THERMAL AND MECHANICAL ANALYSIS OF 3 GHz SIDE COUPLED RF CAVITY FOR MEDICAL LINACS

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Abstract

Medical linear accelerators have wide applications for cancer treatment in the world. Side coupled RF cavities was used in this accelerators for production of X-ray in range of energies between 4 to 25 MeV. Usually, the RF source is magnetron with lower cost in comparison to klystron in this type of applications. Side coupled cavity is a biperiodic structure with sensitive performance to operational thermal and mechanical conditions. In this paper, thermal and mechanical simulations for a period of the structure are presented.

INTRODUCTION

Linear electron accelerators are designed in various types and have wide applications in medicine, industry, agriculture and physics researches [1-2]. Standing wave RF cavities are so attractive for applications with emphasizes on compactness, portability and low weight to be applicable in portable industrial and rotatable medical accelerators [3-4].

Side coupled RF cavities used widely in medical and radiography linear accelerators. They are high quality factor structures (14000) with very narrow resonance band (100 kHz). If the resonant frequency is detuned, approximately all of RF power will be reflected. So, in linacs with standing wave RF structures, automatic frequency controller system (AFC) and Circulators are used commonly. Detuning of RF resonant frequency also effects on operation efficiency of linac. If detuning generated in coupled cells of cavity, the variation in amplitude and phase of RF cavities can cause to variation of electron and X-ray energy distribution and related delivered dose during therapy applications[5].

The variation of resonant frequency can be generated statically or dynamically. The static variations are generated based on mechanical tolerances and errors. Dynamic variation generated mainly because of cooling system stability limitations and lack of it's design efficiency.

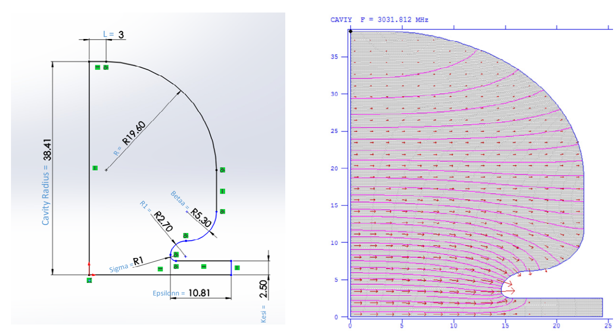
Recently, FEM thermal and mechanical multiphysics design and simulation software such as Ansys [6] and Comsol [7] was developed such that can be used for calculation and optimization of cooling system conditions.

Based on thermal losses obtained from Superfish [8], the required cooling system was evaluated for 6 MeV medical side coupled RF cavity.

These studies were done for evaluation of RF cavity static errors in dependence with geometrical parameters and investigation of better cooling system selection for our 6 MeV standing wave RF cavity.

GEOMETRICAL AND RESONANT FREQUENCY

The geometrical parameters of RF cavity were optimized for operation in 2998 MHz in coupled cells configuration. The geometrical configuration and electric field profiles after optimization in superfish simulation software were shown in figure 1.



(a) (b)

Figure 1: Main accelerating cell (a) geometrical parameters, (b) electric field profiles in superfish.

The most important electromagnetic parameters that effected by variation of geometrical parameters are resonant frequency, quality factor, effective shunt impedance and the ratio of maximum surface to normalized axial electric field (E_{max}/E_0). The geometrical parameter variations effect on resonant frequency was shown in figure 2.

The errors in parameters of cavity radius, R1 and Betaa have maximum effects on resonant frequency. Also Epsilonn and L have minimum effects on resonant frequency. Also, sigma and R have maximum and minimum effects on Emax/E0, respectively.

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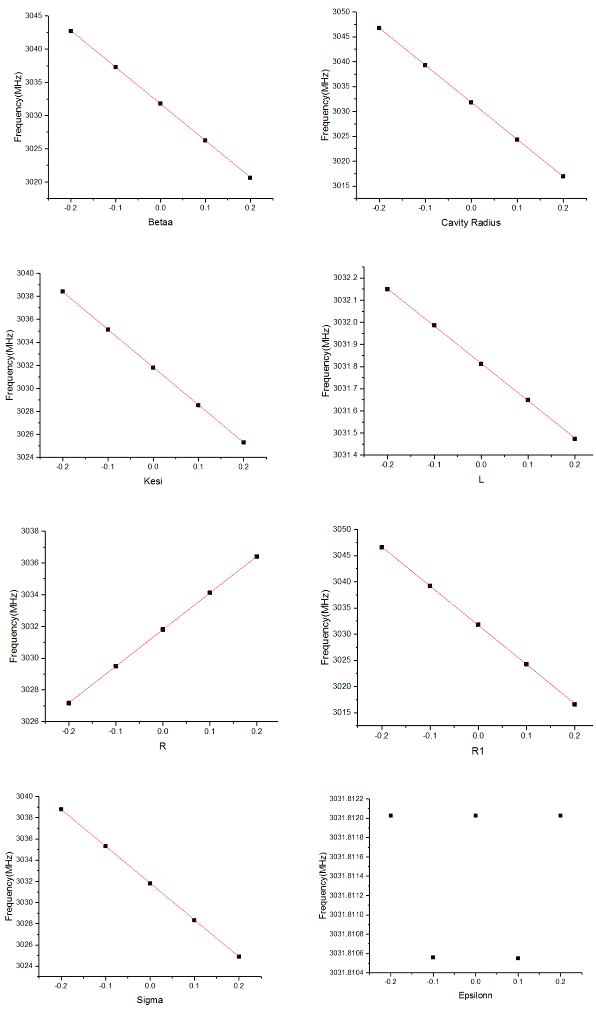

