Tyre Modelling for NVH Engineering in ADAMS

Marc DUVERNIER, Patrice FRAYSSE, Viviane BOMBLAIN and Eric DORMEGNIE

Michelin Tyre Company
Manufacture Francaise des Pneumatiques Michelin
23, Place des Carmes Dechaux
63040 Clermont-Ferrand
Cedex 9,France

ABSTRACT

There is a growing need to use NVH analysis tools at all stages in the design process of a vehicle. The tyre is a key element for the assessment of the NVH performance of a vehicle. This paper presents how a detailed tyre model can be coupled to a vehicle model in ADAMS to simulate various NVH tests. Both standing and rolling tyres are investigated in the time and frequency domain. The detailed tyre model is first built using a non-linear FEA method. It is then sub-structured using the Craig-Bampton technique (fixed interface) to be translated into a flexible body in ADAMS. Different approaches are then used to simulate the tyre and suspension behaviour when rolling over obstacles or standing on a shaker.

A tyre modelling for dynamic studies : What is it for ?

MICHELIN has been developing for many years dynamic modelling of passenger car tyres dedicated to the comfort and noise performance.
In this field, modelling links vehicle models and tyre models. It appears now clearly that the tyre is too complicated a structure to be considered as a simple spring for refined dynamic vehicle simulations. For example, some tuning between vehicle and tyre modes seems necessary to maximise ride comfort.

Tyre materials taken into account

Departing from the geometry description of the tyre, a 2D axisymmetric mesh is defined. It permits the calculation of the inflated state thanks to the knowledge of the geometry, the non-linear orthotropic materials laws and all the necessary extrinsic conditions (temperature, pressure, ...). At this stage, we generate a 3D mesh in order to apply load on the tyre. The deflected state is then obtained, from which the two dynamic reference states needed for the non-rolling and rolling models are derived. The resulting 3D tyre mesh commonly reaches 150000 dofs because for modal analysis it must be refined enough to catch correctly all the required modal wavelengths of the tyre.

Modelling principles

Dynamic substructuring methods are commonly used nowadays to describe the dynamic behaviour of components belonging to large structures. Application to tyre is particularly well adapted, in spite of the presence of non-linear visco-elastic materials, because the tyre looks very simple from a geometrical point of view, the modal density is low at low frequencies and two interfaces only are present (tyre/hub and tyre/ground).

Dynamic substructuring methods offer a way to build reduced models for any structure, based on the assumption that low frequency dynamic motions are dominated by low frequency eigenmodes.

