EXPERIMENTAL AND NUMERICAL STUDIES ON INDOOR AIR QUALITY IN A REAL ENVIRONMENT

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Abstract.

This paper deals with the relevance of Computational Fluids Dynamics (CFD) results confronted to measurements carried out in real environment. Experimental tests have been undertaken in a room and in a kitchen of an experimental house. Although the wall surface temperatures and the air intake temperature have not been imposed, the air change rates have been controlled during the measurements. Moreover, since measurements have been carried out in a real environment, air leakage has occurred at the walls.

Thereafter, measurements have been used to define boundary conditions of CFD simulations.
As a result, it can be stated that as long as air leakage is slight during the experimental tests, satisfactory agreement is observed between CFD results and in situ experimental results.
However, in the case where experimental tests are carried out in a room with significant air leakage, the quality of the numerical results has been decreased since this phenomenon is not considered in the CFD model. For this case, a simplified hypothesis of modelling has been proposed and validated to deal with air leakage effects and thus to improve the accuracy of CFD results.

Keywords : Indoor Air Quality (IAQ), measurements, CFD, air leakage modelling.

Nomenclature

C_e, C_P air pollutants concentration respectively at the exhaust opening and at a point P (kg/m^3)

Dc, D_e laminar molecular diffusivity of respectively air contaminant and the mixture component e in air (m^2/s)

g gravity acceleration (m/s^2)

H height of the test room (m)

m_e mass fraction of the element e of the mixture: air + air pollutant (-)

M_{air} molar mass of air (= 28.97 g/mol at T=293 K and P =101325 Pa)

M_C, M_e molar mass respectively of an air contaminant and of the element e of the mixture studied (kg/mole)

P pressure of the fluid (Pa)

q mass flow rate of air pollutant (kg/s)

Q air flow rate (m^3/s)

$Q_{leakage}$, Q_{red} air leakage rate and reduced air flow rate (m^3/s)

R universal constant of gas

S_I area of the natural air inlet (m^2)

T temperature (K)

T_{inlet}, T_{room} Air temperature at inlet and averaged temperature of the room (K)

$-\overline{u_i^{'}m_e^{'}}$ turbulent mass fluxes (m^2/s^2)

V_a Air molar volume (=20.1 cm^3/mol at T=293 K and P =101325 Pa)

V_C contaminant molar volume (cm^3/mol)

x_i coordinates: x, y, z (m)

Greek symbols

β_T coefficient of volumetric expansion due to temperature change (K^{-1})

β_C coefficient of volumetric expansion due to concentration change (m^3/kg)

μ laminar dynamic viscosity (kg/m.s)

ρ_{air} air density (kg/m^3)

Dimensionless number

Ar_T thermal Archimedes number (-)

Ar_c solutal Archimedes number (-)

N_B ratio of thermal convection force on solutal convection force

1. INTRODUCTION

Since we spend 90 % of our time in enclosed spaces, indoor air quality becomes an important parameter, especially for our health. It is thus important to handle effective ventilation systems in order to obtain and/or to maintain a good quality of indoor air. Ventilation efficiency can be experimentally and numerically analysed. The numerical analysis is useful to deals with details of the internal flow and contaminant distribution. In so doing, Computational Fluids Dynamics (CFD) constitutes one of the best numerical tools. It has been validated by confronting its results to measurements [Nielsen (1974), Allard (1990)]. However, excepted the researches of Zeng et al (2000) which compared CFD results with measurements realized in a portable classroom, few studies have confronted CFD results to in situ measurements.

This paper deals with the relevance of CFD results confronted to in situ measurements.
Indeed, measurements have been carried out in a room and in a kitchen of an experimental house. The wall surface temperatures and the air intake temperature have not been imposed during the experimental test. Nevertheless, the air change rates have been controlled during the measurements.
Measurements have been used to define boundary conditions of CFD simulations. Thereafter, the quality of the CFD results is analysed.

