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COMPUTATIONAL AND EXPERIMENTAL REDUCED-SCALE MODELLING OF AIR-CONDITIONED ROOMS

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ABSTRACT

An improper air distribution within air-conditioned rooms is one of the largest causes of inadequate indoor air quality and thermal comfort. A good knowledge of the phenomena allows for the advance of eventual deficiencies, thus becoming a powerful tool for the optimization of new projects or for the improvement of the operation conditions of the projects already implemented.

In this study two methods were applied, one computational and the other experimental, for modelling of non-isothermal turbulent flows in air-conditioned rooms. The computational model consists of a numerical procedure, which solves in finite-difference form using a control volume technique, the three-dimensional time-averaged equations expressing the conservation of mass, momentum, energy and concentration of species. The turbulence is modelled by the k-ε model and thermal radiation by the discrete transfer method. The experimental validation was accomplished through comparison of the numerical predictions with measurements obtained in a laboratory model, designed to provide similarity with the real room. The good agreement obtained suggests a good accuracy for engineering purposes. A simple physical modelling technique, based on dimensional analysis, was used to derive the physical properties of the experimental (reduced) model. The simulation of the humidity, the thermal and air flow patterns in a forced ventilated cooling room was performed.

INTRODUCTION

In the past the design of air-conditioned rooms and the choice of operating conditions have largely been achieved by trial and error. Nowadays the air-conditioning systems are, to a great extent, the product of years of engineering evolution. They perform their functions reasonably well and, until recently, there was not much need to question seriously their design. In the last decade, however,

this need has emerged in the wake of the soaring importance of energy saving and good indoor air quality, in order to achieve the most economic and the best indoor conditions. The indoor air quality is controlled by a building's mechanical ventilation system. The objective of a mechanical ventilation system in a building is usually to provide fresh air and remove contaminants from the air-conditioned space as quickly as possible, as well as meet the heating and/or cooling load of the buildings.

A healthy and pleasant climate usually has a fairly low air velocity, small velocity and temperature gradients throughout the room, and also a low concentration of pollutants. The pattern of air circulation, and therefore mixing, is, in turn, influenced by the location of air supply outlets, exhausts, windows, room geometry, and interior furnishings. It is also affected by the building's heating and cooling system. An improper air distribution within the air-conditioned rooms is one of the largest causes of inadequate indoor air quality and thermal comfort. An accurate understanding of indoor air motion is crucial to the design of building heating, ventilating and air-conditioning systems in providing thermal comfort and indoor air quality, as well as in increasing the energy efficiency of mechanical and electrical systems.

There are many configurations of ventilation air distribution systems and a wide range of potential conditions within residential or office buildings. The prediction of thermal comfort and ventilation performance for any specific situation is difficult. While full-scale measurements of room air motion, distribution of temperature and gaseous components are very time consuming and expensive, requiring sophisticated sensors and instrumentation, three-dimensional numerical simulation offers the ability to predict ventilation characteristics over a wide range of parameters and physical configurations at much less cost. Murakami and Kato (1988) have reviewed the status of numerical and experimental methods for analysis of the flow field in a room. Xu et al. (1994) compared numerical predictions and experimental measurements of room ventilation. The use of small-scale models to study the dynamic response of the built environmental offers also an attractive and viable solution, Imbabi (1990). In fact, the existence of reliable experimental results is fundamentally important for the validation and improvement of the mathematical models. However, experimental validation must be obtained in real conditions or in their similarity.

The objective of the present paper is to describe physical and mathematical models to study the three-dimensional turbulent air flow patterns with thermal buoyant effects, the heat transfer, including radiation between walls, the gas contaminant transport and the moist air transport within mechanically or naturally ventilated spaces. A case study is outlined.

EXPERIMENTAL MODELLING

Modelling Technique

Bioclimatic full-scale tests of buildings are often prohibitively expensive and time-consuming. Small-scale model studies are usually used in order to resolve these difficulties, and to allow the testing of a particular design, or concept, in the laboratory. The generally accepted statement from similarity theory is that a scale model will perfectly replicate the kinematic response of its prototype, if the Prandtl, Reynolds and Archimedes numbers are identical in both systems. This constraint will invariably lead to the conclusion that convective heat transfer can only be accurately modelled on a real 1:1 scale (see, for instance, Rolloos (1977), Awbi (1990)). However, such research is very expensive. Due to the limitations mentioned, another method to obtain similarity can be proposed. Dimensional analysis is commonly used in experimental studies to transform physical parameters from prototype to model values. Buckingham's π-theorem is ideally suited for the manipulation of large numbers of variables. In fact, if the physical phenomenon at hand is governed by a functional relation φ, which involves n variables

