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MICROPHONIC NOISE SUPPRESSION WITH OBSERVER BASED FEEDBACK*

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Abstract

In particle accelerators it's important to have a stable accelerating system for the beams of particles. The detuning of superconducting radio frequency (SRF) cavities is mainly caused by the Lorentz force. The detuning is the radiation pressure induced by a high radio frequency (RF) field, and microphonic noise are environmental vibrations that induce undesirable unwanted signals. These factors are described by a second order differential equation of the mechanical vibration modes of the cavity. In this paper the three dominant mechanical vibration modes for the system has been considered, then an observer based feedback control scheme has been designed based on input-output linearization. It is shown through simulation studies that the proposed control technique can successfully the suppress the microphonics due to the SRF cavity's dynamic.

INTRODUCTION

In a radio frequency(RF) cavity, the electromagnetic field which induces surface current causes the surface charges on the wall of the cavity. Also this electromagnetic field exerts a Lorentz force on the currents induced in a thin surface layer. Interaction of the surface currents with a standing wave inside cavity generates a pressure, which mechanically deforms the cavity and detunes it from the nominal frequency. This effect is known as Lorentz force detuning (LFD). The Super-Conducting Radio Frequency(SRF) cavities, when compared to RF cavities, have much higher sensitivity to microphonics noise and Lorentz Force. Microphonics are the time domain variations in cavity frequency driven by external vibrational sources and instability and transients due to control. Microphonics are effects due to vibrations which can appear in a wide range of amplitudes at low frequency, mostly in the range of zero to 1 kHz. Detuned cavities are more expensive to build and to operate. Resonance control is one of the most important parts of these SRF cavity systems [1–4] and there is a necessity for implementing microphonic noise cancellation in these systems. Vibration damping is required to improve the beam energy stability [5]. Different studies on supression of mechanical vibration has been conducted in accelerator labs all around the world [6–10]. In [11] studies show that feedback and feed-forward methods are necessary to suppress these unknown unwanted mechanical vibrations. The output voltage of the controller is being excreted to a piezo electric input which is attached to the cavity and the correspondent piezo displacement result from input voltage signal will damp the mechanical vibration to the cavity.

SRF CAVITY

The model of the cavity under study is based on the RLC description of the cavity's RF passband modes [12–15]. An equivalent electric circuit can be used to model the SRF cavity, where the RF signal generator and beam are represented as current generators. A model for describing the behaviour of the RF field envelope with time can be directly derived from the circuit with the cavity related parameters using Kirchhoff's law. Hence a second order differential equation for the cavity voltage is obtained as follows

$$\ddot{V}_{cav}(t) + \frac{\omega_0}{Q_L}\dot{V}_{cav}(t) + \omega_0^2 V_{cav}(t) = \frac{R_L\omega_0}{Q_L}(\dot{I}_g(t) - \dot{I}_b(t)), \tag{1}$$

where V_{cav} is the voltage signal within the cavity, I_g is the input current generator signal, I_b is the beam current signal, and $I_{in} = I_g - I_b$ is the input current signal to the circuit. The half-bandwidth of a resonance cavity is defined as the point where the power ($\propto V^2$) drops by −3 dB. The half-bandwidth can be expressed by

$$\omega_{1/2} = \frac{\omega_0}{2Q_L}. \tag{2}$$

The envelope field vector is described by its real and imaginary components or phasor representation. In RF control theory, these parameters are called in-phase or real part "re" and in-quadrature or imaginary part "im". The time varying behaviour of a Continuous Wave RF driven cavity is described by the field's vector envelope $V_{re}(t)$ and $V_{im}(t)$ [16]:

$$V_{cav} = \begin{bmatrix} V_{cav,re} \\ V_{cav,im} \end{bmatrix}, \tag{3}$$

$$I_{in} = \begin{bmatrix} I_{g,re} - I_{b,re} \\ I_{g,im} - I_{b,im} \end{bmatrix} = \begin{bmatrix} I_{re} \\ I_{im} \end{bmatrix}. \tag{4}$$

