VOL. 11, NO. 10, MAY 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

OPTIMIZATION OF VIBRATION REDUCTION BY DE-COUPLING METHOD IN POWER TRAIN OF TRUCKS

Kumar Yogeesh. D[1], Chandrasekaran. M[2], Eriki AnandaK[3], and Jayakiran Reddy[4]
[1]Department of Mechanical & Automotive Engineering, FEAM, Nilai University, Nilai, Malaysia
[2]Director and Department of Mechanical Engineering, Vels University, Pallavaram, Chennai, India
[3]Department of Mechanical & Industrial Engineering, Caledonian College of Engineering, Muscat, Oman
[4]Research Scholar, Department of Mechanical Engineering, JNT University Anantapur, India
E-Mail:dkyogeesh@gmail.com

ABSTRACT

In this paper, optimize the vibrational analysis of power train by decoupling method in the field of automotive. Power train (Engine and gear box) is typically connected to the chassis by three or four rubber mounts. Ideal engine mounts should isolate engine vibrations caused by engine disturbances in the engine speed range and prevent large engine movement from shock excitations from vehicle accelerations, braking and road undulations. The design of power train/mount system involves selection of stiffness coefficients, location and orientation of the individual rubber mounts. Rubber mount location and orientation is decided by designer based on packaging constraints. The goal in this project is to find the engine mount stiffness values corresponding to natural frequencies and mode shapes of the power train system/mount system. CAE find stiffness values and coordinates corresponding to natural frequencies by optimization.

Keywords: decoupling, power train of trucks, vibrational analysis.

INTRODUCTION

In quality control, the statistical process control (SPC) chart patterns should represent the relationship between process variation and its specific causes. In practice, the shift patterns indicate there are changes in calculation of engine mounts co-ordinates and stiffness for ‘N’ series Power train (Engine and Gear box) for Heavy Commercial Truck (Figure-1). The following assumptions are made:

1. Engine is considered as rigid body. It is reasonable to model engine as rigid body because its natural frequency is much higher than the mounting system.
2. The rubber mounts have nonlinear stiffness as well as viscosity and structural damping, but our purpose here; they will be modeled as linear elastic elements.
3. The chassis is modeled as a rigid body. The reasons to exclude the vibrations of the foundation are:
   - The displacement on the chassis side of the engine mount is small compared to the engine side so that the engine and mounting system can be treated outside the context of full vehicle.
   - It is difficult due to its complexity to fully optimize the mounting system in the context of full vehicle.
4. Rotational stiffness of rubber mounts is neglected. The reason being contribution of rotational stiffness to the global stiffness matrix is negligible compared to contribution of translational stiffness.

The pulsating torque generated by the multi-cylinder engines is one of the major sources of vibration. This becomes even more critical in future vehicles which will employ high power density power trains with lighter and more compact body frames (Eriki, 2015). Mathematical models of Most dynamic design principles attempt to place the natural frequencies of the system below and above the excitation frequency range (Van T.k, 2012). However, if a system has several resonances within a narrow band, it becomes a rather difficult task to achieve. Therefore, a physically decoupled system has a better chance of producing fewer resonances over the operating range. For example, if a system is completely decoupled in physical modes, excitation along one physical co-ordinate should excite only one mode, as shown in Figure-2 (a), (b) and (c).

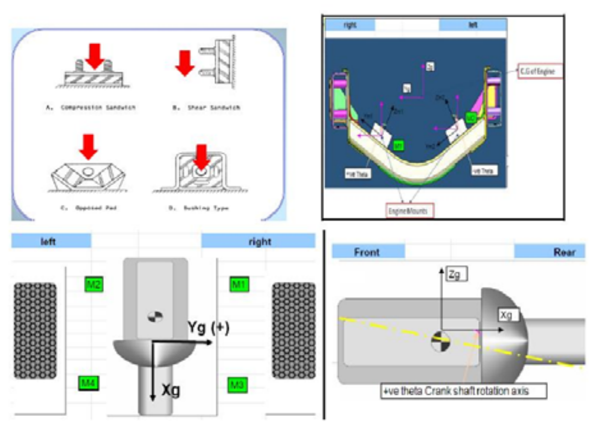

Figure-1. Engine mount structure & properties of and co-ordinate system (front view).

