



The Valmont Mitigator LP3 Vibration Damper Mounted to Round Tapered Light Pole

Valmont Mitigator™ LP3

RESEARCH VERIFICATION OF THE VALMONT LP3 DAMPER PERFORMANCE

MITIGATOR LP3 VIBRATION DAMPER

Light pole structures have been observed to have significant first mode vibration. The Valmont Mitigator LP3 Damper is intended to increase damping and reduce vibration in the first mode of these lightly damped structures. This report describes results from laboratory and numerical studies to examine and predict the performance of the LP3 Damper.

LABORATORY TESTING OF LIGHT POLE STRUCTURES

The light pole was a 30 foot round aluminum pole, 6 inches in diameter, tested in the Structures Research Laboratory at the University of Connecticut, as shown in **Figure 1**. A triaxle accelerometer, PCB 356A17 triaxial accelerometer with a sensitivity of 500 mV/g and frequency range from 0.5 – 3000 Hz, was placed at the location of the damper on the pole to measure the acceleration response of the pole.

Free vibration tests were conducted to identify the natural frequency and damping ratio of the light pole, before the LP3 damper performance was measured. The free vibration time history response is shown in **Figure 2**.

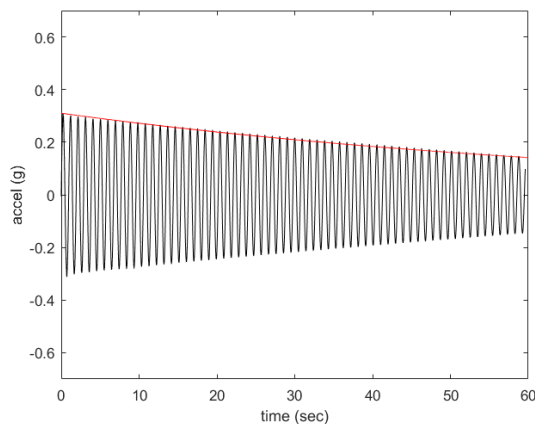


Figure 2: Free Vibration Time History of Light Pole with No Damping Device

The natural frequency, f , was determined as

$$f = \frac{n}{T_n} \quad (1)$$

where n is the number of oscillations and T_n is the time for the n oscillations. The natural frequency of the light pole, f , was observed to be 1.04 Hz.

The damping ratio was determined from the logarithmic decrement

$$\zeta = \frac{1}{2\pi n} \ln \left(\frac{a_o}{a_n} \right) \quad (2)$$

where a_o is the initial response and a_n is the response after n oscillations. The damping ratio was experimentally calculated to be 0.2% (0.002). A plot of the exponential decay of the free vibration response is illustrated in **Figure 2** as a red exponential curve, tracking the decay of the peak of oscillations, thus visually confirming the appropriateness of a 0.2% damping ratio.



Figure 1: Pole Structure Tested at University of Connecticut

Numerical Model of Pole Structure

A low order numerical model of the pole structure is used to evaluate the performance of the proposed LP3 damper. To determine the performance of the LP3 damper, the dynamic weight of the light pole is needed. This dynamic weight is determined numerically using the process described here.

A FEM is developed of the pole structure using in-plane beam elements and 300 nodes (900 degrees-of-freedom). The horizontal, vertical and rotational degrees-of-freedom at the base of the structure are fixed in accordance with the fixed boundary condition at the base of the pole structure. Using the global assembled mass and stiffness matrices, an eigenvalue problem is solved to determine the natural frequencies and mode shapes of the system. The fundamental (first) natural frequency, f_1 , and mode shape corresponding to lateral motion of the pole structure was determined. The discrete mass corresponding to the 52.72 lbs was then added to the horizontal and vertical degrees-of-freedom at the top of the pole to account for the eccentric mass shaker and bracket to support the LP3 damper. The fundamental natural frequency of the pole structure with an added mass of Δm added to the top of the pole is again determined from an eigenvalue problem to be f_2 (where 2 indicates the second condition examined).

