DEVELOPMENT OF A METHOD TO PREDICT STRESSES IN RAILS

using ADAMS/Rail and ABAQUS

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Abstract

Multi-body system models of two typical UK railway locomotives running on a range of curves have been simulated in ADAMS/Rail and the predicted wheel-rail contact forces have been used to generate stresses using two ABAQUS finite element models. The predicted forces were validated against test measurements from one site on the UK rail network. The first ABAQUS model represents a global length of railway track with ballast and sleepers. The second model is a more detailed 3 dimensional contact model that includes a section of wheel running on a length of rail. This method allows integration between the two software packages, with dynamic forces from the vehicle simulation being transferred to the FE models. The global track model predicts the bending stresses in the rail while the wheel-rail contact model predicts the contact stresses in the rail head. This information can then be used to develop an understanding of the process of rolling contact fatigue, crack initiation and crack growth.

1. BACKGROUND

A major project involving track studies of rolling contact fatigue (RCF) has been undertaken by Corus Rail Technologies (CRT) with support from Manchester Metropolitan University (MMU). It adopts a practical approach to understand the occurrence of RCF and the behaviour of the track as a system. The objectives of the project are to develop and validate the CRT Track System Model that will be used to evaluate the influence of a range of parameters related to traffic, track design and construction, track integrity and, in particular, the development of RCF. To do this, sites have been selected and monitoring is being carried out using strain gauges and accelerometers to capture wheel-rail forces and rail displacements respectively under various types of traffic. Rail profiles are also being monitored using a MiniProf device, as well as wheel profiles from vehicles running over the sites. Additionally, measurements will be made to determine the static and dynamic responses of the track structure.

2. TRACK SYSTEM MODEL

The Track System Model (TSM) comprises a series of separate numerical and analytical models that have been developed using commercial software packages. Each model is used to assess a specific part of the railway track system. The structure of the TSM is such that the results from one model can be fed into another allowing specific components to be assessed in more detail. The separate models, model developers, and the software used are as follows:

1. MMU Vehicle Dynamics Model – using Adams/Rail
2. CRT Global Model – using Abaqus
3. CRT Contact Stress Model – using Abaqus

4. CRT Detailed Sleeper and Component Model – using Abaqus
5. CRT/Irsid Fatigue Model – using Visual Basic

Together the models can be used as an effective tool for assessing the in-service performance of track components and to optimise the track system. The TSM may also be used as a tool to investigate problems such as rolling contact fatigue (RCF) to identify the root causes of failure and to evaluate the impact of the introduction of new vehicles. A flow chart of the TSM is shown in Figure 1. This paper will only focus on the first three models.

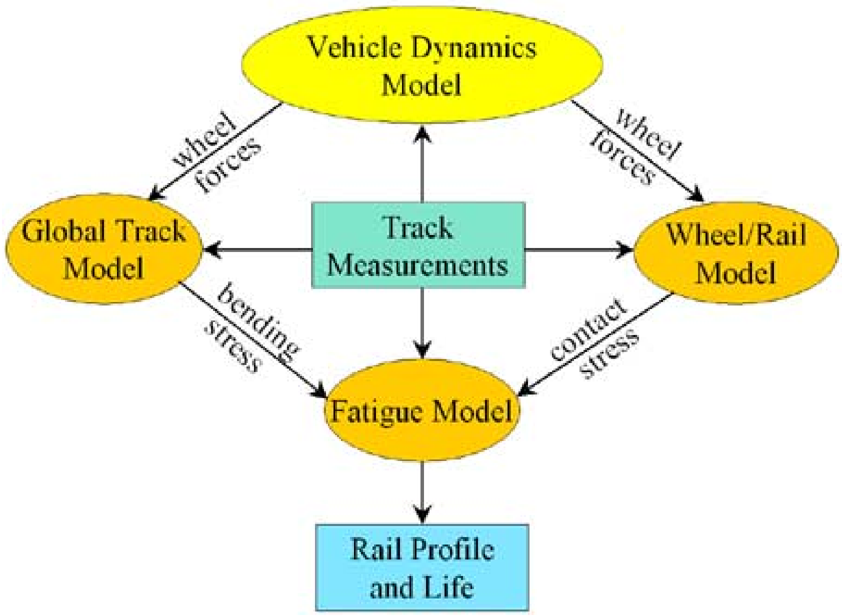

Figure 1, Flow chart of Track System

2.1. Vehicle Dynamics Studies

Two typical UK locomotives were modelled, the Class 91 and the Class 43 as they were the most representative case running on the selected sites. Only the class 91 is presented here.

