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DESIGN AND OPTIMIZATION OF A HIGH TEMPERATURE WATER COOLED PROBE FOR GAS ANALYSIS MEASUREMENT ON K11 COMBUSTION TEST RIG

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ABSTRACT

DGA Aero-engine Testing offers an experience of more than 60 years in airbreathing engines testing. Components and especially combustion chambers are part of this experience from altitude simulated conditions to very high pressure and very high temperature requested for the next-generation engines. The K11, one of the most powerful combustion facilities worldwide, allows tests on full scale annular combustion chambers with a wide range of measurement devices. Its performance (100 kg/s, 60 bars, inlet 1073 K, outlet 2573 K) makes it a unique tool in the world for developing civil and military combustors. An internal traverse gear enables 360° mapping of combustor outlet, with pressure or temperature measurements and gas analysis (CO/CO2, UHC, NOx, Smoke Number) in accordance with ICAO and SAE standards. New probes have been designed and optimized to perform tests with an outlet combustor temperature up to (2123 K) with a high level of reliability in order to ensure the continuity of emissions measurement in the most severe conditions. The main results and the lessons learned during this study concerning the thermal and the structural behavior of the probe will be shown and discussed in this paper.

INTRODUCTION

Even though high temperature gas analysis probes have been used for decades, their manufacturing is often the result of a complex study based on numerous compromises, especially for a combustion test rig like the K11 facility where some of the most extreme test conditions can be met (Fig. 1). In the framework of the European founded research program NEWAC where one of the main objective is to develop and validate lean fuel injection technology that demonstrate 60% to 70% reduction of NOx emissions in the LTO cycle versus the ICAO CAEP/2 limit, DGA Aero-engines Testing is improving the performances of its own gas analysis probes in order to reach the test conditions required by this program.

Figure 1 – K11 combustion test rig (max. inlet test conditions: 100 kg/s, 60 bars, 1073 K, max. outlet temperature: 2573 K).

G. Lengelle and C. Verdier dedicated a whole chapter to the problem of the gas sampling in document [1]. Two different kinds of probes are presented in their book. The first one is a micro-probe for studying laminar flames in laboratory and is typically made of quartz. Freezing the sample is obtained by a quick expansion of the gas inside a divergent part the probe. The second type of probe is used for a large scale facility and the sample is simply frozen by the water-cooling of the probe.

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The principal types of probes employed in combustion research are also discussed in reference [2] and [3] : isokinetic or aerodynamic, cooled or uncooled, for a tiny laboratory flame or an industrial burner, etc. More recent references like [4] or [5] shows that this topic is still explored for imaging applications or unsteady pressure measurements. As it is said by N. P. Cernansky in [6], “no one type of probe is best for all sampling problems” and generally speaking, each probe is designed for a very specific use.

Considering the capacity of the K11 which is an industrial facility and the harsh exhaust environment a water-cooled probe is an obvious solution. However, the quenching of chemical reactions is one of the major functions of the probe if a representative measurement of gas composition at the exit of the flame chamber is expected. Aerodynamic-quenching sampling probes are known [7] to have a good gas cooling rate near the inlet (~10^8 K/s). Consequently, the use of a combined probe involving aerodynamic and water cooled quenching has been chosen for our combustion test rig for several years. On K11, tests are often performed with a brand new combustor with its own dimensions and specifications. Consequently, the manufacturing of a new probe is specific and must be considered each time as a prototype. That implies high development costs and increasing risks of failure especially if the capabilities of the probe must be increased. The idea of this study is to capitalize experience in order to make easier the next development of gas analysis probes for further tests at very high temperatures. For NEWAC tests, a previous design has been reused. Two probes with this old design were successfully used by the past at temperature up to 1950 K during approximately 10 hours with no significant visual impact (Fig. 2).

Figure 2 – Leading edge of an old design gas analysis probes after tests (max. outlet temperature around 1950 K during 10h).

The position of immersion at ambient temperature before and after tests has been checked and the results are summarized in table 1 for future comparison. The height of this old design probe has been affected by the flame. We have observed a small diminution with an impact on the position of the five immersions of less than 0.16 %.

Immersion number		Variation of position before & after tests
Old design probe #1	1	- 0.11 %
	2	- 0.11 %
	3	- 0.10 %
	4	- 0.16%
	5	- 0.10 %
Old design probe #2	1	- 0.14 %
	2	- 0.15 %
	3	- 0.08 %
	4	- 0.08 %
	5	- 0.13 %

Table 1 – Differences in the radial position of immersions of the old design probe before and after tests at 1950 K during 10h.

In order to reach temperature higher than 2100 K for the NEWAC project an upgrade of this design has been studied that will be presented in the present paper.

ENVIRONMENTAL CONSTRAINTS

Probe requirements

The primary objective of this study is to develop a probe based on the following requirements:

- the sampling of combustion gas at the exit of the combustor on a specified measurement plan with five immersions in order to perform emission and temperature measurement by calculating the enthalpy of the different chemical species;
- the sampling must follow as much as possible SAE recommended practice [8] and OACI International Standards [9];
- the smallest size as possible for a minimal flow disturbance;
- and finally, a high reliability considering the harsh conditions at the exit of the combustor.

