Dynamic Behavior of Worn Wheels in a Track Containing Several Sharp Curves Based on Field Data Measurements and Simulation

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Abstract

Study of the wheel and rail wear phenomenon can provide the optimal use of wheel profile which results cost efficiency, dynamic stability, travel comfort, and safety to prevent the derailment especially in curves. In this paper, the experimental data is recorded from the field measurements worn wheels of a passenger wagon in the “Southern line” of Iran’s railway system and is combined with the dynamic simulations to study the effects of severe wheel flange wear on the dynamics of wagon. The results show that the amount of wheel wear (especially the wheel flange) directly impacts the dynamic behavior of the wagon in curves. In addition, based on the history of wear index and the peak derailment ratio, the appropriate range of the wheel flange thickness in order to repair or replace the worn wheels is suggested in the range of 25 to 27 mm.

Keywords: dynamic simulation, sharp curves, field data measurement, wear index, derailment ratio, wheel flange wear

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Introduction

Study on the wheel and rail wear phenomenon can provide the optimal use of wheel profile which results cost efficiency, dynamic stability, travel comfort, and safety to prevent the derailment specially in curves as well as determining more efficient maintenance schedules for track and wagons [1]. Furthermore, wheels’ flange wear is a major factor in the maintenance costs in railways consisting several sharp curves and would be minimized in order to enhance the competitiveness in the business of the transportation [2]. Although rail/wheel lubrication considerably decreases the wear rate of both elements, still wear is considerably great especially under heavy haul railroad conditions, in steep curves and when the lubricant film is worn out [3, 4].

In the contact of wheel/rail, particularly in sharp curves, the wheels flange are the subject of intensive wear [5]. The risk of severe or catastrophic wear resulting from increased train speeds and axle loads increases due to the form change of the wheel–rail contacts in sharp curves causing wagon lifting and hence its derailment [6]. The derailment processes discussed here occur only due to the loss of lateral constraints of the interactions of the rail and wheels. Therefore, any state decreasing the lateral guidance of rail could increase the derailment probability.

Historically, the subject of rail /wheel interaction attracted attention from 1847, when a British commission was appointed to study dynamic effects in railway bridges [7]. The wheel/rail wear has also been extensively studied previously and the important experimental researches have been reviewed by Lewis and Olofsson [8], Sheinman [9], [10], etc. Some of the experimental studies have been conducted through laboratory tests [11-15], some of them through simulations [16-20], and others based on experience and measurements on in-service vehicles on tracks (field experiments) [13,
The difficulties and expenses involved in the field experiments force researchers to use, whenever possible, the laboratory tests and simulations [6]. Since 1970s, the numerical simulations of the dynamic behavior of rail vehicles and the interaction between wheels and rail have been performed. They have often associated with the modelling of the dynamics of trailer car with nonlinear component behavior [23]. Some of the earlier software developed were Vampire developed by British Rail, Medyna by Deutsche Luft und Raumfahrt, and Nucars in the USA [24]. Recently, general purpose software programs for dynamic simulations of multi-body systems such as Gensys, Adams/Rail, Simpack, Dads, Nucars and Vampire have included features that enable efficient dynamic simulations of wagons and wheel/rail interaction [25] and some of them have been benchmarked [26]. Using the engineering software programs, the study of the dynamics of the rail and the wagon are often classified as two relatively independent problems by assuming the rail as 1) a rigid support or as 2) an elastic foundation [27]. By each assumption in the software, the longitudinal, lateral and vertical dynamics of a moving train or wagon are examined. However, the rail dynamics, is also investigated by two assumptions: 1) the simplified beams on elastic foundation approach [28] or 2) the finite element model (FEM) of railway system [16, 29]. Furthermore, these rail models have almost been assumed to be excited by 1) a single wheel or by 2) a single bogie with two wheelsets rolling on the rail [27]. Software for vehicle motion simulations, are normally concerned with the orientation of each wheel relative to the rail, the contact point between wheel tread and rail head, and the contact forces that are caused by the dynamic interaction [30].

In this paper, the experimental data is recorded from the field measurements of the worn wheel of a passenger wagon in the “Southern line” of Iran’s railway system and is combined with the dynamic simulations to study the wear index and derailment ratio.
parameters based on standards UIC510-2, UIC60, and UIC703. In addition, two parameters called wear index and derailment ratio are studied based on the mutual forces of wheel and rail originating from the wagon loads (loads on the wheels from the wagon) and the curves (change of the wheels’ loads due to the track’s shape). Here a model of wagon with 66 degrees of freedom is considered using the standard UIC518. The dynamic simulation software is utilized to modeling the wagon and track and analyzing the effects of wheels’ wear on the dynamic behavior of the wagon. This study can help to consider and suggest strategies which can be used in order to examine the safety of the wheel based on the derailment forces and also employing efficient maintenance strategies.

