DEVELOPMENT OF AN EXPERIMENTAL
LPP GAS TURBINE COMBUSTOR

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ABSTRACT

The design of a generic experimental LPP-combustor (Lean Premixed Prevaporized) is described. The combustor was tested at inlet pressures between 3 and 7 bar and with an inlet temperature between 500 K and 700 K. Tests were run with Diesel No.2 and Jet A-1 fuels comparing emissions of CO and NOx. The tests showed low NOx and CO values for a range of conditions although difficulties of using the LPP-technique close to the final boiling point of the fuel were encountered. To increase the understanding of the flow situation inside the combustor, numerical simulations using a large eddy simulation code was also performed. The main interest in the calculations was the evaporation process of the liquid fuel.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>Ti</td>
<td>Combustor inlet temperature</td>
</tr>
<tr>
<td>P</td>
<td>Combustor pressure</td>
</tr>
<tr>
<td>Tf</td>
<td>Adiabatic equilibrium flame temperature in the zone upstream of the dilution holes</td>
</tr>
<tr>
<td>El</td>
<td>Emission Index (gram emission per kg of fuel)</td>
</tr>
<tr>
<td>T*</td>
<td>Temperature for which ElNOx and ElCO are equal</td>
</tr>
<tr>
<td>El*</td>
<td>Value of ElNOx and ElCO at T*</td>
</tr>
<tr>
<td>w*</td>
<td>Non-dimensionalized channel width</td>
</tr>
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</table>

INTRODUCTION

Low emission combustors are now being introduced in stationary gas turbines for power generation. Ultra low emissions of both NOx and CO are obtained for natural gas fuel with the lean burn technique, Willis et al. (1993), Leonard (1993), Aigner (1991) and Bauermeister et al. (1993).

For aircraft, marine and automotive engines liquid fuels have to be used. Therefore much research is now being focused on RQL- (Rich Burn Quick Quench Lean Burn), Catalytic- and LPP-combustors (Lean Premixed Prevaporized) for burning liquid fuels with low emissions. Examples of work that have been carried out for the three low emission techniques can be found in Zarzalis et al. (1992), Itch et al. (1993) and Cowell (1992).

To support the design process for LPP-combustors Volvo Flygmotor has developed a generic experimental LPP-combustor as a complement to the more complex engine design combustors. The combustor is used to generate data and design criteria on emissions, stability and wall temperature for different liquid fuels and geometry variations. The combustor is not designed to study variable geometry, or staged combustion as these problems depend on the engine application and the control system to be used. This paper concerns design issues, numerical simulations of the flowfield and initial tests of the combustor.

COMBUSTOR DESIGN

To achieve low NOx emissions in gas turbine combustors the flame temperature has to be kept at temperatures well below stoichiometric, but high enough to completely burn the hydrocarbon fuel. In an LPP combustor the flame temperature is kept low by burning a lean uniform air/fuel-mixture without fuel droplets. This is accomplished by vaporizing the fuel and mixing it with air in a separate zone before entering the combustor.

The process of fuel preparation needs a certain residence time to be completed. On the other hand, at high pressures and temperatures, the residence time must be minimized to avoid autoignition in the mixing zone. In addition, wakes or separated regions must be avoided in the mixing region as they may result in local autoignition or flashback into the vaporizing/mixing zone. At low power and idle conditions the vaporizer performance will be reduced because of low air temperatures close to the final boiling point for the fuel. Insufficient vaporizer performance will affect the emission characteristics of the combustor as will be described under test results.

The experimental LPP combustor used for the study presented in this paper is illustrated in Figure 1. The design concept consists of a cylindrical reverse flow can combustor, two counter-rotating radial swirlers, seven pressure swirl fuel atomizers, an annular vaporizing/mixing zone and a lip for secondary airflow fuel atomization. The diameter of the can is 120 mm.
**Can combustor**

The reverse flow circular can combustor is designed with one combustion zone so that equivalence ratios and flame temperatures easily can be evaluated. The combustor wall is convectively cooled and the combustor dome and centerbody are cooled with angled effusion holes.

