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Ground Wind Dynamic Load Alleviation Using Vibration Absorber & Lateral Support for Typical Launch Vehicle

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Abstract— Launch vehicles are subjected to ground wind loads while on launch pad. The response of the vehicle to the steady winds, vortex shedding effect and turbulence can produce significant steady and dynamic loads. Vortices are shedding due to periodic unsteady separation flow along a portion, or entire length of the vehicle. It causes oscillating pressure distribution on the vehicle. Oscillating force distribution due to this pressure excites the vehicle and causes dynamic load on the vehicle. The present work focuses on accurately estimating and reducing the ground wind dynamic load by introducing lateral support and mounting vibration absorber. A finite element model of a typical launch vehicle using beam elements with base fixed condition is modelled in MSC PATRAN software, considering mass distribution, material and geometric properties. Modal (Eigen value) analysis is carried out for natural frequencies and mode shape estimation on basic FE model and on basic FE model with lateral support. Transient response analysis is carried out for the dynamic load estimation due to vortex shedding for basic model, basic model with lateral support and basic model with tuned mass damper. Displacement is also estimated for the three cases. Estimation of vortex caused dynamic loads and the analysis were done by using MATLAB code. Introduction of lateral support at 35% of vehicle length from nose tip makes the frequency higher. Dynamic bending moment (BM) decreased due to this frequency shift. Mounting of tuned mass damper (TMD) system to the vehicle model at 25% and 35% of vehicle length from nose tip. For two tuned mass systems, normalised tip displacement is less than that of base fixed condition. Normalised dynamic BM is reduced for excitation corresponding to first and second natural frequency. Thus it is concluded from this study that dynamic BM is effectively reduced by using TMD.

Index Terms— Launch vehicle, Vortex shedding, Strouhal number, Tuned mass damper, Lateral support, Modal analysis, Transient response analysis

1 INTRODUCTION

A launch vehicle is a rocket propelled vehicle used to carry a payload from Earth's surface, either into orbit around Earth or to some other destination in outer space. A launch system includes the launch vehicle, launch pad, vehicle assembly and fueling systems range safety, and other related infrastructure. A launch vehicle is subjected to different environments during launch operation and its flight, and that must be considered for the design and safe operation of the system. These include environments such as wind loading, acoustic vibration, and vibration due to engine thrust. These environments cause the launch vehicle to experience forces that cause structural deformations and vibrations. These forces produce the internal structural loads and stresses that represent the principal design requirements for the launch vehicle. Launch vehicles prior to launch are exposed to steady wind loads and to the vortex turbulent wake of nearby structures such as a launch tower, masts, and the buildings. The response of the vehicle to the winds and turbulence can produce significant steady and oscillatory loads which are dynamic loads. These loads must be accounted for in the design of the vehicle and support structures and in the development of ground and launch operation plans. In addition, the natural wind environment can be unpredictable and severe at times so a vehicle must be designed to withstand the wide range of wind conditions that could occur over the time. Ground wind loads are of three components. The first component is steady state wind loading that is defined in terms of steady state lift and drag coefficients that can be used to define loads on the launch vehicle. Drag is defined to act on the vehicle cross section, parallel to the wind velocity and lift is defined to act perpendicular to the wind velocity. The second wind loading component is a result of wind gust and turbulence. The third component is the dynamic loading component that results when vortices created by periodic unsteady separation flow along a portion, or entire length of the vehicle, forms the wake of a launch vehicle.

2 VORTEX SHEDDING

When the wind blows across a bluff body, vortices are shed alternately from one side and then the other, giving rise to a fluctuating force acting at right angles to the wind direction. This organized pattern of vortices is referred to as von Karman vortex street. This vortex shedding force will be almost periodic and some wind speeds, it is resonant in character. Vortex shedding is the instance where alternating low pressure zones are generated. These alternating low pressure zones cause the circular section to move towards the low pressure zone, causing movement perpendicular to the direction of the wind. When the critical wind speed of the circular section is reached, these forces can cause to resonate where large forces and deflections are experienced.

Vortex shedding forces are a function of wind velocity, characteristic diameter of which there can be several depending on vehicle stage geometry, wind profile and flow characteristics, and the frequency and amplitude at which a vehicle responds. For lightly damped structures, which are free to oscillate, large amplitude vibrations in the plane normal to the wind may develop when the vortex shedding is near or at resonance with one of the natural frequencies of vibration. This phenomenon is occurring for a range of wind speed and referred to as "lock-on" or "lock- in". It can produce oscillatory loads, enough to damage the structure or levels of motion. These oscillations, which can lead to loads that are an order of magnitude greater than those associated with steady loads. Strouhal number is often used to describe the occurrence of vortex shedding.

$$\text{Strouhal number } S_t = f_s{*}D/V \qquad (1)$$

Where fs is the vortex shedding frequency in Hz, D is the cross section diameter, and V is wind velocity

3 FINITE ELEMENT MODEL

Launch vehicle at launch pad is idealized in to beam element with fixed at one end. MSC/PATRAN software is used for finite element modeling and shown in Figure 1. Modal characteristics of the launch vehicle like frequency, mode shape etc., are generated through finite element analysis. The finite element analysis of a typical launch vehicle using beam element is carried out using MSC / PATRAN software, considering mass distribution, material and geometric properties.

