NUMERICAL ASSESSMENT OF THE EFFECT OF INFLOW TURBULATORS ON THE THERMAL BEHAVIOR OF A COMBUSTION CHAMBER

by

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This work numerically investigates the effects of turbulators at the air and fuel (methane) inlets on the thermal behavior of a combustion chamber. Conservation equations for mass, momentum, energy, gaseous chemical species, soot, and temperature fluctuation variance in cylindrical axysimmetric co-ordinates were solved using the finite volume method. Chemical reaction rates were computed through the Arrhenius-Magnussen model, with two-step combustion reaction. The turbulence closure model, to compute the turbulent viscosity, was the standard k-ε. The modelling of turbulence-radiation interactions considered the absorption coefficient-temperature correlation and the temperature self-correlation. The radiative heat source was calculated using the discrete ordinates method, considering the weighted-sum-of-gray-gases model with the superposition method to compute the radiation from the gaseous species and soot. The effect of inlet turbulators was studied by varying the turbulence intensity (TI) of both inlet streams (air and fuel), encompassing mild to severe turbulators (TI = 3%, 6%, 15%, and 20%). The results showed that temperature and radiative heat source fields, and heat transfer rates on the chamber wall and radiative fraction were importantly affected by the different turbulators intensities (e. g. radiative fraction was increased from 20.6% to 32.8% when the TI was varied from 3% to 20%). Comparisons of results obtained when turbulence-radiation interactions modelling was neglected in relation to results obtained when turbulence-radiation interactions modelling was computed showed that turbulence-radiation interactions influenced the thermal field (temperature and radiative exchange) in a similar way independently of the turbulator intensity (e. g. radiative fraction decreased 20% when turbulence-radiation interactions modelling was neglected, for both turbulators intensities).

Key words: *turbulators, radiation, turbulence-radiation interactions*

Introduction

In turbulent flows the distributions of velocity, temperature and concentrations of species vary over time, and turbulent fluctuations can be very significant. The fluctuations are due to vortices generated by shear stresses in the flow. Turbulence causes an increase in the mixing rate, a marked increase in the rate of reactants consumption, thus increasing the rate of release of chemical energy and the power of a given equipment is much higher than that would be in a laminar flow [1].

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Numerical analysis is a valuable tool to predict and analyse the behavior of flows, allowing to reduce costs and time of projects, as well as testing a greater range of design possibilities. A turbulence promoter can be any kind of body placed at the inlet stream (with shape varying from a simple pin, a wire, a rod, or even a winglet). According to [2], the effect of a turbulator placed at the inlet stream is to enhance the inflow turbulence characteristics, while in a simulation the turbulator can be replaced by its side effect, *i. e.*, the related TI. This means that the variation of different turbulators can be numerically studied by varying the TI at the respective inflow where the turbulator would be installed.

A literature review revealed that there are few works related to the study of TI effects on combustion systems. Saqr *et al.* [3] computed the effect of increasing air-flow turbulence on NO and soot formation in a cylindrical combustion chamber with a methane diffusion flame. Ilbas *et al.* [4] analysed how changing the fuel and air intake angles in a combustor would affect the combustion characteristics of a synthesis gas. Darbandi and Ghafourizadeh [2] studied the effects of turbulators on the air and fuel input currents on the behavior of a jet fuel combustor (JP), as well as on the emission of pollutants. Sadiki *et al.* [5] analysed the effects of TI variations on turbulent droplet dispersion, vaporization and mixing for non-reactive sprays, and also investigated the effects of turbulence modulation coupling with external parameters such as vortex intensity.

The present numerical work has as main objective to evaluate the effects of turbulators in the air and fuel (methane) inlet currents of a combustion chamber, taking into account the radiative heat transfer of the gaseous species and soot, as well as the effects of turbulence-radiation interactions (TRI). This study considered the effects of varying inflow turbulence intensities as a mean to verify the behavior of different turbulence promoters placed at the chamber inlet. These turbulators were considered to produce turbulence intensities of 3%, 6%, 15%, and 20% in both air and fuel inlet streams. As described in the literature review [2-5], there are no studies that address the effects of turbulence promoters on the radiative heat transfer in combustion chambers, and the present research aims to contribute significantly and objectively to the literature in this important area of knowledge.

