Finite Element Analysis of Rail Wheel Interaction

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Abstract-Mechanics of the Rail–Wheel contact is one of the fundamental areas of the study in Railway Engineering, requiring both vast application expertise and dependable analysis approaches. The contact area and pressure distribution in a wheel/rail contact is essential information required in any fatigue or wear calculation and to determine design life, regrinding and maintenance scheme. The main aim of this project is to understand, formulate and simulate wheel-rail interaction analysis at static conditions with a view to optimize the wheel and to optimize the mass of the railway wheel subjected to both vehicle load and contact pressures. In railway, engineers applied one of the numerical computation techniques known as Finite Element Analysis (FEA) into Rail–Wheel contact problems to validate their results by comparing them to their real life data obtained over the years. So we have used software ANSYS in this research for Finite analysis of rail wheel interaction.

Keywords: Rolling Contact Fatigue, Adhesive Wear and Preventive Maintenance, Pressure distribution, Elliptic contact area, Finite element analysis, Rail Wheel profile & Stresses.

I. Introduction

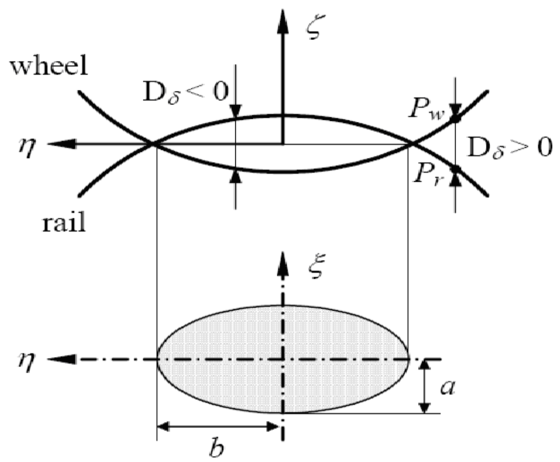
Wheel-rail analysis has focused mainly on what is known as rolling contact fatigue. A simulation of a so-called ratchetting model using finite elements carried out by Ringsberg et al. [1] identified asymptotic values of the friction coefficient at which crack initiation would occur. Continuum rolling contact theory started with a publication by Carter [2], in which he approximated the wheel by a cylinder and the rail by an infinite half-space. The analysis was two-dimensional and an exact solution was found. Carter's theory is adequate for describing the action of driven wheels (for example, it is capable of predicting the frictional losses in a locomotive driving wheel). However, it is not sufficient for vehicle motion simulations that involve lateral forces as well as the motion in rolling direction [3].

In the contact zone between railway wheel and rail the surfaces and bulk material must be strong enough to resist the normal (vertical) forces introduced by heavy loads and the dynamic response induced by track and wheel irregularities. The tangential forces in the contact zone must be low enough to allow moving heavy loads with little resistance, at the same time the tangential loads must be high enough to provide traction, braking, and steering of the trains

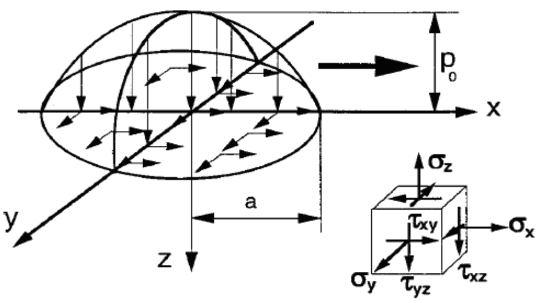
The contact zone (roughly 1cm^2) between a railway wheel and rail is small compared with their overall dimensions and its shape depends not only on the rail and wheel geometry but al soon how the wheel meets the rail influence, i.e., lateral position and angle of wheel relative to the rail, as shown by LeThe Hung[4]

