MBS-Simulation in the Engineering Process at Steyr-Daimler-Puch with Focus on Powertrain Applications

Anton Riepl, Stefan Hofbauer

1. Introduction

At Steyr-Daimler-Puch, MBS-simulation is used for simulating rough roads, NVH and vehicle dynamic problems. In the last 18 months the software ADAMS has been more and more used to execute these tasks. The paper on hand presents two tasks in the field of NVH calculation that have been dealt with by using ADAMS. The varying complexity of the models shows the respective development status during vehicle development. In both cases the items

- modeling
- description of excitation
- parameter variations and discussion on result

have been focused on.

2. Linear vehicle model for evaluating the phenomenon "engine stutter"

A decision concerning the engine support concept is made at a rather early stage of a project. Multi-body systems are an excellent tool for evaluation of various support concepts with regard to their static and dynamic characteristics. First investigations are static load cases such as load due to the driveline aggregate's own weight, maximum load in vertical direction (pothole) and loads entailed by maximum drive torques, which, of course, require non-linear characteristics in the supports. Beside, Fig. 1 shows a simple, linear vehicle model that allows early statements concerning possible comfort problems with regard to engine stutter. During the concept phase these statements are of utmost importance, since at that point it is relatively easy to undertake suitable measures for reducing these comfort drawbacks.

Engine stutter frequently occurs on highways with a concrete slab surface. The regular distance between the transverse joints excites natural frequencies of the engine and the body [1]. The relation between the angular frequency ω, vehicle speed v and concrete slab length L, is given with: $\omega = (2\pi v)/L$.

The rigid body model (Fig. 1) used for these analyses consists of the following bodies:

- wheel substitute masses 1, 2, 3 and 4: simplified representation of the wheel suspensions by translatoric guidance of the wheel substitute masses with respect to the body

- engine 5: elastically connected to the body
- body 6

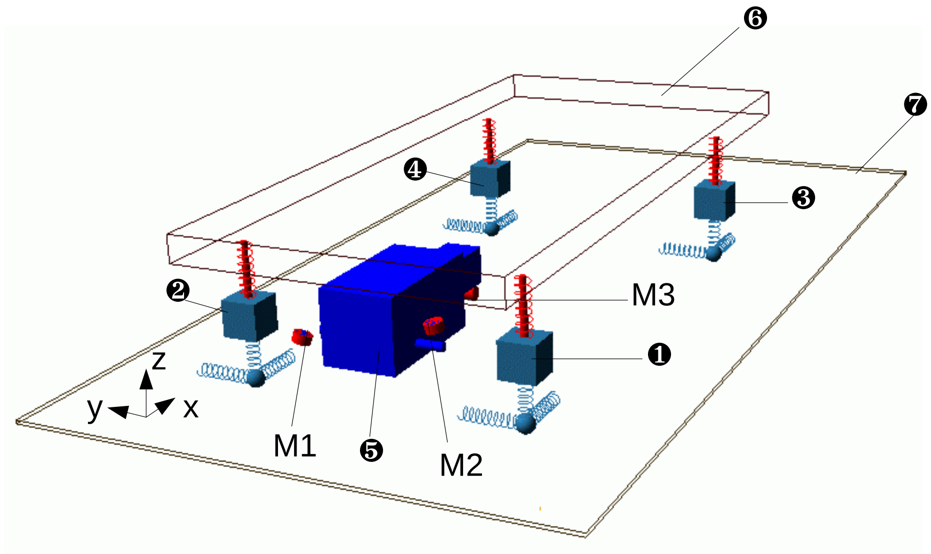


Figure 1: Linear vehicle model for engine stutter

The number of degrees of freedom of the system is:

$$DOF = 6*n - q = 6*6 - 20 = 16. \qquad (1)$$

In Eq. (1), n is the number of parts and q ist the number of constraints of the model. The characteristics of the body springs and the body dampers, which act between the individual substitute masses, are considered to be constant. The elastic connection of the individual wheel substitute masses to the ground 7 (Fig. 1) is achieved by means of wheel stiffnesses c_z and c_y in vertical and lateral direction, respectively. The connection in the circumferential direction is: $c_x = c_\varphi/r^2$, with c_φ being the torsional stiffness of the side shaft and r being the tire radius. The method of connecting in the longitudinal direction is only possible in the case of a 4x4 drive. For the observed frequency range, the constant values for engine mount stiffness c_{Mi} and damping d_{Mi} were calculated by means of the equations

$$c = \sqrt{\frac{c_{dyn}^{\;2}}{1+\tan^2\delta}} \;, \qquad d = \frac{c}{2\pi f}.\tan\delta\,. \qquad (2)$$

In Eq. (2), c_{dyn} is the dynamic stiffness, δ is the loss angle and f is the frequency. The tool ADAMS/Linear [2], is used to calculate the eigenvalues of the system matrix **A**. For the given oscillatory system, these are complex conjugate and have negative real parts.

