OPTIMAL DESIGN OF VOICE COIL MOTOR FOR APPLICATION IN ACTIVE VIBRATION ISOLATION

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ABSTRACT
Vibration isolators typically reduce the vibration transmitted to the nano-precision measuring instrument providing managed stiffness and damping. Active isolators use feedback to provide the necessary control signal. In this paper we propose an optimized design of a Voice coil motor that can be used to implement a suitable controller for active isolation to attenuate the low frequency vibration.

1. INTRODUCTION
Precision measuring instruments such as Atomic Force Microscopes with resolution in nano-meter range are always affected by ground borne vibrations that have typical amplitude in sub micrometer region. A vibration isolation system reduces the effect of such disturbance to transmit to the instrument usually by using a spring damper system with very low cut off frequency which works as a low pass filter for the disturbance. A passive system can be designed to provide the necessary attenuation to this vibration. To effectively attenuate the low frequency ground vibration, a hybrid active passive system is necessary([1], [3]).

The passive system attenuates high frequency disturbance with reasonable attenuation rate and then the active system works to attenuate the low frequency vibration effect. The active system implements active control using sensors and actuators. The actuator of active vibration isolator can be of several types: mechanical mechanisms, piezoelectric actuators, pneumatic springs, electromagnetic motors and electrical linear motor.

This research proposes implementation of actuator, a Voice Coil Motor (VCM) on such an active vibration isolator which works to control the vibration typically in range from 0.1 to 100 Hz. The VCM was designed and optimized to provide necessary feedback force to nullify vibration effect. Necessary cost function was evaluated depending on passive isolation effect and system dynamics. Force generation was calculated using magnetic flux reluctance method. Optimized result was then evaluated comparing with the simulation result.

2. DESIGN OF THE SYSTEM
The isolation system is composed of four elastomer mounts to support upper plate with the instrument to be isolated(around 150Kg) along with three vertical and three horizontal VCMs mounted such that they can control all six rigid body modes using feedback signal from the accelerometers connected to the upper plate as shown in figure 1. Accelerometers measure transmitted vibration from ground to upper plate and sends feedback signal to the actuators.

![Fig. 1: Complete system with actuators and sensors](image)

![Fig. 2: (a)Vertical actuator and (b)Horizontal actuator](image)

3. OPTIMIZATION OF VERTICAL VCM
In order to optimize these VCMs, a cost function needs to be defined along with design parameters that can be varied to arrive at the desired optimization value. Force requirement depends on passive system dynamics as the actuators work against stiffness and
damping of the passive elastomers to provide necessary dynamic force to attenuate vibration.

The complete system model with two inputs (control signal input, \( F_u(s) \) and vibration input, \( b(s) \)) contributing to output variable, \( p(s) \) is as follows:

\[
\begin{align*}
(\overline{M} s^2 + [C] s + [K]) p(s) &= \overline{C} s b(s) + [K] b(s) + F_u(s) \\
p(s) &= \frac{1}{(\overline{M} s^2 + [C] s + [K])} F_u(s) + \frac{[C] s b(s) + [K] b(s)}{(\overline{M} s^2 + [C] s + [K])} b(s) \quad (1)
\end{align*}
\]

\( M, C \) and \( K \) are mass, damping and stiffness matrix respectively.

To find the dynamic force required in vertical direction, we use single degree of freedom model with \( K \) and \( C \) being the vertical directional stiffness and damping respectively as:

\[
K = 8.6 \times 10^5 \text{ N/m} \quad \text{and} \quad C = 1.42 \times 10^3 \text{ Ns/m}
\]

Fig. 3: Single DOF system with actuation force \( F_t \).

From figure 3 we get force requirement for the vertical actuator as follows [2]:

\[
\begin{align*}
M s^2 P(s) + C s P(s) - B(s) + K (P(s) - B(s)) &= F_v \\
3 F_v &= 3 n B_j i = 3 K_j i \\
\Rightarrow M s^2 T(s) X_g + C s X_g (T(s) - 1) + K X_g (T(s) - 1) &= 3 F_v
\end{align*}
\]

where \( T(s) = \text{Transmissibility of passive system} \)

\[
P(s) = \frac{C s + K}{M s^2 + C s + K}
\]

Where \( K_j \) = Force constant \( = n^2 B_j \). \( n \) = No. of coil turns, \( B_j \) = Flux density at the air gap and \( l \) = effective length of coil.

