Numerical analysis of periodic 3D convective heat transfer in fenestration with between-the-glass louvered blinds

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ABSTRACT

A three-dimensional unsteady laminar natural convection in fenestration with between-the-glass louvered blinds. The considered system is a 3D vertical cavity with differentially heated vertical walls and with rotatable louvered blinds located exactly midway between the walls. The temperature imposed to the hot wall is periodic to be closer to real conditions. The effects of, louvered blinds inclination, thermal conductivity ratio, period and Rayleigh number on the temperature field, flow structure and heat transfer were examined.

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1. Introduction

Because of their impact on energy performances, fenestration with incorporated louvered blinds exists in a non-negligible number of buildings. The current numerical study focuses on the 3D transient natural convection within a tall window with between-panes louvered blinds.

The physical model is shown in Fig. 1. A blinds consisting of a set of inclined slats enclosed between two differentially heated glazings. The imposed hot wall temperature is periodic. The enclosure has as height, H, and a glazing spacing; W. The aspect ratio is fixed at \( Ar = \frac{H}{W} = 5 \). Blinds have as length \( L_b \) and width \( \Delta \).

Several studies have been conducted about window with between-panes louvered blinds. However most of these studies are for tow-dimensional configurations.

Urbikain and Sala [1] studied the influence of louvered blinds in a double-glazed unit during night-time conditions. They compared results for two types of blinds material using analytical and CFD studies. Almeida and Naylor [2] conducted an experimental study of free convection in a window with an enclosed aluminum venetian-type blinds. Convective heat transfer measurements and temperature field visualization were obtained using a Mach–Zehnder laser interferometer. They found that the temperature field was unsteady and strongly periodic in the center region of the enclosure. Collins et al. [3] developed a two-dimensional steady laminar natural convection model of a window cavity with between-panes louveres. The configuration was approximated to a vertical cavity with isothermal walls at different temperatures, and with rotatable baffles located midway between the walls. It was found that the system is suited to a traditional one-dimensional analysis, and that the convective heat transfer is independent of the Rayleigh number. Collins [4] examined the convective heat...
transfer around equally spaced heated, horizontal, and rotateable louvers. Results were obtained using a steady, laminar, two-dimensional, conjugate conduction/convection finite element model for a range of Raleigh numbers, heating levels, and blinds placements. Shahid and Naylor [5] solved the combined convection-radiation heat transfer in a window equipped with Venetian blinds using a two-dimensional finite volume model. The results show that the presence of a Venetian blinds significantly improves the energy performance of a single and double glazed window. The blinds reduce the overall heat transfer rate through the window by reducing the thermal radiation from the indoor glazing. Clark et al. [6], made an experimental study of the effect of blinds on convection heat transfer at interior window surfaces. Results shows that heat transfer depend on supply flow rate, blinds angle, diffuser location and window configuration. A numerical study of free
convection in a tall vertical enclosure with an internal louvered metal blinds has been conducted by Avedissian and Naylor [7]. They considered the effects of Rayleigh number, enclosure aspect ratio, and blinds geometry to present an empirical correlation for the average Nusselt number. Dalal et al. [8], presented a simplified model of the coupled convective and radiative heat transfer to study the steady free convection in a double glazed window with a between-panes pleated cloth blinds. The numerical study considers the effects of Rayleigh number, enclosure aspect ratio, and blinds geometry on the convective heat transfer. Costa et al. [9], conducted A numerical study for laminar natural convection heat transfer occurring in a vertical stack of parallelogrammic partial enclosures which has an aspect ratio of 5. They analyzed the overall Nusselt number, the flow field and heat transfer mechanisms, by using the isotherms, the streamlines and the heat lines. Results show that the thermal performance of an enclosure is changed by the shutters introduction, and that the thermal conductivity of the shutters has an important effect on results. Cuevas et al. [10], used an empirical model to determine the convective heat loss, at an indoor glazing surface. The effect of blinds was studied experimentally at four different distances from the window frame. They predicted correlations for the Nusselt number and for the air mass flow rate incoming to the window cavity.

2. Governing equations and numerical solution

The 3D vorticity-vector potential formalism \((\vec{\omega} - \vec{\omega}')\) is used as numerical method, in order to eliminate the pressure term, which is delicate to treat. \((\vec{\omega})\) and \((\vec{\omega}')\) are respectively defined by the two following relations:

\[
\vec{\omega}' = \vec{V} \times \vec{u}' \quad \text{and} \quad \vec{\omega} = \vec{V} \times \vec{u}.
\]

(1)

The setting in equation is described with more details in the article of Ghachem et al. [11].