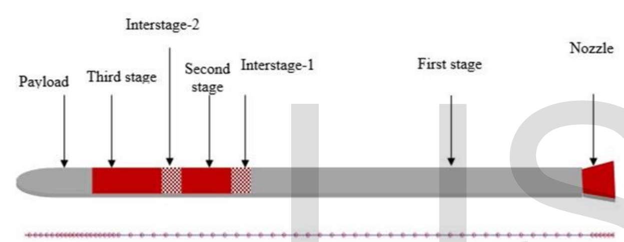

Figure 1: Finite Element Model of a Launch Vehicle

3.1 Modal or Eigen Value Analysis

A modal analysis calculates the natural frequencies and its mode shape of a given system. The analysis of the model is done for base fixed condition (cantilever condition) and for base fixed with lateral support condition.

3.2 Natural Frequencies

The natural frequencies of a structure are the frequencies at which the structure naturally tends to vibrate if it is subjected to a disturbance. Natural frequencies and the corresponding modes are computed using normal mode analysis option (SOL 103). First natural frequency of this typical vehicle for base fixed condition and lateral support condition is nearly 0.8 Hz and 2.5 Hz respectively.

3.3 Mode Shapes

The deformed shape of the structure at a specific natural frequency of vibration is termed its normal mode of vibration. Some other terms used to describe the normal mode are mode shape, characteristic shape, and fundamental shape. Each mode shape is associated with a specific natural frequency. First three mode shapes of the typical launch vehicle for base fixed condition (cantilever condition) and for base fixed with lateral support condition is shown in Figure 2 and Figure 3 respectively.

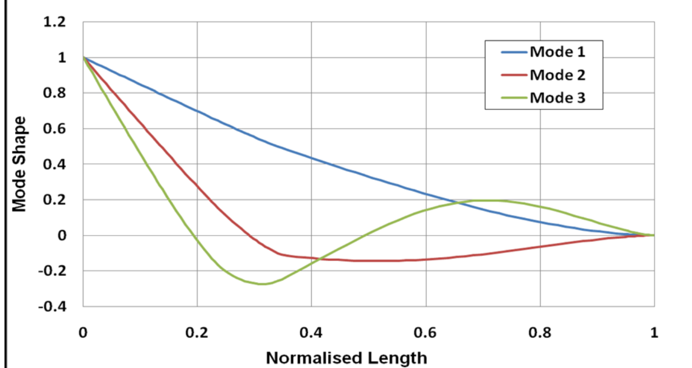

Figure 2: Normalised mode shape for the model with cantilever condition

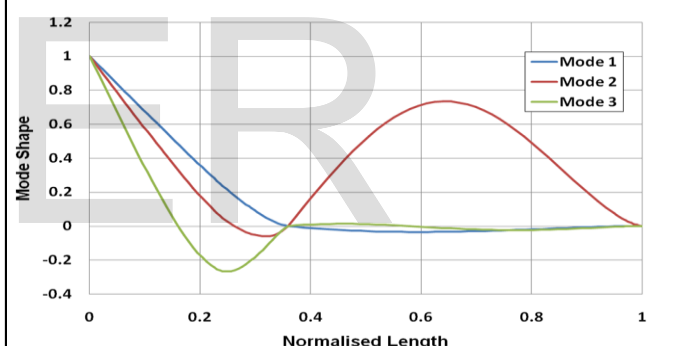

Figure 3: Normalised mode shape for the model of cantilever condition with lateral support

4 TRANSIENT RESPONSE ANALYSIS

Transient response analysis is the most general method for computing forced dynamic response. The purpose of a transient response analysis is to compute the behavior of a structure subjected to time-varying excitation.

Transient load estimated below is acting on entire length of the vehicle. Critical damping of 1% is considered.

$$\text{Transient load} = \sin\omega t \qquad (2)$$

$$\text{Where} \quad A = C_L{*}D{*}\delta x \qquad (3)$$

$$C_L = 0.35$$

$$D = \text{Diameter}$$

$$\delta x = \text{Element length}$$

$$\omega = 2\pi f \qquad (4)$$

$$f = S_t{*}\,V/D \qquad (5)$$

$$S_t = \text{Strouhal number}$$

$$V = \text{Wind speed}$$

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4.1 For Base Fixed Condition

First mode of frequency ≈ 0.8 Hz
Using equation (5)
Critical wind speed = 6.3 m/ s
Second mode of frequency ≈ 3.0 Hz
Critical wind speed = 23.8 m/s

Transient analysis is carried out using NASTRAN for 3 m/s, 6.3 m/s, 10 m/s, 15 m/s, 20 m/s, 23.8 m/s, 25 m/s, 30 m/s and displacement response (in m) is normalised by vehicle diameter and plotted as shown in Figure 4. In Figure 4, 6.3 m/s and 23.8 m/s are the critical wind speeds corresponding to first and second mode of frequency. So response in these speeds is higher.

