Flexible Fuel Operation of a Dry-Low-NO_x Micromix Combustor with Variable Hydrogen Methane Mixtures
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ABSTRACT

The role of hydrogen (H₂) as a carbon-free energy carrier is discussed since decades for reducing greenhouse gas emissions. As bridge technology towards a hydrogen-based energy supply, fuel mixtures of natural gas or methane (CH₄) and hydrogen are possible.

The paper presents the first test results of a low-emission Micromix combustor designed for flexible-fuel operation with variable H₂/CH₄ mixtures. The numerical and experimental approach for considering variable fuel mixtures instead of recently investigated pure hydrogen is described.

In the experimental studies, a first generation FuelFlex Micromix combustor geometry is tested at atmospheric pressure at gas turbine operating conditions corresponding to part- and full-load. The H₂/CH₄ fuel mixture composition is varied between 57 and 100 vol.% hydrogen content.

Despite the challenges flexible-fuel operation poses onto the design of a combustion system, the evaluated FuelFlex Micromix prototype shows a significant low NOₓ performance and high combustion efficiency over a wide fuel range.

NOMENCLATURE

c [m/s] velocity

dₙ [mm] nozzle diameter

g [-] mass fraction

J [-] momentum flux ratio

J_m [%] relative momentum flux ratio

LHV [MJ/kg] lower heating value

m [kg/s] mass flow

P₆th [W] thermal power

q_rel [W/m³] volumetric heat release rate

r [-] volume fraction

SAR [kg₆th/kg₆fuel] stoichiometric air requirement

Tₘ [K] combustor air inlet temperature

Tₑ [K] exhaust gas temperature

Wᵣ [MJ/m³] Wobbe index

y, y_crit [mm] (critical) injection depth

y⁺ [-] normalized wall distance

Φ [-] equivalence ratio

Φₙ [-] power-normalized equivalence ratio

η [-] combustion efficiency

ρ [kg/m³] density

ψ [-] mole fraction

INTRODUCTION

To remedy the adverse effects of fossil fuel combustion on the earth’s climate, technologies for a sustainable and low-emission energy provision have to be developed and supported. Gas turbine systems fueled with hydrogen represent a carbon dioxide-free alternative to conventional power generating facilities if the fuel is produced with excess energy from renewable energy sources by power-to-gas applications.

As bridge technology towards a hydrogen-based energy supply, the admixture of hydrogen into natural gas for combustion in gas turbines is possible [1]. With an increase of hydrogen in the fuel mixture, the emission of carbon dioxide (CO₂) is reduced (cf. Figure 1). The figure shows the CO₂ reduction potential that high-hydrogen combustion offers if the fuel composition of a reference combustor is changed at constant thermal power output. It shows that high-hydrogen combustion is a very effective way of eliminating CO₂ emissions of gas turbine systems.

Figure 1: Relative CO₂ emissions of variable hydrogen methane fuel mixtures at constant thermal power output

In addition to the apparent CO₂ reduction potential, the admixture of hydrogen enables leaner combustion due to the higher reactivity and flame stability of hydrogen. This improves gas turbine turndown capabilities, required in times of peak load of renewable energies [2]. When changing from natural gas to
hydrogen combustion at constant thermal power output, the combustion temperature of hydrogen-rich fuel mixtures decreases (cf. Table 1). When constant thermal power output is defined as boundary condition for a combustor design, this fuel characteristic benefits the lifetime of combustor and especially turbine parts. When constant turbine inlet temperature is the design goal, higher thermal power output is generated by a hydrogen-fueled gas turbine combustor in comparison to a natural gas alternative.

Despite reducing CO₂ emissions significantly, high hydrogen combustion promotes the formation of nitrogen oxides (NOₓ). This is due to hydrogen’s high reactivity, leading to high heat release rates and peak temperatures. In combination with the temperature dependence of NOₓ formation via the thermal NO route [3, 4]. The significant low NOₓ performance of premixed combustion systems is counteracted by the characteristic danger of flashback that is increased when highly reactive hydrogen is admixed to natural gas fuel.

For achieving low NOₓ performance with the inherent safety against flashbacks, Aachen University of Applied Sciences (AcUAS) investigates gas turbine combustion of hydrogen and hydrogen-rich fuels since the European research projects EQHHPP [5] and CRYOPLANE [6]. In these research projects, the Dry Low NOₓ (DLN) Micromix (MMX) combustion principle has been developed initially, and continuously investigated and improved by several follow-up projects since then [7].