Mechanical constraints

The probe should be placed as close as possible to the exit of the combustor with no interference with it in cold and hot conditions. That defines the upper geometric limits of the probe. The radial position of the five immersions is precisely specified for hot conditions and the design of the probe should take into account the thermal expansion so the immersions will be perfectly positioned in hot conditions. The probe is fixed on a traverse gear (Fig. 3) that defines the lower geometry of the probe with an interface on the water network and the gas analysis sampling lines. The traverse gear can receive 2 gas analysis probes and two thermocouples probes. If a slot is not used by a probe, a dummy one is installed as we can see on Fig. 3.

The internal design of the traverse gear is complex and summarized by Fig. 4. The water cooling network of the probes is designed to follow the pressure of the test rig so the ΔP between water and air remains constant (around 15 bars) with a maximal flow of 37 700 l/h for four probes. The use of a dilution probe as mentioned by Jimenez & Ballester [7] is impossible because the traverse gear is not equipped with a nitrogen line.

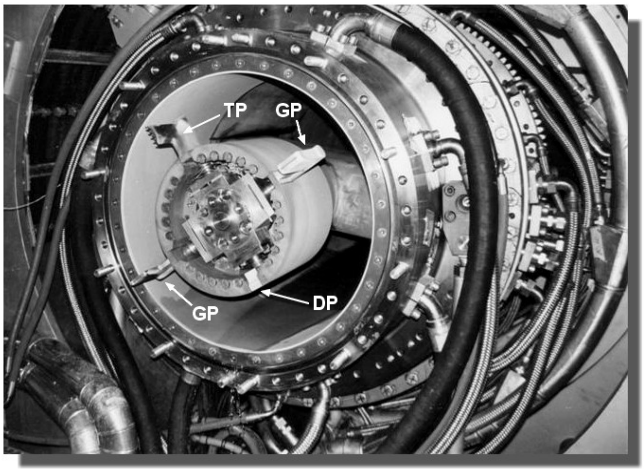

Figure 3 – Traverse gear of K11 equipped with a temperature probe (TP), 2 gas analysis probes (GP) and a dummy probe (DP).

The traverse gear is also water cooled with internal air ventilation. The gas sample is maintained at 433 K outside the probe by a water system and an electric heated line. The probe is used to perform gas analysis sampling on a defined measurement plan and on five different radii (R1 to R5) with no interference with the combustor and its test unit. All these constraints lead to an external design presented in Fig. 5 with no reference size due to confidentiality reasons.

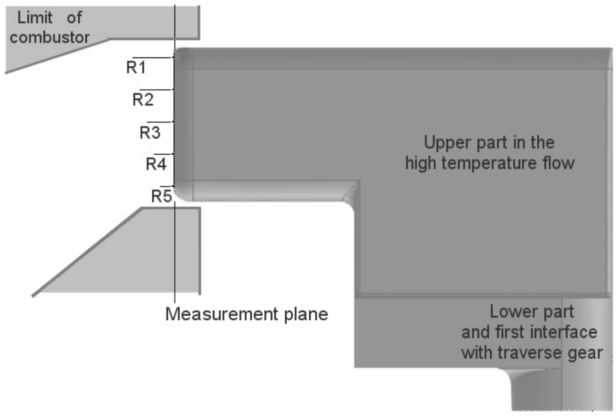

Figure 5 – External design of upper part considering the interference with the combustor, the position of the measurement plane and the radial position of the five gas sampling inlet.

Aerothermal environment

The probes will be put at the exit of the combustor. The flow conditions (design point) at this location are 32 bars and 2123 K and the velocity of the flow is around 66 m/s. The density of gases is computed with the perfect gas law and is equal to 5.252 kg/m^3. All other properties are computed with CEAgui software [10].

