Comfort study of railcar A-TER X 73500 - Simulation with ADAMS/Rail

Figure 1 : Maquette of the railcar A-TER X 73500..

I) Introduction

Whether in public transports or private transports, people display an ever increasing inclination for enhanced vibrational, acoustic visual and thermal comfort. In the rolling stock industry, this trend lead to new researches in the field of vibrational comfort, all the more acute because of the new requisites featured by modern vehicles :

- weight reduction allowing to reduce operating costs in terms of energy consumption and fixed installation maintenance costs,
- enhanced light-levels and roominess of coach compartments,
- easy access.

Within the framework of the development of the new A-TER X 73500 railcar for the German and French Railways, these requisites led to a number of design choices :

- Using aluminum, a lightweight material, for the bodyshell passenger compartment. The complex driver's cabs are made of steel and bolted on the aluminum passenger compartment,
- Large-size windows and access doors,
- Powerplants located underneath the driver's cabs in order to keep the passenger compartment as large as possible and at a height of 500 mm from the upper surface of the rail.

These new requisites induce a noticeable reduction in stiffness as regards the bodyshell. In terms of vibrational comfort, they no longer allow to apply the usual rules which recommended bodyshell modes with high frequencies, that were indeed much higher than suspension modes.

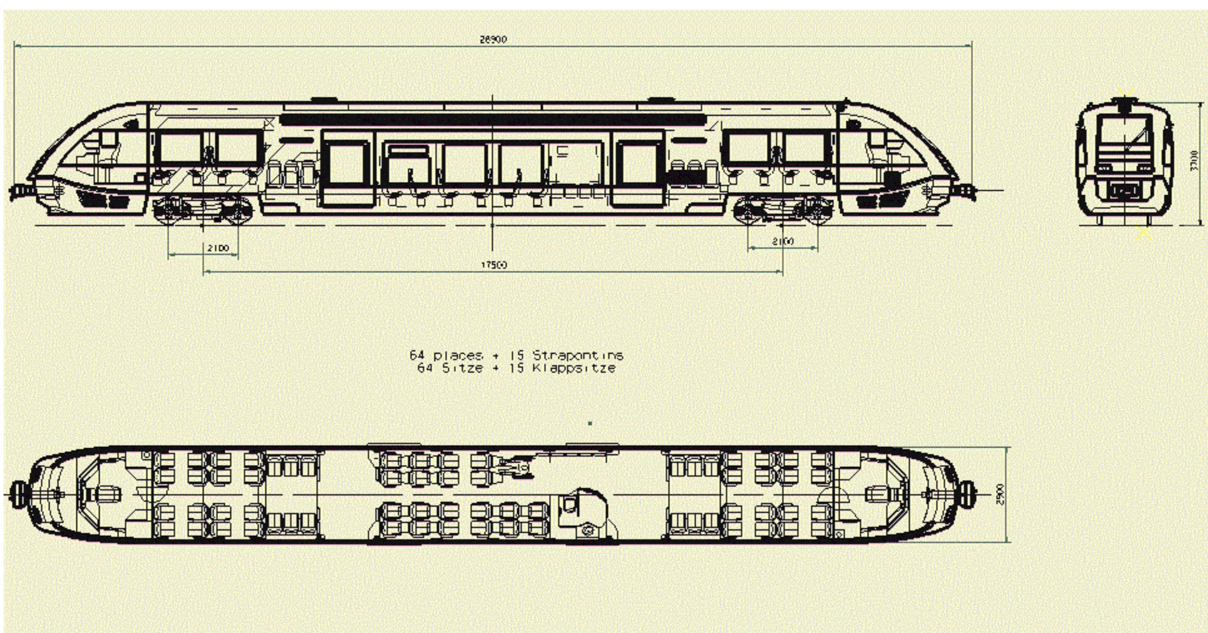


Figure 2 : The A-TER X 73500.

On the A-TER X 73500 this is even worsened by the heavy weights at both bodyshell ends which reduce the bending mode frequencies.

After performing bodyshell structural calculations on the basis of maximum admissible stress and static camber criteria, a bodyshell modal analysis was carried out. These calculations demonstrated indeed that there were bodyshell mode frequencies, lower than 10 Hz, that were close to suspension mode frequencies, especially for the first vertical bending mode which is usually one of the important factors as regards the vertical comfort of a railway vehicle.

However, even if the elastic line of a bodyshell mode is known precisely, knowing whether this mode will be excited by a given bogie mode and above all defining its level of response is quite difficult.

Moreover, the coupling channels are bodyshell-to-bogie connection parameters that can be optimized.

Therefore, it was decided to investigate the comprehensive dynamic behavior of the A-TER railcar running on a reference track under ADAMS/RAIL in order to check the excitability of bodyshell modes and there influence on the bodyshell vibrational level using numerical simulations.

First, the study concentrated on the excitability of the first two vertical bending modes, whose low frequencies are close to the suspension modes, since these seem to be the most damaging to vertical comfort, which is an essential component of overall comfort.

Second, the global vibrational response of the bodyshell, taking account of the whole modes set of the calculated modal basis, was studied according to two successive ways :

a) creep forces on wheel/rail contact not taken into account (« level I »),
b) creep forces on wheel/rail contact taken into account according to the Kalker linear theory (« level IIa »).

