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Rubber Pad Intensity and Track Stiffness of Double Slip Turnout in Heavy Haul Railway

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Abstract: Double slip turnout is common in heavy haul railway and due to its complex structure, its maintenance is difficult as well as important. Focusing on No.12 double slip turnout (75 kg/m rail), this study analyzed the intensity of the different slotted rubber pad under iron plate and track stiffness of turnout zone based on the finite element method with a purpose of improving the performance of the double slip turnout. Results show that: the depth of groove on rubber pad and groove center distance have great influences on the equivalent stresses of iron plate and the pad; under pressure of 85.52 kN on sleeper, the maximal equivalent stresses of iron plate and the pad under iron plate are 72.959 and 2.141 MPa, respectively, both smaller than their allowable value; the maximal longitudinal variation rates of entire track stiffness are 1.42 and 1.34 respectively; the stiffness uniformity of the diamond-shaped double slip turnout can meet the requirement that the train runs at or below 120 km/h. All these achievements can be used as the theoretical guidance for performance optimization of double slip turnout and improving the running stability of heavy haul trains as well as extending service life of the turnout

Keywords: Double slip turnout, heavy haul railway, intensity, rubber pad, track stiffness

INTRODUCTION

Double slip turnout is a good equipment to shorten the length of station throat area, reduce the station size and improve efficiency of shunting operation. It is a bit longer than simple turnout, but its function is equivalent to two groups of simple turnouts. Another feature of the double slip turnout is that it crosses a line straightly to reduce hunting movement of trains and traveling is relatively stable (He *et al*., 2009; Yuan and Liu, 2006). However, due to big axle load of heavy haul railway and complex structure of double slip turnout, there exists many deficiencies in its daily maintenance, so some components are damaged and become worse, which is maintenance difficulty (Zuo, 2005). Under the severe traffic conditions, non-uniform stiffness of turnout zone will intensify wheel/rail dynamic response, so the running stability of the train is affected and the track components stress is increased (Lopez, 2011; Luo and Zhu, 2004; Wang *et al*., 2010). Therefore, it is necessary to study the distribution law of track stiffness and to carry out homogenization measures to uniform distribution of track stiffness longitudinally along the railway line so as to increase running stability of the train and extend the service life of the turnout (Chen and Wang, 2006).

Fig. 1: Diamond-shaped double slip turnout

METHODOLOGY

Diamond-shaped double slip structure: Figure 1 show the structure layout of diamond-shaped cross-points and this part is made by double slip switches and obtuse fork. The stock rail and wing rail adopted is made by 75 kg/m rail (Chinese manufacturing); the switch rail and nose rail are made by 60AT rail. Fastener system is elastic separated structure and there are rubber pads under the rail and iron plate. The rubber pad under rail is single-sided slotted type; the rubber pad under iron plate is double-sided slotted type; the fastening is ω shaped fastening elastic rod; turnout sleeper is reinforced concrete structure.

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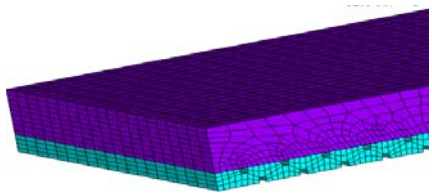

Fig. 2: Local calculation model of rubber pad under iron plate

This study studies on the intensity of key components and track stiffness of the double slip turnout to evaluate whether the diamond-shaped structure is reasonable so as to provide related reference for the optimization of double slip turnout.

Intensity analysis of the rubber pad under iron plate: The rubber pad under rail is single-sided slotted type, small groove depth and moderate stress level. The rubber pad under iron plate is double-sided slotted type with deep groove, so the stress concentration is easy to be induced at groove bottom edges and the deformation of the rubber is changed from simple compression (before being slotted) to combination of compression, bending and shear (after being slotted), which results in increasing of stress level of the pad and its service life is shortened. Therefore, it is necessary to analyze the intensity of rubber pad under the iron plate to ensure its intensity meets requirements.

Calculation model: The pressure on sleeper is transferred to the rubber pad through the iron plate, which means the pressure area of the pad is increased and this is favorable to reducing its stress level. So in calculation of the stress, the iron pad should be considered as the boundary conditions to establish a common bearing system of iron plate and rubber pad. Besides, the rubber is super elastic material and its stress/strain relation is non-linear.

According to above explanation, a calculation model is established (Fig. 2) to analyze the stress of the rubber pad under iron plate. In this model, both the iron plate and rubber pad are simulated by three-dimensional solid elements and the purple part presents iron plate while the green part the rubber pad.

