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Research paper

The influence of worn and nominal rail profiles on railway vehicle motion and dynamics

Krzysztof Zboiński[1], Jacek Kukulski[2], Milena Gołofit-Stawińska[3], Piotr Woźnica[4]

Abstract: In this study the rail wear influence on safety of rail vehicle motion is assessed. The profile measurements for the nominal rail head and for the slightly worn and heavily worn rail heads were performed. Then, these measured real values were entered into a computer program designed to study the motion (dynamics) of rail vehicles. The studied model of passenger vehicle moved in simulations along a track consisting of straight and curved sections with transition curve between them at two velocities. After obtaining the simulation results, they were compared for different stages of the rail wear process based on profile measurement results and the suitability of the track was assessed, too. The aim of present paper was to show that extension of the traditional approach with the simulation studies of vehicle motion and dynamics leads to different and much more precise information about the need of the rails replacement. The proposed idea and approach in the current work is original and not used in practice by the track infrastructure operators.

Keywords: rail profile wear, railway vehicle dynamics

[1]Prof., PhD., Eng., Warsaw Universityof Technology, Faculty of Transport, ul. Koszykowa 75, 00-662 Warsaw, Poland, e-mail: krzysztof.zboinski@pw.edu.pl, ORCID: 0000-0002-6967-4244
[2]Prof., PhD., Eng., Warsaw University of Technology, Faculty of Transport, ul. Koszykowa 75, 00-662 Warsaw, Poland, e-mail: jacek.kukulski@pw.edu.pl, ORCID: 0000-0002-9449-7614
[3]PhD., Eng., Warsaw University of Technology, Faculty of Transport, ul. Koszykowa 75, 00-662 Warsaw, Poland, e-mail: milena.stawinska@pw.edu.pl, ORCID: 0000-0003-2166-1724
[4]PhD., Eng., Warsaw University of Technology, Faculty of Transport, ul. Koszykowa 75, 00-662 Warsaw, Poland, e-mail: piotr.woznica@wp.pl, ORCID: 0000-0002-3715-966X

1. Introduction

Technological progress and following with it rapidly increasing need for rail transport usage, has caused the railways to be heavily exploited for years, what leads to various types of faults, breakdowns and wear processes of both the vehicle and the track. So, since railways exist, there is a problem of rail wear (being the topic of this article), which is a result of cumulated, enormous loads exerted on the rails. Any loss in material or unevenness endangers the safe motion of the rail vehicle. Therefore, it is so important to keep the track fully operational, which is closely related to: diagnostics, maintenance, and repair.

It is known that situation is the safest when one deals with rails in their nominal state (completely new rails). Consequently the question appears here. Namely, to what stage of wear and tear one can assume that situation is still safe. And when the red light should be switched on because the wear is so great that the rails need to be replaced while the vehicle motion on them becomes dangerous.

Nowadays the decision about the moment of rails replacement is based on on-site measurements of their wear. When the wear exceeds values adopted by the infrastructure operator as limit then the rails should be replaced. In this paper different approach is proposed and checked for its usability. The aim of present paper is to show that extension of the above described traditional approach with the simulation studies of vehicle motion and dynamics leads to different and much more precise information about the need of the rails displacement. The proposed idea and approach is original and not used in practice by the track infrastructure operators. Therefore it can be treated as novelty and some contribution of the present authors into the track maintenance issues and methodologies.

It should be noted that under sleeper pads (USP), despite their many advantages and contemporary common use, are not included in the analysis of present paper. This is mainly due to fact that the onsite measurements and simulation analysis concern the structure built and modernized in the 1980s. The literature profiled to this issue is cited further on, however.

2. Literature survey

Today, a large number of scientific works dealing with the problem of wheel and rail profiles wear in wheel-rail contact can be noted. It is visible, mainly in the light of development of both high-speed train lines construction [1] and rapid increase of railway freight transport [2, 3].

Generally, the greatest number of works represent the classical approach to the problem of wheel/rail wear. In such works, the problem of rolling contact in the presence of dry friction is treated as a complicated physical phenomenon. The wheel/rail contact has a crucial influence on a vehicle dynamics. The examples of such works let be the works: [4–7]. As the result of work [6], the magnitude of rail profile wear and wheel wear (both left and right wheels) for different track sections can be determined for the operational process. In [7], its authors examined high-frequency train-track interactions and the mechanism of wheel/rail wear that is non-uniform in magnitude around/along the running surface. Suggested remedies to relieve the problems were presented for three types of irregular wheel/rail wear. There were: short-pitch rail corrugation on tangent track, wheel corrugation caused by the breaking, wheel polygonization.

