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Finite Element Analysis of Rail Wheel Interaction

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Abstract*: The rail wheel interaction is one of the important mechanics for 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 objective 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 Fatigure, Adhesive Wear, Surface Defects, Preventive Maintenance, Pressure Distribution, Elliptic Contact Area, Tangential Force, Continumm Rolling Theory, Hertz Theory, Surface Parameterization , Numerical Computation Techniques

I. INTRODUCTION

This research paper includes wheel-rail contact analysis based on Rolling contact fatigue. This phenomenon arises primarily where there are contacts with low relative sliding, such as on the rail head on straight tracks. A simulation of a so-called ratcheting model using finite elements carried out by Ringsberg et al. [1] identified asymptotic values of the friction coefficient at which crack initiation would occur. In northern Sweden, 60% of rail replacements were found to be due to problems caused by rolling contact fatigue and surface defects, while only 5% were due to flange wear [2]. Preventive grinding, in which rails are ground at regular intervals, is one method adopted by rail operators to extend the service life of rails. A model of the effects of adhesive wear makes it clear why preventive grinding is necessary. Continuum rolling contact theory started with a publication by Carter [3], 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 formulated a creep-force law relating the driving–braking couple and the velocity difference. 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 [4]. Adhesive wear occurs primarily under sliding conditions, where asperities are beating each other under a transient load and stress-related effects may also be present. Vermeulen and Johnson [5] gene realized this theory to elliptical contact areas. Shen et al. [6] improved the results by replacing the approximate values for the creep coefficients given by Vermeulen and Johnson with more accurate values. All of this work is Hertzian-based, giving contact solutions for a class of geometrical objects satisfying the half-space restriction [7]. 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[8]. The contact zone (roughly 1cm2) 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[9]. Bulk material must be strong enough to resist forces generate by 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

II. STATIC ANALYSIS

The size and shape of the contact zone where the railway wheel meets the rail can be calculated with different techniques. Traditionally, the Hertz theory of elliptical contacts has been used implying the following assumptions: the contact surfaces are smooth and can be described by second degree surfaces; the material model is linear elastic and there is no friction between the contacting surfaces; and the contacting bodies are assumed to deform as infinite half spaces.

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

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

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

$$(2.3)$$

Where R is the radius of the curvature.

A. 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 $R_e = \left(R'R''\right)^{0.5} = \frac{1}{2}(AB)^{-0.5}$

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

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.

B. 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:

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

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

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

III.DEFINITION OF WHEEL AND RAIL SURFACES

In the multi body contact model is required to solve the problem of the wheel –rail interaction, it is necessary to define the parametric form of geometry of the surfaces that are in contact. Here the definition needs to satisfy two main requirements.

First the surfaces have to be defined in a global coordinate system since the equations of motion of multi-body system are written with respect to the global inertial frame.

The definition needs in the sense that the parametric equations must be able to represent any spatial configuration of the wheels and rails and wheel and rail profiles obtained by the direct measurements or with the design requirements.

A. Rail Surfaces in a General Track

The surface geometry of the rail is described using the two surface parameters s_r and u_r. The arc length of the rail surface curve, denoted by s_r, defines the rail cross section on which the contact point lies. The parameter u_r defines the lateral position of the contact point in the rail profile coordinate system (ξ_r, η_r, ζ_r). u_r is used to define the lateral position rail profile curve at each cross section as shown in figures

