Vibrations Demo Rig

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ME-360 M3

Professor Wright and Professor Wootton

MOTIVATION

- CLIENT: Professor Baglione
- NEED: A rig demonstrating mechanical vibrations phenomena to support classroom instruction in ME-301
- IMPROVING ON PAST PROJECTS:
  - Demonstrating multiple concepts
  - Incorporating constant force as opposed to direct human intervention
- OBJECTIVE: To demonstrate the following main concepts
  - Vibration suppression with an undamped, tuned vibration absorber (DVA)
  - Excitation via a rotating eccentric mass (i.e. unbalanced force)
  - Multiple-degrees-of-freedom

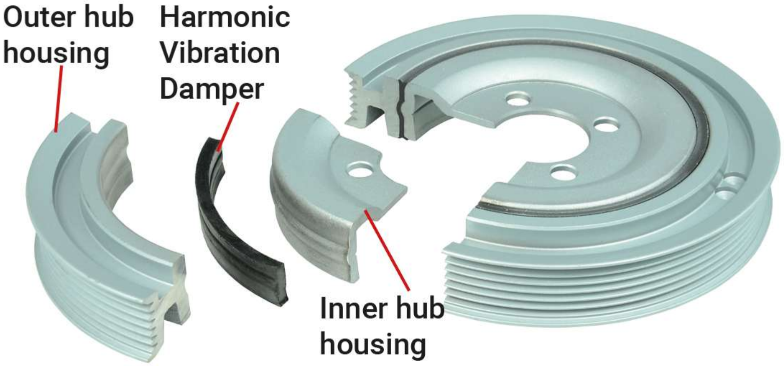
APPLICATIONS

Civil Engineering

Taipei 101 uses a 660-ton tuned mass damped to reduce sway due to wind/earthquakes.

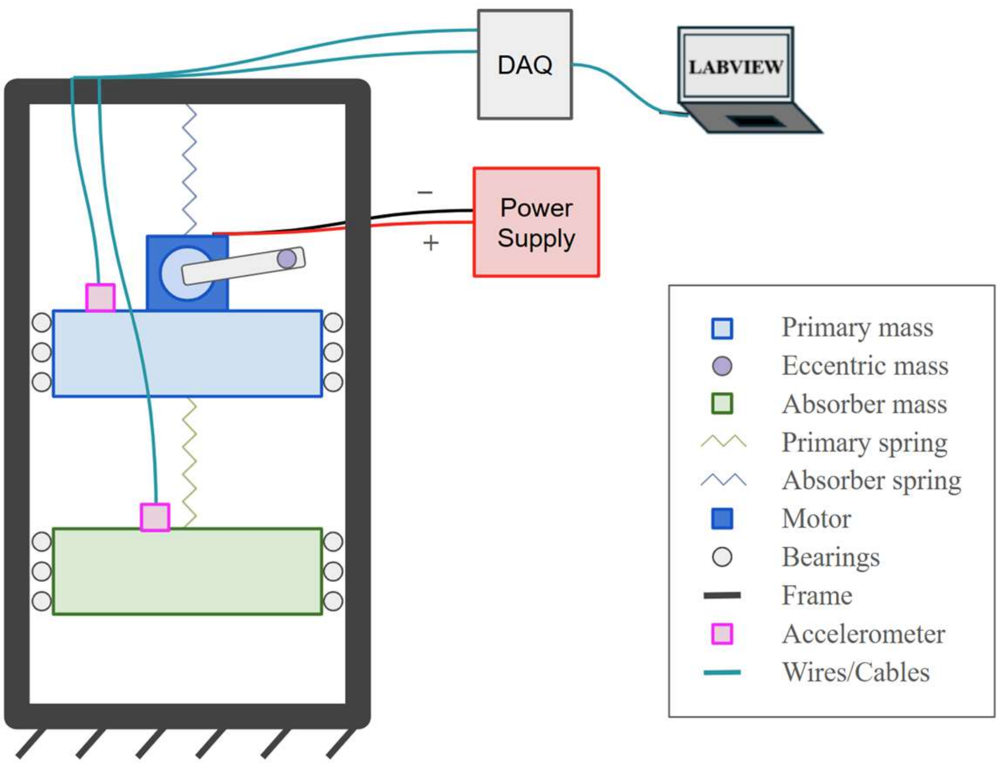
Automotive

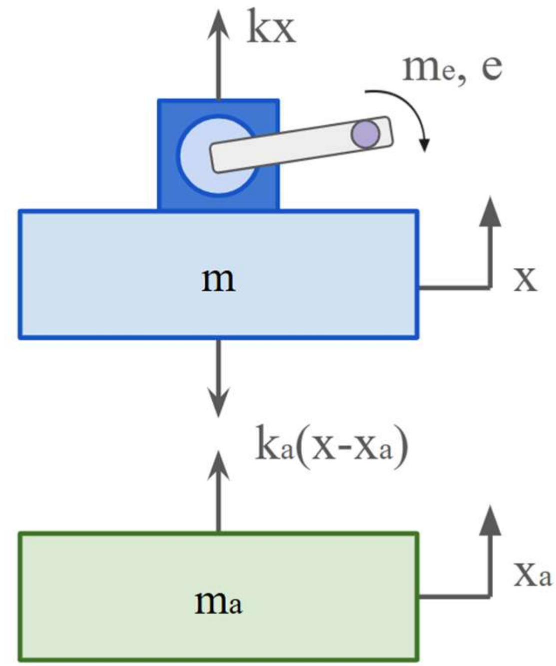
Torsional vibration absorbers in engine crankshafts/drive shafts minimize vibrations transmitted to the frame and reduce noise, vibration, and harshness (NVH).



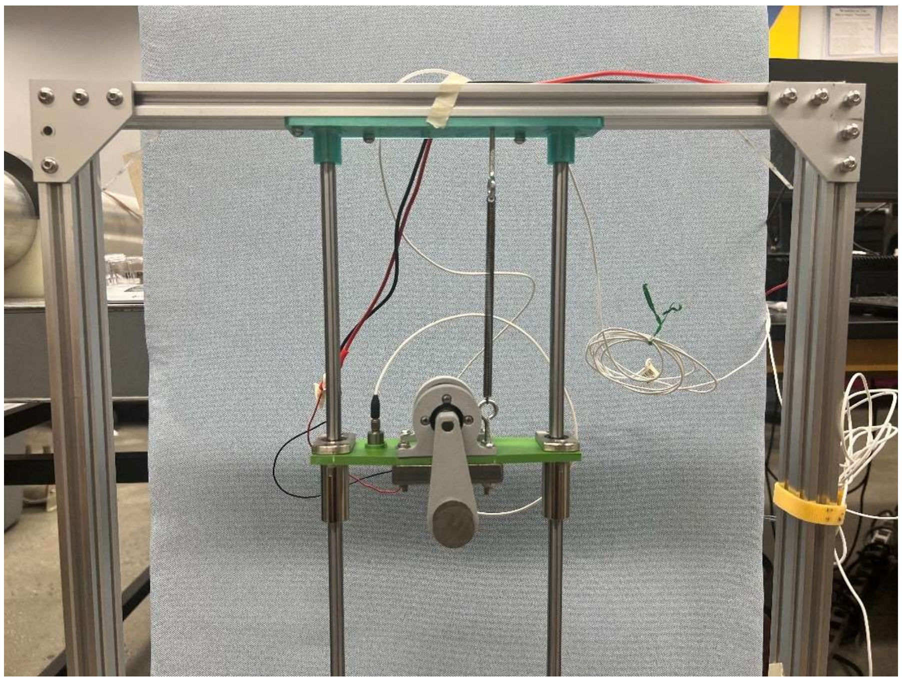
- Reduce resonance effects
- Improve comfort, safety, or precision
- Minimize fatigue and wear

