Optimisation of the indoor thermal comfort in private houses using advanced turbulent models

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ABSTRACT

The simulation has been performed using the CFD package SWIFT. In the first phase, mean flow and thermal comfort parameters of the integral living room of the modern family house have been calculated. In the second phase the effect of turbulence was the subject of interest. Two turbulence models, the k–ε and the HTM (Hybrid Turbulence Model), were applied. HTM already showed good results in other application areas, e.g. aerodynamics of cars. The present study demonstrates the new approach in modelling and calculating air conditioning of the rooms.

INDEX TERMS

CFD; PMV; Turbulence; Thermal comfort; Heat transfer

INTRODUCTION

There is a growing interest among architects, building service engineers and environmental engineers in the analysis of building airflow and related phenomena such as thermal comfort, air quality and atmospheric wind effects on buildings. This interest has been pronounced not only for industrial and public buildings, but also for modern private houses. The present paper shows the CFD analysis of the heating concepts of a family house, which is in the virtual design phase, according to the latest standards of thermal comfort and ventilation effectiveness.

The factors that determine human comfort are numerous and complex, but by controlling certain key parameters, the occupied space can be made quite comfortable. These parameters include thermal comfort (space temperature, relative humidity, thermal radiation and local air velocity), indoor air quality (air change rate and fresh air delivered to breathing zone), and acoustic quality (noise criteria or room criteria). The activity and clothing level of the occupants are also parameters and assumed to be average.

From previous studies, where the basic difference between several heating concepts with natural (floor heating) and forced convection (floor convectors) has been investigated, the most promising variant with forced convection was further investigated more in detail, with three added thermal manikins in different positions.

Most numerical simulations based on Reynolds Averaged Navier–Stokes equations (RANS) start with the standard k–ε models predictions as the model is well tested and all its advantages and disadvantages are understood well. Limitations of the linear stress–strain relation and insensitivity to stress anisotropy become apparent in this type of flow where impingement and separation regions together with strong swirling and tumble motions are present. However, the k–ε models results can serve as a reference point for other model performances. Calculations were also performed using the HTM turbulence model, which is a combination of k–ε and the full second-order turbulence closure model, RSM. Turbulence

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kinetic energy is obtained after solving the full set of Reynolds stress transport equations. The dissipation rate equation is also solved in the form commonly used in the framework of the Reynolds-stress closures. This should better represent the dynamics of the interactions between turbulence and mean flow under the conditions where the flow is adjusting to rapidly evolving strain rates. Also, an ability of this model to calculate the turbulence production rate exactly, results in a better basis for modelling flows in air conditioned rooms.

METHODS

The Computational Fluid Dynamics (CFD) code AVL SWIFT is used for the present simulations. The code employs the finite volume discretisation method, which rests on the integral form of the general conservation law applied to the polyhedral control volumes (cells). The integral form can be written as:

$$\underbrace{\frac{\mathrm{d}}{\mathrm{d}t}\int_V \rho\phi\,\mathrm{d}V}_{\text{Rate of Change:}R} + \underbrace{\int_A \rho\phi U_k \mathrm{d}A_k}_{\text{Convection:}C} = \underbrace{\int_A \Gamma_\phi^{kk}\frac{\partial\phi}{\partial x_k}\mathrm{d}A_k}_{\text{Diffusion:}D} + \underbrace{\int_V S_\phi^V \mathrm{d}V + \int_A S_{\phi k}^A \mathrm{d}A_k}_{\text{Sources:}S} \tag{1}$$

where a general variable $\phi(x_k, t)$ can represent either scalars or vector and tensor field components. Here, the Cartesian co-ordinate system (x, y, z) with the unit vectors $(\vec{i}, \vec{j}, \vec{k})$ is used and tensor notation is employed. In the above equation, ρ is the fluid density, t the time, U_k are components of the fluid velocity vector, Γ_ϕ^{kk} is the diffusion coefficient for the variable ϕ (in this case repeated indices do not imply summation), s_ϕ^V and $s_{\phi k}^A$ are the volumetric and surface source terms, respectively.