II) A-TER X 73500 model

The theoretical model (refer to fig. 3 hereunder) compiled using ADAMS/Rail is comprised of the following :

- the flexible bodyshell with its internal damping,
- the traction units attached on each bodyshell end via flexible pads,
- both bogies comprised of rigid bodies and flexible connections with detailed models of the connections to the bodyshell in order to perfectly feed back the efforts transmitted,
- two bolsters (modeled by rigid bodies) bolted underneath the bodyshell providing all the connections with the bogie frame,
- the track, characterized by its geometrical defects (longitudinal and transverse leveling defects, track straightening defects).

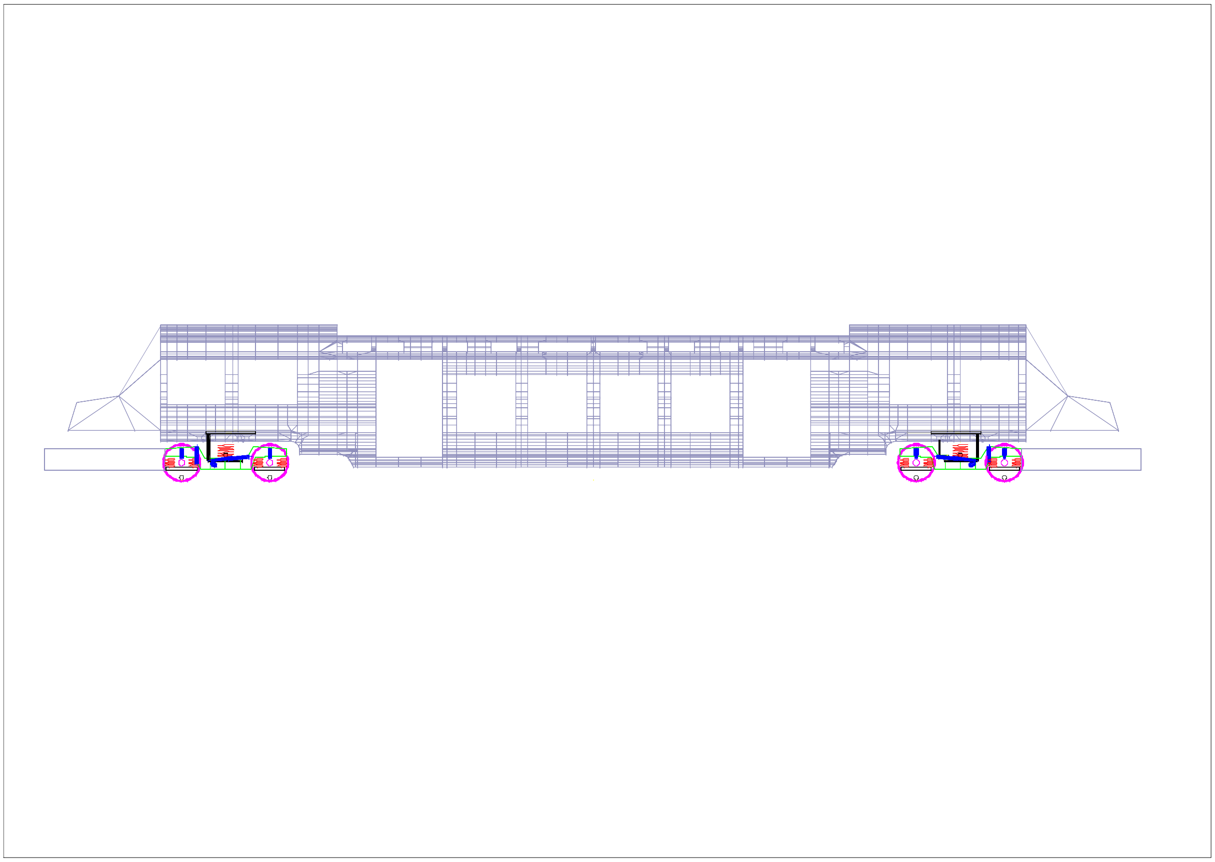


Figure 3 : A-TER X 73500 model compiled using ADAMS / Rail.

➢ *Flexible bodyshell :*

The first step consisted in compiling a finite element model of the bodyshell as a whole (aluminum passenger compartment and steel driver's cabs) that is considered as a flexible item under ANSYS. The aluminum passenger compartment was modeled using shell elements (shell 43), the driver's cab was modeled using beam elements (beam 4), inside lining and miscellaneous items of equipment were integrated as lump masses (mass 21) arranged non-symmetrically. The actual arrangement of these masses was complied with in order to achieve realistic modal elastic lines. The stiffness induced by items of lining is disregarded.