After nondimensionalizing the system of equations controlling the phenomenon are:

\[
\frac{\partial \vec{\omega}_i}{\partial t} + (\vec{u} \cdot \nabla) \vec{\omega}_i - (\vec{\omega}_i \cdot \nabla) \vec{u}_i = \Delta \vec{\omega}_i + Ra \cdot Pr \frac{\partial T}{\partial x_i}
\]

(2)

\[
\frac{\partial T}{\partial t} + \vec{u} \cdot \nabla T = \nabla^2 T \quad \text{in fluid}
\]

(3)

\[
\frac{\partial T}{\partial t} = \frac{\alpha_s}{\alpha_f} \nabla^2 T \quad \text{in solid}
\]

(4)

With: \(Pr = \frac{\nu_s}{\nu_f}, Ra = \frac{g \beta_f (T_b - T_c) \rho_f}{\nu_s \alpha_f \nu_f^3}\).

The energy equation (conduction) needs to be solved in the solid portion of the domain and the blinds conductivity \(k_s\) is assumed constant.

At the solid-fluid interface the temperature and heat flux must be continuous. The latter requirement is mathematically expressed as:

\[
\left( \frac{\partial T}{\partial n} \right)_f = \left( \frac{\partial T}{\partial n} \right)_s
\]

(5)

where \(R_s = k_s/k_f\) is the thermal conductivity ratio between the material of blinds and the medium that fills the fenestration.

There is no flow in the fraction of the volume occupied by the blinds (solid). The control volume finite difference method is used to discretize Eqs. (1)–(4). The central-difference scheme for treating convective terms and the fully implicit procedure to discretize the temporal derivatives are retained. The grid is uniform in all directions with additional nodes on boundaries. The successive relaxation iterating scheme is used to solve the resulting non-linear algebraic equations. The time step \(10^{-4}\) and spatial mesh \(41 \times 201 \times 81\) are retained to carry out all numerical tests. The solution is considered acceptable when the following convergence criterion is satisfied for each step of time:

\[
\sum_{i=1}^{1,2,3} \max \left| \frac{\left| T^n_i - T^{n-1}_i \right|}{\max |T^n_i|} \right| + \max \left| T^n_i - T^{n-1}_i \right| \leq 10^{-5}
\]

(6)

Boundary conditions for considered model are given as follows:

Temperature:

\(T = 0\) for \(x = 1\) and \(T = 1 + A \sin \left( \frac{\pi x}{L} \right)\) for \(x = 0\),

\(\frac{\partial T}{\partial n} = 0\) on all other walls (adiabatic).

Vorticity

\(\omega_x = 0, \omega_y = 0, \omega_z = \frac{\partial T}{\partial x}, \omega_z = \frac{\partial T}{\partial y}\) at \(x = 0\) and \(1\)

\(\omega_x = \frac{\partial T}{\partial x}, \omega_y = 0, \omega_z = \frac{\partial T}{\partial z}\) at \(y = 0\) and \(1\)

\(\omega_x = \frac{\partial T}{\partial x}, \omega_y = \frac{\partial T}{\partial y}, \omega_z = 0\) at \(z = 0\) and \(1\)

Vector potential

\[\vec{V} = \left( \begin{array}{c} \omega_x \\ \omega_y \\ \omega_z \end{array} \right)\]
\[ \frac{\partial \psi}{\partial x} = \frac{\partial \psi}{\partial y} = 0 \text{ at } x = 0 \text{ and } 1 \]
\[ \frac{\partial \psi}{\partial y} = \frac{\partial \psi}{\partial x} = 0 \text{ at } y = 0 \text{ and } 1 \]
\[ \psi = \frac{\partial \psi}{\partial z} = 0 \text{ at } z = 0 \text{ and } 1 \]

Velocity
\[ u_x = u_y = u_z = 0 \text{ on all walls.} \]
Local and average instantaneous Nusselt at hot wall are given as follows
\[ Nu = \left. \frac{\partial T}{\partial x} \right|_{x=0}, \quad Nu_m = \frac{1}{H \cdot D} \int_0^H \int_0^D Nu \cdot \partial y \cdot \partial z \]

The average heat transfer per period (overall Nusselt number) is evaluated by:
\[ \bar{Nu}_m = \frac{1}{\tau_p} \int_0^{\tau_p} Nu_m dt \]

3. Results and discussion

In this section impacts of different parameters on the temperature field, flow structure and heat transfer on the fenestration are presented.