**Materials and Methods**

In this work the dynamic simulations are accomplished on the wheel and rail models that have the same specifications, materials and mechanical properties according to UIC standards (Table 1). The wear in wheel profiles has been measured experimentally on the rail wheel of the train traveling between two points where the path contains several sharp curves. The numerical simulations to calculate the dynamic parameters of the train have achieved by considering the experimentally obtained data of the worn profiles of wheels which highly increases the accuracy of the analysis that may not be achieved simply by a complete simulation process. However, the limitations of empirical work, practical problems, and high costs prevent a complete experimental study. The dynamic analysis is also carried out by appropriate software to such studies. The measurements are recorded on the standard wheel and rail profiles through the travel time.
3.1. Experimental data

The field data measurement path

Experimental data from the field measurements of the wheel profiles are obtained through the traveling of the train in one of the five major rail lines of Iran i.e. the “Southern line”[31]. This line is very important for passenger and freight transportation from Iran’s capital, Tehran, to the three important international transit ports of Iran, i.e. "Imam Khomeini", "Mahshahr", "Khorramshahr". This line is widely known for the number of sharp curves leading to high rate of wheel flange wear.

Measuring instrument and recording procedure

The laser view instrument is used for an automatic measurement of the wheel profiles’ parameters. The measurements are performed in real scale and in service conditions [31]. The geometry of the worn wheel profiles as well as the wheel parameters such as flange thickness (Sd) are recorded through each round trip of the train until the flange thickness of the worn wheel reaches the limit of 22 mm (the limit in the standard UIC-510-2 [32] for the detachment of the wagon). The least squares method is utilized as a curve fitting tool in MATLAB software to express the mathematical relationship between the flange thickness (Sd) parameter and the travelled distance of the wagon.

3.2. Modelling and simulation procedure

The characteristics of the test path

The dynamic simulations are carried out by the dynamic simulation software. Part of the path with the highest report of wheel/rail wear which includes a curve with the shortest radius of curvature within the path is chosen for the purpose of the simulation. The test path is created considering the conditions defined in the standards UIC518 and UIC703
from the real parameters of the Southern line of Iran’s railway system with the length of 600 m. According to the standards, the path contains following parts:

- A straight line with the length of 100 m.
- An interface curve with the length of 50 m.
- A curve with the radius of 220 m and the length of 300 m at an angle of 75°.
- An interface curve with the length of 50 m.
- A straight line with the length of 100 m.

Fig. 1 shows the definition of the test traveling path, the travelled distance and speed of the model wagon in time, and the schematics of wagon’s movement and different views of the test path.

**Wagon modelling**

A wagon from the “Plur e Sabz” type train with bogies of type MD523 is modelled for the simulation with the same characteristics of the wagon for which the experimental data are recorded. The wagon consists of many components which their geometry and mechanical properties should be modelled for wagon–rail dynamic simulations. The wagon components can be subdivided to: 1) body components which hold the wagon mass (weight) including the wagon’s body, wheelsets and bogie frames, and 2) suspension components which are various physical springs and dampers including traction rods, antiroll bars, bump stops, linkages, trailing arms, etc.[33]. The model of detailed components of wagon is created in the simulation software. This model consists of sixty-six degrees of freedom. The mass of all of components are concentrated at the center of gravity of the wagon and their flexibilities are ignored. The friction model used for the dynamic friction force between the various surfaces such as pads (between body and bolster) is defined by a friction coefficient as a function of speed and compressing force of the contact surfaces.
A bogie is a set of mechanical parts which includes wheels and axles, suspension system, brake system and bogie frame. Each wagon consists of two bogies, so each bogie bears half the weight of the wagon. Obviously, when a wagon brakes or when it creates acceleration or deceleration or when the wagon is in ascending or descending slopes, or in curves, the load that each bogie bears is more than its static loadings at rest.

In general, several factors including the speed of the train and the radius of the curve are considered for the design of the bogie. The general geometrical parameters of the bogie and wagon are listed in Table 2 and Table 3. The modelled bogie and wagon are shown in Fig. 2.

