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SIMULATION OF RAIL DYNAMICS AT POLITECNICO OF TORINO

A. Gugliotta, A. Somà, P. Arrus, A. Lombardi

Dipartimento di Meccanica
Politecnico di Torino
C.so Duca degli Abruzzi 24 - 10129 Torino
E-mail: soma@polito.it

Abstract

In this work the simulation of rail dynamics at mechanical department of Politecnico of Torino, is presented. Object of the study is a tilting train realised by Fiat Ferroviaria and named "Pendolino", used for high speed dedicated lines and conventional lines.

High speed train safety is strictly dependent by its lateral stability. Such performance is primarily affected by the structural parameters of the bogie and by the wheel-rail contact.

In the present paper a first numerical model is used to evaluate the lateral stability for straight lines. The simulation analysis is performed for different bogie configurations and uses the results of an appropriate identification procedure for the anti-yaw damper non linear component. Transient results obtained from the numerical model are compared with some experimental data.

Dynamic behaviour taking into account tilting is also analysed for vehicle models using curved lines.

1. Introduction

The dynamic behaviour of an high speed train must be characterised by safe lateral stability and low derailment ratio Y/Q (lateral by vertical force). Furthermore all ride quality standard requirements shall be met. More attention is necessary to such requirements for a "Pendolino" train due to its versatility for use on hybrid track lines (high speed dedicated lines and conventional lines): lateral stability must be provided as in straight lines as in curve. For this reason two different types of models are realised. Only the second model is provided with the tilting system and in this case only a validation procedure of its lateral behaviour in curve has been performed.

The lateral stability in straight lines without irregularities is analised by the calculation of the critical speed. This parameter is strictly dominated by the bogie structural characteristics and by the wheel-rail creep forces [1] that, starting from this speed, are likely to supply energy to the bogie hunting motion that become unstable, or with divergent amplitude. From a numerical point of view the critical speed is obtained by solving the eigenvalues problem after the linearization of the set of differential equations describing the dynamic behaviour of the vehicle (multibody elements) and the wheel-rail interaction. This linear analysis allows to perform in a simple way a parametric study on the main structural parameters of the bogie and on the wheel-rail contact properties.

To properly simulate the overall vehicle dynamics each component requires a good characterisation for both linear and non linear behaviour. This is primarily valid for the anti-yaw damper component, usually connecting the bogie with the coach, which is necessary to avoid or to reduce the bogie hunting instability oscillations.

Transient results obtained from the numerical model in a straight line with a switch, compared with experimental data, reveal the goodness of numerical analysis results.

The availability of reliable numerical models and structural optimisation codes can result of great importance in vehicle design so to reduce to a minimum costs for in-line tests. Numerical models and simulations described in the present paper has been performed with the multibody Adams™ code that uses the wheel-rail contact models derived from the Kalker theory [2].

The efficiency of this code has been verified in a previous work [3] where the results of a numerical simulation were compared with those available from a benchmark proposed by ERRI (European Rail Research Institute) [4].

2. Numerical models

According to the "multibody" classic formulation the modelling of the vehicle consists of rigid bodies connected by means of elastic elements having stiffness and damping characteristics. Figure 1 shows the schematic view of the "Pendolino" bogie.
The main geometry includes: pivot spacing of 19 m, wheelbase of 2.70 m, primary suspensions gauge of 1.00 m, wheel radius of 0.46 m, wheel profile S1002, rail profile UIC 60, rail conical tread 1:20 or 1:40. Some values of mass and inertia characteristics are given in Table I.

