STUDY OF A NEW COMBUSTION CHAMBER CONCEPT WITH PREMIXING (PREVAPORISING) DELIVERY TUBES

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ABSTRACT

This paper presents a preliminary investigation of the new combustion chamber concept with double reversed flow, incorporating premixing (prevaporising) delivery tubes. This combustor was developed as a part of the project SATE (Sophisticated-Small Aircraft Turbine Engine) as a possible solution for reduction of pollutant emission in small gas turbine engines (up to 1MW output). In contrast with most used small gas turbine combustors, where fuel or fuel mixture is injected in the same direction with the main flow within the liner, in this concept the partially prepared fuel mixture is injected in opposed direction, this almost doubles the effective path of the fuel compared to the classical design and prevents escape of unburned fuel drops.

Presented combustor concept has been developed and investigated by use of numerical simulation methods with methane as the model fuel. Through the preliminary research it was found, that this combustor could operate at very lean primary zone conditions, starting at the equivalence ratio of 0.5 and accordingly low combustion temperatures of 1700 K, what promises low NO_x formation. Operation was simulated under high power density of 120 MW/m^3 in ISA conditions and 540 MW/m^3 at pressure of 460 kPa.

KEYWORDS

Combustion chamber, double reversed flow combustor, vaporising tubes, lean, low NOx, CO emission.

NOMENCLATURE

c	reaction progress variable
c_p	specific heat at constant pressure
Hv	fuel heating value
$\dot{m}$	mass flow
MW	molecular weight
Sc	reaction progress source term
Sc_t	turbulent Schmidt number
T	temperature
T_{act}	activation temperature
u	velocity
U_t	turbulent flame speed
t	time
Y	mass fraction to the total mass
ρ	density
μ_t	turbulent viscosity

subscripts

fu	fuel
ox	oxidizer
in	inlet
-	mean parameter
‘	fluctuating parameter
i	direction component

INTRODUCTION

An intensive research has been done during past years to minimize pollutant emissions from gas turbines combustors, notably nitrogen oxides NO_x and carbon monoxide CO. Most attention has been devoted to big size engines as the main source of pollutant, but increasing pressure is presently seen also to reduce emissions of small gas turbine engines.
The problem of pollutant reduction is, that UHC and CO burning rates are accelerated by high combustion temperatures and pressure, but this simultaneously leads to higher NOx emission formed through Zeldovich mechanism. To obtain low NOx and CO emission, it is needed to keep the combustion temperature within the range of about 1670-1800K [2] and simultaneously keeping high turbulence in reaction zone to prevent hot spots, this reduction method is used by the state of art lean combustion systems [2]. Another possible method of NOx reduction is burning under rich condition, where all oxygen is consumed for fuel burning, because of higher fuel affinity to oxidation than that of nitrogen, then the products are cooled enough fast (quenched) by adding big portion of cold air and then burning completion under lean conditions. This principle is used in RQL (Rich burn – Quick quench-Lean burn) combustors [2].
Difficulties in development of a small combustor are coming out of a small size its components and accordingly high technological problems, while the price is very important in this engine class, so use of classical big size combustion systems is very limited. Based on these factors a new combustion chamber concept has been designed and is recently examined and developed.

BASIC DESIGN - DESCRIBTION

As seen in the fig. 1, this concept is in basic principle a double reverse flow combustor, where fuel mixture is injected against the main flow within the liner. This has the advantage, that the unvaporised fuel drops path length is almost doubled compared to classical combustors, where fuel is injected co-flow. Another advantage is, that the fuel mixture passes through the hot exhaust area before its combustion, this intensively preheats the fuel mixture, what enables leaner blow off limit. Also this type of flow pattern creates strong recirculation and stagnation zones within the rear part of the liner fig. 6, needed for stable and efficient operation.
Fuel mixture is partially premixed and prevaporised within the delivery tube, which is heated by the outgoing combustion products, this tube also supplies prepared mixture to the reaction zone and in combination with auxiliary air inlets produces desired mixture fraction pattern and flow field for optimal combustion.