The equations of momentum conservation in the Cartesian tensor notation are:

$$\rho\frac{\mathrm{D}U_i}{\mathrm{D}t} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} - \frac{2}{3}\frac{\partial U_k}{\partial x_k}\delta_{ij}\right) - \rho\overline{u_i u_j}\right] \tag{2}$$

The operator $\mathrm{D}/\mathrm{D}t = \partial/\partial t + U_j\partial/\partial x_j$ stands for the material derivative, U_i is the mean-velocity vector, P is the static pressure, ρ is the fluid density and $\mu = \rho\nu$ is its dynamic viscosity.

The $k-\varepsilon$ model

In the $k-\varepsilon$ model of turbulence, the Reynolds stresses are obtained from the Boussinesq's eddy viscosity formulation:

$$-\rho\overline{u_i u_j} = -\frac{2}{3}\rho k\delta_{ij} + 2\mu_t S_{ij}, \qquad S_{ij} = \frac{1}{2}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) \tag{3}$$

where k is the turbulent kinetic energy, S_{ij} is the mean rate of strain tensor and μ_t is the turbulent viscosity, which is evaluated from the expression:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{4}$$

With reference to Eq (1), the corresponding diffusion coefficients and source terms that describe the $k-\varepsilon$ model (ε being the dissipation rate of k by viscous action) are as follows:

$$\phi = k \Rightarrow \Gamma_\phi = \mu + \frac{\mu_t}{\sigma_k}, \quad s_\phi^V = \rho(P_k - \varepsilon), \quad s_{\phi k}^A = 0, \quad P_k = -\overline{u_i u_j}\frac{\partial U_i}{\partial x_j} \tag{5}$$

$$\phi = \varepsilon \Rightarrow \Gamma_\phi = \mu + \frac{\mu_t}{\sigma_k}, \quad s_\phi^V = \rho(C_{\varepsilon 1} P_k - C_{\varepsilon 2}\varepsilon)\frac{\varepsilon}{k}, \quad s_{\phi k}^A = 0 \tag{6}$$

where P_k is the production of the turbulent kinetic energy.

Among others, there are two standard ways of improvements of the standard k–ε model. The first one is to work out a better formulation for the dissipation rate equation and the second is to modify the stress–strain relation.

The hybrid turbulence model (HTM)

Although it has been shown in the past that the Reynolds-stress model (RSM) models can be used for real industrial applications, it is also inevitable that CFD users seek less complicated and more robust solutions. Seeking for such solutions, Basara and Jakirlić (2001) and Basara *et al.* (2001) recently proposed and validated a new and simpler scheme for turbulence model employment.

The k–ε models use the turbulence kinetic energy and its dissipation rate to define characteristic turbulence scales. The C_μ coefficient is derived from the measured ratio $\overline{u_i u_j}/k$ for the wall boundary layers and then used as a constant value. With this assumption, the turbulence viscosity is determined and then used in momentum equations. A weak point of such a formulation, beside its isotropic form, lies in a derivation of the constant C_μ, which in reality does not vary just from one to another type of the flow but also across the same flow. The commonly used value is 0.09. An approach advanced by Basara and Jakirlic (2001) suggests a derivation of C_μ by equalising the production of turbulence kinetic energy predicted by the Reynolds stress model and with the production obtained by the k–ε model, thus

$$C\mu = \left(-\overline{u_i u_j}\frac{\partial U_i}{\partial x_j}\right) \Bigg/ \left(\frac{k^2}{\varepsilon}S^2\right), \qquad S = \sqrt{2S_{ij}S_{ij}} \tag{7}$$

Therefore, in the proposed model, the stress and mean strain tensors are coupled via Boussinesq's formula, as in the standard k–ε model. However, the turbulence kinetic energy is now obtained after solving the full set of Reynolds stress transport equations. The dissipation rate equation is also solved in the form commonly used in the framework of the Reynolds-stress closures. Finally, the structural parameter C_μ is calculated as a function given above rather then kept constant. This 'constant free' eddy-viscosity model greatly improves results compared to its standard k–ε counterpart. Several, very well-known weaknesses of the k–ε modelling practice, pertinent especially to the rotating and swirling flows, separated flows, as well as flows with strong dilatational effects, are removed in such a way. On the other hand, this approach improves significantly the convergence rate in comparison to the

Reynolds-stress model groups. Such a formulation of the eddy viscosity model offers very robust computational procedure and accurate solutions. The weak point of this method is the larger CPU time required.

