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Linear Characterization of Engine Mount and Body Mount for Crash Analysis

Abdolvahab Agharkakli, Digvijay Pradip Wagh

Abstract— ***This study summarizes the methodology to find the linear mount characteristics with the help of mathematical models and comparison of these results with results from MATLAB simulations. The mounts are treated at the component level, and mathematical models for the same are evaluated to get the required characteristics. The mounts are modelled as spring and damper system subjected to impact loading that occurs during crash events. The approximation of input pulse has been described mathematically, which then serves to find the characteristics of the mounts. The change in the characteristics of mounts with the change in the velocity of impact has also been studied.***

Index terms— ***Engine Mount, Body Mount, Vibration and Harshness (NVH), Crash Pulse Approximation***

I. INTRODUCTION

The engine mount and body mount is a complex joint assembly comprising of rubber bushings on the top and bottom of the frame bracket, a bolt, and retainer. The engine/body mounts are designed to carry the horizontal impact load in an impact and to isolate the noise, vibration and harshness (NVH), occurring during driving from entering the passenger compartment. The design of an engine mount is basically dependent on the capacity of the rubber bushing to withstand the loads coming, the loads that an engine or body mount is subjected to is compression during normal functioning of the vehicle and tensile and shear during crash events. For a mount to sustain these loads coming on it, it must have enough strength in the prescribed directions.

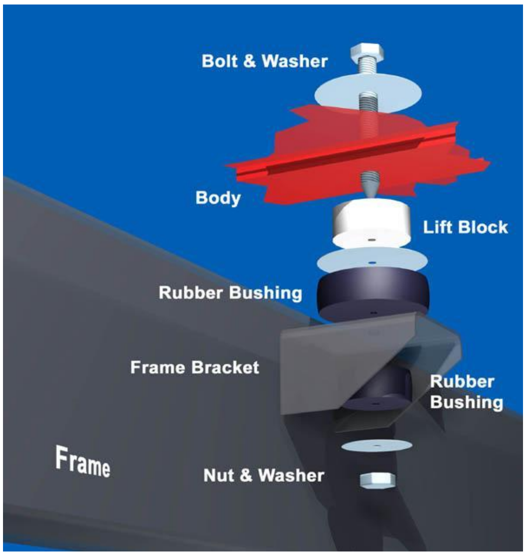

Fig. (1): Schematic model of mount

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The problem in the analysis of mounts is that, during an impact, the impact forces from chassis system are transferred to the cab or engine, while transferring impact forces the body mounts influence the 'crash pulse' to the occupant compartment, during this processes of crash the mounts may fail and this would result in endangering the lives of passengers, thus in order to prevent this the mounts are designed to take the horizontal loads, from the safety point of view, the dismantling of the component assembly is not acceptable. To test the engine/body mount for their functional capabilities full vehicle crash tests are conducted, a crash test is a form of destructive testing usually performed in order to ensure safe design standards in crashworthiness and crash compatibility for various modes of transportation or related systems and components.

CAE is again a new way of predicting the behaviour of a crash vehicle during crash events, the crash characteristics of engine/body mount can also be determined from CAE analysis. But, the time and cost involved in CAE analysis is also huge, and hence we are interested in determining the characteristics of mounts by analytical computations, which would result in saving the time and cost involved. The various functionalities and dynamic characteristics of mounts in vehicles have long been investigated both experimentally and theoretically. However, most of these studies are intended for the NVH (Noise Vibration and Harshness) analysis, only a few are for the crash safety applications. P. Michael Miller, J. C. Lee of MGA Research Corporation and R. Krishna Murthy, James Cheng of Ford Motor Company, in the year 1998, studied the role of the body mount on the passenger compartment response of a frame/body structured vehicle in frontal crash [3], which was a comprehensive strategy to investigate the role of the body mounts on the passenger compartment response in a frontal crash event is presented. Methodology On The Testing Of The Automobile Mount Dynamic Response, this paper by Yijung Chen, Xiaodong Zhang, Tau Tyan and, Omar Faruque [7], reports the latest development of methodologies for testing and CAE modelling of the automobile mounts. The objective of this study is to provide dynamic mount properties for product evaluation and CAE modelling guideline for crashworthiness simulations. In the year 2003, a paper authored by researchers from the ford motor company revealed a systematic investigation of the body mounts' dynamic characteristics in component, sub-system and full system levels and its application in the frontal impact analysis of a body-on-frame (BOF) vehicle [8],[9]. The research done so far has its focus on determining the mount dynamic characteristics by CAE and practical testing of mounts. Though proper mathematical treatments are not available to find the dynamic characteristics of mount and also the methodologies adopted so far are time consuming and costly. Thus, the study is needed to be done in two steps, firstly finding the linear characteristics and then the non-linear (Dynamic) characteristics.