We have then

$$\begin{bmatrix} \dot{V}_{cav,re} \\ \dot{V}_{cav,im} \end{bmatrix} = \begin{bmatrix} -\omega_{\frac{1}{2}} & -\Delta\omega_m \\ \Delta\omega_m & -\omega_{\frac{1}{2}} \end{bmatrix} \begin{bmatrix} V_{cav,re} \\ V_{cav,im} \end{bmatrix} + \omega_{1/2} R_L \begin{bmatrix} I_{re} \\ I_{im} \end{bmatrix}. \tag{5}$$

MECHANICAL MODEL

TRIUMF's SRF resonator has an extremely high loaded quality factor $Q_L \sim 3\times10^6$ and a narrow bandwidth of about

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200 Hz. Hence, the cavity is very sensitive to the mechanical distortion caused by microphonics and the Lorentz force, changing the resonator's frequency. Due to the Lorentz force, the cavity resonance frequency changes with the rising field gradient. The mechanical model of the SRF cavity describes the Lorentz force detuning, which is a function of the square of time varying field gradient [17, 18]. It is based on the relationship of cavity's mechanical modes with the resonance frequency ω_m and the mechanical quality factor Q_m for the given mode. The time domain description of the mechanical system is given by

$$\begin{aligned}&\Delta\ddot{\omega}_{m,i}(t) + \frac{\omega_{m,i}}{Q_{m,i}}\Delta\dot{\omega}_{m,i}(t) + \omega_{m,i}^2\Delta\omega_{m,i}(t) \\ &= -\,k_{LFi}2\pi\omega_{m,i}^2 F(t).\end{aligned} \tag{6}$$

The total detuning experienced by the cavity is the sum of the individual terms as follows

$$\Delta\omega_{cav}(t) = \sum_i \Delta\omega_{m,i}(t). \tag{7}$$

Each of the mechanical modes is driven by the square of the cavity field gradient E_{acc} and microphonics vibration V_{mic}. Lower mechanical modes introduce instability into the system. Thus the state space mechanical model considering one mechanical mode is given by

$$\begin{aligned}\begin{bmatrix}\Delta\dot{\omega}_m(t)\\ \Delta\ddot{\omega}_m(t)\end{bmatrix} &= \begin{bmatrix}0 & 1\\ -\omega_m^2 & -\frac{\omega_m}{Q_m}\end{bmatrix}\begin{bmatrix}\Delta\omega_m(t)\\ \Delta\dot{\omega}_m(t)\end{bmatrix} + \begin{bmatrix}0\\ -k_{LF}2\pi\omega_m^2\end{bmatrix}\\ &\times(E_{acc}^2 + V_{mic}),\end{aligned} \tag{8}$$

where $E_{acc}^2 \propto V_{cav}^2$. Considering all mechanical modes, we have

$$\begin{aligned}\begin{bmatrix}\Delta\dot{\omega}_{m,1}\\ \Delta\ddot{\omega}_{m,1}\\ \vdots\\ \vdots\\ \Delta\dot{\omega}_{m,n}\\ \Delta\ddot{\omega}_{m,n}\end{bmatrix} &= \begin{bmatrix}0 & 1 & \cdots & 0 & 0\\ -\omega_{m,1}^2 & -\frac{\omega_{m,1}}{Q_{m,1}} & \cdots & 0 & 0\\ \vdots & \vdots & \vdots & \vdots & \vdots\\ \vdots & \vdots & \vdots & \vdots & \vdots\\ 0 & 0 & \cdots & 0 & 1\\ 0 & 0 & \cdots & -\omega_{m,n}^2 & -\frac{\omega_{m,n}}{Q_{m,n}}\end{bmatrix}\\ &\times\begin{bmatrix}\Delta\omega_{m,1}\\ \Delta\dot{\omega}_{m,1}\\ \vdots\\ \vdots\\ \Delta\omega_{m,n}\\ \Delta\dot{\omega}_{m,n}\end{bmatrix} + 2\pi\begin{bmatrix}0\\ -k_{LF1}\omega_{m,1}^2\\ \vdots\\ \vdots\\ 0\\ -k_{LFn}\omega_{m,n}^2\end{bmatrix}\\ &\times(V_{cav}^2 + V_{mic}).\end{aligned} \tag{9}$$

where $\Delta\omega_k(t)$ is the detuning, ω_m is the eigen-mode angular frequency, Q_m is the quality factor of the mode, and k_{LF} is the Lorentz force constant of each mode.