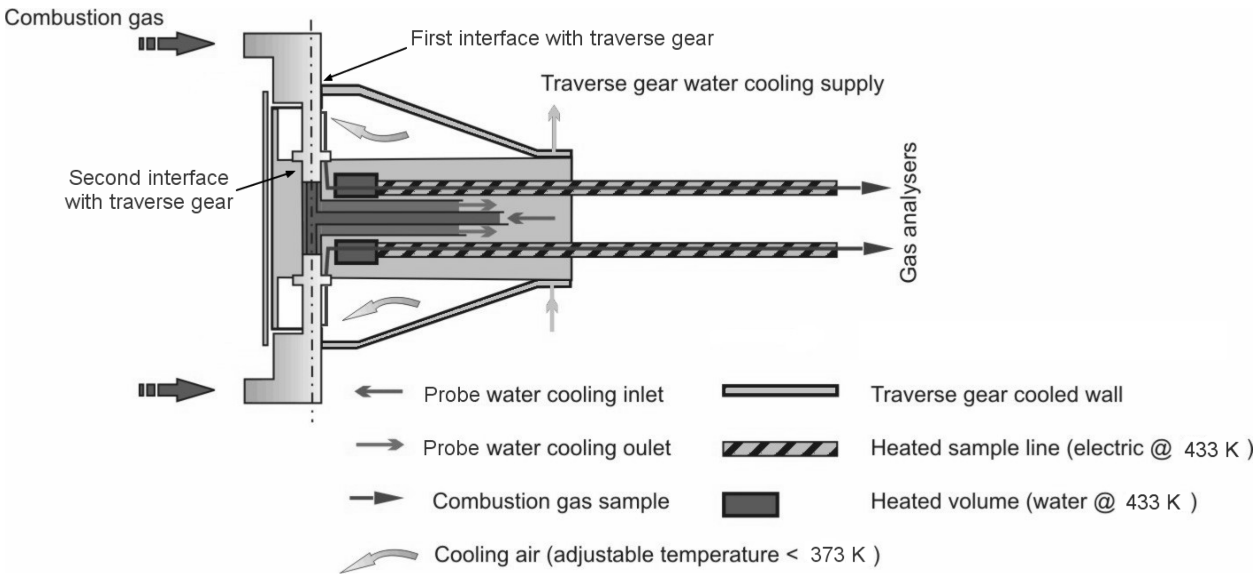

Figure 4 – Internal scheme of the K11 traverse gear with all its ancillaries.

Properties	Value
Molar mass [kg/mol]	0.028968
Dynamic viscosity [kg.m^{-1}.s^{-1}]	6.476×10^{-5}
Heat capacity [J.kg^{-1}.K^{-1}]	1350.5
Conductivity [W.m^{-1}.K^{-1}]	0.117
Prandlt number	0.745

Table 2 – Combustion gas properties at equilibrium.

A classic Nusselt number correlation [11] used for the design of the very first probes of K11 has been used in order to estimate the heat flux received by the leading edge of the probe.

$$Nu_{[11]}(\theta) = 1.14 \cdot Re^{1/2} \cdot Pr^{0.4} \cdot \left(1-(\theta/90)^3\right) \quad (1)$$

θ is the angle from the leading edge in degree and we have a stagnation point at θ = 0. This correlation is valid up to 80° with turbulence intensity less than 1%. With a Reynolds number around 75 000, the convective heat flux computed from Eq. (1) gives at the stagnation point of the leading edge a value of 3.59 MW.m^{-2}. With a turbulent intensity of 10%, we can assume from data collected in document [11] that the heat flux would increase at least 50%. That's consistent with the Nusselt number correlation mentioned in paper [5].

A comparison has been made with the Lowery and Vachon formulation Eq. (2) and with the same Reynolds number and a turbulence intensity of 10%, the convective heat flux is equal to 5.33 MW.m^{-2}.

$$\overline{Nu_{LV}} = Re^{0.5} \cdot \left[\begin{array}{c} 1.01 + 2.64\left(\dfrac{Tu \cdot Re^{0.5}}{100}\right) \\ -3.07\left(\dfrac{Tu \cdot Re^{0.5}}{100}\right)^2 \end{array} \right] \quad (2)$$

PRELIMINARY DESIGN

Heat flux applied on the probe at design point

Considering the very high level of heat flux on the leading edge of the probe and its shape, a CFD computation has been performed in order to have a better idea of the heat flux applied on all the parts of the gas analysis probe: the leading edge, the side area and the rear area (Fig. 7). The simulation has been performed with Fluent© on a 2D shape with a second order implicit segregated solver and with a k-ε turbulent model with standard wall functions and no radiation. The flow is considered as a perfect gas at 2123 K with a velocity of 66m/s and its properties are given by table 2.

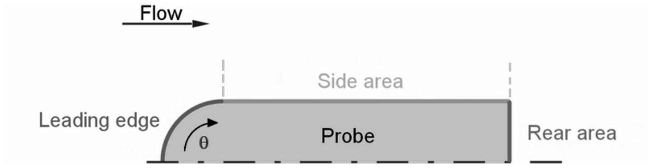

Figure 7 – 2D geometry of the probe for computing the convective heat flux.

The contours of velocity magnitude and the value of the convective heat flux along the probe is given in Fig. 8. The peak value is at 4.98 MW/m^2. The mean value on the leading edge is at 3.52 MW/m^2, the mean value on the side area is at 2.05 MW/m^2 and the mean value on the rear area is at 0.54 MW/m^2.

The radiative flux is computed in a conservative way due to the lack of information on the flame emissivity (considered as a black body) and the complexity of the geometry (form factor is equal to 1). It gives an additional heat flux contribution around 1 MW/m^2 excepting in the rear area.

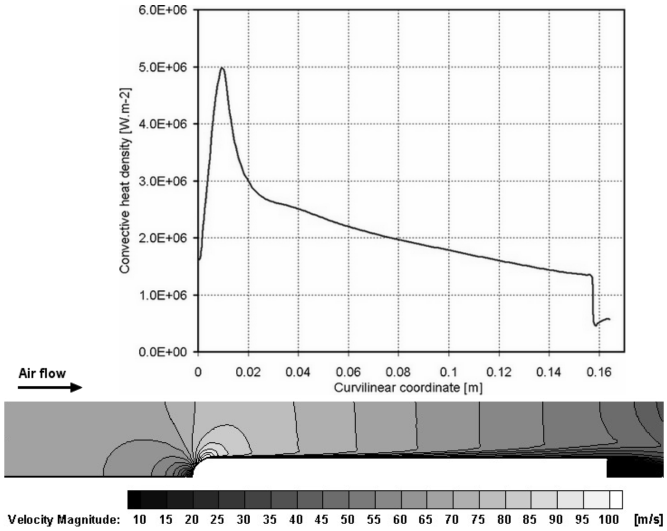

Figure 8 – Contours of velocity magnitude around the probe and convective heat flux at design point.