It is difficult to make direct measurements of the contact area between the wheel and the rail. An interesting approach for measuring the contact area for full-scale worn wheel and rail pieces is presented by Marshall et al[5]. The surface topographies of the ultrasonic measured surfaces were measured with a stylus instrument and used as input to a contact mechanics method for rough surfaces (for details see Bjo¨rklund et al[6].). Poole[7] used low-pressure air passing through 1mmdiameter holes drilled into the rail head to measure the contact area as the holes being blocked by the passing wheel. Mehmet Ali Arslan ,Og˘uz Kayabas [8] show Mechanics of the Rail–Wheel contact is one of the fundamental areas of the study in Railway Engineering, required both vast application expertise and dependable analysis approaches. Analytical formulations described the physics of this phenomenon are only defined for certain type of simple contact geometries, thus more complicated geometries the analytical models utilizing closed formulations remain elusive. Remaining option is utilized numerical computation methods. Martina Wiest et al [9] Observed that in a fully three-dimensional is developed a dynamic model using an explicit finite element program , for a wheel running over a crossing . They considered the full mass of the wheel and the crossing and elastic-plastic material behaviour. Hiromichi Kanehara, Takehiko Fujioka [10] Observed that it is difficult to detect lateral contact point of rail and wheel, though technology of instrumented wheel set has advanced. They tried to develop a method of measuring rail/wheel contact point by improving conventional method of measuring wheel load and lateral force in which strain of the disk surface is used for measuring these forces.

II. Wheel-Rail Interaction Analysis:



Elliptical contact area between wheel and rail interaction



The contact pressure and stress distribution at the

A. *Static Analysis Case*

The wheel-rail contact can be described by the general case of elliptic contact area. Considering the a and b semi-axles of the ellipse formed in the contact region of two bodies with arbitrary curvature, the pressure distribution has the given by

$$p_z(x, y) = p_0\left\{1-(x/a)^2-(y/b)^2\right\}^{1/2} \qquad (2.1)$$

This pressure acting on the elliptic region by:

$$(x/a)^2+(y/b)^2-1=0 \qquad (2.2)$$

The classical approach, using the potential functions of Boussinesq, is usually followed

This pressure produces displacements within the ellipse given by

$$\phi(x, y, z) = \iint_s \left\{1-(\xi/a)^2-(\eta/b)^2\right\}^{1/2} R^{-1} d\xi d\eta \qquad (2.3)$$

Where R is the radius of the curvature.

B. *Elliptic contact area*

$$\delta - Ax^2 - By^2 = \frac{1}{\pi E^*}\left(L - Mx^2 - Ny^2\right)$$

Where $A = \frac{M}{pE^*}$ $B = \frac{N}{pE^*}$ and $d = \frac{L}{pE^*}$ are geometric constants

$$A = \frac{p_0}{E^*}\frac{b}{e^2a^2}\{K(e) - E(e)\}$$

$$B = \frac{p_0}{E^*}\frac{b}{e^2a^2}\left\{\frac{a^2}{b^2}K(e) - E(e)\right\}$$

$$\delta = \frac{p_0}{E^*}bK(e)$$

Where $K(e)$ and $E(e)$ are complete elliptical integrals of argument and the pressure distribution from the known volume of an ellipsoid

$$e = \left(1 - b^2/a^2\right)^{1/2} b < a$$

The total load acting on the ellipse is given by:

$$N = 2\pi abp_0/3$$

The shape and size of the ellipse of contact,

$$\frac{B}{A} = \left(\frac{a}{b}\right)^2 \frac{E(e) - K(e)}{K(e) - E(e)} = \left(\frac{R^'}{R^{''}}\right)$$

Where $R^'$ and $R^{''}$ are the principal radii of curvature of the surface at the origin

Equivalent radius is given by $R_e = \left(R^' R^{''}\right)^{0.5} = \frac{1}{2}(AB)^{-0.5}$

$$(AB)^{0.5} = \frac{1}{2}\left(\frac{1}{R^'R^{''}}\right)^{0.5} = \frac{p_0}{E^*}\frac{b}{a^2b^2}\left[\left\{(a/b)^2 E(e) - K(e)\right\}\right]^{1/2}$$