NUMERICAL MODEL

The computational domain (integrated kitchen and living room, Figure 1) is a hybrid mesh and contains 600.000 cells where 80% are hexahedrons and the rest tetras, prisms and pyramids. Local grid refinement is applied (Figure 2) to improve a grid resolution around the thermal manikins, convectors and windows. The grid is carefully checked for numerical errors and it represents the minimum size acceptable for turbulent model testing. The first next to wall cells ensures y^+ values on the manikins to be less then 100. For all cases, the grid is the same.

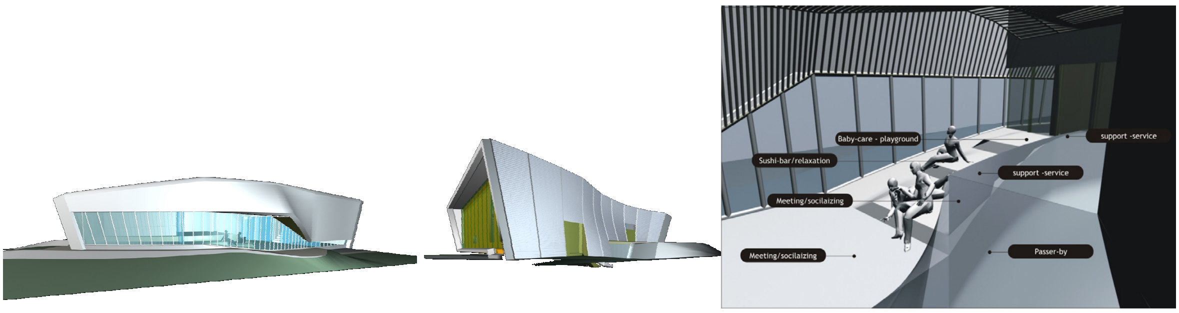

Figure 1 Residential family house and integral living room.

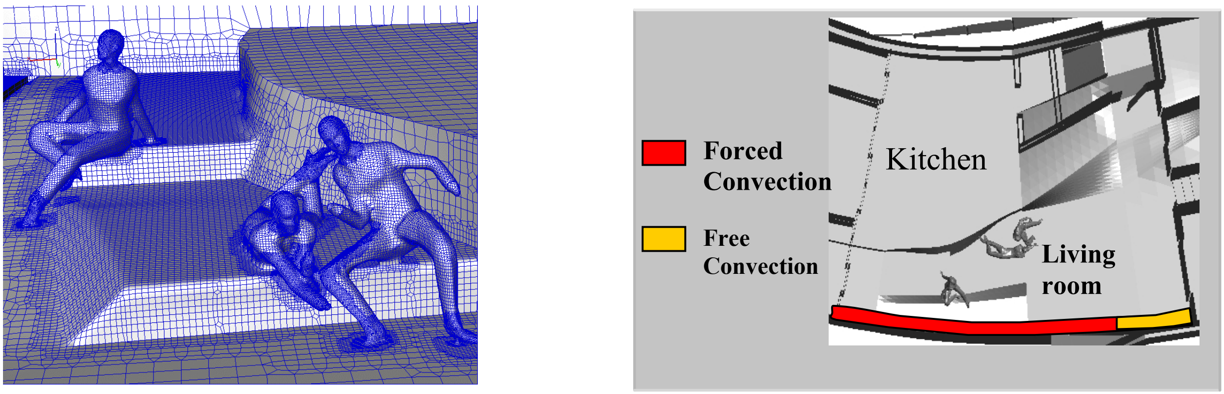

Figure 2 Computational grid - thermal manikins detail and used boundary conditions.

A previous study (Tibaut *et al.*, 2002) shows, that between three types of heating (floor heating, heating with floor convectors and combination between these two ones) the most reliable one with regard to thermal comfort factors is the one with floor convectors heating. This variant was selected for further study and optimisation. To reduce regions with negative PMV and high (>10%) PPD values, the load point of the convector fans has been increased. To reach lower costs, one part of the convector's line has been replaced with free convection convectors. This is in the NE part of the living room, where an inner garden has to be designed.