Internal water flow and thermal behavior

From the heat flux previously computed, the internal modelization of the probe has been performed in order to check the quality of the water cooling and to allow the assessment of the mechanical constraints with the knowledge of the wall temperature of the probe.

A part of the fluid volume implied in the CFD simulation is given in Fig. 9 where the water path is mentioned. Five guide vanes are visible on this figure. They have a key role in the improvement of the quality of the flow comparing to the very first design without them where big stagnation areas have been noticed in the leading edge part of the probe. The total number of cells is 2 144 000. The simulation has been performed with Fluent© with a second order implicit segregated solver and with a k-ε turbulent model with standard wall functions. The water properties have been chosen for a pressure of 40 bars and a temperature around 380 K. Four simulations cases have been performed and are summarized in table 3 with inlet water temperature at 300 K in order to study the influence of the pressure and the water velocity. Case Sim2 with the highest water mass flow is the most representative for a normal behavior of the traverse gear and its internal water cooling system.

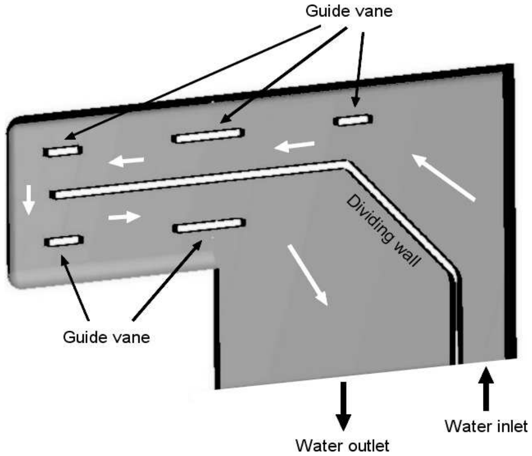

Figure 9 – Simulated volume inside the gas analysis probe with the water path.

Case	Water pressure	Water flow
Sim1	22 bar	1368 l/h
Sim2	22 bar	6840 l/h
Sim3	25 bar	3420 l/h
Sim4	18 bar	3420 l/h

Table 3 – Simulation cases (inlet water temperature 300 K).

Results for case Sim2 are given in Fig. 10 to 13. In case Sim2, different stagnation areas can be located (Fig. 10). The first one (0a) is not a problem due to the low level of heat flux in this area. Stagnation areas 0b and 0c are more problematic because the heat flux is maximal on the leading edge of the probe. Guide vane #4 and #5 are also concerned by stagnation area (0d & 0e) but the biggest one is 0f just before the water outlet. The water temperature stays between 293 and 330 K with a maximal value located in the middle of the stagnation area 0f (Fig. 11). Stagnation areas 0b and 0c have a weak impact on the global water temperature due to the small area of contact. The problem is more related to the wall temperature in order to avoid any water boiling. Considering the water pressure and the maximal internal wall temperature of 446 K (Fig. 12), the risks of boiling are low. Even on the side of the probe where stagnation area 0f is located, the temperature doesn't exceed the boiling temperature of water at 22 bars which is around 489 K. The external wall temperature of the probe is given in Fig. 13. The temperature of the leading edge can reach 662 K. Even if all the external skin of the probe is made with the same material the difference of temperature between the leading edge and the side area would probably imply a high level of stress due to differential thermal expansion of the two parts.

The pressure loss in the test case Sim2 is given in table 4 where total inlet pressure in the probe is considered equal to 100 % and total outlet pressure is equal to 0 %. Majority of the pressure loss is located in the 180° elbow between third and fourth guide vane.

Area involved	Water inlet total pressure
Inlet of the probe	100 %
At first guide vane	87.2 %
At second guide vane	85.7 %
After the 180° elbow	32.3 %
At fifth guide vane	31.5 %
Outlet of the probe	0 %

Table 4 – Pressure loss in the probe for Sim2.

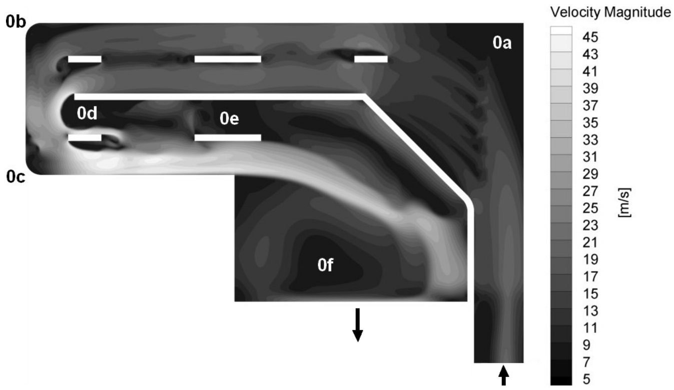

Figure 10 – Velocity magnitude of water inside probe in case Sim2.