And by assuming $c = (ab)^{0.5}$ and by substituting for p_0 in this equation we will get

$$c^3 \equiv (ab)^{3/2} = \left(\frac{3NR_e}{4E^*}\right)\frac{4}{\pi e^2}\left(\frac{b}{a}\right)^{3/2}\left[\left\{(a/b)^2 E(e) - K(e)\right\}\right]^{1/2}$$

$$c = (ab)^{1/2} = \left(\frac{3NR_e}{4E^*}\right)^{1/3} F_1(e)$$

$$F_1(e) = \left(\frac{4}{\pi e^2}\left(\frac{b}{a}\right)^{3/2}\left[\left\{(a/b)^2 E(e) - K(e)\right\}\right]^{1/2}\right)^{1/3}$$

The compression is obtained as

$$\delta = \frac{3N}{2\pi abE^*}bK(e)$$

$$= \left(\frac{9N^2}{16E^{*2}R_e}\right)^{1/3}\frac{2}{\pi}\left(\frac{b}{a}\right)^{1/2}\{F_1(e)\}^{1.5}K(e)$$

From this normal force acting in the wheel is given by

$$N = K\delta^{3/2}$$

Where K is stiffness of the wheel surface of the elliptical contact region.

C. Tangential contact forces resulting from the wheel -rail interaction

The Hertz theory does not consider the surface shear traction p that, in terms of railways ,is called tangential traction

$$p = \{T_\xi \quad T_\eta\}^T$$

Where T_ξ and T_η are the longitudinal and lateral components of tangential traction exerted on the wheel at the contact point, defined as:

$$T_\eta \equiv T_\eta(\xi,\eta) = -\tau_{\xi\zeta}$$

$$\text{at}\,\zeta = 0$$

$$T_\xi \equiv T_\xi(\xi,\eta) = -\tau_{\xi\zeta}$$

The tangential traction vanishes on the surfaces of the bodies outside the contact area **C** . But, inside **C,** the tangential traction is governed by coulomb's law of dry friction, which relates the slip of the wheel over the rail with the tangential traction.

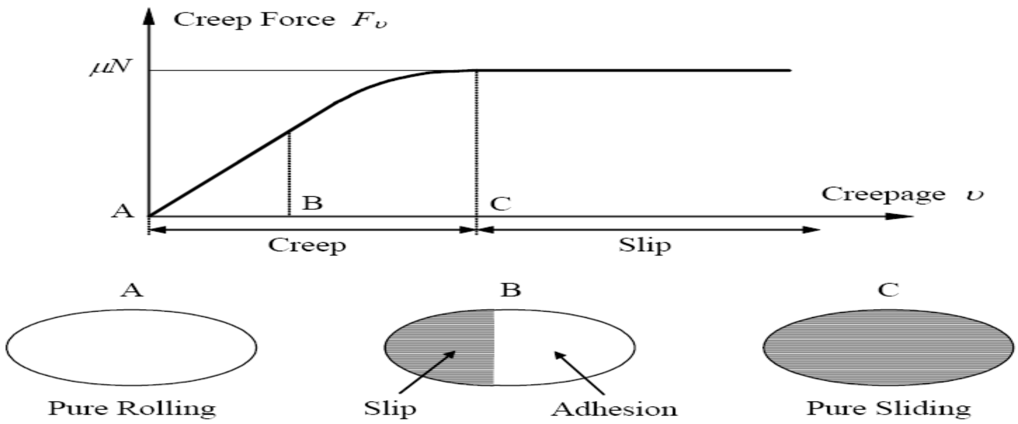
As a consequence of compressive and frictional forces in the contact region, deformations occur in the wheel and rail surfaces. These deformations referred with respect to a reference coordinate system that moves with the contact point as shown in Fig

The wheel –rail rolling contact problem can be formulated as follows: for a given slip $\dot{w}$, determine the tangential traction $p = \{T_\xi \quad T_\eta\}^T$ and, in particular, the longitudinal F_ξ and lateral F_η creep forces, as well as the spin creep moment M_ϕ are given as:

$$F_\xi = \iint_C T_\xi d\xi d\eta$$

$$F_\eta = \iint_C T_\eta d\xi d\eta$$

$$M_\phi = \iint_C (\xi T_\eta - \eta T_\xi) d\xi d\eta$$



Creep forces vs creep age characteristic curve and the area of adhesion and slip

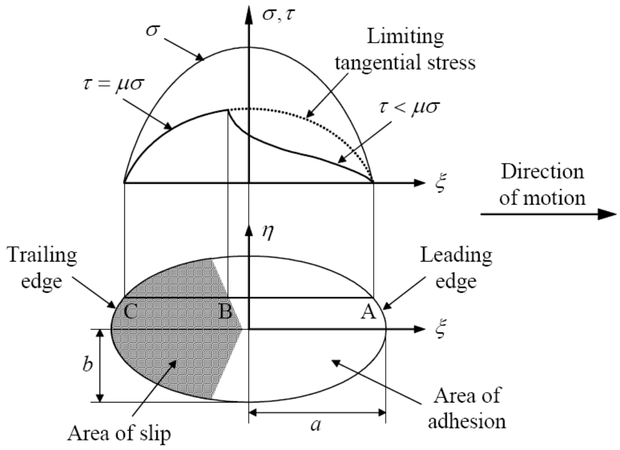
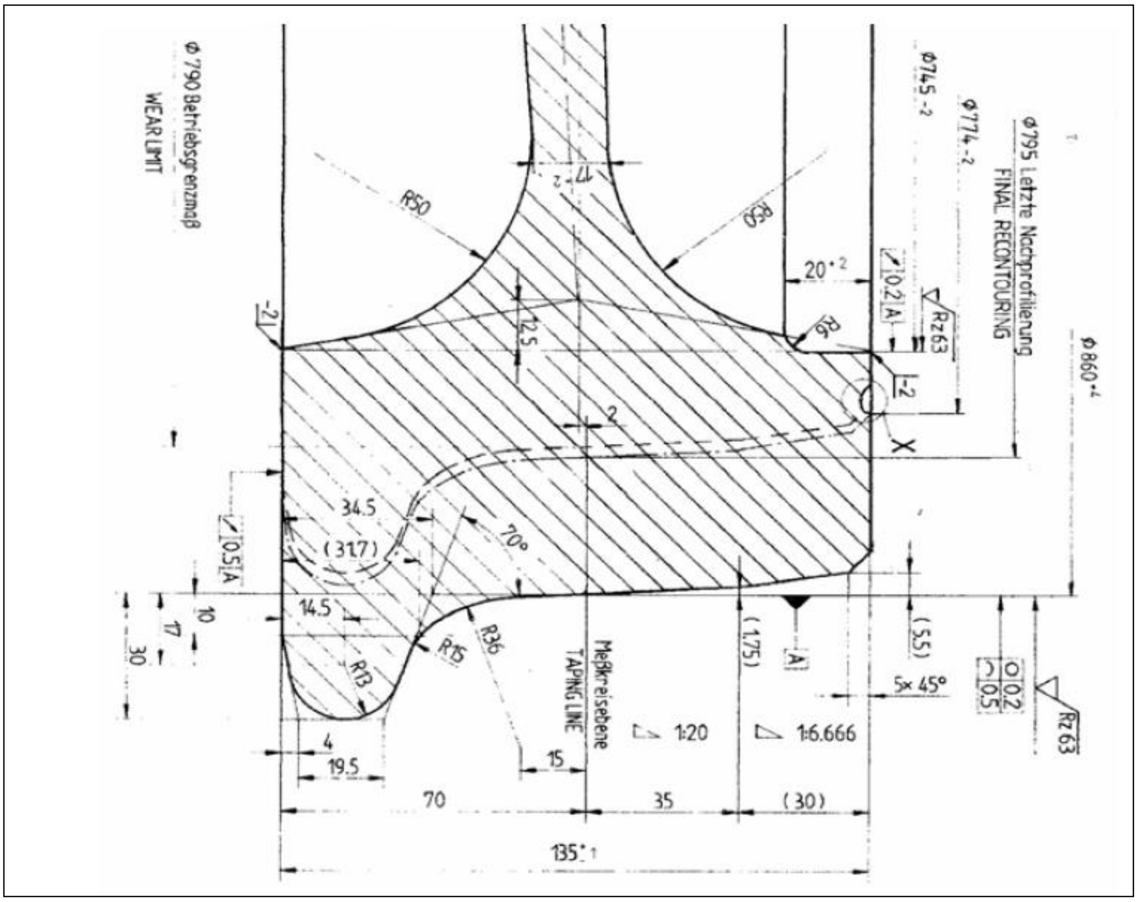