The next group of the works are the works, where the optimization of the wheel/rail profile is performed. As the examples, the works [8–11] can be given. In [9], the minimum value of wear, being the product of the creep forces and creepages is searched. In [10], its authors formulated the objective function making the discrepancy between the target rolling radii difference function and the calculated rolling radii difference function for the design profile. The minimum value of this function was searched, using the so-called MARS method.

In some works, the wear in rail/wear contact is examined only in a curved track. Such typical works can be: [12, 13], and also mentioned earlier work [9]. The work [9] presents an optimum design procedure for asymmetric rail head profile wherein the design profiles of high and low rails are simultaneously determined by minimizing the wear on the curved track. The procedure employs a genetic-algorithm-based method.

Mentioned earlier works analyse the wear in the straight track and curved track. The separate group of work are the works, where the authors examine the wear in turnouts. The examples let be: [14] and [15]. The hollow-worn wheel phenomenon has attracted the attention of the authors of [14]. They also determined the methodology to evaluate the dynamic wheel-rail interactions using SIMPACK software. The vehicle-turnout simulation model integrated the dynamic model of a high-speed vehicle capable of velocity 350 km/h with 1:18.5 turnout model via the wheel-rail contact model. The measured and recorded in files wheel worn profiles were applied in the mentioned model. In work [16], its authors applied the computer aided analysis for wear investigation of railway wheel running surface. They presented the wheel profile wear by Archard wear law, when the computational model of railway vehicle was operating on track with a constant velocity. The rail UIC 60 with the inclination of 1:40 was used. The simulations were also made with SIMPACK software. In [17], the correlation between rail wear rates and operating conditions in a commercial rail track is presented. According to the authors of the mentioned work, Rolling Contact Fatigue (RCF) and artificial abrasion due to rail grinding caused the material removal from the rails. Also, a synergistic effect was found to be significant as inadequate rail grindings caused defects that acted as preferential nucleation sites for fatigue cracks. The corrugation was also accused of rails wear.

The work [18] is an example of the works, where the application of the computational tool to study the influence of worn wheels on railway vehicle dynamics was presented. The mentioned application was VAMPIRE software. The core of the wear prediction was the wear computation, which calculated the worn material amount removed from the wheel surfaces. In the work, the wear function developed by University of Sheffield was used. The comparative analysis of railway vehicle dynamics was performed. The purpose of the analysis was to assess, how the wheels' wear growth influences the vehicle dynamical behaviour.

The valuable work from the point of view of rail track maintenance is the work [19]. The mentioned work treats the application and the performance assessment of a wheel profile measurement system. It presents case studies focusing on this system use and measurement performance. The proposed application relates to the problem how the information from the system can be integrated with physical models to obtain the real wheel behaviour.

The next works worth of mention are the works [20] and [21]. In [20], the performed study investigated the phenomenon of the interaction between wheel and rail wear, including influencing factors such as: friction, adhesion, abrasion, the methods to reduce these phenomena in contact, like lubrication.

The authors of [21] used the meta-models the determine the relation between input parameters (like vehicle speed and axle load) and the amount rail wear. The coefficient of determination was higher than 0.95, so the accuracy of the model proposed by the authors was very high.

In [22], the authors presented the results of the experimental and numerical investigations on regular and worn railway wheel profiles. Experimental tests were made using a full-scale roller rig. It allowed to measure the evolution of wheel profiles, and also contact forces.

The work [23] was aimed to examine the behaviour of metro vehicle model equipped with differentials. The use of mentioned differentials was intended to improve the interaction of the constrained wheelsets with rails in small-radii curves. As the results of the work, the authors concluded that the use of differential in a light train can be limited only to the small curves (not greater than 150 m). In curves of greater radii, the use of differential can, however, increase the wear in wheel/rail contact.

Many topics relevant to the subject of the current article were also recently raised in works like [24–27]. Interesting are also the works, where under sleeper pads (USP) are analysed [28–32], mainly on the resistance analysis of several prototypical under sleeper pads to severe environmental conditions [31].

In the end, it is also worthy of noting the works [33, 34] of two authors of the current work. In [34], they optimized the shape of the railway transition curves, taking into account the minimization of the wear in wheel/rail contact. As the objective functions creepages in wheel/rail contact, as well contact forces were assumed and minimized. The authors found shape of the optimum transition curves, which gave the minimum value of the objective functions.

Analyzing the literature relating to the problem of wear in wheel/rail contact, the authors of the current work may conclude that today are two main reasons, why the wear in wheel/rail contact is investigated. The first reason is the safety of the passenger journey and the impact on the dynamical behavior of the vehicle. The second reason is the fact that wear has a strong connection with the railway track and vehicle maintenance organization. The smaller wear of wheels and rail results in smaller maintenance costs in the life cycle of railway infrastructure and vehicle, too.