2. EXPERIMENTAL SET-UP AND PHENOMENOLOGY OF THE FLOWS

2.1 Case of the room

Experimental tests are carried out in a room of the experimental house MARIA of CSTB, Riberon et al (2002). The wall surfaces temperatures and the air intake temperature are not imposed. Nevertheless, the air flow rate is controlled at 18 m^3/h via a mechanical exhaust system.

This experimental test thus constitutes a transition between measurements in cell tests where all boundaries conditions are controlled and in situ measurements.

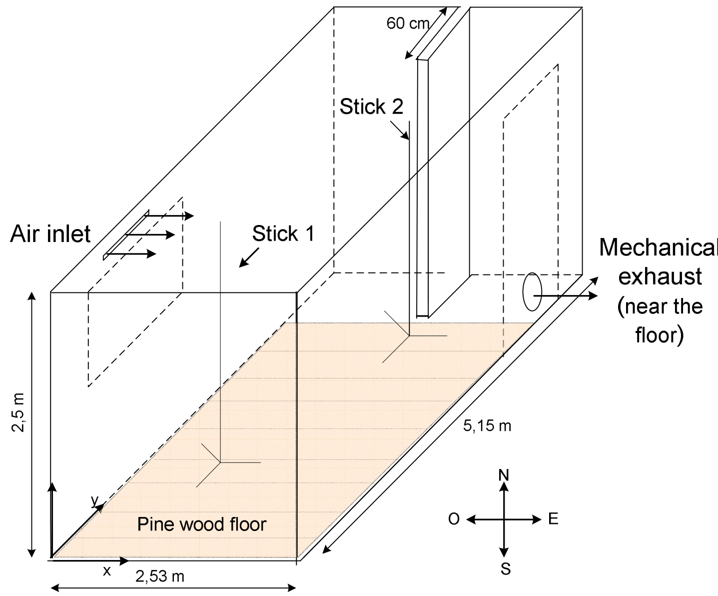


Figure 1- Layout of the ventilated room

The air exhaust opening is located near the floor and the natural air inlet is above the top of the room window (see figure 1).

The room has also been equipped with a pine wood floor which emits several VOCs considered as air pollutants here. The major identified VOC is α-pinene. Since its concentration in the inlet air is very slight, it has been selected as the VOC tracer of the VOCs emitted by the pin wood floor. More details of this experimental set-up is available in Akoua et al (2003).

The mass flow rate of VOC tracer emitted by the pine wood floor is evaluated as follows:

$$q = Q.C_e \tag{1}$$

$q = 0.2 \times 10^{-9} kg/s$.

The flow physical characteristics can be revealed with the following dimensionless numbers:

- Reynolds number, Re:

$$\mathrm{Re} = \frac{\rho (Q/S_i) H}{\mu} \tag{2}$$

- Ratio N_B:

$$N_B = \frac{\beta_T (T_{room} - T_{inlet})}{\beta_C ((q/Q) - C_{inlet})} \tag{3}$$

- Thermal Archimedes number, Ar_T:

$$Ar_T = \frac{g \beta_T (T_{room} - T_{inlet}) H}{(Q/S_i)^2} \tag{4}$$

- Solutal Archimedes number Ar_C:

$$Ar_C = \frac{g.\beta_C\left(\left(q/Q\right)-C_{inlet}\right).H}{\left(Q/S_i\right)^2} \tag{5}$$

The solutal Archimedes number for mass transfer is analogous to the thermal Archimedes number for heat transfer. It characterises the motion of fluid due to density differences generated by variations of the air pollutant concentration. Expression of the solutal Archimedes number comprises the coefficient of volumetric expansion due to concentration change:

$$\beta_C = -\left(\frac{1}{\rho}\right).\left(\frac{\partial\rho}{\partial C}\right)_{P,T} \tag{6}$$

Based on ideal-gas theory, β_C can be calculated as follows, Tiffonet (2000):

$$\beta_C = -\frac{V_M\left(M_C - M_{air}\right)}{V_M C_C\left(M_C - M_{air}\right) + M_C M_{air}} \tag{7}$$

For α-pinene:

β_C = - 0.65 m^3/kg and $\rho_{air}.\beta_C = -0.78$

Physical characteristics of the experimental test are summarized in table 1.