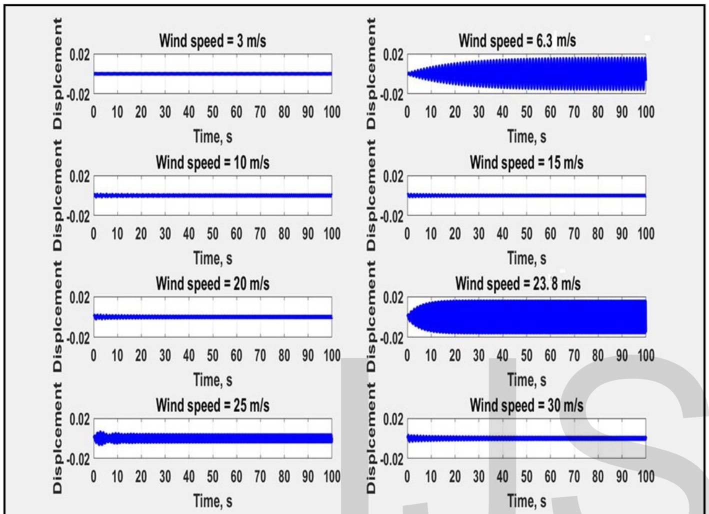

Figure 4: Displacement history for base fixed condition

Dynamic bending moment (BM) at the base of vehicle corresponding to each wind speed is shown in Figure 5 to Figure 8. Excitation and dynamic BM are normalised. For critcal wind speed of 6.3 m/s and 23.8 m/s, dynamic BM is 50 and 3.7 times higher than static part of vortex shedding force caused bending moment. For other wind speeds, BM at base is almost equal static part of vortex shedding BM since these wind speeds are not exciting vehicle to first or second natural frequencies.