3. Experimental research

The experimental tests covered measurements of the rail head profile removed from the track section located in track no. 1 on the section between the Warszawa Zachodnia station and the Warszawa Włochy station. Figures 1 and 2 show the location of the track and the circular arc from which the test samples were cut out.

Fig. 1. View of the track no. 1 and the turnout no. 575

Fig. 2. A view of the track curve located on the track no. 1

The track was built of: 60E1 rails (former UIC60), concrete sleepers PS-83, elastic fastenings SB-3 and thickness of ballast bed up to 35 cm. The track geometry from the turnout no. 575 to the Warszawa Włochy station includes three circular arcs with radii $R_1 = 1100$ m, $R_2 = 800$ m, and $R_3 = 1100$ m and straight sections with a total length of 2 684 m.

Measurements of the profiles of the worn sections of rails were made with the use of the X-Y profiler. This profiler is used to measure the cross-section of rails and various elements of railway turnouts. It can be used to measure both new and in-built turnouts. The instrument can also be used to measure the cross-section of other elements within its measuring range. The measuring system includes: X-Y profiler with the measurement results recording system (Fig. 3) and elements for fixing and setting the profiler in the track.

Fig. 3. X-Y profiler measuring system [Photo. J. Kukulski]

The profiler is based on the track for the time of measurement by means of a retaining beam. After the profilometer is in the correct position, the operator manually guides the measuring tip along the surface to be measured (Fig. 4). Measurement results are stored in the recorder memory as separate ones for each started and finished measurement.

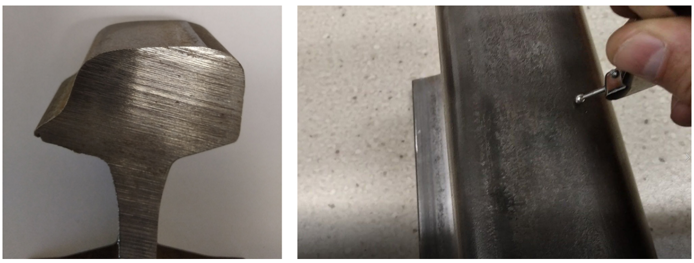

Fig. 4. Measurement of rail head profile and manual guidance of the measuring tip [Photo. J. Kukulski]

The obtained results of measurements of rail head profiles are presented in Chapter 5 of the current work.

4. Simulation research

The model used in the work is the model of the 4-axle Mk III passenger car described in [35]. The nominal model of this vehicle is shown in Fig. 5a. The parameters of this car model are also given in [35]. The model consists of 7 rigid bodies. The ULYSSES program [36] of AGEM type (Automatic Generation of Equation of Motion) was used to build the model. Kane's equations adapted to the description of relative motion were used [35, 37]. The original form of these equations, including vectors and tensors, was not used. In the automatic generation of the equations, a matrix form based on Huston's results was used [38], instead.

The 4-axle vehicle model was supplemented with laterally and vertically flexible discrete track models. These models are shown in Figs. 5b and 5c, respectively. The nominal track gauge of $2b = 1.435$ m was used for the tests. The track models increase the number of degrees of freedom of the system by 3 for each of the four wheelsets. Another adopted in the

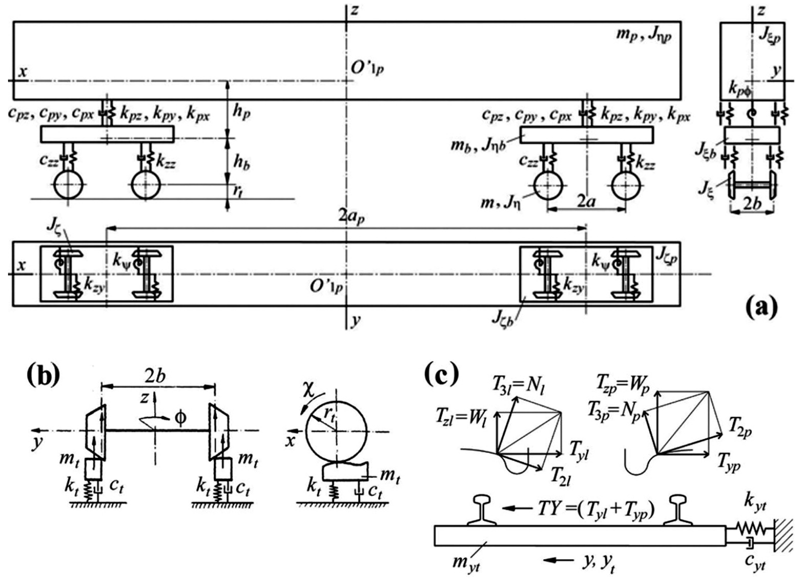

Fig. 5. Nominal models [26]: a) Mk III vehicle model; b) track laterally, c) track vertically