Table 1. Physical characteristics of the internal flow of the room.

Q	Re	Ar_T	Ar_C	$\lvert N_B \rvert$
18 m^3/h	2 x 10^5	0.3	5 x 10^{-9}	6 x 10^7

Implications:

- Value of thermal Archimedes number is small ($Ar_T < 1$). This thus suggests that effects of thermal convection are slight compared to convective effects of the forced flow due to the mechanical ventilation. Furthermore, the flow is almost isothermal. Nevertheless, this value of the thermal Archimedes number does not allow us to consider that the internal flow is completely isothermal. For this, it would have been necessary to have $Ar_T << 1$.

- Value of solutal Archimedes number is very small ($Ar_c << 1$). Therefore, effects of solutal convection can be neglected compared to the convective effects of the forced flow. Moreover, N_B value indicates that effects of solutal convection are slight compared to effects of thermal convection. As a result, the VOC tracer can be considered as a passive gas whose distribution is mainly governed by the thermoaeraulic field.

2.2 Case of the kitchen

Measurements are also carried out in the kitchen of the experimental house MARIA of CSTB. Air flow rate of the kitchen is controlled at 120 m^3/h via a hood. This ventilation rate corresponds to recommendations of French standards for high speed of mechanical exhaust in dwelling kitchens, journal official (1982).

Schematic diagram of the kitchen investigated is shown in figure 2.

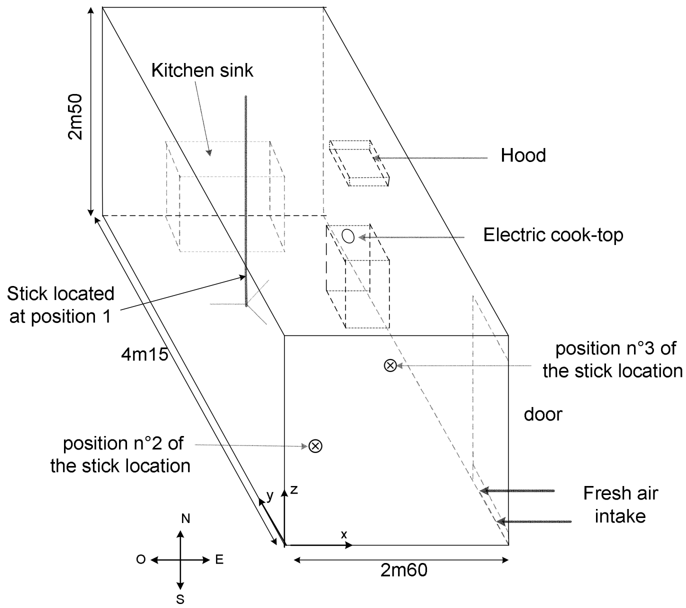


Figure 2 - Schematic diagram of the kitchen investigated

This experimental set-up is based on a French normative test used to assess efficiency of kitchen hoods, NF E 51-704.

However, this experimental test does not exactly correspond to the normative test. Indeed, unlike the normative test relevant to the study of the dynamics of pollutant extraction, the stationary field of pollutant concentration is studied here. In addition, the experiments were carried out in a real kitchen, not in an experimental cell as in the normative test. Therefore, walls surface temperatures and the air intake temperature could are not imposed.

The kitchen studied is also equipped with an electric cook-top. Its temperature is controlled at 110℃ as in the normative test NF E 51-704. A pan is put on it.

The vapours of cooking are studied using a tracer gas (SF_6) which passes through a pan.

More details on this experimental set-up are available in Akoua et al (2004).

Moreover, physical characteristics of the internal flow of the kitchen are dealt with dimensionless number. Table 2 indicates values of these dimensionless numbers.

Table 2. Physical characteristics of the kitchen internal flow.

