Modelling of automotive antivibration rubber parts

1. The nature of rubber

Natural rubber is a high polymer: it is made up of molecules of great length composed of hydrogen and carbon. Naturally, molecular chains are flexible, independent of each over. In its raw state, natural rubber if a tough material which deforms in part by viscous flow and in part elastically. It becomes soft and sticky when warms. Its practical uses in the raw state are limited.

The practical potential of rubber is achieved by the process known as vulcanisation. Vulcanisation with sulphur links the rubber molecules together at intervals along their length by means of short chains of sulphur atoms (other vulcanising systems as peroxides can be used). It gives strength, elasticity and mechanical behaviour less sensitive to temperature.

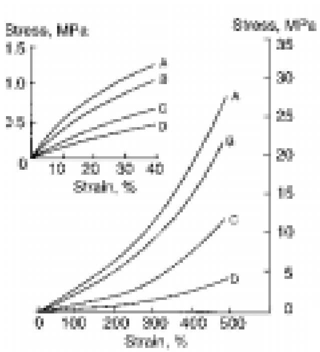
Rubbers also contains a filler, generally carbon black, which improves tear and abrasion properties as well as increasing the modulus, hysteresis and creep.

2. Mechanical properties

Here is a table which sums up physical properties of vulcanized natural rubber.

Property	Unfilled rubber	Filled rubber	Mild steel
Elongation at break, %	770	520	40
Young's modulus, MPa	1.5	Strain dependent	210000
Shear modulus, MPa	0.49	1.3	82000
Density	0.97	1.15	7.9

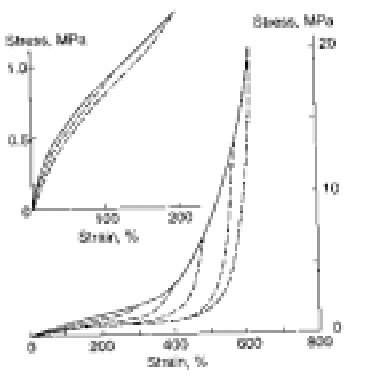
2.1 Stress-strain behaviour



The load deflection curve for unfilled vulcanizates deformed in tension and compression are approximately linear for strains up to a few per cent, and values of Young's modulus E0 can be ascribed to these low strain regions. The stiffness at first decreases with increasing strain, but two features lead to an increasing stiffness at high strain. One is the approach of the molecular chains in the network to their limited extension. The other is strain-induced crystallization.

The A, B, C and D curves correspond to four natural rubber vulcanizates. A and B are two reinforcing black, while C and D are unfilled and have different degree of vulcanization.

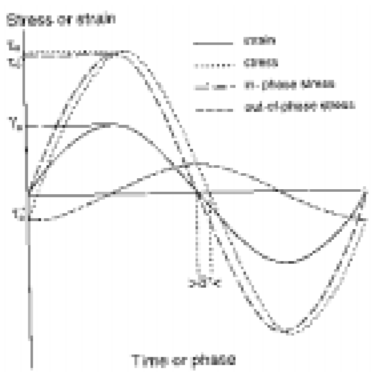
2.2 Hysteresis



Energy dissipation through hysteresis is represented by the area between the loading and unloading curves in load-deformation cycle, and occurs with all rubbers. It depends on the type of polymer, of the filler and the other compounding ingredients. For natural rubber there is little hysteresis up to moderate extensions.

The degree of damping is commonly expressed in terms of the loss angle.

2.3 Dynamic behaviour



If the relation between the force and the deformation is approximately linear the application of a sinusoidal deformation will result in a sinusoidal force of the same frequency but displaced in terms of phase by an amount termed the loss angle.

It is convenient to consider the elastic (in phase response) and the viscous (out-of-phase response) in terms of two moduli. The overall response can be expressed as a complex modulus:

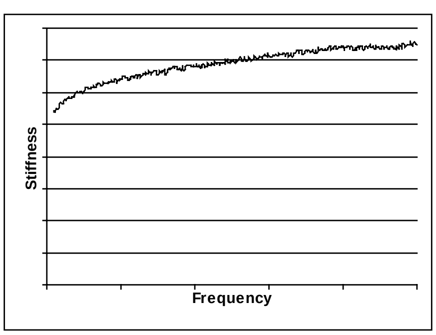
$$G^* = G' + jG''$$

The dynamic stiffness is given by $K = |G^*|$, and the phase or loss angle δ by $\tan\delta = G''/G'$.