$$\varphi(x_1, x_2, x_3, \ldots, x_n)=0 \qquad (1)$$

this problem can be equally described by another functional relation ψ, more compact, just involving k=n-m dimensionless groups (π's)

$$\psi(\pi_1, \pi_2, \ldots, \pi_k)=0 \qquad (2)$$

where m is the number of independent variables in the equation (1). The similarity between model and prototype is obtained through the establishment of the equalities (see Imbabi (1990), Pitarma (1998) for further details)

$$\pi_{model} = \pi_{prototype} \qquad (3)$$

The described technique is used here to obtain similarity between model and prototype.

Prototype and Model

The prototype room used in this study is a 3,80x1,80x1,65 m^3 very insulated light structure room with the common method of air distribution in rooms, represented in schematic diagram in Figure 1. The air flows into the room, in the longitudinal direction, through a supply square opening placed close to the ceiling and leaves the chamber through another square hole below. The geometrical and functional parameters, relevant for the bioclimatic response of the room, considered in the present work are shown in Table 1.

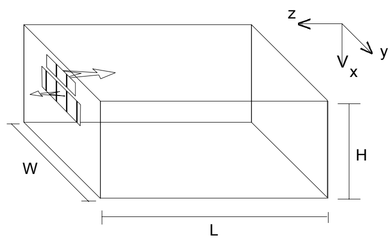

Figure 1 - Sketch of the prototype and experimental room configuration

Table 1 - Dimensionless groups

Parameter	Symbol	SI Units	Dimension	Dimensionless groups
Outside air temperature	T_e	K	$[\theta]$	-
Density of air	ρ	kg m^{-3}	$[ML^{-3}]$	-
Specific heat cap. of air	c_p	J kg^{-1} K^{-1}	$[L^2 T^{-2}\theta^{-1}]$	-
Thermal load	Q	W	$[ML^2T^{-3}]$	$\pi_1 = Q\, T_e^{-1.5}\, c_p^{-1.5}\, \rho^{-1}\, L^{-2}$
Wall thermal resistance	R	m^2 K W^{-1}	$[T^3\theta M^{-1}]$	$\pi_2 = R\, T_e^{0.5}\, c_p^{1.5}\, \rho$
Length	L	m	[L]	-
Width	W	m	[L]	$\pi_3 = W\, L^{-1}$
Height	H	m	[L]	$\pi_4 = H\, L^{-1}$
Time	t	s	[T]	$\pi_5 = t\, T_e^{0.5}\, c_p^{0.5}\, L^{-1}$
Inlet air temperature	T_i	K	$[\theta]$	$\pi_6 = T_i\, T_e^{-1}$
Inlet width	W_i	m	[L]	$\pi_7 = W_i\, L^{-1}$
Inlet height	H_i	m	[L]	$\pi_8 = H_i\, L^{-1}$
Outlet width	W_o	m	[L]	$\pi_9 = W_o\, L^{-1}$
Outlet height	H_o	m	[L]	$\pi_{10} = H_o\, L^{-1}$
Air flow rate	V	m^3 s^{-1}	$[L^3T^{-1}]$	$\pi_{11} = V\, T_e^{-0.5}\, c_p^{-0.5}\, L^{-2}$
Air velocity	v	m s^{-1}	$[LT^{-1}]$	$\pi_{12} = v\, T_e^{-0.5}\, c_p^{-0.5}$
Convection coefficient	h	W m^{-2} K^{-1}	$[MT^{-3}\theta^{-1}]$	$\pi_{13} = h\, T_e^{-0.5}\, c_p^{-1.5}\, \rho^{-1}$

Thus, if L, T, Cp and ρ are chosen as independent variables, the various dimensionless groups, detailed in the same table, may be deduced. Finally, the similarity between model and prototype is obtained through the establishment of the equalities $\pi_{model}=\pi_{prototype}$, that were used to derive the model parameters listed in Table 2.

In accordance with Table 2, the experimental enclosure consists of 1,52x0,72x0,66 m3 box with 6mm thick "Perspex" glass walls. The entire apparatus is insulated with two snugly fitting layers of 50mm thick expanded polystyrene, in order to achieve insulating thermal conditions given in the same table.

