Simulation of Non-Premixed Natural Gas Flame

Ashish Puriya[1], Rajesh Gupta[2]

[1]P.G. Student, Department of Mechanical Engineering, M.A.N.I.T, Bhopal, India

[2]Associate Professor, Department of Mechanical Engineering, M.A.N.I.T, Bhopal, India

Abstract: ***A numerical study to model a non-premixed natural gas flame in a 300 kW gas turbine combustor was undertaken by solving the appropriate form of governing equations in an axi-symmetric, two-dimensional domain. The numerical simulations are performed using RNG k-ε for turbulent transport along with a suitable radiation model for various wall conditions. The results are then compared with measured data for mean flow field temperature and species concentrations. The comparison of computed results with experimental data is found plausible.***

Keywords: Natural gas, Non-premixed flame, 300 KW combustor, Spices concentration, Swirl flow.

Nomenclature

C_1, C_2, σ_k, σ_ϵ, α Constants used in the k-ϵ model
C_p Specific heat of the gas mixture
g Acceleration due to gravity
m_l Mass fraction of species l
μ_t Turbulent viscosity
mw_l Molecular weight of species l
mw_{mix} Molecular weight of the gas mixture
R_{ij} Turbulent stress tensor
τ Stress tensor
U $(u,v)^T$

1. Introduction

Recently interest in modeling of furnace has been increased greatly to achieve better mixing and improve performance of furnace and reduce pollutant emission, the development of industrial burners, boilers, and furnaces with higher performance, (particularly higher efficiency, and lower pollutant emissions) is typically the main goal in the design of combustion equipment. The most effective way of achieving the goal is to increase the thermal efficiency of industrial furnaces and, by doing so, to decrease the CO_2 emissions per ton of product. An increase in the thermal efficiency requires both high process temperatures and efficient heat recovery systems.

In this paper presents non-premixed flame of natural gas in combustion model for an industrial furnace by using numerical simulations. One of the most important practical problems of furnaces is to fulfill the limitations of pollution concentration of the flue-gas. The end product of combustion can contain CO_2, CO, and pollutants, some of which can be poisonous, e.g. CO, etc and predictions for reacting flow in a non-premixed natural gas flame, species concentration of [CO, O_2, CO_2] and axial, tangential velocity, temperature profile compared to an initial set of data collected by [[1]] and a recent set of additional measurements of the same flame documented by [[3]]. In recent study, the stoichiometry and the flame structure of the leading edge, an anchor point, of a non-premixed methane flame were studied [[5]]. CFD simulations of soot formation in various hydrocarbon non-premixed flames were studied [[6]]. The experimental and numerical study of non premixed flame behavior, like temperature profile and their stabilization limits were investigated for various fuel-air flow rates in narrow combustor is studied [[7],[8]]. The structural measurements of the opposed tubular non-premixed flames generated by hydrocarbon fuels as well as understanding the effect of curvature on flame temperature is studied [[9]].

The main goal of this work is to evaluate the use of a low-Mach number model for combustion and a higher order projection methodology in simulation of a natural gas-fired furnace. As such, we assume that the perfect gas law applies. We also use several relatively simple, standard sub-model namely, a two step reduced mechanism, a RNG k-ε for turbulent transport, and eddy-dissipation model for turbulent combustion.

2. Mathematical Model

For a multi-component perfect gas in a low Mach number [M<0.3] axi-symmetric swirl flow, the following constraint on divergence of the velocity is effectively satisfied [5, 6].

$$\nabla.(u,v)^T = \frac{1}{T}\frac{DT}{Dt} + mw_{mix}\sum_l \frac{1}{mw_l}\frac{Dm_l}{Dt} \quad [1]$$