EXPERIMENTAL: SCHEMATIC AND FBD

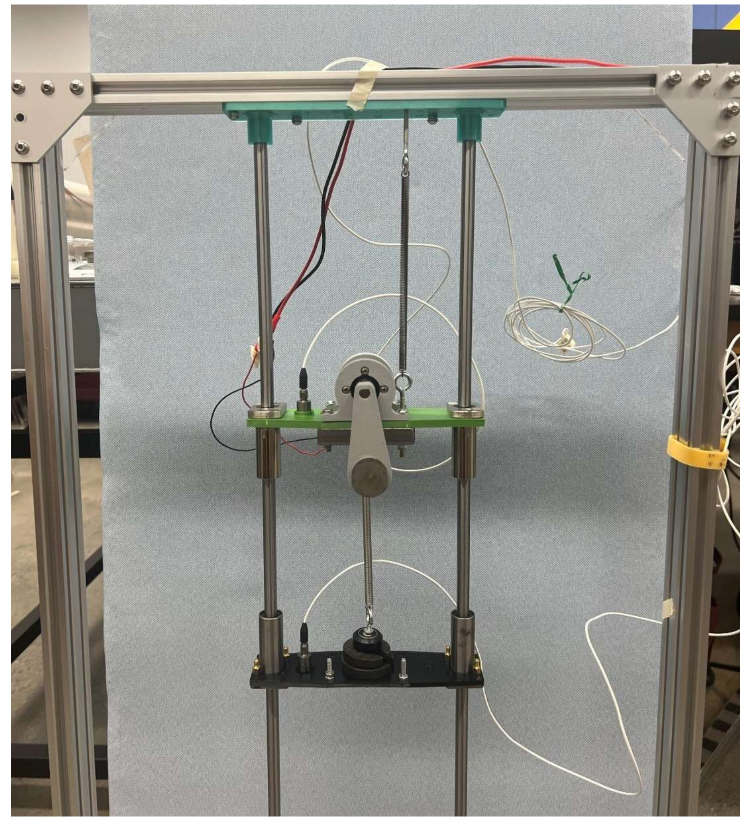




EXPERIMENTAL: APPARATUS



Without DVA



With DVA

EXPERIMENTAL: INSTRUMENTATION

Instrumentation	**Model Number**	**Uncertainty**	**Units**
Tape Measure	Stanley Powerlock 12’	± 0.03125	In
Digital Calipers	Mitutoyo 500-196-30 Digimatic O-6”/150mm	± 0.00025	In
Scale	Royal ds5	± 0.5	g
Accelerometer	PCB Piezotronics 333B31	± 0.05	Hz
Data Acquisition (DAQ) Device	National Instruments NI USB-4431	N/A	N/A

EXPERIMENTAL: BOM

Category	Component	Cost/Unit	Quantity	Cost
Frame	T-slot aluminum (4658N21)	$12.68	10	$126.80
	Linear Rail	$59.29	1	$59.29
Motor	12V Brushed Motor (6331K11)	$122.40	1	$122.40
Modular Masses	Machined with metal stock	-	4	$0.00
Misc	Lubricant	$14.99	1	$14.99
	Hardware (i.e. eye hooks, nuts, bolts)	$0.11 - $1.00	36	$20.00
Springs	4” Spring, 1.3 lbf/in (5108N266)	$14.24	1	$14.24
	2.25” Spring, 1.25 lbf/in (9654K193)	$3.76	1	$3.76
	3.5” Spring, 0.429 lbf/in	$12.07	1	$12.07
Data Aquisition	Accelerometer	-	2	$0.00
	DAQ	-	1	$0.00
			Total Cost:	**$373.55**

KEY CHOICES: Springs

Type	Option	Rating (N/m)	Length (in)	Range (lbs)	OD (in)
Primary	1	219	6	1.6 - 17.5	1.125
	2	228	4	0.5 – 5.3	0.24
Absorber	1	63	6	0.6 – 2.4	0.25
	2	112	2.25	0.64 – 3.8	0.375
	3	75	3.5	Max: 1.9	0.25
	4 (lab)	45	12	N/A	0.25

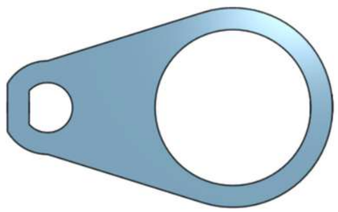
*Note that this table omits all the original spring combinations attempted prior to ordering!

- Ultimately one spring is more effective than multiple in series (less room for error)

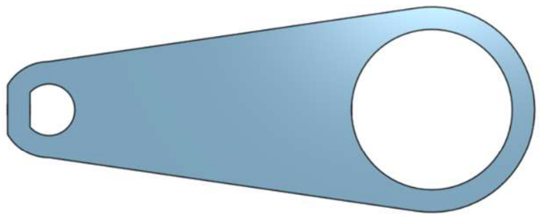
*No spring was perfect→ Primary Spring 2 and Absorber Spring 3 were chosen

KEY CHOICES: Eccentricity

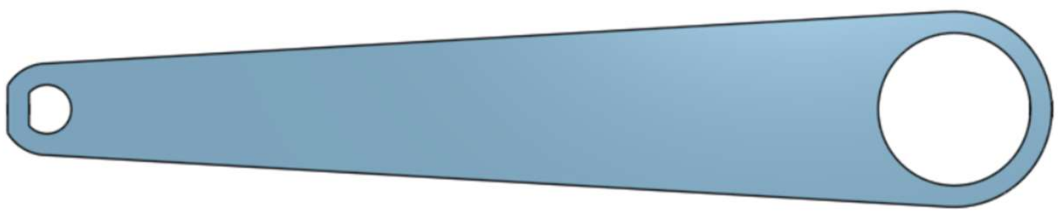
$e = 3\ cm$



$e = 6\ cm$



$e = 15\ cm$



The eccentricity affected two main components:

1) Motor Stall
2) Amplitude of Vibrations

Too much eccentricity induced motor stall, but too little did not induce sufficiently visible amplitude of vibrations.

