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A COMPUTATIONAL STUDY OF HYDROGEN SUBSTITUTION EFFECTS ON THE COMBUSTION PERFORMANCE FOR A MICRO GAS TURBINE

Hsin-Yi Shih and Chi-Rong Liu

Department of Mechanical Engineering Chang Gung University, Taoyuan, Taiwan

ABSTRACT

The effects of hydrogen substitution on methane/air combustion in a micro gas turbine were studied in this work. The combustion performance and emission characteristics of a can type combustor were investigated with model simulations using the commercial code STAR-CD, in which the three-dimension compressible k-E turbulent flow model and presumed probability density function for chemical reaction between methane/hydrogen/air mixtures were used. With hydrogen being the substituent, not a supplement to methane, the detailed flame structures, distributions of flame temperature and flow velocity, and gas emissions were presented and compared by using a fraction of hydrogen to substitute methane in the combustor. For the scenarios from pure methane to pure hydrogen, results show the flame temperature and exit gas temperature increase when only 10% methane is substituted. But as hydrogen substitution percentage increases, the flame temperature and exit gas temperature decrease because of a power shortage caused by lower mass flow rate and heating value of the resulting blended fuels, although the pattern factor drops drastically compared to that of pure methane. As the fuel inlet velocity decreases from 100 m/s to 20 m/s, the high temperature region shifts to the side of the combustor due to the high diffusivity of hydrogen. Increasing hydrogen substitution percentage at a fixed fuel injection velocity reduces NO_x emission due to lower flame temperature, but CO emissions increase continually with increasing hydrogen substitution percentage because oxygen depletion for methane/air combustion. Before hydrogen blended fuels or pure hydrogen are used as an alternative fuel for the micro gas turbine, further experimental testing are needed as the CFD modeling results provide a guidance for the improved designs of the combustor.

Keywords: Hydrogen substitution effects, Hydrogen blended fuels, Gas turbine combustion, Micro gas turbine.

INTRODUCTION

Due to the shortage of fossil energy and increasing concerns about greenhouse gas emission, hydrogen or hydrogen blended fuels has been considered as one of the most promising clean fuels in the applications of engines. Hydrogen has a number of properties that make it an attractive alternative, such as lower ignition energy and flammability limits, higher flame speed (i.e. higher reactivity) and heating value per unit mass compared to conventional hydrocarbon fuels. Research works have showed adding hydrogen in hydrocarbon fuels could allow stable lean burn at a lower temperature [1]. Hydrogen addition to natural gas feedstock or hydrogen-natural gas blended fuel has the potential to reduce thermal NO_x production [2], and mostly, hydrogen can be utilized as an additional supply besides the hydrocarbon fuels to improve the ignitibility and flammability for lean premixed flames [3, 4].

There have been many studies regarding hydrogen fuel in spark-ignition engines [5, 6]. However, hydrogen has seen limited use in gas turbine engines. Adding hydrogen to augment natural gas for gas turbine combustion has been investigated in the past. For example, fuels containing up to 10% hydrogen were utilized in a commercial gas turbine [7]. Conceptual studies and development of gas turbine power generators were conducted with hydrogen addition up to 40% to improve the combustor performance [8, 9]. Also, hydrogen addition effects on the combustion performance of a micro gas turbine were studied with model analysis [10]. The combustion and emission characteristics were analyzed with hydrogen concentrations from 10% to 80%. On the more fundamental bases, the effects of hydrogen enrichment on a lean premixed methane flame at hydrogen concentrations up to 20% were studied, and the addition of hydrogen extended the lean stability domain [11]. Significant research efforts have been done using a lean premixed swirl-stabilized combustor [12-14]. The hydrogen addition effects, lean blowout limits and emissions of CO, NO_x on methane-air flames were investigated

for blended fuels with various hydrogen concentrations up to 80%. Also, some experimental and numerical simulations were made to study the flame characteristics of hydrogen-methane hybrid fuels in jet diffusion flames [15-17]. Detailed structure of non-premixed flames of hydrogen-methane blended fuels with certain compositions or various hydrogen fractions were compared.

