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MODELLING THE SPATIO-TEMPORAL TEMPERATURE DISTRIBUTION IN AN IMPERFECTLY MIXED VENTILATED ROOM

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ABSTRACT

In this study the spatio-dynamic temperature response in a ventilated room to variations of the supply air temperature was modelled for a wide range of ventilation rates. The model structure was first formulated by applying standard heat transfer theory to zones of better mixing. Spatio-temporal temperature data were then exploited in statistical terms to estimate the physically meaningful model parameters. The dynamic model yielded an excellent fit to the experimental data and was found to characterise the spatially heterogeneous nature of the air flow pattern quite well.

KEYWORDS

imperfect mixing, ventilated room, spatio-temporal temperature distribution, air flow pattern, well mixed zone, heat transfer theory, experimental data

INTRODUCTION

Every living organism (man, animal, plant) lives in an imperfectly mixed fluid which is characterised by spatio-temporal temperature gradients under non-isothermal conditions. In many process rooms (livestock buildings, greenhouses,...) it is desirable to control these temperature gradients in order to achieve optimum process quality (production results, comfort,...) with a minimum use of energy. For this purpose, advanced model-based control theory can be applied. Before this becomes possible, it is first required to have a dynamic mathematical model which describes the spatio-temporal temperature distribution in the process room. The objective of this study was to develop such a model and to evaluate its efficacy in a laboratory test room under controlled conditions.

LABORATORY TEST ROOM

The study was carried out in a large instrumented laboratory test room (length 3 m, width 1.5 m, height 2 m, volume 9 m^3) which is schematically depicted in figure 1. It is a mechanically ventilated room with a slot inlet in the left sidewall just beneath the ceiling (1 in figure 1) and an asymmetrically positioned, circular air outlet in the right sidewall just above the floor (2 in figure 1). An envelope chamber (6 in figure 1) is built around the primary test room to minimise disturbing effects of varying laboratory conditions. The volume of the buffering interspace or buffer zone is 21 m^3. The primary test room and the envelope chamber are both constructed of transparent Plexiglass through which the air flow pattern can be observed during flow visualisation experiments.

A series of five aluminium heating elements (3 in figure 1) and a shallow hot water reservoir (4 in figure 1) are placed at the floor to physically simulate the heat and moisture production of the occupant(s). A mechanical ventilation system enables an accurate control of the ventilation rate in the range 70 - 420 m^3/h. A heat exchanger in the supply air duct is used to regulate the temperature of the inflowing air. Supply air temperatures in the range 10 - 30 °C can be achieved.

To measure the spatio-temporal temperature distribution in the test room, 36 thermocouples are positioned in a 3-D measuring grid (5 in figure 1). Thermocouples are further located in the air inlet and outlet, in the buffer zone and in the laboratory hall. The accuracy of the thermocouples is 0.1 °C. An intelligent data logger with programmable measurement speed is used for the data acquisition.

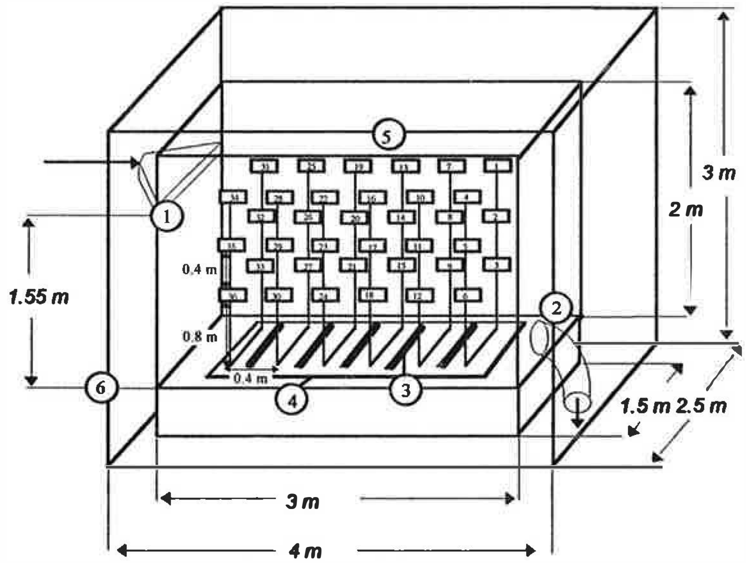


Figure 1: Schematic representation of the test room: 1. slotted air inlet; 2. air outlet; 3. heating element; 4. hot water reservoir; 5. grid of 36 thermocouples; 6. envelope chamber.