To compromise, $e = 6\ cm$ was chosen as the eccentricity with an eccentric mass of 100g as opposed to the original 50g.

As $F_o = m_o e w^2$, increasing m_o accounts for the loss in amplitude from decreasing the eccentricity while avoiding motor stall at low ω

KEY CHOICES: Adding Mass & Frame Material

Adding Mass

- The springs readily available do not hit all specs necessary for our rig
- To accommodate this, mass is added to both platforms to bring each spring further into its respective loading range
- One must maintain an acceptable mass ratio while doing so
- In the end, about 400g and 120g were added to the primary and absorber platforms respectively
- The 400g mass was machined such that it could be fastened to the platform

Frame Material

- T-slot aluminum rail was used as the frame's material
- What drove this choice?
  - Availability
  - Easy Connections
- Not only did we have a lot of this material prior to starting this project, but more importantly:
- The rails allow for easy joining through T-nuts, 90° joining plates, and custom-made joining plates
- These custom joining pieces were laser cut, allowing us to avoid using L-brackets

EXPERIMENTAL: PROCEDURE

- Connect the springs, accelerometers, power supply, and DAQ device according to the schematic.
- With ONLY the primary mass attached, manually displace it and determine the damping ratio and natural frequency from the time response and FFT respectively
- With ONLY the primary mass and primary spring attached, set the power supply to the desired voltage value.
- Collect and record data for 17 well spaced amplitude measurements for the corresponding voltage inputs ranging from 9V to 18V using the LabView VI file titled DVA.vi.
- Turn off the power supply and attach the absorber mass and absorber spring to the system.
- Securing the primary mass to the frame's top cross brace, manually displace the absorber and determine the damping ratio and natural frequency from the time response and FFT respectively
- Set the power supply to the desired voltage value. Collect and record 17 well spaced amplitude measurements over the same voltage range for BOTH the primary mass and absorber mass using the LabView VI file.
- To convert the measured amplitude from dB to m, use the following equation:

$$A_m = \frac{g}{\omega_d} \times 10^{\left(\frac{A_{dB}}{20}\right)}$$

RAW DATA: ABSORBER MASS FREE RESPONSE

Log Decrement

$$\boxed{\delta = \frac{1}{n}\ln\frac{B_1}{B_{n+1}} = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}}$$

$\rightarrow$

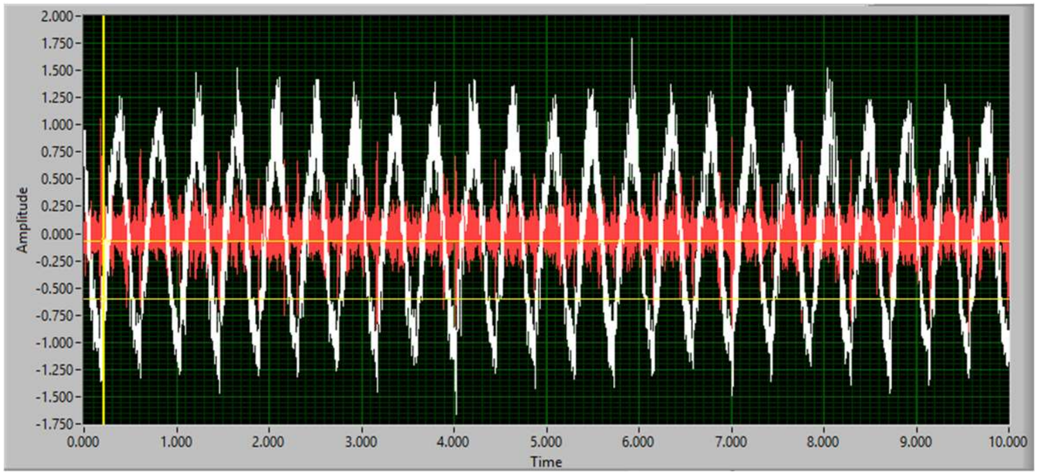
Peak	Amplitude (cm)
1	5.21
6	1.43

$$\boxed{\zeta = 0.035}$$

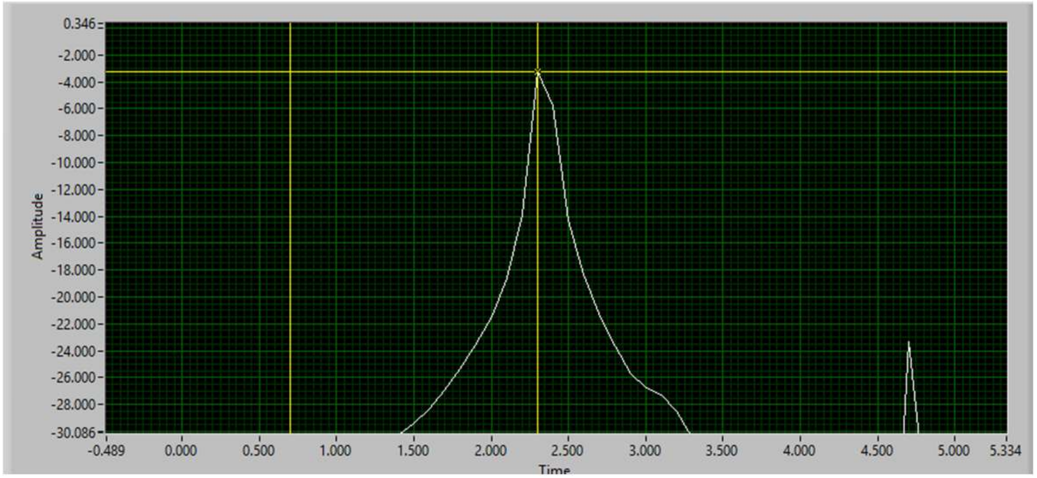
RAW DATA: PRIMARY MASS WITHOUT ABSORBER

Primary Mass (white)

Absorber Mass (red)



Amplitude vs. Time



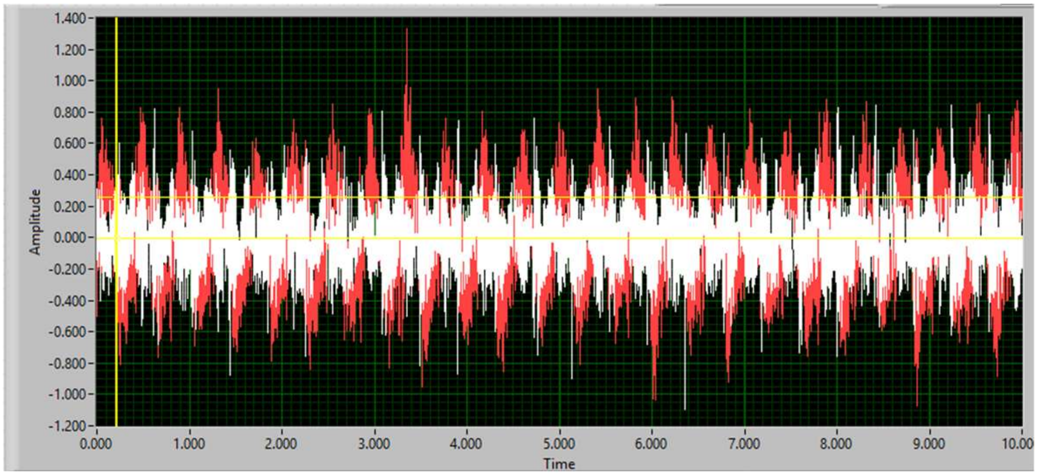
Amplitude vs. Driving Frequency (FFT plot)

