ISBN: 978-3-95450-191-5 doi:10.18429/JACoW-SRF2017-MOPB104

DEVELOPMENT OF A NOVEL SUPPORTING SYSTEM FOR HIGH LUMINOSITY LHC SRF CRAB CAVITIES

T. Jones[1], STFC Daresbury Laboratory, Warrington, UK
G. Burt, University of Lancaster, Lancaster, UK
K. Artoos, R. Calaga, O. Capatina, T. Capelli, M. Sosin, J. Swieszek, C. Zanoni, CERN, Geneva, Switzerland
[1]also at University of Lancaster, Lancaster, UK

Abstract

Compact SRF Crab Cavities are integral to the HL-LHC upgrade. This paper details the design of support structures within the SPS (Super Proton Synchrotron) Crab Cavity Cryomodule. For ease of alignment each cavity is supported with the mechanical tuner and RF Fundamental Power Coupler (FPC) via a common support plate. To reduce heat leak and remove bellows in the FPC it was determined that this would be the fixed support for the cavity. In addition, novel flexural blades were designed to give increased stiffness yet allow for thermal contraction of the cavity towards the fixed point of the FPC. This approach was superior when compared via simulation to several alternative techniques. A detailed simulation model was used for optimisation of directional stiffness, identification of vibration modes and minimising thermal stresses. A transmission matrix was developed to assess modal deflection for given ground vibration conditions. The spreadsheet gives an instantaneous yet comparable result to time consuming random vibration FE Analyses. The final engineering design of the supporting system is now complete and will also be described in this paper.

INTRODUCTION

There are currently 2 crab cavity designs envisaged for the HL-LHC upgrade. These are the Double Quarter Wave Resonator (DQW) and the RF Dipole (RFD), the designs for which are shown in Fig. 1. The design of the DQW RF structure has been led by Brookhaven National Laboratory, the RFD by Old Dominion University. Each cavity type has a different deflecting plane, the DQW rotates bunches vertically, whereas the RFD deflects bunches horizontally. The cavities sit within a Grade 2 Titanium liquid helium tank. This material was chosen as it has an almost identical co-efficient of thermal expansion to the Niobium cavity. There are mechanical interfaces on the cavity for connection to the Cavity Tuner. The SRF Compact Crab Cavities designed for HL-LHC have never been tested with beam. The risk of installing unqualified cavities into the LHC was deemed unacceptable; therefore a test of 2 DQW cavities in the Super Proton Synchrotron is planned for 2018.

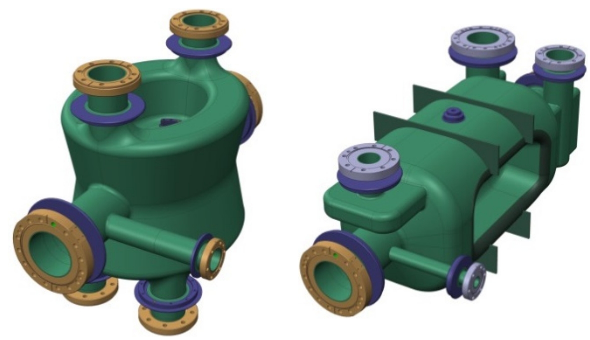

Figure 1: DQW (left) and RFD (right) prototype crab cavity models.

The cryomodule designed to house the cavities during testing on the SPS is shown in Fig. 2. This paper describes the process by which the cavity support structures were designed and integrated into the cryomodule. The cavity support system is to be capable of supporting the ~250 Kg dressed cavity, yet with low cross section in order to minimise conductive heat losses into the 2 Kelvin cryogenic system. The support system should allow for thermal contraction of the 2 K components so that there are no stresses above the respective material yield, and the cavity should not be deformed during cool down. In addition, cavity RF detuning due to vibration (microphonics) is to be minimised to <5.5 Hz [1]. The choice of operating below the lambda point of helium, i.e. the operation at 2 K is partially driven by this desire to minimise microphonics as in this superfluid state helium should not boil [2]. Microphonics can be compensated by the use of a fast acting Piezo tuner, however, it is believed that through careful design of the cryomodule this should not be required for the HL-LHC Crab Cavities.

