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NUMERICAL INVESTIGATION OF TURBULENT INTERNAL FLOW WITH COMBINED CONVECTIVE AND RADIATIVE HEAT TRANSFER IN A PARTICIPANT MEDIUM

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Abstract. *The present work reports the influence of combined convective and thermal radiation heat transfer over the temperature field of turbulent internal flows in a participant medium. In order to achieve this purpose, two numerical simulations are performed for a turbulent lid-driven cavity flow: 1) with combined convective and thermal radiation heat transfer and 2) without thermal radiation (purely forced convective flow). For both simulations the conservation equations of mass, momentum and energy are discretized with the finite volume method (FVM). Moreover, turbulence was tackled with dynamical Smagorinsky subgrid-scale model into the large eddy simulation (LES) framework. For the case with thermal radiation heat transfer, the radiative transfer equation (RTE) is solved with the discrete ordinates method (DOM). The participant medium is treated as a gray gas with an optical thickness of* τ_0 *= 10.0. For both simulated cases, the Reynolds and Prandtl numbers are* Re_H *= 10000 and Pr = 0.71, respectively. The results revealed that three-dimensional patterns of the flow, as well as, the multiplicity of scales are strongly affected when thermal radiation is taken into account, showing the importance of thermal radiation for the phenomenology of turbulent flows with heat transfer. For the studied case, the time-averaged temperature profiles and statistics of turbulence into the cavity domain are smoothed when thermal radiation is considered. Differences of nearly 30% and 63% were noticed for time-averaged and statistics of thermal field, respectively.*

Keywords: *Thermal radiation, convective, participant medium, turbulence, LES.*

1. INTRODUCTION

Cavities represent a geometrical idealization for various engineering fluid flows. Some examples include interstitial flows between fins in heat exchangers, solar thermal applications, cooling of electronic components in an integrated board, ventilation of rooms and air flow in buildings. The main advantages for the study of cavities is its simple geometry, well posed boundary conditions and the presence of complex phenomena such as the boundary detachment and reattachment and the generation of primary and secondary vortices (Prasad and Koseff, 1989; Prasad and Koseff, 1996). As a consequence, the enclosed cavities has served as prototype for several development of numerical algorithms, examples are shown in Vahl Davis (1983), Guia *et al.* (1982), Ertuk and Gökçöl (2006) and Dos Santos *et al.* (2011).

It is worthy to mention that several numerical and experimental studies in laminar and turbulent cavity flows have been performed to achieve a better understand of the fluid dynamic behavior of isothermal flows (Guia *et al.*, 1982; Prasad and Koseff, 1989; Prasad and Koseff, 1996; Kim and Menon, 1999), as well as, the thermal behavior of forced convective (Iwatsu and Hyun, 1995; Dos Santos *et al.*, 2008; Dos Santos *et al.*, 2013), mixed convective (Dos Santos *et al.*, 2011) and free convective flows (Peng and Davidson, 2001; Trias *et al.*, 2013).

Some studies about coupled convection and thermal radiation heat transfer problems have also been reported into literature (Chang *et al.*, 1983, Ibrahim and Lemonier, 2009; Saravanan and Sivaraj, 2013). For instance, Chang *et al.* (1983) studied numerically the natural convection and thermal radiation in a square cavity at two levels of Grashoff numbers for participating and non-participating gases. Ibrahim and Lemonier (2009) studied numerically the coupling of transient convection with radiation in a square cavity filled with a participant medium (mixture of N_2 and CO_2). In this work, the temporal evolution of the flow has been computed in both aiding and opposing situations. The numerical results showed that gas radiation modifies significantly the structure of the transient velocity and thermal fields. Afterwards, Saravanan and Sivaraj (2013) reported a theoretical study to understand the interaction of surface radiation

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and natural convection in a transparent medium (air) filled cavity with a centrally placed thin heated plate for several values of Rayleigh number ($10^5 \leq \text{Ra} \leq 10^7$) and surfaces emissivity ($0 \leq \varepsilon \leq 1$). However, in all of the above mentioned studies the fluid flow is laminar and the thermal behavior (turbulent patterns, time-averaged fields and statistics of turbulence) of combined convective and radiative transfer in turbulent cavity flows has not been investigated into literature.

In this sense, the main purpose of the present work is to evaluate the influence of combined forced convective and thermal radiation heat transfer over the transient temperature field of turbulent internal flows in a participant medium. More precisely, two numerical simulations are performed for a turbulent lid-driven cavity flow: 1) with combined convective and thermal radiation heat transfer and 2) without thermal radiation (purely forced convective problem). For both cases, the Reynolds and Prandtl number are given by $\text{Re}_H = 10000$ and $\text{Pr} = 0.71$ and for the case with thermal radiation it is considered a gray gas with an optical thickness of $\tau_0 = 10.0$. The conservation equations of mass, momentum and energy are discretized with the finite volume method (FVM) (Patankar, 1980; Versteeg and Malalasekera, 2007) and turbulence is tackled by means of large eddy simulation (LES) with a dynamical Smagorinsky subgrid-scale model (Germano et al, 1991; Lilly, 1992; Lesieur et al., 2005; Sagaut, 2006). For the case with thermal radiation heat transfer, the radiative transfer equation (RTE) is solved with the discrete ordinates method (DOM). All the simulations of the present work are performed with the use of FLUENT™ CFD code (Fluent, 2007).

