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Des outils de simulation pour les systèmes véhicule ferroviaire/voie.

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Simulation Tools for Railway Vehicle/Track-Systems

Simulationstools für Schienenfahrzeug/Gleis-Systeme

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Abstract

This paper presents an overview of some of the main simulation tools used today for analysis of railway vehicle dynamics and a brief history of the development of this field. The methods typically used to handle the difficult and highly non-linear aspects of wheel-rail contact and other suspension non-linearities are outlined. The various approaches to interface with the user and links to other types of simulation tools are mentioned and future developments in this area discussed.

Zusammenfassung

Dieser Beitrag bietet eine Übersicht über einige der wichtigsten zur Schienenfahrzeugdynamik-Analyse verwendeten Tools sowie eine kurze Geschichte der Entwicklung auf diesem Gebiet. Die zur Bearbeitung der schwierigen und höchst nicht-linearen Aspekte des Rad/Schiene-Kontaktes und anderer Nichtlinearitäten der Federung typisch angewendeten Methoden werden behandelt. Die verschiedenen zum Informationsaustausch mit dem Anwender verwendeten Methoden und die Verbindung mit anderen Arten von Simulationstools werden erwähnt und zukünftige Entwicklungen auf diesem Gebiet erörtert.

1. Early development of the understanding of wheel-rail interaction

The earliest railway vehicles had simple cylindrical wheels and were guided by the plateways on which they ran. As the technology developed, conical wheels were introduced and the flanges were moved from the rail to the inside of the wheel. This may initially have been due to manufacturing process and material cost considerations but the benefits to vehicle running were soon noticed and the basic wheelset which is still in use on almost all railway vehicles today was established.

The motion of a wheelset with conical wheels was understood in principle as early as 1821 when George Stephenson wrote: *„The wheels are conical, that is the outer is rather less than the inner diameter by about 3/16 of an inch. Then from a small irregularity of the railway the wheels may be thrown a little to the right or a little to the left, when the former happens the right wheel will expose a larger and the left one a smaller diameter to the bearing surface of the rail which will cause the latter to loose ground of the former...alternately gaining and loosing ground of each... will cause the wheels to proceed in an oscillatory but easy motion on the rails.“* (quoted from Wickens [1] who gives a comprehensive and definitive account of the development of this field).

The equations that described this motion were gradually laid out with increasing levels of complexity. Redtenbacher [2] in 1855 derived an equation for the displaced rolling line of a wheelset in a curve based purely on the radii of the wheels at the point of contact and the rails at the inner and outer wheel. Klingel [3] in 1883, derived an equation for the kinematic oscillations of the wheelset.

Precise prediction of the motion of the wheelset and the vehicle in response to discrete or random track irregularities was, however, much more difficult.

2. Modelling wheel-rail contact – the contact patch and normal forces

The classical theory of contact was developed by Hertz [4] in 1882 when he was a 24 year old research assistant at the University of Berlin. He proved that the contact area between two non-conformal bodies of revolution would be elliptical and he established a method for calculating the semi-axes of the ellipse and the pressure distribution within the contact patch. The Hertz theory is strictly restricted to frictionless surfaces and perfectly elastic solids but it still provides a valuable starting point for most contact problems and is included in most computer programs which deal with wheel-rail contact.

In analysis of the contact between a railway wheel and a rail the first step is to establish the location and the size and shape of the contact patch (or patches). As the cross sectional profiles of the wheel and the rail can be quite complex shapes most computer simulation packages have a pre-processor, which puts the wheel and rail profiles together for a given wheelset and track and establishes where the contact will occur. A description of the cross sectional profiles is prepared from the designs or measured using a device such as the widely used 'Miniprof'. The contact parameters are established for the required lateral displacement and yaw angle of the wheelset. Sometimes this is done considering the bodies to be rigid or their flexibility can be allowed to influence the position of the contact. Iteration may be required as the roll angle of the wheelset both influences and is a function of the contact position. This contact pre-processor is run whenever the contact details are required or can be used to set up a table of data from which the properties can be interpolated.

Some software packages then use Hertz theory to establish elliptical contact patches around the contact point. The normal load on the contact point is required and the calculation may be iterative to allow the correct load distribution between the contact points to be found. In tread contact the radii of curvature are only changing slowly with position and the contact patch is often close to elliptical in shape. However, if the radii are changing sharply or the contact is very conformal the contact patch may be quite non-elliptical and the Hertz method does not produce good results. Knothe and Le-The Hung [5] set out a numerical method for calculating the tangential stresses for non-elliptical contact in 1985.