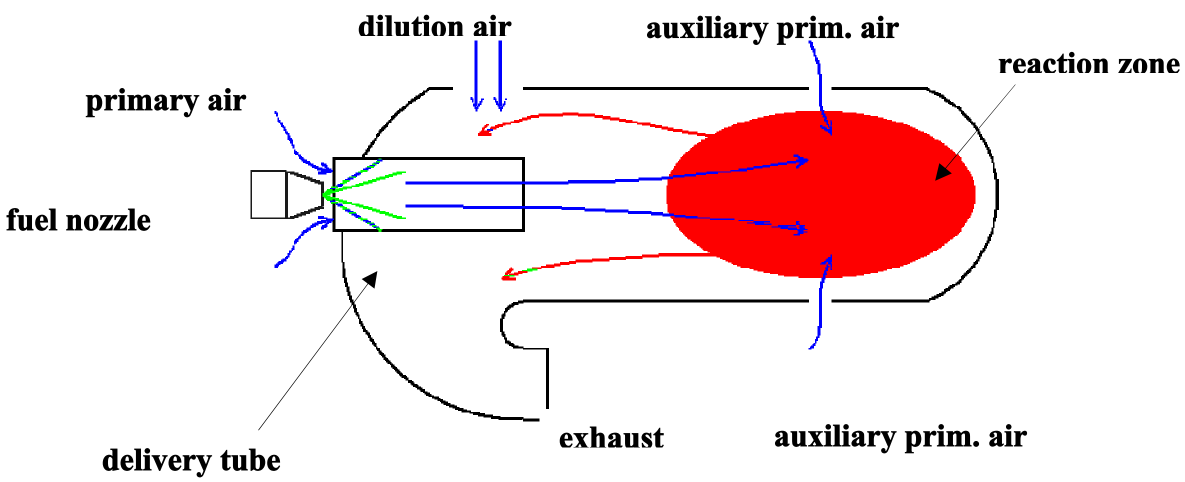

fig. 1 – combustor schematics (cross section)

FUEL INJECTION

For the primary tests and evaluation of the CFD model methane gas will be used, in the next development stage liquid fuel (Jet A1) is planed.

gas fuel

Gas fuel mixture preparation is relatively easy, because the fuel is already evaporated, however from the CFD preliminary results it was found, that the mixture ratio at the exit of the delivery tube should have certain pattern shown in fig. 2, this is needed for protecting the main fuel stream from mixing with the outgoing exhaust products before reaching main reaction zone, what causes carrying the fuel away unburned and can also cause premature mixture ignition and creation near stoichiometric hot spots.

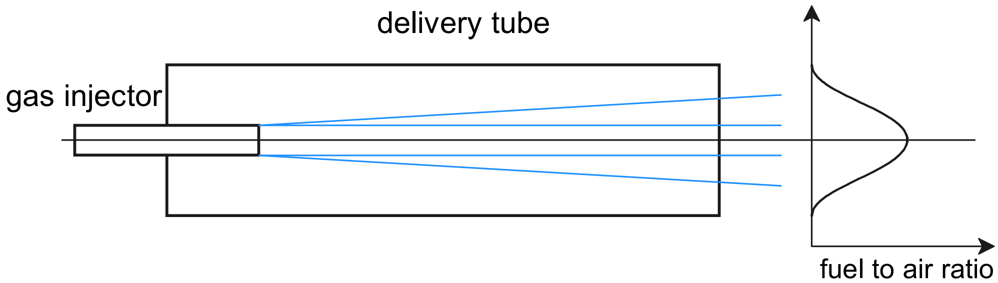

fig. 2 – gas fuel injector

liquid fuel

Liquid fuel has the disadvantage, that it needs to be vaporized prior combustion, this is effectively done by atomizing the fuel to small drops, what enlarges a surface to volume ratio and correspondingly shortens an evaporation time. State of art big combustors use an Airblast atomizer for this purpose, this atomizer has the advantage of producing very small fuel drops at low fuel pressure, it's disadvantage is poor atomization capabilities at start up conditions, where air pressure drop across the atomizer is low. Another possible method is use of pressure swirl atomizer, which has advantage of producing small droplets "independently" on the air pressure drop across the liner, but requires high fuel pressure at high throughput.
From these statements a combination of pressure swirl and airblast atomizer was chosen, this should provide sufficient atomization at the start up, desired fuel to air ratio pattern at the delivery tube exit and additional airblast atomization. Possible arrangement of the delivery tube for liquid fuel is shown in the fig. 3. Retarding tube except of prefilming has the purpose of stopping the fuel drops to fly on the delivery tube wall and prevent formation of rich area in the outer diameter.