2. MATHEMATICAL MODELING

2.1. Large Eddy Simulation (LES)

The modeling of transient, incompressible, non-isothermal flows is based on the solution of the conservation equations of the problem together with its boundary and initial conditions. In the LES approach, the mass, momentum and energy equations are spatially filtered with a box filter (Findikakis and Street, 1982). These equations can be written as (Lesieur et al., 2005; Sagaut, 2006):

$$\frac{\partial \bar{v}_i}{\partial x_i} = 0 \qquad (i = 1, 2 \text{ and } 3) \text{ in } t \times \Omega \qquad (1)$$

$$\frac{\partial \bar{v}_i}{\partial t} + \frac{\partial (\bar{v}_i \bar{v}_j)}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \bar{p}}{\partial x_j}\delta_{ij} + \frac{\partial}{\partial x_j}\left\{ \upsilon \left(\frac{\partial \bar{v}_i}{\partial x_j} + \frac{\partial \bar{v}_j}{\partial x_i} \right) - \tau_{ij} \right\} \qquad (i, j = 1, 2 \text{ and } 3) \text{ in } t \times \Omega \qquad (2)$$

$$\frac{\partial \bar{T}}{\partial t} + \frac{\partial}{\partial x_j}(\bar{v}_j \bar{T}) = \frac{\partial}{\partial x_j}\left\{ \alpha \frac{\partial \bar{T}}{\partial x_j} - q_j \right\} - \overline{\nabla \cdot q_r} \qquad (i, j = 1, 2 \text{ and } 3) \text{ in } t \times \Omega \qquad (3)$$

where $\overline{(\ )}$ represents the large (filtered) scales; ρ is the density of the fluid (kg/m^3); υ is the kinematic viscosity (m^2/s); α is the thermal diffusivity (m^2/s); v_i is the velocity in *i*-direction, i = 1, 2 and 3 (m/s); x_i corresponds to the spatial coordinate, i = 1, 2 and 3 (m); p is the pressure (N/m^2); T is the temperature (K); δ_{ij} is the Kronecker delta, Ω is the spatial domain (m); t represents the time domain (s); $\overline{\nabla \cdot q_r}$ is the filtered divergence of the radiative transfer (W/m^3), which is inserted as a source term in the energy equation. The terms τ_{ij} and q_j that arise in the filtering process of the momentum and energy conservation equation, respectively, need to be modeled and can be written as:

$$\tau_{ij} = \overline{v_i v_j} - \bar{v}_i \bar{v}_j \qquad (4)$$

$$q_j = \overline{v_j T} - \bar{v}_j \bar{T} \qquad (5)$$

2.1.1. The dynamic Smagorinsky subgrid-scale model (DSSGS)

The DSSGS is based on the hypothesis of Boussinesq's eddy viscosity (Lesieur et al., 2005). For incompressible flows, the turbulent tensor can be written as:

$$\tau_{ij} = \upsilon_{sgs}\left(\frac{\partial \bar{v}_i}{\partial x_j} + \frac{\partial \bar{v}_j}{\partial x_i} \right) - \frac{2}{3}k\delta_{ij} \qquad (6)$$

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where υ_{sgs} is the kinematic eddy viscosity (m^2/s^1) and k is the turbulent kinetic energy (m^2/s^2). The turbulent transport of the temperature is obtained by an analogy with the subgrid Reynolds tensor (Lesieur et al., 2005; Sagaut, 2006), and is given by:

$$q_j = \alpha_{sgs} \frac{\partial \overline{T}}{\partial x_j} \tag{7}$$

in which α_{sgs} is the thermal eddy diffusivity (m²/s).

According to the model, the kinematic eddy viscosity and the thermal eddy diffusivity are given by:

$$\upsilon_{sgs} = C(x,t)^2 \overline{\Delta}^2 \left|\overline{S}\right| \tag{8}$$

$$\alpha_{sgs} = \frac{C(x,t)^2}{\mathrm{Pr}_{sgs}(x,t)} \overline{\Delta}^2 \left|\overline{S}\right| \tag{9}$$

where $\overline{\Delta}$ is the subgrid-scale characteristic length (m); $\left|\overline{S}\right|$ is the strain-rate of the filtered field (s^{-1}) and $\overline{S}_{ij}$ is the filtered-field deformation tensor (s^{-1}), which are given by:

$$\overline{\Delta} = \sqrt[3]{\prod_{i=1}^{3} \Delta x_i} \tag{10}$$

$$\left|\overline{S}\right| = \sqrt{2 \overline{S}_{ij} \overline{S}_{ij}} \tag{11}$$

$$\overline{S}_{ij} = \frac{1}{2} \left(\frac{\partial \overline{v}_i}{\partial x_j} + \frac{\partial \overline{v}_j}{\partial x_i} \right) \tag{12}$$

The Smagorinsky constant, $C(x,t)$, and the SGS turbulent Prandtl number, $\mathrm{Pr}_{sgs}(x,t)$, are dynamically computed based on the approach proposed by Germano *et al.* (1991) and modified by Lilly (1992). This modeling is based on the use of two spatial filters with different lengths, providing information on the energy transfer between the solved (obtained in the test filtering region) and not solved scales of motion (Lesieur *et al.*, 2005). More details on the SGS model can be found in the works of Germano *et al.* (1991), Lilly (1992) and Lesieur *et al.* (2005).

2.2. Radiation modeling

For the solution of the thermal radiation field it is necessary to solve the radiative transfer equation (RTE) (Siegel and Howell, 2002). The filtered radiative transfer equation is obtained after the employment of the spatial filtering operation (Jones and Paul, 2005; Gupta *et al.*, 2009). For a gray gas medium this equation can be written by:

$$\frac{d\overline{i}}{ds} = -\overline{\kappa i} + \overline{\kappa i_b} \tag{13}$$

where s is the coordinate along path of radiation (m), κ is the absorption coefficient (m^{-1}), i_b is the total blackbody intensity (W/(m$^2\cdot$sr)) and i is the total radiative intensity (W/(m$^2\cdot$sr)).

