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NUMERICAL STUDY OF THE EFFECT OF DIFFERENT PARAMETERS ON COMBUSTION CHARACTERISTICS OF METHANE FOR DIFFERENT COMBUSTION CHAMBER GEOMETRIES*

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ABSTRACT

The present work investigates and compares between three models developed to simulate the flow and combustion in a typical gas turbine reversed combustor. The first model relies on conventional combustion which uses preheated air using a recuperative heat exchanger. The second and third models investigate the Mild and Intense Low oxygen Dilution "MILD" or Flameless oxidation (High Temperature Air combustion(HiTAC)). HiTAC is based on very efficient preheating of the combustion air by means of regenerators. The regenerators are heated by the exhaust flue gases. The Combustion is carried out using internal Flue gas recirculation coupled with high temperature air under diluted oxidant conditions. The three models use methane as fuel. The models rely on the computational code "FLUENT 6.2.3" which was used to solve the conservation equations of mass, momentum, energy, and transport equations of species concentrations. Turbulence, combustion and radiation modeling in addition to NO_x modeling equations were solved together to represent finally temperature and NO_x distribution inside the burner. All the models are considered 2D axisymmetric. The combustion process has been simulated as non-premixed and non-adiabatic. The effects of excess air, preheated air, diluted air and diluted fuel on the combustion process are also discussed.

KEY WORDS: MILD combustion, Flameless, Combustion modeling, NO_x reduction

ETUDE NUMERIQUE DE L'EFFET DE DIFFERENTS PARAMETRES SUR LES CARACTÉRISTIQUES COMBUSTION DU METHANE POUR DIFFERENTES GEOMETRIES CHAMBRE DE COMBUSTION *

RÉSUMÉ:

Ce travail étudie et compare entre trois modèles développés pour simuler l'écoulement et de la combustion dans une turbine à gaz de combustion inversée typique. Le premier modèle repose sur la combustion classique qui utilise de l'air préchauffé à l'aide d'un échangeur de chaleur à récupération. Les deuxième et troisième modèles enquêter sur la dilution d'oxygène doux et intense faible oxydation "légère" ou sans flamme (Haute Température de l'air de combustion (Hitac)). Hitac sont basées sur le préchauffage très efficace de l'air de combustion au moyen de régénérateurs. Les régénérateurs sont chauffés par les fumées d'échappement. La combustion est réalisée à l'aide de recirculation interne des gaz de combustion avec de l'air couplé à haute température dans des conditions oxydantes dilué. Les trois modèles utilisent le méthane comme combustible. Les modèles reposent sur le calcul de code "FLUENT 6.2.3» qui a été utilisée pour résoudre les équations de conservation de masse équations, l'élan, l'énergie et le transport de concentrations des espèces. Modélisation de la turbulence, combustion et rayonnement en outre équations de modélisation de NO_x ont été résolus en même temps pour représenter la distribution de température et enfin de NO_x à l'intérieur du brûleur. Tous les modèles sont considérés comme 2D axisymétrique. Le processus de combustion a été simulée comme non prémélangée et non adiabatique. Les effets de l'excès d'air, l'air préchauffé, l'air et le carburant dilué dilué sur le processus de combustion sont également discutés.

MOTS-CLÉS: la combustion DOUX, sans flamme, modélisation de la combustion, la réduction des NO_x.

* Received: 1/7/2012, Accepted: 23/9/2012, Ref. No. 119, (Original Paper)

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1. INTRODUCTION

Numerical simulation plays today a major role in the design of new concepts of combustion chamber in gas turbines. The numerical simulation of combustion processes are including the formation of pollutants, has become increasingly important in the recent years. Today, the simulation of those processes has already become an indispensable tool when developing new combustion chamber designs.

MILD combustion or flameless combustion is one of the latest technologies operated. MILD combustion is defined as process that includes widely states of temperature range and overall oxygen mass fraction.

Major target of this technique is to achieve very small NO_x production, avoid high temperatures and achieve energy saving. Hence, it is required to keep most of the combustion process at or below a limiting temperature that NO_x formation is suppressed.

The MILD technique is an effective method to enhance efficiency and diminish fuel consumption by preheating the reactants or the combustion air alone using heat recovery techniques. Preheating using recycled heat from the exhaust gases without mixing the reactants and products streams has been initially called excess enthalpy combustion, and later more generically, heat-recirculating combustion [1].

The advantage of the flameless technology for temperatures > 850 °C with respect to the best low- NO_x combustion chamber designs is quite clear with no visible or audible flame [2-3]. Hiroshi Tsuji et al [4] had studied Flue Gas Recirculation (FGR). This process may be either external or induced, depending on the method used to move the exhaust gas. FGR may also minimize CO levels while reducing NO_x levels. When the reactants are diluted with products at a high temperature, it is well established that significant reduction in NO_x are possible even with high levels of air preheating [1].

Gyung-Min Choi and Masashi Katsuki [5] studied advanced low NO_x combustion using highly preheated air. This method has recently received much attention for its accomplishment not only in energy saving, but also for low NO_x emissions. The characteristics of combustion with highly preheated air were studied to understand the

change of combustion regime and the reason for the compatibility between high performance and low NO_x emissions. Sivaji Seepana and Sreenivas Jayanti [6] investigated the effect of High Temperature Air Combustion (HiTAC), observed that the flame volume expands, the peak temperature decreases and NO_x formation rate can be significantly reduced as thermal NO_x is avoided. Flame volume increases with a reduction in oxygen concentration. This trend is clearer if oxygen concentration in the preheated air is below 10%. The temperature profile becomes more uniform when oxygen concentration in preheated air decreases, especially at low oxygen levels [7-8].

