Indian Journal of Engineering

ANALYSIS

ISSN 2319 – 7757 EISSN 2319 –7765

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Publication History
Received: 29 April 2016
Accepted: 26 May 2016
Published: 1 July 2016

Citation
Rajesh Gupta, Pankaj Dubey, Gopal Kumar Deshmukh. Combustion simulation in a differentially heated diffuser under free convection. *Indian Journal of Engineering*, 2016, 13(33), 437-445

COMBUSTION SIMULATION IN A DIFFERENTIALLY HEATED DIFFUSER UNDER FREE CONVECTION

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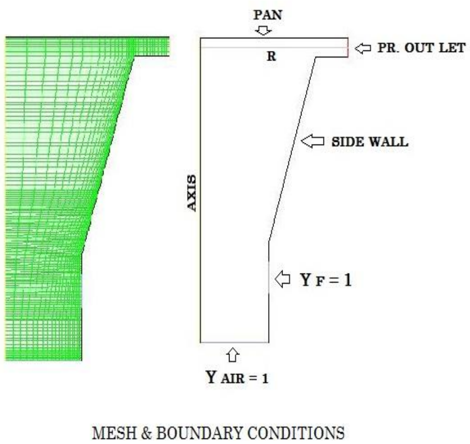
ABSTRACT

The present work is aimed to explore combustion in a vertical diffuser with an emphasis on the buoyancy-induced nature of the flow. This combustion problem is similar to the combustion of pyrolysis gases in an annular vertical diffuser which is widely used in rural areas for cooking. The analysis of the problem is complicated owing to the coupling amongst natural convection flow, heat transfer and combustion. A simple finite reaction rate model for a single component fuel is presented for the combustion process. Then, the fuel is taken to be a mixture of combustible gases and an Arrhenius-type single step reaction is assumed between each of the combustible gases and oxygen. In each stage of modelling, temperature profiles are generated for various tube wall temperatures and fuel inflow rates. Results indicate that the maximum flame temperature is located close to the radial diffuser. The flame tends to move away from the wall with higher volatiles flow rate and higher wall temperatures also. It was also seen that maximum flame temperature increases with increase in tube wall temperature and power input, maximum flame temperature region spread in radial diffuser. Heat transfer efficiency increase with increase in power input.

Keywords: Heat transfer efficiency, wood-volatiles, vertical diffuser, natural convection, mixture fraction formulation.

1. INTRODUCTION

Research and development of cook stoves and vertical combustors have been receiving its due share of attention at least in the developing world where a large chunk of the population still resides below a standard of living. This part of the population generally resorts to the use of wood, wood volatiles or any other bio mass fuel available easily for its cooking needs. The issue of stove or combustor is therefore quite sensitive and has direct implications on the socio-economic developments of the poorest of the poor human beings. A general layout of vertical diffuser combustor is shown in Figure 1 and 2. A combustion zone is sustained in the central passage by a buoyancy-induced upward flow of air and a radial inflow of combustible volatile gases from the vertical combustor. The hot flue gases transfer heat to the pan placed above the conical tube. Scientific investigation of such combustor is difficult because of the coupled phenomena of natural convection flow, heat transfer, and flaming combustion. In this work a certain radial flow rate of combustible volatiles has been prescribed from the inner surface of a vertical tube. In the present work, the combustion phenomenon of fuel gas in a buoyancy-induced airflow in a vertical combustor is modeled under the assumptions of a finite reaction rate. The volatile flow rate and the inner wall temperature were parametrically varied over a restricted range for studying their effects on combustion.