For a real gas, the behavior of the absorption coefficient as a function of wavelength is significantly more complex than the gray gas model, which assumes the absorption coefficient to be independent of the wavelength. More advanced models, such as weighted-sum-of-gray-gases (WSGG) (Smith *et al*, 1982), the spectral line based WSGG (SLW) (Denison and Webb, 1993), and cumulative wavenumber (CW) (Galarça *et al.*, 2011), are available for non-gray media. Despite its simplicity, the gas is considered gray to reduce the difficulties and high computational demands regarding the spectral effects. This same aspect leads to other simplification hypotheses, such as the treatment of density and viscosity as constants.

With the purpose to couple the thermal radiation field and the convection heat transfer in the LES framework, it is necessary to take into account the filtered divergence of the radiative flux ($\overline{\nabla \cdot q_r}$), which for a gray medium is given by:

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$$\overline{\nabla \cdot q_r} = 4\pi\overline{\kappa i_b} - \int_{4\pi} \overline{\kappa i} d\Omega \tag{14}$$

Coelho (2009) and Roger et al. (2010; 2011) observed that for the simulation of reactive turbulent flows the average errors associated with neglecting the SGS radiative absorption and emission terms are consistently small, especially for flows with low turbulence intensity. However, the assumption of neglecting the SGS-TRI should be taken carefully when the optical thickness becomes large (τ_0 ~ 100 or larger). Then, in the present study, the modeling of SGS radiative absorption and emission terms will be modeled simply by: $\overline{\kappa i} = \overline{\kappa}\,\overline{i}$, $\overline{\kappa i_b} = \overline{\kappa}\,\overline{i_b}$. This simplification was previously performed for the simulation of non-reactive and reactive flows (Gupta *et al.*, 2009).

3. NUMERICAL PROCEDURES

Concerning the simulation of turbulent flows, Eqs. (1) to (3) are solved using a CFD package based on hexahedral finite volume method (FLUENT, 2007). The solver is pressure based and all simulations reported were performed with second-order spatial (bounded central differencing) and implicit temporal discretizations. The bounded central differencing scheme employed consists of a mixture of two advection schemes: central differencing for regions where the flow is diffusive and upwind of second order for regions where advection is dominant (Gaskell and Lau, 1988; Leonard, 1991; Zhu and Rodi, 1991). The velocity-pressure coupling is performed with SIMPLE method. More details concerned with the FVM can be found in Patankar (1980) and Versteeg and Malalasekera (2007).

The radiative transfer equation, Eq. (13), is solved with the discrete ordinates method (DOM) (Siegel and Howell, 2002; Kim *et al.*, 2001). For the solution, it is employed the approximation (S_4), n = 4. For three-dimensional enclosures, the total number of different discrete directions, M, to be considered at each computational node is defined as $M = n(n+2)$. Then, in the present work the number of discrete directions for each computational node is M = 24.

The numerical simulations are performed using a computer with two dual-core Intel processors with 2.67 GHz clock and the memory amount of 8GB. For parallelization it is used the library Message Passing Interface (MPI). The time processing for the simulations combining convection and thermal radiation and only with convection are nearly 5.40×10^6s and 1.20×10^6s, respectively.

The calculations are considered converged when the residuals for the mass, momentum, energy and intensity of radiation between two consecutive iterations were less than 10^{-4}, 10^{-6}, 10^{-8} and 10^{-6}, respectively. Moreover, an under-relaxation factor of 0.7 is imposed for all conservation equations.

4. PROBLEM DESCRIPTION AND VERIFICATION OF THE NUMERICAL METHOD

The simulations performed here consider a three-dimensional, incompressible and turbulent flow with combined convective and radiative heat transfer in a participant medium. Figure 1 depicts the cavity domain with the following dimensions: $H = L$ = 1 m and W = 0.5 m, as well as, the imposed boundary conditions for the studied case. Concerning the convective problem, the flow is generated by the movement of a superior plate (plane xz), where the imposed velocity is v_1 = 100 m/s. Moreover, this surface has a non-slip and impermeability conditions. For laterals surfaces (plane yz) and for posterior surface (plane xy at z = 0 m) the velocities are prescribed null ($v_1 = v_2 = v_3$ = 0 m/s). The frontal surface (plane xy at z = 0.5 m) has a symmetry condition. Concerning the thermal field, the flow is heated by the imposition of a temperature of T_S = 1000 K at the superior surface, which is higher than those of the fluid flow and inferior surface, which is prescribed at T_I = 300 K. The lateral surfaces, frontal and posterior are treated as adiabatic. Into the thermal radiation framework, the cavity surfaces are treated as perfectly black, i.e., the total hemispheric emissivity is ε_w = 1. For the purely convective problem the same fluid dynamic and thermal conditions are imposed and thermal radiation field is not solved. The flows are considered with Re_H = 10000, Pr = 0.71 (for both cases) and τ_0 = 10.0 (for the case with thermal radiation).

Concerning the initial conditions, the fluid is considered at rest, with velocity and pressure fields null along all domain and with a stratified thermal field, i.e., the temperature varies linearly from T_I = 300 K at the lower surface of the cavity until T_S = 1000 K at its upper surface.

For all simulations, the cavity domain is discretized with 100 × 100 × 30 finite volumes in x, y and z directions, respectively. Moreover, the grid is more refined near the cavity surfaces than in the center of cavity and the most refined volume has a minimal cut-off wavenumber of $k_{c,\min} = \pi/\Delta x_{\min}$ = 800 m^{-1}. For the transient problem it is employed a time-step of $\Delta t = 1.0 \times 10^{-4}$ s.