To stabilize the combustion process a recirculating flowfield is set up in the combustor by strong swirling flows in combination with a sudden area change between the annular vaporizing/mixing zone and the combustor.

**Primary air swirler and annular vaporizing mixing zone**

The primary air swirler consists of 10 radial vanes giving a strong swirl in the annular vaporizing/mixing zone. At the exit of the swirler vanes, six Delavan pressure swirl atomizers are located. The size of the fuel injectors are chosen to create small droplets in the mixing zone for fuel pressures ranging from 2.5 to 20 bars.

The annular vaporizing/mixing zone with a strong swirling air flow creates high shear forces on the fuel droplets for quick vaporization.

A contraction that will reduce the boundary layer thickness is located at the outlet of the annular zone reducing the probability of the flame to propagate backwards into the mixing zone.

**Secondary air swirler and secondary fuel atomization**

Large droplets from the main fuel injector will be transported radially outwards because of the centrifugal acceleration induced by the rotating air in the vaporizing/mixing zone. These large droplets will hit the lip which separates the air from the primary and secondary swirler, and create a thin fuel film which slowly will be transported by the airflow to the edge of the lip. At the edge the thin fuel film will be atomized by the two rotating flows.

The secondary air swirler consists of 10 radial vanes and is counter rotating to the primary air swirler. In addition to secondary atomization the counter-rotating flows are used to induce streamwise vortices between the primary and secondary swirling flows. The axial vortices will improve the uniformity of the fuel/air-mixture in the annular vaporizing/mixing zone and increase the turbulence level in the flow entering the combustor. The fine scale turbulence will increase the flame speed in the first part of the combustor and help complete the combustion process before the jets from the dilution holes quenches the flame.

**Pilot fuel injector**

The pilot fuel injector is a Delavan pressure swirl atomizer which is used for smooth ignition of the combustor and for stabilizing the flame before reaching the correct test condition. At the test condition it will be turned off and purged with air before the gas analysis measurements are performed.

**Combustor flowsplit**

The combustor was designed to have an air flowsplit according to Table 1. The accuracy of the figures are within 5% of each value.

<table>
<thead>
<tr>
<th>Air passage</th>
<th>Flowsplit</th>
</tr>
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<tbody>
<tr>
<td>Primary swirl</td>
<td>34%</td>
</tr>
<tr>
<td>Secondary swirl</td>
<td>7%</td>
</tr>
<tr>
<td>Pilot fuel injector</td>
<td>2%</td>
</tr>
<tr>
<td>Effusion cooling</td>
<td>5%</td>
</tr>
<tr>
<td>Dilution holes</td>
<td>52%</td>
</tr>
</tbody>
</table>

**NUMERICAL SIMULATION**

**Method**

An in-house developed computer code which is based on the Large Eddy Simulation (LES) technique has been used for numerical studies of the flow field inside the combustor. The flow solution procedure is based on a cell-centred finite-volume scheme with explicit Runge-Kutta time-marching. The advantage with the LES technique is that large scale structures are captured by the use of a fine grid, however a turbulence model is still needed for the scales smaller than the grid cells.
The code has capabilities to handle reacting flows including evaporation of droplets. The combustion model is based on the EDC (Eddy Dissipation Concept) model by Magnussen (1985), which is a general concept for treating the interaction between turbulence and chemistry in flames. It is a concept which identifies a reactor to the fine structures and the reactor is treated as homogeneous, exchanging mass and energy with the surrounding fluid. A detailed description on how the EDC model was modified for the LES code is found in Rydén et al. (1993a), which also includes calculations on a premixed test case. Further applications of the code is given by Rydén et al. (1993b). The calculation results presented in this paper are from non-reacting cases, but include studies of evaporating liquid fuel sprays.