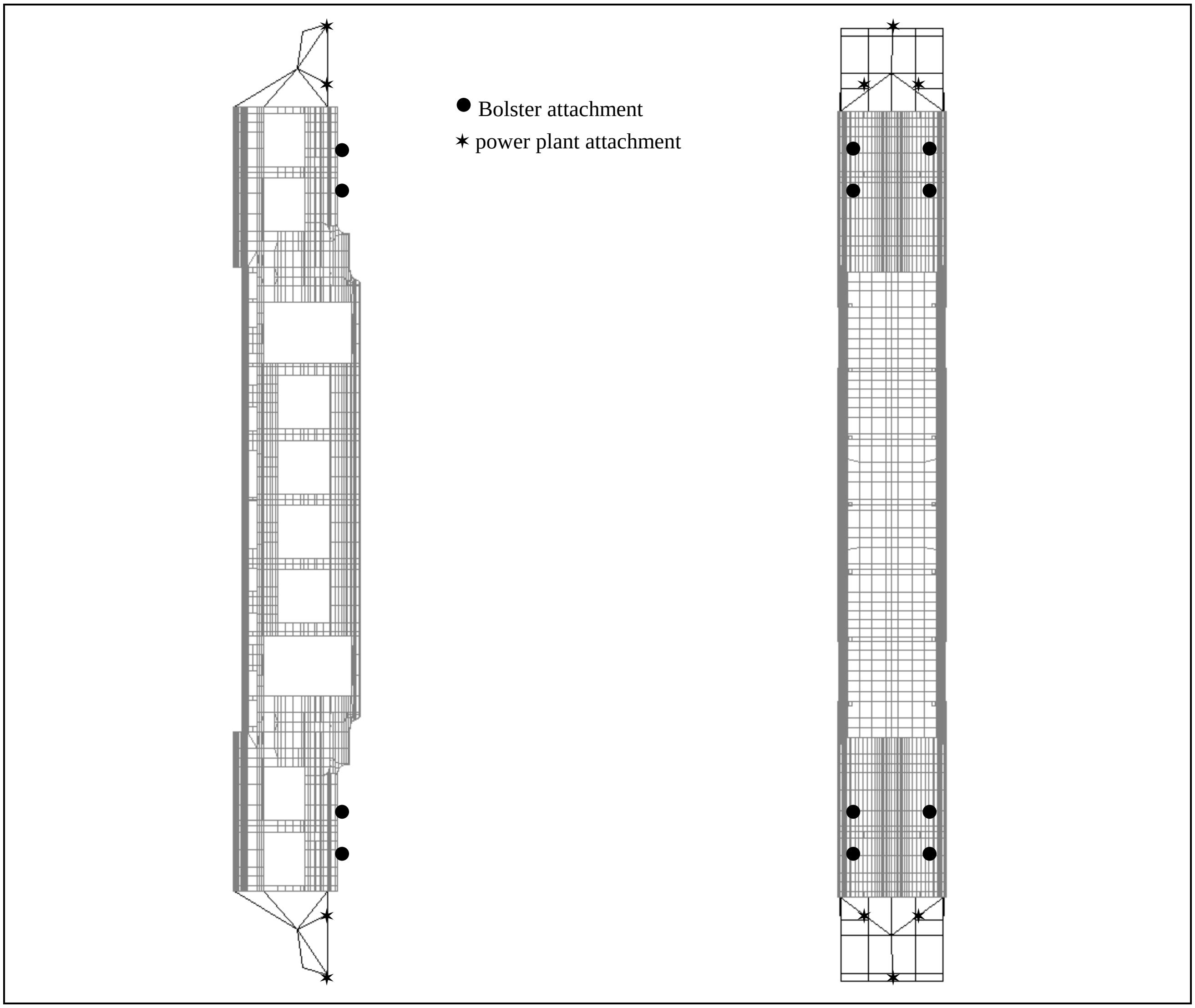


Figure 4 : ANSYS bodyshell finite element model.

Bodyshell components	Finite element	Number of finite elements
Aluminum passenger compt.	SHELL 43	18798
2 steel driver's cabs	BEAM 4	666
Lining, glass windows, access doors	Nodal mass (MASS 21)	2903

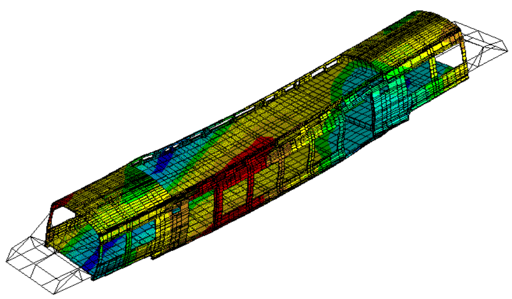
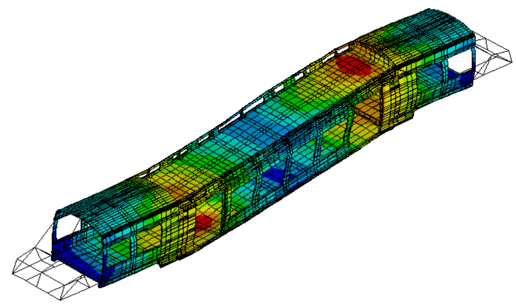
On the whole the finite element model comprises 14136 nodes and 22367 elements.

The weight of the bodyshell, i.e. the aluminum passenger compartment, both steel driver's cabs, items of lining and equipment amounts to:

M = 32000 kg

On the bodyshell described herein (i.e. no bogies, no engines and no bolsters), a free-free modal analysis gives the following results for the first two vertical bending modes.

Figure 5 : First two bending modes of the bodyshell.

1		**2**	

The table hereafter compares the results (frequencies) achieved via this modal analysis with those achieved using the modal basis generation method used by the ADAMS software (Mode Component Synthesis Analysis, MCSA) or Craig - Bampton method.

Tableau 6 : Eigenfrequencies of the first two bending modes of the bodyshell.