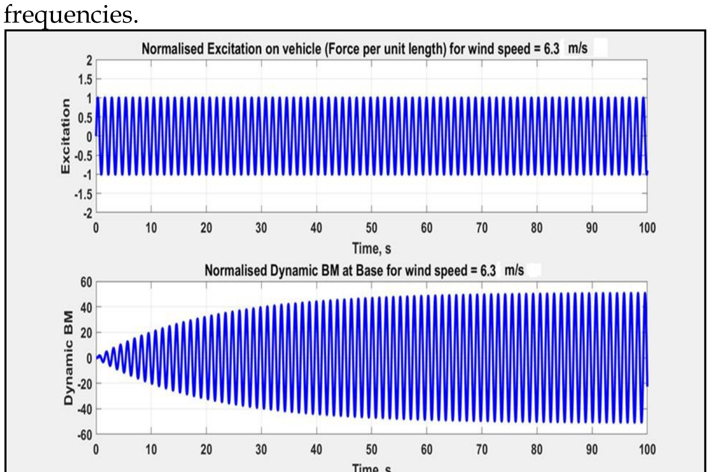

Figure 51: Dynamic bending moment at base for V = 6.3 m/s

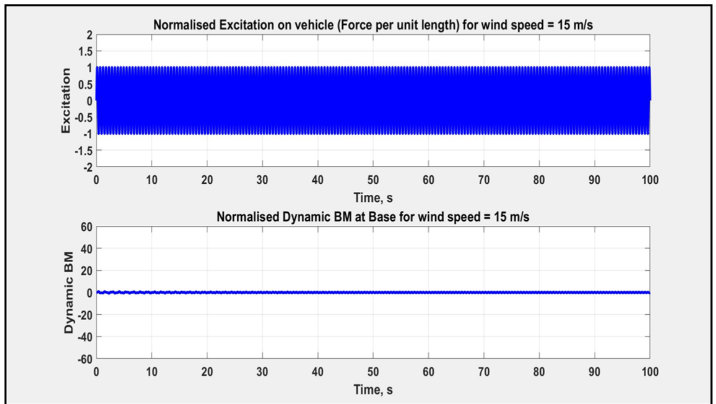

Figure 6: Dynamic bending moment at base for V = 15 m/s

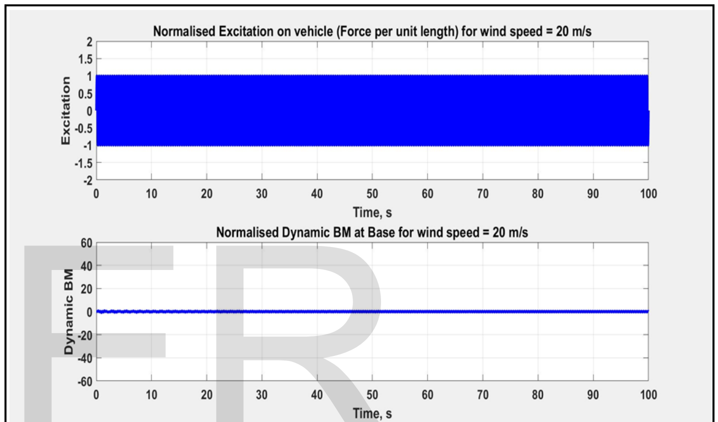

Figure 7: Dynamic bending moment at base for V = 20 m/s

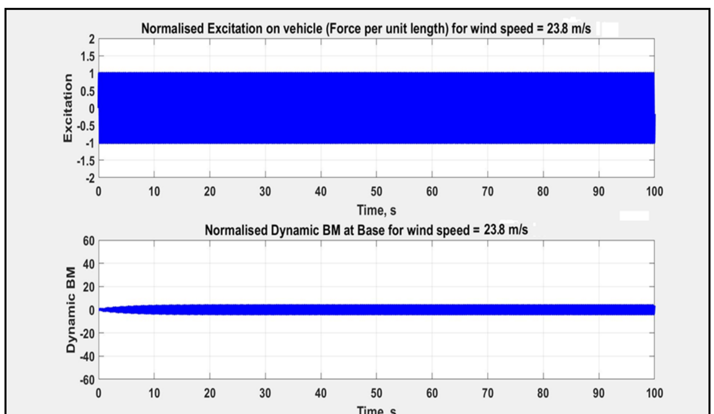

Figure 8: Dynamic bending moment at base for V = 23.8 m/s

4.2 For Base Fixed with Lateral Support

Critical wind speed is estimated considering first bending frequency near 2.5 Hz.

Critical velocity = 20.5 m/s

Transient analysis is carried out using NASTRAN for wind speed of 3 m/s, 6.3 m/s, 10 m/s, 15 m/s, 20 m/s, 20.5 m/s, 25 m/s, 30 m/s. Normalised displacement with respect to vehicle diameter is plotted for above wind speeds and shown in Figure 3. From Figure 9, Normalised displacement response is higher at wind speed 20.5 m/s. Response at 20 m/s is slightly greater than other velocities.

Dynamic BM at base of vehicle corresponding to each wind speed is shown in Figure 9 to Figure 13. Excitation and dynamic bending moment are normalised. Dynamic bending moment is higher at wind speed 20.5 m/s and then 20 m/s. When compared to the base fixed condition, the response and the dynamic load of the lateral support are lower for wind speed below 20 m/s. Normalised dynamic BM for 20.5 m/s is 3.8. So it is clear that we can reduce the vibration of the launch vehicle upto a level by introducing a lateral support.