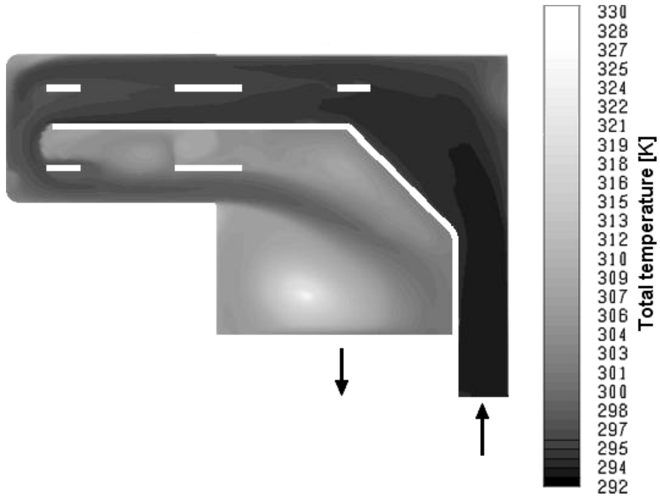

Figure 11 – Total temperature of water inside probe in case Sim2.

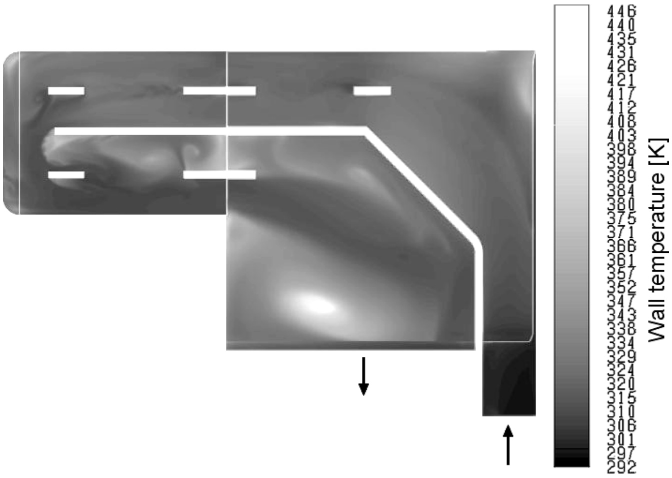

Figure 12 – Internal wall temperature of probe in case Sim2.

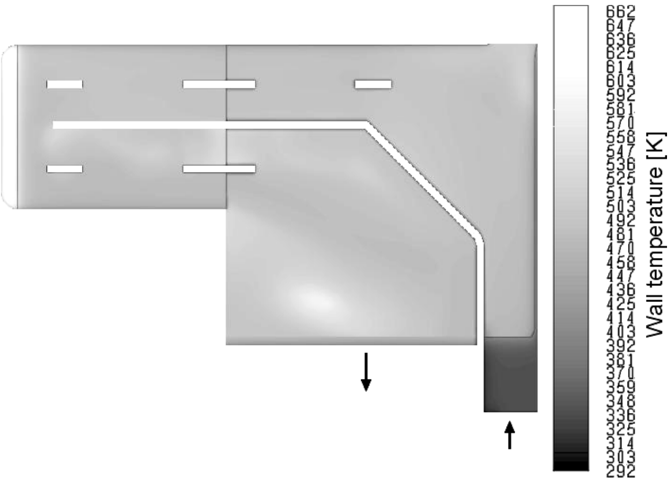

Figure 13 – External wall temperature of probe in case Sim2.

Assessment of mechanical constraints

From the temperature field computed with CFD code, a mechanical study has been performed on the probe with a thermal loading and a differential pressure of 15 bars between water inside the probe and combustion gas outside the probe. Even if some areas (S1 and S2 on Fig. 14) are highly stressed, the resistance of the probe has been confirmed because the safe load has not been exceeded. We can assume that deformation will exceed the elastic limit of the material so definitive deformations will be probably observed especially in area D1 and D2 (Fig. 15). The definitive position of each immersion can be optimized regarding the deformation in hot test condition. Consequently, the radial positions of the five immersions in cold condition are smaller than the specifications in order to compensate the expansion of the probe during hot test conditions.

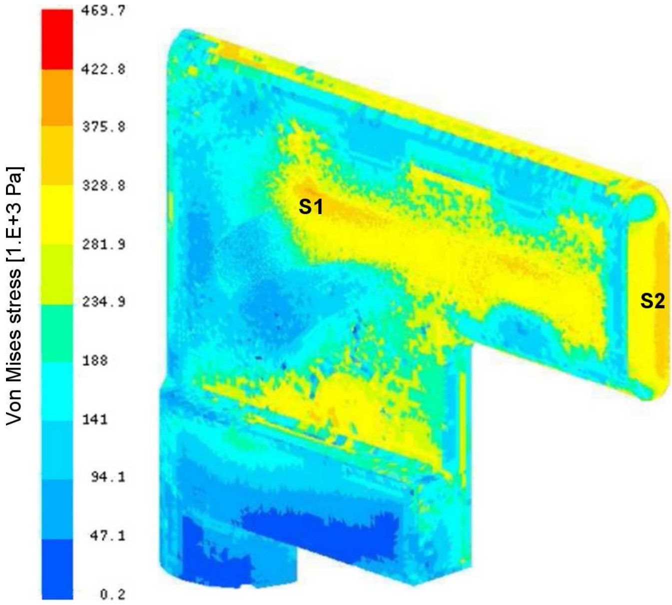

Figure 14 – Von Mises stress on the probe due to thermal expansion and internal water pressure.

