag{3}$$

The continuity equation (Eq. 1) represents a mass balance within an air control volume. The momentum equation (Eq. 2) states that the momentum transferred into an air control volume through convection is balanced by the pressure gradient, molecular viscous stress, turbulence stress, thermal buoyancy and the gravity. The thermal buoyancy and gravity terms exist only in the vertical direction (i.e., when i=2). The energy equation (Eq. 3) states that energy transported into the air control volume is balanced by the molecular diffusion and turbulence diffusion. The energy loss due to dissipation is neglected in the equation since it is insignificant in room ventilation flows.

Similarity Parameters

The air flow fields in two geometrically similar rooms (i.e., a reduced scale model and its full scale prototype) are completely similar to each other if they have the same distributions of dimensionless velocity (U^*), temperature (Θ), pressure (P^*) and turbulence stress. This would require that the two flow fields have the same boundary conditions and the same pressure number (Pn), Reynolds number (Re_d), Archimedes number (Ar_d), Froude number (Fr_d) and the Prandtl number (Pr) in the governing equations (Eqs. 2 and 3). These numbers are called similarity parameters of the flow field and are discussed as follows.

Pressure Number: (Pn) It represents the ratio of pressure to the double of the dynamic head. Similarity between the motion in a model and its prototype requires:

---

[1] All symbols are defined in the NOMENCLATURE section.

$$(\frac{P_{ref}-P_d}{\rho U_d^2})_m = (\frac{P_{ref}-P_d}{\rho U_d^2})_p \quad (4)$$

where, subscripts m and p denote model and prototype (i.e., in the full scale room), respectively.

If P_{ref} is defined as the pressure outside the test room, we then have, based on the Bernoulli equation:

$$P_{ref}-P_d = \frac{1}{2}\rho U_d^2 \quad (5)$$

In this case, Pn=1/2 which does not depend on the room scale. In other words, Eq. (4) is 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:

$$(\frac{U_d w_d}{\nu})_m = (\frac{U_d w_d}{\nu})_p \quad (6)$$

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

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

where, n is the geometric scale of the model relative to the full scale 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:

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

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

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

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:

$$[\frac{U_d}{(gw_d)^{1/2}}]_m = [\frac{U_d}{(gw_d)^{1/2}}]_p \tag{10}$$

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:

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

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[8].

Isothermal and Non-Fully Developed Turbulence Ventilation Flows

In isothermal flow cases, the term with the Archimedes number vanishes in equation (2). Assuming there is a balance between the weight of the air and its hydrostatic buoyancy, the term with the Froude number can be combined with the pressure gradient term by taking the pressure as the difference between the thermodynamic pressure when air is in motion and that its hydrostatic pressure. This assumption is valid for incompressible flows with no free surface in the flow field, since a change in pressure does not result in a change in volume[9]. Therefore, Reynolds number becomes the only important similarity parameter for scaling model experiments (Eq. 7). The following equation can then be derived from eqution 7 used to extrapole model results to its prototype:

$$(U)_p = n(U)_m \tag{12}$$

where, $0<n<1$.

Isothermal and Fully Developed Turbulence Ventilation Flows

The Reynolds number becomes large when the room airflow is fully turbulent. Which makes the molecular diffusion in the governing equation (2) negligible as compared to the turbulent diffusion term. Therefore, the Reynolds number used in the reduced scale tests does not have to be the same as that in the full scale room as long as it is higher than a threshold value above which the flow in the reduced scale model is assured to be turbulent. In this case, model tests can be done at any convenient diffuser air velocity with a Reynolds number higher than the threshold value. The following equation can then be used to extrapolate the results from the model to its prototype:[2]

$$(U)_p = \frac{(U_d)_p}{(U_d)_m}(U)_m \tag{13}$$

where, $(U_d)_p/(U_d)_m$ dose not have to be equal to n.

Timmons[2] compared measured air velocity distribution from a reduced scale model to that from a full scale room, indicating excellent agreement between the two when equation (13) was used for the extrapolation of the model results. He also noted that the threshold Reynolds number increased with the room dimensions.

Non-Isothermal and Fully Developed Turbulence Ventilation Flows

When internal heat loads are present as in most realistic rooms, the thermal buoyancy force due to the heat loads can have a significant effect on the room airflow pattern and distributions of air velocities and temperatures. The Archimedes number becomes important since it represents the ratio between thermal buoyancy and inertial force within the flow field. The molecular diffusion terms (with Reynolds number) in the governing equations (2) and (3) can again be neglected because the flow is fully turbulent. Therefore, the Archimedes number becomes the only important term for scaling model experiments (Eq. 9), as long as the Reynolds number is higher than a threshold value which insures a fully developed turbulence flow in the model. Results from the model tests can then be extrapolated to the full scale room with the following equation:

$$(U)_p = n^{-1/2}(U)_m \tag{14}$$

Batturin[5] used this approach and obtained reasonable agreement between the model tests and the full scale measurements.