In order to validate the numerical method (FVM and LES) for simulation of turbulent flows, the time-averaged velocity field and the statistics of turbulence for velocity field when the flow reaches the steady state are compared with experimental results of Prasad and Koseff (1989). Once this case has been studied solely in the isothermal framework, only results for velocity fields are presented here.

Figures 2(a) and 2(b) show the comparison between the time-averaged velocity profiles and RMS (Root Mean Square) obtained in the cavity center with the present method and those of Prasad and Koseff (1989). The simulations

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are performed until a physical time of t = 4.5 s. From this interval of time, the last t = 2.0 s are considered for achievement of statistics of turbulence. Moreover, the RMS velocities are defined by:

$$v_{i,rms} = 10\sqrt{\frac{\overline{v_i'^2}}{v_{i\,\max}^2}} \tag{15}$$

Figure 2(a) shows that time averaged velocity profiles, $v^*_1(y^*)$ and $v^*_2(x)$, obtained numerically are in excellent agreement with those predicted experimentally, except by slight differences for profile $v^*_2(x^*)$ in the range $0.75 \le x^* \le 0.90$. The mean difference for the profile $v^*_2(x^*)$ is lower than 5.0 %. Figure 2(b) exhibits the RMS velocity profiles, $v^*_{1,\mathrm{rms}}$ (y^*) and $v^*_{2,\mathrm{rms}}$ (x^*). The numerical results are also in very good agreement with those obtained by Prasad and Koseff (1989). In the worst situation, the mean deviation for profile $v^*_{1,\mathrm{rms}}(y^*)$ reached by the present numerical method and the experimental results are lower than 10.0 %. Here v^* represents the ratio between $v_i/v_{\max}$ and x^* and y^* represents the ratio between x/H and y/H, respectively.

For the thermal field, the method used here was previously validated for the simulation of temperature field in laminar and turbulent cavity flows (Dos Santos et al., 2008; Dos Santos et al., 2013) and for the sake of brevity will not be re-exhibited here.

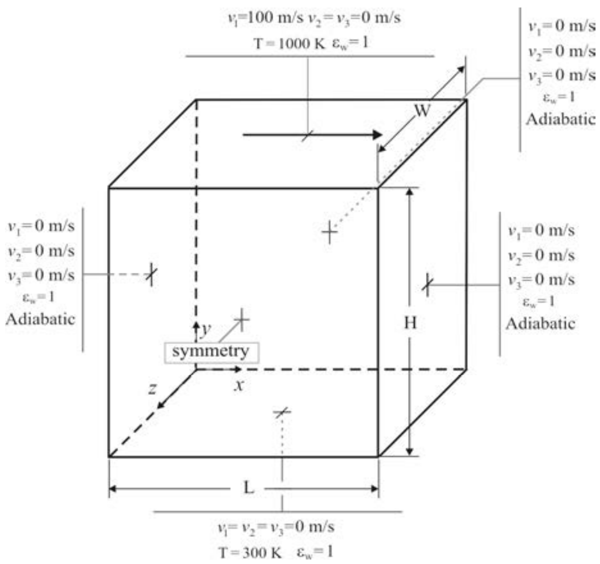
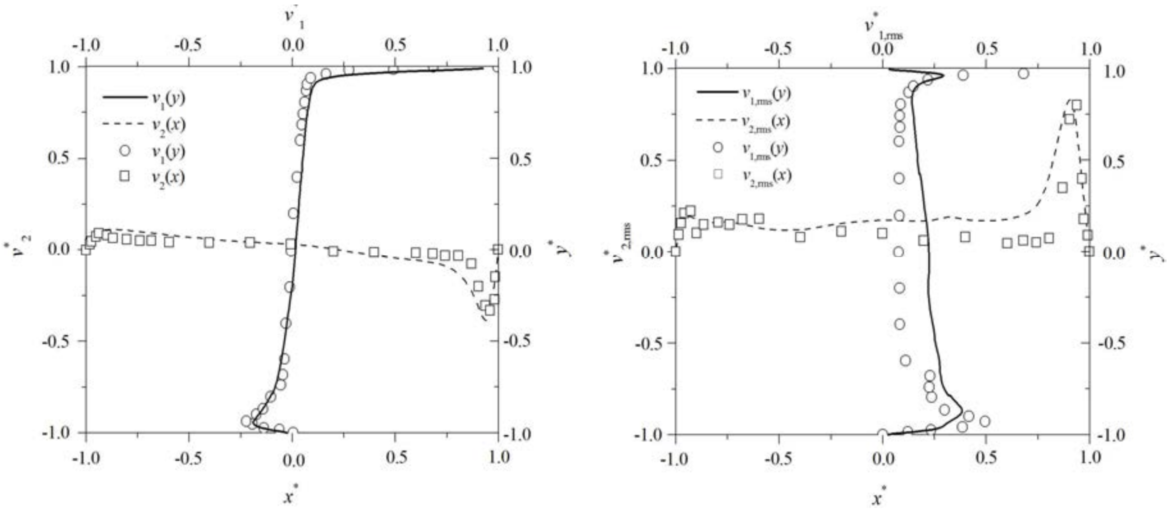


Figure 1. Computational Domain for the flow with Re_H = 10,000, Pr = 0.71 and τ_0 = 10.0.