Fig. 2 shows the coupling of the translational degrees of freedom (X, Y, Z) and the rotational degrees of freedom (RX, RY, RZ) of the 6 rigid body modes of the engine. Engine stutter is mainly caused by the displacements in z-direction and the twist about the y-axis. The graphical representation (Fig. 2) shows that these motion modes are very distinctive at the frequencies 5.9, 7.2 and 10.1 Hz.

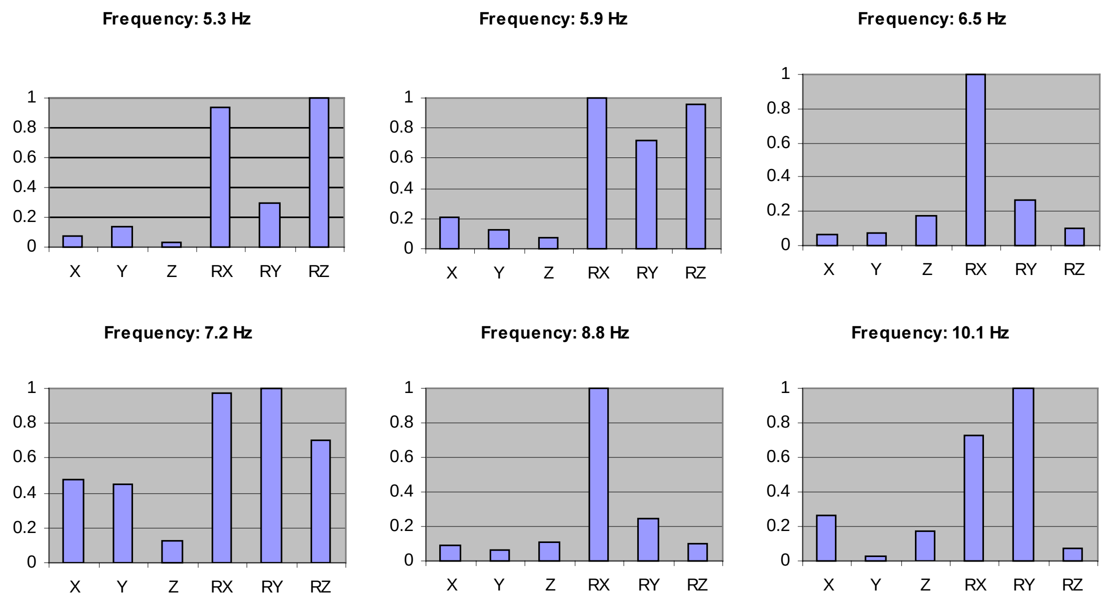


Figure 2: Coupling of the six rigid body motions of the engine

In order to evaluate the vibrational behavior of the engine during driving on bumpy roads, forces in z-direction are applied on the front wheels simultaneously and in phase. Fig. 3 shows the amplitude gain = FZ_{M3}/FZ_{wheel} for the basis model and the parameter variations. FZ_{wheel} stands for the identical forces at both front wheels, and FZ_{M3} is the z-force at engine mount M_3 (Fig. 1). The curves in Fig. 3 mean:

- M_3 basis: basis model
- M_3 position 1: symmetrical arrangement of supports M_1 and M_2 in front of the side shafts (in driving direction)
- M_3 position 2: symmetrical arrangement of supports M_1 and M_2 behind the side shafts (in driving direction)
- M_3 damping*2: mount M_3 with increased damping effect

At approximately 3 Hz, all 4 curves show a peak caused by the excitation of a body eigenmode (longitudinal stutter). The maxima at approximately 7 Hz and 10.5 Hz are caused by the engine's rigid body eigenmodes. The peak at approximately 7 Hz must be considered as very critical because the human response is very sensitive in this frequency range. Variation of mount positions M_1 and M_2 - both mounts in front of the side shafts - caused a distinct deterioration (Fig. 3). With regard to the engine stutter, the variation with increased damping effect ranks best.