ISBN: 978-3-95450-191-5 doi:10.18429/JACoW-SRF2017-MOPB104

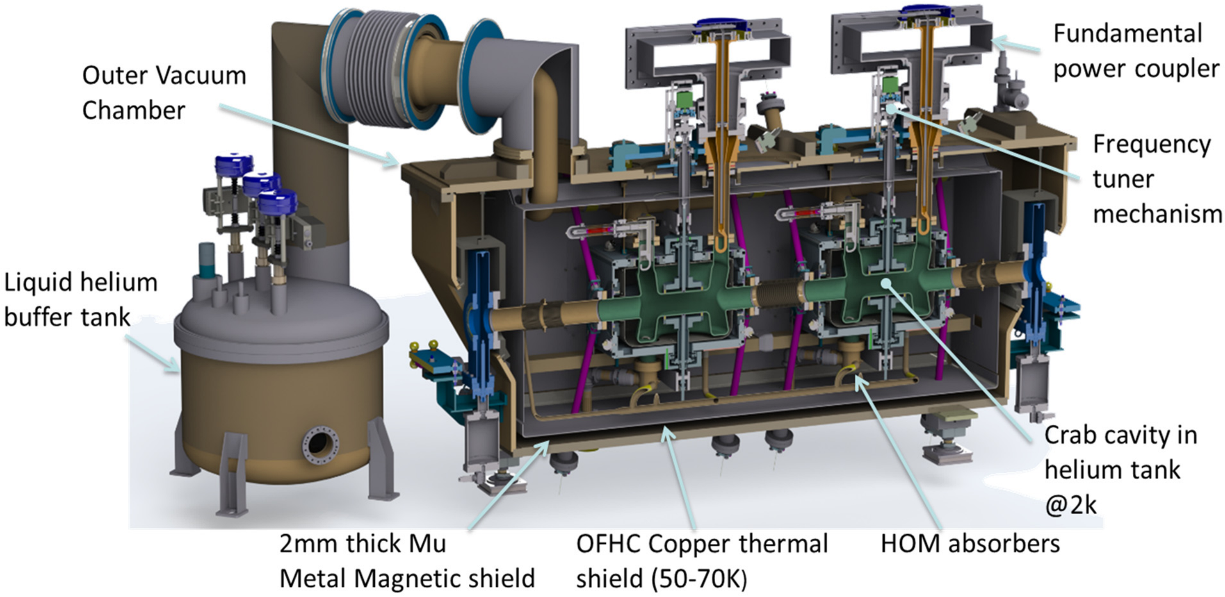

Figure 3: DQW Crab Cavity SPS Cryomodule cross section view.

CAVITY SUPPORT SYSTEM

It was decided that for ease of alignment the cavity should be supported, along with the mechanical tuning mechanism, via a common support plate which itself is mounted to the upper vacuum flange of the outer vacuum vessel of the cryomodule. To remove the need for bellows in the design of the RF input coupler it was determined that this would be the main support for the cavity, similar to the technique used in the SPL Cryomodule [3]. Stainless Steel bellows in the RF line require copper coating and there is a risk of this copper coating flaking off and contaminating the coupler or cavity [4], therefore eliminating the bellows removes this risk completely. Doing this also has the advantage of reducing the static heat leak to the cavity by minimising the number of rigid support structures. Novel cavity support flexures were designed and compared to various alternative support options using a simplified FEA model in ANSYS workbench. The RFD cavity was chosen for the comparative studies as its larger mass and more cantilevered support would give 'worst case' results.

Cavity Support Flexure Design

It was proposed that the use of flexural blades as the supplementary support to the fundamental power coupler could dramatically improve the performance of the cavity support system. They act to rigidify the system but will not induce stress in the coupler upon cooldown. This is a technique previously used successfully on the interface plate for the SPICE instrument thermal testing enclosure shown in Fig. 3 [5].