At the inlet, a mass flow of 200m^3/h and 40ºC have been defined. Thermal manikins transmit 1clo, windows have HTC 1W/m^{2}K. Two-layer wall function and a central differencing convective scheme have been used. As in the previous study, 600 sec of physical time has been simulated.

RESULTS AND DISCUSION

The comparison between k–ε and HTM models in the thermal manikins area, shows different velocity levels (Figure 3) and also different PMV and DR values (Figure 4). It is to be expected that k–ε gives higher velocities and HTM shows more intensive separation and recirculation flow phenomena. This can be explained with a larger kinetic energy dissipation term in the turbulence production equation and lower overall velocities by the HTM model. This means that k–ε will probably over-predict the velocity on the thermal manikins and consequently the thermal comfort factors, where velocity is a more significant influencing factor, will be higher as in HTM. On the other hand, by HTM in recirculation zones (for example behind the thermal manikins) velocity and heat transfer coefficients as well as some comfort coefficients will be higher. From Figures 3 and 4, it is seen that PMV and DR on the front facing side are higher with the k–ε model and on the back facing side higher with the HTM model.

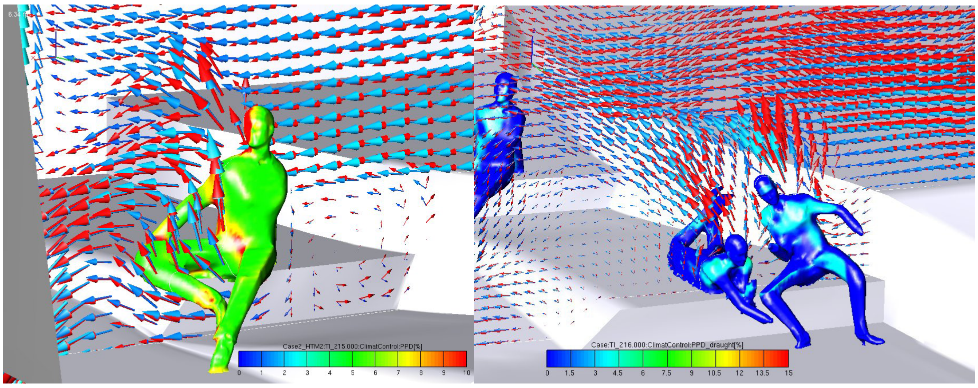

Figure 3 Velocity vectors by k–ε and HTM turbulence model (blue—k–ε, red—HTM).

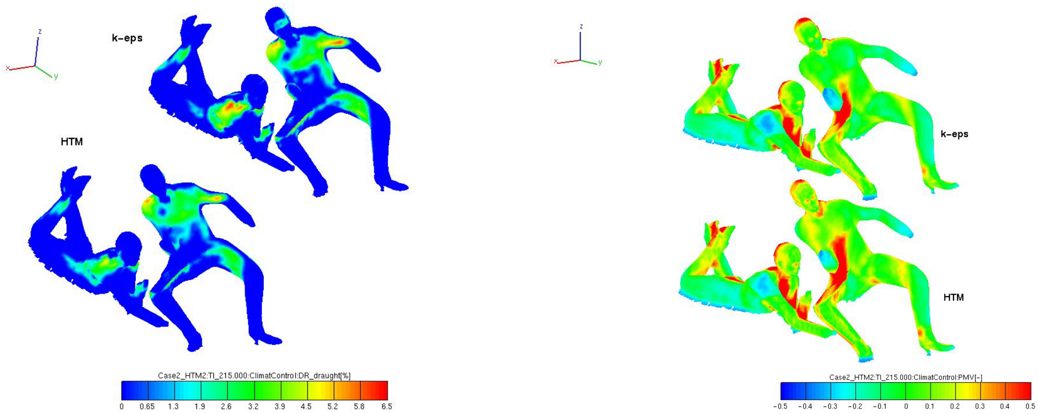

Figure 4 DR and PMV on the thermal manikins for k–ε and HTM turbulence model.