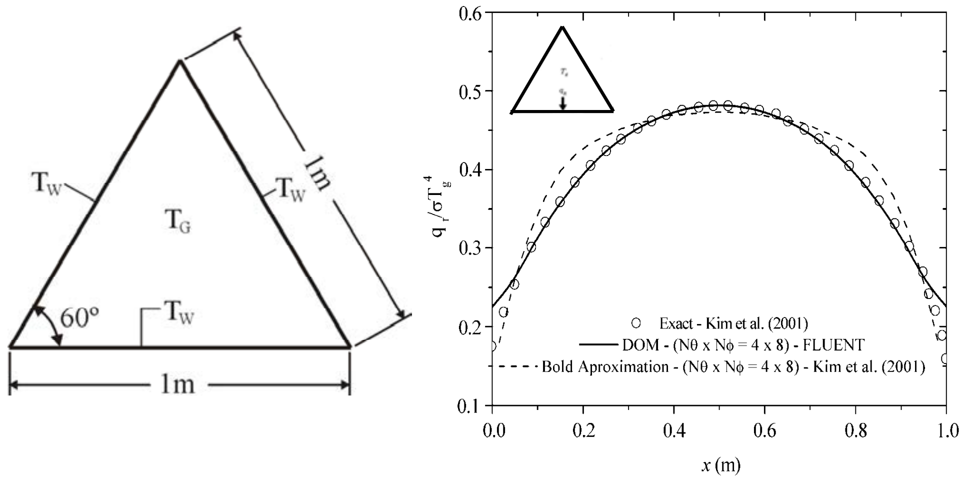
a)

b)

Figure 2. Dimensionless velocity profiles in the cavity center for isothermal flow with Re_H = 10000: a) time-averaged profiles, b) statistics of turbulence. Results of Prasad and Koseff (1989): ○ $v^*_1(y^*)$, $v^*_{1,\mathrm{rms}}(y^*)$; □ $v_2(x^*)$, $v^*_{2,\mathrm{rms}}(x^*)$. Presente trabalho: − $v^*_1(y^*)$, $v^*_{1,\mathrm{rms}}(y^*)$; − − $v^*_2(x^*)$, $v^*_{2,\mathrm{rms}}(x^*)$.

In the absence of available exact results for the three-dimensional rectangular configuration that will be studied here, the present methodology to compute the radiative heat transfer in participant media is evaluated for an equilateral triangular enclosure with an absorbing and emitting, but non-scattering, medium. For this simulation the gas is maintained at constant temperature of T_G = 1000 K and the walls are held cold (T_W = 0 K). The enclosure surfaces of length L = 1 m are treated as perfectly black, ε_w = 1, Fig. 3(a). The medium is treated as a gray gas with an absorption coefficient of κ = 1.0 m^{-1}. The results obtained here are compared with the exact and numerical solution presented in Kim et al. (2001). The spatial independent grid employed in the present simulation consists of 290 quadrilateral cells with an angular discretization of $N_\theta \times N_\phi = 4 \times 8$.

Figure 3(b) presents the dimensionless incident radiative flux as a function of the dimensionless coordinate ($x^* = x/L$) on the bottom wall. The result achieved with the present solution is in close agreement with the exact solution, except in the corners regions of the triangular enclosure ($x^* \leq 0.05$ and $x^* \geq 0.95$) where differences of nearly 12 % are noticed. In spite of this, the average difference was lower than 3%.



a) b)

Figure 3. Radiative heat transfer in a triangular enclosure with participant medium: (a) domain; (b) dimensionless radiative heat flux in the lower surface.

5. RESULTS AND DISCUSSION

Firstly, it is verified the transient behavior of the radiative heat flux for the simulation of combined convective and radiation transfer. It is believed that the fluctuations of instantaneous radiative flux behave similarly to the fluctuations of temperature field in convective turbulent flows. In order to perform this evaluation, the spatial-averaged radiative fluxes in the lower and upper surfaces, q_r (W·m^{-2}), are depicted in Fig. 4. As expected, the results illustrate a strong influence of temperature field fluctuations over the radiative fluxes in the cavity surfaces. The fluctuations of radiative fluxes are nearly 10.0 % and 3.0 % for the inferior and superior surfaces, respectively. It is also noticed that the time-averaged radiative flux is approximately two times higher than that noticed for the lower surface, showing the large amount of radiative energy which is absorbed by the participant medium.

The next step consists on the investigation of the influence of the radiative transfer in participant medium over the transient thermal field. Firstly, it is performed a qualitative evaluation. In this sense, the transient temperature topologies obtained for the case with combined convective and radiative and for purely forced convective problem are compared (Fig. 5). For the quantitative evaluation, the transient temperature field as function of time is monitored for two points of cavity domain: point 1 (x = 0.5 m, y = 0.27 m and z = 0.5 m) and point 2 (x = 0.5 m, y = 0.94 m and z = 0.5 m) as depicted in Fig. 6. These placements allow the observation of the transient behavior in regions near lower and upper surfaces.

Figure 5 exhibits qualitatively the temporal evolution of temperature field for the flows with and without thermal radiation in a participant medium. Figures 5(a – d) present the temperature topologies for the following instants of time: t = 0.1 s, t = 0.2 s, t = 0.4 s and t = 1.0 s and for the flow with forced convection, while Figs. 5(e – h) show the topologies for the same instants of time and for the case with thermal radiation. Figures 5(a) and 5(e) show the initial formation of the main vortex near the right and upper corner of the cavity. For this instant of time, the temperature field does not seem affected by the presence of the radiative transfer. In Figures 5(b) and 5(f), t = 0.2 s, it is still possible to observe first differences between the temperature fields obtained with and without thermal radiation. For the simulation with thermal radiation, three-dimensional patterns, as well as, secondary flows near the lower right corner are strongly smoothed in comparison with the simulation with forced convective heat transfer. The presence of thermal radiation for

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a participant medium with τ_0 = 10.0 also affects the development of the main vortex of the flow. Afterwards, the effect of thermal radiation over the temperature patterns is even more evident, especially near the right-hand side surface, where structures like *Tollmien-Schilichting*, vortices of *Taylor-Göertler* and turbulent streaks are more intense for the forced convective flow. As the turbulent flow tends to the steady state, Figs. 5(d) and 5(h), it is noticed that thermal radiation suppressed the multiplicity of scales. Moreover, the thermal field also is stabilized when thermal radiation is taken into account. In other words, the phenomenology of turbulent thermal field is strongly modified for the combined convective and radiative heat transfer in comparison with purely forced convective flow. This behavior can influence the characterization of turbulent flow, e.g., transition of the flow from laminar to turbulent regime.