Non-Isothermal and Non-Fully Developed Turbulence Ventilation Flows

When the entire flow field is neither isothermal nor fully turbulent as in most rooms under realistic ventilation conditions, both the molecular diffusion and thermal buoyancy terms are important. Therefore, a proper scaling method needs to consider both Reynolds number similarity (Eq. 6) and Archimedes number similarity (Eq. 8) in addition to the Prandtl number similarity (Eqn. 11) and similar boundary conditions. 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[3]. 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 Reynolds number describes the degree of turbulence of the flow field, which can be considered as the micro-feature of the flow for a given design and arrangement of deffuser. The Archimedes number directly affects the trajectory of the diffuser air jet and thus the overall airflow pattern within the room, which is the macro-feature of the flow. Therefore, it is generally agreed that the Archimedes number is a more important parameter than the Reynolds number in terms of predicting room airflow patterns and mean quantities (velocity and temperature) in the occupied regions[4][5].

A critical Archimedes number can be defined as the Archimedes number at which the diffuser air jet drops immediately after entering the room if one gradually increases the Archimedes number (either by decreasing the diffuser air velocity or by increasing the internal heat load). This critical Archimedes number was found to decrease with the room size[6].

Zhang[6] proposed a scaling method based on the equality of the relative deviation of Archimedes number from its critical value between the reduced scale model and its prototype. In expression,

$$\left(\frac{Ar_{fdc}-Ar_{fd}}{Ar_{fdc}}\right)_m = \left(\frac{Ar_{fdc}-Ar_{fd}}{Ar_{fdc}}\right)_p \qquad (15)$$

Therefore,

$$(Ar_{fd})_m = \frac{(Ar_{fdc})_m}{(Ar_{fdc})_p}(Ar_{fd})_p \qquad (16)$$

i.e.,

$$\left(\frac{\beta g w_d (T_r - T_d)}{U_d^2}\right)_m = \frac{(Ar_{fdc})_m}{(Ar_{fdc})_p}\left(\frac{\beta g w_d (T_r - T_d)}{U_d^2}\right)_p \tag{17}$$

Let $(T_r\text{-}T_d)_m=(T_r\text{-}T_d)_p$, $(\beta)_m=(\beta)_p$, $(g)_m=(g)_p$ and $(w_d)_m=n(w_d)_p$, we have

$$(U_d)_m = \left[\frac{(Ar_{fdc})_p}{(Ar_{fdc})_m}\right]^{1/2} n^{1/2} (U_d)_p \tag{18}$$

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. (18) is between $n^{1/2}$ and $1/n$, which are the scaling factors derived from Archimedes number similarity (Eq. 8) and Reynolds number similarity (Eq. 6), respectively. Therefore, the new scaling equation (Eq. 18) appears to be a compromise between the Archimedes number similarity and the Reynolds number similarity.

Eq. (18) is a equation for determining the diffuser velocity to be used in a reduced scale model test. For a given room air velocity, $(U)_m$ measured at a given location or region in a reduced scale model, the velocity at the corresponding location or region in the full scale prototype can then be predicted by

$$U_p = \frac{1}{n^{1/2}} \frac{(Ar_{fdc})_m}{(Ar_{fdc})_p} U_m \tag{19}$$

Comparisons of airflow patterns between the full scale tests and 1/4th scale tests scaled with equation (18) and extrapolated with equation (19) are shown in Figures 1, 2 and 3, indicating a good agreement. The accuracy in predicting mean air velocity, temperature, and turbulent kinetic energy and turbulence intensity in the occupied region were, on average, 11%, 15%, 14% and 26%. Tests also indicated that predicting performance varied with the internal heat load and the need for further investigation regarding how diffuser location, opening size and velocity, and internal heat loads and obstructions affect the value of the critical Archimedes number for a given room before the above method can be generally used as a proper scaling method for predicting non-isothermal and non-fully developed turbulence ventilation flows. Experiments have been conducted in an 1/4th scale test room to study these effects and results are presented in the following sections.

REDUCED SCALE MODEL TESTS

Experimental Facilities and Procedures

The experimental setup is shown in Figure 4. The test rooms had a continuous slot diffuser opening and exhaust, resulting in two dimensional room ventilation flows[6]. One of the end walls was made with plexiglass for flow visualization. Three 0.61x1.22 m (2x4 ft) electrical heating panels were used to simulate uniform floor heating.

Airflow rate was measured with a flow rate measurement chamber precalibrated with a fan test chamber designed according to ASHRAE standard[10]. The diffuser air velocity was calculated based on the measured flow rate and the diffuser opening area and verified with a hot wire anemometer. The floor surface temperature, air temperature at the diffuser and at the exhaust were measured with copper-constantan T type thermocouples.

Neutrally thermal buoyant helium bubbles were injected at the diffuser to trace the air movement within the test room. A 24x27 mesh (figure 5) formed with fish wire was placed at the central cross section of the test room and illuminated with a light sheet. Airflow patterns at the central cross section were video taped and photographed together with the mesh to measure the horizontal length of the diffuser air jet.

For each specified diffuser location (Y_d), diffuser opening size (W_d), internal heat load (q^*) and location of internal obstruction $(L)_o$, the diffuser air velocity (U_d) was adjusted first from high to low and then from low to high with an interval of 0.025m/s (5 ft/min). The flow was allow to reach the steady state for each diffuser velocity setting. The horizontal length of the jet (L_x) was then measured. The condition at which the horizontal length of the jet reached its minimum value (i.e., the jet dropped immediately after entering the room) was used to calculate the critical Archimedes number. Figure 6 shows an example of the measuring results.