2.3.1 Temperature dependency

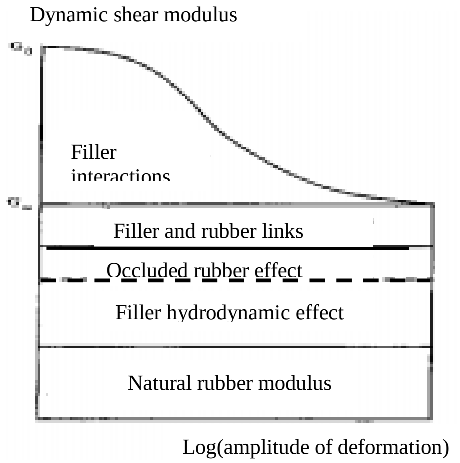
Temperature has very low effect on dynamic stiffness in the range of temperature usual in the automobile industry.

2.3.2 Frequency dependency



The stiffness of natural rubber is increasing lightly with frequency. This is called "dynamic stiffening". The phase is quite independent of frequency. Dynamic stiffening is strongly correlated with the phase angle, which is itself correlated with the filler content. It shows that the static stiffness of rubber needs to be very precisely defined (amplitude and frequency or speed of measurement).

2.3.3 Amplitude dependency ("PAYNE effect")

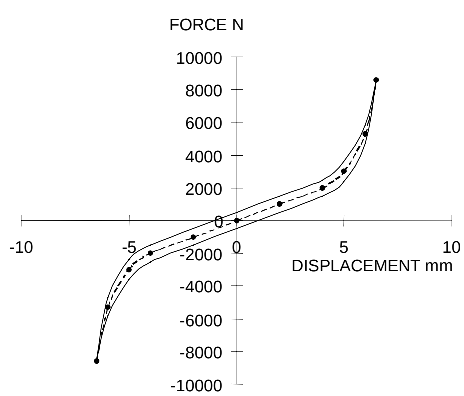


The dynamic shear modulus of filled rubber shows a dramatic decrease when the amplitude of deformation is increasing. This phenomenon (Payne effect) is linked to the breakdown of interaction of carbon black aggregates with rubber.

3. Antivibration rubber parts

- Hydraulic and rubber engine mounts (between engine and body): they support the engine, and avoid the transmission of vibrations from the engine. Hydraulic mount capability to dissipate energy in a low frequency band is used to damp the vertical mode of the powertrain.
- Tie bars: they are used as torque rod or connecting rod. They act principally in the longitudinal direction, because of their low stiffness in the other directions.
- Bushings: kinematic and filtration roles are ensured by these parts.
- Strut mounts are installed between the damper and the body, to prevent transmission of vibrations due to the increasing stiffness of the damper at high frequency.
- Cradle mount: these rubber mounts link sub-frame to body. In that case, filtration of engine vibration is ensured in two steps: engine vibration are transmitted to the cradle via the engine mounts, which is isolated of body by cradle mounts.
- Exhaust mounts: these rubber parts support exhaust line, and filtrate vibrations. They must accomodate deformation due to heat elongation of the exhaust line.