**Wheelsets and rail**

The wheelsets consist mainly from the axles and wheels. Each axle connects rigidly two wheels. The wheel on each axle has a conical shape and not a cylindrical shape. The flange of a wheel is its inner edge which is designed to prevent the derailment of the wagon during the travel and specially in curves. The wheel geometry is such that if any small lateral movements occur when the wagon travelling through the rail path, a centripetal force toward the center of the wheel and axle returns the wagon to its initial position and compensating the produced temporal deviation. Moreover, implementing the restoring force can increase the radial consistency of the wheelset in curves. This consistency, can increase the rolling and decrease the sliding of the wheels and hence can decrease the wheel and rail wear phenomenon producing from the sliding of the wheels on the rail. Either in the straight line or curved tracks, the gap between the wheel and rail cannot exceed a specific limit (Fig. 3b). This specific limit, constraints the lateral movements. To model the wheel/rail contact in the software, given wheel and rail profiles must be defined. UIC60 profiles is applied for rails and for wheels, new and worn profiles from standard S1002 are used. Fig. 3 represents the characteristics of the
wheelset and rail in contact, the wheel and rail profiles and their definition in the software, and the naming convention of the wheels and bogies.

**The wheel/rail contact problem**

Since the 1960s much research has attended to the fundamental issues of wheel/rail contact problem. The reason that the wheel/rail contact characteristics mainly affect the dynamic interaction between vehicle and track. Hertz [34, 35] used the early elastic theories which were related particularly with the calculation of surface tractions of wheel/rail in a track. He presented the contact surface formula as a prediction of an ellipse by calculation of its small and large diameters. Finally, Hertz analytically obtained the calculations of force and stress in the contact point or contact surface with simplifying assumptions such as: 1) The contact surfaces are flat. 2) The contact between the bodies is frictionless. 3) The contact is assumed to be elliptic. 4) The important dimensions of the contact region are very small with respect to the dimensions of the bodies which are in contact.

In general, the creep forces are created between the contact of two rolling bodies where are not considered in Hertz Contact Theory. When two bodies are compressed on each other and are then rolled by a given force, due to the difference between their strain rates in the contact point, some forces are created in the contact region. These forces are non-conservative and may cause the instability of the dynamic system. In addition, these forces depend on the factors such as environmental conditions, surface smoothness, and the material and geometry of the wheel and rail. The creep motions mainly occur in the curved paths to define a deviation from the pure rolling motion. In particular, the difference between the lengths of inner and outer rails in a curve makes the two wheels in an axle of the wagon to produce the creep motions and hence causes the wear of the wheels.
In 1967, Kalker [36] suggested a linear theory describing that for the creep, the slip zone is so small to have a considerable effect and hence its effect might be negligible. Therefore it could be assumed that the sticky area represents the contact region. In this theory, a constant normal load could be assumed in the straight line, but the influence of the contact of the wheel flange on the distribution of normal load in curves should be included. The linear theory of Kalker for the rail vehicles is widely used in the calculations of lateral stability and steady state forces in curves. Kalker formulated two nonlinear creep laws which include the effects of rotational creep in his later works [37]. These two laws are added to the simple theory of rolling contact. The difference between the solutions originates from the two simplifying assumptions which are related to the tangential stress of the displacement and the distribution of the normal stress in the contact region. Kalker’s theory [37] offers the best results for the deviation from the pure rolling motion.

The linear theory of Kalker is used for the wheel/rail contact in our simulations. In this analysis, a multi-point connection model is used for the contact of the wheel and rail. In a multipoint contact, the analyzing procedure is based on the non-linear polynomial equations due to the full consideration of the profile of the wheel/rail contact. The contact type is modelled based on the existing standards which have been prescribed as geometrical tables.

**The forces acting on the wheel and rail in contact**

Specifying the forces between wheel and rail in their contact region is one of the most important issues related to the dynamics of train motion. The mutual effects between track and vehicle is determined from these forces. The forces acting on the wheel and rail are divided into tangential and normal forces acting on the contact surface. In the contact forces between the wheel and rail, the created stresses may cause the transition
of the material behavior from the elastic mode to the plastic mode. Tangential forces are the main reasons of the wheel damages such as wear and fatigue. Fig. 4 shows the schematics of the longitudinal and translational radii of curvature and the distribution of the longitudinal and translational forces.

