EXPERIMENTAL AND NUMERICAL INVESTIGATION ON TEMPERATURE AND AIR VELOCITY DISTRIBUTION IN A ROOM EQUIPPED WITH SPLIT-SYSTEM AIR CONDITIONER

M. De Carli[1], F. Peron[2], R. Zecchin[1]

[1]Dipartimento di Fisica Tecnica, Università di Padova, Padova, Italy
[2]Istituto Universitario di Architettura di Venezia, Venezia, Italy

ABSTRACT

Split-system air conditioning is increasingly used both for residential and commercial applications, owing to its low cost and installation ease. The indoor split-system unit is commonly of the wall-mounted type and, due to its dimensions and position, very often it gives rise to appreciable air velocities and temperature gradients in the occupied zone of the room. This work reports and discusses some experimental data collected in a test room with wall-mounted indoor unit, under different operating conditions. A CFD numerical model has been developed and assessed on the basis of the experimental data; the model can be applied to investigate the influence of different parameters such as air flow rate and velocity, inlet air temperature, position of the unit, direction of air flow etc. From the results of parametric analyses some conclusions can be drawn, with reference to comfort conditions, useful to improve the design and construction of this type of air conditioning equipment and to develop more efficient installation criteria.

INTRODUCTION

The parameters influencing the thermal comfort in the indoor environment have been widely studied and evaluation criteria such as PMV ("predicted mean vote", ranging between -3 for "very cold" and +3 for "very hot") and PPD ("percentage people dissatisfied", with a reference upper value of 10% suggested by several standards) are nowadays of common use [1] [2] [3] [4]. Fanger's approach easily allows to predict users reactions (PMV and PPD), as a function of activity level (expressed by the metabolic rate M), clothing thermal resistance R_c, mean radiant temperature t_r of the surrounding surfaces, as well as dry-bulb temperature, relative velocity and humidity ratio of the air (t_a, v_a and W_a respectively):

$$PMV = f\,(M,\ t_a,\ t_r,\ R_c,\ W_a,\ v_a) \tag{1a}$$

$$PPD = 100 - 95\ exp\,(\,-\,0.03353\ PMV^4 - 0.2179\ PMV^2\,) \tag{1b}$$

For radiantly grey surfaces with emissivities close to one, the mean radiant temperature, in the absolute scale, can be expressed as:

$$T_r=[\Sigma_{k=1,n}\, F_{p\text{-}k}\,(T_s)_k^{\,4}]^{0.25} \tag{2}$$

where $F_{p\text{-}k}$ are the view factors between the human body and the surfaces and T_s their absolute temperatures; view factors can be determined by means of graphs [1] or, for computerised calculations, by means of algorithms based on spherical trigonometry [5]. In this study the distribution of radiant temperature has been derived from view factors calculated by this last method.

If the air velocity is appreciable, the parameter DR ("Draft Risk", expressed as percent of people dissatisfied due to draught) should be evaluated as well [3] [4]; the following expression can be used:

$$DR = (34 - t_a)(v - 0.05)\,0.62\,(0.37\, v\, T_u + 3.14) \tag{3}$$

where t_a is the local air temperature [°C], v is the local mean air velocity [m/s] and T_u is the local turbulence intensity [%], defined as ratio between standard deviation and mean value of the local air velocity, which can be calculated as:

$$T_u = (2\,k)^{0.5} / v \tag{4}$$

where k [m^2/s^2] is the local kinetic turbulence energy [6].

In the case study here considered, the air temperatures and velocities can vary quite considerably from point to point within the occupied zone. An experimental investigation on this phenomenon has been carried out in order to evaluate its significance and decide possible improvements [7]. The test room was 5.5 m by 5 m, 2.7 m height, and the indoor units of two different trademarks have been examined. The air temperatures were measured by means of precision differential thermocouples, having the air inlet of the unit as reference point; the air velocities were measured by means of a hot-wire anemometer. Details on measurement grids, instrumentation, accuracy, data logging and processing techniques are reported in [7]. The case of summer air conditioning (cooling) is developed here, as this appear to be the most critical one as for the comfort conditions. An example of the experimental results is shown in Fig. 1 as patterns of equal air temperature in a vertical plane through the air jet central line (A1), air velocity in the same plane (A2) and PPD on an horizontal plane at 1.1 m above the floor (A3). It can be seen that the zone in front of the internal unit is affected by a cold air stream.

