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Numerical Analysis of Combustion and Thermal Performance of a Bluff-Body and Swirl-Stabilized Micro-Combustor with Premixed $\text{NH}_3/\text{H}_2/\text{Air}$ Flames

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Abstract: This research presents a novel bluff-body and swirl-stabilized micro-combustor fueled by an ammonia/hydrogen mixture, aimed at enhancing flame stabilization for zero-carbon micro-combustion-based power generators. Employing numerical simulations, the study examines the effects of bluff-body geometry, inlet mass flow rate, vane angle, and combustor material on combustion and thermal efficiencies. Key findings demonstrate that the shape of the bluff-body significantly influences the combustion outcomes, with cone-shaped designs showing the lowest radiation efficiency among the tested geometries. The study identifies an optimal inlet mass flow rate of 9×10^{-6} kg/s, which achieves a combustion efficiency of 99% and superior uniformity in the mean outer wall temperature. While variations in flow rate primarily affect NO emissions and outer wall temperatures, they have minimal impact on combustion efficiency. Further analysis reveals that adjusting the vane angle from 15 to 60 degrees significantly improves mean outer wall temperatures, temperature uniformity, and combustion and radiation efficiencies, while also reducing NO emissions. The 60-degree angle is particularly effective, achieving approximately 44% radiation efficiency. Additionally, material selection plays a pivotal role, with silicon carbide outperforming others by delivering an optimized mean outer wall temperature (approximately 910 K), radiation efficiency (38.5%), and achieving the most uniform outer wall temperature. Conversely, quartz exhibits significantly lower thermal performance metrics.

Keywords: swirling flames; bluff-body; premixed combustion; micro-combustor; ammonia/hydrogen mixture; micro-power generation



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1. Introduction

The development of miniature mechanical and electromechanical devices has been significantly enhanced by micro-machining technology. However, developments in Micro-Electro-Mechanical Systems (MEMSs) have encountered an obstacle, primarily as a result of technology constraints associated with power supplies. Conventional chemical batteries, however widespread as the principal energy source for micro-devices, exhibit some significant limitations. These include substantial volume and mass, poor energy density, shortened durability, and lengthy recharge durations [1]. In recent years, there has been an increase in global interest in micro-power generators that operate through combustion processes. These generators offer advantages such as a high energy output, a prolonged operational capacity, and compact dimensions [2,3].

The classification of these micro-power generators is bifurcated based on their distinct modes of energy transformation. The first category includes micro-thermoelectric (MTE) and micro-thermophotovoltaic (MTPV) systems, which harness the heat generated through fuel combustion and convert it directly into electrical power using thermoelectric or thermophotovoltaic techniques. In contrast, the second category includes micro-thermomechanical systems, such as micro gas turbines and rotary engines. These mechanisms convert combustion-induced thermal energy into mechanical work, which is subsequently converted into electricity [3,4].

The MTPV generator has distinct advantages compared to indirect generators. Firstly, it is free from any mechanical components, which improves its reliability. Secondly, it features simplified production and assembly methods [5]. The operational mechanism of an MTPV system is shown in Figure 1. A filter, an array of PV cells, and a heat source are all components of the MTPV system. The emitter is situated on the outer surface of a micro/mesoscale combustor. A portion of the heat generated from micro- or mesoscale combustion is emitted through the outer surface of the combustor towards the filter. The filter captures some of this radiant energy and transfers it to the PV cell array. In the last stage, the PV cell array converts the energy received from the filter into electric power [2]. The micro-combustor is a crucial element in a typical MTPV system, as it plays a key role in maintaining a stable combustion flame and determining the temperature distribution of the outer wall [6]. These factors significantly impact the power output density and overall efficiency of the system's energy conversion [7].

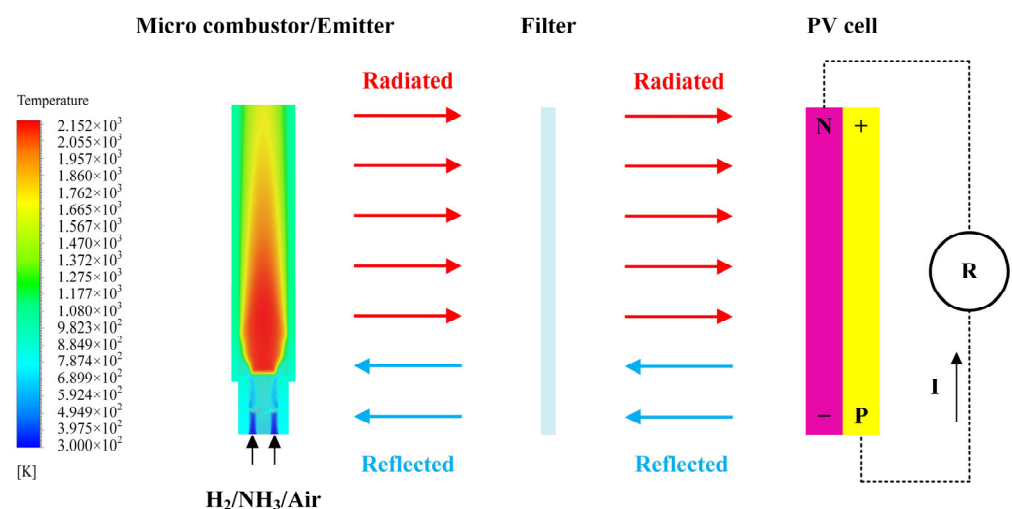


Figure 1. Schematic of an MTPV system.

Stable combustion in micro/mesoscale devices faces major challenges due to wall-induced suppression of heat and radical processes. Thermal quenching, from excessive heat loss, and radical quenching, where wall interactions extinguish reactive species, are key issues. Furthermore, limited residence time aggravates flame extinction at these dimensions. Integrating micro/mesoscale combustors into micro-power systems necessitates enhancing stability and sustainability [8,9]. Numerous researchers have developed a variety of techniques aimed at optimizing micro-combustors for MTPV systems, with a focus on enhancing combustion efficiency and ensuring flame stability. These methodologies can be classified into two main categories: (I) advancements in combustion technology, which includes catalytic combustion [10], hydrogen addition [11,12], and electrospray techniques [13]; and (II) innovations in burner geometry, further subdivided into strategies centered around heat recirculation and flow recirculation [3,14–17]. Additionally, innovative designs have also been explored, such as multichannel combustors [18], cross-plate

insertion [19], tube outlets [20], and other specialized designs [21], demonstrating the dynamic evolution of micro-combustor technologies for MTPV applications.

Incorporating catalysts onto the walls of micro-combustors is a popular method to improve flame stability and broaden flammability limits in microscale combustion applications [22,23]. Wenming et al. [24] evaluated a micro-TPV power generator prototype using different emitter materials, employing micro-cylindrical combustors made from silicon carbide (SiC) and platinum. The platinum combustor exhibited significantly higher wall temperatures than its SiC counterpart due to platinum's catalytic properties, under the same experimental conditions. Li et al. [10] designed a symmetrical catalytic micro-combustor with a 1.2 mm gap for non-premixed CH₄ and air mixtures, achieving maximum combustion and surface radiation efficiencies of 98.5% and 25.9%, respectively. This design was intended as a heat source for micro-TPV systems and could operate at velocities over 9 m/s.

Additionally, blending hydrogen with methane has been shown to significantly enhance flame stability and expand the flammable range from 0.6 m/s to 4 m/s with just a 10% hydrogen blend [11]. Besides, the use of liquid hydrocarbon fuels is particularly advantageous for micro-combustion-based power generation [25], where the electrospray technique has been crucial. This technique involves applying a strong electric field to liquid fuel, creating a fine aerosol of charged droplets. Gan et al. [13] utilized this method in a mesoscale combustor designed to work with energy conversion modules, improving the efficiency of TPV systems that operate on liquid fuels.

Preventing flame extinction in small-scale combustion devices is a significant challenge due to substantial heat losses, which are more pronounced at smaller scales due to an increased surface area-to-volume ratio. This leads to a higher ratio of heat loss to heat generation as the device size decreases. To counteract these effects, many researchers have adopted heat-recirculating combustors as an effective mitigation strategy. This approach often involves capturing heat from combustion products and using porous media to enhance thermal management in micro/mesoscale combustors [14]. Tang et al. [26] found that a heat recirculation micro-combustor significantly outperformed a straight-channel combustor in an experimental study, showing a maximum blowout limit over three times greater and a significantly larger high-temperature zone with a mean temperature increase of 25–120 K on the external wall. In another study, Bani et al. [27] explored integrating a TPV system with a porous media combustor equipped with a stainless steel (SS) mesh featuring a porosity of 0.9. This research underscores the potential of innovative combustor designs in enhancing the efficiency and viability of micro/mesoscale combustion-based energy systems.

In advancing flame stabilization and thermal performance in TPV systems, adaptive combustor geometries such as bluff bodies, swirlers, and innovative designs like Y-shaped fins and cavity configurations play crucial roles. These features create recirculation zones and anchoring points that enhance flame stability and extend flammability limits [28]. For instance, Fan et al. [29] and Wenming et al. [30] demonstrated improved blow-off limits and thermal performance by using bluff-body and stepped and micro-combustors for micro-combustion-based TPV power generators. Moreover, vortex and swirling flow techniques are key strategies for promoting flow recirculation, improving mixing and thermal distribution, and enhancing the efficiency and stability of TPV systems. Lee et al. [31] showcased increased system performance with NH₃-H₂ blends in a micro-TPV device using a cyclone adapter in the fuel-air mixture supply system, achieving a power output of 5.2 W and an emitter efficiency of 37%, with the micro-emitter's outer surface reaching up to 1408 K. These findings underscore the importance of innovative combustor designs in advancing micro/mesoscale combustion-based TPV systems. Furthermore, researchers

like Yang et al. [4] and Sheykhabglou et al. [32] have developed micro/mesoscale combustors utilizing swirling flow techniques for flame stabilization, underscoring the pivotal role of innovative combustor designs in advancing micro/mesoscale combustion-based TPV systems.

In addition, in the field of TPV systems and micro/mesoscale combustion optimization, considerable research efforts have focused on improving flame stability and the efficiency of thermal energy conversion by exploring a range of innovative, integrated flame stabilization techniques aimed at addressing the intrinsic challenges associated with enhancing combustor performance. For example, Li et al. [33] proposed a compact TPV system featuring a central porous-medium combustor and heat recuperator. Zhang et al. [34] developed a micro-step helix combustor equipped with helical fins, specifically designed to enhance the performance of micro-combustors under high flow conditions. Li et al. [35] explored the impact of incorporating a swirling combustor with porous media and heat recuperation encircled by GaSb PV cells within a mesoscale TPV system, investigating its thermal and operational dynamics. Ni et al. [36] investigated the insertion of porous media into I-, T-, and Y-shaped micro-combustors, analyzing their influence on combustion efficiency and stability. Qian's research on a divergent porous media combustor demonstrated its capability to increase the flammability limit by 186% and enhance radiation efficiency by 70% compared to a conventional straight-channel combustor [37]. Peng et al. [38] carried out both experimental and numerical analyses to assess the combustion characteristics and thermal performance of a premixed $\text{H}_2/\text{C}_3\text{H}_8/\text{air}$ mixture in a micro-planar combustor with a block insert. Furthermore, Amani et al. [39] performed a numerical investigation on an innovative baffle-cavity micro-combustor, tailored for micro-TPV applications, revealing significant improvements in flame stabilization and system efficiency. Additionally, in one of the latest scholarly contributions, Sheykhabglou and Dimitriou [40] proposed a novel bluff-body swirl-stabilized micro-combustor for micro-power generation. Their findings indicated that larger bluff-bodies significantly enhance combustion and radiation efficiencies and achieve the most uniform outer wall temperatures. These results underscore the crucial role of bluff-body size in optimizing the thermal performance of combustors and its significant impact on the design of micro-power generators. These diverse approaches to single and integrated flame stabilization in the development of micro/mesoscale combustion-based TPV systems are summarized in Figure 2, illustrating the depth of recent advancements in this field.

The emission of greenhouse gasses, particularly carbon dioxide, into the atmosphere as a consequence of the combustion of fossil fuels, contributes to global warming and triggers a variety of environmental challenges [41,42]. This has emphasized the necessity of reducing carbon emissions and pollution as part of a global research agenda [43]. Ammonia exhibits great potential as a novel fuel source, offering numerous advantages. Due to its absence of carbon, ammonia does not generate carbon dioxide emissions [44], thereby reducing its environmental impact. In addition, ammonia possesses a high energy density [45], allowing it to hold a greater amount of energy per unit volume. Consequently, the process of transporting and storing [46] becomes less complicated, which is essential for practical purposes. Despite its potential, the combustion of ammonia faces notable challenges, namely its narrow flammability spectrum [47], slow combustion rate, and the emission of considerable amounts of pollutants, notably nitrogen oxides, nitrous oxide, and unburnt ammonia [48,49]. To tackle these issues, combusting ammonia in conjunction with more reactive fuels and utilizing ammonia-oxygen-enriched flames are two advanced strategies under consideration to boost the efficiency and cleanliness of ammonia combustion processes [48,50–52].

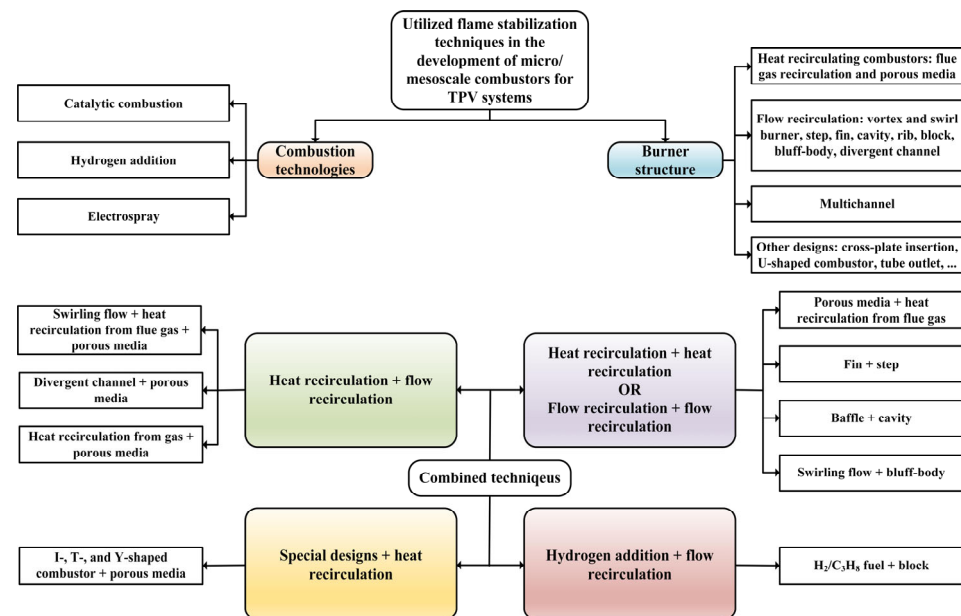


Figure 2. Utilized single and combined flame stabilization techniques in developing micro/mesoscale combustion-based TPV generators.

