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Study of the Aeraulic Flows in a Building Including Heating and Air Conditioning Systems

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Abstract This study is based on the modeling of the air flow in the hall building including heating and air-conditioning systems. The building contains two converter stations "valves" considered as heat sources. Heat transfer in the hall is numerically simulated using the standard k- ε model of turbulence. For a very hot weather, this study aims to evaluate the local temperatures in the ambient air of the hall, with assuming running valves and air conditioning device in open loop with a 35°C inlet temperature. The study has shown that the air conditioning is efficient enough to maintain low level of temperature disparity. It has been found that the maximum air temperature around bushings does not exceed 40°C which is the highest temperature supported by the bushings.

Keywords: Aeraulic, thermal insulating, forced convection, building materials, thermal conductivity.

Nomenclature

thermal diffusivity a= $\lambda/\rho c_p$ ($m^2 s^{-1}$)
Height of Hall (m)
masse (kg)
molecular weight of air (kg. $kmol^{-1}$)
thermodynamic Pressure (Pa)
Specific heat at constant pressure ($J.kg^{-1}.K^{-1}$)
flow rate ($m^3.s^{-1}$)
surface (m^2)
temperature (K)
$\overline{T}$ mean temperature (K)
Rayleigh number Ra=$\rho g\beta\Delta TH^3/\mu.a$

Grashof number Gr = $g\beta\Delta T\, H^3/\nu^2$
C_3 coefficients in (k-ε) models
Greek symbols
density ($kg.m^{-3}$)
λ thermal conductivity ($Wm^{-1}K^{-1}$)
μ dynamic viscosity (kg m^{-1} s^{-1})
μ_t turbulent viscosity (kg m^{-1} s^{-1})
ε dissipation rate (m^2s^{-3})
k turbulent kinetic energy ($m^2.s^{-2}$)
β coefficient of thermal expansion (k^{-1})
v_i inlet flow velocity ($m.s^{-1}$)

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1 Introduction

The buoyancy forces of heat transfer resulting from temperature gradient is shown to have large influences on heat transfer of fluid in many engineering systems, drying processes and building technology, for example.

Two-equations turbulence models allow the determination of both, a turbulent length- and time scale by solving two separate transport equations. The standard k-ε model in ANSYS [Ansys] falls within this class of models and has become the workhorse of practical engineering flow calculations in the time since it was proposed by Launder and Spalding [Launder and Spalding (1972)]. The standard k-ε model is a model based on model transport equations for the turbulence kinetic energy (k) and its dissipation rate (ε). The model transport equation for (k) is derived from the exact equation, while the model transport equation for (ε) was obtained using physical reasoning and bears little resemblance to its mathematically exact counterpart. In the derivation of the k-ε model, the assumption is that the flow is fully turbulent, and the effects of molecular viscosity are negligible. For a much more practical approach, the standard k- ε turbulence model is used which is based on our best understanding of the relevant processes, thus minimizing unknowns and presenting a set of equations which can be applied to a large number of turbulent applications. Therefore, we employ in our study a standard k–ε turbulence model [Launder and Spalding (1972)]. In fact, the standard k–ε model [Jones and Launder (1972); Launder and Spalding (1974)], is till now by far the most widely used and validated of the classical models.

The site object of work connects sellindge, England and les Mandarins by a national Grid and RTE to allow the transfer of high voltage electricity from England to France and vice versa. The goal of the project studied is the renovation of two converter stations having a capacity of 2x1000MW (HVDC), high voltage Direct current, with an electric potential of 270 KV, link connecting the GB and French transmission systems, was attributed to AREVA T&D [PC 8892 A AREVA(2009)]. After this the electricity is produced at a power station where it will be distributed to the customers. The produced electricity is transferred towards the outside via the overhead conductors which operate at high voltage and connect the valves with the bushings.

This study concerns the building of the Valve Halls Mandarins, France, as shown in Fig.1. The valves have to be replaced because the existing valves discharge all of their heat into the halls, whereas the new valves will have water cooling systems to remove most of the heat at source. However, some heat will still be discharged into the halls, so the ventilation systems must be changed to assure the requirements of the new valves and to maintain the halls at the suitable conditions. The computational fluid dynamics (CFD) method is used to investigate the temperature fields in a mechanically ventilated hall. The maximum air temperature around bushings must be lower than 40°C. The high degree of temperature is not supported by the bushings and will be damaged.