Most of these previous studies focus on the hydrogen addition effects. It means the hydrogen is a supplement or an additional fuel besides natural gas to augment the combustion in gas turbine engine or modeled combustor. But when hydrogen is not a supplement, but a substituent for hydrocarbon fuel, the power decrease in engines will occur since hydrogen has a lower heating value per mole than the hydrocarbon fuels. The high reactivity and low ignition energy of hydrogen have led to concern of flashback and auto-ignition for premixed systems. Besides, the NO_x emissions produced by a stoichiometric hydrogen reaction will exceed current power industry standards [18, 19], although several experimental works demonstrated improved fuel injection and mixing strategy could help to reduce the emissions of hydrogen fueled gas turbine [20, 21]. Particularly in [21], not just for pure hydrogen, additional experimental tests were run at full load using a varying mixture of natural gas and hydrogen, which is pertinent to current study. Therefore, the purpose of this work is to numerically study the combustion performance and evaluate the hydrogen substitution effects for a micro gas turbine. The 3D numerical simulations of the can combustor were performed using hydrogen substitution percentage and fuel injection velocity as parameters. The fundamental characteristics, resulting operability and emission performance for using hydrogen to substitute methane were discussed.

THE MICRO GAS TURBINE AND CAN COMBUSTOR

A micro gas turbine engine with a thermal power output of 60 kW was proposed and developed, which was designed using methane (CH₄) as the fuel. The feasibility of the system and the combustion performance of the can combustor with pure methane have been modeled previously [22], and the hydrogen addition effects were also discussed in a later study [10]. Current simulation adapted the same model of the can combustor, but investigating the effects of hydrogen substitution for methane on the combustion and emission. A brief overview of the micro gas turbine and the can combustor is provided below.

The schematic of the engine is shown in Fig. 1, which includes the centrifugal compressor, can combustor, axial turbine and the enlarged hollow shaft. Because the combustor is mounted directly inside the shaft, the proposed system greatly reduces the dimensions and weight of the engine core rotor, in comparison with conventional designs of equal power. For the proposed micro gas turbine engine to have power output near 60 kW, the designed efficiency for compressor, combustor and turbine are 0.81, 0.9, and 0.84 respectively. The compression ratio is 3.28, and the expansion ratio is 1.73. The overall thermal efficiency without any heat recuperation will be 12.8% at nominal speed of 58000 rpm. Here the target combustion efficiency of 90% is set lower than the normal application in gas turbine engine because a conservative design was

considered for the newly proposed system in compact size [22]. This invention has granted the U.S. and R.O.C patents [23, 24].



Fig. 1. Schematic of the innovative micro gas turbine.

Figure 2 shows the picture of the can combustor, which was designed based on the design method for conventional combustor with natural gas as the fuel. From the cycle analysis for the proposed micro gas turbine, the airflow rate at the inlet of the combustor is 0.473 kg/sec at 431K and 3.2 atm. The fuel is methane with a flow rate of 0.0096 kg/sec. The overall equivalence ratio is 0.35, and the Reynolds number is around 3.4×10^5 . The design criteria for the can combustor are to achieve the combustion efficiency larger than 90% with exit gas temperature of 1200 K and 6% pressure drop.



Fig. 2. The modeled can combustor.

The combustor is 75 mm in diameter and 180 mm in length, consisting of primary, secondary and dilution zones, followed by a bullet nosed cap to form a transition duct. The diameter of the casing is 95 mm. The air admission holes are determined by evaluating the air distribution to achieve the highest flame temperature in the primary zone, weaker reaction zone in the secondary zone and extinction region in the dilution zone. The air holes are distributed equally around the combustor, eight for primary zone, each 7 mm in diameter, eight for secondary zone, each 5 mm in diameter, and six for dilution zone, each has a diameter of 11 mm. Most of the main air stream flows through the gap between casing and combustor liner and then enter the combustor through the holes. Only about 10% of the entrance air flow directly into the combustor through a swirler.

A simple and compact design with heavy combustion loading was made. Therefore, the swirler was simply designed as twelve rectangular slots circling around the inlet of the combustor, with a vane angle of 30° . The swirl number is estimated near 1. The film cooling was replaced with two rows of small cooling holes located at the combustor dome and dilution zone just ahead of the dilution holes. The gaseous fuel nozzle is inserted axially, and the fuel is injected radially from the small holes that equally spaced around the circumference of the nozzle.

THE MODELING

The commercial CFD software STAR-CD [25] was used to model the combustor. The computational domain includes the interior of the can-combustor, combustor liner and the space between liner and casing. The girds were generated using multi-block method. The meshes for fuel injection holes and the liner holes were created and refined in centric coordinates. Finer cells are located in the primary zone where strong reaction is expected, and also around the air entrance holes where the physical properties change more dramatically. The grids are much denser inside the combustor compared to those for the air stream outside the liner.