Problem statement

The combustion chamber under study consists of the natural gas-fueled one as reported in [6]. This physical system was a good choice for being the base-scenario for the current investigation since it involves a turbulent flame, so turbulence-radiation effects are present, as well as a highly non-isothermal, non-homogeneous participating medium. In addition, several experimental data for temperature and species concentrations profiles along axial and radial coordinates were presented in [6], in addition to the numerical results provided in [7-11], which are based on the same geometry and boundary conditions of the experiment in [6] and report studies about combustion and radiation modelling for that chamber.

The geometric parameters, boundary conditions, and fuel/air compositions were set:

- The cylindrical chamber had 1.7 m length and 0.5 m diameter, as shown in fig. 1.
- A duct aligned with the chamber centerline was employed to inject natural gas into the domain, so the burner/flame did not present swirl.
- The inlet boundary condition of air and natural gas modelled the burner streams, prescribing a fuel excess of 5% (fuel mass-flow rate of 0.01453 kg/s at a temperature of 313.15 K, and air mass-flow rate of 0.1988 kg/s, at a temperature of 323.15 K).

- The cylindrical duct in which the fuel enters the chamber had 0.06 m diameter, and an annular duct having a spacing of 0.02 m, outside to the fuel inlet duct, was used to inject air into the chamber.
- Considering the geometric configurations of inlet ducts and mass-flow rates described, the fuel velocity was 7.23 m/s and the air velocity was 36.29 m/s. Then the Reynolds number at the entrance was approximately $1.8 \cdot 10^4$, classifying the flow as turbulent.
- The inlet air composition was (in mass fraction): 23% oxygen, 76% nitrogen, and 1% water vapor. The fuel was considered to be composed as: 90% methane and 10% nitrogen. Taking into account the fuel composition and its mass-flow rate at the entrance, the burner power was about 600 kW. Inlet profiles of velocity and mass fractions were assumed uniform in the axial direction.
- The operational pressure of the combustion chamber was taken as p = 101325 Pa.
- For the numerical analysis, buoyancy effects could be neglected since the burner provided high velocity streams of air and fuel.
- It was considered symmetry at the chamber centerline (radial velocity and velocity gradient are null).
- It was considered prescribed temperature on the walls (T_w = 393.15 K). Regarding the chamber walls, it was also assumed impermeability and non-slip conditions.
- The outlet condition assumed: null diffusive fluxes for all variables, null radial velocity, and the axial velocity component was numerically corrected to satisfy mass conservation.
- Concerning the radiative boundary condition: chamber walls, inlet, and outlet ducts were modelled as black surfaces. Temperatures of the fuel and oxidant at the inlet were imposed to the inlet ducts, while an outlet flow bulk temperature was imposed to the outlet duct.

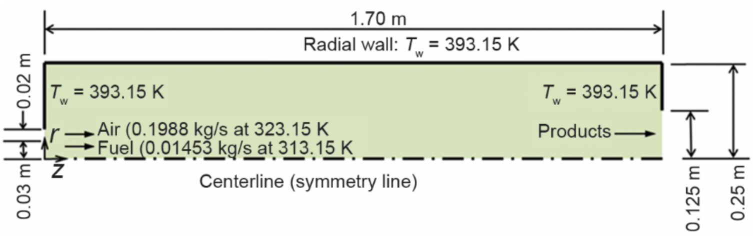

Figure 1. Combustion chamber geometry

Turbulent boundary conditions at the chamber inlet were computed as: turbulent kinetic energy was prescribed as $k = 3/2(u_{in}\mathrm{TI})^2$, where TI is the turbulence intensity (6% for the oxidant stream and 10% for the fuel stream – these values were used only at the validation step, when comparisons were made between the numerical solution and experimental data in fig. 2(a), while to comply with the aims of the present study, TI needed to be modified) and u_{in} was the inlet axial mean velocity, and turbulent kinetic energy dissipation rate was prescribed as $\varepsilon = (C_\mu^{3/4} k^{3/2})/l$, where $C\mu$ = 0.09, and l is the turbulence characteristic length scale (0.04 m for the oxidant stream and 0.03 m for the fuel stream).