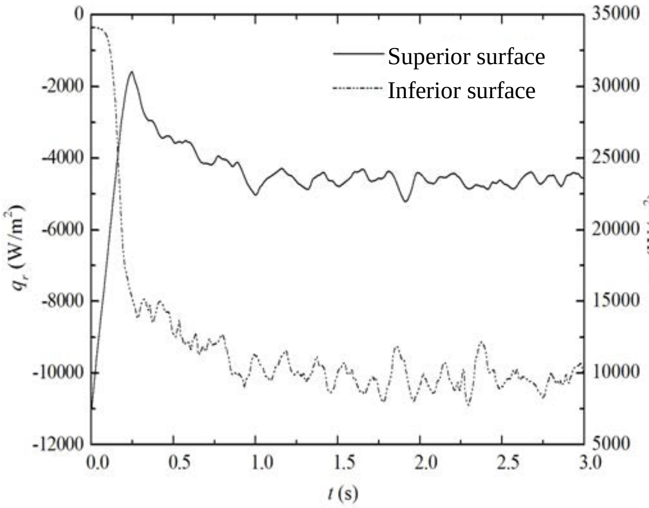

Figure 3. Radiative fluxes in the upper and lower surfaces for the combined convective and radiative turbulent cavity flow with Re_H = 10000, Pr = 0.71 and τ_0 = 10.0.

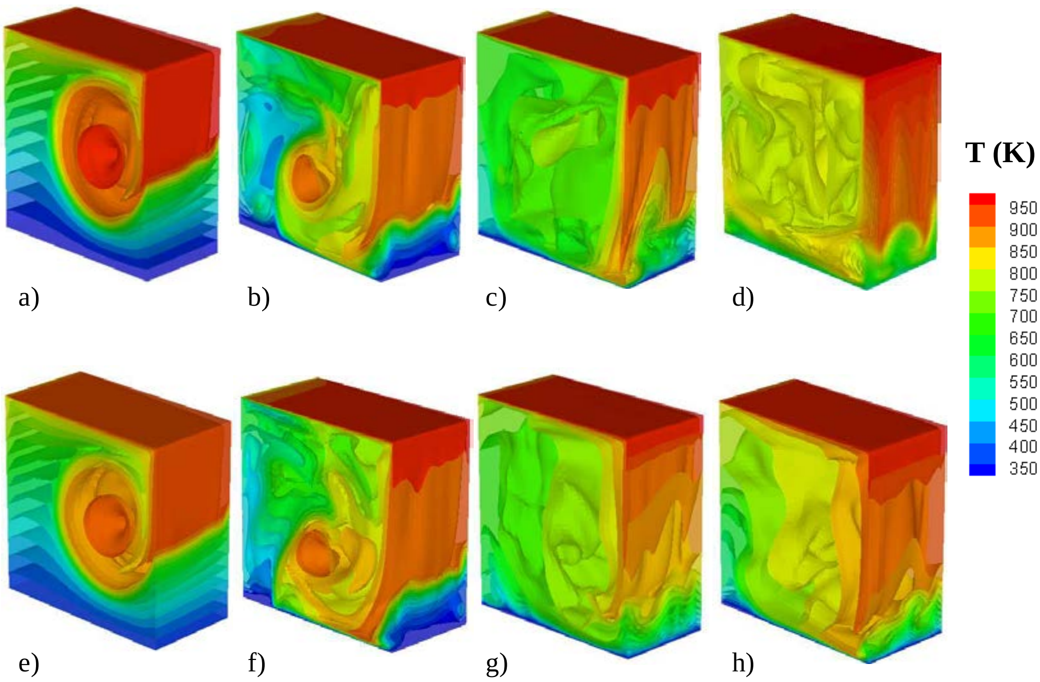

Figure 4. Turbulent thermal Field (K) as function of time for the flow with Re_H = 10000 and Pr = 0.71. Forced convective: (a) t = 0.10 s, (b) t = 0.20 s, (c) t = 0.40 s, (d) t = 1.00 s. With thermal radiation (τ_0 = 10.0): (e) t = 0.10 s, (f) t = 0.20 s, (g) t = 0.40 s, (h) t = 1.00 s.

In order to investigate quantitatively the influence of thermal radiation over the transient thermal field, Fig. 6(a) and 6(b) show a comparison of the evolution of temperature field for the simulations with and without thermal radiation as a function of time for the two monitoring points described before: point 1 and point 2. In general, the qualitative behavior of transient temperature is similar for both simulations. In Figure 6(a) it is noticed that in the lower region of the cavity

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between the time-averaged temperature predicted with and without thermal radiation is nearly ΔT = 65 K, i.e., the time-averaged temperature for the simulation with thermal radiation is approximately 9.0 % higher than that reached for the purely convective flow. Concerning the fluctuations of temperature field, for the case evaluated here, the fluctuations for the forced convective flow is 8.8 % of the time-averaged magnitude of temperature while for the simulation with thermal radiation, this percentage decreases to only 4.4 %. For point 2, Fig. 6(b), the difference between the time-averaged temperatures is even higher: ΔT = 100 K (13.5 %). Moreover, the fluctuations of temperature field in point 2 are also less intensive for the case with thermal radiation. The magnitudes of fluctuations are 13.7 % and 9.8 % for the simulations without and with thermal radiation. The behavior obtained in the present simulations for the turbulent flow is similar to that one noticed by Ibrahim and Lemonnier (2009) for the simulation of laminar cavity flows with stable stratification.