Calculation parameters and schemes: In calculation, the iron plate is 26 mm thick; the constitutive relation of the material is linear; the elastic modulus is 206 GPa; and Poisson's ratio is 0.3; the yield strength is 235 MPa; the shore hardness of rubber pad is 75; allowable stress is 3 MPa. The stress/strain relation is determined according to the Mooney-Rivlin hyperelastic theory (Huang *et al*., 2008; Xu, 2002).

According to Winkle elastic foundation beam theory, when the wheel load on the rail is 235.5 kN, the maximal pressure on the sleeper is 85.52 kN (the supporting point stiffness is 60 kN/mm), which is used to check whether the stress of the rubber pad exceeds the limit.

Table 1: Slotting schemes of the rubber pad

Schemes	Upper width (mm)	Lower width (mm)	Depth (mm)	Groove center distance (mm)
A	4	3	3	16
B	4	3	4	16
C	4	3	3	12
D	4	3	4	12

In order to analyze the influence of slotted type on the stress of rubber pad under the plate, the stress of rubber pad under different conditions (Table 1) is calculated.

RESULTS ANALYSIS

The equivalent stress distribution of the iron plate and the rubber pad under plate is shown in Fig. 3 and 4. Table 2 shows the maximal equivalent stresses under different conditions.

The following results can be concluded:

- The maximal equivalent stress of the iron plate appears on the plate surfaces at rail base center line; the maximal equivalent stress of the rubber pad under plate appears at the groove edges close to the rail bottom center line.
- The deeper the groove is, the bigger the equivalent stress becomes; the smaller the center distance between neighboring grooves, the bigger the equivalent stress becomes.
- As for scheme A, maximal equivalent stresses of the iron plate and the rubber pad is 72.959 and 2.141 MPa, respectively, both smaller than their allowable stress and their intensity satisfies the application requirements.

Analysis of track stiffness distribution in turnout:

Influence factors: The track stiffness of the diamond-shaped double slip turnout is composed of the following parts: stiffness of rubber pad under rail, stiffness of the rubber pad under iron plate, rail flexural stiffness, flexural stiffness of spacer block, supporting stiffness of ballast bed. The track stiffness will change longitudinally along the turnout due to the influences of the following factors:

- The stock rail, nose rail and switch rail are made by different rail type
- The nose rail and switch rail is sliced at the moveable part The end of switch rail and nose rail is connected by the spacer block
- The length of turnout sleeper differs
- The connectivity between sleeper and rail changes longitudinally along the line

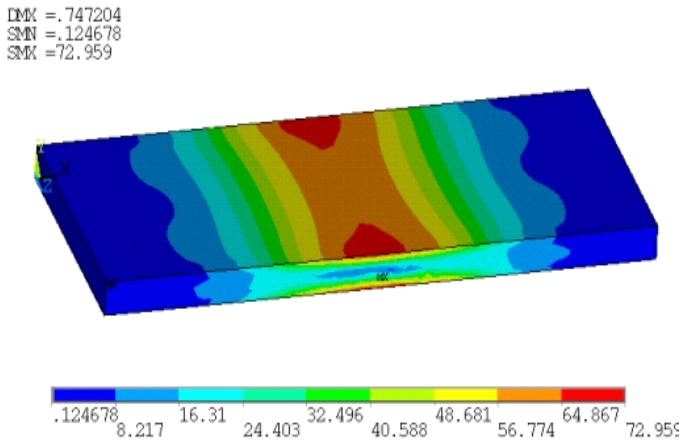

Fig. 3: Equivalent stress of the iron plate of scheme A

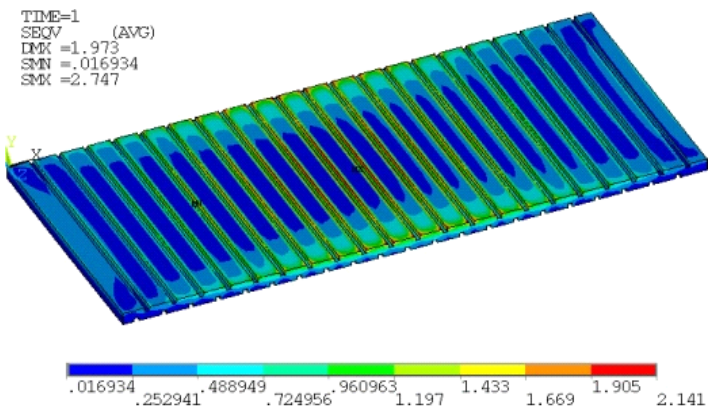

Fig. 4: Equivalent stress of the rubber pad of scheme A

Table 2: The maximum equivalent stress of different schemes (MPa)

Schemes	A	B	C	D
Iron plate	72.959	82.103	77.441	86.127
Rubber pad	2.141	2.521	2.568	2.637

- The stiffness of rubber pad under general iron plate is different from that of pad under sliding platen

Calculation method: The diamond-shaped double slip switches is much more complex than the interval railway track. For example, various types of rails, different length of the sleeper and pad, variable stiffness of the rubber pad under the plate and the spacer block and sliding plate, etc., that can't be found in the interval railway track. Uneven distribution of these components in range of double slip turnout results in changes of track stiffness (longitudinally along the line).