Since control bandwidth is 0.1-100.0 Hz, to find maximum force requirement, 100 Hz frequency was considered since at that particular frequency maximum acceleration and velocity is transferred to the upper plate considering peak to peak ground vibration displacement \( B(s) = 100 \) micro meter.

Transmissibility of passive system also varies with frequency and it can be calculated from passive system dynamics as plotted in figure 4 and the value is \( T(s) = 0.0143 = -36.9 \text{dB} \)

Fig. 4: Transmissibility of passive system (vertical direction)

Maximum force requirement is:

\[
3 F_v = M s^2 X_g + C s X_g (T(s) - 1) + K X_g (T(s) - 1) \Rightarrow F_v = -29.3 \text{ N}
\]

The VCM has to produce less than or equal to the maximum force requirement \( F_v \) = 30 N. So the cost function is:

\[
\begin{align*}
\text{minimize} \quad \| F_v - 3 F_{vCM} \|_2^2 &\Rightarrow \text{minimize} \quad (88 - 3 F_{vCM})^2
\end{align*}
\]

To choose the constraints we decide that the designed actuator should have compact size along with maximum allowable temperature, saturated flux density and maximum current through the coil as constraints. The list is as follows:

**Table 1: Cost function and constraints**

<table>
<thead>
<tr>
<th>COST FUNCTION</th>
<th>( \text{minimize} \quad (88 - 3 F_{vCM})^2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>CONSTRAINTS</td>
<td></td>
</tr>
<tr>
<td>Temperature (T)</td>
<td>( \leq 70 ) (C)</td>
</tr>
<tr>
<td>VCM size (Height\times Length\times width)</td>
<td>( \leq 40 ) (mm)\times 47 (mm) \times 46 (mm)</td>
</tr>
<tr>
<td>Generated force (N)</td>
<td>( \leq 30 ) N</td>
</tr>
<tr>
<td>Saturated Flux density (Hs)</td>
<td>( \leq 1.8 ) T</td>
</tr>
<tr>
<td>Coil Current (I)</td>
<td>( \leq 1.0 ) Amp.</td>
</tr>
</tbody>
</table>

In order to optimize size of VCM, some geometric parameters are considered constant and the others are varied at some defined range with the objective to minimize the cost function. Considering symmetric half part of the VCM, the geometric parameters are as shown below:

Fig. 5: Symmetric half structure with geometric parameters.

**Table 2: The fixed geometric parameters**

<table>
<thead>
<tr>
<th>Fixed constraints</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_y ) (Magnet length)</td>
<td>40.0 mm</td>
</tr>
<tr>
<td>( T_c ) (Coil thickness)</td>
<td>2.0 mm</td>
</tr>
<tr>
<td>( W_c ) (Coil width)</td>
<td>0.5 * Wm</td>
</tr>
<tr>
<td>( W_a ) (Width of side air gap)</td>
<td>0.1 * Wm</td>
</tr>
</tbody>
</table>
| \( T_y \) (central yoke thickness) | 2 \* T
| \( T_c \) (coil holder width) | 2.00mm |
| \( L_c \) (Coil side length) | 49.0mm |
| \( L_e \) (Coil opposite side length) | 14.00mm |
| \( L_e \) (Coil effective unit length) | 2 \* ((L_e-Tc)+(L_e-Tc)) |

Table 3 shows the design variables with their range.

**Table 3: Design Variables**

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Design range</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_m ) (Magnet thickness)</td>
<td>2.0 - 5.0 mm</td>
</tr>
<tr>
<td>( W_m ) (Magnet width)</td>
<td>10.0 - 25.0 mm</td>
</tr>
<tr>
<td>( T_y ) (Yoke thickness)</td>
<td>1.0 - 5.0 mm</td>
</tr>
<tr>
<td>( T_g ) (Air gap thickness)</td>
<td>5.0 - 6.0 mm</td>
</tr>
<tr>
<td>( d_c ) (coil diameter)</td>
<td>0.1 - 1.0 mm</td>
</tr>
<tr>
<td>( I_c ) (coil current)</td>
<td>0.3 - 1.0 Amp</td>
</tr>
</tbody>
</table>

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Geometric constraints:

\[ L_y = 40 \text{ mm} \quad \text{[combined length]} \]
\[ 14 \leq T_y \leq 47 \quad \text{[combined thickness]} \]
\[ 18 \leq W_y \leq 46 \quad \text{[combined width]} \]