Effect of three diluents (H_2O , CO_2 and N_2) on NO_x formation rate was studied by Saeid J. [9]. The results show that rate of formation of NO_x is reduced by the addition of any diluents at constant combustion temperature. Effect of N_2 inert on soot and NO_x emission levels was experimentally investigated by D.P. Mishra and P. Kumar [10]. They observed the reduction in NO_x emission level for air-diluted stream much higher than fuel-diluted stream.

Several studies investigated dilution in air and fuel sides on MILD combustion burner. The obtained results from these studies are reported as follow [11]:

- 1) If inlet air flow rate increases, the maximum and average temperature in the combustion chamber reduces.
- 2) Increase of exhaust gas recirculation and reactants dilution, NO_x production decreases.
- 3) NO_x production is reduced by the decrease in oxidizer concentration in the Flue Gas Recirculation conditions.
- 4) H_2O diluter because of its high specific heat is more effective in the reduction of NO_x than N_2 and CO_2 .
- 5) Fuel dilution in the Fuel Induced Recirculation (FIR) condition causes a reduction in NO_x emission and suppresses any flame propagation inside the combustion chamber.
- 6) Comparing the FGR and FIR conditions in the case of N_2 , CO_2 and H_2O dilution indicates that the FIR case is more effective in NO_x reduction than the FGR case with small amounts of dilution.

Cécile Cohe' et al [12] observed that the combustion intensity increases with decreasing CO_2 rate and the mean fuel

consumption rate decreases with the CO₂ addition rate.

When the oxygen concentration is high, the reaction zone is formed near the fuel nozzle and the NO_x formation by the thermal mechanism becomes dominant, due to the increase in flame temperature. On the other hand, when the oxygen concentration is low, the reaction is spread out more uniformly in the combustion chamber. The NO_x formation by the prompt mechanism is dominant compared with other mechanism especially when the air is diluted with nitrogen [13].

When burning methane or propane with various conditions of inlet air temperature and oxygen concentration, the results show that when the oxygen concentration is high, the maximum flame temperature becomes higher and the two fuels show quite different characteristics in the downstream region. On the other hand, for low oxygen concentration, the temperature difference between the two fuels is relatively small and remains fairly constant throughout the combustion chamber. The propane gives a higher NO formation compared to the methane especially when the oxygen concentration is high [14].

low-NO_x burners operated at high air preheating temperature is unlike conventional flame mode, the flameless mode is insensitive to air preheating temperature as far as NO_x is concerned, and this is very important for application to high temperature industrial processes or furnaces [15].

The results also show that the lean premixed combustor is more effective to reduce the NO_x emission levels, and NO_x emission reduces with decrease in residence time of the hot gases in the combustor [16].

2. FLOX® BURNER

The burner is cylindrical with radius = 0.045 m and length = 0.58 m it delimited by a radiant tube closed at the upper end. A flame tube radius = 0.02 m and length = 0.41 m, flame tube equipped with three windows and positioned inside the burner. The combustion chamber is delimited by a radiant tube closed at the upper end.. It operates with an internal recirculation of exhaust gases. The burner can operate by radiating heat, as shown in Fig. (1) [17].

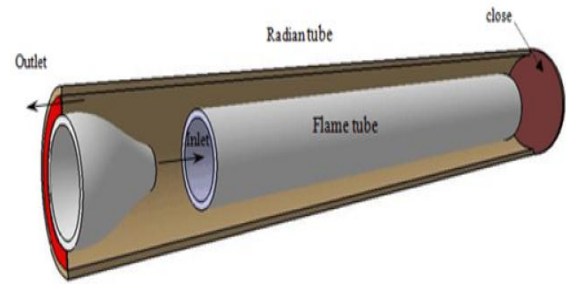


Fig. (1) configuration of the combustion chamber

2.1. Applications of Flox® Burners

There are many applications for FLOX burner such as follows:

2.1.1 Energy savings in steel furnaces

The thermal efficiency of high temperature furnaces can be increased by means of efficient heat recovery through air preheating of high efficiency is equivalent to reduction in fuel consumption and to a corresponding saving in greenhouse gas emissions [18].

2.1.2 Power generating equipment

The FLOX® principle is not limited to steel furnaces and can be applied to several high temperature processes. Examples are the Stirling engines, where heat is made available at high temperature with high efficiency, with the purpose of providing combined heat and power in small power generating units [2]. Gas turbine combustors are operated under very specific conditions. Very high combustion densities are common and extremely low NO_x emissions are required. Prototype tests were very promising with NO_x emissions in the single digit [15].

3. PHYSICAL MODEL

In the present work using a burner of FLOX design [19,20] that operates in MILD combustion model, was modeled. Table 1 show details of typical data.

In the paragraphs to follow, the models used in this investigation are explained;

3.1 Model (A) Conventional Combustion Chamber

Air and fuel enter directly into the combustion chamber and the combustion products get out at the right boundary of combustion chamber. This chamber has no flame tube, as shown in Fig. (2).