Results and Discussion

Selection of the Reference Temperature Differential The internal heat load had more effect on the critical Archimedes number defined with the temperature differential (T_{fd}) between the floor surface and diffuser air than that defined with the temperature differential (T_{ed}) between the exhaust air and diffuser air (Figure 7). Such a phenomenon was consistent for all the tests conducted. This indicates that using T_{ed} as reference temperature differential can reduce the dependence of the critical Archimedes number on the level of internal heat load, which is desired for forming a universal scaling parameter. Therefore, the critical Archimedes number defined with T_{ed} is used in the following discussion.

Effect of the Internal Heat Load As shown in Figure 7, the critical Archimedes number decreased slightly when the internal heat load increased. Such a phenomenon was consistent for all the tests conducted. The slight decrease of the critical Archimedes number may be due to the effect of turbulence. A higher internal heat load would result in higher turbulence production in the room due to the thermal buoyancy[11]. The turbulence acted as a disturbance to the diffuser air jet and made the jet drop at a smaller Archimedes number. However, such effect was very small because the turbulence generated due to the internal heat production was much less than that produced due to the mixing of the diffuser air jet with the room air[11].

Effect of the Diffuser Location The effect of the diffuser location on the critical

Archimedes number was studied by varying the distance (y_d) between the slot opening and the ceiling surface of the test room. As shown in Figure 8, the critical Archimedes number decreased when y_d increased. The horizontal length of the diffuser air jet is determined by the balance between the Coanda effect (resulted from the inertial force and the restriction of the room geometry) and thermal buoyancy effect. As y_d increased, the restriction to the jet entrainment of air above the jet decreased, which decreased the Coanda effect. Therefore, only a smaller thermal buoyancy was needed to make the diffuser air jet drop, resulting in a smaller critical Archimedes number.

Effect of the Diffuser Opening Size The effect of the diffuser opening size on the critical Archimedes number was studied by varying the diffuser opening width (w_d) when y_d = 7.62cm (3"). As shown in Figure 9, the critical Archimedes number increased when w_d increased, indicating that thinner jet is more likely to be affected by thermal buoyancy as expected.

Effect of the Internal Obstruction The effect of the internal obstruction on the critical Archimedes number was examined by positioning a 0.254x0.254x1.83m (0.833x0.833x6 ft) block at three different locations. As shown in Figure 10, the critical Archimedes number decreased slightly when L_o increased from 1/3 L to 1/2 L, but remained almost unchanged when L_o increased further. This indicates that the obstruction may have a significant effect on the value of the critical Archimedes number only when it is within certain distance from the diffuser air jet. Otherwise, the effect of obstruction on the diffuser air jet may be neglected.

SUMMARY AND CONCLUSIONS

Scaling methods for predicting various room ventilation flows including isothermal, non-isothermal, fully developed and non-fully developed turbulence flows are reviewed and discussed based on the governing flow equations. When air is used as the working fluid in a reduced scale model for predicting non-isothermal and low turbulence ventilation flows, scaling based on the relative deviation of Archimedes number from its critical value was found to be promising.

In 1/4th scale model test room, neutrally thermal buoyant helium bubbles were used to visualize the airflow patterns and study the effects of diffuser location, opening size, internal heat load and location of obstruction on the critical Archimedes number. The following may be concluded:

1. The critical Archimedes number (Ar_{edc}) defined with the temperature differential between the exhaust and the diffuser air had less variation with the internal heat load as compared to that (Ar_{fdc}) defined with the temperature differential between the floor surface and the exhaust. Ar_{edc} slightly decreased with the increase of the internal heat load due to the additional turbulence generated by the increased thermal buoyancy.

2. When the distance between the diffuser and the ceiling increased, the critical Archimedes number decreased due to the decrease of Coanda effect.

3. The critical Archimedes number decreased when the diffuser opening size increased.

4. The internal obstruction did not have significant effect on the diffuser air jet and hence the critical Archimedes number for the tests conducted.

NOMENCLATURE

Variables

Ar_{fd}	= Archimedes number defined as $\frac{\beta g w_d (T_f - T_d)}{U_d^2}$;
Ar_{ed}	= Archimedes number using ΔT_{ed} as reterence temperature differential;
Ar_{edc}	= Critical Archimedes number using ΔT_{ed} as reterence temperature;
Ar_{fdc}	= Critical Archimedes number using ΔT_{fd} as reterence temperature differential;
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, ft/min^2 (m/s^2);
H	= Room height, ft (m);
k	= Turbulent kinetic energy, $(ft/min)^2$ or $(m/s)^2$;
L	= Length of the room (in Z direction), ft (m);
n	= Geometrie scale of model relative to its full scale prototype ($o<n<1$)
l_d	= Length of the diffuser slot (in Z direction), ft (m);
P	= Thermodynamic pressure, psi (Pa);
P^*	= Dimensionless pressure (ratio between the pressure at a point and P_d);
P_d	= Diffuser pressure, psi (Pa);
Pn	= Pressure number defined as $\frac{P_d}{\rho U_d^2}$;
Pr	= Prandtl number defined as ν/α;
P_{ref}	= A reference pressure (e.g., the pressure outside the test room, psi (Pa).
Re	= Reynolds number defined as $\frac{U_d w_d}{\nu}$;
T, t	= Mean temperature and fluctuation component, F (°C);
T_f	= Maximum temperature in room (e.g., on the heated surface), F (°C);
T_d	= Diffuser air temperature F (°C);
T_e	= Air temperature at the exhaust, F (°C);
ΔT	= T_f-T, F (°C);
ΔT_{ed}	= T_e-T_d, F (°C);
ΔT_{fd}	= T_f-T_d, F (°C);
ΔT_{rd}	= Temperature difference between the diffuser air and

