Effects of wall emissivity and fin blockage ratio on micro combustor performance

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Abstract. Combustion system design is a complex mechanism involving several physical mechanisms such as chemical reactions, fluid dynamics and heat transfer. At micro scale, short flow residence time adds further to the complexity. Enhancement of heat transfer to the outer surface of the combustor is desirable in several power generating systems. The present work focuses on this aspect by geometrical modification in the form of wall fin and observing the change in outer wall temperature level along with heat transfer rate for various wall surface emissivities and blockage ratio (BR) of wall fin. Lower surface emissivity and higher fin blockage ratio was found to give better combustor performance in terms of outer surface temperature.

1. Introduction

An increased demand for miniaturized electronic and mechanical systems have led to widespread research activity for combustion based power supply with longer working hours, higher energy densities and shorter recharging durations as compared to conventional batteries[1], [2]. The micro combustion systems are used to generate electrical power through micro gas turbines (MGT), micro thermo photovoltaic (MTPV), micro thermo electric (MTE) generators, etc[3]–[7]. The small size of combustor geometry incurs large amounts of heat and radical loss due to large surface to volume ratio. At higher velocities, the residence time of reacting species reduces remarkably and is of the same characteristic time scale of chemical reaction. This deteriorates combustion efficiency and flame stability significantly.

Various efforts in terms of geometries, materials, operating conditions have been made by a number of researchers in order to improve combustion characteristics; and recirculation zones generated because of geometrical modulation is one of them. Recirculation of reactants and products enhances mixing which helps in better heat transmission to different parts of micro combustors. Better heat recirculation helps in preheating of reactants coming in and more heat is transferred to outer wall of the combustor.

At high mixture velocities, stable flame sustenance becomes difficult due to reduced residence time of reactant species. Flame stabilization is then achieved by incorporating flame holders inside the combustion chamber such as backward facing step [8], [9], bluff bodies [10], [11], cavity [12], [13], porous medium [14], [15], etc.
In the present work, triangular wall fin and backward facing step configuration in a micro combustor is investigated for a range of external wall surface emissivities. The coupling effects of recirculation zones created by backward step and wall fin helps improve the combustor performance.

2. Geometrical and Computational model
Micro combustors with combined backward facing step and variable blockage ratio triangular wall fin at a distance of 2 mm from the step are investigated and the configuration is shown in Figure 1(a). Blockage ratio (BR) is defined as the ratio of fin height to combustion chamber height (BR = b/(h_i+h_s)). Case MCW04 signifies micro combustor with blockage ratio of 0.4 and so on. Aspect ratio of the fin is kept as unity for the present case but further effects of different aspect ratios can also be investigated. Flame anchoring due to step and wall fin is expected to enhance combustion due to flow and heat recirculation. Table 1 encapsulates the geometrical considerations for simulations.

| Geometrical parameter | Parameter name (mm) | Constant/variable |
|-----------------------|---------------------|-------------------|
| L                     | Combustor length    | 27                |
| L_i                   | Inlet length        | 7                 |
| h                     | Block location      | 2                 |
| h_i                   | Inlet height        | 0.5               |
| h_s                   | Step height         | 0.5               |
| h_t                   | Combustor wall thickness | 0.5          |
| b                     | Wall fin height     | 0.4, 0.5, 0.6     |
| d                     | Wall fin length     | 0.4, 0.5, 0.6     |
| AR                    | Aspect ratio = d/b  | 1                 |
| BR                    | Blockage ratio = b/(h_i+h_s) | 0.4, 0.5, 0.6 |

2.1. Numerical scheme
A global H_2/air reaction mechanism was implemented for combustion modelling while ignoring the surface reactions on the combustor wall. ANSYS FLUENT 16.0 was used to discretise and solve the governing conservation equations [16] of mass, momentum, energy and species along with relevant boundary conditions. Second order upwind discretization scheme and SIMPLE pressure-velocity coupling algorithm were employed. Turbulence-chemistry interaction was simulated using Finite Rate/Eddy Dissipation Model. The computational area comprises of the fluid and solid regions which makes it necessary for the heat conduction equation in the solid walls to be solved in order to account for the strong thermal interaction between solid and fluid domains in the micro combustors. Stability and convergence issues occur as reacting flows are difficult to solve numerically due to a strong coupling between the conservation equations. Also enormous density fluctuations and accelerations in flow occur due to excessive heat release from combustion reactions.

