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Performance Analysis of Engine Mount System for Vibration Reduction using 16 DOF Mathematical Model

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ABSTRACT

Vehicle engine mounting system, generally, consists of an engine and several mounts connected to the vehicle structure. The major function of engine mounts is to isolate the unbalanced engine disturbance force from the vehicle structure. The behavior of the mounting system not only depends on the performance of individual mounts but also on the complete system. The design of an engine mount system involves the selection of stiffness coefficients, location and orientation of the individual mounts. Traditionally, the mounts designed based on experience and extensive analysis procedures. Defining a near optimum initial mounting configuration is not an easy task due to the complex nature of the engine inertia properties and the packaging constraints on the mount locations as imposed by manufacturability considerations. To decide mount stiffness and mount location in initial design phase we developed 16 Degree of freedom mathematical model of vehicle considering engine, chassis, and wheels as mass and engine mounts, suspensions and tyres as stiffness. These 16 equations of motions were solved using MATLAB to calculate Natural frequencies and Kinetic energy fraction, also Harmonic analysis of Driver's seat rail (DSR) is done by mode super position method in MATLAB and validated same with experimental results on actual system. Finally effect of engine mount stiffness on DSR vibration is analyzed. Hence MATLAB based tool is developed to predict DSR vibrations and engine natural frequencies in initial design phase of vehicle.

Keywords— **DSR, Degree of freedom (DOF), Engine mount, Kinetic energy fraction, Natural frequency, vibration.**

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I. INTRODUCTION

Engine of any vehicle is a main source of vibration, in an internal combustion engine, there are two basic dynamic disturbances: the firing pulse due to the explosion of the fuel in the cylinder; the inertia force and torque caused by the rotating and reciprocating parts like piston, connecting rod and crank. These two disturbances cause large vibration in an internal combustion engine. These vibrations get transferred to structure of vehicle and passenger in vehicle feels uncomfortable hence there is need to isolate these vibrations from vehicle structure. To isolate vibration produced due to engine of a vehicle from vehicle structure engine mounting system is used.

The primary function of the engine mounting system is to support the weight of the engine. The plan view location of the engine center of gravity should not only be contained within the support base, but the engine weight should also be well distributed among the load carrying mounts. This will ensure that the engine can be freely maintained in its specific design position. Other than supporting the engine weight, the major function of engine mounts is to isolate the unbalanced engine disturbance force from the vehicle structure. For an internal combustion engine, there exist two basic dynamic disturbances: the firing pulse due to the explosion of the fuel in the cylinder; the inertia force and torque caused by the rotating and reciprocating parts. The firing pulses will cause a torque to act on the engine block about an axis parallel to the crank. The directions of the inertia forces are both parallel to the piston axis and

perpendicular to the crank and piston axes. The inertia torque acts about an axis, which is parallel to the crankshaft.

An engine mount system generally consists of three to four engine mounts. The behavior of the mounting system not only depends on the performance of individual mounts but also on the complete system. The design of an engine mount system involves the selection of stiffness coefficients, location and orientations of the individual mounts. Traditionally, the mounts designed based on experience and extensive analysis procedures. Defining a near optimum initial mounting configuration is not an easy task due to the complex nature of the engine inertia properties and the packaging constraints on the mount locations as imposed by manufacturability considerations[1].

Yunhe Yu et al. [1] studied all types of engine mount and reported that though active engine mounts give good amount of vibration isolation, due to cost and weight they are not popular as elastomeric engine mounts. Kevin A. et al. [2] formulated the problem of vehicle modeling and ride quality optimization design within linear quadratic gausian control theory using kalman filter to identify vehicle parameters. John Bretl [3] determined response of driver seat hip using computerized optimization algoritham. D.S. sachdeva et al [4] compared three strategies all decoupled modes, coupled bounce-pitch modes and coupled bounce-roll modes to optimized to reduce vehicle response due to engine and road inputs.

The current industrial strategies use model approach to analyze the harmonic response of engine or resilient supports attached to ground, and 6 DOF model is used in modal analysis. Here we present 16 DOF model approach to find Natural frequency Kinetic energy fraction and to analyze harmonic response of driver's seat rail.