One layer of zero-thickness cells called "Baffles" simulated the liner of the combustor. The baffles were assumed to be smooth, stationary, impermeable walls with no-slip condition. Thermal resistance of 6.67×10^{-5} m² K/W was also specified at the baffle, so the amount of heat transferred across the liner can be calculated. Total amount of 302139 grids were used, and grid independence has been checked to balance the computational time and accuracy. The grid structure is shown in Fig. 3, and the baffle representing the combustor linear is shown in Fig. 4. The boundary conditions for casing need to be specified. Except no-slip, smooth, impermeable, the casing was assumed non-adiabatic and at a temperature of 300 K, which indicates the scenario of ambient air cooling on the outside wall of the casing. However, a test result for adiabatic wall condition showed the changes of combustion characteristics are not significant.



Fig. 3. The grid structure of the model.



Fig. 4. The combustor liner represented by baffle.

The theoretical model consists of three-dimensional compressible k-ɛ/high Reynolds number turbulent flow model and chemical reaction between hydrogen/methane and air. Gravitational effect was ignored. The steady-state Navier-Stokes equations were solved using finite-volume method and SIMPLE algorithm [26]. The chemical reaction mechanism adopted the Presumed Probability Density Function (PPDF) for diffusion flames in the chemical module to simulate the combustion process, where ten species (CH₄, H₂, O₂, N₂, H₂O, CO₂, CO, NO, NO₂, N₂O) were considered and solved based on a local instantaneous chemical equilibrium. The mode of nonpremixed flames was chosen because of better understanding of the different diffusion abilities between methane and hydrogen. The turbulent fluctuations of reactant concentrations and temperature are described using a presumed PDF of threedimensional Gaussian distribution based on first and secondorder moments of reactant concentration and temperature. Mean reaction rates are obtained by mean of integration under the presumed PDF. The PPDF model has the advantage of the PDF method, which takes into account the influence of turbulent fluctuation on the mean reaction rates, as well as not requiring computing the full PDF equation.

For turbulent combustion, it has been shown [27] the predicted results of velocity, temperature and species concentrations, especially for NO formation using PPDF model agree better with the experiment than that using the conventional eddy breakup (EBU) model, and comparable with laminar flamelet model. However, laminar flamelet model is more restricted to circumstances for which Damkohler number is very large and the reaction is confined to a thin sheet of laminar structure whose thickness is small in comparison with the scales of turbulence. The NO_x formation is then further calculated with post-processing method [28], because the NO_x concentration is normally low and it has little influence on the flow field and combustion of hydrocarbon. Hence computations of NO_x can be decoupled from the predictions of main reacting flow. In this paper, CHEMKIN [29] was used to post-process the NO_x emissions with GRI 3.0 mechanism [30], where thermal NO_x based on the extended Zeldovich mechanism [31] and prompt NO_x [32] are adopted. Although the model does not represent the state-of-art in simulation of gas turbine reacting flows compared with the model-free approach, such as direct numerical simulation and large eddy simulation, it could provide a reasonable qualitative trend prediction of the complex combustion process and the considerable saving in computing time in studying hydrogen/methane combustion and hydrogen substitution effects in the early developing stage.

RESULTS AND DISCUSSIONS <u>Computational Parameters</u>

All the cases were computed with air flow rate of 0.473 kg/sec at 431 K and 3.2 atm. As for the fuel inlet conditions, the original design of fuel flow rate for methane was 0.0096 kg/sec with fuel inlet velocity of 60 m/s, an overall equivalence ratio of 0.35. To understand the effect of hydrogen substitution, here we used hydrogen substitution percentage to represent the amount of methane which is substituted by hydrogen. In other words, it also indicates the amount of hydrogen in the consequent hydrogen-methane blended fuel. A variable volumetric fraction of hydrogen substitution percentages from

0% (pure methane case) to 100% (pure hydrogen case) were studied. Because of the differences in density and heating value for different hydrogen-methane blended fuels, heat input to the engine could be significantly different. In order to keep up the power output of the engine, the fuel flow rate of the resulting blended fuels would be changed. Therefore, the fuel injection velocities of this blended fuel ranging from 20 m/s to 100 m/s were also parametrically studied.

