# THERMAL DESIGN OF AN RFQ CELL FOR THE RADIO FREQUENCY QUADRUPOLE UNDER CONSTRUCTION FOR ISIS.

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*Abstract*

A high duty cycle Radio Frequency Quadrupole (RFQ) accelerator is being designed and constructed under a collaboration between the Johann Wolfgang Goethe Universitat, Frankfurt, Germany and the CCLRC, Rutherford Appleton Laboratory, UK. This paper discusses the design of the cooling of an RFQ cell, the finite elements analysis of a thermal model under predicted heat load conditions and experiments to confirm the validity of the model.

## 1 INTRODUCTION

The RFQ has a design peak power of 200 kW operating at a 10% duty cycle, resulting in a surface heating of 20 kW distributed across 14 cells. Within each cell 34% appears on the electrodes, 44% on the electrode support stem and 22% on the rest of the cavity but mainly the ground plate [1].

To maintain the alignment and stability of an accelerated ion beam this heat must be dissipated to enable the RFQ to operate at a very stable temperature and limited thermal expansion.

The cells at each end of the vessel (numbers 1,2,13 & 14) have stems which are half the thickness of the rest and only one annular coaxial cooling channel. Potentially these are the most difficult to cool, hence they are taken as a worst case scenario. Cells 3 - 12 have stems with an inlet and outlet coaxial channel linked by a straight drilled channel. This configuration is shown in Figure 1 with the flow path identified.

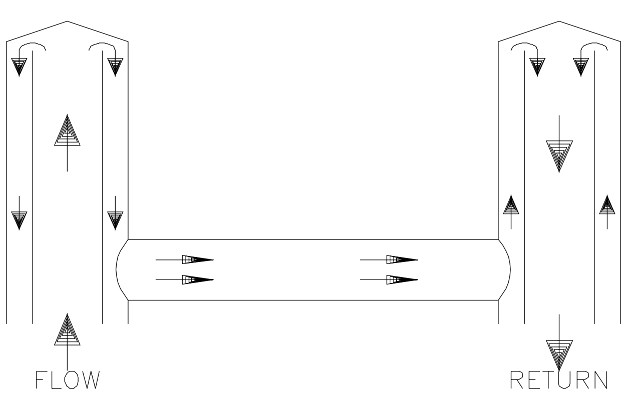


Figure 1.

The thermal finite element model yields a temperature profile for a specific cell, the nodal results file is then used to analyse the thermal expansion due to temperature gradients.

## 2 THERMAL CALCULATIONS

With a cooling water velocity, v, of 2.5 m/s in the coaxial inlet pipe an estimation of the pressure drop across a stem gives a value of ~0.8 bar. This suggests that each stem needs to be fed in parallel from a manifold in order to reduce the overall pressure drop across the RFQ vessel.

At this velocity the mass flow rate $\dot{m} = 0.047\, kg/s$, (2.8 litres/min), the specific heat capacity of water $c_p$ = 4200 J/kg K and with an expected heat load of 1.43 kW/cell the temperature rise across each cell is:

$$\Delta T = \frac{Q}{\dot{m} c_p} = \sim 7^\circ C$$

Calculating the heat transfer coefficient, h, for an annular coaxial channel is complex but an approximation can be obtained by substituting the equivalent diameter, $d_e$ ,for the nominal diameter in expressions for the Reynolds and Prantl numbers [2].

