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THEORETICAL AND EXPERIMENTAL THERMAL ANALYSIS OF X-BAND ACCELERATING STRUCTURES FOR SPARC

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Abstract

This paper presents the study of the thermal behavior of RF cavities operating at the frequency of 11.424 GHz, for linearizing the longitudinal phase space in the Frascati Linac Coherent Light Source (SPARC). Two different structures have been analyzed, the first one is a 9-cell cavity operating on a π standing wave mode, and the second one is bi-periodic accelerating section operating on the π/2 standing wave mode. Numerical simulations have been carried out with SUPERFISH and ANSYS codes [1] on both structures considering different constructive solutions. The use of a different material for some parts of the structure has been proposed, studied and compared to the case of a structure totally made out of copper. A solution for a cooling system has been studied and simulated and some experimental tests have been prepared.

1 INTRODUCTION

A possible schematic layout to be used for enhancing the bunch compression performances for the X-FEL project to be constructed at the Frascati Laboratories is reported in Fig. 1.

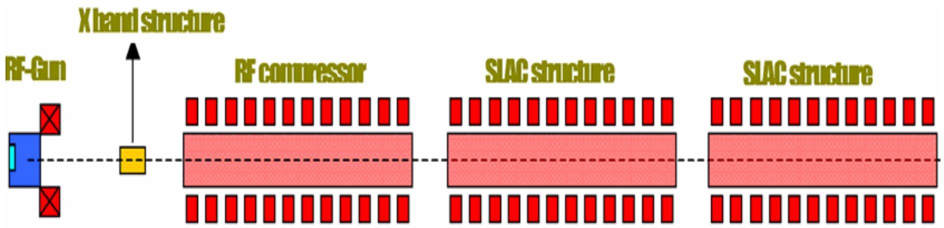

Figure 1

It is mainly constituted by a photo-cathode RF gun which works in the S-Band at a frequency F=2.856 GHz and delivers bunches with a 5 MeV kinetic energy, 1 nC charge and 10 psec bunch length; an X-Band accelerating structure with frequency equal to the fourth harmonic of the S-Band frequency; a travelling wave velocity buncher which works at a frequency F=2.856 GHz; two $2\pi/3$ travelling wave constant gradient structures operating at F=2.856 GHz.

The TW accelerating sections induce a chirp of energy in the bunch that remains even after the compression. The bunch will therefore present a non-linear energy-time correlation, for this reason the compression will not take place properly and this might lead to some instabilities in the beam dynamics. For compensating the non-linearity distortions due to the RF curvature of the 2.856 GHz accelerating cavities, the use of a higher harmonic standing wave structure operating at 11.424 GHz is required. The choice of using an X-band electron accelerating structure is dictated mainly by the poor availability of space, and therefore by the need to use a small and compact structure. But using X-band accelerating cavities presents also other advantages such as higher shunt impedance, higher breakdown threshold level and short filling time. On the other hand it has to be considered that RF stability during operation and tuning tolerances are crucial points for the RF structure design in the high frequency range. The complexity of machining, tight mechanical tolerances and alignments, are therefore important aspects which have to be taken into account in the design activity. For this same reason it is important to study and, if necessary to modify and keep under control the thermal behavior of the structure. In fact a rise in temperature will vary the accelerator dimensions and the frequency characteristic will change accordingly.

The main objective of the work described in this paper is to analyze the temperature distribution in the structure in operating conditions and to estimate the frequency change caused by a change in temperature. We will assume that a closed cooling water system is used in order to keep the operating temperature on the external surface of the structure at 40°C.

We will study two different cavities and we will analyze the differences in their thermal behavior. We will also consider the possibility of using a different material, Molybdenum, for the irises of the structure.

2 RF CHARACTERISTICS AND GEOMETRIC DESCRIPTION

The global RF requirements for an accelerating SW structure are summarized in the following:

1) high accelerating field gradient to reduce the accelerator length;
2) high shunt impedance to reduce the requirement of RF power;
3) low ratios E_p/E_0 and B_p/E_0 (where E_p and B_p represent the surface peak electric and magnetic fields respectively and E_0 is the average accelerating field) to reduce dark currents, break down conditions and thermal effects;
4) high group velocity in order to be less sensitive to the mechanical fabrications errors;
5) low content of longitudinal and transverse higher order modes that can affect the beam dynamics;
6) appropriate shape profile to avoid multipactoring phenomena that could limit the accelerating section performances.

In the SPARC project the structure must have the requirements reported below

1) accelerating voltage V = 5 MV;
2) length L = 12 cm;
3) beam aperture hole φ = 8 mm;
4) operating frequency f_{CAV} = 11.424 GHz;

with the following beam parameters:

1) pulse charge, Q = 1 nC;
2) full pulse length, τ = 10 psec;
3) single pulse operation;
4) pulse repetition rate frequency, f_{REP} = 50 Hz.

On the basis of these considerations two different structures have been designed, simulated and realized [2,3,4] at the LNF in Frascati and they are the object of the thermal analysis described in this paper.

The first one is a periodic accelerating cavity operating on a π mode. It is made of 9 cells axially coupled and each one separated from the next by an iris. The cells are all exactly identical (radius R_c= 1.054 cm, axial length L_c= 1.112cm) except for the two end cells that have been slightly reduced (R_e= 1.047 cm) in order to obtain field-flatness. The irises have a radius R_i= 0.4 cm and a length L_i= 0.2 cm. Two beam tubes with a radius R_t=R_i= 0.4 cm are coupled respectively to the first cell and to the last cell, in order to guide the beam in and out of the cavity. The other structure that was considered is a bi-periodic cavity operating on a $\pi/2$ mode. This structure is quite similar to the first, with the exception that the 9 large accelerating cells are separated not only by irises, but also by smaller coupling cells. The whole structure is therefore made of 9 large cells and 8 small cells. The radius of the smaller coupling cells has been increased a bit (R_s= 1.172cm) in order to close the so-called stop-band.

The main advantage of the bi-periodic structure is the reduced sensitivity of the field-flatness and of the resonant frequency with respect to mechanical errors, cell-to-cell temperature variation and assembly errors. Nevertheless the process of tuning is more difficult because it is necessary to close the stop-band. On the other hand, one of the main advantages of the structure operating on the π mode is that, having a simpler geometry, it is cheaper and it allows an easier construction with satisfactory mechanical tolerances.

3 ELECTROMAGNETIC ANALYSIS AND RF PARAMETERS

As we have seen in paragraph 2, one of the most important requirements for an RF cavity is to have a high accelerating field. Unfortunately an upper limit for the axial electric field is given by the maximum value of surface field possible before breakdown. Experimental tests prove that a possible way to raise this breakdown limit is to use different materials for the inner surface of the structure containing the cavity. For this reason we have decided to consider, study and compare three different constructive solutions:

1) the classical solution with the structure totally made out of copper;
2) the solution with irises made out of Molybdenum;
3) mixed solution, with only half of the irises (the nose) made out of Mo, and the outer half made out of Cu;

An electromagnetic analysis of the π mode operating structure has been performed for each one of the three cases above with the SUPERFISH numerical code, and the main RF parameters are shown below.