II. CRASH PULSE APPROXIMATION

To supplement full scale dynamic testing of vehicle crashworthiness, mathematical models and laboratory tests (such as those using a huge sled or a vehicle crash simulator) are frequently employed. The objective of these tests is the prediction of changes in overall safety performance as vehicle structural and occupant restraint parameters are varied. To achieve this objective, it is frequently desirable to characterize vehicle crash pulses such that parametric optimization of the crash performance can be defined. Crash pulse characterization greatly simplifies the representation of crash pulse time histories and yet maintains as many response parameters as possible. The response parameters used to characterize the crash pulse are those describing the physical events occurring during the crash such as (maximum) dynamic crush, velocity change, time of dynamic crush, centroid time, static crush, and separation (rebound)velocity. A number of crash pulse approximations and techniques have been developed for the characterization [2]. These are divided into two major categories according to whether or not the initial deceleration is zero. The crash pulse here is a crucial factor. It determines the RHS of the governing equation of motion. The crash pulse has been approximated by the following ways:

1) Excitation with Constant Acceleration:-

This approximation uses a constant value of acceleration which is obtained by averaging the accelerations of the moving barrier. Again, we may use the initial velocity for the moving parts of the fixture, thus we have two cases, one with zero initial velocity and the other with initial velocity equal to the barrier velocity.

Here, forcing function is given as:

$$F(t) = Mr \times A \tag{1}$$

Initial velocity is: $V_0 = V_{12}$ = barrier approach velocity.

2) Excitation with Halfsine Approximation:-

Halfsine wave is half of the sine pulse and the values for the acceleration are obtained by solving the equations as follows. The detailed derivation is not considered here but for further knowledge we may see the appendix.

Here, forcing function is given as:

$$F(t) = Mr \times A(t) \tag{2}$$

Initial velocity is: $V_0 = V_{12}$ = barrier approach velocity.

Where,

$$A(t) = A_p \times \sin(\omega t) \tag{3}$$

$$A_P = \frac{-V_{12}\omega}{1-\cos(\omega t_m)} \tag{4}$$

And, A_p = Peak Amplitude.

ω = Forcing frequency.

t_m = time of dynamic crush.

3) Excitation with Haversine Approximation:-

The haversine approximation approximates the given crash pulse at three points and thus is more appropriate than the halfsine.

Here, forcing function is given as:

$$F(t) = Mr \times A(t) \tag{5}$$

Initial velocity is: $V_0 = V_{12}$ = barrier approach velocity.

Where,

$$A(t) = \frac{A_p}{2}[1 - \cos(\omega t)] \tag{6}$$

$$A_p = \frac{-2V_{12}\omega}{(\omega t_m) - \sin(\omega t_m)} \tag{7}$$

And, A_p = Peak Amplitude.

ω = Forcing frequency.

t_m = time of dynamic crush.

4) Excitation with Fourier Equivalent Wave:-

The FEW is most appropriate to use when we have a given crash pulse and it approximates the crash pulse accurately and thus a nonlinear curve can be easily represented in a mathematical form.

Here, forcing function is given as:

$$F(t) = Mr \times A(t) \tag{8}$$

Initial velocity is: $V_0 = V_{12}$ = barrier approach velocity.

Where,

$$A(t) = \sum_{i=1}^{n}[a_i \times \sin(\omega t)] \tag{9}$$

$$a_i = \frac{2t_k}{(i\pi)^2}\sum_{j=1}^{k} m_j\left[\sin\left(\frac{i\pi t_j}{t_k}\right) - \sin\left(\frac{i\pi t_{j-1}}{t_k}\right)\right] \tag{10}$$

$$m_j = \frac{A_j - A_{j-1}}{t_j - t_{j-1}} \tag{11}$$

III. MATHEMATICAL MODELLING

The mounts are assumed to be a spring and damper system, and the model proposed here is for component level test of mounts on a test fixture subjected to impact loading. Let us now consider the linear mount model.