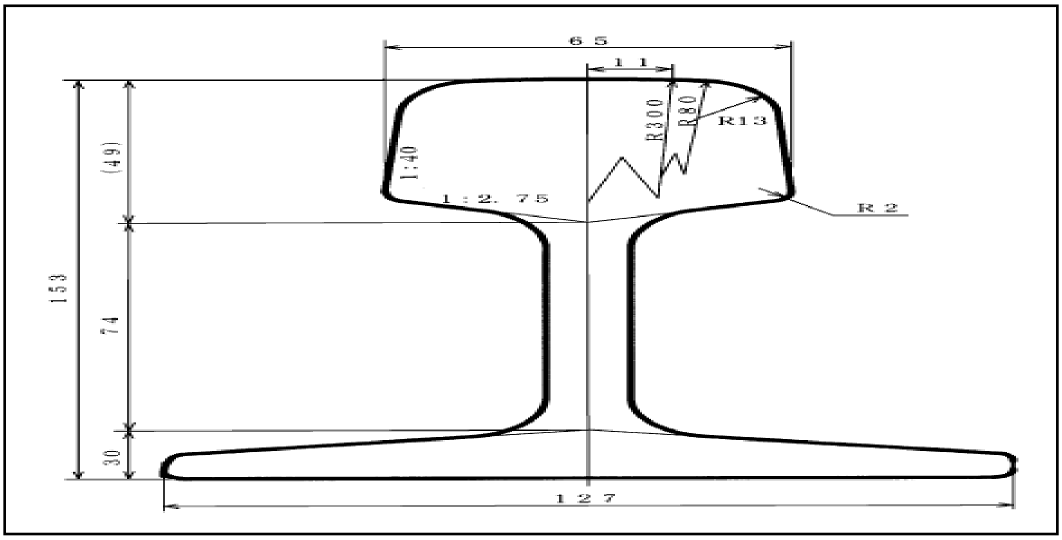
Fig. A-Distribution of normal and tangential in adhesion and slip contact area