The mass transfer from an evaporating droplet is described by Fick's law, which relates the flux of droplet vapour into the gas phase to the gradient of the vapour concentration between the droplet surface and the surrounding gas. The droplet temperature is updated at every time-step by the use of a heat balance which relates the droplet temperature to the convective heat transfer at the droplet surface. The details of the implemented evaporation droplet model into the LES code is given by Olovsson (1993).

As the method is transient, new droplets have to be introduced at predetermined intervals. It is not possible to follow every physical droplet and instead so called numerical droplets, or parcels, are used where each parcel has to represent a number of real droplets. The number of real droplets each parcel will represent depends on the droplet size and its corresponding massflow. With large droplets each parcel only represents a few real droplets. In order to describe a spray, as in the calculations presented in this paper, several different droplet sizes are needed together with a mass flux distribution.

The computational grid used in the simulations is illustrated in Figure 2. Although the main interest was to study the vaporizing/mixing zone the can combustor was also included in the calculations in order to give realistic boundary conditions at the outlet of the vaporizing channel.

**Results**

Time averaged fuel mass fraction distribution in the vaporizing/mixing channel, with an inlet temperature of 700 K to the combustor and the pressure 7 bar, is given in Figure 3. The initial droplet sizes varied from 20 µm up to 60 µm with an SMD (Sauter Mean Diameter) of 40 µm. A radial gradient of the fuel fraction appears right after the main fuel injectors that is probably due to the fast evaporation of fuel which results in a bad penetration of droplets into the surrounding air. The effect of the secondary air to enhance the mixing is illustrated in the figure.

The development of the fuel/air mixing is illustrated in Figure 4 which gives the profiles of the calculated equivalence ratios, based on evaporated fuel, at the three locations marked in Figure 3. The curves represent arithmetic mean values of the studied 180 degrees sector and the equivalence ratios are given as function of a non-dimensionalized channel width, w*, at the present location. The change in profile in the relative short distance from II to III is due to the high turbulent region, created by the two counter rotating swirls that meet downstream of the lip. The mixing is then further enhanced by the sudden expansion in the can combustor leading to a well premixed system.
Although no tests were performed at higher inlet temperatures than 700 K a numerical simulation was performed at an inlet temperature of 900 K. Figure 5 gives profiles of equivalence ratios at the inlet temperatures 700 K and 900 K. As expected a higher inlet temperature will give faster evaporation and thus less penetration of fuel into the surrounding air. However, because of the highly turbulent region downstream the lip, the fuel/air distribution at the inlet to the combustor is fairly independent of the inlet temperature. Another situation may be found at low inlet temperatures were all fuel might not have evaporated before entering the combustor.

In the evaporation model a single temperature is used to define the droplet temperature when the evaporation process should start. As long as the temperature is higher than this value, mass transfer from liquid to gas will continue. In a real fuel, different fractions of the fuel will evaporate at different temperatures, but this effect has so far not been included in the model.

TEST SET-UP AND DATA REDUCTION

Preheated non-vitiated air enters the combustor test rig in a reverse flow direction after being measured with a sub-critical orifice with an accuracy of 3%. The combustor exhaust gases pass a water-cooled, stainless steel, gas analysis probe and a variable geometry back-pressure valve before exiting to the atmosphere. The combustor pressure and airflow are individually controlled with an upstream regulating valve and the downstream back-pressure valve. A small quartz window is used for visual monitoring of the flame inside the combustor.

The test rig instrumentation allows measurements of air flow, fuel flow, inlet air temperature, air pressure, combustor pressure drop and exhaust gas composition (CO2, CO, O2, NOx and UHC). The water-cooled gas analysis probe has three radially mounted tubes with a total of 10 holes. The gas analysis equipment consists of a chemiluminescence analyser for NOx, two non-dispersive infrared instruments for CO and CO2, a paramagnetic analyser for O2 and a flame ionization detector for total unburned hydrocarbons (UHC). The accuracy of a single species concentration was estimated to 5%.