The models used in this analysis are simple containing single degree of freedom system. The model consists of energy absorbing (EA) elements with masses connected to both ends for full vehicle models and one end is fixed and mass is attached to the other end. Since most engineering materials exhibit viscous and elastic characteristics, the EA can be represented by a Kelvin element, Maxwell element, or combinations of the two elements. Schematically, the masses in the models are connected by springs and/or dampers in a simple arrangement. The equation of motion (EOM) for the model has been solved explicitly with closed-form solutions. The application of such a closed-form formulas used for impact analysis.

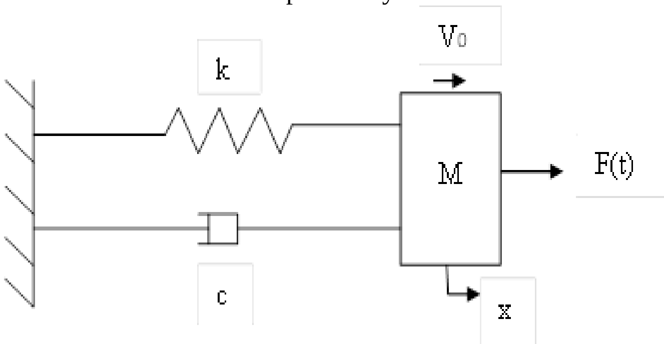

Fig. (2): Spring-damper model

Let,
- k = spring stiffness, (N/m) (unknown).
- c = coefficient of damping, (N-s/m) (unknown).
- M = Mass of moving parts of fixture.
- Mr = Mass of moving barrier.
- V_0 = Initial Velocity.
- X = Displacement of mass 'M', (m).
- $\dot{X}$ = dX/dt = Velocity of mass 'M', (m/s).
- $\ddot{X}$ = d^2X/dt^2 = Acceleration of mass 'M', (m/s^2).
- δ = Maximum Displacement.
- t_m = time at dynamic crush (when dX/dt = 0).
- t_c = estimated time at dynamic crush = $\frac{\delta}{V_0 t_m}$

$$\zeta = \text{damping factor} = \frac{c}{C_c} = \frac{c}{2M\omega_e}$$

$$\omega_e = \text{undamped natural frequency} = \sqrt{\frac{K}{M}}$$

Equation of motion considering condition at equilibrium,

$$\ddot{X} + 2\zeta\omega_e\dot{X} + \omega_e^2 X = \frac{F(t)}{M}$$

The second order linear differential equation is solved using the method of Particular Integrals.

$$X = e^{at}[C_1\cos(\omega_d t) + C_2\sin(\omega_d t)]$$

Differentiating the above equation, we have,

$$\dot{X} = e^{at}[(aC_1 + \omega_d C_2)\cos(\omega_d t) + (aC_2 - \omega_d C_1)\sin(\omega_d t)]$$

Now, applying the initial conditions and finding the constants C_1 and C_2.

I.C. (1) at $t = 0$; $X = 0$;

$$C_1 = 0 \quad (12)$$

I.C. (2) at $t = 0$; $dX/dt = V_0$;

$$C_2 = \frac{V_0}{\omega_d} \quad (13)$$

Substituting these values in the equation of 'X', we have:

$$X = e^{at}\left(\frac{V_0}{\omega_d}\right)\sin(\omega_d t) \quad (14)$$

$$\dot{X} = e^{at}\left[\frac{V_0}{\omega_d}a\sin(\omega_d t) + V_0\cos(\omega_d t)\right] \quad (15)$$

Now, applying the boundary conditions:

B.C. (1) at $t = t_m$; $X = \delta$; Let, $t_m = \frac{\pi}{2\omega_d} \Rightarrow$

$\sin(\omega_d t_m) = 1$

$$\frac{\delta}{V_0 t_m} = e^{-\zeta\omega_e t_m}\left(\frac{1}{(\omega_e t_m)\sqrt{1-\zeta^2}}\right) \quad (16)$$

B.C. (2) at $t = t_m$; $dX/dt = 0$;

Simplifying which we get;

$$\omega_e t_m = \frac{1}{\sqrt{1-\zeta^2}}\tan^{-1}\left(\frac{\sqrt{1-\zeta^2}}{\zeta}\right) \quad (17)$$