RF parameters for Copper structure

	20°C	**40°C**	**60°C**
Frequency [MHz]	11424.65	11424.65	11424.65
E_0 [MV/m] (average)	69	69	69
Transit-time factor	0.765	0.765	0.765
Stored energy [joules/m]	7.2	7.2	7.2
Power dissipation [W]	692.3	719.00	744.74
Q	8814.27	8487.03	8193.74
R_{sh}*T^2 [MOhm/m]	80.75	77.75	75.07
R_{sh} *T^2/Q [MOhm/m]	9161.37	9161.37	9161.37
Duty cycle, D.C.	10^{-4}	10^{-4}	10^{-4}

RF parameters for structure with Molybdenum irises

	20°C	**40°C**	**60°C**
Frequency [MHz]	11424.65	11424.65	11424.65
E_0 [MV/m] (average)	69	69	69
Transit-time factor	0.765	0.765	0.765
Stored energy [joules/m]	7.2	7.2	7.2
Power dissipation [W]	942.77	979.52	1014.91
Q	6472.60	6229.77	6012.54
R_{sh} *T^2 [MOhm/m]	59.30	57.07	55.08
R_{sh} *T^2/Q [MOhm/m]	9161.37	9161.37	9161.37
Duty cycle, D.C.	10^{-4}	10^{-4}	10^{-4}

RF parameters for structure with mixed Molybdenum Copper irises

	20°C	**40°C**	**60°C**
Frequency [MHz]	11424.65	11424.65	11424.65
E_0 [MV/m] (average)	69	69	69
Transit-time factor	0.765	0.765	0.765
Stored energy [joules/m]	7.2	7.2	7.2
Power dissipation [W]	734.50	762.89	790.25
Q	8307.92	7998.78	7721.81
R_{sh} *T^2 [MOhm/m]	76.11	73.28	70.74
R_{sh} *T^2/Q [MOhm/m]	9161.37	9161.37	9161.37
Duty cycle, D.C.	10^{-4}	10^{-4}	10^{-4}

4 THERMAL SIMULATIONS

The use of Molybdenum surely brings advantages in terms of the electromagnetic behavior of the cavity since, as we have said, it allows to operate with higher accelerating fields. It is also true, on the other hand, that molybdenum has a lower value of both electric and thermal conductivity. This will obviously mean that by using molybdenum we will have greater power losses due to higher surface resistance, and a more difficult heat evacuation because of the lower rate of heat transfer. In accordance with this we expect to have higher temperature gradients, and therefore greater thermal expansion, which means also greater frequency variations. This fact implies the necessity to carry out thermal analyses on these structures containing molybdenum, in order to keep under control the temperature distributions, and compare them with the data referred to the classical copper structure.

Numerical simulations performed with ANSYS show that the maximum temperature variation within the structure, in the classical solution (i.e. with a structure totally made out of copper) is less than 2°C as we can see in Fig.2 In our thermal simulations we have not considered the entire cavity, but only one full and one half cell, applying all the necessary symmetry conditions. This has obviously been done in order to reduce and simplify the model to be analyzed with the numerical code.

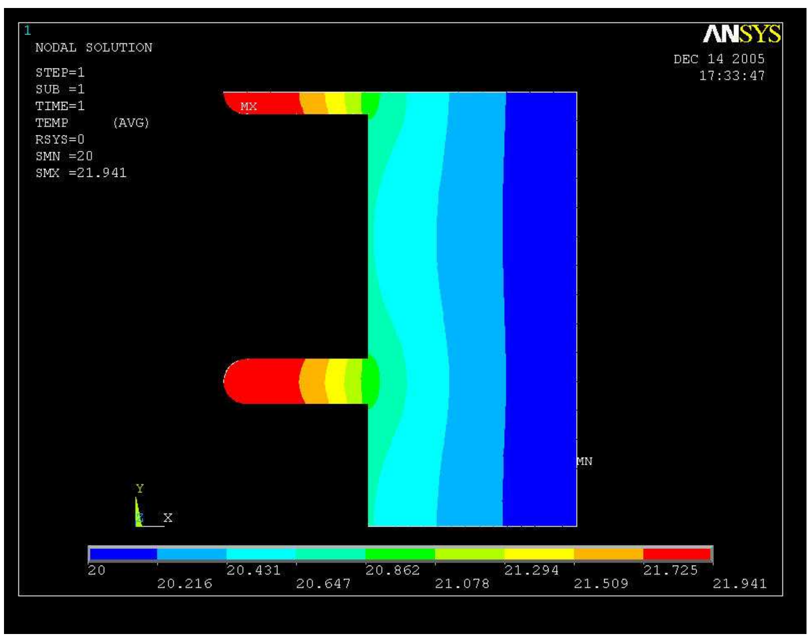

Figure 2

The maximum temperature variation goes up to more than 7°C in the case of using molybdenum for the entire irises as shown in fig. 3

The mixed copper-molybdenum solution appears to be the best since it has both the advantages of allowing higher accelerating fields and small temperature variations. The highest temperature reached in this type of structure in operating conditions, supposing to fix the outer temperature at 20°C, is 22.627°C.