Q	Re	Ar_T	Ar_C	N_B
120 m^3/h	$3.4 \times 10^{+5}$	2.6	1.5×10^{-3}	$3.9 \times 10^{+3}$

Implications :

- Value of Reynolds number indicates turbulent flow in the kitchen.
- Value of thermal Archimedes number indicates that the flow is non-isothermal. Effects of natural thermal convection and the convective effects due to the mechanical ventilation have the same order of magnitude.
- Solutal Archimedes number value is very small than 1. As a result, natural solutal convection due to non-uniformity of SF_6 distributions has globally slight effects on the forced convection due to the mechanical ventilation. Moreover, $N_B >> 1$. Effects of the natural solutal convection are also slight compared to effects of the natural thermal convection.

3. CFD SIMULATIONS

3.1 The numerical model

A CFD code based on "Finite Volume Method" (Fluent®) is used to numerically explore the internal flows.

In the case of the room studied, the pine wood floor is considered as a solid body source of VOCs. It is assumed that no adsorption phenomena of VOCs are occurred.
Moreover, α-pinene is selected as tracer of VOCs emission. It is considered as a pollutant gas which has no chemical interactions with indoor air.

It is previously shown that the VOC can also be considered as a passive gas (cf.: section 2.1). However, there may be significant concentration differences close to the pollutant source. Therefore, in the numerical model, the internal fluid is considered as a species mixture: air plus α-pinene.

In the case of the kitchen, the internal fluid is also considered as species mixture: air plus SF_6.

Whatever the case studied, the species mixture are supposed to be completely mixed. The flow is also supposed to be weakly turbulent and non-isothermal.

The mathematical equations to be solved are Navier-Stokes equations: conservation equations of continuity, of momentum and of energy.
In addition to these equations, an equation of species conservation is solved to deal with the distribution of the air pollutants (α-pinene or SF_6). It takes the following formulation:

$$\rho \overline{u}_j \frac{\partial m_e}{\partial x_j} = \frac{\partial}{\partial x_i}\left(\rho D_e \frac{\partial m_e}{\partial x_i} - \rho \overline{u_i' m_e'} \right) \tag{8}$$

The mixture density is defined as follows:

$$\rho(T, m_e) = \frac{P}{RT\sum_e (m_e/M_e)} \tag{9}$$

In order to solve the conservation equation of pollutant concentration, the mass diffusivity of the pollutant (α-pinene or SF_6) in air is needed. Tucker et *al* (1990) recommend using FSG method to evaluate the molecular coefficient of diffusion of a contaminant in air:

$$D_C = \frac{10^{-3} T^{1.75} \sqrt{(M_a + M_C)/(M_a M_C)}}{P\left(V_a^{1/3} + V_C^{1/3}\right)^2} \tag{10}$$

Where *P* is the pressure (in *atm.*)

At T = 25 °C (298 K), the coefficients of diffusion of α-pinene, $(C_5H_8)_2$, and of SF_6 in air are respectively:

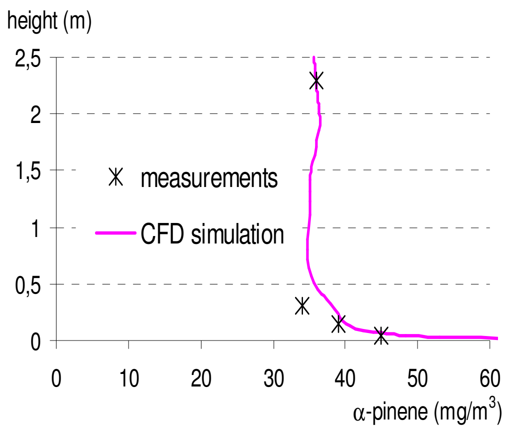
$D_{\alpha-pinene} = 6 \times 10^{-6}$ m²/s and $D_{SF_6} = 8.96 \times 10^{-6}$ m²/s.

Moreover, effects of turbulence on the flow are dealt with "realizable" k-ε model, Shih et al (1995). This is an improvement of the standard k–ε model, Lauder et al (1972).

