STUDY OF THE COOLING FOR THE SPARC RF DEFLECTOR

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Abstract

The tight requests for mechanical stability of the RF deflector for the SPARC project has been accomplished by a careful design of its cooling. A FEM model linking fluido-dynamic, thermal and structural aspects has been realized. Procedures and results are here revised.

1. Introduction

The Project SPARC (Sorgente Pulsata e Amplificata di Radiazione Coerente) will be built at LNF. The aim of the project is to promote an R&D activity on the development of a coherent ultra-brilliant X-ray source.
The RF deflector is part of the diagnostic of the SPARC photo-injector and it is used to characterize the beam. The deflector consists in five cell, separated by irides.
Its working requires a high dimensional stability. In fact, the RF frequency has to be directly matched to the deflector resonance, depending on the volume of the cell cavity, because no active feedback will be used.
Aim of this work was to investigate the feasibility of the requirement from both the thermo-mechanical and the hydronic point of view.

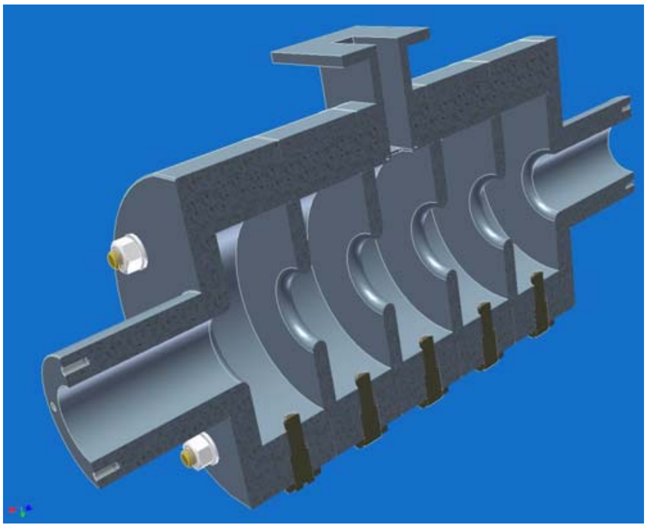

Figure 1: RF deflector solid model

2. Boundary conditions

1. RF heating: 2 MW peak pulses of 4.5 μs duration at a repetition rate of 10 Hz, uniformely distributed at the internal surface of the cell; cos2θ polar distribution.
2. Thermal exchange with air with T=23°C, v<0.2 m/s, film coefficient h_{air}=3.75 W/m^2 °C, evaluated via Dittus-Boelter correlation (Appendix A).

3. Cooling by copper tubes wrapped in machined grooves around the body, with water at 45°± 0.1°C (see "Cooling design").
4. Radial dimensional stability along operation: $\Delta R<1$ μm (at the internal surface of the cell), comprehending initial thermal transition from at rest conditions (i.e.: no RF power, water temperature of 45°C).

A dramatic simplification of the model can be achieved owing to the following statements:

- The axial asymmetry (geometrical as well as of load) can be neglected, because of high aluminium thermal diffusivity compared with the dimensions of the body;
- the thermal transmission at the ends of body, connected to thin steel bellows, should be negligible;
- the frequency of RF heating produces a purely surface load for skin effect;
- due to thermal time constant of the body, the pulsed load can be represented as a constant load of mean value.
- both the convection coefficients of the heat exchange inside the tubes and at the external surface of the body are constant, assuming a little or no variation of temperature and flow conditions in air and water.

3. Cooling design

To avoid exceeding variation of the body diameter from start of loading to steady state, the water temperature rising between inlet and outlet should be limited to something less than the general accuracy estimated. So a tentative ΔT_w=0.12°C was imposed, corresponding to 10 tubes of inner diameter of 6 mm, at 1 m/s coolant velocity and a film coefficient evaluated with Churchill-Chu correlation (Appendix A) as 6684 W/m^2 °C, on an useful total exchange surface of 0.112 m^2.

The steady state thermal calculation done with a FEM model under such conditions gives the results plotted in figure 2 and 3, showing a maximum heating of 0.234°C, and a corresponding radial displacement at the cell internal boundary of less than 0.2 μm.

The transitory from the beginning of RF heating was evaluated with the same model. The graph of figure 4 shows the temperature at the hottest point (internal surface) and at the external surface vs. time.