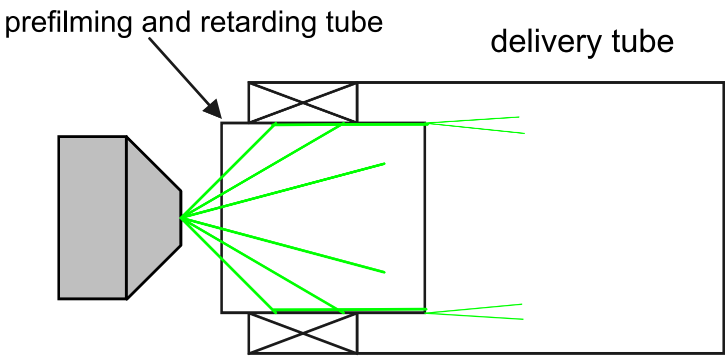

fig. 3 – liquid fuel injector

WELL STIRRED REACTOR CALCULATIONS

To determine operation ranges of the combustor for known parameters stated in tab. 1 (derived from operating conditions of the TJ100 test engine) a single step CH_4 reaction WSR model [1],[2] was used, described by equation 1 and 2

$$\dot{w}_{fu} = -A \frac{MW_{fu}}{MW_{fu} MW_{ox}} \rho^2 Y_{fu} Y_{ox} \exp\left(\frac{-T_{act}}{T}\right) \quad 1$$

$$\dot{m} c_p \left(T - T_{in}\right) = -\dot{w}_{fu} V H v \quad 2$$

With $A=1x10^9$ kmol m^{-3} , T_{act}=20000K.

Operating pressure	Pa	479800
Inflow temperature	K	498
Fuel mass flow rate	kg/s	0,038
Air mass flow rate	kg/s	1,756
Combustion volume	m^3	0,0031

tab. 1 – input parameters

The reaction temperature and fuel burning efficiency across different equivalence ratios ranging from 0,4 to 1 were then calculated. Equivalence ratio was adjusted by changing of the combustion air flow rate with constant fuel mass flow. The results are shown in fig. 4.

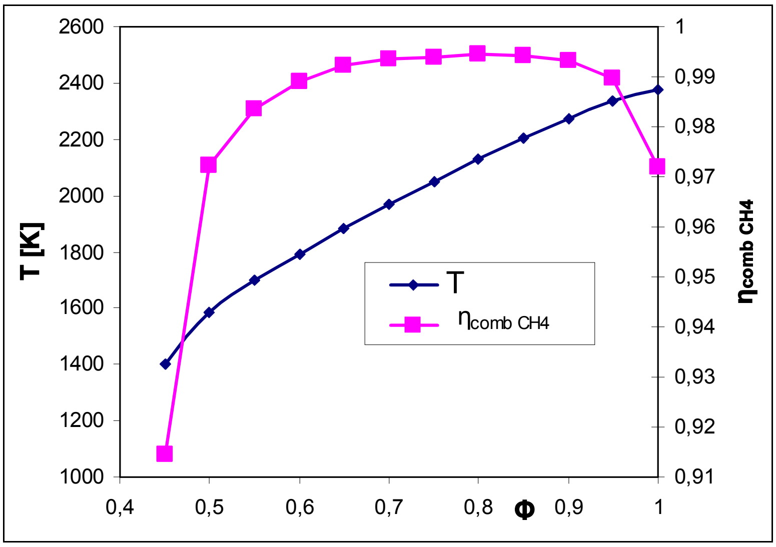

fig. 4 – combustion temperature, CH_4 burning efficiency vs. equivalence ratio

From the WSR calculations a lean blow off limit was determined to be at equivalence ratio of 0.46 for power density of $630MW/m^3$. Optimal temperature for low NOx and CO emissions is in the range of 1670-1800K [2], what corresponds to combustion zone equivalence ratio of 0.55-0.6 . Based on these results an optimal ratio of the primary to dilution air was determined and stated as the main leading point for the combustor air passages design.