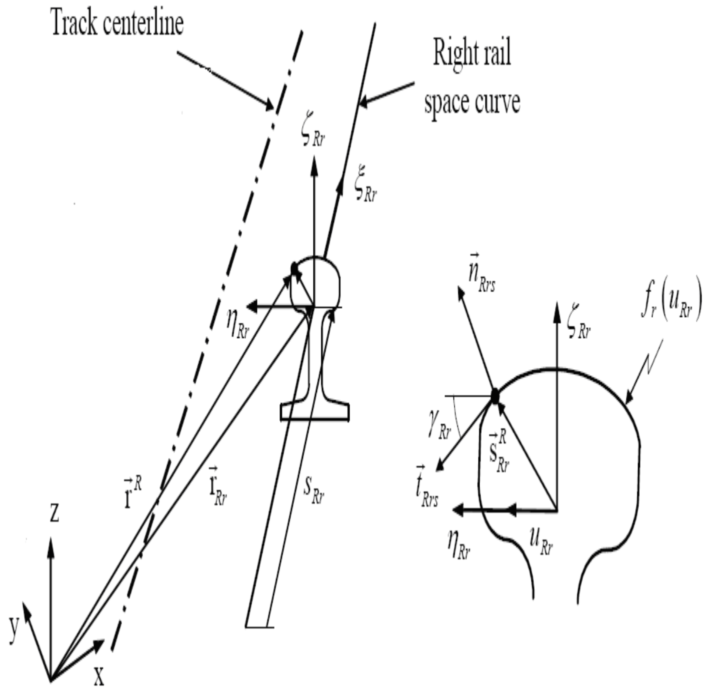

Fig .1-Parameterization of the rail surface

B. General wheel surface

The surface geometry of the wheel can be described using the two surface parameters s_w and u_w. The surface parameters s_w represent the rotation of the wheel profile coordinate system (ξ_w, η_w, ζ_w) at the centre of the wheel. It defines the rotation of the contact point as shown in fig above. The parameter u_w not only defines the lateral position of the contact point in the wheel profile coordinate system but also defines the wheel profile curve at each cross-section as shown in figure2.

C. Identification of the Wheel-Rail Profile Points

The minimum distance conditions in equation are not sufficient to in the candidates to contact points between wheel and rail surfaces. In fact the geometric equations do not cover all possible situations that may occur in the wheel-rail contact problem. The possible positions are given in the fig3. where i and j represents the wheel and rail surfaces

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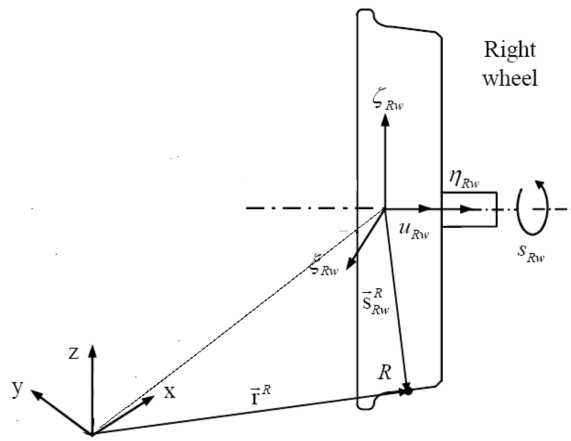

Fig.2 -Parameterization of the wheel surface

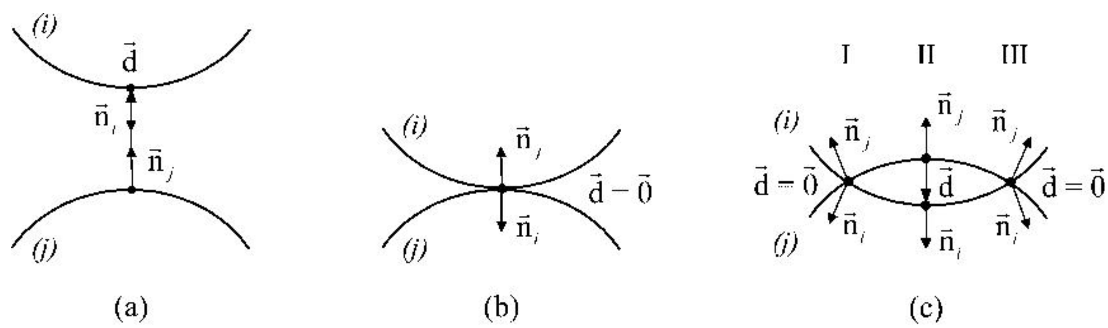

FIG.3 Wheel-rail contact situations: a) No contact b) Contact at a single point without penetration c) Contact with penetration