		the room air, F (°C);
U, u	=	Mean air velocity and fluctuation component, ft/min (m/s);
u′	=	Standard deviation of velocity, ft/min (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), ft/min (m/s);
W	=	Width of the test room (in X direction), ft (m);
w_d	=	Width of the diffuser slot (in Y direction), ft (m);
w_e	=	Slot width of the exhaust, ft (m);
x, y, z	=	Eulerian Cartesian coordinates with the origin at the upper left corner of the 2-D room flow (Figure 5), ft (m);
x^*,y^*,z^*	=	Dimensionless Enlerian Cartesian coordinates based on W_d;
y_d	=	Distance from the ceiling to the diffuser upper edge, ft (m);
y_e	=	Distance from the ceiling to the upper edge of the exhaust, ft (m);
α	=	Thermal diffusion coefficient, ft²/min (m²/s);
β	=	Thermal expansion coefficient, 1/R (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, ft²/min (m²/s);
ρ	=	Air density, lb_m/ft^3 (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.

REFERENCES

[1] Pattie, D.R. and W.R. Milne. 1966. "Ventilation airflow patterns by use of models". Transactions of ASAE 9(5):646-649.

[2] Timmons, M.B. 1984. "Use of physical models to predict the fluid motion in slot-ventilated livestock structures". Transactions of ASAE 27(2):502-507.

[3] Moog, W. 1981. "Room flow tests in a reduced-scale". Transactions of ASHRAE 87(4):1162-1181.

[4] Christianson, L.L., G.L. Riskowski, J.S. Zhang and R. Koca. 1988. "Predicting air velocity at the pig level". In Proceedings of the third International Livestock Environment Symposium. Toronto, Canada.

[5] Baturin, V.V. 1972. Fundamentals of Industrial Ventilation. 3rd edition. Pergamon Press, Oxford, U.K.

[6] Zhang, J.S. 1991. An Fundamental Study of Two Dimensional Room Ventilation Flows under isothermal and Non-isothermal Conditions. PhD thesis, University of Illinois at Urbana-Champaign, Urbana, IL., U.S.A.

[7] Int-Hout, D. 1989. "Physical modelling". In Building Systems: Room Air and Air Contaminant Distribution, ed. L.L. Christianson, 22-23. ASHRAE, Atlanta, GA.

[8] Zhang, J.S., L.L. Christianson and G.L. Riskowski. 1991. Effect of diffuser air velocity profiles on scaling methods for predicting room air motion. ASAE Paper No. 91-4558. American Society of Agricultural Engineers. St. Joseph, MI, U.S.A.

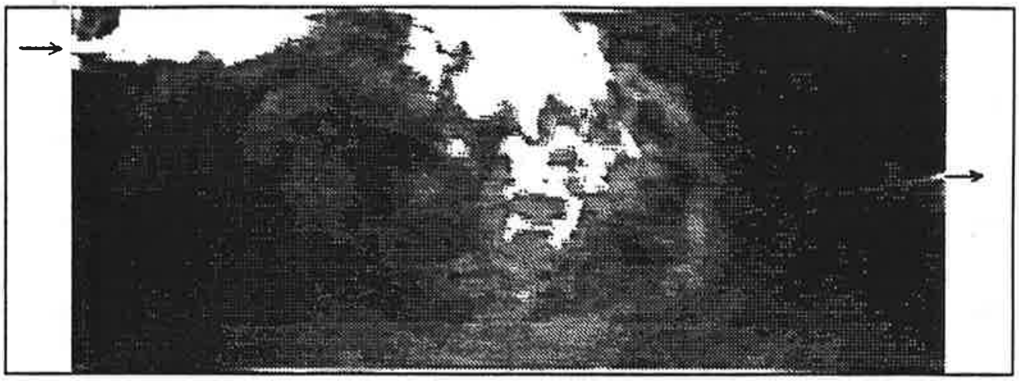
[9] Schlichting, H 1979. Boundary-Layer Theory. Mcloraw-Hill, Inc. New York.

[10] ASHRAE. 1985. Laboratory Methods of Testing Fans for Rating. Standard 51-1985. American Society of Heating, Refrigeration and Air-Conditioning Engineers, Atlanta, GA, U.S.A.