The design of the railroad tracks and wheels is based on the specifications and orientation of the vehicle in the track line (Rail). In a curve, the radius of the curvature in one hand and the effect of centripetal forces on the other hand specify the characteristics of the contact between the wheel and rail. Centripetal forces cause the lateral compressing forces of the wheel on the outer rail. The centripetal forces depend on the vehicle mass, the radius of curvature and especially the vehicle speed. The wear in the rail/wheel contact depends on the friction force the in contact region. The friction force depends also on the coefficient of friction and the compressive normal forces. To reduce the friction, it is beneficial to reduce the impact of each of these two factors. Lubrication is used to reduce the friction coefficient and the transverse slope is used to reduce the normal compressive forces. The forces are shown in Fig. 5.

The centripetal force and weight of the wagon along the line $AA'$ acting on the wheel yield:

$$\sum f_{AA'} = m \frac{V^2}{R} \cos(\alpha) - mg \sin(\alpha) \quad (1)$$

where $\alpha$ is the transverse slope of the track line, $g$ is the acceleration of gravity, $U$ is the vehicle speed, $R$ is the radius of curvature and $m$ is the mass of the vehicle. If there is no transvers slope of the track line, $\sum f_{AA'}$ (the resultant of the forces acting on the wheels) is presented by:
\[ \sum f_{AA'} = m \times \frac{V^2}{R} \]  

If the transverse slope is increased, the resultant force acting on the wheel is reduced. The ideal condition is when the magnitude of this force is zero. This occurs when the angle takes the following value:

\[ \alpha = \tan^{-1}\left(\frac{V^2}{Rg}\right) \]  

The wear index

As presented in previous sections, creep forces (deviation from the pure rolling motion) are produced in the contact region between two rotating objects (wheel and rail). These non-conservative forces depend on the following factors: 1) Environmental conditions, 2) Surface smoothness of rotating objects, 3) Materials of rotating objects, 4) Geometry of rotating objects. The dimensionless parameter, creep, is defined along the longitudinal and transverse directions with \( \zeta_x \) and \( \zeta_y \), respectively. The equation of the wear index is then defined as following:

\[ \text{Wear Index} = F_x \times \zeta_y + F_y \times \zeta_x \]  

where, from this definition, the dimension of the wear index is same as the dimension of force.

Effect of wear on derailment
The flange on the inner edge of the wheel prevents the derailment of the wagon during the motions and its deviations producing from the lateral forces in curves, etc. Error! Reference source not found. shows a diagram of forces acting on the wheel flange when passing through a curve. In this condition, the wheel is deviated from its balanced position and its flange creates one or more contact points with the rail edge. According to Error! Reference source not found., the lateral force \( Y \) and the vertical force \( Q \) are the components of the resultant force \( R \) acting on the wheel flange. The parameter which cause the derailment is force \( Q \) producing the lifting force for the wheel. Therefore, in wheel and its flange design, attempts are focused to reduce the ratio \( (Y \text{ (Lateral Force)}) / (Q \text{ (Normal Force)}) \) which is called “derailment ratio” [38]. In the worst cases, the components of the exerting forces are large enough to overcome the friction force and the wheel moves relative to the rail. \( (Y/Q)_{\text{limit}} \) is the maximum allowable amount of \( Y/Q \) in the safe mode. Practically, the limit of \( Y/Q \) is achieved experimentally. By approaching the amount of \( Y/Q \) to \( (Y/Q)_{\text{limit}} \), the probability of the derailment is increased.

**Transient solution**

In order to wear analysis and software simulation, the time step size or number of the time divisions should be chosen small enough to obtain the desirable results when applying different profiles. If time is greater than the required amount, the obtained answers will not be sufficiently accurate. The time step size must also be chosen in such a way that the results would be converged. For this purpose, the time step is set by using a trial and error procedure.

**Results and discussions**
The model of wagon and the path is created and the simulations are conducted on the wheel profiles measured from the field data of the real wagon in service traveling through the Southern line of Iran. In the next step, the dynamic analyzing of the curved path based on the given speed same as data measurement conditions, and the effects of wear of wheel profiles on the dynamic behavior of the wagon are considered with respect to the derailment ratio and wear index.

