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DISPLACEMENT INVESTIGATION OF WHEEL AND RAIL IN TEHRAN SUBWAY SYSTEM

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ABSTRACT

Due to the wear event and its impact on the metro system, studying the factors influencing this phenomenon has been always regarded. Based on this paper, a better maintenance schedule can be resulted in terms of understanding of how deformation and displacement occur and conduct during the fleet operation. Indeed, first by implementing the field measurements, the precision profile evolution of wheel/rail can be discovered with mileage. Secondly, the worst profile evolution behaviours are modelled and the simulations will be approached to find the lateral/vertical displacements and their behaviours by logged distance. Moreover, for better analysis and comparison of the outcomes, contact theories have been also applied. Simulation results indicate that the maximum transverse displacement for rail and wheel occur on the rail's head and on the wheel's web on the S shape profile, respectively. The simulation presents about -16.7% and 7% changings in displacements by the logged distances.

KEYWORDS

Wear, Profile evolution, Displacement, Contact theory, Finite element

1. INTRODUCTION

The major characteristic of the railway transportation is its contact's condition in which the interactions cause sharp stresses. Subsequently, the railway's defects are so expectable, and they can be seen in every parts of the transportation system. Yet, the principal concern is about the defects created in contact patch, resulted from rail/wheel interaction, which force a significant maintenance and repair costs; For instance, crack, break, fatigue, deformation [1].

In addition to burdening the costs, such detects beget the safety issues such as derailment, and also decrease the speed limit, dynamic stability, and comfort coefficient as well which can be affected directly from profile evolution and deformation [2,6]. Shear stresses have been recognized as the principal reasons of defects. Nevertheless, the final causes are the micro displacements of wheel/rail, as results of stresses. The aim of this paper, particularly, focuses on the investigation of displacements and their conduct by logged distance, as the profile evolution occurs. Indeed, if that become possible to forecast the amounts of displacements and also the critical areas in which the displacements are considerable, it can be worthwhile for a better maintenance managing come from being more informed. All gathered info and investigations are based on the track 2 of Tehran metro system.

As the first step, a precious field measurement has been done in Tehran metro transportation system by which profile evolutions of a large collection of experimented wheels/rails are measured; this procedure lasts more than three years. Among the all eroded wheels/rails the critical deformation behaviour is picked up in order to be modelled in the software. To simulate the rolling contact phenomenon, the wheels and rails are meshed first, and then imported to the FEM software. Subsequently, the amounts of lateral and vertical displacements are obtained. Last, as a theoretical investigation the amounts of vertical displacement are determined by applying the Hertzian contact theory [7].

2. PROFILE EVALUATION MEASUREMENT

In order to obtain the profile shapes of wheels and rails by logged distance, there are two basic ways: laboratory and field studies. Laboratory studies are so easier and faster, whilst not so precision. On the other hand, field measurements need a long time to be approached and also they are expensive, yet the results are appreciably precise.

Whereas the contact phenomenon is so sensitive to the shapes of wheels/rails, it seems significantly less accurate the results obtained from the laboratory measurements [8, 10]. Therefore in this paper, the field measurements are applied [11]. By selecting an inverse sets of wheels and rails, and by a long experiment era, the principal behaviour of the profile evolution can be attained. The device implemented in this project is Miniprof by which it can be feasible to discover the wheel/rail's profiles exactly. This device is an electro mechanic scanner can move across the wheel and rail's contact surface by a small magnetic wheel rolling on the surface. Device's sensors are sensitive to the displacements and therefore while the magnetic wheel rolls on the surface the coordinates are transferred to the computer through a coaxial cable. A general view of the Miniprof implemented in the present paper is displayed in Fig 1.