Table 2 - Prototype and model parameters

Parameter	SI Units	Prototype value	Scale factor	Model value
T_e	K	298	1,0	298
ρ	kg m^{-3}	1,20	1,0	1,2
c_p	J kg^{-1} K^{-1}	1006	1,0	1006
Q	W	309	6,25	49,4
R	m^2 K W^{-1}	3,45	1,0	3,45
L	m	3,80	2,5	1,52
W	m	1,80	2,5	0,72
H	m	1,65	2,5	0,66
t	S	1,00	2,5	0,4
T_i	K	278	1,0	278
W_i	m	0,60	2,5	0,24
H_i	m	0,15	2,5	0,06
W_o	m	0,80	2,5	0,32
H_o	m	0,30	2,5	0,12
V	m^3 s^{-1}	0,279	6,25	4,464E-2
v	m s^{-1}	(Test)	1,0	(Test)
h	W m^{-2} K^{-1}	(Test)	1,0	(Test)

Auxiliary Equipment and Measurements

An auxiliary experimental installation which applies a computer linked air conditioning laboratory unit schematically illustrated in Figure 2, will be used to reproduce the inlet air flow conditions in the experimental room, given in Table 2. A complete description of the experimental installation can be found in Pitarma (1998). The inlet air temperature, Ti=5,0ºC, was controlled with a precision of 0,25ºC. The mean inlet velocity was Vi=3,1m/s, but velocity measurements were made to characterize the velocity distribution at the inlet section. The internal room load was produced, with an accuracy of 2%, by electrically heated tapes laid over the floor area to produce a uniform load distribution.

A vertical rake of 7 thermocouples (T type, 200μm wire diameter) was used for temperature measurements. Temperature signals were acquired by a Data Translation board (DT2811/DT756Y), connected to an HP Vectra microcomputer, with an accuracy of 0,25ºC (±5%). The sampling rate, selected after few preliminary runs, was 1Hz (i.e. each one of the 7 channels was sampled every 7 seconds).

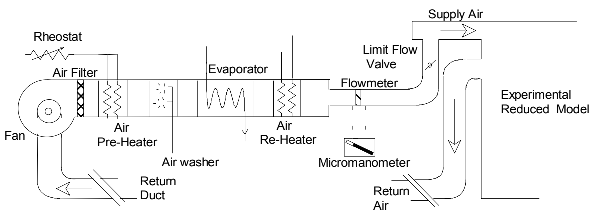

Figure 2 - Sketch of auxiliary installation to conditioning supply air

The air velocity magnitude ($V=\sqrt{u^2+v^2+w^2}$) measurements were obtained by a hot-wire anemometer system connected to an HP Vectra microcomputer, with an omnidirectional hot-film sensor. This hot-wire anemometer system uses miniature probes and therefore low disturbance to the flow occurs. The control of the whole system, including acquired data and its treatment, was made by a software support, in Windows environment (see Pitarma (1998) for a detailed description of this system). The accuracy of measurements is within ±1% (typically 0,5%)±0,02m/s.

Complementary information of the airflow was obtained by flow visualization. Due to its mixing, flow visualization was carried out through orientation acquired by nylon fibre placed inside the model, in two longitudinal planes (close to symmetry plane and close to lateral wall) and in a traverse plane near the downstream wall.

MATHEMATICAL MODELLING

Mathematical Formulation

Airflow in air-conditioned space is normally three-dimensional, recirculating, and turbulent. The mathematical model consists of the continuity equation, momentum equations, enthalpy equation, moisture concentration equation, concentration of species equation and the k-ε turbulence equations. For incompressible steady-state flow the model is represented by the following time-averaged equation expressed in tensor notation:

$$\frac{\partial}{\partial x_i}(\rho U_i \phi) = \frac{\partial}{\partial x_i}\left(\Gamma_t \frac{\partial \phi}{\partial x_i}\right) + S_\phi \qquad (4)$$

where ϕ represents the dependent variable (=1 for the continuity equation; = U, V, W, h, ω and C for momentum, energy, moisture and contaminant equations). Γ_ϕ is the diffusion coefficient and S_ϕ represents the source term. A detailed description of the transport equations is given, for example, by Patankar (1980). The turbulence model employed to calculate the turbulent fluxes is the k-ε two-equation turbulence model representing the kinetic energy, k, and its rate of dissipation, ε (k-ε turbulence model, Launder and Spalding (1974)). This k-ε turbulence model is relatively efficient and stable computationally compared with the more complicated Reynolds stress models, yet is also reasonably accurate for a wide range of turbulent flows, Awbi (1995). The effects of buoyancy are included both on the vertical component of velocity and on the turbulence model. Because of the damping effect of the wall, the transport equation for the turbulence quantities does not apply close to the wall. So, the boundary conditions at the walls for velocity components, k-ε and thermal energy are specified using algebraic relations, so-called wall functions. Details about the origin and applications of wall functions can be found in Launder and Spalding (1972).