Table 1. Typical data of MILD combustion

Parameter	Value
Fuel	CH4
Fuel flow rate	0.000267 kg/s
Air flow rate	0.0067 kg/s
Radiant tube diameter	0.09 m
Flame tube diameter	0.04 m
Burner length	0.58 m
Flame tube length	0.41 m
Air inlet section area	88mm ²



Fig. (2) Conventional model

3.2 Model (B) mild combustion chamber (FLOX® combustion chamber)

Air and fuel enter directly into the combustion chamber and the combustion products get out at the front of the chamber. The combustion chamber contains flame tube, as shown in Fig. (3).

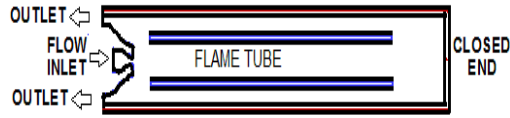


Fig. (3) Combustion chamber using internal recirculation

3.3 Model (C) mild combustion chamber with primary and secondary air.

Air and fuel are premixed at the stoichiometric A/F ratio before being delivered to the combustor. The fire is completed in the combustion chamber using the secondary air that represents excess air and the combustion products get out at the front of combustion chamber. The combustion chamber contains flame tube, as shown in Fig. (4).

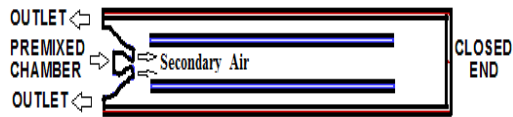


Fig. (4) Combustion chamber with primary and secondary air using internal recirculation.

4. MATHEMATICAL MODELING

This model enables the prediction of combustion chamber performance under different operating conditions. The combustion chamber was considered two-dimensional axi-symmetric and studied using FLUENT 6.2.3 CFD software. Within this study the following models were adopted; the k-ε turbulent viscous model, non-premixed combustion species model, P1 radiation model, thermal and prompt NO_x emission models.

4.1 Governing Equations

Fluent CFD code solves conservation equations for mass, momentum and energy. For flows involving species mixing or reactions, species conservation equations are solved. Additional transport equations are also solved for the turbulent flow [18].

The governing equations of this model can be expressed as follows:

4.1.1 Mass conservation in 2-D cylindrical coordinates.

$$\frac{\partial}{\partial x}(\rho v_x) + \frac{\partial}{\partial r}(\rho v_r) + \frac{\rho v_r}{r} = S_m \quad (1)$$

4.1.2 Species conservation equation.

It predicts the local mass fraction of each species Y_i , through the solution of a convection-diffusion equation for the i th species. This conservation equation takes the following general form:

$$\nabla \cdot (\rho \vec{v} Y_i) = -\nabla \cdot \vec{J}_i + R_i + S_i \quad (2)$$

4.1.3 Momentum conservation equation

For 2-D axisymmetric geometries, the axial and radial momentum conservation equations are given by;

$$\frac{1}{r} \frac{\partial}{\partial x}(r \rho v_x v_x) + \frac{1}{r} \frac{\partial}{\partial r}(r \rho v_r v_x) = -\frac{\partial p}{\partial x} + \frac{1}{r} \frac{\partial}{\partial x} \left[r \mu \left(2 \frac{\partial v_x}{\partial x} - \frac{2}{3} (\nabla \cdot \vec{v}) \right) \right] + \frac{1}{r} \frac{\partial}{\partial r} \left[r \mu \left(\frac{\partial v_x}{\partial r} + \frac{\partial v_r}{\partial x} \right) \right] + F_x \quad (3)$$

and

$$\frac{1}{r} \frac{\partial}{\partial x}(r \rho v_x v_r) + \frac{1}{r} \frac{\partial}{\partial r}(r \rho v_r v_r) = -\frac{\partial p}{\partial r} + \frac{1}{r} \frac{\partial}{\partial x} \left[r \mu \left(2 \frac{\partial v_r}{\partial x} + \frac{\partial v_x}{\partial r} \right) \right] + \frac{1}{r} \frac{\partial}{\partial r} \left[r \mu \left(2 \frac{\partial v_r}{\partial r} + \frac{2}{3} (\nabla \cdot \vec{v}) \right) \right] - 2 \mu \frac{v_r}{r^2} + \frac{2}{3} \mu (\nabla \cdot \vec{v}) + \rho \frac{v_r^2}{r} + F_r \quad (4)$$

where

$$\nabla \cdot \vec{v} = \frac{\partial v_x}{\partial x} + \frac{\partial v_r}{\partial r} + \frac{v_r}{r} \quad (5)$$

4.1.4 Energy conservation equation

For a non-adiabatic, non-premixed combustion model, the total enthalpy can be obtained from the following equation:

$$\nabla \cdot (\rho \vec{v} H) = \nabla \cdot \left(\frac{k_t}{cp} \nabla H \right) + S_h \quad (6)$$

Under the assumption that the Lewis number (Le) = 1, the conduction and species diffusion terms combine to give the first term on the right-hand side of the above equation while the contribution from viscous dissipation appears in the non-conservative form as the second term. The total enthalpy H is defined as

$$H = \sum_i Y_i H_i \quad (7)$$

where Y_i is the mass fraction of species i , and

$$H_i = \int_{T_{ref,i}}^T c_{p,i} dT + h_i^0(T_{ref,i}) \quad (8)$$

C_p is the specific heat at constant pressure, $h_i^0(T_{ref,i})$ is the enthalpy of formation of species i at the reference temperature $T_{ref,i}$ and $T_{ref,i} = 298.15$ K.