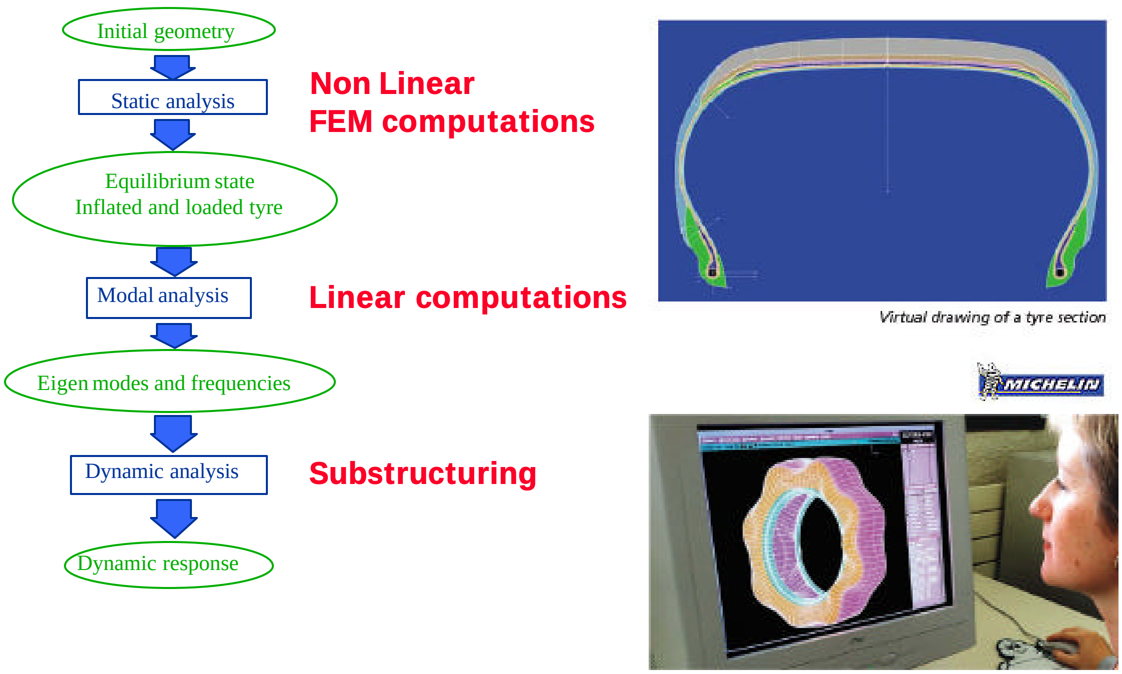

Figure 1 : Model Building Scheme

Craig Bampton methods

As MSC/ADAMS for the Flex Body approach, the Craig-Bampton substructuring method was chosen to build our tyre models ([1], [8], [11]). The Craig-Bampton method meets our preference for 3 main practical reasons :

1. No wheel characteristics are required to apply the dynamic reduction.
2. The interface physical degrees of freedom are preserved during reduction, implying that whatever the number of retained modes is, the static information (i.e. rigidities) is not sensitive to the modal truncature; it is particularly useful if we want to deal with a limited number of modes (between 15 and 30 for example).
3. Validation measurements with fixed-wheel boundary conditions are easier to carry out.

Moreover, Craig-Bampton models are often easier to understand, handle and connect for our customers.

Description of the models

A dynamic tyre model is a dataset that describes in a condensed way the dynamic behaviour of the tyre under certain working conditions (non-rolling or rolling state) and certain boundary conditions (fixed, pinned or free wheel, wheel connected to a suspension, ... etc). Most of the model is contained in three reduced matrices $\overline{\boldsymbol{K}}$, $\overline{\boldsymbol{C}}$, $\overline{\boldsymbol{M}}$, which describe the reduced dynamics of the tyre :

$$\overline{K}\,\overline{u}+\overline{C}\,\dot{\overline{u}}+\overline{M}\,\ddot{\overline{u}}=\overline{f} \quad \text{with} \quad \overline{u}=\begin{Bmatrix} a_i \\ \overline{u}_{wc} \\ \overline{u}_{cp} \end{Bmatrix}$$

Here "reduced" means that a Craig-Bampton type dynamics substructuring has been applied in order to introduce modal coordinates a_i (referring to a special modal basis) into the reduced reference vector $\overline{u}$ where

- $a_i, i = 1, nv$ refer to the modal co-ordinates of the considered eigenmodes
- $\overline{u}_{wc}$ refers to the 6 dofs at Wheel-Centre (3 translations X_{wc} , Y_{wc} , Z_{wc} and 3 rotations $\hat{x}_{wc}$, $\hat{y}_{wc}$, $\hat{z}_{wc}$)
- $\overline{u}_{cp}$ refers to the dofs considered for the Contact Patch area. By default, they are global dofs in case of non-rolling models, or "meridians" dofs in case of rolling models

A mesh for visualisation is optionally added, which allows to look at the tyre modal deformation amplitudes.

Validation and applications

This modelling has been extensively validated by comparisons with experimental measurements.

Modal Analysis

The first step of validation is the modal analysis. The model is fully predictive in the [0-200Hz] frequency range (Figure 2). It can be extended in the [0-300 Hz] frequency range if the dynamics of the wheel and the cavity are considered [9]. This last point can be of great interest for the simulation of noise inside the cabin since the first cavity resonance cavity is one of the contributors to the interior noise of passenger cars.