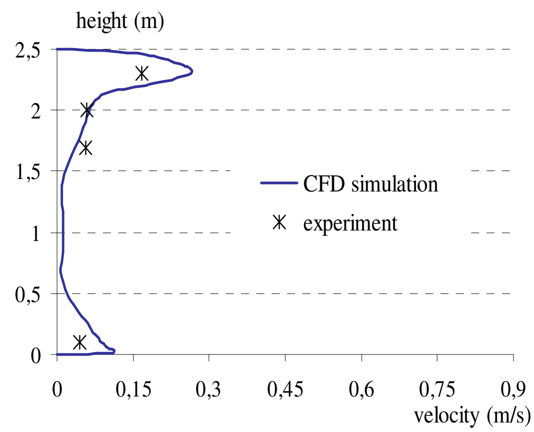
It proposes a new formulation of the eddy viscosity and a new model of the equation of dissipation rate (ε). This model is supposed to allow a better representation of the spectral energy transfer. It has been extensively used for a wide range of flows where its superiority has been established for flows including boundary layers under strong adverse pressure gradients, separation and recirculation, Shih et al (1995).

3.2 CFD results compared to measurement carried out in the room

Numerical profiles of α-pinene concentration and of air velocity are compared to measurements on figure 3. Measurements probes of α-pinene concentrations are set on the vertical stick 2 located in the volume of the room. Thermo anemometrical probes are set on the vertical stick 1 located front of the air inlet (see figure 1).



(a) α-pinene concentration



(b) air velocity

Figure 3 - CFD profiles compared to measurements

Satisfactory agreements are noted between CFD results and in situ measurements carried out in the room.

3.3 CFD results compared to measurement carried out in the kitchen

Figure 4 compares numerical profile of air velocity and measurements carried out front of the kitchen air inlet (see figure 2).

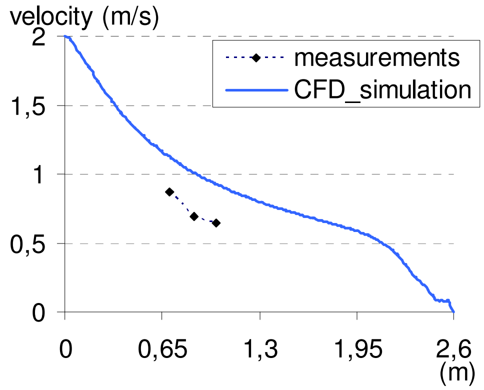
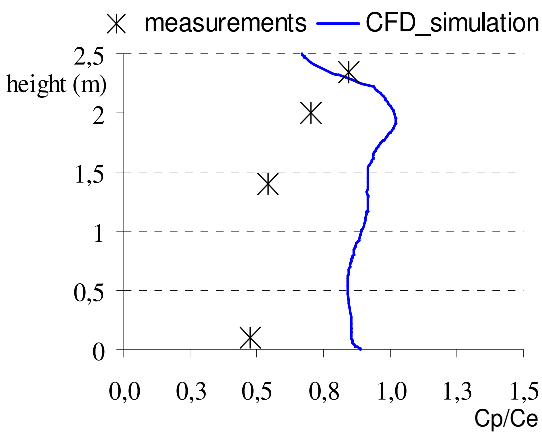
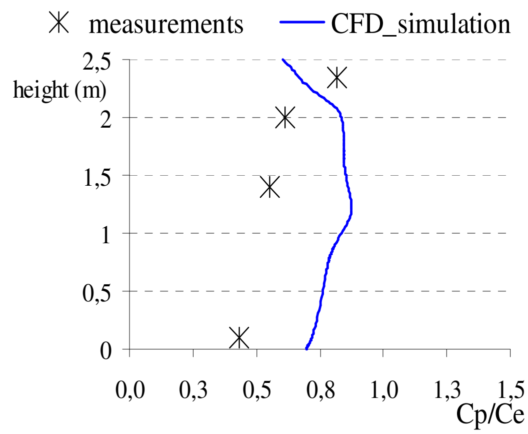


Figure 4 - CFD profile of air velocity compared to measurements realized front of the air inlet