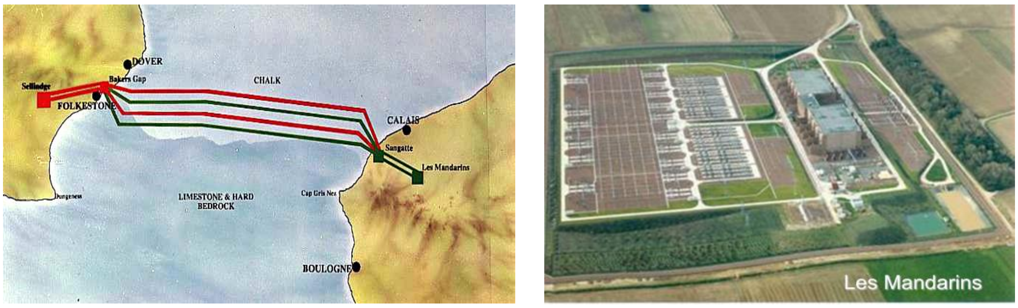


Figure 1: The located site at " Les Mandarins", France

2 Mathematical formulation

Computations are made on the three-dimensional Cartesian coordinate system using the standard k–ε model of turbulence for incompressible flow. There are no chemical reactions and the radiative heat transfer in the hall is neglected. The air is invoked as ideal gas ($\rho = \frac{PM_a}{RT}$) and the thermophysical properties of air are assumed to be constant.

In Reynolds averaging, the solution variables in the Navier-Stokes equations are decomposed into the mean ϕ and fluctuating ϕ' components.

The fluid is considered as incompressible with an ideal-gas law depending only with temperature. The equations for continuity, momentum and energy are as follows:

The continuity equation:

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

The momentum equation:

$$\rho\frac{\partial}{\partial x_j}(u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \rho\frac{\partial}{\partial x_j}\left(-\overline{u'_i u'_j}\right) + \rho g_i \tag{2}$$

The energy equation:

$$\rho\, c_p \frac{\partial}{\partial x_i}(u_i T) = -\frac{\partial}{\partial x_i}\left(\lambda\frac{\partial T}{\partial x_i}\right) + \rho\, c_p \frac{\partial}{\partial x_i}\left(-\overline{u'_i T'}\right) + S_h \tag{3}$$

The source term S_h in the energy equation is the volumetric power produced within the valves.

The turbulence kinetic energy, k, and its rate of dissipation, ε, are obtained from the following transport equations:

$$\rho\frac{\partial}{\partial x_i}(k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon \tag{4}$$

and

$$\rho\frac{\partial}{\partial x_i}(\varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \frac{\varepsilon}{k}(C_1 G_k + C_3 G_b) - C_2\rho\frac{\varepsilon^2}{k} \tag{5}$$

In these equations, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients. G_b is the generation of turbulence kinetic energy due to buoyancy. C_1, C_2, and C_3 are constants. σ_k and σ_ε are the turbulent Prandtl numbers for k and ε, respectively.

$$G_k = -\overline{u'_i u'_j}\frac{\partial u_j}{\partial x_i} \qquad \text{with } \overline{u'_i u'_j} = -\nu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{6}$$

$$G_b = \beta g_i \frac{\mu_t}{Pr_t}\frac{\partial T}{\partial x_i} \tag{7}$$

The coefficient of thermal expansion, β, is defined as: $\beta = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p$. For ideal gas, the equation (7) reduces to:

$$G_b = -g_i \frac{\mu_t}{\rho Pr_t}\frac{\partial \rho}{\partial x_i} \tag{8}$$

Where Pr_t is the turbulent Prandtl number for energy and g_i is the component of the gravitational vector in the *i* direction. For the standard k- ε models, the default value of Pr_t is 0.85.

The turbulent viscosity, μ_t, is computed by combining k and ε as follows:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{9}$$

Where ρ represents the density; k and ε correspond to the turbulent kinetic energy and its rate of dissipation, respectively. C_μ= 0.09 is the empirically value.

The model constants C_1, C_2, σ_k and σ_ε, have the following default values determined from experiments for fundamental turbulent flows: $C_1 = 1.44$, $C_2 = 1.92$, $\sigma_k = 1$ and $\sigma_\varepsilon = 1.3$.