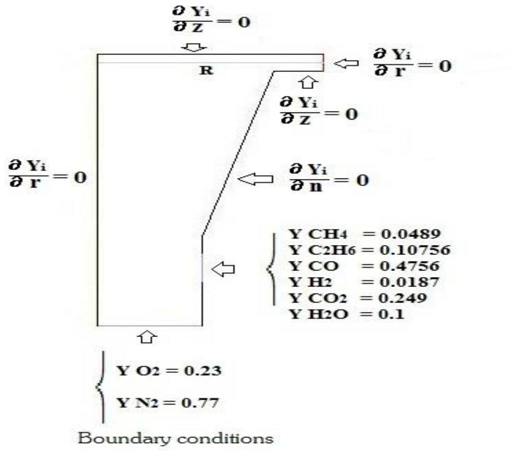


Fig. 1 Mesh & general layout Fig.2 Vertical diffuser combustor

2. MATHEMATICAL FORMULATION

The flow, through the geometry shown in Figure 1, is modeled as axisymmetric. The computational domain is depicted in Figure 2. It consists of three regions, viz., straight tubular portion, and diffuser like conical portion known as axial diffuser and finally region between the stove and top disk is known as radial diffuser. The cylindrical polar coordinates r-z is used the present analysis. H is the total height of stove; s is the spacing between the stove and the pan. The flow is considered steady and turbulent and fluid is assumed to be incompressible. Since large variation of temperatures is encountered in stove, the Boussinesq approximation commonly used in natural convection flows is not employed in the present analysis. Consequently, all the fluid and thermodynamic properties including density are taken as functions of temperature. The equations of mass, momentum and energy describing the flow are written in non-dimensional form, represented by hat on the top of the variable.

The present investigation mainly concentrates on modeling the actual combustion process along with fluid flow heat transfer. The fuel mixture is taken as combination of CH_4, C_2H_6, CO, H_2, CO_2, and H_2O which are typically part of volatile emerging from pyrolysis of wood. Composition of volatiles for CH_4, C_2H_6, CO, H_2, CO_2, and H_2O is taken as 0.0489, 0.10756, 0.4756, 0.0187, 0.249 and 0.1 respectively such that their combined calorific value becomes equal to that of typical wood sample.

MASS CONSERVATION EQUATION:

$$\frac{1}{\hat{r}} \frac{\partial \hat{\rho} r \hat{V}_r}{\partial \hat{r}} + \frac{\partial \hat{\rho} \hat{V}_z}{\partial \hat{z}} = 0 \qquad (1)$$

RADIAL MOMENTUM EQUATION:

$$\frac{1}{\hat r}\frac{\partial \hat\rho\,\hat r\hat V_r^2}{\partial \hat r}+\frac{\partial \hat\rho\hat V_r\hat V_z}{\partial \hat z}=-\frac{\partial \hat p}{\partial \hat r}-\hat\mu\hat V_r Da+\frac{4}{3}\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat r\hat\mu\frac{\partial \hat V_r}{\partial \hat r}\right)-\frac{2}{3}\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat r\hat\mu\frac{\partial \hat V_z}{\partial \hat z}\right)$$
$$-\frac{2}{3}\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat r\hat\mu\frac{\partial \hat V_r}{\partial \hat r}\right)-\frac{4}{3}\frac{\hat\mu}{\hat r}\hat V_r^2+\frac{2}{3}\frac{\hat\mu}{\hat r}\frac{\partial \hat V_r}{\partial \hat r}$$
$$+\frac{2}{3}\frac{\hat\mu}{\hat r}\frac{\partial \hat V_z}{\partial \hat z}+\frac{\partial}{\partial \hat z}\left(\hat\mu\frac{\partial \hat V_z}{\partial \hat r}\right)+\frac{\partial}{\partial \hat z}\left(\hat\mu\frac{\partial \hat V_r}{\partial \hat z}\right) \qquad (2)$$

AXIAL MOMENTUM EQUATION:

$$\frac{1}{\hat r}\frac{\partial \hat\rho\,\hat r\hat V_r\hat V_z}{\partial \hat r}+\frac{\partial \hat\rho\hat V_z^2}{\partial \hat z}=-\frac{\partial \hat p}{\partial \hat z}+\hat\rho\,Gr\theta-\hat\mu\hat V_z Da+\frac{4}{3}\frac{\partial}{\partial \hat z}\left(\hat\mu\frac{\partial \hat V_z}{\partial \hat z}\right)$$
$$-\frac{2}{3}\frac{\partial}{\partial \hat z}\left(\hat\mu\frac{1}{\hat r}\frac{\partial \hat r\hat V_r}{\partial \hat r}\right)+\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat r\hat\mu\frac{\partial \hat V_z}{\partial \hat r}\right)+\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat r\hat\mu\frac{\partial \hat V_r}{\partial \hat z}\right) \qquad (3)$$