Figure 2: Resonant frequency variation based on errors of geometrical parameters.

THERMAL AND MECHANICAL CALCULATION

The cavity will be operated with 2.6 MW input power with 5 μs beam length and 100, 200 and 300 repetition rate. The average loss of RF power on inner surface of cavity calculated based on equation 1.

$$P_{ave} = (P_{peak} - P_{beam}) \times \text{pulse width} \times \text{pulse repetition rate} \quad (1)$$

The loss of each accelerating cell will be calculated for mentioned repletion rate equal to 250, 350 and 500 watts for each accelerating cell for beam current of 100 mA by dividing to number of accelerating cells respectively.

The percent of power loss on inner surfaces of accelerating cell was calculated based on Superfish calculation results. The surface area numbering was shown in figure 3 and calculation results for input powers were reported in table 1.

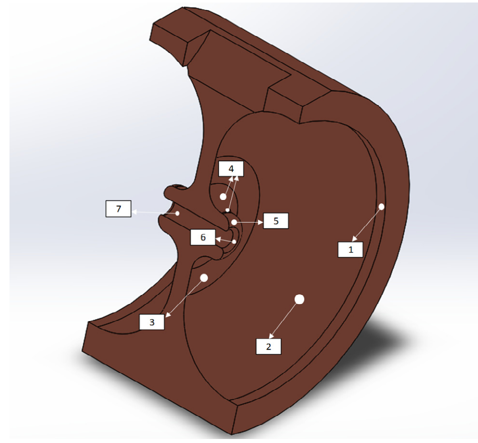

Figure 3: Number of each inner accelerating cell surfaces.

Table 1:Calculation Results for Input Powers

Surface Num.	Total surface area(m^2)	Power Loss Ratio(%)	Surface Heat Flux(W/m^2)		
			500W	350W	250W
1	0.144802	14.70242183	30540	21378	15270
2	1.190038	15.36677488	31920	22344	15960
3	0.139214	19.28068088	40050	28035	20025
4	0.088206	32.1489106	66780	46746	33390
5	0.027814	17.48981614	36330	25431	18165
6	0.005652	1.007120709	2092	1464.4	1046
7	0.030819	0.004274967	8.88	6.216	4.44

For thermal simulation of water flow in cooling pipes and their effect on removal of RF loss in cavity, the heat transfer coefficient should be determined and used in simulations.

$$h = k\frac{Nu}{D} \quad (3)$$

In this equation k is thermal conductivity of water (0.606 $W \cdot m^{-1} \cdot K^{-1}$ at 295 K), Nu is dimensionless Nusselt parameter and D is the hydraulic pipe diameter (for non-circular pipes = 4*area/perimeter) and was supposed equal to 0.008 m. The Nusselt number principally relates the heat transferred between two surfaces at different temperatures for a flowing coolant (convection) to the coolant at rest (conduction). Thus it depends mostly on the flow state. Flow rates should be high enough to produce a turbulent flow. A criterion for turbulent water flow (Re ≥ 10000) in a water tube is given by the Reynolds number. Mainly three regimes characterize the flow situation:

(i) Laminar regime: Re ≤ 2320

(ii) Transient regime: 2320 < Re < 10000

(iii) Turbulent regime: Re ≥ 10000

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The Reynolds number calculated based on equation 4 and 5[9].

$$Re = \frac{VD}{\nu} \qquad (4)$$

$$\nu = \frac{\mu}{\rho} \qquad (5)$$

Where ν is the kinematic viscosity (in $m^2.s^{-1}$) obtained from the dynamic viscosity μ (0.000959 $kg.s^{-1}.m^{-1}$ at 295K) and the water density ρ (998.8 Kg/m^3 at 295 K), and V is the water velocity. For ensuring turbulent water flow, the velocity of 2 $m.s^{-1}$ gives the Reynolds number of Re=16664.