When part of the methane is substituted by hydrogen, the resulting methane-hydrogen blended fuel has less mass compared to the original methane. Therefore, the fuel flow rate and overall equivalence ratio depend on hydrogen substitution percentage and injection velocity of the blended fuels. The overall equivalence ratios in the computations are then shown in table 1. Lower fuel injection velocities and higher hydrogen substitution percentages decrease the equivalence ratios. The computed range of equivalence ratios is then from 0.58 to 0.03, a relatively wide combustion region. Note that the hydrogen substitution percentage is the volumetric fraction of hydrogen in the methane-hydrogen blended fuels. Since the density of hydrogen is much smaller than that of methane, only when hydrogen substitution percentage is higher than 90%, hydrogen is more than methane in the fuel mixture in terms of mass bases.

Table 1. Overall equivalence ratios for hydrogen substitution percentage (0-100%) and fuel inlet velocity (20-100 m/s).

V _{jul} (n	n/s) 20	40	60	80	100
X _{B1} (%)					
0	0.12	0.26	0.35	0.47	0.58
10	0.11	0.24	0.32	0.43	0.54
20	0.10	0.22	0.30	0.39	0.49
30	0.09	0.20	0.27	0.36	0.45
40	0.08	0.18	0.24	0.33	0.41
50	0.07	0.16	0.22	0.29	0.36
60	0.06	0.14	0.19	0.26	0.32
70	0.06	0.12	0.17	0.22	0.28
80	0.05	0.10	0.14	0.19	0.23
90	0.04	0.08	0.11	0.15	0.19
100	0.03	0.06	0.09	0.12	0.15

Comparisons of Flame Structures

The flame structures of the can combustor are compared first for different hydrogen substitution percentages and fuel inlet velocities. The original design of the combustor was for natural gas fired engine and the fuel injection velocity was 60 m/s. As mentioned earlier, the hydrogen substitution percentage is defined as the molar fraction (or volumetric fraction) of hydrogen in the methane-hydrogen blended mixture. The fuel inlet velocity is then the injection velocity of the blended fuels.

Figure 5 shows the distributions of flame temperature and flow velocity on the axial centerline planes, which are sliced through one of the primary holes, for the amounts of 0% (pure methane), 20%, 40% and 60% hydrogen substitution. The fuel inlet velocities for all these cases are fixed at 60 m/s. Those results are chosen to make the comparisons because it demonstrates clearly a flame structure change qualitatively and quantitatively. When pure methane is used, the flame is stabilized in the core region of the primary zone since the fuel

and air mixes rapidly in the recirculation zone there. The flow recirculation in the primary zone is induced by the vortex breakdown in swirling flows when the rotation imparted to the flow is high. The swirling flows are produced by the swirler, which is sited around the fuel injector. This type of recirculation promotes better mixing and has been widely used in industrial burners and gas turbine combustor. The highest temperature is around 2376K, which is a little bit higher than the adiabatic flame temperature (2310 K) for methane-air mixture at equivalence ratio near 1. Going downstream, temperature decreases due to air-cooling and dilution. The super-adiabatic flame temperature has been found in rich hydrocarbon premixed flame [33] and diffusion flame of CH₄/H₂ blended fuel [34], which is attributed to preferential diffusion of H₂ and H radical. Besides, in previous study, the predicted maximum flame temperature from PPDF model was higher than the experimental data in jet diffusion flame [27].



Fig. 5. Flame structures on the centerline planes for 0%, 20%, 40%, and 60% hydrogen substitution, 60 m/s fuel inlet velocity (from top to the bottom).

When the hydrogen substitution in fuel is increased to 20%, the flame temperature increases drastically and the high temperature region becomes wider and shifting to the sides. The combustion gases are then pushed downstream due to strong thermal expansion effect. The maximum temperature is almost 400 K higher than that of pure methane. Usually the adiabatic flame temperature of H2-air is about 200-250 K higher than that of CH₄-air mixture. The preferential diffusion of H₂ and H radical, as well as the PPDF model may cause such a high temperature. As the hydrogen substitution percentage increases further to 40% and 60%, the flame shape is basically similar to that at 20% hydrogen. But the high temperature zone becomes smaller and moves further to the sides of the combustor. The maximum flame temperature does not change too much, however, the average temperature in the primary zone decreases because of the lower heating value per mole of hydrogen compared to that of methane. The detailed temperature comparisons will be discussed in next section.