So \( g(1) = 2T_y + W_y + 2T_y \leq 46 \text{mm} \Rightarrow g(1) = 2T_y + 2W_y \leq 46 \text{mm} \)
and \( g(2) = 2T_y + T_y + T_y > 47 \Rightarrow g(2) = 4T_y + 2(T_y + T_y) \leq 47 \text{mm} \)

Current constraint:

Current is considered to be less than or equal to 1.0Amp to reduce the effect of electrical dynamics (i.e. effect of coil resistance and inductance)

\[ I_c \leq 1 \]

\( g(3) : I_c - 1.0 \leq 0 \)

Heat dissipation constraint:

Heat dissipation is selected on elastomer temperature limit. Since coil current is responsible for heat dissipation, heat flow circuit is as shown in figure 7.

Fig. 6: Coil assembly and its cross section along with heat flow direction

Fig. 7: Heat flow circuit

Here \( h = \) Heat conductivity of air, \( k = \) Convection coefficient of coil holder material and \( A_s \) are respective cross sections.

From the figures we get the heat dissipation constraint

\[ q_{out} = q_{1out} + 2q_{2out} + q_{3out} + 2q_{5out} \leq q_s \]

\[ g(4) = \frac{q_{out}}{q_s} - 1.0 \leq 0 \text{ here } q_s = I_c^2 * R_c \]

To optimize system parameters, reluctance method was used. The equivalent reluctance circuit for the magnetic flux can be represented as in figure 8.

Fig. 8: Equivalent reluctance circuit of VCM

Relationship of different parameters in the reluctance circuit is as follows:

Fig. 9: Relationship of parameters

Implementing all relations we performed the optimization using Matlab toolbox with objective to minimize the cost function.

4. OPTIMIZATION RESULT

Optimization was performed using the design variables and minimizing the cost function where \( g(1), g(2) \) and \( g(3) \) are linear inequality constraints and \( g(4) \) is non linear inequality constraint.

To find the global converged value we iterate the optimization process using many initial value combinations of the design parameters within the bound and then the cost function value was plotted as in figure below for all the iterations.

Fig. 10: Global Minima

From figure 10 the global minima was found and the optimized design parameters were chosen with respect to this global minima. Result of the optimization process is as follows:

Fig. 11: Cost function convergence
Figure 11 shows the cost function convergence and the design variable convergence are shown below in figure 12.

The converged values are summarized in table 4.

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Design value after optimization</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tm (Magnet thickness)</td>
<td>4.2617 mm – 4.25 mm</td>
</tr>
<tr>
<td>Wm (Magnet width)</td>
<td>24.5851 mm – 24.6 mm</td>
</tr>
<tr>
<td>Tuy (Yoke thickness)</td>
<td>4.9527 mm – 4.95mm</td>
</tr>
<tr>
<td>Tg (Air gap thickness)</td>
<td>5.1207 mm = 5.1 mm</td>
</tr>
<tr>
<td>dc (coil diameter)</td>
<td>0.19 mm</td>
</tr>
<tr>
<td>ic (Coil current)</td>
<td>0.9592 Amp. = 0.96 Amp</td>
</tr>
</tbody>
</table>

Force generated is $F_{\text{VCM}} = 2*F = 29.12 \text{ N}$ and Flux density is also within saturation limit ($\geq 1.8T$).

5. SIMULATION

To verify the optimization result we perform FEM simulation using Maxwell software and compared the result. The simulation model and result is as follows:

Fig. 13: VCM model and flux distribution in the cross section.

Flux density distribution at the air gap is as follows:

Fig. 14: Flux density distribution at air gap

The simulation result is shown in table 5.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Percent error from optimization result</th>
</tr>
</thead>
<tbody>
<tr>
<td>N (No. of turns)</td>
<td>800</td>
<td>3.625</td>
</tr>
<tr>
<td>Force</td>
<td>13.5 N</td>
<td>-8.59</td>
</tr>
<tr>
<td>Flux Density</td>
<td>0.47 T</td>
<td>-4.25</td>
</tr>
</tbody>
</table>

From table 5 we see that simulation result conforms to the optimization result.

The horizontal VCM can also be optimized in a similar manner considering different parameters for the horizontal isolation.

6. CONCLUSION

The optimized design of a Voice Coil Motor was performed which can be used as an actuator for the Active Vibration Isolation system. The result of the optimization was verified using FEM simulation. This optimized VCM can easily be implemented to control the low frequency vibration typically within the range 0.1 to 100 Hz.

7. REFERENCE