Since the aim of this work is to understand the effects of different inlet turbulence promoters on the thermal behavior of the combustion chamber, while those turbulators are numerically studied by varying the TI at the respective inflow where the turbulator would be installed, at this point it is interesting to quote the role of the TI on the mathematical modelling: in RANS methodology, as in the present study, the turbulence closure is made by means of an enhanced viscosity, known as the turbulent viscosity, μ_t. The turbulent viscosity is a function of the turbulent kinetic energy, k, and the turbulent kinetic energy dissipation, ε, *i. e.*, $\mu_t = C_\mu \rho k^2/\varepsilon$, and the boundary condition for the computation of k is a function of TI at the inflow stream, as described in the previous paragraph. The turbulent viscosity, μ_t, is added to the molecular viscosity, μ, in each fundamental equation (mass, energy, momentum, and chemical species), meaning that there will be an enhancement in the turbulent mixing due to an increase on TI at the chamber entrance.

Standard wall functions were employed for the treatment of chamber walls for both energy and momentum conservation equations [12].

Mathematical formulation

The present investigation considers a calculation of a turbulent, non-premixed, methane-air flame in a cylindrical chamber aiming to study the influence of different inflow turbulators on the process, mainly on the radiative heat exchange.

The conservation equations for mass, momentum in the axial and radial directions, k-ε turbulence closure model, energy, and chemical species mass fractions for steady low-Mach flow in 2-D axisymmetric co-ordinates were solved using the finite volume method by means of a Fortran code. Detailed description of the fundamental equations is found in [7, 11].

It was considered that the gas-phase combustion process occurs at finite rates with methane oxidation taking two global steps, where the rate of formation or consumption of each species was obtained by the Arrhenius-Magnussen's model [13-15]. The soot kinetics employed here was based on the solution of a transport equation for soot mass fraction together with equations for soot formation and oxidation rates according to [15, 16], respectively. This gas-soot kinetics formulation was also successfully employed in [7, 9, 10], where additional details about its implementation are available.

Radiation heat transfer in participating media is governed by the radiative transfer equation (RTE), which establishes a relation for the variation of the spectral radiation intensity along a certain path in the medium.

The solution of the radiative transfer requires the angular and spectral integrations of the RTE. In this study, the integration of the RTE over the wavenumber spectrum was carried out with the WSGG model based on the superposition of the correlations for H_2O/CO_2 mixtures and soot [17, 18], for a partial pressure ratio of $p_{H_2O}/p_{CO_2} = 2$, which is the ratio typically considered for methane combustion. This formulation was previously tested in [10], where it was shown that this formulation provides very good agreement with the most accurate solution of the RTE, obtained with the LBL integration, providing maximum and average deviations of 4.8% and 1.2%, respectively, between the reference solution (LBL integration) and the weighted-sum-of-gray-gases (WSGG) model.

The RTE for non-scattering media, with the WSGG model, is given by eq. (1), where $\kappa_{m,j}$ is the local absorption coefficient of the participant mixture and $a_{m,j}(T)$ is the blackbody emission factor at the local temperature, T. These two terms were computed with the superposition WSGG model based on the combination of correlations obtained for the individual components of the mixture [18]. According to this method, the absorption coefficient and the emission factor of the mixture, $\kappa_{m,j}$ and $a_{m,j}(T)$, can be obtained by eqs. (2) and (3).