The DLN MMX principle reduces the retention time of NOₓ precursors in high temperature regions by miniaturization of the individual flames to sizes of 10-40mm in length. According to Lefebvre [8], NOₓ production is a function of retention time, reaction rate and mixing rate. Thus, reduction of the residence time of NOₓ precursors and intense mixing of fuel and oxidizer results in reduced emissions. Despite being a non-premixed combustion system, intense mixing of fuel and oxidizer directly after fuel injection is achieved by jet-in-crossflow-mixing (JICF).

Figure 2: Sketch of a jet-in-crossflow, with primary (blue) and secondary jet (yellow)

In this mixing process, the fuel gas jet (secondary) is injected transversely into the stream of air (primary) at an injection angle of 90° (cf. Figure 2). The secondary hydrogen jet is deflected and entrained by the air in the primary flow channel due to the pressure gradient between the upstream and downstream side of the injected fuel jet. Downstream of the injection point a characteristic counter rotating vortex pair is shaped that deforms the fuel jet and accelerates mixing between fuel and oxidizer. Directly after fuel injection into the crossflow of air, combustion occurs in miniaturized, diffusion-like flames with an inherent safety against flashbacks.

Figure 3: Geometry of a typical Micromix combustor [9]

In Figure 3, the geometry of a Micromix test burner mounted in an atmospheric test rig is presented. Via fuel supply segments the fuel is distributed before it is injected through multiple small nozzles into a crossflow of air. The recirculation zones that are formed in the wake region of the fuel supply segments and the air guiding panels (cf. Figure 4) facilitate aerodynamic flame stabilization and prevent adjacent flames from merging. Flame merging is to be avoided since it increases the flame expansion and thus the residence time of NOₓ precursors in the hot reaction zone, causing an increased NOₓ formation.

Figure 4: Schematics of the Micromix combustor geometry, detailing the recirculation zones and aerodynamic flame stabilization and the jet-in-crossflow mixing process [10]

Based on previous research work targeting combustion of pure hydrogen [11, 12] and hydrogen-rich syngas [13], the current development aims at flexible-fuel operation with various hydrogen methane mixtures over a wide range of gas turbine operating conditions at a constant combustor geometry. The paper presents first test results of an innovative FuelFlex Micromix combustor operated under gas turbine full-load and part-load conditions with variable hydrogen methane fuel mixtures. Initial exhaust gas measurements from atmospheric combustion chamber testing are presented along with results of combustion and flow simulations for hydrogen contents between 57 vol.% and 100 vol.% in the fuel mixture.

DESIGN CONSIDERATIONS

The Micromix combustion principle is designed for application in industrial-scale gas turbines. For validation of the combustion characteristics under high-pressure operation, the auxiliary power unit Honeywell/Garrett GTCP 36-300 is used (cf. Figure 5). This small aviation gas turbine is operated as experimental test rig at Aachen University of Applied Sciences for investigating the feasibility of alternative fuels such as hydrogen, hydrogen-rich synthesis gases and methane in gas turbine engines and their impact on engine control strategy. The GTCP 36-300 is a
constant-speed single spool gas turbine engine with a single-stage radial compressor and a single-stage radial turbine. It requires approx. 1.6 MW thermal energy converted to shaft power for producing electrical and pneumatic power up to 370 kW. Electrical power is provided by an auxiliary generator, pneumatic power by an additional single-stage radial load compressor.

Figure 5: APU GTCP 36-300 mounted in engine test rig [14]

The challenges in the flexible-fuel adaption of a gas turbine combustion system are the combustion characteristics of the applied fuel mixtures that change significantly over the investigated mixture range. Table 1 summarizes the fuel characteristics that are most important for the combustor design process for a fuel range between 0 and 100 vol.% of hydrogen in the fuel mixture. The stated design point $\Phi_{\text{design}}$ is based on operating conditions of the APU gas turbine Garrett GTCP 36-300 with pure hydrogen fuel.

