Numerical Simulation Of Turbulence-Radiation Interaction In A Cylindrical Incinerator Of The Smokes Produced By Household Wastes

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Abstract: A numerical study of the thermal destruction of the pollutant components of smoke generated by wastes combustion is presented. Calculations are performed for a cylindrical incinerator with an inlet and outlet port. Turbulent transfers are modeling using RANS equations and the k-ε model. These equation linked to the radiative transfer equation are solved using an implicit scheme based on the finite volume method. We analyze the effects of Reynolds number, inlet and outlet port position and radiative transfers on the velocity and temperature fields, and on the cleaning up of the smokes effectiveness. The results show that the thermo destruction process is more efficient for low Reynolds number. The radiative transfer intensifies transfers in the incinerator and enhances the thermo destruction process efficiency. The cleaning up of the smokes is better with an outlet port position shifted with respect to the inlet port location.

Index Terms: Turbulence, radiation, Eddy-dissipation, k-ε model, pollutants.

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1 INTRODUCTION

THE process of incineration is a waste treatment technology that involves the dispersion in the atmosphere a number of pollutants, especially carbon oxides (COx), sulfur dioxide (SO2), nitrogen oxides (NOx), ammonia (NH3), acids and metals which toxic effects and environmental risks affect people living conditions. Among several processes of treatment of such effluent, the thermo destruction of the gas pollutants remains at present the most promising technique of smokes depollution. Experimental and theoretical works have shown that household wastes combustion is characterized by different mechanisms, which strongly interact with each other. These are the chemical reactions of gas mixture oxidation and of the pollutant formation/destruction processes, convective heat and mass transfers as well as radiative transfers. Also, turbulent combustion remains an important and timely subject in engineering science. Many of the most urgent energy efficiency, climate change, and pollutant emission issues worldwide are related to the conversion of chemical energy to sensible energy (heat) via a combustion process in a turbulent flow environment. The combustion process in devices as engines, stationary and aircraft gas-turbine combustors, and industrial burners is characterized by complex turbulence-chemistry interactions that span multiple combustion regimes: premixed flame propagation, mixing-controlled burning, and chemical-kinetics-controlled processes. In the same way, the incineration of combustive smoke components in the thermo destruction process involves complex turbulence-chemistry-radiative interactions. Turbulence is the most common state of fluid flow in a wide range of technologies and natural conditions [1]. A wide range of turbulence models is available, encompassing turbulent-viscosity models, Reynolds stress models, probability density function (PDF) methods and large eddy simulation (LES), direct numerical simulation (DNS) [2]. Among the models used to predict turbulent flows, the most popular is the two-equation k-ε model .The standard k-ε model developed by Launder and Spalding [3] is used for free shear layer flows with small pressure gradient and high Reynolds number. However, this model is not suitable for low Reynolds number flows as boundary flows or flows with buoyancy forces. Lam and Bremhorst [4] and several other researchers [5-8] have extended the original k-ε model to the low-Reynolds number flows. In order to take into account the radiative heat transfer effect on transfers which occurs in combustion systems, including boilers, furnaces, internal combustion and rocket engines, and also in fires [9], many radiation models have been developed to calculate radiative heat transfer [10]. Most of these models are based on the resolution of the radiative transfer equation (RTE). Some detailed numerical studies on thermal radiation and radiation models are reported in [11-19] for hydrocarbon fuel combustion. The interaction between turbulence and radiation (TRI) is well established, both theoretically and experimentally [20-21]. The flow and species concentration fields are influenced by radiation transfer because of the fluid density is depending on the fluid temperature and consequently on the radiative heat transfer. A theoretical analysis in which radiative transfer, thermal

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conduction and velocity fluctuations in post- combustion gases are linked showed that the temperature fluctuations of any amplitude induced fluctuations of the radiative flux and the flow velocity [22]. It has also been shown that in a turbulent flow the radiative transfer modifies the temperature spectrum [23]. In reactive flows, the turbulence promotes fluctuations of the temperature and the species concentration [21]. Radiative transfer in reactive flows acts as a dissipative process, especially for large-scale structures for which the optical thickness becomes important. This was first pointed out by Spiegel [24] and Townsend [25], in an analysis of the equations of the mean-square fluctuations of the velocity and temperature and the radiation field. The influence of radiative transfer on turbulence in high temperature flows was investigated by Soufiani [23] for the case of a homogeneous and isotropic turbulence. This study has shown that the effects due to radiative transfer increase as temperature increases, but decrease as the dissipation rate of turbulent kinetic energy decreases. As seen, the combustion process has been the subject of various studies in order to analyze the effect of radiation on heat and mass transfer. Thus, the purpose of the present work is to study numerically the thermo-destruction in an incinerator of smokes produced by the combustion of household wastes. One of the aims of this study is to optimize the efficiency of the thermo destruction process.