MECHANISTIC FORMULATION OF MODEL STRUCTURE

In previous research (De Moor and Berckmans, 1993) it has been demonstrated that the test room is an imperfectly mixed airspace with considerable temperature gradients. Within this imperfectly mixed airspace it is always possible to define a well mixed zone (WMZ) around a temperature sensor in which there exists a good mixing and an acceptably low temperature gradient (Berckmans et al., 1992). To describe the dynamic behaviour of temperature in the WMZ, standard heat transfer theory can be applied. In case of a constant ventilation rate this suggests a linear, first-order heat balance differential equation of the form:

$$\frac{dT_i(t).vol_I.\gamma_I.cl_I}{dt} = V_c.T_o(t).\gamma_o.cl_o - V_c.T_i(t).\gamma_I.cl_I + Q_c + k_I.S_I.(T_{buff}(t) - T_i(t)) \qquad (1)$$

where t (s) is the time; T_i (°C) is the temperature in the WMZ; T_o (°C) is the supply air temperature; T_{buff} (°C) is the buffer zone temperature; vol_I (m^3) is the volume of the WMZ; V_c (m^3/s) is the part of the ventilation rate entering the WMZ; Q_c (J/s) is the part of the internal heat production of the 5 heating elements entering the WMZ; γ_I (kg/m^3) and cl_I (J/kg.°C) are the density and the heat capacity of the air in the WMZ; γ_o (kg/m^3) and cl_o (J/kg.°C) are the density and the heat capacity of the supply air; and k_I (J/s.m^2.°C) and S_I (m^2) are the heat transfer coefficient and the surface area of heat exchange between the WMZ and the buffer zone. A schematic representation of the WMZ concept is given in figure 2.

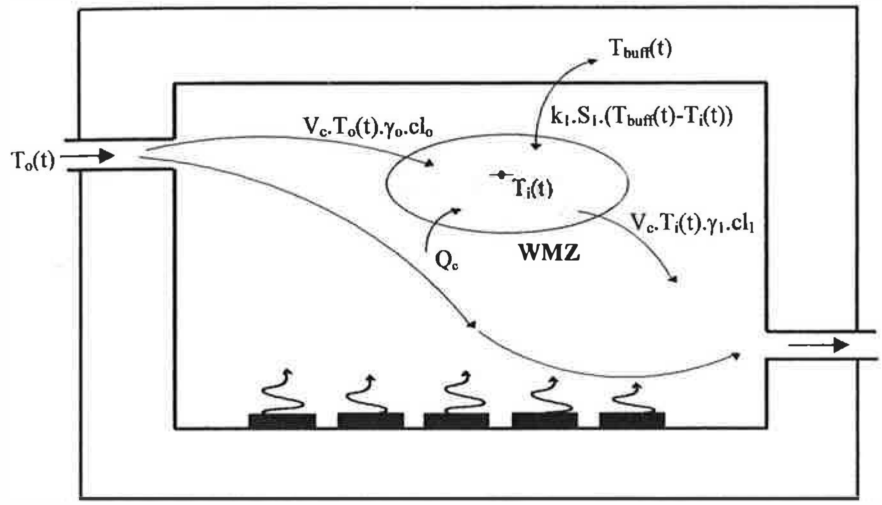


Figure 2: Schematic representation of the WMZ concept

If it is assumed that $\gamma_o \approx \gamma_I \approx \gamma$ and $cl_o \approx cl_I \approx cl$ and if only small temperature perturbations ($t_o(t)$, $t_{buff}(t)$ and $t_i(t)$) are considered about steady state, heat balance differential equation (1) can be written more concisely as follows (Janssens, 1999):

$$\frac{dt_i(t)}{dt} = \beta_I.t_o(t) + K_I.t_{buff}(t) - \alpha_I.t_i(t) \qquad (2)$$

where $\beta_I = \frac{V_c}{vol_I}$

$K_I = \frac{k_I.S_I}{vol_I.\gamma.cl}$

$\alpha_I = \frac{V_c}{vol_I} + \frac{k_I.S_I}{vol_I.\gamma.cl}$

The WMZ concept, represented in figure 2, can be applied to each of the 36 spatially distributed sensor positions in the test room. By doing so, we obtain a set of 36 first-order differential equations of the form (2) which describe the spatio-dynamic temperature behaviour.