The Nusselt number is a complex measure of the Reynolds number Re, the given geometry and the material dependent Prandtl number which mainly describes the relation of the heat in a viscous fluid generated by friction related to the conducted heat (Pr=6.6 for water at 295 K). The Nusselt number Nu can only be determined accurately by experiments. For tubulent flow the following empirical formulas have been used depending also on the tube friction factor f:

$$\mathrm{Nu} = \frac{f}{8}\frac{\mathrm{Re.Pr}}{1.07+12.7\sqrt{\frac{f}{8}}(\mathrm{Pr}^{2/3}-1)} \qquad (6)$$

$0.5 < \mathrm{Pr} < 2000 \qquad 10^4 < \mathrm{Re} < 5\times10^6 \qquad f = 0.02$

With having the temperature distribution, a structural analysis can be carried out. Structural analyses give information about the thermal deformation of the cavity, producing frequency resonant shifts and intrinsic stresses. Material properties, boundary constraints and forces acting on the geometry as caused by temperature loads have to be set before the analysis is performed.

The structural material properties to be defined for the copper are the thermal expansion factor at a reference temperature T=293K ($1.8*10^{-5}$ $°C^{-1}$), the Young's modulus (ratio of stress to strain = $1.1*10^{11}$ Pa) and the Poisson's ratio (the ratio of the transverse strain perpendicular to the applied load to the axial strain in the direction of the applied load = 0.34).

THERMAL AND MECHANICAL SIMULATIONS

Thermal and mechanical calculations of one period of RF cavity were done by ANSYS. For thermal and mechanical calculations, two types of cooling pipes with circular and rectangular cross sections were selected. The simulation results for 350 watts surface loss with 25 °C input water temperature for two different cooling pipes were shown in figures 4 and 5.

The simulation results show that the maximum and minimum temperatures for circular and rectangular cooling pipe cross sections are 45.7-30.3 and 49.6-28.9 °C respectively. Also maximum von-mises stress is 50.1×10^7 Pa and 30.4×10^7 Pa for circular and rectangular cooling pipe cross sections. According to these stresses the maximum deformation for the circular and rectangular cross section pipes are about 22 and 18 μm respectively. But, the value of deformation on the surfaces with RF loss are very low and the performance of cooling system is acceptable. Based on this calculations the cooling pipes with rectangular cross section have better efficiency because of it's further surface connection. Also, better cooling of rectangular cross section pipes decrease von-mises stress considerably.

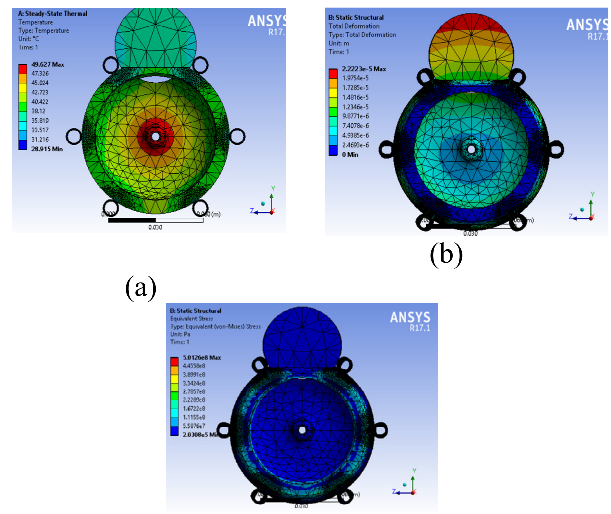

Figure 4: Cooling pipes with circular cross section (a) Temperature, (b) Deformation, (c) Von-mises Stress.

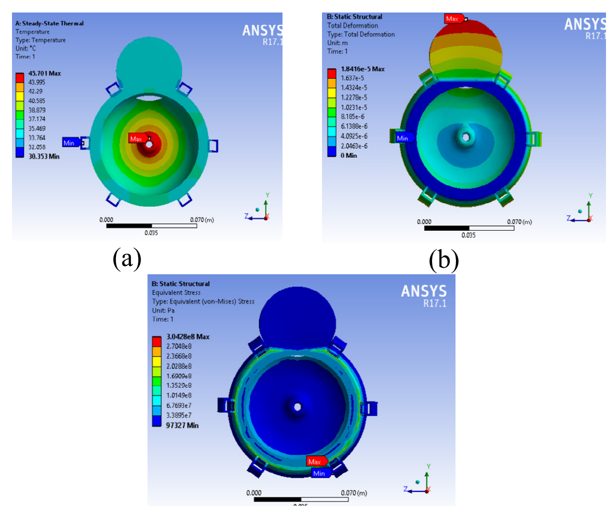

Figure 5: Cooling pipes with rectangular cross section (a) Temperature, (b) Deformation, (c) Von-mises Stress.

CONCLUSION

The effect of variation of Geometrical parameters of side coupled RF cavity was evaluated on resonant frequency and E_{max}/E_0. Also, percent of RF power loss on each surface of RF cavity was calculated with Superfish. In addition thermal and mechanical simulations were done by cooling pipes with two different types of cross sections.

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