ENERGY EQUATION:

$$\rho C_p\left[\hat V_r\frac{\partial \hat T}{\partial \hat r}+\hat V_z\frac{\partial \hat T}{\partial \hat z}\right]=\frac{1}{\Pr}\left[\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat r\hat k_e\frac{\partial \hat T}{\partial \hat r}\right)+\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat r\hat k_e\frac{\partial \hat T}{\partial \hat z}\right)\right]+\dot Q gen \qquad (4)$$

The symbols Gr, Pr and Da indicate the Grashoff number, the Prandtl number and the Darcy numbers respectively. Q_{gen} refers to the source term representing uniform volumetric heat generation in the domain during combustion.

SPECIES EQUATION:

The combustion model consists of conservation of species equations. The fuel mixture contains species namely CH_4, C_2H_6, CO_2, H_2, CO, H_2O and O_2, N_2. These species are assumed to undergo a finite rate, single-step irreversible reaction kinetics.

$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$

$C_2H_6 + 3.5\,O_2 \longrightarrow 2\,CO_2 + 3H_2O$

$CO + 0.5\,O_2 \longrightarrow CO_2$

$H_2 + 0.5\,O_2 \longrightarrow H_2O$

The equation conserving the species is then written as:

$$\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat\rho\,\hat r\hat V_r\hat Y_i\right)+\frac{\partial}{\partial \hat z}\left(\hat\rho\hat V_z\hat Y_i\right)=\frac{1}{Le\Pr}\left[\frac{1}{\hat r}\frac{\partial}{\partial \hat r}\left(\hat\rho\hat r\hat D\frac{\partial \hat Y_i}{\partial \hat r}\right)+\frac{\partial}{\partial \hat z}\left(\hat\rho\hat D\frac{\partial \hat Y_i}{\partial \hat z}\right)\right]+\hat\omega_i \qquad (5)$$

Le and D refer to the Lewis number and diffusion coefficient respectively while Y_i and ω_i represent mass fraction and the rate of generation or depletion of the i^{th} species respectively. The rate of generation or depletion non-dimensional ω_i for each individual species mentioned above can be expressed in terms of Arrhenius equation.

3. BOUNDARY CONDITIONS

3.1 AT THE FUEL INLET:

The pressure at fuel inlet is taken equal to the stagnation pressure to the ambient pressure at the same elevation; the viscous losses due to acceleration of the fluid from surroundings to the stove inlet are neglected. Further, the fluid is considered entering the stove radially thus; axial velocity at the fuel inlet is taken to be zero. Then, continuity equation dictates the normal gradient of axial velocity is also zero. The temperature at the stove inlet can be assumed to the ambient temperature,

$$\hat{p}=-\frac{1}{2}\hat{\rho}\hat{V}_z^2,\ \hat{V}z=0,\ \frac{\partial\hat{\rho}\hat{V}_z}{\partial z}=0,\quad \theta=0\quad \text{for } 0\le\hat{r}<1,\ \hat{z}=0 \tag{6}$$

3.2 AT THE AXIS OF SYMMETRY:

Due to assumed axi-symmetry of the flow, computations may be carried out for only one axi-symmetric plane of the domain as shown in Fig. 1. Symmetry, then, dictates the radial velocity, normal gradients of the axial-velocity and the temperature respectively to be zero,

$$\hat{V}_r=0,\ \frac{\partial\hat{\rho}\hat{V}_z}{\partial r}=0,\quad \frac{\partial\hat{T}}{\partial\hat{r}}=0\quad \text{for } \hat{r}=0,\ 0\le\hat{z}\le\hat{s}+\hat{H} \tag{7}$$