The true TRA mode decoupling strategy has also been sought by designers and researchers over the past two decades. Requirements for the existence of a decoupled (Jomehzadeh E, 2011) TRA mode are suggested by Geck and Patton and these have been implemented in a numerical optimization scheme. But they could not obtain a more complete decoupling of the TRA mode. The TRA decoupling mechanisms therefore remain poorly

ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

understood and inadequately analyzed (Fai Ma, 2010). Figure-3 shows the elastics axis Decoupling scheme and relevant coordinate systems.

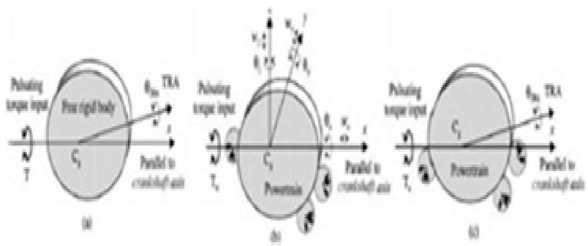

Figure-2. (a).TRA-free rigid body (b) Coupled power train mounts (c) TRA mode decoupled.

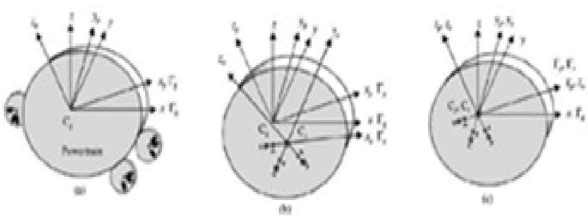

Figure-3. Elastics axis decoupling scheme and relevant coordinate systems.

METHODOLOGY

Dynamic analysis

The analyst must evaluate the finite element model in terms of the type of dynamic loading to be applied to the structure. This dynamic load is known as the dynamic environment. The dynamic environment governs the solution approach (i.e., normal modes, transient response, frequency response, etc.). This environment also indicates the dominant behavior that must be included in the analysis (i.e., contact, large displacements, etc.). Proper assessment of the dynamic environment leads to the creation of a more refined finite element model and more meaningful results.

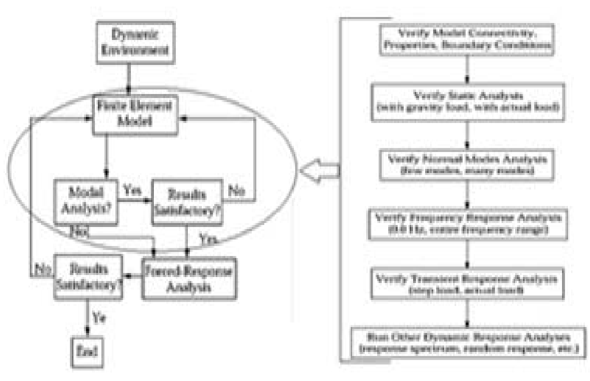

Figure-4. Flowchart of dynamic analysis strategy.

Figure.4 shows the flowchart of dynamic analysis strategy. The primary steps in performing a dynamic analysis are summarized as follows:

- Define the dynamic environment (loading).
- Formulate the proper finite element model.
- Select and apply the appropriate analysis approach to determine the behavior of the structure.
- Evaluate the results

Engine mount reaction force

When conducting an MSC. Nastran dynamic analysis, the formulation of the model depends upon the desired results and the type of dynamic loading. Figure 5 shows the geometric vehicle mount coordinate system.