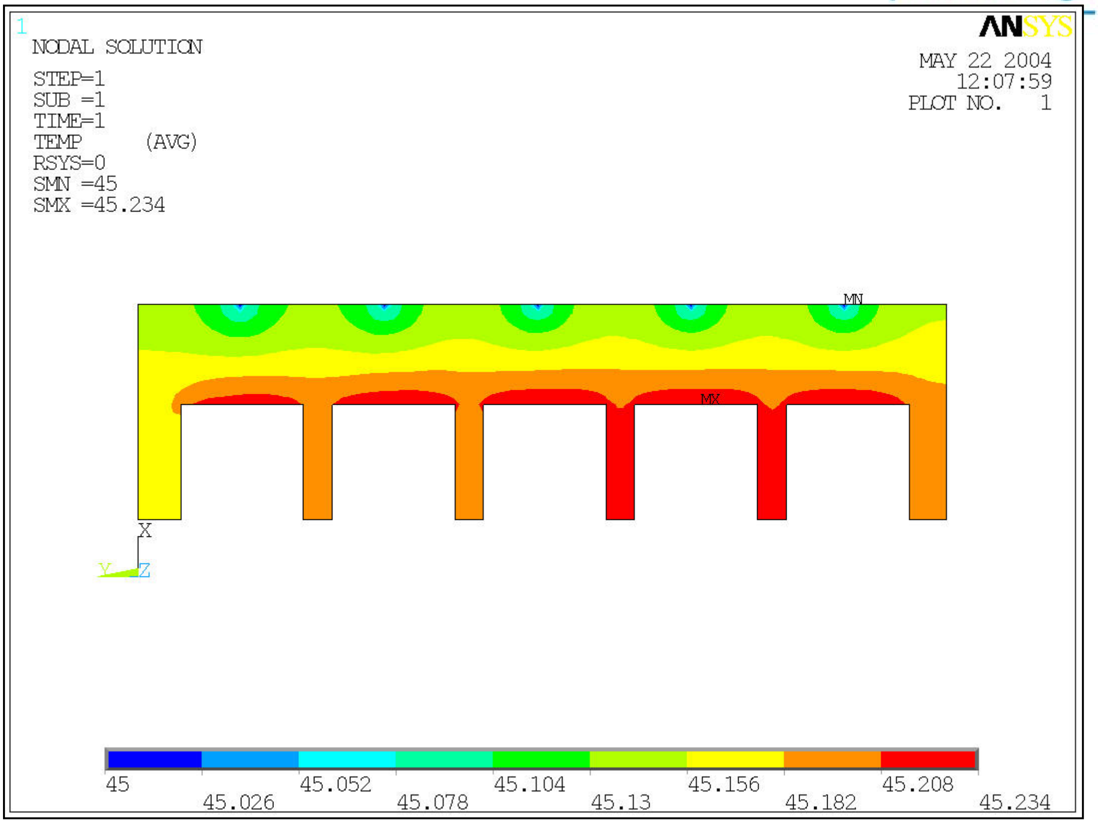

Figure 2: Temperature (°C) at steady state

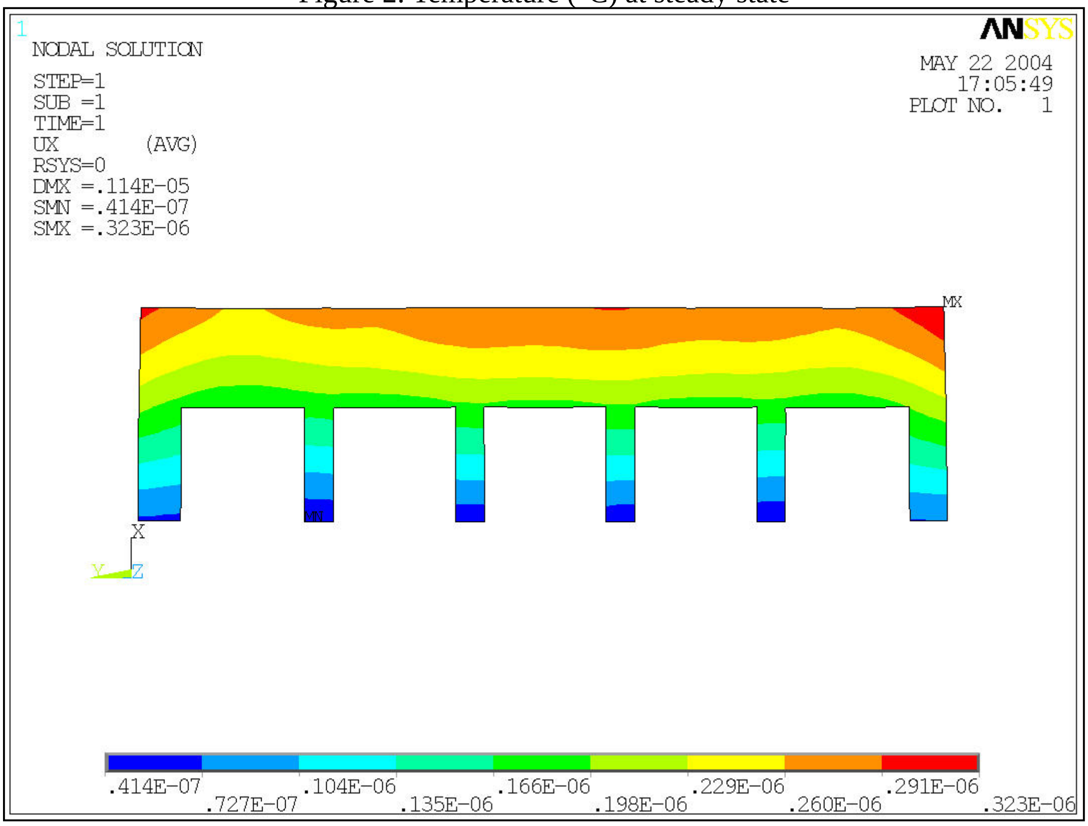

Figure 3: displacements (m) at steady state.

4. Analytical solution

To quickly verify the FEM model, simple calculations were done, under the hypotheses of coolant temperature stabilized with an accuracy of ± 0.1°C, that means a maximum variation of the body temperature ΔT_{Bmax} of 0.2°C, and plane stresses condition (i.e.: body free to expand axially due to the supporting system). The foreseeable maximum steady state thermal expansion will be therefore in the order of magnitude of

$\Delta R/R=\alpha\Delta T_{Bmax}=0.23\ 10^{-4} * 0.2=0.46\ 10^{-5}$

corresponding to $\Delta R= 0.28\ 10^{-6}$ m, just close to the FEM results, and inside the requested limit.