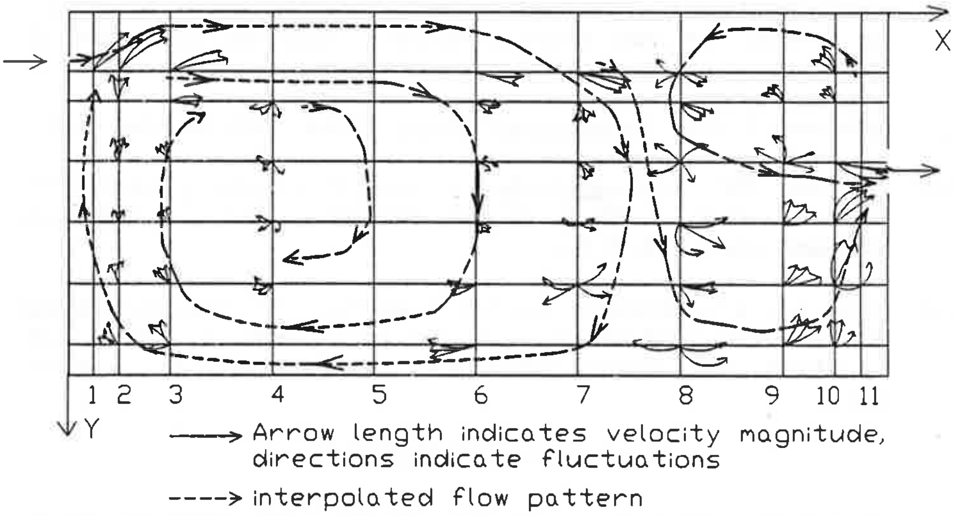
[11] Zhang, J.S., L.L. Christianson and G.L. Riskowski. 1991. Mean and turbulent flow characteristics of isothermal and non-isothermal ventilation flows. ASAE Paper No. 91-4535. American Society of Agricultural Engineers. St. Joseph, MI, U.S.A.

ACKNOWLEDGEMENT

This study was sponsored by the United States National Science Foundation and the University of Illinois Campus Research Board. The authors are very grateful for their financial support.

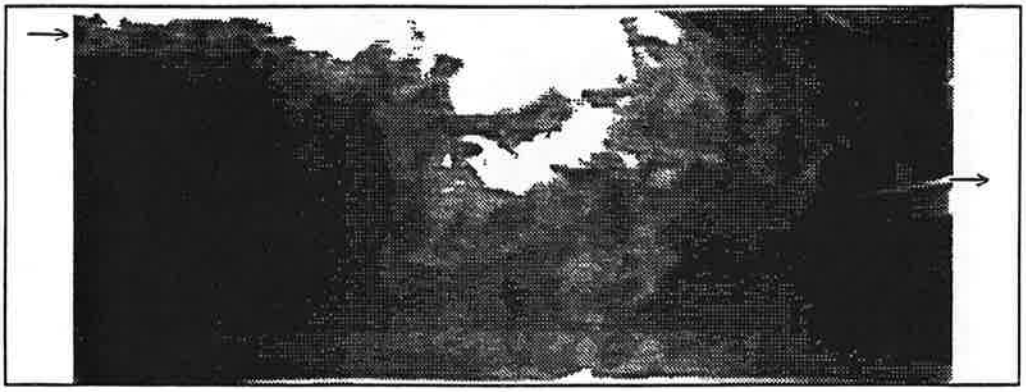


a. Model test: U_d=1.19 m/s, ΔT_{fd}=37.5 °C

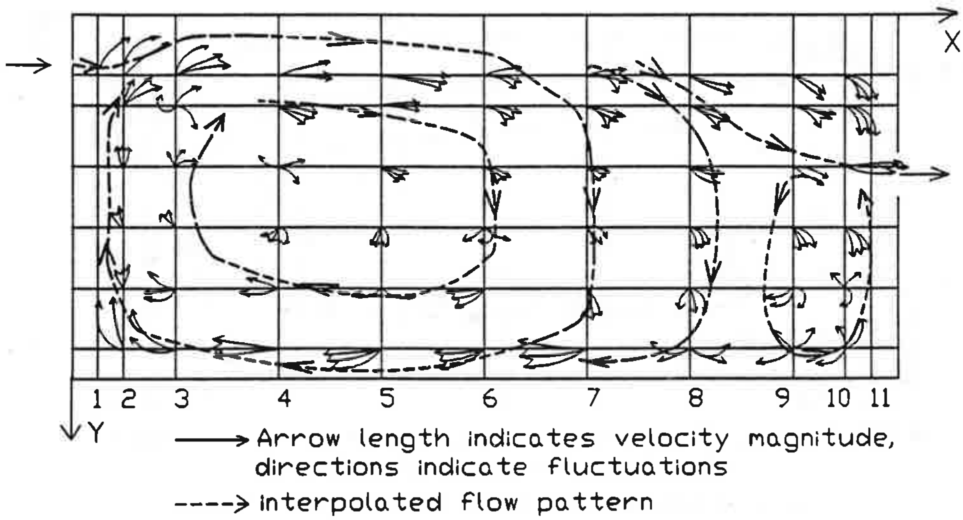


b. Full scale test: U_d=1.78 m/s, ΔT_{fd}=37.4 °C

Figure 1 Comparison of flow patterns between 1/4th scale model and full scale tests #1 - high internal heat load

