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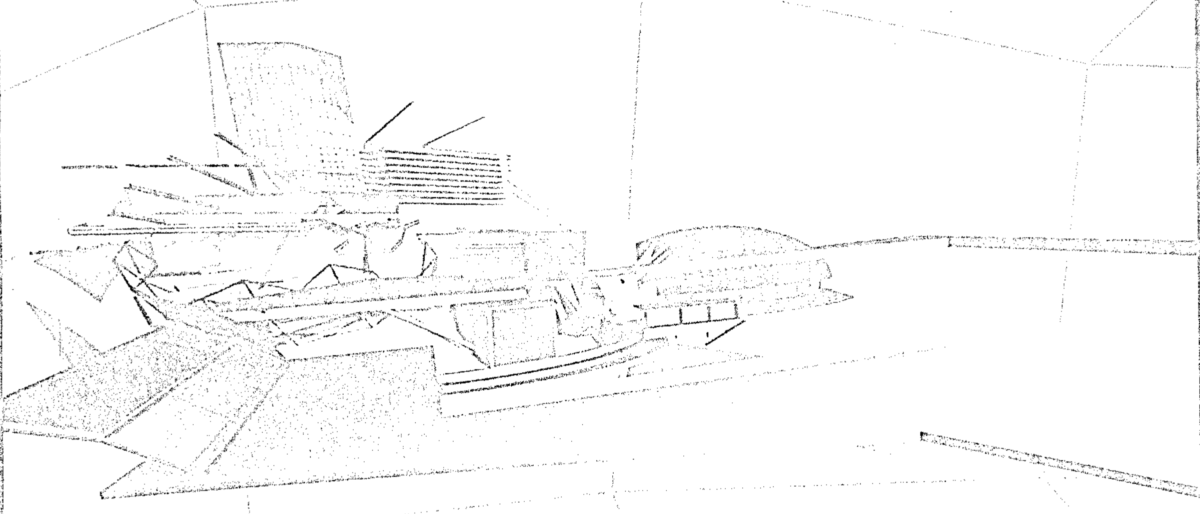

HOUSING CONSTRUCTION
An Interdisciplinary Task

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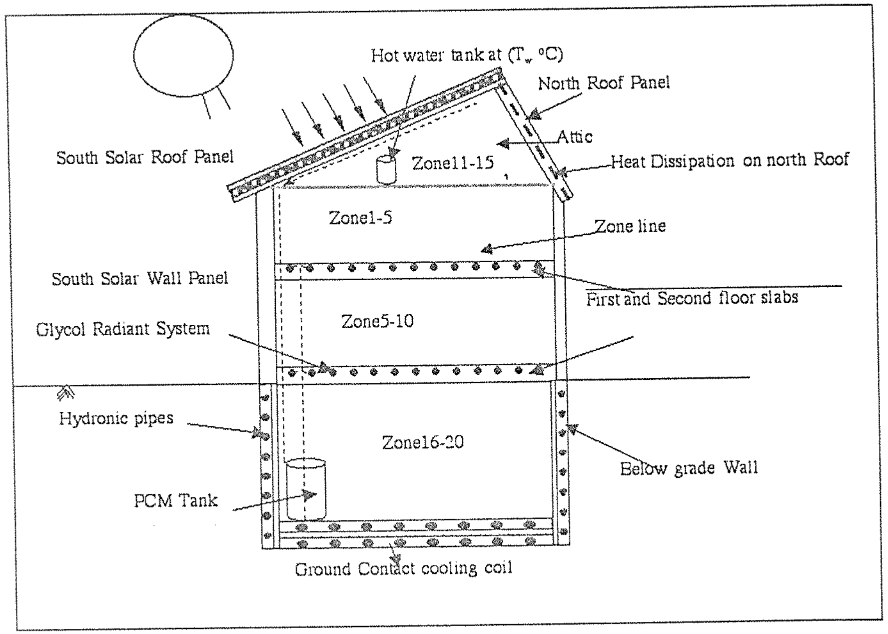

Figure 1: Schematic of the Holistic House System

4 Conclusions

There is an opportunity to do much more for energy savings and building life-cycle cost if we can look at the issues form a holistic and integrated perspective. This effort will continue for the next year with a continuation of a Holistic House class that has been offered to students for the past year. These students are at every level of undergraduate and graduate curriculum as well as representing architecture, building construction, civil engineering and mechanical engineering.