$$\frac{\mathrm{d}I_j}{\mathrm{d}s} = -\kappa_{m,j} I_j + \kappa_{m,j} a_{m,j}(T) I_b(T) \tag{1}$$

$$\kappa_{m,j} = \kappa_{\mathrm{wc},j_{\mathrm{wc}}} + \kappa_{s,j_s} \tag{2}$$

$$a_{m,j}(T) = a_{\mathrm{wc},j_{\mathrm{wc}}}(T)a_{s,j_s}(T) \tag{3}$$

where $\kappa_{\mathrm{wc},j\mathrm{wc}}$ is the local absorption coefficient of the j_{wc}-th gray gas of the H_2O/CO_2 mixture (treated as a single species), $\kappa_{s,js}$ – the local absorption coefficient of the j_s-th gray gas of soot, $a_{\mathrm{wc},j\mathrm{wc}}(T)$ and $a_{s,js}(T)$ – the emission factors for H_2O/CO_2 mixture and soot, respectively, both at the local medium temperature, T. The WSGG parameters for the H_2O/CO_2 mixture, $\kappa_{\mathrm{wc},j\mathrm{wc}}$ and $a_{\mathrm{wc},j\mathrm{wc}}(T)$, were taken from [17], while the parameters for soot, $\kappa_{s,js}$ and $a_{s,js}(T)$, were taken from [18].

The solution of the RTE also requires its angular integration. The angular integration of the radiative transfer equation was accomplished with the discrete ordinates method with S_6 quadrature, employing the diamond discretization scheme to express unknown intensities in terms of upstream values [19]. Other studies have shown that the use of S_6 quadrature can lead to satisfactory results when compared with S_8 nd S_{14} quadratures for cylindrical enclosures [20-22]. Once the RTE was solved, the radiative heat source, S_{RAD}, which is required to solve the energy equation in the participating medium, was calculated as the divergence of the radiative heat flux. More information about the RTE solution with the superposition-WSGG model and DOM method is available in [10].

The temperature and concentration fields of species may present large fluctuations in turbulent reactive flows, generating variations in the radiative field; the so-called TRI deals with such phenomena. Decomposition of variables (temperature and species concentrations) into mean and fluctuating components followed by time-averaging reveals several terms which require modelling [23]: temperature self-correlation, absorption coefficient self-correlation, absorption coefficient-temperature correlation, and absorption coefficient-radiation intensity correlation. The TRI modelling is an ongoing research field with many interesting papers available in the literature, *e. g.* [24, 25].

The absorption coefficient-radiation intensity correlation, expressed as $\overline{\kappa I} = \bar{\kappa}\bar{I} + \overline{\kappa' I'}$, is generally computed as $\overline{\kappa I} = \bar{\kappa}\bar{I}$ based on the arguments of Kabashnikov and Kmit [26], known as the optically thin fluctuation approximation. This approximation relies on the assumption that the absorption coefficient fluctuations, κ', are weakly correlated with the radiation intensity fluctuations, I', *i. e.*, $\overline{\kappa' I'} \approx 0$, if the mean free path for radiation is much larger than turbulence integral length scale. This assumption was adopted here and in many other studies dealing with TRI based on the time-averaged form of the RTE [23, 27, 28]. Additionally, in the present study it was applied the approximation proposed in [23, 7], in which both absorption coefficient-temperature correlation and temperature self-correlation are considered. These two TRI correlations have been found to be the most important in reactive flows [27-30]. In order to implement this TRI formulation, an additional transport equation for temperature fluctuation variance was solved. The complete set of equations used for incorporating TRI effects into the present simulations can be found in [7, 10, 23].

Numerical schemes and grid quality analysis

The set of equations were solved using the finite volume method [12] by means of a Fortran code. The power-law was applied as the diffusive-advective interpolation function on the faces of the control volumes. The pressure-velocity coupling was made by the SIMPLE method [12]. The resulting system of algebraic equations was solved by the TDMA algorithm, with block correction in all equations except the equations for k and ε. Convergence criteria were based on the imposition that the normalized residual mass in the SIMPLE method was 10^{-8}. For the other equations the maximum relative variation between iterations was 10^{-6}. The complete set of equations (mass, energy, momentum, chemical species, turbulent kinetic energy, turbulent kinetic energy dissipation, and temperature fluctuation variance) was solved with the appropriate models for turbulence, combustion and radiation (and TRI) for closure of source terms present in those governing equations. That solution provided fields of velocity, turbulent kinetic energy, turbulent kinetic energy dissipation, temperature fluctuation variance, temperature and chemical species (being CO_2, H_2O and soot the most representative for the present study, due to the radiative heat exchange), and also heat transfer fields (radiative heat source field) and heat fluxes (radiative and convective heat fluxes).