$$\omega_1^2 = \frac{k}{m} , \quad \omega_2^2 = \frac{k}{m + \Delta m} \quad (3)$$

The effective mass, m , of the light pole was determined by solving (3) for stiffness and setting the two equations equal to each other such that

$$k = \omega_1^2 m , \quad k = \omega_2^2 (m + \Delta m) \quad (4)$$

where

$$\omega_1^2 m = \omega_2^2 (m + \Delta m) \quad \text{or} \quad f_1^2 m = f_2^2 (m + \Delta m) \quad (5)$$

and solving for mass, m ,

$$m = \Delta m \frac{f_2^2}{f_1^2 - f_2^2} \quad (6)$$

The dynamic weight W_d is defined as: $W_d = mg$ where g is gravity, 32.2 ft/sec² or 386.4 in/sec.

The numerically determined frequency and dynamic weight of the light pole tested at the University of Connecticut is calculated to be 1.08 Hz (compared to the 1.04 Hz measured in the lab) with an associated dynamic weight of 82.94 lbs.

Proposed Luminaire Monopole Shot Ball Damper (Mitigator LP3)

The LP3 damper, shown in **Figure 3**, consists of a shot filled ball rolling on a curved surface which at large amplitudes impacts the walls of the damper housing. This device dissipates energy through the motion of the shot inside the ball and the impact of the ball on the walls of the dampen housing. The unit operates as a vibration absorber as the ball translates on the curved surface when given sufficient space to roll. A proposed mass size is 3.43 lbs, with a 30% inherent damping ratio and a 8 in radius curved surface that provides for an absorber frequency of 0.93 Hz to insure substantial performance for a wide range of the light pole structures (with multiple devices insuring performance for the heavier light pole structures). The inside diameter of the bowl for the 3.54 in diameter ball to travel inside is 5.5 inches.



Figure 3: LP3 Damper Mounted to Square Light Pole

Numerical Model of LP3 Damper

With the dampen parameters identified, a numerical nonlinear model based on energy conservation is developed for the impact behavior of the LP3 damper. An idealized impact damper applied to a simplified undamped single degree-of-freedom light pole, where the mass and stiffness of the light pole, m and k , have an impact damper mass, m_d , in a housing with gap d . The vibration absorber action is not modeled to provide a conservative estimate of performance.

To develop a model of this system, assume a forced sinusoidal motion of the light pole, such that the displacement and velocity of the light pole are:

$$x(t) = a \sin \omega t \quad (7)$$

$$\dot{x}(t) = a \omega \cos \omega t \quad (8)$$

where $x(t)$ is the light pole displacement as a function of time t , a is the amplitude of displacement and ω is the forcing frequency in rad/sec ($\omega = 2\pi f$), where f is the forcing frequency in Hz).

The energy present in this system can be represented completely by strain energy at maximum displacement (zero velocity).

$$E_o = \frac{1}{2} k a_o^2 \quad (9)$$

where a_o is the amplitude of the displacement without impacting.

During an impacting event, the damper mass leaves the wall of the housing when the velocity is maximum (zero displacement, $\mathbf{x}(t) = 0$) and begins to travel across the canister. The damper mass will impact the other side of the canister after some time t_i , where

$$\dot{x}_{max}t_i = a\sin\omega t + d \quad (10)$$

Where d is the housing gap, or difference between the width of the housing and diameter of the ball, from (8) $\dot{x}_{max} = a\omega$ at $t = 0$ or $t = \frac{\pi}{\omega}$ and (10) can be written as

$$a\omega t_i = a\sin\omega t_i + d \quad (11)$$

which can be rearranged as

$$\omega t_i = \sin\omega t_i + \frac{d}{a} \quad (12)$$

With some rearrangement of (12), the time t_i is determined numerically as the solution to

$$\sin\omega t_i + \frac{d}{a} - \omega t_i = 0 \quad (13)$$

but t_i is larger than $\frac{\pi}{\omega}$ as (the time to the end of a half period) then the impact mass has not reached the opposite wall during that half cycle and no impact occurs – no energy is dissipated due to the impact. For $t_i = \frac{\pi}{\omega}$ impacting occurs and the device dissipates energy.