2.1.1. Railway Vehicle Model

The Class 91 (shown in figure 2a & 2b) is an electric locomotive, with an axle load of 215 kN, a maximum operating speed of 201km/h (designed for 225km/h), generating a tractive effort of 3765 kW at the maximum operating speed.

The primary suspension includes:

- 4 coil springs (per axle)
- 4 vertical dampers (per axle)
- 2 vertical bumpstops (per axle)

The secondary suspension includes:

- 4 coil springs
- 1 traction centre equivalent bush
- 1 anti roll bar equivalent bush
- 2 vertical damper
- 2 lateral dampers
- 4 yaw dampers
- 2 lateral bumpstops

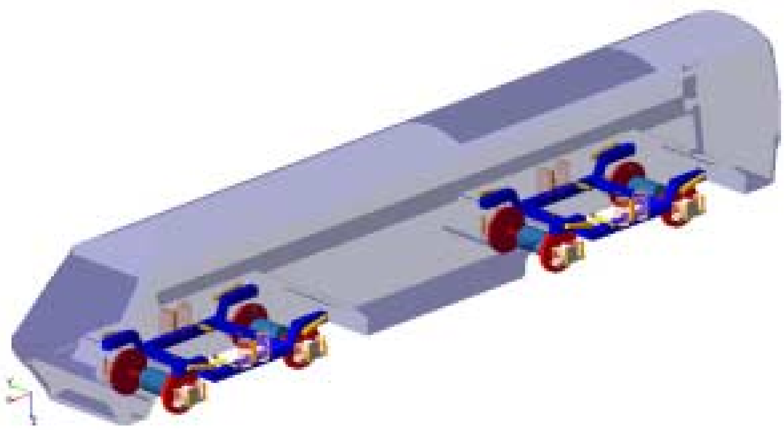

Figure 2a, Class 91 model

A MiniProf device was used to measure the wheel profiles from one axle of each vehicle at the particular site of interest. The profiles were then imported into the Adams/Rail vehicle model.

2.1.2. Model Validation

The model was validated against static test measures reproduced in ADAMS/Rail such as:

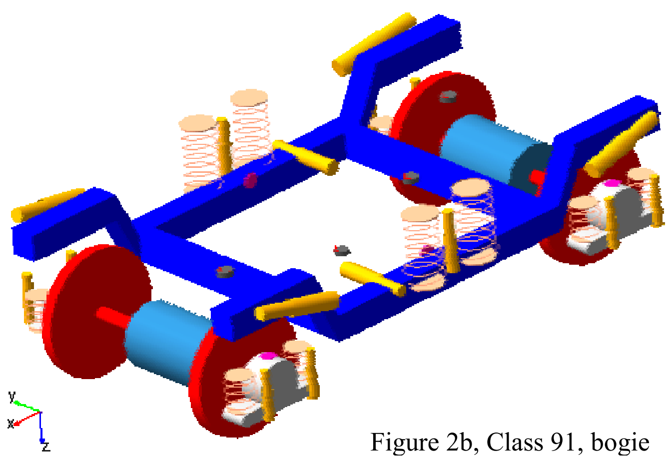

Figure 2b, Class 91, bogie

- ΔQ/Q (wheel unloading test) using specified twisted track geometry.
- X-factor (bogie rotational resistance test).

On-site measurement, presented in the section *2.1.5.*, were also used for validation purposes.

2.1.3. The Site

One site at Aycliffe (East Coast Main Line) was chosen for the modelling exercise in order to validate the model against measured data. The track data was recorded by a track recording coach. This includes the track distance, cross-level & curvature irregularities, lateral & vertical irregularities and gauge variation. The data around the measuring site location was selected and further processed for inclusion in ADAMS/Rail. The general design layout was defined (curvature and cant elevation), and the irregularities were converted to vertical and lateral irregularities at each rail.

The track set up for the simulation is composed of a small length of straight track - transition - curve to the right - transition. Figure 3 shows the track cant angle used for the simulation (equivalent to a 155mm cant elevation) with the approximate location of the track force detector (TFD) site shown. The radius of the curve is 765m. Rail profiles were measured by Corus on site using a MiniProf and a selected profile was used in ADAMS/Rail for both left and right rails.