Inclusion of the production term due to buoyancy in the k-ε model was first proposed by Rodi [Rodi (1978)]. Cox et al. [Markatos, Malin and Cox (1982)] conducted a detailed study to determine the influence of C_3 to the k-ε equations. Viollet has demonstrated the effectiveness of G_k, C_1, and C_3 mentioned above in flow through reactor coolant analysis [Viollet (1987)]. Viollet [Viollet (1987)] has adopted the following procedure for determining the value of C_3 based on the sign of G_k; the term denoting the buoyancy production of turbulence in k. If $G_k > 0$; $C_3 = C_1 = 1.44$ and $C_3=0$ if $G_k < 0$. Note that the sign of G_k is related to the stability of flow.

Finally, the two equations for the kinetic energy and the dissipation rate of the standard k–ε turbulence model are added to the continuity equation, the Navier-Stokes momentum equations and the thermal energy equation in order to capture the air flow in the hall.

The dimensionless governing parameters are de Reynolds number Re, Grashof number Gr, Prandtl number Pr and Richardson number Ri which are defined respectively as:

$$Re = \frac{u_i H}{\nu}, \quad Gr = \frac{g\,\beta \Delta T\, H^3}{\nu^2}, \; Pr = \frac{\nu}{a}, \quad Ri = \frac{Gr}{Re^2}$$

Where, $\nu = \frac{\mu}{\rho}$ indicates the kinematic viscosity of the fluid and H is the height of the building.

After discretization, the conservation equation for a general variable ϕ at a cell P can be written as:

$$a_p \phi_p = \sum_{nb} a_{nb} \emptyset_{nb} - b \tag{10}$$

a_p is the center coefficient, a_{nb} refers to the influence coefficients for the neighboring cells, and *b* is the contribution of the constant part of the source term. The coefficients a_{nb} and *b* will be different for every cell in the domain at every iteration. The scaled residual at the center of cell P as a function of the variables in the surrounding cells, is defined as:

$$R^{\emptyset} = \frac{\sum_{cells,P}|\sum_{nb} a_{nb}\emptyset_{nb}+b-a_P\emptyset_P|}{\sum_{cells,P}|a_P\emptyset_P|} \tag{11}$$

3 Solution procedure

3.1 Computational conditions

Heat transfer within the valves hall is numerically simulated using the standard k-ε model of turbulence. Considering the geometrical symmetry of the valves hall, only one-half of the entire field is used as the solution domain, as shown in Fig.2.

However, the studied building consists of two identical rooms, called "pole1" and "pole2". The building contains three valves and the same number of bushings inside each room. Thus, the symmetry plane is formed by the common wall separating the two rooms.

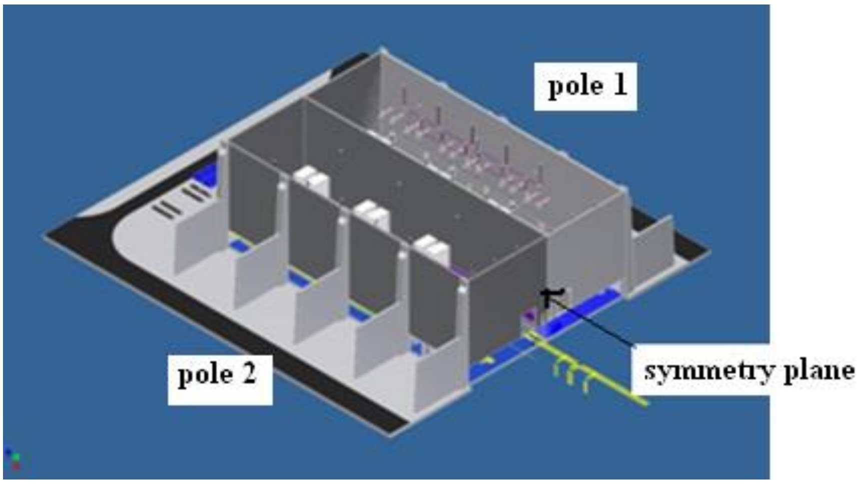


Figure 2: The building of the Valves Hall Mandarins, France

The dimensions of each Valve Hall are 18.3 m wide, 60.5 m length and 18.1 m height. The construction of the buildings is assumed to be as follows;

- Valve Hall Floor 200 mm reinforced concrete with a density of 2500 kg m^{-3}
- Valve Hall Walls 350 mm reinforced concrete with a density of 2500 kg m^{-3}
- Valve Hall Roof 150 mm reinforced concrete with a density of 2500 kg m^{-3} with bitumen felt on top. The insulation under roof felt 60 mm thick foam glass with a thermal conductivity of 0.042 W $m^{-1}K^{-1}$.