Numerical Procedure

The numerical procedure is based on a finite-volume discretization of the governing equations, employing a staggered grid for mean-velocity components relative to scalar properties. The hybrid central/upwind differencing scheme is used to approximate the convection terms. One method based on the Simple algorithm was chosen for the pressure-velocity coupling correction, Patankar (1980). The solution of the individual equation sets was obtained by a form of Gauss-Seidel line-by-line iteration. The simulation model was used to predict the velocity and temperature fields in the reduced-scale model, described above. The computations were performed using an 11x9x19 control volumes grid. In accordance with the symmetry of the experimental chamber and their operating conditions, only half of the flowfield was covered by computational domain. Grid-dependence tests were carried out indicating that the differences between the results in this grid and a 22x18x38 control volumes grid are not significant. The sums of the absolute residuals of mean field variables were used for monitoring convergence. The iterative process was terminated after the normalized residuals had fallen below 0,05%. The computations were performed on a HP Apollo 720 workstation, and the required time of CPU to achieve convergence was 39 minutes.

RESULTS

The predictions were validated against experimental data acquired in the reduced model. Experiments include measurements of mean air velocity and mean air temperature distribution that allow testing the experimental and numerical models performance. Figures 3 and 4 show, respectively, a comparison between computed and measured velocity and temperature nondimensional profiles (attention to the assumed symmetry). Inlet conditions were used to normalize the computational and measured results.

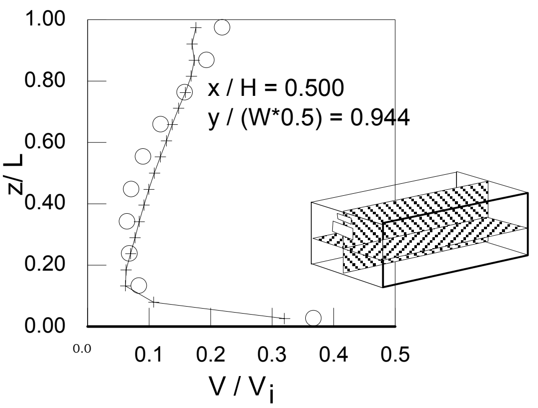

Figure 3 - Calculated and measured non-dimensional velocity (magnitude) profiles (o measurements; +– predictions)

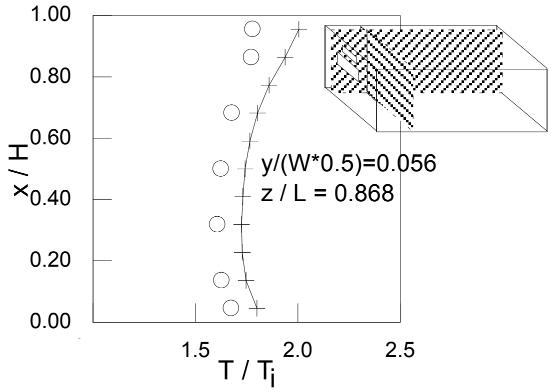

Figure 4 - Calculated and measured non-dimensional temperature profiles (o measurements;+– predictions)

It can be seen from these figures that predicted values of air velocity and temperature are consistent with measurements. In addition, the predicted and experimentally observed airflow patterns are qualitatively consistent.

Figure 5 gives an example of the calculated velocity and temperature fields at the symmetry plane. The velocity field reveals the flow patterns and the vortices formation of the most common method of air distribution in rooms. In spite of the high airflow, this rate of air circulation will result in an air temperature range of ≈4K. As expected, due to high inlet velocity and the so-called Coanda effect, the wall jet from the supply opening follows the ceiling and entrains air from the occupied zone to induce a recirculating air movement (see figure 5). The velocity in the wall jet accordingly decreases as it reaches the occupied zone and the jet appears to detach from the ceiling close to the opposite wall. The return air circulation close to the floor due to suction effect of the outlet can be seen in the longitudinal plane presented, where the velocity vectors are essentially horizontal. In accordance with the convective effect of the flow pattern, the temperature contours near the floor indicate that the temperature is decreasing from inlet wall to opposite wall.