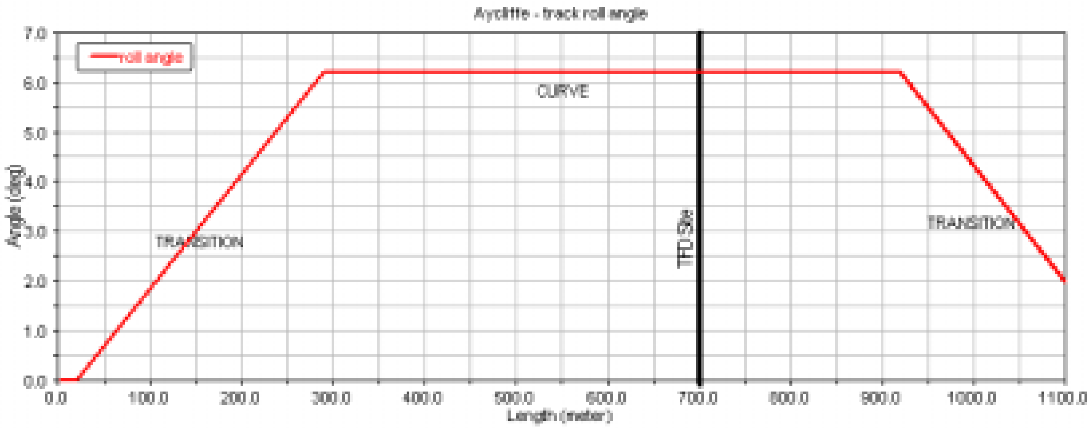

Figure 3, Aycliffe track roll

2.1.4. Simulation Cases

The ADAMS/Rail track flexibility option was used to include the vertical, lateral and roll stiffness and damping of the rails and of the track substructure. The main values were taken from the Eurobalt study (report RR-TCE-35) and others are estimated.

The contact level used in ADAMS/Rail was the 'Tabular' contact which uses a pre-computed kinematic table based on the measured wheel and rail profiles and on a static vertical force. For any lateral shift of the wheel relative to the rail, the following data are stored in a look-up table:

- Relative distance between wheel and rail.
- Rolling radius difference to the nominal rolling radius.
- Contact angle.
- Contact patch ellipse size (semi axis a, b and area).
- Contact point on the rail and on the wheel (in lateral coordinate).
- The contact table is then used on-line during the simulation. The coefficient of friction was 0.45.

The class 91 model was run on the Aycliffe site at a speed of 108.6km/h (30.17m/s), which corresponds to one of the highest speeds collected from the measured data.

2.1.5. Outputs

Forces generated by the class 91 running at Aycliffe were compared with site measurements for validating the model. A 100 metre section of track around the track force detector site was used to extract the mean values for the vertical and lateral forces under all eight wheels. They were compared with the on-site measurement with vehicles running at the same order of speed. The vertical forces at each wheel are within 12.8% of the measured results with the difference in the total force on each side being 2%. The lateral forces, taken individually, are very difficult to compare with measured data because values can vary significantly from one wheel to another. However the orders of magnitude observed give a certain confidence in the predicted results. Figure 4 show an example of the vertical forces predicted for the front bogie.

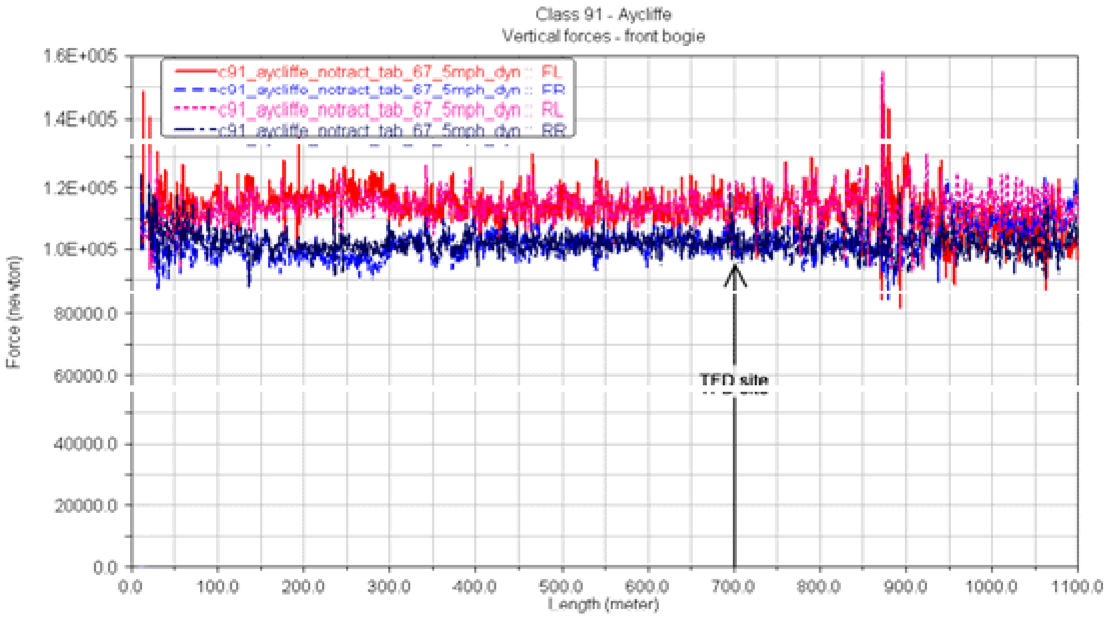

Figure 4, Vertical forces predicted at the front bogie.