5. COMPUTATIONAL MODEL

The computational domain consists of three subdomains: a fluid domain representing the real combustion chamber and two solid domains representing the flame and the radiant tubes. It was chosen to simulate those two solid subdomains to take into account heat transfer, due mainly to radiation effects between the reacting gas

and the solid boundaries.

Chiara Galletti [12] studied a 3-dimensional model and an axisymmetric model and mentioned that the axisymmetric simulations could be performed, allowing lower computational cost, and predicted NO_x corrected by assuming 15–20% larger recirculation degrees. At present work, 2-D axisymmetric model is used since the combustion chamber is symmetric around the centerline.

5.1 Mesh Generation

A sector of the combustion chamber with angle of 1° is considered. This is used to reduce computational time in case of symmetric geometries. The geometry and mesh of this model were generated separately using GAMBIT; the preprocessor tool for computational mesh generation of the FLUENT 6.3.2 package. GAMBIT generates structured (mapped), unstructured (paved), and hybrid meshes. The mesh structure considered is the unstructured (paved). The mesh data for the three models are shown in table 2.

6 RESULTS AND DISCUSSION

6.1 Comparison between Three Combustion Chamber Geometries

Comparison between model (A) which adopts the conventional concept and models (B,C) which adopt MILD combustion concept is present. Using the commercial Ansys Fluent 6.3 and the data in table (1) above. Figures. (5) to (8) show the temperatures profile and NO_x emissions profile along the length of chamber for the different models. A thorough investigation of these profiles clarifies the following:

- (1) For model (A) the temperature increases from 660K at $X/L=0$ to reach the maximum temperature of 1956 K at $X/L=0.5$ and decreases

Table 2. Mesh data for three models

Cases		Cell-quadrilateral	Faces	Nodes
Conventional Combustion Concept	Model (A)	6968	14252	7285
MILD Combustion Concept	Model (B)	7258	14832	7575
	Model (C)	7340	15015	7676

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to 1710 K at the exit of the chamber. in model (B) the temperature decreases from 1437 K at $X/L=0$ to reach the minimum temperature 1330 K at $X/L=0.1$ and increases to 1887 K at $X/L=1$. in model (C) the temperature increases from 1424 K at $X/L=0$ to reach maximum 1742 K at $X/L=1$. Note that from Fig. (5), the temperature is nearly constant from $X/L=0.5$ to $X/L=1$.

For models (B,C), mixing inlet air and fuel with the exhaust gases tend to more uniform temperature.

- (2) The difference between the maximum and minimum temperature for model (A) is 1300K, model (B) is 560K and model (C) is 320 K. From these results, it is evident that model (C) has more uniform for temperature distribution and consequently low a thermal stresses.