Example @ 11.5 V

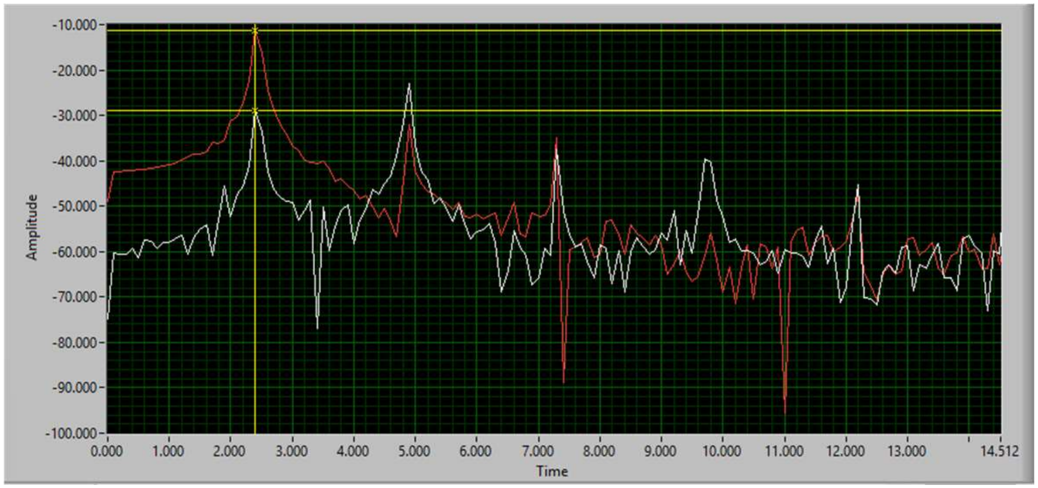
RAW DATA: PRIMARY MASS WITH ABSORBER MASS

Primary Mass (white)

Absorber Mass (red)



Amplitude vs. Time



Amplitude vs. Driving Frequency (FFT plot)

Example @ 11.5 V

EXPERIMENTAL RESULTS

Trial	Voltage (V)	Driving Frequency (rad/s)	Amplitude of Primary Mass without DVA (mm)	Amplitude of Primary Mass with DVA (mm)	Amplitude of Absorber Mass (mm)
1	10	13.22	13.16	3.55	13.90
2	10.5	13.97	18.32	2.75	13.40
3	11	14.70	24.22	1.51	11.40
4	11.5	15.50	28.08	1.53	11.19
5	12	16.23	26.62	2.43	13.64
6	12.5	16.92	31.32	3.09	13.00
7	13	17.51	38.64	4.01	16.82
8	13.5	18.32	37.96	4.76	18.59
9	14	19.10	18.33	5.32	19.72
10	14.5	19.72	16.43	5.08	19.56
11	15	20.47	11.09	4.04	15.30
12	15.5	21.15	9.53	3.97	15.36
13	16	21.93	10.48	4.19	15.79
14	16.5	22.57	11.69	4.29	16.66
15	17	23.33	10.57	5.03	19.10
16	17.5	24.06	8.26	5.00	16.15
17	18	24.79	7.77	6.35	17.65

Resonance occurs close to 11 V or a driving frequency of 14.70 rad/s

THEORETICAL MODEL

- Our theoretical model is based on an undamped tuned vibration absorber. The amplitudes in each scenario are as follows:

$$X_{without\ DVA} = \frac{m_o e}{m} \frac{r^2}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}$$

Resonance theoretically occurs at a driving frequency of 15.21455 rad/s or voltage input of 11.3 V.

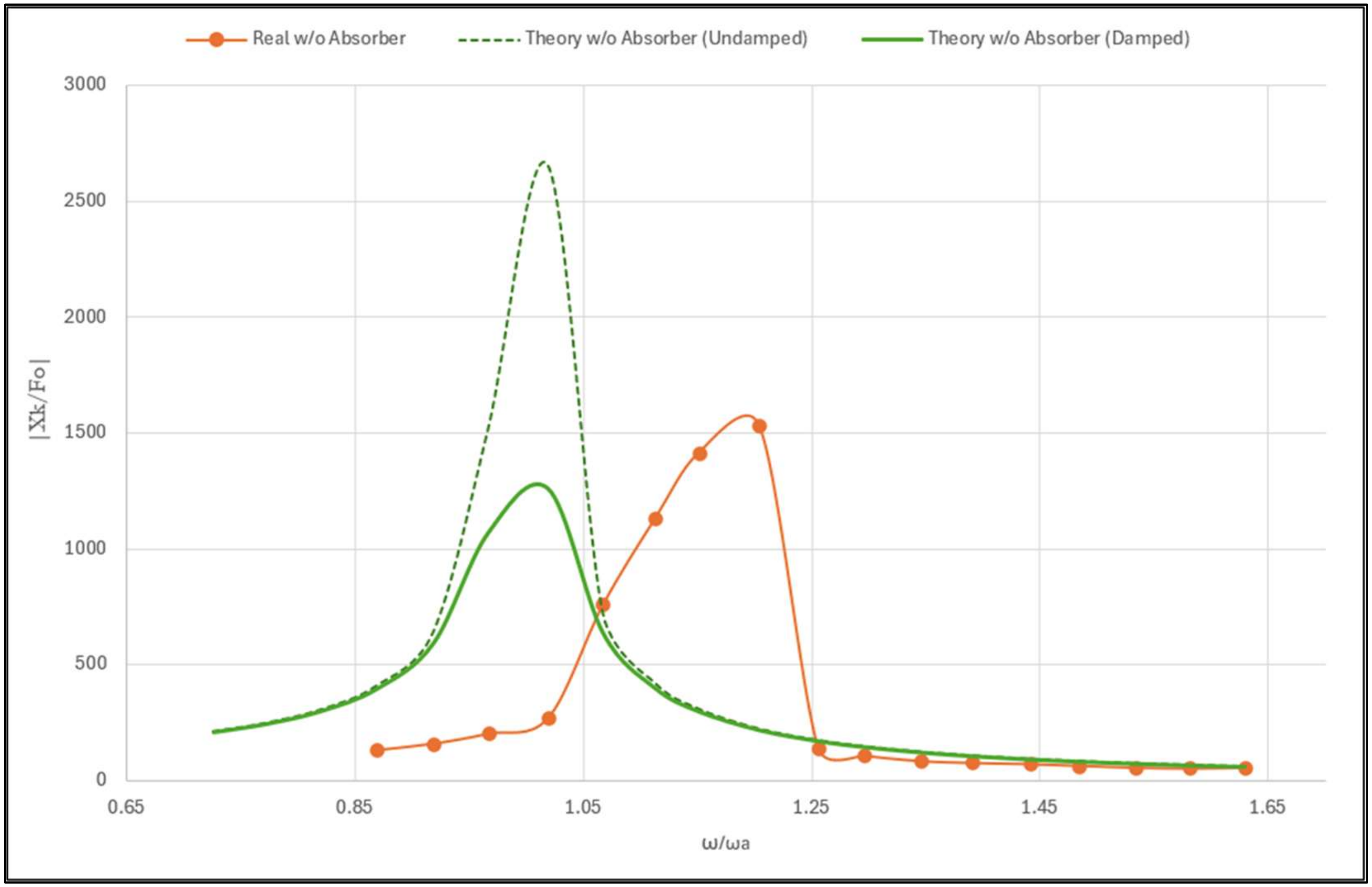
$$X_{with\ DVA} = \frac{F_0}{k} \frac{1-r_a^2}{\left(1+\mu\beta^2 - r_p^2\right)(1-r_a^2) - \mu\beta^2}$$