The blades are designed such that the flexural bending plane is tangential to the cylindrical surface of the fundamental power coupler (FPC) which acts as the fixed

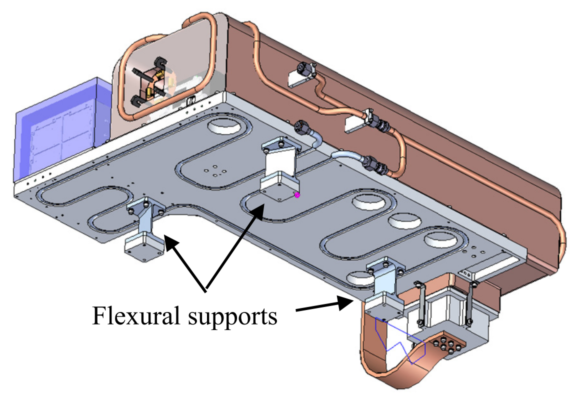

Figure 2: SPICE instrument within thermal testing environment on flexural supports.

point. This allows for thermal contraction towards the FPC, whilst the system remains stable in all other directions. For the design of such flexures first one must consider the amount of thermal contraction of the helium vessel. The distance from the coupler to the flexures was initially set to 650 mm for comparative studies. This gives a contraction of 1.3 mm in the Grade 2 Titanium, therefore the flexures must deform by this amount. Taking this value one can calculate the required geometry of the flexure using the following formulae;

$$S = \frac{W}{Z}\left(\frac{1}{2}l - x\right)$$

$$y = \frac{Wx^2}{12EI}(3l - 2x)$$

ISBN: 978-3-95450-191-5 doi:10.18429/JACoW-SRF2017-MOPB104

Where, S is the stress at a given location x from the 'moving' end, W is the force acting perpendicular to the flexure, l is the flexure length, E is the Young's modulus of the flexure material, I is the second moment of area of the flexure geometry and Z is the section modulus. The deflection of the flexure is denoted by y which in this case is known. Therefore one can re-arrange the first equation and solve for W. Then by using W in the second equation one can obtain the maximum stresses in the structure, which are equal and opposite and found at the support ends.

The flexures will be manufactured from 316L Stainless Steel to give identical vertical thermal contraction to the fundamental power coupler. In order to minimise the heat leak to the 2 K system the flexures will need to be intercepted by a passive cooling strap to give intermediate cooling in the equivalent location to that of the fundamental power coupler. The approximate thermal profile and dimensions used for the comparative studies are shown in Fig. 4.

Integrated thermal conductivity values [6] can be used in combination with the geometry of the blades to calculate the heat leak to each of the cooling circuits using the Fourier rate equation;

$$\dot{Q} = (K_h - K_c)\left(\frac{A}{L}\right)$$

Where;

$$K = \int_{4K}^{T_h} k_t \, dT$$

K_h is the thermal conductivity integral value at the higher temperature, K_c that of the lower temperature. A is the cross sectional area of the blade and L is the length between the two temperatures. If one calculates $\dot{Q}$ based upon the geometry shown in Fig. 4 the values are 7.12 W at 80 K and 0.11 W at 2 K per flexure.

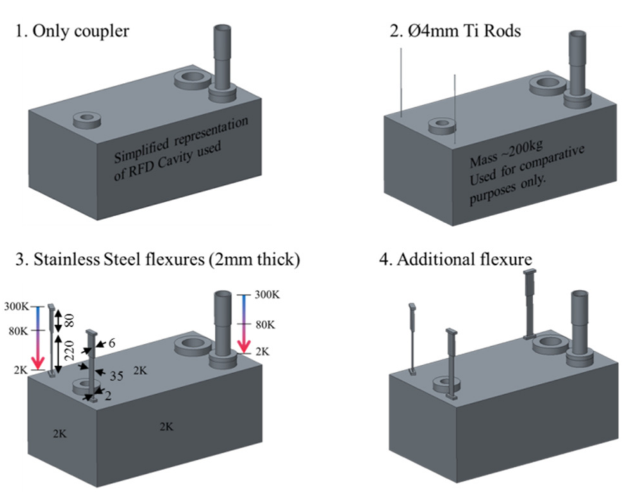

Figure 4: Support system models for comparative studies.

Comparative Cavity Support Studies

An acceleration of 9.81 m/s^2 was applied to each model to represent standard earth gravity. The models were simply boxes with the correct size and mass of the cavity. The power coupler, rods and flexures were all fully fixed at the location of the common support plate, therefore assuming that this plate is infinitely rigid. Static total deformation, maximum von-Mises Stress and the first 4 mechanical vibration modes were found for each option shown in Fig. 4.