Fig. 6. Flame structures on the centerline planes for 40, 60, 80, 100 m/s fuel inlet velocities, 20% hydrogen (from top to the bottom).

At the same fuel injection velocity, the mass flow rate of the methane-hydrogen blended fuel becomes less when the hydrogen substitution percentage is higher. Thus, the flame structures are then calculated for different fuel inlet velocities. The results are shown in Fig. 6, where temperature and velocity profiles are compared for inlet velocities of 40 m/s, 60 m/s, 80 m/s and 100 m/s. The hydrogen substitution percentages are fixed at 20% for those cases. Interestingly, as the fuel injection velocity increases from 40 m/s to 100 m/s, the high temperature zone shifts back to the center of the combustor. The high temperature region is wider and toward downstream as the fuel inlet velocity is higher.

From Fig. 5 and Fig. 6, we can see that a higher hydrogen substitution percentage or a lower fuel inlet velocity, resulting in a combustible mixture with a smaller overall equivalence ratio, may move the high temperature zone toward the sides. The reason to cause this shifting is because of low density and high diffusivity of hydrogen. Evidence can be seen in Fig. 7, where mass fractions of methane and hydrogen are shown for the case of fuel injection velocity of 60 m/s and hydrogen substitution percentage of 20%. When the blended fuels are injected radially into the combustor at the same injection velocity, methane is heavier and consumed in the front part of primary zone. However, the hydrogen is then flown easily to the sides and around the combustor liner, and it can even diffuses to the secondary zone unless the injection velocity is high enough so that hydrogen is driven by recirculation flows to the center of primary zone. Therefore, when the fuel injection velocity is lower or the hydrogen percentage is higher, the hydrogen flows along the sides of the combustor, where the residence time of air flow is shorter because of the boundary layer, resulting in a better mixing of hydrogen/air and consequently, a high temperature zone mainly due to hydrogen combustion. Although higher diffusivity of hydrogen compared to hydrocarbon fuels improves mixing, enhances turbulence and increases homogeneity, this indicates the modification of fuel injector and the injection strategy for hydrogen-methane hybrid fuels are needed.



Fig. 7. Mass fraction of methane (top) and hydrogen (bottom) on the centerline planes for fuel inlet velocity of 60 m/s, 20% hydrogen substitution.

Hydrogen Substitution Effects on the Performance

The effects of hydrogen substitution on the flame temperature including maximum temperature, average temperature in the primary zone and exit temperature of the combustor are analyzed in this section. These parameters are related to the combustion performance for gas turbine engines. The maximum flame temperatures and average temperature in the primary zone for various hydrogen substitution percentages and fuel inlet velocities are shown in Fig. 8. A variable fraction of hydrogen in the blended fuel, between 0 and 100% in volume, were injected at the fuel inlet velocities ranging from 40 m/s to 100 m/s. Results show that, with only 10% hydrogen substitution, the maximum flame temperature increases approximately 400 K compared to that of pure methane at the same inlet velocity. This demonstrates the intrinsic differences in flame temperature between hydrogen and methane. However, the maximum flame temperatures do not vary too much with hydrogen substitution percentages in the fuels, except at rather high hydrogen percentage above 90% and pure hydrogen case.



Fig. 8. Hydrogen substitution effects on maximum and average flame temperature at different fuel inlet velocities.

Apparently, due to high reactivity of hydrogen, the maximum flame temperature may be increased by substituting hydrogen for methane in the blended fuel. However, at the same fuel injection velocity, the mass flow rate of the blended fuel is less when the hydrogen substitution percentage is higher. Consequently, the combustion heat release and average temperature are decreased so that the engine may suffer a power loss. This can be demonstrated in Fig. 8, where the average temperatures in the primary zone are also shown. The average temperatures increase when a small amount of hydrogen is substituted. But when the hydrogen substitution percentages are too high, the average temperatures begin to drop quickly, indicating a reduction of high temperature region in the primary zone and a decrease of heat release, although the maximum flame temperatures do not vary accordingly. Also, the influence of hydrogen percentage on the average flame temperature in the primary zone exhibits a non-monotonic behavior with a peak at certain hydrogen percentage, and the percentage is higher for higher fuel injection velocity.