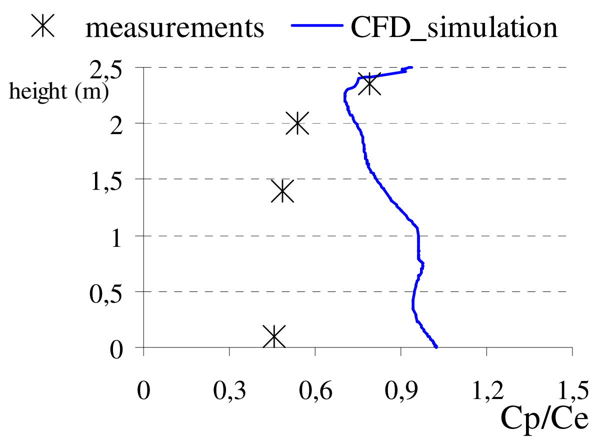
Figure 5 compared numerical profiles of SF_6 concentration to measurements realized in the kitchen.



(a): stick located on the position 1



(b): stick located on the position 2



(c): stick located on the position 3

Figure 5: CFD profiles of SF_6 concentration compared to measurements

Where C_P and C_e represent SF_6 concentrations respectively at a measurement point P and at the air exhaust. Moreover, within the kitchen, the measurements probes are situated on a vertical stick. Its different locations during the measurements series are indicated on figure 2.

Figure 4 and figure 5 show that the numerical results do not correspond to the in situ experimental results. This can be explained by the air leakage influence. Indeed, the measurements series have been realized in real environment. Therefore, air leakage has occurred at the walls. However, the numerical model handled in this paper does not take into account the air leakage. The next section analyses air leakage influence on the accuracy of the numerical predictions.

4. TOWARD AN IMPROVEMENT OF THE CFD RESULTS QUALITY

4.1 Influence of the air leakage

This section deals with analysis of the air leakage influence on the quality of CFD simulations. In so doing, the air leakage rates at the walls of the room and the kitchen are evaluated as follows, Moyé (1985):

$$Q_{leakage} = K.(\Delta P)^n \tag{11}$$

Where n is a constant which is generally equal to 2/3. K is also a constant. ΔP indicates the pressure difference between indoor environment and the exterior of the local studied.

Air leakage rates at the walls of the room and the kitchen are indicated on table 3.

Table 3. Air leakage rates through the walls of the local studied (the room and the kitchen)

	$\Delta P(Pa)$	$Q_{leakage}$	Q	$Q_{leakage}/Q$
room	0.1	1.2 m^3/h	18,2 m^3/h	6 %
kitchen	5	17 m^3/h	120 m^3/h	14 %

Based on the data of the table 3, the followings points can be noted:

- Air leakage rate at the room walls is estimated to 1.2 m^3/h. This corresponds to 6 % of the fresh air which does not pass by the inlet air. Fortunately, when performing CFD simulations with boundaries conditions given by the measurements carried out in the room, satisfactory quality of the CFD results is obtained although the air leakage is not taken into account in the numerical model (cf.: figure 3). Therefore, air leakage rate which represents 6 % of the room air flow rate has slight influence on the internal flow.

- Air leakage rate at the kitchen walls is estimated to 17 m^3/h. This indicates that, during the in situ measurements, 14 % of the fresh air does not pass by the inlet located under the door (see figure 2). This air leakage is significant enough to decrease the quality of the numerical results since this phenomenon is not considered in the numerical model (cf.: section 3.3).

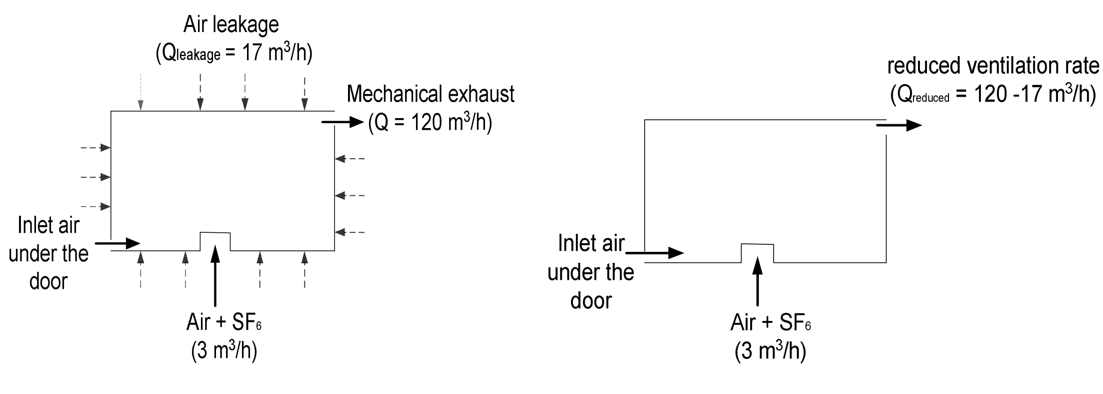
4.2 Simplified model to take into account the air leakage