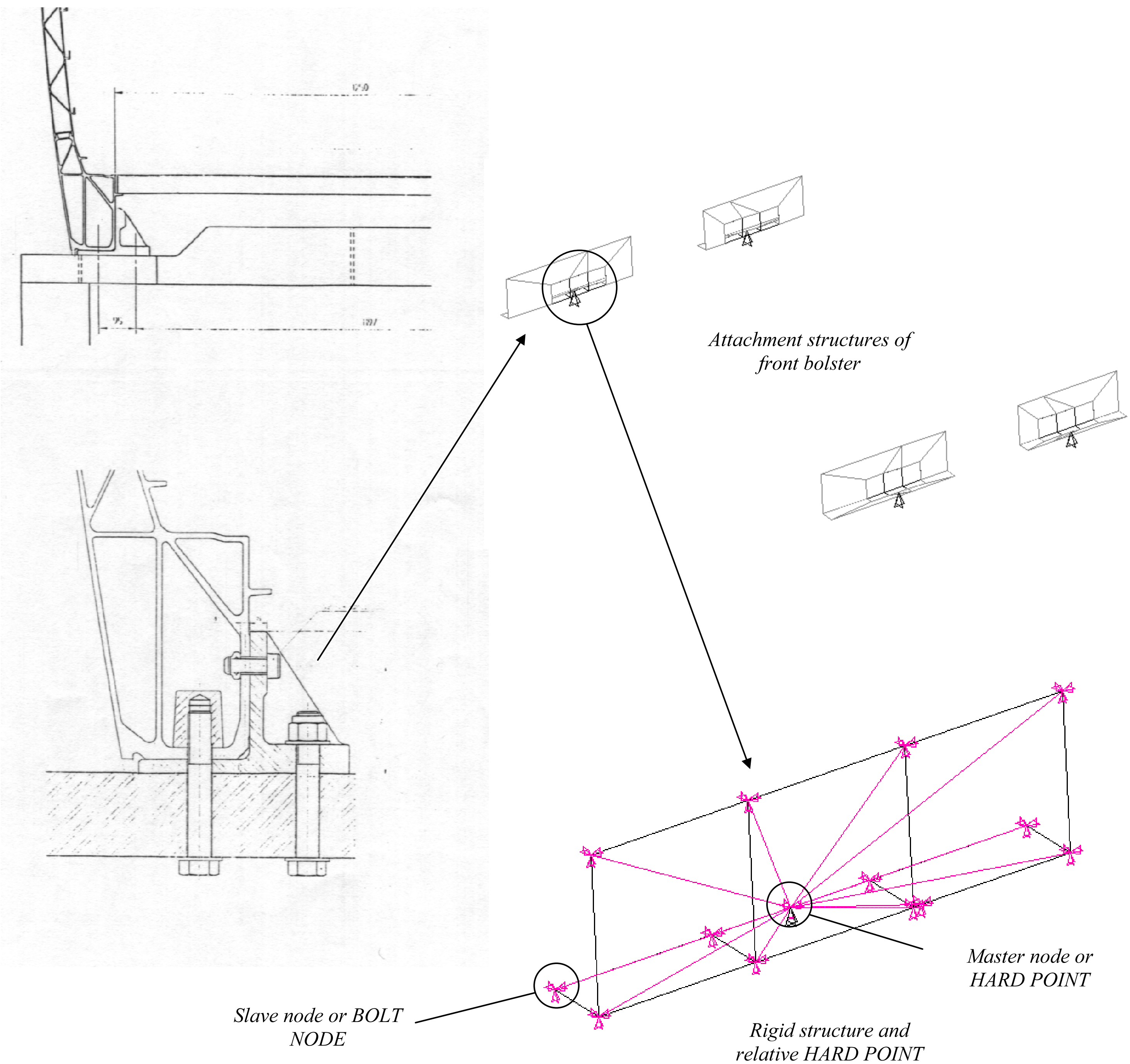
Mode	Type	ANSYS (SUBSPACE)	MCSA
1	1st vertical bending mode + slight twist	7.6 Hz	7.6 Hz
2	2nd vertical bending mode	10.7 Hz	11.0 Hz

➣ *Connections between the flexible bodyshell and the solid bodies :*

The connection between the bodyshell and the powerplant is achieved via three attachment points underneath the driver’s cabs.
The connection between the bodyshell and the bolster is achieved via four attachment points.

Therefore, the bodyshell model features 14 interface points.

The areas in which the bogie bolsters are attached to the bodyshell (each of the two bolsters is secured to the bodyshell by 4 series of 6 bolts) were stiffened on the finite element model so as to excite the flexible bodyshell realistically.



When using the modal basis generation method of Craig-Bampton (see further), the way interfaces are modeled greatly influences the modes of the modal bases compiled. As a matter of fact, the areas in which the bolsters are attached to the bodyshell had to be stiffened in order to obtain the same elastic lines and the same eigenfrequencies for the first six modes as those determined when using the free-free modal analysis of the bodyshell.