The data were analysed to determine the combustor equivalence ratio in three independent ways, from O2 concentration, from CO2 concentration and from measured air and fuel flow. Only gas analysis measurements within 5% agreement of the equivalence ratio based on air and fuel flow were accepted. The results reported in this paper are based on CO2 concentrations.

To estimate the adiabatic equilibrium flame temperature, $T_f$, in the main combustion zone, upstream of the dilution holes, the equivalence ratio was estimated from the total air and fuel flow and the airflow split. The flame temperature was then calculated by a computer code developed by Gordon and McBride (1971). The heat loss from the flame were considered not to be significant due to the reverse flow geometry that makes the primary and secondary air absorb most of the heat from the combustor walls before entering the combustor. Due to the uncertainties in air flow split the accuracy of the flame temperature calculations was estimated to +/- 50 K.

TEST CONDITIONS

The combustor was run at a constant pressure drop of 3% over the range of conditions listed in table 2.

<table>
<thead>
<tr>
<th>Table 2. Combustor operating range</th>
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<tbody>
<tr>
<td>Compressor pressure</td>
</tr>
<tr>
<td>300 - 700 kPa</td>
</tr>
<tr>
<td>Inlet temperature</td>
</tr>
<tr>
<td>500 - 700 K</td>
</tr>
<tr>
<td>Air massflow</td>
</tr>
<tr>
<td>0.4 - 1.0 kg/s</td>
</tr>
</tbody>
</table>

Jet A-1 and Diesel No.2 fuel was used. Properties of the fuels are listed in table 3.
Figure 6  Emissions of NOx and CO at an inlet condition of 500 kPa, 700 K (Jet A-1).

TEST RESULTS

The combustor was run at nine different combinations of inlet conditions for each fuel. For each condition the combustor equivalence ratio was varied to change the flame temperature, \( T_f \), in the reacting zone upstream of the dilution zone.

Figure 6 shows how the emissions of NOx and CO vary with the estimated flame temperature in the combustor. High flame temperatures produce high NOx emissions and low flame temperatures result in incomplete combustion of CO. At 1890 K the Emission Index values of NOx and CO are around 1 (EI=1 would correspond to approximately 13 ppm NOx and 22 ppm CO at 15 % O2). A conventional diffusion type combustor would yield approximately ten times as much NOx at the corresponding condition and equal CO levels. An increase in the CO-level due to dissociation of CO2 can be noted at temperatures above 2050 K.

To make simple comparisons between different inlet conditions possible, \( T^* \) and \( EI^* \) are defined. \( T^* \) defines, for a specific condition, the estimated adiabatic flame temperature for which the same amount of NOx and CO is produced per unit mass of fuel. \( EI^* \) defines the level of emission production at \( T^* \). \( EI^* \) and \( T^* \) for an inlet condition of 500 kPa, 700K can be seen in figure 6. \( T^* \) can be interpreted as an optimum design temperature for a fixed inlet condition. But, this design temperature will only become relevant if the combustor design needs to meet both low NOx and CO emissions. The definition is for the moment relevant in consideration of current CO and NOx levels that have to be met for stationary gas turbines, see for instance Bauermeister et al. (1993).

The effect of different inlet temperatures on NOx production can be seen in figure 7. The diagram shows results from a test where the combustor pressure was held constant while the inlet temperature and fuel flow were varied. The trend from the test is in good agreement with correlations that states that the NOx-formation would increase exponentially with flame temperature at constant pressure and residence time, Odgers (1988).

Figure 8 shows the corresponding CO-levels to the NOx emissions presented in figure 7. Note how the region with low CO emissions stretches towards lower flame temperatures as the inlet temperature is increased. An increase in inlet temperature will widen the low NOx - low CO window thus increasing the control range of the combustor.