Substituting this value in the equation obtained from B.C. (1), we have;

$$\frac{\delta}{V_0 t_m} = \frac{t_c}{t_m} = \frac{1}{tan^{-1}\left(\frac{\sqrt{1-\zeta^2}}{\zeta}\right)} e^{\frac{-\zeta}{\sqrt{1-\zeta^2}}\left[tan^{-1}\left(\frac{\sqrt{1-\zeta^2}}{\zeta}\right)\right]} \quad (18)$$

The above two equations are helpful in plotting graphs, for known values of $\frac{t_c}{t_m}$ we can estimate the value of 'ζ' and the corresponding value of 'ω_e' as follows.

Table 1: Frequency response for variation in zeta

Zeta	t_c/t_m	$\omega_e t_m$	ω_e(rad/sec)
0	0.636619772	1.570796327	24.1661
0.1	0.58655203	1.478037662	22.73904
0.2	0.552149643	1.39767722	21.50273
0.3	0.530404479	1.327237282	20.41904
0.4	0.520091917	1.264877612	19.45966
0.5	0.521671403	1.209199576	18.60307
0.6	0.537950831	1.159119023	17.8326
0.7	0.576525803	1.113781141	17.13509
0.8	0.658908929	1.072501848	16.50003
0.9	0.873693779	1.03472647	15.91887
1	∞	0	0

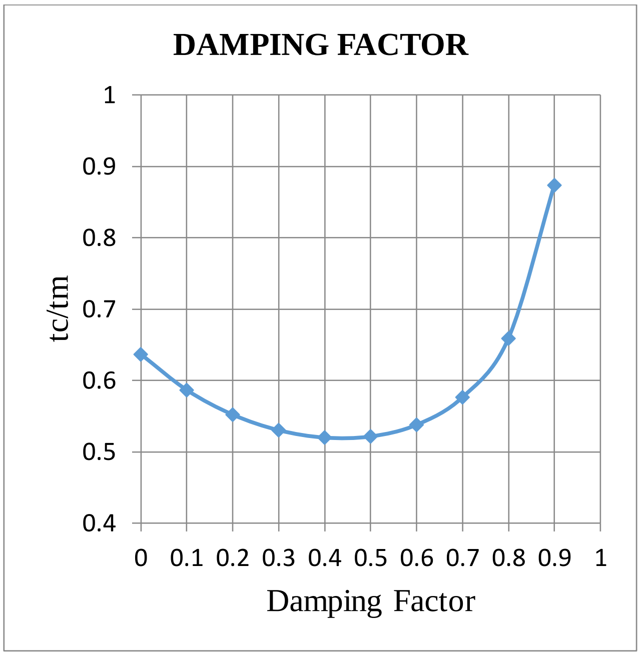

Fig. (3): Graph to compute zeta with known tc/tm

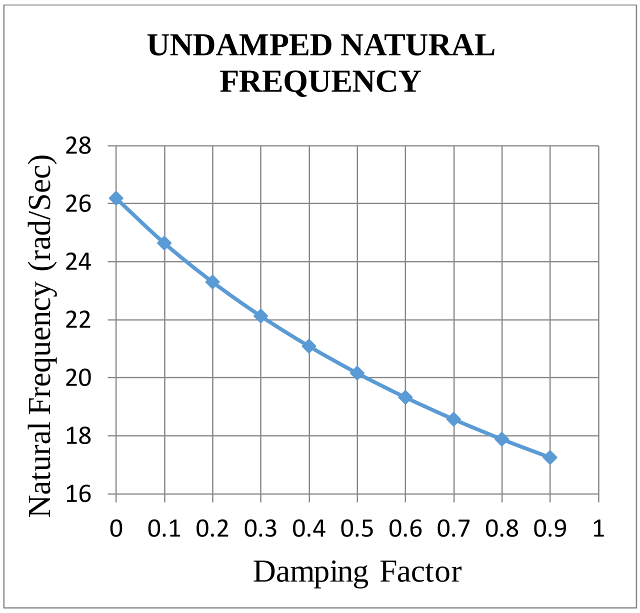

Fig. (4): Graph to find Natural frequency

The above graphs are helpful in finding the spring stiffness and damping coefficient. Now that we know the spring stiffness and coefficient of damping, we can solve the equation of motion with different crash pulses.