Given the aforementioned considerations, it is worthwhile examining the potential of $\text{NH}_3\text{-H}_2$ blends as a clean energy solution in micro-combustion-driven power generation systems. Moreover, despite the application of various techniques for flame stabilization in the development of micro/mesoscale combustion-based TPV generators, to the authors' knowledge, there has been limited investigation into the synergistic effects of bluff-body configurations combined with swirling flows, with the exception of our preliminary studies [40]. Formation of a recirculation zone through a bluff-body and swirling flow serves as an efficient strategy for maintaining flame stability. This is achieved by enabling the mixing of recently combusted hot gasses with the incoming unreacted mixture, thereby sustaining the combustion process effectively [49,53].

Building on our previous work [40], which investigated the influence of various flow parameters, bluff-body size, and the half-angle for a cone-shaped geometry, there remained gaps in understanding the impacts of vane angle, bluff-body geometry, and wall material. Consequently, this research seeks to provide a deeper analysis of the combustion characteristics, thermal performance, and emissions of a micro-combustor fueled by ammonia and hydrogen. Our comprehensive evaluation explores critical variables such as the bluff-body geometry, swirler vane angle, inlet mass flow rate, and the selection of combustor materials. The insights garnered from this study are instrumental in advancing the development of micro-combustion technology aimed at achieving environmentally sustainable, zero-carbon micro-power generation.

2. Physical and Mathematical Model

2.1. Physical Model

The general layout of the burner and micro-combustor is depicted in Figure 3. From the inlet to the bluff-body exit, the entry section measures 3.5 mm, whereas the combustion chamber (CC) with an inner diameter of 3 mm extends 16 mm in length and has a thickness of 0.4 mm. The bluff-body (cone-shaped in Figure 3) and axial swirler are positioned upstream of the CC. The axial swirler introduces a counterclockwise swirling flow through six vanes, each 0.1 mm thick, set at an angle θ relative to the burner axis from the nozzle's

upper edge. The swirler comprises a hub diameter (D_h) of 0.8 mm and a tip diameter (D_{sw}) reaching 1.8 mm. The swirl number (SN) is obtained using Equation (1) [54]:

$$SN = \frac{2}{3} \left[\frac{1 - (D_h/D_{sw})^3}{1 - (D_h/D_{sw})^2} \right] \tan \theta \quad (1)$$

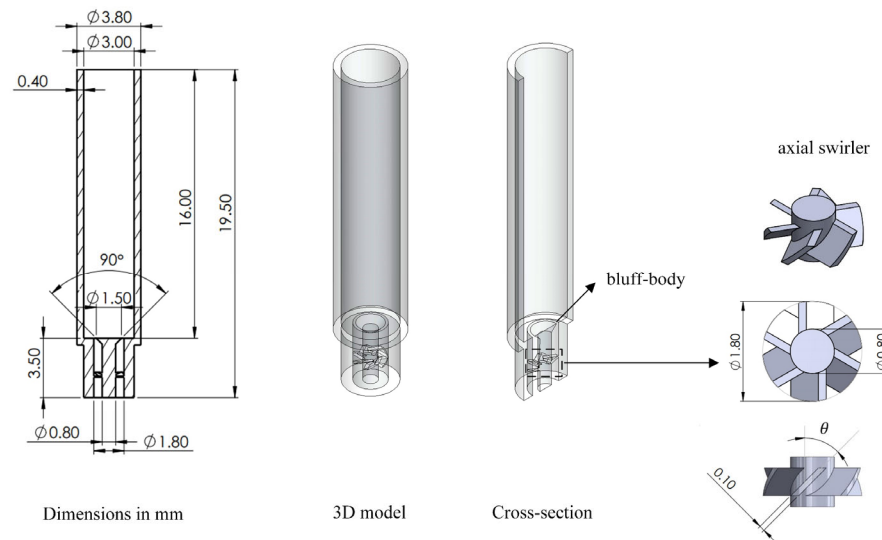


Figure 3. Schematic of the micro-combustor with the bluff-body and swirl-stabilized configuration.

In our simulations, various bluff-body configurations, vane angles for axial swirlers, and combustor materials are employed. The geometrical specifications of studied swirlers are listed in Table 1, and selected combustor materials and their physical and thermal properties are presented in Table 2 [4,28,55,56].

Table 1. Geometrical specifications of the studied swirlers.

Swirler	Vane Angle [°]	Swirl Number, SN [–]	Hub Diameter, D_h [mm]	Tip Diameter, D_{sw} [mm]	Number of Vanes [–]	Vane Thickness [mm]
1	15	0.2031	0.8	1.8	6	0.1
2	30	0.4375				
3	45	0.7578				
4	60	1.313				

Table 2. Physical and thermal properties of selected materials for micro-combustor [4,27,54,55].

Material	Density, ρ [kg/m ³]	Specific Heat, c_p [J/(kg.K)]	Thermal Conductivity, λ [W/(m.K)]
Stainless steel	8030	502.48	16.27
Quartz	2650	750	1.05
Ceramic	3440	710	22.7
Silicon carbide	3217	2352	32.8

2.2. Mathematical Model

To develop the mathematical model encompassing fluid flow, heat transfer, and combustion reactions, several key assumptions were made [57,58]: (a) the system is in a steady state; (b) Dufour effects are neglected; (c) there is no work performed by pressure or viscous forces; (d) wall surface reactions are absent; and (e) gas radiation is not considered. Additionally, the Knudsen number (K_n) for the gas combustion was calculated prior to further simulation, revealing that the K_n is well below the threshold value of 0.001 [59,60]. Consequently, this validates the use of the Navier–Stokes equations for subsequent analyses, as the assumptions of fluid continuity are satisfied. Based on these premises, the governing equations for mass, momentum, energy, and species conservation are significantly simplified [15].

The following are the governing equations for the gaseous mixture [4,52]:

- Mass conservation:

$$\nabla \cdot (\rho \vec{v}) = 0 \quad (2)$$

Symbol ρ represents the premixed gas density, while $\vec{v}$ is the velocity vector.

- Momentum conservation is

$$\rho (\vec{v} \cdot \nabla \vec{v}) = -\nabla p + \nabla \cdot \left\{ \mu \left[\nabla \vec{v} + (\nabla \vec{v})^T \right] - \frac{2}{3} \nabla \cdot \vec{v} I \right\} \quad (3)$$

where p represents the gas mixture pressure, μ indicates the molecular viscosity, and the unit tensor is denoted by I .

- Energy conservation:

$$\nabla \cdot \vec{v} (\rho E_f + p) = \nabla \cdot \left[k_{eff} \nabla T_f - \sum_i h_i \vec{D}_i + \left\{ \mu \left[\nabla \vec{v} + (\nabla \vec{v})^T \right] - \frac{2}{3} \nabla \cdot \vec{v} I \right\} \cdot \vec{v} \right] + S_f^h \quad (4)$$

The variable E_f signifies the overall energy of the fluid, while k_{eff} denotes the effective conductivity. T_f represents the fluid's temperature, and S_f^h is the source term of the fluid enthalpy. Additionally, $\vec{D}_i$ denotes the diffusion flux of species i , with h_i being the specific enthalpy of that species i .

- Species conservation:

$$\frac{\partial}{\partial t} (\rho \vec{v} Y_i) = -\nabla \cdot \vec{D}_i + \omega_i \quad (5)$$

The species i mass fraction is denoted as Y_i , and ω_i shows the total rate at which species i is produced or consumed through chemical reactions.

The energy equation in the solid phase is as follows:

$$\nabla \cdot (k_s \nabla T_s) = 0 \quad (6)$$

k_s and T_s denote the thermal conductivity and temperature of the solid phase, respectively.

The energy loss from the outer surface of the micro-combustor's wall to the environment occurs through two mechanisms: radiation heat transfer ($\dot{Q}_{rad}$) and convection heat transfer ($\dot{Q}_{conv}$). The mathematical expressions representing the total energy loss from the exterior surface ($\dot{Q}_{loss}$) is provided as follows [7]:

$$\dot{Q}_{loss} = \dot{Q}_{rad} + \dot{Q}_{conv} = \varepsilon \sigma A (T_w^4 - T_0^4) + h_c A (T_w - T_0) \quad (7)$$

In this equation, h_c denotes the natural convection heat transfer coefficient, which has a value of $10 \text{ W}/(\text{m}^2 \cdot \text{K})$ in our study according to the references [4,61]. Additionally, A represents the outer surface area, while T_w and T_0 signify the outer wall temperature and the ambient environment (300 K), respectively. In addition, ε describes the wall emissivity and σ is the Stefan–Boltzmann constant, which has a value of $5.67 \times 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K}^4)$.

When assessing the thermal performance of a micro-combustor (bluff-body and swirl-stabilized micro-combustor in our research), three crucial parameters come into play: the mean outer wall temperature (T_{mean}), the radiation efficiency (η_{rad}), and normalized temperature standard deviation (NTSD), which is an indication of the uniformity of outer wall temperature [4,62]:

$$T_{mean} = \frac{\sum A_{w,i} T_{w,i}}{A} \quad (8)$$

$$\eta_{rad} = \frac{\dot{Q}_{rad}}{\dot{m}_{fuel} LHV} \times 100\% \quad (9)$$

$$NTSD = \frac{\sum A_{w,i} |T_{w,i} - T_{mean}|}{T_{mean} \sum A_{w,i}} \times 100\% \quad (10)$$

The variables $A_{w,i}$ and $T_{w,i}$ indicate the area and temperature of the surface of the element i , respectively. Meanwhile, $\dot{m}_{fuel}$ denotes the hydrogen/ammonia mixture mass flow rate. LHV signifies to lower heating value of this fuel mixture.

According to the references [63,64], the combustion efficiency (η_C) is obtained using the following equation:

$$\begin{aligned} \eta_C &= \frac{\dot{m}_{fuel,in} - \dot{m}_{fuel,out}}{\dot{m}_{fuel,in}} \times 100\% = \left[1 - \frac{\dot{m}_{fuel,out}}{\dot{m}_{fuel,in}} \right] \times 100\% \\ &= \left[1 - \frac{Y_{H_2,out} + Y_{NH_3,out}}{Y_{H_2,in} + Y_{NH_3,in}} \right] \times 100\% \end{aligned} \quad (11)$$

In this equation, $\dot{m}_{fuel,in}$ and $\dot{m}_{fuel,out}$ represent the mass flow rates of fuel at the inlet and outlet, respectively.

2.3. Numerical Model and Boundary Conditions

ANSYS Fluent is utilized to simulate the physical and mathematical models concerning fluid flow, heat transfer, and combustion within the micro-combustor. The micro-combustion process, involving a hydrogen and air mixture, employs a reduced chemistry model, incorporating 28 species and 213 reversible reactions [65]. Consistent with existing literature [55,66] and the recommendations of Kuo and Ronney [67], a turbulent flow model is adopted for gas flow simulations in micro-combustors when the Reynolds number exceeds 500. For modeling turbulence, the realizable $k - \varepsilon$ model is selected based on its compatibility with previous studies [55,66]. To address the turbulence–chemistry interaction, the Eddy Dissipation Concept (EDC) model is employed, which is preferred for its ability to balance computational cost while considering finite-rate kinetics effects. This model effectively captures how turbulence influences reaction rates, providing a robust framework for simulating these complex interactions.

Regarding the properties of the mixture, the law of incompressible ideal gas is applied to calculate density, while the mixing law is used for specific heat calculations. Thermal conductivity and viscosity are determined using the mass-fraction-weighted mean method, and the kinetic theory is utilized to calculate mass diffusivity. For individual species, specific heat is calculated via piecewise polynomial fitting of temperature, and kinetic theory is again applied to determine thermal conductivity and viscosity [15].

The simulation employs the SIMPLE (Semi-Implicit Method for Pressure Linked Equations) algorithm to decouple pressure from velocity. All governing equations are

discretized using the second-order upwind scheme. The convergence criteria for continuity, momentum, and species equations are set at less than 1×10^{-3} , while the energy equation requires a stricter convergence criterion of less than 1×10^{-6} [15,57]. Finally, the location of boundary conditions and the settings of them are presented in Figure 4 and Table 3, respectively.

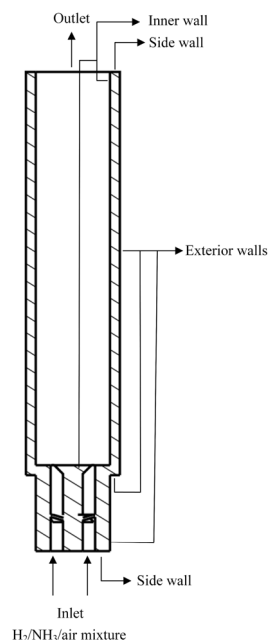


Figure 4. Location of boundary conditions.

Table 3. The settings of boundary conditions.

Location	Boundary Conditions	Values
Inlet	Mass flow inlet	50% NH ₃ + 50% H ₂ for fuel and air as oxidizer; equivalence ratio of 1
	Mass flow rates	$7 \times 10^{-6} \frac{\text{kg}}{\text{s}}$, $9 \times 10^{-6} \frac{\text{kg}}{\text{s}}$, $11 \times 10^{-6} \frac{\text{kg}}{\text{s}}$, $13 \times 10^{-6} \frac{\text{kg}}{\text{s}}$
	Temperature	300 K
Outlet	Pressure outlet	Guage pressure of 0 Pa
Exterior walls	Material	Stainless steel, quartz, ceramic, and silicon carbide
	Mixed thermal condition (radiation and convection heat losses)	Free stream temperature of 300 K, heat transfer coefficient of $10 \text{ W}/(\text{m}^2 \cdot \text{K})$ [4,61]
Side walls	Material	Stainless steel, quartz, ceramic, and silicon carbide
	Adiabatic thermal condition	Heat flux of $0 \text{ W}/\text{m}^2$
Inner wall and gas-solid interface	Coupled thermal conditions, no-slip shear condition, stationary wall, zero diffusive flux for species boundary conditions	-
Operating pressure	-	101,325 Pa

2.4. Analysis of Grid Independence

To confirm the simulation outputs are unaffected by mesh size and to pinpoint the smallest sufficient element count, a grid independence study was performed on the bluff-

body and swirl-stabilized micro-combustor with a cone-shaped bluff-body. This process entailed examining four separate mesh configurations, initially comprising 1,591,748, 2,328,748, 3,123,364, and 3,709,222 elements, respectively, prior to their conversion to polyhedral format within ANSYS Fluent (Figure 5a). The simulations maintained an equivalence ratio of 1 and an inlet mass flow rate of 7×10^{-6} kg/s, employing a fuel mixture composed equally of ammonia and hydrogen blends (50% each by volume).