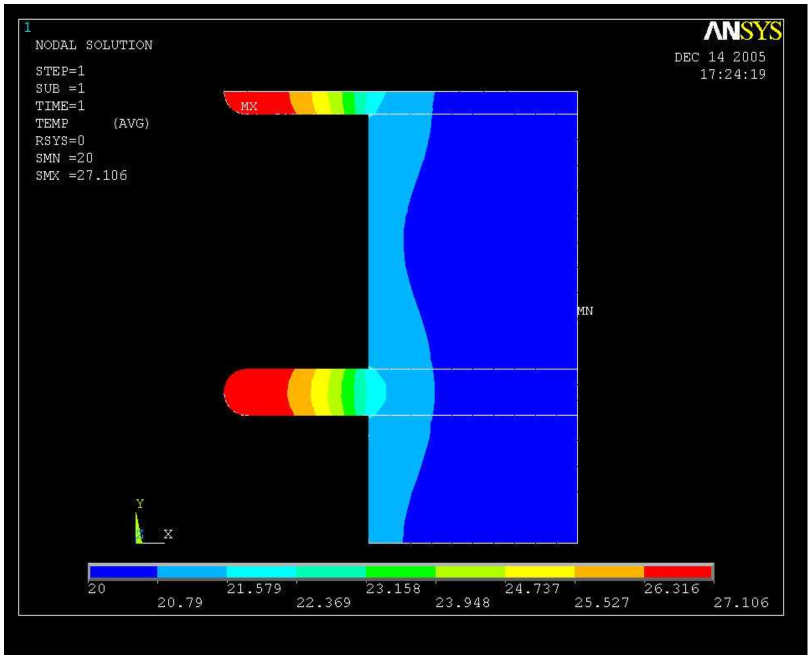

Figure 3

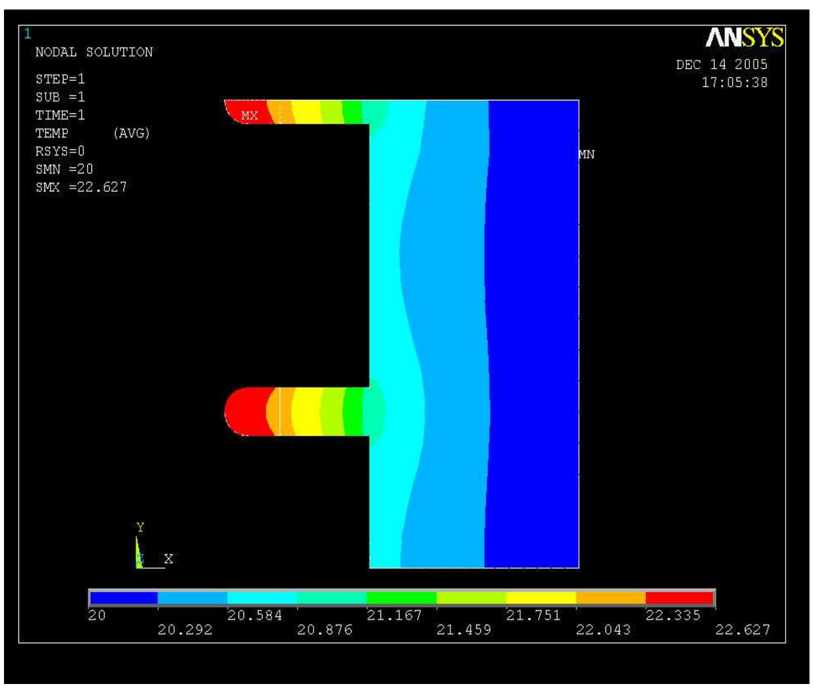

Figure 4

5 THE COOLING SYSTEM

In order to materially obtain the temperature boundary condition on the external surface of the structure, which in our previous considerations we have assumed as a fixed temperature value, we need to supply the accelerating structure with a cooling system. This will be a closed water device with 4 tubes, parallel to the axis of the cavity, positioned on the outer surface of the structure. Two different duty cycles and two different angular arrangements of the tubes have been considered and simulated with ANSYS. The two duty cycles represent the different levels of power which the structure may be supplied with, and the analysis of different angular positions for the tubes allows us to understand whether this parameter is strongly influent, or not, on the temperature distribution, and therefore to decide for the best solution according also to the availability of space around the structure. The results, as we can see in figures 5,6,7 and 8, show that the temperature distributions do not differ much from the previous analyses obtained with the approximation of considering a constant temperature on the whole external surface.

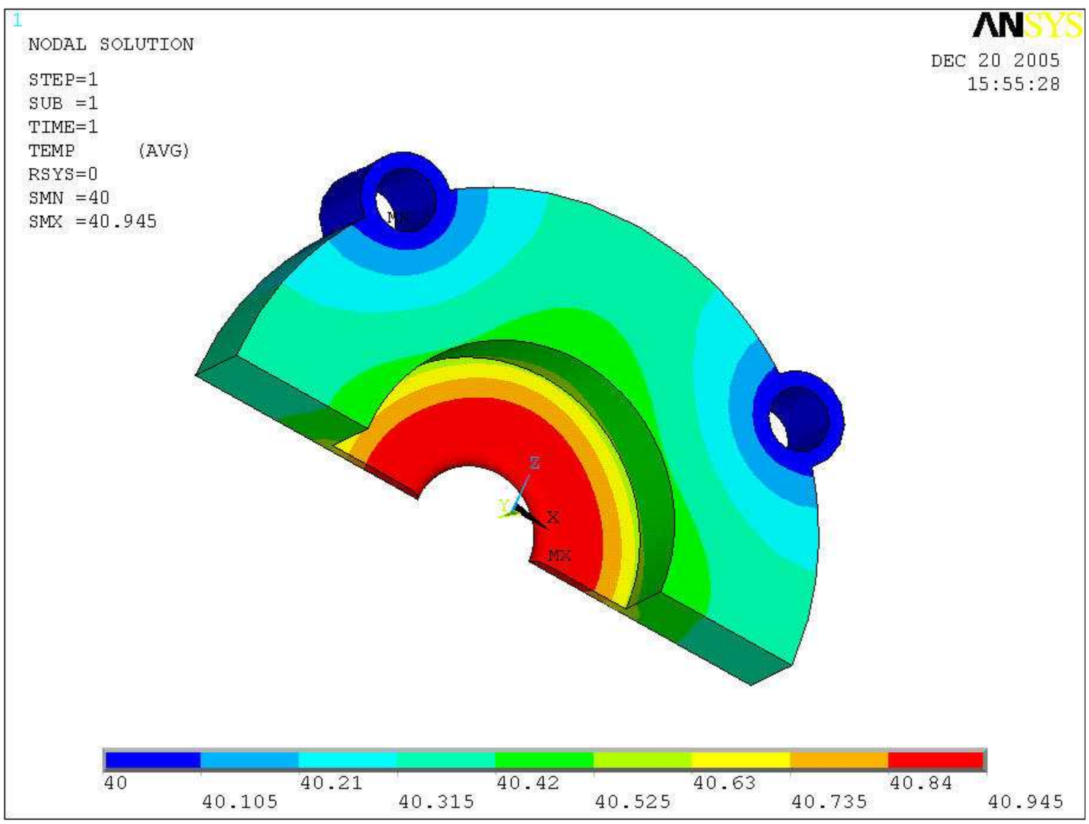

Figure 5: π mode cavity with tubes at 45° working on duty cycle 1 (1E-4)