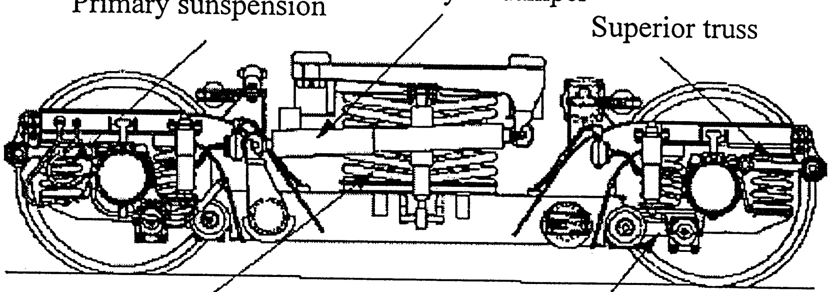

Figure 1 - Schematic view of "Pendolino" bogie.

COMPONENT	QUANTITY	MASS [kg]	Ixx [kg m^2]	Iyy [kg m^2]	Izz [kg m^2]
BODY	1	39000	65000	1870000	1850000
BOGIE	2	3100	2000	2150	4130
WHEELSET 1,4	2	1250	820	90	820
WHEELSET 2,3	2	1550	820	90	820
AXLEBOX	8	90	1	4	3
SUPERIOR TRUSS	8	11	0,5	0,4	0,4
INFERIOR TRUSS	8	20	0,1	0,3	0,3
TOTAL	31	51770	82389	3027380	300300

Table I - Vehicle mass and inertia characteristics.

A previous activity was carried out [5] to clearly quantify the anti-yaw damper dynamic behaviour. Specific tests were performed using the test bench reported in figure 2 with the aim to evaluate the damper force as function of relative speed and displacement of the two ends. The achieved data were used to determine a methodology for the derivation of a damper simple model to be used in Adams/Rail simulation analysis. Synthetically the damper mathematical simulation has been performed by means of two models: the viscous damper and the Maxwell element consisting of a spring-damper series elements.

The first model is build up using the interpolation function of the force vs. speed relation (see figure 3). The second model requires the additional definition of the series stiffness which can be constant with speed or a non linear function of the speed (see figure 4). The energy method was find to be the most efficiency tool for data analysis.

The comparison of the two Maxwell model variants (constant stiffness or non linear stiffness) with the experimental results is given in figure 5 and 6 for two different values of displacement amplitude. It can be remarked that for low relative displacements between the two end of the damper (figure 5) the introduction into the model of a non linear stiffness has the only effect to increase the computation time being the damper load cycle almost unchanged passing from constant series stiffness to non linear series stiffness. For higher values of damper movements the Maxwell model with non linear series stiffness gives better match with experimental data (figure 6).

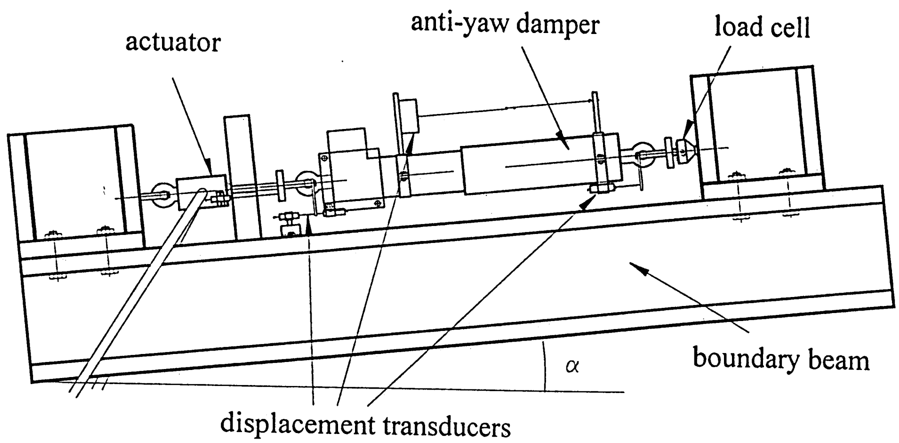

Figure 2 - Anti-yaw damper test set up

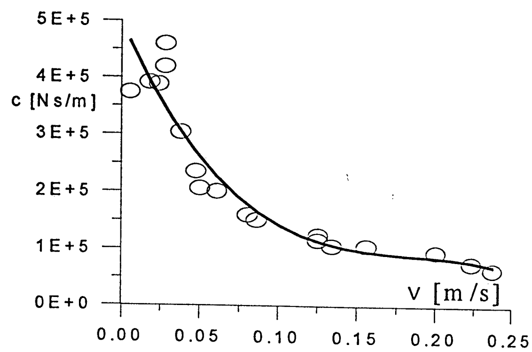

Figure 3 - Damping vs. damper ends relative velocity.