The simulation results were then processed to be used by the CRT models. Predicted forces and wheelset positions were used as input to both the global track model and the contact stress model.

Global Model Requirements:

- Vertical forces for all 4 wheels for front and rear bogies.
- Lateral forces for all 4 wheels for front and rear bogies.
- Longitudinal creep forces for all 4 wheels for front and rear bogies.

Contact Model Requirements:

- Vertical and lateral forces.
- Axle torque.
- Lateral position of the wheel relative to the rail.
- Axle roll angle.

2.2. Global Track Model

The objective of the global model is to predict the bending stresses and strains in the rail and sleeper components when subjected to vehicle forces. The results are then used to characterise the stress cycle and therefore provide data to help predict the fatigue life of components. Sensitivity studies can also be performed to assess the effect of using different track geometry and/or component properties, such as sleeper spacing or sleeper type. Additionally, the predicted displacements of the rail and sleeper components can be used to feed into more detailed models ensuring that components react more realistically thus improving accuracy.

The global model refers to an in-house developed finite element (FE) model that represents a section of railway track. It has been developed using the software package MSC Patran and is analysed using Abaqus Standard Version 6.2. A total track length of 50 metres has been constructed enabling the forces from all eight wheels of a single locomotive to be applied simultaneously. These can be applied as a single set of forces, i.e., to produce a 'worst case' snapshot, or as a series of forces to simulate the effect of a vehicle moving over a specified distance. The model incorporates all of the major components associated with ballasted track and includes the rails, sleepers, rail pads and track bed elements. Currently, the global model is a static analysis of the railway track and uses linear elastic values for the track response. However, it can also be modified to simulate dynamic train-track interactions and include both linear and non-linear track responses.

2.2.1. *Track Definition*

The track components are represented in different ways in order to model them in an accurate but efficient manner. The rail and sleeper components are modelled as Timoshenko beam elements, that rely on beam theory to describe the elastic deformation of a particular shape when forces are applied. The beams are one-dimensional line elements in a 3-dimensional space and their stiffness is defined by section properties. Bending stresses can be calculated for the head, web and foot areas of the rail section. The rail pad, ballast and subgrade are represented using spring elements that have an appropriate stiffness as defined by test measurements. Accurate stiffness values are difficult to achieve and therefore values determined from previous projects such as the 'Eurobalt Project' have been used where no other data is available. It is intended as part of this project to determine more realistic values of the track and associated component response at the monitoring sites in order that these can be input into the models. For example, vertical track stiffness values will be determined from using falling-weight deflectometers. A section of the global model and the component layout is shown in Figure 5.