The grid convergence index (GCI), first proposed in [31] and updated in [32], was employed in this study. The GCI of the fine grid is defined as $GCI = F_S|\varepsilon|/(r^p - 1)$, where F_S is a safety factor (it is recommended in [31, 32] that the safety factor be $F_s = 3.0$ for comparisons of two meshes), r – the mesh refinement factor ($r = 4$ in the present study), and p – the order of convergence ($p = 1$ in the present study). The relative error, ε, is defined as $\varepsilon = (\psi_2 - \psi_1)/\psi_1$, being ψ a selected output of the numerical solution (subscript 1 refers to the fine mesh and 2 refers to the coarse mesh).

Two meshes were tested in this work, with 48 × 140 volumes (fine mesh) and 12 × 35 volumes (coarse mesh); both meshes were uniformly spaced. Three parameters ψ were selected to evaluate the mesh quality: radiative fraction, f_{rad}, maximum temperature, T_{max}, and maximum mole fraction of H_2O ($x_{H_2O,max}$). The results of the GCI for each of the three selected parameters are: GCI(f_{rad}) = 3.4%, GCI(T_{max}) = 0.18%, GCI($x_{H_2O,max}$) = 0.25%, where it is noted that the highest GCI was obtained for the radiative fraction, f_{rad}, as 3.4%, which is a relatively low GCI value (since the GCI can be interpreted as the difference between the present result and the result that would be obtained with an infinitely refined mesh). In this manner, the mesh with 48 × 140 volumes was considered appropriate to be used in the present work.

The results of the present simulations were compared to experimental data from [6] and it was obtained good agreement for centerline and axial and radial profiles of temperature and concentrations of CH_4, CO_2, O_2, and CO. As an example, fig. 2(a) show a comparison between the present results and experiments for temperature at the chamber centerline (comparisons for CH_4, CO_2 , O_2, and CO are not shown here for brevity).

Results and discussions

The impact of turbulators on the combustor thermal behavior

The first part of this study concerns to the investigation of the influence of different inlet turbulators on the thermal behavior of the combustion chamber, focusing on the thermal radiation exchange. Those turbulators produce 3%, 6%, 15%, and 20% of turbulence intensities each (it was considered the same TI for both fuel and air inlet streams). Figure 3 shows the fields of the radiative heat source and the temperature for the four scenarios studied. In figs. 3(a)-3(d) it is observed that with the use of more intense turbulators the region dominated by the emission of thermal radiation (negative radiative heat source) was shifted towards the combustor inlet with a considerable increase of the heat source term. In the same way, it is observed in figs. 3(e)-3(h) that the region with the highest temperatures was also displaced towards the combustion chamber inlet. It was verified that the average temperature in the entire domain of the chamber was increased significantly, thus explaining the increase of the radiative heat source term, and thus implying that the use of more intense turbulence promoters led to an increased heat transfer in the combustion chamber.

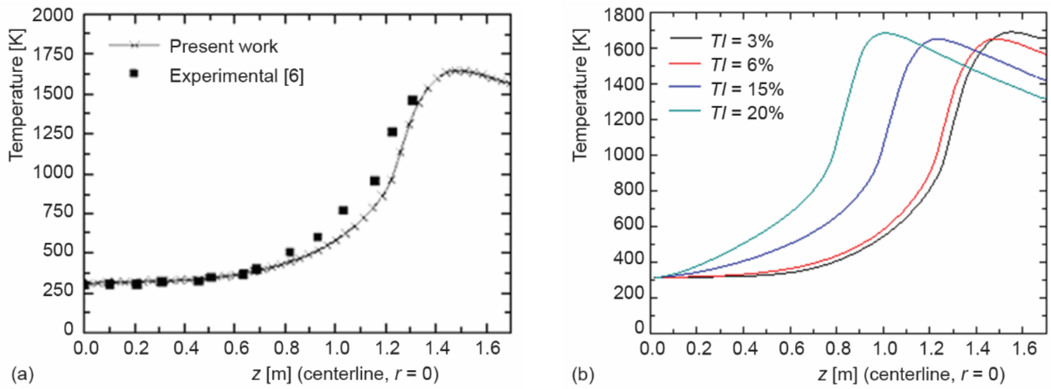

Figure 2. Temperature profiles at the chamber centerline (*r* = 0); (a) numerical results and experimental data from [6], (b) effect of TI variation