CFD SIMULATION

Through the development phase many different combustor arrangements have been simulated using CFD code. The changes have been done on the position and dimensions of the delivery tube, setup of the auxiliary and dilution inlets and fuel nozzle position. Results have been compared each other to determine optimal layout and to get better understanding of processes inside this combustor. Some of the arrangements tested are shown in fig. 5.
1/12 segment of the whole combustor annulus has been solved and a combination of structured un unstructured grid with total amount of 400-500 thousands cells has been used. Boundary conditions have been taken the same as for the WSR model - tab. 1.

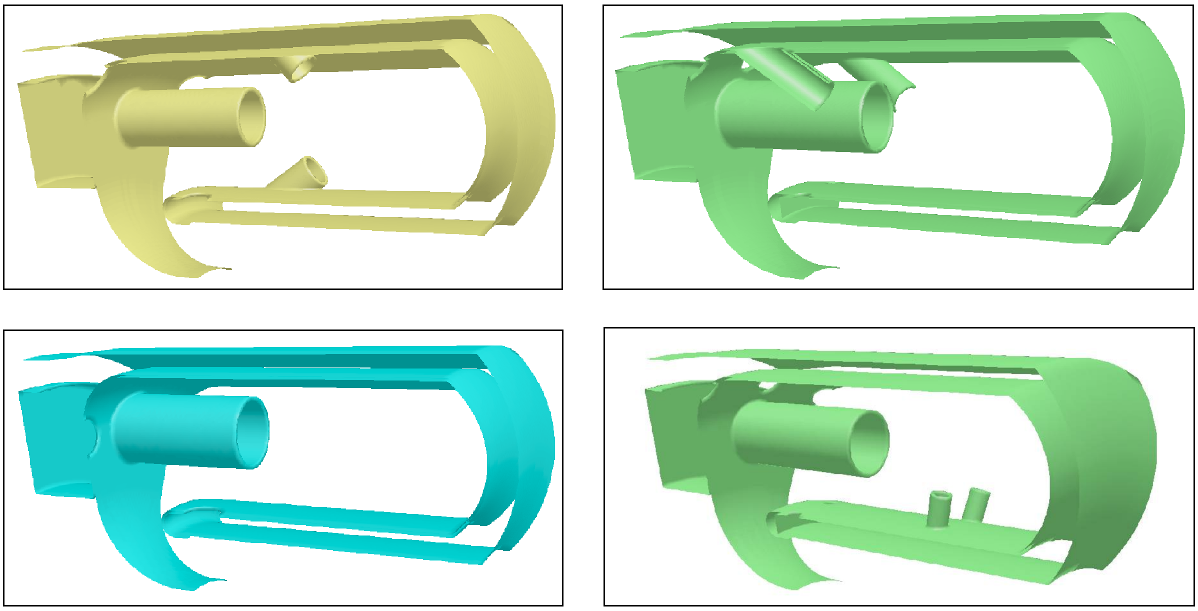

fig. 5 – some of the simulated variants

solver

For modeling of the combustion processes a partially premixed flamelet-pdf/FANS solver was used, this solver was chosen due to it's ability of incorporating finite rate chemistry and on the other hand significant reduction of the computation time, because chemistry is reduced and completely described by two conserved scalar quantities, mixture fraction f and scalar dissipation χ. The PDF look up tables of chemical species, density and temperature were calculated using 17 species 25 step approach of global CH_4 – air reaction (CHEMKIN). The system was for this development stage assumed adiabatic, so neither heat losses nor radiation were incorporated. Also no model of slow reacting species like NOx and CO was included.

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\upsilon u_i) = 0 \qquad 3$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_l}{\partial x_l}\right)\right] + \frac{\partial}{\partial x_j}(-\rho \overline{u_i' u_j'}) \qquad 4$$

Within the partially premixed model a scalar quantity c, – favre averaged reaction progress variable, is solved to determine flame front position. Behind the flame front (c=1), the mixture is burnt and the laminar flamelet mixture fraction solution is used. Ahead of the flame front (c = 0), the species mass fractions, temperature, and density are calculated from the mixed but unburnt mixture fraction. Within the flame (0 < c < 1), a linear combination of the unburnt and burnt mixtures is used.