Theoretical resonance is higher than experimental.

$$X_a = \frac{k_a F_0}{(k+k_a-m\omega^2)(k_a-m_a\omega^2) - k_a^2}$$

*Note that as DVAs are designed such that $\omega_p = \omega_a$, $\beta = 1$ for all data acquired

PROCESSED DATA $(\mu = 0.31)$



Theoretically Damped, Undamped, and Real Normalized Magnitudes of Primary Mass for $\mu = 0.31$ Without Absorber vs. Normalized Frequency

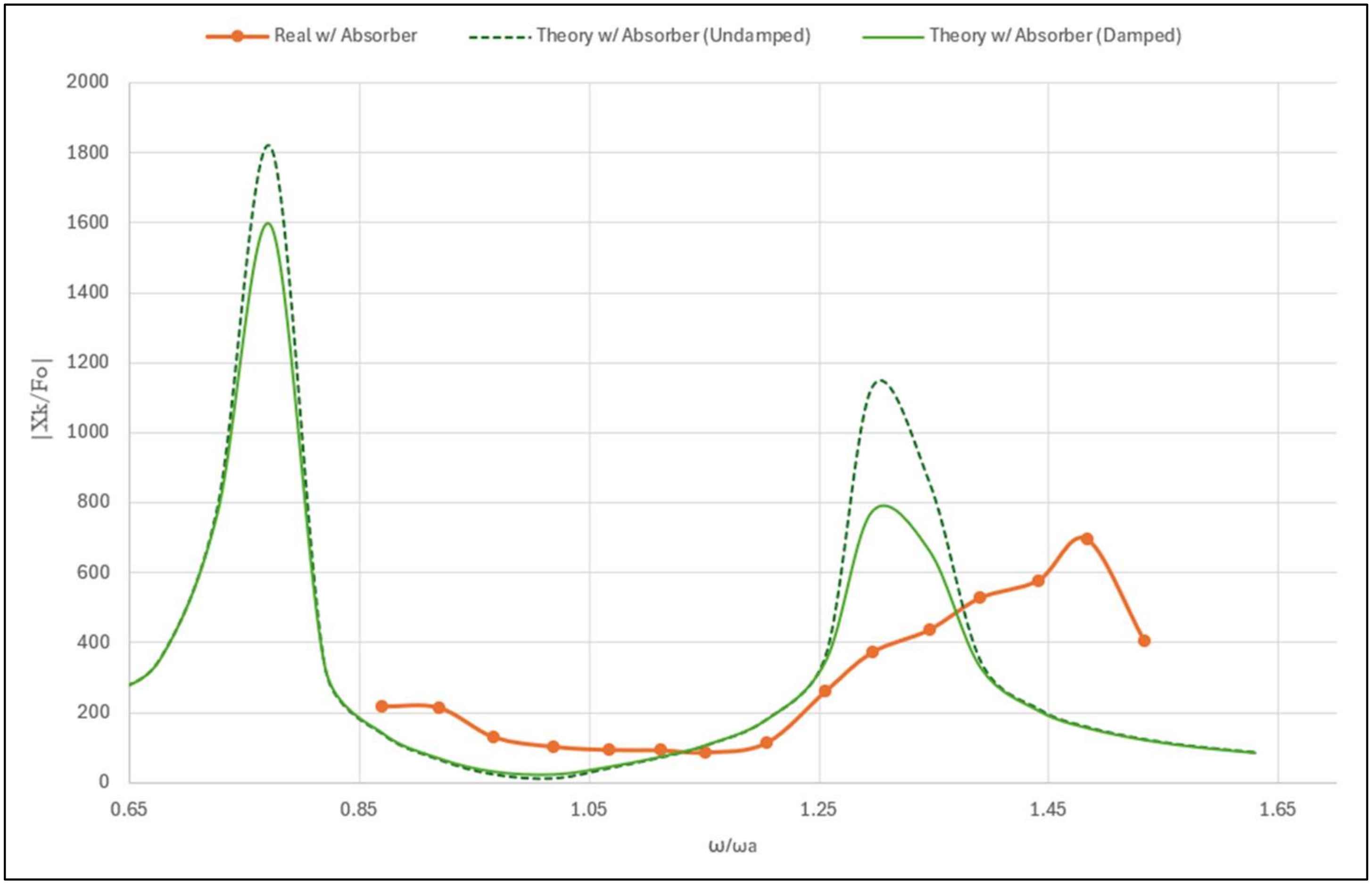
<u>Key Formulas</u>

$$X = \frac{F_0}{|k - m\omega^2|}$$

$$\overbrace{\frac{f_0}{\sqrt{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2}}}^{X}$$

$$X = \frac{(k_a - m_a\omega^2)F_0}{(k + k_a - m\omega^2)(k_a - m_a\omega^2) - k_a^2}$$

PROCESSED DATA ($\mu = 0.31$)



Theoretically Damped, Undamped, and Real Normalized Magnitudes of Primary Mass for $\mu = 0.31$ With Absorber vs. Normalized Frequency

