Gauge Face Wear Caused with Vehicle/Track Interaction

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Abstract

The authors are focussing on the influence of vehicle/track interaction such as attack angle of wheelset and lateral force interacting between wheel and rail on gauge face wear of rails and flange vertical wear of wheels. So several track site dynamic measurements have been carried out for two years at Shinkansen sharp curved tracks to understand the actual vehicle/track lateral and vertical interactions and to estimate the influence of gauge face wear on the vehicle/track interaction. Also, the experiments of gauge face wear have been carried out using a large twin-disc machine to study the effect of attack angle and lateral force on the amount of wear. Then, 3-D FEM analysis of contact stresses has been performed to estimate the possibility of plastic deformation and wear taking place at rail gauge corner and wheel flange contact.

In this study, the large variations of vehicle/track interaction with types of bogies and the great influence of gauge face wear shape on the vehicle track interactions were obtained from the track site measurements. Also, the significant effect of attack angle on the amount of gauge face wear was obtained in the experiments using a large twin-disc machine, which meant the increase of attack angle let the contact point between wheel and rail specimens moved toward the increase of sliding speed. In general, sliding speed is roughly proportional to wear amount. From the FEM analysis, the possibility of very large plastic deformation and wear on the contact surface between wheel and rail was identified so that the effect of profile on reducing the amount of wear was estimated.

Keyword: Gauge face wear, Attack angle, Contact stress, 3-D FEM analysis, Sharp curve
1. Introduction

Renewing worn high rails at sharp curved tracks relatively has a significant influence on maintenance costs. Because preventive grinding has a great effect on reducing rolling contact fatigue defects called squats [1][2]. It is important to clarify some factors that progress the wear of high rails, and predict and estimate the gauge face wear based on the effect of those factors [3]. It is expected that the prediction of gauge face wear will effectively reduce the scheduled maintenance work and costs. The authors are focusing on the effect of dynamic vehicle/track interaction, such as lateral force, attack angle, and others, on the amount of wear.

Then, the authors measured the vehicle/track interaction when cars ran on new rails and worn rails at sharp curved tracks of Shinkansen to understand the actual rail/wheel behaviors. Also the authors carried out FEM contact stress analyses and laboratory experiments to study wear phenomenon. This paper describes discussions on the results and information obtained from these analyses and experiments.

2. Dynamic measurements of vehicle/track interaction at sharp curved track

2.1 Test arrangements

The authors measured wheel load, lateral force, the lateral deflection of railhead and attack angle at curved tracks with a radius of 400m and 900m in Shinkansen. The test arrangements at track site are shown in Table 1. The design profile of JIS60, which was adapted for this track, is shown in Fig. 1 and the progress of gauge face wear is shown in Fig. 2. Specifications of the rolling stock used for the measurement are shown in Table 2, and the wheel design profiles are shown in Fig. 3. The balancing cant of curve radius 400m is 120mm and that of curve radius 900m is 55mm. The train speed at these points is about 65km/h.

<table>
<thead>
<tr>
<th>Curve radius (m)</th>
<th>Rail type</th>
<th>Cant (mm)</th>
<th>Accumulated passing tonnage (MGT)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>New</td>
</tr>
<tr>
<td>400</td>
<td>JIS60</td>
<td>95</td>
<td>0.5</td>
</tr>
<tr>
<td>900</td>
<td>(Heat-treated)</td>
<td>35</td>
<td>1.4</td>
</tr>
</tbody>
</table>

Table 1 Test arrangement of the track
Fig. 1 Design profile of JIS60 rail

Fig. 2 Progress of gauge face wear
Table 2 Specifications of rolling stock used for the measurement

<table>
<thead>
<tr>
<th>Vehicle type</th>
<th>Old</th>
<th>New</th>
<th>Special</th>
</tr>
</thead>
<tbody>
<tr>
<td>Static wheel load</td>
<td>8.5t</td>
<td>6.5t</td>
<td>5.5t</td>
</tr>
<tr>
<td>Wheel tread</td>
<td>Cone</td>
<td>Arc</td>
<td>Arc (400type)</td>
</tr>
<tr>
<td>Wheel diameter</td>
<td>910mm</td>
<td>860mm</td>
<td>860mm</td>
</tr>
<tr>
<td>Wheel base</td>
<td>2500mm</td>
<td>2500mm</td>
<td>2250mm</td>
</tr>
<tr>
<td>Bogie type</td>
<td>Direct mount</td>
<td>Bolsterless</td>
<td>Bolsterless</td>
</tr>
</tbody>
</table>

Fig. 3 Wheel design profiles

(1) Cone tread

(2) Arc tread

(3) Arc tread (400 type)

2.2 Results of measurement

(1) Profiles of worn rail

Fig. 4 shows the radii at the running band of a high rail at the curved track with a radius of 400m. The origin of abscissa, which represents railhead points, is the gauge corner at 14mm below the rail crown because wear normally doesn’t progress under 14mm below the rail crown. The radius of worn rail at design radius of 13mm is almost the same as that of new rail, but that of worn rail at design radius of 50mm and 600mm are different from that of new rail. The radius of worn rail at design radius of 50mm is larger than 100mm and that of worn rail at design tread radius of 600mm is almost flat and partly less than 100mm. The worn running band of rail is not smooth because of rail grinding. Therefore, different profiles of worn rail must be checked to evaluate the worn running band of rail.
Fig. 4 Radii at the running band of a high rail (The track of curve radius 400m)