Assuming that upon impact with the opposite canister wall the damper mass dissipates energy to become fixed to the far wall, until such a time that light pole passes back through zero displacement, and that the second impact on the return oscillation dissipates approximately the same energy as the first impact, the energy of the system after one oscillation including the impact is

$$E = \frac{1}{2}ka^2 + 2\left(\frac{1}{2}m_a a^2 \omega^2 - \frac{1}{2}m_a (a\omega \cos\omega t_i)^2\right) \quad (14)$$

where m_a is the moving mass of the impact damper and (14) can be simplified to

$$E = \frac{1}{2}ka^2 + m_a a^2 \omega^2 (1 - \cos^2\omega t_i) \quad (15)$$

The reduction in the light pole displacement amplitude over a single oscillation can be determined by setting the initial energy in the system (9) to the energy after one one oscillation (two impacts) (15), such that

$$\frac{1}{2}ka_o^2 = \frac{1}{2}ka^2 + m_a a^2 \omega^2 (1 - \cos^2\omega t_i) \quad (16)$$

which can be rearranged as

$$a_o^2 = a^2 \left(1 + 2\frac{m_a}{k}\omega^2(1 - \cos^2\omega t_i)\right) \quad (17)$$

The ratio of the amplitude after one cycle a to the original displacement a_o can then be determined, where the mass ratio is defined as the mass of the impact damper over the mass of the main structure, $\mu = m_a/m$ or $m_a = \mu m$, where

$$\frac{a^2}{a_o^2} = \frac{1}{1 + 2\mu \frac{\omega^2}{\omega^2} (1 - \cos^2 \omega t_i)} \quad (18)$$

such that

$$\frac{a}{a_o} = \frac{1}{\sqrt{1 + 2\mu \left(\frac{\omega}{\omega}\right)^2 (1 - \cos^2 \omega t_i)}} \quad (19)$$

Assuming an equivalent system with viscous damping, the response is attenuated as

$$a = a_o e^{-\zeta \omega t} \quad (20)$$

where after one cycle of the forced vibration, $t = \frac{2\pi}{\omega}$ such that

$$a = a_o e^{-2\pi \zeta \frac{\omega}{\omega}} \quad (21)$$

For the equivalent system with viscous damping, the ratio of the original displacement amplitude to amplitude after one cycle can then be determined as

$$\frac{a}{a_o} = e^{-2\pi \zeta \frac{\omega}{\omega}} \quad (22)$$

The equivalent viscous damping ratio is then found to be

$$\zeta = -\frac{1}{2\pi} \frac{\omega}{\omega} \ln\left(\frac{a}{a_o}\right) \quad (23)$$

where $\frac{a}{a_o}$ is determined in (19) and t_i found from the numerical solution of (13).

Performance of LP3 Damper

The LP3 damper was placed at the top of the light pole and both forced and free vibration measurements taken. The pole is initially excited with an eccentric mass shaker with: (a) 1 pound placed at an eccentricity of 3 inches; (b) 1 pound placed at an eccentricity of 6 inches; (c) 2 pounds placed at an eccentricity of 6 inches; and (d) 2 pounds placed at an eccentricity of 12 inches. As such, four levels of excitation are provided to this system (each twice as large an input force as the prior) to examine the amplitude dependence of the LP3 damper. From the acceleration measurement of the pole tip free vibration response, the damping ratio is also determined. The time history plots for each of the poles is shown in **Figure 4**.