As shown in the section above, in the case of the kitchen, the air leakage is significant during the experimental tests. It has to be taken into account in the numerical model when performing CFD simulations with boundaries conditions given by in situ experimental tests.

In this purpose, it would have been necessary to solve $Q_{leakage} = K.(\Delta P)^n$ at the walls of the calculation field. This needs to know the pressure difference between indoor environment and the exterior of the local studied. This also needs the accuracy knowledge of air leakage characteristics (assessment of *K* and n). That is difficult to realize in real environment like the kitchen studied here.

As an alternative, a first level of air leakage modelling is proposed in this section. It consists in using a reduced ventilation rate at the mechanical exhaust:

$$Q_{reduced} = Q - Q_{leakage} \tag{12}$$



(a): basic model (b): simplified hypothesis of modelling

Figure 6 - Simplified model to take into account the air leakage rate of the kitchen

Numerical profiles obtained with the simplified model of air leakage taking into account are compared to in situ experimental results on figure 7 and figure 8.

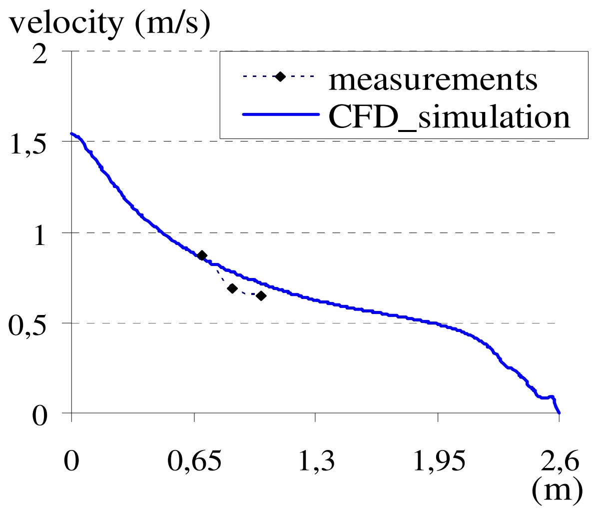
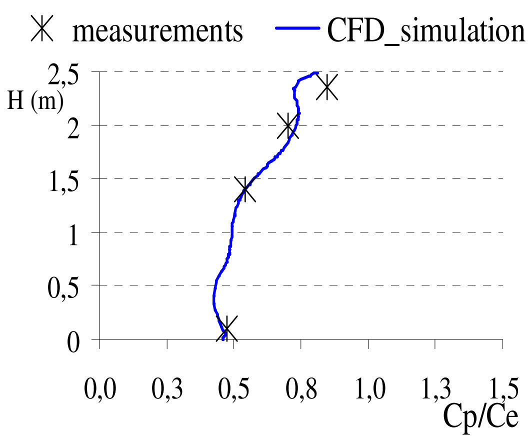
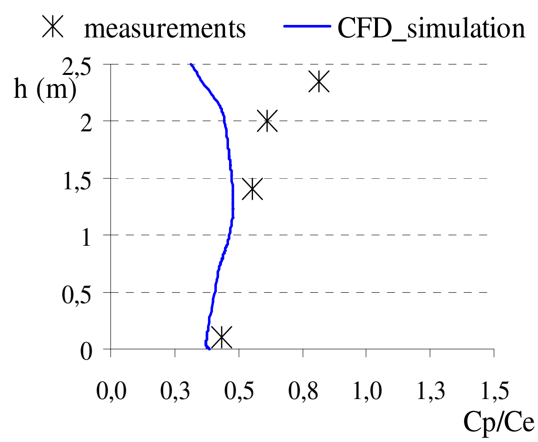