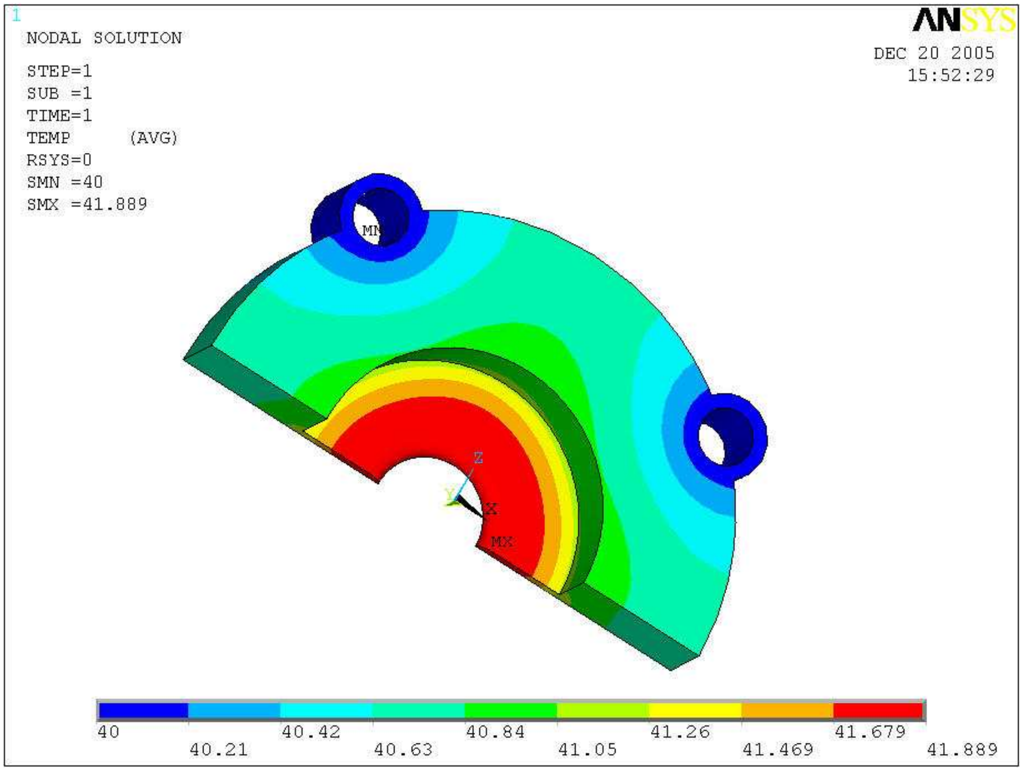

Figure 6: π mode cavity with tubes at 45° working on duty cycle 2 (5E-5)

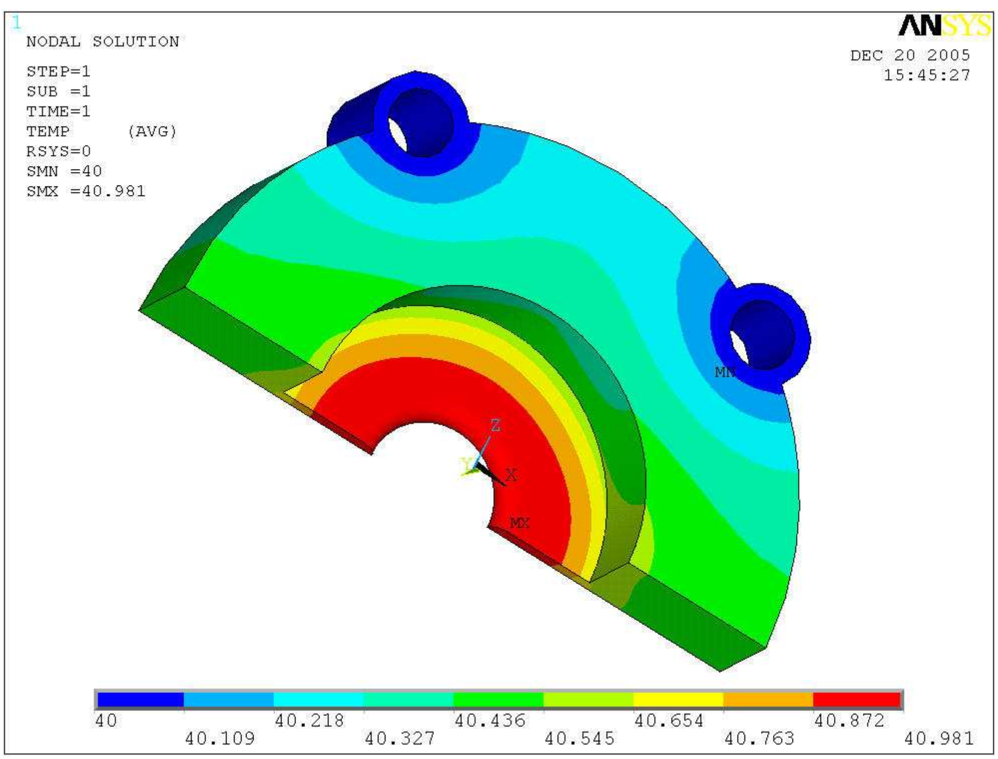

Figure 7: π mode cavity with tubes at 30° working on duty cycle 1 (1E-4)

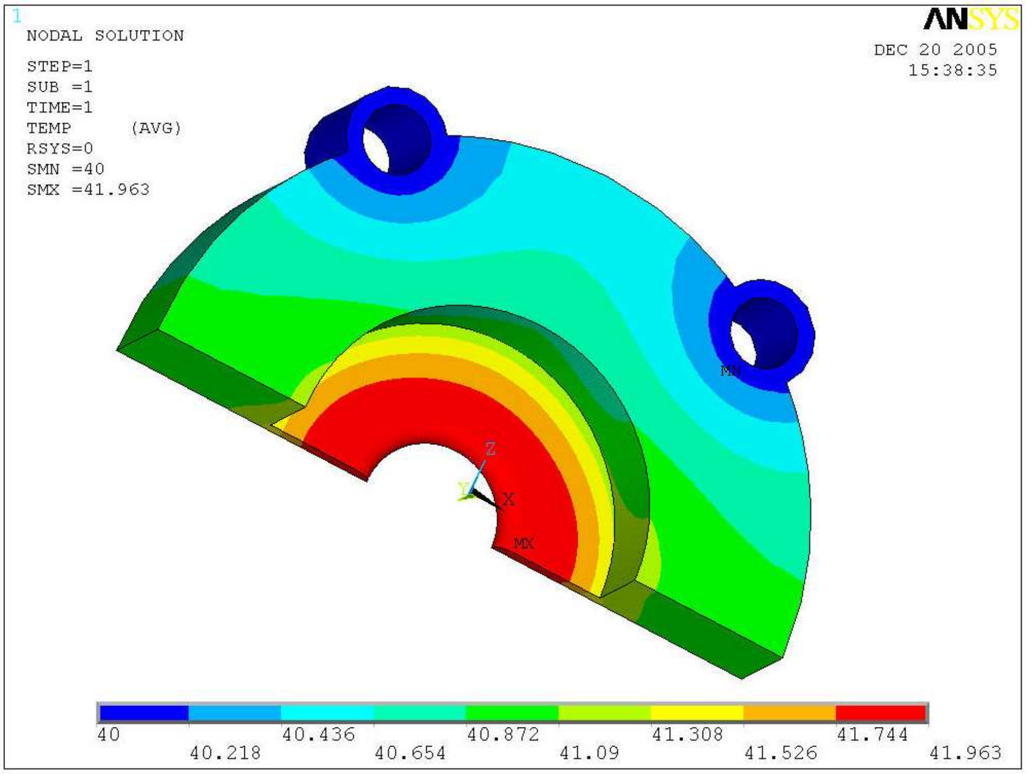

Figure 8: π mode cavity with tubes at 30° working on duty cycle 2 (5E-5)

Analogous simulations have also been performed on the biperiodic structure. The results, as we can see in figures 9,10,11 and 12, show that the thermal behavior of this accelerating cavity is very similar to the one of the π mode structure previously studied. The highest temperature value obtained, even in the most critical conditions (duty cycle 5E-5 with tubes at 30°), is not greater than 42°C.