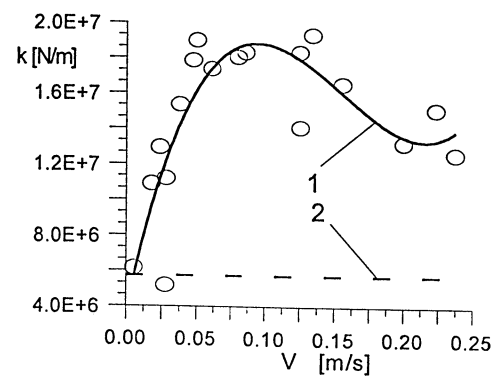

Figure 4 - Stiffness vs. damper ends relative velocity: 1 - non linear stiffness; 2 - constant stiffness (5E6 N/m).

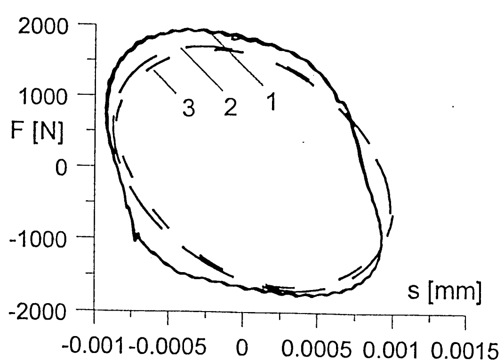

Figure 5 - Force vs. damper ends relative displacement (f = 1 Hz, X = 1 mm): 1 - experimental data; 2 - constant stiffness (5E6 N/m); 3 - non linear stiffness.

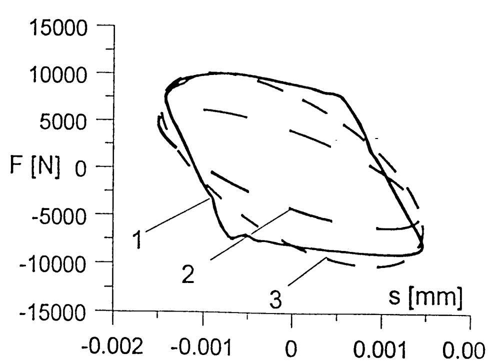

Figure 6 - Force vs. damper ends relative displacement (f = 3 Hz, X = 1.5 mm): 1 - experimental data; 2 – constant stiffness (5E6 N/m); 3 - non linear stiffness.

The tilting system in Adams of the model for dynamic simulation in curve lines is presented in figure 7.