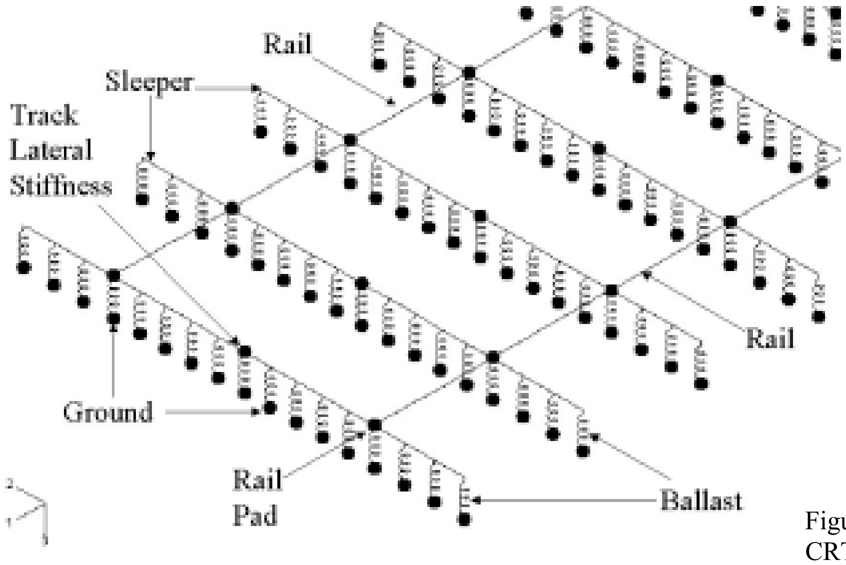

Figure 5, a section of the CRT Global track model

2.2.2. *Global Model Simulation*

The average wheel forces from the Class 91 vehicle simulation were applied in the appropriate positions in the global model. They include both the vertical and lateral wheel-rail contact forces. If necessary, peak forces can also be selected from the graph and applied in the global model simulation. The track and component properties used for the Aycliffe site is shown in Table 1.

Site	Rail Type	Sleeper Type	Rail Pad Type/Stiffness (MN/m)	Ballast Stiffness (MN/m)	Ground Stiffness (MN/m)	Track Lateral Stiffness (MN/m)
Aycliffe	BS113A	F27 Concrete	Rubber/200	80	20	40

Table 1.

2.2.3. Global Model Results

Maximum rail displacement predictions were approximately 1.9mm for a wheel loads of 108 kN. Fig 6 shows the corresponding stress predictions for the head of the rail. These results are very useful for characterising the stress cycle experienced by the rail when a vehicle passes over it. The plot clearly shows compressive stress values occurring under each wheel but also shows the stress rapidly changing to a positive value (tensile) in between the axles. The rail also experiences tensile stresses at the front and rear of the vehicle and also between the bogies.

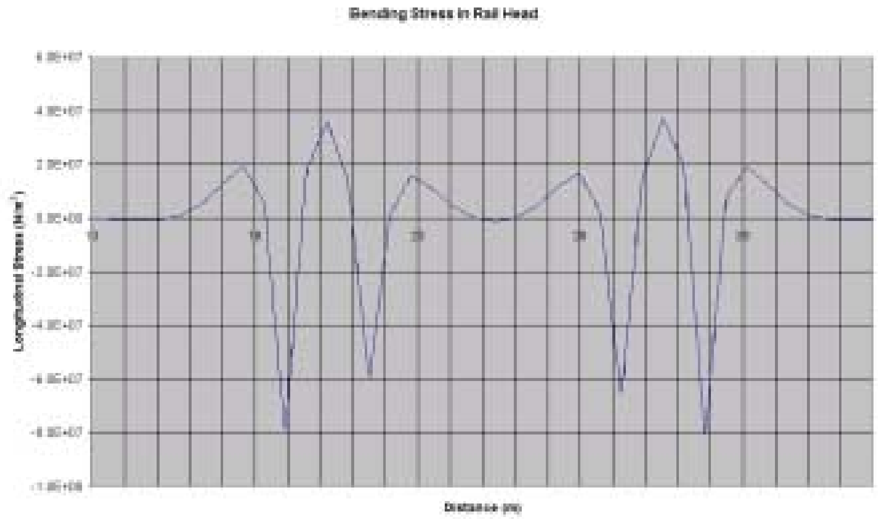

Figure 6, bending stresses in rail head.

The stress values, taken in isolation, are considered to be quite low and well below the yield of rail steel. However, the compressive stresses in the rail head (under each axle) are additive to the compressive stresses caused by the wheel-rail contact and therefore should be included in any assessment. Similarly, tensile stresses are at their maximum at the bogie centres, i.e., in between the axles, and these will interact with both residual stresses and the lane stress. It is these tensile stresses that are considered to be the primary cause of rail fracture once the crack has reached a critical length.

2.3. Contact Stress Model

2.3.1. Model Development

The contact stress model consists of a short length (200mm) of rail head and a segment of wheel rim with both profiles being based on actual measured data rather than using as-new nominal dimensions. The MiniProf data was used to produce an accurate 2D curve that was input into the Abaqus finite element software package. The next stage of the process was to convert the 2D geometry into 3D solid models of the two components. This was accomplished using Abaqus CAE. As well as the detailed geometry of the wheel and rail the program requires the material properties for the two components to be specified. The rail in question was manufactured from Mill Heat Treated steel and the corresponding true stress-true strain data was used in the analysis. Another important feature of this phase of the model building process is determining the relative location of the two components. During cornering the centripetal forces on the vehicle cause the wheelset to move laterally and the contact patch between the rail and wheel to move accordingly from this central position. The position of the contact patch was provided by the MMU vehicle dynamics model. Figure 7 illustrates the position used for the model.