track-wheelset system, there are two constraints linking the vertical displacements of rails with the vertical and lateral displacements of a given wheel. These constraints reduce the number of system degrees of freedom by 2 for each of wheelsets. There are also zeroing constraints for the longitudinal displacement x and pitch angle χ coordinates of wheelset (2 constraints for each of 4 wheelsets). Thus the number of degrees of freedom lst of the Mk III car-track system equals:

$$lst = [(6 \cdot 7) + 3 \cdot 4] - (2 \cdot 4 + 2 \cdot 4) = 38. \tag{4.1}$$

The spring and damping elements of the vehicle and track models were assumed to be linear.

One of the most important elements of modelling the dynamics of a rail vehicle is the problem of wheel-rail contact. From the point of view of this work, the publications [39] and [40] are important, as they describe the modelling of the wheel-rail contact adopted in the used vehicle model. The work [40] describes contact forces calculation in detail. In general, in the issues of contact, the following elements are sought to be resolved:

– contact geometry description,
– the so-called normal contact problem, including the determination of normal contact forces,
– determination of tangential contact forces,
– description of forces and torques acting on wheelset.

The description of the contact geometry is entered into the model using the table of contact parameters generated by the ArgeCare RSGEO program. It is a single table generated for the zero tire relaxation angle. The table contains the rolling radii of the wheels and the contact angles as a function of the relative displacement of the wheel and rail (wheelset and track). The model uses a pair of S1002/UIC60 wheel profiles/rails running on a track with a rail inclination of 1:40.

5. Results of the research

During the research in the experimental part, wear measurements of the 60E1 rail elements were carried out. These measurements were made for three variants of rails wear: the new (nominal) profiles, profiles of medium wear, and quite heavily worn ones. The measurement results of the nominal profile and the worn rail head profiles as well as the change of rail head wear angle are shown in Figs. 6–8.