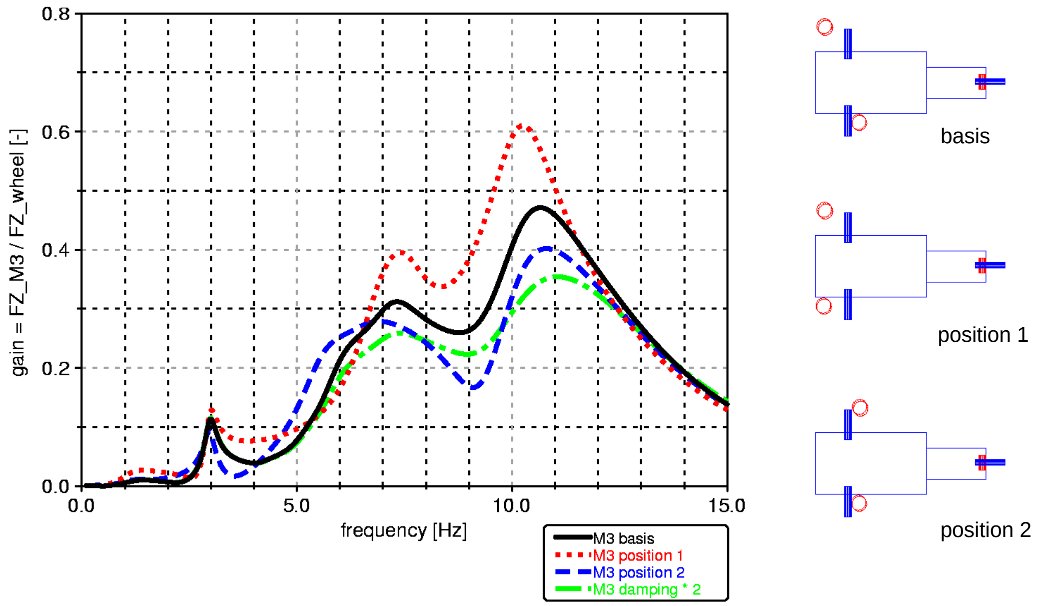


Figure 3: Gain of the ratio vertical force of mount 3 / vertical force on front wheels

3. Non-linear powertrain model

The following example deals with the low speed boom. Prototype vehicles showed that during WOT accelerations the interior noise increased excessively within the engine speed range between 900 and 1800 rpm. The excessive level is caused by irregular engine torque exciting a torsional eigenmode of the powertrain.

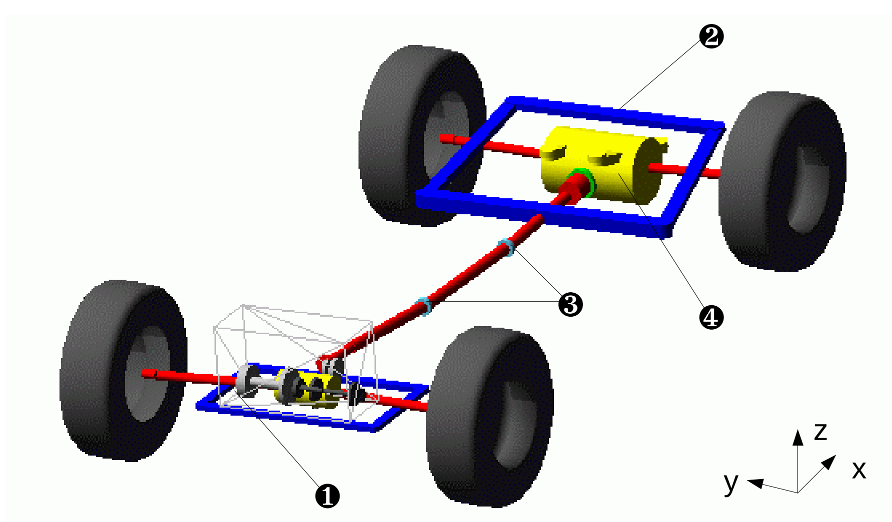


Figure 4: Non-linear powertrain model

Fig. 4 shows the non-linear 4x4 powertrain model applied for solving the problem. The number of the system`s degrees of freedom, see Eq. (1), DOF = 6*n – q = 6*34 - 166 = 38. The engine housing 1, the rear subframe 2 and both intermediate bearings 3 of the propshaft are elastically supported by the body. The rear axle drive 4 is elastically supported by the subframe 2. For the given problem, the torsional behavior of the powertrain is of utmost importance.