The thermophysical properties of air were evaluated at reference temperature T_0=313K.

Table 1: Thermophysical properties of air at T_0=313K and atmospheric pressure

ρ_a(kg.m^{-3})	C_{pa}(J.kg^{-1} K^{-1})	μ_a(kg.m^{-1} s^{-1})	k_a(W.m^{-1} K^{-1})
1.132	1006	2.006×10^{-5}	0.0272

Calculation were carried out by utilizing the commercial, control-volume based code Ansys13. Results of the simulations were collected and processed by employing in-house softwares. A second-order upwind scheme was used for the advective and transport terms. The velocity-pressure coupling was solved with Simple algorithm and the pressure was calculated with a body-force weighted scheme. The convergence criterion required that the scaled residuals equation (Eq.11) be smaller than10^{-5} for mass and momentum equations and smaller than 10^{-6} for energy and transport equations.

The isotherms and mean velocity vectors have been analyzed to explain the flow and thermal fields structure of the heat transfer. The Reynolds number Re of the flow based on the inlet velocity $v_i \approx 4.86$m/s and the building height H and the Prandtl number Pr were calculated according to the working conditions of the installation and were kept fixed during the computation (Re=5×10^6 and Pr=0.71), respectively.

Table 2: The dimensionless numbers

Pr	Ra	Gr	Re	Ri
0.71	4.39×10^{12}	5.92×10^{12}	5×10^6	0.24

3.2 Boundary conditions

The studied hall contains three valves. When the valves are operating, the internal heat gain is 40 kW per the three valves hall at temperature of 50°C local to the valves. However, some heat will still be discharged into the hall, and the air conditioning systems are designed to remove this heat and maintain the hall at the correct conditions. The system can run continuously for 1 year without maintenance necessitating a shutdown, apart from filter changing. The ventilation shall operate 24 hours per day to maintain conditions

Ventilation includes both the exchange of air to the outside as well as circulation of air within the hall. It is one of the most important factors for maintaining acceptable condition in the hall. To maintain the area around the valve hall at a reasonable temperature, we have assumed that several fans would be installed in the external walls and roof around each hall. The ventilation rate is expressed by the volumetric flow rate of outside air being introduced to the hall. The total flow rate delivered by the ventilator during inspiration is $q_v = 1.53\ m^3\ s^{-1}$.

At Les Mandarins, to achieve the air temperature around the bushings at 40°C, air will be directed to the bushings at approximately 35°C or lower. This will generally be achieved using ambient air, however, when the ambient temperature rises above 35°C, the cooling system will be activated. Also, the natural ventilation of the hall is achieved through openings in the lower areas. The jet is maintained at a fixed temperature T_e by an air-treatment system. Also, the ventilation system allows us to impose inlet flow rate q_m.

Table 3: Boundary conditions of the simulation

Heat source	$V_{valve}= 178\ m^3$ *and* $S_h=74.7\ Wm^{-3}$
Mass flow rate and temperature of ventilation	$q_m= 3\times0.577\ kg.s^{-1}$ and T_e=35°C
Opening of aeration	Outflow
Floor	adiabatic for temperature and $\vec{u}=0$
Roof	h_T=0.67 W $m^{-2}K^{-1}$, T_{ex}=45°C and $\vec{u}=0$

External vertical walls	h_{ex}=25 W $m^{-2}K^{-1}$, T_{ex}=45°C and $\vec{u}=0$
Symmetry plans	Normal velocity to the plan is set at 0 and free slip, otherwise (normal gradient of zero)

4 Computational results and discussions

Simulation of heat transfer in a full computational flow domain requires a high computational effort; hence, many studies made use of the symmetry of the flow problem and applied half plane computation to reduce the computational demand [Lu, Lo and Fang (2001), Lu and Yuen (2001)]. As a result, very fine grid size was achieved inside the simplified model to improve the numerical accuracy.
The section presents the numerical results for the simplified and the complete models as shown in Figs.4 and 7. The comparison concerns the mean temperature fields. Several plans have been chosen for the analysis: The vertical plans perpendicular to the bushings. The bushings are the cylindrical bars of metal having a diameter of 0.356 m and length of 2.46 m.

4.1 Simplified model

The simplified model is represented in Fig.4. The model consists of one enclosure, whose dimensions are 7.08m, 18.1m and 18.3m according to the coordinate directions (x,y,z). The vertical wall with fixed bushings is in contact with a hot ambient temperature at T_{ex}=45°C with uniform surface heat transfer coefficient h_{ex} and the opposite wall is the symmetry plane as described in section 3.1.