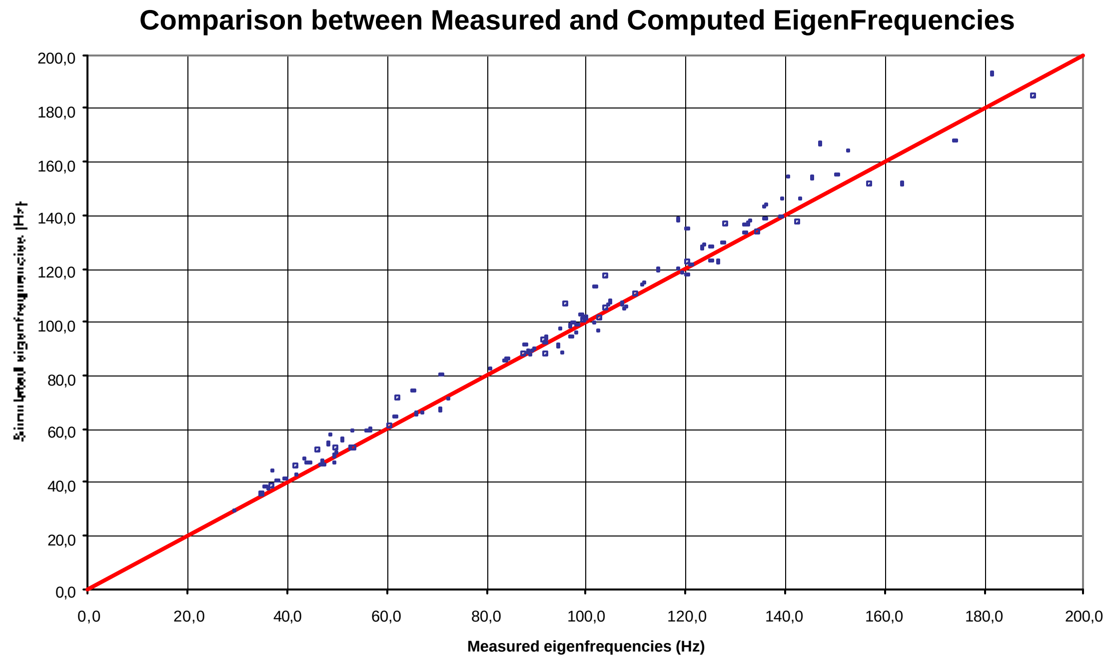

Figure 2 : Comparison between measured and computed frequencies

For the non rolling tyre

For validation of the tyre-alone models, reference measurements are dynamic stiffness transfer functions. The "dynamic stiffness" of a tyre can be defined as the transfer function between a global harmonic displacement applied on one interface (wheel centre or contact patch), and the resulting dynamic force measured on the other interface (contact patch or wheel centre). In main cases the contact patch degrees of freedom are condensed into global degrees of freedom analogously to what is done for the Wheel-Centre (Figure 3).

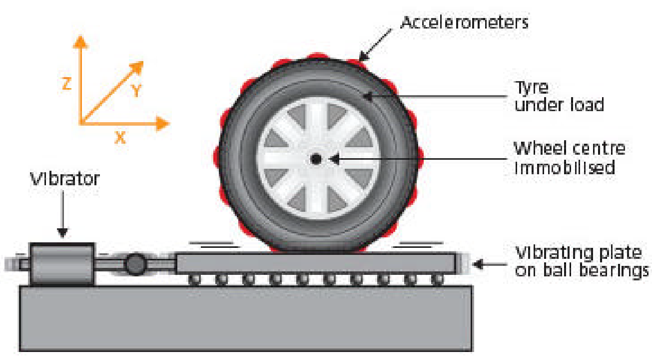

Figure 3: a dynamic shaker excites the tyre.

The measured dynamic stiffness is used as reference validation curves to insure that the tyre model gives a good representation of the reality (see example on Figure 4 in the X direction).

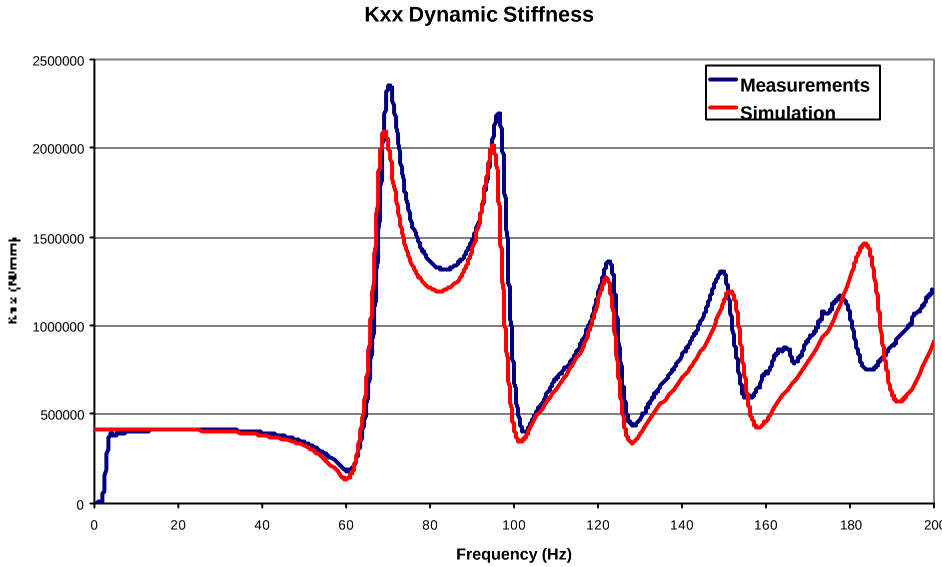

Figure 4 : tyre longitudinal dynamic stiffness.

Such a non-rolling model can be used to study the modal behaviour of a standing vehicle, or part of it (suspension, etc…). In this use the whole structure can be excited thanks to a shaker through the contact patch of the tyre.