Table 1: Summary of general gas composition and properties

<table>
<thead>
<tr>
<th>$\text{CH}_4$</th>
<th>[-]</th>
<th>1</th>
<th>0.9</th>
<th>0.8</th>
<th>0.57</th>
<th>0</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\text{LHV}$</td>
<td>[MJ/$\text{kg}$]</td>
<td>119.6</td>
<td>86.9</td>
<td>73.2</td>
<td>59.9</td>
<td>49.9</td>
</tr>
<tr>
<td>$\text{SAR}$</td>
<td>[-]</td>
<td>34.3</td>
<td>26.3</td>
<td>22.9</td>
<td>19.7</td>
<td>17.2</td>
</tr>
<tr>
<td>$\rho$</td>
<td>[kg/$\text{m}^3$]</td>
<td>0.082</td>
<td>0.140</td>
<td>0.197</td>
<td>0.329</td>
<td>0.656</td>
</tr>
<tr>
<td>$\text{Wi}$</td>
<td>[MJ/$\text{kg}$]</td>
<td>37.4</td>
<td>35.3</td>
<td>35.4</td>
<td>37.4</td>
<td>44.0</td>
</tr>
<tr>
<td>$\Phi_{\text{design at }}$</td>
<td>[-]</td>
<td>0.375</td>
<td>0.395</td>
<td>0.410</td>
<td>0.429</td>
<td>0.450</td>
</tr>
<tr>
<td>$T_{\text{ad at }}$</td>
<td>[K]</td>
<td>1563</td>
<td>1568</td>
<td>1572</td>
<td>1576</td>
<td>1580</td>
</tr>
<tr>
<td>$J_{\text{rel}}$</td>
<td>[%]</td>
<td>100</td>
<td>111.7</td>
<td>111.5</td>
<td>100</td>
<td>72</td>
</tr>
</tbody>
</table>

Changing the mixture composition leads to a change of the lower heating value (LHV) and the stoichiometric air requirement (SAR). This, in turn, results in a shift of the design equivalence ratio $\Phi_{\text{design}}$, if constant thermal power output is applied as boundary condition. Adding methane to the fuel mixture, shifts the design equivalence ratio towards fuel-rich conditions, whereas high hydrogen contents enable leaner combustion, which also benefits gas turbine turnaround capabilities.

For comparing gas turbine operating points with different fuel compositions, the power-normalized equivalence ratio $\Phi_n$ with reference to pure hydrogen is introduced. It is defined according to Eq. (1).

$$\Phi_n = \Phi_{\text{mix}} \times \frac{\text{SAR}_{\text{LHV}}}{\text{LHV}_{\text{H}_2}} \times \frac{\text{LHV}_{\text{mix}}}{\text{SAR}_{\text{H}_2}}$$ (1)

The dependence is derived by the requirement of constant thermal power output between a combustor fuelled with a specific mixture of hydrogen and methane (index “mix” in Eq. (1)) and the same combustor fuelled with pure hydrogen at constant air mass flow. For one gas turbine load condition the normalized equivalence ratio is constant for all mixture compositions. All fuel compositions yield the same thermal power output as hydrogen combustion at a given equivalence ratio $\Phi$, if for these mixtures the normalized equivalence ratio $\Phi_n$ is set to the hydrogen value.

In the fuel range between 100 vol.% and 57 vol.% of hydrogen in the mixture, the Wobbe index stays nearly constant with deviations less than 6%, suggesting good interchangeability between the applied fuel mixtures. Towards methane rich mixtures, the change in density, stoichiometric air requirement and lower heating value leads to a significant increase in the Wobbe index. This disproportionately lowers the flow velocity through the multiple fuel nozzles when constant thermal power output and a constant geometry are applied as boundary conditions. The lowered fuel velocity affects the jet-in-crossflow mixing of fuel and air, which is characterized by the injection depth $y$. According to Eqs. (2) and (3), $y$ is controlled by the nozzle diameter, the fuel and air velocities, and their respective densities. These quantities are determined by the combustor geometry and the boundary conditions. The density and velocity ratios are summarized in the momentum flux ratio $J$.

$$y \propto d_i \cdot \sqrt{J}$$ (2)

$$J = \frac{\rho_{\text{fuel}} \cdot c^2_{\text{fuel}}}{\rho_{\text{air}} \cdot c^2_{\text{air}}}$$ (3)

At sufficiently low injection depth of the fuel jet into the air crossflow, the fuel-air-mixture discharges freely into the combustion zone. The residence time of NOx precursors is low, resulting in low NOx emissions of the combustor. At a critical injection depth $y_{\text{crit}}$, the fuel jet penetrates the shear layer and enters the inner recirculation vortex (cf. Figure 4). The fuel-air-mixture that is formed in the vortex ignites and leads to hot gas recirculation and vertical flame merging with extended retention times for NOx precursors at elevated temperatures, resulting in increased NOx emissions. In contrast, insufficient injection depth reduces the fuel-air-mixture quality and ultimately, part-load stability.