Multi-Hertzian methods split the contact patch into strips with Hertz contact being calculated for each strip. Some iteration may be required to establish the correct normal load distribution across the whole contact patch if the contact is being treated as a constraint. Pascal and Sauvage [6] developed a method using an equivalent ellipse which first calculated the multi-Hertzian contact and then replaced this with a single ellipse which gives equivalent forces. In the methods developed by Kik and Piotrowski [7] an approximate one step method is used and some results for an S1002 wheel and a UIC60 rail are shown in *Figure 1*. In the 'semi-Hertzian' methods developed by Ayasse, Maupu and Chollet [8] the contact is treated as Hertzian for the longitudinal curvatures (along the rail) and non-Hertzian for the lateral curvatures (across the rail).

3. Modelling wheel-rail contact – simulation of creep forces

Carter [9] in 1916 introduced the concept of creepage or microslip between the wheel and the rail and the corresponding creep force which was generated. A fuller treatment of the creep forces was given by Vermeulen and Johnson [10] in 1964 and in 1967 Kalker [11] gave a full solution for the general three-dimensional case with arbitrary creepage and spin.

Kalker produced various computer algorithms for the calculation of creep forces according to his theory. The program CONTACT based on Kalker's 'exact' theory which includes non-Hertzian contact but which is relatively slow and not practical for use at every time step in a numerical integration. Table interpolation routines are available such as USETAB [12] which interpolate between values of creep force pre-calculated by CONTACT. FASTSIM [13] is based on Kalker's 'simplified theory', which assumes an elliptical contact patch with a flexible layer between the two rigid bodies.

Experimental measurements taken by Brickle [14] with a twin disc rig and Illingworth [15] with a roller rig validated Kalker's results provided that the surfaces in contact were free from contamination. Hobbs [16] proposed that the Kalker coefficients be factored by 0,6 to take normal levels of contamination into account.

Heuristic methods for predicting creep force were developed initially by Johnson and Vermeulen [10] based on a cubic equation for creep force as it nears saturation. The method developed by Shen, Hedrick and Elkins [17] is widely used and is very fast but results are approximate and become less accurate when spin (relative rotation about the normal axis) is high, Polach [18] has developed a method which works well at high levels of creepage and spin and also includes the falling value of the coefficient of friction as slip velocity increases.

When surface roughness effects are significant the creepage creepforce relationship is affected and Bucher, Knothe and Theiler [19] have proposed methods for dealing with this situation. Knothe, together with Gross-Thebing [20] has also looked at the effect of rapidly varying creepages which were not previously considered. When the contact characteristics have been calculated by non-Hertzian methods (see above) a modified version of FASTSIM has to be used as explained by Kik and Piotrowski[7].

4. The development of computer simulation tools for Railway Vehicle dynamics

Once an understanding of the wheel-rail contact had been established the way was open for a full analysis of the dynamic behaviour of a railway vehicle. This was encouraged by a prize offered by the Office for Research and Experiments

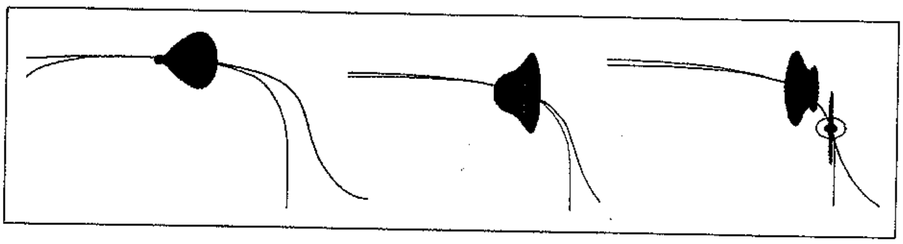

Figure 1: Non elliptical contact patches for profile combination S1002/UIC60 (from Kik and Piotrowski)

(ORE) of the Union of International Railways (UIC) in 1950 for the best analysis of the stability of a two axle railway vehicle. The prize winners were Possel, Boutefoy and Matsudaira [21].

All of the prize winners had used a linear analysis of the problem but de Pater [22] formulated the hunting behaviour as a nonlinear problem. Van Bommel [23] later published nonlinear equations for a two-axle vehicle using wheel and rail profiles and a creep force-creepage law measured by Müller for the ORE committee.

With the advent of analogue and then digital computers it became possible for these equations to be solved for real problems and for non-linearities to be included more easily.

Wickens [24] at British Rail led a group who improved the analysis to include an understanding of the wheelset as a feedback mechanism and applied first analogue and then digital computer methods to the problem. This resulted in a new high speed two axle freight vehicle with much improved stability and provided a basis for the work on the Advanced Passenger Train (APT) and for the development of the software tools used today in the UK.

Müller [25] carried out one of the first simulations with analogue computers of a bogie vehicle running into a curve. Müller [26] also recognized the importance of the inclusion of non-linear wheel profiles and included tabulated geometric data which was measured for a combination of worn wheels and rails. This was taken further by Cooperider et al. [27] who produced an algorithm for combining measured wheel and rail profiles to produce the non-linear parameters such as rolling radius difference and contact angles as the wheelset moved laterally across the track.