For the rolling tyre

For rolling tyre models, the validation test is the classical cleat test (Figure 5). A tyre pinned on a rotating dynamometric hub is deflected on a drum, which supports a cleat. The tyre is lead to roll when the drum rotates, and it impacts the cleat at each revolution. The speed is controlled, and the forces at the hub are time-recorded for every impact (see example on Figure 7 for a 5x20mm cleat at 30kph). Rolling tyre models may be coupled with vehicle models to allow rolling vehicle simulations of dynamics (hardness, harshness, ... etc).

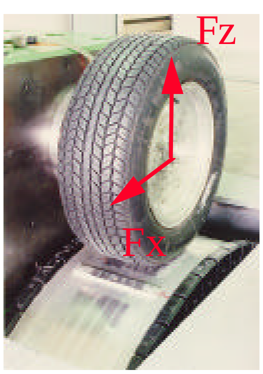

Figure 5 : tyre cleat test.

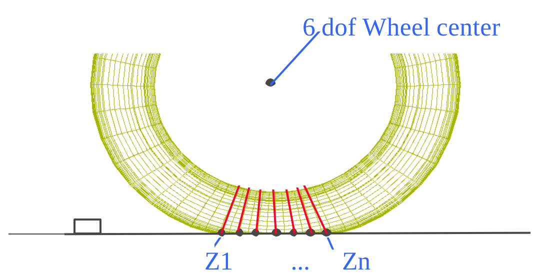

Figure 6 : vertical displacement of the meridians.

For the simulation of these tests, the Contact Patch area is most of the time described by "meridians" (Figure 6). It is then possible to simulate the rolling of the tyre over any kind of obstacle (low frequency ground profiles, cleats, ... etc) by passing their profile under the tyre.

Each meridian is raised up and down as the obstacle passes progressively under it, seeing a constant geometry in the lateral direction Y. A 3D contact patch can also be used for the simulation of a macro-rough surface excitation.

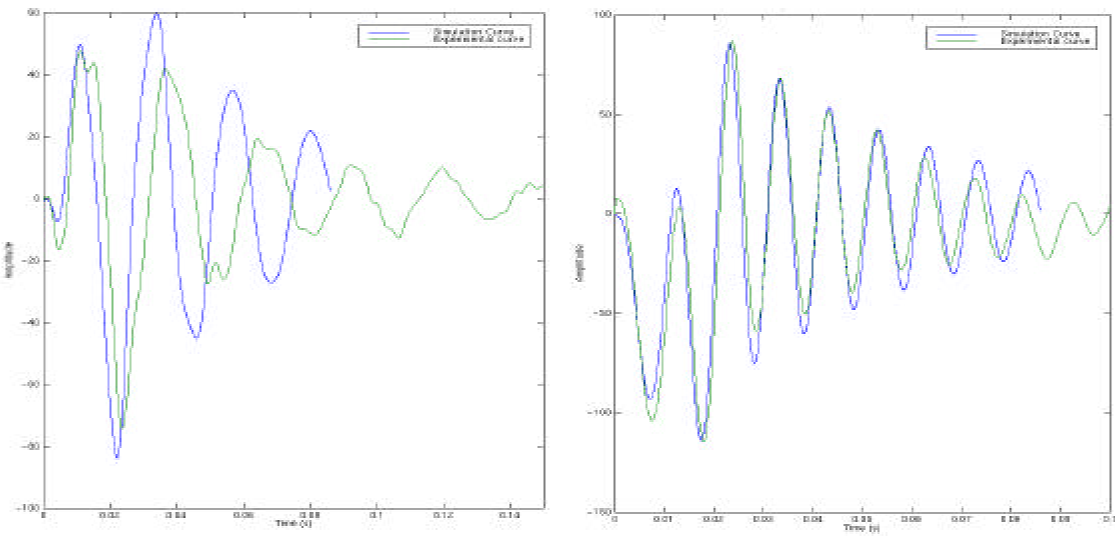

Figure 7 : Fx longitudinal force and Fz, vertical force, at the hub centre versus time at 30kph (experiment and simulation).

The structure of the model permits easy coupling with FE models of axle or vehicle. The Figure 8 shows an example of cleat test simulation performed at 30kph on a 10x20mm cleat with a twist-beam axle with MSC.NASTRAN.