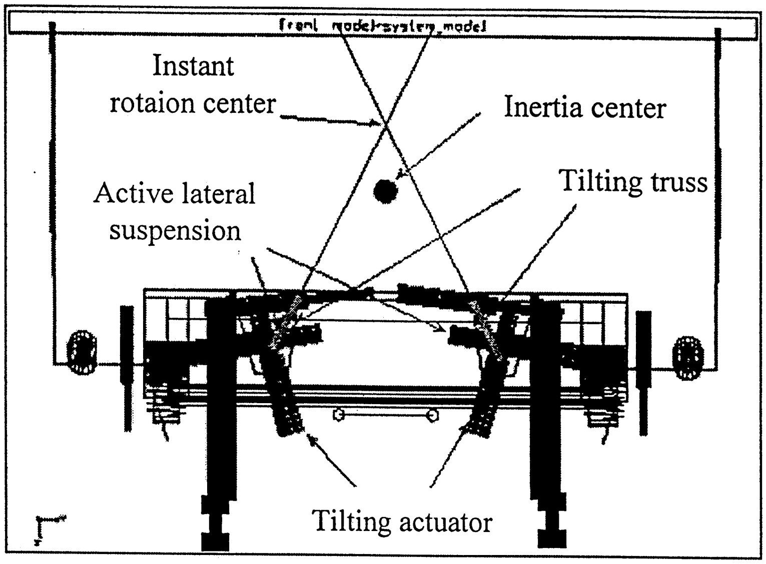

Figure 7 – Tilting system for dynamic simulation in curve lines

2. Stability analysis in straight lines

Computation is performed through the eigenvalue analysis of the complete vehicle model for different running velocity. System vibration modes are damped, or stable, if the real part of the relevant eigenvalue are negative. The speed value that causes the real part reaching zero is the critical speed and the associated eigenvector defines the unstable mode (usually the kinematics hunting mode).

The study is addressed to the evaluation conicity on the vehicle critical speed of the influence of the equivalent which higher values for given wheel and rail basic profiles may be assumed to be indicative of accumulated vehicle service or of track irregularities (low gauge, local irregular rail profile, unbalanced bogie running, different national rail conical tread (1:20 or 1:40), etc.) in case of new wheel profile.

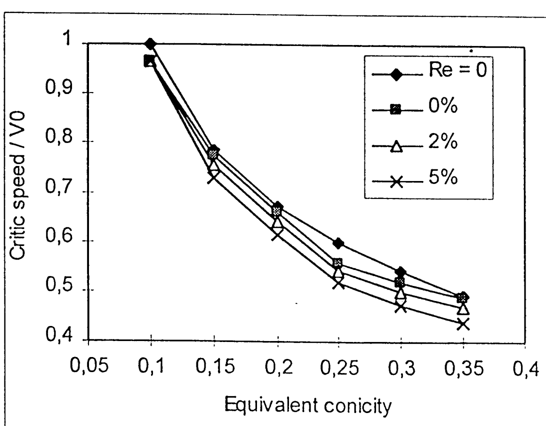

Figure 8 – Critical speed vs. equivalent conicity for the nominal model

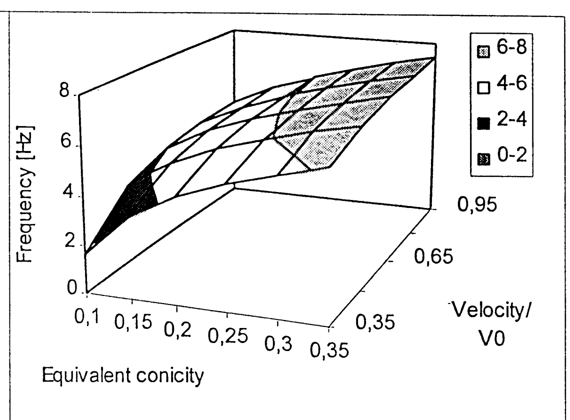

Figure 9 – Frequency vs. Equivalent conicity and velocity for the nominal model

The nominal vehicle model (vehicle_AS_K in table II) is composed only by a single body with mass charateristic in table I, with Maxwell damper's model and damper coefficient for the wheel-rail contact equal 0,3. Track has no irregularities. Contact table is calculated with the module RSGEO of Adams/Rail for the used profiles UIC60 e S1002.

In figure 8 is represented the stability diagram of the nominal vehicle model for different methods of evaluating the critical speed (Re = 0, real part null; A%, damping ratio = A). In figure 9 the relative frequency of the instable mode for different equivalent conicity and velocity is described.

The stability analysis has been performed for different bogie configurations (see table II), to evaluate the influence of some parameters on lateral stability. Main part of the analysis is concentrated on the presence and on the simulated characteristics of anti-yaw dampers. The malfunctioning (absence) of one anti-yaw damper is also investigated.