A. Excitation With Constant Acceleration

The right hand side of the equation of motion then can be written as

$F(t) = Mr \times A$, $V_0 = V_{12}$.

The governing equation of this case may be written as:

$$\ddot{X} + 2\zeta\omega_e\dot{X} + {\omega_e}^2X = \frac{M_r}{M}A \qquad (19)$$

This equation can be solved by the method of particular integrals with initial conditions as:

When, time t = 0, displacement X = 0, and velocity $V_0 = V_{12}$ = barrier velocity.

The general solution of thus obtained is then expressed as:

Displacement is:

$$X = e^{-\zeta\omega_e t}\left[-\frac{\left(\frac{Mr}{M}A\right)}{{\omega_e}^2}\cos(\omega_d t) + \left(\frac{V_0}{\omega_d} - \frac{\zeta\omega_e\left(\frac{Mr}{M}A\right)}{\omega_d{\omega_e}^2}\right)\sin(\omega_d t)\right] \qquad (20)$$

Velocity is:

$$\dot{X} = e^{-\zeta\omega_e t}\left\{\left[\frac{M_r}{M}\frac{A}{{\omega_e}^2}\left(\frac{\zeta}{\sqrt{1-\zeta^2}}\omega_d + \zeta\omega_e\right) - V_0\right]\cos(\omega_d t) - \right.$$
$V0\zeta1-\zeta2+MrMA\omega e2-\zeta2\omega e1-\zeta2+\omega d\sin\omega dt$
(21)

B. Excitation With Halfsine Pulse

The right hand side of the equation of motion then can be written as

$F(t) = Mr \times A_p\sin(\omega t)$.

The governing equation of this case may be written as:

$$\ddot{X} + 2\zeta\omega_e\dot{X} + {\omega_e}^2X = \frac{M_r}{M}A_P\sin(\omega t) \qquad (22)$$

This equation can be solved by the method of particular integrals with initial conditions as:

When, time t = 0, displacement X = 0, and velocity $V_0 = V_{12}$ = barrier velocity.

The general solution of thus obtained is then expressed as:

Displacement is:

$$X = e^{-\zeta\omega_e t}\left\{\left[\frac{M_r}{M}A_pP_2\right]\cos(\omega_d t) + \left[\frac{V_0}{\omega_d} - \frac{M_r}{M}\frac{A_p}{\omega_d}(-\zeta\omega_eP_2 + \right.\right.$$
$\omega P1\sin\omega dt+MrMApP1\sin\omega t-P2\cos\omega t$ (23)

Velocity is:

$$\dot{X} = e^{-\zeta\omega_e t}\left\{\left[-V_0 + \frac{M_r}{M}A_p(\omega P_1 - 2\zeta\omega_eP_2)\right]\cos(\omega_d t) + \right.$$
$-\zeta V01-\zeta2+MrMAp\omega dP2-MrMAp\omega d\zeta2\omega e2P2-\zeta\omega e\omega P1\sin\omega dt-MrMAp\omega P1\cos\omega t+P2\sin\omega t$
(24)

Where,

$$P_1 = \frac{P_{11}}{P} \qquad (25)$$
$$P_2 = \frac{P_{22}}{P} \qquad (26)$$
$$P_{11} = ({\omega_e}^2 - \omega^2) \qquad (27)$$
$$P_{22} = 2\zeta\omega_e\omega \qquad (28)$$
$$P = {P_{11}}^2 + {P_{22}}^2 = ({\omega_e}^2 - \omega^2)^2 + (2\zeta\omega_e\omega)^2 \qquad (29)$$

C. Excitation With Haversine Pulse

The right hand side of the equation of motion then can be written as

$$F(t) = Mr \times A(t) = Mr \times \frac{A_p}{2}[1 - cos(2\omega t)]$$

The governing equation of this case may be written as:

$$\ddot{X} + 2\zeta\omega_e\dot{X} + {\omega_e}^2X = \frac{M_r}{M}\times\frac{A_p}{2}[1-\cos(2\omega t)] \qquad (30)$$

This equation can be solved by the method of particular integrals with initial conditions as:

When, time t = 0, displacement X = 0, and velocity $V_0 = V_{12}$ = barrier velocity.