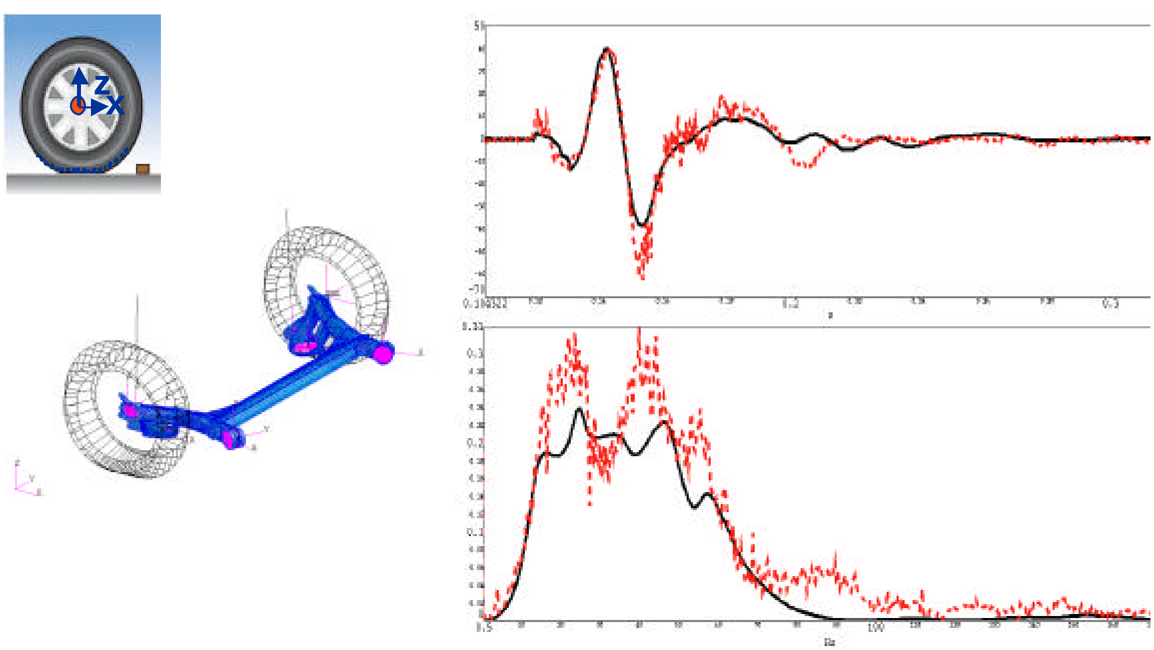

Figure 8 : MSC.NASTRAN simulation of cleat test, comparison with experiments, acceleration at the wheel centre (upper curve: X , lower curve: Z)

ADAMS Application

A formulation closely linked to the Craig-Bampton formulation is supported by MSC.ADAMS through the use of flexible bodies. It was therefore very tempting to introduce the MICHELIN modal model into ADAMS using the Flexible Body approach. A toolkit has therefore been developed by MSC and MICHELIN to translate the MICHELIN modal models into mnf files (modal neutral file) which is the file format supported by the MSC.ADAMS/Flex module. The format of the modal model input file for the translator has been defined following the TIME ORBIT requirements in order to allow anybody willing to use the same format to do so.

Below are described several applications of a tyre Flex model in ADAMS/View and ADAMS/Car.

Modal analysis

As for the tyre alone validation, the first step is to perform a modal analysis of the system using ADAMS/Linear functionality.

The example of Figure 9 shows a simple ¼ vehicle model with a flex-tyre. The mode described is the 18th mode of the system corresponding to the 5th radial non-symmetric mode of the tyre appearing around 190Hz. In this plot, a crown has been added to the tyre description in order to easily visualise the mode shapes of the tyre. This crown has no effect on the mechanical behaviour of the system.

As the modal description of the tyre is the exact copy of the modal basis derived from the FE Analysis, there is no loss of information and the flex-tyre describes exactly the modal behaviour of the tyre.

