Development of a hydrogen fuelled 1 kW ultra micro gas turbine with special respect to designing, testing and mapping of the μ-scale combustor

A. E. Robinson, H. H.-W. Funke, P. Hendrick, E. Recker, J. Peirs

***Abstract*— This paper will provide an insight into the ongoing development of an ultra micro gas turbine rated for an estimated electrical power output of 1 kW. For a safe operation of this gas turbine with hydrogen as a fuel a new combustion chamber has to be developed and tested using the proven micromix burning principle. Detailed investigations on the burning characteristics for different chamber configurations were carried out for an optimization of the burner concept and combustor mapping for the later gas turbine integration.**

I. Introduction

Although with today's widely used batteries and accumulators there is still some slight improvement possible, they could hardly fulfill the growing demand for portable electricity which should be easily available and serviceable even under the harshest environmental surroundings. The so called energy harvesting or power scavenging systems are one of the new attempts currently under research worldwide to solve this problem. But with a power output of 100 μW up to approximately 1 mW there would be still a gap to applications needing a few Watts up to 1 kW of electrical energy. This is where ultra micro gas turbines burning liquid or gaseous fuels could bridge the gap. Used within the Brayton cycle the – compared to batteries – much higher energy density of fuels can be effectively converted into electricity.

Driven by these factors the project "powerMEMS" was founded back in 2003 with the target to develop an ultra micro gas turbine [1]. Because of the stronger regulations in terms of pollutant emissions and in search for non coal based alternative fuel the decision was made to use gaseous hydrogen as fuel. With nearly 20 years of experience in hydrogen combustion Aachen University of Applied Sciences started a new project funded by German Ministry of Education and Research (BMBF) in cooperation with the Royal Military School (RMS) in Brussels [2], [3] to develop a hydrogen fuelled combustion chamber for the potential use in an ultra micro gas turbine.

II. The Ultra Micro Gas Turbine – General Layout

The general layout of the "powerMEMS" ultra micro gas turbine (Fig. 1) consists of a conventional 3D-design with radial compressor and turbine [4]. Including the electrical generator the estimated length is 110 mm with an outer diameter of 100 mm. Rotor diameter is 20 mm running at 500,000 rpm to reach the targeted pressure ratio of 3 (Table I). The chosen material for the compressor rotor is a Titanium alloy (Ti-6Al-4V) fabricated by a 5-axis micro milling machine. In contrast the turbine rotor is fabricated from a composite ceramic (Kersit 601) to withstand the 1200 K turbine inlet temperature without cooling. The turbine is fabricated by die sinking EDM process. Figure 2 shows examples of fabricated compressor and turbine rotors.

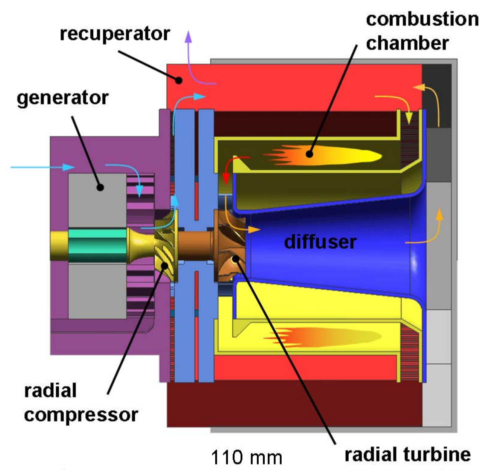

Fig. 1. Sectional drawing of basic gas turbine layout with generator

The two rotors are coupled together by a short shaft supported by air bearings. The generator drive shaft is also directly coupled to the compressor shaft thus moving the generator outside of the hot engine parts and allow cooling by the cold inlet air.

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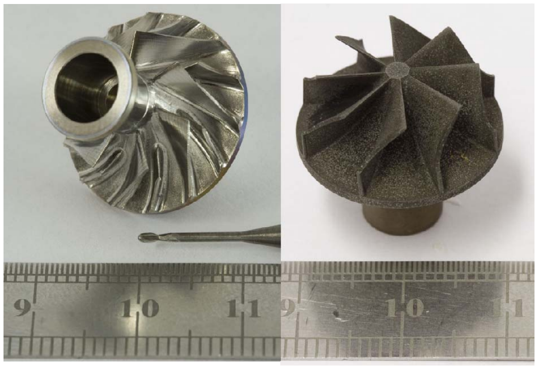

Fig. 2. Fabricated original size impeller prototypes.
Left: Titanium compressor.
Right: Ceramic turbine.