The governing differential equations consist of the divergence constraint [1] and the following evolution equations for density, velocity, enthalpy, temperature, species mass fractions, and turbulent kinetic energy and dissipation rate:

$$\frac{\partial\rho}{\partial t} + \nabla.\rho U = 0 \quad [2]$$

$$\frac{\partial\rho(u,v,w)^T}{\partial t} + (\nabla\rho Uu, \nabla.\rho Uv, \nabla.\rho Uw)^T + \left(-\frac{\rho w^2}{r}, \rho g, \frac{\rho uw}{r}\right)^T = -\nabla\pi + \nabla.\tau \quad [3]$$

$$\frac{\partial\rho h}{\partial t} + \nabla.\rho Uh = \nabla.\lambda_e\nabla T - \nabla.q_{rad} + \nabla.\sum_l \rho h_l(T)D_e\nabla m_l \quad [4]$$

$$\rho c_p\frac{DT}{Dt} = \nabla.\lambda_e\nabla T - \nabla.q_{rad} + \nabla.\sum_l \rho h_l(T)D_e\nabla m_l - \rho\sum_l \frac{Dm_l}{Dt}h_l(T) \quad [5]$$

$$\frac{\partial\rho m_l}{\partial t} + \nabla.\rho U m_l = \nabla.D_e\rho\nabla m_l - R_l \quad [6]$$

$$\frac{\partial\rho k}{\partial t} + \nabla.\rho Uk = \nabla.(\alpha\mu\nabla.k) - R_{ij}\frac{\partial U_i}{\partial x_j} - \rho\epsilon \quad [7]$$

$$\frac{\partial\rho\epsilon}{\partial t} + \nabla.\rho U\epsilon = \nabla.\left(\frac{\mu_t}{\sigma_t} + \mu\right)\nabla\epsilon - C_1.\frac{\epsilon}{k}R_{ij}\frac{\partial U_i}{\partial x_j} - C_2\rho\frac{\epsilon^2}{k} \quad [8]$$

The right hand sides of [5] and [6] are used to evaluate DT/Dt and Dm_l/Dt in [1]. The RNG values of the k-ϵ model constants are used, and effective diffusivities μ_e, λ_e, and D_e are defined in the usual way [8]. The enthalpy h is defined by:

$h = \sum_l m_l h_l(T)$ [9]

Where $h_l(T)$, the specific enthalpy of species l, includes the heat of formation of l. When needed, T is computed using h, m_l and [9]; equation [5] is used only in evaluating [1].

3. Problem Description

The flame considered is un-staged natural gas fired in a combustor (as shown in fig. 1) with swirl air. The combustor is vertically fired and of octagonal cross-section, with a conical furnace hood and cylindrical exhaust duct. The combustor wall may either be lined with refractory material or water-cooled. The burner features 24 radial fuel injection holes and a bluff center body. Air is introduced through an annular inlet and movable swirl blocks are used to impart swirl.

Two configurations were considered during experiments, one with water cooled furnace walls and the other with the furnace wall lined with refractory. The current study presents results only for the case with the refractory lining, i.e., the so-called "hot-wall" configuration. The problem was modeled as axi-symmetric and appropriate area adjustments were made to account for the 2D representation of an inherently 3D problem.

4. Boundary Conditions

The boundary conditions are given as follows:

At inlet natural gas enters at 22.7 kg/hr at 308 K, while air enters the burner at 31.35 m/sec with parabolic profile and swirl number is 0.56. Dirichlet boundary conditions are applied at the inlet of chamber.

Inlet:

At inlet (x=0), $\partial p/\partial x=0$

Wall:

No-slip boundary conditions apply at walls, and the experimental data are put as wall temperature distribution, while the rest of the problem parameters, e.g. species mass fraction, are considered to be zero gradient along the furnace diameter.

At $r = r_0$, $u = 0$, $v = 0$, $w=0$, $\partial m_l/\partial r=0$

Outlet:

At the outlet, the atmospheric pressure is fixed. And $\partial/\partial x=0$ for other variables.

The boundary conditions applied on the furnace are sketched in Fig. 1.

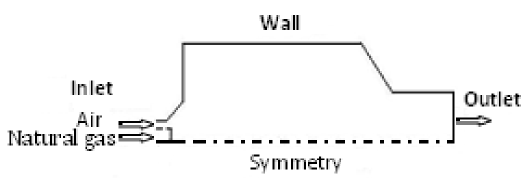


Figure 1: Schematic of the 2-D computational domain.

5. Results

The computational results with RNG k-ϵ and DO [Discrete ordinates] model are compared with measured radial profile of axial and tangential component of velocity, temperature, and mole fractions of CO, O_2, and CO_2 ,At 27 mm and 343 mm downstream of burner throat in transverse direction, shown in fig.2.