4.1. Experimental data

The wagon sweeps along the Southern line of Iran’s railway system with a length of 905 km. The wheels’ profile data are recorded for each round trip (1810 km). Seven worn profiles are selected for the simulations. These wheel profiles are history of the severe flange wear from 32 mm for a new profile to the minimum of 22 mm for the last worn profile. This limit has been defined in the standard UIC510-2 for the detachment of the wheel in order to repairing or replacement with a new wheel. After traveling about 40,000 km, the wheel number 1L (called “critical wheel”) is detached due to severe wear in the flange thickness (Sd) of 22.9 mm. Fig. 6 represents the new and worn profiles which are defined into the software for the simulations [3]. The technique of curve fitting is used to model the wear procedure of the critical wheel. Fig. 7 represents the 1st to 5th order polynomial functions fitted to the experimental data of flange thickness which the evaluation parameters show that among them, the 5th order polynomial function is the best fitted model [3].

4.2. The derailment analysis

One of the outputs of the dynamic analysis in the dynamic simulation software is the safety analysis related to the derailment by the lateral and vertical forces between wheel
and rail. The derailment ratio must be smaller than 0.8 according to the standard UIC-518. Here, the derailment ration critical is only considered for the critical wheel.

**Derailment ratio**

According to Fig. 1b, by a steeply variation of the wagon speed in time $t = 6$ s (which is the state in which the wagon passes the first transient distance (interface curve) from the length 100 m to the length 150 m), corresponding peaks for the new and selected worn profiles of the critical wheel can be observed (Fig. 8). Fig. 9 shows the history of the peak derailment ratio through the worn profiles of the critical wheel. It is found that by increasing the amount of wear of the profiles, the derailment ratio increases. However, according to UIC518, it must not exceed the limit of 0.8. The results show that the derailment ratio is always less than this limit in our simulations. Hence, the derailment does not occur for the wheels of this wagon through the prescribed path and test conditions.

The plot of the peak derailment ratio (Fig. 9) can be divided into three zones. The first zone ($28 \, mm < Sd < 32 \, mm$) shows that the flange thickness wear is relatively small in this region. In this zone, the lateral forces increase continuously with respect to the vertical forces. The amount of wheel flange thickness ($Sd$) for the detachment of the critical wheel in order to repairing or replacement is not efficient in this zone. However, in the second zone, the range of wheel flange thickness is $25 \, mm < Sd < 28 \, mm$ and the plot shows local minimums with respect to the third zone, in the range of $22 \, mm < Sd < 25 \, mm$. In the third zone, a continuous increasing trend can be observed for the derailment ratio which can greatly increase the risk of derailment.

**Wear index**
Similar to the derailment ratio, according to Fig. 10, the wear index plots show corresponding peaks in their plots for the new and worn profiles of the critical wheel (1L). Fig. 11 shows the variation of the peak wear index of the critical wheel. It is found that by increasing the amount of wear in the profiles, the wear index increases but not for all the range. Fig. 11 represents the zone $25 \text{ mm} < Sd < 27 \text{ mm}$, where the wear index is minimum. This range can be used in maintenance strategies to optimize the maintenance costs and schedules.

Based on the wear index and derailment ratio from the dynamic analysis, an appropriate range of the wheel flange thickness in order to repairing or replacement of the worn wheel can be suggested in the range of $25 \text{ mm} < Sd < 27 \text{ mm}$.

**Conclusion**

In this paper, the experimental data was recorded from the field measurements of the worn wheels of a passenger wagon in the Southern line of Iran’s railway system and was combined with the dynamic simulations in the software to study the dynamic behavior of the wagon by two parameters: wear index and derailment ratio. The wheels’ profile data where recorded for each round trip and seven worn profiles of the critical wheel (the first wheel which is detached due to the wear) were selected for the considerations. The technique of curve fitting was used to model the wear procedure of the critical wheel. The evaluation parameters showed that the 5th order polynomial equation was the best fitted model.

The results of the variations of the derailment ratio and wear index of the wheel profiles with respect to the wheel flange thickness (Sd) during the travelling of the train showed that the amount of wheel wear directly impacts the dynamic behavior of the wagon in curves. In general, increase in the amount of wear in profiles, increased the derailment
ratio. It was found that the derailment ratio for the tests were always below the limit of 0.8 according to standard UIC518 where exceeding the limit increase the hazard of the derailment of the wagon. However, when wheel wear is in a certain range, the wear index and derailment ratio are reduced.

The appropriate range of the wheel flange thickness in order to repairing or replacement of the worn wheel is suggested in the range of 25 to 27 mm.

References

Biography of Authors

Seyyed Miad Salehi received his PhD degree in Mechanical Engineering in 2016 from Sharif University of Technology in Tehran. He has received several national and international awards and prizes in his various activities such as the nation's leading student in 2012 and Kharazmi Young Award in 2007 and 2012.