3.1. Grid sensitivity and code validation

To ensure that the numerical results were grid independent a grid sensitivity testing was done. Calculations for \( Ra = 10^5 \) were performed for four grids; \( 21 \times 101 \times 41, 31 \times 151 \times 61, 41 \times 201 \times 81 \) and \( 51 \times 251 \times 101 \). The overall Nusselt numbers for \( 41 \times 201 \times 81 \) and \( 51 \times 251 \times 101 \) grids, differed by less than 0.5%. Thus for all simulations, the grid \( 41 \times 201 \times 81 \) was retained.

To validate the used code a comparison with the 2D results of Sourtiji et al. [14] was made using the same periodic boundary conditions considered by the authors. In fact Fig. 2, presents for \( Ra = 10^3 \) the projection of the velocity vectors in the XY plan and the stream lines found by Sourtiji et al. [14]. It is noticed the great similarity between the results, despite of the difference between the considered geometries (2D and 3D). In addition the code used in this research was fully validated for the case of 3D differentially heated cavities vertical without blinds and served for the publication of several papers [11, 15–17].

3.2. Temperature field and flow structure

In this part the effects of blinds inclination and Rayleigh numbers on temperature field and flow structure for fixed thermal conductivity ratio (\( Rc = 100 \)), hot temperature amplitude (\( A = 0.5 \)) and period (\( \tau_p = 0.5 \)) are studied.

Figs. 3–5 shows the numerically predicted iso-surfaces of temperature at \( t = \tau_p \) and isotherms in the XY plan for \( Ra = 5.10^5 \) at different instants of the period, it can be seen that, except near the top and bottom of the fenestration, the temperature field is repetitive in the \( y \)-direction (This observation is also valid for particle trajectories (Figs. 6–8)).

These figures show the existence of a horizontal stratification near of active walls, which is more pronounced for \( \theta = 0^\circ \) (open blinds) implying that open blinds significantly inhibits the convection. The intensity of this stratification periodically switches between active walls. It is also noticed the absence of the central symmetry except for \( t = \tau_p / 4 \), indicating a varied amount of heat transfer as function of time at both surfaces. Isotherms present a brusque slope change crossing the blinds, what is expected due to the chosen value of \( Rc (Rc = 100) \).

Figs. 3.a, 4.a and 5.a are plotted to show the 3D aspect of the temperature field, it’s clear from these figures that elapsing from a constant \( z \)-plan to another the isotherms are similar except near of the front and back walls.

For \( \theta = 0^\circ \) isotherms present a central vertical stratification. Changing the blinds angle (\( \theta = 30^\circ \) and \( -30^\circ \)) this stratification becomes in a direction normal to blinds. In all cases there is an intensification of the temperature gradient near of blinds tips. This intensification is due to the increase of the fluid flow when the blinds tips are in close proximity to the walls (see Figs. 9–11). The intensification is more noticed when \( \theta = 0^\circ \) because blinds tips are nearest. This result was also found by Huang [12] and Garnet [13] who noticed that for open blinds, when the louvers and glass were in close proximity, resulted in larger thermal transmission.

The effects of blinds angle and Rayleigh number on the flow at \( t = \tau_p \) are presented in Figs. 6–8. It’s clear that the increase of Rayleigh number increase the 3D aspect of the flow. For all cases the internal flow converges directly from the frontal wall toward the XY plan, then from this plan toward the frontal wall. The overall external flow is also convergent from the frontal walls toward the XY plan and occurs very near of active walls because of the existence of the blinds guiding the external flow far to center of the fenestration. The external flow is slightly away from the active walls for \( \theta = -30^\circ \) and \( \theta = 30^\circ \) because of the enlargement of the space between the blinds tips and the active walls when the blinds are fully open.

Fenestration is split into vertical stacks, so two types of external flow occur one in the overall of the cavity and one in each
Present work Results and Schematic diagram of Sourtiji et al [14]

Fig. 2. Validation of the code with the results of Sourtiji et al. [14]; for \( \tau = \frac{t}{p} \); \( \text{A}=1 \) and \( \text{Ra}=10^4 \).