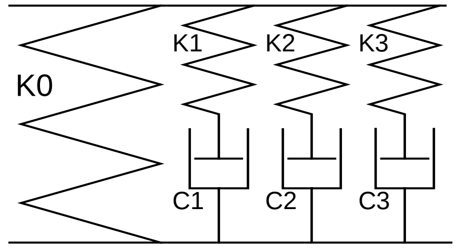
4. Static modelling of rubber parts



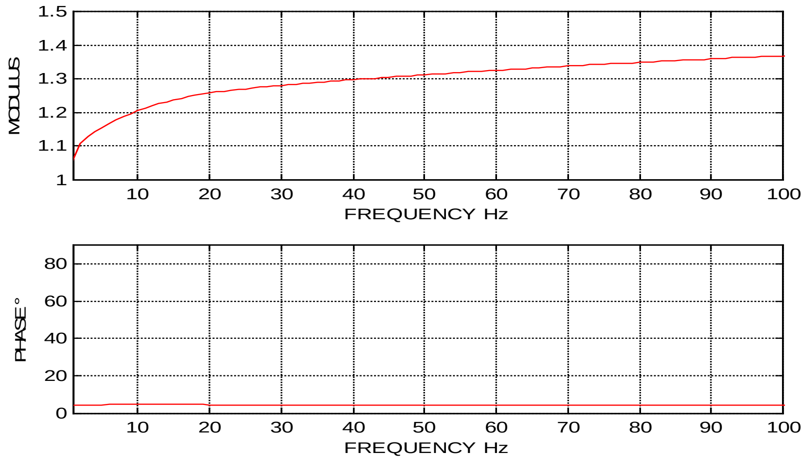
The force versus displacement relationship is not linear. Some of the non linearity comes from the rubber itself (cf. 2.1), some from the geometry of the part. These curves can be predicted by finite element analysis (rubber specific code). We use a spline to represent this behaviour. Three splines must be used to completely model one bushing: one for each of the stiffness principal directions. We assume that all directions are decoupled: the force in one direction depends only on the displacement in the same direction.

5. Dynamic modelling of rubber parts (linear)

To take into account the viscoelastic damping of rubber, we use a generalised Maxwell model. It is a combination of springs and dampers.

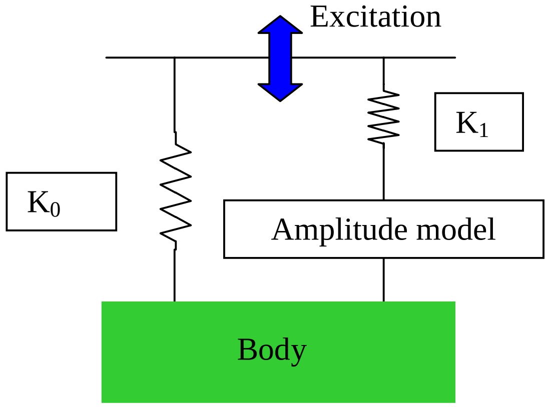


Three Maxwell cells are a good compromise for simulation. In this case, there are seven parameters which must be adapted to the rubber .

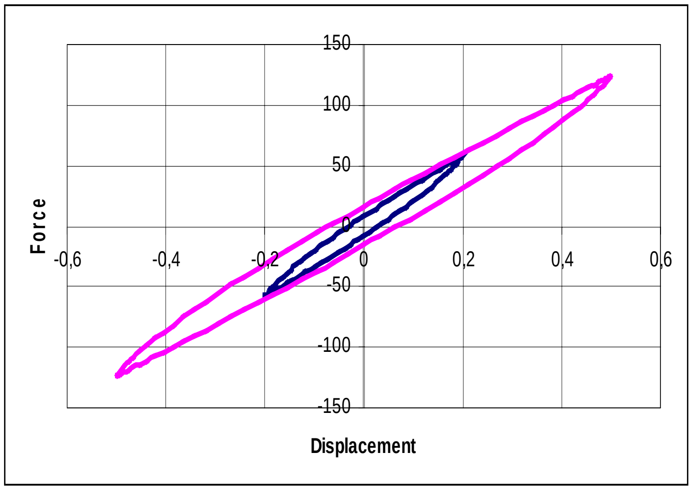


6. Dynamic modelling of rubber parts (non linear)

It takes into account the decreasing of the dynamic modulus of filled rubbers as the amplitude of the applied strain increases.



Here is Adams simulation results for two amplitudes of excitation of the amplitude model:



The higher dynamic stiffness is obtained with the lower amplitude.