Thermal and Environmental Simulation of Buildings

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Key words: thermal comfort, indoor air quality, ventilation, thermal radiation, physical modelling, numerical simulation, experimental validation

Abstract

The purpose of most buildings is to provide a comfortable and healthy indoor environment for the occupants. As a result of increasing energy prices in the last decades, much effort has been done to reduce the cost arising from mechanical and natural ventilation of buildings. This often involves lowering the outdoor air intake. At the same time, many new pollution sources have been introduced in the buildings. These issues are more important if we take into consideration that today the majority of the people spends more than 90% of their time indoors.

In this work will be presented a 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 into mechanical or natural ventilated spaces. The calculations made use of a numerical procedure, which solves the three-dimensional equations for the conservation of mass, momentum, thermal energy and air contaminant transport. The turbulence is modelled by the k-ε model and thermal radiation by the discrete transfer method. To validate the mathematical model, the predictions results were compared with experimental data for two different scales - full-scale office room and 1/4 laboratory scale model. A physical modelling technique, based on dimensional analysis, was used to derive the physical properties of the laboratory model, designed to provide similarity with the room. The simulation of the air contaminant distribution, the thermal and the air flow patterns in an office natural ventilated room (prototype and model) was performed.

1 Introduction

The purpose of natural or mechanical ventilation of buildings is to supply fresh air, remove heat load or supply heating and create a health and uniform climate in the occupied zone. A health and pleasant climate is defined as a fairly low air velocity, small velocity and temperature gradients throughout the room, and a low concentration of pollutants. Airflow through doorways, windows and other large openings are significant ways in which air, pollutants and thermal energy are transferred from one zone of a building to another or to outside. 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.

Measurements of room air motion, distribution of temperature and gaseous components are very time consuming and expensive, requiring sophisticated sensors and instrumentation techniques. The use of small-scale models to study the dynamic response of the built environmental offers an attractive and viable solution [1]. In the other hand, with the increasing computer power and the fluid dynamics research, several methods have been developed based on numerical calculation of fundamental field equations for the analysis of fluid phenomena. Since the first application to room air distribution in 1973 [2], there has been great interest in developing CFD computer codes for predicting the airflow in ventilating rooms [3,4]. The proposed 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 into mechanical or natural ventilated spaces, are described, and a case study outlined, in the course of this paper.

2 Experimental modelling

2.1 The prototype

The prototype room used in this study is shown in an schematic layout in Figure 1. It is a typical 18 m² naturally ventilated office room, communicating with a 2,5m high corridor via a 2m high hinged wooden door opening. Inside the room, near the front wall, there are 4 electric air heater convective and radiant units. Two of them were 500 W and the other were 800 W each. Experimental techniques are used to measure the air temperature distribution, to measure the velocity air flow pattern and to have flow visualisations (see [5]). A vertical support with 6 Cu/Cu-Ni isolated thermocouples (0,5 mm wire diameter) was used for air temperature measurements inside the room and an other similar support with 8 thermocouples was used for the door opening section. Thermocouple signals were acquired by a 12 bits Data Translation board (DT 2811 / Dt 756 Y), with a precision of 0,5°C. The air velocity components were measured using an Airflow TA 400 temperature compensated small sphere probe with a 0,06 m/s precision. Injection of smoke in the air is probably the most common method of visualising the air motion in a room. The smoke particles when illuminated cause scattering of light and their movement can then be traced and photographed. In this work flow visualisation was carried out in the door opening section and inside the room.

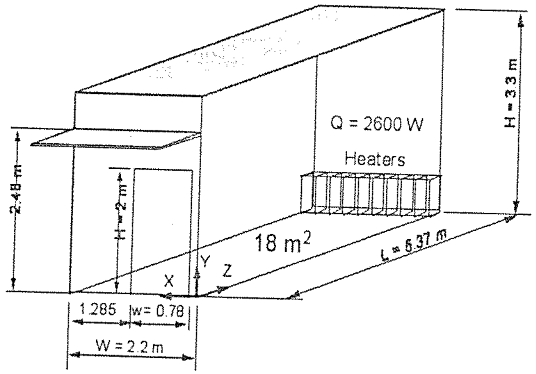

Figure 1: Room configuration.