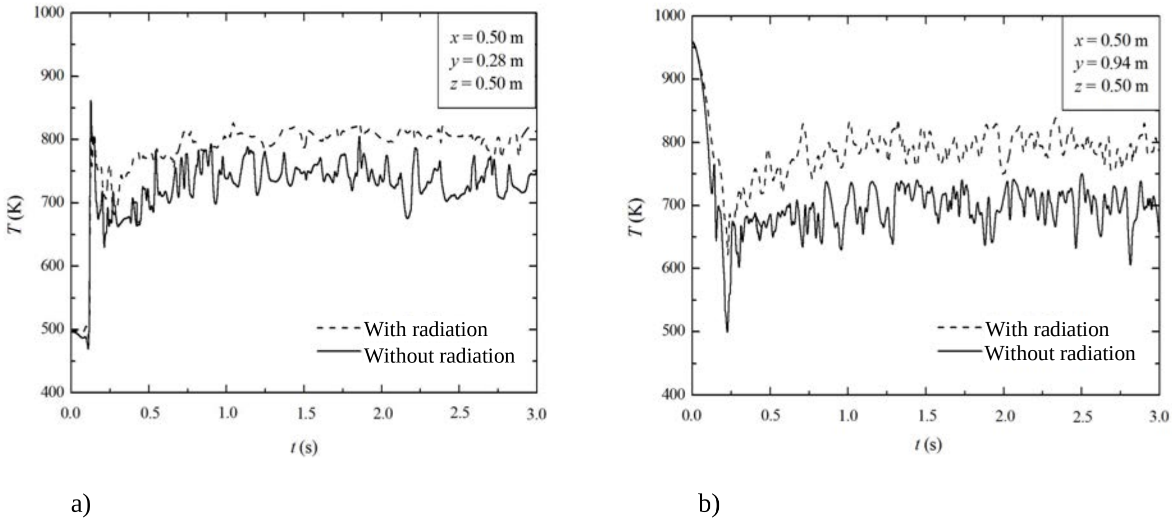

Figure 5. Comparison between transient temperature fields with and without thermal radiation for the flow with Re_H = 10000, Pr = 0.71 and τ_0 = 10.0 (for the case with thermal radiation) obtained in the following monitoring points: (a) point 1, (b) point 2.

In order to investigate the effect of thermal radiation over the thermal field when the flow reaches the steady state, dimensionless time-averaged temperature profiles in the center of cavity ($T^*(y^*)$ at x^* = 0.5 and $T^*(x^*)$ at y^* = 0.5) are obtained with thermal radiation and with forced convective heat transfer. As performed for the validation of the method for simulation of isothermal turbulent flow, the last t = 2.0 s are considered to obtain the statistics of turbulence.

The dimensionless temperature presented here is given by:

$$T^* = \frac{T - T_I}{T_S - T_I} \tag{16}$$

Figure 6 depicts the dimensionless temperature profiles at the cavity center. Both vertical and horizontal profiles, $T^*(y^*)$ e $T^*(x^*)$, have a higher magnitude of temperature for the simulation with thermal radiation heat transfer than that obtained for forced convective flow. The highest differences for the vertical and horizontal profiles are 32.3 % and 29.7 %, respectively. Other important observation is that thermal radiation not only affects the magnitude of temperature field but also the behavior of profiles in the cavity center. This fact is more evident for the profile $T^*(x^*)$ where the temperature field predicted with thermal radiation varied almost linearly for $x^* \geq 0.25$. Meanwhile, for the simulation with forced convection the temperature profile is almost constant in the central region of the cavity, $0.15 \leq x^* \leq 0.85$, with a steep increase of temperature near the right-hand side surface. Concerning the temperature gradient near the cavity surfaces, in general, they are smoothed when thermal radiation is taken into account, probably due to part of energy be transferred by thermal radiation. The only exception occurred near the lower surface, where the temperature gradient is slightly increased for the case with thermal radiation. This fact can be related with the radiative energy which is emitted from the hottest regions of the cavity towards the lower region of the cavity, causing a general augmentation of temperature in this region.

Figure 7 shows a comparison between the statistics of turbulence for the temperature field obtained with and without thermal radiation. Figure 7(a) shows the RMS temperature profile (T_{rms}) as a function of y^* at x^* = 0.5 and Fig. 7(b) presents T_{rms} as a function of x^* for y^* = 0.5. The results revealed that the mean magnitude decreases for both profiles of T_{rms} for the case with thermal radiation, corroborating the previous qualitative observations obtained with the temperature topologies, i.e., the multiple scales of turbulence are smoothed when thermal radiation is taken into account

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for this case (with a participant medium of τ_0 = 10.0). The highest differences reached for the profiles $T_{rms}(y^*)$ and $T_{rms}(x^*)$ for the simulations with and without thermal radiation are 63.0 % and 60.0 %, respectively.

In general, the results revealed that the no consideration of thermal radiation can led to significant deviations for prediction of temperature field of turbulent flows in participant media. Moreover, the magnitude and the behavior of time-averaged and statistics of temperature field are strongly affected by the presence of thermal radiation. It is also important to mention that the phenomenology of temperature field of turbulent flows was also influenced by the radiative field.