CONTROLLER DESIGN

Resonance control is an important part of every SRF cavity system. In a particle accelerator, the frequency of the system should be controlled in order to have the highest amplitude of the signal with preferably least phase difference. The controller should be stable to be desirable to cancel the unwanted detuning frequency. According to Lyapunov stability analysis, the stability of (9) is not affected by the presence of $\Delta\omega$ and this ensure the stability of the system. Therefore, in control design the focus is on the mechanical system given in (6). For notational simplicity, the equation (6) can be rewritten as follows

$$\begin{aligned}\Delta\ddot{\omega}_i =& -\frac{\omega_{m,i}}{Q_{m,i}}\Delta\dot{\omega}_i - \omega_{m,i}^2\Delta\omega_i\\ &- 2\pi k_{LFi}\omega_{m,i}^2(V_p^2 + V_{mic} + u_p)\\ =& f_i(\Delta\omega_i, u_p), i = 1, \dots, N;\end{aligned} \tag{10}$$

$$y_p = \Delta\omega = \sum_{i=1}^{N}\Delta\omega_i, \tag{11}$$

where u_p and y_p are the system input and output, respectively.

The control objective is to bring the output of the system y to zero. Note that this second order dynamic system is a SISO system. Let us use the input-output linearization method by obtaining the second-order time derivative of the output, which yields

$$\ddot{y}_p = \sum_{i=1}^{N}\Delta\ddot{\omega}_i. \tag{12}$$

By imposing

$$k_p v_p + k_v v_v = \sum_{i=1}^{N} f_i(\Delta\omega_i, u_p), \tag{13}$$

the input-output dynamics become

$$\ddot{y}_p = k_p v_p + k_v v_v, \tag{14}$$

$$\begin{aligned}\begin{bmatrix}\Delta\dot{\omega}_m(t)\\ \Delta\ddot{\omega}_m(t)\end{bmatrix} &= \begin{bmatrix}0 & 1\\ -\omega_m^2 & -\frac{\omega_m}{Q_m}\end{bmatrix}\begin{bmatrix}\Delta\omega_m(t)\\ \Delta\dot{\omega}_m(t)\end{bmatrix} + \begin{bmatrix}0\\ -k_{LF}2\pi\omega_m^2\end{bmatrix}\\ &\times u,\end{aligned} \tag{15}$$

$$u = \begin{bmatrix}k_p & k_v\end{bmatrix}\begin{bmatrix}\Delta\omega_m(t)\\ \Delta\dot{\omega}_m(t)\end{bmatrix} \tag{16}$$

$$\begin{aligned}\begin{bmatrix}\Delta\dot{\omega}_m(t)\\ \Delta\ddot{\omega}_m(t)\end{bmatrix} &= \begin{bmatrix}0 & 1\\ -\omega_m^2 & -\frac{\omega_m}{Q_m}\end{bmatrix}\begin{bmatrix}\Delta\omega_m(t)\\ \Delta\dot{\omega}_m(t)\end{bmatrix} + 2\pi\begin{bmatrix}0\\ -k_{LF}\omega_m^2\end{bmatrix}\\ &\times\begin{bmatrix}k_p & k_v\end{bmatrix}\begin{bmatrix}\Delta\omega_m(t)\\ \Delta\dot{\omega}_m(t)\end{bmatrix},\end{aligned} \tag{17}$$

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$$\begin{bmatrix} \Delta\dot{\omega}_m(t) \\ \Delta\ddot{\omega}_m(t) \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_m^2 & -\frac{\omega_m}{Q_m} \end{bmatrix} \begin{bmatrix} \Delta\omega_m(t) \\ \Delta\dot{\omega}_m(t) \end{bmatrix} + 2\pi \times \begin{bmatrix} 0 & 0 \\ -k_{LF}\omega_m^2 k_p & -k_{LF}\omega_m^2 k_v \end{bmatrix} \begin{bmatrix} \Delta\omega_m(t) \\ \Delta\dot{\omega}_m(t) \end{bmatrix}, \tag{18}$$

$$\begin{bmatrix} \Delta\dot{\omega}_m(t) \\ \Delta\ddot{\omega}_m(t) \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_m^2 - 2\pi k_{LF}\omega_m^2 k_p & -\frac{\omega_m}{Q_m} - 2\pi k_{LF}\omega_m^2 k_v \end{bmatrix} \times \begin{bmatrix} \Delta\omega_m(t) \\ \Delta\dot{\omega}_m(t) \end{bmatrix}, \tag{19}$$

$$S^2 + (\frac{\omega_m}{Q_m} + 2\pi k_{LF}\omega_m^2 k_v)S + 2\pi k_{LF}\omega_m^2 k_p = 0 \tag{20}$$