The behaviour of the cavity body could be modelled with a lumped mass approximation, assuming a constant water temperature and neglecting the air cooling, by an expression like (1), where hA/ρVc at the exponent can be viewed as the time constant of the cooled body and W/hA as the steady state ΔT.

$$(1)\quad T_B = T_w + (T_{B0} - T_w - \frac{W}{hA})e^{-\frac{hA}{\rho Vc}t} + \frac{W}{hA}$$

The plot of T_B vs. time is in figure 4 together with the FEM results. In the limits of hypotheses, the expression (1) is therefore a useful approximation.

Different steady state load scenarios (different RF power and/or pulse repetition rate) can be easily estimated (table 1), due to the linearity of the model proposed (thermal exchange coefficients constant).

Table 1: DR at different load scenarios

ΔR (μm)	*W_{peak} (MW)*	*Freq (Hz)*
0.28	*2*	*10*
0.14	*1*	*10*
0.07	*1*	*5*
0.14	*2*	*5*

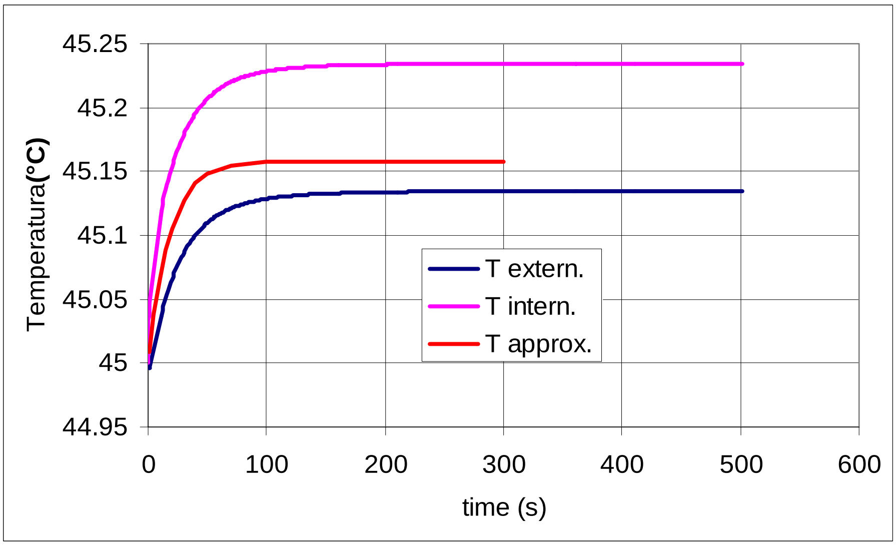

Figure 4: Temperature (°C) transitory (FEM model and hend estimation..