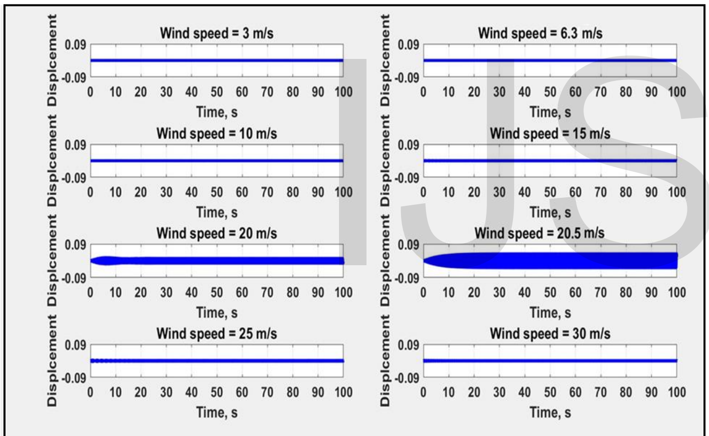

Figure 9: Displacement history for base fixed with lateral support condition

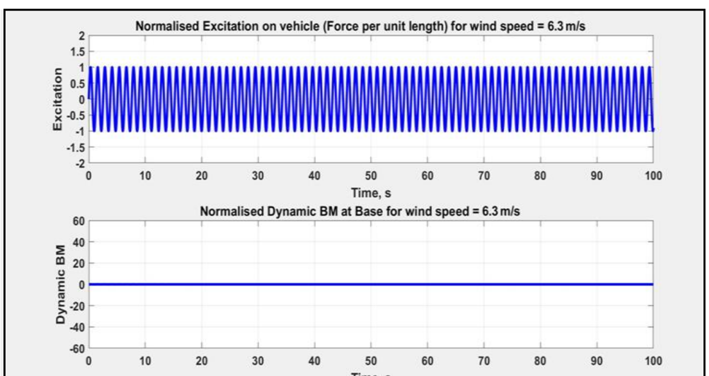

Figure 10: Dynamic bending moment at base for V = 6.3 m/s

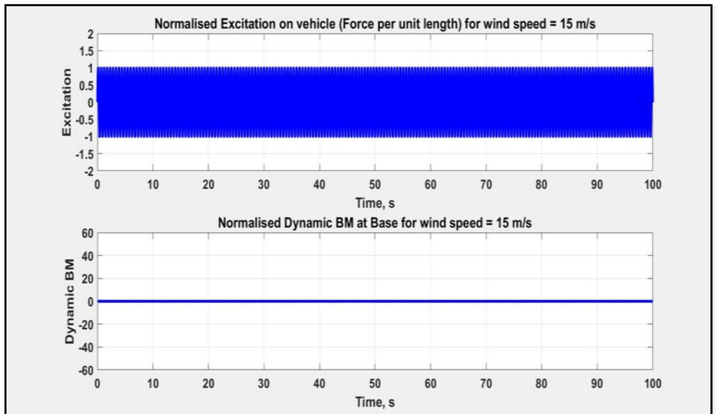

Figure 11: Dynamic bending moment at base for V = 15 m/s

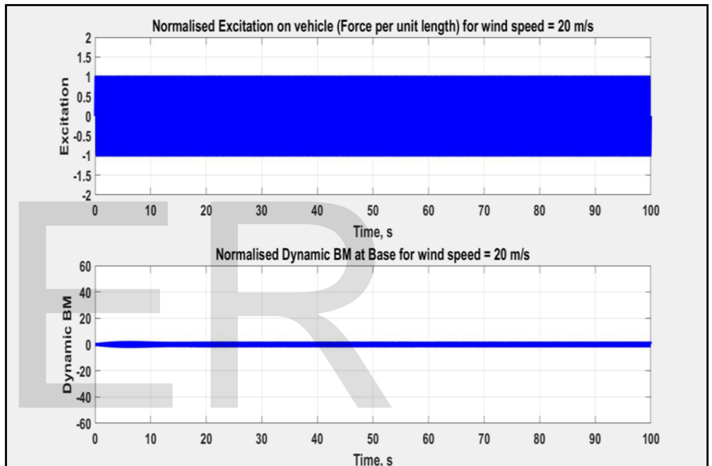

Figure 12: Dynamic bending moment at base for V = 20 m/s

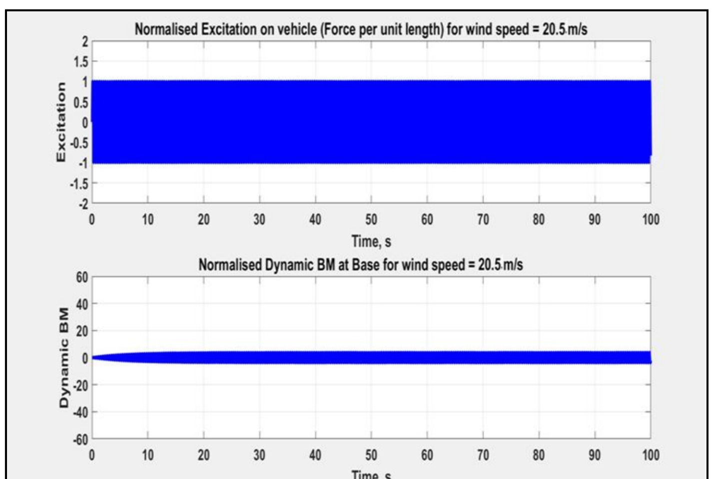

Figure 13: Dynamic bending moment at base for V = 20.5 m/s

5 Tuned Mass Damper

The Tuned mass damper (TMD) is a secondary system consisting of a mass, spring and damper and it is connected to primary system (i.e. the structure) to reduce mechanical vibrations of primary system. Figure 14 shows a typical tuned mass damper with primary system. The basic concept of tuned mass damper is that when the resonance of the tuned mass damper is tuned to match that of the primary system, the motion of the primary system is reduced to zero or to acceptable level at its resonance frequency. Therefore, the frequency and damping of the TMD is tuned so that when the structure resonates at a particular frequency, the TMD vibrates out-of-phase transferring the energy from the primary system to the secondary system (i.e. the TMD) and dissipating it in the damper.

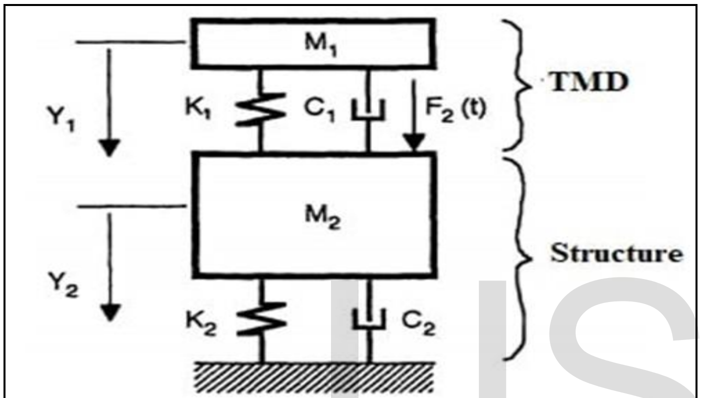

Figure 14: Tuned mass damper with primary structure

Tuned mass damper is mounted to a specific location in a structure so as to reduce the amplitude of vibration to an acceptable level whenever a strong lateral force such as an earthquake or high winds hit. Their application can prevent discomfort, damage, or outright structural failure. While on launch pad, launch vehicle will experience vibrations due to vortex shedding and tuned mass damper is the next attempt to alleviate the vibration.

5.1 Transient Response Analysis with TMD

Modal analysis of the model with tuned mass damper is carried out for finding natural frequencies. Mass ratio and damping ratio (ζ) considered are 0.0082 and 0.01 respectively. Hence total mass of the tuned mass is 0.0082 times the total mass of vehicle. To analyse the model with TMD, three parameters are to be considered.