The early programs tended to split up the types of behaviour to simplify the task of calculation. Programs for calculation of the vehicle attitude and the forces developed during steady state curving were one example of this. An eigenvalue analysis of the linear or linearised equations of motion was used to give information about the natural frequencies and mode shapes of the oscillations and to predict limits to stable running. Time stepping integration could be carried out if the systems were non-linear but it was usually necessary to separate the vertical behaviour (involving bounce and pitch of the bodies) and lateral behaviour (yaw, roll, sway). Longitudinal dynamics, which

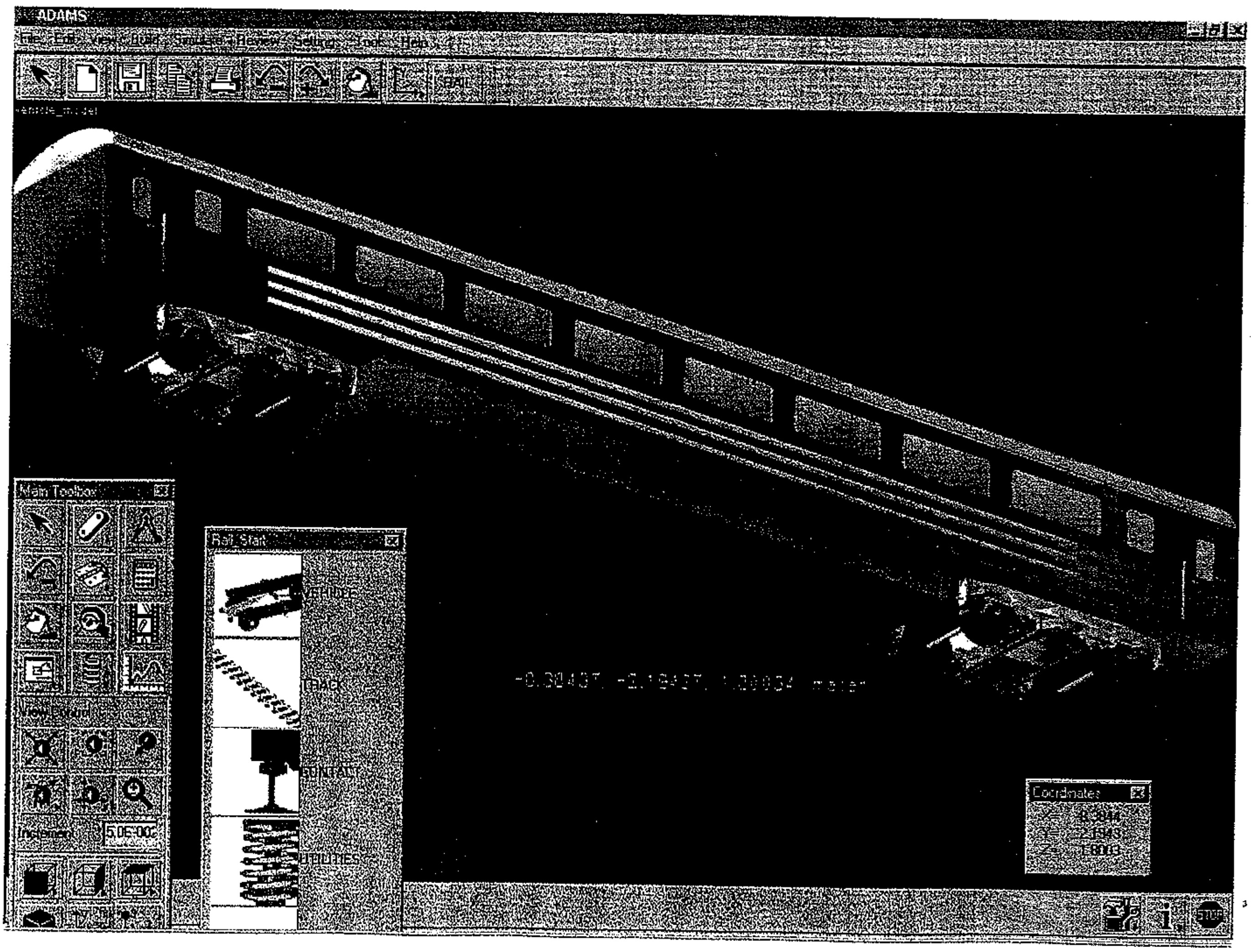

Figure 2: ADAMS/Rail Graphical User Interface with vehicle model

is more important when dealing with long freight trains, was also handled separately. As computing power developed it became less necessary to handle each aspect of the vehicle behaviour separately and powerful numerical methods were applied in the time domain unless a frequency domain output was required.

Multibody dynamics theory is used to develop the equations of motion for the system and these are processed by a solver which produces the results of interest. A review of the main multibody simulation packages and the methods that they used was carried out by Schiehlen [28].