2 PROBLEM FORMULATION

The physical model and coordinates system are shown in Fig.1. The furnace considered is a vertical cylinder of height H=1 m and diameter D=1 m. The top and bottom walls are insulated with a 0.20m thick glass wool; the vertical walls are composed of three layers: an internal refractory brick layer (0.10 m); a 0.005 m thick metallic sheet and a layer of glass wool of thickness equal to 0.10 m. The smokes enters in the furnace through a port (din=0. 20 m) located in the middle of the bottom wall. After the thermo-destruction process the gas mixture is rejected through a port (dout=0.20 m) in the middle of the top wall.

It is assumed that:

- the viscous dissipation is neglected;
- the Dufour and Soret effects are neglected;
- the furnace walls are gray, diffusively emitting and reflecting;
- the fumes are composed of CH_4, C_2H_4, CO, and H_2
- the smokes are assimilated to an ideal gas and supposed to be a grey and diffusing medium
- transfers are two dimensional.

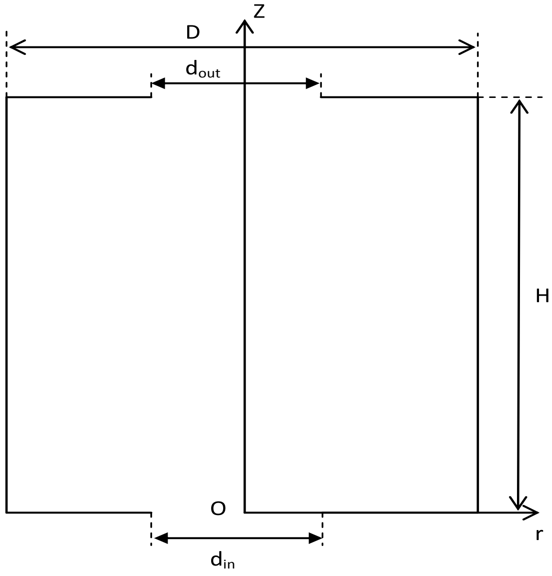

Fig.1. Schematic of the system studied.

In a two dimensional cylindrical coordinates, the RANS equations for a turbulent flow of incompressible fluids, heat and mass transfer linked to radiative transfer are expressed in conservative differential form in cylindrical coordinates (r, z), in terms of variables such as velocities (u, v), pressure (P), density (ρ), masse fraction of species "i" (Y_i), temperature (T), turbulence kinetic energy (k) and the dissipation of turbulent kinetic energy (ε) .

Continuity equation

$$\partial\rho/\partial t + \partial(\rho u)/\partial z + \partial(\rho r v)/r\partial r = 0 \qquad (1)$$

Momentum equation

$$\begin{aligned}\partial(\rho u)/\partial t + \partial(\rho u u)/\partial z + \partial(\rho r v u)/(r\partial r)\\ = 2[\partial(\mu_e \partial u/\partial z)/\partial z + \partial(r\mu_e \partial u/\partial r)/(r\partial r)]\\ -\partial P/\partial z + \rho g\end{aligned} \qquad (2)$$

$$\begin{aligned}\partial(\rho v)/\partial t + \partial(\rho u v)/\partial z + \partial(\rho r v v)/(r\partial r)\\ = 2[\partial(\mu_e \partial v/\partial z)/\partial z + \partial(r\mu_e \partial v/\partial r)/(r\partial r)]\\ -\partial P/\partial r - 2\mu_e v/r^2\end{aligned} \qquad (3)$$

Where μ_e is the effective viscosity defined by $\mu_e = \mu + \mu_t$, with $\mu_t = C_\mu \; k^2/\varepsilon$, the turbulent viscosity, and μ the dynamic fluid viscosity. C_μ denotes an experimental constant.

Energy equation

$$\begin{aligned}\partial(\rho T)/\partial t + \partial(\rho u T)/\partial z + \partial(\rho r v T)/(r\partial r)\\ = \partial[(\lambda/C_P + \mu_t/Pr_t)(\partial T/\partial z)]/\partial z\\ +\partial[r(\lambda/C_P + \mu_t/Pr_t)(\partial T/\partial r)]/(r\partial r) + S_h\end{aligned} \qquad (4)$$

Where:

- S_h denotes the volumetric heat quantity released by the chemical reactions and the radiative transfer,
- λ is the thermal conductivity
- C_P is the specific heat at constant pressure.
- Pr_t is the turbulent Prandtl number

Mass transfer equation

$$\begin{aligned}\partial(\rho Y_i)/\partial t + \partial(\rho u Y_i)/\partial z + \partial(\rho r v Y_i)/(r\partial r) \\ = \partial[(\rho D_i + \mu_t/Sc_t)\partial Y_i/\partial z]/\partial z \\ +\partial[r(\rho D_i + \mu_t/Sc_t)(\partial Y_i/\partial r)]/(r\partial r) + R_i\ ,\end{aligned} \tag{5}$$

R_i denoting the volumetric mass rate of species i *(CH_4, C_2H_4, CO,H_2, O_2, CO_2, H_2O)*