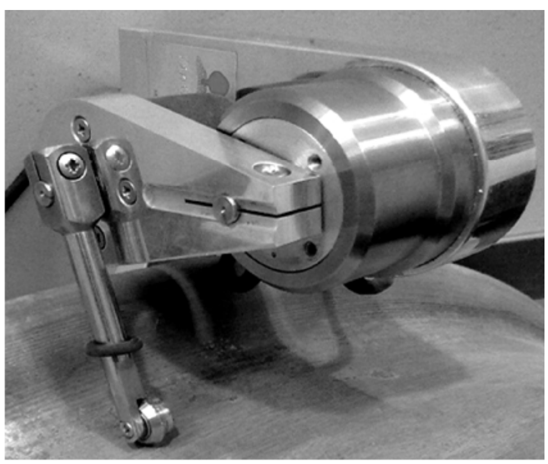

Figure 1.An image of miniprof device applied on the wheel's contact surface

The device should be implemented carefully and far from magnetic and electric fields which exist in every subway system. Moreover, the coaxial cable should be without tie and be straight as much as possible to noises be prevented.

During the field measurement procedure the trains work as usual and the measurements occurs in the periodic schedules. By measuring the marked wheels/rails in three years, consequently the outcomes are categorized in the Miniprof's software. As a result, we will be able to determine the

worst wear behaviour having been led to a critical profile evolution. The elected wheel and rail, with the worst wear behaviour, are the bases of the simulation approached in the FEM software.
To compute the amounts of displacements for the worn rails/wheels, four couples of them obtained from the critical wear behaviour are prepared to be applied in the software. The numbers 1 to 4 for the worn couples are named and introduced in Table 1 in which the amounts of wheel's flange thicknesses have been added as well regarding the worn wheel/rail.

Table 1.Heading and text fonts.

The worn wheels/rails couples IDs	**Logged distance of the worn rail (Km)**	**Logged distance of the worn wheel (Km)**	**Wheel flange thickness (mm)**
#1	122000	81430	31
#2	227000	152320	29
#3	262000	175000	27
#4	298000	197500	24

3. ROLLING CONTACT SIMULATION

Modelling the worn wheels/rails, first, the measured profiles are imported to the software. After importing the coordinates to SolidWorks [12], they become standard based on the real dimensions of the wheel/rail by which the points far from contact surface might be modified. Moreover, the new wheel and rail profiles are modelled as the 2D sketches by considering their factory's map. Such new wheel/rail's sketches are to be applied as the bases for the worn wheels and rails. Indeed, the worn curves resulted from field measurements are to be fixed on the new wheels/rails contact surfaces, instead of their new contact curves. By applying the worn curves on the top of the wheels/rails' profiles, the worn shapes are approached in forms of some shells. Fig. 2 is an image of the final worn shells for a couple of wheel and rail.

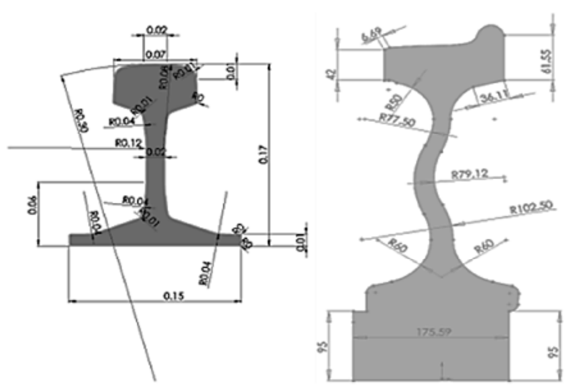

Figure 2. Worn wheel/rail's shell model resulted from SolidWorks software

Since the main analysis is based on the Finite Element Method, the models are required to be meshed. There are, generally, two ways for applying the meshes on the models. First, meshing the 3D wheels/rails, and second meshing their 2D shells. The first way can be so elaborated and therefore the quality of the elements will not be very satisfactory. Yet, the second way provides the tidier meshes in 2D shells led to the meshed solid models by extruding and revolving them along the normal vector and about the symmetry axes of the rail and wheel, respectively. This method can be only applied when the shapes possess a symmetric geometry like wheel and rail. Since the contact phenomenon is so sensitive, therefore the elements should be fine enough. In present article the amounts of meshes are considered about 0.25 millimetres. Nevertheless, it is wholly logical to design the elements coarser in the areas far from the contact patch to eliminate the extra time efforts.