While higher fuel inlet velocity increases the overall equivalence ratio, higher hydrogen substitution percentage then decreases it. Figure 9 then re-plots the maximum and average flame temperature in terms of overall equivalence ratio, ranging from 0.6 down to 0.05. For original designs using methane, the overall equivalence ratio is about 0.35. By substituting hydrogen for methane or using hydrogen-methane blended fuel, stable combustion and reasonable temperature distribution for gas turbine can be obtained for the equivalence ratio as low as 0.15, a rather fuel-lean condition. This could help to reduce CO and NO_x emissions.



Fig. 9. Maximum and average flame temperature at different equivalence ratios.

In order to achieve a satisfactory and consistent distribution of temperature in the efflux gases discharging into the turbines, the most important parameters are combustor exit temperature (T_{4avg}) and pattern factor (PF). T_{4avg} represents the turbine inlet temperature, which is calculated by the mass-flow-weighted mean of all the exit temperatures here. Limited by the material properties of the turbine, higher T_{4avg} indicates higher power output of the turbine. On the other hand, too high a temperature or too severe a temperature fluctuation at the exit is a problem for turbine. Therefore, the pattern factor highlights the maximum exit temperature and is normally defined as $(T_{4max} - T_{4avg})/(T_{4avg} - T_{3avg})$ to indicate the temperature fluctuation or the possible hot spots at exit.

The effects of hydrogen substitution on the exit temperature and the pattern factor for fuel inlet velocity of 60 m/s are shown in Fig. 10. Compared to the case with pure methane (0% hydrogen percentage), the exit temperature increases near 600 K with only 10% hydrogen substitution. Such a high exit temperature is caused by the combination of high reactivity and high diffusivity of hydrogen and the combustion gaseous are pushed further downstream near the exit. The combustor exit temperatures then decreases with increasing amount of hydrogen substitution in fuel, indicating a decrease of power output, especially for the case of pure hydrogen. Although the exit temperature may be too high for moderate hydrogen substitution percentage, the pattern factor is continually decreased from 0.64 for pure methane to below 0.2 for methane-hydrogen blended fuels, except for the pure hydrogen case, where the pattern factor rises drastically. The temperature distributions at combustor exit for the cases without hydrogen and with 20% hydrogen substitution is shown in Fig. 11. Six hot spots are clearly found at several places near the outside circumference, which is close to the target exit temperature of 1200 K, but the temperature fluctuation is evident. This demonstrates the serious problem of cooling for current combustor design with six large dilution holes. However, with 20% hydrogen substitution, the temperature distribution becomes more uniform and the pattern factor decreases, but the exit temperature is too high for a combustor in satisfaction. The cooling strategy certainly needs to be reconsidered for moderate hydrogen substitution percentage. On the other hand, at higher hydrogen substitution percentages (around 80%, 90%), the exit temperature and pattern factor seems to be promising, but it would be suitable for the engines with smaller power output.



Fig. 10. Combustor exit temperature and pattern factor for various hydrogen percentage (fuel inlet velocity of 60 m/s).



Fig. 11. The exit temperature distribution for fuel without hydrogen (top) and with 20% hydrogen (bottom).

Figure 12 shows the exit temperature and pattern factor for different fuel inlet velocities with the fuels of pure methane, 20% hydrogen and 80% hydrogen substitutions. For pure methane, the exit temperature increases with increasing fuel inlet velocity drastically. With 20% hydrogen fuel, the trends of exit temperature with fuel inlet velocity are similar. But when the hydrogen substitution percentage is up to 80%, the influence of fuel inlet velocity on the exit temperature becomes mild. As for the pattern factor, with hydrogen substitution, the pattern factor decreases for all the cases of fuel inlet velocities computed. A non-monotonic behavior is also found for the trend of pattern factor with fuel inlet velocity for the cases of pure methane and lower hydrogen substitution percentage.



Fig. 12. Combustor exit temperature and pattern factor for various fuel inlet velocity.

Emissions

Figures 13 and 14 illustrate the emission index of CO and NO_x for the fuel inlet velocities of 20 m/s to 80 m/s, respectively. The emission index is defined as the amount of emission species in gram per unit kilogram of fuel burned, and it is calculated as $\dot{m}_K / \dot{m}_{fuel}$, where \dot{m}_K and \dot{m}_{fuel} are the mass flow rate of emission species K and mass flow rate of the blended fuel, respectively.