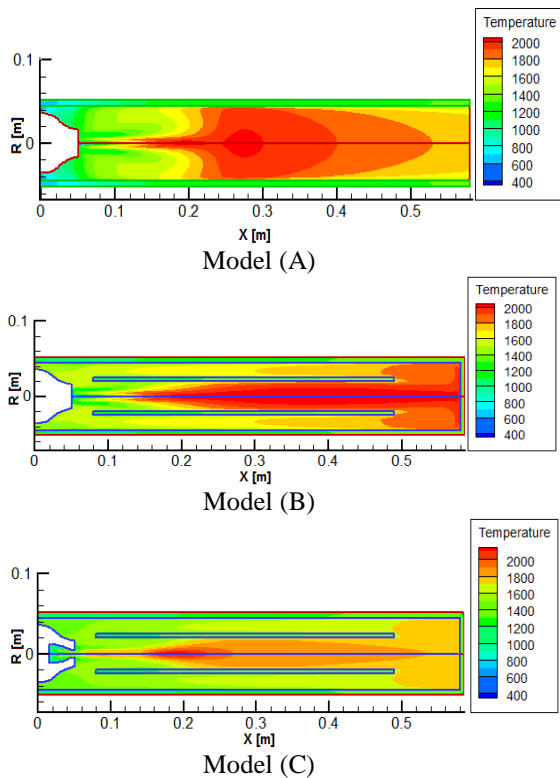


Fig. (5) Temperature contours for different models

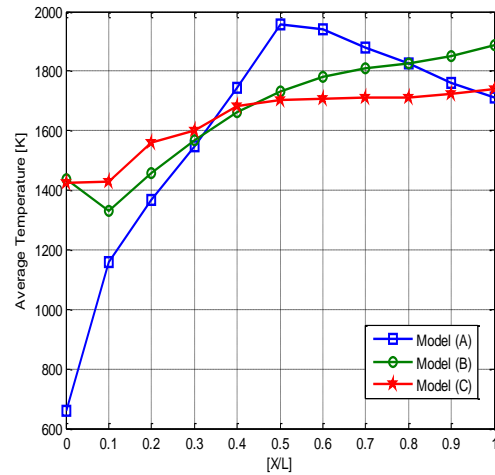


Fig. (6) Models temperature distribution along combustor

- (3) From Fig. (6), it can be noticed that the change in temperature from $X/L=0$ to $X/L=1$ is more uniform for model (C) compared with other models the maximum temperature is reached at $X/L=0.5$ for model (A), $X/L=0.3$ for model (B) and $X/L=1$ for model (C).
- (4) Model (C) has nearly the lowest temperature among the different models.
- (5) Model (C) has minimum outlet temperature. Note that the outlet for models (A) at $X/L=1$ but models (B) and (C) at $X/L=0.0$.
- (6) For model (B), a part of the reaction zone out from the combustor (flame out from combustor).
- (7) NO_x concentration is minimum for model (C) is nearly constant along combustion chamber as shown in Fig. (7) and (8).
- (8) The combustion using internal exhaust gas recirculation is more stable, smooth and flameless.
- (9) Conventional model (A) outer wall is directly exposed to the maximum temperatures, as shown in Fig. (5), but the outer walls in models (B) and (C) are protected by the flame tube.
- (10) Model (C) has the minimum emission of CO Fig. (11) and (12).

According to the aforementioned results combustion with MILD concept gives lower temperature gradient, compared with conventional combustion. Thus, in a properly designed HiTAC combustion

chamber, the combustion takes place almost "isothermally" where avoid local high temperature spots.

NO_x are one of the most regulated air pollutants due to their smog formation.

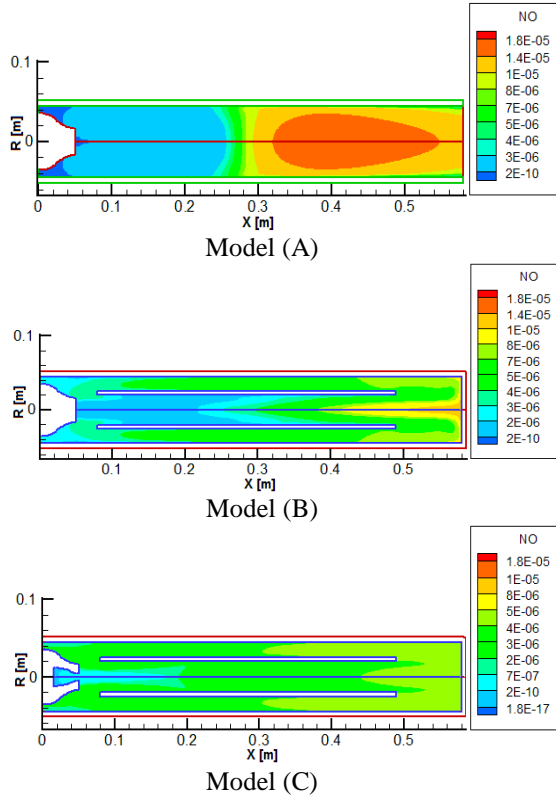


Fig. (7) NO_x emissions contours for different models

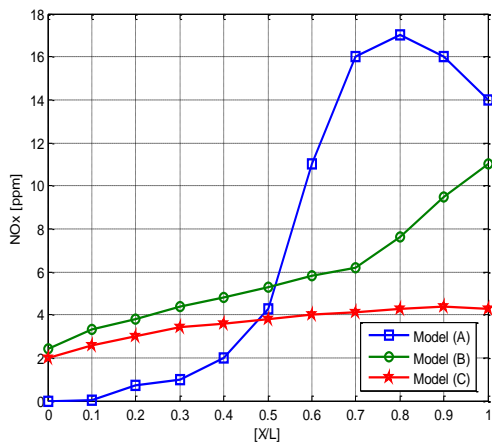
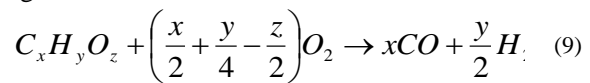


Fig. (8) Models NO_x distribution along combustor

In general, fractions higher than 90% of the NO_x present in the exhaust of a combustion system are in the form of NO. Most of the NO is produced around the reaction zone. Oxidation to NO₂ eventually occurs away from the combustion region.

This oxidation increases as the combustion temperature increases. Accordingly, NO_x emissions for the conventional model are maximum with levels nearly 17ppm and 14ppm at outlet while MILD model (B) produces less than 11ppm. MILD model (C) with primary and secondary air gives NO_x levels less than 5ppm and less than 3ppm at outlet. This is shown in Fig. (8).

The complete combustion of hydrocarbon fuels produces CO₂ and H₂O. CO emission is a result of incomplete combustion. Despite the complexity of hydrocarbon combustion chemistry, the oxidation process can be simplistically described by a two-step global mechanism;



In equation (9) fuel is partially oxidized to carbon monoxide and then in equation (10) the final conversion of carbon monoxide to carbon dioxide occurs. Figure (9) shows CO₂ emissions for the three models.