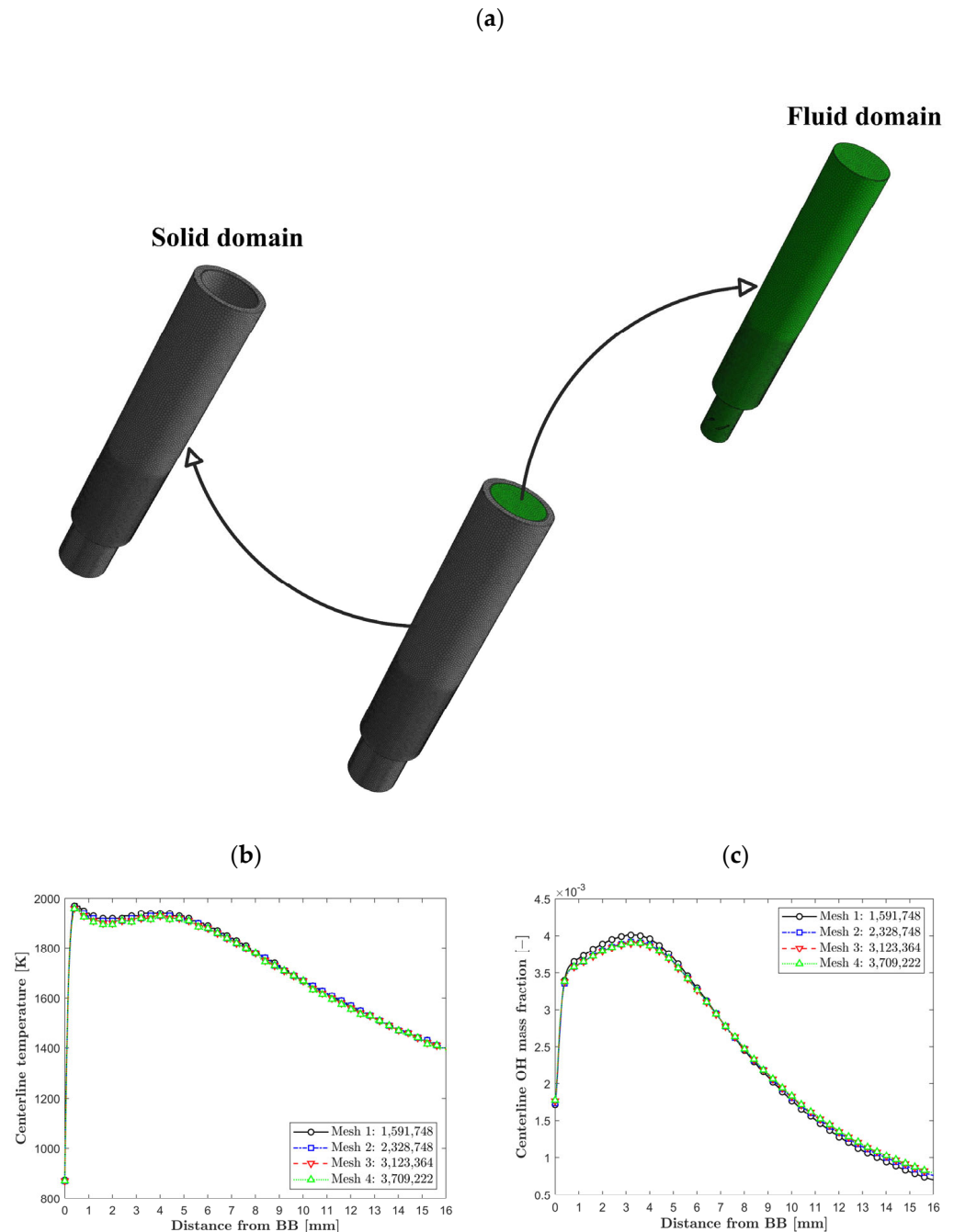


Figure 5. (a) Generated mesh; mesh independence study; (b) centerline temperature; (c) centerline OH mass fraction.

Figure 5b,c visually depict the temperature and OH mass fraction profiles along the centerline, initiating at the bluff-body's exit plane, across the four discrete mesh configurations. The temperature distribution profile exhibits two distinct peaks: there is a significant increase in temperature in the vicinity of the burner exit plane, which is followed by a

pattern of reduction, increase, and another decrease. However, the second increase is relatively minor. From Figure 5, it is evident that the temperature and OH mass fraction distributions along the centerline remain almost the same as the element count rises beyond 2,328,748. Supporting data in Table 4 illustrate that the deviations in temperature and OH mass fraction along the axis, and the exhaust gas temperature, are 0.20%, 0.88%, and 0.09% between Mesh 2 and Mesh 3. Additionally, the deviations in temperature and OH mass fraction along the axis, and the exhaust gas temperature, are 0.20%, 0.88%, and 0.09% between Mesh 3 and Mesh 4. Consequently, given these negligible variations observed, Mesh 2 has been selected for pursuing deeper analysis within this study.

Table 4. Study of mesh independence.

Mesh	Initial Element Size	Element Size After Polyhedral Conversion	Percentage Difference [%] Compared with Previous Mesh		
			Centerline Temperature	Centerline OH Mass Fraction	Outlet Temperature
Mesh 1	1,591,748	325,946	-	-	-
Mesh 2	2,328,748	458,376	0.30	2.81	0.22
Mesh 3	3,123,364	618,586	0.20	0.88	0.09
Mesh 4	3,709,222	717,783	0.1	0.29	0.04

2.5. Validation

To guarantee the accuracy of the numerical simulations, it is crucial to validate them against available experimental data. Due to the lack of experimental data on NH_3/H_2 combustion at the micro-scale combustion, the outcomes of hydrogen/air micro-combustion are utilized for validation purposes as conducted in [40,68]. The employed numerical model and chemical mechanism have been applied to the experimental results presented by Wan et al. [69] and Yang et al. [30], enabling a direct comparison between the experimental findings (exhaust gas temperature and mean outer wall temperature) and the simulated results. For experiments conducted by Wan et al. [69], the micro-combustor is constructed of quartz walls, which have a density of 2650 kg/m^3 . The thermal conductivity of the material is $1.05 \text{ W/(m}\cdot\text{K)}$, its specific heat capacity is $750 \text{ J/(kg}\cdot\text{K)}$, and its emissivity is 0.92. In experiments related to Yang et al. [30], the combustor material is silicon carbide with a density of 3217 kg/m^3 , the thermal conductivity of $32.8 \text{ W/(m}\cdot\text{K)}$, the specific heat of $2352 \text{ J/(kg}\cdot\text{K)}$, and the emissivity of 0.9 [4,60]. Figure 6a,b illustrate the comparison of experimental and simulation results at various inlet velocities when the equivalence ratio is 0.5 and various equivalence ratios when the inlet velocity is 12 m/s , respectively. It is evident that the results of the experiments and simulation become increasingly similar as the inlet velocity and equivalence ratio increase. The accuracy and effectiveness of numerical methods are indicated by the fact that the maximal errors of experimental and simulation results related to exhaust gas temperature are less than 10% at the inlet velocity of 5 m/s , narrowing down to about 4.5% when the inlet velocity rises to 13 m/s . Similarly, the most significant deviation in the mean outer wall temperature measurements occurs within an 8% margin at an equivalence ratio of 0.5, substantially tightening to approximately 3% as the equivalence ratio increases to 1. These minor deviations underscore the robustness and reliability of the numerical models across varying operational conditions.

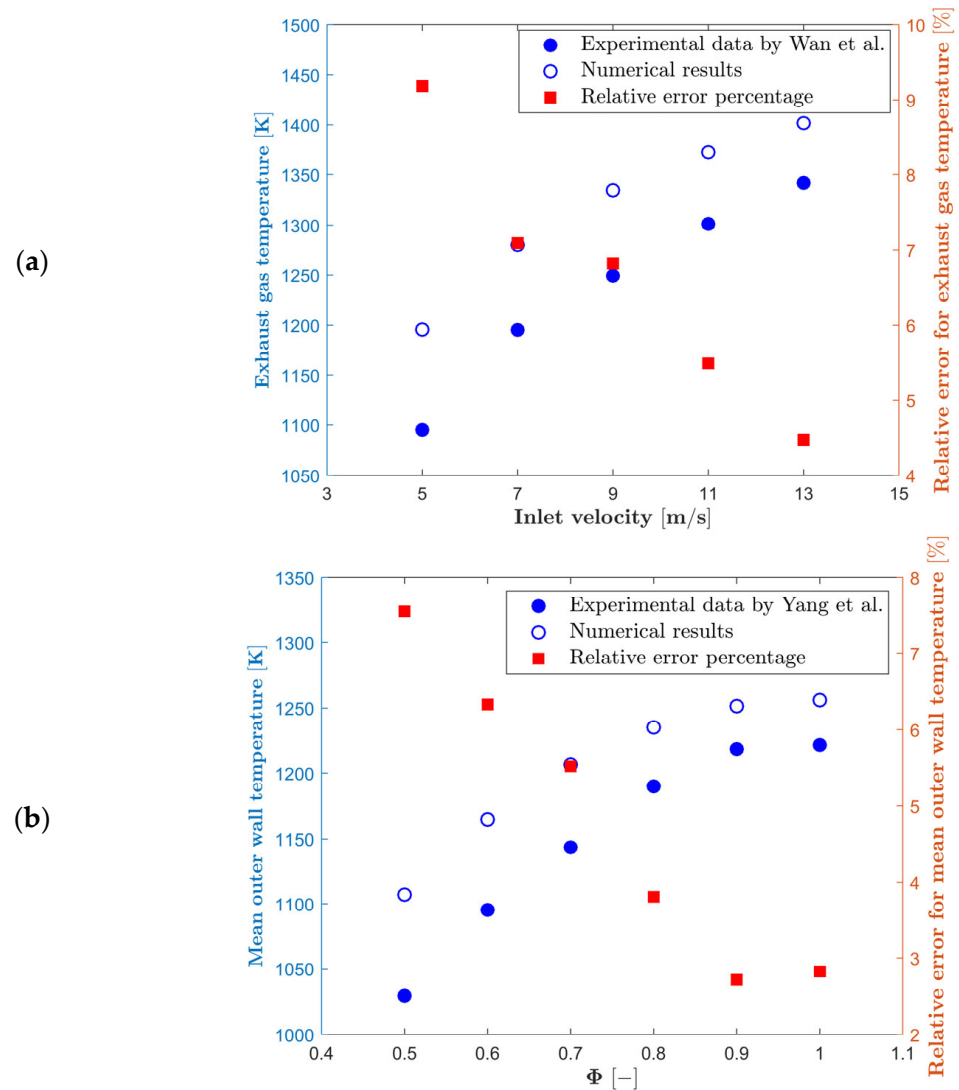


Figure 6. Validation: (a) comparison of the simulated and experimental results for exhaust gas temperature by Wan et al. [69]; (b) comparison of the simulated and experimental results for mean outer wall temperature by Yang et al. [30].

3. Results and Discussion

The results section examines the impact of several factors, such as the bluff-body geometry, vane angle of the swirler, inlet mass flow rate, and combustor material on the combustion and thermal performances of the bluff-body and swirl-stabilized micro-combustor.

3.1. Effects of Bluff-Body (BB) Geometry

The effects of different geometries of bluff-bodies on combustion and thermal performance of the bluff-body and swirl-stabilized micro-combustor are investigated numerically in this section. In these simulations, the material of the micro-combustor is silicon carbide and the inlet mass flow rate of 7×10^{-6} kg/s is chosen. In addition, selected geometrical parameters are a bluff-body size of 1.5 mm, and a swirler vane angle of 45° (swirl number of 0.7578). Figure 7 displays four distinct types of BBs with the same size that have been chosen for investigation: cone-shaped with a cone angle of 90° (as presented in Figure 3), cone-disk-shaped, disk-shaped, and hemisphere-shaped.

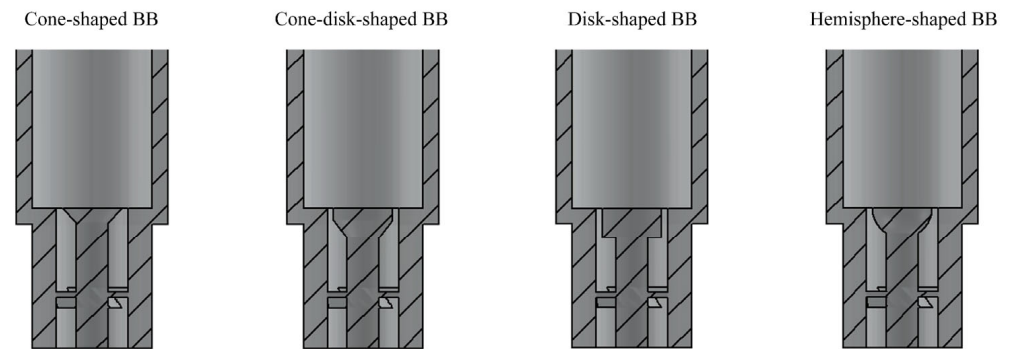


Figure 7. Physical models of four different shapes of bluff-bodies.

Figure 8 illustrates the effects of different shapes of BBs on the temperature and OH mass fraction distributions both as contours and along the combustor centerline. The OH mass fraction serves as an indicator of the reaction zone's position [2]. It has been noted that the geometry of the bluff-body (BB) significantly influences both the temperature distribution and the OH mass fraction within the combustor. Specifically, micro-combustors featuring a cone-shaped BB exhibit the lowest temperatures and OH mass fraction at the lower section of the combustor which can be noted in Figure 8a,c. Additionally, the region of highest temperature displays a distinctive triphasic width variation, characterized initially by a decrease, followed by an increase, and then a subsequent decrease again. Furthermore, it is noted that the peak distribution of the OH mass fraction differs among the BB geometries. Specifically, in the cone-disk, disk, and hemisphere-shaped BB configurations, the greatest concentration of OH mass fraction is observed along the lateral edges of the burner's central axis. Conversely, the cone-shaped BB exhibits a central concentration of the OH mass fraction, primarily surrounding the centerline. This distinct distribution pattern can be ascribed to the unique flow characteristics and velocity profiles inherent to each design, as depicted in Figure 9. Based on the data presented in Figure 9, the cone-shaped bluff-body demonstrates the most expansive inner recirculation zone among the configurations examined. This extensive recirculation zone contributes significantly to a central concentration of OH radicals, a key observation that can be directly attributed to the specific flow pattern induced by this geometry. In proximity to the BB, it is observed that the high-temperature zone reaches its broadest extent in the case of the cone-shaped configuration (Figure 8a), while other BB configurations result in a narrower high-temperature region near the BB and they attain the maximum width of the high-temperature zone. Furthermore, the OH mass fraction distribution contour illustrates that the flame within the cone-shaped design extends further longitudinally (as depicted in Figure 8b).

An analysis of Figure 8c reveals the presence of two prominent temperature peaks along the central axis. An immediate rise in temperature is detected just after the burner exit (at about 0.5 mm far from BB), likely due to the formation of a recirculation zone and succeeded by a descending trend. This is then followed by another increase and decrease, with the latter elevation being notably less pronounced for the cone-shaped BB. The initial peak is consistently positioned just downstream of the BB, with minimal variation across all BB geometries examined. Conversely, the second peak emerges further downstream and is situated closer to the BB outlet plane for the cone-shaped design compared to other configurations.