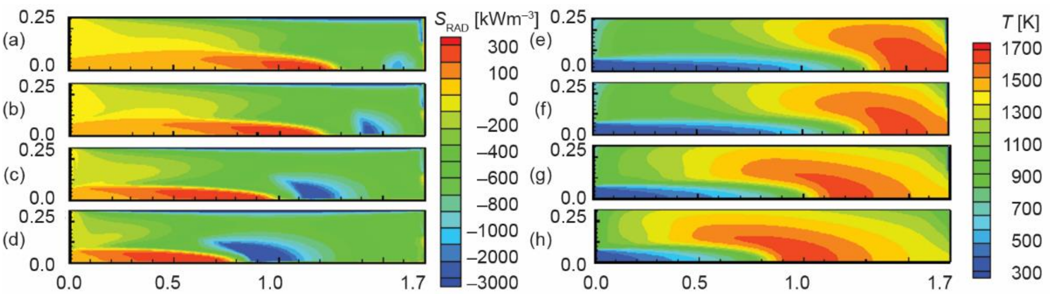

Figure 3. Effect of different turbulators on (a)-(d) radiative heat sources and (e)-(h) temperature: (a), (e) TI = 3%, (b), (f) TI = 6%, (c), (g) TI = 15%, (d), (h) TI = 20%

Figure 2(b) shows the temperature profile along the chamber centerline considering the scenarios studied. It was verified that the distribution of temperatures along the centerline varied according to the use of different turbulators. Such change indicates that the exhaust temperature decreased, being 1653.5 K, 1564.9 K, 1420.1 K, and 1315.3 K for the mildest (TI = 3%) to the most intense (TI = 20%) turbulence promoter, respectively. That reduction in the exhaust temperature of the gases can be considered as an improvement in the thermal exchange of the chamber and can be attributed to the increase of the mixture of fuel and oxidant in the region closer to the entrance of the combustion chamber, resulting in a more efficient combustion process inside the combustor, and consequently a better heat transfer to the walls of the chamber.

Figure 4 depicts the mole fraction field of CO_2 and the soot volume fraction field. The field of H_2O mole fraction is similar to the x_{CO_2}, with the same distribution contours but with values doubled ($x_{H_2O} \approx 2x_{CO_2}$); it is not shown here for brevity. According to the use of turbulence promoters, there was an increase of the region with significant values in the mole fraction field of both H_2O and CO_2, and the production of these species occured in the region closest to the entrance of the chamber with the increase of the TI, as occurred for the temperature. However, the variation of maximum values in these fields was negligible when TI was changed. The same displacement was observed for the soot field as the turbulator was intensified, but soot amount was increased (peak soot volume fraction increased from 0.17 ppm to 0.33 ppm when comparing the weakest and the strongest turbulator, which is in the range of soot production presented in [33]).

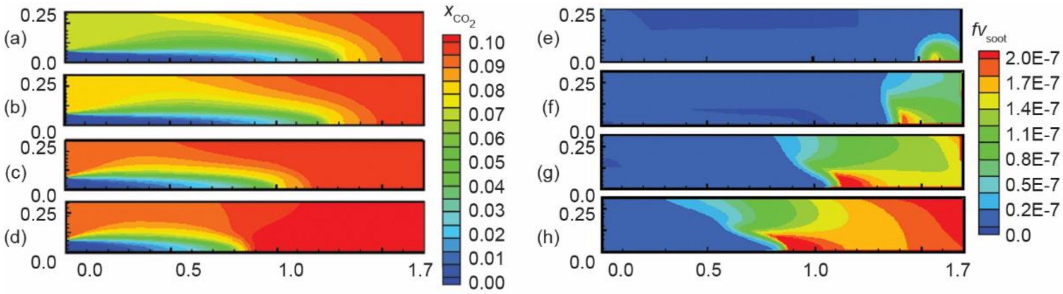

Figure 4. Effect of different turbulators on (a)-(d) mole fraction field of CO_2 and (e)-(h) soot volume fraction field: (a), (e) TI = 3%, (b), (f) TI = 6%, (c), (g) TI = 15%, (d), (h) TI = 20%