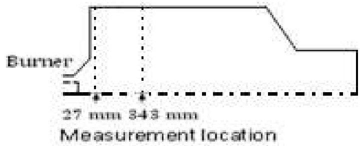


Figure 2: Location of measurements

Fig .3 shows the various parameters contours of flame in furnace, fig 3(a, b) shows contours of axial and tangential velocity and direction shows the flow direction in furnace. Fig. 3(c) shows temperature with highest temperatures occurring in the mixing layer between the fuel and oxidant streams. Fig. 3(d) shows the oxygen contour downstream of burner throat. The CO profile is shown in figure 3 (f) with largest concentrations shown in red color in narrow reaction layer near the burner throat.

A selection of results obtained at 27 mm downstream from the burner throat is shown in Fig 4. The predicted axial velocity, temperature, and CO, O_2, CO_2 mole fraction show reasonable agreement with the experiment implying accurate predictions of the flow and the concentration fields. Negative values for axial velocity indicate that there is an internal recirculation zone in front of the fuel nozzle due to the sudden expansion and swirl velocity of air exiting the throat. This zone entrains most of the fuel exiting, resulting in a very high reaction rate; this produces a higher temperature and higher concentration of product gases compared to the other parts of the furnace.

Similar results were obtained for 343 mm downstream from the burner exit as shown in Fig 4. Indicate that the largest discrepancy between the model and experimental results is in the reaction zone. This is likely due to the two-step reduced mechanism, which overestimate the reaction rates. The other sources of error are the under prediction in the turbulent viscosity of the RNG k-ε, the approximation of constant radiative properties in the radiation model, and the modeling of the fuel inlet holes as an annular slot. CO concentrations are over-predicted at 343 mm. A possible explanation for this disagreement is that our implementation of the eddy dissipation model uses the same rate constant in both the natural gas and the CO steps.

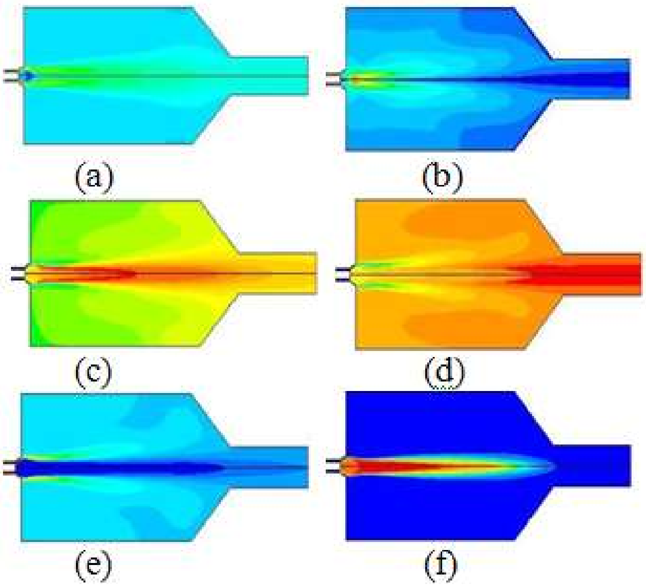
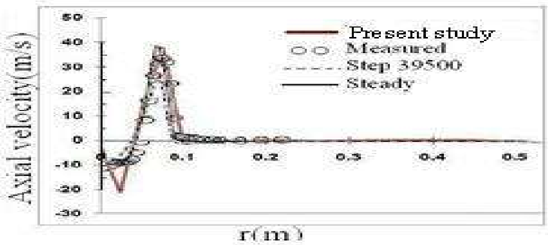
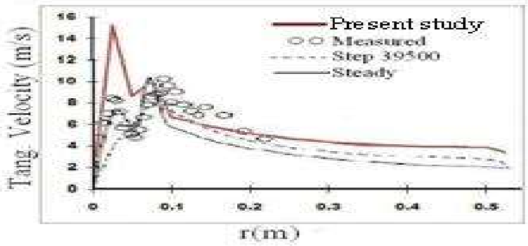


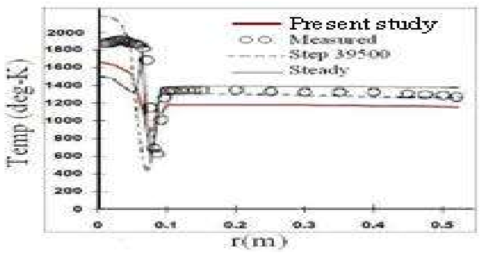
Figure 3: Contours diagram of (a) axial velocity, (b) tangential velocity, (c) temperature, (d) mol fraction CO_2, (e) mol fraction O_2 (f) mol fraction CO.