The building is in contact with a hot external ambient through the roof and the floor is assumed adiabatic. The jet is maintained at the fixed temperature T_e=35°C and the inlet flow rate q_m is insured by the ventilation system. The valve heating source is located at the center of computed domain which is delimited by two other vertical planes of symmetries as illustrated in Fig.3. The computational conditions are summarized in Tab.3. Details of the geometry and boundary conditions are shown in Fig.4.

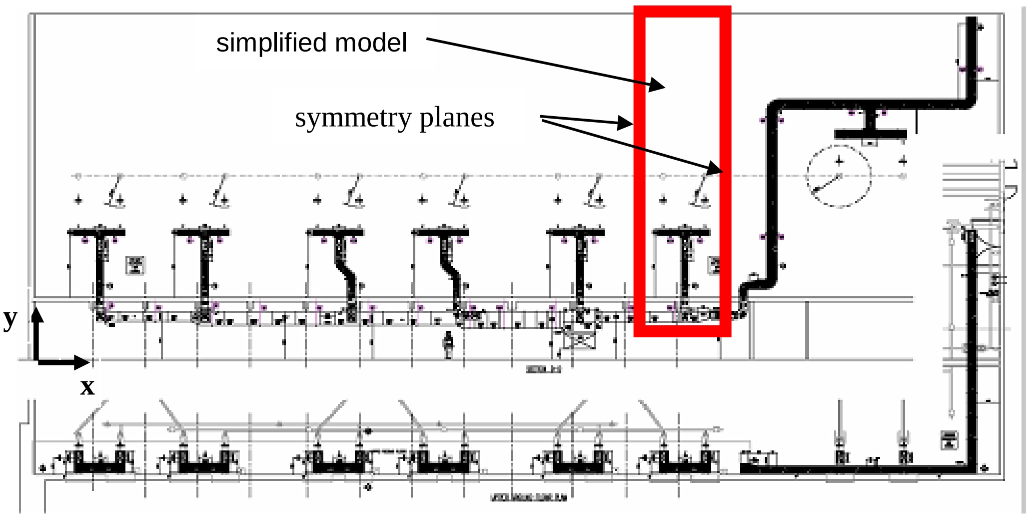


Figure 3: The ventilation plan of the valves hall

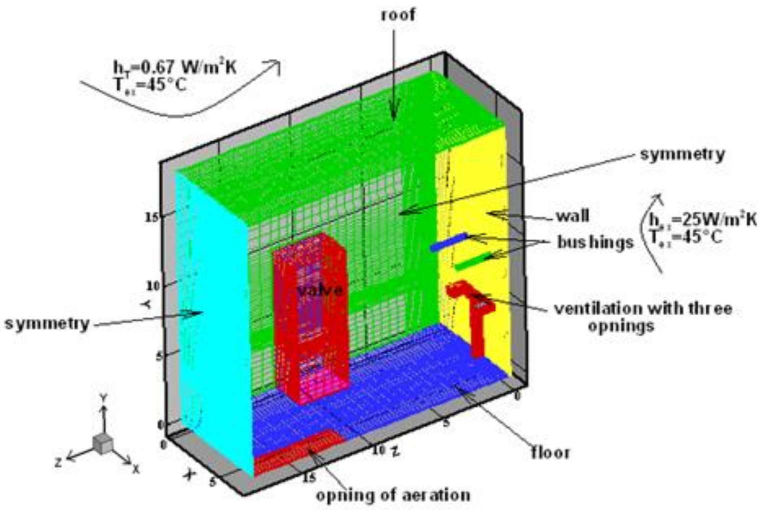
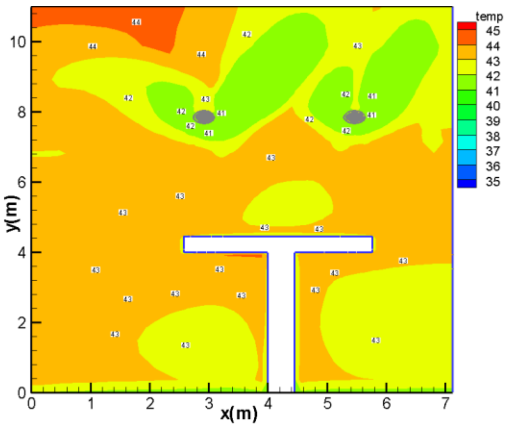