After preparing the models by fine meshes, they will be applied on the FEM solver software in which some applications are done to simulate the real rolling contact phenomenon, the FEM software implemented in this paper is LS-DYNA EXPLICIT [13]. Since, the principal aim of the present paper is investigating of the amounts of displacements by logged distance in the worst application of the system; therefore the simulation is approached on the critical curve of the track which is 190 meters.

The first step in simulation is determining the boundary conditions by which the rail, primarily, spins about its longitudinal axis 1.43 degrees, applied based on the track 2 of Tehran metro system. Then, the rail's foot base becomes fixed by selecting some nodes on that area. By preparing the rail's state, the wheel is located above the rail so that on the curve analysis, the wheel's flange and rail's corner will be touched. In spite of the rail, wheel has freedom to move and is not fixed. This state is indeed a primarily geometry and the final state will be approached when all external forces are applied. Before applying the forces, all mechanical and material characteristics of the wheel and rail should be applied, based on the Tehran subway criteria. Such properties can be mentioned as following examples: elasticity and rigidity modulus, poisson's ratio, friction coefficient, the elastic-plastic diagrams' steeps of material, density, grades of the steels, determining the coefficients regarding the slave and master bodies in the contact phenomenon.

The external forces and accelerations are applied to the wheel in order to simulate the real loading conditions. Such loading cause the final state of the contact between wheel and rail. In the software, the loading is approached by defining the curves by which the equivalent accelerations are applied to the wheel. Such curves should possess a rational rate to create a stable dynamic condition. The accelerations are computed by considering the maximum axle load possible, the worst curve of the track, the maximum speed, and the weight of the wheel. The maximum axle load by considering the net weight of the train, the quantity of the axles, and the nominal and maximum passengers result in 13.7 tons per axle. The train speed is the other dynamic factor which should be applied in two options: 40 Km/hr and 80 Km/hr for straight and curved lines respectively.

Finally, the simulated elements should be saved in a text file. The exported file can be opened by notepad software where every numeral values and other settings can be checked and controlled. In addition to controlling the numbers, there are some dominant features which need the attention. In this paper, three kinds of software are implemented for modelling, meshing, and simulation and solving. When the several types of software are used, the primary attention in spite of considering their compatibility is related to controlling the dimensions and defining the proper units by which the model can be in proper dimension in the final software. The other point is checking the notepad to be free of very small or big numbers, since such numbers cause errors in running process. To solve this problem the scale factors should be defined in the notepad file. Moreover, based on the importance of the simulation it is possible to increase or decrease the calculation time in Control Termination unit, and also it is possible to control the rate of D3plots, which are the computational results in specific running times. At the end, based on the needed outcomes the types of calculation results can be added or modified to the notepad.

Subsequently, after modifying and controlling the notepad file, the simulated file is imported to the solver unit of the software. The running time is completely affected by the numbers of meshes and the running and D3plot timing. Each model in the present paper possesses approximately 350,000 to 400,000 elements

4. THEORETICAL FORMULATION OF DISPLACEMENT

To investigate the contact phenomenon there are several formulations which definitely all of them have some limitations resulted from their specific consumptions. Therefore, it is wholly feasible to have error in theoretical calculations. The FEM method as a new way of calculation made an evolution in engineering and can conclude much more precision results. This method applied in the last section.

Among the all theories regarding the contact phenomenon, Hertzian formulation [7] has been very renowned. Nevertheless, recently, with the advent of the non-linear theories there are some non-Hertzian formulation led to a more precision outcomes, the Hertz's theory is still satisfactory [14, 15]

Subsequently, the Hertzian formulations are applied here to calculate the contact patch square and the cultivated pressure and displacement. Some of the major assumption of the Hertz's theory in the contact event can be mentioned as follows: continuous contact patch, no friction, pour rolling contact without any slipping, no plastic deformation, minute amounts of strains, and a very small contact area.