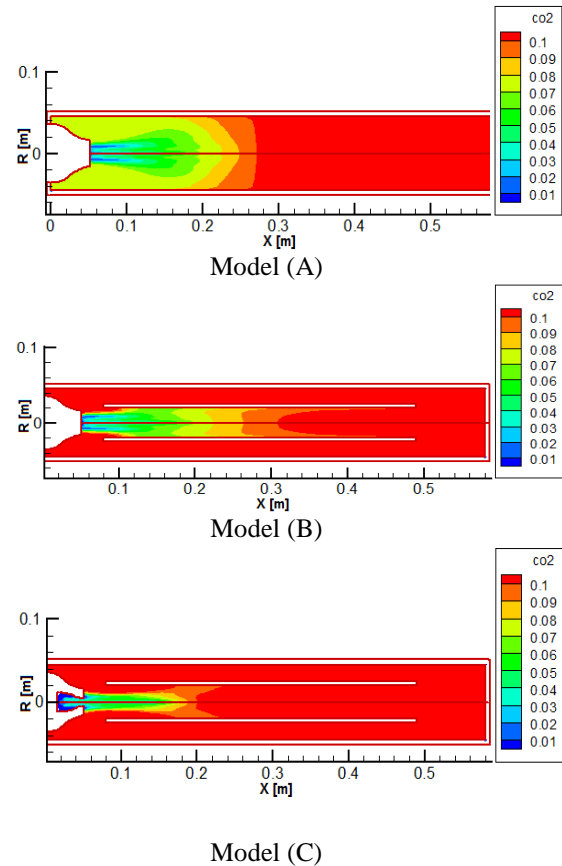


Fig. (9) CO₂ emissions contours for different models

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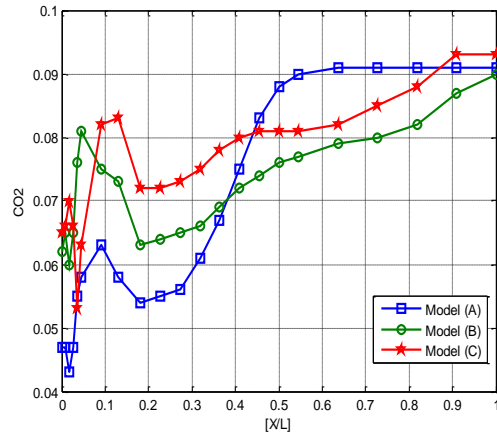


Fig. (10) Models CO₂ distribution along combustor

Although CO₂ emissions show moderate concentration near the inlet of the chamber of model (A), it is increased rapidly and becomes the highest rate among the models by the exit as shown in Figs. (9) and (10). This is attributed to the highest temperature achieved by model (A) as indicated in Figs. (5) and (6).

On the other hand the peak of CO emission corresponds to lower temperatures. Accordingly, it can be concluded that model (C) has the lowest CO concentration throughout the combustion chamber. This is shown in Figs. (11) and (12)

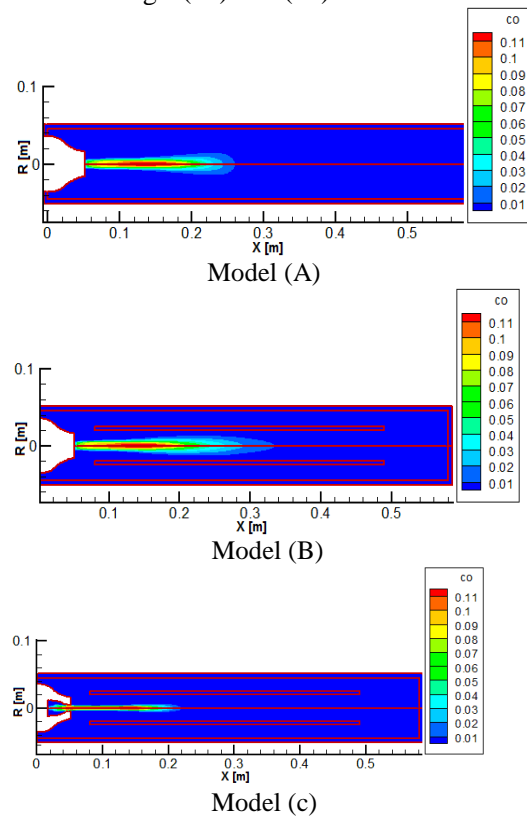


Fig. (11) CO emissions contours for different models

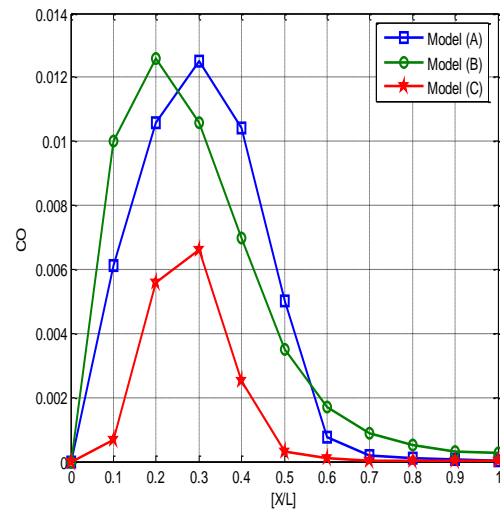


Fig. (12) Models CO distribution along combustor

6.2 Factors Affecting Combustion Characteristics

6.2.1 Effect of preheated air

Figures (13) and (14) shows the effect of combustion air temperature on combustion chamber temperature and NO_x emission. As the inlet combustion air temperature increases from 1050K to 1200K, the average temperature increases linearly from 1674K to 1770K for model (A), from 1726K to 1805K for model (B) and from 1673K to 1746 for model (C).

However NO_x emissions, they increased from 17ppm to 69ppm for model (A), from 12ppm to 37ppm for model (B) and from 4.5ppm to 12ppm for model (C) .