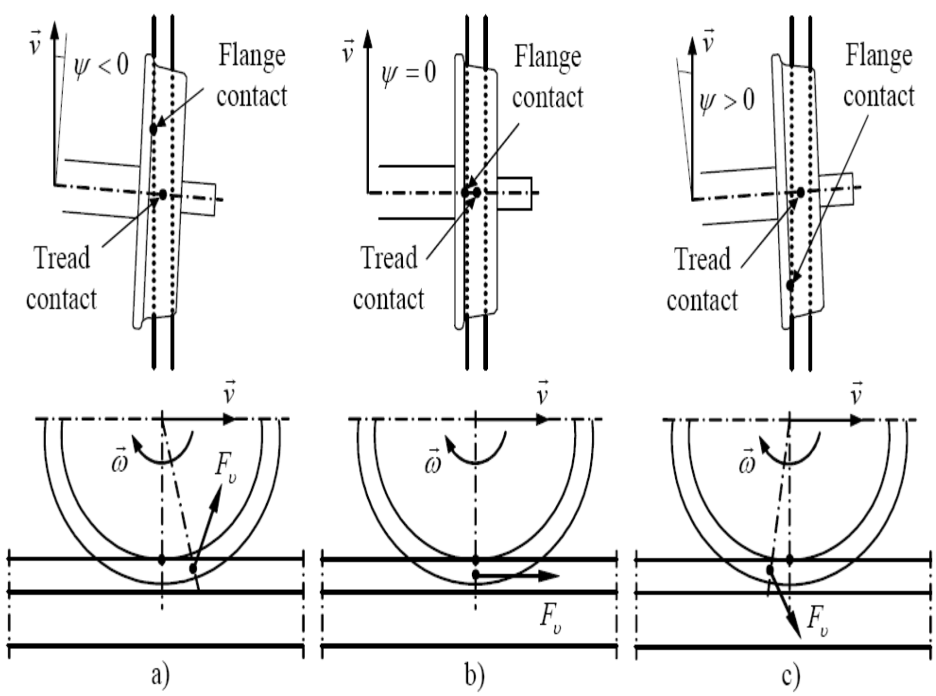

Fig .4- Different contact positions

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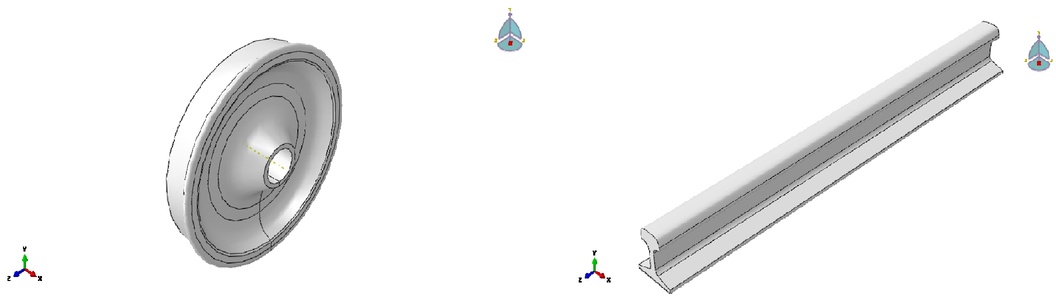
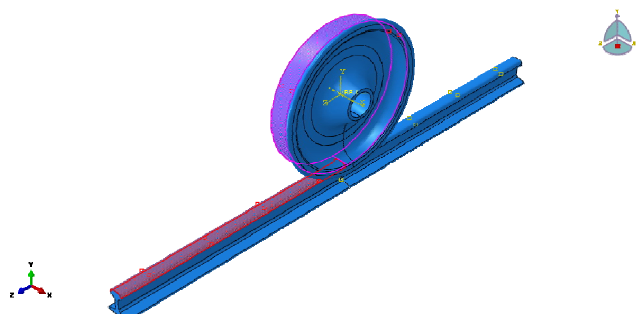
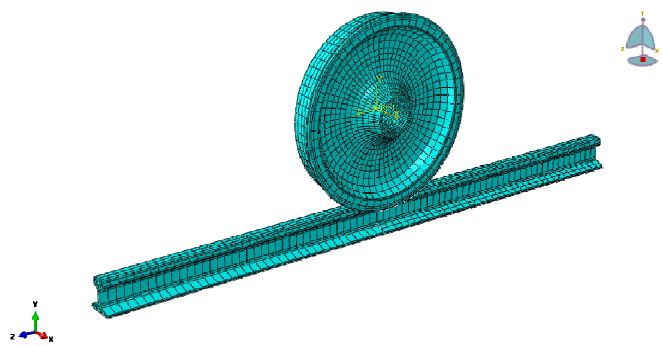
D. Modelling of Rail and Wheel Profile



