Linear Motor Active Damper (LMAD) for Vibration Reduction in Precision Manufacturing System

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Contents

1. Intro – Machining Vibration issue

bmmagazine

- **Chatter vibration**

- Free vibration
- Forced vibration

Widely developed

- **Self-excited vibration** TARGET

Made by interaction between tool & workpiece

- **Negative Effect of Chatter**

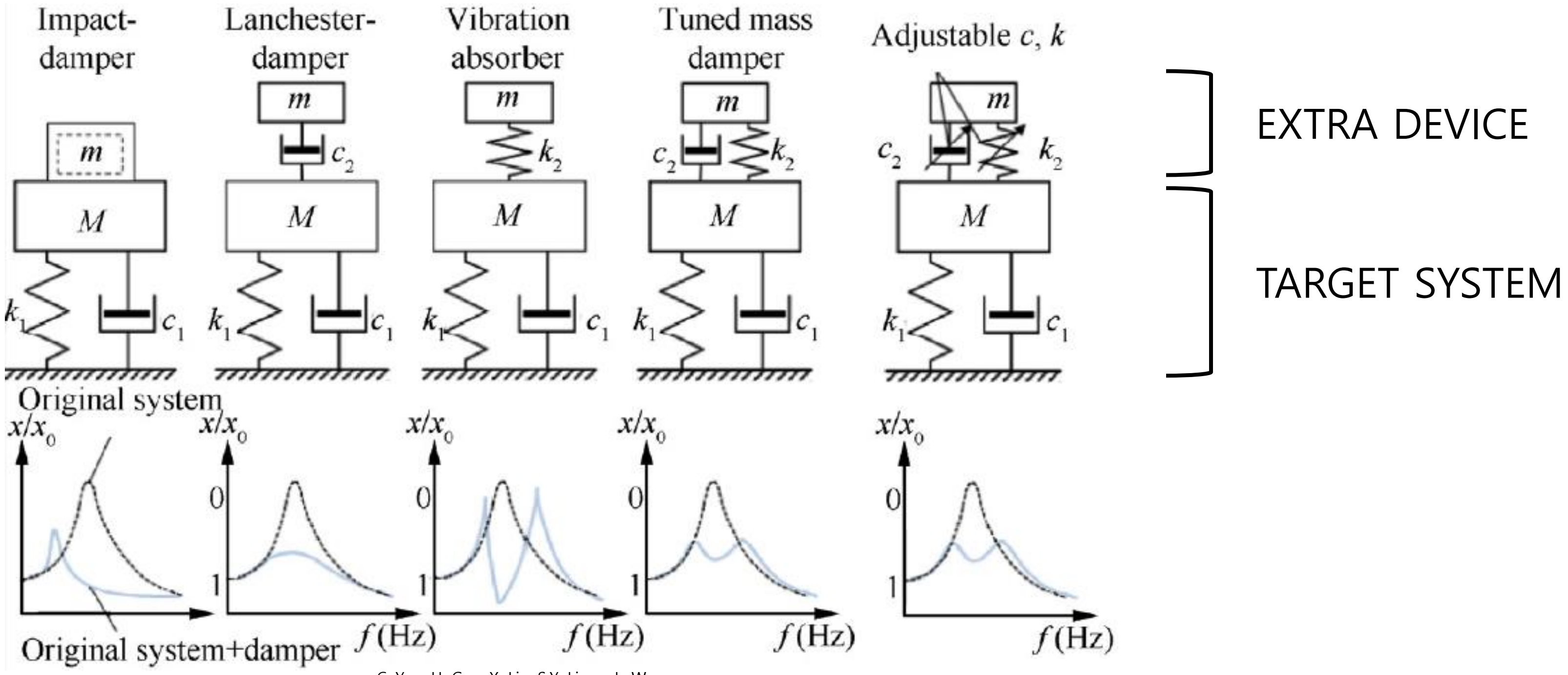
- Poor surface quality
- Inaccuracy
- Noise
- Machine tool damage
- And so on….

1. Intro – Passive strategies

Strategies for ensuring stable machining processes

- Using the lobbing effect
- Changing the system behavior
  - **Passive strategies**
  - Active strategies

- Use of extra devices (absorb energy)
- **Limited response to change after design**



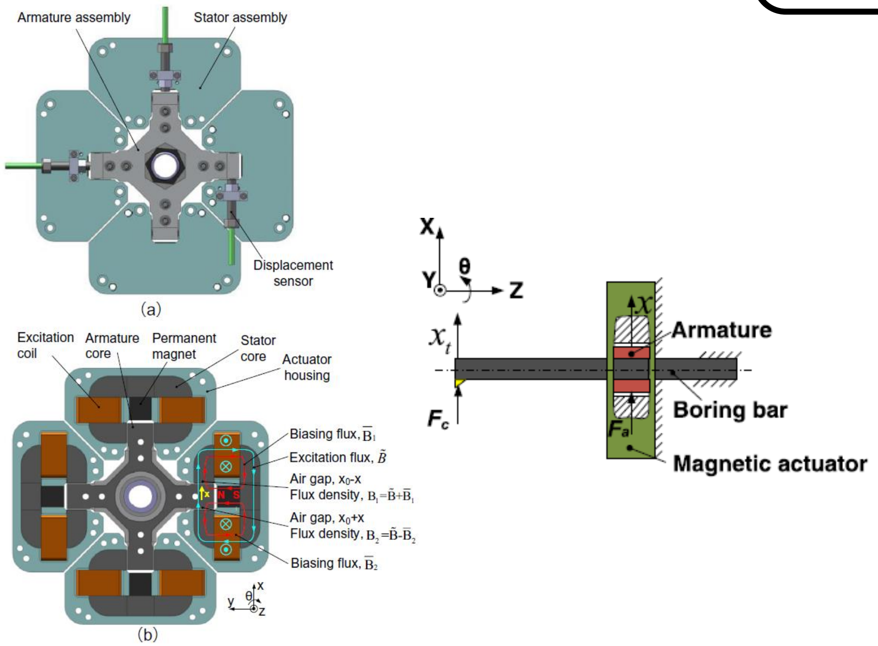
C. Yue, H. Gao, X. Liu, S.Y. Liang, L. Wang
A review of chatter vibration research in milling
Chin J Aeronaut, 32 (2) (2019), pp. 215-242

1. Intro – Active strategies

Strategies for ensuring stable machining processes

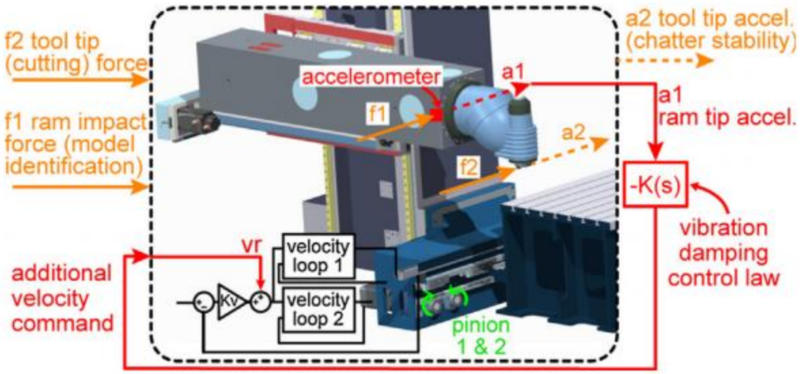
- Using the lobbing effect
- Changing the system behavior
  - Passive strategies
  - **Active strategies**

- Monitor dynamic behavior of tools
- Adjust with actuator
- **Active response to change**



Chen F, Liu G (2016) Active damping of machine tool vibrations and cutting force measurement with a magnetic actuator. Int J Adv Manuf Technol:1–10.