AN APPROACH BASED ON COMPUTATIONAL FLUID DYNAMICS

An exhaustive analysis of the comfort conditions cannot be easily accomplished by experimental tests only, as it is very difficult to obtain the desired combination of the many different parameters involved. As in other similar fields, it is thus useful to resort to numerical methods, commonly called CFD (computational fluid dynamics) [8]. Conservation laws of mass (continuity equation), energy (energy equation) and momentum (Navier-Stokes equations) are the starting base of such technique. From these laws a set of partial differential equations can be obtained, which, numerically integrated with the relevant boundary conditions (e.g. values at inlets and outlets of an air conditioning device) and adopting a suitable turbulence model, allows to evaluate the temperature and velocity fields inside a room. In this study a commercial software, namely PHOENICS [6], based on finite volumes discretisation method, and the classic k-epsilon turbulent model have been fitted to test case and used to determine temperature and velocity fields in the considered room.

The above described experiments permitted to face a fundamental problem of numerical simulation: the tuning of the model on the specific physical system by comparing measurements and calculations. In fact the problem of evaluating the ability of a computer code to correctly predict thermal and fluid dynamics parameters is not a simple task. The criterion of appreciation depends strongly on the specific system considered and on the purpose of the calculation. Some comparisons are reported and discussed in the literature [9], [10], [11], [12] but they do not permit to draw final conclusions, valid for this work, as the pertinent systems are quite different from the one here considered. Again in Fig. 1 are shown the patterns of temperature (B1) and air velocities (B2) calculated by numerical simulation for the same operating conditions as in the experiments. The presented results refer to a k-epsilon model; the "low Reynolds number" model has been used as well, but no significant improvement has been found in the agreement between experimental and calculated values for the cases here considered. The general agreement and the local differences between the two series of patterns (measured and calculated) can be deemed and some indications could be drawn to refine numerical methods (this last aspect being nevertheless beyond the scope of this work). When assessing models for evaluating comfort conditions it seems however more reasonable to attach more importance to the effect of such differences on the comfort parameters, namely PPD: the patterns A3 and B3 in Fig 1 show an acceptable concordance. A further element of judgement is presented in Fig. 2 as "temperature ratio" defined as:

$$\Theta = (t_c - t_i) / (t_m - t_i) \qquad (5)$$

where t_c is the calculated air temperature, t_m is the measured air temperature and t_i is the inlet air temperature, which in both cases was 8°C; this parameter, which includes a reference magnitude (the inlet air temperature t_i), in this context, seems to be more significant than the simple difference $t_c - t_m$.

APPLICATIONS AND RESULTS

In order to perform the analyses planned for this study, a model describing a room subject to summer conditions has been set up. The room is 5 m length, 5.5 m width and 2.7 m height, with one external surface including a 3 m by 1.5 m window. The space domain has been discretised in 31 x 33 x 40 cells. In the assessment of the model, it has been seen that a more detailed grid did not modify the results. Details on thermal characteristics of the building and external climatic data (45° Lat. North, July) are given in [13]; the thermal behaviour of the building has been handled by NBSLD [14], a computer code based on the response-factors technique which takes into account the mutual radiation exchanges between the internal surfaces; the resulting peak cooling load, for the cases here illustrated, has been calculated as 1500 W. The air flow rate has been set up at 250 m^3/h. The comfort analysis refers to sedentary activity (1.2 met) and light clothing (0.5 clo).

Once the temperature and air distribution inside the room are known, the assessment of the thermal comfort becomes possible by suitably implementing the comfort algorithms in a computer program: different design solutions (in terms of position of the internal unit, air flow rate and temperature, air jet direction etc.) can be easily analysed and some cases are reported in this work. Further analysis are reported in [13].

In Fig. 3 (A1, A2, A3 and A4) the temperature and air velocity distributions on the vertical central plane, the PPD distribution on a horizontal plane (at 1.1 m height) and the DR distribution on the same plane are shown respectively; in this case the direction of the air jet has been modified, in comparison with the unsatisfying situation dealt with in the experiments (Fig. 1), by tilting it 5° upward from the horizontal. A further case is shown again in Fig. 3

(B1, B2, B3 and B4) referring to the same operating conditions and air flow rate, with the same air jet inclination (5° upward), but with the jet itself horizontally divided in two streams, 45° apart from the axis of the unit. It can be seen that the case of Fig. 3A shows an appreciable improvement in comparison of the case of Fig. 1, but still it exhibits non negligible areas of discomforts, specially for DR; the case of Fig. 3B shows a much better result, more uniform thermal comfort conditions and negligible DR. In this way it was possible to draw some indications for a more efficient design of the indoor unit.

DISCUSSION

The comparison between experimental data and numerical calculations indicates that CFD models, although admitting improvements, are suitable to describe the behaviour of air conditioning equipment such as split indoor units. The CFD model can be used as a design tool for improving the characteristics of the considered equipment, from the comfort point of view, or for air conditioning design purposes.