We design k_p and k_v from solving the above second order equation to have stable system, linearize the system and the result would be a second-order LTI system. The linearization controller u_p can be derived from (10), which is given by

$$u_p = k_p v_p + k_v v_v - \frac{\sum_{i=1}^{N}(2\zeta_{m,i}\omega_{m,i}\Delta\dot{\omega}_i + \omega_{m,i}^2\Delta\omega_i)}{\sum_{i=1}^{N} 2\pi k_{LFi}\omega_{m,i}^2} - (V_p^2) + V_{mic}) \tag{21}$$

$$u_p = k_p v_p + k_v \dot{v}_p - \frac{\sum_{i=1}^{N}(2\zeta_{m,i}\omega_{m,i}\Delta\dot{\omega}_i + \omega_{m,i}^2\Delta\omega_i)}{\sum_{i=1}^{N} 2\pi k_{LFi}\omega_{m,i}^2} - (V_p^2 + V_{mic})) \tag{22}$$

This technique of observer based feedback control has been used in the design of linear (auxiliary) control v_p and v_v in state-feedback control.

MODEL ANALYSIS AND SIMULATION

We need to find out at least 3 dominant mechanical modes in response to mechanical microphonics and vibration. Through some measurement techniques and exciting the cavity with shaker, 3 dominant mechanical modes were detected by using a phase noise analyzer [19]. These microphonics vibration has been measured using a signal analyzer. Therefore the dynamics of the system are defined as a combination of the three mechanical modes and their respective coefficients.

The main source of vibration and microphonics identified as cooling water supply and cryomodule valves. The state-space specification of the linearized system is given by

$$\dot{x} = Ax + Bu \tag{23}$$

$$y = Cx \tag{24}$$

where

$$x = \begin{bmatrix} \Delta\omega_{m,1} \\ \Delta\dot{\omega}_{m,1} \\ \Delta\omega_{m,2} \\ \Delta\dot{\omega}_{m,2} \\ \Delta\omega_{m,3} \\ \Delta\dot{\omega}_{m,3} \end{bmatrix},$$

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\omega_{m,1}^2 & -\frac{\omega_{m,1}}{Q_{m,1}} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & -\omega_{m,2}^2 & -\frac{\omega_{m,2}}{Q_{m,2}} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & -\omega_{m,3}^2 & -\frac{\omega_{m,3}}{Q_{m,3}} \end{bmatrix},$$

$$B = \begin{bmatrix} 0 \\ -k_{LF1}\omega_{m,1}^2 \\ 0 \\ -k_{LF2}\omega_{m,2}^2 \\ 0 \\ -k_{LF3}\omega_{m,3}^2 \end{bmatrix},$$

$$C = \begin{bmatrix} 1 & 0 & 1 & 0 & 1 & 0 \end{bmatrix}.$$

The system parameters are defined in Table 1.

Table 1: Three Mechanical Modes Parameters

	Mode1	***Mode2***	***Mode3***
ω_m	39	157	224
Q_m	100	128	200
k_{LF}	0.4	0.3	0.7

It is straightforward to verify that for the linearized system, (A, B) is completely controllable and (A, C) is completely observable.

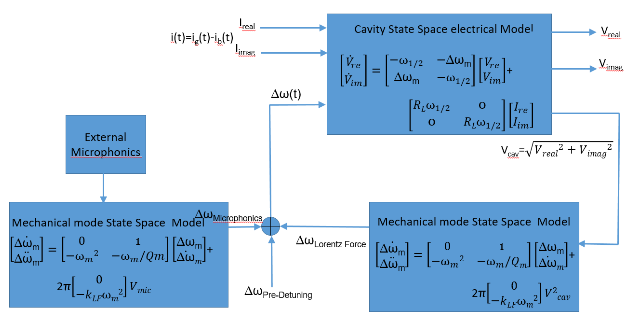

Figure 1: Coupling between electrical and mechanical model for one mode.