7. Modelling of hydraulic parts

7.1 Linear modelling of hydraulic parts

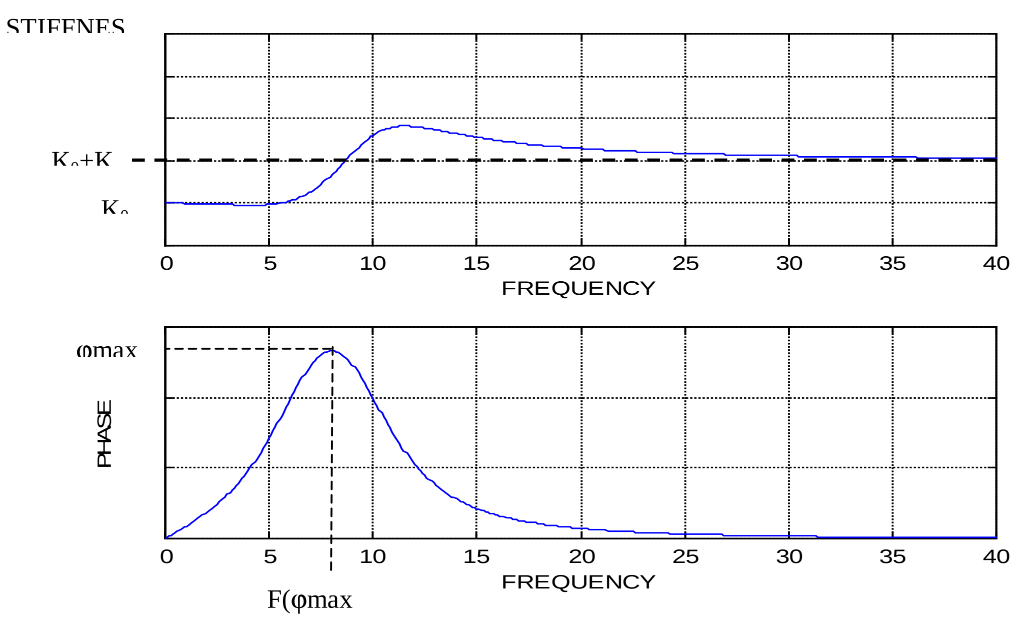
An hydraulic mount can be described with four parameters in the frequency domain: the maximum of phase (φmax) and its frequency (f(φmax)), the static stiffness (K_0) and the stiffness increase at high frequency (K_1). These parameters are particular to one excitation amplitude, but of course the linear model is amplitude independent.

A practical tool in Adams to represent such linear behaviour is the transfer function:

$$TFS = \frac{n2.p^2 + n1.p + 1}{d2.p^2 + d1.p + 1},$$

where n1, n2, d1, d2 are real numbers estimated from K_0, K_1, φmax and f(φmax).

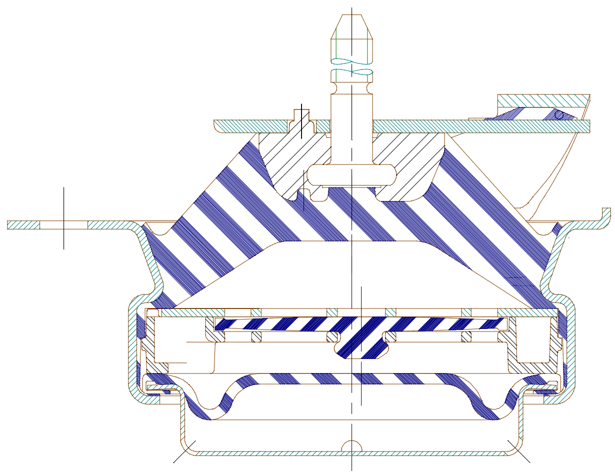
Example:

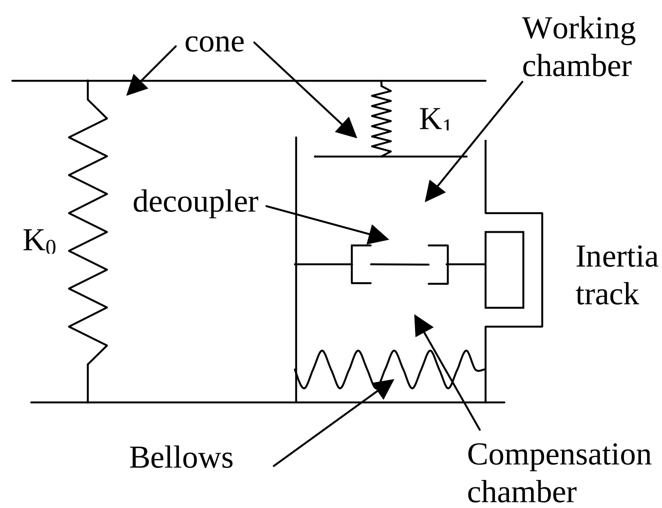


7.2 Non linear modelling of hydraulic parts

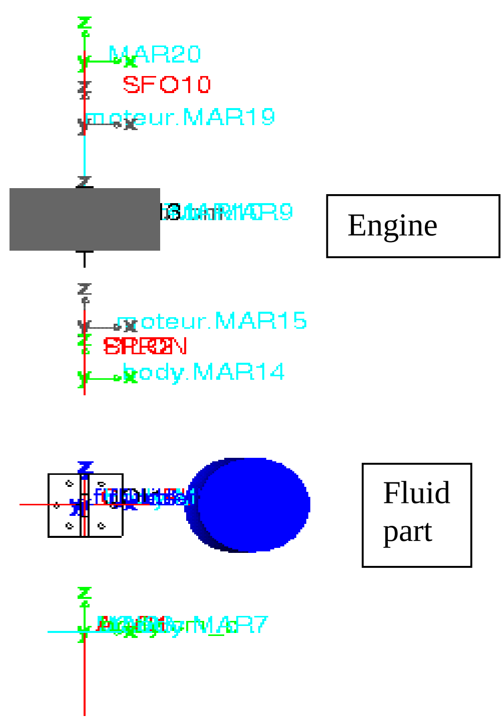
Real parts are not linear for two reasons: the flow is not laminar but turbulent; there is often a non-linear device, the decoupler, to reduce the high frequency stiffness.

Consequently, a non linear ADAMS model of an hydromount has been developed.





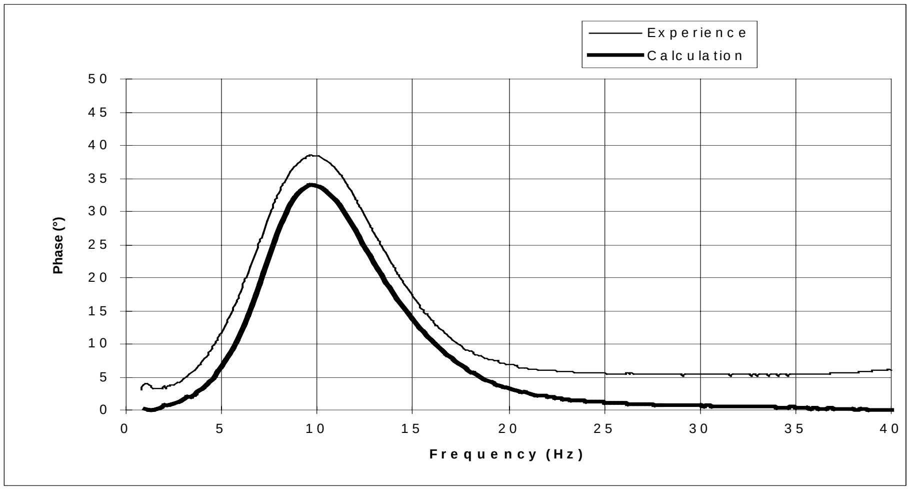
Fluid is enclosed within two chambers linked by a pipe. The decoupler also links the two chambers. As long as the amplitude is low, the decoupler moves freely and the dynamic stiffness remains K_0. For larger amplitudes, the decoupler is limited and we have to modelize fluid flow in the inertia track.