Table 1(a) presents the results obtained for the convective and radiative heat transfer rates on the cylindrical wall of the combustion chamber. The convective heat transfer rate was computed with Fourier's Law, considering that the fluid is stagnant at the chamber wall. It is observed that the convective rate remained practically constant, with an increase of only 1.9 kW, comparing the extreme cases, while the radiative rate increased significantly (increase of 68.2 kW, equivalent to 54%); as a result, the total rate increased significantly. Thus, the more intense turbulator provided a net increase in the total rate of heat transfer between the combustion products and the chamber wall of 70.1 kW, equivalent to 38%, in comparison with the weakest turbulator. The increase on the turbulence intensity generated by the inflow turbulators led to an increase to the turbulent viscosity close to the chamber entrance and then promoted a better reactant mixing inside the combustion chamber, enhancing its thermal performance, so that the average temperature of the products was increased, larger portions of the chamber had significant amounts of H_2O, CO_2, and soot, leading to a higher radiative heat transfer rate on the chamber wall.

An important factor that describes the global radiative field in a flame is the net rate of radiative transfer in the flame and its normalized variable, the radiative fraction, f_{rad}. The net rate of radiative transfer corresponds to the integral of the radiative heat source over the entire

Table 1. (a) Heat transfer rates on the cylindrical wall of the combustion chamber and (b) net rate of radiative transfer and radiative fraction: results for different inlet TI

	(a) Heat transfer rates on the cylindrical wall of the combustion chamber			(b) Net rate of radiative transfer and f_{rad}	
TI [%]	Convective heat transfer rate [kW]	Radiative heat transfer rate [kW]	Total rate (rad + conv) [kW]	Net rate of radiative transfer [kW]	f_{rad} [%]
3	56.8	125.7	182.5	128.1	20.6
6	56.1	134.9	190.9	138.5	23.0
15	57.8	171.8	229.6	177.7	29.0
20	58.7	193.9	252.6	198.4	32.8

computational domain; f_{rad} is the ratio of this value to the rate of heat released in the combustion [7]. Table 1(b) presents the net rate of radiative transfer and f_{rad} for the inlet turbulators studied, both presenting significant values, confirming the relevance of the study of thermal radiation in this chamber. The augmentation in these variables agrees with the increase of the radiative heat source, figs. 3(a)-3(d).

Effect of turbulence promoters on TRI

This section concerns to the influence of different turbulent intensities on the modelling of TRI. In order to make a clearer comparison of the scenarios, the results are presented considering the effects of two turbulators, which generate TI = 6% and TI = 20%, in both cases considering and neglecting the modelling of TRI (again, it was considered the same TI for both fuel and air inlet streams).

Figure 5 presents the radiative heat source and temperature fields, respectively. It was found that the radiative heat source calculated including TRI was greater than that calculated when TRI was neglected. As can be seen in fig. 5, TRI also played a relevant role over the temperature field. The maximum temperature decreased 42 K with the weakest turbulator and 47.3 K with the most intense turbulator when considering TRI. The influence of TRI on the maximum flame temperature was shown to be roughly of the same importance for both weak and strong turbulators.