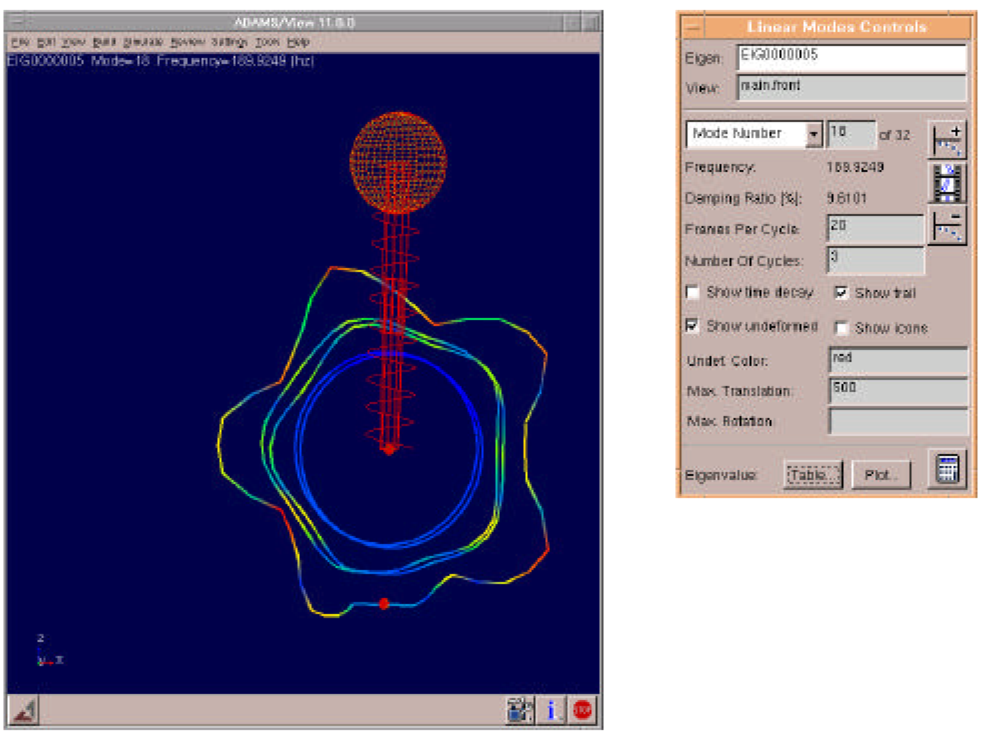

Figure 9 : Radial mode of a loaded tyre at 190Hz with linear analysis in MSC.ADAMS

For the non rolling Tyre

On Figure 10 are plotted the comparison of Dynamic Stiffness simulation made in ADAMS and using home-made code. The ADAMS plot is in blue and the other plots corresponds to several damping definitions.

The agreement is perfect both for the "static" part of the curve (below 30Hz) and for the "modal" part of the curve (above 30Hz). It shows the ability of the modal tyre in ADAMS to accurately reproduce the forces at the hub centre for a given excitation in the contact patch.

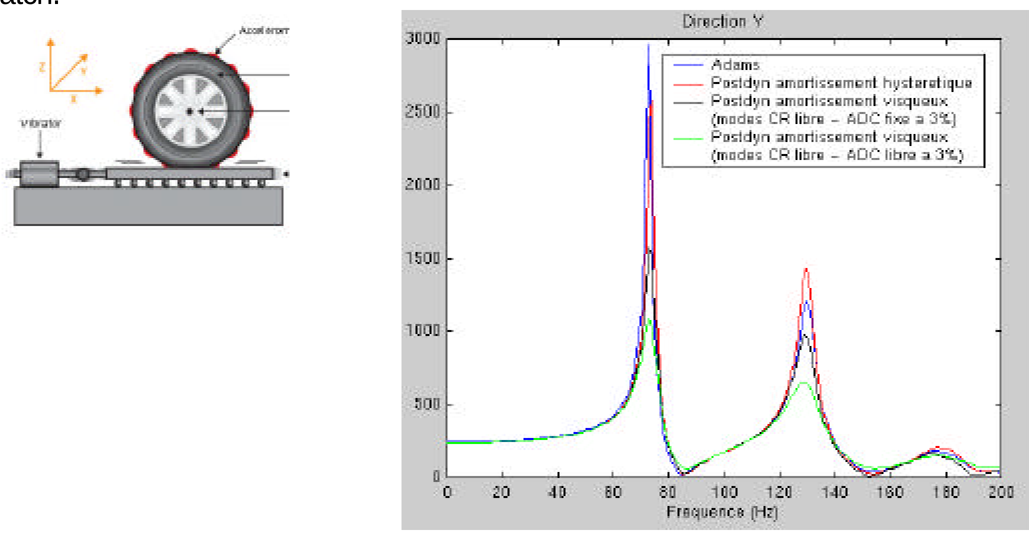

Figure 10 : ADAMS "dynamic stiffness" and home-made code "dynamic stiffness"

This kind of simulation can be extended to a full vehicle simulation where each contact patch of the vehicle's tyres is excited by a given signal in the time or frequency domain.

For the rolling Tyre

The tyre model can of course be coupled with an axle model. The Figure 11 shows the excellent agreement for ADAMS and NASTRAN simulation of a 10x20mm rectangular cleat test at 40kph. In this simulation, the tyre is coupled with a twist-beam axle model both in ADAMS and NASTRAN.