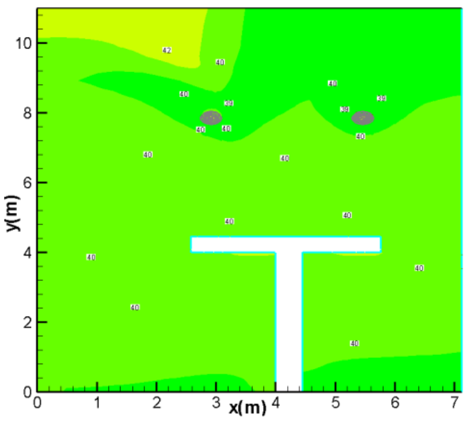
Figure 4: The simplified model of the hall valve

To conduct our study, we start with the simplified model; this approach seems attractive because the increase in size of the computational domain proves to be expensive, both in memory and in computational time. For these reasons, the domain extensions are often either relatively reduced or large but coarsely discretized. In order to evaluate the accuracy of our models, we then compare the mean temperature and the velocity magnitude in the jet zone. In all cases, the simulation models prove that the expansion of the jet and then the dynamic of the fluid can correctly predicted.

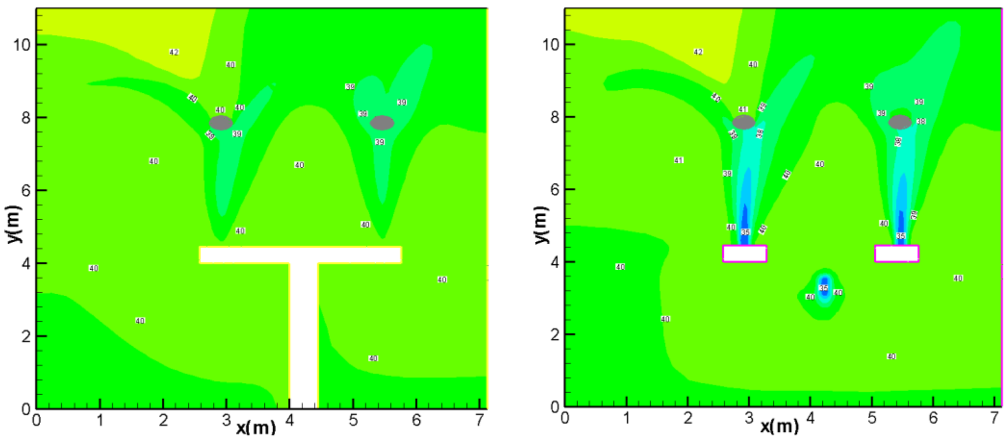
The figure 5 indicates that the maximum valves hall temperature is about 50°C and the maximum air temperature around bushings is about 40°C. In the ventilated hall, the dynamic of the flow is quite simple because there are two principal zones: the jet zone and the non-moving fluid zone where the fluid velocity is less than 0.05 m/s, as displayed in Fig.6.



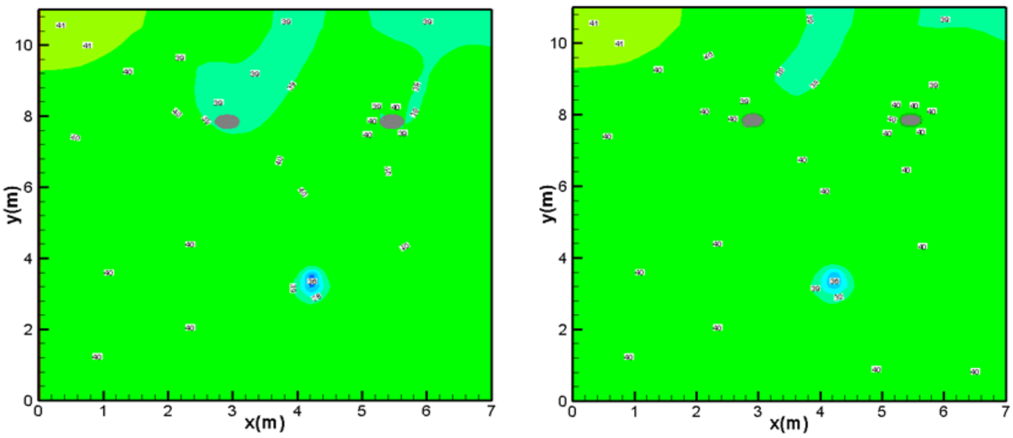
(a) at the surface of the wall (z=0)