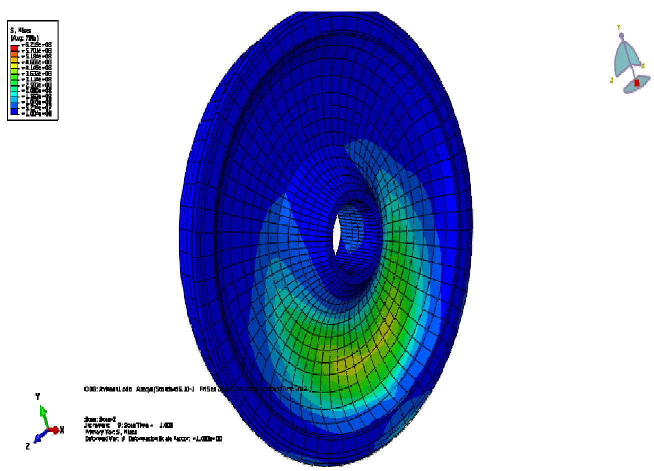
E. Finite Analysis of rail-wheel interaction



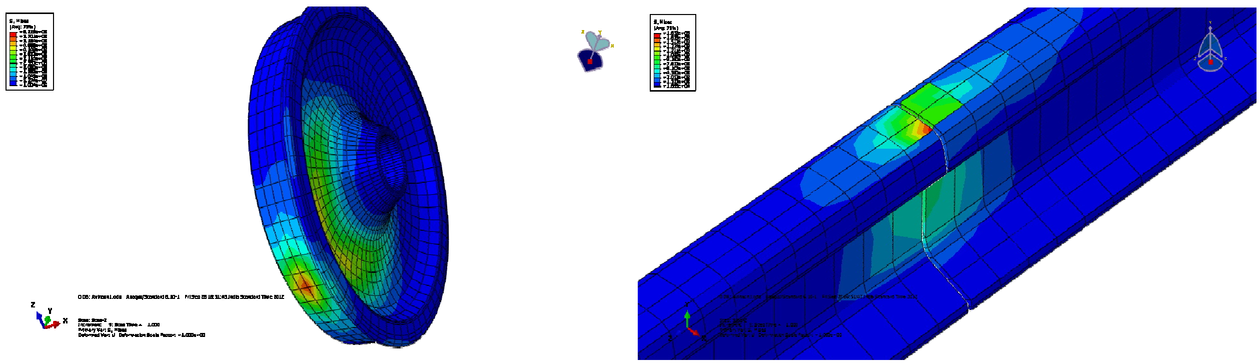
F. Meshing Of The Rail-Wheel Interaction Model