The coaxial cooling channel has an equivalent diameter $d_e$= 0.002m, for a dynamic viscosity $\mu$ = 0.9x10$^{-3}$kg/ms, density $\rho$ = 1000kg/m$^3$ and a velocity of 4.6m/s in the annulus the Reynolds No. is:

$$\mathrm{Re} = \frac{\rho v d_e}{\mu} = 11231$$

For forced convection in a closed conduit the Nusselt number, Nu, is given by the modified Dittus-Boelter equation for turbulent flow:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \qquad [2]$$

when $\mathrm{Re} \geq 10000$ & $0.7 \leq \mathrm{Pr} \leq 160$
with the thermal conductivity of water k = 0.6W/mK and the Prantl number being:

$$\Pr = \frac{c_p \mu}{k} = 6.3 \qquad \text{the} \qquad Nu = 83.5$$

The heat transfer coefficient is given by:

$$h = \frac{Nu\,k}{d_e} = 25056 W/m^2 K = 0.025\, W/mm^2 K$$

The expected heat loss distribution on each cell gives heat load /unit area values of 0.0336, 0.0320 and 0.0324 $W/mm^2$ on the electrodes, stems and ground plate respectively. These values along with the heat transfer coefficient for the coaxial channel are used as the basic parameters to model an end cell with one coaxial cooling channel.

## 3 FINITE ELEMENT RESULTS

A 3-D ANSYS™ model of an end cell including its ground plate, stem and electrodes was analysed. Solid 87 elements which are 10-node tetrahedral in shape are used as they are ideally suited to meshing irregular shapes. Solid 92 are the equivalent structural element required for modelling the thermal expansion [3]. The meshing procedure is optimised by utilising element size and shape options. Applying the heat load and convection coefficient to the single coaxial channel model gives Figure 2.

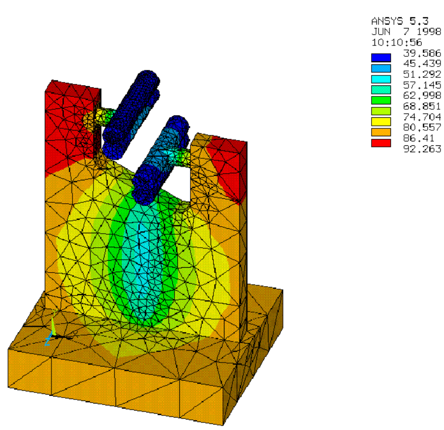


Figure 2.

A maximum temperature of 92°C is seen on the stem legs which support the electrodes. The ground plate temperature is around 86°C, but as the ground plate is rigidly bolted to the RFQ vessel any thermal expansion will be seen predominately on the stem. This temperature gradient produces a maximum movement on the longer leg and is in the order of +0.2 mm in the 'Y' axis i.e. vertically upwards.

Running the RFQ with stems at such a high temperature presents many operational problems and is unacceptable.

Space for any modification to the cell design is limited because the cooling feed pipes for the electrodes run up through the ground plate via the stem legs, although, this may provide some level of cooling to the leg tips. However, there is sufficient space for the addition of another coaxial cooling channel inboard of the legs. Figure 3 shows the modified design with twin cooling channels. The maximum temperature is once again on the leg tips but has dropped to 68°C with the ground plate now being cooled more efficiently and seeing a temperature of 54°C.

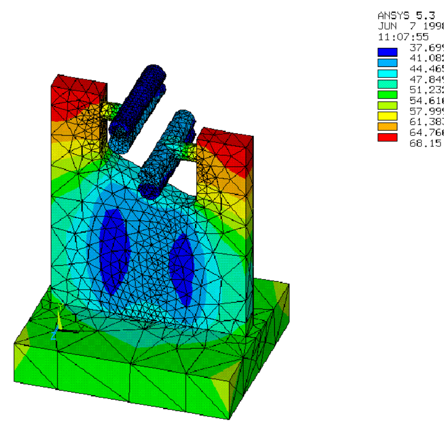


Figure 3.

Analysing the same model structurally results in a maximum deflection in the 'Y' axis of 0.13 mm. This is shown in Figure 4.