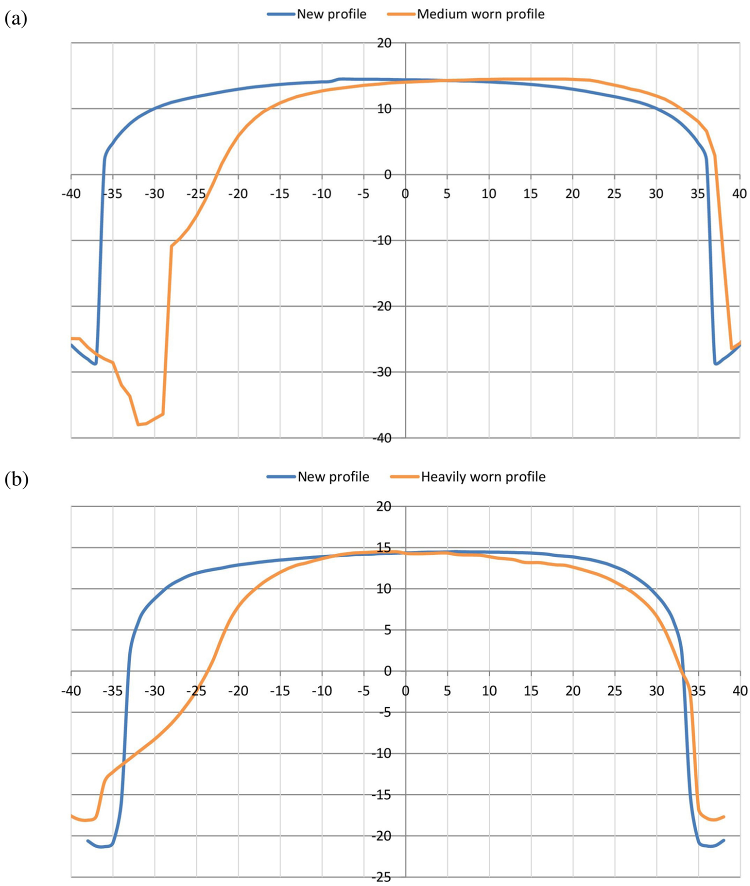

Fig. 6. Rail profiles 60E1 new, medium (a) and heavily (b) worn

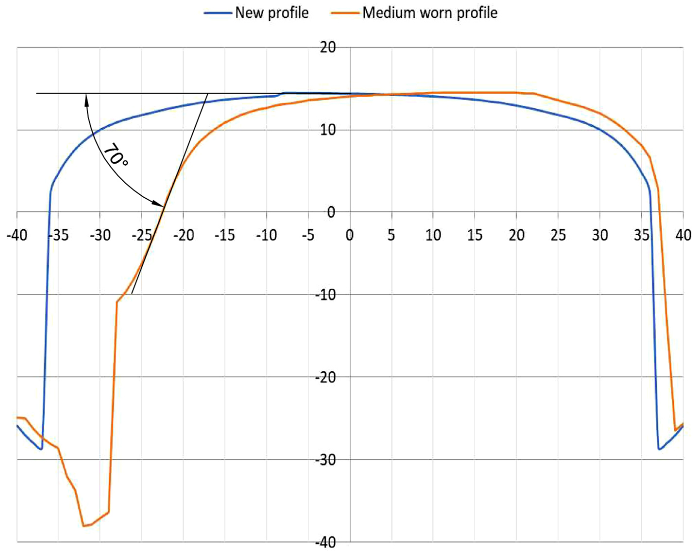

Fig. 7. Wear angle of 60E1 rail head medium worn

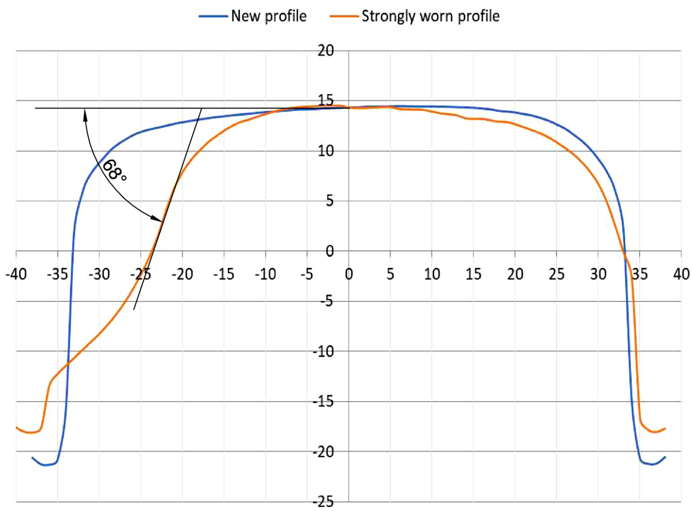

Fig. 8. Wear angle of 60E1 rail head heavily worn

The obtained measurement results show that the analysed rails should be counted to eligible for replacement. The lateral wear was 9.70–13.76 mm, and the inclination angle of the lateral surface was 68–70°. Despite the fact that the permissible values were not exceeded, dismantling of the rails done due to modernisation works was purposeful, as it leaded to an increase in traffic safety, e.g. by preventing an eventual train derailment. The limit values for the side wear of the rail head are defined in the documents of the Infrastructure Manager

PKP PLK S.A. Instruction Id-1 (Technical conditions for the maintenance of pavement on railway lines) in Annex 14 represents the permissible values of rail cracks per km of line, acceptable vertical and lateral wear of the rail head and the angle of inclination of the side surface of the rail head.

During the research, in the second part, simulations were performed on complex routes similar to the first part of the straight track (ST), transition curve (TC) and circular curve (CC). In the simulation studies, the transition curve used was always 3rd degree parabolic curve. The vehicle motion always proceeded at a constant velocity v, which was varied. The individual vehicle was considered in this part. Formally, lateral displacements of isolated vehicle are different than for the vehicle within the train. On the other hand, just professionals in railway vehicle dynamics realize how weakly vehicle lateral dynamics is coupled with vehicle vertical and longitudinal dynamics. This is due to domination of the contact forces (especially tangential ones) over the external forces acting on the vehicle. This especially concerns lateral displacements (paths) of wheelsets. That is why so often when lateral dynamics is in view single vehicle is studied. This is even true in curved track for railway vehicles. Only in tram tracks, at extremely small curve radii the contact forces cannot beat the external forces. The whole train dynamics is especially important for the longitudinal dynamics. The simulations were performed for two velocities $v = 11$ and 22 m/s in each of the three track wear variants. The radius of the circular curve was always $R = 1100$ m. Initial lateral displacements $y_1(0) = y_2(0) = y_3(0) = y_4(0) = y_{b1}(0) = y_{b2}(0) = y_p(0) = 0$ m were imposed on all 7 rigid bodies of the model. The symbols used in sequence refer to the wheelsets (front to rear), bogie frames (front to rear) and car body. The track superelevation h is selected as equal to $h = 0.075$ m. This means almost equilibrium of components of gravity and centrifugal forces in the track plane for $v = 22$ m/s. The route parameters are given in the captions under the figures. Precisely, however, very small superelevation excess exists for this velocity. In case of $v = 11$ m/s considerable superelevation excess exists. The diagrams in the figures show the lateral displacements y of the wheelsets (y_1, y_2, y_3, y_4) and bogie frames (y_{b1}, y_{b2}) as a function of the road. Charts for body (y_p) are omitted.