II. MATHEMATICAL MODELING

For analytical analysis of any system first a mathematical model is required. Here also mathematical model for a vehicle system has been developed considering undammed lumped mass system. A 16 degree-of -freedom mathematical model is created of vehicle using engine, chassis and 4 wheels as masses and engine mounts, suspensions and tyre as spring.

A. *Mathematical Model*

Fig. 1 shows simple spring mass system 16 degree-of-freedom mathematical model which consists of 6 masses i.e. 4 wheels, chassis and engine. In this model we have considered 6 engine mounts, 4 suspensions and 4 tyres as stiffness.

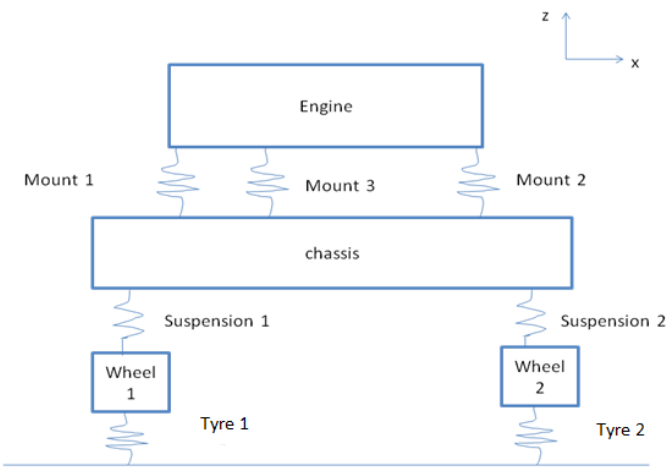

Fig. 1 16 DOF Mathematical model of vehicle.

In this mathematical model 16 DOF consider are as follows:

TABLE I
16 DEGREES OF FREEDOM

No.	Mass	Degree of freedom considered	Directions
1	Wheel (FL)	1	Translation in **Z** direction only
2	Wheel (RL)	1	Translation in **Z** direction only
3	Wheel (RR)	1	Translation in **Z** direction only
4	Wheel (FR)	1	Translation in **Z** direction only
5	Chassis	6	Translation in **X,Y** and **Z** directions Rotational in **X,Y** and **Z** directions
6	Engine	6	Translation in **X,Y** and **Z** directions Rotational in **X,Y** and **Z** directions

B. *Equation of Motions*

Equations of motion is derived using Newton's second law of motion which will give 'n' second order differential equations of motion for 'n' degree of freedom system. To derive equations of motion we will consider system which only considers 3 masses i.e. wheel 1, wheel 2 and chassis. Using FBD as shown in fig.2 and fig.3 where Z_1, Z_2, Z_3 and are displacement of wheel 1, wheel 2, wheel 3 and chassis in Z direction and and are rotational displacement of chassis about X and Y axis respectively we will derive equation of motion for mass 1(wheel 1) and mass 2 (wheel 2) by Newton's second law of motion as follow [7].

Hence equation of motion for mass 1 (wheel 1) and mass 2 (wheel 2) are as follow:

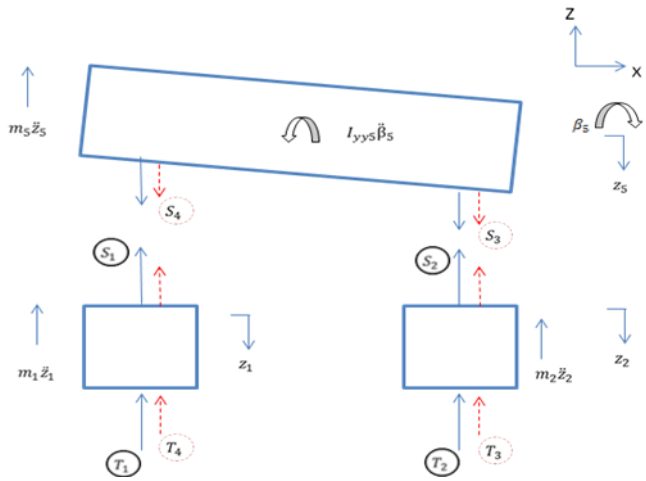

Fig.2 FBD considering wheel and chassis in Z-X plane.