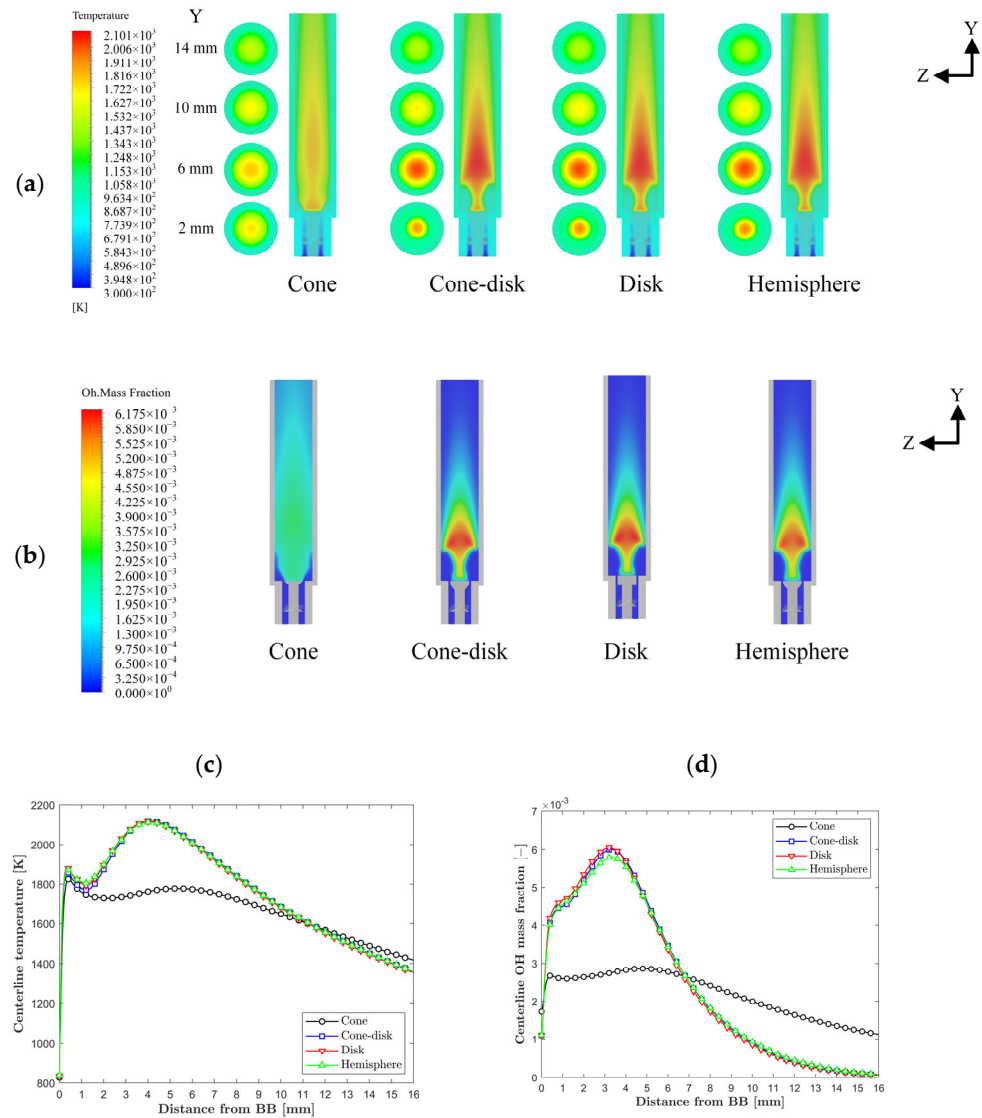


Figure 8. Effect of bluff-body geometry on (a) temperature contour; (b) OH mass fraction contour; (c) centerline temperature; (d) centerline OH mass fraction.

The temperature profile is segmented into two distinct sections: Section I spans from 0 to approximately 11 mm, while Section II begins around 11 mm and ends at 16 mm. In Section I, elevated temperatures along the central line for combustors with cone-disk, disk, and hemisphere-shaped BBs, relative to the cone-shaped BB, suggest heightened heat release rates and greater OH mass fractions (evident in Figure 8d). Conversely, Section II exhibits an opposing trend, which can likewise be correlated with variations in heat release and differences in flame elongation. This trend is further elucidated by the data presented in Figure 9. It appears that a larger outer recirculation zone significantly influences the flame size, typically resulting in shorter flames. Specifically, the cone-shaped structure exhibits lower temperatures in Section I and higher temperatures in Section II compared to other configurations, which correlates with its longer flame size. Additionally, the distribution of OH along the centerline indicates that from 0 to approximately 7 mm, the OH concentrations are higher in the cone-disk, disk, and hemisphere-shaped configurations than in the cone-shaped geometry. However, from 7 mm to 16 mm, the cone-shaped configuration exhibits higher OH values. This pattern is indicative of the longer flame associated with the cone-shaped configuration, highlighting how geometric factors can impact flame characteristics and chemical species distribution in combustion processes.

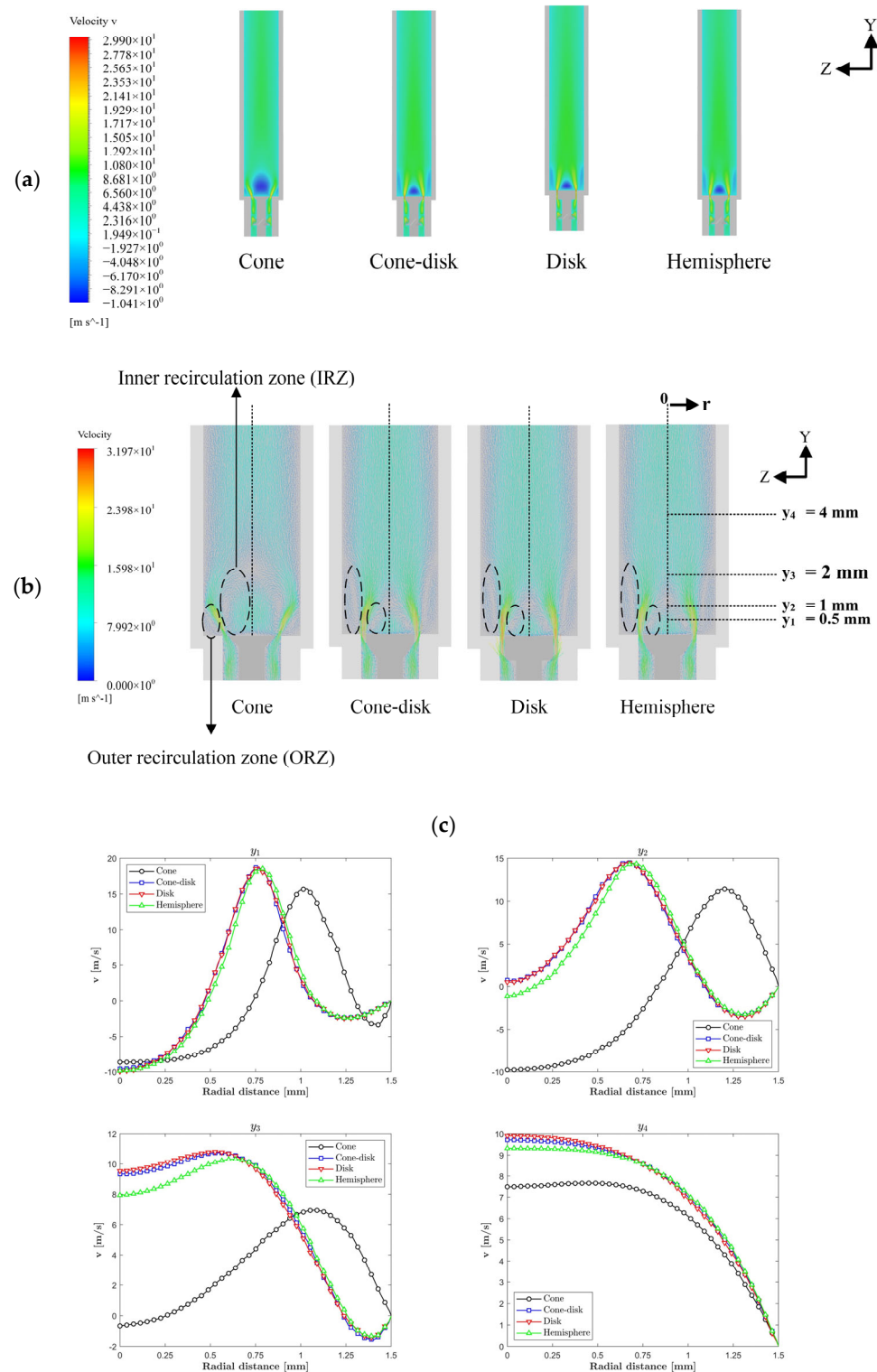


Figure 9. Effect of bluff-body geometry on (a) axial velocity contours; (b) flow-field; and (c) axial velocities at different axial distances.

Figure 9a–c present axial velocity contours, typical velocity fields, and axial velocity profiles along different axial distances for studied BBs. As depicted in Figure 9a,b, two distinct recirculation zones exist: (I) an inner recirculation zone (IRZ) that forms just behind the BB, and (II) an outer (corner) recirculation zone (ORZ) that forms near the wall of the combustor. The flame stabilization is substantially influenced by the IRZ that is located above the bluff-body. This is a result of the persistent ignition of the premixed reactants

discharged from the annular outflow, which is caused by the accumulation of combustion products in the surrounding area [70]. The analysis reveals that the cone-shaped BB exhibits the largest IRZ and the smallest ORZ among the various BB geometries considered. It is observed that by employing cone-disk-shaped, disk-shaped, and hemisphere-shaped BBs the IRZ shrinks and ORZ lengthens compared with cone-shaped BB.

Two peaks are visible in the symmetrical axial velocity profiles, as illustrated in Figure 9c (although only one side of the symmetry is depicted). These peaks, which result from annular fluid flows, are evenly distributed on both sides of the central axis. As the axial position increases from 0.5 mm to 2 mm, the distance of these side peaks (along the radial direction) from the burner axis is observed to increase for cone-shaped BB, while a reverse trend is observed in other BB configurations. It is observed that the side peak axial velocities of cone-shaped BB are the most distant, while the peaks of cone-disk-shaped and disk-shaped are the closest to the burner axis when the different BB designs are compared. This indicates that the flow downstream from the BB is narrowing and will continue to a specific point. Furthermore, for cone-disk-shaped, disk-shaped, and hemisphere-shaped BBs, the intensities of the side peaks in the axial velocity profile are higher compared with cone-shaped BBs. It is also observed that the peak intensities will gradually decrease as the axial distance from the burner exit plane increases. Finally, the elevated temperature and OH concentration in the vicinity of the burner exit plane are the result of the accumulation of high-temperature combustion products and the presence of an inner recirculation zone (as illustrated in Figure 9). Cone-disk-shaped, disk-shaped, and hemisphere-shaped BBs exhibit the highest positive (~ 18.8 m/s) and reverse velocities (~ -9.9 m/s) at the axial location of 0.5 mm. The larger reverse velocities are a representation of the longer residence time and stronger recirculation IRZ which can result in higher temperature and OH mass fractions near the BB for the mentioned configurations.

The performance parameters of the BB designs under investigation are compared in Figure 10, including the mean outer wall temperature (T_{mean}), normalized temperature standard deviation (NTSD), combustion efficiency (η_C), radiation efficiency (η_{rad}), and NO emissions at the outlet of the combustor. Pivotal factors for modules that employ micro-combustion for direct energy conversion include the mean temperature of the outer wall and radiation efficiency. The investigation reveals that the combustor featuring a cone-shaped BB demonstrates the lowest metrics for mean outer wall temperature, temperature uniformity across the outer wall (lower values for NTSD are an indication of the more uniform temperature of the outer wall), combustion efficiency, radiation efficiency, and NO mass fraction at the outlet. According to Figure 9a,b, cone-disk, disk, and hemisphere-shaped BB configurations exhibit the largest ORZ compared with the disk-shaped one. The outer recirculation zone helps in the radial spreading of heat and assists in a more even distribution of thermal energy within the combustor. This distribution plays a crucial role in the thermal performance of the combustor, impacting the overall heat transfer to the combustor walls and to the working fluid in micro-thermophotovoltaic systems. Additionally, the swirling motion and the outer recirculation zones increase the residence time of the gasses within the combustor. Increased residence time allows for the complete combustion of fuel, leading to improved combustion efficiency. The mean outer wall temperature for cone-shaped BB is almost 30 K lower than other configurations. The mass fraction of NO at the combustor's outlet for the cone-shaped bluff-body (BB) is 76% lower than that of the cone-disk-shaped, 76.4% lower compared to the disk-shaped, and around 75.3% less than the hemisphere-shaped BB configurations. The lower NO values observed in the cone-shaped configuration, compared to other designs, can be attributed to a more uniform temperature distribution within the combustor and generally lower temperatures. These conditions are conducive to reducing the production of thermal NOx. Furthermore,

a lower concentration of OH radicals in the cone-shaped configuration may also play a role in suppressing NO_x generation through the HNO pathway and the oxidation of N/NH. The presence of OH radicals typically facilitates the conversion of NH_3 to NH_2 , which subsequently activates pathways leading to increased NO_x production. For a more detailed explanation of these mechanisms, please refer to our previous publication [40]. Alterations in the BB design from cone-shaped to alternative configurations yield noticeable enhancements, with combustion and radiation efficiencies experiencing improvements of approximately 5% and 15%, respectively. Moreover, it is observed that configurations adopting a cone-disk, disk, and hemisphere shape exhibit comparable levels in terms of mean outer wall temperature, outer wall temperature uniformity, combustion efficiency, and radiation efficiency. In addition, among the configurations examined, the disk-shaped BB configuration registers the highest concentrations of NO at the outlet and achieves the greatest outer wall temperature uniformity.

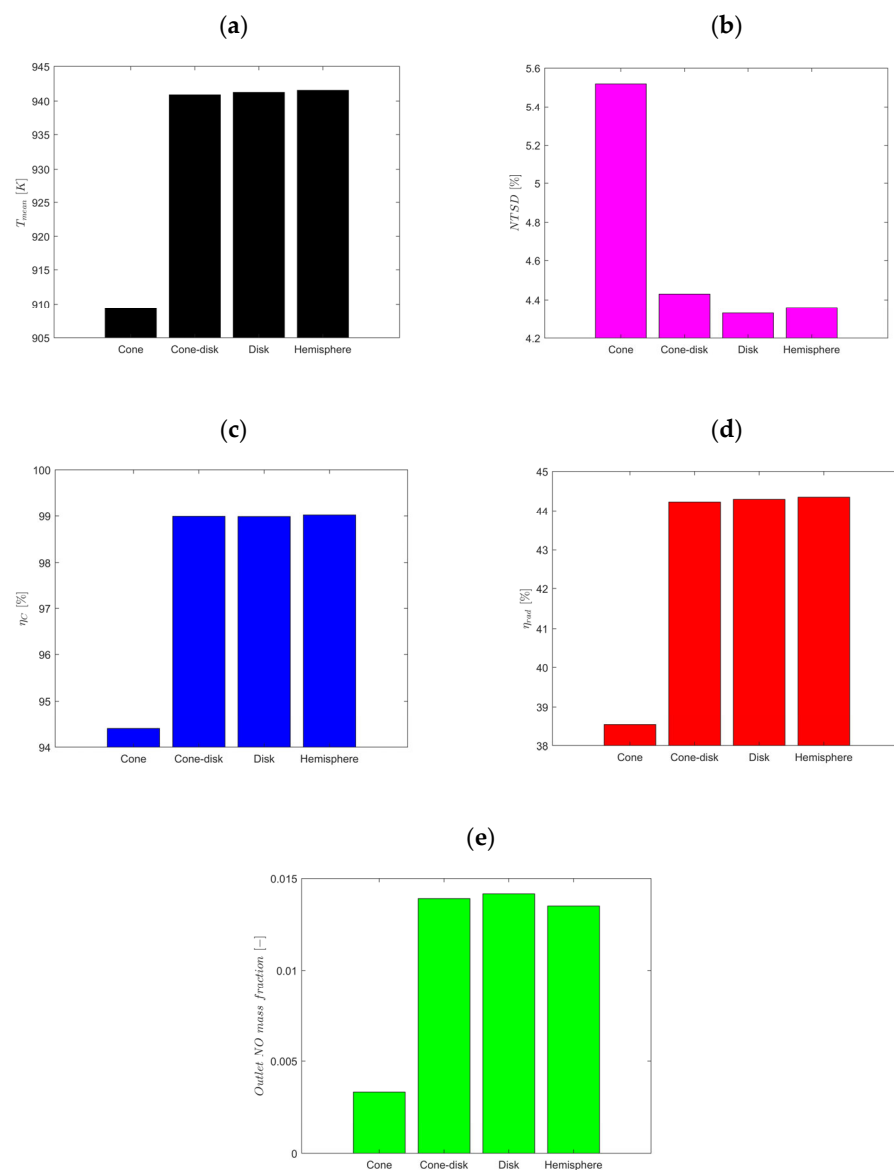


Figure 10. Effect of bluff-body geometry on (a) mean outer wall temperature; (b) NTSD; (c) combustion efficiency; (d) radiation efficiency; (e) NO emission.