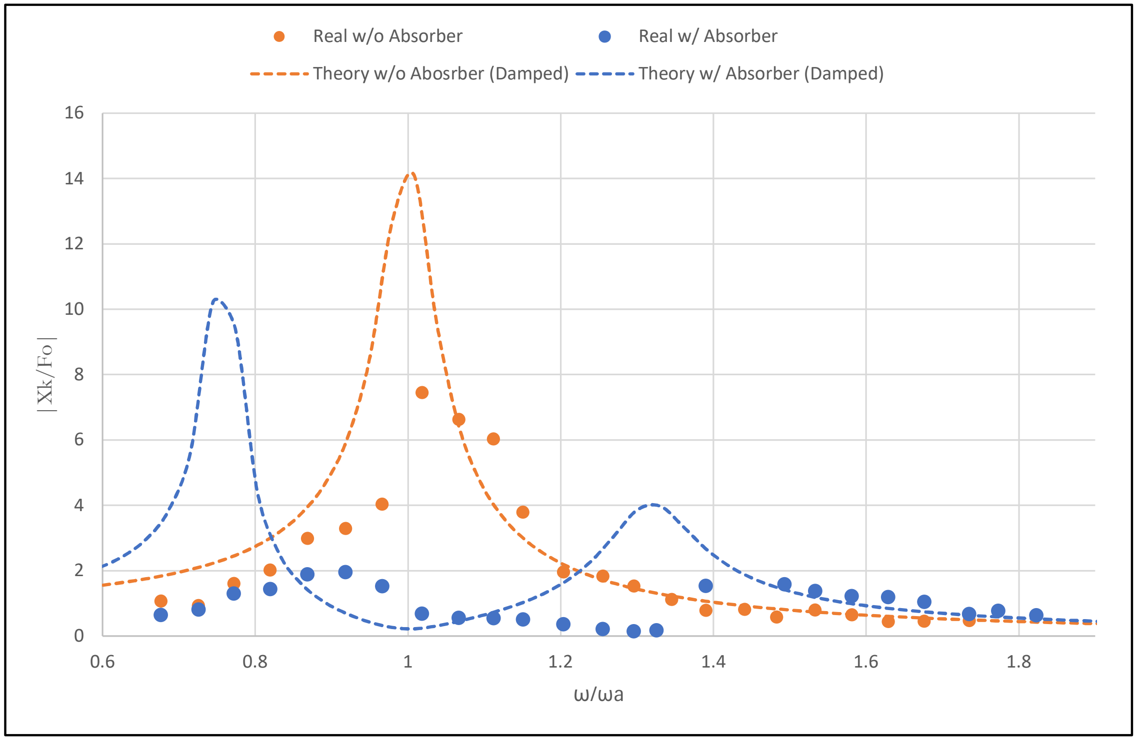
Key Formulas

$$X = \frac{F_0}{|k - m\omega^2|}$$

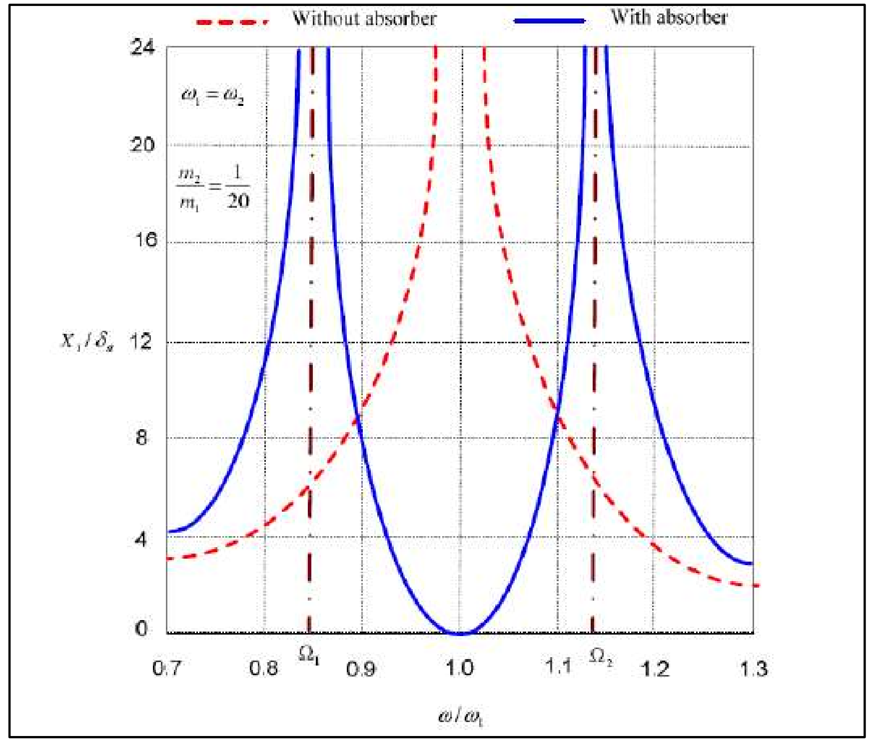
$$\overbrace{\qquad\qquad\qquad\qquad}^{X} \atop \frac{f_0}{\sqrt{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2}}$$

$$X = \frac{(k_a - m_a\omega^2)F_0}{(k + k_a - m\omega^2)(k_a - m_a\omega^2) - k_a^2}$$