Front left wheel (Displacement in Z direction only)

- $$m_1\ddot{Z}_1 + \boldsymbol{Z_1}(Kz_{S1} + Kz_{T1}) - \boldsymbol{Z_5}(Kz_{S1}) - \boldsymbol{\beta_5}(S_{1xc} * Kz_{S1}) + \boldsymbol{\alpha_5}(S_{1yc} * Kz_{S1}) = 0$$

(1)

Rear left wheel (Displacement in Z direction only)

- $$m_2\ddot{Z}_2 + \boldsymbol{Z_2}(Kz_{S2} + Kz_{T2}) - \boldsymbol{Z_5}(Kz_{S2}) - \boldsymbol{\beta_5}(S_{2xc} * Kz_{S2}) + \boldsymbol{\alpha_5}(S_{2yc} * Kz_{S2}) = 0$$

(2)

Similarly for mass 3 (wheel 3) and (wheel 4) we will derive equations.

FBD of chassis and engine is considered to derive all remaining equation of motions for mass 5 (chassis) and mass 6 (engine), where Z_5 *and* Z_6 are displacement of chassis and engine in Z direction and α_5 and β_5 are rotational displacement of chassis about X and Y axis respectively and similarly α_6 and β_6 are rotational displacement of engine about X and Y axis respectively.

Fig.3 shows FBD of chassis and engine in Z-X plane

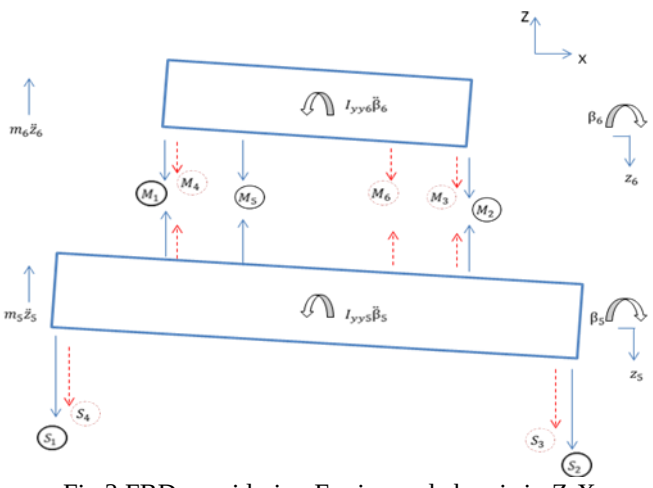

Fig.3 FBD considering Engine and chassis in Z-X plane.

Translation of Engine in Z direction.

- $$m_6\ddot{Z}_6 - \boldsymbol{Z_5}(\textstyle\sum_{i=1}^{6} Kz_{Mi} * Cos\theta i) + \boldsymbol{Z_6}(\textstyle\sum_{i=1}^{6} Kz_{Mi} * Cos\theta i) - \boldsymbol{\beta_5}(\textstyle\sum_{i=1}^{6} Kz_{Mi} * Cos\theta i * M_{ixp}) + \boldsymbol{\beta_6}(\textstyle\sum_{i=1}^{6} Kz_{Mi} * Cos\theta i * M_{ixp}) + \boldsymbol{\alpha_5}(\textstyle\sum_{i=1}^{6} Kz_{Mi} * Cos\theta i * M_{iyp}) - \boldsymbol{\alpha_6}(\textstyle\sum_{i=1}^{6} Kz_{Mi} * Cos\theta i * M_{iyp}) = 0$$

(3)

Similarly all 12 equations of motion 6 for Engine and 6 for chassis are derived.

III. Solution using MATLAB program

Derived 16 equations of motion solved using MATLAB script. A MATLAB program is generated to calculate Natural frequency, % Kinetic energy fraction and to determine Harmonic response of Driver's seat rail.

A. *Natural Frequency:*

16 X 16 Mass Matrix [M] and 16 X 16 stiffness Matrix [K] is generated using MATLAB program and Dynamic Matrix [D] is calculated as follow

$$[D] = [M]^{-1}[K] \quad (4)$$

Eigen values of matrix [D] will give natural frequency and Eigen vector of matrix [D] will give mode shapes.