HTC surface values on the east and top part of the computational domain (Figure 5a) follow the above properties. Because of higher near wall velocities, local HTC is higher by k–ε (on the big east window HTC is bigger for 5.3% and heat flux for 2.7% in comparison with HTM). Higher local velocities influence the temperature distribution (Figure 5b). Different temperature profiles result with HTM, higher mean PMV and PPD values (PMV +3.7% and

PPD +4%) and lower DR (–32%) on thermal manikins. The redirection of the mean flow and velocity influence surface PMV values (Figure 6) on thermal manikins. In the observed room, the main vortex with clockwise rotation is created (Figure 6). Because of coupled work with architectural design, the recirculation and dead-water zones are reduced to the minimum.

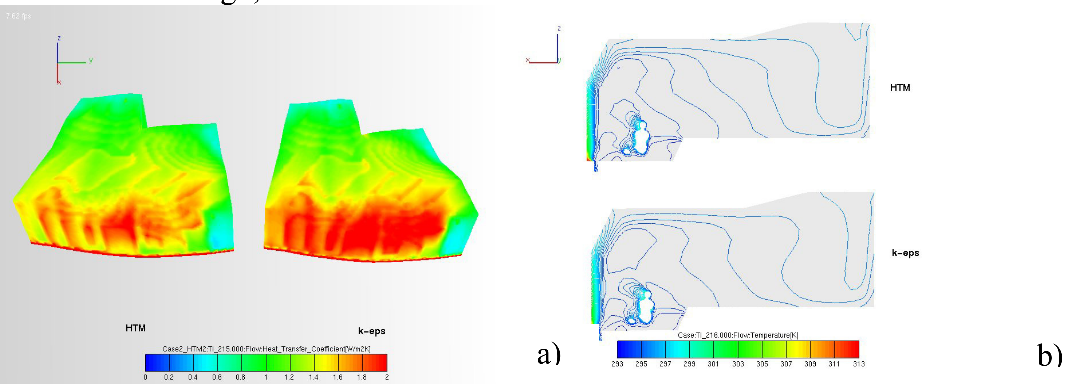

Figure 5 Heat transfer coefficient (HTC) on east façade (a) and temperature distribution in the plane cut trough the kitchen and single thermal manikin (b).

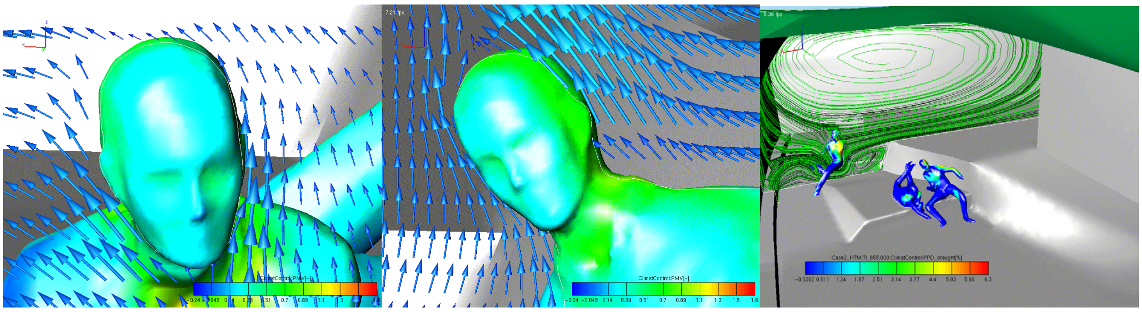

Figure 6 PMV, velocity vectors and streamlines on and around thermal manikins by HTM.

CONCLUSION AND IMPLICATIONS

The results of numerical computations of the air flow in a integral living room of a modern family house were fairly conclusive: there is a substantial difference in prediction of both turbulence energy and velocity field when the standard $k–\varepsilon$ and the HTM model are used. With that all mean thermal comfort parameters are influenced. The use of alternative turbulence models in complex flows is suggested by the use of Reynolds-stress equations (HTM) to calculate the turbulence energy. Some limitations of the proposed hybrid turbulent model are the higher CPU time, which was in our case +40% in comparison with the standard $k–\varepsilon$ model.

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