3.2. Effects of Vane Angle

This section investigates the impact of an increasing swirl number (reflected by vane angle) on the combustion and thermal performance of hydrogen/ammonia/air flames in a cone-shaped BB and swirl-stabilized micro-combustor under the inlet mass flow rate of 7×10^{-6} kg/s. For these simulations, the silicon carbide was chosen as the material of the combustor and chosen geometrical parameters include a bluff-body size of 1.5 mm and a cone angle of 90° , and the swirler vane angle spans from 15° to 60° .

Figure 11 illustrates the effects of changing the vane angle which is indicative of swirl strength, on the distribution of temperature and OH mass fraction within the combustor. The information is given by contour plots and profile plots along the centerline. Figure 11a,b demonstrate that the regions of high temperature and high OH mass fraction are predominantly localized along the axial centerline, and increasing the vane angle leads to a reduction in the length of these distributions (high temperature and high OH mass fraction zones). Furthermore, it is noted that the width of high temperature region extends and the concentration of the high OH mass fraction region is also seen to shorten longitudinally and move closer to the burner exit plane with larger vane angles. This indicates the presence of shorter flames.

According to Figure 11c,d, there are two peaks for temperature and OH mass fraction along the centerline. Immediately following the BB, a sudden increase in temperature and OH mass fraction is observed, which can be attributed to the formation of a recirculation zone. This is subsequently followed by a descending trend and an additional increase and decrease. The initial peak is located just downstream of the BB, with minimal variation across all vane angles that have been analyzed. Conversely, the second peak moves toward the BB as the vane angle of the swirler increases (5.5 mm for 15° vane angle and 3.5 mm for 60° vane angle). The distribution of OH radicals within the combustor can be categorized into two distinct sections: Section I, spanning from 0 to 8 mm, and Section II, extending from 8 to 16 mm. It has been observed that as the vane angle of the swirler increases, there is a corresponding increase in the OH distribution within Section I. Conversely, in Section II, a different trend is evident, where the swirler with a 60-degree vane angle displays the lowest OH values. This phenomenon can be attributed to the formation of a shorter flame when using a 60-degree vane angle, as clearly depicted in the OH contour (Figure 11b). In addition, Figure 11c,d illustrate the trend where an increase in the vane angle of the swirler leads to heightened peak temperatures and OH mass fractions. Specifically, when the vane angles are set at 15° , 30° , 45° , and 60° , the respective peak gas temperatures reach 1768, 1784, 1825, and 1965 K.

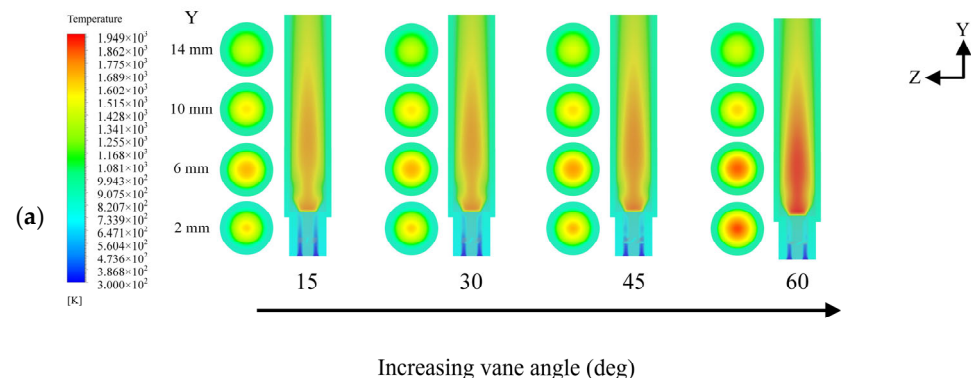


Figure 11. Cont.

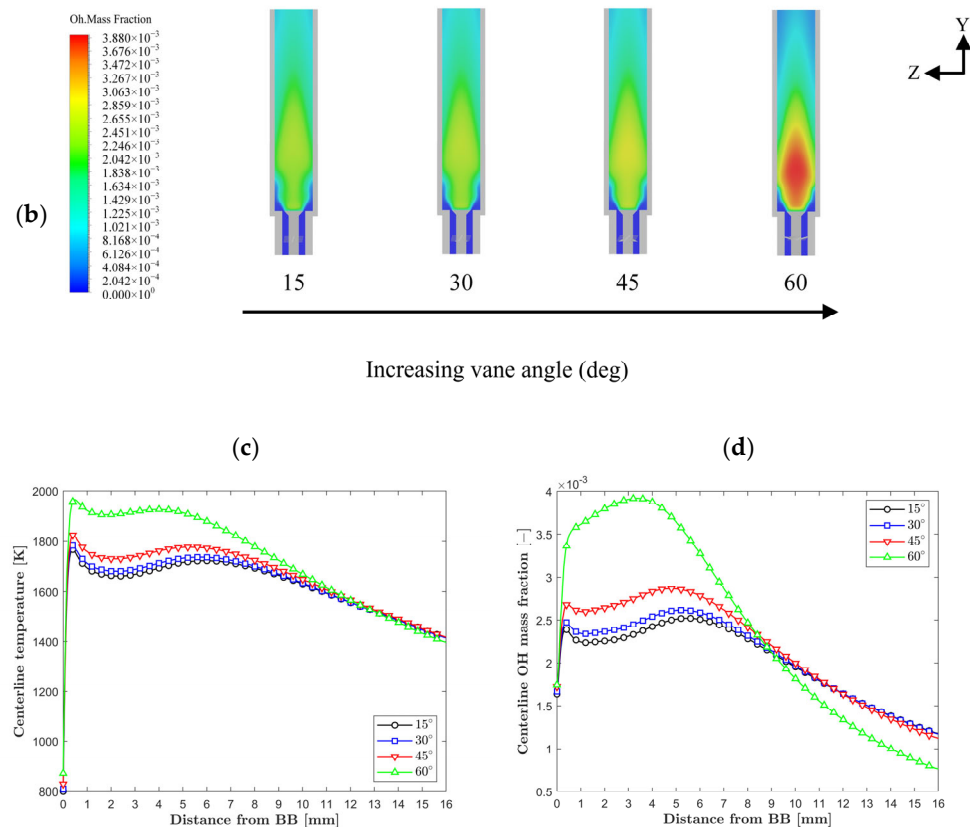


Figure 11. Impact of increasing vane angle of the swirler on (a) temperature contours; (b) OH mass fraction; (c) centerline temperature; (d) centerline OH mass fraction.

Figure 12a,b present the axial velocity contours and their corresponding profiles along the radial direction at varying axial distances for the vane angles under study. From Figure 12a, it is evident that changes in the swirler vane angle do not significantly impact the strength of the IRZ, a phenomenon that is consistently observed across axial velocity profiles at four distinct axial locations, as illustrated in Figure 12b. According to Figure 12b, the distance of the axial velocity's side peak—attributable to annular flow—increases from the burner axis as the axial position extends up to 2 mm, signifying the flow's expansion downstream from the burner exit. It is observed that side peak values (positive axial velocity) increase with higher swirl numbers (vane angles), so that the setup with a 60-degree vane angle is distinguished by achieving the most significant positive axial velocity, while the reverse axial velocities remain almost the same across all configurations. Moreover, when analyzing four scenarios with distinct vane angles, it becomes apparent that the side peak of axial velocity for the configuration employing a 60-degree vane angle is positioned closer to the burner axis compared to the others, up to an axial distance of 2 mm. As observed in Figure 12a, near the burner exit plane, the influence of the bluff-body on the flow pattern appears to be more significant than that of the swirl strength. However, as the distance from the burner increases, the effect of the bluff-body diminishes, and the differences in axial velocity distribution become more pronounced. As depicted in Figure 12b, closer to the centerline, the axial velocity decreases with an increase in the swirl number, whereas the opposite trend is observed further from the centerline. This phenomenon can be explained by the following mechanism: Introducing a swirl into a combustor imparts a rotational motion to the airstream. With higher swirl numbers, this rotational motion intensifies, generating an outward centrifugal force on the flow. This force causes the flow to expand radially, and as it moves away from the burner and spreads

outward, the axial component of the velocity naturally decreases due to the conversion of flow energy from axial to radial momentum.

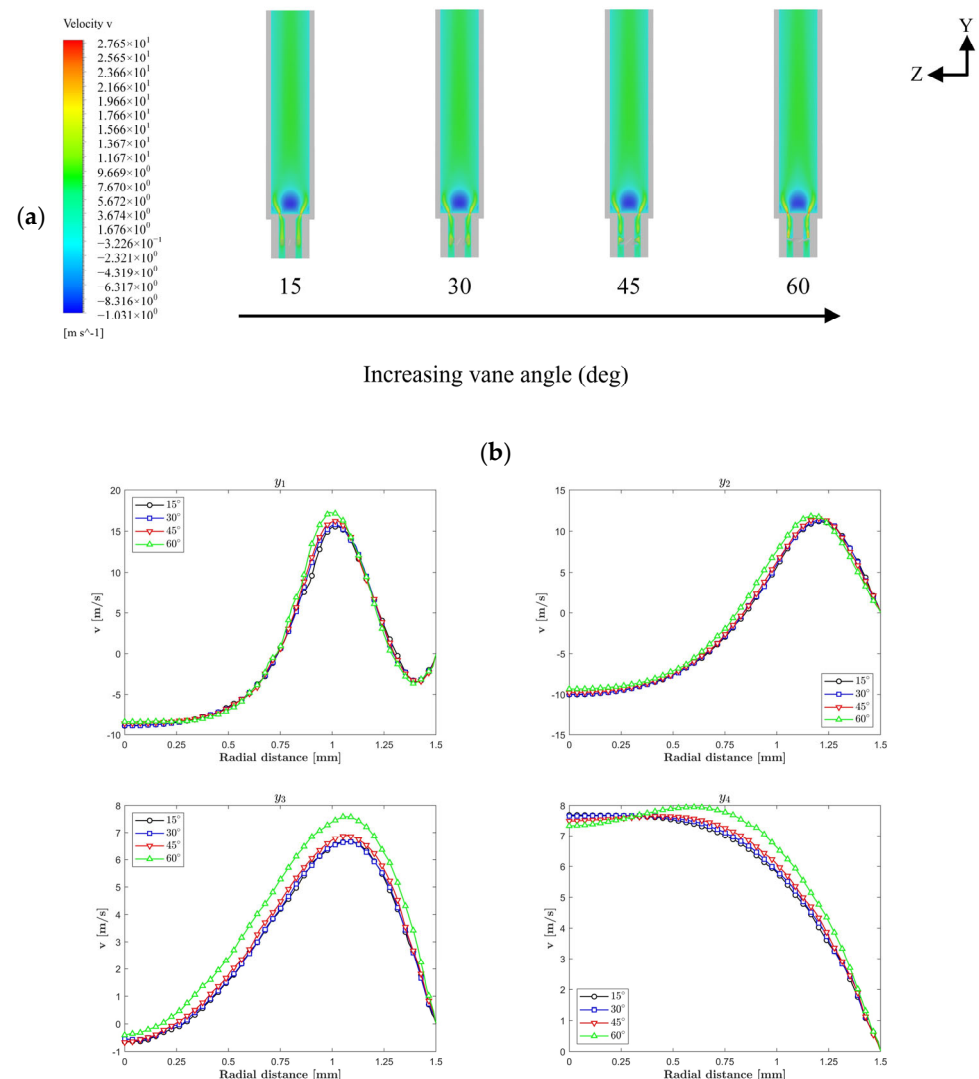


Figure 12. Effect of increasing the vane angle of the swirler on (a) axial velocity contours; (b) axial velocities at different axial distances.

Figure 13 illustrates that alterations in the vane angle of the swirler significantly impact the combustion and thermal performance parameters. Observations reveal enhancements in mean outer wall temperature, outer wall temperature uniformity, combustion efficiency, and radiation efficiency with an increase in the swirler's vane angle (or swirl number). An elevated vane angle leads to a longer flow residence time, more complete combustion, and increased heat release, thereby improving the mean outer wall temperature and, consequently, the radiation efficiency at a given inlet mass flow rate. Notably, transitioning from a swirler with a 15-degree vane angle to one with a 60-degree angle yields improvements of 4.6%, 24.5%, 6.7%, and 20.4% in mean outer wall temperature, outer wall temperature uniformity, combustion efficiency, and radiation efficiency, respectively. Moreover, a micro-combustor equipped with a 60-degree vane angle swirler exhibits the highest NO mass fraction in the exhaust gas (approximately 36% more than other vane angles), attributable to the higher combustion temperatures. Such temperature increases are expected to enhance thermal NO_x emissions from hydrogen and fuel NO_x emissions from ammonia. This is

attributed to higher concentrations of the radicals OH, O, and H in higher combustion temperatures that promote NO formation via HNO and NH oxidation pathways [71].