Calculated Performance is determined by considering the amplitude given various levels of excitation. For the LP3 damper, the housing gap d is 1.96 in. The excitation frequency $\omega = 2\pi f$, where f is the forcing frequency is set equal to the natural frequency of the light pole, 1.08 Hz. The mass ratio, $\mu = m_a/m$, for this light pole with a shot filled ball weight of 3.43 lbs and a dynamic weight of the pole of 82.94 lbs is, $\mu = m_a/m = 0.0414$. Given these parameters, the time t_i is determined numerically from (13) for different levels of amplitude, a . The attenuated response ratio, $\frac{a}{a_o}$, is determined from (19), which then allows for the equivalent viscous damping ratio, ζ , to be determined from (23). The equivalent viscous damping ratio is iteratively identified for this specific mass ratio for amplitudes of vibration from 0 to 25 inches. The peak damping ratio is calculated to be 0.84%

Laboratory Measured Performance

The LP3 damper is observed in these free vibration experimental tests to increase the damping from 0.2% to 0.83%, 0.83%, 0.80% and 0.60%, for the increasing amplitudes, respectively. The amplitude dependence is shown in **Figure 5** as a plot of the calculated damping ratio as a function of displacement. This corresponds well to the calculated prediction of 0.84%. Further examination is needed to estimate the response reduction for the forced vibration testing.