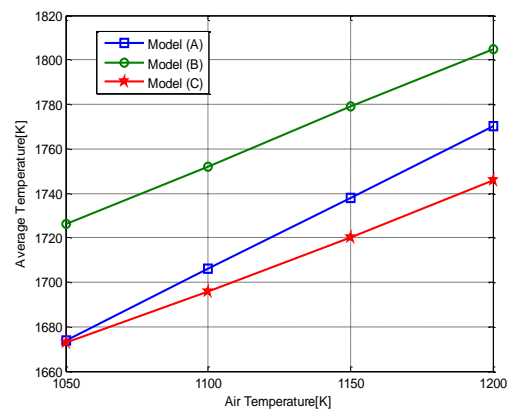


Fig. (13) Effect of air preheating on average temperature

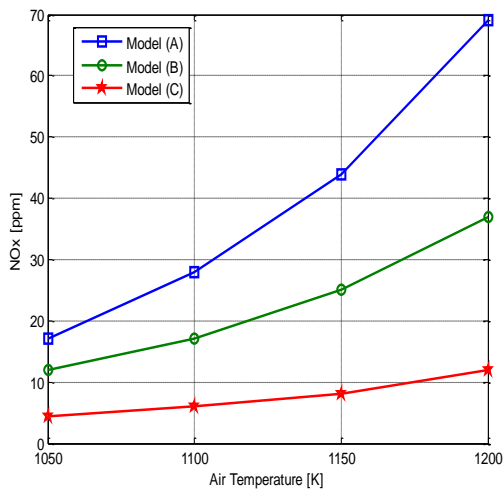


Fig. (14) Effect of air preheating on NO_x emissions

As shown, NO_x increasing rate for model (C) is the smallest compared with model (A) and model (B).

6.2.2 Effect of excess air

Figure (15) shows the temperature profiles and NO_x with air excess variations. It is found that for the same air inlet cross sectional area, when the air excess increases from 0% to 50% it is observed that average temperature for all models decrease. This decrease for model (A) is from 1789 K to 1667 K, model (B) from 1822K to 1719 K, and model (C) from 1707 K to 1670 K. NO_x emissions decrease for model (A) from 150 ppm to 15 ppm, for model (B) from 68 ppm to 10 ppm and for model (C) from 5.2 ppm to 4.4 ppm as indicated in Fig. (16).

6.2.3 Effect of fuel and air dilution with CO₂, H₂O or N₂

The fuel stream was diluted using either N₂, CO₂ or H₂O to test its effect on average temperature and NO_x emissions for conventional and MILD combustion models as shown in figures (17) and (18). NO_x is used here as an indicator for judging of the best combustion model from environmental point of view. Fuel and air mass flow rates were kept constant during these predictions. Fuel dilution with inert gases causes a reduction in NO_x emission and suppresses any flame propagation inside the furnace. Such dilution results in a shift in the stoichiometric mixture fraction toward the rich side which has the highest scalar of dissipation and ensures the mixture of fuel

and air is diluted before it can react. Dilution with H₂O rather than N₂ and CO₂ decreases further the value of NO_x emissions because of the higher specific heats of H₂O. For air dilution, the air flow rate has been fixed at 0.0067 kg/s which corresponds to 46% excess air ratio, and the diluents of CO₂, N₂ and H₂O are added in the oxidizer stream.