The whole gas turbine is encased by a recuperator. Precalculations of the thermodynamic cycle showed an improvement of cycle efficiency from 11% without up to 20% with recuperation [5].

Before entering the recuperator the exhaust gases are guided through a diffuser, which creates a slight sub ambient pressure at the turbine outlet and thus improves the power output of the turbine.

TABLE I
CHARACTERISTIC DATA OF THE "POWERMEMS" GAS TURBINE

nominal air mass flow rate	[g/s]	20
hydrogen mass flow rate	[g/s]	0.097
pressure ratio	[-]	3
rotor speed	[rpm]	500,000
rotor diameter	[mm]	20
combustion chamber inlet temperature	[K]	690
maximum turbine inlet temperature	[K]	1200
blade height at turbine inlet	[mm]	3.6
effective shaft output	[W]	1180

III. THE HYDROGEN COMBUSTION CHAMBER

Referring to the aforementioned layout of the µGT the maximum affordable size of the annular combustion chamber is ~40 mm inner diameter, ~60 mm outer diameter and ~50 mm overall length. Now the challenge was to downscale and fit a proven hydrogen burning principle into this small space.

The "micromix" diffusive burning principle of gaseous hydrogen based on cross flow mixing was first developed for the use in jet engines for aircrafts to significantly reduce NO_x emissions [6]-[8]. The successful realization of this concept as an alternative for existing kerosene fuelled combustion chambers led the way to the idea of scaling down the principle for the potential use in an ultra micro gas turbine. The advantage of the micro mix burning principle is its inherently safety against flashback, because it is a non premixed concept.

Figure 3 shows the realization of the micromix burning principle in the first prototype burner. The air is entering the chamber through U-shaped holes in the guiding panel. In a specific distance aft of the guiding panel (distance "x") the hydrogen is injected via 0.2 mm diameter holes (one for each hole in the air guiding panel). Following this concept air and hydrogen are mixed by cross flow interaction and the mixture burns directly in a diffusive type flame.

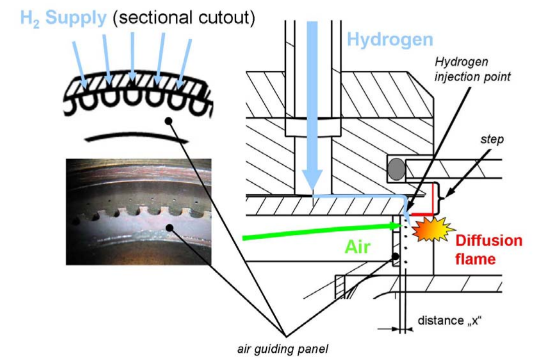

Fig. 3. Detail of the realized micromix burning principle with crossflow injection of hydrogen

IV. EXPERIMENTAL RESULTS OF COMBUSTION TESTS

Based on the state of the art equipped combustion test lab for exhaust gas analysis the test facility was adapted to micro combustion investigations. Due to the requirements of a much smaller air mass flow and preheated and pressurized tests the air supply had to be changed and supplied with an 11 kW electric heater. The pressurized and preheated air simulates the compressor exit conditions of a micro gas turbine.

TABLE II
NOMENCLATURE

mpl	[g/s]	mass flow rate of air
T_{inlet}; T_3	[K]	combustion chamber inlet temperature
A_{ref}	[mm^2]	reference area of chamber
d_{ref}	[mm]	reference diameter of chamber
p_3	[bar]	inlet pressure
$1/\theta$	[kg/(s bar $^{1.75}$ m$^{2.75}$)]	$\frac{mpl}{p_3^{1.75} \cdot A_{ref} \cdot d_{ref}^{0.75} \cdot e^{(T_3/300K)}}$
λ	[-]	$\frac{\text{air to fuel ratio}}{\text{stoichiometric air to fuel ratio}}$
$1/\lambda$	[-]	equivalence ratio (ER)
η_A	[-]	burning efficiency
Q_{H2}	[kJ/s]	applied rate of energy from hydrogen

One of the key features of the first prototype burner [9] is the possibility of visual access to the flame region during operation at atmospheric conditions. For this reason the chamber walls are completely reproduced by quartz glass components. In addition the combustion chamber can be pressurized up to 3 bar with a variable water cooled orifice setup in order to get the same conditions as in the real gas turbine. The ignition system was realized by a tungsten wire cemented from the inside to the inner glass tube.