Figure 1 shows the model for just one mechanical mode. As shown in Fig. 1, there are three sources for detuning $\Delta\omega$. A constant for pre-detuning, microphonics, which is comprised of multiple sinusoidal signals with the frequencies in

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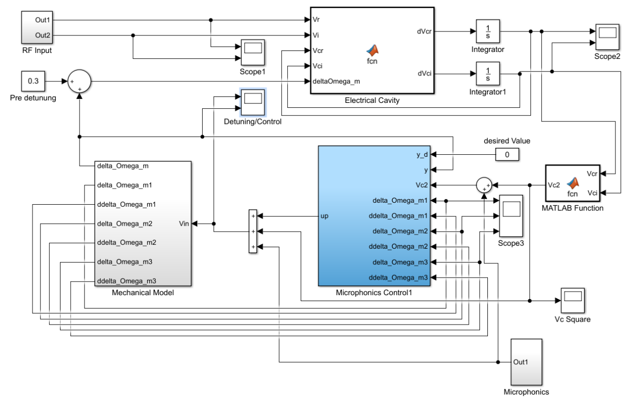

Figure 2: Simulink Model for the system.

the range of kHz and Lorentz force detuning. This model is extended into three mechanical modes.

In Fig. 2, the proposed controller designed for three modes The desired signal for the controller is zero which means that detuning signal should go to zero. Pre-detuning $\Delta\omega_{pre}$= 0.3;
R_L= 50Ω; $\omega_0 = 141 \times 10^6$ Hz; $Q_L = 3 \times 10^6$; $\omega_{1/2} = \omega_0/2Q_L$; The controller gains are $k_p = 0$ and $k_v = 10^5$.
All three mechanical frequencies and coefficients are considered according to the data in Table 1.

Input signal to mechanical dynamic V_{in} and the detuning signal after control are shown in Fig. 3.

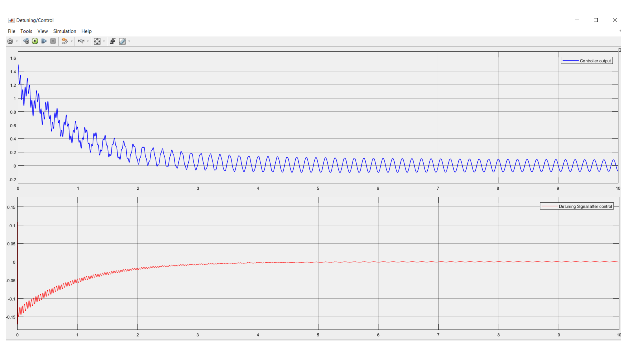

Figure 3: Input signal to the mechanical dynamic and Detuning Signal after Control.

It is observed that the controller can successfully cancel the microphonics and Lorentz force detuning terms. To cancel the unwanted vibrations, the most effective way is installing a piezo electric to the cavity. Applying an input voltage to the piezo electric produce a displacement proportional to the voltage. The signal from the controller is the input signal to the piezo which installed to the cavity. The result displacement will reduce the main microphonics peaks in the cavity and damp the vibrations which are coming from klystron water pumps.

As shown in Fig. 4 the piezo in installed on tuning tower to damp the vibrations.

Figure 4: Installed Piezo on tuning tower

CONCLUSION

In this paper the superconducting RF cavity's dynamic obtained from measuring the signals and microphonics by spectrum analyzer at TRIUMF. Three dominant mechanical vibrations considered as three modes for the mechanical dynamic of the cavity. These microphonics are low modes frequencies which are mostly from the environmental vibrations. A control scheme has been obtained for microphonics noise and Lorentz force detuning in a SRF cavity with these three dominant mechanical modes. The observer based feedback control technique and input-output linearization method by obtaining the second-order time derivative of the output is applied as the controller in this paper. This is an efficient algorithm for active noise suppression in superconducting cavities. By applying this feedback method the three dominant microphonic noise has been suppressed in the simulation. The simulation results indicate the effectiveness of the proposed controller that controls the unwanted detuning to the desired value zero. This obtained signal from controller is a voltage that is being applied as input of a piezo which installed to the cavity and the displacement of the piezo electric result from the controller signal suppresses the mechanical vibration.

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