Figure 10 shows the results of this sensitive analysis for different configurations (AS means anti-yaw dampers; K, series stiffness of damper's Maxwell model): the variation % of critical speed due to the change of one parameter is represented as function of equivalent conicity. This figure reveals that the influence of anti-yaw damper presence is very important: critical speed increase to 40-60%; the effectiveness reduces with equivalent conicity increasing. The malfunctioning of one of them reduces the critical speed to values similar to those obtained without anti-yaw dampers. A proper simulation of anti-yaw damper component has also a great importance. In fact, as can be seen, the use of a viscous model leads to results for the critical speed very different from the nominal model (from 40 to 100%).

Figure 10 also indicates the poor influence on the vehicle critical speed of the complexity of the model used for other dampers (primary and secondary) or, in other words, how much correct is the estimate of their series stiffness.

Finally the influence of the damper coefficient for the wheel-rail contact is analyzed. Changing it from 0,3 to 0,6 the critical speed ever decrease (not for low equivalent conicity in the case of vehicle model without anti-yaw dampers).

Vehicle model	Anti-yaw damper		Absence of an	Damper's model		Wheel-rail
	primary	secondary	anti-yaw damper	Maxwell	Viscous	damper coeff.
vehicle_AS_K	yes	yes	no	yes	no	0,3
vehicle_noASS_K	yes	no	no	yes	no	0,3
vehicle_noAS_K	no	no	no	yes	no	0,3
vehicle_AS_Mal_K	yes	yes	yes	yes	no	0,3
vehicle_AS_noK	yes	yes	no	no	yes	0,3
vehicle_noAS_noK	no	no	no	no	yes	0,3
vehicle_AS_mu06	yes	yes	no	yes	no	0,6
vehicle_noASS_mu06	no	no	no	yes	no	0,6

Table II – Configurations of the vehicle model for stability analysis

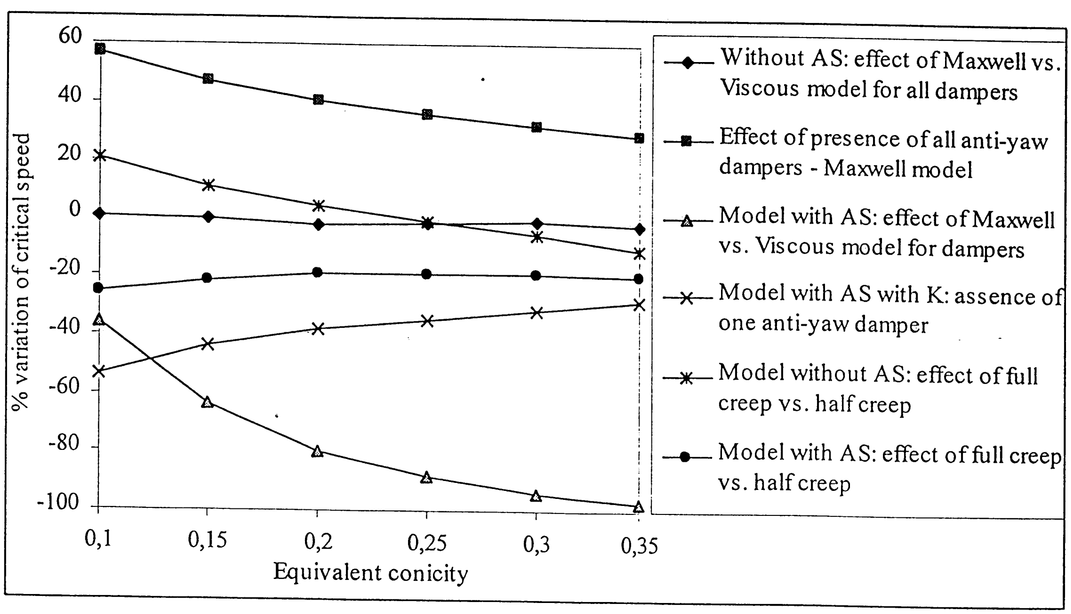

Figure 10 – % variation of critical speed vs. equivalent conicity: effects of some parameters for different configurations

3.1 Comparison with experimental data

The international standards criteria on rail vehicle safety and reliability have been tailored on the basis of theoretical studies and experimental results. Nevertheless it is possible to found some differences between methods developed in North America and Europe [7, 8].