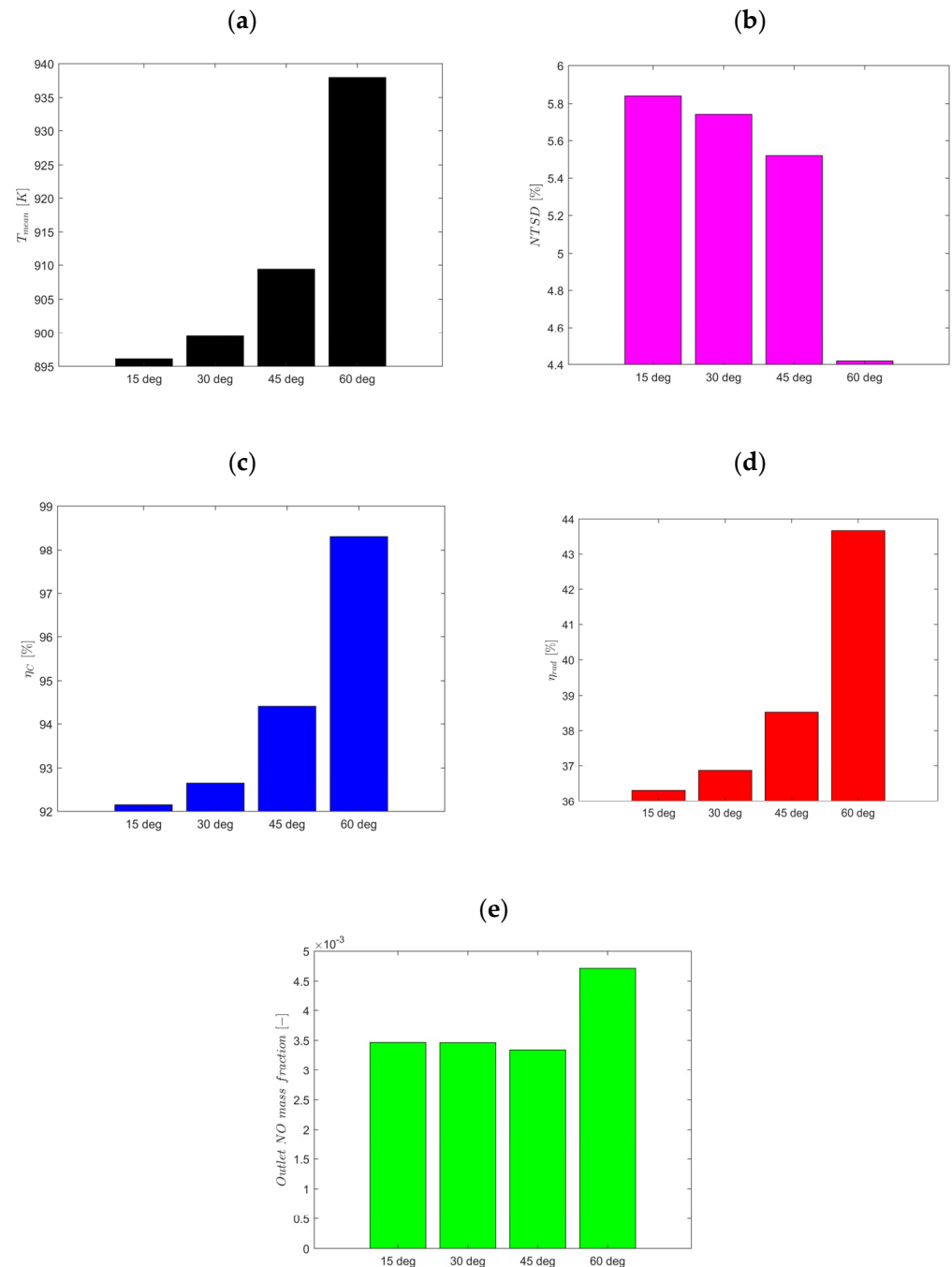


Figure 13. Effect of increasing vane angle of the swirler on (a) mean outer wall temperature; (b) NTSD; (c) combustion efficiency; (d) radiation efficiency; (e) NO emission.

3.3. Inlet Mass Flow Rate Effect

This section investigates the thermal performance and combustion characteristics of the micro-combustor as a function of the inlet mass flow rate. A micro-combustor made of silicon carbide featuring a cone-shaped bluff-body and an axial swirler with a vane angle of 60° (swirl number of 1.313) is chosen to examine the effects of inlet mass flow rate.

Figure 14 illustrates the impact of increasing the inlet mass flow rates on the temperature and OH mass fraction profiles, shown through contour plots and along the combustor's central axis. As seen in Figure 14a, growth in the spatial extent of high-temperature areas,

both wider and taller, accompanies heightened mass flow rates. This effect is fundamentally due to the heightened heat generation when mass and fuel flow rates increase while maintaining an equivalence ratio of 1. Notably, the total nominal heat input rises from approximately 20.5 W to around 38 W as flow rates increase from 7×10^{-6} kg/s to 13×10^{-6} kg/s. Furthermore, the high-temperature zone of the flame is characterized by a flat leading edge, which then evolves into a sharply pointed end in the trailing edge in the burner axis.

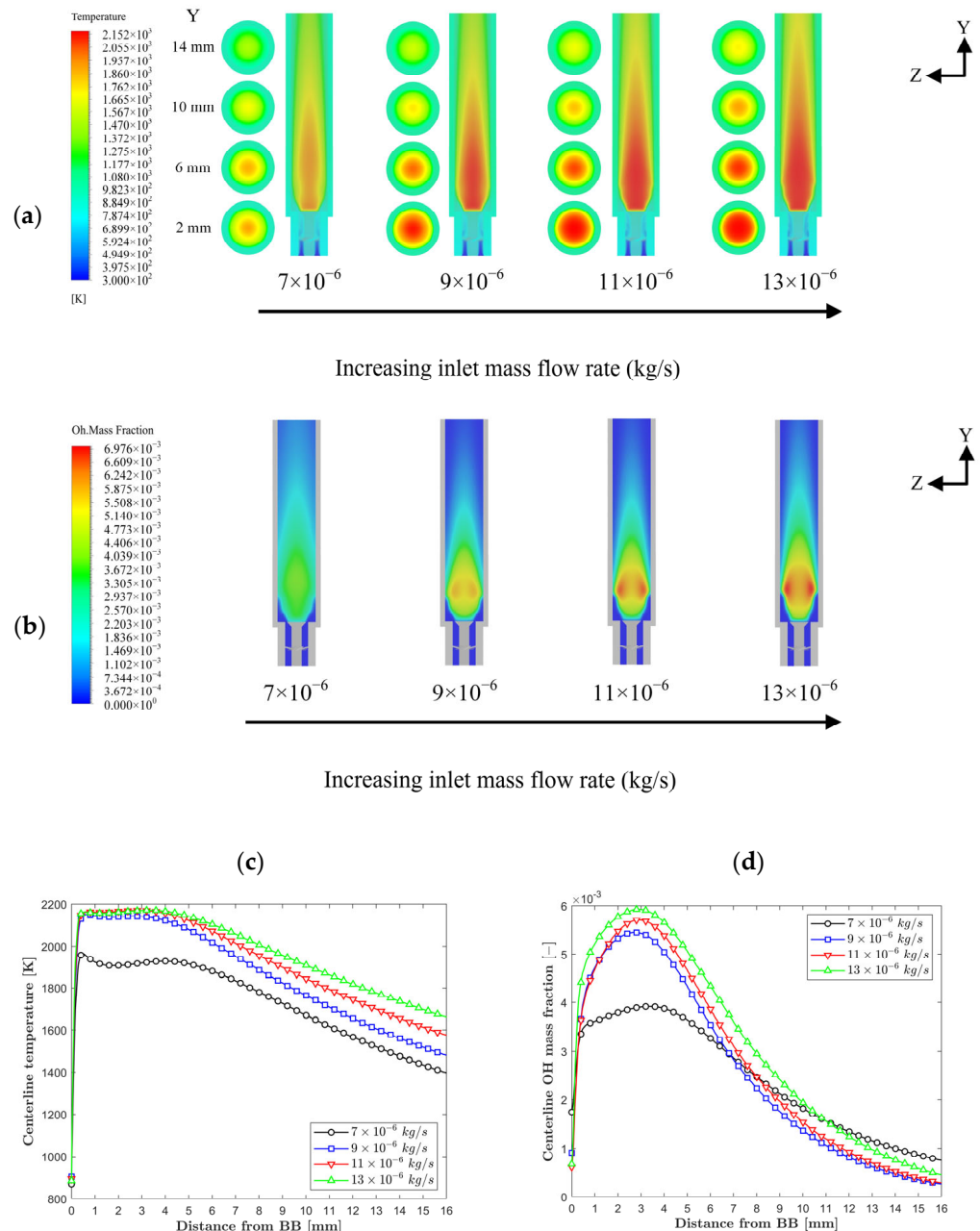


Figure 14. Effect of inlet mass flow rate increase on (a) temperature contours; (b) OH mass fraction; (c) centerline temperature; (d) centerline OH mass fraction.

In Figure 14c, the temperature distribution along the centerline is depicted. It is observed that there is a significant initial temperature rise just after the BB exit plane, succeeded by a sequence of descent, subsequent minor rise, and another descent. It is obvious that a heightened inlet mass flow rate results in an overall increase in temperature along the central line, a phenomenon that can be linked to the greater heat release, accompanying

the augmented heat input from increased mass flow into the combustor. Additionally, an increase in the inlet mass flow rate augments turbulence, which in turn intensifies the combustion process. This leads to an enhancement of the IRZ, positively influencing combustion efficiency and boosting heat release proximate to the burner outlet. Specifically, the maximum temperatures associated with respective mass flow rates of 7×10^{-6} kg/s, 9×10^{-6} kg/s, 11×10^{-6} kg/s, and 13×10^{-6} kg/s reach approximately 1965 K, 2150 K, 2170 K, and 2171 K, illustrating an increasing trend in the peak temperature.

Figure 14b demonstrates an alteration in the distribution of the OH mass fraction as the inlet mass flow rate inside the combustor increases. By increasing the inlet mass flow rate, the high-mass fraction region of OH extends radially and longitudinally, and the high-mass fraction region of OH is going to be distributed on both sides of the centerline. Conversely, at a low airflow rate of 7×10^{-6} kg/s, it will be distributed in the middle and around the centerline.

The OH mass fraction's variation along the centerline is depicted in Figure 14d, which shows a trend of increasing and then diminishing. The OH profile can be divided into two sections: Section I, which spans from 0 to approximately 8 mm and represents the first half of the combustion, and Section II, which spans from approximately 8 mm to 16 mm and represents the second half of the combustion. The OH mass fraction in Section I increases as the mass flow rate increases, whereas the low flow rate of 7×10^{-6} kg/s shows a higher OH mass fraction in most of Section II.

Figure 15 illustrates the impact of varying inlet mass flow rates on the axial velocity contours and profiles at different axial distances for the examined inlet mass flow rates. An increase in the inlet mass flow rate enhances the intensity of the Inner Recirculation Zone (IRZ). Notably, the axial velocity profiles near the bluff-body (BB) exhibit two peaks along the specified axial distances, though only one peak is displayed here due to symmetry, representing half of the axial velocity distribution. The separation between these two peaks expands as the axial position extends from 0.5 mm to 2 mm, signifying a broadening of the flow downstream from the burner exits up to a particular point. Upon comparing different mass flow rates, the peak axial velocity at a mass flow rate of 7×10^{-6} kg/s is observed to be further from the burner axis, whereas peaks corresponding to a mass flow rate of 13×10^{-6} kg/s are closest. Furthermore, the intensity of the side peaks in the axial velocity profile escalates as the inlet mass flow rate increases. The condition with a mass flow rate of 13×10^{-6} kg/s demonstrates the most noticeable positive and reverse axial velocities, indicating the longest residence time near the burner exit plane. Ultimately, the peak intensities gradually decrease as the axial distance from the burner starting point plane increases.

In this section, we examine the impact of varying inlet mass flow rates on the combustor's combustion and thermal performance. According to Figure 16, an increase in the inlet mass flow rate is associated with a noticeable rise in both the mean outer wall temperature and NO emissions at the combustor's outlet, which can be linked to the elevated temperatures of exhaust gasses and combustion (Figure 14a). This temperature escalation is likely to augment thermal NO_x emissions from hydrogen and fuel NO_x emissions from ammonia, given that NO formation is primarily influenced by the radicals OH, O, and H. At higher airflow rates, an excess of O₂ enhances the production of O and OH through the reaction $\text{H} + \text{O}_2 \rightarrow \text{O} + \text{OH}$ (Figure 14b). According to reference [71], this mechanism significantly contributes to the increased formation of NO. Furthermore, it has been noted that the NTSD (normalized temperature standard deviation) exhibits a fluctuating trend, reaching its lowest point at an inlet mass flow rate of 9×10^{-6} kg/s. This suggests a point of optimal uniformity in the outer wall temperature distribution. Additionally, the combustion efficiency records its lowest at a mass flow rate of 7×10^{-6} kg/s compared to higher inlet mass

flow rates, attributed to the weakest inner recirculation zone and the shortest residence time at the burner exit, resulting in the incomplete combustion of hydrogen and ammonia. An increase in mass flow rate from 9×10^{-6} kg/s to 13×10^{-6} kg/s also sees a reduction in combustion efficiency, predominantly due to the shorter residence times caused by higher inlet flows that lead to incomplete combustion. Conversely, radiation efficiency is seen to decrease with an increase in the inlet mass flow rate. This can be explained by the fact that as the inlet mass flow rate rises, heat losses from the combustor's outer wall, through both radiation and convection, increase due to the higher mean outer wall temperature. Although the enthalpy in the exhaust gasses also rises with the temperature of exhaust gasses, the relative contribution of heat loss through radiation decreases.

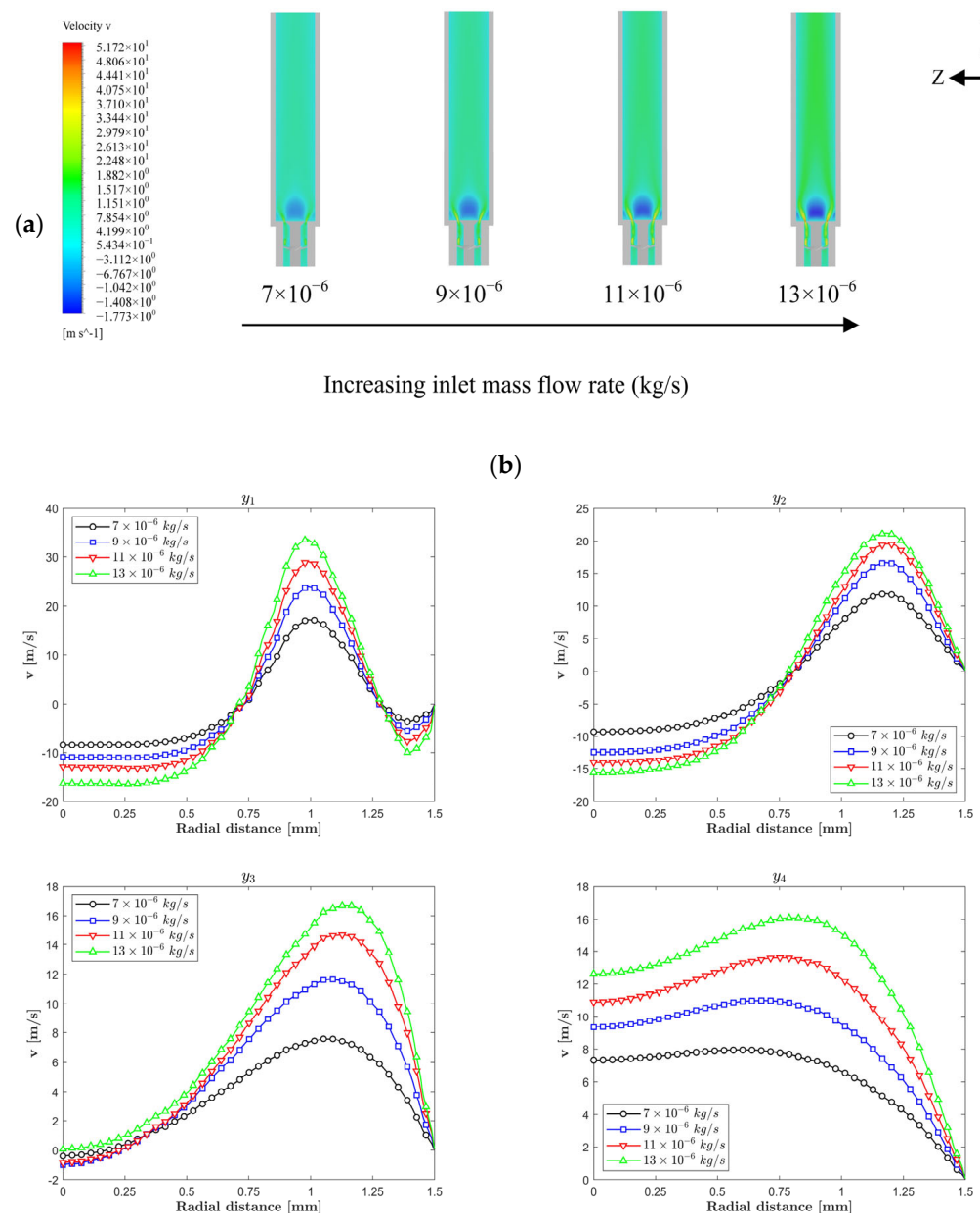


Figure 15. Effect of increasing inlet mass flow rate on (a) axial velocity contours; (b) axial velocities at different axial distances.