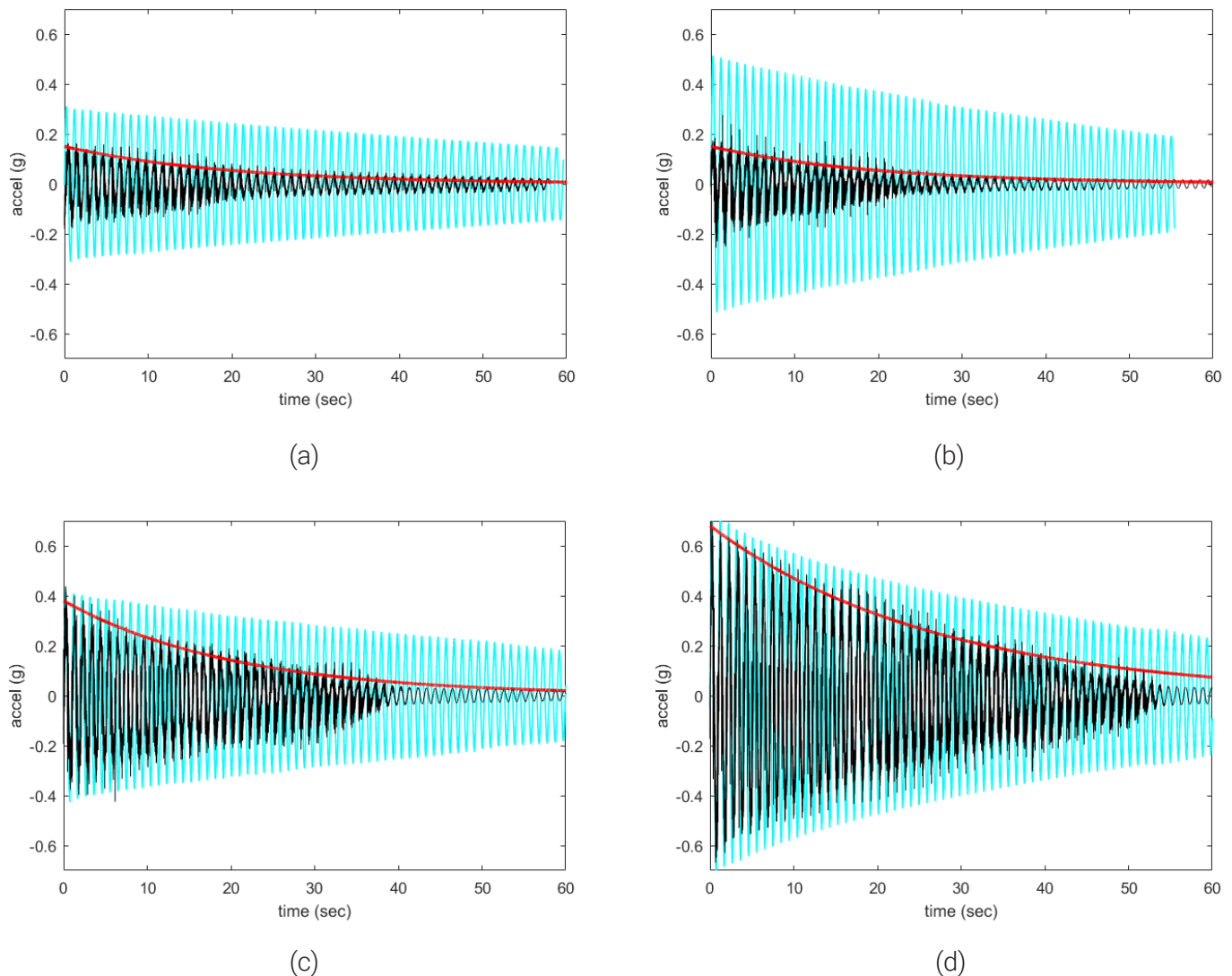


Figure 4: Free vibration time history acceleration for pole with LP3 damper:

(a) $\zeta=0.83\%$ (b) $\zeta=0.83\%$ (c) $\zeta=0.80\%$ (d) $\zeta=0.60\%$

[cyan, uncontrolled measured acceleration; black, LP3 measured acceleration; red, exponential decay]

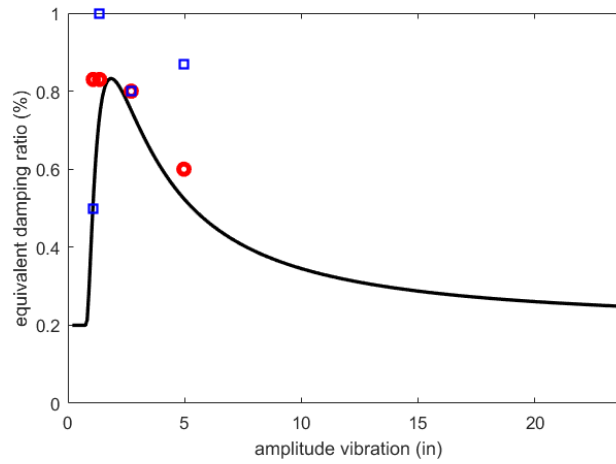


Figure 5: Amplitude Dependence of LP3 damper (free vibration measured data shown as red circles, forced vibration measured data shown as blue squares, predicted calculated performance black line) for the 30' aluminum light pole tested at UConn.

Given the assumed periodic response (7) with amplitude a , the acceleration is

$$\ddot{x}(t) = -a\omega^2 \sin \omega t \quad (24)$$

such that the amplitude of the pole displacement can be calculated from the peak acceleration, \ddot{x}^{peak} (in/sec²), and frequency, ω , as

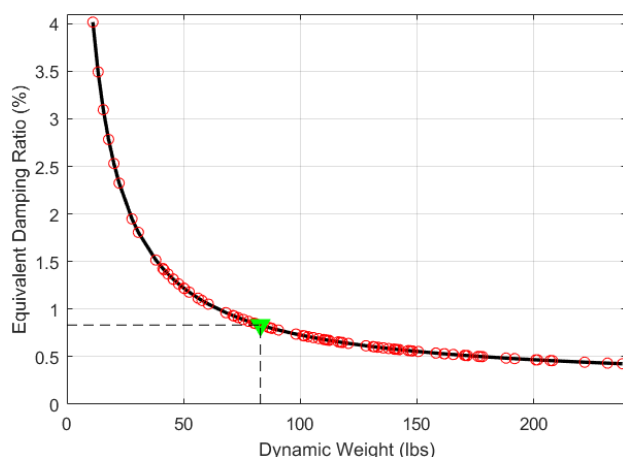
$$a = \ddot{x}^{peak} / \omega^2 \quad (25)$$

The maximum acceleration for the four levels of excitation are 0.12g, 0.12g, 0.30g and 0.55g. From (25) this corresponds to amplitudes of displacement, a , of 1.09 in, 1.18 in, 2.72 in and 4.97 in as measured with the LP3 damper.