G. Stress obtained in rail-wheel profile

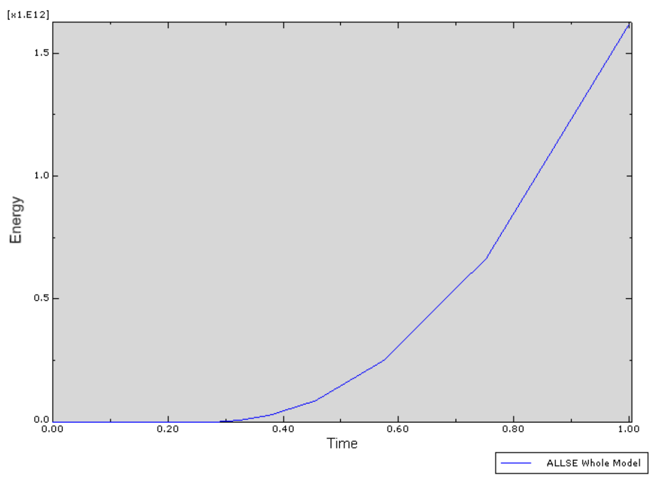


H. Elliptic Contact Area Obtained in Contact Region of Wheel Through Ansys Software

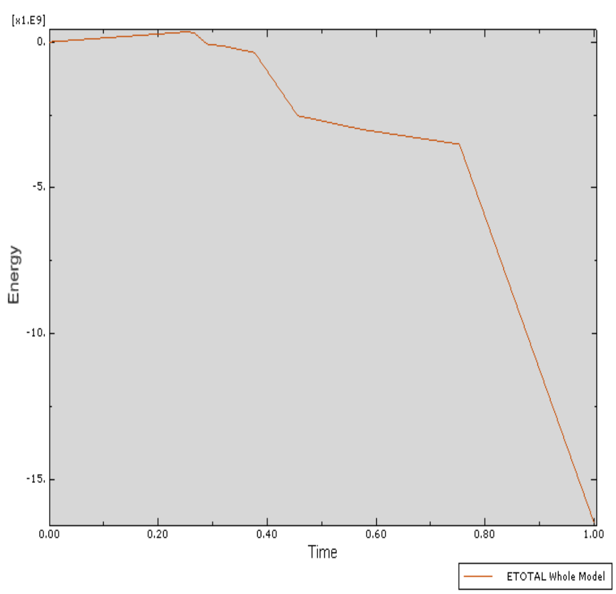


Graph obtained

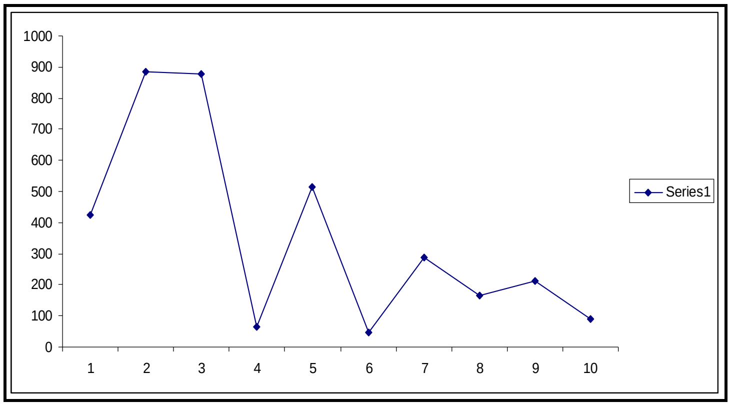
Strain vs Time



Total Energy vs Time



Contact pressure distribution along the elements



IV. CONCLUSIONS

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.

A. From static analysis it can be seen that the contact area depends only on the normal load.
B. 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.
C. The speed of the train largely depends on the rail and wheel interaction forces and coefficient of friction in between them.
D. The contact between the rail and wheel has an elliptical contact area, which is confirmed from contours obtained from the simulation.
E. The stresses generated in the contact region are more than the yield limit, the area in the contact region will plastically deform.

V. ACKNOWLEDGEMENT

I would like to thank my mentor Mr. Kuldeep Kumar for introducing me to this design area of numerical computation techniques. Working under their guidance was a privilege and an excellent learning experience that I will cherish for life time.