True [29] applied the theory of non-linear dynamics to the behaviour of a railway vehicle and showed that failure to consider the non-linearity of the wheel rail contact can lead to an inaccurate estimate of the critical speed of the vehicle. Recently Schupp [30] has described a method using numerical bifurcation analysis to simulate the non-linear behaviour of railway vehicles. These methods have resulted in the software PATH [31] which has been used together with SIMPACK.

The early packages used text based interfaces where vehicle parameters were listed in a particular order or using key words to provide the input to the simulation. User-friendly graphical interfaces were added and packages developed to allow engineers to test the effects of making changes to any part of the system and to animate the output. For example ADAMS/Rail where the user works with a vehicle model through a graphical user interface which allows interaction with the model in the same way as a computer aided design system *(Figures 2 and 3)*.

A large number of computer codes have been developed by railway organisations to assist in the design of suspensions and the optimisation of track and vehicles. Some of these have been combined into general purpose packages and some examples of those currently in widespread use are given here although this is not a comprehensive list and the aim is to illustrate the variety of programs that are in use today.

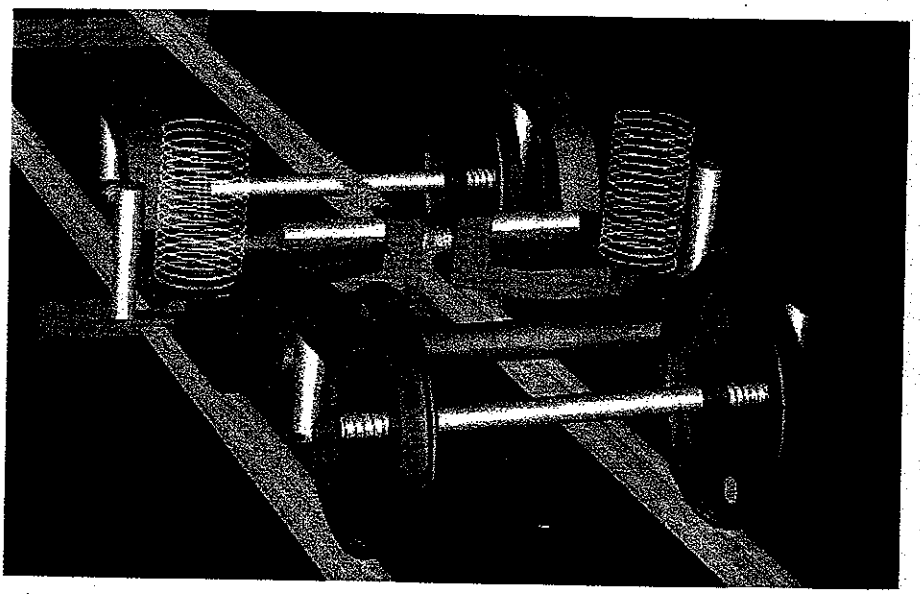

Figure 3: ADAMS/Rail parameterised bogie model

One of the early complete packages, MEDYNA [32] (Mehrkörper-Dynamik) was developed at the German Aerospace Research organisation (DLR) together with MAN and the Technical University of Berlin. MEDYNA was based on a multibody system with small rigid body motions relative to a global reference frame which allowed large motions. The linearised kinematic equations of motion for each body are formulated with respect to the global reference frame. SIMPACK was developed later by the same team at DLR and as it was intended for road as well as rail vehicles it allowed non-linear kinematics from the start. The equations of motion are formulated in terms of relative coordinates and can be generated symbolically and numerically in an implicit and explicit form. The kinematics of elastic bodies are developed including second order terms to allow stress stiffening effects to be taken into account.

ADAMS is one of the most popular dynamic simulation codes worldwide and in 1995 it entered the railway vehicle simulation market as ADAMS/Rail [33], initially by including wheel-rail contact methods developed by NedTrain and later by licensing the wheel-rail contact elements from MEDYNA.

In the USA the Association of American Railroads (AAR) funded the development of a program to simulate the behaviour of