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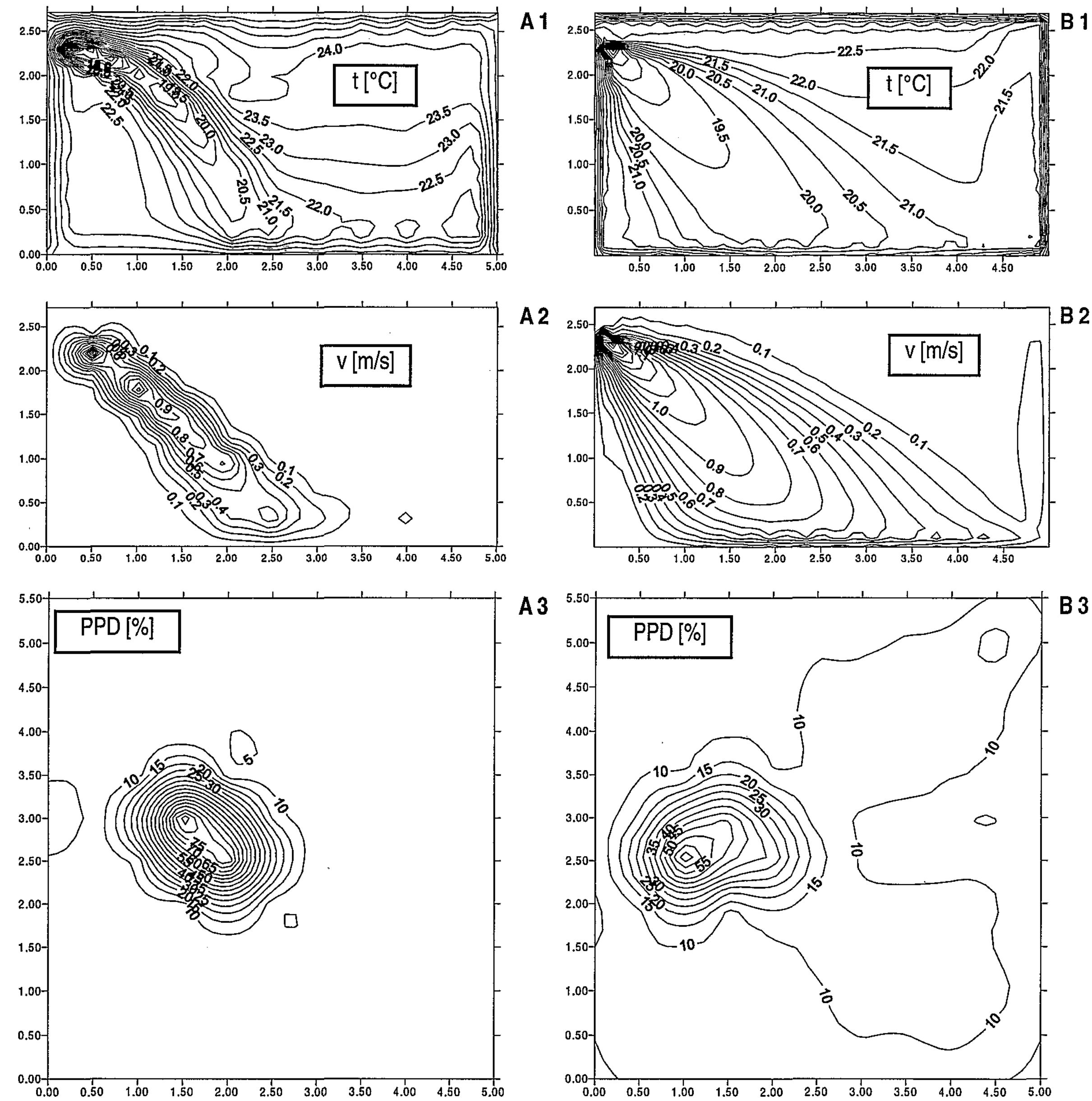
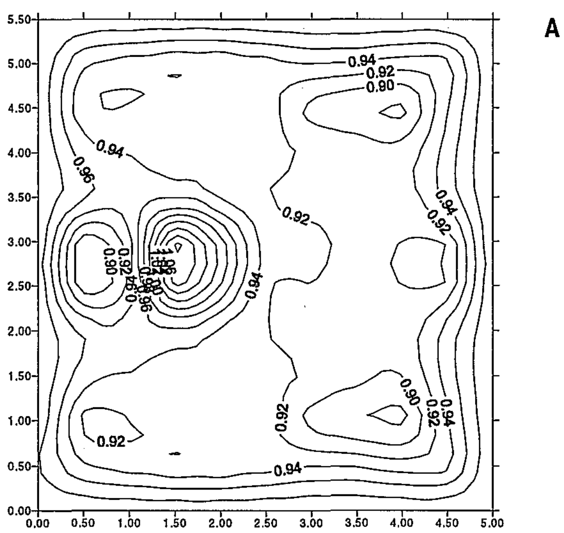


Fig. 1 - Air temperature, air velocities and PPD patterns: measured (A), calculated (B).