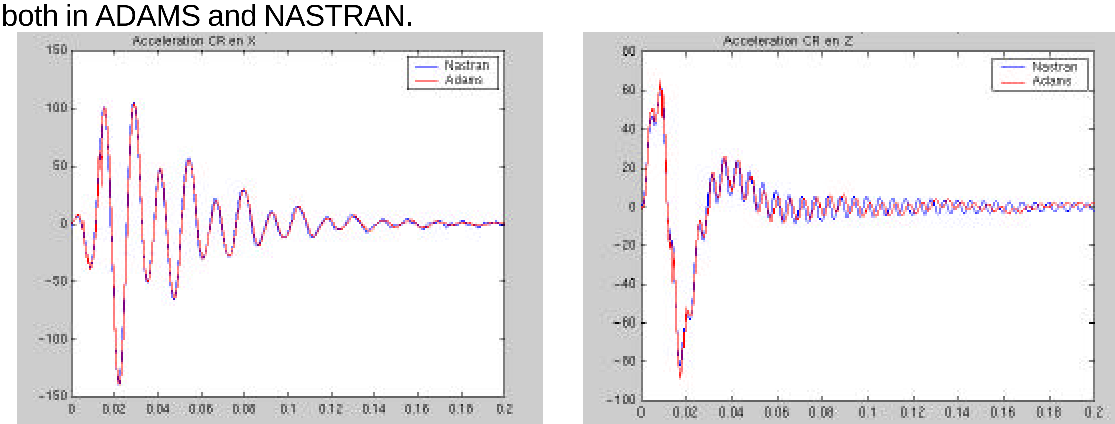

Figure 11 : Comparison of acceleration time record at the hub centre during a cleat test between MSC.ADAMS and MSC.NASTRAN (left: X, right: Z)

Full Vehicle analysis in ADAMS/Car

The tyre model is being introduced in ADAMS/CAR as a standard internal tool.
The aim is to perform standard NVH simulation such as rolling over various obstacles (Figure 12) or standing on shakers with a given time and/or frequency excitation of the contact patch.

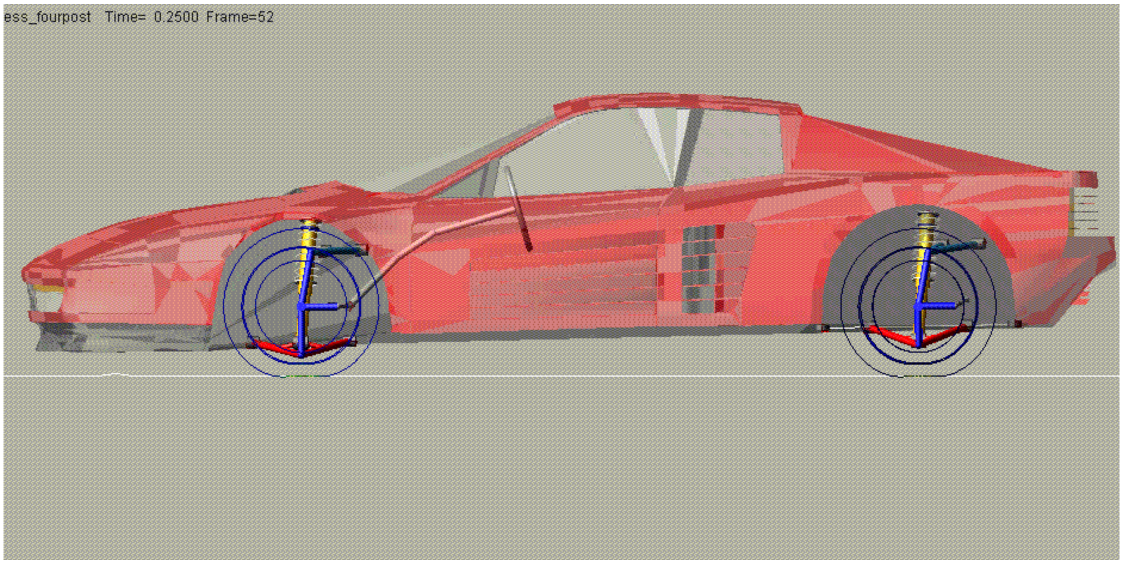

Figure 12 : The tyre modal model in ADAMS/CAR

The Figure 13 describes a cleat test performed in ADAMS/CAR for a full vehicle model. Requests have been made to get the displacement signal in the contact patch and the corresponding displacement at the wheel centre for the front and rear tyres. The results look reasonable and the comparison with experimental measurements are going on.