- Monitor : Displacement sensor
- Actuator : Magnetic actuator



Munoa J, Beudaert X, Erkorkmaz K, Lglesias A, Barrios A. Active suppression of structural chatter vibrations using machine drives and accelerometers. CIRP Ann- Manuf Technol 2015;64(1):385–8.

- Monitor : Accelerometer
- Actuator : tool's own drive

1. Intro – This research?

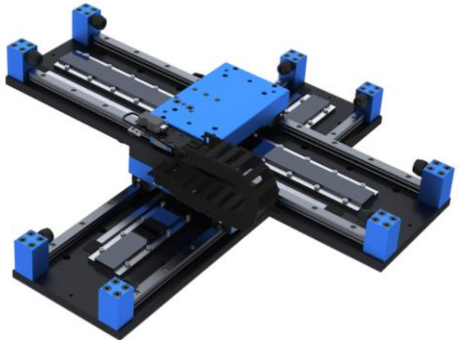
Active strategy

Accelerometer (Monitor)

Digiducer Model 333D01

+

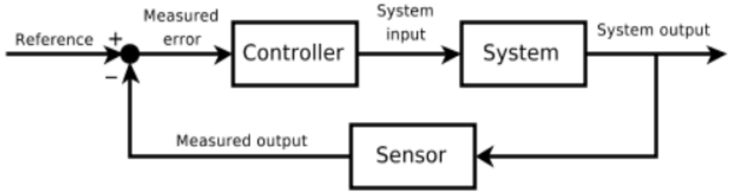
Linear Motor (Actuator)



https://www.linearmotiontips.com/new-generation-of-linear-motor-stages-from-yaskawa/

+

Direct Velocity Feedback (DVF)



https://electronics.stackexchange.com

Method of Vibration reduction

- Single frequency
- Multiple frequency

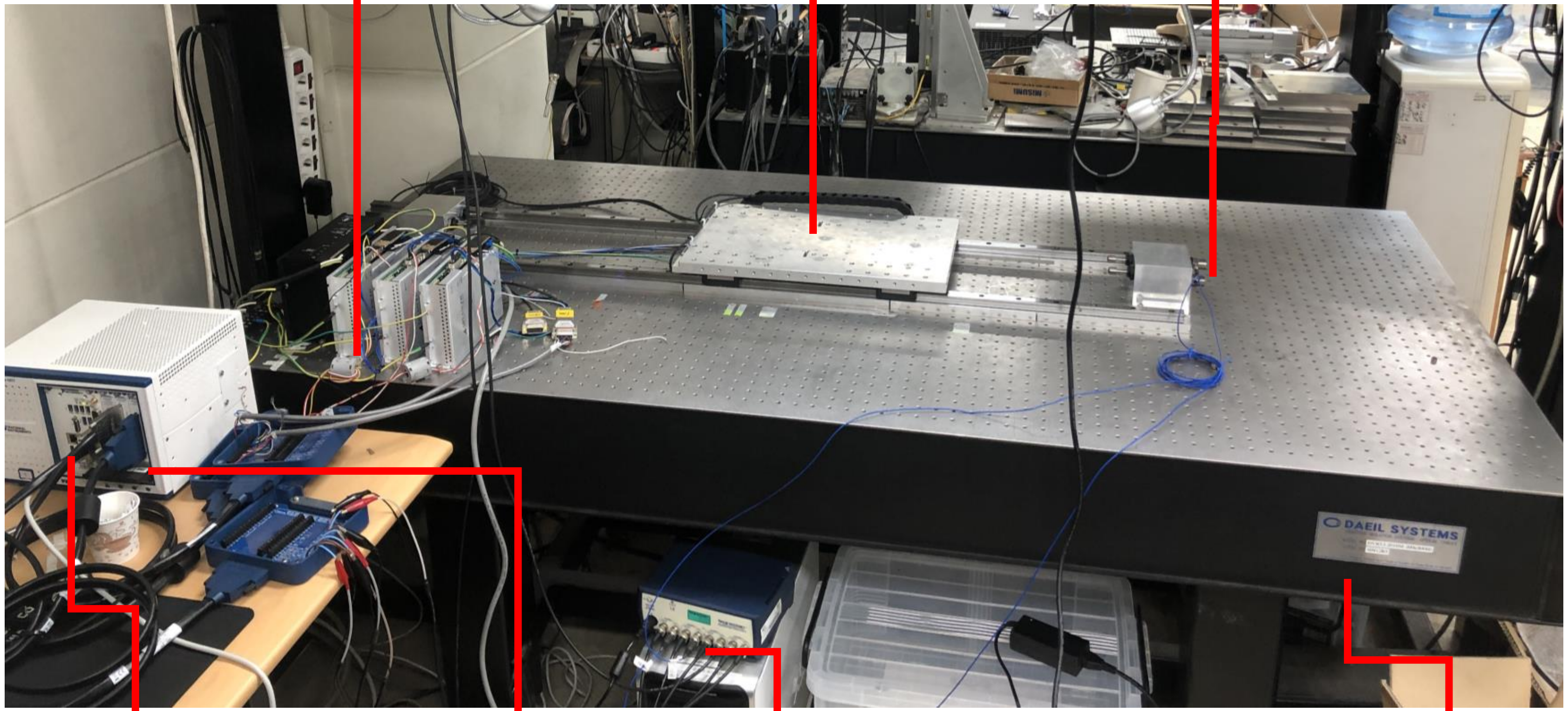
⇨ **Find basis for dealing with the Entire Frequency**

2. Analytical Modeling – Setup

Moving mass
(Rotor part of linear motor)

Motor driver
(current mode)

Accelerometer



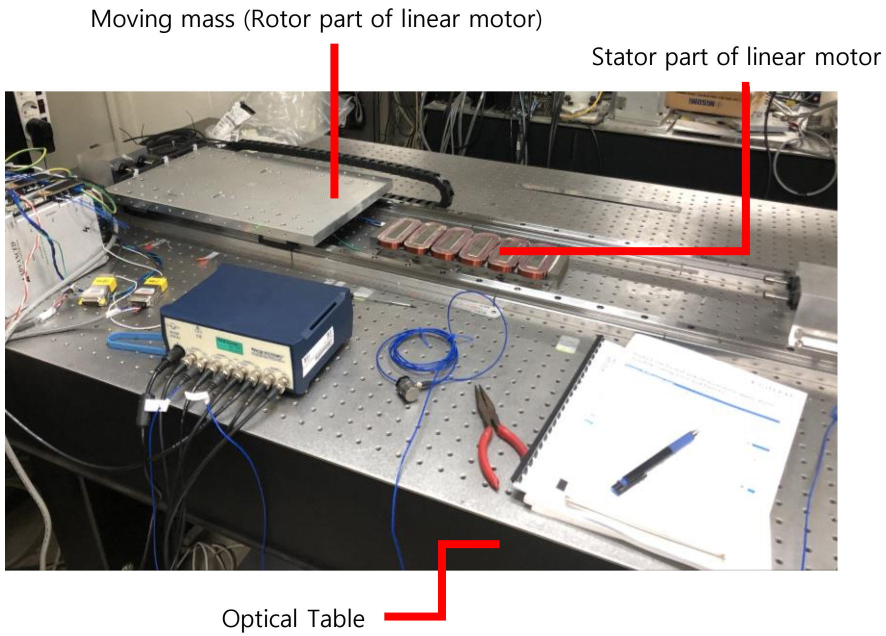
Signal Conditioner
(for Accelerometer)