III) The Craig-Bampton method

➢ *Theory :*

For a structure consisting of 20000 nodes comprising of elements with 6 degrees of freedom per node, the structure stiffness matrix [K] and mass matrix [M] feature 120000 lines and 120000 columns. Solving the following linear differential system based upon the theorem of virtual powers:

$$[M]\{\ddot{Q}\}+[K]\{Q\}=\{F_{ext}\}$$

where :

- [M] stands for the masses of the mechanical assembly (E),
- [K] stands for the stiffnesses of the mechanical assembly (E),
- {Q}stands for the column vector of the node displacement,
- $\{F_{ext}\}$ stands for the column vector of the efforts applied onto the nodes of (E),

in order to define the dynamic motions of the flexible body requires a calculation power and duration that is incompatible with industrial requirements. Hence, the dynamic calculation algorithm under ADAMS uses flexible elements in the form of a modal basis. The modal basis computed with ANSYS via a "Mode Component Synthesis"-type method using the procedure set up by Roy R. Craig Jr. and Mervin C. C. Bampton (1) in 1968 was transferred to ADAMS. In the sequel, constraint modes or static correction modes and retained normal modes were orthogonalized.

Note : (1) Craig, R. R. and Bampton, M. C. C., "Coupling of Substructures for Dynamics Analyses', American Institute of Aeronautics and Astronautics Journal, vol. 6. n°7, 1968, pp. 1313 - 1319.

The system studied under ADAMS contains no more than 160 degrees of freedom compared to approximately 100000 for a dynamical calculation performed using a conventional finite element method that does not allow to select the degrees of freedom, i.e. to reduce the size of matrices [K] and [M]. Solving the 160 equations is a task that the dynamic calculation software (modified GEAR's algorithm) can fulfill easily.

The multi-body calculation code used by ADAMS analyses the movements of material systems by integrating an equation system of movements given in the Euler-Lagrange format, i.e. differential algebraic equations of the second degree.

Bodies with flexible properties are entered this way. The approach used to describe the flexible behavior is a modal approach (i.e. work based upon a modal basis). Deformations are estimated via a discrete displacement function obtained by multiplying the modal matrix [ϕ] by the modal coordinates q. The position of body X is estimated by coupling the small elastic deformations δ with the general movement of the body considered as being rigid x :

$$\{X\}=\{x\}.\{\delta\} \quad \text{with} \quad \{\delta\}=[\Phi].\{q\} \qquad (1)$$

The flexible body equations of Euler-Lagrange are given by format (2) where L stands for the lagrangian (L = T - V, where T is the kinetic energy and V the potential energy) F is the dissipation energy, ψ the connection equations, λ the Lagrange multipliers for the connections, ξ the generalized coordinates of the flexible body, q_k the modal coordinates of the flexible body at number n and m the number of connections.

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\xi}_i}\right)-\frac{\partial L}{\partial \xi_i}+\frac{\partial F}{\partial \dot{\xi}_i}+\sum_{j=1}^{m}\left[\frac{\partial \Psi_j}{\partial \xi_i}\right]^T \lambda_j - Q_i = 0$$

$$\Psi_j = 0$$

with

$$i = 1,\dots,n+6$$

$$j = 1,\dots,m$$

$$\text{and} \quad \xi = \left\{\begin{bmatrix} x \\ y \\ z \\ \psi \\ \vartheta \\ \varphi \\ q_k \end{bmatrix}\right\} \qquad (2)$$

The difficulty with the aforementioned method lays in the capability to choose the best series of generalized modal coordinates, i.e. the modal shapes to be considered when modeling the behavior of the flexible body.

The ADAMS software uses a method of modal synthesis based upon the method developed by Craig and Bampton. Such a method allows to reduce the generalized coordinates to the smallest number possible on the one hand and offers greater freedom when defining the boundary conditions on border points (e.g. conditions of connection or application of a force) describing the effects of local flexibility.

The movement of the flexible body, characterized by N degrees of freedom and a known number of border points (or interfaces) is described by the combination of P normal modes and S static correction modes. The first modes are determined by performing a modal analysis of the body while considering the degrees of freedom of the interfaces as constrained whereas the second modes are merely the S static elastic lines achieved by causing each interface degree of freedom to be displaced individually while maintaining the others fixed.

This allows to compile a $[\phi^c]$ (SxS) matrix for constraint modes or static correction modes and a $[\phi^N]$ (RxR) matrix for normal modes (also called fixed interface normal modes or constrained normal modes). From the latter solely a truncated system of P(<R) modes is retained in order to reduce calculation times and the required computer memory space depending on the highest frequencies to be analyzed.

When writing physical coordinates {δ} as a modal addition and changing coordinates in compliance with formula (3), the result is equation (4) where *p* stands for the new coordinate system and where the mass and stiffness matrices are said to be generalized.

$$\{\delta\} = \begin{Bmatrix} \delta^B \\ \delta^I \end{Bmatrix} = \begin{bmatrix} [I] & [0] \\ [\phi^C] & [\phi^N] \end{bmatrix} \begin{Bmatrix} p^B \\ P^I \end{Bmatrix} = [\phi]\{p\} \qquad (3)$$

$$[\overline{M}]\{\ddot{p}\} + [\overline{K}]\{p\} = \begin{bmatrix} \overline{m}^{BB} & \overline{m}^{BN} \\ \overline{m}^{NB} & \overline{m}^{NN} \end{bmatrix} \begin{Bmatrix} \ddot{p}^B \\ \ddot{p}^I \end{Bmatrix} + \begin{bmatrix} \overline{k}^{BB} & 0 \\ 0 & \overline{k}^{NN} \end{bmatrix} \begin{Bmatrix} p^B \\ p^I \end{Bmatrix} = \begin{Bmatrix} \bar{f}^B \\ \bar{f}^I \end{Bmatrix} = \{\bar{f}\} \qquad (4)$$

Attention should be paid to the noticeable reduction in size of the problem in terms of total degrees of freedom, which from N are cut down to P + S. The solver of the ADAMS software determines modal coordinates *p* using (2) and computes deformations {δ} from (3).