Turbulence kinetic energy equation

$$\begin{aligned}\partial(\rho k)/\partial t + \partial(\rho u k)/\partial z + \partial(\rho r v k)/(r\partial r) \\ = \partial[(\mu + \mu_t/\sigma_k)(\partial k/\partial z)]/\partial z \\ +\partial[r(\mu + \mu_t/\sigma_k)(\partial k/\partial r)]/(r\partial r) + G_k - \rho\varepsilon\end{aligned} \tag{6}$$

where

$$\begin{aligned}G_k = 2\mu_e[(\partial u/\partial z)^2 + (\partial v/\partial r)^2 + (v/r)^2] \\ + \mu_e(\partial u/\partial r + \partial v/\partial z)^2\end{aligned} \tag{7}$$

Dissipation of turbulent kinetic energy equation

$$\begin{aligned}\partial(\rho\varepsilon)/\partial t + \partial(\rho u\varepsilon)/\partial z + \partial(\rho r v\varepsilon)/(r\partial r) \\ = \partial[(\mu + \mu_t/\sigma_\varepsilon)(\partial\varepsilon/\partial z)]/\partial z \\ +\partial[r(\mu + \mu_t/\sigma_\varepsilon)(\partial\varepsilon/\partial r)]/(r\partial r) \\ +\varepsilon(C_{\varepsilon 1}G_k - C_{\varepsilon 2}\rho\varepsilon)/k\end{aligned} \tag{8}$$

with $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$, $C_{\varepsilon 1} = 1.44$, $C_{\varepsilon 2} = 1.92$, $C_\mu = 0.09$

Radiative Transfer equation (RTE)

$$\begin{aligned}\partial(\xi I)/\partial z + \partial(\gamma r I)/(r\partial r) - \partial(\eta I)/(r\partial\phi)= \\ -(k_a + \sigma_s)I + k_a I_b + (\sigma_S/4\pi)\int_{\Omega'=4\pi} I\Phi(\vec{S}',\vec{S})d\Omega\end{aligned} \tag{9}$$

Where
I radiative intensity
I_b blackbody radiative intensity
ξ,γ,η are direction cosines,
$\vec{S},\vec{S}'$ are direction vector and scattering direction vector,
k_a,σ_s are absorption coefficient and scattering coefficient,
Φ is phase function ,
ϕ angular azimuthal angle
Ω,Ω' solid angles.

Combustion model

The fluid flow and chemical reaction must be defined, for the combustion of the components of the smoke generated by household wastes incineration. The smoke is a mixture of hydrocarbons such as methane (CH_4) and ethylene (C_2H_4), hydrogen (H_2) and monoxide of carbon (CO). In the present work, the following global single-step reactions for smoke components combustion in air are used.

$$\begin{gathered}CH_4 + 2O_2 \rightarrow\ 2H_2O + CO_2 \\ C_2H_4 + 3O_2 \rightarrow\ 2H_2O + 2CO_2 \\ CO + (1/2)O_2 \rightarrow CO_2 \\ H_2 + (1/2)O_2 \rightarrow H_2O\end{gathered} \tag{10}$$

Reaction rate in turbulent flow is calculated by the eddy-dissipation model [18] which is given as:

$$R_{i,r} = \nu'_{i,r}M_i A\rho(\varepsilon/k)min\left(Y_R/\nu'_{R,r}M_R\right) \tag{11}$$

or

$$R_{i,r} = \nu'_{i,r}M_i B\rho(\varepsilon/k)\left(\sum Y_P/\sum\nu''_{j,r}M_j\right) \tag{12}$$

Where:
Y_R, Y_P are mass fractions of reactants R and products P respectively; A et B: empirical constants ($A = 4.0; B = 0.5$).
M_i : molecular weight of specie i

Initial and boundaries conditions

The initial flow velocity is taken as zero and the initial fluid temperature and mass fraction are considered to equal to constant values. Therefore, the initial conditions for $t \le t_0$, t_0, being the time from the beginning of the smoke components combustion are written as:

$$u = v = 0, Y_k = Y_{kin}, T = T_{in}, k = k_{in}, \varepsilon = \varepsilon_{in} \tag{13}$$

In this study no slip and no penetration velocity conditions are imposed at the incinerator walls. Turbulent kinetic energy and its dissipation rate are set as:

At walls

$$k = 0 \tag{14}$$

At the vertical walls

$$\varepsilon = 2(\mu/\rho)(\partial k^{1/2}/\partial r)^2 \tag{15}$$

At the horizontal walls

$$\varepsilon = 2(\mu/\rho)(\partial k^{1/2}/\partial z)^2 \tag{16}$$

The top and bottom walls are adiabatic. The heat transfer condition on the vertical walls, the principle of heat flux conservation is applied to both fluid and wall according to Fourier law of heat conduction.