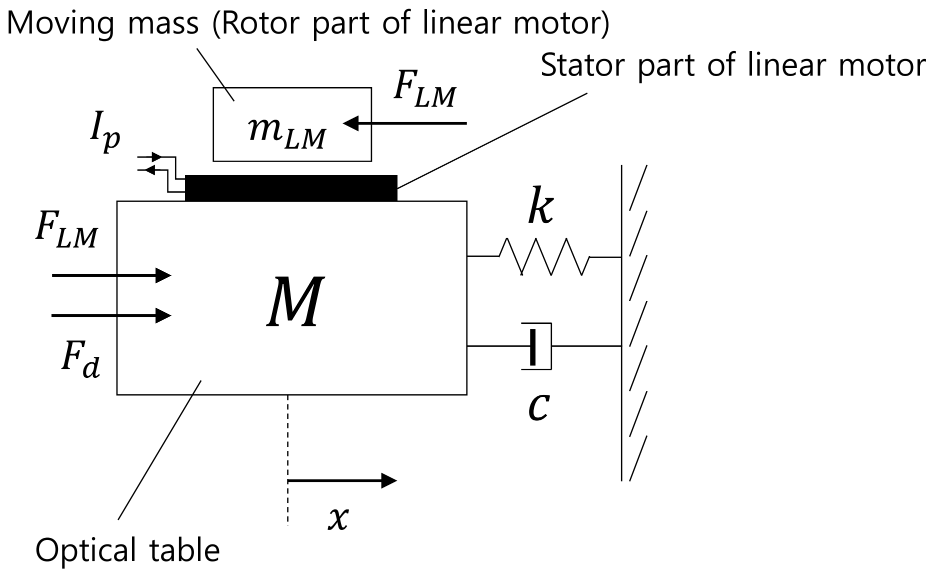
Real-time controller
(PXIe-8840)

Analog I/O module
(PXIe-6733, PXIe 6361)
FPGA module
(PXIe-7846R)

Optical Table
(Large Mass)

2. Analytical Modeling – System Modeling





I_p : Peak current of commutation law for the linear motor. Positive if F_{LM}>0

F_d : Disturbance force

Standard Second–order system of Optical table

$$\mathrm{M}\ddot{x} + \mathrm{c}\dot{x} + \mathrm{k}x = F_{\mathrm{LM}} + F_{\mathrm{disturbance}}$$

Mechanical governing equation of Linear Motor

$$m_{\mathrm{LM}}\ddot{x}_{LM} = \mathrm{K_i I_p} - \mathrm{K_d}\cancel{x}_{LM} - \mathrm{c_d}\cancel{\dot{x}}_{LM} \Rightarrow m_{\mathrm{LM}}\ddot{x}_{LM} = F_{\mathrm{LM}} = \mathrm{K_i I_p}$$

SJ Moon, TY Chung, CW Lim, DH Kim, "A linear motor damper for vibration control of steel structures", Mechatronics, 14(10), 1157-1181, 2004

$\mathrm{M_{LM}}$: moving mass

x_{LM} : Relative position of LM from the table

$\mathrm{K_i}$: linear motor force constant

$\mathrm{I_p}$: peak current of commutation law for the linear motor

2. Analytical Modeling – System Modeling(Identification)

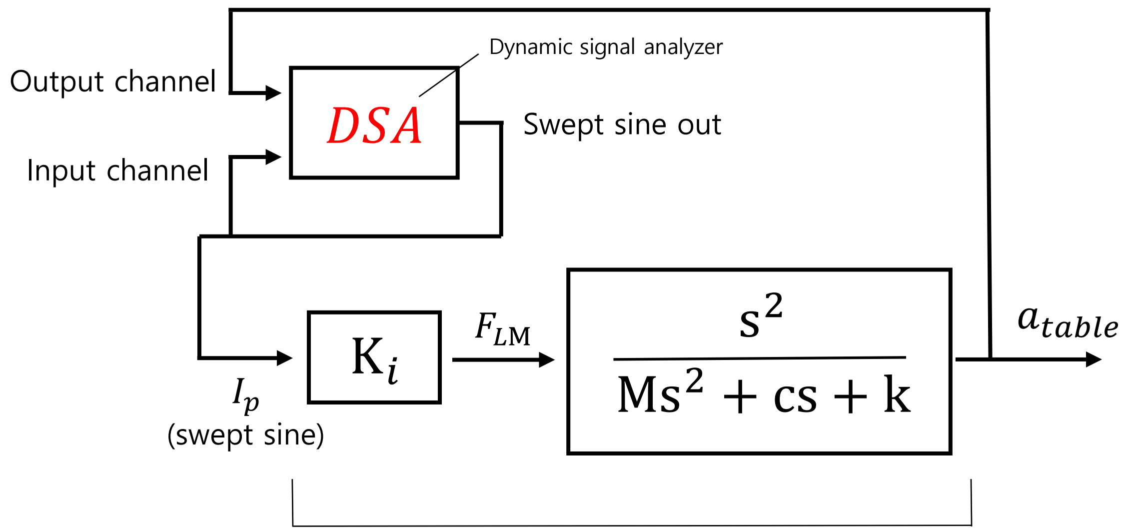
• DSA Experiment

- Frequency domain analysis for Unknown : c, k

Open−loop Transfer Function

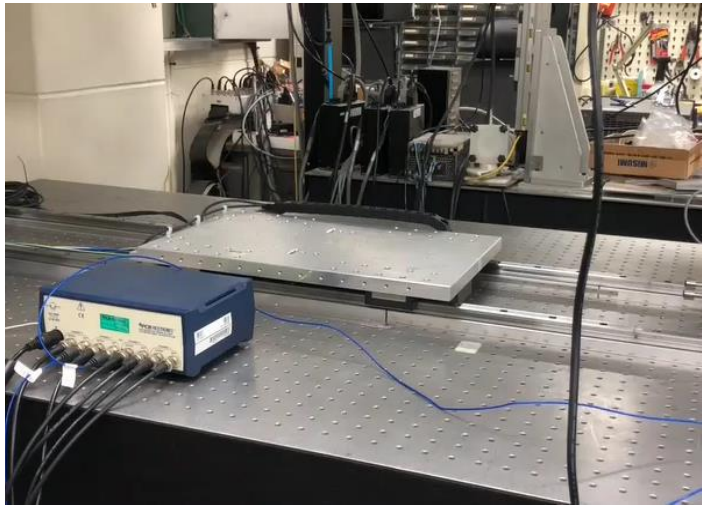
$$\frac{A(s)}{I_p(s)} = \frac{K_i s^2}{Ms^2 + cs + k}$$

Supported by TA



Open loop plant $G(s) = \frac{A(s)}{I_p(s)}$

(Sampling rate of 10kHz / Excitation amplitude of 3.0A for all frequencies)