By considering the contact patch as a circle and a vertical loading, the amounts of the displacements in vertical direction can be calculated by Equation (1),

$$\bar{w}_z = \frac{1-\vartheta^2}{E_{comp}}\left(\frac{\pi P_{max}}{4a}\right)(2a^2 - r^2), \quad r \leq a \qquad (1)$$

where $\bar{\boldsymbol{w}}_{\boldsymbol{z}}$, $\boldsymbol{\vartheta}$, $\boldsymbol{r}$are the amounts of vertical displacement, poisson's ratio, and the radius distance from the center of the the contact circle, respectively. E_{comp}, additionally, is the compound modulus of elasticity regarding the wheel and rail both which can be calculated by Equation (2),

$$\frac{1}{E_{comp}} = \frac{1-{\vartheta_1}^2}{E_1} + \frac{1-{\vartheta_2}^2}{E_2} \qquad (2)$$

where$\boldsymbol{E_1}$, $\boldsymbol{\vartheta_1}$ and $\boldsymbol{E_2}$, $\boldsymbol{\vartheta_2}$ are modulus of elasticity and poisson's ratio of rail and wheel respectively. Moreover, $\boldsymbol{a}$ and $\boldsymbol{P_{max}}$ mention to the circle's radius and maximum pressure in contact patch respectively calculated by Equations (3) and (4),

$$a = \frac{\pi P_{max} R}{2E_{comp}} = \sqrt[3]{\frac{3FR}{4E_{comp}}} \qquad (3)$$

$$P_{max} = \frac{3F}{2\pi a^2} \qquad (4)$$

where$\boldsymbol{R}$ is wheel's radius and $\boldsymbol{F}$ is the force applied to one wheel.

Subsequently to approach the maximum vertical displacement, the value of $\boldsymbol{r}$ should be equal to zero, which leads to maximum vertical deformation occurring in the center of the contact circle. Such calculations are only related to one-point contact state which happens in the straight lines.

5. CONCLUSIONS

By applying the FEM simulation the lateral and vertical displacements are concluded. Since the principal aim of the present paper is to discover the amounts of displacements in the worst condition by logged distance, consequently the simulation's results focus on the rolling contact of wheel/rail on the worst curve. A general view of the simulation is shown in Fig. 3 in which the meshes can be seen obviously too. This Figure displays a final running result regarding the rolling phenomenon on the curved line.

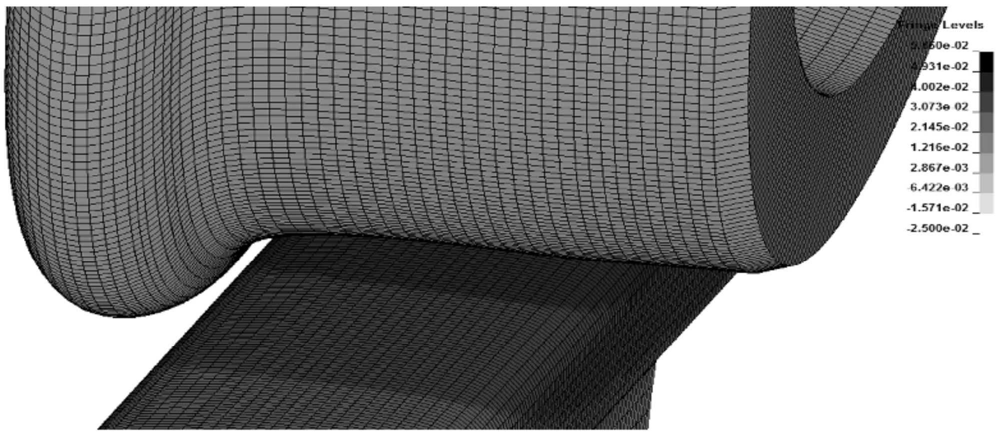

Figure 3. A general view of the simulated rolling contact on the worst curve

In Fig.4(a) it can be revealed which areas possess more lateral displacements. Indeed, the simulation determined that maximum lateral displacements in rail occur on the rail's head which decreases by getting far from the top of the rail's head; and for wheel, the maximum lateral displacement occurs on the middle area of the S-shape profile of the wheel. The lateral displacements for the wheel are almost 0.2 times smaller than the maximum of which on the rail's head; also the direction of the displacements are reverse.