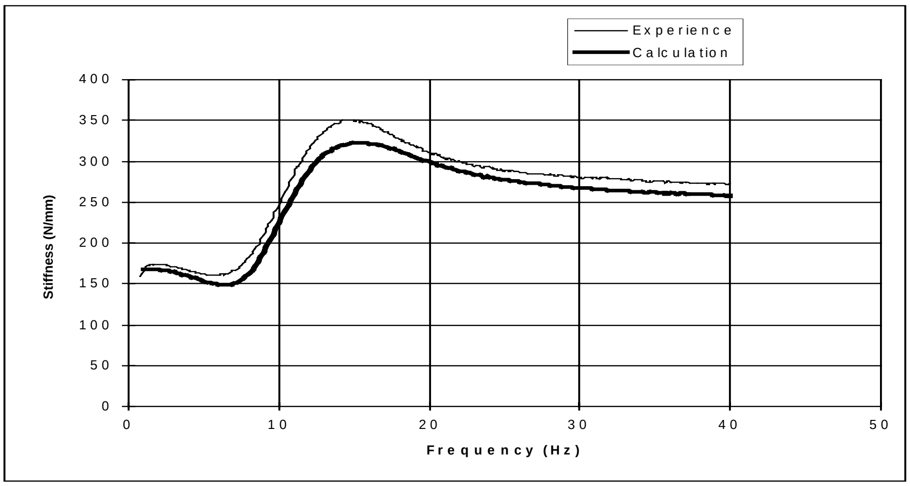


Adams hydromount model

The hydromount model is made of one part representing the fluid in the inertia track, and a general force. The part which represents the fluid has a rotational degree of freedom. Energy dissipation has a term proportional to speed (viscous damping) and a term proportional to square of speed (turbulent damping). The general force applies the calculated force generated by the mount to the body and the engine. Physical characteristics of mount are input in design variables.

Here is a comparison between experience and calculation.



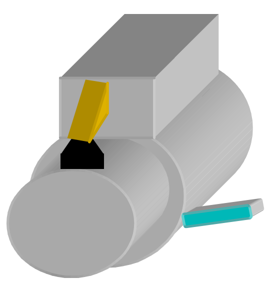


The calculated maximum of phase is 34° instead of 38°. Its frequency is 10 Hz in both cases. The error on stiffness is less than 10% whatever the frequency. Thus the model provides good results.

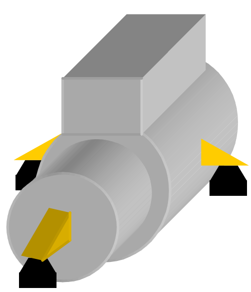
8. Some engine suspensions models

The type of the engine suspensions depends on the vehicle, and on the orientation of the engine in the car (longitudinal or transversal direction). In all cases, engine suspension has to support the engine, and to limit its displacement.

pendulum suspension

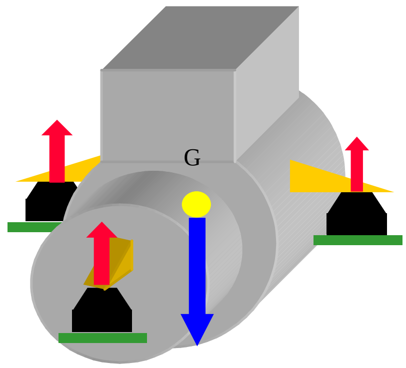


3 points suspension



9. Static loads

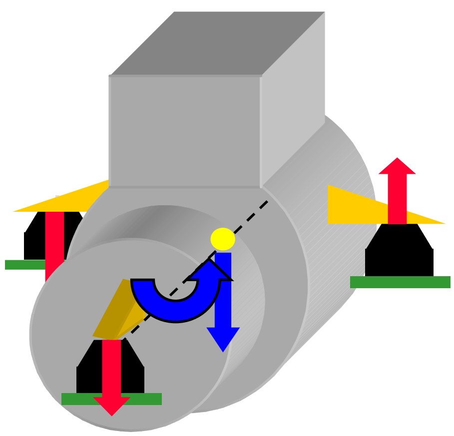
Static load due to engine weight must be known by engineer to design the mount. It gives the working point of each mount. ADAMS provides us this information by a static calculation:



Input: engine weight
Output: static load of each mount

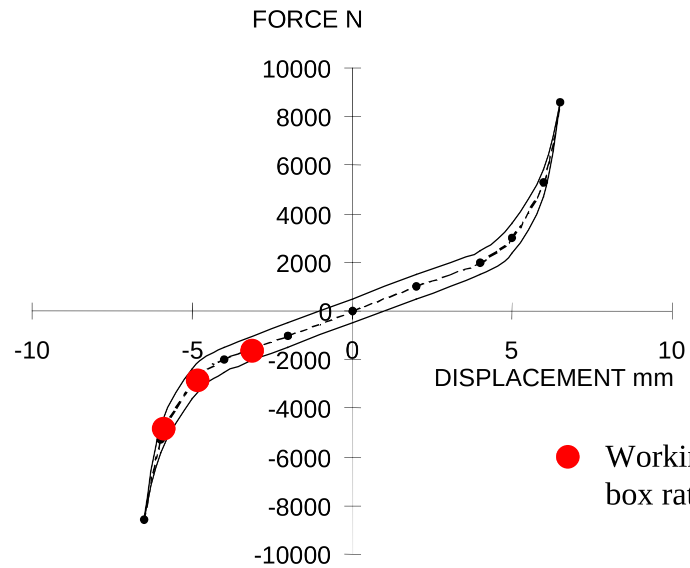
10. Static calculation

The translation pistons motions are transformed into rotation by the crankshaft which delivers a torque. This torque is transmitted to wheels by gearbox and differential. This torque induces a rotational and translatory displacement of the engine which must be evaluate and limit. In Adams, we impose a torque to the engine and we look at its response.



Input: engine weight and torque
Output: displacements of elastic centres, forces

Results are plot on the force-displacement splines:

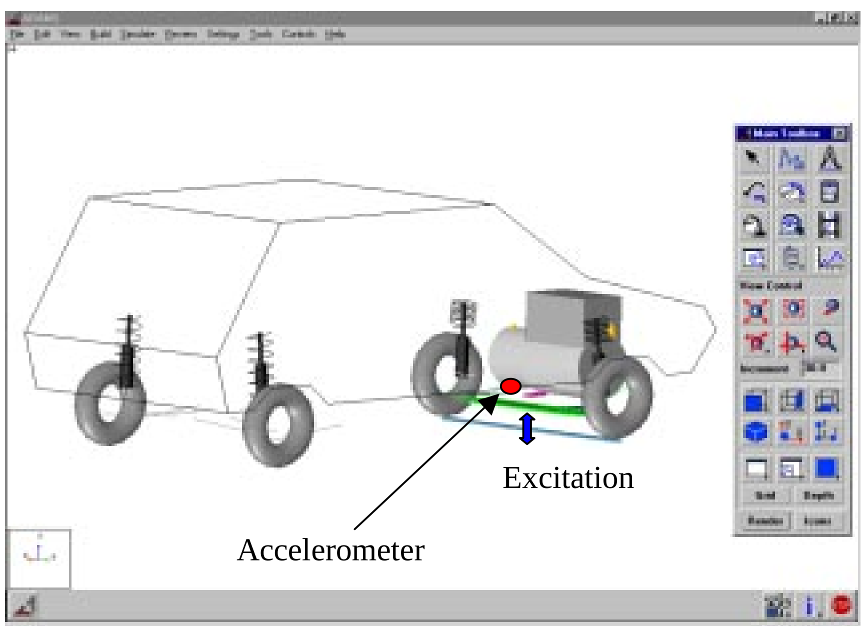


Working points along gear box ratio.