Frequency 별로 Magnitude, Phase 데이터 수집

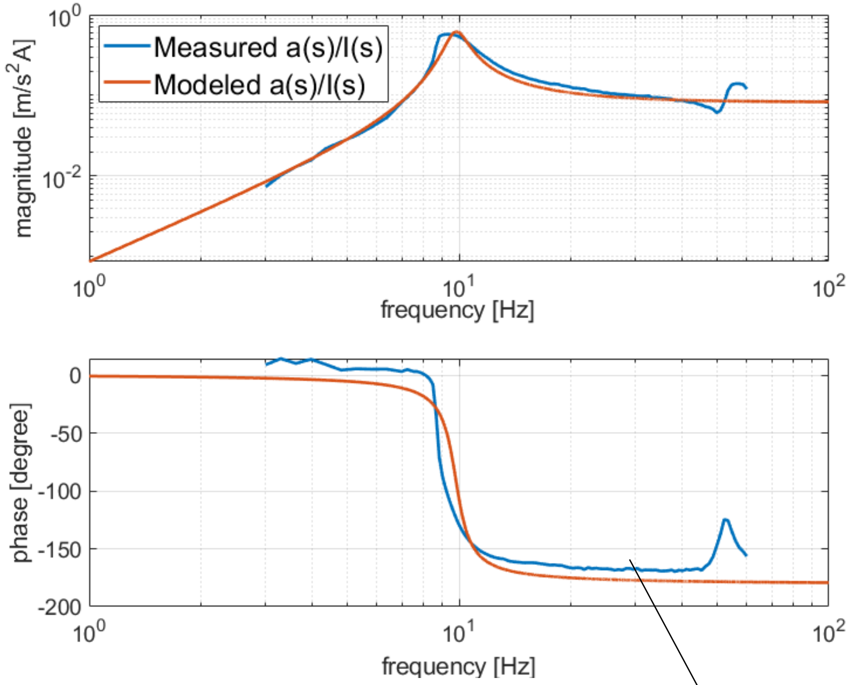
Natural frequency 근접 ⇨ Magnitude ⇧

2. Analytical Modeling – System Modeling(Identification)

- **Estimated parameters (DSA test)**

$M = 280kg,\ c = 2300\ Ns/m\ ,\ \mathrm{k}\ = 1{,}018{,}700\ N/m$

$(\ \mathrm{k}\ =\ \mathrm{M}(2\mathrm{w}_\mathrm{n}\pi)^2\ )$



flipped phase : measurement of $-A(s)/I_p(s)$

Open–loop Transfer Function

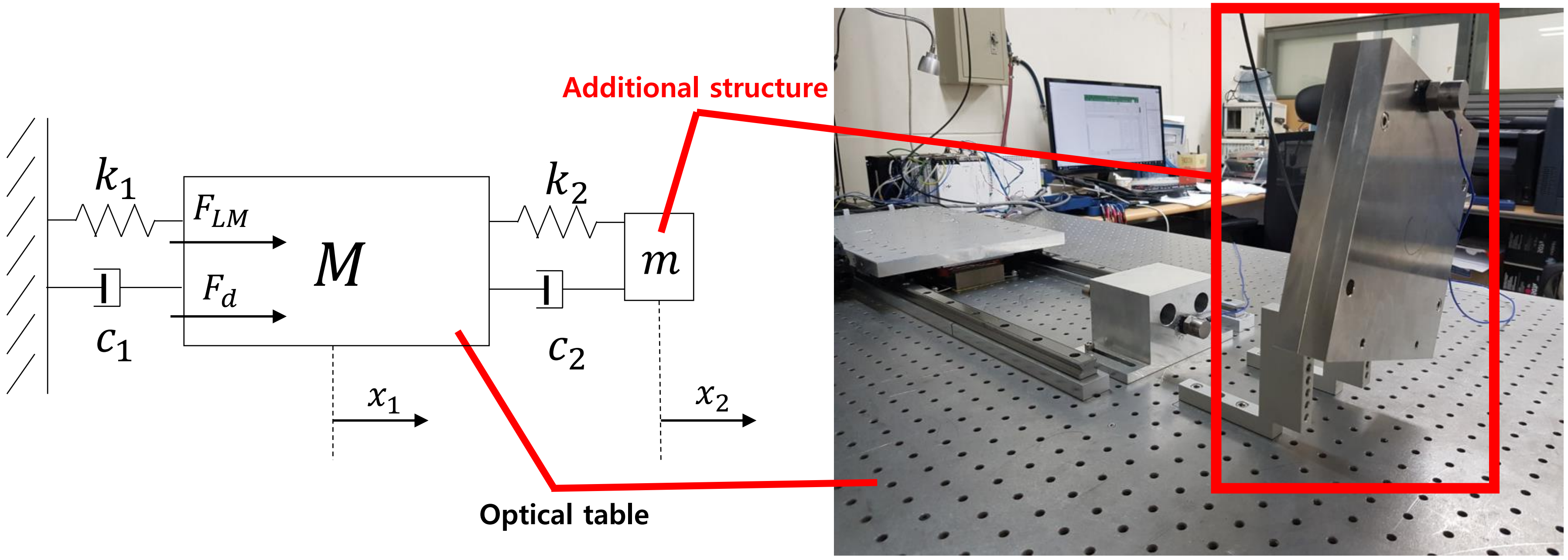
$$\frac{\mathrm{A(s)}}{\mathrm{I_p(s)}} = \frac{\mathrm{K_i s^2}}{\mathrm{Ms^2 + cs + k}}$$

Frequency domain analysis for Unknown : c, k

Modeled Open–loop Transfer Function

$$\frac{\mathrm{A(s)}}{\mathrm{I_p(s)}} = \frac{23\mathrm{s}^2}{280\mathrm{s}^2 + 2300\mathrm{s} + 1061622}$$

2. Analytical Modeling – Multi-Frequency System Modeling



(1) $M\ddot{x}_1 + c_1\dot{x}_1 + c_2(\dot{x}_1 - \dot{x}_2) + k_1 x_1 + k_2(x_1 - x_2) = F_{LM} + F_d$

(2) $m\ddot{x}_2 + c_2(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0$ $(F_{LM} = K_i i)$

(3) $\boldsymbol{i = -[g_1\,\dot{x}_1 + g_2(\dot{x}_1 - \dot{x}_2)]}$

$\Rightarrow$ $M\ddot{x}_1 + (c_1 + K_i g_1)\dot{x}_1 + (c_2 + K_i g_2)(\dot{x}_1 - \dot{x}_2) + k_1 x_1 + k_2(x_1 - x_2) = F_d$

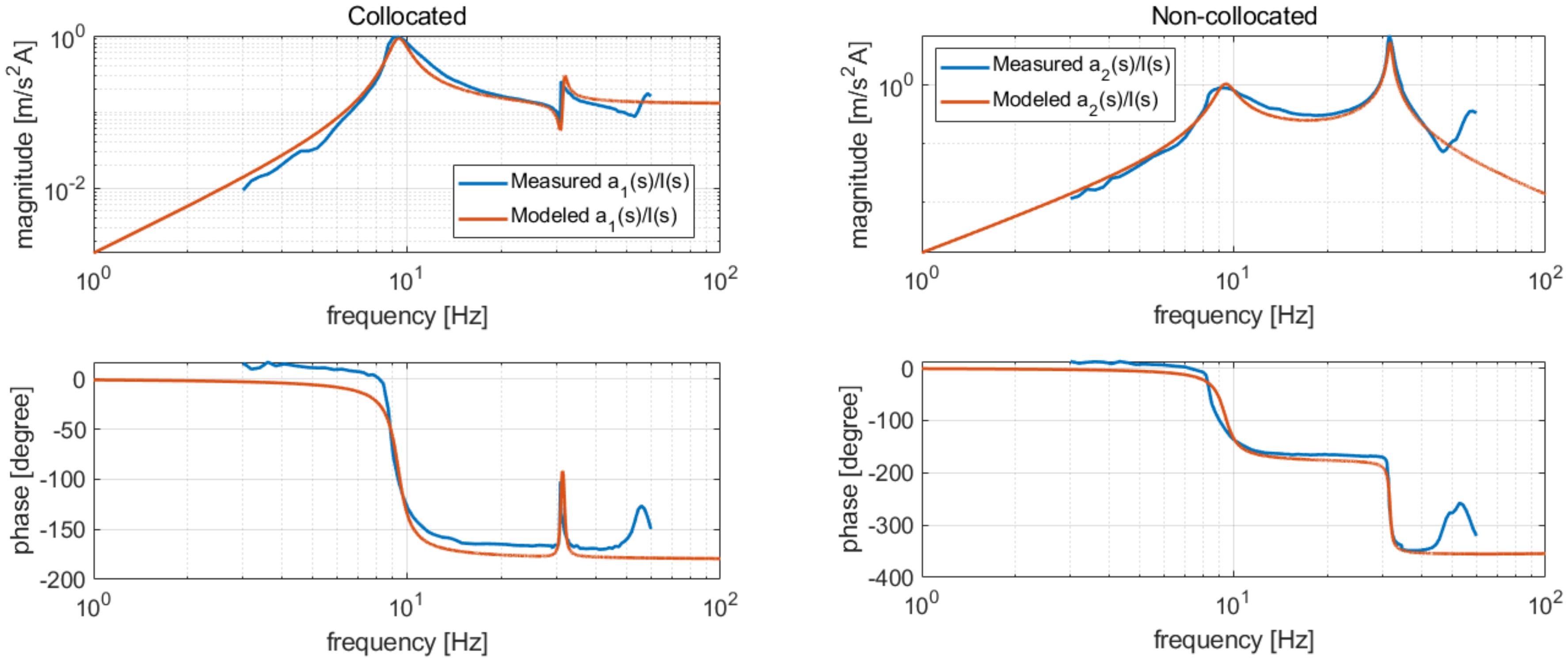
Additional damping coefficient to the collocated mass