| Boundary | Conditions applied                                                                 |
|----------|-------------------------------------------------------------------------------------|
| Fluid-solid boundaries | No-slip velocity condition  
                      | Zero flux of all species  
                      | Coupled heat transfer   |
| Inlet    | Uniform velocity inlet [12 m/s]  
                      | Incoming H_2-air mixture temperature = 300 K |
| Outlet | Zero gauge pressure |
|-------|---------------------|
| Inlet/outlet walls | Adiabatic (Zero heat flux) |
| Outer wall | Mixed boundary condition (Radiation/Convection) |
|           | Wall Surface emissivity, $\varepsilon = 0.2 - 1.0$ |
|           | Material (stainless steel) thermal conductivity = 5 W/m-K |
|           | Heat transfer coefficient = 10 W/m²-K |
|           | Ambient temperature = 300 K |

Gas mixture density was attained from ideal gas equation and mixture specific heat, viscosity and thermal conductivity from mass fraction weighted average of the species properties. Specific heat for each species is calculated by piecewise polynomial fitting of temperature[17]. Values of turbulent kinetic energy, $k = 1 \times 10^{-3}$ for and dissipation, $\varepsilon = 1 \times 10^{-6}$ for the k–\(\varepsilon\) turbulence model were defined as take by Li et al [18]. Residuals of $10^{-6}$ are set for convergence of all the equations. Finite Rate/Eddy Dissipation Model is adopted for chemistry-turbulence interaction modelling.

Figure 1(a) shows the geometrical model of a simple micro combustor with a step used in validation against the experimental data of Li et al [18]. Figure 1(b) depicts the geometrical parameters used in the micro combustor. H₂/air mixture comes into the micro combustor at a particular velocity ($u = 12$ m/s) and equivalence ratio ($\varphi = 0.8$) through the inlet and premixed combustion occurs in the combustion chamber causing large amounts of heat release which gets transferred through the solid wall to the ambient[18]. Exhausted gases leave the combustor from the right side located outlet.

Several assumptions are made to further simplify the problem:

1. Dufour effects are ignored;
2. insignificant gas radiation;
3. nophre and viscous stress work;
4. steady-state combustion process.

$$q_y = h_y(T_{w,o} - T_{a,o}) + \varepsilon(T^4_{w,o} - T^4_{a,o})$$

![Figure 1](image_url)

**Figure 1.** (a) Schematic diagram of micro combustor (b) Geometrical parameters of micro combustor with wall fin
2.2. Computational grid and validation

ANSYS Workbench was used to generate multi zone structured quadrilateral mesh (see Figure 2) with refinement near the wall fin. Three grid sizes were created to confirm mesh convergence. The outer wall temperature profiles were compared for the three mesh systems and a discreet difference for the temperature profiles was observed. Consequently, the medium sized mesh with roughly 1.1E05 number of nodes was chosen for the simulation (not shown here) to keep a balance between the computational time and precision.

![Combustor wall](image)

**Figure 2.** Computational grid system

![Validation of simple micro combustor with experimental data of Li et al. [18]](image)

**Figure 3.** Validation of simple micro combustor with experimental data of Li et al. [18]
For validation of the numerical model selected for the present work, outer wall temperature of the simple micro combustor with backward facing step and without wall fin was validated against the experimental data results by Li et al. [18] for an inlet velocity of 12 m/s and equivalence ratio of 0.8.

As depicted in Figure 3, though the present simulation shows relatively larger differences compared with the experimental data, the trend of the two curves is similar. The global mechanisms over predict the temperatures but the temperature distribution trend lines are similar. Adoption of global mechanism saves large amount of computational time required in detailed reaction mechanisms while predicting better performing geometries.

3. Results and discussion

The results for qualitative as well as quantitative temperature and heat flux plots along with discussions are presented in this section.