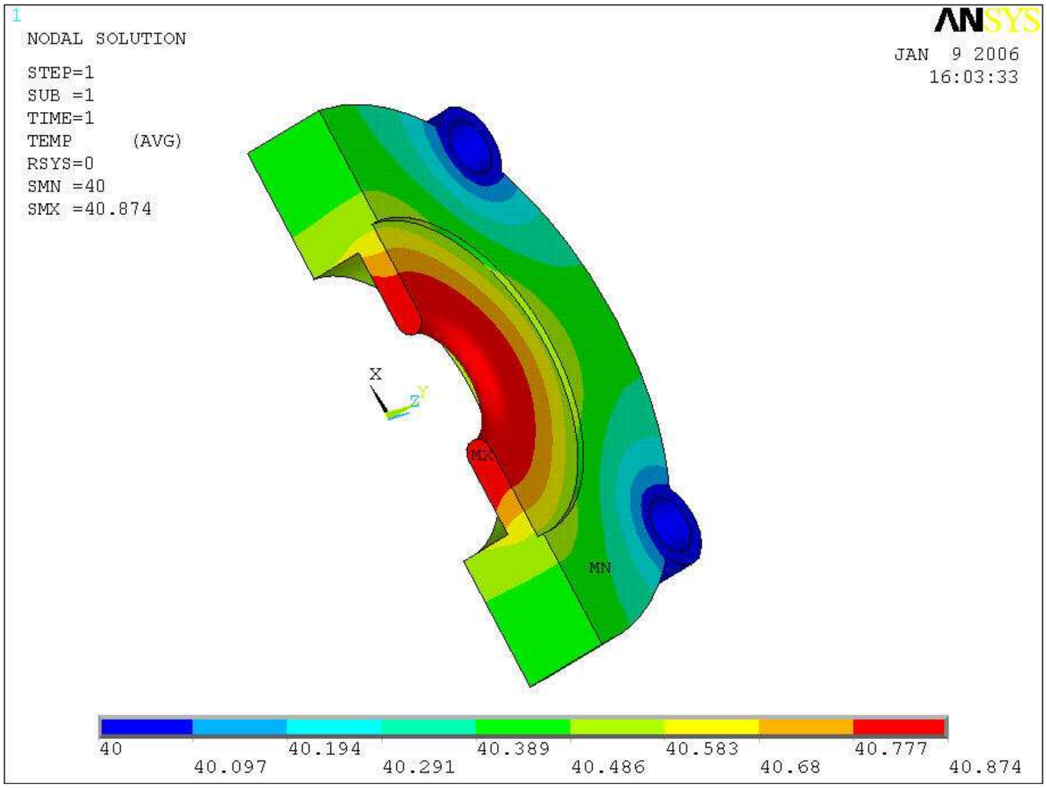

Figure 9: $\pi/2$ mode cavity with tubes at 45° working on duty cycle 1 (1E-4)

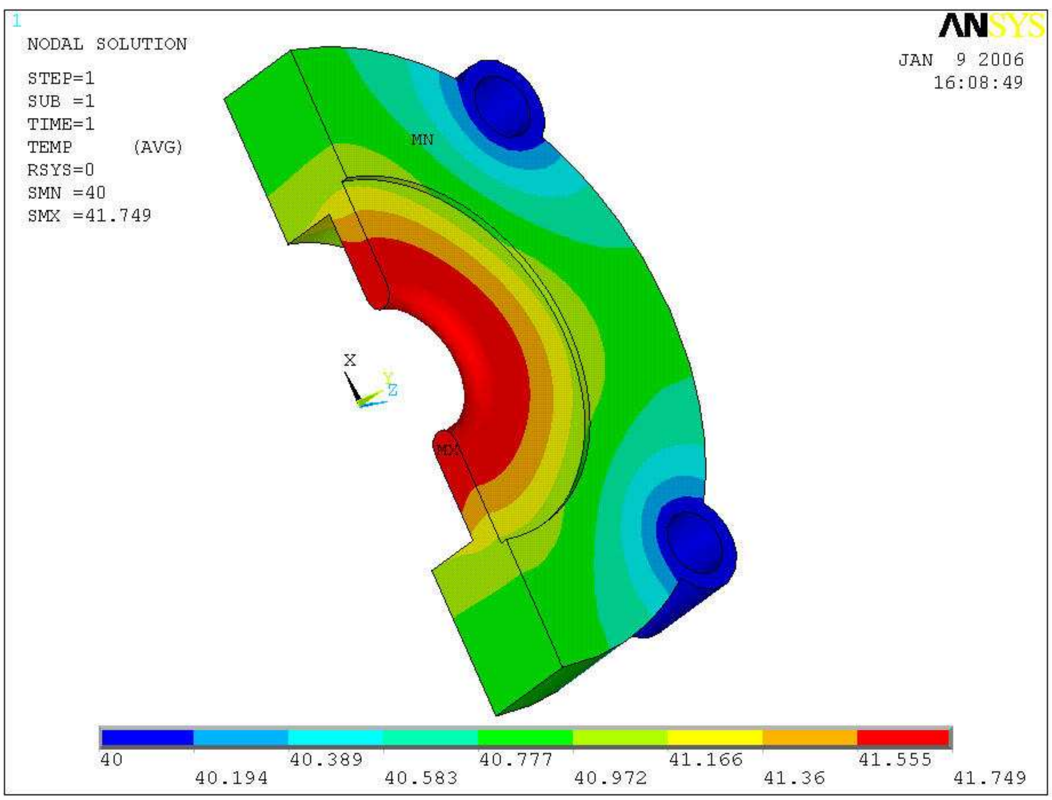

Figure 10: $\pi/2$ mode cavity with tubes at 45° working on duty cycle 2 (5E-5)