$$\begin{aligned}2\pi r_1 H\lambda\,\partial T/\,\partial r = (Tw_{in} - Tw_{out})/R_t \\ = 2\pi r_1 H h_{air}\left(T_{w_{out}} - T_{amb}\right)\end{aligned} \tag{17}$$

with

$$R_t = \left(\sum_{l=1}^{3}(1/3)\log(r_{l+1}/r_l)\right)/2\pi H \tag{18}$$

where λ_l is the layer l conductivity, r_l and r_{l+1} are two consecutive radius of the cylindrical surfaces delimiting the layer l. For the mass fraction condition, we suppose:

$$\partial Y_i/\partial r = 0 \tag{19}$$

Which means that there is no one specie deposit. At inlet port ($0 \le r \le d_{in}/2$, $z = 0$), uniform boundary condition for velocity, temperature, mass fraction, turbulent kinetic energy and its dissipation are specified :

$$\begin{aligned}u = u_{in}, v= 0\ , Y_k = Y_{kin}, T = T_{in}, \\ k = 0.005\,U_{in}^2\ ; \varepsilon = 2C_\mu k^{3/2}/(0{,}03\ d_{in})\end{aligned} \tag{20}$$

We assume flow, thermal and mass fraction fields are developed fully at the outlet port:

$$\partial\emptyset/\partial z = 0 \qquad (21)$$

Where $\emptyset$ can be any unknown variable: $u, v, T, Y_i, k, \varepsilon$ Along incinerator axis is the symmetric boundary condition due to one of our assumptions.

$$\partial\emptyset/\partial r = 0 \qquad (22)$$

Where $\emptyset$ can be any unknown variable: $u, v, T, Y_i, k, \varepsilon, I$ The radiation boundary condition is written as:

At the inlet port: $0 \le r \le d_{in}/2$, $z = 0$

$$I = \varepsilon_g(\sigma T_{in}^4)/\pi \qquad (23)$$

At the outlet port: $0 \le r \le \frac{D}{2}$, $z = H$

$$I = \varepsilon_g(\sigma T_{out}^4)/\pi \qquad (24)$$

On walls:

$$I_w = \varepsilon_w \frac{\sigma T_w^4}{\pi} + \frac{1-\varepsilon_w}{\pi}\int_{\vec{n}_w.\vec{\Omega}} I_w\left|\vec{n}_w.\vec{\Omega}\right| d\Omega' \text{ if } \vec{n}_w.\vec{\Omega} > 0 \qquad (25)$$

where ε_w is the wall emissivity and $\vec{n}_w$ represents the unit normal vector on the wall with:

3. NUMERICAL PROCEDURE

A finite volume scheme is employed to solve the governing equations for the present study. The radiative transfer equation (RTE) is the discrete coordinate method. The obtained discretized equations are solved using Gauss Seidel iterative method and under-relaxation factors (0.6 for u and v; 0.8 for the others parameters). For each time step Δt=0.01s, convergence was declared when the following criterion is verified:

$$|\emptyset^{n+1} - \emptyset^n|/|\emptyset^{n+1}| \le \text{Prec}$$

where the superscripts design the iteration number, $\emptyset$ stands for $u, v, T, Y_i, \varepsilon, k$ and I. Prec is the relative variations of any dependent variable and is equal to 10^{-4}.

3.1. Validation

The accuracy of the numerical model was checked by comparing the results from the present study with those reported in [27]. As shown in Fig.2 ours results are in good agreement with the experimental velocity and temperatures of [27]. The maximum discrepancy is less than 3.5% for the vertical component velocity and about 4% for the temperature.

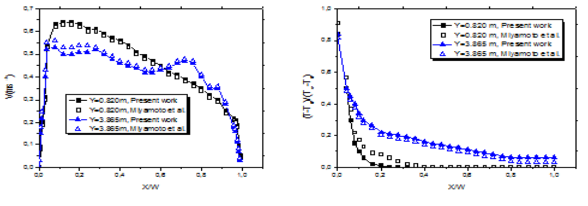

Fig.2. Vertical component velocity (a) and temperature (b) profiles versus the abscissa for q_w=208Wm^{-2}, w=0.1m and Tuin=15%.

3.2. Mesh size sensibility

For various Reynolds number used in ours computations, we ensure that results are grid independent. For Re = 5000 we obtained a grid independent solution for a spatial grid 51 x 51 (Table 2). Whereas a spatial grid refinement for the others Reynolds numbers doesn't contribute to ameliorate the precision of the solution. Consequently, we have retained the spatial grid 51*51. It corresponds to the following grid step: Δr= 0.01; Δz =0.02. In Table 1, Nu_{max}, u_{max}, v_{max}, ψ_{max} and μ_{tmax} are respectively, the maximum local Nusselt number, the axial and radial velocity component, the stream function and the dissipation of turbulent kinetic energy.