Gholam Hossein Farrahi received his PhD degree in Mechanical Engineering in 1985 from (ENSAM), Paris, France. He is currently Professor at the School of Mechanical Engineering, Sharif University of Technology. He is also the Head of Materials Life Assessment and Improvement Laboratory. His research interests include structural integrity assessment, failure analysis, fatigue, wear and life improvement methods.

Saeed Sohrabpour received his PhD degree in Mechanical Engineering in 1971 from University of California, Berkeley, USA. His research interests include large deformations, mechanics of metal forming, and optimal design.
**Figure Captions**

Fig. 1 a) The traveling path definition in software, b) The travelled distance and speed of the model wagon in time.

Fig. 2 The models of a) bogie of type MD523 and b) the wagon of type Plure Sabz, in the simulation software.

Fig. 3 a) The characteristics of wheel and rail on rail with a gauge of 1435 mm. b) Wheel and rail profiles with their balance contact position. c) The defined wheel and rail profiles in the software. d) The naming convention of the wheels and bogies.

Fig. 4 The schematics of the longitudinal and translational radii of curvature and the distribution of the longitudinal and translational forces.

Fig. 5 The wheel and axle set in curve with a lateral slope and the acting forces. Derailment occurs due to lateral forces acting on the wagon in curves.

Fig. 6 The resultant of the forces acting on the wheel flange. Q is the vertical force, Y is the lateral force and R is the resultant force.

Fig. 6 The experimental data of the selected worn profiles obtained from field measurements [3].

Fig. 7 a) the fitting evaluation parameters in order to choose the best fitted model, and b) 1st to 5th order polynomial functions fitted to the experimental data of flange thickness versus the travelled distance [3].

Fig. 8 The derailment ratio of the a) new, b) first worn, and c) last worn, profiles of the critical wheel (1L). The derailment ratio shows corresponding peaks for the new and worn profiles.
Fig. 9 The peak derailment ratio variation for the critical wheel (1L). There are three zones with respect to the wheel flange thickness (Sd). The minimum peak derailment ratio occurs in flange thickness range $25 \, \text{mm} < Sd < 28 \, \text{mm}$.

Fig. 10 The wear index for the a) new, b) first worn, and c) last worn, profiles of the critical wheel (1L). The wear index shows corresponding peaks for the new and worn profiles.

Fig. 11 The wear index of the critical wheel (1L) versus wheel flange thickness (Sd). It represents the minimum zone of the wear index for the range of wheel flange thickness $25 \, \text{mm} < Sd < 27 \, \text{mm}$.
Table Captions

Table 1. Wheel steel specifications according to UIC 812-3 [39].

Table 2. General characteristics of Model bogie MD523.

Table 3. The geometry of the passenger wagon model.
### Table 1

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<th>Steel Category</th>
<th>Carbon content (%)</th>
<th>Yield strength (N/mm²)</th>
<th>Tensile strength (N/mm²)</th>
<th>Elongation (%)</th>
<th>Notch impact energy (J)</th>
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<td>R7 T,E</td>
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<td>≥ 520</td>
<td>820</td>
<td>≥ 14</td>
<td>≥ 15</td>
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### Table 2

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<td>Distance between primary Springs</td>
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<tr>
<td>Distance between secondary Springs</td>
<td>2580</td>
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<td>Distance between side bearers</td>
<td>1410</td>
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<tr>
<td>Distance between pivots</td>
<td>19000</td>
</tr>
<tr>
<td>Wheel Base</td>
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<td>Primary Suspension Spring play</td>
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<td>Secondary Suspension Spring play</td>
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### Table 3

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<td>Car Body(SGP)</td>
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<tr>
<td>Distance Between Pivot Center</td>
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<td>Hard Point</td>
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Figure 1

(a) Rail Profile

(b) Travel Velocity and Distance
Figure 2
Figure 3
Figure 4
Figure 5
Figure 6
Figure 7
Figure 8

Fitting on wheel profiles

Traveled Distance (km)

Residual Value

Order of Polynomial
Figure 10
Figure 11

(a) Graph showing force (N) over time (sec) with a peak at around 5 sec and a gradual decline.

(b) Graph showing force (N) over time (sec) with a peak at around 5 sec and a gradual decline.

(c) Graph showing force (N) over time (sec) with a peak at around 5 sec and a gradual decline.
Figure 12