(2) Lateral force of high rails

Fig. 5 shows lateral forces of new and worn high rails caused by leading axles of various types of cars at the curved tracks with a radius of 400m and 900m. Fig. 6 shows the wheel number to mean that the first and third axles are leading axles. The lateral forces of high rail at curved track whose radius of curvature is 400m are almost twice as large as that at the curve of radius 900m on account of lateral forces due to curve negotiation because the cant deficiency is almost the same for the two curved tracks. In this figure, the top, bottom and middle lines represent maximum, minimum and average lines, respectively. As a whole, there is a tendency for the lateral forces of worn rail at both curved tracks to be more stable than that of new rail. On the other hand, the changes of lateral forces from new rails to worn rails are not constant from these measured data, because the tendency of change is different from the two curved tracks by type of cars.

Next, Fig. 7 shows the averaged lateral forces of new rails caused by the first and third axles of vehicle at the curved track of radii 400m and 900m. The lateral forces caused by the first axle of an old type car at both two curved tracks are larger than that of any other type of car.
Fig. 5 Lateral forces of new and worn high rail caused by leading axles

Fig. 6 Location of wheelset
(3) Lateral deflection of high rail

Fig. 8 shows lateral deflection (lateral displacement of railhead) of new and worn high rails caused by leading axles at the curved tracks with a radius of 400m and 900m. The plus signs of the deflection mean outward direction of the gauge, and the minus signs mean the inward direction in this figure. The lateral deflection is almost proportional to the lateral force, the following which the deflection caused by old type cars, which generate large lateral forces when running on the both tracks, are larger than those by any other type cars. Also, two interesting phenomena were observed at curved track of radius 900m. One was that some lateral deflection of high rail was into the inward direction, and the other was that the deflections of worn rails were scattering more than that of new rails. Some possibility can be understood, but further investigation to clarify the reason why these deflections of worn rails scattering in wide ranges will be expected.

Next, Fig. 9 shows the average lateral deflections of new rails on the curved track with a radius of 400m and 900m which were excited by the first and third axles of bogie. In figures 7 and 9, the railheads deflect almost in proportion to the lateral forces. Generally, the lateral deflection of high rails shifts outward by the lateral force but when the lateral force is small, the lateral deflection can shift inward depending on the position of rail/wheel contact and the magnitude of wheel load.
Fig. 8 Lateral deflection of new and worn high rails caused by leading axles

Fig. 9 Averaged lateral deflections of new rails caused by first and third axles of a bogie
(4) Attack angle of high rail

Fig. 10 shows attack angles of new and worn high rails caused by the leading axles on the curved tracks with a radius of 400m and 900m. The attack angle of worn rails at curved track of radius 900m was measured after accumulated passing tonnage of 7500 MGT, and the amount of gauge face wear was 3.5mm at that time. In this figure, the attack angles of worn rails irrespective of the rolling stock type are smaller than that of new rails.

Next, Fig. 11 shows the average attack angles of new rails excited by the first and third axles at the curved tracks of curve radius of 400m and 900m. In figures 7 and 11, the attack angles excited by the first axle of the old type cars of which the lateral forces are the largest at the both curved tracks are the largest.
3. Laboratory simulation on gauge face wear
3.1 Testing machine and test specimen

The laboratory experiments on gauge face wear were carried out using a large twin-disc testing machine which was developed at RTRI to study rolling contact phenomena between rails and wheels[4]. Fig. 12 shows the schematic structure of testing machine and test arrangement of the rail/wheel test specimen interface. Specimens of wheel and rail disc are 500 mm and 350 mm in diameter, respectively.
During the test, the circumferential speed of rail disc was initially set at 70km/h, but it had to be decreased to 30km/h of minimum speed to prevent from more than 88m/s² of vibration acceleration of axle boxes for protection of the testing machine. Both rail and wheel test specimens are made of the same materials as commercial rails and wheels used in Japan. The initial contact geometry between rail and wheel specimens is the same as that on the Shinkansen line. To simulate actual contact conditions between wheel and rail, the contact angle of 1/40 was taken as equating to the tilting angle in which rail is installed for Shinkansen track. In addition, the attack angles of 0° and 0.3° were set to estimate their effects on gauge face wear. Here 0.3° is generally estimated to be corresponding to the curved track of radius about 400m. Also, in Fig. 10, the averaged attack angle of old type of Shinkansen vehicle at the curved track of radius 400m is roughly 0.3° and a bit more. The other measured data are smaller than 0.3°. The profiles of the cross-section for the rail specimen were taken up with replica after each 0.2 million cycles and measured with a laser system.
3.2 Test arrangements

A 3-D contact model, as shown in Fig. 13, has been developed for estimating the contact stress of actual wheel/rail interface. The linear elastic finite element model was adopted to calculate contact stresses between the ø860 conical wheel and 60kg rail under vertical load of 85kN and lateral load of 34kN. The vertical load of 85 kN is the standard design load of track structures and components in Shinkansen, and the lateral load of 34kN is estimated as two times of standard deviation of lateral load distribution caused by train running at sharp curves in Shinkansen. In Fig. 5, compared with measured lateral forces at track site, the applied lateral force of 34kN to experiments are roughly estimated to be corresponding to the averaged measured data of old type of Shinkansen vehicle at the curved track of radius 400m.