The momentum flux ratio $J$ changes as a function of the fuel mixture composition for a given set of geometrical and operational boundary conditions. The relative momentum flux ratios with reference to pure hydrogen fuel ($J_{\text{rel}}$) are given in Table 1. The highest values of $J_{\text{rel}}$ are present for fuel mixtures between 80 and 90 vol.% H2, implying that these mixtures are most critical for hot gas recirculation as a consequence of the mixture-dependent increased injection depth. Apart from its low LHV and lower reactivity in comparison to hydrogen-rich fuels, pure methane offers the lowest momentum flux ratio. In consequence, part-load stability issues may arise due to a reduced fuel-air mixture quality caused by insufficient injection depth.

**EXPERIMENTAL APPROACH**

In the following, the experimental approach is briefly summarized. A detailed description including an analysis of all major error sources and their consequences on measurement accuracy is presented in [15].

* Adiabatic flame temperature calculated for an ideal combustion with complete fuel conversion at $\Phi_{\text{design}}$ with $T_{\text{ad}} = 298.15K$ and $T_{\text{env}} = 559K$. 
The atmospheric combustion chamber test stand for evaluating the combustion characteristics of the Micromix principle under flexible-fuel operation is displayed in Figure 6. The combustor module (test burner) is mounted on the test burner flange and integrated into the atmospheric test rig.

The test stand provides ambient air as oxidizer via two radial compressors. The air is preheated by an electric heater to \( T_3 = 559 \text{K} \). The fuel mixture is prepared in a gas mixing facility that continuously controls the component mass flows of methane and hydrogen, mixes both streams in a static mixer and supplies it to the test stand at room temperature. The error analysis carried out in [15] yields a mixing accuracy between ±0.6 vol.% and ±1.15 vol.% of \( \text{H}_2 \) or \( \text{CH}_4 \) in an \( \text{H}_2/\text{CH}_4 \) mixture. The equivalence ratio can be determined with a relative accuracy ranging between ±2.9% and ±4.5%.

![Figure 6: Schematics of the atmospheric test rig [14]](image)

For validation of the combustion characteristics, exhaust gas measurements are performed. Samples are extracted by a heated probe and supplied to the analysis modules of the continuous gas analysis system ABB Advanced Optima AO2020.

![Figure 7: Illustration of the applied measurement grid [15]](image)

For measuring the exhaust gas composition, an active heated probe is located behind the combustor outlet and positioned by actuators along three axes (\( x, y, z \) in Figure 7). On a defined measurement grid, exhaust gas samples are extracted when steady state conditions are given. For each combustor operating point (equivalence ratio and fuel mixture composition) the arithmetic mean of measurements at 10 individual locations is obtained.

The exhaust gas samples are supplied to the continuous gas analysis system ABB Advanced Optima AO2020 via heated tubing. The analysis modules determine the exhaust gas composition in terms of unburned \( \text{H}_2 \) (ABB Caldos 27) and unburned hydrocarbons (UHC) (ABB MultifID4), oxygen (\( \text{O}_2 \)) (ABB Magnos 206) CO and \( \text{CO}_2 \) (ABB Uras 26). For the determination of \( \text{NO} \), (i.e. \( \text{NO} \) and \( \text{NO}_2 \)), an Eco Physics CLD 700 EL is used and directly connected to the hot exhaust gas sample.

To avoid zero or sensitivity drifts affecting the measurement accuracy, all exhaust gas analysing devices are calibrated using zero-point and reference-point calibration gases, before each test campaign.

NUMERICAL APPROACH

The experimental investigations are accompanied by combustion and flow simulations that facilitate a phenomenological interpretation of the experimental results.