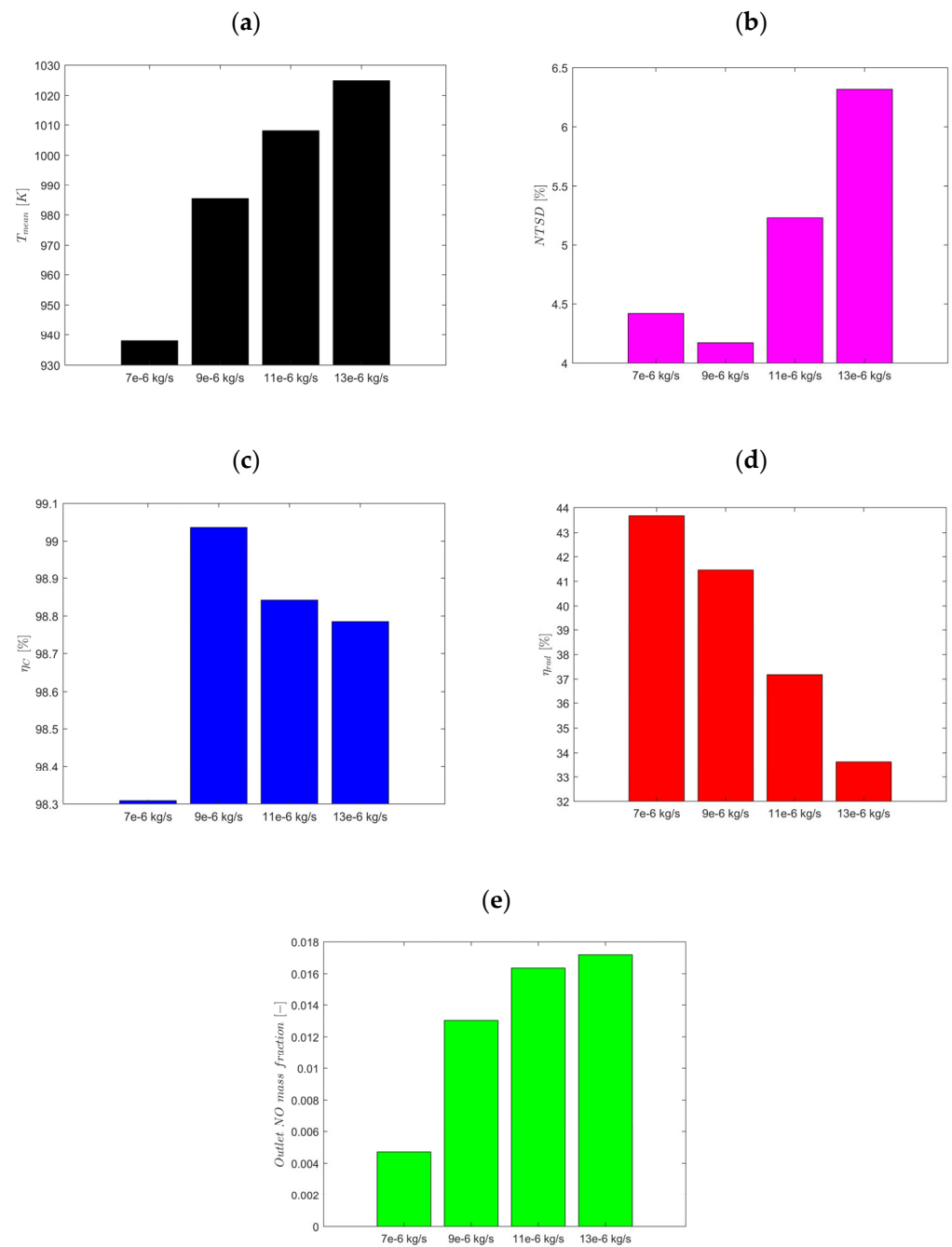


Figure 16. Effect of increasing inlet mass flow rate on (a) mean outer wall temperature; (b) NTSD; (c) combustion efficiency; (d) radiation efficiency; (e) NO emission.

3.4. Effects of Combustor Material

The combustion characteristics and thermal behavior of micro-combustors are greatly influenced by the materials from which they are made [55]. Therefore, a selection of materials for micro-combustors, such as stainless steel, quartz, ceramic, and silicon carbide, has been studied numerically. Table 3 displays the physical and thermal properties of the mentioned materials, including density (ρ), specific heat capacity (c_p), and thermal conductivity (λ). In this section, a cone-shaped BB and a swirler with a 45-degree vane angle are chosen for the micro-combustor, and the inlet mass flow rate is 7×10^{-6} kg/s.

Figure 17 illustrates how different combustor materials impact temperature and the OH mass fraction, as depicted through contour and centerline plots. In Figure 17a, the extent of the high-temperature zone varies along the y -axis, showing expansion, contraction,

a second expansion, and then a final contraction. Notably, this region's lower segment is more constricted for stainless steel and quartz combustors than for those made from ceramic and silicon carbide. The temperature distribution within the solid material, visible in Figure 17a, indicates that combustors made of ceramic and silicon carbide exhibit a more uniform wall temperature compared to those made of stainless steel and quartz. This uniformity is likely due to the materials' thermal conductivity, suggesting that materials with higher thermal conductivity facilitate better lateral (radial) heat distribution across the outer wall, leading to a more uniform external wall temperature. Consequently, selecting a micro-combustor material with superior thermal conductivity is advisable to optimize heat transfer and enhance energy efficiency. As depicted in Figure 17b, the regions with high OH mass fractions in combustors made of stainless steel and quartz are positioned further from the bluff-body (BB) exit plane compared to those in combustors made of ceramic and silicon carbide. Moreover, the region exhibiting high OH mass fractions extends over a greater length in the ceramic and silicon carbide combustors than in those made of stainless steel and quartz. Notably, the combustor constructed from silicon carbide achieves the highest peak in OH mass fraction, a trend that is also evident in Figure 17d. As shown in Figure 17c, there is a significant temperature increase near the bluff-body (BB) plane, which then follows a pattern of decrease, a second increase, and a final decrease, although the second increase is relatively minor for ceramic and silicon carbide materials. When ranking these materials by peak temperature along the centerline, silicon carbide (~1825 K) shows the highest, followed by ceramic (~1798 K), quartz (~1783 K), and stainless steel (~1781 K). The analysis is divided into Section I (0 mm to approximately 7 mm) and Section II (approximately 7 mm to 16 mm). In Section I, flames in stainless steel and quartz combustors exhibit lower temperatures but display higher temperatures in Section II, attributed to differences in flame location and OH mass fraction distribution (as seen in Figure 17b,d). A similar pattern is observed in the OH mass fraction distribution along the centerline, with stainless steel and quartz showing high OH mass fractions more downstream (Section II) compared to ceramic and silicon carbide. Hence, the reactive zone and flame location are positioned more downstream for stainless steel and quartz combustors. Temperature readings for flames in stainless steel and quartz combustors are nearly identical. Furthermore, peak OH mass fraction values are comparable for stainless steel and quartz, while silicon carbide attains the highest OH mass fraction values (Figure 17d).

Figure 18a,b demonstrate the axial velocity contours and their profiles along the radial direction at various axial distances for the materials being analyzed. Observations from Figure 18a reveal that combustors constructed with stainless steel and quartz materials manifest a less pronounced IRZ compared to those made from ceramic and silicon carbide—a trend further supported by the data presented in Figure 18b. According to Figure 18b, as the axial positions increase up to 2 mm, the distance between the axial velocity's side peak associated with the annular flow and burner centerline increases, indicating an expansion of the flow downstream from the burner exit. Moreover, ceramic and silicon carbide materials are identified to produce the highest values for positive axial velocities across all evaluated axial positions, and similarly, they achieve the most significant values for reverse axial velocities up to an axial distance of 2 mm. This phenomenon can be linked to an extended residence time near the bluff-body (BB) and the proximity of flame formation to the burner exit plane, as depicted in Figure 17b through OH mass fraction distributions. In contrast, flames in combustors utilizing stainless steel and quartz are formed further from the burner exit plane, a consequence of their diminished IRZ and lower reverse axial velocities near the burner, as illustrated in Figure 17b. This analysis underscores the significant influence of combustor material on flow dynamics and flame stabilization within the micro-combustor.

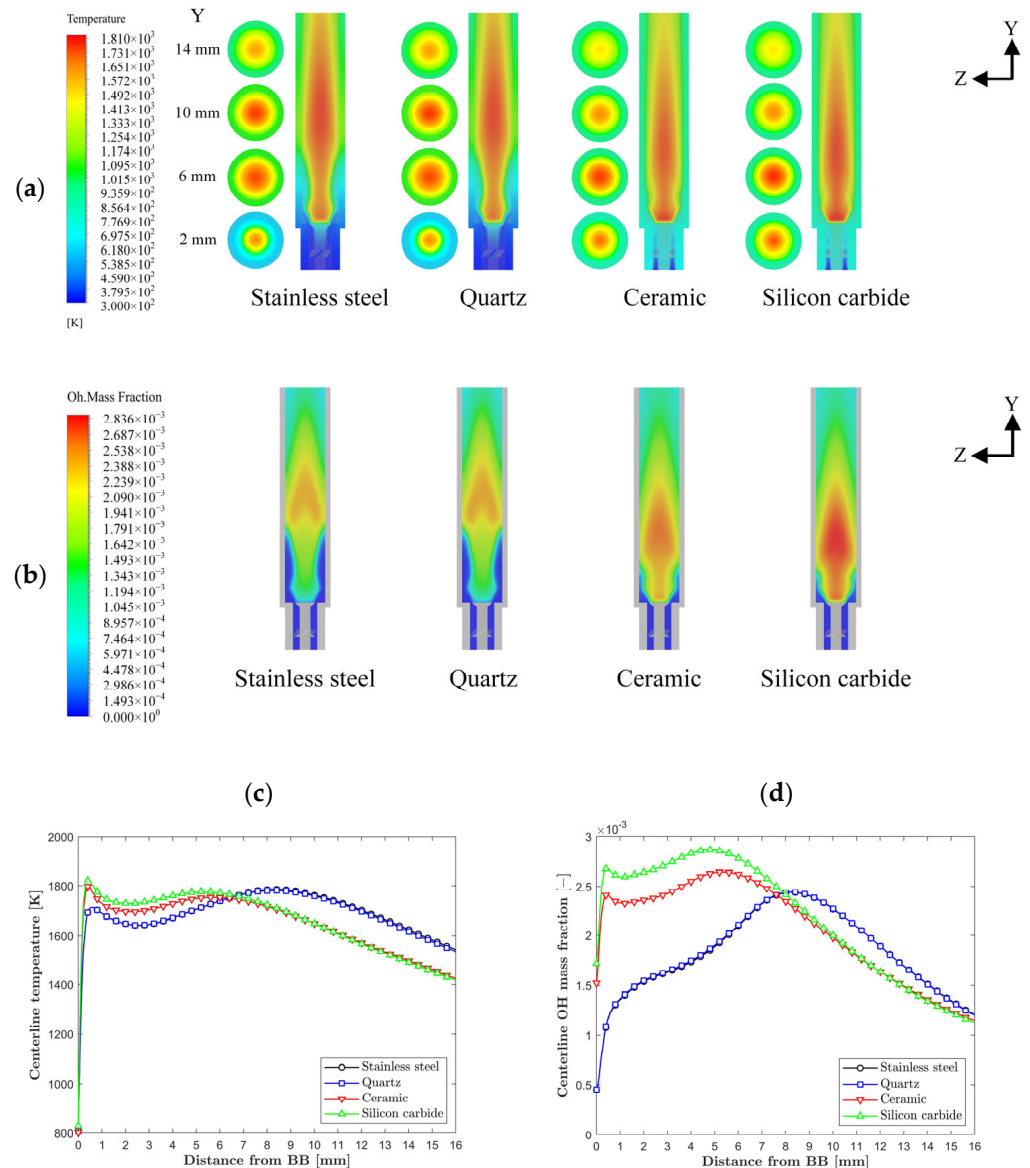


Figure 17. Effect of combustor material on (a) temperature contours; (b) OH mass fraction; (c) centerline temperature; (d) centerline OH mass fraction.

Figure 19 displays the mean outer wall temperature, normalized temperature standard deviation, combustion efficiency, radiation efficiency, and NO emissions for bluff-body (BB) and swirl-stabilized micro-combustors using different materials. According to the data in Figure 19, combustors made with silicon carbide material achieve the highest mean outer wall temperature (approximately 910 K), radiation efficiency (around 38.5%), and the most uniform outer wall temperature. In contrast, those made from quartz exhibit the lowest average outer wall temperature (about 834 K) and radiation efficiency (approximately 27.1%). This means that choosing silicon carbide as the material for the combustor instead of quartz will lead to a radiation efficiency improvement of over 40%. While silicon carbide provides outstanding thermal performance, it is also important to consider the fabrication costs associated with its use in micro-combustor applications when assessing its overall feasibility. According to Figure 19d, arranged in the order of better radiation efficiency, the sequence is silicon carbide (~38.5%), ceramic (~37%), stainless steel (~28.7%), and quartz (~27.1%). This is consistent with the order for mean outer wall temperature depicted in Figure 19a. Furthermore, among the materials evaluated, ceramic demonstrates the lowest

combustion efficiency (Figure 19c). Conversely, combustors made from stainless steel and quartz exhibit the highest combustion efficiencies (about 97%) and the most significant NO emissions (Figure 19e), compared to those made from ceramic and silicon carbide materials.