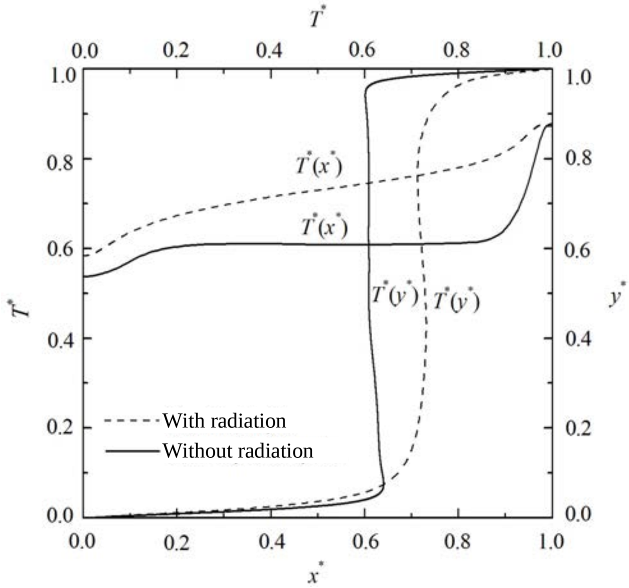
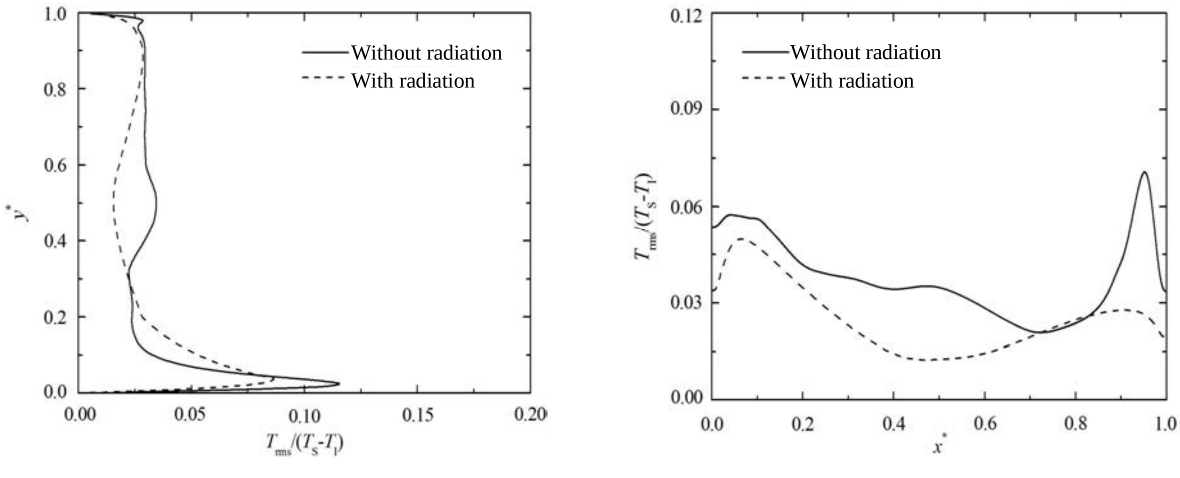


Figure 6. Comparison between the time-averaged temperature fields in the cavity center obtained with forced convective flow (Re_H = 10000 and Pr = 0.71) and with combined convective and thermal radiation (Re_H = 10000, Pr = 0.71 and τ_0 = 10.0).



a) b)

Figure 7. Comparison between the statistics of turbulence for the thermal Field in the cavity center obtained with and without thermal radiation for a flow with Re_H = 10000, Pr = 0.71 and τ_0 = 10.0 (for the case with radiation): (a) T_{rms} (y^*), (b) T_{rms} (x^*).

6. CONCLUSIONS

In the present work it was evaluated the influence of combined convective and thermal radiation heat transfer over the transient temperature field of turbulent internal flows in a participant medium. A lid-driven cavity problem was numerically simulated with combined convective and radiative heat transfer and without thermal radiation (purely forced convective flow). The conservation equations of mass, momentum and energy were discretized with the finite volume method (FVM) and turbulence was tackled with dynamical Smagorinsky subgrid-scale model into the LES framework. For both cases, the Reynolds and Prandtl numbers were Re_H = 10000 and Pr = 0.71, respectively. For the case with thermal radiation, the radiative transfer equation (RTE) was solved with the discrete ordinates method (DOM) and the participant medium was treated as a gray gas (τ_0 = 10.0).

The results showed that transient three-dimensional patterns of thermal field, such as *Tollmien-Schilichting* waves, turbulent streaks and *Taylor-Göertler* vortices, were smoothed when thermal radiation was taken into account for the flow with τ_0 = 10.0, i.e., thermal radiation affects the phenomenology of turbulent flow. This behavior can be important for characterization of turbulent flow, e.g., in definition of transition point from laminar to turbulent regime. The quantitative results for transient temperature field, for time-averaged temperature profiles and statistics of thermal field into the cavity domain corroborated the qualitative observations, i.e., the temperature field was stabilized and the multiplicity of scales was reduced for the simulation with thermal radiation heat transfer. Moreover, the results revealed that the no consideration of thermal radiation can led to significant deviations for prediction of temperature field of turbulent flows in participant media. For the studied case, differences of nearly 30% and 63% were noticed for time-averaged and statistics of thermal field, respectively.

In future studies, it is intended to evaluate the influence of combined convective and radiative heat transfer over the thermal field of turbulent cavity flows for other optical thickness.

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