In particular as far as lateral stability is concerned, European standards [8] require that the lateral acceleration measured on the bogie frame in line with the axlebox shall not reach levels higher than 8.0 m/s^2 in the frequency range between 4.0 and 8.0 Hz. On this basis the experimental critical speed is conventionally identified as that velocity at which lateral acceleration levels higher than 8.0 m/s^2 occur for a number of 6 consecutive cycles.

Concerning the comparison of transient analysis simulation results with experimental data, it is highly complicated by the absence of complete information on the rail status. However an attempt of correlation is made for a switching cross (Figures from 17 and 20). The FFT performed on the acceleration lateral signals shows a good correlation between the experimental frequency of the hunting mode instability of the numerical result.

Figures 17 and 18 report the numerical analysis results performed considering the vehicle cross on a theoretical switch [4] and a non linear wheel-rail contact model. The experimental data refer to a switching cross at a speed of 0.85*Vo (Figure 19) and 0.87*Vo (Figure 20) for a vehicle configuration without anti yaw dampers. It can be seen comparing Figures 17 and 19 that both numerical model and the actual vehicle are stable following the indications of the European standard [8]. Moreover also the acceleration levels are quantitatively comparable. In Figure 17 the acceleration response return to zero after the attenuation of the switch excitation as an idealised perfect rail has been used for simulation. For the actual case (Figure 19) a light new hunting harmonisation can be seen due to rail irregularities.

Figure 20 gives an example of experimental instability occurred at a speed of 0.87*Vo. In the simulation analysis the instability is found at a speed of about 0.90*Vo (Figure 18).

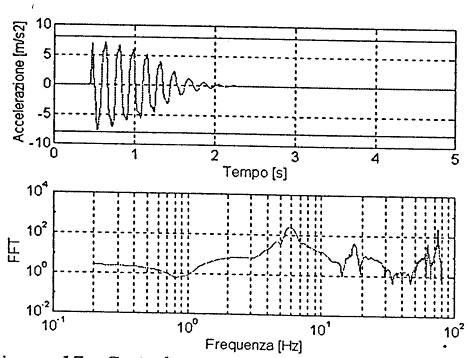

*Figure 17 - Switch crossing simulation with ADAMS/Rail: lateral acceleration Velocity = 0.85*Vo*

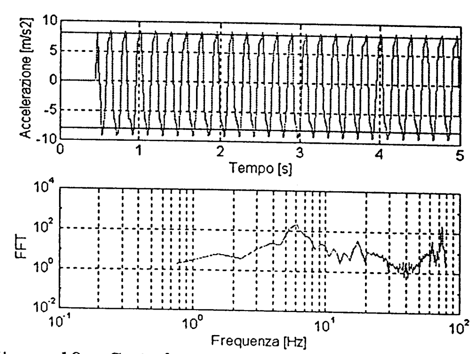

*Figure 18 - Switch crossing simulation with ADAMS/Rail: lateral acceleration Velocity 0.90*Vo*

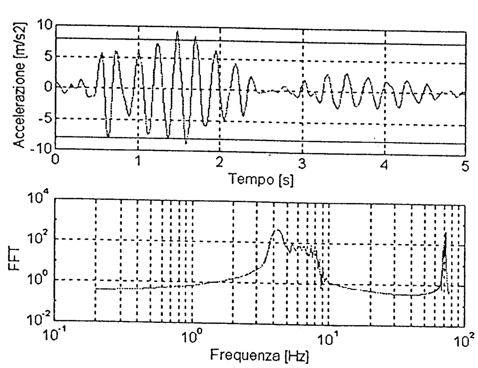

*Figure 19 - Switch crossing experimental data: lateral acceleration Velocity = 0.85*Vo*