2. Analytical Modeling – Multi-Frequency System Modeling(Identification)

- **Estimated parameters (DSA test)**

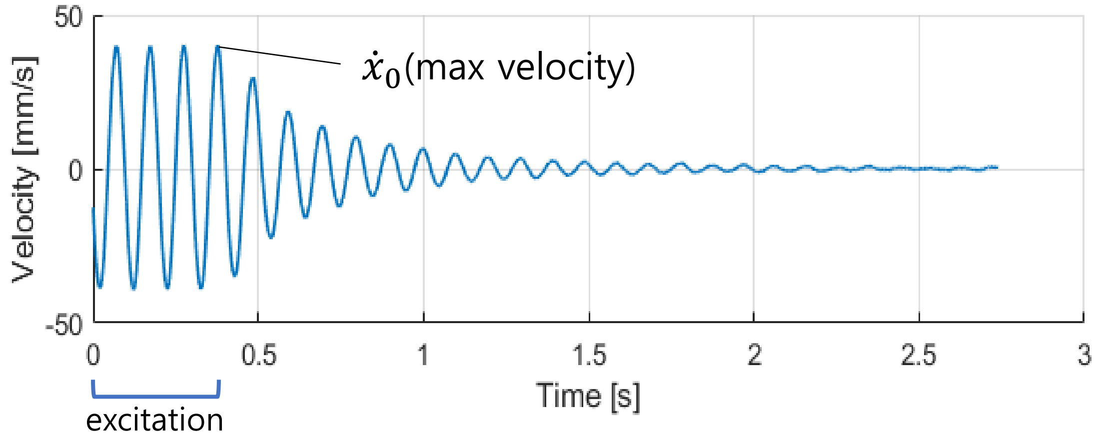
$M = 280kg,\ c_1 = 2300\ Ns/m\ ,\ k_1 = 1{,}018{,}700\ N/m$

$m = 30kg,\ \ c_2 = 55\ Ns/m\ ,\quad k_2 = 455{,}270\ N/m \qquad K_i = 36\ N/A$



2. Analytical Modeling – Required Force Constant(LMAD)

Role of Linear Motor : Exciter (sine wave current)



$$X(s) = \frac{M\dot{x}_0}{Ms^2 + (c + K_v K_i)s + k}$$

w/o Active Damping

$$X_{plant}(s) = \frac{M\dot{x}_0}{Ms^2 + cs + k}$$

Amplitude attenuation ratio($\boldsymbol{\eta}$) at target frequency ($\boldsymbol{w_n}$)

$$\eta = \left|\frac{X(s)}{X_{plant}(s)}\right|_{s=jw_n} = \frac{c}{c + K_v K_i}$$

w_n : Natural frequency of optical table

c : Damping coefficient of optical table

K_i : Linear motor force constant

K_v : Velocity feedback controller gain

2. Analytical Modeling – Required Force Constant(LMAD)

- **Active Damper Design Specification**

  - Required force constant(K_i) of the actuator

$$\eta = \frac{c}{c + K_i K_v} \xrightarrow{K_v = I_p/\dot{x}} \boldsymbol{K_i = \frac{(1-\eta)V_{max}c}{\eta I_{max}}}$$

(1) η : Required amplitude attenuation ratio → 80% reduction target

(2) V_{max} : Maximum treatable velocity amplitude (0.040m/s by test)

(3) c : Obtained by experiment (Linear motor excitation, DSA test, …)

(4) I_{max} : Maximum capacity of current driver ~ 21A

2. Analytical Modeling – Required Force Constant(LMAD)

- **Required force constant(K_i) of the actuator**

$$K_{i,required} = \frac{(1-\eta)V_{max}c}{\eta I_{max}} = 14.4734\ \text{N/A}$$

η : Required amplitude attenuation ratio → 0.20 (**80% reduction**)

V_{max} : Maximum treatable velocity amplitude → 0.040 m/s

c : Damping coefficient of the optical table system → 2300 Ns/m

I_{max} : Maximum capacity of current driver → 21A

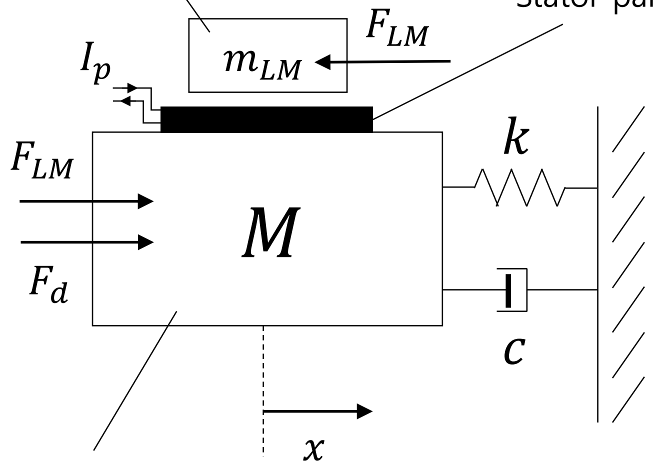
For our system

➡ $K_{i,LMAD} = 36N/A > K_{i,required} = 14.47N/A$

$$\eta = \frac{c}{c+K_v K_i} \rightarrow K_v = 255.6\ As/m$$

2. Analytical Modeling – Required Actuator Mass

- **Required actuator mass**



$M\ddot{x} + c\dot{x} + kx = F_{LM} + F_d$: Table dynamics

$F_{LM} = K_i i$: Linear Motor plant

$$F_{LM} = m_{LM}a = m_{LM}w_L^2 X sin(w_L t) = K_i I_{max} \sin(w_L t)$$

$X < L_{max}/2$

$$\frac{2K_i I_{max}}{L_{max} w_L^2} < m < m_{lim}$$

X : Actuator amplitude

m_{lim} : Maximum actuator mass (system limit)

w_L : Minimum frequency of actuator excitation

L_{max} : Maximum Stroke of actuator

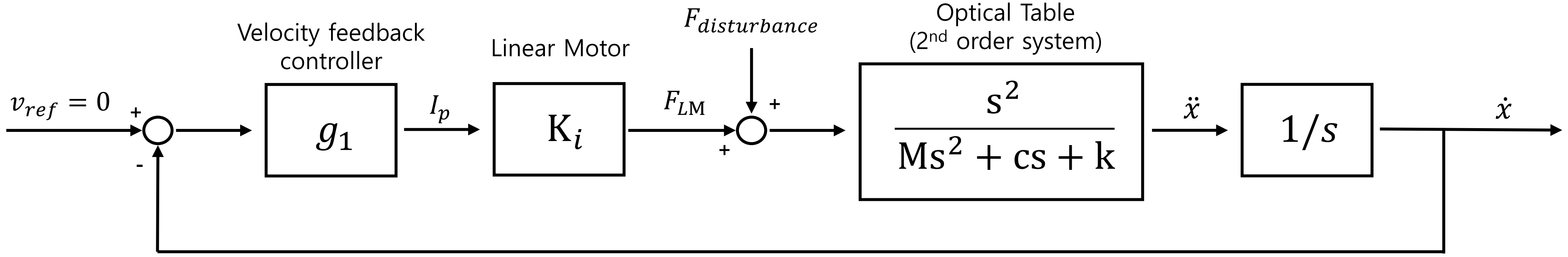
For our system

$$\Rightarrow \quad \frac{2K_i I_{max}}{L_{max} w_L^2} = 3.36kg < m = 10.8kg$$

3. Single Frequency Vib. Reduction – Direct Velocity Feedback

Standard Second-order system of Optical table	Mechanical governing equation of Linear Motor
$M\ddot{x} + c\dot{x} + kx = F_{LM} + F_{\mathrm{disturbance}}$	$m_{LM}\ddot{x}_{LM} = F_{LM} = K_i I_p$