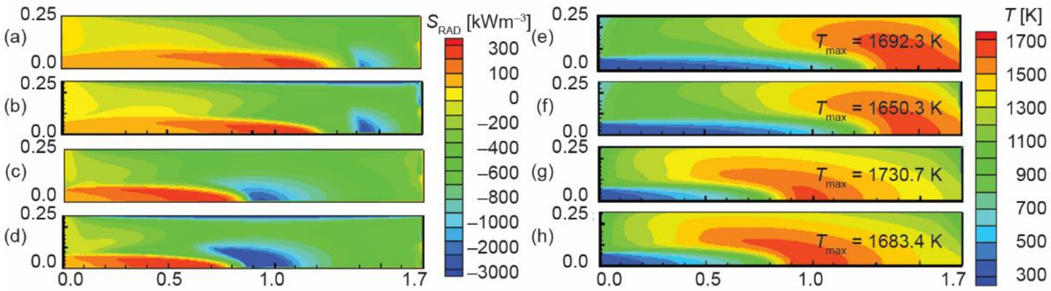

Figure 5. (a)-(d) Radiation heat source and (e)-(h) temperature fields for the following calculation scenarios: (a), (e) neglecting TRI, TI = 6%, (b), (f) computing TRI, TI = 6%, (c), (g) neglecting TRI, TI = 20%, and (d), (h) computing TRI, TI = 20%

Table 2(a) presents the results obtained for the convective and radiative heat transfer rates on the cylindrical wall of the combustion chamber when TRI modelling was neglected (these results must be compared with those in tab. 1(a), which considers TRI modelling). It is observed that the heat transfer by convection increased when TRI was neglected, since the temperature gradients in the chamber were higher, as a consequence of the lower radiative heat source term that was obtained in the non-TRI scenario (when comparing with the scenario computing TRI). As expected, the radiative heat transfer increased when TRI were considered, since the radiative heat source term was greater in that case, refer to fig. 5(a)-5(d).

The net rate of radiative transfer and the radiative fraction were calculated and the results are also shown in tab. 2(b). It is observed that the consideration of TRI led to an important effect on these parameters, just as it did for the heat rates on the wall. When considering the effects of TRI, the radiative fraction increased by 25% (from 18.4% to 23.0%) for the case with TI = 6%, while the increase was 17% (from 28.0% for 32.8%) for TI = 20%.

The results of this section showed that TRI have important effects on the heat transfer in the combustion chamber for both intensities of turbulence promoters, influencing significantly heat transfer rates, temperature field and radiative parameters, independently of the turbulator strength.

Table 2. (a) Heat transfer rate on the cylindrical wall of the combustion chamber and (b) net rate of radiative transfer and radiative fraction: results neglecting TRI calculations

	TI = 6%	TI = 20%
(a) Heat transfer rates on the cylindrical wall of the combustion chamber		
Convective heat transfer rate without TRI [kW]	70.1	74.8
Radiative heat transfer rate without TRI [kW]	106.6	160.3
Total rate (radiation + convection) without TRI [kW]	181.7	235.1
(b) Net rate of radiative transfer and radiative fraction		
Net rate of radiative transfer without TRI [kW]	110.9	169.2
The f_{rad} without TRI [%]	18.4	28.0

Conclusions

The present work presented numerical simulations of a turbulent non-premixed methane flame in a cylindrical chamber, aiming to study the effect of inlet turbulators on the chamber thermal behavior, with emphasis on the thermal radiation exchange. The primary effect of such turbulators is to produce a range of turbulence intensities at the air and fuel inlets, which in turn affect the flow and thermal behavior of the combustion chamber.

The comparison between different turbulators (producing turbulence intensities ranging from 3% to 20% at the inlet streams) showed that temperature and radiative heat source fields, as well as heat transfer rates on the chamber wall and the radiative fraction, were importantly affected by the increase of the inlet turbulence intensity, since the turbulent mixing enhanced the gas mixture, thus increasing the thermal performance of the combustion chamber. It was shown that the radiative fraction increased about 60% when the turbulence intensity was varied from 3% to 20% in both inflow streams (air and fuel).

In a subsequent section, the study concerned to the influence of different turbulence intensities on the modelling of TRI. It was observed that TRI played an important role in the temperature field, heat transfer rates and radiative parameters, independently of the turbulator intensity (with a decrease of about 20% on the radiative fraction when TRI modelling was neglected, for both turbulence intensities simulated), proving the importance of including the TRI modelling into the calculations.

The numerical results of the present study showed a good perspective on how to enhance the heat transfer in a combustion chamber, despite that, it must be mentioned that experiments should be performed to verify and guarantee the trends found here.

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