The general solution of thus obtained is then expressed as:

Displacement is:

$$X = e^{-\zeta\omega_e t}\left\{\left[\frac{M_r}{M}\frac{A_p}{2}\left(Q_1 - \frac{1}{{\omega_e}^2}\right)\right]\cos(\omega_d t) + \left[\frac{V_0}{\omega_d} - \right.\right.$$
$MrMAp2-\zeta1-\zeta2Q1-1\omega e2+2\omega\omega dQ2\sin\omega dt+MrMAp2 1\omega e2-Q1\cos2\omega t+Q2\sin2\omega t$ (31)

Velocity is:

$$\dot{X} = e^{-\zeta\omega_e t}\left\{\left[\frac{M_r}{M}\frac{A_p}{2}(-2\zeta\omega_e)\left(Q_1 - \frac{1}{{\omega_e}^2}\right) + 2\omega Q_2 - \right.\right.$$
$V0\cos\omega dt+-\zeta V01-\zeta2+MrMAp2\omega d-\zeta2\omega e1-\zeta2Q1-1\omega e2-2\zeta\omega1-\zeta2Q2\sin\omega dt-MrMAp22\omega Q1\sin2\omega t+Q2\cos2\omega t$ (32)

Where,

$$Q_1 = \frac{Q_{11}}{Q} \qquad (33)$$
$$Q_2 = \frac{Q_{22}}{Q} \qquad (34)$$
$$Q_{11} = ({\omega_e}^2 - 4\omega^2) \qquad (35)$$
$$Q_{22} = 4\zeta\omega_e\omega \qquad (36)$$
$$Q = {Q_{11}}^2 + {Q_{22}}^2 = ({\omega_e}^2 - 4\omega^2)^2 + (4\zeta\omega_e\omega)^2 \qquad (37)$$

The above equations can be solved by substituting the values of all the parameters to obtain the displacement and velocity curves.

IV. ESULT AND CONCLUSION

The results obtained from the analytical computations are discussed here. To reduce the complexity and for easy Understanding we consider only the most relevant results. The following table details the normalized results of displacement and velocity variation with variation in time.

Table 2: Analytical results

	No Excitation $V_0 = V_{12}$		**Halfsine Excitation $V_0 = V_{12}$**		**Fourier Equivalent Wave $V_0 = V_{12}$**	
Time (Sec.)	Displacement $[X/X_p]$	Velocity $[V/V_p]$	Displacement $[X/X_p]$	Velocity $[V/V_p]$	Displacement $[X/X_p]$	Velocity $[V/V_p]$
0	0	1	0	1	0	1
0.005	0.149450549	0.953732	0.121170865	0.744756	0.118900344	0.685361
0.010	0.290842491	0.895752	0.220558203	0.684761	0.200687285	0.442405
0.015	0.421978022	0.827286	0.320626276	0.733693	0.239862543	0.152268

0.020	0.542857143	0.749796	0.430224643	0.809695	0.284536082	0.62371
0.025	0.650549451	0.664675	0.548672566	0.8509	0.428178694	1.283993
0.030	0.745054945	0.573482	0.667801225	0.82491	0.616494845	1.264747
0.035	0.824908425	0.47773	0.778080327	0.72803	0.775945017	0.938853
0.040	0.89010989	0.378978	0.870660313	0.57617	0.886597938	0.62947
0.045	0.940659341	0.278738	0.939414568	0.39556	0.960824742	0.40012
0.050	0.975091575	0.178498	0.982300885	0.211207	0.997250859	0.113967
0.055	0.994871795	0.07965	1	0.041229	1	-0.05896
0.060	1	-0.01644	0.994554118	-0.1053	0.980756014	-0.20374
0.065	0.98974359	-0.10852	0.971409122	-0.22517	0.945017182	-0.28378
0.070	0.967032967	-0.19546	0.932607216	-0.31848	0.899656357	-0.35306
0.075	0.93040293	-0.27619	0.882232811	-0.38733	0.84604811	-0.3915
0.080	0.882783883	-0.34982	0.823689585	-0.43432	0.788316151	-0.41279
0.085	0.824175824	-0.4156	0.759700477	-0.46213	0.729896907	-0.40343
0.090	0.756043956	-0.47281	0.693669163	-0.47379	0.671477663	-0.41322
0.095	0.67985348	-0.52112	0.626276378	-0.47214	0.611683849	-0.41793
0.100	0.597069597	-0.56009	0.560245065	-0.45978	0.551890034	-0.40821
0.105	0.509157509	-0.58958	0.496255956	-0.43974	0.498969072	-0.33141
0.110	0.417582418	-0.6096	0.435670524	-0.4131	0.448797251	-0.40194
0.115	0.323809524	-0.62013	0.379169503	-0.38289	0.38419244	-0.46676
0.120	0.229304029	-0.6215	0.327433628	-0.34999	0.32371134	-0.3737
0.125	0.134798535	-0.61397	0.2804629	-0.31615	0.274914089	-0.32205
0.130	0.041758242	-0.59806	0.237576583	-0.28219	0.230927835	-0.28479
0.135	-0.047619048	-0.57427	0.200136147	-0.24922	0.193127148	-0.24828
0.140	-0.132600733	-0.54329	0.166780123	-0.21776	0.160137457	-0.20869
0.145	-0.213186813	-0.50581	0.138189244	-0.18872	0.132646048	-0.17723
0.150	-0.287179487	-0.46261	0.113002042	-0.16144	0.109278351	-0.14984