As can be observed in Table 1, the performance of the flexural supports is significantly greater than a rod type arrangement. The static deformation is reduced by a factor of 10 and the natural frequency increased by more than a factor of 3. A thorough study of rod configurations completed by CERN [7] compared various rod thickness, angle and pre-tension values, however, the results showed no particular improvement over the 4mm vertical rods (concept 2) used for this comparison. The 3 blades give optimum performance, however, also give increased heat leak and increased integration complexity within the module. Therefore it was decided to proceed with the coupler as the main support and to have 2 stainless steel blade supports.

Table 1: Results of Cavity Support Comparison (Analysis Options from Fig. 4)

Analysis	Max Deformation (mm)	Max von-Mises stress (MPa)	Mode 1 Frequency (Hz)	Mode 2 Frequency (Hz)	Mode 3 Frequency (Hz)	Mode 4 Frequency (Hz)
1	3.9	183	7.7	8.3	16.1	61.1
2	0.24	65.2	8.5	25.3	38.3	70.9
3	0.025	15.3	25.1	48.3	56.5	122
4	0.01	10.5	27.2	50.15	66.9	174

Structual and Thermal Analysis of Full Engineering Model

A Finite Element Analysis was performed for each design iteration of the dressed DQW cavity, the initial model is shown in Fig. 5. A self-weight condition was applied plus the operational steady state temperature profile i.e. 2 K at the helium vessel. Contraction of components was calculated within the software using non-linear thermal expansion co-efficient values obtained from testing at Fermilab [6]. Low stress of 8.5 MPa was observed in the cavity. In other structures the stresses were below 20% of yield, except at the flange interfaces were differential thermal contraction between the stainless steel flanges and Niobium tubes gave high stresses ~200 MPa (Fig. 6). This is above the yield strength of Niobium at room temperature, however, these components are at 2 K therefore the yield strength of Niobium is ~400 MPa, meaning that these stresses are acceptable.

ISBN: 978-3-95450-191-5 doi:10.18429/JACoW-SRF2017-MOPB104

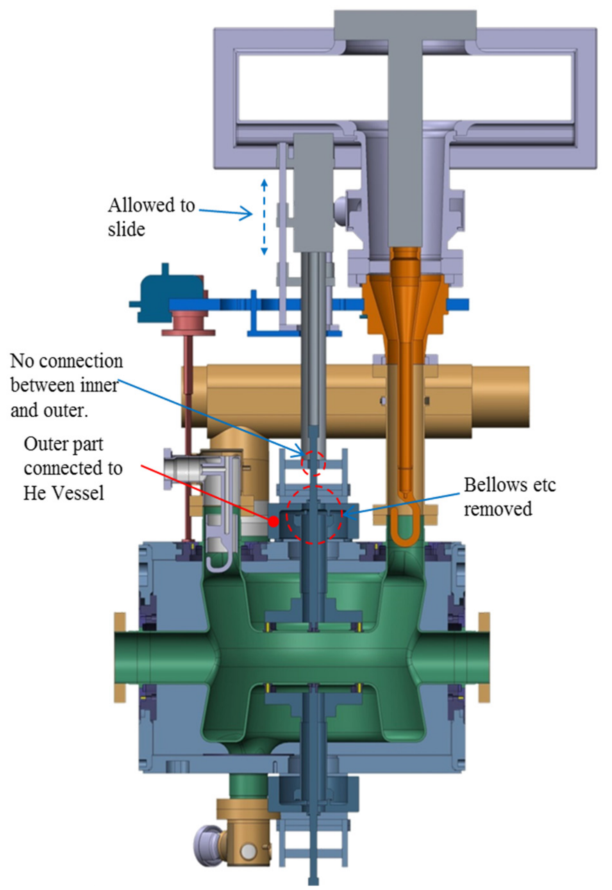

Figure 5: Full detail Finite Element model of DQW Dressed Cavity.

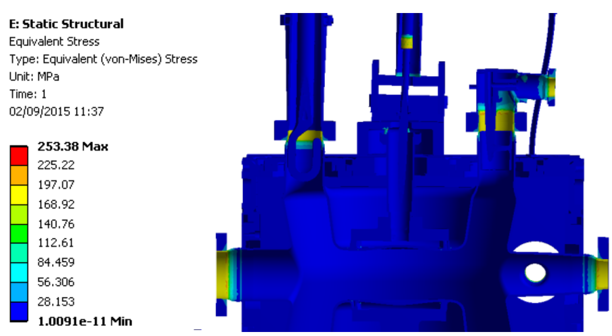

Figure 6: Stress result from self-weight and cooldown.