The obtained results of the simulation tests are shown in Figs. 9–14. The drawings were arranged in pairs depending on the rail wear (three variants of track wear). Figures 9 and 10 were obtained for the nominal track model (unworn rail profiles). Figures 11 and 12 are slightly worn track wear variant and Figures 13 and 14 for a heavily worn track wear variant. In each pair of figures, the first result was obtained for the lower velocity $v = 11$ m/s (approx. 40 km/h), and the second for the higher velocity $v = 22$ m/s (approx. 80 km/h).

In Figures 9 and 10, for the nominal track with no wear, there are no vibrations in ST and TC. In CC there are also no vibrations in Fig. 9, obtained for the velocity $v = 11$ m/s, while for the velocity $v = 22$ m/s (Fig. 10), self-excited vibrations appear for the rear bogie (for its wheelsets and the frame represented by displacements y_3, y_4 and y_{b2}). However, these vibrations have small amplitudes of the order of 0.001 m, so they can be ignored.

The solutions in CC are stationary for both cases, although in Fig. 10 the vibrations of small amplitudes could already be considered a periodic solution. The values of lateral displacements in ST, TC and CC are similar in both cases. In TC and CC, a deviation of the lateral displacements from the track centre line can be observed. It is caused by typical in

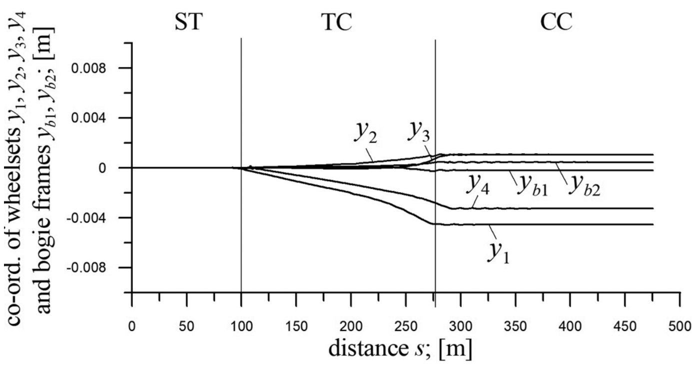

Fig. 9. Co-ordinates of wheelsets and bogie frames of passenger car Mk III; nominal track model (unworn rail profiles); $v = 11$ m/s

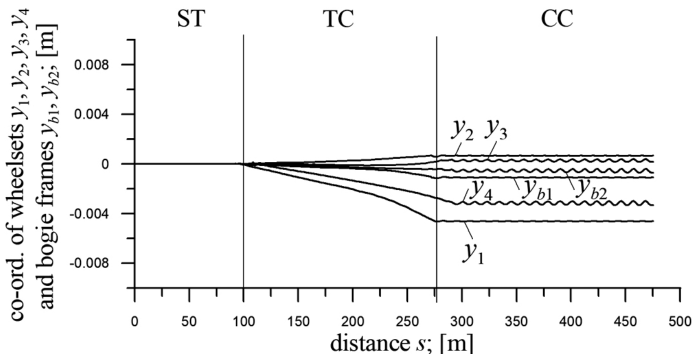

Fig. 10. Co-ordinates of wheelsets and bogie frames of passenger car Mk III; nominal track model (unworn rail profiles); $v = 22$ m/s

curved track differences in tangential contact forces for the inner and outer wheels. In both velocity cases, the vehicle travels below the critical velocity.

The results presented in Figs. 11 and 12 are quite similar to each other. However, generally, the lateral displacements are a bit bigger for the higher velocity $v = 22$ m/s. In both cases there are no vibrations in ST, which is caused by the lack of the forcing factor (this applies to all six figures). In TC, initially the vibrations are excited, but have a decreasing amplitude and they disappear very quickly. In Fig. 11 for a velocity $v = 11$ m/s it is approximately in 1/3 of the TC length, and in Fig. 12 for a velocity of $v = 22$ m/s it is a bit later, just before 1/2 of the TC length. Lack of vibrations in almost whole TC is continued in CC. For both cases of v stationary solutions are observed in CC.