III. Modelling of rail and wheel profile

A. ***Drawings of wheel and rail profiles***



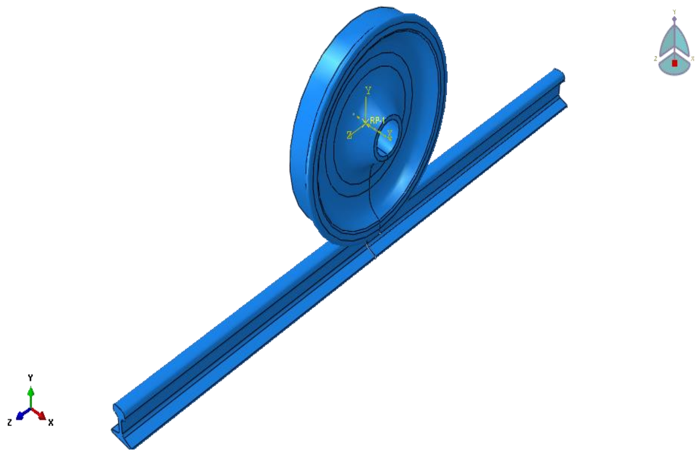
Railway wheel profile



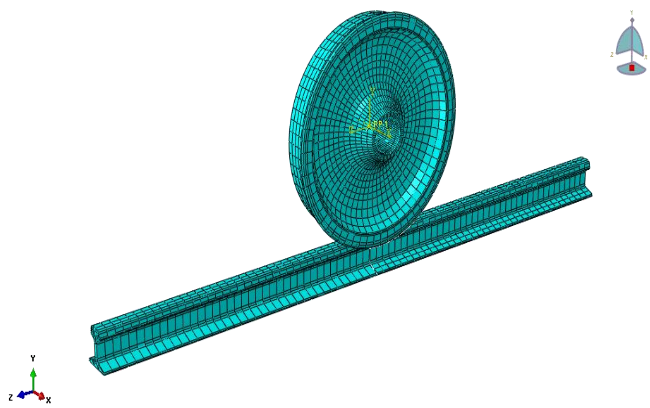
Rail Profile

IV. Finite Analysis of rail-wheel interaction

A. Assembly and reference plane in rail wheel interaction

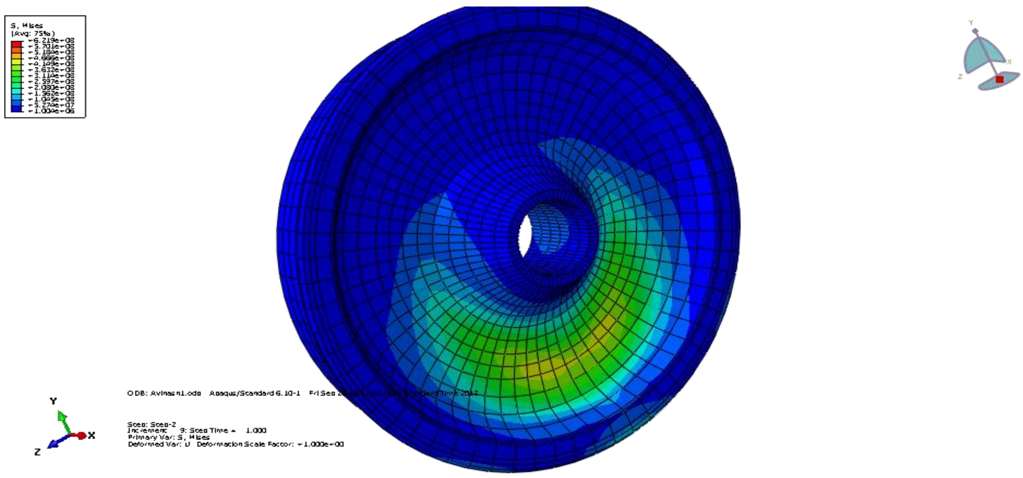


B. Meshing of the rail-wheel interaction model

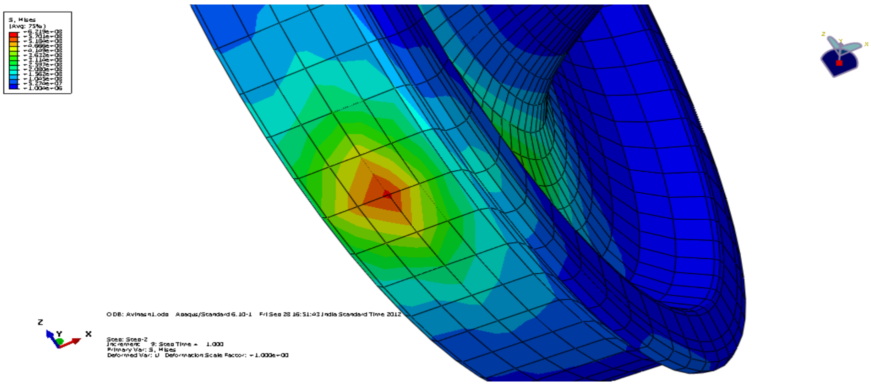


C. Stress obtained in rail-wheel profile

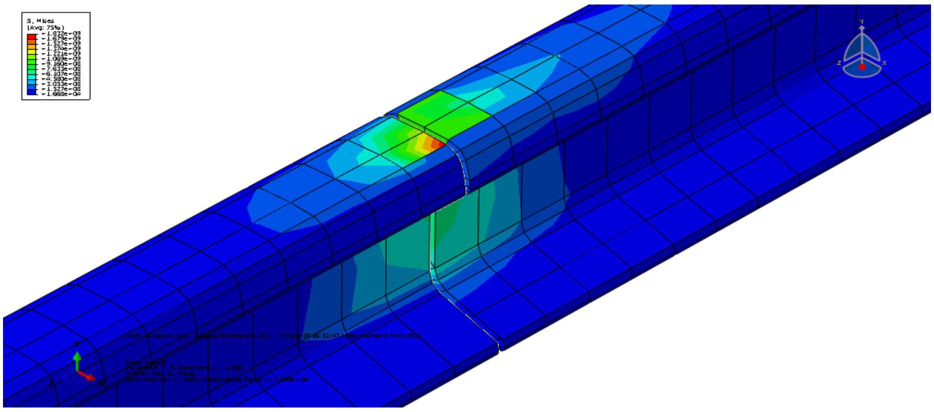
<u>Wheel Profile:</u>



<u>Elliptic contact area obtained in contact region of wheel:</u>

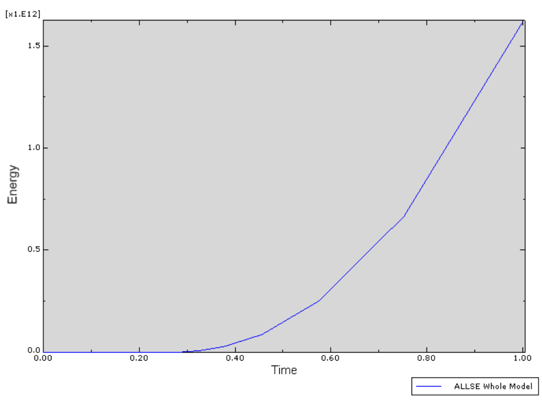


Rail Profile

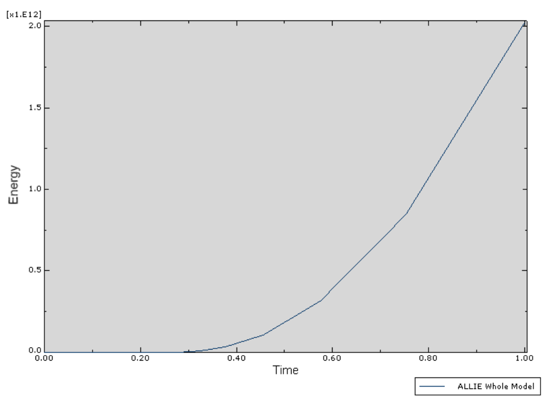


Graph obtained:

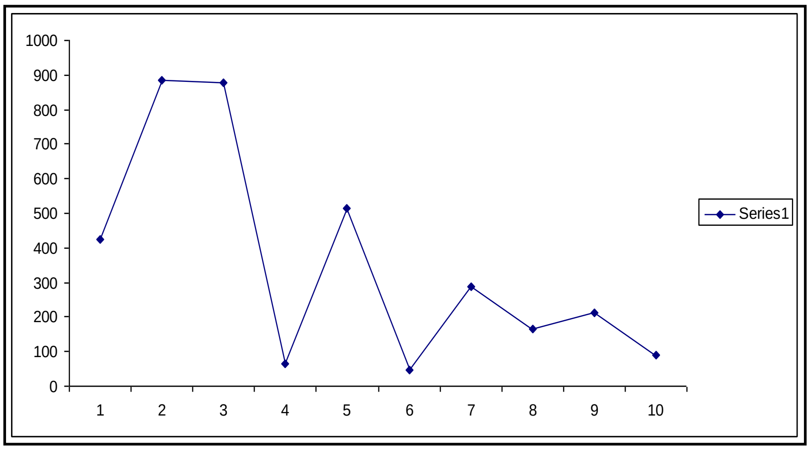
Strain vs Time



Internal Energy vs Time



Contact pressure distribution along the elements



V. Conclusion and Proposed work

A. Static analysis

The static analysis for the railway wheel is carried out by applying the axial load of 18000 kN .Yield strength of the material is 460MPa.

By using the symmetry of the wheel, this geometry is modelled with quarter part of the of the wheel. The axial load of 18000 kN applied at the wheel tread and encaster boundary conditions applied at the hub surface.

1. From static analysis it can be seen that the contact area depends only on the normal load.
2. The Creep age forces acting at the rail and wheel contact region will decides the amount of torque required to turn and steer the train.
3. The speed of the train largely depends on the rail and wheel interaction forces and coefficient of friction in between them.
4. The contact between the rail and wheel has an elliptical contact area, which is confirmed from contours obtained from the simulation.
5. The stresses generated in the contact region are more than the yield limit, the area in the contact region will plastically deform.
6. The above facts can be used for a realistic optimization problem of railway wheel shape and size.

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