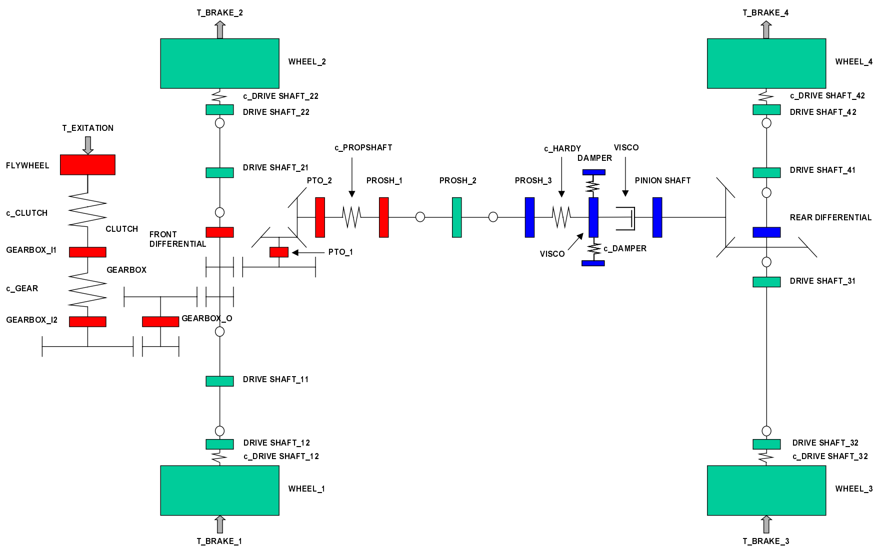


Figure 5: Torsional transfer behavior scheme of the non-linear powertrain model

Fig. 5 shows the scheme for the torsional transfer behavior from engine to the wheels. The colored bodies are components with mass and moments of inertia. The bodies marked red are rotationally supported in the engine/transmission unit, the components marked blue are rotationally mounted to the rear axle drive housing. The points marked by a spring represent the torsional elasticities of the 4x4 powertrain. Torsional elasticities and respective dampings were taken into consideration at the following powertrain components:

- Clutch stiffness c_{CLUTCH}
- Transmission stiffness c_{GEAR}: The complete torsional stiffness of the drive shafts of the transmission was reduced to the input shaft. The torsional stiffness c_{GEAR} depends on the gear that is engaged.
- Propshaft stiffness $c_{PROPSHAFT}$
- Hardy disk stiffness c_{HARDY}
- Driveshaft stiffness $c_{DRIVESHAFT\_i2}$ with i = 1(1)4

The individual transmission ratios were accounted for by means of kinematic constraints in the form $r_1 * \omega_1 = r_2 * \omega_2$:

- Transmission ratio in the individual gears: i_{Gj}, with j = 1(1)5
- Final drive – transmission ratio i_{FD}
- Axle drive transmission front i_{AXF} and rear i_{AXR}

The transfer behavior of the viscous coupling – damper symbol in Fig. 5 – is approximated by means of a non-linear characteristic in the form M = M($\Delta\Omega$). Effects caused by the absolute speed and the temperature were neglected. External loads acting on the MBS model are the excitation torque $T_{ECXITATION}$ acting on the flywheel, and the brake torque T_{BRAKE_i} acting on the four wheels. The torque distribution between front and rear axle is 80% to 20%.

The torque irregularity due to the second engine order is caused by the combustion forces generated in a 4-cylinder combustion engine twice per revolution of the crankshaft. Thus a frequency range of 30 Hz - 60 Hz applies to the above mentioned engine speed range. The engine torque acting as external torque on the flywheel, is calculated by means of the equation

$$T_{EXCITATION} = T_0 + T_1 * \sin[(2*\pi*f_{st}*t) + (\pi* s_f*t^2)]. \qquad (3)$$

The abbreviations in Eq. (3) mean: basic torque T_0, superimposed torque T_1, start frequency f_{st} of the sinus sweep function, and sweep factor s_f. Since the relevant frequency range is known from previous tests, and in order to reduce the computing time as well as the amount of output data , the applied start frequency is set as f_{st} = 30 Hz.

An excerpt from the parameter studies for the 4x4 powertrain model is shown in Tab. 1. The measures can be classified as torsional and vertical ones.

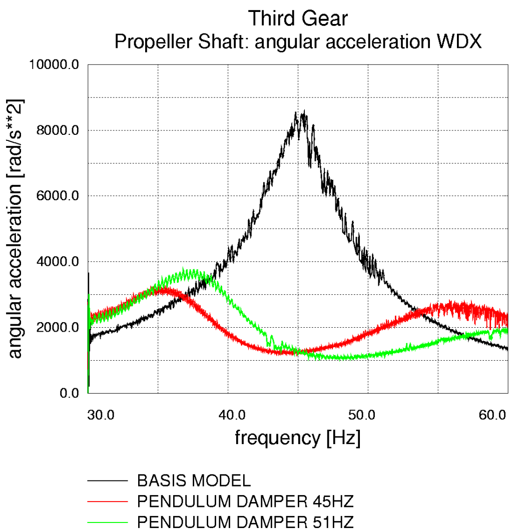
	Measures	Shifting of critical torsional eigenfrequency	Reduction of critical torsional resonance	Reduction of max. rear dive vibration
Torsional	Stiffness variation by ± 20% propshaft, driveshafts and hardy disk	+ *	~	~
	Torsional blocking mass	++ *	~	~
	Torsional damper		++	++
Vertical	Variation of the vertical stiffness of rear axle transmission support	~	~	~
	Vertical damper rear drive	~	~	+