Fig. 3. Iso-surfaces of temperature at \( t = \tau_p (a) \) and isotherms in the XY plan (b) \( t = \tau_p /4 \); (c) \( t = \tau_p /2 \); (d) \( t = 3\tau_p /4 \); (e) \( t = \tau_p \) for \( \text{Ra}=5.10^5 \), \( \text{Rc}=100 \), \( \text{A}=0.5 \), \( \tau_p = 0.5 \) and \( \theta = 0^\circ \).
The 3D flow is complex due to the passage of particles from a vertical plane to another. The projection of velocity vectors are drawn in Figs. 9–11 for a better understanding of the changes of the flow structure over a complete period for \( Ra = 5 \times 10^5 \) and different blinds inclinations. By analyzing these figures it is noticed that projections of velocity vectors are not closed in opposition with stream lines in 2D cases; this is specific to the 3D cases.

It is noticed that the flow structure is periodic due to the imposed periodic boundary condition. For \( \theta = 0^\circ \) and \( t = \tau_p/4 \) the flow is characterized by a diagonal (with positive inclination) two-vortex structure in each stack, progressing in the period there is coalescence the vortexes and the structure becomes mono-vortexes, with a quasi-bidimensional flow specially on the central stack (Fig. 9.d). At the end of the period there is a progressive return to the two-vortex structure.

The same observations are noticed for \( \theta = 30^\circ \). For \( \theta = -30^\circ \), the flow has invariant one vortex structure. This is due to the opposition between the inclination of the blinds and the structure that convective force attempt to impose.

### 3.3. Heat transfer analysis

The dependence of Nusselt number on the blinds inclination angle, thermal conductivity ratio, period and Rayleigh number, is analyzed to conduct the heat transfer analysis. The amplitude is fixed at \( A = 0.5 \).

Fig. 12, presents the temporal variation of \( \bar{Nu}_m \) for \( Ra = 5 \times 10^5 \) it is noticed that for all blinds inclinations the different variations are in phase with each other and are in phase shift with the variation of \( T_B \). This phase shift is due to the transient phase of the temporal variation. For \( \theta = 0^\circ \) the variation of \( \bar{Nu}_m \) is in amplitude offset compared to it variations for \( \theta = 30^\circ \) and \( \theta = -30^\circ \), implying a higher heat transfer rate.

Fig. 13, presents the variation of the overall Nusselt number as function of Rayleigh number for \( Rc = 100, Ra = 5 \times 10^5, A = 0.5, \tau_p = 0.5 \) and different blinds inclinations. For all Rayleigh numbers, \( \bar{Nu}_m \) is maximal for \( \theta = 0^\circ \) and minimal for \( \theta = 30^\circ \). For \( \theta = -30^\circ \) and \( \theta = 30^\circ \), the blinds and active vertical walls act like nozzles for the fluid flow, thus decreasing its intensity. This yields a minimization of \( \bar{Nu}_m \). Increasing Ra induces the increase of \( \bar{Nu}_m \), this is a trivial result caused by the increases of the buoyancy induced flow.

The effect of the thermal conductivity ratio on heat transfer is illustrated in Fig. 14, for \( Ra = 10^5, A = 0.5, \theta = 30^\circ \) and \( \tau_p = 0.5 \). It is noticed that the amplitude of \( \bar{Nu}_m \) is proportional to Rc. Thus the increase of the thermal conductivity ratio
Fig. 5. Iso-surfaces of temperature at $t = t_p$ (a) and isolines in the XY-plan (b) $t = t_p/4$; (c) $t = t_p/2$; (d) $t = 3t_p/4$; (e) $t = t_p$ for $Ra=5.10^5$, $Re=100$, $A=0.5$, $t_p=0.5$ and $\theta = -30^\circ$.

Fig. 6. Particle trajectory for: (a) $Ra=10^4$; (b) $Ra=10^5$; (c) $Ra=5.10^5$; at $t = t_p$ for $A=0.5$, $t_p=0.5$ and $\theta = 0^\circ$. 
Fig. 7. Particle trajectory for: (a) Ra = 10^4; (b) Ra = 10^5; (c) Ra = 5.10^5; at t = τ_p for A = 0.5, τ_p = 0.5 and θ = 30°.

Fig. 8. Particle trajectory for: (a) Ra = 10^4; (b) Ra = 10^5; (c) Ra = 5.10^5; at t = τ_p for A = 0.5, τ_p = 0.5 and θ = −30°.
Fig. 9. Velocity projection in the XY-plan (a) $t = Tp/4$; (b) $t = Tp/2$; (c) $t = 3Tp/4$; (d) $t = Tp/8$; (e) $t = Tp$, for $Ra = 5.10^5$, $Rc = 100$, $A = 0.5$, $T_p = 0.5$ and $\theta = 0^\circ$.