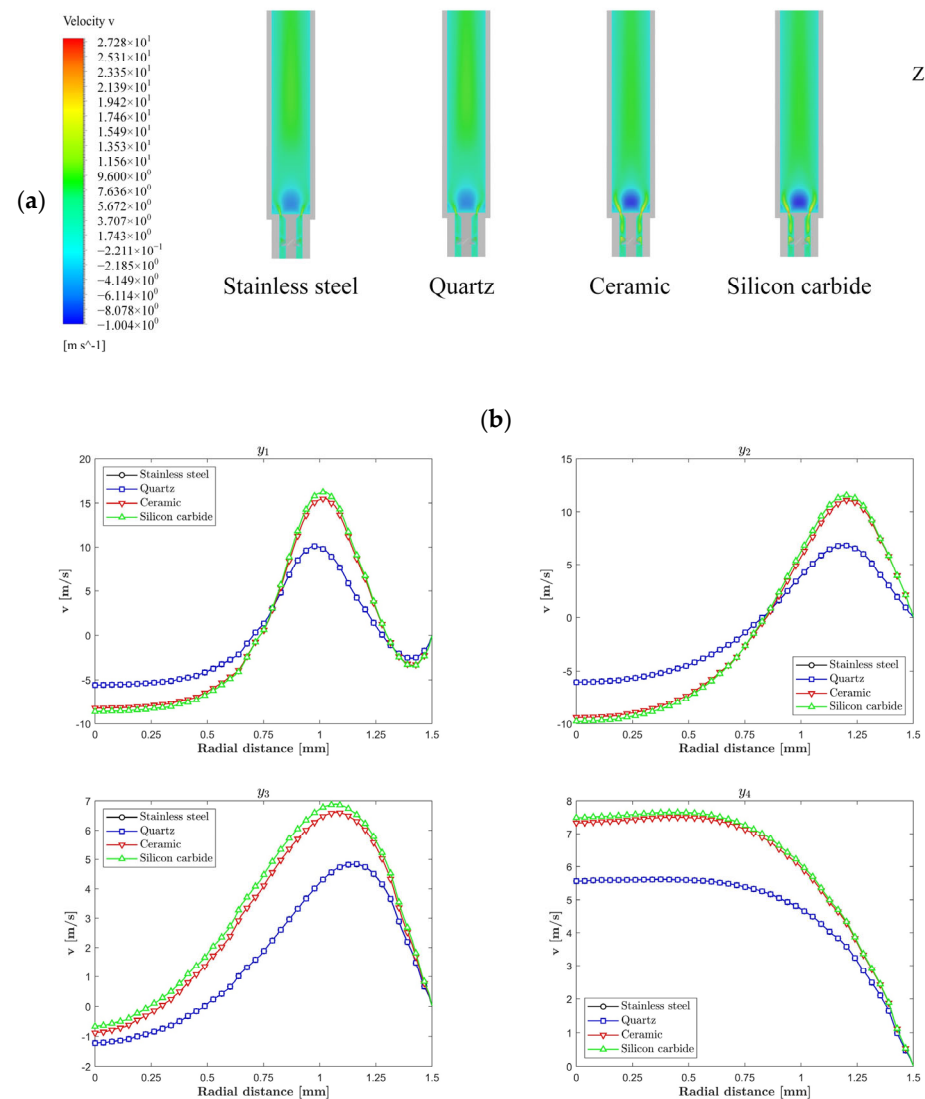


Figure 18. Effect of combustor material on (a) axial velocity contours; (b) axial velocities at different axial distances.

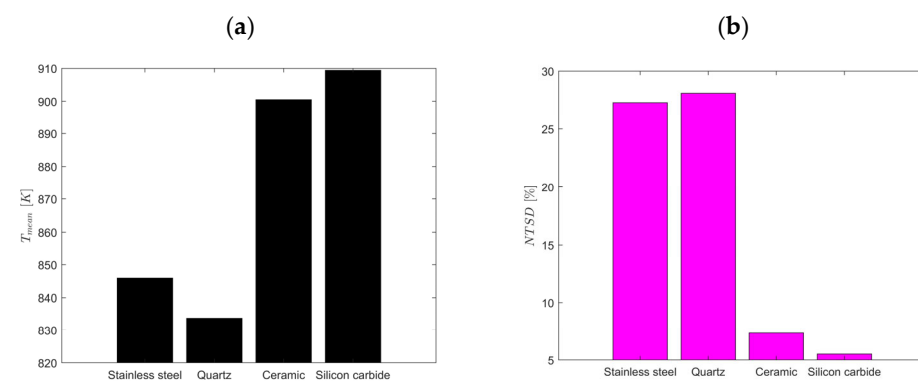


Figure 19. Cont.

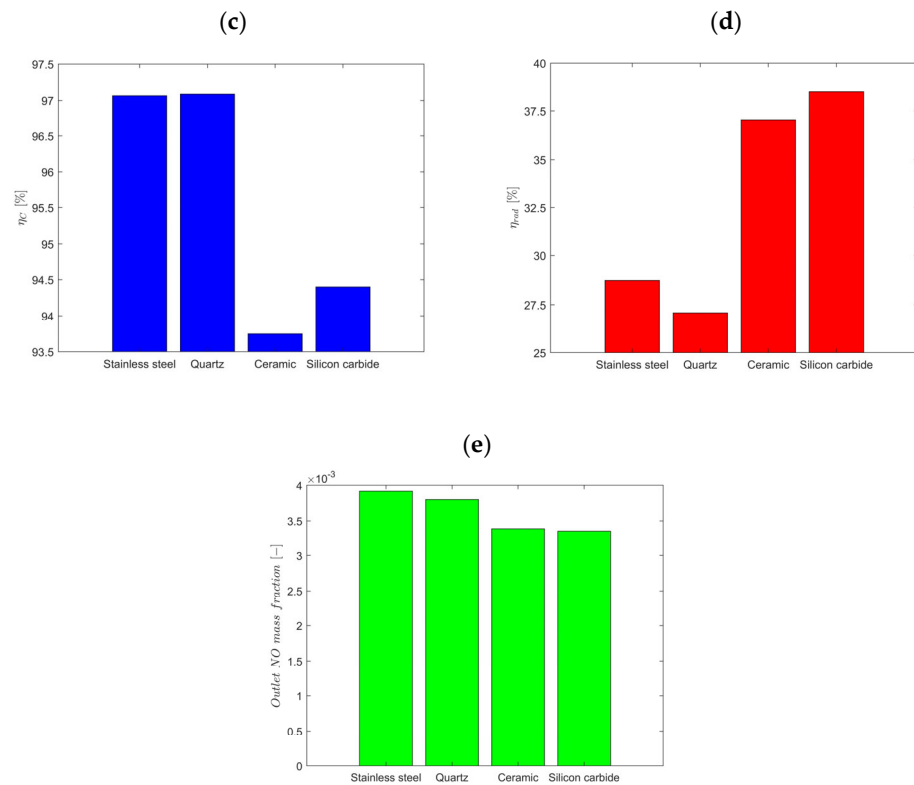


Figure 19. Effect of combustor material on (a) mean outer wall temperature; (b) NTSD; (c) combustion efficiency; (d) radiation efficiency; (e) NO emission.

3.5. Sensitivity of Studied Parameters

This section investigates the sensitivity analysis of thermal performance metrics and combustion with regard to a variety of factors, including the swirler vane angle, inlet mass flow rate, combustor material, and BB shape. The purpose of this research is to identify the factors that have the greatest impact on these important indicators of performance.

Figure 20 illustrates the variation in mean outer wall temperature, normalized temperature standard deviation, combustion efficiency, radiation efficiency, and NO emissions relative to their average values under various BB shapes (indicated by “BB”), swirler vane angle (indicated by “Angle”), inlet mass flow rate (indicated by “Inlet mass flow”), and combustor material (indicated by “Material”). Our results show that among the parameters we examined, the combustor material is the most important one, causing big changes in the mean outer wall temperature ($\sim 3.7\%$), normalized temperature standard deviation ($\sim 62.3\%$), and radiation efficiency ($\sim 15\%$). This emphasizes the critical role of the combustor’s material in optimizing combustion processes and attaining the intended thermal outcome. In addition, the BB shape has been recognized as a substantial determinant of NO emissions ($\sim 35\%$), surpassing other parameters, despite the fact that it exhibits the least change in mean outer wall temperature and radiation efficiency. This finding highlights the BB shape’s critical influence on emission characteristics, making it a useful tool for emission control. In addition, it is observed that alterations in the vane angle of the swirler stand out for their pronounced effects on combustion efficiency.

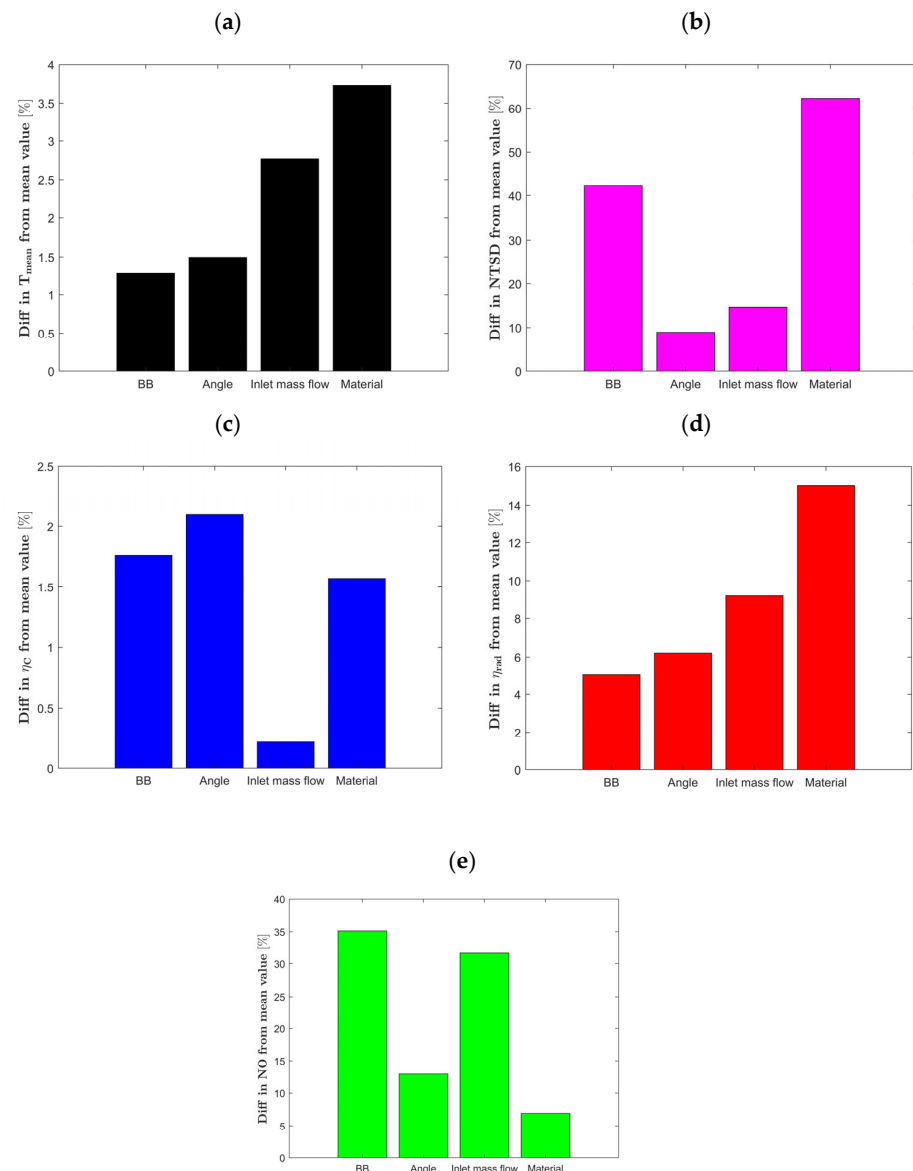


Figure 20. Percent change in (a) mean outer wall temperature; (b) NTSD; (c) combustion efficiency; (d) radiation efficiency; (e) NO from their mean values under studied parameters.

4. Conclusions

This study introduces a bluff-body and swirl-stabilized micro-combustor powered by an ammonia/hydrogen fuel mixture and evaluates its combustion and thermal performance across various parameters such as bluff-body geometry, swirler vane angle, inlet mass flow rate, and combustor material. Key insights from this investigation include the following:

- **Bluff-body Geometry:** The shape of the bluff-body significantly affects the micro-combustor's performance. Cone-shaped bluff bodies demonstrated the lowest performance metrics except NO emissions, while the hemisphere-shaped configuration excelled, achieving over 44% radiation efficiency, thus highlighting its effectiveness in designing combustion-based micro-thermophotovoltaic systems;
- **Vane Angle of the Swirler:** Adjustments to the vane angle significantly impact the combustor's performance. Transitioning from a 15-degree to a 60-degree vane angle swirler resulted in substantial enhancements, increasing the mean outer wall temperature by 4.6%, combustion efficiency by 6.7%, and radiation efficiency by 20.4%. The 60-degree swirler notably achieved approximately 44% radiation efficiency;

- **Inlet Mass Flow Rate:** Changes in the inlet mass flow rate directly influence the combustor's performance, impacting the mean outer wall temperature and NO emissions, while slightly affecting radiation efficiency. Notably, an inlet mass flow rate of 9×10^{-6} kg/s provided the optimal combustion efficiency of 99% and the most uniform mean outer wall temperature;
- **Material Choice:** The selection of combustor material profoundly impacts its thermal characteristics and performance. Silicon carbide outperformed other materials, reaching the highest mean outer wall temperature (approximately 910 K) and radiation efficiency (around 38.5%), and exhibited the most uniform outer wall temperature. Conversely, quartz displayed the lowest performance metrics, with a mean outer wall temperature of about 834 K and radiation efficiency of approximately 27.1%. This suggests that opting for silicon carbide could yield more than a 40% improvement in radiation efficiency over quartz.

These findings are vital for advancing micro-combustion-based power generators aimed at reducing carbon emissions. The insights provided offer valuable direction for future research and improvements in micro-combustion technology, paving the way for more efficient and environmentally friendly energy solutions.

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Nomenclature

Symbol	Parameter	Symbol	Parameter
SN	swirl number	k_s	thermal conductivity of solid phase
D_h	swirler hub diameter	T_s	temperature of solid phase
D_{sw}	swirler tip diameter	$\dot{Q}_{rad}$	radiation heat transfer
θ	vane angle of the swirler	$\dot{Q}_{conv}$	convection heat transfer
ρ	density	$\dot{Q}_{loss}$	total heat loss
c_p	specific heat	ε	emissivity
λ	thermal conductivity	σ	Stefan–Boltzmann constant
K_n	Knudsen number	A	exterior surface of the combustor
$\vec{v}$	velocity vector	T_{mean}	mean outer wall temperature
μ	molecular viscosity	$A_{w,i}$	surface of element i
I	unit vector	$T_{w,i}$	temperature of element i
E_f	overall energy of the fluid	η_{rad}	radiation efficiency
k_{eff}	effective conductivity	$\dot{m}_{fuel}$	fuel mass flow rate
T_f	fluid's temperature	LHV	lower heating value
S_f^h	source term of the fluid enthalpy	$NTSD$	normalized temperature standard deviation
$\vec{D}_i$	diffusion flux of species i	η_C	combustion efficiency
h_i	specific enthalpy of that species i	$\dot{m}_{fuel,in}$	inlet fuel flow rate
Y_i	species i mass fraction	$\dot{m}_{fuel,out}$	outlet fuel flow rate
ω_i	total rate at which species i is produced or consumed through chemical reactions		

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