Direct Velocity feedback



$M\ddot{x} + c\dot{x} + kx = F_{LM} + F_{\mathrm{disturbance}}$

$F_{LM} = K_i I_p$

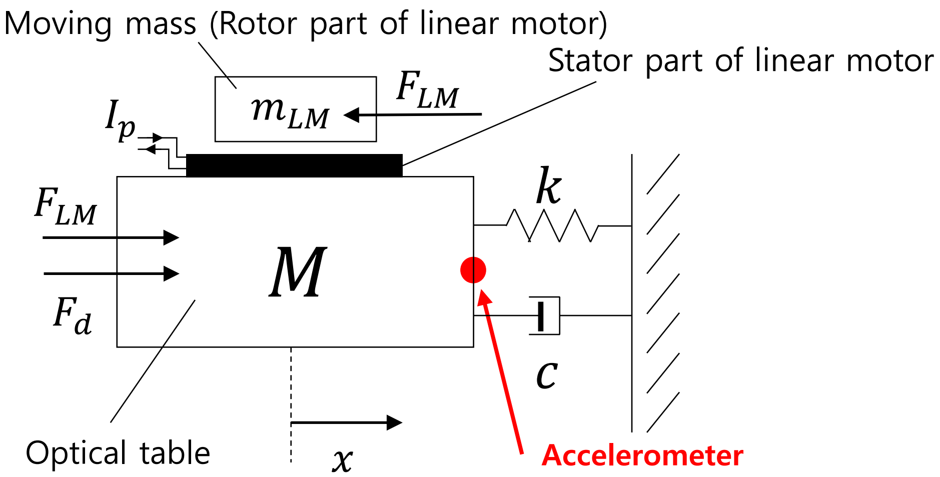
$I_p = -g_1\dot{x}$ (Velocity feedback control)

$$\boldsymbol{\eta} = \frac{c}{c + g_1 K_i} \rightarrow g_1 = \mathbf{255.6\ As/m}$$

$$M\ddot{x} + (c + g_1 K_i)\dot{x} + kx = F_{\mathrm{disturbance}}$$

Adjustable Damping Term

3. Single Frequency Vib. Reduction – Direct Velocity Feedback



- **Test Results**

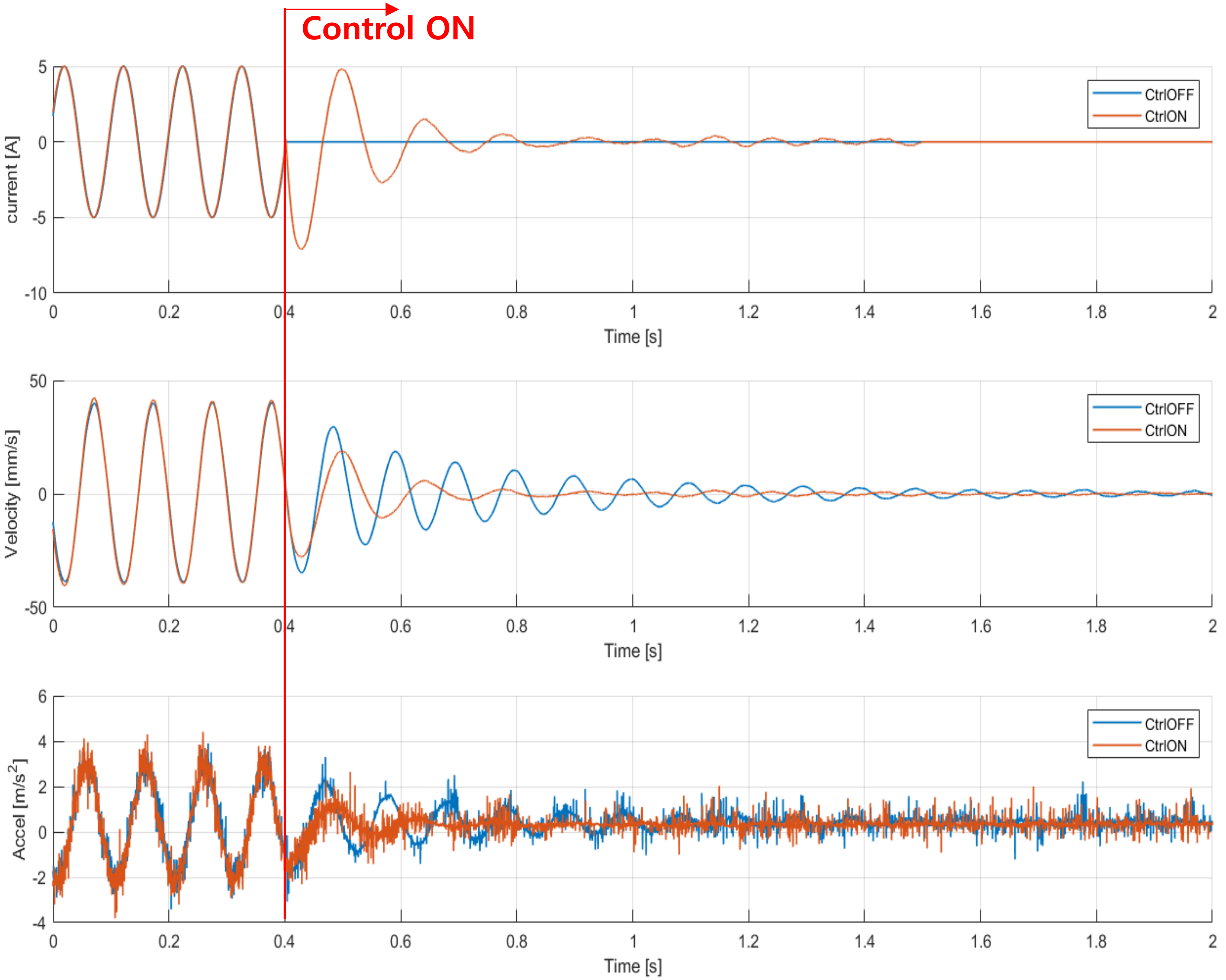
- Role of Linear Motor (Absence of exciter)
Control OFF : Exciter (sine wave current)
Control ON : Active Damper

- Excitation
9.8 Hz : 5.0A excitation (by LM)