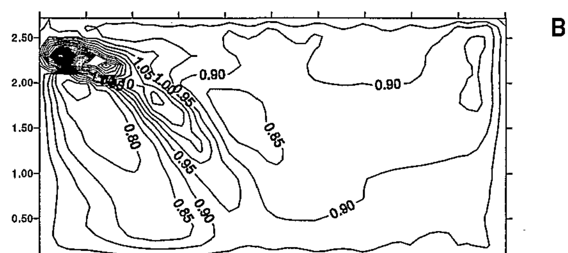


Fig. 2 - Patterns of Θ on a horizontal plane at 1.1 m height (A) and in a vertical plane on the air jet axis (B).

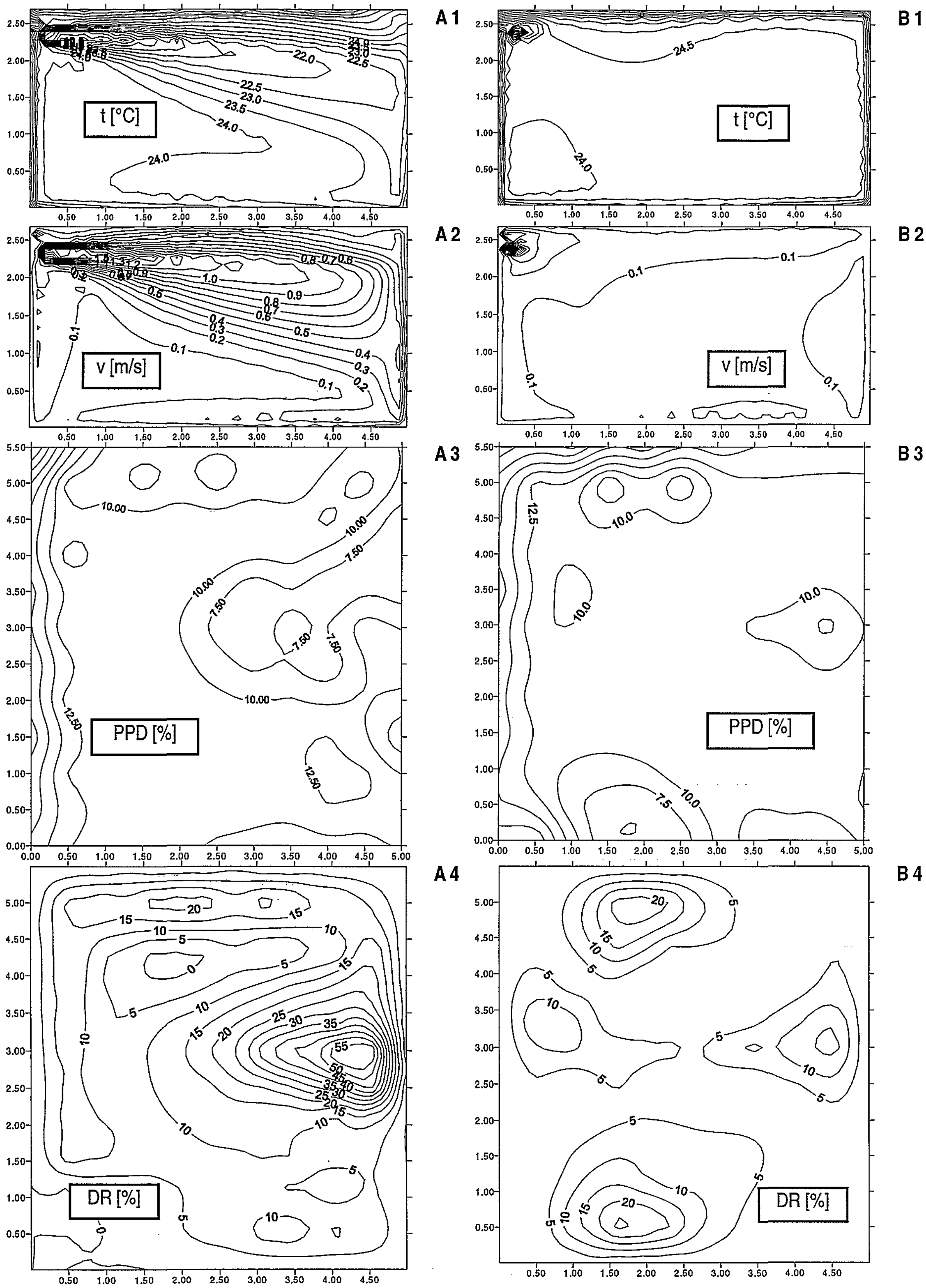


Fig. 3 - Air temperature, air velocity, PPD and DR patterns for two different indoor unit configurations: horizontal air jet (A) and 5° upward distributed air jet (B).