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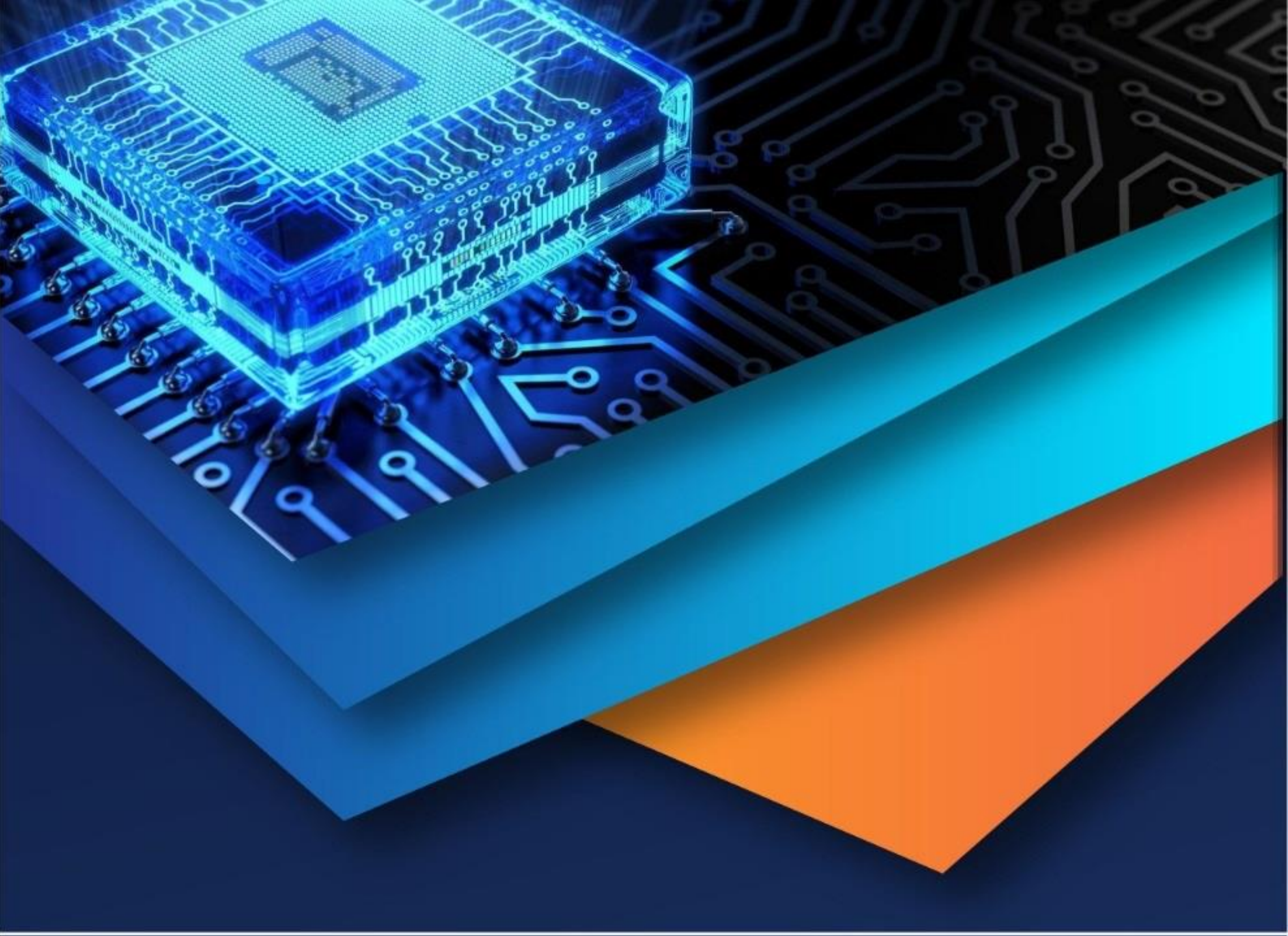

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