Scalar quantity c is modeled by solving a transport equation:

$$\frac{\partial}{\partial t}(\rho c) + \nabla \cdot (\rho \vec{u} c) = \nabla \cdot \left(\frac{\mu_t}{Sc_t} \nabla c \right) + \rho S_c \qquad 5$$

where the mean reaction rate ρS_c is modeled as

$$\rho S_c = \rho_u U_t |\nabla c| \qquad 6$$

results - velocity field

In the fig. 6 and fig. 7 is clearly seen, that strong recirculation areas are formed on sides of delivery tube jet and a stagnation area is formed at the proximity of the rear liner wall. These recirculation areas are crucial for stable and efficient combustion.

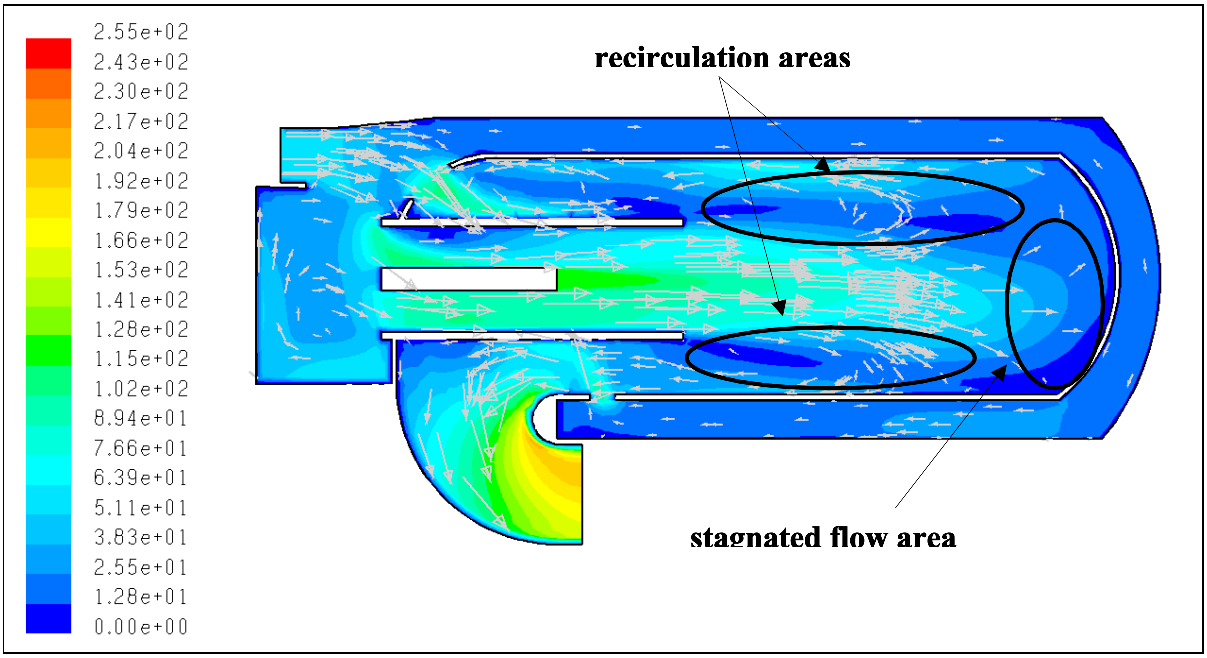

fig. 6 – velocity field – middle section

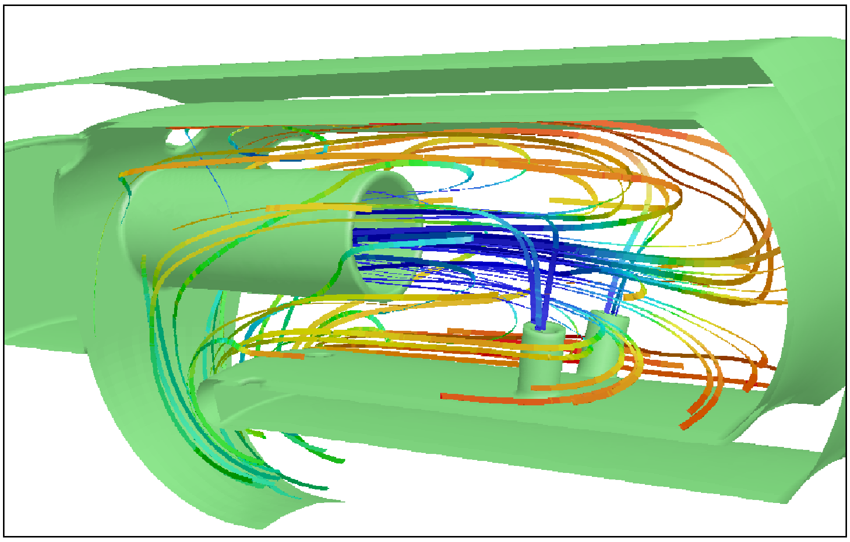

fig. 7 – path lines

Also has been found, that the fuel nozzle position and arrangement must supply the fuel only to the inner part of the delivery tube air jet and the outer portion of the air must sustain lean at least to the main reaction zone. In the other case, the fuel can be blown out by the outgoing flow without burning as can be seen in the fig. 8.