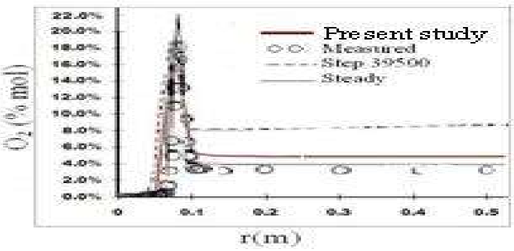
(a)



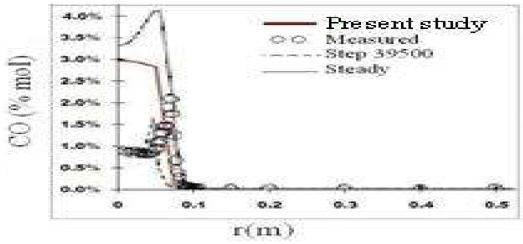
(b)



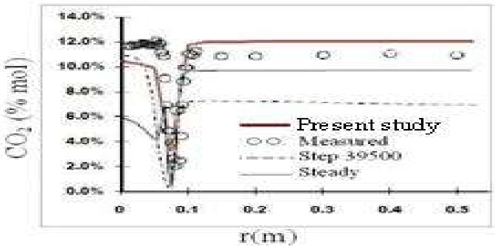
(c)



(d)

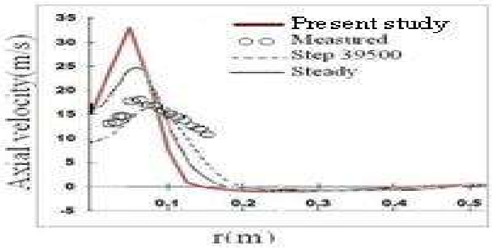


(e)

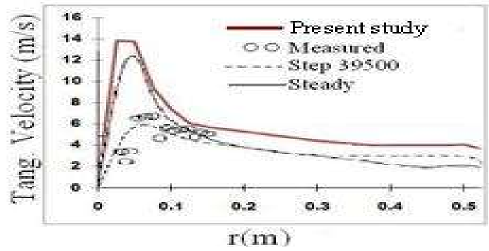


(f)

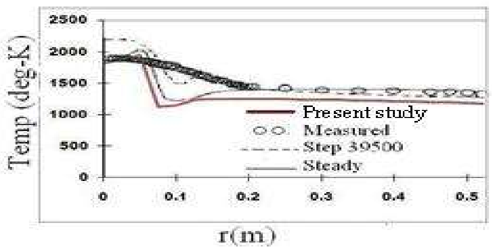
Figure 4: Comparison of various flame parameters with measurements of [[2]] at 27 mm burner throat, (a) axial velocity, (b) tangential velocity, (c) temperature, (d) mol fraction of CO_2, (e) mol fraction of O_2, (f) mol fraction of CO.



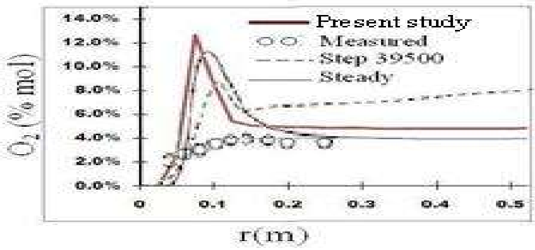
(a)



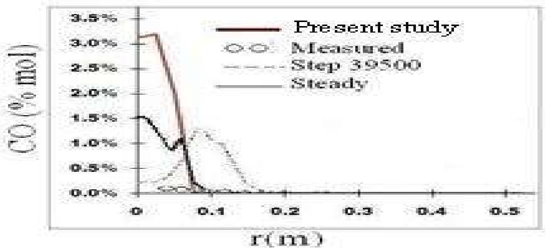
(b)