3.3 AT THE STOVES EXIT:

The exhaust gases leaving the combustion chamber can be treated as a jet entering a still medium i.e. static pressure is equal to the local ambient pressure. Also, assuming that the flow leaves the exit radially, consequently, axial velocity is zero. Thus, boundary conditions at the exit can be written as;

$$\hat{p}=\frac{(p_0-\rho_0 gz)r_1^2}{\rho_0 v_0^2},\ \hat{V}_z=0,\ \frac{\partial\hat{\rho}\hat{V}_r}{\partial\hat{r}}=0,\quad \frac{\partial\hat{T}}{\partial\hat{r}}=0 \tag{8}$$

3.4 AT SOLID WALLS:

Ceramic liner is a solid wall boundary at all locations other than the primary and secondary air slots. At solid walls, the no slip boundary condition would apply, i.e., both radial and axial velocities are zero. For temperature, the convective boundary condition is imposed,

$$\hat{V}_r=0,\ \hat{V}_z=0,\quad \hat{k}\frac{\partial\hat{T}}{\partial\hat{r}}-\hat{q}_w=Nu\hat{T}\quad \text{for } r=r_{max}(z),\ 0\le\hat{z}<\hat{H} \tag{9}$$

3.5 AT THE AIR INLETS:

The air can be assumed to enter in the combustor axial direction and stagnation pressure is taken equal to the ambient pressure. The temperature at port can be taken as the ambient temperature. Air will flow due to free convection.

$$\hat{p}=-\frac{1}{2}\hat{\rho}\hat{V}_r^2,\ \hat{V}_r=0,\ \frac{\partial\hat{\rho}\hat{V}_r}{\partial\hat{r}}=0,\quad \hat{T}=0 \tag{10}$$

3.6 EQUATION FOR PROPERTY VARIATION:

The fluid properties such as viscosity and thermal conductivity are assumed to be varying according to the Sutherland's relation as follows

$$\hat{k} = \hat{T}^{3/2}\left[\frac{1+\hat{S}_k}{\theta+\hat{S}_k}\right], \quad \hat{\mu} = \hat{T}^{3/2}\left[\frac{1+\hat{S}_\mu}{\theta+\hat{S}_\mu}\right], \quad \hat{S}_k = \frac{194.44}{T_0}, \quad \hat{S}_\mu = \frac{110.56}{T_0} \tag{11}$$

The variation of specific heat in terms of temperature is expressed by a cubic polynomial as:

$$\hat{C}_p = -0.1291\hat{T}^3 + 0.3408\hat{T}^2 - 0.0691\hat{T} + 0.9987 \tag{12}$$

3.7 EFFICIENCY OF HEAT TRANSFER:

The efficiency of heat transfer is defined in terms of the net heat input to the gas as follows:

$$\eta_{ht} = \frac{Q_{pan}}{Q_{gen}-Q_{loss}} \tag{13}$$

The heat transfer efficiency being an important result as far as performance of the combustor or stove is concerned. Since the variation in two quantities Q_{pan} and Q_{loss} depends on combustor geometry & volatiles flow rates. It has been seen in most cases that higher mass flow rates through the system leads to lower Q_{pan} as well as lower Q_{loss} due to lower temperatures. While a lower value of Q_{pan} tends to decrease the efficiency of heat transfer lower Q_{loss} tends to increase the efficiency. The net variation in $\eta h.t.$ depends up on which quantity has more dominating influence.

4. SOLUTION PROCEDURE

A finite volume method was employed to discretize the governing equations. The SIMPLER algorithm, and standard κ-ε turbulence model were used. The discretized sets of algebraic equations were solved by Ansys–Fluent 13.0. To predict the effect near the wall clearly, uniform grids are selected in such a way that more number of grid points are near the wall.

5. RESULTS AND DISCUSSION

The variation in wall temperature and power input gives different combustion temperature. The temperature contours are generated of 800 K, 900K, and 1000K for 0.5 kW, 1.0 kW and 1.2 kW for each case. The temperature profiles in the domain show that maximum temperature region moving away from the wall. This region also corresponds to the flame location in the combustion chamber. The maximum combustion temperatures for various cases are shown in Table 1. Maximum combustion temperature obtained is 1263 K. This observation also matches with maximum velocity region.