~ slight effect; + positive effect; * still in operation range

Table 1: Parameter variations for non-linear powertrain model

Generally it can be stated, that the critical eigenfrequency, which in the first gear is approximately 50 Hz, decreases with the gear number (e.g. in third gear approx. 45 Hz). Tab. 1 also differentiates whether the respective measures shifted the torsional eigenfrequency

and reduced the critical torsional resonance, and how the measures effected the dive motion of the rear axle transmission caused by the torsional resonance.
Out of all analyzed measures, the torsional vibration damper showed the best results. The torsional vibration damper is rotationally mounted on the viscous coupling (Fig. 5).



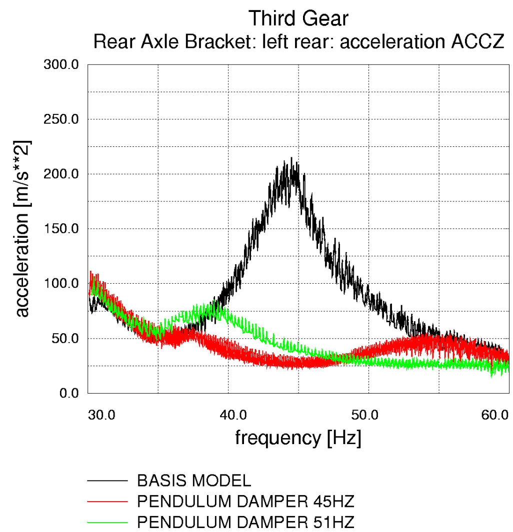


Figure 6a: Angular acceleration of propshaft

Figure 6b: Acceleration at a mounting of the rear axle transmission

Fig. 6a shows the effect of a torsional vibration damper in the frequency domain. The variant without torsional vibration damper (basis model) shows a distinct increase in angular acceleration of the propshaft in third gear at approximately 45 Hz. A torsional vibration damper tuned to 51 Hz as well as one tuned to 45 Hz can reduce this resonance. The well-known phenomenon when using a torsional vibration damper, namely the splitting of the resonance frequency, occurs in this case, too [3]. The effect of the torsional resonance on the dive motion of the rear axle transmission is shown in Fig. 6b. Both torsional vibration dampers considerably reduce the acceleration of the rear axle transmission. A two-mass flywheel, which could have a distinct positive effect on the source of excitation, could not be used due to package reasons. For Fig. 6a and 6b, the results of the simulations carried out in the time domain were transformed to the frequency range by means of the in-house-tool t2f [4].

4. Comments, outlook

During the work on the applications presented in this paper it became obvious, that staying in the ADAMS enviroment during the whole simulation process would be very beneficial. Graphical representation of the coupling of the degrees of freedom (Fig. 2) as well as of the kinetic and potential energy [3], [5] should be an integral part of the tool ADAMS/Linear.

Another interesting item is the generation of Bode representations in the case of out-of-phase excitation. Furthermore, without too much effort it should be possible to convert models generated in ADAMS/Car so that they can be used in ADAMS/View.

Steyr-Daimler-Puch's future focal points with regard to further development activities in the field of power-train dynamics/NVH are:

- Implementation of evaluation criteria that approximate the human perception of vibrations
- Description of the gearwheel contact for analyzing gearwheel noises, for which ADAMS/Engine can be an excellent tool
- Implementation of active elements via the interface to MATLAB / SIMULINK

5. Literature

[1] Klingner, B.: Einfluß der Motorlagerung auf Schwingungskomfort und Geräuschanregung im Kraftfahrzeug. Dissertation TU-Braunschweig, 1996.
[2] Using ADAMS/Solver; Version 9.
[3] Holzweißig, F., Dresig H.: Lehrbuch der Maschinendynamik, Fachbuchverlag Leipzig-Köln, 1994.
[4] Hofbauer, S.: Modellbildung und Simulation von nichtlinearen Motorlagern. Diplomarbeit TU-Graz, 1998.
[5] Laschet, A.: Simulation von Antriebssystemen, Fachberichte Simulation, Band 9 Springer-Verlag,1988.