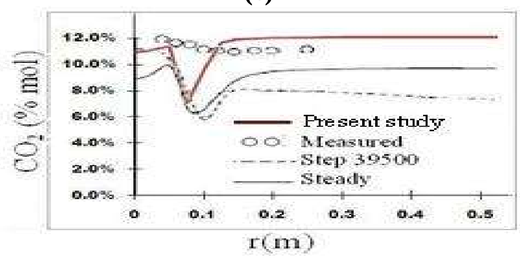
(c)



(d)



(e)



(f)

Figure 5: Comparison of various flame parameters with measurements of [[2]] at 343mm burner throat, (a) axial velocity, (b) tangential velocity, (c) temperature, (d) mol fraction of CO_2, (e) mol fraction of O_2, (f) mol fraction of CO.

6. Conclusion

In this study, we have done modeling of non-premixed flame of natural gas with swirl air combustor and compared the result with measured data. We used RNG k-ε model for turbulence and discrete ordinate model for radiation. The model capture most of the trends well including the velocity, temperature, and species profile for CO, O_2, and CO_2. In this study The RNG k-(ϵ) model is used because the RNG model has an additional term in its (ϵ) equation that significantly improves the accuracy for rapidly strained flow and the effect of swirl on turbulence is included in the RNG model, so enhancing accuracy for swirling flow.

References

[1] A. Sayre, N. Lallemant, J. Dugue and R. Weber, Scaling Characteristics of Aerodynamics and Low NOx Properties of Industrial Natural Gas Burners. The Scaling 400 StudyPart IV: The 300 kW BERL Test Results, International Flame Research Foundation, 1994.

[2] R.B. Pember, P. Collela,L.H. Howell, A.S. Almgren, J.B. Bell, W.Y. crutchfield, V.E. Beckner, K.C. Kaufman, W.A. Fiveland, J.P. Jessee. Lawrence Livermore National Laboratory, The modeling of a laboratory natural gas fired furnace with a higher-order projection method for unsteady combustion, 1996.

[3] Kaufman, K.C., and Fiveland, W.A., 1996, "Pilot Scale Data Collection and Burner Model Numerical Code Validation," topical report for GR! Contract 5093-260-2729, to be published.

[4] M. Sinoda, R. Tanaka and N. Arai, "Optimization of Heat Transfer Performances of a Heat- Recirculating Ceramic Burner during Methane/Air and Low-Calorific-Fuel/Air Combustion," Energy Conversion and Management, vol. 43, pp. 1479-1491, 2002.

[5] Attachment structure of a non-premixed laminar methane flame.(Yuji Ikedaa,*, Jean Luc Beduneaub, 2004)

[6] A simplified model for soot formation and oxidation in CFD simulation of non-premixed hydrocarbon flames.(Christopher W. Lautenbergera,, John L. de Risb, Nicholas A. Dembseya, Jonathan R. Barnetta, Howard R. Baumc, 2005).

[7] Non-premixed flame characteristics of opposed methane jets in coaxial narrow air stream tubes (Nam Il Kim *, Young Min Yun, Min Jung Lee, 15 March 2010).

[8] Numerical study of opposed non-premixed jet flames of methane in coaxial narrow air tube. (Nam Il kim, 14 September 2011).

[9] Structural study of non-premixed tubular hydrocarbon flames.(ShengtengHu1, Robert W. Pitz,17 October 2008)

[10] G. J. Hesselmann, Optimization of Combustion by Fuel Testing in a NOx Reduction Test Facility, Fuel, vol. 76(13), pp. 1269-1275

Author Profile

Ashish Puriya received B.E degree in Mechanical Engineering from I.E.T D.A.V.V. Indore in 2010. He is presently pursuing P.G. with Thermal engineering from M.A.N.I.T Bhopal, India.

Rajesh Gupta received PhD in Thermal engineering, M.Sc in Applied Mathematics from Technical University of Nova Scotia, Canada in 1990, M.Sc. Engg., Mechanical Engineering from University of New Brunswick, Canada, 1993, B.E mechanical from M.A.CT, 1984.Presently he is associate professor in M.A.N.I.T (Deemed University).