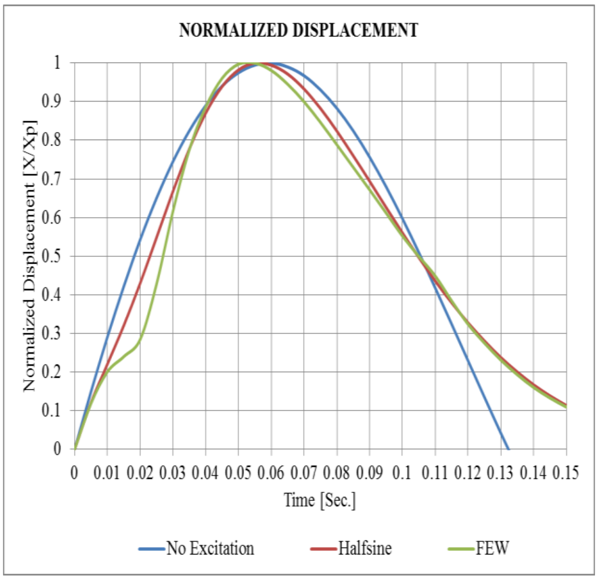

Fig. (5): Displacement curve

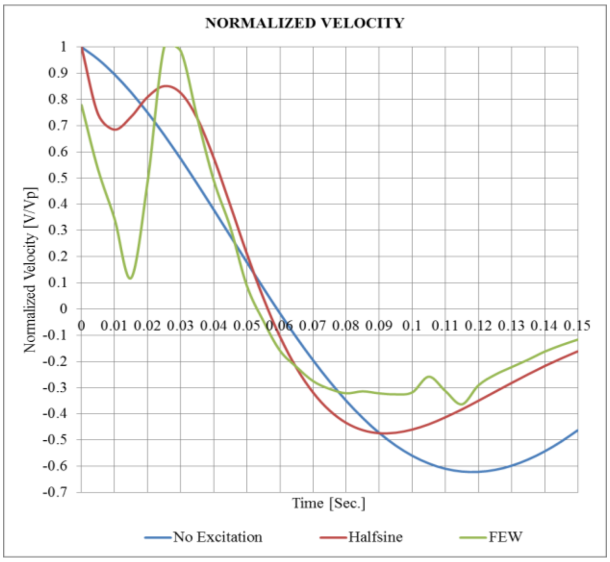

Fig. (6): Velocity curve

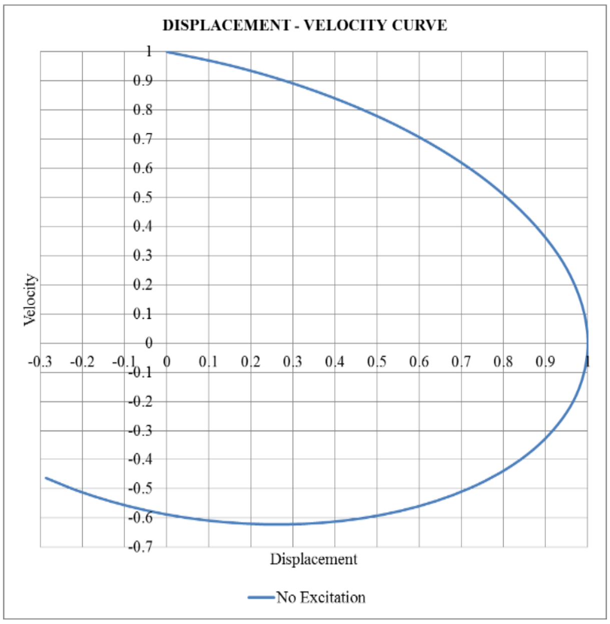

Fig. (7): Displacement-velocity curve for no excitation

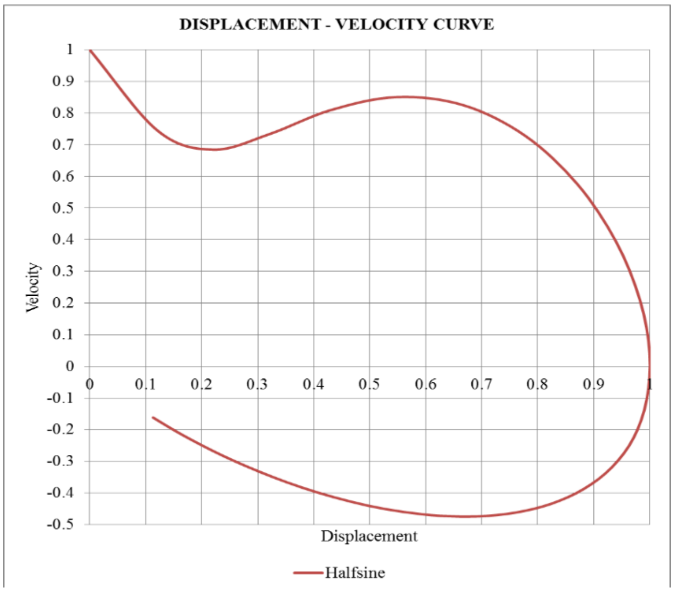

Fig. (8): Displacement-velocity curve with halfsine excitation

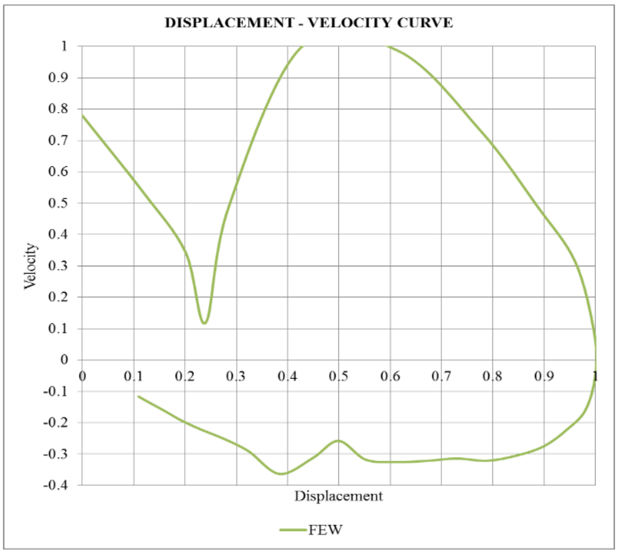

Fig. (9): Displacement-velocity curve with Fourier Equivalent Wave excitation

The curves shown above are compared i.e. from the comparison of curves obtained from analytical computations for no excitation, half-sine excitation and Fourier equivalent wave, we can draw some conclusions. These are:

1) The displacement predicted by half-sine excitation is equivalent to Fourier equivalent wave (FEW represents the actual crash pulse)
2) The breakage of the mount may occur after it has displaced as predicted by half-sine excitation.
3) The boundary conditions are well satisfied by the model, and thus the mathematical model can be used in future as standard tool to predict the deformations and velocity of the mounts at the initial stage.
4) The velocity curve obtained by the half-sine excitation when compared with the one obtained by FEW is comparable and this proves the success of adopted methodology, because Fourier equivalent wave in this case is representing the actual crash pulse.
5) The maximum displacement always occurs when the velocity is zero, which is the absolute theoretical

prediction, and CAE or experimental results would be in well accordance to this theory.

6) The process of finding the stiffness and damping coefficients now can be permanently modified to the current method, which give more appropriate results and this satisfies the purpose of the research.

The final understanding from the research is that, the dynamic characteristics of mounts can be predicted by this methodology, i.e. by using a half-sine wave to predict the stiffness and damping coefficient.

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