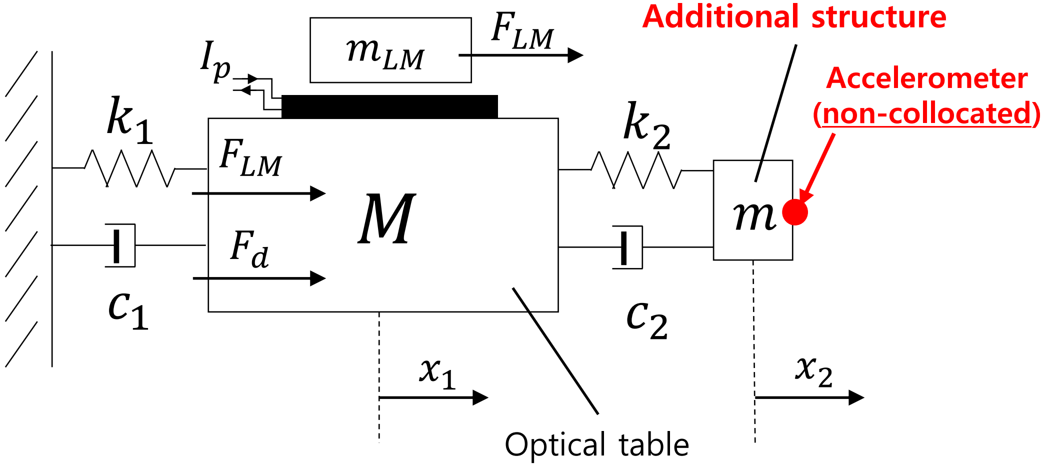
- Control gain
$g_1 = 255.6$

$\boldsymbol{2\%\ settling\ time\ 43.3\%\ reduced}$

$\boldsymbol{t_{s,natural} = 1.68 \ll t_{s,control} = 0.95}$



4. Multi-Frequency Vib. Reduction – Direct Velocity Feedback



- **Test Results**

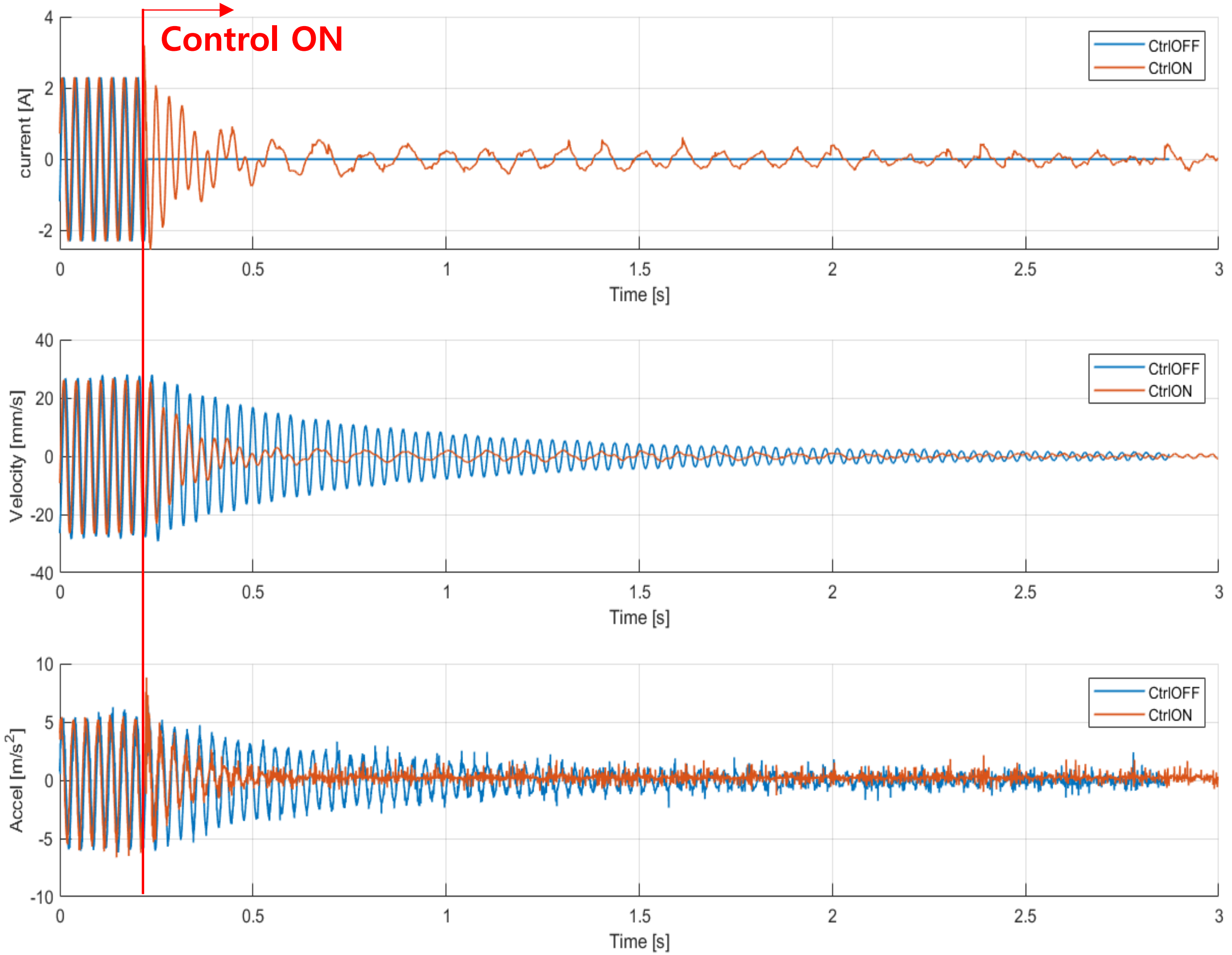
- **Role of Linear Motor** (Absence of exciter)
  Control OFF : Exciter (sine wave current)
  Control ON : Active Damper

- **Excitation**
  9.8 Hz : no excitation
  31.3 Hz : 2.3A excitation (by LM)

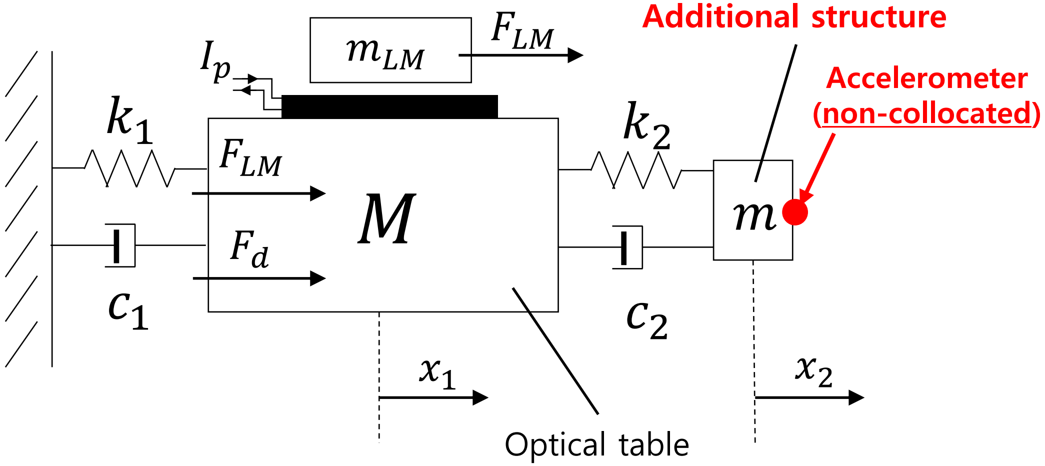
- **Control gain**
  $g_1 = 255.6, \ g_2 = 100$