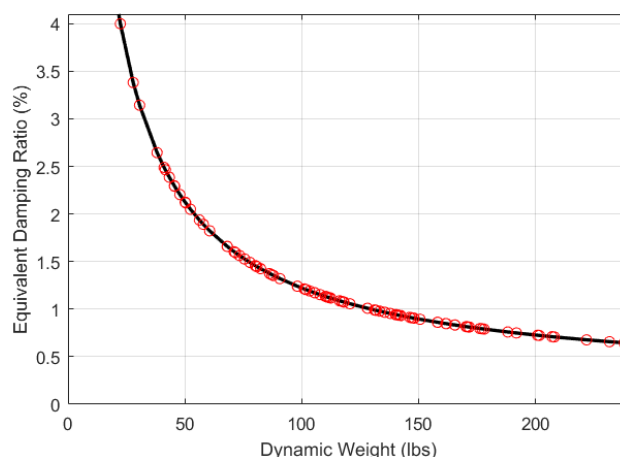
Considering the forced vibration test results, the light pole with no damper attached, for the first excitation (a) the maximum acceleration is measured as 0.30g, which from (26) corresponds to a displacement of 2.72 in and for the larger excitations it can be estimated as 5.43 in, 10.86 in and 21.73 in. This corresponds to response reductions of 60% for low amplitudes of uncontrolled pole vibration less than 3 in (reduced from 2.72 in to 1.09 in). For moderate amplitudes of the uncontrolled pole vibration of 6-12 inches the response reduction is 80% and 75% (reduced from 5.43 in and 10.86 in to 1.09 in and 2.72 in, respectively), and a 77% response reduction is observed for poles with uncontrolled large amplitudes of over 20 inches (reduced from 21.73 in to 4.97 in). This level of performance is determined by the dynamic weight of the light pole and the amplitude of vibration. Using (23), an equivalent damping ratio can be determined to be 0.50%, 1.00%, 0.80%, 0.87%. The measured increased damping ratio is shown in **Figure 5** for these corresponding levels of displacement. The data measured from the laboratory corresponds closely to what is predicted in the model.

Application to Broad Range of Light Poles

The numerical model is shown to quite accurately predict the response of the light pole in the laboratory while being conservative. This model was then used to determine the performance for a range of light poles from dynamic weights from 10 lbs to 300 lbs. The amount of damping in a pole with inherent damping of 0.2% is shown in **Figure 6** for using one LP3 damper and using two LP3 dampers. For comparison, the dynamic weights of the DS330 square steel light poles are shown as circles. The DS330 poles are steel and shown with configurations of 1, 2, 3 and 4 luminaires, representing an extreme and broad range of pole structures. The aluminum poles, which are more susceptible to fatigue, will in general be of less dynamic weight.



one (1) LP3 damper



two (2) LP3 damper

Figure 6: Peak damping performance of LP3 damper as a function of dynamic weight [black, LP3; red circles are DS330 poles; green triangle is 30' aluminum pole tested in laboratory]. (left – 1 device; right – 2 devices)

These results indicate that for a wide range of light poles, the LP3 damper will be extremely effective. The LP3 damper is demonstrated in the laboratory to reduce the vibration in a 30 foot aluminum pole by 80% in a quiet and effective manner. Smaller light poles will experience further improved performance. Most light pole configurations will see over 50% reduction in the damaging vibrations that lead to fatigue, providing a level of safety and security for your pole infrastructure.



The Valmont Mitigator LP3 Vibration Damper Mounted to Square Light Pole



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