B. *Kinetic Energy fraction:*

The modal kinetic energy fraction is calculated to identify the significant local modes of subsystem for improving dynamic response prediction. KEF is measurement of amount of system kinetic energy contained within each subsystem for any given mode [8]. KEF is calculated as follow

$$[KEF] = \frac{[\varnothing_c]^T[M_c]\cdot[\varnothing_c]}{[\varnothing_s]^T[M_s]\cdot[\varnothing_s]} \quad (5)$$

Where,

$[\varnothing_s]$ is system mode shape matrix

$[\varnothing_c]$ component partition of the system mode shape matrix

$[M_c]$ is component mass matrix

$[M_s]$ is system mass matrix

(.) is element by element multiplication operator and (–) is element by element division operator.

C. *Harmonic Analysis of Driver's seat rail (DSR):*

Harmonic analysis of Driver's seat rail is done using mode superposition method and acceleration of Driver's seat rail in z direction at engine idling is computed using MATLAB program.

IV. Results and Discussion

MATLAB program is solved using all input data for a 3 cylinder engine small car to calculate results.

A. *Natural Frequency and KEF*

Following natural frequencies and KEF is obtained using MATLAB program for 3 cylinder engine small car.

TABLE III
16 DOF NATURAL FREQUENCIES AND %KEF

Engine Mode	**NF (Hz)**	**% KEF**	**Engine Mode**	**NF (Hz)**	**% KEF**
Engine F/A	7.1	66	Chassis vertical	1.5	71
Engine lateral	6.16	46	Chassis Roll	2.4	66
Engine vertical	8.47	52	Chassis pitch	1.8	57
Engine Roll	10.4	52	Chassis yaw	2.1	57
Engine Pitch	5.9	46	Wheel (FL)	12.3	57
Engine yaw	12.9	40	Wheel (RL)	13.1	68
Chassis F/a	1.6	65	Wheel (RR)	13.1	69
Chassis lateral	1.1	39	Wheel (FR)	12.3	57

All Natural frequencies are below 20 Hz and exactly equal to Natural frequencies calculated through CAE simulation.

B. *Harmonic Analysis of Driver's seat rail*

Driver's seat rail acceleration of 3 cylinder small car is as shown in fig. 4 which is calculated solving 16 equations of motion by mode superposition method in MATLAB program.

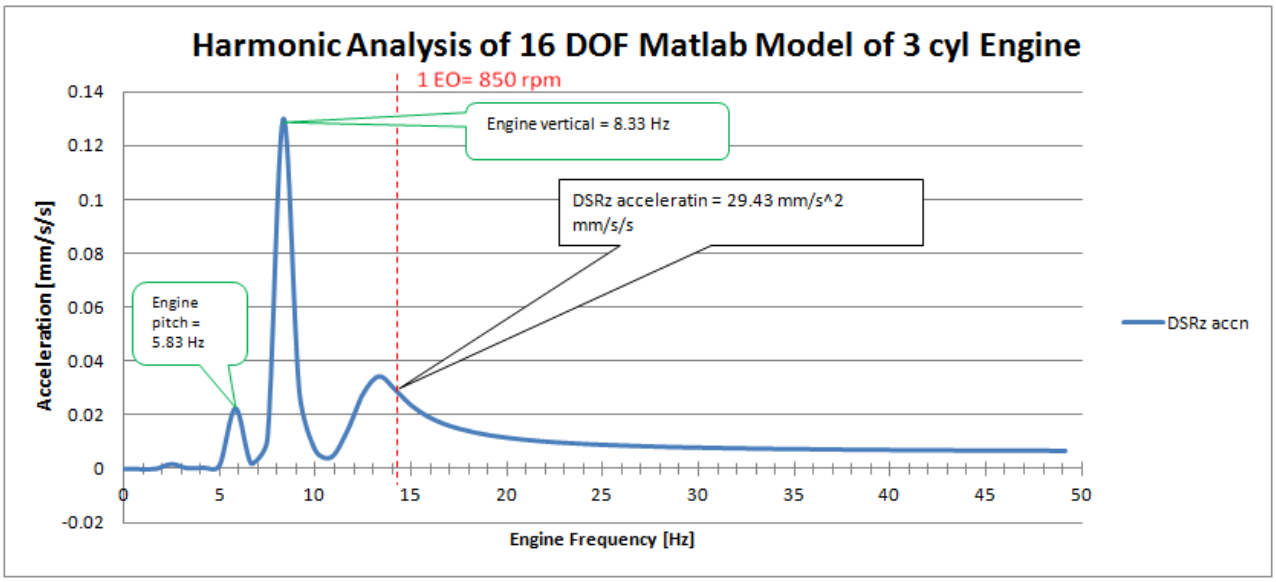

Fig. 4 Harmonic analysis of 16 DOF MATLAB model of 3 cyl engine.