2.2 Modelling strategy

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. For fluid flow phenomena analysis it has been suggested that similarity between the model and prototype exist if the Prandtl, Reynolds and Grashof requirements are satisfied. This constraint will invariably lead to the conclusion that convective heat transfer can only be accurately modelled on a real 1:1 scale (see [6]). However, such research is very expensive. Due to the presented limitations, 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. Therefore, this technique is used to obtained similarity between model and prototype (see [5,6] for further details).
The geometrical and functional parameters, relevant for the bioclimatic response of the building, considered in the present work are shown in Table 1. Thus, if L, T, Cp e ρ are chosen as independent variables, the various dimensionless groups, detailed in the same table, may be derived. Finally, the similarity between model and prototype is obtained through the establishment of the equalities $\pi_{model}=\pi_{prototype}$, that was used to derive the model parameters listed in Table 2.

2.3 The laboratory model

Model specifications are presented in Table 1. Naturally that the geometric form of room is preserved by the model (Fig. 1). In accordance with the modelling strategy (Table 1), the experimental enclosure consists of a 0,55x1,34x0,80 m³ box with 6 mm thick "perspex" glass walls. The apparatus is insulated with polyurethane plates in order to achieved insulating thermal conditions given in the same table. Miniature thermocouples and hot-wire anemometer was used for temperature and air-movement measurements, respectively.

Table 1: Dimensionless groups.

Parameter	Symbol	SI Units	Dimension	Dimensionless groups
Reference temperature	ΔT_0	K	$[\theta]$	-
Thermal local	Q	W	$[ML^2T]$	$\pi_1 = Q\,\Delta T_0^{-1.5}\,c_p^{-1.5}\,\rho^{-1}\,L^{-2}$
Density of air	ρ	$kg\ m^{-3}$	$[ML^{-3}]$	-
U_w – value of walls	U_w	$W\ m^{-2}\ K^{-1}$	$[ML^{-3}\theta^{-1}]$	$\pi_2 = U_w\,\Delta T_0^{-0.5}\,c_p^{-1.5}\,\rho^{-1}$
U_{fw} – value of front wall	U_{fw}	m	[L]	$\pi_3 = U_{fw}\,\Delta T_0^{-0.5}\,c_p^{-1.5}\,\rho^{-1}$
U_f – value of floor	U_f	m	[L]	$\pi_4 = U_f\,\Delta T_0^{-0.5}\,c_p^{-1.5}\,\rho^{-1}$
U_c – value of ceiling	U_c	m	[L]	$\pi_5 = U_c\,\Delta T_0^{-0.5}\,c_p^{-1.5}\,\rho^{-1}$
Length	L	m	[L]	-
Width	W	m	[L]	$\pi_6 = W\,L^{-1}$
Height	H	m	[L]	$\pi_7 = H\,L^{-1}$
Door width	H_d	m	[L]	$\pi_8 = H_d\,L^{-1}$
Door height	W_d	m	[L]	$\pi_9 = W_d\,L^{-1}$
Time	t	s	[T]	$\pi_{10} = t\,\Delta T_0^{0.5}\,c_p^{0.5}\,L^{-1}$
Specific heat cap. of air	c_p	$J\ kg^{-1}\ K^{-1}$	$[L^2\,T^{-2}\theta^{-1}]$	-
Air flow rate	V	$m^3\ s^{-1}$	$[L^3T^{-1}]$	$\pi_{11} = V\,\Delta T_0^{-0.5}\,c_p^{-0.5}\,L^{-2}$
Air velocity	v	$m\ s^{-1}$	$[LT^{-1}]$	$\pi_{12} = v\,\Delta T_0^{-0.5}\,c_p^{-0.5}$
Convection coefficient	h	$W\ m^{-2}\ K^{-1}$	$[MT^{-3}\theta^{-1}]$	$\pi_{13} = h\,\Delta T_0^{-0.5}\,c_p^{-1.5}\,\rho^{-1}$

Table 2: Prototype and model parameters.