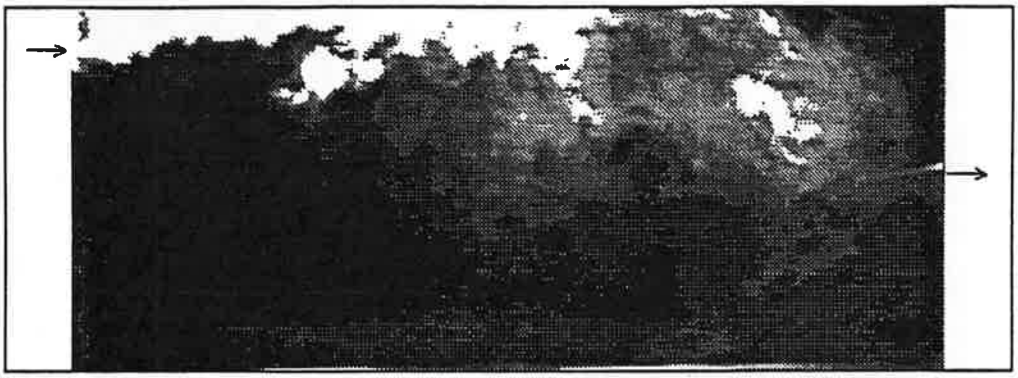


a. Model test: U_d=1.19 m/s, ΔT_{fd}=26.8 °C

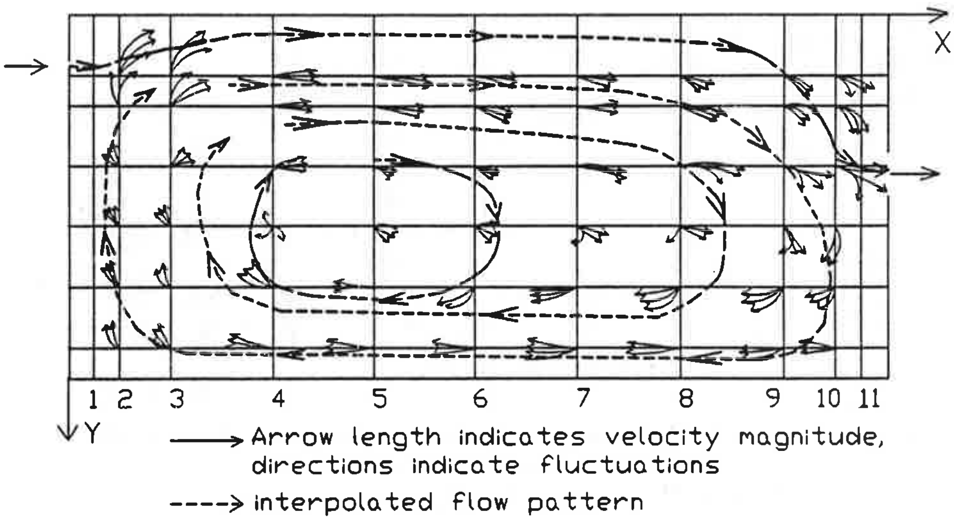


b. Full scale test: U_d=1.78 m/s, ΔT_{fd}=26.6 °C

Figure 2 Comparison of flow patterns between 1/4th scale model and full scale tests #2 - medium internal heat load



a. Model test: U_d=1.19 m/s, ΔT_{fd}=16.8 °C



b. Full scale test: U_d=1.78 m/s, ΔT_{fd}=16.4 °C

Figure 3 Comparison of flow patterns between 1/4th scale model and full scale tests #3 - low internal heat load

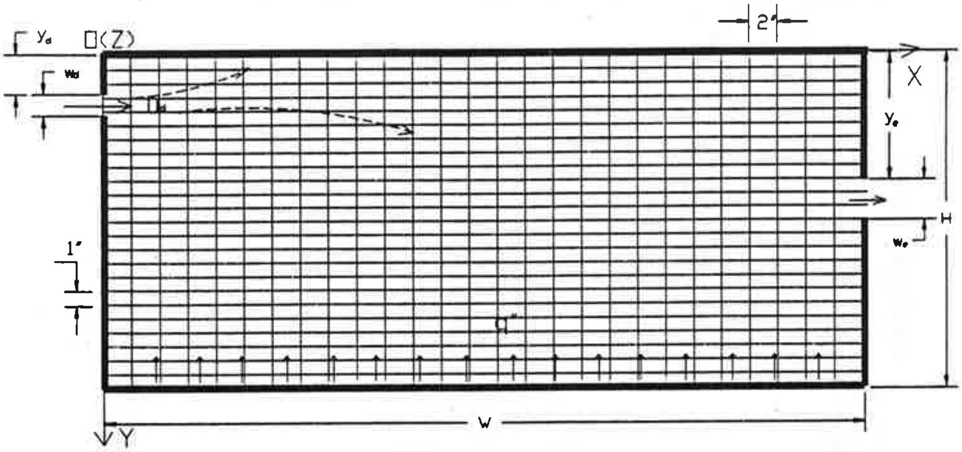


Figure 4 Schematic of the room geometry studied and the definition of coordinates (Z is into the page)