In fig.4 we can see acceleration at idling i.e. at 850 Engine RPM is 29.43 mm/s^2 which is near to 35 mm/s^2 which is obtain experimentally using FFT analyzer and data acquisition system.

[c] Parameter optimization of DSRz vibration:

In this section we will see effect of different parameters on DSRz vibration. In fig. 5 we can see effect of engine mount stiffness, suspension stiffness and tyre stiffness on DSRz vibration levels. Here we once increased and once decreased engine mount stiffness, suspension stiffness and tyre stiffness with respect to baseline configuration where we got 29.4 mm/s^2 acceleration.

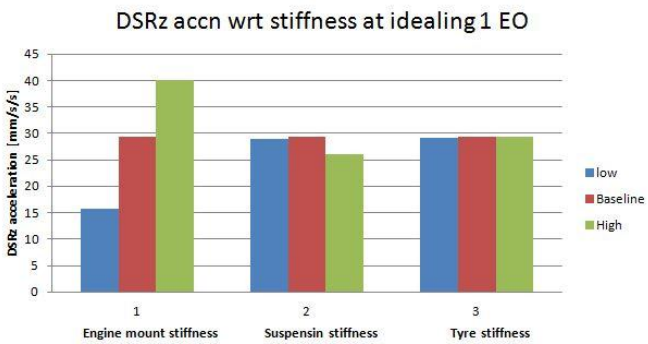

Fig.5 Effect of stiffness on DSRz vibrations

We can see in fig. 5 suspension stiffness and tyre stiffness is not contributing much in DSRz vibration levels as we can see vibration level is more or less constant but DSRz vibrations are more sensitive to Engine mount stiffness as vibration level is changing with respect to mount stiffness.

V. CONCLUSION

- A powerful MATLAB tool is developed to determine natural frequencies, KEF of Engine.
- Designer can predict vibrations of Driver's seat rail in initial design phase to predict mount stiffness and locations.
- Hence further multidisciplinary optimization required to predict correct mount stiffness and mount locations in initial design stage.

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REFERENCES

[1] Kevin A. Wise, Robert E. Reid, Modeling and identification of a light Truck engine mounting system for ride quality optimization, SAE paper 841142, 1984

[2] John Bretl, Optimization of engine mounting systems to minimize vehicle vibration, SAE paper 931322, 1993.

[3] Yunhe Yu, Nagi G. Naganathan, A literature review of automotive vehicle engine mounting systems, Rao Dukkipati, mechanisum and machine theory 36 (2001) pp.123-142

[4] D.S.Sachdeva,R.Hadi, Effect of engine mounting strategy on vehicle NVH, SAE paper 2003-01-1467.Mohammad Sirafi, Mohamad Qatu, Accurate Modeling for the Powertrain and Subframe Modes, SAE paper 2003-01-1469.

[5] Magnus Olsson, Mikael Törmänen , Sylvain Sauvage and Catharina Hansen, Systematic multi-disciplinary optimization of engine mount, SAE paper 2011-01-1674.

[6] Cyirl M. Harris, Harri's shock and vibration handbook, 5th edition, McGraow Hill, 2002, pp. 2.1-3.56

[7] Y.T.Chung, L.L. Kahre, A general procedure for finite element model check and model identification

[8] Kwon-Hee suh, A study on balancing of the 3 cylinder engine with balance shaft, SAE 2000-010601

[9] Jianghua Fu1, Jian Pang ; Chengtai Hu1; Xiaomin Xu1, Renwei Deng Xiaohong Kuang, Study on startup transient vibration of a vehicle with 3-cylinder engine, inter.noise 2014 melbourn, austrelia