Parameter	SI Units	Prototype value	Scale factor	Model value
ΔT_0	K	(Test)	1,0	(Test)
Q	W	2600	-	16,5
ρ	$kg\ m^{-3}$	1,2 (Test)	1,0	1,20
U_w	$W\ m^{-2}\ K^{-1}$	2,40	1,0	2,40
U_{fw}	m	1,40	1,0	1,40
U_f	m	1,70	1,0	1,70
U_c	m	1,90	1,0	1,90
L	m	5,37	4,0	1,34
W	m	2,20	4,0	0,55
H	m	3,30	4,0	0,80
H_d	m	2,00	4,0	0,50
W_d	m	0,78	4,0	0,194
t	s	1,00	2,5	0,40
c_p	$J\ kg^{-1}\ K^{-1}$	1006	1,0	1006
V	$m^3\ s^{-1}$	(Test)	16,0	(Test)
v	$m\ s^{-1}$	(Test)	1,0	(Test)
h	$W\ m^{-2}\ K^{-1}$	(Test)	1,0	(Test)

3 Mathematical modelling

The Reynolds-averaged form of the governing equations for a steady, incompressible and turbulent flow expressed in tensor notation read:

$$\frac{\partial U_i}{\partial x_i} = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}\left(U_j \Theta\right) = -\frac{\partial}{\partial x_j}\left(<u_i \theta>\right) \tag{2}$$

$$\frac{\partial}{\partial x_j}\left(U_j U_i\right) = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} - \frac{\partial}{\partial x_j}\left(<u_i u_j>\right) + \beta g_i \Theta \tag{3}$$

where U_i and u_i represent the x_i component of mean and fluctuating velocity, respectively; P is the pressure and ρ the fluid density; Θ represents the mean temperature difference, $T-T_0$, where T_0 is the room average temperature. Θ is the temperature fluctuation; β is the coefficient of thermal expansion and g the gravitation constant. Finally, < > stands for the temporal average of the quantity. A detailed description of the transport equations is given, for example, by [5,6]. The turbulence model employed to calculate the turbulent fluxes is a two-equation turbulence model representing the kinetic energy, k, and its rate of dissipation, ε (k-ε turbulence model [7]). The effects of buoyancy are included both on the vertical component of velocity and on the turbulence model. Further details of the k-ε turbulence models are found in [8]. The boundary conditions at the walls for velocity components, k-ε, and thermal energy are specified using wall functions. Details about the wall functions can be found in [8].

The numerical procedure is based on a finite-volume discretization of the governing equations, employing a staggered grid for mean-velocity components relatively to scalar properties. The hybrid central/upwind differencing scheme is used to approximate the convection terms. One method based in the Simple algorithm was chosen for the pressure-velocity coupling correction [9]. The solution of the individual equation sets was obtained by a form of Gauss-Seidel line-by-line iteration.

3.1 Results

The predictions were validated against experimental data acquired in the prototype room and laboratory model. Experiments include measurements of mean air velocity and mean air temperature distribution, that allow testing the experimental and numerical models performance. The numerical results presented here (prototype and model) are for a non-uniform mesh grid size of 13x37x30 (8x15x10 for the radiation model) and non-isothermal airflow conditions. The maximum residual in the mass conservation was $< 5.10^{-3}$. Figure 2 shows the comparisons between the predicted and its data, on the same plane, in the door opening section (z/L=0,0). The comparisons between the measurements and the predictions indicated that the predicted flow pattern follows closely the experimental values. The Figure 3 shows the predicted steady-state longitudinal mean air velocity vectors on a longitudinal plane (x=0,53m).

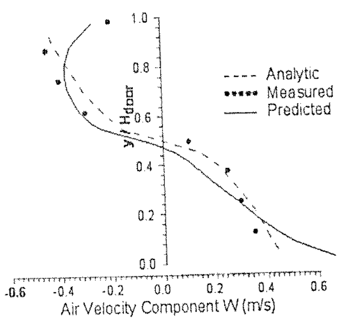

Figure 2: Comparison between calculated and measured results.

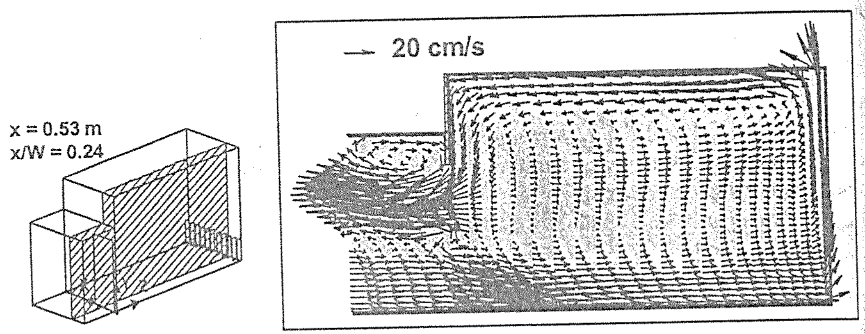

Figure 3: Predicted velocity vectors on a longitudinal plane (prototype).