According to the full consideration of there influence factors, a calculation model of track stiffness is established on basis of the finite element method, as seen

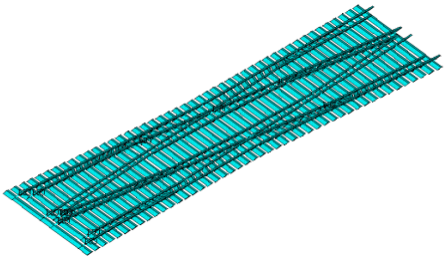

Fig. 5: Track stiffness calculation model

in Fig. 5. In this model, the stock rail and spacer block are simulated as uniform beams; the switch rail and nose rail simulated by variable cross-section beams; the fastener, the rubber pads under rail and plate are simplified as linear spring; the sleeper is simulated by limited long beam on elastic foundation.

3-d uniform beam and elastic foundation beam element have two nodes and each node has 6 Degree of

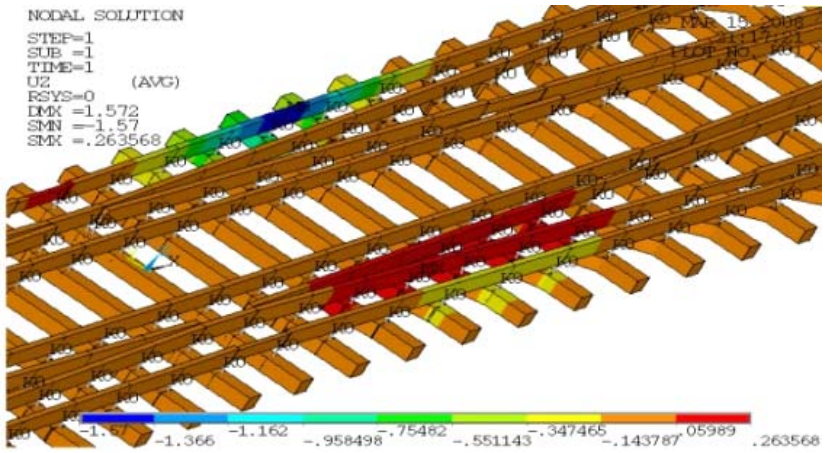

Fig. 6: Vertical displacement of the stock rail

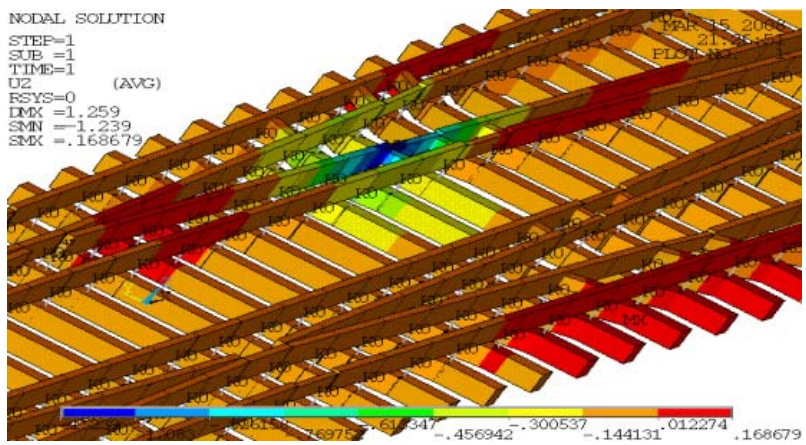

Fig. 7: Vertical displacement of the inner rail

Freedom (DOF): transitional DOFs along x, y, z and rotational DOFs around x, y, z (Wang, 2003).

There are two types of spring elements in the model. One is linear spring, which is used to simulate the fastener and rubber pad under rail. The spring element is composed of two nodes and each node only has transitional DOFs along the element's length. The same shape function can be used for both spring elements and the following is the expression:

$$u = \frac{1}{2}(u_i(1-s) + u_j(1+s)) \tag{1}$$

According to formula (1) and the minimum potential energy principle, stiffness matrix of the linear spring unit can be obtained:

$$[K_e] = k\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \tag{2}$$

Here, k is stiffness of the spring.