TABLE 1

EFFECT OF GRID SIZE FOR RE=5000

Grids	Nu_{zmax} at r=R	U_{max} at z=0.5	V_{max} at z=0.5	ψ_{max}	μ_{tmax}
51x51	55.8261	0.413744	0.0181427	0.04853	7.5629×10^{-4}
61x61	55.8284	0.4142	0.0176445	0.04862	7.5788×10^{-4}
81x81	75.9318	0.413267	0.0186186	0.04872	7.6482×10^{-4}
101x101	94.0672	0.412576	0.0166763	0.04896	7.8162×10^{-4}
131x131	98.0465	0.407878	0.0118862	0.04953	8.2979×10^{-4}

4. Results and discussion

Numerical computations are carried out to illustrate the influence of the physical parameter on the resulting streamlines, isotherms and iso-concentrations as well as turbulent kinetic energy and its dissipation rate, the thermo-destruction efficiency and the mass fraction of the smoke pollutant component at the outlet port of the incinerator. Reynolds number was varied in the range of 5000 to 20000, the aspect ratio H/D and din/D, respectively equal to 1 and 1/5. The emissivity of the incinerator wall is taken equal 0.8. The smoke is a mixture of CH_4, C_2H_4, CO, O_2, N_2, CO_2, H_2O whose mass fractions are respectively equal to: 0.38, 0.23, 1.29, 0.03, 16.07, 2.55 and 11.66.

4.1. Flow and temperature fields

Fig.3 and 4 show the distribution of streamlines for two Reynolds numbers (Re = 10000, Re = 20000). It can be seen that the flow structure is depicted by two counter-rotating cells, located on both side of the symmetrical vertical axis of the incinerator along which the mean flow is developed between the inlet and outlet ports (Fig.3 and 4). The increase of the Reynolds number leads to an increase of the streamlines values without modifying significantly the size of the counter-rotating cells. That can be explained by the fact that the increase of Reynolds number results in the intensification of convective transfers. The corresponding isotherms are presented in Fig.5 and 6 for respectively Re =10000 and Re= 20000. Regarding to the temperature contour, it can be noted that isotherms start at the inlet port difference places on both side of the inlet mean flow and then are warped by the flow along the symmetrical vertical axis and the heat loss through the vertical walls. The Comparison of the isotherms illustrated in Fig. 5 and 6, shows that the isotherms fields are weakly modified when the Reynolds number increases. The examination of isotherms values for the two Reynolds number reveals that they decreases when the Reynolds number increase because of the reduction of the residence time which affects the combustion of smoke pollutants. The effect of radiation on isotherms is illustrated in Fig. 7. It can be observed that isotherms fields are weakly modified (Fig. 5 and 7). The isotherms values increase notably in the zones located on both side of the incinerator symmetrical vertical axis (Fig.5, Fig.6 and 7). This result is due to the fact that the combustion of smoke components is more intense in these zones. It can be observed that the maximum value which is equal to 895 K (Fig.5), when taking into account combustion and neglecting radiation, increases up to 1230 K when the radiative heat transfer is took into account in the combustion of smoke pollutants (Fig.7). The radiation enhances the thermo destruction of smoke components.

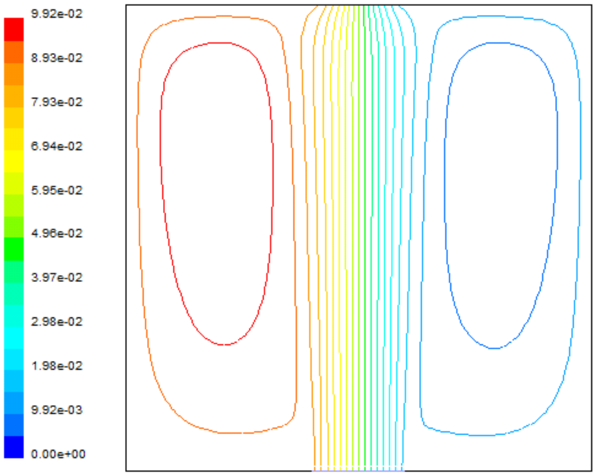

Fig.3. Streamlines at t=30s for Re = 10000 with combustion without radiation.

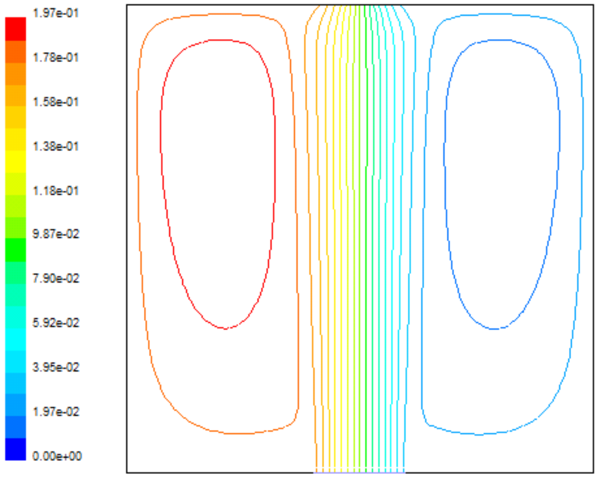

Fig.4. Streamlines at t=30s for Re = 20000 with combustion without radiation.