A sample of the room flow pattern description is shown in Figure 3. The incoming air flows from the corridor through the lower half-height door opening and then takes the form of a gravity current flowing horizontally along the floor in the direction of the air heaters. Due to buoyancy effect, the air goes up to the ceiling. The outgoing air flows to the doorway direction where it leaves the room to the corridor (cold zone), as a turbulent jet, by the upper half-height door opening.

The measurements of air temperature distribution are also consistent with the CFD predictions (see Fig. 4, relative to the model). In fact, the predicted vertical temperature stratification is generally in good agreement with the experiment. The numerical results show a cooler hot zone, near the air heaters, than that observed experimentally (Fig. 5).

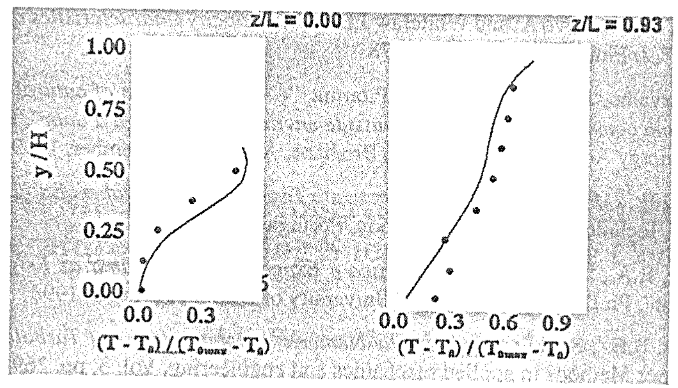

Figure 4: Comparison between the air temperature prediction and its data in two sections (model).

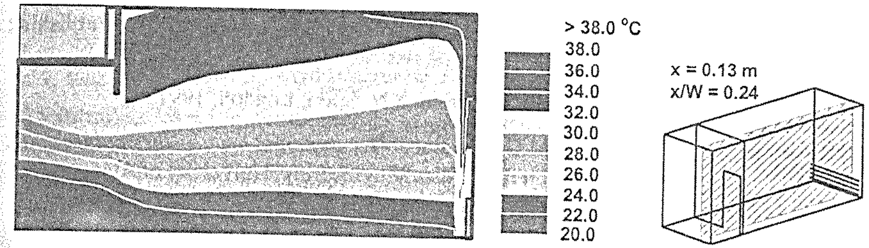

Figure 5: Predicted air distribution temperature (model).

4 Conclusions

Computational and experimental reduced-scale models for the simulation of bioclimatic building response are presented. A physical modelling technique, based on dimensional analysis, was used to derive similarity between laboratory model and prototype room. To validate the computer model, which solves in finite-volume method the three-dimensional time-averaged equations for the conservation of mass, momentum, energy and contaminants, results are compared with those from experimental tests. The good agreement obtained suggested their good accuracy for engineering purposes. This suggests that before real construction, still in design phase, it is highly desirable that as preliminary assessment of thermal comfort and indoor air quality conditions based on these types of predictions methods be made. Here, numerical prediction is very promising.

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Steel Mechanical Properties Evaluated at Room Temperature after Being Submitted at Fire Conditions

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Key words: mechanical properties, fire conditions, residual stress relief, metallurgic phase transformation

Abstract

Several researchers have already studied the effect of temperature increase in steel structures regarding load-bearing capacity and some material models have been developed in fire conditions, based on experimental results. The mechanical properties of a S275 JR steel construction material has been tested at elevated temperatures after a natural and forced cooling. The major focus has been made on yield and ultimate strength behavior, hardness material and residual stress relief. Experimental results will be presented for each temperature level, dwell time and cooling rate. Test results were compared with the normal room temperature condition. The material specimens were cutted from the web of each unit length IPE 100 profiles, after being heated by means of electro ceramic mat resistance and tested at the universal machine, according to NP EN 10002-1 standard. Microstructure analysis has been done regarding each different steel state, based on the standard NP-1467. The hardness tests has been determined for some steel specimen conditions over the entire cross section, according to NP 4072 standard. The results shown that material yield strength from the tested heated specimens are smaller and grater when compared with the yield strength of the used material at room temperature, depending on the temperature level during fire and respective cooling rate. Residual stresses were measured as closed as possible to ASTM E837-01. A stress relief was thermally induced and the final stress state measured and compared with the initial state.