The values of lateral displacements along the entire route length are similar in both cases. As before, in TC and CC a deviation of lateral displacements from the track centre line can be observed, which is caused by the tangential contact forces values. In both cases, the vehicle travels below the critical velocity.

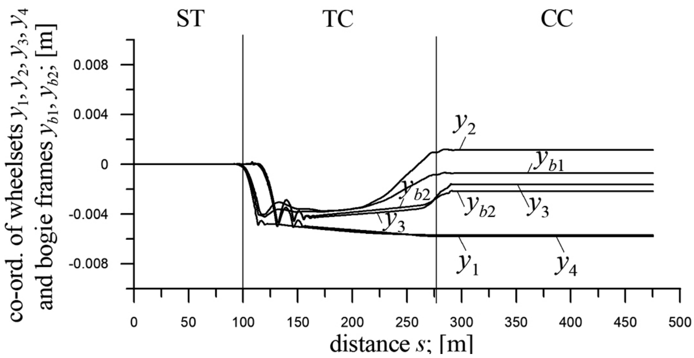

Fig. 11. Co-ordinates of wheelsets and bogie frames of passenger car Mk III; slightly worn track variant; $v = 11$ m/s

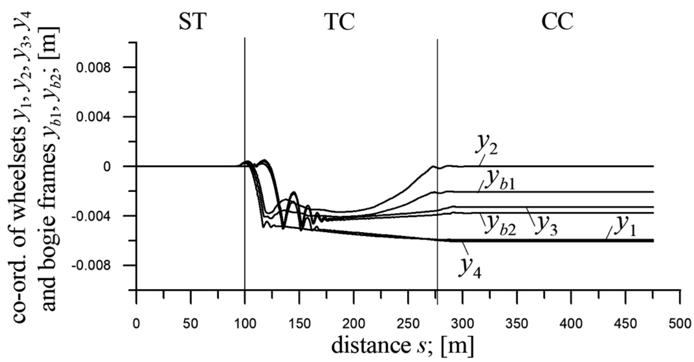

Fig. 12. Co-ordinates of wheelsets and bogie frames of passenger car Mk III; slightly worn track variant; $v = 22$ m/s

The third pair of results, Figs. 13 and 14, obtained for a heavily worn track variant shows different vehicle behavior for the velocity $v = 11$ m/s (Fig. 13) and $v = 22$ m/s (Fig. 14). In the first case (Fig. 13), there are no vibrations in ST. In TC, vibrations appear immediately after entering it and disappear at its end. However, the amplitudes of these vibrations are very small and the vibrations are irregular. There are no vibrations in CC. One can observe stationary solutions there. An interesting phenomenon in this case is the occurrence of slight vibrations in TC despite the lack of vibrations in ST and CC. The vehicle, as before, is moving at a velocity below the critical velocity. The completely different behavior of the vehicle is shown in Fig. 14. Here vibrations are excited already in ST just after 1/2 of ST length. A limit cycle with a fairly large amplitude is established in ST. This could be excited by contact conditions due to rails wear applied a bit asymmetric for the left and right track side. In TC, the vibrations have a decreasing tendency. In CC, a boundary cycle is established for both wheelsets and bogie frames, so we observe periodic solutions. For the front wheelset and bogie frame (y_1, y_2 and y_{b1}) there is a lower amplitude limit cycle, while for the rear wheelset and bogie frame (y_3, y_4 and y_{b2}) there is a higher amplitude limit cycle. In Fig. 14 one can observe

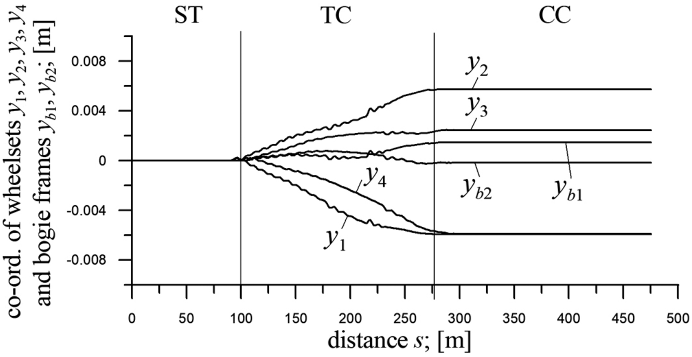

Fig. 13. Co-ordinates of wheelsets and bogie frames of pass. car Mk III; heavily worn track variant; v = 11 m/s