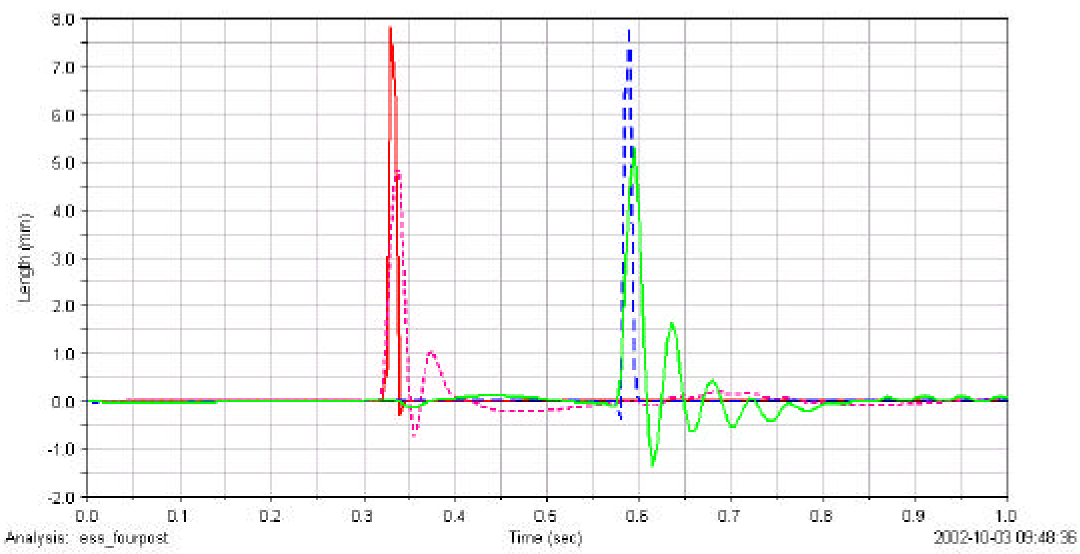

Figure 13 : Displacements at the contact patch and wheel centre (front and rear axles)

Conclusion

For NVH studies, MICHELIN has been intensively using modal models based on the Craig-Bampton substructuring method for many years. Those models are derived from internal non-linear FE analysis and are fully predictive for non-rolling and rolling conditions in the [0-200Hz] frequency range.

With the collaboration of MSC, the MICHELIN modal model has been translated into a Flexible Body in order to be introduced into ADAMS.

The use of flexible bodies to simulate the tyre dynamic behaviour permits to easily simulate the NVH behaviour of a full vehicle in ADAMS and is particularly suitable to take advantage of the other NVH-related ADAMS modules such as the linear analysis and the ADAMS/Vibration modules.

The results of flex-tyre simulations have been compared with FEA simulations for non-rolling and rolling tyres coupled with vehicle models and showed a perfect agreement.

Full vehicle simulations in ADAMS/Car have now to be compared with experimental results but are likely to accurately represent the tyre behaviour since using the flex approach implies having no loss in the dynamic description of the component.

Acknowledgement

This work has been conducted in collaboration with MSC/France.

Reference

[1] R.R. Jr. Craig and M.C.C. Bampton, « **Coupling of Substructures for Dynamic Analysis** », *AIAA Journal*, Vol 6, July 1968

[2] R. MacNeal, « **A Hybrid method of component mode synthesis** », *Computers and Structures*, Vol 1, 1971

[3] Sundaram S., Hohman R., Richards T, « **Vibration modes of a tire using MSC/NASTRAN** », *User's conference*, March 1985

[4] Richards T., Charek, Scavuzo R., « **The effects of spindle and Patch Boundary conditions on Tire vibration modes** » *SAE Paper*, Feb 1987

[5] AC. Deneuvy, « **Modal analysis of a pneumatic Tyre and dynamic simulation by a sub-structuring method** », *Proceedings 7th IMAC*, Las Vegas 1989.

[6] E.P. Vinesse, P. Peruzzetto, « **Simulations of Tire Dynamics Over Small Obstacles Using Virtual Profiles and a Sub-structuring Method** », *Proceedings of IPC-6*, Seoul 1991.

[7] E.P. Vinesse, Masahiro Imai, « **Tire-Suspension Coupled Vibration Model** », *JSAE symposium "Tire performance Theory and practice"*, July 1992.

[8] E.P. Vinesse, « **Sub-structuring method and eigen function representation for a rolling tyre coupled to a secondary mechanical system** », *Proceedings of ImechE* 1996, Vol 210.

[9] W.B. Cayton, R. Saint-Cyr, « **Incorporation of the Tire Air Cavity Resonance into The Modal Tire Model** », *Proceedings IMAC XVI*, January 1998.

[10] D. Tsihlas, T. Lacroix, B. Clayton, « **A Comparison of Two Sub-Structuring Techniques for Representing the Modal Properties of Tires** », *Tire Science and Technology*, TSTCA, Vol. 29, N°1, Jan-March 2001, pp.23-43

[11] ***ADAMS/Flex*** Reference Manual