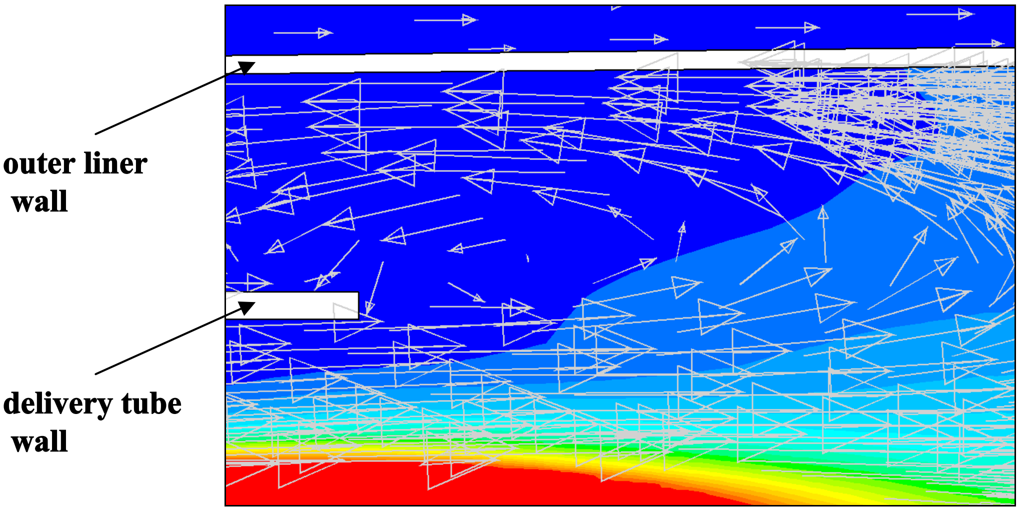

fig. 8 – delivery tube exit CH_4 mass ratio + velocity vectors

comparison of different equivalence ratios

Comparisons of the same primary zone and delivery tube arrangement with different primary zone equivalence ratios of 0.9 and 0.55 were done, to determine how the combustion would change. Primary zone equivalence ratio was related to the ratio of the fuel to the total primary air. Lower equivalence ratio was obtained, by closing dilution air inlet, what correspondingly means rise of primary air throughput. Fuel mass flow and other parameters sustained same to keep same power density for both cases.
As seen in the fig. 9 showing product formation rate (6), the mean flame area is stably anchored thanks to the stagnation zone within the rear part of the combustor.