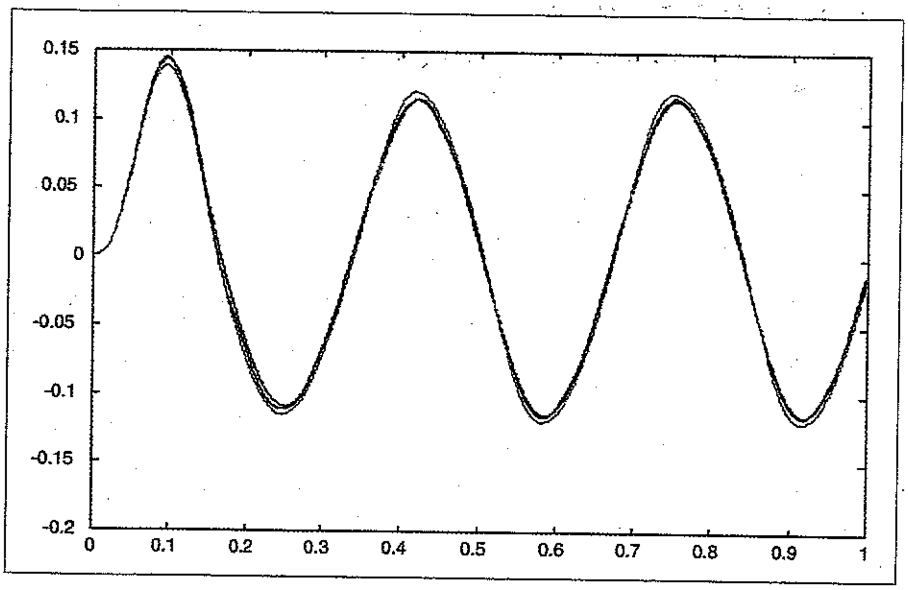

Figure 4: Friction benchmark - displacement of mass [m] against time [s]

a railway vehicle negotiating a curve. This was developed into the general purpose simulation package NUCARS [34] (New and Untried Car Analytic Regime Simulation). NUCARS has been used to improve the dynamic behaviour of the 3 piece freight bogie.

The French National Transport Research Institute INRETS developed a multibody simulation code VOCO (Voiture en Courbe) in 1987 with a reference frame that permitted simulation on long curves. The inclusion of friction damping was possible from the beginning because the code was initially used to simulate the Y25 bogie. A commercial version of this named VOCOLIN [35] in 1991 allowed simulation of the wheel-rail contact with a multi-Hertzian approach. A second approach was made by Sauvage, Pascal et al. [36] and this was incorporated into the code VOCODYM.

In the UK British Rail Research developed a number of computer programs to analyse different aspects of railway vehicle dynamic behaviour as has been mentioned above. These have now been brought together into one coherent package VAMPIRE [37], which is now supported by AEA Technology Rail.

In Sweden modelling of railway vehicles using computers started at ASEA in 1971. Initially the analysis was carried out in the frequency domain with linear models and then, in 1973, a non-linear, time stepping integration program was developed. This program separated lateral and vertical modes and was used in the development of the X15 high-speed test train and the Rc4 locomotive in 1975 [38]. In 1992 the development of a new three-dimensional calculation program started and software development was transferred to a new company called DEsolver. This new three-dimensional, general computer code, together with all earlier pre- and post-programs became in 1993 the new railway vehicle analysis tool called GENSYS.

5. Modelling friction

Friction is quite widely used in railway vehicle suspensions to dissipate energy especially in freight vehicles where its relatively low cost and simplicity make it an alternative to viscous damping which is more commonly used in passenger vehicle suspensions. Unlike viscous damping the use of friction results in a non-linearity which can sometimes lead to harsh riding behaviour.

Friction elements can be arranged to give damping that varies with vehicle load as in the common Y25 bogies used in Western Europe or the '3-piece' type freight bogies used in North America and Eastern Europe. This variation in damping with load is especially important in freight vehicles as the ratio between tare and laden weights can be very great and would otherwise result in too great a conflict in desired parameters. Multiple leaf springs are also commonly used in freight vehicles with the friction between the individual leaves again providing energy dissipation.

The basic phenomenon of dry friction is well understood from observations and is governed by the Coulomb equation: $F = \mu \cdot N$. The value of the coefficient of friction μ is more difficult to be precise about and is sometimes taken to have two values, one before sliding takes place (the coefficient of static friction) and one during sliding (the coefficient of dynamic friction) or recognised as being dependent on the sliding velocity.

In computer simulation of vehicle suspensions the numerical methods used tend to have problems with highly non-linear equations and a great deal of effort has been expended to find a suitable algorithm to calculate the friction force in a robust and yet efficient way. It is also clear that the significance of the friction force in the vehicle suspension makes it important that the results of the algorithm are accurate.

By way of an illustration of the differences between the existing algorithms the author, together with Hugues Chollet and Jean Bernard Ayasse at INRETS has collected information about the exact nature of the algorithm used by a number of simulation packages. These algorithms were all transcribed to the Matlab and Simulink environment and the predicted friction forces evaluated for a simple case. A mass spring system was used and excitation was generated by a displacement of one end of the spring:

a) The Massing model – used in Medyna and ADAMS/Rail: This model is based on a series of Jenkin elements [39]. Switching between 'stick' and 'slip' motion is controlled by the force across the element against the limiting value of the friction force;
b) The Bosso, Gugliotta, Soma [40] model: This model uses a single mathematical expression to evaluate the friction force so that no switching is required;
c) The Ayasse, Maupu [41] model – used in VOCOLIN: This method calculates

the friction force from an internal series stiffness and the relative displacement if the force is below $\mu \cdot N$ and a small internal damping value and the relative velocity if the force is greater than $\mu \cdot N$.