For atmospheric tests the original µGT design point of 20 g/s air mass flow at 3 bar pressure with an inlet temperature of 690 K and a Lambda value of 6 had to be recalculated according to the Mach similarity [9]. The corresponding air mass flow for the design point at ambient pressure results in 6.7 g/s keeping all other parameters like inlet temperature and Lambda value unchanged.

The combustion tests were started with a 50 mm long annular chamber (chamber 1) which uses the maximum space available in the "powerMEMS" micro gas turbine. The variation of the chamber volume was focused in two additional prototype burners: chamber No. 2 also with 50 mm length, but smaller outer diameter (this eliminates the step shown in Fig. 3), and chamber No. 3 shortened to 20 mm in length but with the step. With preheated conditions for a constant air mass flow of 6.7 g/s tests were made for all three configurations with a full range of hydrogen mass flow variation (λ-variation) between 2.5 and 14 down to the extinction limit.

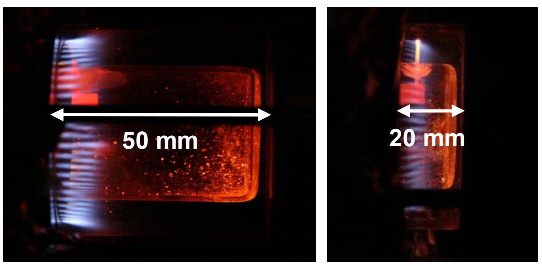

Fig. 4. Pictures of atmospheric tests at a constant mass flow rate of 6.7 g/s and λ=6
Left: chamber 1 configuration (50 mm length)
Right: chamber 3 configuration (20 mm length)

Figure 4 shows some pictures from the atmospheric tests of chamber 1 and chamber 3 configuration at a constant mass flow rate of 6.7 g/s and λ=6. The miniaturized diffusive hydrogen flames are clearly visible. The flame length does not vary significantly with the chamber length.

After careful comparison of the three tested configurations [10] the decision was made to only use the chamber 3 configuration for further investigations as it especially offers more potential for a possible integration into the "powerMEMS" gas turbine with its shorter length.

A. Ignition and Extinction Limits

The starting investigations consists in performing ignition and extinction trials. Thus the ignition and extinction limits could be set and the whole combustion chamber could be classified regarding its starting ability. This information is important for setting the control parameters of the starting sequence for the µ-gas turbine.

So the ignition tests were started under atmospheric pressure and ambient temperature (~293 K) resembling a kind of worst case scenario for starting the combustion process. In Fig. 5 you can see the results for the chamber 1 and 3 configuration plotted in a diagram where you will find the equivalence ratio $1/\lambda$(ER) on the vertical axis and the so called air-loading-parameter $1/\theta$ on the horizontal axis. This parameter is derived as reciprocal from the "theta" parameter introduced by Lefebvre [11] as an indicator of combustion loading or intensity (see Table II for the complete formula). The cold ignition limit is based on several measurements in order to get statistic data for the instant and unsteady combustion startup process during the ignition. There is an apparent correlation between air loading of the chamber and ER, the higher the air mass flow became the more fuel is needed to ignite the chamber indicated by the rising value of the ER. Starting of the µ-scale gas turbine will take place at $1/\theta$-values below 40, because even at a cold start temperature and pressure will rise quickly with the startup of the compressor and thus compensating the likewise rising air mass flow in the formula of the air-loading-parameter (Table II).