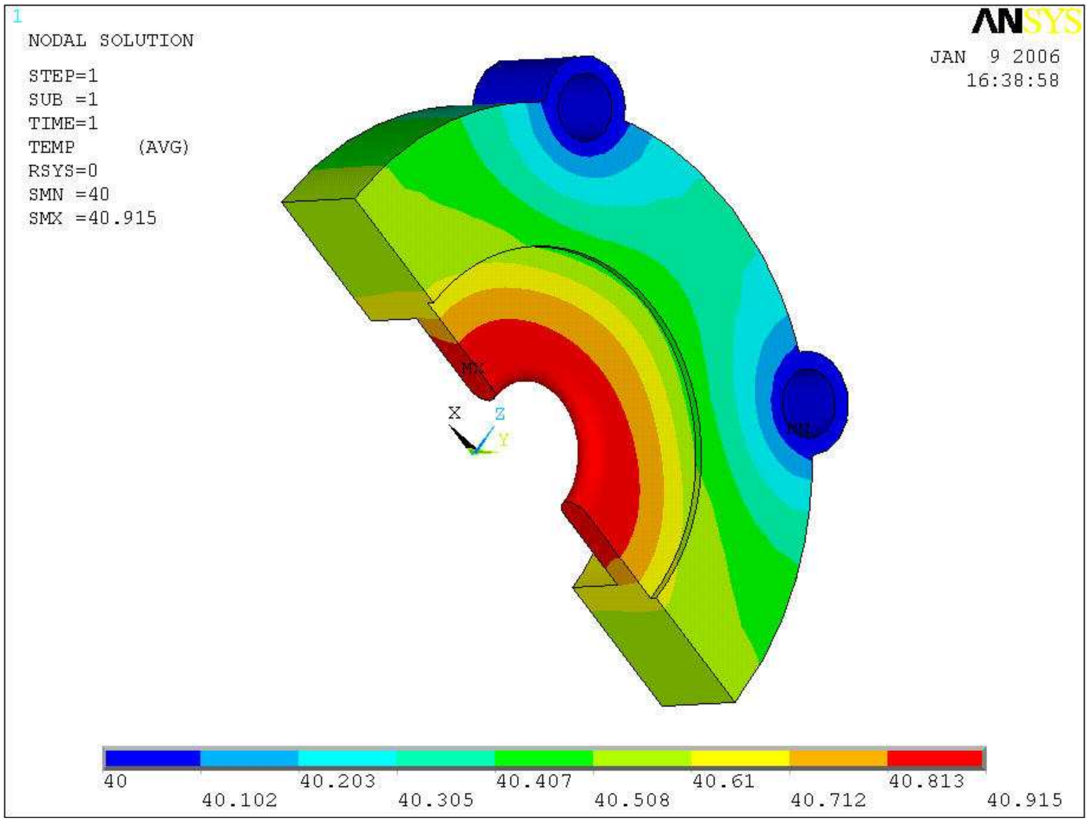

Figure 11: π/2 mode cavity with tubes at 30° working on duty cycle 1 (1E-4)

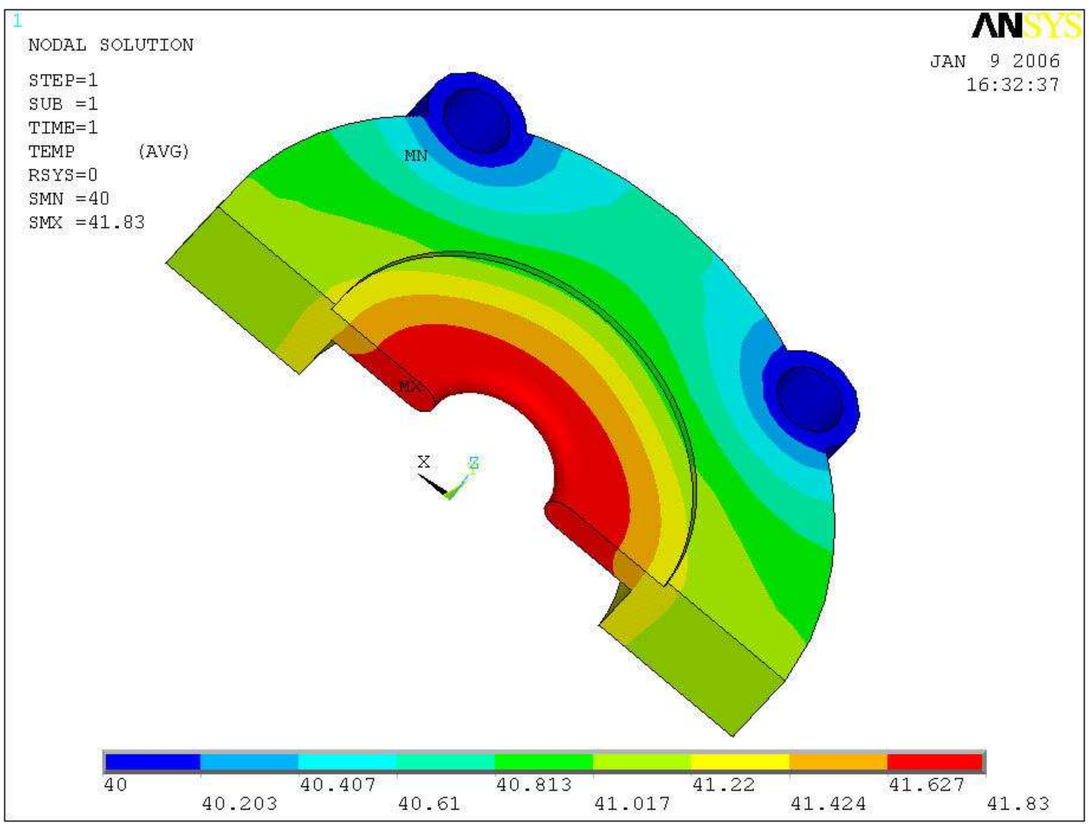

Figure 12: π/2 mode cavity with tubes at 30° working on duty cycle 2 (5E-5)

6 COOLING SYSTEM CALCULATIONS

Our previous thermal analyses performed with the numerical code ANSYS show that the use of a cooling system made up of four axial tubes with 6mm inner diameter should be able to keep the maximum rise of temperature in the structure lower than 2°C. In those simulations we have considered a steady value for the wall temperature of the tubes. By doing so we have approximated the cooling system as a perfect heat absorber. Actually it is not so, and it is necessary and important to calculate the needed speed and temperature values for the water, in order to have a good efficiency for the cooling system. A first approximation of the water flow rate may be given by the following equation

$$W = P \cdot \delta \cdot c \cdot \Delta T \qquad (1)$$

in which W is the heat transferred to the water per unit time, P is the volumetric water flow rate, δ is the water density, c is the specific heat and ΔT is the difference of temperature of the water from the beginning to the end of the tubes. With equation (1), knowing how much heat per second is transferred to the cooling system, knowing the main physical parameters of water (tab.3), and choosing a value of maximum raise of temperature acceptable it is possible to establish an approximate value of the flow rate necessary.