Figure 13 indicates that basically, CO emission index increases with increasing hydrogen substitution percentage and it decreases with increasing fuel injection velocity. The main factors to increase the CO emission are the hydrogen substitution and the chemical kinetics involving methane/hydrogen/air. The significance of chemical kinetics increases with the fraction of hydrogen, and the competition for air between hydrogen and methane favors the hydrogen-air reactions. Therefore, CO emission increases due to the lack of oxygen for methane combustion. However, it is expected at 100% hydrogen, the CO emission drops to zero. Taking the case at 80 m/s injection velocity as an example, Fig. 13 shows the emission index of CO increases continually up to near 99%, but on the other hand, mole fraction of CO begins to drop after 90%. It is because the definition of emission index is based on per unit mass of fuel burned. When hydrogen substitution is over 90%, CO mole fraction is less, but CO emission index still increases due to a significantly reduced fuel mass at high oxygen percentage. However, with a small amount of hydrogen

substitution at lower fuel injection velocities, CO emission decreases, such as the cases with less than 10% hydrogen under 40 m/s inlet velocity. It is because the mixture of methane/air may become flammable again in the region of the primary zone, when hydrogen is present to serve as an additional fuel [35].



Fig. 13. Effect of hydrogen substitution percentage on CO emissions (fuel inlet velocity of 20 m/s to 80 m/s).

Figure 14 shows NO_x emission index (also NO_x mole fraction for the case of 80 m/s injection velocity) continually decreases with increasing hydrogen substitution percentage and fuel inlet velocity, which is basically consistent with the trend of average flame temperature in the combustor (shown in Fig. 8 previously). At very high hydrogen percentage and low inlet velocity, the NO_x emissions are relatively small because of low flame temperature at rather lean combustion. As the fuel injection velocity lowers to 20 m/s, results in Fig. 14 shows NO_x emission first increases, and then decreases with the increase in the fraction of hydrogen. With a small amount of hydrogen substitution (less than 10% in the figure), NO_x emission increases due to high reactivity and the promotion of methane/air combustion by hydrogen. It happens when the mixing time between fuel and air becomes longer in the leaner primary zone at lower fuel injection velocity, the same reason for CO emission to be reduced. This non-monotonic behavior was also found in [4] for the flames closer to extinction limits in ultra-lean CH₄/air combustion with hydrogen addition.



Fig. 14. Effect of hydrogen substitution percentage on NO_x emissions (fuel inlet velocity of 20 m/s to 80 m/s).

The hydrogen substitution effects on the emission characteristics are quite different from the hydrogen addition effects, which demonstrate an increase of NO_x formation and the reduction of CO emission for hydrogen addition [10]. The NO_x emission decreases with increasing hydrogen substitution percentage mainly due to less mass flow rate of blended fuel and much leaner combustion. One thing has to be noted that, by comparisons between Fig. 13 and Fig. 14, the trends of NO_x and CO emission with fuel inlet velocity (or fuel mass flow rate or equivalence ratio) are opposite. To reduce NO_x emission by decreasing injection velocity could raise the CO emission. It has been estimated that an increment on CO emission index means a decrease of combustion efficiency. Therefore, a balance between combustion efficiency and NO_x emission requires further evaluations when hydrogen is substituted for methane.

CONCLUSIONS

The effects of hydrogen substitution on combustion performance of a micro gas turbine were studied numerically. The simulations were conducted with various fraction of hydrogen substitution from 0% to 100% and fuel injection velocities between 20 m/s to 100 m/s. Flame structures were compared and the combustor performance were analyzed to understand the potential of partial hydrogen substitution and full hydrogen application for the innovative micro gas turbine.

The results show the flame temperature increase when a small amount of methane is replaced by hydrogen, which is conducive to the combustion efficiency of the engine. However, further increasing the hydrogen substitution percentage then decreases the equivalence ratio, consequently the flame temperature. Higher temperature benefits the performance of gas turbine, but the cooling and the NO_x emissions are the primary concerns. To reduce the level of NO_x emission and the exit temperature, a viable choice is to increase the hydrogen percentage or decrease the fuel injection velocity, which produces a leaner combustion zone. But substantial increase of CO emission indicates a decrease in combustion efficiency. Although further improved designs and experimental testing are still needed to employ the blended fuels, the model simulation paves an important step to understand the combustion characteristics for the hydrogen fueled micro gas turbine.

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