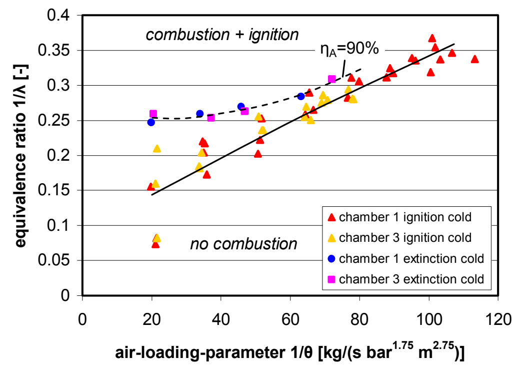

Fig. 5. Ignition and extinction limits for ambient conditions of chamber 1 and 3 configuration

Comparing the results of the two chamber configurations at ambient conditions there is barely any difference. So it is to assume that stability at ambient conditions is mainly influenced by the burning principle and not by combustion chamber geometry. The same could be said about the extinction points, too. As criteria for the extinction limit a burning efficiency (calculated from the exhaust gas analysis) of η_A=90% was set. Below that value extinction of the flame is defined. Although the chamber still burns below this value, burning efficiency and stability drops down significantly making it improper for continuous gas turbine operation. As a result the ignition limit is slightly below the extinction limit, meaning that although the chamber is not properly burning below the extinction limit anymore you can still ignite the chamber at a lower ER reducing the risk of an explosive ignition during startup. For the later gas turbine application the ignition line as well as the (dotted) extinction line are important reference parameters for setting the control laws at startup conditions of the gas turbine concerning the minimal amount of fuel that is necessary to safely ignite the chamber and then going into a stable modus of operation.

B. Effect of Mass Flow Variation

In order to get an idea of the full potential from the chosen chamber 3 configuration extensive testing for this chamber was accomplished including a full mass flow variation ranging from part load (45%) to extreme overload (150%) of gas turbine operation.

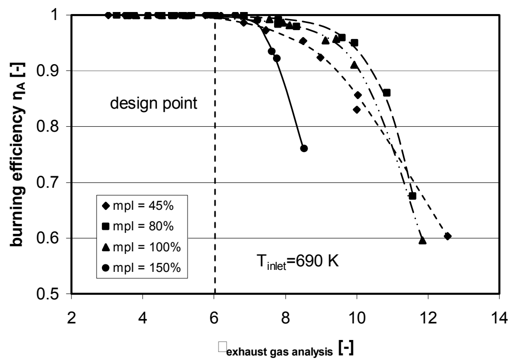

Fig. 6. Burning efficiency against λ-variation at different air mass flow rates (atmospheric, preheated conditions)

Figure 6 shows the burning efficiencies η_A=(heat released in combustion)/(heat available in fuel) of the different mass flow rates over a λ-variation. Values are calculated from the measured exhaust gas mass fractions gathered by the exhaust gas analysis.

The chamber shows a wide stability range also for this significant variation in mass flow. Up to λ=6 the burning efficiency is 100%. At lower mass flow rates (45% and 80%) similar curves to the design point mass flow rate of 6.7 g/s (100%) are visible, all reaching an efficiency of more than 70 % up to λ=12. Only at an extremely increased mass flow rate (150%) the combustion chamber looses stability significantly earlier at λ=7.5.

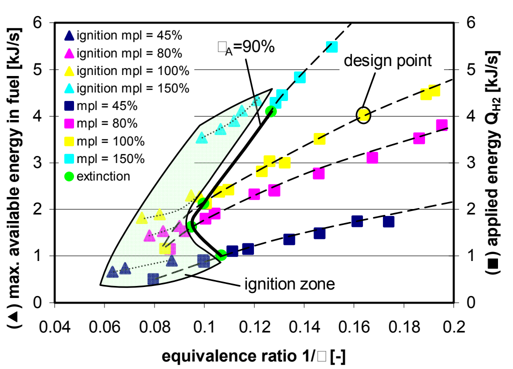

Fig. 7. Applied energy Q_{H2} against ER for different air mass flow rates (atmospheric, preheated conditions). Maximum available energy in fuel against ER for hot ignition points.