DATA-BASED PARAMETER ESTIMATION

To estimate the model parameters β_l, K_l and α_l for each of the 36 sensor positions in the test chamber, 30 identification experiments were carried out. In each experiment the supply air temperature was switched from 11.5 to 17 °C as sharply as possible, whilst maintaining a constant ventilation rate. The experiments were carried out over a range of low (130, 140, 150 and 160 m^3/h) and high (200, 210, 220, 230, 240, 250, 260, 270, 280, 290 and 300 m^3/h) ventilation rates with two runs at each rate. The internal heat and moisture production were maintained constant at 300 J/s and 0.5 l_{water}/h. In all experiments the supply air temperature, the buffer zone temperature, the ventilation rate and the temperature response at each of the 36 sensor positions in the test room were measured every second over a time period of 2 hours. Further the air flow pattern was visualised and filmed. In the low ventilation rate experiments (130 - 160 m^3/h) the incoming air jet deflected downward to the floor resulting in an airflow pattern with stable anti-clockwise direction of rotation. In the high ventilation rate experiments (200 - 300 m^3/h) the air jet remained at inlet level after entry producing an airflow pattern with stable clockwise direction of rotation. Experiments at intermediate ventilation rates (170, 180 and 190 m^3/h) were not carried out, because preliminary research revealed that the airflow pattern and the spatio-dynamic temperature behaviour were unstable under these conditions.

As a typical example, figures 3(a) and 3(b) show the output of the estimated model for sensor positions 12 and 31 in comparison with the measured temperature response data for an identification experiment with ventilation rate of 290 m^3/h. The model clearly yields an excellent fit to the experimental data at both sensor positions. This was the case for each of the 36 sensor positions in each of the 30 experiments (Janssens, 1999). The average model accuracy was 0.15 °C.

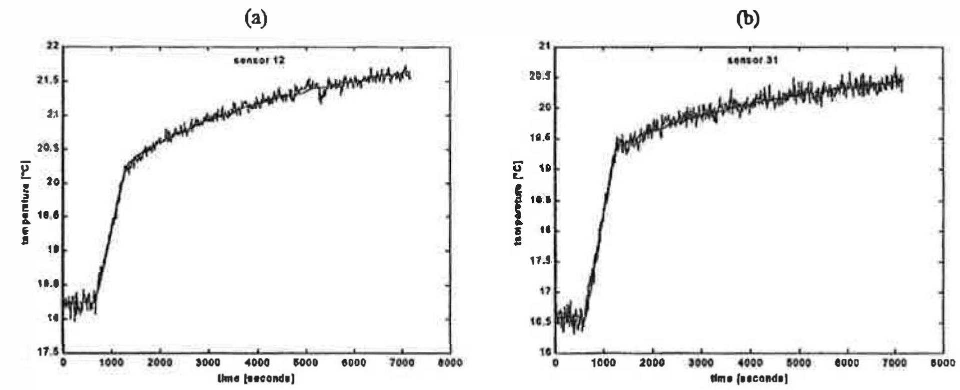


Figure 3: The output of the estimated model for sensor positions 12 (a) and 31 (b) in comparison with the measured temperature response data for an experiment with ventilation rate of 290 m^3/h.

SPATIAL DISTRIBUTION OF FRESH OUTSIDE AIR

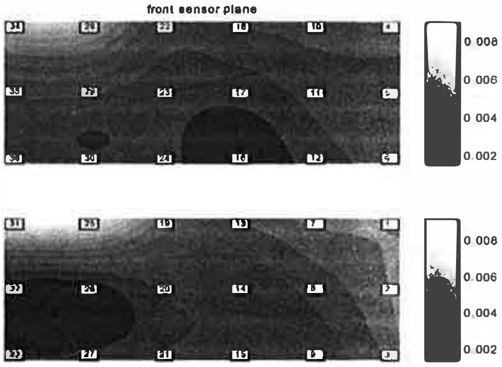
A very interesting aspect of the model lies in the physical meaning of model parameter $\beta_l = V_c/vol_l$ ($m^3/s.m^3$) which is the local outside air change rate. By modelling the temperature responses at each of the 36 sensor positions in the 30 identification experiments, we obtained very useful information about the spatial distribution of fresh outside air in the test room for a wide of ventilation rates.