PROCESSED DATA $(\mu = 0.31)$

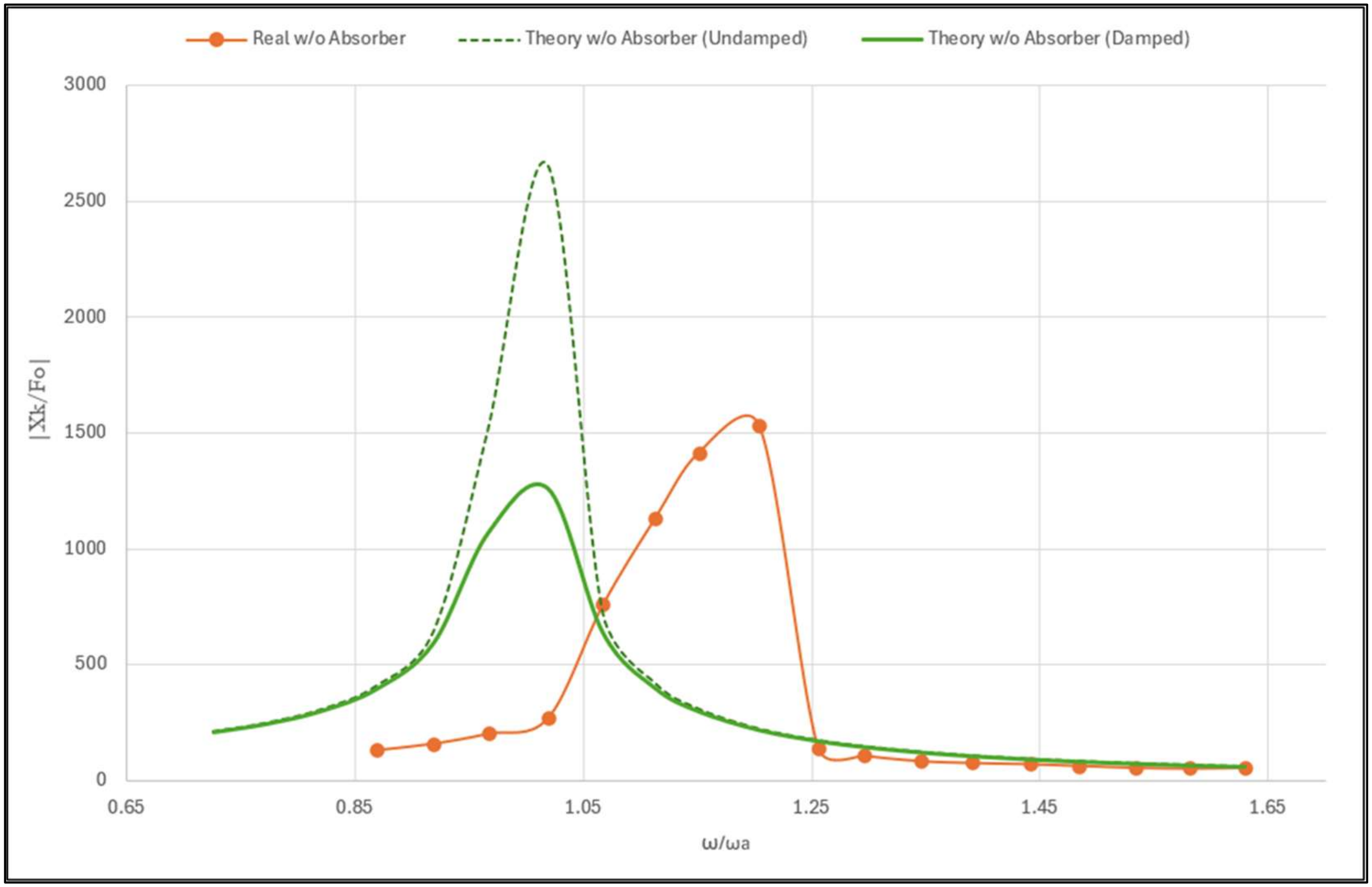


Theoretically Damped and Real Normalized Magnitudes of Primary Mass for $\mu = 0.31$ *With and Without Absorber vs. Normalized Frequency*



Effect of DVA on Response of a Machine ([*Mirsanei, 2012*]*)*

PROCESSED DATA $(\mu = 0.25)$



Theoretically Damped, Undamped, and Real Normalized Magnitudes of Primary Mass for $\mu = 0.25$ Without Absorber vs. Normalized Frequency

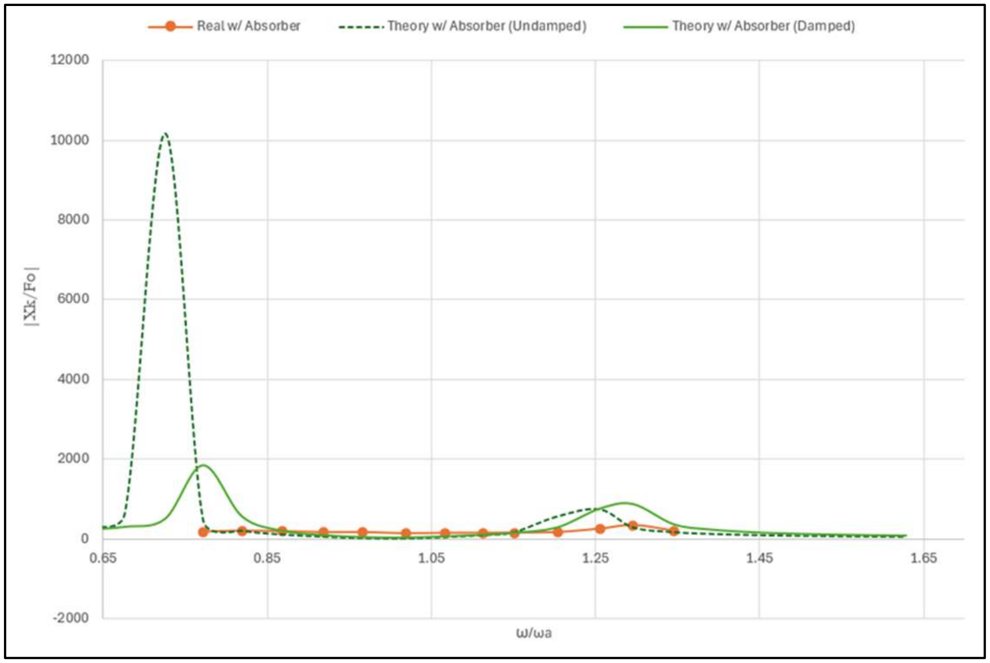
Key Formulas

$$X = \frac{F_0}{|k - m\omega^2|}$$

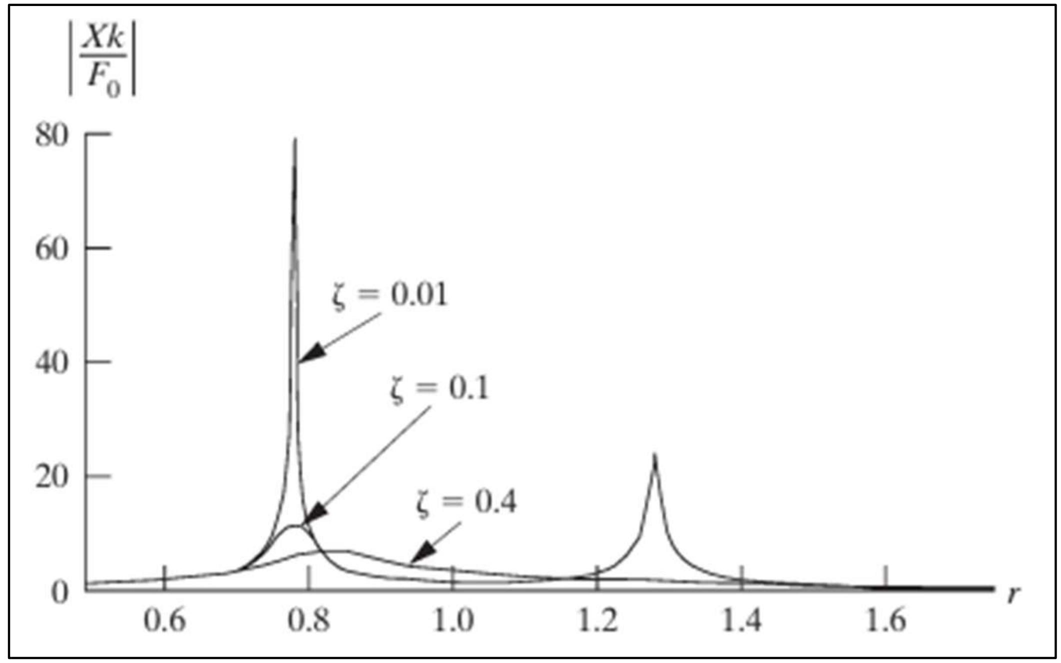
$$\overbrace{\frac{f_0}{\sqrt{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2}}}^{X}$$

$$X = \frac{(k_a - m_a\omega^2)F_0}{(k + k_a - m\omega^2)(k_a - m_a\omega^2) - k_a^2}$$