(b) at 0.10m from the wall (z=0.10m)



(c) at 0.50m from the wall (z=0.50m) (d) at 1m from the wall (z=1m)



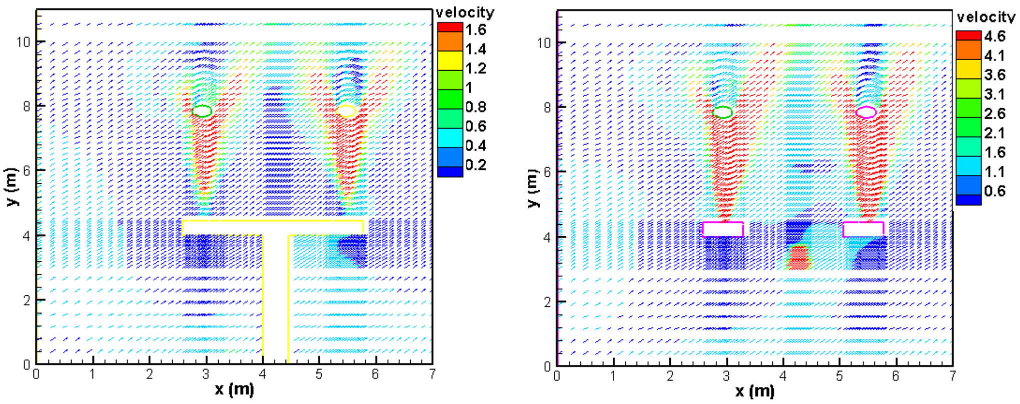
(e) at 1.50m from the wall (z=1.50m) (f) at 2m from the wall (z=2m)

Figure 5: The mean temperature fields

The thermal convection leads to a vertical temperature stratification within the hall with the lowest temperatures along the bushings estimated between 38°C and 40°C, fed by the ventilation air. The high temperature region has approximately 42°C fed by discharged heat of the valves into the hall.

The changes in the temperature contours indicate the effect of the dominated forced convection on the heat transfer process. It noted that the Richardson number Ri=0.24 indicates the dominance of the forced convection (Tab.2).

At very low values of Richardson number, such as 0 and 0.1, the forced convection due to the driven force dominates the flow structure. Knowing that the inertia force of the fluid is dominant compared to the buoyancy force according to the magnitude of Ri (Ri<1) [Sumon and al. (2006)]. Thus, as the Richardson number increases to Ri ≈ 0.24 as the following studied case, the forced convection dominated regime and the fluid flow intensity increases with the forced convection flow conditions imposed by providing the ventilation air.



(a) at 0.5 m from the wall (z=0.5m) (b) at 1 m from the wall (z=1m)

Figure 6: The mean velocity vectors

Figure 6 provides a representation of the evolution of the two jets simulated. Colored velocity vectors represented along vertical planes illustrate the growth of each jet along its centerline. The flow advances in the "y" direction and is deviated around the bushings. Close to the opening, the centerline velocity of jets remains equal to the initial velocity, denoting the core region of the jets. The velocity starts decreasing at the certain distance from the jet centerline and an axisymmetric turbulent jet is developed.

4.2 Complete model

The calculation is performed over the complete model presented in Fig.7. Figure 9 shows that the complete model provides results in agreement with the simplified model. An analysis of the results obtained for the simplified model proves to exhibit very similar properties and coincides very well to those of the complete model based of the simulation of the entire hall, even if for slightly different parameters. Hence, the simplified model presents the advantage of having fewer grids, which required half day computing time on a workstation.

The temperature distributions are displayed in Fig.10 for planes at z=0 and z=1m from the wall, respectively. The possible symmetry of the flow fields about the vertical middle plans are evidenced at steady state.