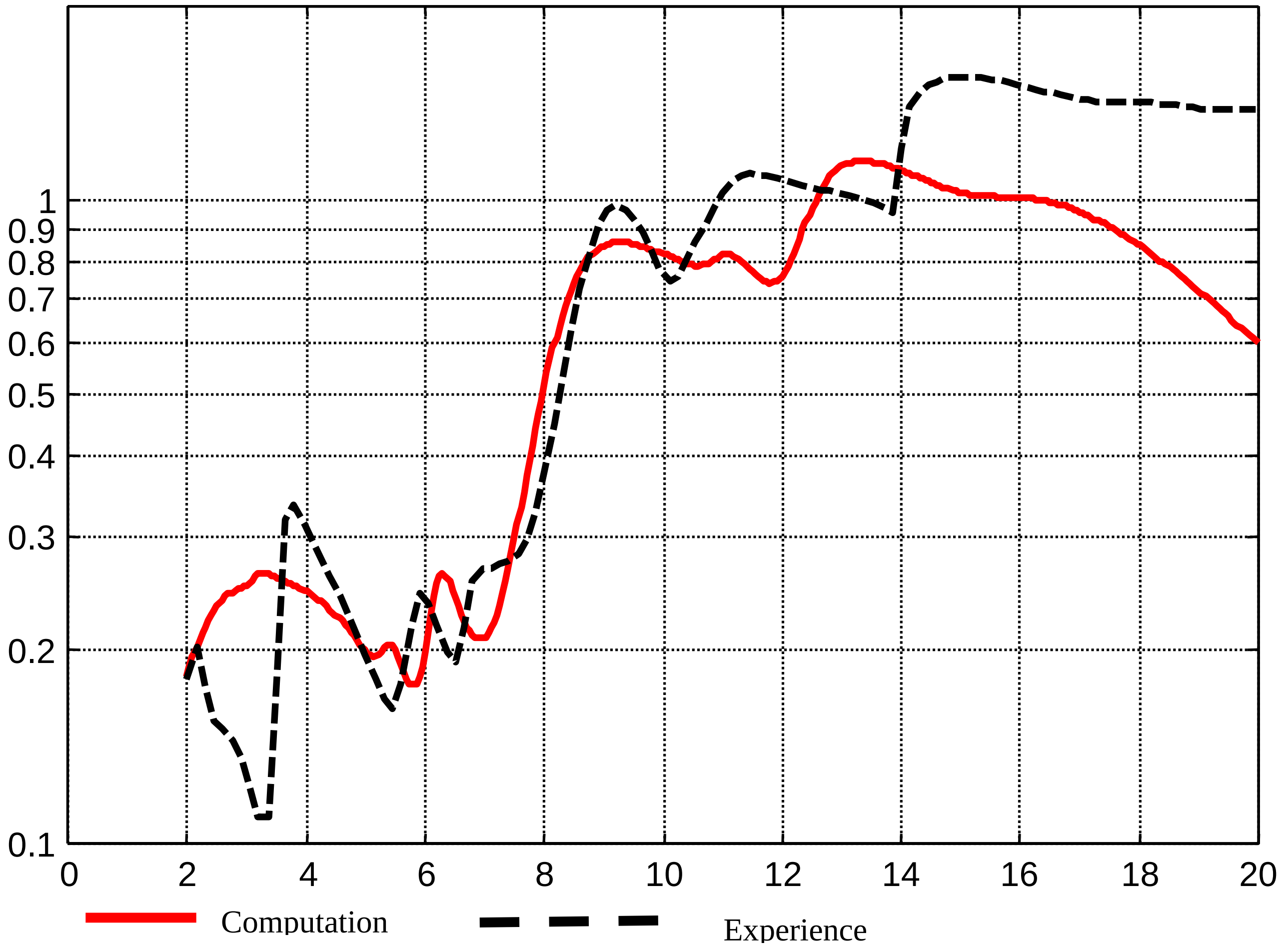
11. Harshness calculation

Vehicle comfort is characterised by seat rail acceleration. These accelerations are the results of an excitation at the front wheels. Along the criteria of comfort, excitation can be at amplitude, velocity or acceleration constant, and we do a frequency swept. Results are plot in the frequency domain.

A complete vehicle is modelled in Adams for this type of calculation: wheels, vehicle suspension, engine, engine suspension, subframe and body. Here is an example of an Adams model:



Calculation and experimental (dotted line) results are compared in the frequency domain.



12. Conclusion

Paulstra develops Adams models of the parts it produces: bushings, tie bar, hydraulic mounts... These models are incorporated in more complete model of vehicle to analyse behaviour of the whole engine suspension, to determine characteristics of each engine suspension part to achieve automobile makers criteria.