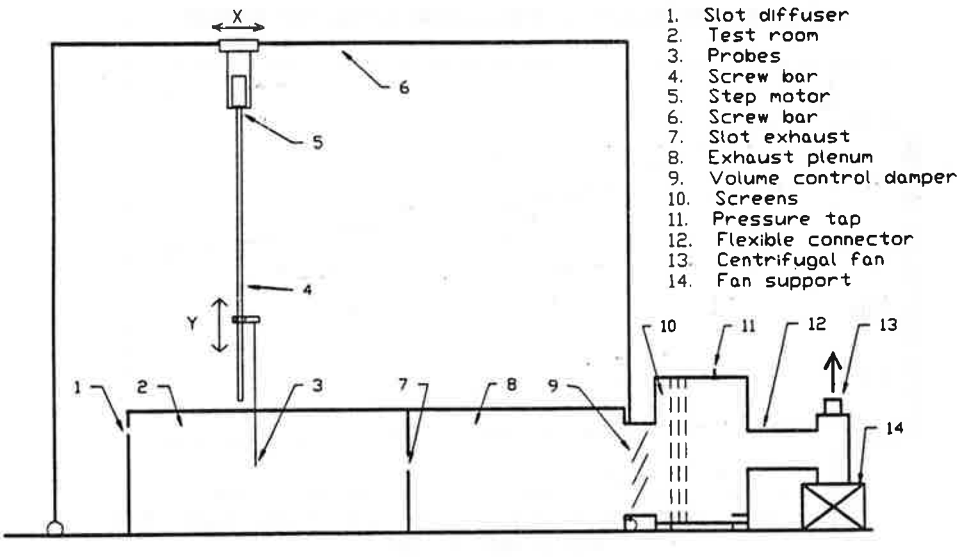


Figure 5 Experimental set up for the 1/4th scale tests

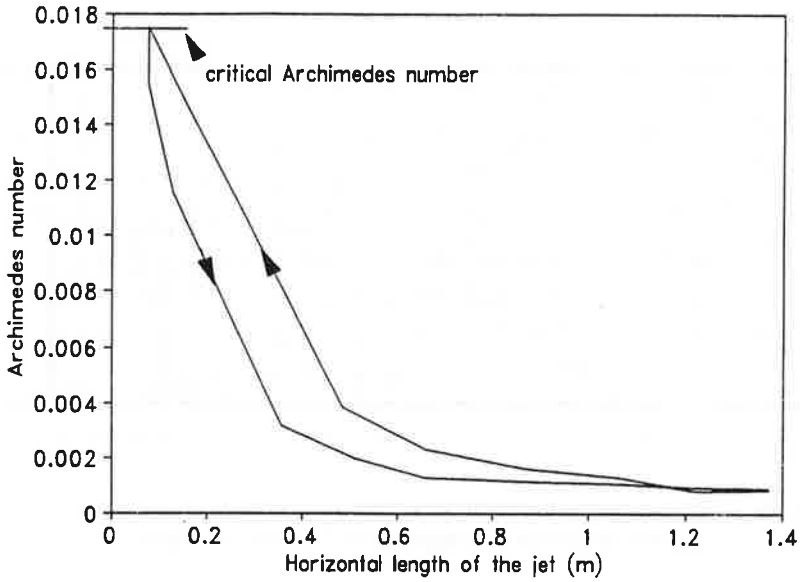


Figure 6 Measurement of the critical Archimedes number

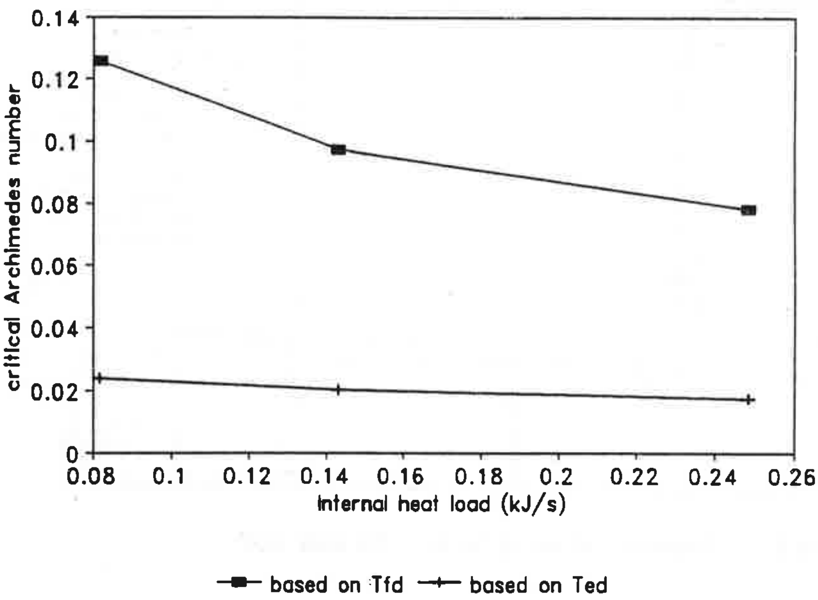


Figure 7 Variation of critical Archimedes number with the internal heat load

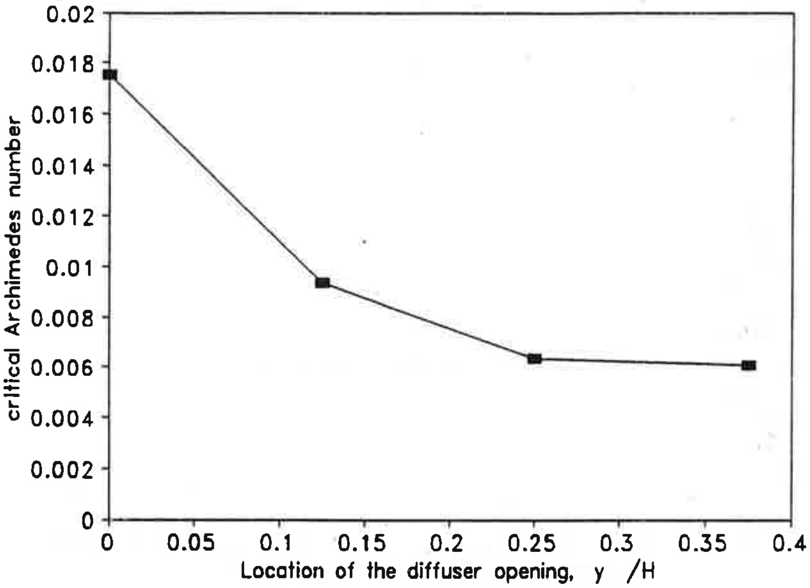