In order to manage the flexible body more easily than in the original Craig-Bampton method, the ADAMS software orthogonalizes the modes of the reduced system described in (4). This last operation leads to generalized mass and diagonal stiffness matrices which show the behavior of the free flexible body while additionally accounting for the flexible deformability of the border points. Prior to orthogonalizing, modes consist of fixed interface retained normal modes (or retained normal modes) and constraint modes (or static correction modes). Subsequent to orthogonalizing modes are close to free body normal modes and interface modes. The three advantages of orthogonalization read as follows:

1. all modes have an associated frequency
2. high frequency modes may be deactivated with an effect of lesser importance on stresses
3. rigid modes (which are part of free body normal modes) may be easily determined and deactivated.

The flexible deformation of border points is no longer visible on interface modes, but reappears on high frequency modes generated by orthogonalization.

➢ *Validity assessment of this last method :*

ALSTOM DDF, in conjunction with ENSAIS (Engineering School of Strasbourg), initiated a study aimed at evaluating the assets and the validity of the Craig-Bampton method applied to railway vehicles.

IV) Simulation calculations

5.1) Study of the response of the first two vertical bending modes

➢ *Objective :*

The main goal of the study consisted in highlighting the influence of a certain number of parameters regarding the excitability of the first two vertical bending modes of the bodyshell (refer to fig. 7 hereunder) whose low frequency is close to the suspension modes:

- traction unit flexible support vertical stiffness
- bodyshell-to-bogie drive rod stiffness
- secondary suspension vertical stiffness
- secondary suspension damping

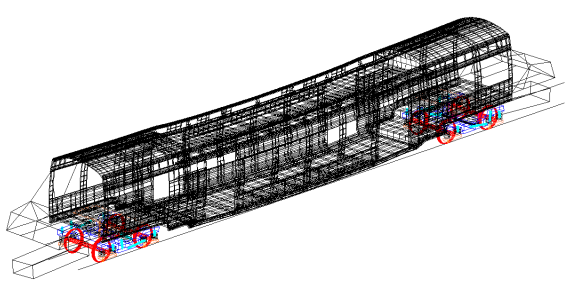
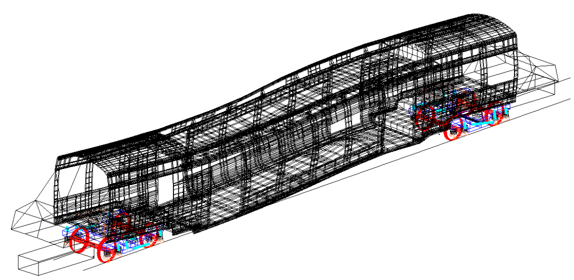
	
1st vertical bending mode on the system as a whole {bodyshell + bogies + bolster + engines}	**2nd vertical bending mode on the system as a whole {bodyshell + bogies + bolster + engines}.**

Figure 7

In order to meet this goal, a comprehensive design of experiment comprising several variables and several levels was designed and run under ADAMS/Rail. Analyzing the results allowed to study the dynamical behavior of the vehicle and define the optimum parameters so as to curtail the vibrational levels on the bodyshell.

➢ Hypotheses and calculation conditions :

a) Track

The simulation calculation for a vehicle running on a straight track were performed using the vehicle model described earlier. The geometrical defects of the track are defined by ERRI B 176 in the form of PSD:

Defect type	Formula
Longitudinal leveling	$S_{zz} = (A_v*\Omega_c^2)^2/((\Omega^2+\Omega_r^2)*(\Omega^2+\Omega_c^2))$ $[m^2/(rad/m)]$
Straightness	$S_{yy} = (A_a*\Omega_c^2)^2/((\Omega^2+\Omega_r^2)*(\Omega^2+\Omega_c^2))$ $[m^2/(rad/m)]$
Roll	$S_{\varphi\varphi} = ((A_v*\Omega_c^2*\Omega^2)^2/(\Omega^2+\Omega_r^2)*(\Omega^2+\Omega_c^2)*(\Omega^2+\Omega_s^2)))/f^2$ $[m^2/(rad/m)]$

where: $\Omega_c = 0.8246$ rad / m
$\Omega_r = 0.0206$ rad / m
$\Omega_s = 0.4380$ rad / m
$A_v = 1.080.10^{-6}$ m*rad
$A_a = 6.125.10^{-7}$ m*rad

These track is entered into ADAMS via its spatial coordinates compiled from the PSD by introducing a random phase and by accounting for defects whose wavelength ranges between 2 m and 100 m (2 m < λ < 100 m).

b) Damping ratio

Using ADAMS to model a flexible body allows a posteriori to define a damping rate for each of the mode of the modal basis.

As regards the bodyshell, the following values were selected :
-3.5 % for the first two vertical bending modes,
-100 % for all other modes.