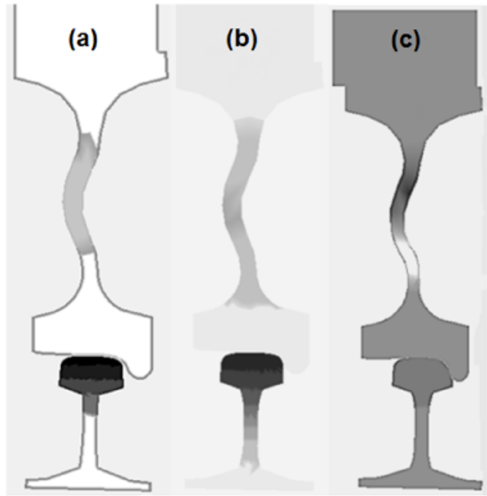

Figure 4. Lateral displacements contours for: (a) new wheel/rail contact on the straight line, (b) new wheel/rail contact on the curved line. (c) worn wheel/rail contact #4 on the curved line

Fig. 4(b) and 4(c) also present the lateral displacement contours for new and worn wheel/rail contact on the curve 190 meters. In the new wheel/rail contact, for the curved line, the maximum lateral displacements on the rail's head are severer and also more broaden than the worn ones. Furthermore, by increasing the erosions in the wheel/rail, the amounts of lateral displacements decrease slightly in the rail's head, nevertheless, the area of maximum lateral displacement is significantly decreased and the most of the rail's profile possess low lateral displacements.

Regarding the vertical displacement, the Fig. 5 displays the maximum changings possible in the new wheel/rail contact plotted by recognizing the worst points on the rail's surface. The amounts of vertical displacements on the curved line decrease considerably, yet such displacements for the worn wheel/rail contacts do not alter significantly.

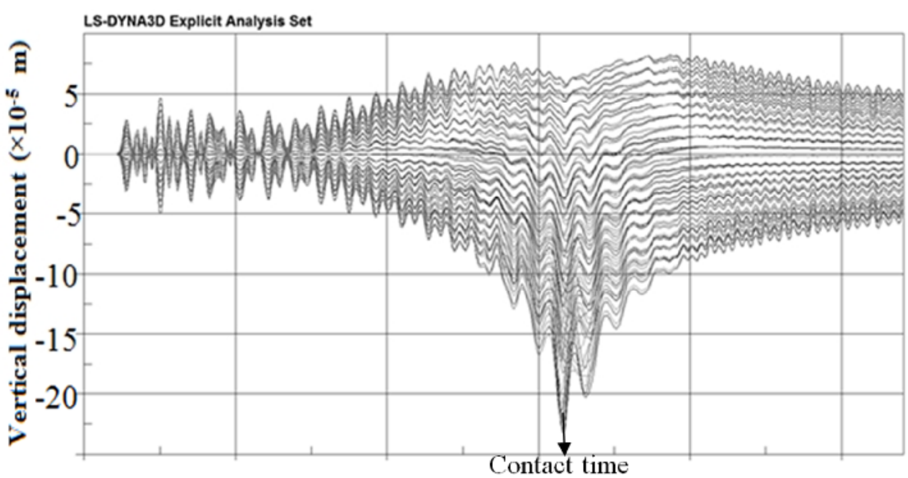

Figure 5.The maximum vertical displacements for the critical points on the rail's surface. The diagram is plotted before, during, and after the contact of the several critical points on the rail's surface possessing the high magnitudes of the vertical displacements.

The horizontal axis in figure 4 is not important; the main aim of this diagram is the amounts of maximum vertical displacements occurring in contact time for some elected points which possess the highest amounts of the displacements. The complete results of the simulations are presented in Table 2 in comparison to theoretical outcomes.