PROCESSED DATA $(\mu = 0.25)$

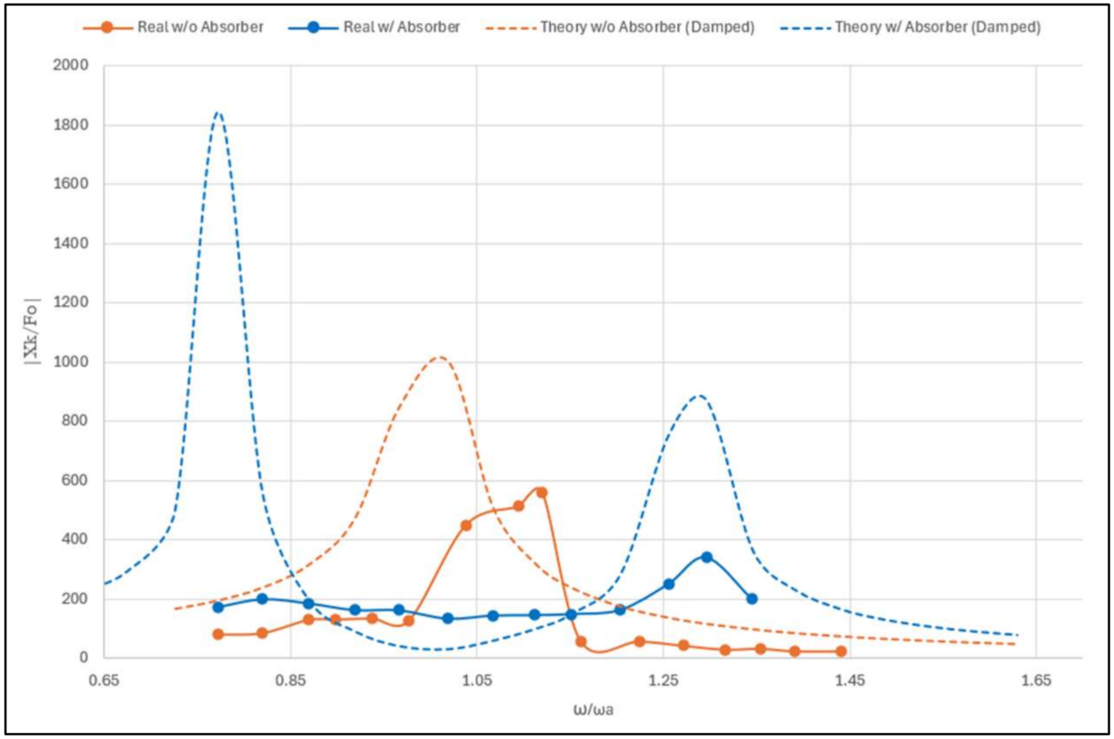


Theoretically Damped, Undamped, and Real Normalized Magnitudes of Primary Mass for $\mu = 0.25$ With Absorber vs. Normalized Frequency

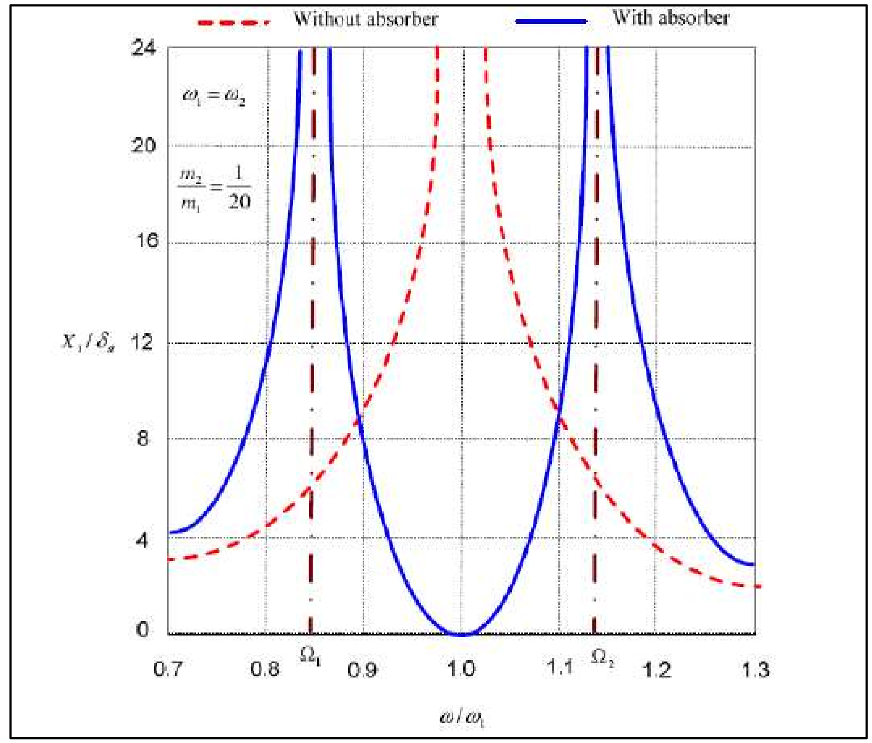


Theoretical Normalized Magnitudes of Primary Mass for $\mu = 0.25$ With Absorber vs. Normalized Frequency for Various Damping Ratios ([Inman, Fig. 5.20]())

PROCESSED DATA $(\mu = 0.25)$



Theoretically Damped and Real Normalized Magnitudes of Primary Mass for $\mu = 0.25$ *With and Without Absorber vs. Normalized Frequency*



Effect of DVA on Response of a Machine (Mirsanei, 2012)

DISCUSSION & TAKEAWAYS

- We modeled the system as an undamped tuned vibration absorber. In reality, some damping exists due to some air drag and primarily friction on the bearings.
- This procedure incorporates experimentation methods highlighted in this course and key to ME301
  - Log decrement
  - Understanding Natural frequencies vs. Driving Frequencies in FFT plots (specifically when taking the free response)
  - Forced response amplitude
- The user should grasp several key factors driving tuned DVAs
  - What do vibration absorbers accomplish? Why might this be useful?
  - What effect does the mass ratio have on a DVAs effectiveness?
  - What could be done differently to fully represent theory in practice with this rig?
  - What was the frequency ratio for this rig? Why was this ratio chosen?
  - What practical limitations deviate experimental results from theory?

CONCLUSIONS & FUTURE WORK

- Our rig successfully demonstrates vibration suppression at resonance with the addition of a DVA.
- Our experimental resonance does not perfectly match the theoretical resonance due to damping.
- Our rig is designed to demonstrate resonance; it would be interesting to design parameters that show the primary and absorber masses oscillating in-phase and out-of-phase, as well as beyond bandwidth.
- As an M1 experiment, each group could ideally test a unique mass ratio so a range from 0.1 – 0.4 could be compared!
- Another option is testing various frequency ratios!