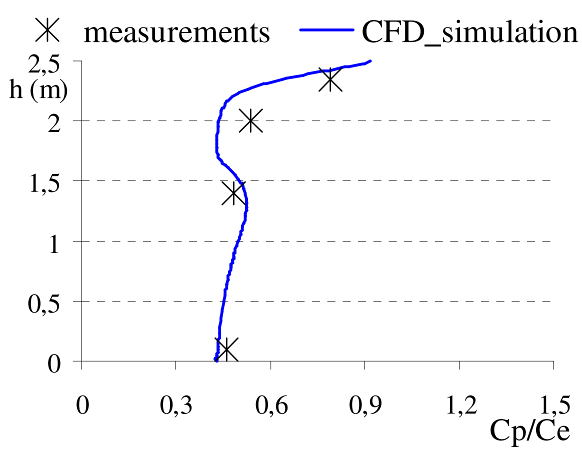
Figure 7 - CFD profile of air velocity compared to measurements realized front of the air inlet



(a): stick located on the position 1



(b): stick located on the position 2



(c): stick located on the position 3

Figure 8 - CFD profiles of SF_6 concentration compared to measurements

As shown on the figure 7 and the figure 8, the quality of the CFD results is globally improved by the air leakage modelling which uses a reduced ventilation rate: $Q_{red} = Q - Q_{leakage}$. However, numerical profiles of SF_6 concentrations do not correspond to measurements realized front of the air inlet (cf: figure 8_(b)). Indeed, the CFD calculations underestimate the SF_6 concentration in the air located front of the kitchen door.

This can be explained by the conservation equations of mass, momentum, energy, etc., used to model the internal flow (cf.: section 3.1). Indeed, since the mass flow rate is modified when using a reduced ventilation rate ($Q_{reduced} = Q - Q_{leakage}$), the solution of these equations is also modified. Therefore, the internal flow pattern obtained with the simplified hypothesis of air leakage modelling can not correspond to the reality.

To numerically deal with the real internal flow of the kitchen, it would have been necessary to solve equation $Q_{leakage} = K.(\Delta P)^n$ at the walls of the calculation field. This is currently very difficult to realize with numerical calculations.

5. CONCLUSION

This paper dealt with the accuracy of CFD predictions confronted to in situ measurements. In so doing, measurements have been carried out in a room and a kitchen of the experimental house MARIA. These in situ measurements have been used as boundaries conditions of CFD simulations. Thereafter, qualities of the numerical results have been analysed.

As a result, it is stated that as long as the CFD simulations are performed with boundaries conditions given by measurements carried out in a local with slight air leakage, satisfactory quality of numerical results is noted although air leakage is not considered in the numerical model. This situation is verified in the case of the room studied.

However, in the case of CFD simulations performed with boundaries conditions given by measurements carried out in a local with significant air leakage, the quality of the numerical results are decreased since this phenomenon is not considered in the numerical model. This situation is verified in the case of the kitchen studied in this paper.
To consider the air leakage in the numerical model, it would have been necessary to solve the equation $Q_{leakage} = K.(\Delta P)^n$ at the walls of the calculation field. Unfortunately, this is currently very difficult with CFD code.

As an alternative, a simplified model of air leakage taking into account has been proposed. It allowed an improvement of numerical predictions quality.
However, the internal flow modelled with this last hypothesis can not exactly correspond to the reality.
Indeed, the mass flow rate changes when using a reduced ventilation rate ($Q_{red} = Q - Q_{leakage}$). As a result, due to the Navier-Stokes equations handled in the CFD calculations, the internal flow pattern is also modified.
Nevertheless, in the case of CFD simulations with boundaries conditions given by in situ measurements, the simplified model proposed in this paper can constitute a first level of air leakage taking into account to improve the numerical predictions quality.

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