c (specific heat)	4186 [J/(kg*K)]
δ (density)	1000 [kg/m^3]
λ (thermal conductivity)	0.63 [W/(m*K)]
μ (dynamic viscosity)	0.001 [Pa*s]

Table 1: physical properties for demineralized water

In our case the quite small dimensions of the structure to be cooled do not give the possibility of choosing bigger sizes for the tubes or a different number of tubes, therefore the flow rate will depend uniquely on the water speed. Once the speed of the water has been established we can evaluate the water temperature with the equation of convection heat transfer

$$W = h_c \cdot S \cdot (T_w - T_b) \qquad (2)$$

where h_c is the convection film coefficient, S is the total exchange surface, T_w is the wall temperature and T_b is the bulk temperature. The film coefficient is to be calculated with the dimensional analysis and with the help of empirical formulas. For the study of water flowing through pipes in turbulent regime the correlation to use is the Dittus-Boelter equation

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \quad (3)$$

in which Nu is the Nusselt number ($Nu = h_c \cdot D / \lambda$), Re is the Reynolds number ($Re = v \cdot D \cdot \delta / v$) and Pr is the Prandtl number ($Pr = c \cdot \mu / \lambda$).

Diagrams 1 and 2 show the values of the water temperature rising between inlet and outlet, and temperature difference between wall and bulk, corresponding to different values of water speed (diagram 1) and of water volumetric flow rate (diagram 2).

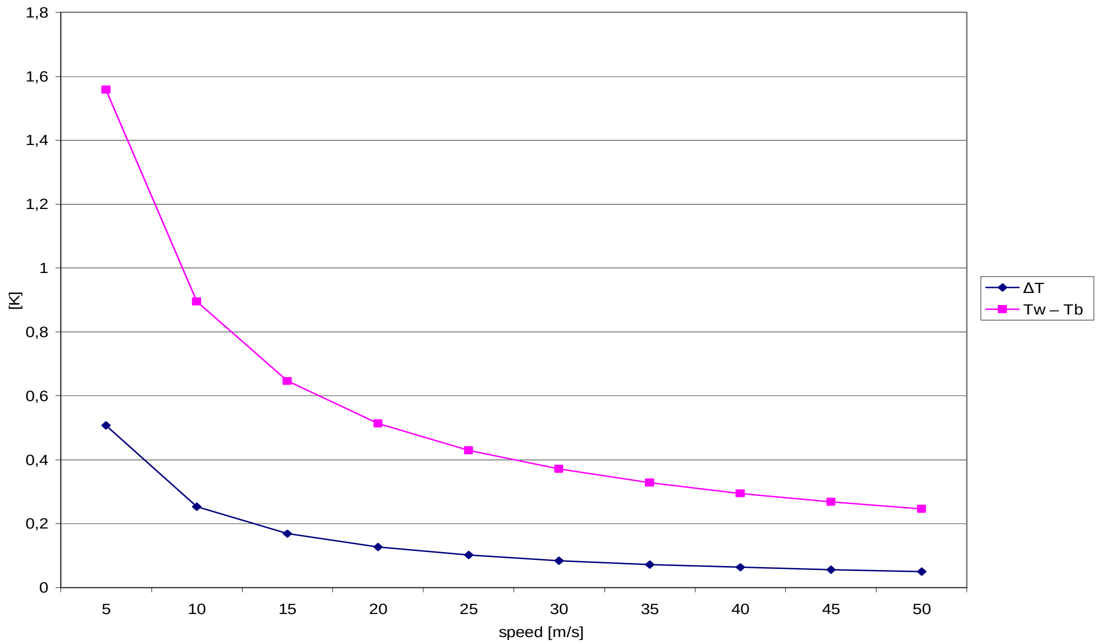

Diagram 1

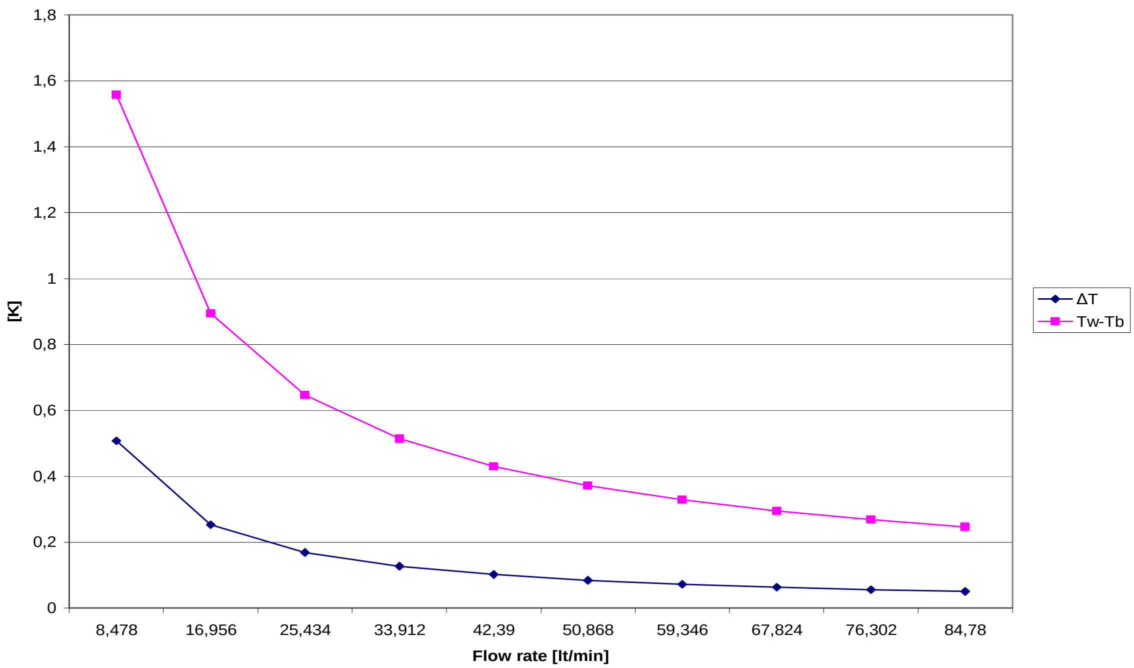

Diagram 2

With the help of empirical formulas it is possible to evaluate also the pressure loss through the pipes. We consider only the distributed losses since the concentrated losses may be neglected. The distributed pressure loss may be calculated with the following expression

$$\Delta p = \zeta \cdot L \cdot \delta \cdot v^2 / 2D \quad (4)$$

in which ζ is a dimensionless coefficient depending on the characteristics of the tube. For turbulent regime and smooth pipes ζ depends only on Re and it can be evaluated with the empirical formula

$$\zeta = 0.316 \cdot Re^{-0.25}$$

The results of our calculations are shown in the following diagram.