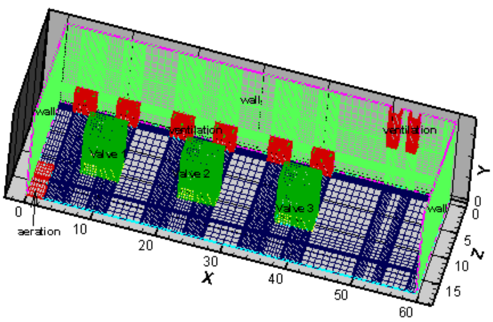


Figure 7: The complete model of the hall valves

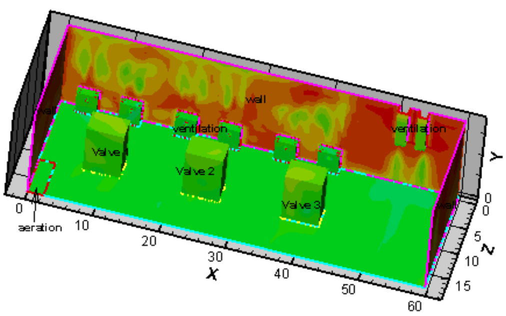


Figure 8: The surface temperatures of the walls hall

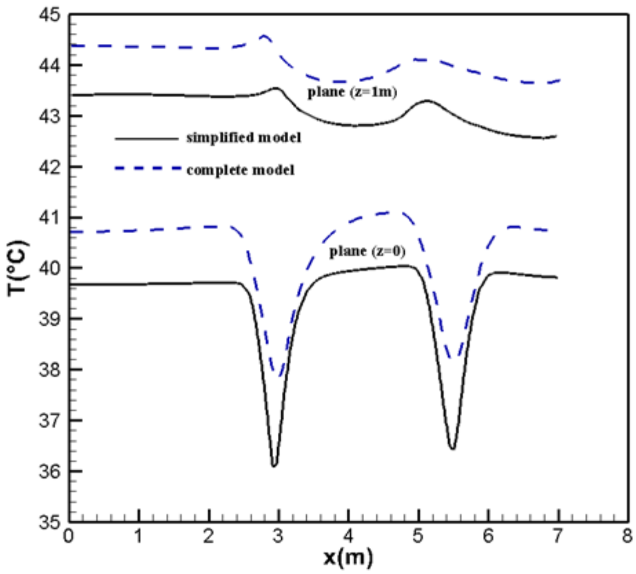
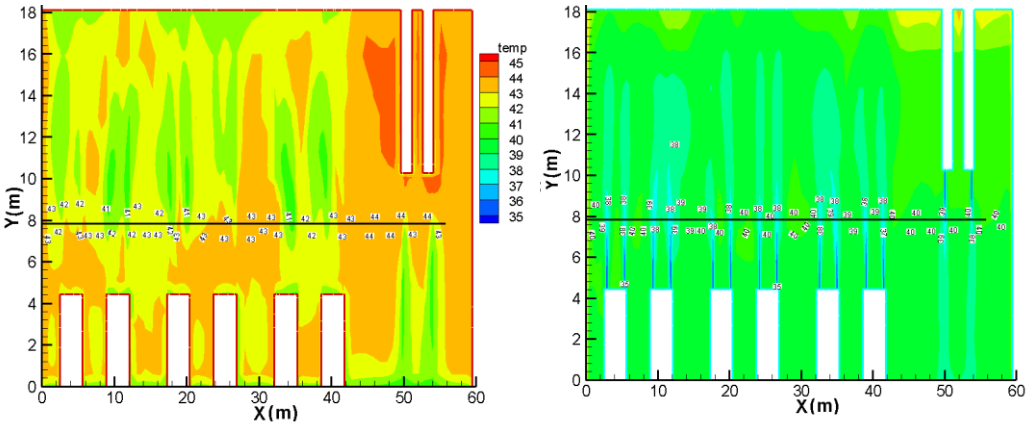


Figure 9: The evolutions of temperature in plans (z=0 and z=1) at y=6m



(a) at the surface of the wall (z=0) (b) at 1m from the wall (z=1m)

Figure 10: The mean temperature fields

5 Conclusions

The simulations presented were carried out for both simplified and complete models. The standard k– ε model seems to predict well the velocity and temperature fields in the valves hall. So, the global comportment of the fluid is correctly solved. The prediction of the airflow and the slope of the jet is well identified. However, this parameter is very important for the thermal comfort conditions of a building hall.

Knowing that the design of all thermal loads and selections of equipment will be based on the external design conditions which indicate that during summer, the average maximum of temperature reaches 45°C. The calculated mean air temperatures reach 39.5°C inside the valves hall. The local air temperatures variation around the double bushing is about +/-1°C from mean value but the simple bushing located at 16.85m high is more exposed to warm air at 42°C than the doubled bushing. The study indicates that the staff will wait two hours before entering the valves hall within the air temperatures become lower than 25°C when the valves are stopped and air conditioning loop closed.

It should be noted that the cost of air conditioning of the valves hall is increasing rapidly. The best way to reduce the cooling cost is to add insulation to walls of the hall that is below recommended levels. Finally, the study aims was to ensure a satisfactory installation and the feasibility of the project.

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