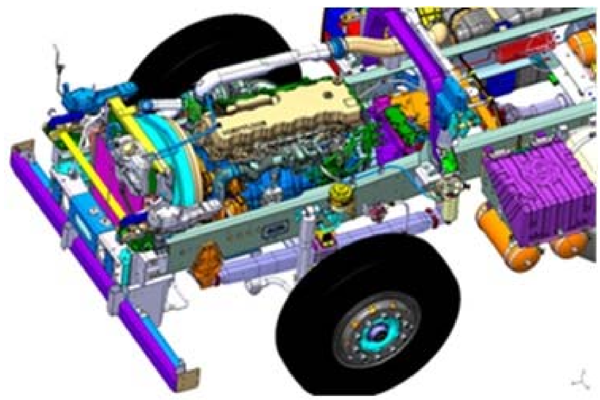

Figure-5. Geometric vehicle mount coordinate system.

Mass input

Mass input is one of the major entries in a dynamic analysis. Mass can be represented in a number of ways in MSC.Nastran. The mass matrix is automatically computed when mass density or nonstructural mass is specified for any of the standard finite elements (CBAR, CQUAD4, etc.) in MSC.Nastran, when concentrated mass elements are entered, and/or when full or partial mass matrices are entered.

Lumped and coupled mass

Mass is formulated as either lumped mass or coupled mass. Lumped mass matrices contain uncoupled, translational components of mass. Coupled mass matrices contain translational components of mass with coupling between the components. The CBAR, CBEAM, and CBEND elements contain rotational masses in their coupled formulations, although torsional inertias are not considered for the CBAR element. Coupled mass can be more accurate than lumped mass. However, lumped mass is more efficient and is preferred for its computational speed in dynamic analysis. The mass matrix formulation is a user-selectable option in MSC.Nastran. The default mass formulation is lumped mass for most MSC.

VOL. 11, NO. 10, MAY 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

Nastran finite elements

The coupled mass matrix formulation is selected using PARAM,COUPMASS,1 in the Bulk Data. Table-1 shows the input values of engine mounted on vehicle for the mass options available for each element type. Dynamic analysis is more complicated than static analysis because of more input (mass, damping, and time- and frequency-varying loads) and more output (time- and frequency varying results). Table-2 shows the stiffness value of mounts with different load cases, while Table-3 shows the element of mass type, and Figure-6 shows the geometric mount after load system.

Table1. Input values of engine mounted on vehicle.

Geometric Data in Vehicle Coordinate System [mm]				Mass [kg]
Coordinates of mounts	X	Y	Z	
Engine CG [E]	0	0	0	553
Gear box CG [G]	0	0	0	0
Front Right Mount [M1]	-456	202	-133	NA
Front Left Mount [M2]	-456	-214	-133	NA
Rear Right Mount [M3]	153	238.5	31	NA
Rear Left Mount [M4]	153	-250.5	31	NA
Power train CG (Combined E&G)	0	0	0	553

	IXX	IYY	IZZ	IXY	IXZ	IYZ
MI	24.3	88.7	71.1	7.5	6.9	6.4

Table2. Stiffness values of mounts with different load cases.

Stiffness values [N/mm]				Rotation about y-axis [deg]	x-axis
Location	X	Y	Z		
Front Right Mount [M1]	35	35	500	10	30
Front Left Mount [M2]	35	35	250	10	-30
Rear Right Mount [M3]	670	300	580	10	0
Rear Left Mount [M4]	670	300	580	10	0
Dynamic Index [DI]	1				

Crank shaft Rotation axis w.r.t Y-axis [deg]		10
Refer side view		

Load Case	g Loads			Torque *FGR [N-mm]
	Vertical	Braking	Cornering	
Load Case 1	1	0	0	6240000
Load Case 2	3	0	0	0
Load Case 3	1	0.6	0	0
Load Case 4	1	0	0.3	0
Load Case 5	1	-0.6	0.3	0

Results from static analysis are usually easier to interpret, and there are numerous textbook solutions for static analysis that make it relatively easy to verify certain static analyses. Nevertheless, the guidelines in this paper shall help researcher to perform dynamic analysis in a manner that will give the same level of confidence in the dynamic results that the researchers would have with static results. The guideline are:

- Create the initial model only; do not apply any loads. Verify the model's connectivity, element and material properties, and boundary conditions. Use a graphical tool such as MSC/PATRAN to assist in this. Make sure that mass is specified for this model.