Transient response analysis is carried out for two cases :

a) Model with one mass damper system (For tuning first natural frquency)

Tuned mass is mounted at the 25% of vehicle length from nose tip with the following mass, stiffness and damping force:

Mass , m_1 : 0.0082* Total vehicle mass
Stiffness , $k_1 = m_1/\ \omega_1^2$
Damping force $c_1 = 2\zeta m_1\ \omega_1$

b) Model with two mass damper systems (For tuning first & second natural frquencies)

First tuned mass is mounted at the 25% of vehicle length from nose tip for tunning first natural frequncy and second one is at 35% of vehicle length for tunning second natural frequency. Mass, stiffness and damping force for each TMD are given below:

Total Mass , m_t : 0.0082* Total vehicle mass
Stiffness, $k = m_t/\ \omega_1^2+\omega_2^2$

Parameters for first TMD:
Stiffness , $k_1=k$
Mass , $m_1=k_1\ \omega_1^2$
Damping force , $c_1= 2\zeta m_1\ \omega_1$

Parameters for second TMD:
Stiffness , $k_2=k$
Mass , $m_2=k_2\ \omega_2^2$
Damping force $c_2=2\zeta m_2\ \omega_2$

Critical wind speeds for vehicle with tuned mass damper are 5.8 m/s, 6.9 m/s and near 23.8 m/s. Normalised displacement at the vehicle tip is shown in Figure 15. It is observed that introduction of the damper system reduces the tip displacement. Dynamic bending moment for each critical velocity case is plotted for one TMD. Normalised dynamic BM plots are shown in Figure 16 to Figure 19. Normalised dynamic BM is reduced from 50 to 20 for wind speeds which excite the vehicle in the first natural frequency. No change in the BM corresponding for exciation in the second natural frequency since there is no TMD mounted for this frequency.