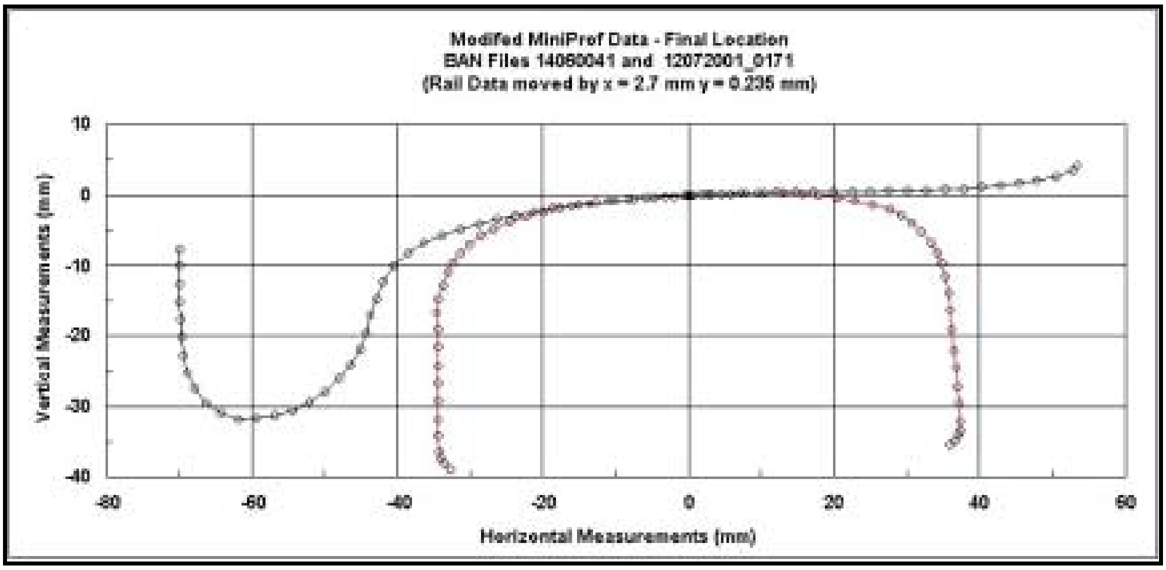

Figure 7, contact position modelled

The next stage of the model development involves dividing each component into a mesh of discrete elements that can then be used in finite element analysis. This inevitably involves some compromise. In order to reflect the complex geometry of the two components a fine mesh is necessary and this, in turn, adds to the size of the problem both in processing time and the space required to store the results of the analysis. Figure 8 shows the finite element mesh of the rail wheel assembly used in the analysis.

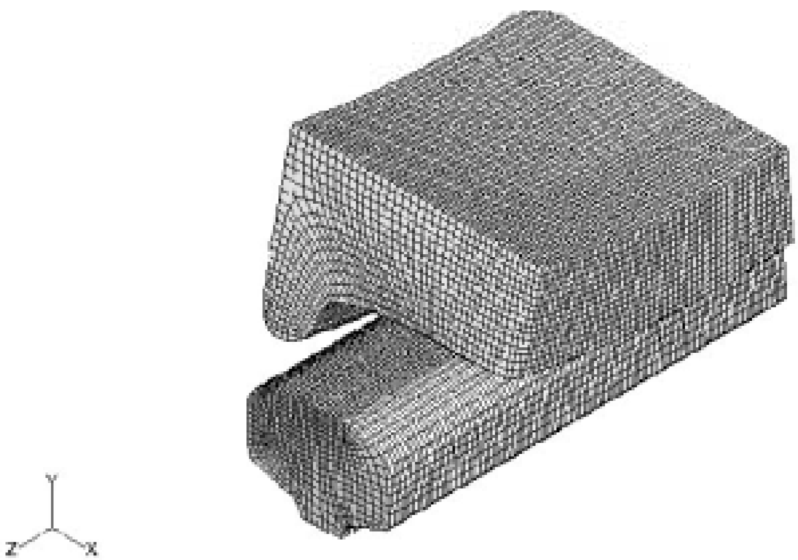

Figure 8, FE Model Wheel and Rail Profile

The mesh has been refined to give a greater element density in the areas where contact is likely. All of the elements used in this work are 8-noded linear solids.

2.3.2. *Finite Element Analysis*

The loading information was supplied from the vehicle dynamics studies conducted in ADAMS/Rail at MMU. A snapshot from the time history was taken around the TFD site. The loads were applied to the wheel vertically and horizontally together with a driving torque. The analysis was conducted as a dynamic analysis using Abaqus Explicit during which the segment of wheel rim is rolled along the head of the rail under the appropriate loading and torque. The speed of the locomotive was assumed to be constant and a value of 0.45 taken as the coefficient of friction between the two components.