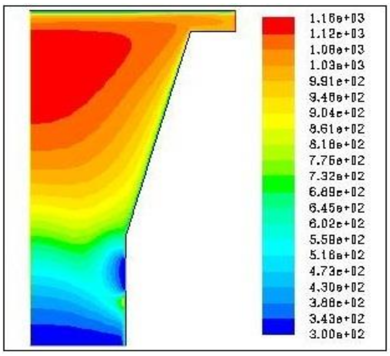
Figure 4 shows variation of heat transfer efficiency for various input power. The simulations results are shown in Figure (a) to (i) for conditions given in Table 1. In general, these figures show that maximum combustion temperature increases with increase in wall temperature for the same power input. It is clear that with increase in wall temperature flame region is increasing and flame is moving away from the wall and spread in radial diffuser.

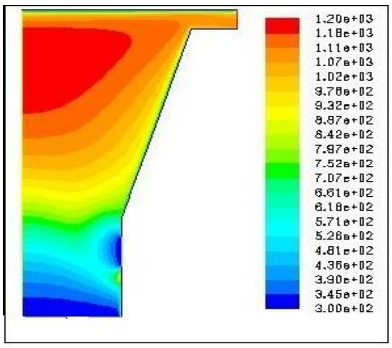
Similar effects are obtained with increase of power with constant wall temperature. It clearly indicates that maximum combustion temperature increases with increase in power and increase in wall temperature.

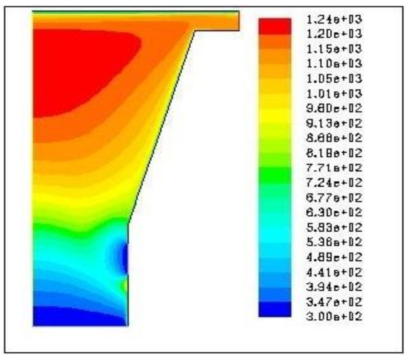
The heat transfer efficiency being an important result as far as the performance of the combustor or stove is concerned. Since the variation in two quantities Q_{pan} and Q_{loss} depends on combustor geometry and volatiles flow rates. It has been seen in most of cases that higher mass flow rates through the system leads to lower Q_{pan} as well as lower Q_{loss} due to lower temperatures. While a lower value of Q_{pan} tends to decrease the efficiency of heat transfer lower Q_{loss} tends to increase the efficiency. The effect of variation in radial diffuser and stove power is depicted in Figure 4. The Q_{pan} and Q_{loss} are found to increase monotonically, so increase in power input combustion temperature increases, due to this fact heat transfer efficiency increases.
The contours below show to temperature variation in vertical combustor. The contours arranged in the manner of case id according to Table 1.1.

Table 1. Maximum combustion temperatures at glance for various simulations.