This was done in order to study the excitability of the bending modes directly.

c) Interfaces between the track and the wheelsets

The model of the wheel-rail contact was compiled in level 1 with ADAMS/Rail. This boils down to defining a degree of freedom in yaw for each wheelset. In this case, no pseudo-slide effort at the contact point is defined.

Vertical, transverse and roll displacements are transmitted from the track to the wheelset via a pivot connection located in the center of the wheelset following the average geometrical profile of the track.

This model type remains nevertheless adapted to the studies of vertical comfort and more particularly to the study of bending mode excitability. Furthermore, it reduces the calculation time compared to levels including pseudo-slide efforts.

d) Calculation parameters

Speed: 38.89 m / s (140 kph, maximum speed of A-TER X 73500 on SNCF network)
Analysis duration (end time): 20 s
Time intervals: 0.01 s approximately (2050 steps)
Integrator: WSTIFF (H_{max} = 0.01)

Remark :

When modifying the number of sample points, no modification in the contents of the frequency spectrum is noted. Hence, there are no spurious lines or interferences.

e) Parameters studied in DOE, levels

Parameter	Appellation	Levels
Secondary suspension stiffness z	RSSZ	2.5E+05 N/m 4.0E+05 N/m 5.5E+05 N/m
Engine support stiffness z	RPMZ	1.0E+05 N/m 6.0E+05 N/m 1.2E+06 N/m
Drive stiffness along x, y and z axes	REXYZ	2.5E+06 N/m 5.0E+06 N/m 8.0E+06 N/m
Secondary suspension damping	SV	2.0E+04 N/(m/s) 6.0E+04 N/(m/s)

Figure 8: Parameters studied, levels.

RSSZ, RPMZ, REXYZ are three-level variables, SV a two-level variable. Hence, the number of simulation calculations run with ADAMS/RAIL, which amounts to the number of possible parameter combinations reads 54 (i.e. 3 x 3 x 3 x 2).

f) Results studied:

The areas finally investigated read as follows:

- Rear driver's cab center area ➠ Point I
- Rear bogie area ➠ Point II
- Bodyshell bottom center area ➠ Point III

For each of these areas and each of the possible parameter combinations of the design of experiment, the following data were analyzed :

- Power Spectral Density of the weighted vertical acceleration (weighted according to UIC leaflet 513 so as to account for the differential sensitivity of people depending on the frequency). These data allowed to perform a frequency analysis of the bodyshell response and to define modes that feature a response.

- Weighted RMS values for the corresponding vertical acceleration. These data quantifies the energy level of a response. It was processed statistically by calculating the mean effects and the interactions of the 4 variables of the design of experiment for each of the 3 areas.

➢ *DOE Conclusions :*

These investigations allowed to better understand the vibrational behavior of a flexible bodyshell when running and to define the coupling channels with the bogie. Moreover, several facts and the related behaviors could be highlighted, e.g. :

- the importance of the secondary vertical dampers which are offset with respect to the bogie centerline. These secondary dampers induce a coupling between the bogie pitch mode and the bodyshell bending mode. Figures 9 and 10 show the PSD as regards vertical acceleration in the bodyshell center area for two different damping values.

Figure 9 : PSD in bodyshell center area, first damping setting of vertical dampers.

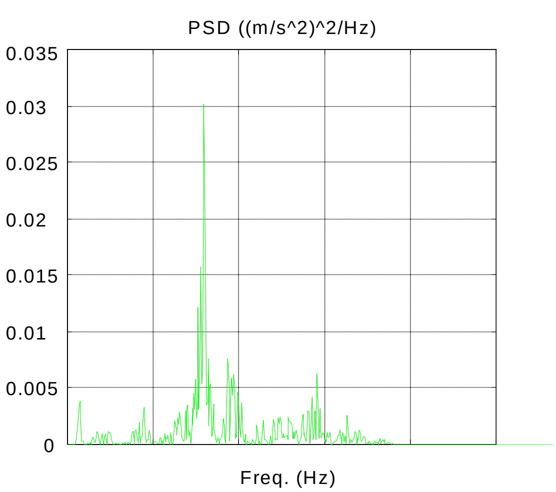
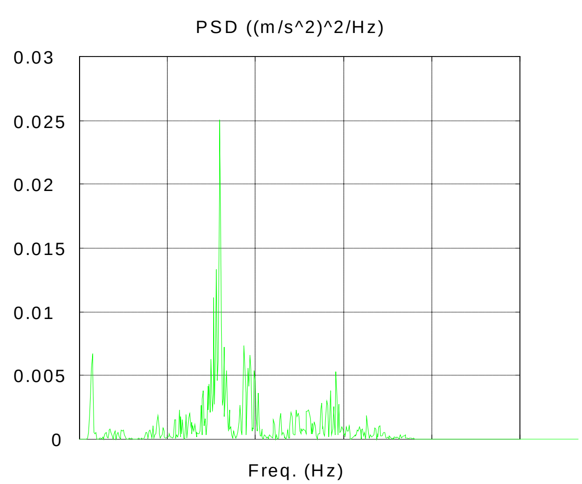


Figure 10 : PSD in bodyshell center area, second damping setting of vertical dampers.



- The vertical stiffness of the power plant flexible supports. Increasing this stiffness, although it induces a coupling of the coupling vertical modes of the engines and the first vertical bending mode, leads to improved RMS values. Figures 11 and 12 show the PSD as regards vertical acceleration in the center of the bodyshell for 2 stiffness values.