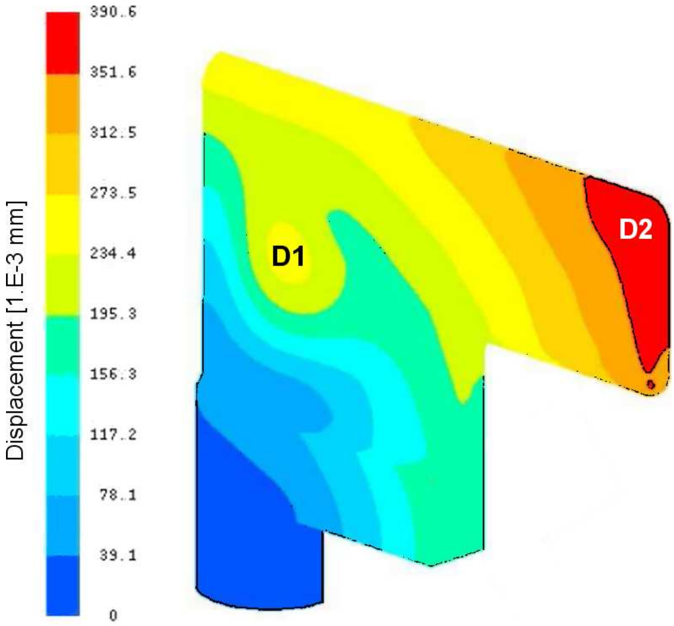

Figure 15 – Maximal displacement of the probe at design point.

INTERNAL COOLING OPTIMIZATION

Description of the test case

It has been demonstrated that the preliminary design is adapted to the test condition for the NEWAC project. Nevertheless, some improvements can be obtained regarding some stagnation areas inside the probe and its wall cooling. Three strategies have been defined (Fig. 16):

(1) Improving the shape of the guide vanes.
(2) Putting additional guide vane in the leading edge.
(3) Optimizing the position of guide vane in the lower part.

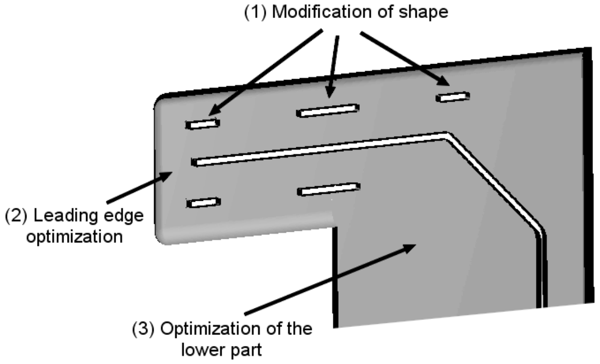

Figure 16 – Strategy of optimization on the preliminary design.

Five different designs have been tested on Sim2. Velocity fields of Sim2 are shown on Fig. 17 to 21. Leading edge and sidewall temperature changes for each design using Sim2 are shown in Tables 5 and 6.

Design #1 helps us to fight against the stagnation area 0d and especially 0e of Fig. 10 but with a negative impact on the temperature on area 1a (Fig. 17) especially in Sim3 with a raise of the internal wall temperature of +15%. This negative impact totally disappears if the mass flow is higher like in Sim2. Comparing with Design #0, the maximal internal wall temperature on leading edge also increases in Sim3 (+3 %).

Design #2 brings an improvement on the temperature of side area (2a & 2b) especially for Sim3 where internal highest temperature decreases by 5% compared to initial design. This is also the best configuration for Sim3 regarding the temperature on the leading edge (internal wall: –5 % & external wall: –2 %).

Results on designs #3 and #4 are quite similar. Shaped guide vanes increase the homogeneity of the flow in the upper part of the probe (4a) with no significant impact on its cooling.

Figure 21 shows one of the unusual designs that have been tested in this study with no success. This design led to the highest internal wall temperature on the leading edge due to the stagnation area 5a.

Concerning the temperature on the leading edge (Table 5), designs from 1 to 4 have a weak impact on it excepting for design #5. The improvement of the velocity field on the 180° return part for design 1 to 4 only implies a decrease in the stagnation area 0d of Fig. 10.

The most impressive improvements are obtained on the side area (0f on Fig. 10). Wall temperature decreases from 4% to 15% inside or outside the probe excepting for design #5 which is clearly not adapted.