As an example, figure 4(a) shows the spatial contours of the local outside air change rate in the front and rear sensor plane of the test chamber at a high ventilation rate of 290 m^3/h. The front sensor plane is the vertical xy-plane of the test room which consists of the temperature sensors 4, 5, 6, 10, 11, 12, 16, 17, 18, 22, 23, 24, 28, 29, 30, 34, 35 and 36 and which lies at a z-distance of 0.375 m from the front wall of the room. The rear sensor plane consists of the sensors 1, 2, 3, 7, 8, 9, 13, 14, 15, 19, 20, 21, 25, 26, 27, 31, 32 and 33 and lies at a z-distance of 0.375 m from the back wall. The contour plots in figure 4(a) illustrate rather well how, in this particular situation of high ventilation rate, the fresh supply air, which enters at the upper left, flows along the top of the chamber and then descends to the exit at the lower right.

Quite different behaviour occurs at low ventilation rates, as shown in figure 5(a). Here, with an airflow rate of 150 m^3/h, the high local outside air change rates concentrated towards the base of the chamber show how the fresh incoming air sinks to the bottom and then moves across the chamber to the outlet at the lower right, leaving more stagnant air above it.

The visualised air flow pattern at high and low ventilation rates, as shown in figures 4(b) and 5(b), is clearly mirrored in the spatial contour plots. It is clear, therefore, that the WMZ model underlying these plots is characterising the flow behaviour quite well.

(a)



(b)

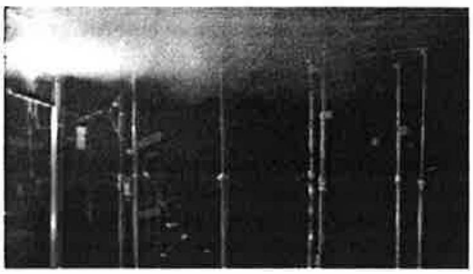


Figure 4: The spatial contours of the local outside air change rate in the front and rear sensor plane of the test chamber (a) and the visualised air flow pattern (b) at a high ventilation rate of 290 m^3/h.

(a)

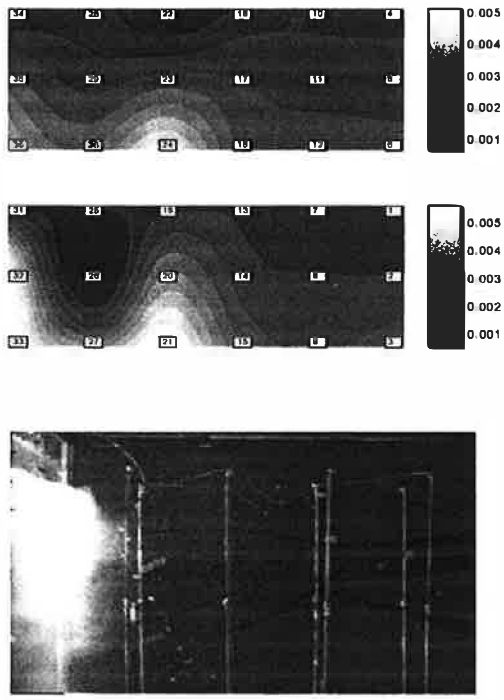


Figure 5: The spatial contours of the local outside air change rate in the front and rear sensor plane of the test chamber (a) and the visualised air flow pattern (b) at a low ventilation rate of 150 m^3/h.

CONCLUSIONS

The spatio-dynamic temperature response in an imperfectly mixed ventilated test room to variations of the supply air temperature was successfully modelled for a wide range of ventilation rates. The model provided an excellent fit to the spatio-temporal temperature response data with an average accuracy of 0.15 °C and characterised the spatially heterogeneous nature of the air flow pattern quite well.

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