Modal Analysis of Full Engineering Model

Each iteration of the model was also analised using the ANSYS workbench modal solver in order to assess the first 10 mechanical vibration modes. In the first iteration the lowest modes were due to the cavity RF tuner mechanism 'rocking' both laterally (Fig. 7) and longitudinally at 8.9 Hz and 9.9 Hz. It can be observed that tuner vibrations have an impact on the cavity shape and that the deformation is in one of the most sensitive regions of the cavity for RF stability. The deflection values shown on the left hand side of Fig. 7 can be disregarded as in this modal analysis the driving frequencies and amplitudes had not been set. However, the relative values of deformations are valid and show that the cavity would deflect ~9-12% of maximum tuner vibration amplitude in the RF tuning location. In order to assess the true deformation of the tuner, dressed cavity and other associated components, a novel transmissibility matrix was created in MS Excel based upon the following function [8];

$$Transmissibility = \frac{Y}{X}$$

$$\frac{Y}{X} = \sqrt{\frac{4\xi^2(\omega/\omega_n)^2 + 1}{[1-(\omega/\omega_n)^2]^2 + 4\xi^2(\omega/\omega_n)^2}}$$

Where; Y is displacement of the supported mass, X is displacement of the base of the support, ξ is the fraction of critical damping, ω/ω_n is the ratio of ground forcing frequency to the natural frequency of the supported mass.

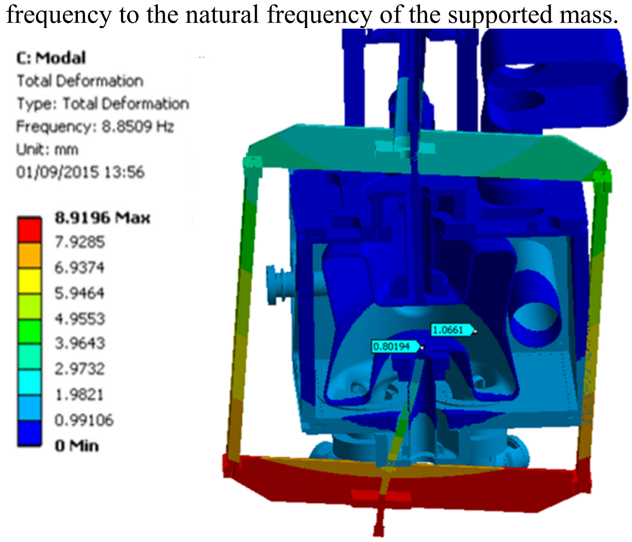

Figure 7: Tuner deflection mode at 8.9Hz.

The values for X were obtained by calculating ground displacement from a displacement PSD of the Diamond Light Source Facility [9] with data from 1 Hz to 100 Hz. The displacement Y for each mode was then obtained at each frequency band. These results were then combined for each mode using the integrated RMS method. From this one can anticipate a worst case tuner frame integrated RMS displacement of 307nm from 1 to 100 Hz. As discussed previously, cavity deformation is 9-12% of the total tuner movement. In the central region of the cavity that is affected by the tuner it has been shown that cavity RF frequency sensitivity is 372 kHz/mm [10]. If we assume 9% of total movement of 307 nm, this would give deflection of the region of 27.6 nm, which gives an RF detuning at a level of 10.3 Hz. There are 2 directions of tuner motion which would couple, plus additional modes which could add to this giving a higher value. Peak values of deformation due to seismic spikes in ground vibration may also be as high as 6x this value [11], therefore peak detuning could have been as high as 60 Hz. The tuner does not have a fast tuning mechanism (such as a piezo type device) to mitigate these microphonics, therefore there is a requirement to limit this detuning to ~5.5 Hz at

ISBN: 978-3-95450-191-5 doi:10.18429/JACoW-SRF2017-MOPB104

1σ [1]. Due to the relatively high values observed it was determined from this initial work that the first 4 cavity modes were too low and therefore additional stiffness was required in both the cavity support and tuner mechanism.