equivalence ratio 0.9 **equivalence ratio 0.55**

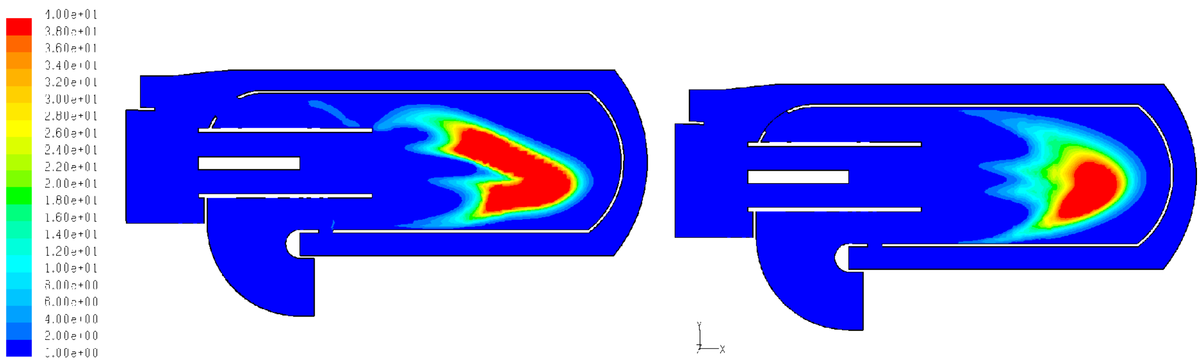

fig. 9 – product formation rate

In the fig. 8 and fig. 9 showing temperature fields across different sections of the combustor, can bee seen how the max. temperature is reduced from about 2100K for the equivalence ratio of 0.9 to about 1700K and the temperature field became also more uniform, but it must be noted, that this higher uniformity is influenced by higher turbulence, due to the higher pressure drop across the liner. Also it is seen that the maximal temperature is close to the liner rear wall, what can cause problems with wall durability and another research needs to be done to solve this problem.

equivalence ratio 0.9 **equivalence ratio 0.55**

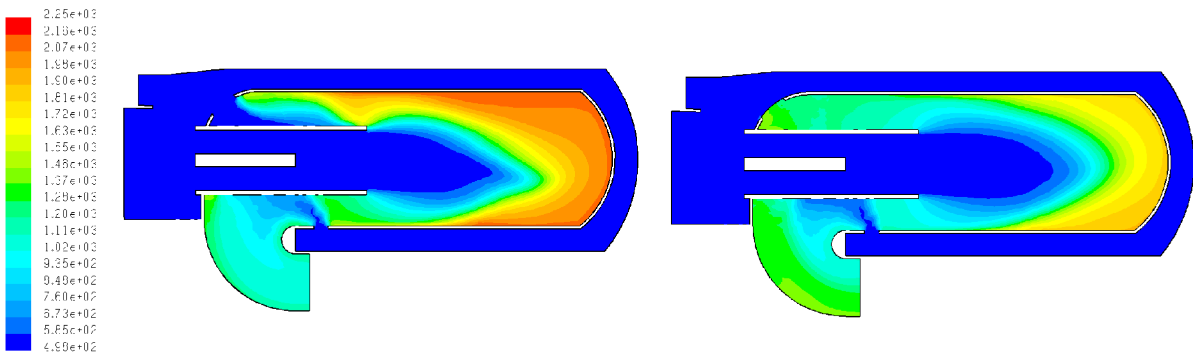

fig. 10 – temperature field –section through delivery tube

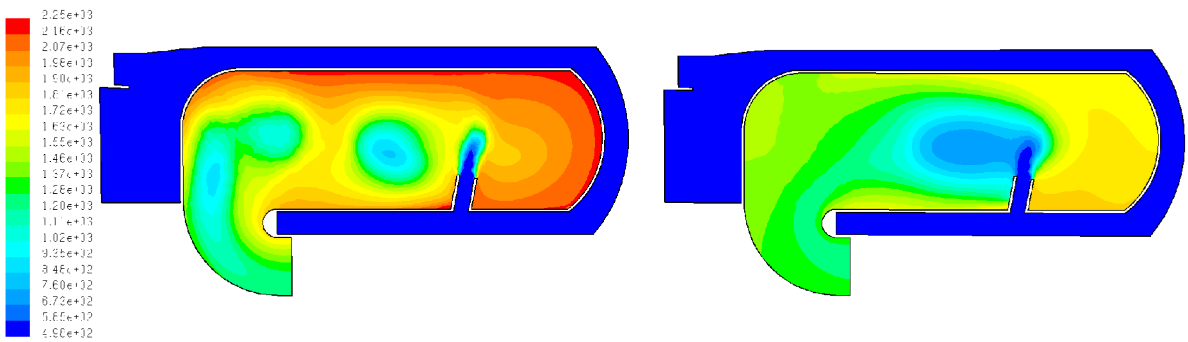

fig. 11 – temperature field – section through auxiliary tube

CONCLUSION

A new combustor concept has been deigned and preliminary calculations proved, that the combustor is able to provide stable combustion under very lean conditions and accordingly low combustion temperatures at high power density of 120MW/m^3 (100kPa), what is needed by small gas turbine engines. At this development stage the combustor has relatively high pressure loss (about 6-7%) and another work is needed to be done to precisely determine and reduce the pressure losses to the smallest level while keeping good combustion.
Also during these preliminary calculations no heat losses, wall heating and slow reacting species like NOx and CO have been modeled, so next work is needed to be done to get answers to these problems.

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