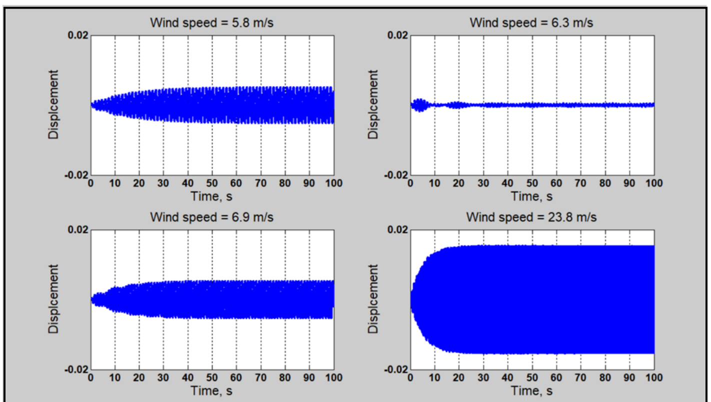

Figure 15: Displacement history at vehicle tip for one tuned mass system

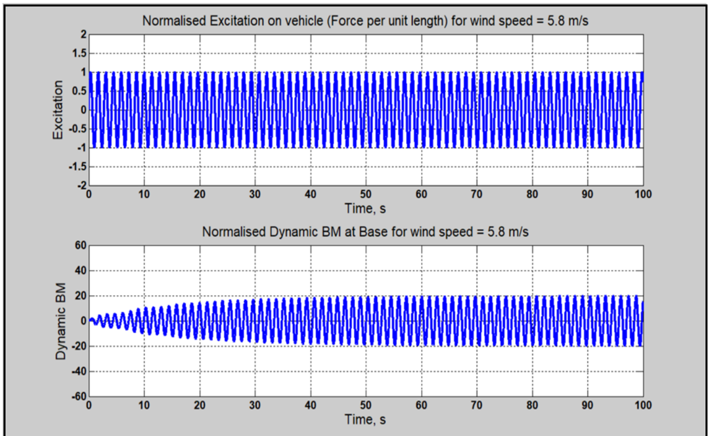

Figure 16: Dynamic bending moment at base V = 5.8 m/s for one tuned mass system

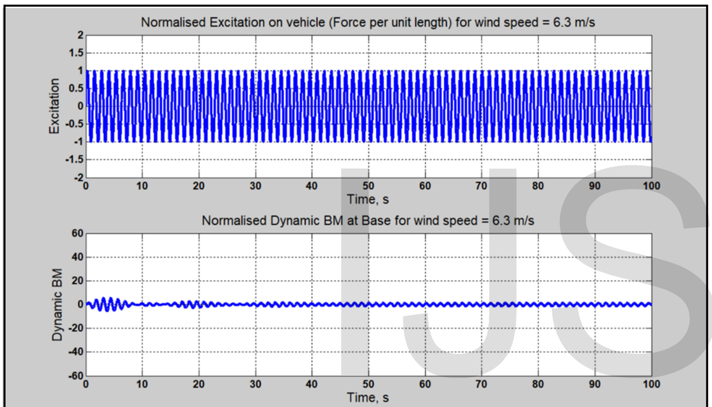

Figure 17: Dynamic bending moment at base V = 6.3 m/s for one tuned mass system

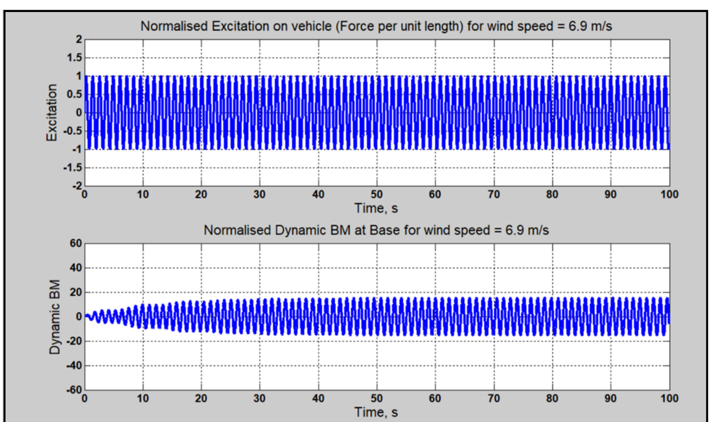

Figure 18: Dynamic bending moment at base V = 6.9 m/s for one tuned mass system

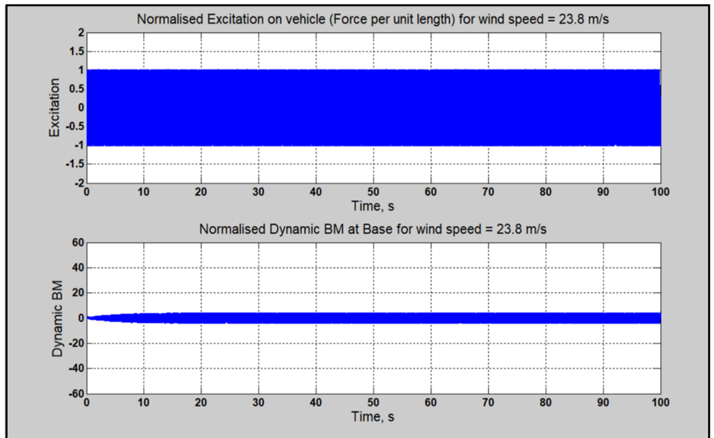

Figure 19: Dynamic bending moment at base V = 23.8 m/s for one tuned mass system

When two TMD systems are introduced, vehicle tip displacement response is reduced for excitation corresponding to first and second natural frequencies. Normalised displacement response at tip is shown Figure 20. Dynamic bending moment for each critical velocity case is plotted. Normalised plots are shown in Figure 21 to Figure 24. Normalised dynamic BM is reduced from 50 to 20 for wind speeds which excite the vehicle in the first natural frequency. BM value of 3.7 to 1.3 is reduced for excitation corresponding to second natural frequency. Hence TMD reduces the dynamic BM.