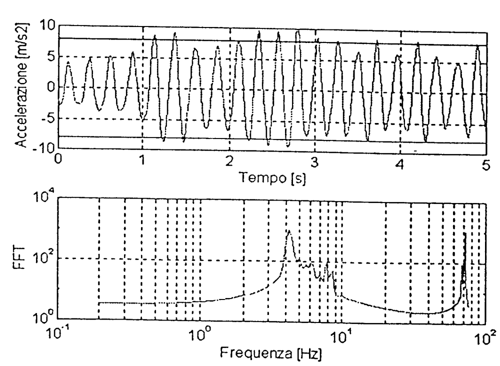

*Figure 20 - Switch crossing experimental data: lateral acceleration Velocity = 0.87*Vo*

4. Tilting train control

It is well known that the tilting control allows to reduce the acceleration on passengers during a curved trajectory. This is shown in figure 21 (passive control) with 22 (active control): the centripeter acceleration decreases as it is function of the angle α.

$$a_{BODY\_PASSIVE} = a_{nc} = \frac{V^2}{R} - g\alpha \qquad\qquad a_{BODY\_ACTIVE} = a_{nc} = \frac{V^2}{R} - g\alpha_T$$

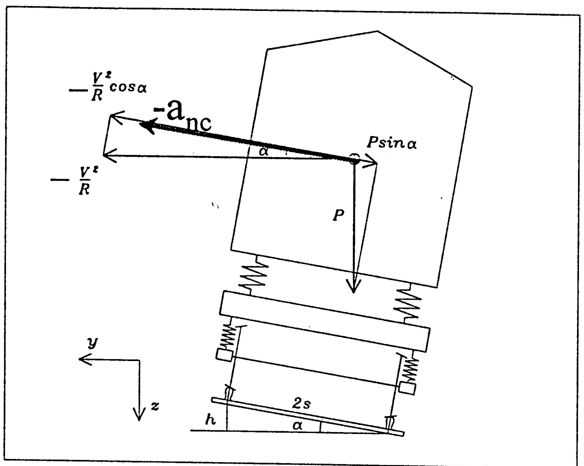

Figure 21 - Passive control

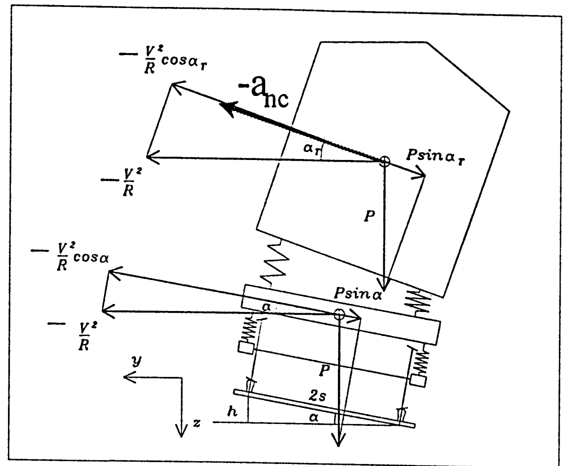

Figure 22 – Active control

In order to simulate the dynamic behaviour of the "Pendolino" a control scheme (P.I.D. control) has been applied, see figure 23. It uses a gyroscopy and lateral accelerators to determine the beginning of the curve. It is to be noticed that this control used is not the actual real control of the train.

Different preliminary simulations has been performed by using various tracks, some analytical, other experimental. In figure 24 is presented the result of a simulation in a s-curve; only a wagon has been modelled: passenger lateral acceleration for passive and active control is compared. As it can be seen, the active control is not yet very efficient for fast change of trajectory.