5. Hydronic system design

The basic configuration chosen is shown in figure 5. An "injection" scheme is employed, with a mixing three way valve on the return line, spilling from mains with head available. The secondary circuit as well as the primary are at constant flow; the valve is sized with authority P_v=0.5.

The valve system leads to a temperature accuracy of ± 1°C. Considering a final set point of 45°C at the RF deflector, the choose for its temperature set is 44°C.
The electric heater should be therefore sized at least for a maximum temperature rising of 2°C, which means 2.4 kW power, and should be driven by a thyristor unit with a good linearity. The whole control chain should then have a total accuracy better than 0.1°C. Simulations are being performed to design the system.

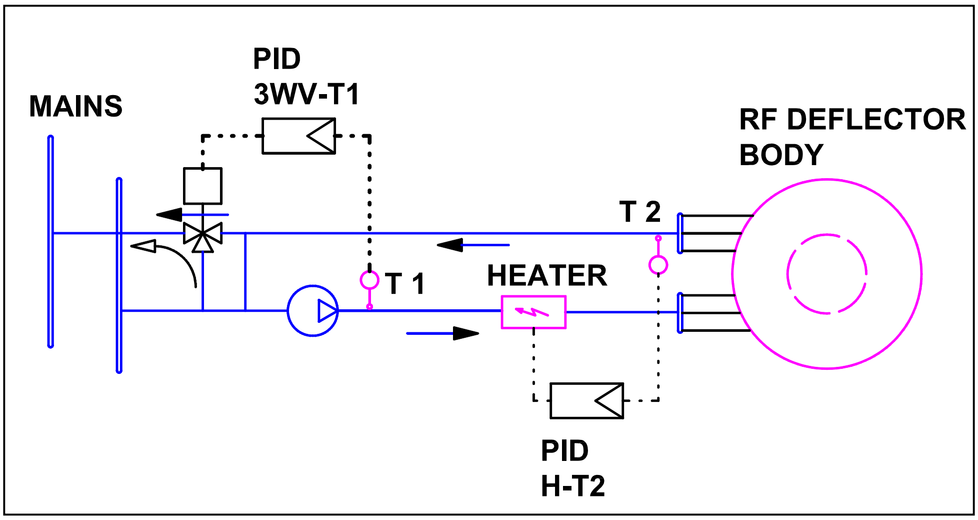

Figure 5: Cooling hydronic system

6. Conclusions

The required dimensional stability is feasible. Its achievement strongly depends on the accuracy of the design and the commissioning of the cooling system.
Accurate tests will be therefore done to match the PID parameters to the system characteristics, as well as to trim all the set-up.

7. Acknowledgements

Thanks are due to Mr. Valerio Lollo for the mechanical design work and the picture in figure 1.

8. References

[1] D. Alesini, C. Vaccarezza, “Longitudinal and transverse phase space characterization”, SPARC Technical Note, SPARC-BD-03-006, 25th November 2003, INFN/LNF, Frascati, Italia.
[2] F.P. Incropera and D.P. DeWitt, “Fundamentals of Heat and Mass Transfer”, 4th Ed, J. Wiley & sons, USA.

9. Glossary

T_B body temperature (°C)
T_w water bulk temperature (°C)
W input RF power (W)
h convection film coefficient (W/m^2 °C)
ρ density (kg/m^3)
V body volume (m^3)
c specific heat (W/m^3 °C)
A heat exchange surface (m2)
α thermal expansion coefficient

number of Nusselt $Nu = \frac{\mathrm{hD}}{k}$

number of Prandtl $\mathrm{Pr} = \frac{\mu c_p}{k}$

number of Raleygh $\mathrm{Ra} = \mathrm{Gr}\,\mathrm{Pr} = \frac{\mathrm{g}\beta\Delta\mathrm{TD}^3}{\alpha\nu}$

10. Appendix A

(1) $Nu_D = 0.023 \cdot \mathrm{Re}_D^{0.8} \cdot \mathrm{Pr}^{0.4}$ Dittus-Boelter correlation (Re> 104)

(2) $Nu = 0.36 + \frac{0.518 \cdot Ra^{0.25}}{\left[1 + \left[\frac{0.559}{\mathrm{Pr}}\right]^{\frac{9}{16}}\right]^{\frac{4}{9}}}$ Churchill-Chu correlation (nat. conv. around horizontal cylinder)