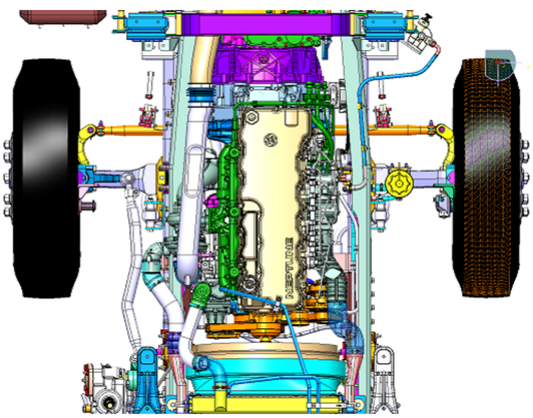

Figure-6. Geometric vehicle mount after load system

Table3. Element mass type.

Element Type	Lumped Mass	Coupled Mass*
CBAR	x	x
CBEAM	x	x
CBEND		x
CONEAX	x	
CONM1	x	x
CONROD	x	x
CRAC2D	x	x
CRAC3D	x	x
CHEXA	x	x
CMASS1	x	
CPENTA	x	x
CQUAD4	x	x
CQUAD8	x	x
CQUADR	x	x
CROD	x	x
CSHEAR	x	
CTETRA	x	x

- Perform a static analysis (SOL 101) first in order to verify proper load paths and overall model integrity. (Note that the researcher have to constrain the structure for static analysis even if you were not planning to do so for dynamic analysis.) For a three-dimensional model, the researcher should run three load cases, each with a 1g gravity load applied in a different direction. Compute displacements and SPC forces, and verify the results. Check for unusually large grid point displacements and unreasonable SPC forces. The use of a graphical postprocessor can aid the researcher at this step. Next, apply static loads that have the same spatial distribution that your

 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

subsequent dynamic loads have. Verify the results for reasonableness. Do not go to dynamic analysis until the researcher are satisfied with the results from the static analysis. It is recommended at this stage that the model contain PARAM,GRDPNT,n (where n is a reference grid point or 0, the origin of the basic coordinate system). Verify the results from the grid point weight generator in order to ensure that the model's rigid body mass and inertia look reasonable. This step, in conjunction with the static analysis results, helps to ensure that the proper mass units are specified.