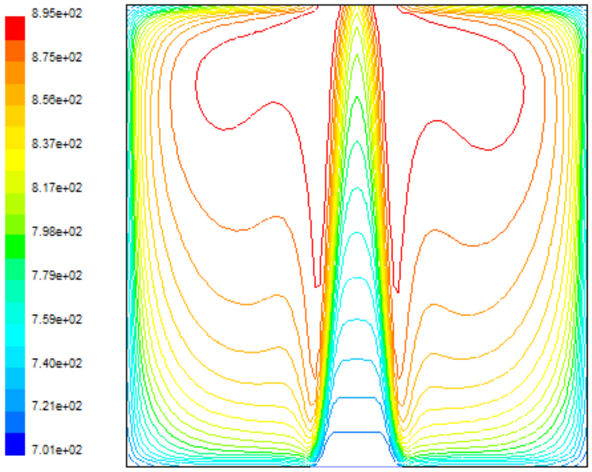

Fig.5 Isotherms distribution at t=30s for Re=10000 with combustion without radiation

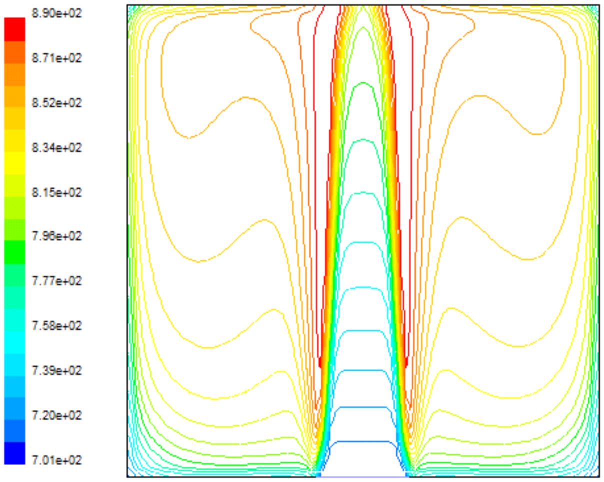

Fig.6 Isotherms distribution at t=30s for Re=20000 with combustion without radiation

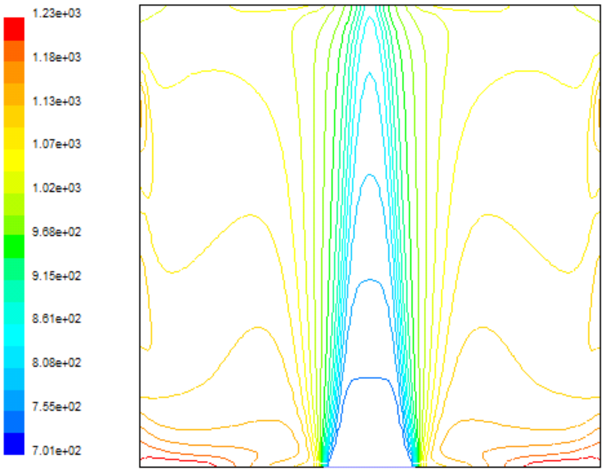

Fig.7 Isotherms distribution at t=30s for Re=10000 with both combustion and radiation.

4.2. Effect of the location of inlet and outlet ports on the flow and temperature fields

The effect of the outlet port position on the flow field and the heat and mass transfers at is displayed in Fig. 8 and 9. It is observed that the inlet port location modifies strongly the flow structure (Fig.3 and Fig.8) and the distribution of temperature (Fig.5 and Fig.9). It is noted an increase of the isotherms values when the inlet and outlet ports are not opposite.

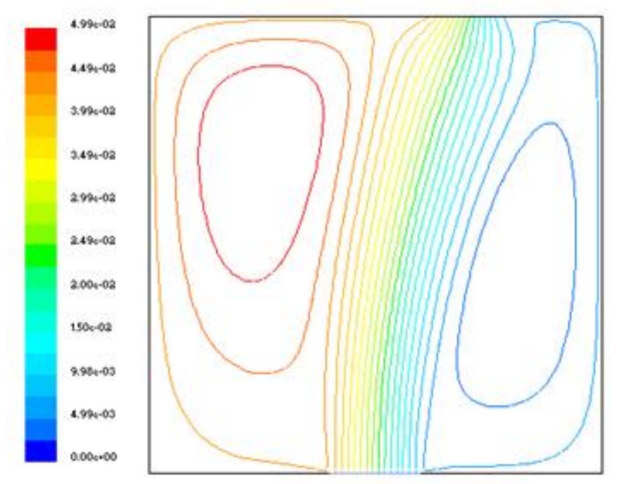

Fig.8. Streamlines at t=30s for Re = 10000 with combustion without radiation. Effect of the inlet and outlet ports shifted