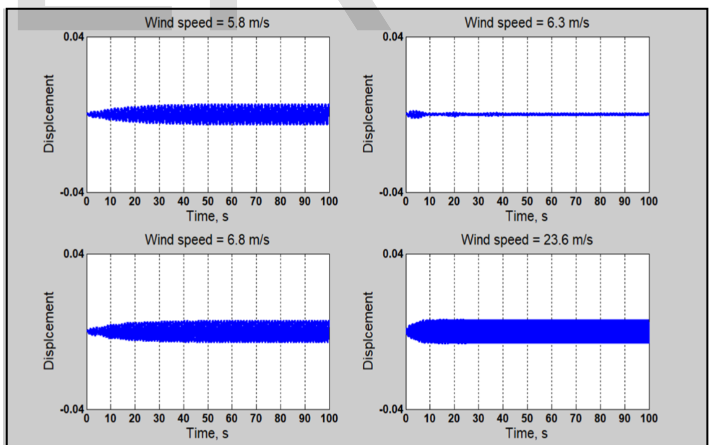

Figure 20: Displacement history at vehicle tip for two tuned mass system

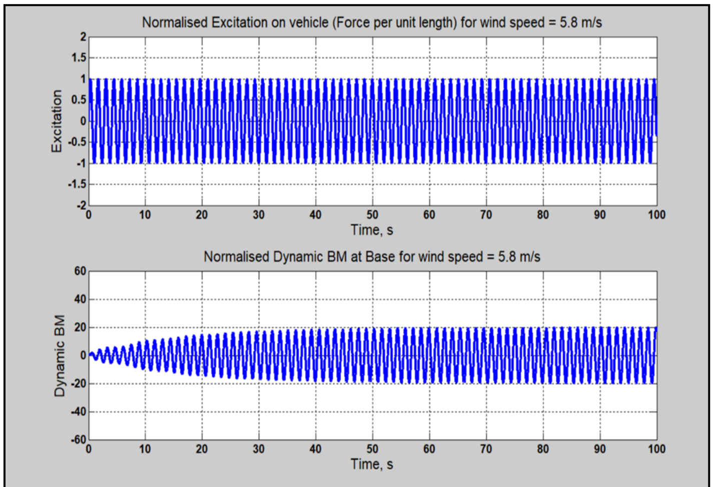

Figure 21: Dynamic bending moment at base V = 5.8 m/s for two tuned mass system

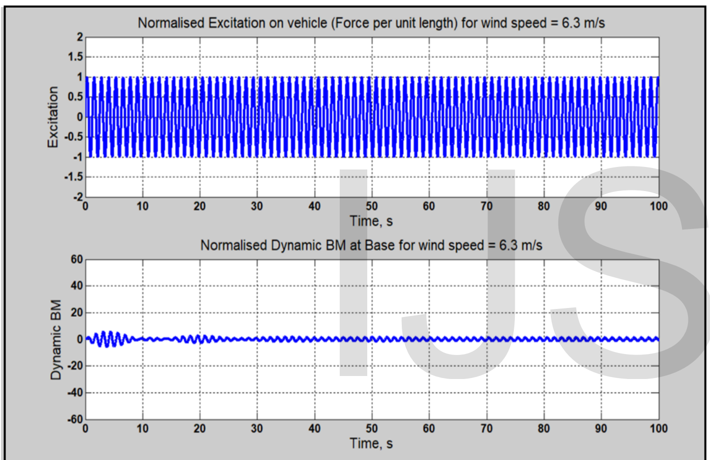

Figure 22: Dynamic bending moment at base V = 6.3 m/s for two tuned mass system

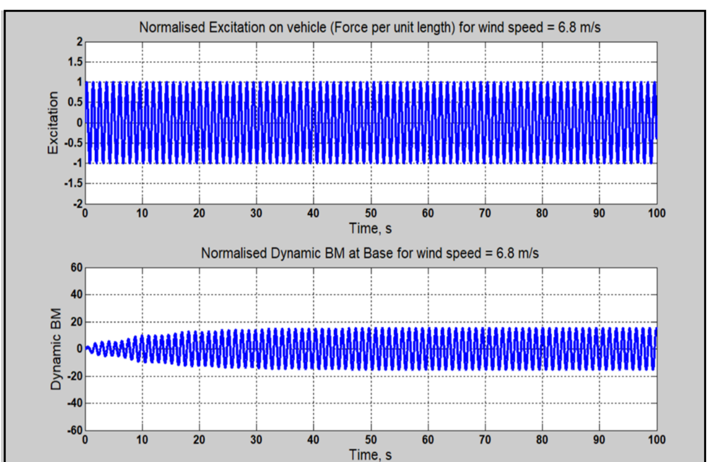

Figure 23: Dynamic bending moment at base V = 6.8 m/s for two tuned mass system

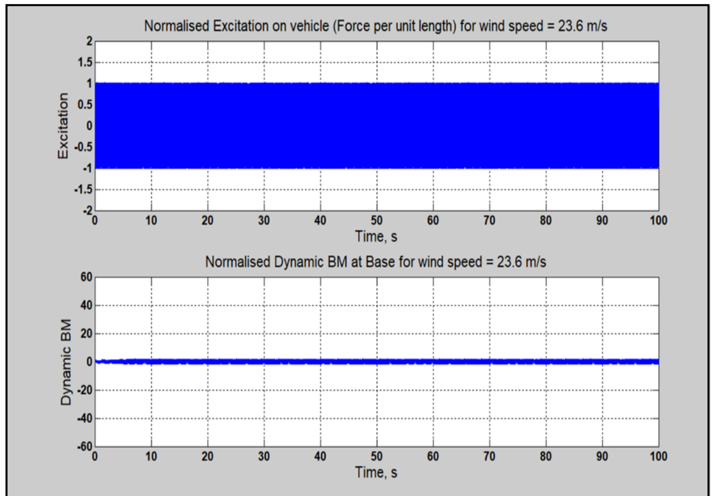

Figure 24: Dynamic bending moment at base V = 23.6 m/s for two tuned mass system

6 CONCLUSIONS

Launch vehicles are subjected to ground wind loads during launch operation. The response of the vehicle to the steady winds, vortex shedding effect and turbulence can produce significant steady and dynamic loads. Vortices are shedding due to periodic unsteady separation flow along a portion, or entire length of the vehicle. It causes oscillating pressure distribution on the vehicle. While on launch pad, vehicle is at base fixed condition. Oscillating force distribution due to this pressure excites the vehicle and causes dynamic load on it.

Critical wind speed of 6.3 m/s and 23.8 m/s is corresponding to first and second natural frequencies of vehicle at base fixed condition. For these critical wind speeds, dynamic BM is 50 and 3.7 times higher than static part of vortex shedding force caused bending moment. For other wind speeds, BM at base is almost equal static part of vortex shedding BM since these wind speeds are not exciting vehicle to first or second natural frequencies. In the present work, two attempts are adopted to alleviate the dynamic load due to vortex shedding.

The first attempt is the introduction of lateral support at 35% of vehicle length from nose tip. Lateral support shifts the frequency to higher. Due to this frequency shift dynamic bending moment (BM) is decreased and Normalised dynamic BM for 20.5 m/s is 3.8.

The second attempt is the mounting of tuned mass damper (TMD) system which was mounted to the vehicle model at 25% and 35% of vehicle length from nose tip. For two tunned mass systems, normalised tip displacement is less than that of base fixed condition. Normalised dynamic BM is reduced from 50 to 20 for wind speeds which excite the vehicle in the first natural frequency. BM value of 3.7 to 1.3 is reduced for excitation corresponding to second natural frequency. Hence the dynamic BM is effectively reduced by TMD.

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