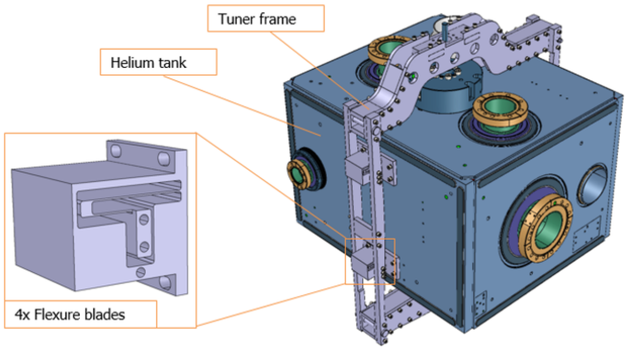

Figure 8: Improved Tuner design with flexural links to Helium Vessel.

Flexural links have been introduced between the tuner frame and the helium vessel (Fig. 8) which prevent movement of the frame and subsequent detuning of the cavity. Revised modal analyses of this configuration show that all modes of the tuner are now in excess of 40 Hz. This reduces the integrated displacement of the tuner from 307nm to 28nm (from 1 Hz to 100 Hz), a factor of 11 improvement, reducing the microphonics by this amount brings them to within allowable limits. The width of the flexural blades of the cavity were increased to 75 mm and the mounting of the common support plate was changed to a 5 point mount to raise the cavity swinging mode from 12.7 Hz to 19 Hz, above the 15 Hz which is considered best practice for accelerator components [12]. The stiffness of the system is now considerably improved over the concept design as can be observed in Table 2.

Table 2: Stiffness of Support System

Direction	Concept Stiffness	Final Stiffness
Lateral	0.62 kN/mm	1.83 kN/mm
Longitudinal	1.58 kN/mm	1.70 kN/mm
Vertical	6.87 kN/mm	25.79 kN/mm

CONCLUSION

A thorough design process was undertaken to determine the optimum configuration for the HL-LHC Crab Cavity support system. This system has been analised and the stresses and deformations are acceptable. Potential microphonics issues were identified early in the design process and these have since been improved. The final design provides enhanced performance over the concept and shows the benefit of the optimisation process employed.

REFERENCES

[1] P. Baudrenghien, "Functional Specifications of the LHC Prototype Crab Cavity System," CERN, Geneva, 2013.

[2] F. Carra, "Crab Cavity and Cryomodule Development for HL-LHC,"in *Proc. SRF2015*, Whistler, BC, Canada, Sept 2015, paper FRBA02, pp. 1460-1466.

[3] V. Parma, "Status of the Superconducting Proton Linac (SPL) Cryomodule," in *Proc. SRF2013*, Paris, France, Sept 2013, paper MOP085, pp. 345-348.

[4] L. Popielarski, "Testing Of Copper Plating Quality On Rea3 Coupler Bellows And Approach To Improved Plating For FRIB Production," in *Proc. SRF2013*, Paris, France, Sept 2013, paper THP067, pp. 1077-1080.

[5] T. Jones, "RFS-1017-FEA-002-V3-Interface plate thermal analysis," STFC, 2013 [Report]

[6] C. Zanoni, He Tank Support system, EDMS, CERN, 2015 [Report]

[7] T. Nicol, Thermal Expansion Data for use in ANSYS, Private communication with T.Nicol, 2013.

[8] S. Sharma, "Design of Accelerator Girder System for Vibration Suppression," 2005.

[9] H.C, Huang, "Vibration Measurement at Diamond and the Storage Ring," in Proc. EPAC 06, Edinburgh, Scotland, June 2006, paper THPLS027, pp. 3338-3340.

[10] S. Verdu-Andres, "Design and prototyping of HL-LHC double quarter wave crab cavities for SPS test", in Proc. IPAC15, Richmond, Virginia, May 2015, paper MOBD2, pp. 64-66.

[11] G.K. Davis, "Vibration response testing of the CEBAF 12 GeV upgrade Cryomodules," in Proc. LINAC 2012, Tel-Aviv, Israel, Sept 2012, paper MOPB031, pp. MOPB031, pp. 240-242.

[12] A. W. Chao, Handbook of Accelerator Physics and Engineering, Stanford, California: World Scientific Publishing Co. Pte. Ltd, 1999.