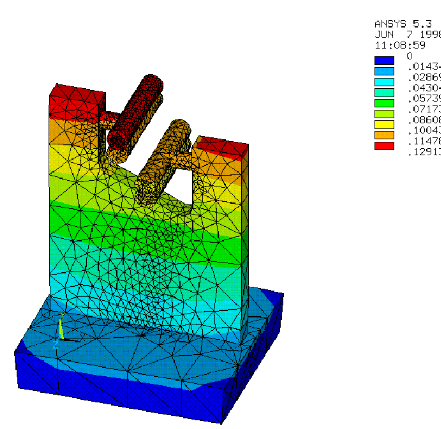


Figure 4.

Figure 5 shows a model of cells 3 - 12 which have a cooling channel design as shown in Figure 1. An overall maximum temperature of 47°C is seen on the ground plate with the leg tips reaching 40°C. This is lower than the twin channel end cell and consequently the thermal movement is less being in the order of +0.1mm.

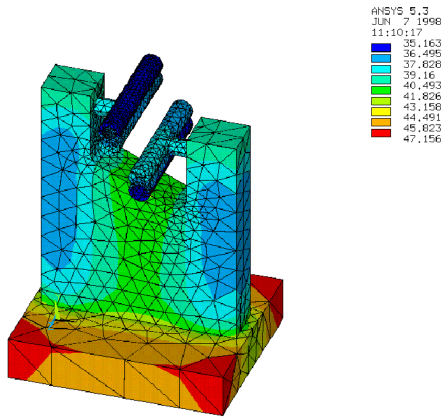


Figure 5.

## 4 EXPERIMENTAL TESTS & RESULTS

A full scale end cell stem is used as a test piece to verify the finite element model. A heater jacket applies a measured heat load to the test piece with 2.6 litres/min running through the cooling channel. Three type ‘K’ thermocouples are used to monitor temperature 10mm below the tips of the stem legs and mid-way along the slope between the stem legs. Two further thermocouples are used to monitor the input and output water temperatures. During tests the stem and jacket are wrapped in an insulating blanket to limit heat convection to air.

The stem was soaked for approximately 45 minutes with a total electrical input power of 770 kW applied evenly to either side. The temperatures on the long stem leg, short stem leg and slope were 41.0, 44.1 and 43.7 respectively with the inlet and outlet temperatures being 27.3 and 31.4°C. This temperature rise equates to 746kW of heat being dissipated to the cooling water.

A finite element model is used to simulate the exact conditions of the test. Applying the same heat load to the finite element model and using revised heat transfer coefficients with a staggered bulk fluid temperature produces Figure 6. There is good correlation between the experimental results and finite element model with the long stem leg, short stem leg and slope showing temperatures of ~46, 48 and 50°C respectively i.e. within 12% of the experimental values.

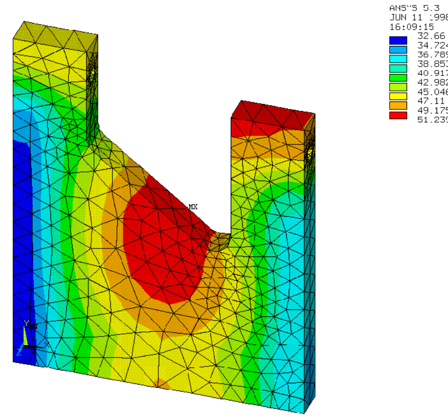


Figure 6.

## 5 CONCLUSIONS

The addition of another coaxial cooling channel in the end cells drops the maximum temperature of the stems to a more acceptable operational level. This temperature drop, by definition, decreases any movement of the electrodes resulting in a more stable RFQ structure. Movement of cells 3 -12 is smaller but if development tests highlight problems with uneven thermal expansions then the flow to these cells may be adjusted so that all cell movements are the same.

The experimental results show that the finite element model is sound albeit slightly pessimistic and gives confidence in its use during further developments.

## REFERENCES

[1] Hartmut Vormann, Johann Wolfgang Goethe Universitat, Private Communication.
[2] William S Janna , Engineering Heat Transfer, SI Edition, Chapter 11.
[3] ANSYS™ Elements User’s Manual Revision 5.0, Volume 3.