**Figure 4.** Temperature contours (Top) and Outer wall temperature distribution (Bottom) with respect to axial displacement for different wall emissivities (0.2 – 1)
As can be seen in the outer wall temperature plots shown in Figure 4, the temperature levels are higher for smaller emissivities and get reduced as the emissivity value is increased. This happens because of an increased amount of heat loss due to radiation when surface emissivity is high. Since radiation heat transfer follows the fourth power law of temperature difference so for higher ε values, large amounts of heat losses can take place and this is quantitatively depicted in Figure 5. The negative values are for heat loss occurring from a system.

At the outer wall surface, the total heat loss rate (heat flux) to the ambient via natural convection and thermal radiation is given as:

\[ q_o = h_o(T_{w,o} - T_\infty) + e\sigma(T_{w,o}^4 - T_\infty^4) \]  

1

where \( h_o \) is the natural convection heat transfer coefficient (10 W/(m² K)), \( T_{w,o} \) is the outer wall temperature, \( T_\infty \) is the ambient temperature (300 K), \( e \) is the emissivity of the solid surface and \( \sigma \) is the Stephan Boltzmann constant, 5.67 x 10⁻⁸ W/(m² K⁴).

It is observed that more amount of heat loss is occurring from MCW06 design which may be due to more amount of heat transferred to the solid wall from the combustion zone as explained below.

In Figure 6, the recirculation zones formed in micro combustors with different blockage ratios at an inlet mixture velocity of 12 m/s, equivalence ratio of 0.8 and external surface emissivity of 0.6 are shown with the help of flow path lines. Two recirculation zones are formed in the micro combustor – one near the step wall and another behind the wall fin. The recirculation zone helps in better mixing of fluids causing enhanced heat transfer from one region to another. As the blockage ratio of fin is increased, the recirculation zone is seen to grow larger while no significant changes are observed in the zone created near the step wall. The enlargement of this zone is the main reason for increased outer wall temperatures for a particular emissivity.

In Figure 7, it is observed that for a particular value of emissivity, MCW06 has highest peak temperature for the whole range of ε. This is due to the enlarged recirculation zone formed as

![Figure 5](image-url)
explained previously which causes better heat recirculation along with the radicals. Thus larger sized wall fin is an effective means for enhancing outer wall temperature.

Two important aspects need to be discussed at this point. One is that for small $\epsilon$, radiation heat loss is smaller. Secondly, when heat loss is less, the combustion process intensifies and more heat gets generated. And more heat is available to be conducted upstream through the solid wall which is evident from the temperature contours and plots shown.

![Recirculation zones](image)

**Figure 6.** Streamlines and recirculation zone depiction in different micro combustor geometries at $\epsilon = 0.6$, $u = 12$ m/s

![Maximum outer wall temperature](image)

**Figure 7.** Comparison of maximum outer wall temperatures at different wall emissivities for various geometries

**Conclusions**

Micro combustors having a backward facing step and triangular wall fin were studied numerically for different blockage ratios (0.4, 0.5 and 0.6) with premixed $\text{H}_2$–air combustion for a range of outer wall surface emissivities (0.2 – 1).

- The outer wall temperature was observed to be higher for lower emissivities due to lesser amount of radiation heat loss through the outer surface of combustor.
For a particular outer wall surface emissivity, MCW06 displayed highest peak temperatures for the whole range of emissivities due to enlargement of recirculation zone behind the wall fin when blockage ratio is increased.

From the results, it can be concluded that lower emissivity and higher blockage ratio contribute to greater outer wall temperatures and would provide better MTPV performance.

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Nomenclature

- $q_o$: convection and radiation heat transfer through the outer wall
- $h_o$: convective heat transfer coefficient
- $\varepsilon$: outer wall surface emissivity
- $\sigma$: Stephan-Boltzmann constant = $5.67 \times 10^{-8}$ W/(m$^2$K$^4$)
- $T_{w,o}$: Outer wall surface temperature
- $T_a$: Ambient temperature
- $k$: turbulent kinetic energy
- $\varepsilon$: energy dissipation