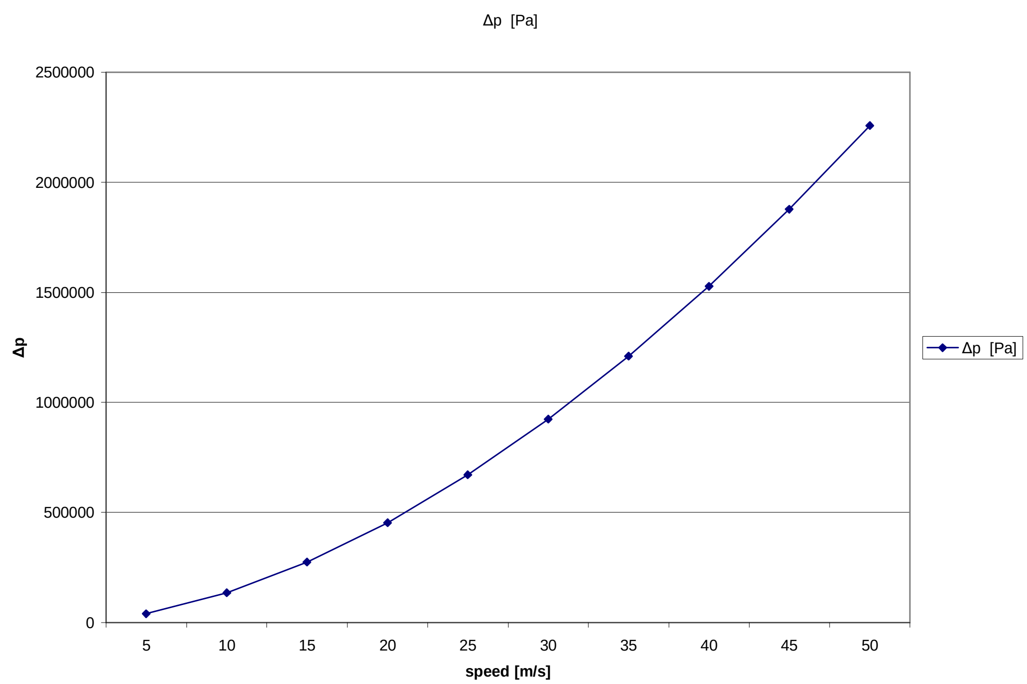

Diagram 3

7 EXPERIMENTAL TESTS

In fig.13 a five cells brazed structure is shown. It will be dedicated to temperature tests as a function of the power dissipated on the inner surface by the RF field. A Ni-Chrome heater, able to generate a power up to 400W, will be mounted on the axis of the structure in order to simulate the power produced by the RF field. A 1mm diameter hole has been drilled up to 2mm from the inner surface of the central iris. A thermocouple will be inserted in this hole. Another thermocouple will be connected to the external surface, where the cooling system is located. With this configuration the difference in temperature between the hottest point and the water can be measured and then compared to the calculated numbers.

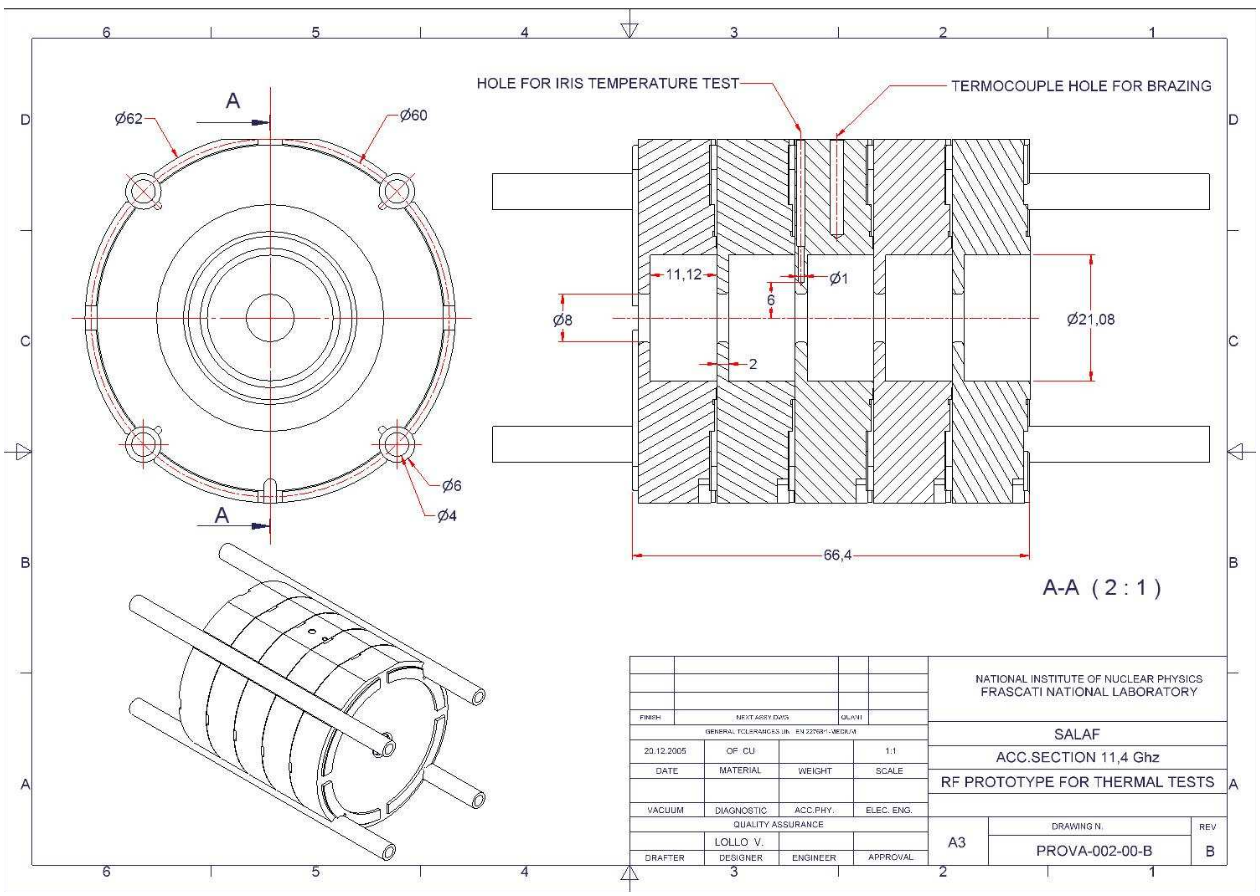

Figure 13

8 REFERENCES

[1] ANSYS is a trademark of SAS Inc. www.ansys.com.

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