The results shown in *Figures 4 and 5* are for a sinusoidal displacement excitation with a frequency of 3 Hz and an amplitude of 0,1 m, a mass of 50 kg and a spring stiffness of 100 kN/m. The results show some significant differences in the forces although the variation in the displacement of the mass is generally small. This smaller variation in the displacement is due to the double integration between acceleration and position. Some of the algorithms have a tendency to numerical oscillation with certain parameters. Such oscillation can cause the multibody solver to fail and must be avoided. The calculation time of the different algorithms also varies significantly.

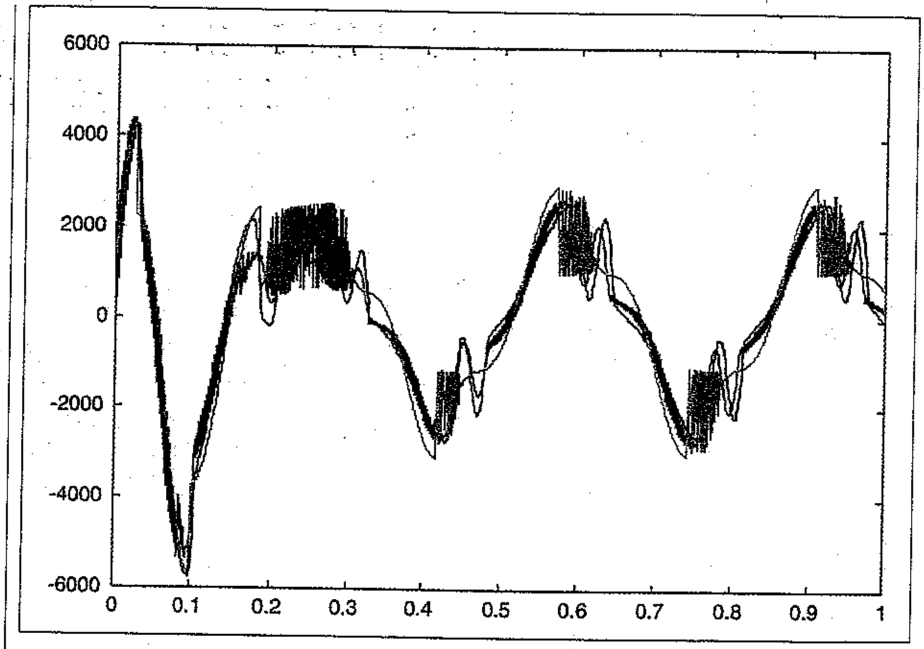

Figure 5: Friction benchmark - force acting on the mass [N] against time [s] green = Bosso et al., red = Massing, blue = Ayasse, Maupu

6. Flexible bodies

Most of the multibody simulation packages allow the inclusion of flexible bodies such as car bodies where bending or torsion can be significant. Information about each mode for the flexible body must be included and this is often taken from a finite element analysis of the body carried out outside the multibody simulation package.

Track models used in multibody simulations are generally relatively simple one or two layer rigid body models and the track support conditions are usually constant along the rail. Although this is likely to be adequate for many types of simulation it may not allow full representation of all the dynamic response modes of the rail. Knothe [42] has reviewed developments in this area. More detailed track models have been developed, for example by Corus Rail Technologies as part of their 'Track System Model'[43] shown in *Figure 6*.

An alternative to separate vehicle and track models is a finite element model (FEM) of the track integrated into the vehicle dynamics software. This should enable the track response to be accurately captured and subsequently improve the accuracy of the vehicle response, which in turn should improve the accuracy of wheel-rail forces. This approach is currently being developed by some of the vehicle software packages. FE flexible models of a bridge structure have been successfully developed using ANSYS and incorporated into ADAMS/Rail. SIMPACK includes flexible bodies using an inbuilt flexible element called SIMBEAM, and also using an FE interface called FEMBS. FEMBS uses the Standard Input Data (SID) file present in the majority of FE packages to create a reduced modal representation of the complete FEM. With the continuing increase in processing speed it may be that the multibody system