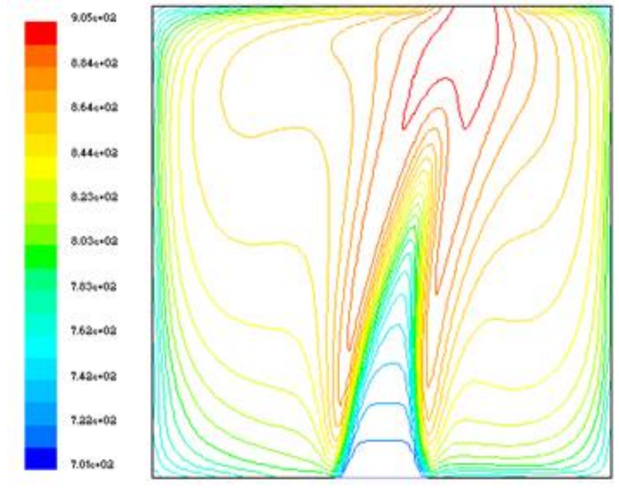

Fig.9. Isotherms at t=30s for Re = 10000 with combustion without radiation. Effect of the inlet and outlet ports shifted

4.3 Thermo destruction effectiveness

Components thermal destruction efficiency was calculated from the following equation:

$$\mathrm{TED} = (\mathrm{Y_{in}} - \mathrm{Y_{out}})/\mathrm{Y_{in}}$$

where Y_{out} is the amount of pollutant component at the outlet and Y_{in} is the amount of pollutant component supply to the incinerator. The effect of the smoke flow rate on the thermal destruction efficiency is shown in Fig. 10. It can be noted that the thermal destruction efficiency of the pollutant components of the smoke decreases with the increase of the Reynolds number. This result can be explained by the inlet and outlet port position which guide the mean flow along the symmetry axis. This configuration leads to the small residence time of the pollutants in the incinerator. It is smaller when the Reynolds number is great (Fig. 10). For an inlet H_2 mass fraction equal to 3.010^{-4}, the thermo destruction of this component reduces its mass fraction to 2.1.10^{-4} for Re=5000 (Table 2). It can be noted that this mass fraction is reduced to 2.175.10^{-4} for Re = 20000 and to 2.2.10^{-4} for Re=20000 (Table 2).

TABLE 2
MASS FRACTIONS OF SMOKE COMPONENTS

components		CH_4	C_2H_4	CO	H_2
Mass fractions before incineration		3.8.10^{-3}	2.30.10^{-3}	1.29.10^{-2}	3.0. 10^{-4}
Mass fractions after incinera -tion	Re=5000	3.07.10^{-3}	1.25.10^{-3}	1.16.10-2	2.1.10^{-4}
	Re=10000	3.15.10^{-3}	1.75.10^{-3}	1.17.10^{-2}	2.17.10^{-4}
	Re=20000	3.17.10^{-3}	1.20.10^{-3}	1.18.10^{-2}	2.20.10^{-4}

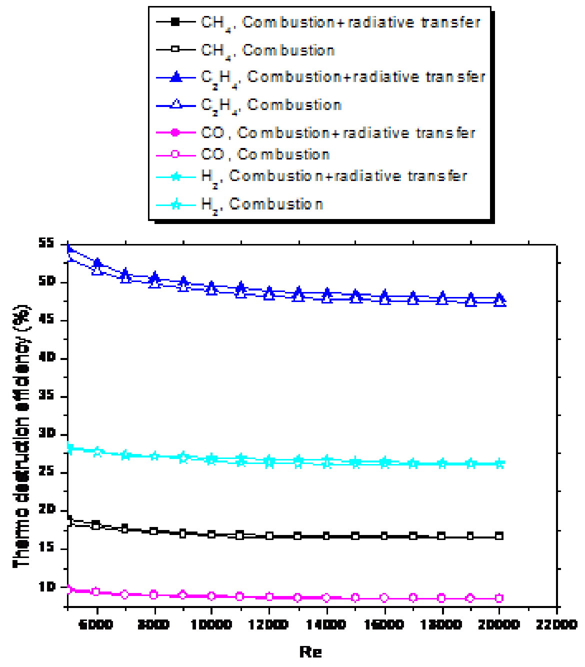

Fig.10. Thermo destruction efficiency of smokes components versus Re at t=30s. Effect of radiative transfer

The effect of the radiative heat transfer on the thermo destruction of the components of the smoke is illustrated on Fig.10. It can be noted that the increase of the gas mixture temperature due to the radiation heat transfer effect improves the thermal destruction of the smoke components. The thermo destruction in turbulent regime is only partially efficient (Table 3) as whereas in a laminar regime it has been shown [29] that all the pollutants disappear after a few seconds. It can be deduced that this mode of thermal treatment for this kind of cylindrical geometry is only efficient in the case of laminar regime (Reynolds numbers less than 2000). The position of inlet and outlet ports affects strongly the thermal destruction of smokes components. Fig. 11 shows that when the outlet position is shifted with respect to the inlet port, the thermal destruction is greater than the case with outlet and inlet in opposite for different Reynolds numbers. For instance, for Re=20000 the thermo destruction efficiency of C_2H_4 is about 50% when the inlet and outlet ports are in opposite but it is equal to 95% with the outlet port shifted. It can be explained by the modification of the flow structure when the outlet and inlet ports are not in opposite. The flow structure imposed by the location of the outlet position leads to an increase of the residence time and consequently of the thermal efficiency (Fig. 11).