The numerical analyses are carried out using the commercial CFD code STAR-CCM+. The 3D combustion simulations base on a fully symmetrical slice model that features 2 mass flow inlets for fuel and air and a pressure outlet (cf. Figure 8). The model offers a fluid region for the combustion and flow simulation and solid regions that account for conjugate heat transfer through the combustor walls. For the spatial discretization, adaptive mesh refinement of a polyhedral mesh is applied. For the numerical test-burner investigation, the design point of the combustor (\( T_0 = 0.375 \)) and off-design points at part-load conditions are analyzed. The equivalence ratios are set at constant air mass flow by adjusting the fuel mass flow accordingly.

The reactive flow regime is solved by a three-dimensional, steady, pressure-based RANS solver using the realizable \( k-\varepsilon \) turbulence model with a universally applicable "all \( y^+ \) wall treatment" approach. For high \( y^+ \) values, characterized by a low mesh resolution near walls, the approach applies a wall function model for resolving the boundary layer. For low \( y^+ \) values, characterized by a fine grid near walls, no wall functions are applied, and the boundary layer is resolved explicitly.

![Figure 8: Computational domain and coordinate system of the derived slice model [12]](image)

To account for flexible fuel operation, the numerical approach is expanded towards modelling of hydrocarbon combustion. A numerical preliminary study has been performed on the Sandia / Sydney bluff-body stabilized flame HM1 fueled with a mixture of 50 vol.% hydrogen and 50 vol.% methane [16]. In this study, the performance of 7 detailed reaction mechanisms ranging from a skeletal mechanism with 16 species to a comprehensive mechanism with 118 species has been evaluated. The study identified the 22 species DRM22 reaction mechanism [17] as best compromise between modelling accuracy and calculation time. The chosen combustion mechanism reduces the calculation time by 80% compared to the well-established GRI 3.0 mechanism while achieving comparable accuracy. This mechanism is now applied in the Micromix combustion simulations.

Turbulence-chemistry interactions are treated by the Eddy Dissipation Concept [18]. For evaluating NO emissions, a thermal NO model is used.

The numerical boundary conditions \( p_i, T_3, \Phi \) and fuel mixture composition correspond to the experimental test case. A detailed description of the numerical approach is presented in [13, 19].

RESULTS

In Figure 9, the combustion efficiency \( \eta \) is displayed for full- and part-load operation with 4 different hydrogen methane fuel mixtures. For determining \( \eta \), the thermal power lost by emission of CO and the unburned fuel components \( \text{H}_2 \) and \( \text{CH}_4 \) is put in relation to the potential thermal power introduced to the combustor by both fuel components.
For fuel compositions between 90 and 100 vol.% H₂ in the fuel, the combustion efficiencies exceed 0.995 for the design point at a normalized equivalence ratio φ₀ of 0.375 and both part-load conditions at φ₀ = 0.3125 and φ₀ = 0.25. When increasing the methane share in the fuel, the combustion efficiencies at part-load operation are reduced, as can be seen for the 80% H₂ fuel mixture at φ₀ = 0.25. For the 57% H₂ fuel mixture, combustion efficiencies are further reduced to 0.99 at the design point and 0.966 at lean off design.

\[
\eta = 1 - \frac{\sum_{i=1}^{m} (\dot{m}_{\text{fuel, tot}} \cdot g_i \cdot LHV_i)}{\sum_{i=1}^{n} \dot{m}_{\text{fuel, i, tot}} \cdot LHV_i}
\]  

For hydrogen injection, the increased turbulent kinetic energy in the jet-in-crossflow zone enhances mixing of fuel and oxidizer significantly. Further downstream in the main reaction zone, turbulence chemistry interactions affect the flame structure and the combustion progress. Hydrogen’s higher diffusivity and the generally increased turbulence level intensify the mixing of reactants in the flame. The resulting higher heat release rates lead to a faster and more homogeneous combustion progress with overall higher combustion efficiencies.

The change in overall reactivity is evaluated by simulating the local heat release rates at φ₀ = 0.3125 with pure hydrogen and a 57 vol.% hydrogen fuel mixture. The local distribution on a combustor cross-section is shown in Figure 11. The maximum heat release rate for hydrogen combustion is almost tripled in comparison to the 57 vol.% H₂ case, whereas the overall thermal power output remains nearly constant. Additionally, the heat release zone is much more confined.