Figure 9a and 9b show how \( T^* \) varies with pressure and inlet temperature for Jet A-1 and Diesel No.2 fuels. \( T^* \) is reduced as the inlet temperature and combustor pressure are increased. This is mainly due to a shift in the CO-curves as can be expected from figure 8 and general theories for oxidation of hydrocarbon fuels. The difference between the two fuels is most pronounced at 500 K inlet temperature where incomplete vaporization makes the Diesel No.2 burn partly in a premixed prevaporized mode and partly as a spray flame. The shift in burning mode was noticed as a change from a light blue flame to a transparent yellow flame. The fact that \( T^* \) is lower for Diesel No.2 than for Jet A-1 for 500 K inlet temperature may be explained by a reduced CO level due to a higher flame propagation in the fuel/air mixture containing more and larger droplets. The effect of faster flame propagation can be explained with the fact that the chemical reactions takes place close to the droplets where the equivalence ratio is near unity. Experiments concerning the effect of flame propagation in liquid aerosols can be found in Burgoyne and Cohen (1954). A comparison of \( T^* \) at 3 Bar and 600 to 700 K shows the difficulty of burning diesel
compared to kerosene. These conclusions may be affected by different shapes of the equivalence ratio profiles at the outlet of the vaporizing channel due to differences in evaporation behaviour and droplet sizes.

Figure 10 shows the corresponding EI* values to the T* values in figure 9. The EI* values are low for inlet temperatures of 600 and 700 K indicating that the LPP-principle works for these inlet conditions. At the 500 K inlet temperature the EI* values are higher depending on both incomplete vaporization and the need of a higher flame temperature to completely oxidize the fuel. A comparison of T* and EI* for Jet A-1 and Diesel No.2 at 500 K indicates that the diesel flame partly burns at higher temperatures than the mean flame temperature thus generating more NOx. The difference between Diesel No.2 and Jet A-1 combustion should be most pronounced in the low temperature regime where the combustor behaviour is more dependent on the evaporation process. Although evaporation does occur below the initial boiling point a comparison of distillation curves at 500 K reveals a big difference between the fuels. According to the fuel properties in Table 3 approximately 70% of Jet A-1 while only 25% of Diesel No.2 was recovered at 500 K.

CONCLUSIONS
The numerical simulation does not predict a perfectly uniform equivalence ratio profile at the entry to the combustor. This indicates that the NOx-levels probably can be reduced with an optimised vaporizing mixing channel. The evaporation model used in this study can not be used to study effects of different fuel properties. The next step in code development would be to incorporate a better description of the distillation curve.

The LPP combustor described in this paper has proven that the LPP technique is capable of reducing NOx emissions by more than 90% compared to a standard diffusion type combustor. The tests have also shown the problems associated with using an LPP combustor at low inlet temperatures and the difference in burning Jet A-1 and Diesel No.2. Flashback or autoignition have not occurred in the vaporizing/mixing zone during the tests. However, further tests at higher temperatures and pressures are needed to exploit the limit where flashback or autoignition occurs.

As can be concluded from the tests the low emission window is not wide enough to make this combustor work in a low emission gas turbine engine. A practical variable combustor geometry or a staged combustion system must be developed before it can cope with wide changes in engine power setting.
Another way to further reduce the NOx emissions, not discussed previously in the paper, would be to increase the combustor volume thus increasing the residence time. An increased residence time would allow the CO to be completely oxidized for a lower flame temperature and consequently reducing the NOx production rate. An example of a large volume combustor yielding below 10 ppm NOx at 15% O2 (\(E_{\text{NOx}} < 0.75\)) can be found in Bauermeister et al. (1993).

Considering the tests described in this paper, performed up to 7 bars, it seems likely that it is easier to achieve low emissions of NOx with a high pressure ratio simple cycle gas turbine, as long as autoignition and flashback can be avoided. The improved CO oxidation at higher pressure makes combustion possible at lower flame temperature. This counteracts the questionable increase in NOx-formation at higher pressure. For different views on pressure dependence of NOx formation see, for instance, Odgers (1988), Leonard (1993) and Willis et al. (1993). For a regenerative gas turbine cycle with high inlet temperatures and modest pressures the best NOx-emissions will be achieved for the highest possible inlet temperatures.

ACKNOWLEDGEMENTS

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REFERENCES