Generally, the lateral displacements in the curved lines are 44% more than which in the straight lines. The amounts of the lateral displacements in curved line by logged distance, as the erosion occurs, change approximately -16.7% in proportional to the new wheel/rail contact in the curve. Such changings in the lateral displacements are not entirely linear, since the real worn profiles measured by field measurements do not possess the uniform behavior.

Indeed, during the application of the subway, rails/wheels are exchanged or re-shaped led to a non-unique deformation behavior. If the laboratory measurements were applied, instead of field measurements, the results became completely decreasing and significantly linear which definitely do not happen in the reality.

Table 2.The final outcomes, simulation and theoretical calculations.

<table>
<tr><td colspan="5">(a) Theoretical Outcomes</td></tr>
<tr><td>Circle Radius (m)</td><td>Compound Elasticity modulus (Gpa)</td><td colspan="2">Maximum Pressure (Mpa)</td><td>Vertical Displacement (m)</td></tr>
<tr><td>0.0058</td><td>113.048</td><td colspan="2">965.05</td><td>0.00007</td></tr>
<tr><td colspan="5">(b) Simulation Outcomes</td></tr>
<tr><td colspan="2"></td><td>Maximum Pressure (Mpa)</td><td>Lateral Displacement (m)</td><td>Vertical Displacement (m)</td></tr>
<tr><td colspan="2">New wheel/rail, straight line</td><td>988</td><td>0.000070</td><td>0.00016</td></tr>
<tr><td colspan="2">New wheel/rail, curved line</td><td>1302</td><td>0.000150</td><td>0.00012</td></tr>
<tr><td colspan="2">Worn wheel/rail #1, curved line</td><td>1566</td><td>0.000145</td><td>0.00012</td></tr>
<tr><td colspan="2">Worn wheel/rail #2, curved line</td><td>1575</td><td>0.000155</td><td>0.00011</td></tr>
<tr><td colspan="2">Worn wheel/rail #3, curved line</td><td>1210</td><td>0.000125</td><td>0.00013</td></tr>
<tr><td colspan="2">Worn wheel/rail #4, curved line</td><td>990</td><td>0.000125</td><td>0.00013</td></tr>
</table>

The vertical displacements, on the other hand, are bigger in the straight lines in proportional to the curved lines, which is because of the distribution of the forces on the rail. In the worn wheel/rail interactions on the worst curve, the amounts of the vertical displacements can be varied not considerably by logged distance, about 7%. Yet, they do not have a linear behavior either, and they might even increase as the profile evolution develops. The major reason for having the similar amounts of vertical displacements by logged distance is: The displacements are generally affected from the applied forces and the rigidity coefficient of the material. Therefore, since the amounts of the loaded forces remain similar in every new and worn rail, and since the rail's material will be the same, the amounts of displacement cannot be different significantly. Indeed, changing the profiles' geometries will not affect the displacements significantly. The small difference caused by logged distance is because of the distribution of the forces on the rail's corner or on the rail's head created by the erosion.

The pressures and the stresses, however, will be affected obviously by logged distance and they are a function of the profiles' geometry directly, which can be changed about 32%. Such amounts pressure changings can be much more visible if the laboratory measurements were applied. Yet, in the reality, because of maintenance actions let to replacing and re-arranging the rails and wheels the profile evolution will not be uniform. Subsequently, the declension of stresses will not be significant even the erosion develops highly and therefore the defects remain considerable in worn rails, unlikely the vertical displacements which alter insignificantly.

The theoretical and simulation outcomes have a good agreement on the calculation of the maximum pressure on the straight line resulted in 2.4% error. On the other hand, regarding the vertical displacements, theoretical and FEM calculations do not meet the same outcomes. The main reason for this difference is that the rail's vertical displacement, in the reality, is affected by whole dimension of the rail and its specific shape, however in Hertz's theory the amounts of displacements are related to an enormous and solid body that its contact patch is wholly minuscule let to a smaller displacement.

It should be considered that the whole outcomes come from simulation never consider the effects of possible defects might be occur by logged distance in the worn rails and wheels which can differ the general behaviours of the displacements.

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