Since heat release rate and local combustion temperature are interconnected, also the peak temperatures are reduced as methane is added to the hydrogen fuel (cf. Figure 13). With lower temperatures, all reaction processes within the flame occur at increased timescales, leading to expanded Micromix flames and increased emissions of unburned fuel components and CO. The presence of hydrocarbons, hydrogen, or carbon monoxide in the exhaust gas, reduces the combustion efficiency since their chemical energy is not released during the combustion process.}

![Figure 9: Experimental results of combustion efficiency at full- and part-load operation for variable fuel mixtures](image)

The reduced combustion efficiencies for methane-rich fuel mixtures are the result of reduced reactivity (lower heat release rate) and reduced mixing intensity between fuel and oxidizer for higher methane contents. Despite having the same Wobbe index (cf. Table 1), which suggests good interchangeability between 100 vol.% H₂ and the 57 vol.% H₂ fuel, the mixing intensity in the jet in crossflow region is altered. Due to the lower density of pure hydrogen fuel in comparison to methane-rich fuels, the injection velocities are significantly increased for hydrogen. Taking also into account the reduced viscosity as well as significantly increased diffusivity of pure hydrogen, higher turbulence levels upon injection are the consequence. This characteristic is illustrated by the turbulent kinetic energy levels displayed in Figure 10.

![Figure 10: Simulated turbulent kinetic energy in the jet-in-crossflow mixing zone for for rH₂=1 (left) and rH₂=0.57 (right)](image)

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![Figure 11: Simulated heat release on combustor cross-section at φ₀=0.3125 for rH₂=1 (top) and rH₂=0.57 (bottom)](image)
Enlargement of the flames for optimizing the combustion process at increased CH₄ shares is a conflicting design goal towards low NOₓ emissions with hydrogen-rich fuels.

![Flame images of Micromix flames at lean off-design operation with variable fuel mixtures](Image)

Figure 12: Flame images of Micromix flames at lean off-design operation with variable fuel mixtures

In Figure 13, the simulated temperature distribution of a Micromix combustor fuelled with 100 vol.% hydrogen and a 57 vol.% hydrogen fuel at a fixed operating point are presented. Despite the higher exhaust gas temperatures for the methane-rich case, the peak temperatures are reduced by 165 K. This difference in the temperature profile implies significant alterations in the NOₓ emission characteristic between hydrogen and methane-rich fuels.

![Simulated temperature on combustor cross-section](Image)

Figure 13: Simulated temperature on combustor cross-section at $\Phi_v=0.3125$ for $r_H=1$ (top) and $r_H=0.57$ (bottom)

In Figure 14, the obtained NOₓ emissions are depicted for full- and part-load operation. The Micromix FuelFlex prototype shows a significant low NOₓ performance with peak emission levels of 4 ppm at the design point $\Phi_v = 0.375$, despite the design challenge of multi-fuel operation.

![Experimental results of NOₓ emissions for variable fuel mixtures](Image)

Figure 14: Experimental results of NOₓ emissions for variable fuel mixtures, corrected to 15 vol.% O₂

CONCLUSION

An experimental combustor test campaign has been conducted, targeting flexible-fuel operation between 57 vol.% and 100 vol.% H₂ in a hydrogen/methane fuel mixture, at high combustion efficiency and low NOₓ emissions with a single combustor geometry.

In the presented studies a single FuelFlex Micromix combustor geometry has been tested at atmospheric pressure over a range of fuel compositions at part-load and full-load gas turbine conditions. Despite the design compromise, that takes into account the significantly different fuel and combustion properties of the applied gas mixtures, the initial results confirm promising operating behavior, combustion efficiency and pollutant emission levels for flexible-fuel operation. The investigated combustor module exceeds 99.5% combustion efficiency for hydrogen contents of 90-100 vol.% in the fuel mixture for the investigated operating range. Higher amounts of methane in the fuel increase the flame expansion as consequence of lower heat release rates. NOₓ emissions for increased methane content fuels are significantly reduced as local peak temperatures are decreased in comparison to pure hydrogen fuel combustion. The first generation Micromix FuelFlex combustor shows NOₓ emissions less than 4 ppm corrected to 15 vol.% O₂ at the design point and atmospheric conditions.

Future research will be directed towards exploration of the entire design space at varying gas turbine operating conditions and optimization of the pollutant emission level and combustion efficiency at high methane contents in the fuel.

ACKNOWLEDGEMENTS

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REFERENCES