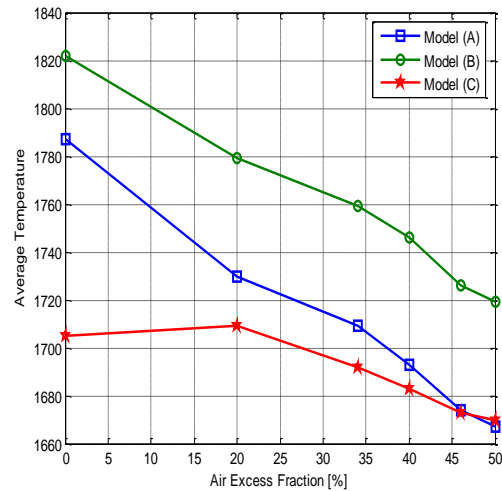


Fig. (15) Effect of excess air percentage on average temperature

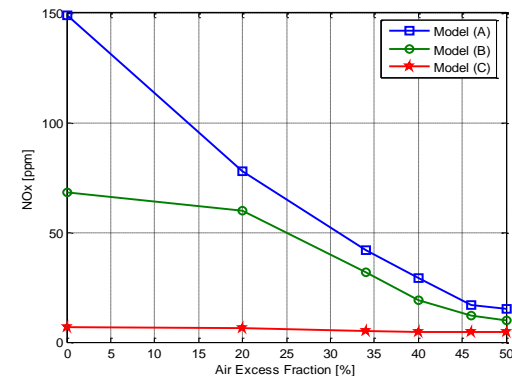


Fig. (16) Effect of excess air percentage on NO_x emissions

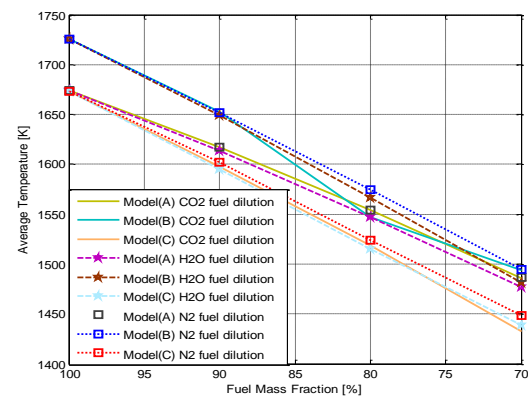


Fig. (17) Effect of fuel dilution by CO₂, H₂O and N₂ on average temperature

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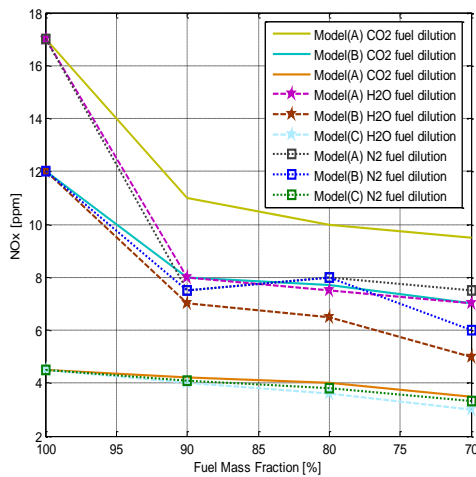


Fig. (18) Effect of fuel dilution by CO₂, H₂O and N₂ on NO_x emissions

Figures (19) and (20) show temperature profiles and values of NO_x product as function of oxygen mass fraction (X_{O₂}). X_{O₂} = 0.23 has been considered as the base case without dilution. The average temperature and NO_x emission of burner have been reduced with the decrease of X_{O₂}. For example dilution with CO₂ reduces the average temperature from 1674 K to 1605 K for model (A), from 1726 K to 1630 K for model (B) and from 1673 K to 1597 K for model (C). The flame temperature gradually reduces in the downstream and maximum average temperature position is shifted downstream with reduction of X_{O₂}. NO_x emission decreases from 17ppm to 5 ppm for model (A), from 12ppm to 0.25ppm for model (B) and from 4.5ppm to 0.03ppm for model (C).

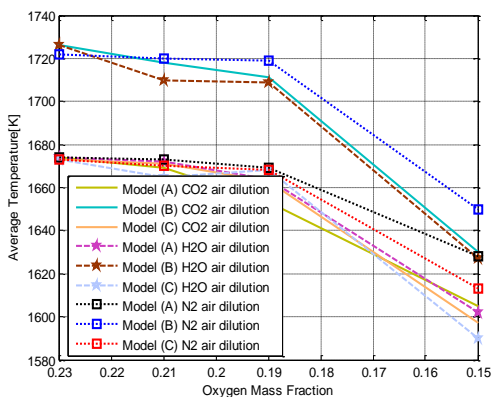


Fig. (19) Effect of air dilution by CO₂, H₂O and N₂ on average temperature

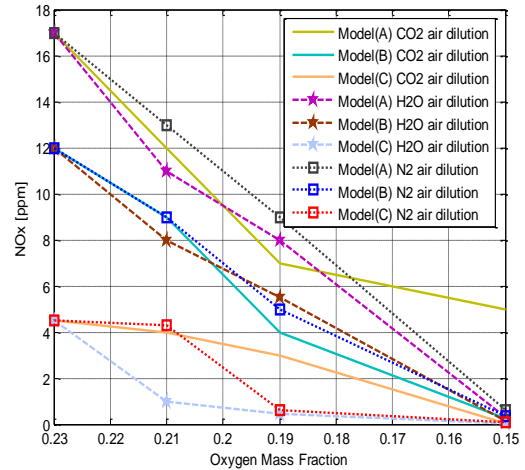


Fig. (20) Effect of air dilution by CO₂, H₂O and N₂ on NO_x emissions

7 CONCLUSIONS

According to the computational study and the analysis for studying conventional and FLOX® burner models we obtain the following results have been obtained:

1. MILD combustion produces an invisible flame and very low NO_x emissions.
2. MILD combustion avoids exit high temperatures of flue gases.
3. Temperature distribution is more homogenous for MILD models than conventional models. This produces minimum thermal stresses in burner walls.
4. MILD combustion has less restrictions on fuels because less flame stability is required constraints
5. For the same conditions and length, the conventional combustor (model A) has higher exhaust temperature and often flame exist at out the burner.
6. Air preheating results in an increase in average temperature. This leads to higher thermal NO_x emissions rate for conventional combustor (A) compared with FLOX burner models (A&B).
7. When dilution gases enter with air from air cross sectional area, the value of average temperature and NO_x products decrease.
8. Dilution of fuel stream with inert gases reduces NO_x emission and

- suppresses any flame propagation inside the combustion chamber.
9. Dilution with H₂O rather than N₂ and CO₂ decreases further the value of NO_x because of the higher specific heats of H₂O compared with N₂ and CO₂.
 10. When excess air percentage increases, the average temperature observed in the burner decreases and consequently NO_x emissions decrease too.
 11. Using primary and secondary air (Model C) leads to more uniform temperature combustion and lower NO_x among the models.

Nomenclatures

F	Force, [N]
H	Total Enthalpy [J]
P	Pressure [pa]
Le	Lewis No.
R _i	Net Rate of Production of Species <i>i</i> by chemical reaction
S _i	Source term of the <i>i</i> th species
Sh	Heat of Chemical Reaction [J/mol]
Sm	Mass source term from the dispersed phase
T	Temperature, [K]
v	Velocity [m/sec]
x	Mole fraction
Y	Mass fraction of the <i>i</i> th species in the mixture
ρ	Density [kg/m ³]
subscript	
av	Average
r	Radial
x	Axial

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