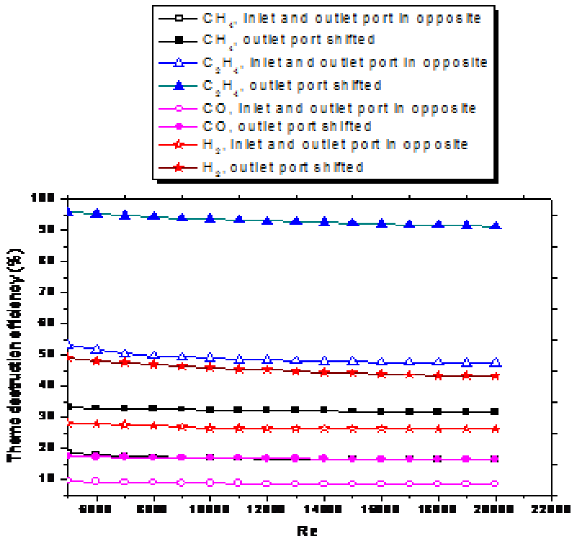

Fig. 11. Thermo destruction efficiency versus Re. t=30s. Effect of the position of the inlet and outlet ports

5. Conclusion

A numerical study has been carried out to investigate the thermo destruction in turbulent regime, of the pollutant components of smoke produced by the household wastes combustion. The model is based on turbulent forced convection equations associated to the radiative transfer equation and to a one step-global chemical kinetics. Transfers equations are discretized using an implicit scheme based on the finite volume method. The algebraic equations obtained are solved using Gauss Seidel iterative method. The main results can be summarizes as follows:

- The amount of heat released by the combustion of the pollutant components intensifies transfers but modifies weakly the flow structure.
- The radiative heat transfer has no effect on the flow structure but leads to an increase of the streamline values. The isotherms fields are weakly modified by the radiative transfer.
- The thermo destruction efficiency of the pollutant components of the smoke decreases as the Reynolds number increases.
- The thermo destruction efficiency depends on the inlet and outlet ports position.

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NOMENCLATURE

Cp	specific heat of the fluid, (Jkg^{-1} K^{-1})
din	inlet port size ,(m)
dout	outlet port size, (m)
D	furnace diameter, (m)
D_k	coefficient of diffusion, (m^2s^{-1})
G	gravitational acceleration, (m s^{-2})
H	furnace height, (m)
hair	heat transfer coefficient by natural convection, ($Wm^{-2}K^{-1}$)
I	radiative intensity, (Wm^{-2} sr^{-1})
K	turbulent kinetic energy, (m^2s^{-2})
P	pressure, (Pa)
q_r	radial radiative heat flux, (Wm^{-2})
Q_r	dimensionless radial radiative heat flux, ($Q_r = \pi q_r/n^2\sigma T_0^4$)
R	radial coordinate, (m)
$\vec{S}, \vec{S}'$	outward and inward radiation directions
R_k	Volumetric mass rate of species i, (kgm^{-3} s^{-1})
R_t	thermal resistance, (KW^{-1})
S_R	volumetric radiative heat rate, (Wm^{-3})
S_h	volumetric heat rate due to chemical reactions and radiation, (Wm^{-3})
T	time, (s)
T	temperature, (K)
Tu	turbulent intensity, (%)
U	Axial velocity, (m s^{-1})
V	radial velocity, (m s^{-1})
Mk	molecular weight of species k,(kg mol^{-1})
Yk	species mass fraction
Z	axial coordinate, (m)

Greek symbols

Β	extinction coefficient, (m^{-1})
Ε	turbulent kinetic energy dissipation rate, (m^2 s^{-3})
σ_k	turbulent Prandtl number
σ_ε	Schmidt number
Ω	solid angle, (sr)
Φ	scattering phase function
Λ	fluid thermal conductivity, (W $m^{-1}K^{-1}$)
Μ	Dynamic fluid viscosity, (kg m^{-1} s^{-1})
ρ	fluid density, (kgm^{-3})
κ_a	absorption coefficient, (m^{-1})
σ_s	scattering coefficient, (m^{-1})

γ, η,ξ direction cosines

Subcripts

amb	ambient
E	effective
b	blackbody
in	inlet quantity
out	outlet quantity
t	turbulent
w	Wall

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