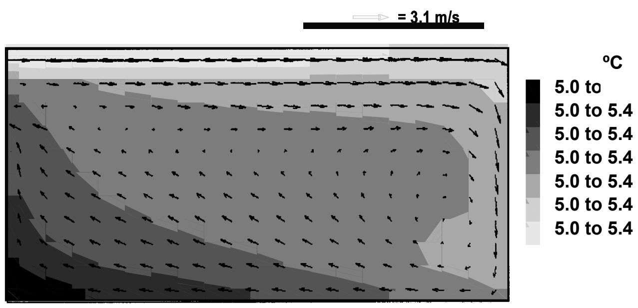

Figure 5 - Predicted velocity vectors and distribution temperature contours

Therefore, the temperature remains high near the outlet wall, particulary close to the corners, which suggests that this is the critical zone of the room, certainly characterized by maximum levels of temperature and pollutants. These results reveal the need of an improvement in the common systems of air distribution in order to obtain major uniform properties.

CONCLUSIONS

Computational and experimental reduced-scale models, for the simulation of three-dimensional non isothermal flows in air-conditioned rooms, are presented. The similarity between reduced-model and prototype room was achieved through a simple physical modelling technique, based on dimensional analysis. To validate the computer model, the calculated results are compared with those from the experimental tests. The agreement obtained suggested their accuracy for engineering purposes.

According to the results of both techniques, the following main conclusions can be drawn: i) The suggested modelling techniques offers a means of studying a wide range of air-conditioned room problems, with low costs and simplicity; ii) Before the real construction, still in the design phase, it is highly desirable that a preliminary assessment of indoor environmental conditions based on these types of prediction methods be made. Here, numerical prediction is very promising; iii) Due to its lower costs and acceptable accuracy, the computational models can be conveniently used in practice; iv) But, the experimental validation must be obtained in real conditions or in their similarity. The use of small-scale models offers an attractive and viable solution.

REFERENCES

Awbi, H.B., Nemri, M.M. 1990. Scale effect in room airflow studies, Energy and Buildings, 14, pp. 207-210.

Awbi, H. B. 1995. Ventilation of buildings, E. & F.N. Spoon Press, Second Edition, London.

Carvalho, M.G., Ramos, J.E., Pitarma, R.A. 1999. Characterization of natural convection in a room communicating with the outside environment through a door, Passive and Low Energy Architecture-PLEA'99, Brisbane, Australia, September.

Imbabi, M.S. 1990. A general procedure for the small-scale modelling of buildings, International Journal of Energy Research, Vol. 14, pp. 311-321.

Launder, B.E., Spalding, D.B. 1972. Mathematical models of turbulence, Academic Press, London.

Launder, B.E., Spalding, D.B. 1974. The Numerical Computations of Turbulent Flows, Computer Methods in applied mechanics and engineering, Vol. 3, pp. 269-289.

Murakami, S., Kato, S. 1988. Current status of numerical and experimental methods for analysing flow field and diffusion field in a room, Building Systems: Room air and air contaminant distribution, pp. 39-55, ASHRAE, Atlanta.

Patankar, S.V. 1980. Numerical heat transfer & fluid flow, Hemisphere Publishing Corporation, Washington.

Pitarma, R.A. 1998. Modelação Matemática e Experimental de Câmaras Frigoríficas de Veículos, Ph.D. Thesis, Technical University of Lisbon, Portugal.

Ramos, J.E. 1998. Modelação do Comportamento Térmico e Ambiental em Edifícios, Ph.D. Thesis, Technical University of Lisbon, Portugal.

Rodi, W. 1982. Examples of Turbulence Models for Incompressible Flows, AIAA Journal, vol. 20, nº 7.

Rolloos, M. 1977. Possibilities and limitations for the prediction of air flow patterns, temperature and velocities in large halls using scale models, Proc. Meeting of Commission E1 of the International Institute of Refrigeration, Yugoslavia, pp. 245-256.

Van Doormaal, J.P., Raithby, G.D. 1984. Enhancements of the SIMPLE method for predicting incompressible fluid flows, Numerical Heat Transfer, vol. 7, pp. 147-163.

Xu, J., Liang, H., Kuehn, T.H. 1994. Comparison of numerical predictions and experimental measurements of ventilation in a room, Proceedings of RoomVent '94: Air distribution in rooms, Fourth International Conference, vol.2, pp. 214-227, Krakow, Poland.