Figure 8 Variation of critical Archimedes number with the location of diffuser opening

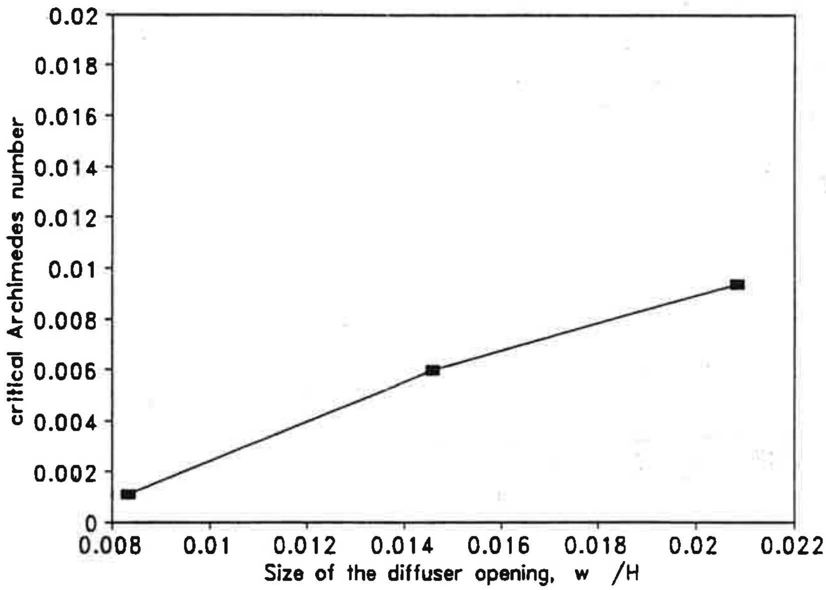


Figure 9 Variation of critical Archimedes number with diffuser opening size

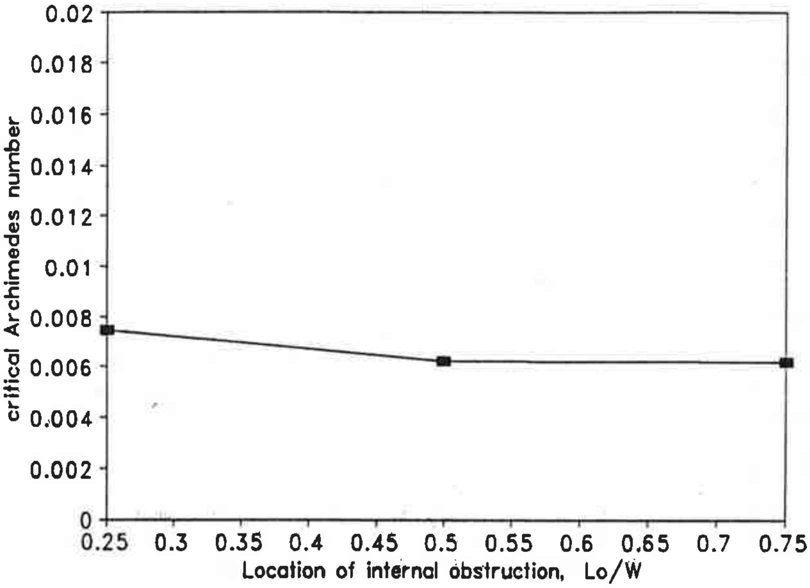


Figure 10 Variation of critical Archimedes number with the location of obstruction