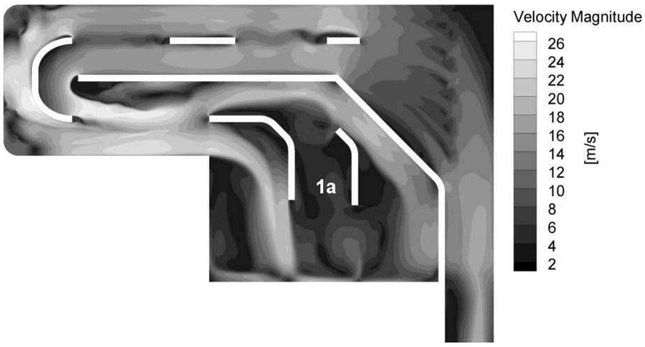

Figure 17 – Velocity magnitude of water inside the probe in case Sim2 on design #1.

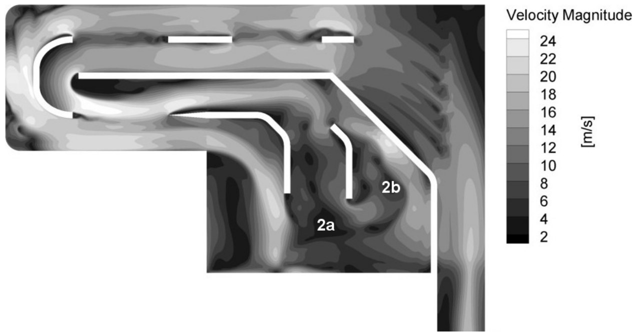

Figure 18 – Velocity magnitude of water inside the probe in case Sim2 on design #2.

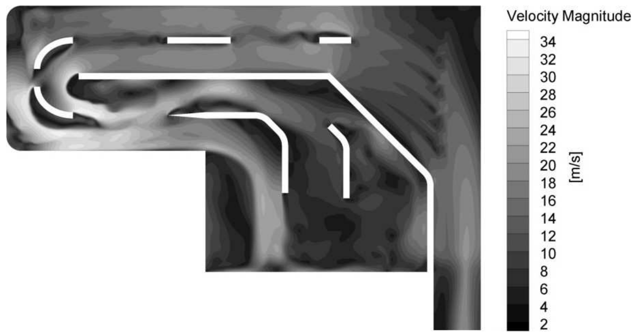

Figure 19 – Velocity magnitude of water inside the probe in case Sim2 on design #3.

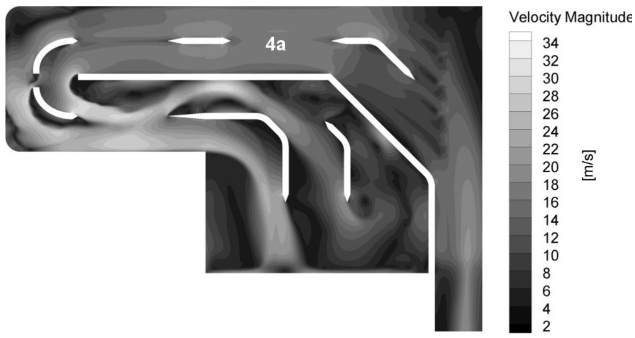

Figure 20 – Velocity magnitude of water inside the probe in case Sim2 on design #4.

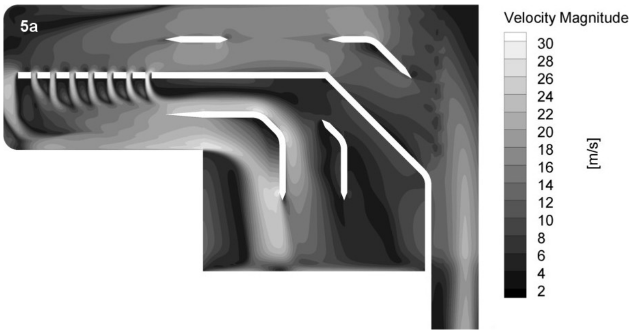

Figure 21 – Velocity magnitude of water inside the probe in case Sim2 on design #5.

Design	Leading edge temperature (inside probe)	Leading edge temperature (outside probe)
#1	+1 %	+1 %
#2	+1 %	0 %
#3	–1 %	+2 %
#4	+1 %	+2 %
#5	+17 %	+9 %

Table 5 – Leading edge maximal temperature on Sim2 for optimized design compared with maximal temperature on the same area on initial design.

Design	Side area wall temperature (inside probe)	Side area wall temperature (outside probe)
#1	–11 %	–14 %
#2	–6 %	–15 %
#3	–8 %	–15 %
#4	–4 %	–14 %
#5	+10 %	–8 %

Table 6 – Side area maximal temperature on Sim2 for optimized design compared with maximal temperature on the same area on initial design.

Optimal design definition

From a thermal point of view, the most promising design is #3. Design #4 has better pressure loss with no significant improvements on the cooling of the probe. Sharp shape on each guide vane (design #4) could have a significant impact on the price of the probe for negligible improvement and can be definitively discarded. Due to hard constraints on the production schedule and on the manufacturing of the internal part of the probe (sharp edge, vertical guide vane, 180° round elbow), the final design focused on a new solution inspired from design #3 and given by Fig. 22 with a round shape on 6a and a deflection plate on 6b in order to lower the temperature on the side area of the probe.