- Perform an Eigen value analysis (SOL 103) next. Compute only a few modes first, verify their frequencies, and view their mode shapes for reasonableness. If the researcher graphical postprocessor can animate the mode shapes, do so because that helps the researcher to visualize them. Things to check at this step are local mode shapes, in which one or a few grid points are moving a very large amount relative to the rest of the model (this can indicate poor stiffness modeling in that region), and unwanted rigid-body modes (which can arise due to improper specification of the boundary conditions or a mechanism).Once the researcher are satisfied with these results, perform the full eigen value analysis (for as many modes as you need).
- Frequency-dependent loads, perform frequency response analysis (SOL 108 or SOL 111) using the dynamic load spatial distribution. If structure is constrained, then apply the dynamic load at only one results are not equal, then there is probably an error in the specification of the dynamic load, and the researcher should check the LSEQ and DAREA entries. If modal frequency response (without structural damping) is used, then the 0.0 Hz results should be close to the static results; the difference is due to modal truncation, then apply the load across the entire frequency range of interest, the running modal frequency response, enough modes to ensure accurate results for even the highest forcing frequency. Also to have a small enough if in order to accurately capture the peak response. Verify these results for reasonableness (it may be easier to look at magnitude and phase results instead of real and imaginary results, which are the default values). The ultimate goal is a transient response analysis for which damping is to be neglected, then the frequency response analysis can also omit damping. However, if damping is to be included, then use the correct damping in frequency response analysis. The proper specification of damping can be verified by looking at the half power bandwidth. Plots are important at this stage to assist in results interpretation. X-Y plots are necessary in order to see the variation in response versus frequency, deformed structure plots at a frequency near a resonant frequency can also help to interpret the results. The structure plots are made, in the imaginary component because the single degree-of-freedom (SDOF) displacement response at resonance is purely imaginary when damping is present (this response does not occur in practice because the response is usually due to several modes).
- Time-dependent loads, perform transient response (SOL 109 or SOL 112) analysis. The structure is constrained, apply the load "very quickly" (over one or two time steps) as a step function and look at the displacement results, the duration of the analysis needs to be as long as the period of the lowest frequency mode. For an SDOF system, a quickly applied load results in a peak displacement response that is twice the response resulting from the same load applied statically, this peak response does not occur in your actual model because of multiple modes and damping, but the results should be close. If the structure is unconstrained, the displacements will grow with time (unless the rigid-body modes are excluded in a modal transient response analysis). However, the stresses should be roughly twice those from the static analysis. In any event, examine the results via X-Y plots to ensure reasonableness. Apply the correct time variation to the load and compute the results. Again, use X-Y plots to verify the accuracy of the results.
- Finally, perform any other dynamic analyses, such as response spectrum analysis, random response, nonlinear transient response, or dynamic response optimization.

RESULTS

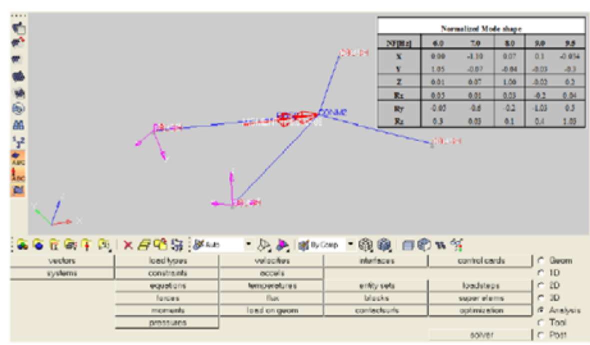

Figure-7. Results obtain after optimization.

Figure-7 shows the result obtain after optimization. It was found that at each frequency, the value engine is having only one degree of freedom, so decoupling is achieved.

CONCLUSIONS

In this research is analytically examines the multi- dimensional mounting schemes of an automotive engine- gearbox system when excited by oscillating torques, and new dynamic decoupling axioms are presented and compared with the CAE results. It explain the design concepts for the 4-point Engine mounting scheme, also the methodology can be used for optimizing Engine mount location new vehicle with reduced Vibration.

REFERENCES

[1] Sinha, A.K., Eriki, A.K., Lokanadham, R. and Babu, A.R. (2015). Vibrational Analysis of Propeller Shaft with Transfer Crack. International Journal of Mechanical and Production Engineering (IJMPE). 3(3), 70-75.

[2] Van, T.K., Van, D.M., Bram, D.J. and Maarten, S. (2012). Design, implementation and experimental validation of optimal power split control for hybrid electric trucks. Control Engineering Practice. 20,547-558.

[3] Jomehzadeh, E., and Saidi, A.R. (2011). Decoupling the nonlocal elasticity equations for three dimensional vibration analysis of nano-plates. Composite Structures. 93, 1015-1020.

[4] Fai Ma, Matthias, M., and Ali, I. (2010). The decoupling of damped linear systems in free or forced vibration. Journal of Sound and Vibration. 329, 3182-3202.