Figure 11 : PSD in bodyshell center area, first vertical stiffness setting for powerplant flexible supports.

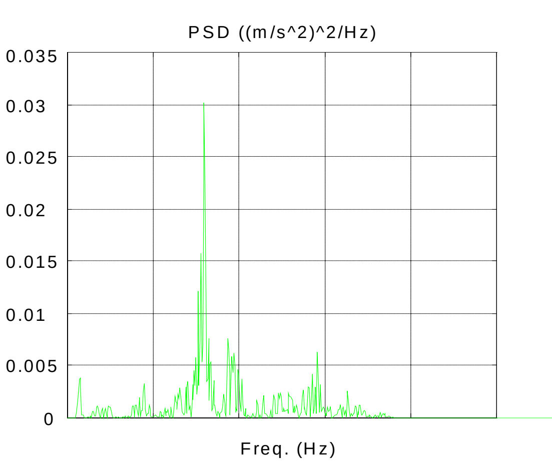
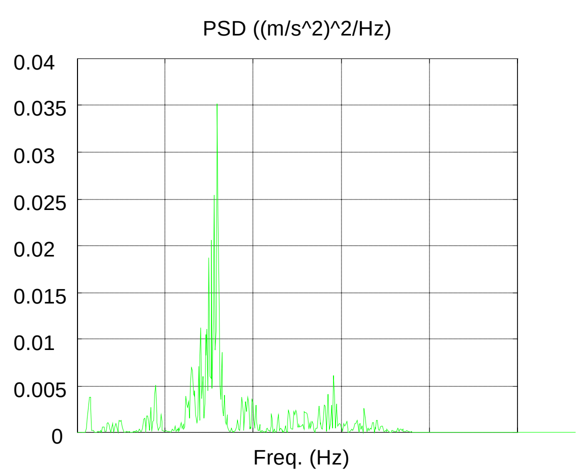


Figure 12 : PSD in bodyshell center area, second vertical stiffness setting for powerplant flexible supports.



- the low interaction between the parameters studied which may be optimized independently from one another within the ranges studied.

5.2) Study of the level I overall modal response (wheel / rail contact creep forces are disregarded)

➢ *Aim*

The aim of this study was to highlight the influence of several parameters (primary and secondary suspension damping rate, position of vertical dampers) on the overall bodyshell modal response and to assess the vertical comfort level by calculating the vertical acceleration RMS value.

➢ *Hypotheses and specific calculation conditions*

a) Results analysed :

The points analysed are the same as those analysed in section 5.1. The calculation results were also analysed as power spectral densities of weighted vertical accelerations and related vertical acceleration weighted RMS values.

b) Damping rate of bodyshell modal basis modes (32 metric tons, no bolster) :

The study of the flexible bodyshell overall response led to assume realistic damping rates for each mode of the Graig-Bampton modal basis of the flexible bodyshell:

- 3.5 % for the first two vertical bending modes,
- 3 % for all other frequency modes ranging between 0 and 12 Hz,
- 15 % for all frequency modes ranging between 12 and 1000 Hz,
- 100 % for all frequency modes greater than 1000 Hz.

➢ *Main conclusions*

The results of the level I overall response calculations for the bodyshell confirm the results achieved in section 5.1, especially the slight response of the first vertical bending mode.

Other parameters affecting the response level of the first bending mode were highlighted (e.g. primary and secondary damping rates, position of secondary vertical dampers).

Hence, this response level may be optimised while accounting for the level of vertical acceleration RMS values.

However, since these values are dependent upon the damping rates selected for the bodyshell modes, calculation results must be considered cautiously.

Except for the latter comment, the vibrational level of the bodyshell remains satisfactory.

5.3) Study of the level IIa overall modal response (wheel / rail contact creep forces are accounted for)

➢ *Aim*

The aim consists in analysing the evolution of the overall modal response and of level of RMS value of vertical acceleration when the wheel / rail contact creep forces are accounted for.

➢ *Hypotheses and specific calculation conditions*

The calculation hypotheses are identical to those of section 5.2 (level I calculations). The equivalent conicity values that were accounted for in level IIa calculations read: 0.1; 0.2; 0.3; 0.4 and 0.5.

➢ *Main conclusions*

The level IIa vertical acceleration PSD and RMS values for the bodyshell bottom, driver's cab rear and bogie rear midpoints proved similar to those achieved in level I. However, a slight response of a transverse bending mode coupled to a vertical shearing of the sidewalls showed on the lateral acceleration PSD in the lower bodyshell midpoint and on the vertical acceleration PSD of the sidewalls.

Additional calculations showed that this mode responds when both yaw dampers of a same bogie are excited in opposition of phases. Hence, the response of this mode is dependent upon the wheel / rail creep forces, therefore it cannot be assessed in level I calculations.

The quick decrease in critical speed as the equivalent conicity increases on the range studied was also highlighted.

➢ *Model limits :*

The complexity of the structures modeled (especially as regards the items of lining) obliges to make hypotheses which may lessen the accuracy of the simulations performed. Therefore, investigation must be carried on in order to validate a number of selected models. Plans were made to:
- model the bolsters using flexible bodies
- experimentally adjust the calculated modal basis and measure the modal damping ratios.

However, the model compiled remains a tools that helps making design choices.