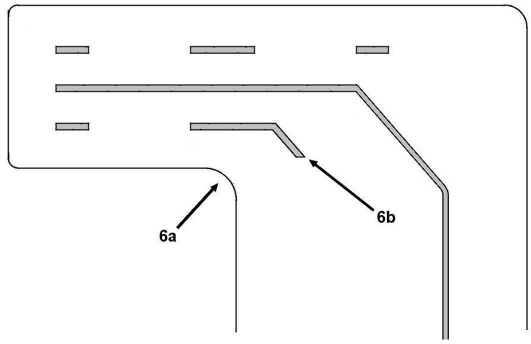

Figure 22 – Final optimized design for the probe.

During functional checks before the installation of the probe on the traverse gear, a quick comparison between the old design (fig. 9) and the new one (fig. 22) has been made on the K11 facility with the maximal water flow in order to assess the pressure loss of the two probes. Some extension lines have been added on the water system of the traverse gear according to the figure 23 in order to allow the use of a flow meter and some pressure plugs for testing the probes. Even if a complete numerical study of this new design hasn't been made due to limited computing resources, preliminary observations on the finished probes are promising. An old design probe has been put on the water extension line equipped with a flow meter and two pressure sensors (P_{in} and P_{out}) in order to quantify its maximal water-cooling flow and its pressure loss. Then, the new design probe has been tested during these functional tests and it has demonstrated a better maximal water flow (+11 %) and a better pressure loss (–22 %) compared with the old design considered as the reference.

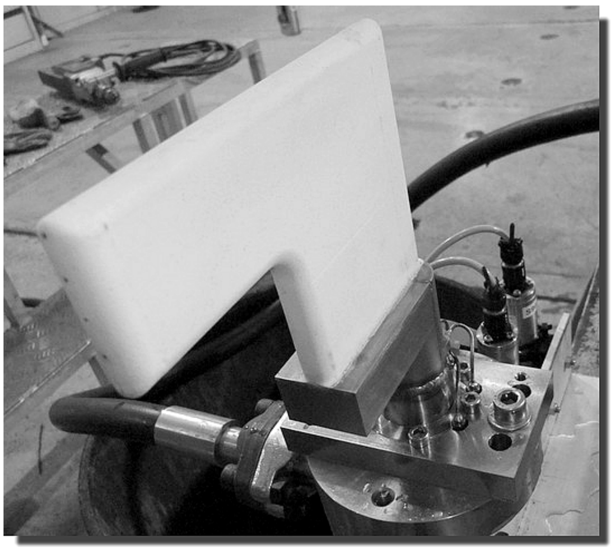

Figure 22 – NEWAC gas analysis probe during functional tests of the water cooling system. White part of the probe is a thermal barrier coating added after this study for specific test purpose.

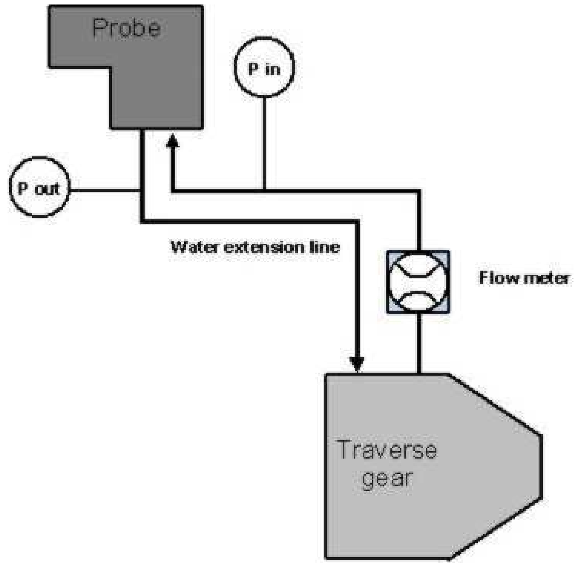

Figure 23 – Scheme of the experimental setup used for preliminary functional tests on the gas analysis probes.

CONCLUSION

This paper describes the design and the optimization from a thermal and structural point of view of a high temperature water cooled probe for gas analysis measurements on a high performance combustion test rig. The first part describes the environmental constraints around the probe: interfaces with water system and gas analysis sampling lines, interference with the combustor, heat flux at design point. Calculation of the cooling performances has been performed on a preliminary design in order to obtain the temperature of the probe in steady state conditions. Then, temperatures of the walls in different areas of the probe have been used in a structural analysis to check the mechanical resistance of the probe and its distortions. Finally, parametric studies to optimize the internal design of the probe have been performed in order to improve the cooling flow pattern. Manufacturing constraints led to a new design that minimizes the pressure loss (-22 % compared to the old design) and guarantees a correct water cooling at temperature up to 2123 K with no significant impact on the manufacturing complexity of the probe.

ACKNOWLEDGMENTS

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This work has been also supported by DGA Aeronautical Systems (Toulouse) for complex thermo-mechanical computation and expertise.

NOMENCLATURE

Latin letters

Nu	=	Nusselt number [-]
Pr	=	Prandlt number [-]
Re	=	Reynolds number [-]
Tu	=	Turbulent intensity [-]

Greek letters

θ	=	angular position [degree]

Subscript

LV	=	related to Lowery and Vachon correlation

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