With the constant net calorific value of hydrogen, the hydrogen mass flow and the burning efficiency the applied energy Q_{H2} for each measurement point was calculated. Figure 7 shows these values plotted against the corresponding ER. Furthermore the extinction limit (set at 90% burning efficiency as described above) is put into this diagram. So this illustration shows clearly that the best stability range for the operation of this combustion chamber is around 80% to 100% of the design point mass flow rate. Lower or higher air mass flow rates will end up with a reduction in burning efficiency and stability. This is shown in Fig. 7 by the extinction limit line bending towards higher ERs for the mass flows differing from the design point mass flow rate and thus also indicating the limits for safe and economic operation of this chamber design.

In addition to the operating lines the recorded points of hot ignition summarized to an ignition zone are displayed in Fig. 7. As it is not possible to get an instant exhaust gas analysis and calculated burning efficiency during an ignition attempt, the maximum available energy in fuel, calculated from the amount of hydrogen mass flow at the time of ignition and the net calorific value, is used as the y-axis scale parameter. Contrary to Fig. 5 this ignition zone shows the hot starting points and thus giving information about the relight capabilities of the chamber after a flame out for example. Like under cold conditions the chamber will ignite at ERs below the chosen extinction limit for every tested mass flow, but for a stable burning the hydrogen mass flow has to be increased to reach ERs above the extinction line.

Next step in testing was the application of 3 bar pressure and 20 g/s air mass flow to reach the desired design point conditions. A comparison between the atmospheric and pressurized tests both at 100 % air mass flow rate shows a significant drop down in burning efficiency for the pressurized tests. Together with numerical investigations the pressure influence was further investigated and the chamber length was found to be the possible cause for reduced burning stability under pressurized conditions [12]. Therefore for further optimization the chamber length has to be increased again to achieve the desired burning efficiency and stability in all operating ranges.

V. Conclusions and Outlook

Although the key components of the ultra micro gas turbine have been fully designed and manufactured further testing of each component is necessary before they can be mounted together in a real engine.

For the combustion chamber as one of the most crucial parts of the whole engine design the non-premixed micromix burning principle has proven its capability even at this small scale. Different geometry variants were successfully tested at the new test rig for micro scale combustion chambers under ambient pressure.

The results from ignition and extinction investigations show a wide range of safe ignition under every possible design and off design mass flow rates nearly unaffected by the chosen chamber length for atmospheric conditions, thus making the chamber applicable for the most common gas turbine starting procedures. Coupled with the results from the mass flow variations a complete mapping of the μ-scale micromix combustion chamber could be made containing all information needed for setting the control laws and parameters as well as the start up sequence for the later use in the ultra micro gas turbine application. Furthermore the results gave important information about the control regions in which the chamber could be operated safely and economic.

One configuration was chosen for further investigations including pressurized tests. Even though the atmospheric results of this chamber showed a very good burning stability for design and off-design operation, some losses in stability occurred at higher pressure. Further experimental tests linked with numerical investigations direct towards the chamber volume as possible cause for the occurred behaviour.

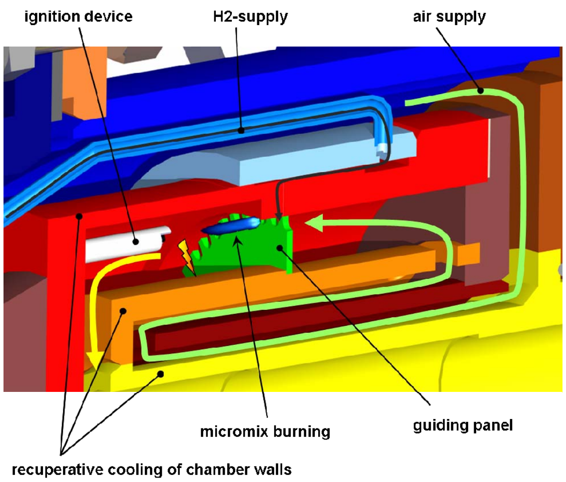

Fig. 8. Cutaway of latest combustion chamber prototype with recuperative wall cooling

To comply with these findings a new prototype burner has been designed which incorporates a slight increase in chamber length (Fig. 8). It also features recuperative wall cooling which allows the use of inox alloys for the chamber design. This prototype also considers the possible integration of the micromix burner concept into the real "powerMEMS" gas turbine.

As the manufacturing of the new prototype burner is almost finished further work will focus on extensive pressurized tests for this prototype chamber.

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