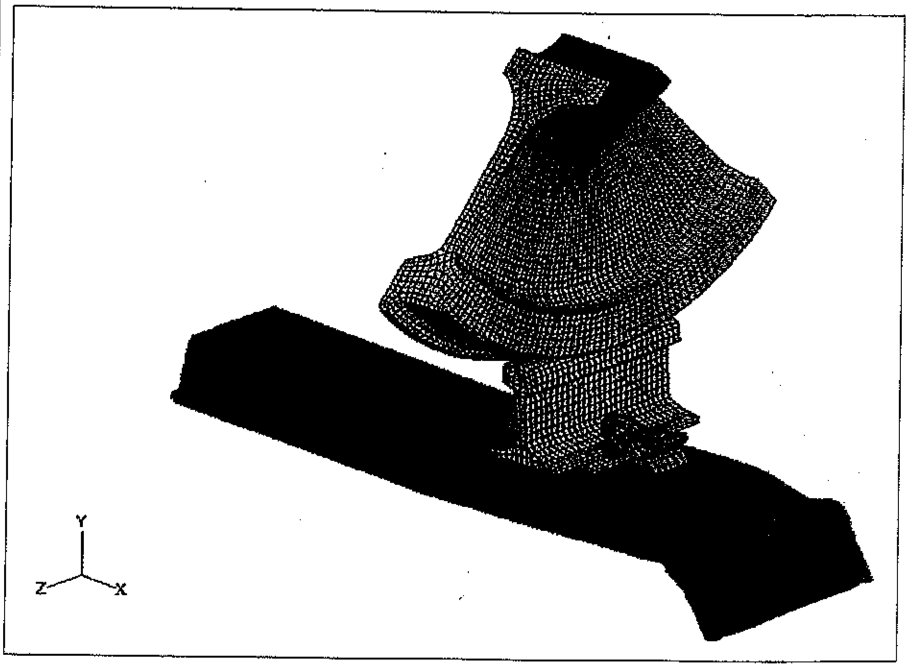

Figure 6: The CRT Track System Model

type program will be superseded by software based on finite element analysis which allows easy treatment of flexible bodies and locates stress and fatigue problems within bodies.

7. Benchmarking of railway vehicle dynamics computer packages

Due to the high level of complexity of the software codes developed for simulation of railway vehicle dynamics there is a high level of interest in comparing the results of the different codes for certain test cases.

In the exercise initiated at the Herbertov workshop on 'Multibody systems applications to problems in vehicle system dynamics' [44] in 1990 and reported on by Kortüm and Sharp [45] the computer codes that were able to handle wheel-rail contact were asked to simulate a single wheelset and a bogie. The wheelset benchmark was proposed by Pascal and participants were required to calculate the lateral deflection of the specified wheelset in response to a lateral force of 20 kN and to find the level of lateral force at which the wheelset would derail. In the bogie benchmark defined by Kik and Pascal, participants were required to predict the behaviour of the bogie in a vehicle running in straight and curved track at several speeds. Not all codes participated fully in the exercise but some interesting results were shown by some participants.

In the Manchester Benchmarks published in 1999 [46] two simple vehicles and four matching track cases were defined to allow comparison of the capabilities of computer simulation packages to model the dynamic behaviour of railway vehicles. One of the aims of this benchmark was to try to encourage railway organisations to accept simulations carried out using any reliable computer simulation package and not to insist on one particular tool. Simulations were carried out with five of the major packages (VAMPIRE; GENSYS; SIMPACK; ADAMS/Rail-MEDYNA and NUCARS) and the results and statements of methods were presented.

A number of outputs were requested for each of the track cases each of which was designed to test a particular potential vehicle problem. One of the most useful indicators of derailment potential is the Lateral/Vertical (L/V or Y/Q) force ratio at each wheel. In a curve it is usually the outer wheel where derailment takes place