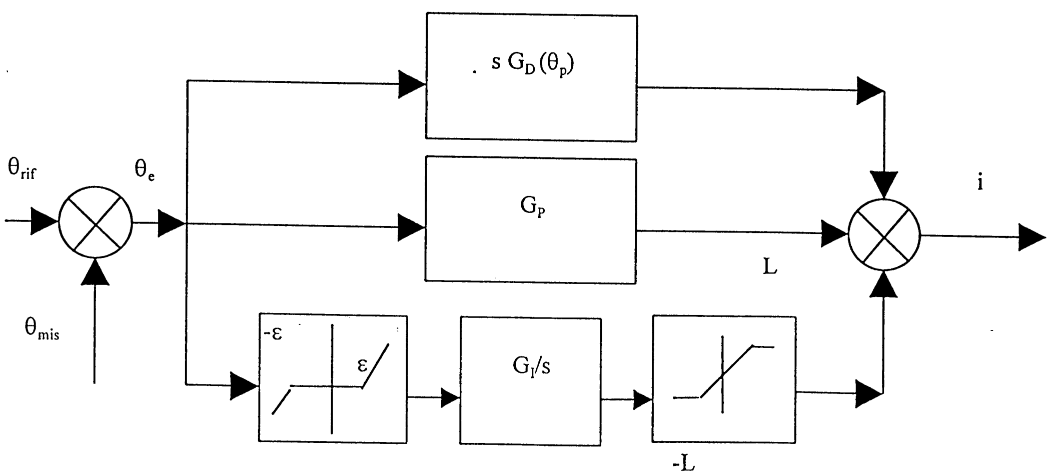

Figure 23 – The control scheme

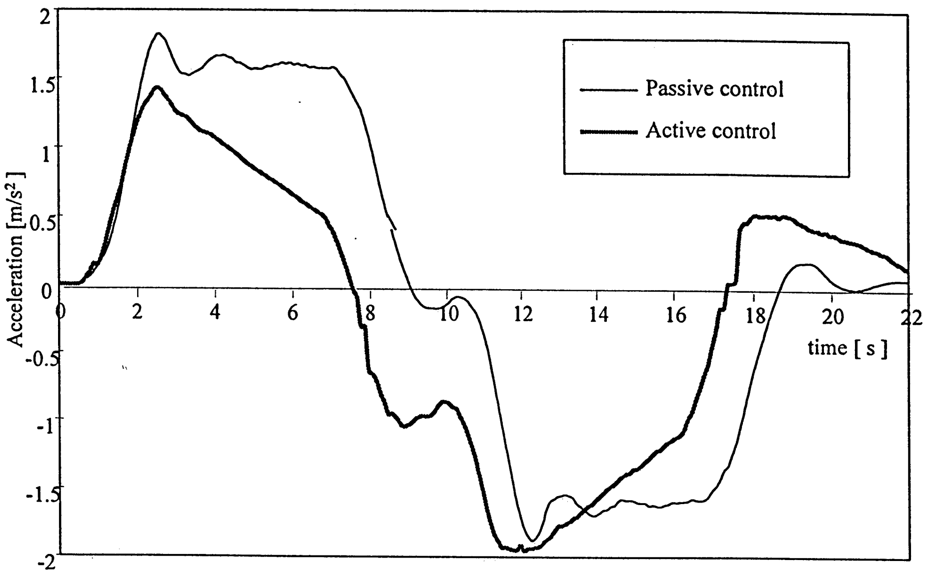

Figure 24 – Passenger lateral acceleration: active and passive control (S-curve, velocity = 105 Km/h, radius = 500 m)

5. Conclusions

A numerical model of the "Pendolino" train, in service on several European track lines, has been generated with the ADAMS/Rail multibody code for lateral stability analysis. The critical speed calculations have been performed using both linear (eigenvalue analysis) and non linear (transient analysis) wheel-rail contact models. Linear calculations are considered a good design tool allowing fast and easy sensitivity analysis on bogie configuration parameters. The almost good match between the transient analysis results and the experimental data relevant to a switch crossing allows a further model validation. Finally the validation of a tilting vehicle necessary for simulating curve lines has been performed.

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