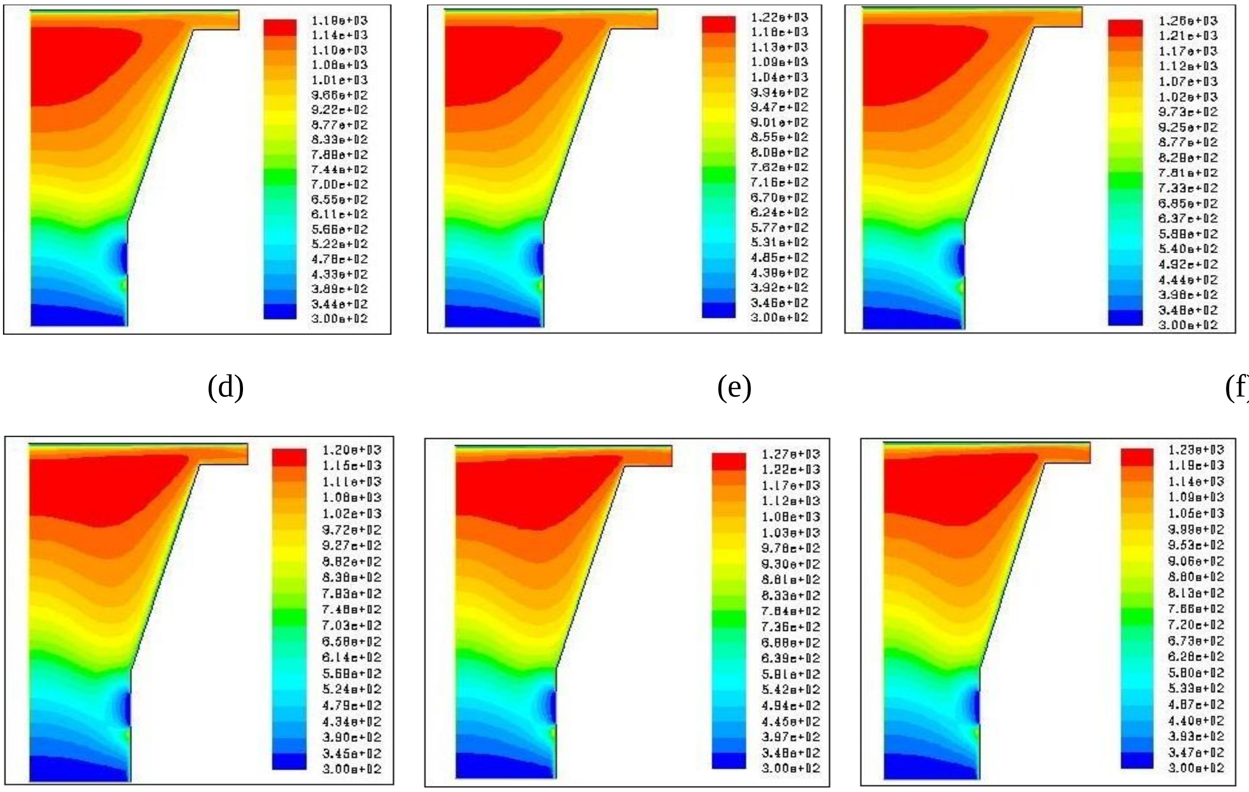
S.n.	Case Id	Wall Temperature	Power (KW)	Maximum Combustion Temperature (K)	Volatiles Flow Velocity (mm/s)
1	P1T1	800	0.5	1190	5.20
2	P2T1	800	1	1200	10.40
3	P3T1	800	1.2	1206	12.48
4	P1T2	900	0.5	1219	5.20
5	P2T2	900	1	1230	10.40
6	P3T2	900	1.2	1232	12.48
7	P1T3	1000	0.5	1249	5.20
8	P2T3	1000	1	1262	10.40
9	P3T3	1000	1.2	1263	12.48







(a) (b) (c)



(d) (e) (f)

(g) (h) (i)

Fig. 3. Iso-contours of temperatures in the computational domain for cases (a) P1T1, (b) P2T1, (c) P3T1, (d) P1T2, (e) P2T2, (f) P3T2, (g) P1T3, (h) P2T3, (i) P3T3 listed Table1.

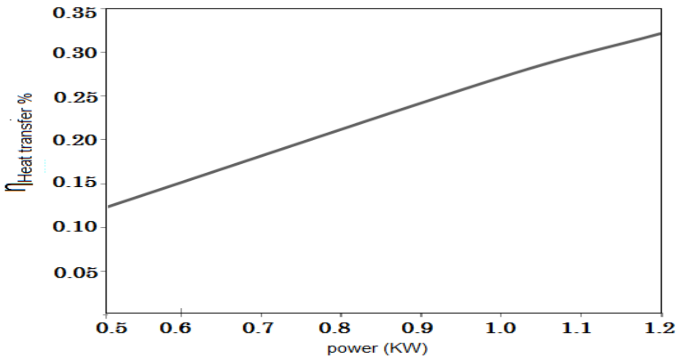


Fig.4. Variation of heat transfer efficiency with input power.

6. CONCLUSION

In the present work, the results of the combustion simulation are summarized as follows:

(1) The temperature contours are predicted by the present combustion model. Results indicate movement of the flame away from the wall for higher volatiles flow rates and higher wall temperatures.
(2) The region of maximum combustion temperature also corresponds the zone where the velocities are maximum.

(3) In general, maximum temperature in the flame region tends to increase with increase in wall temperature as well as power input. Maximum flame temperature region tends to move in radial diffuser.

(4) Heat transfer efficiency monotonically increases with increase in power input.

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