Fig. 10. Velocity projection in the XY-plan (a) $t = Tp/4$; (b) $t = Tp/2$; (c) $t = 3Tp/4$; (d) $t = Tp/8$; (e) $t = Tp$, for $Ra = 5.10^5$, $Rc = 100$, $A = 0.5$, $T_p = 0.5$ and $\theta = 30^\circ$. 
induces an increase of the heat transfer by convection. This result is confirmed by Fig. 15, presenting the variation of $N_u$ as function of $R_c$, for $\theta = 30^\circ$, $Ra = 10^5$, $A = 0.5$ and $\tau_p = 0.5$. Therefore if the objective is to increase the heat transfer through the fenestration, blinds with $\theta = 0^\circ$ and high thermal conduction should be used.

In Fig. 16 the effect of the variation of the period on $N_u$ is presented for $Ra = 10^5$, $A = 0.5$, $\theta = 0^\circ$ and $R_c = 100$. It is noticed that for $\tau_p = 0.5$, $N_u$ is in considerable amplitude offset regarding to other values of $\tau_p$. In this case $N_u$ has the maximal maximum and the maximal minimum.

Fig. 17 presents Variation of $N_u$ as function of $\tau_p$ for $\theta = 30^\circ$, $Ra = 5.10^5$, $A = 0.5$ and $R_c = 100$. This variation has an

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**Fig. 11.** Velocity projection in the XY-plan (a) $t = \tau_p/4$; (b) $t = \tau_p/2$; (c) $t = 3\tau_p/4$; (d) $t = 7\tau_p/8$; (e) $t = \tau_p$, for $Ra = 5.10^5$, $R_c = 100$, $A = 0.5$, $\tau_p = 0.5$ and $\theta = -30^\circ$.

**Fig. 12.** Variation of $N_u$ as function of time for different louvered blinds inclinations for $R_c = 100$, $Ra = 5.10^5$, $A = 0.5$ and $\tau_p = 0.5$. 

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Fig. 13. Variation of $N_{um}$ as function of $Ra$ for different louvered blinds inclinations for $Rc = 100$, $Ra = 5 \times 10^5$, $A = 0.5$ and $\tau_p = 0.5$.

Fig. 14. Variation of $N_{um}$ as function of time for different $Rc$ for, $Ra = 10^5$, $A = 0.5$, $\theta = 30^\circ$ and $\tau_p = 0.5$.

Fig. 15. Variation of $N_{um}$ as function of $Rc$ for $\theta = 30^\circ$, $Ra = 10^5$, $A = 0.5$ and $\tau_p = 0.5$. 
extremum for $\tau_p = 0.5$ and remains constant for the other values. This result is very interesting from a practical point of view. Indeed, if the aim is to enhance the heat transfer, the option for a geometric configuration resulting in a value of $\tau_p = 0.5$ would be beneficial.

4. Conclusion

The thermal performances of a fenestration with different internal louvered blinds inclination were investigated by means of numerical simulations of the flow, temperature fields and heat transfer. The flow is considered unsteady and periodic temperature was imposed to the hot wall. The effect of several parameters such as temperature period and amplitude, conductivities ratio and blinds inclination were studied.

The main obtained results are:

- Except near the top and bottom of the fenestration, the temperature field and the flow structure are repetitive in the $y$-direction.
- Open blinds significantly inhibits the convection.
- The temperature field presents stratification in a direction normal to blinds.
- The increase of Rayleigh number increase the 3D aspect of the flow which is complex due to the passage of particles from a vertical plane to another.
- The internal flow converges directly from the frontal wall toward the XY plan, then from this plan toward the frontal wall. The overall external flow is also convergent from the frontal walls toward the XY plan and occurs very near of active walls.
- The flow structure is periodic due to the imposed periodic boundary condition.

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Fig. 16. Variation of $\text{Nu}_{\text{in}}$ as function of time for different $\tau_p$ for $Ra=10^5$, $A=0.5$, $\theta = 30^\circ$ and $Rc=100$.

Fig. 17. Variation of $\text{Nu}_{\text{in}}$ as function of $\tau_p$ for $\theta = 30^\circ$, $Ra=10^5$, $A=0.5$ and $Rc=100$. 

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• For $\theta = 0^\circ$ the variation of $Nu_m$ is in amplitude offset compared to it variations for $\theta = 30^\circ$ and $\theta = -30^\circ$, implying a higher heat transfer rate.

• The increase of the thermal conductivity ratio induces an increase of the heat transfer.

References