➡ ***5% settling time 39.5% reduced***

$$\boldsymbol{t_{s,natural} = 2.26 \ll t_{s,control} = 1.36}$$



4. Multi-Frequency Vib. Reduction – Direct Velocity Feedback



- **Test Results**

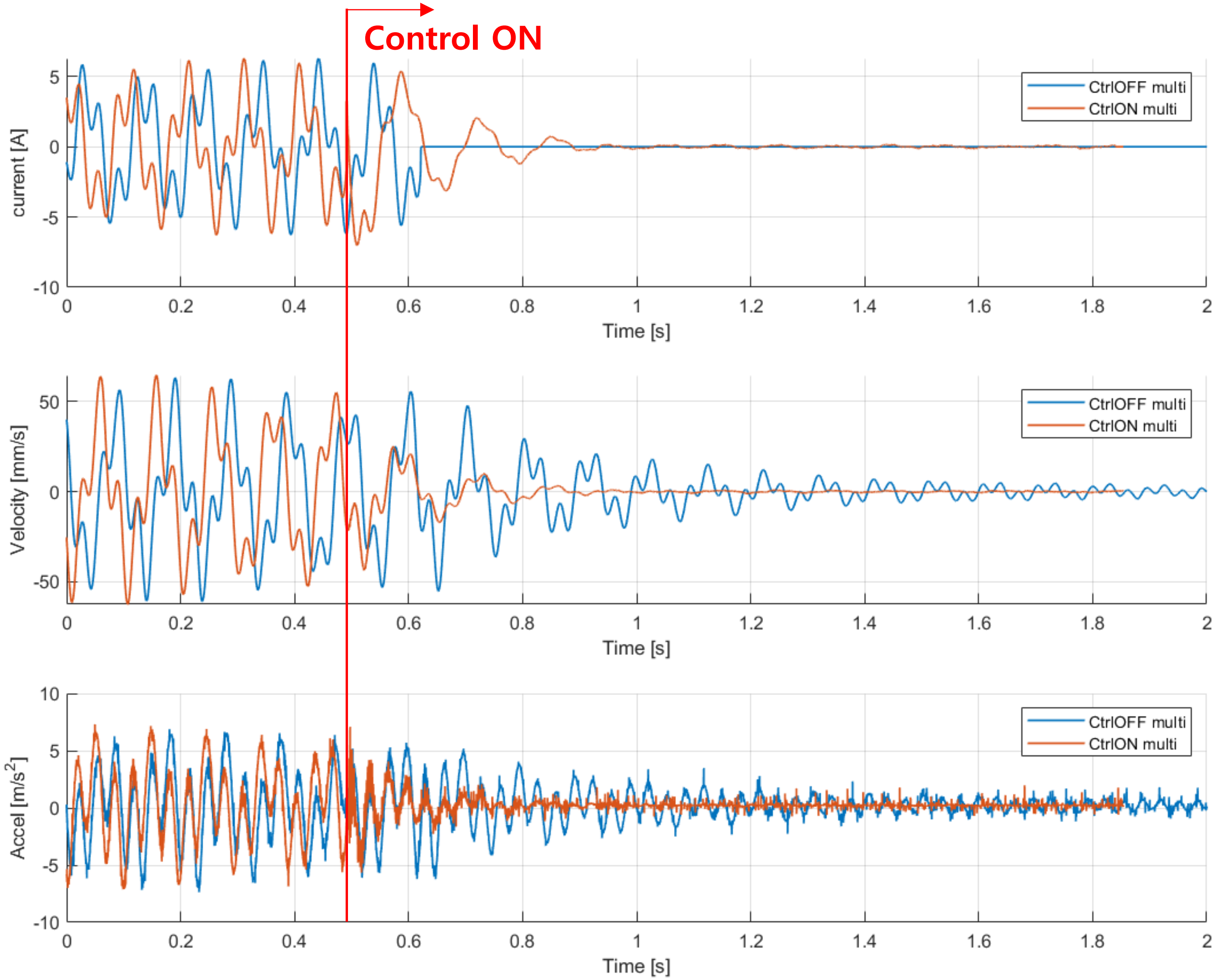
- **Role of Linear Motor** (Absence of exciter)
  Control OFF : Exciter (sine wave current)
  Control ON : Active Damper

- **Excitation**
  9.8 Hz : 4.0A excitation
  31.3 Hz : 2.3A excitation (by LM)

- **Control gain**
  $g_1 = 255.6,\ g_2 = 100$

➡ $\mathbf{5\%\ settling\ time\ 71.5\%\ reduced}$

$t_{s,natural} = 1.71 \ll t_{s,control} = 0.49$



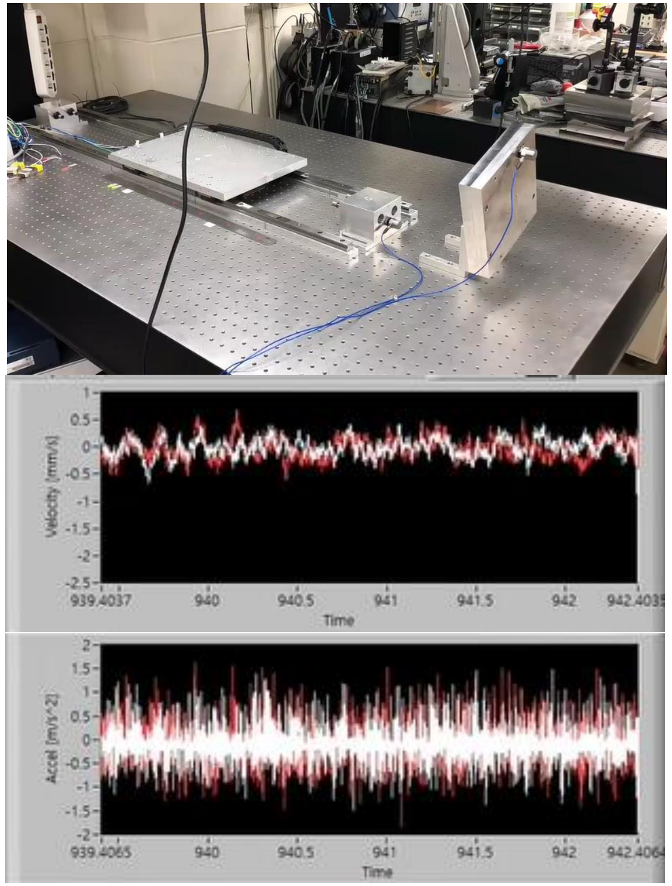
5. Conclusion

Speed X0.25

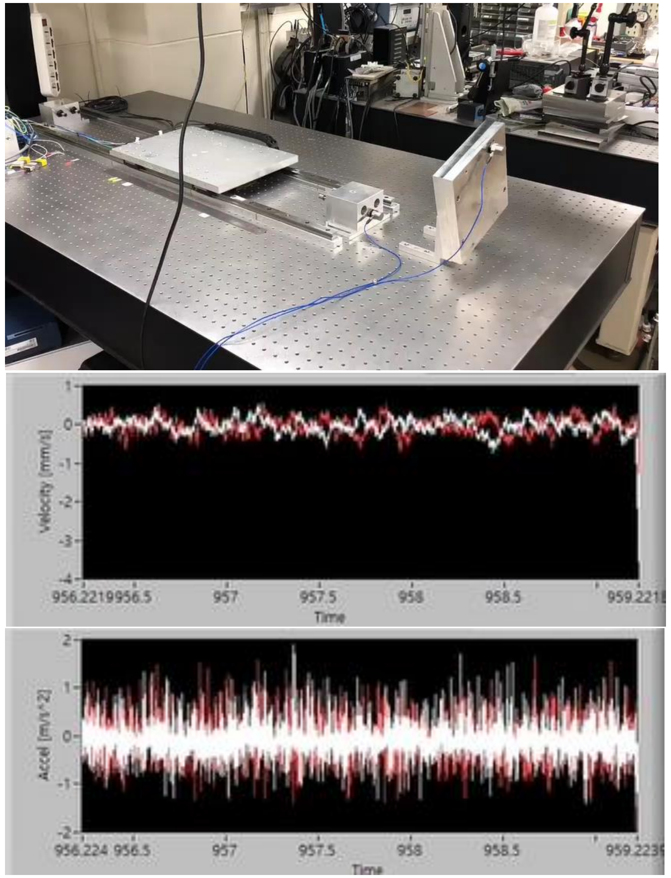
Excitation
9.8Hz, 31.3Hz

Red : non-collocated
White : collocated

Natural dissipate



Control



Found feasibility for dealing with the Entire Frequency

6. Future work

- **During control, LMAD deviates from the center**
  - Apply parallel controller (Separate effect on performance)
  - LMAD position controller with low frequency bandwidth

- **Only 1DOF & 2DOF applications (modeling, controller)**
  - Generalize n-DOF applications

- **Used simple controller (DVF, P control)**
  - Apply specific stability conditions
  - Specification of control methods

7. Acknowledgements

Special thanks to

Professor Min

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T.A. Kim