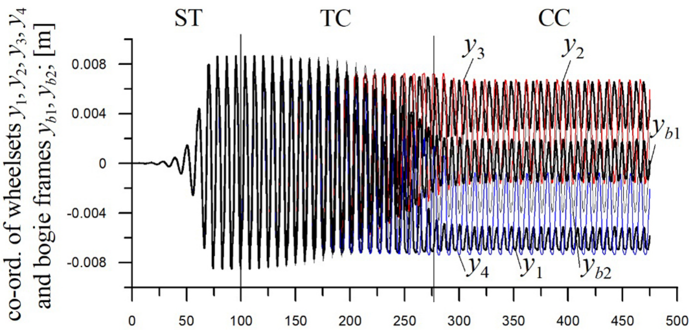

Fig. 14. Co-ordinates of wheelsets and bogie frames of pass. car Mk III; heavily worn track variant; v = 22 m/s

a smooth transition of vibrations from ST to CC. This result is similar to the behavior of the vehicle in Fig. 10, where while no vibrations were observed for the front bogie frame and its wheelsets, a very low amplitude limit cycle was observed for the rear bogie frame and its wheelsets. In Fig. 14, the vehicle is traveling at a velocity above the critical velocity, both in ST and in CC.

6. Discussion

Track wear has a direct impact on the safety of the vehicle motion, and thus also on the maximum permissible velocity with which the vehicle can travel on this track. There is no need to convince anyone that these are two of the features most expected by producers and users. Everyone, on the other hand, wants to move quickly and safely. For many years, there has been a trend towards a continuous increase in the velocity of rail vehicles. However, this cannot be done without neglecting safety. It is clear from Figs. 9–14 of this article that the critical velocity v_n decreases along with the increase in rail wear. It is the velocity at which the vehicle starts to perform hunting motion and its behaviour is represented by stable periodic solutions less favourable than stationary behaviour in terms of safety of the moving vehicle. The tests were performed for two velocities $v = 11$ and 22 m/s. For the nominal track and both velocity cases, the vehicle moves at a velocity below the critical velocity (see Figs. 9 and 10). In terms of critical velocity, the vehicle behaves similarly in the case of a slightly worn track (see Figs. 11 and 12). Here also, the critical velocity for both straight track and circular curve is above $v = 22$ m/s. On the other hand, the vehicle behaves differently in the case of motion on a heavily worn track and higher velocity $v = 22$ m/s (Fig. 14). For the velocity $v = 11$ m/s, we still observe stationary solutions (Fig. 13), while for the $v = 22$ m/s, the vehicle is already moving with a hunting motion with a constant amplitude, both in a straight track and in a circular curve, which means that the vehicle is traveling above the critical velocity. The critical velocity for this case is between $v = 11$ m/s and $v = 22$ m/s, but closer to 22 m/s. This means that it is very dangerous to drive the vehicle at such a velocity on a heavily worn track. On the other hand, it remains safe to move at the same velocity along the nominal and slightly worn track. In conclusion, in order for the rail vehicle to be able to move at a higher velocity and safely, it is recommended to replace heavily worn rails with new ones.

It is also reasonable to expand the traditional criteria of rails replacement based on just on measuring the wear with simulation studies in order to get higher accuracy and recommendation to replace the rails earlier.

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Wpływ zużytych i nominalnych profili szyn na ruch i dynamikę pojazdów szynowych

Słowa kluczowe: zużycie profilu szyny, dynamika pojazdów szynowych

Streszczenie:

W pracy oceniono wpływ zużycia szyn na bezpieczeństwo ruchu pojazdów szynowych. Wykonano pomiary profili dla nominalnej główki szyny oraz dla lekko i mocno zużytej główki szyny. Następnie te zmierzone wartości rzeczywiste wprowadzono do programu komputerowego przeznaczonego do badania ruchu (dynamiki) pojazdów szynowych. Badany model pojazdu pasażerskiego poruszałsię w symulacjach po torze składającym się z odcinków prostych i zakrzywionych z krzywą przejściową między nimi przy dwóch prędkościach. Po uzyskaniu wyników symulacji porównano je dla różnych etapów procesu zużywania się szyn na podstawie wyników pomiarów profilu oraz oceniono przydatność toru. Celem niniejszego artykułu było wykazanie, że rozszerzenie tradycyjnego podejścia o badania symulacyjne ruchu i dynamiki pojazdów prowadzi do uzyskania odmiennej i znacznie dokładniejszej informacji o potrzebie wymiany szyn. Zaproponowana koncepcja i podejście jest oryginalne i nie jest stosowane w praktyce przez zarządców infrastruktury szynowej. Można go zatem traktować jako nowość i pewien wkład autorów w problematykę i metodykę utrzymania torów.

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