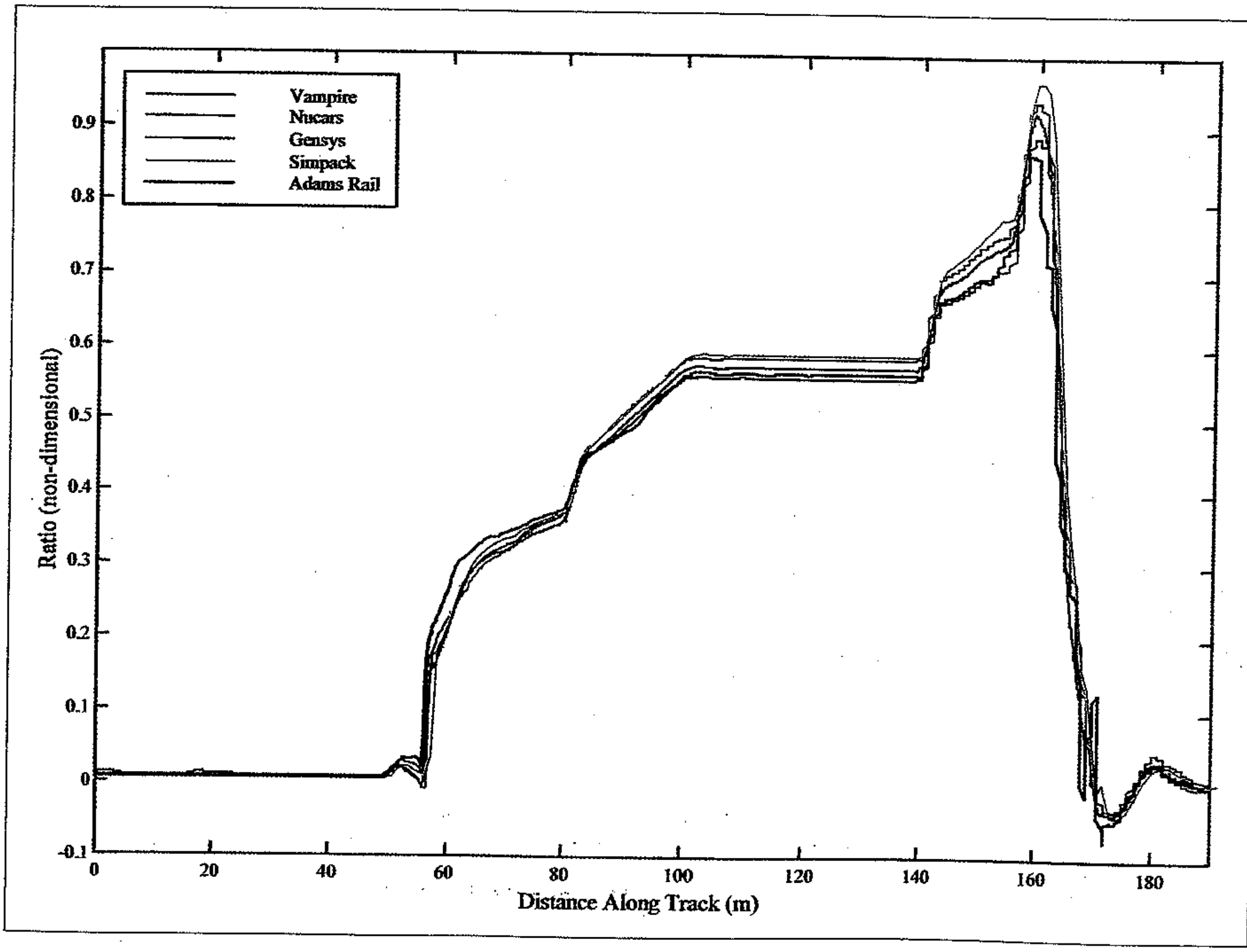

Figure 7: L/V ratio from the Manchester Benchmarks (Vehicle 1, Track case 1, 4.4 m/s)

and *figure 7* shows the L/V ratio for the outer wheel on the first wheelset for one of the benchmark vehicles. The peak value occurs at a dip designed to test the vehicle suspension and shows that all five packages give good agreement on the nearness to derailment of this vehicle.

8. Future developments

8.1 Prediction of rolling contact fatigue

The development of cracks in rails and wheels due to rolling contact fatigue is a real problem for railway engineers. The basic mechanisms are understood but the prediction of exactly when cracks will initiate and the rate at which they will grow has been difficult. In recent years several accidents have focussed efforts on this problem and vehicle dynamics tools are being used to provide the inputs to several fatigue tools. Ringsberg [47]and his colleagues at Chalmers University in Sweden and Burstow et al. [48] at AEA Technology in the UK have prototype models that are currently being validated against real data.

8.2 Simulation of wear

The work done in the contact patch, often evaluated by the product of creepage and creep force (known as T · gamma), has been used to predict the amount of wear that will occur at the wheel and rail surface. Jendel and Berg [49] have shown that the use of a 'load collective' made up of simulation results from a batch of vehicle dynamic runs can be used in an iterative process to predict the development of worn wheel profiles. Johansson [50] has used wear simulation based on a discretised contact patch to predict the growth of out of roundness and Nielsen [51] has looked at the link between wear on the rail head and growth of corrugation.

8.3 Real Time Simulation

Real Time VAMPIRE is an interesting development which has been made possible by advances in computer technology and the very high simulation speed possible with VAMPIRE. Track data can be fed into VAMPIRE directly from a track recording coach and the behaviour of vehicles on that section of the track predicted in real time. All the typical vehicle dynamics outputs can be generated for the specific vehicle model that has been loaded. This allows derailment risk, passenger comfort, track force or any other normal output to be produced and these values are available immediately for track engineers to use, for example in prioritising maintenance.

9. Conclusions

The development of software tools for simulation of wheel-rail contact and the dynamic behaviour of railway vehicles on track has grown with the development of computing power. From the earliest analogue computers to the modern powerful digital processors, the equations that govern the contact location, pressure distribution and tangential creep forces have been developed and then coded into computer programs. These programs are now combined into a number of powerful and reliable computer simulation packages and several of these have been described above.

These tools have been used to help engineers design, maintain and operate vehicles and to understand the issues around vehicle dynamic behaviour. Further developments in computing power and solution techniques are now allowing the solution of problems involving flexible wheels and rails, contact problems where the surfaces are not smooth surfaces or where the contact is highly conformal or where the higher frequency dynamic motions must be considered. The fastest computer tools now allow simulations to take place in real time so that assessment of control or other operating strategies can now be based on the behaviour of the virtual vehicle before putting it into practice on the real system. - A 126 -

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(K index words: railway vehicle simulation, wheel/rail modelling, computer simulation)

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