2.3.3. Results & Outputs

Figure 9 shows the contact pressure that has been generated between the two components. The diagram illustrates that the contact patch is spread over a complex shape with high contact pressures located in two places. Maximum contact pressure reaches 993 N/mm². Figure 10 illustrates the vertical stress calculated for the rail in the immediate vicinity of the rail/wheel contact area. It shows again the stress, spreading away from the two locations at the surface of the rail, in an oval shape.

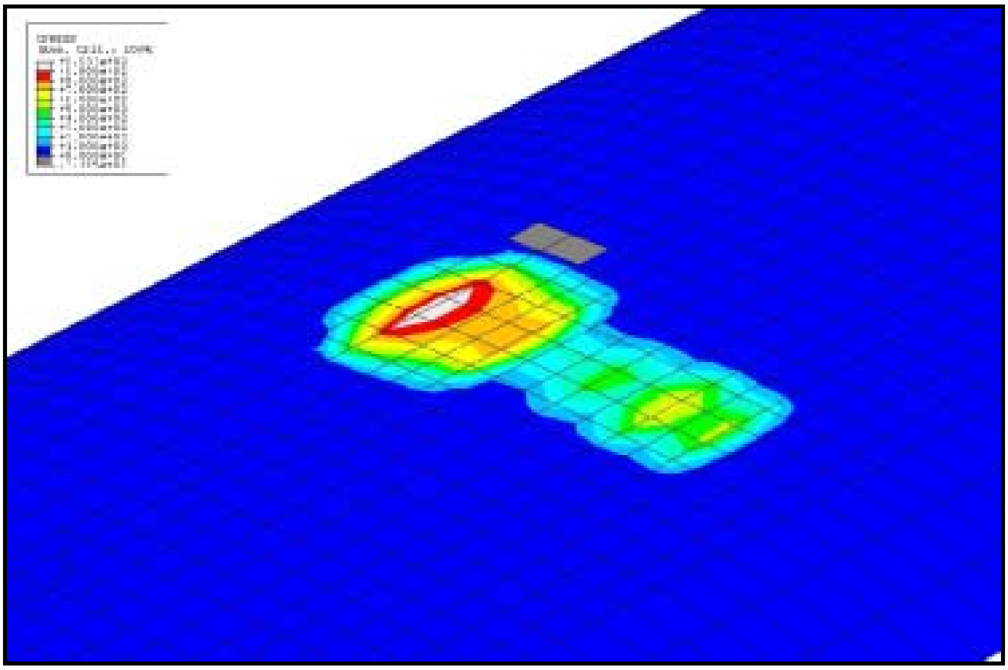

Figure 9, Contact pressure

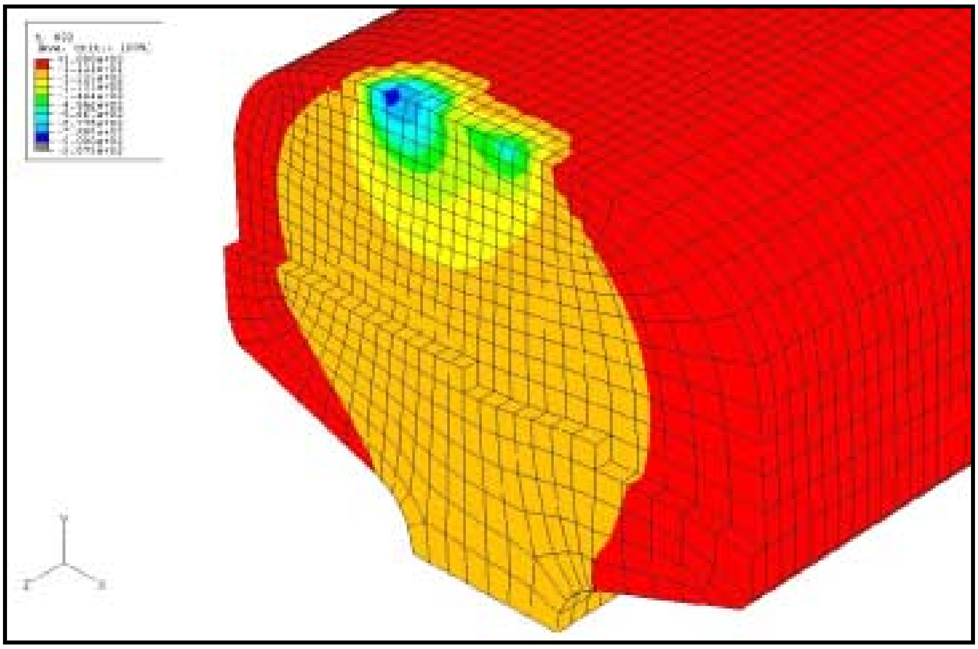

Figure 10, Vertical stress

Figure 11 shows the longitudinal shear stress through the centre of the contact patch in the rail component. The maximum shear stress of 172 N/mm^2 occurs at the front of the contact patch (appears in red in the picture) whilst the minimum value at the rear of the contact patch is –235 N/mm^2 (appears in blue in the picture). It is important to note that these maximum and minimum values occur below the surface of the rail. This is because the rail material is extended in front of the wheel and compressed behind it, as it is driven. It is believed that these longitudinal shear stresses have a significant influence on the development of 'cracks' once they reach a certain depth.

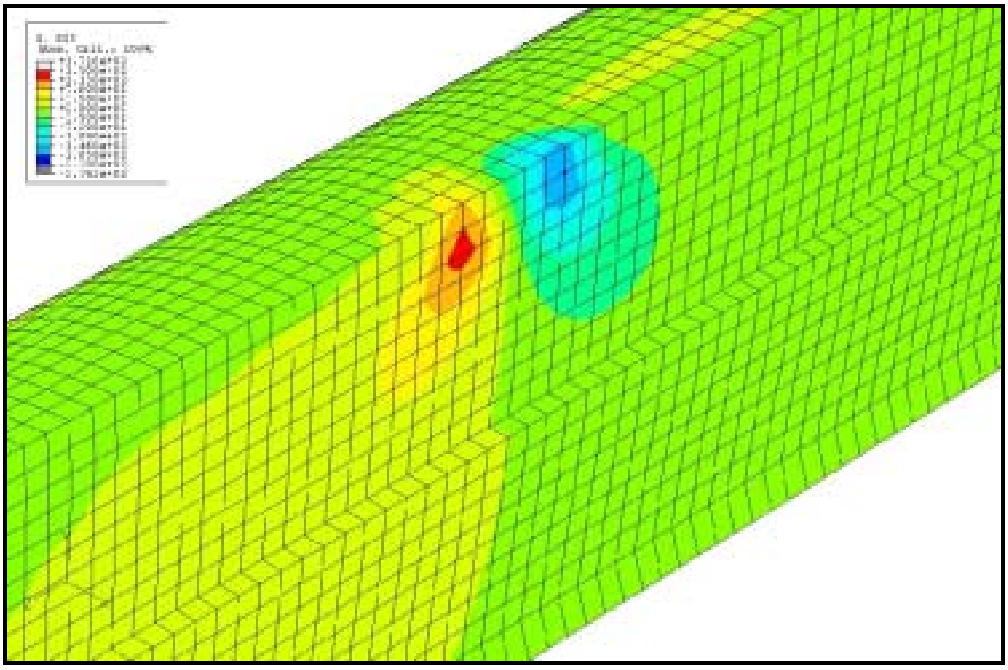

Figure 11, Longitudinal shear stress

3. CONCLUSIONS

The forces predicted from the vehicle dynamic modelling using Adams/Rail are in broad agreement with measured data from on site instrumentation. The predicted forces can then be used with confidence in both the global track and contact stress models.

The global model has predicted the typical response of the rail when subjected to vehicle forces. It has clearly shown how the rail component experiences a number of tensile and compressive cycles every time a vehicle travels over it. In isolation the stresses could be considered as relatively low. However, these stresses will play an important role in the component life cycle, particularly if a RCF crack is allowed to become too large. It is essential, however, that the global model is validated against on-site measurements in order to build confidence in the results it is predicting.

The CRT contact stress model has shown that a complex shaped contact patch can be produced and that two areas of high contact pressure may occur. The model provides a distribution of all stresses through the section of the rail head and highlights maximum sub surface shear stresses. The magnitude and directions of stress should therefore facilitate the understanding of the early growth of fatigue cracks.

Monitoring of the current sites continues and additional CRT sites will be instrumented. The measurements will be used to further validate both the vehicle and track system models. Both the CRT global and contact stress models will be used to predict RCF crack initiation and crack growth.

Eventually this work will allow further benchmarking of the ADAMS wheel-rail contact prediction such as contact patch position, stress prediction (using the normal force and the contact patch area). Such exercise would require the use of the full non-linear contact definition in ADAMS/Rail to take into account for multi-point contact and non-elliptical contact patch shape.

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