![Fig. 13 3-D FEM model of rail/wheel contact](image)

From the results of stress analysis, the maximum compressive stresses are 1025MPa at railhead and 2378MPa at gauge corner for attack angle of 0° and 1040MPa at railhead and 2680MPa at gauge corner for attack angle of 0.3° under actual wheel/rail contact. The stresses of railhead show a good agreement with Hertzian contact stress. It is found that the stresses of gauge corner are extremely large, more than two times those of railhead. And with increasing the attack angle, the contact stresses at railhead do not change very much, but that at gauge corner increases 12.7 percent.

On the other hand, the stress analysis of experimental ø500wheel/ø350rail discs contact was performed in order to determine applied test loads according to the maximum compressive contact stress in gauge corner and rail head of actual wheel/rail. Based on the results of stress analysis, with equivalent to full scale wheel load of 85kN and lateral load 34kN, radial load and thrust load in laboratory simulation were 25.5kN and 14.6kN for the attack angle of 0°, 26.4kN and 16.7kN for the attack angle of 0.3° respectively. The test arrangements are shown in Table 3.
4. Discussions on experimental results

Fig. 14 shows the variations of cross-sectional profile of rail disc specimens with the increase of rolling cycles. For the attack angle of 0°, the smooth wear profiles of rail disc until 30 MGT (1.8 million cycles) were shown in Fig. 14(a). At 40 MGT (2.4 million cycles), the wear profile was not smooth due to the repeated contact of rail/wheel discs in the same positions. In the case of the attack angle of 0.3°, the wear profiles seem more similar to the actual situation in the rail wear process. It is found that the wear in initial stage mainly occurs at gauge corner, then develops gradually down to gauge face.
Fig. 14 Worn profile at gauge face obtained from laboratory simulation
Fig. 15 shows the relationships between wear amount and number of repeated cycles for the attack angle of 0° and 0.3°. Here, the wear amount of railhead was measured at the position of 32.5mm away from gauge corner, and the maximum value of the wear was recorded at the gauge corner. It can be seen that the wear amount of the rail head increases gradually with repeated rolling cycles, and has no obvious difference between the attack angle of 0° and 0.3° until 40 MGT. However, the wear at gauge face increases rapidly both for attack angles of 0° and 0.3°. The wear amount of 0.3° attack angle was larger than that of the attack angle of 0°. The maximum wear amount of about 7.23mm and 4.83mm at gauge face were obtained for the attack angle of 0° and 0.3° after 40 MGT, which was corresponding to 40 million passing tonnage. Compared with the wear amount of actual curved track obtained in this study, the results of laboratory simulation, in which the attack angles of 0° and 0.3° were applied, located between the data of wear amount obtained at the actual curved tracks of radii 400m and 900m. Also, the wear amount of actual curved track of radius 400m is similar to that of laboratory simulation in which the attack angle of 0.3° and the lateral force of 34kN were applied. There are absolutely many different aspects from each other between the mechanical conditions of actual track and the mechanical arrangements of experiments. However, roughly speaking, the measured data of wear amount at the actual track are similar to the experimental data of rail discs, the following which means the appropriate lateral force, attack angle and other arrangements may be applied in the laboratory simulation.

5. Concluding Remarks
Some measured data at track site in this study have some variations so that the influence of wear on lateral force is not clearly identified. However, some good achievements were obtained from dynamic measurements as follows:

The lateral forces of high rail at curved track whose radius of curvature is 900m are almost twice as large as that at the curve of radius 400m on account of lateral forces due to curve negotiation because the cant deficiency is almost the same for the two curved tracks. As a whole, there is a
tendency for the lateral forces of worn rail at both curved tracks to be more stable than that of new rail. The lateral deflection is almost proportional to the lateral force, the following which the deflection caused by old type cars, which generate large lateral forces when running on the both tracks, are larger than those by any other type cars. Generally, the lateral deflection of high rails shifts outward by the lateral force but when the lateral force is small, the lateral deflection can shift inward depending on the position of rail/wheel contact and the magnitude of wheel load. The attack angle of worn rails irrespective of the vehicle types is smaller than that of new rails. The attack angles excited by the first axle of the old type cars of which the lateral forces are the largest at the both curved tracks are the largest.

Roughly speaking, the measured data of wear amount at actual tracks are similar to the experimental data of rail discs, the following which means the appropriate lateral force, attack angle and other arrangements may be applied in the laboratory simulation.

In this study, the influence of vehicle/track interaction on gauge face wear was investigated and roughly understood. Then, the possibility of prediction of gauge face wear based on vehicle/track interaction can be identified form the achievements of this study. Further study will be expected.

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Reference