The entire turnout model is dispersed into the above-mentioned elements and the nodes are numbered. Finally, the stiffness matrix of each element can be used to compose the stiffness matrix of the system and to solve.

Calculation parameters: In calculation, elastic module of concrete is 36.5 GPa, Poisson's ratio 0.15, stiffness of rubber pad under rail of movable rail 380 kN/mm (with groove), stiffness of rubber pad under rail of unmovable rail 650 kN/mm (without groove); stiffness of rubber pad under ordinary plate 120 kN/mm; supporting stiffness of half sleeper is 120 kN/mm (sleeper length is 2.6 m), which means the sleeper supporting stiffness is 92.307 kN/mm/m. The rail parameters are shown in Table 3.

Quasi-static wheel load 235.5 kN is used in calculation of each node of track entire stiffness.

Track stiffness distribution of turnout: Figure 6 and 7 shows the vertical displacement distribution of the stock

Table 3: Calculation parameters of rail

Type	Cross-sectional area (cm^2)	Height (mm)	Vertical inertia (cm^2)	Lateral inertia (cm^2)
75 kg/m rail	95.04	192	4489	661
60AT rail	89.20	152	2041	767

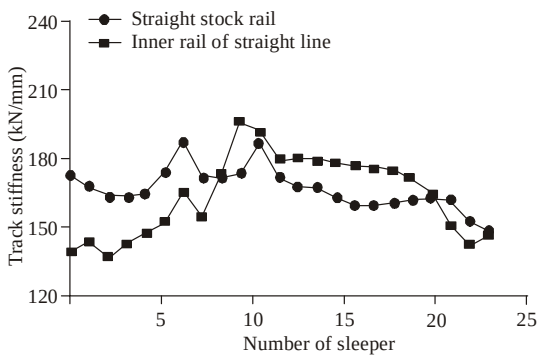

Fig. 8: Track stiffness distribution of straight line

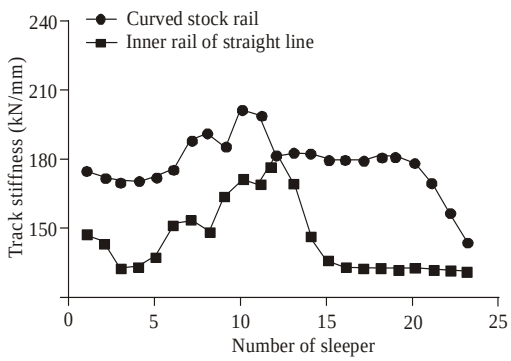

Fig. 9: Track stiffness distribution of branch line

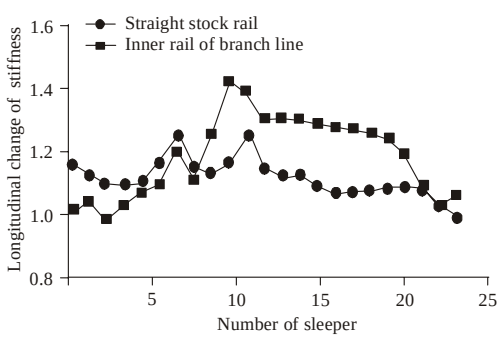

Fig. 10: Longitudinal track stiffness change rate of straight line

rail and inner rail under the 235.5 kN wheel load. Figure 8 and 9 show the longitudinal distribution of track stiffness in turnout. Figure 10 and 11 show the longitudinal change rate of track stiffness (the minimal track stiffness as the base). The extreme values of each calculation result are shown in Table 4.

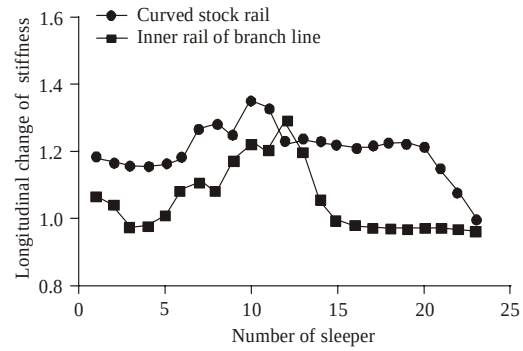

Fig. 11: Longitudinal track stiffness change rate of of branch line

Figure 6 to 11 and Table 4 show that:

- Under quasi-static wheel load 235.5 kN, the deflection range of the stock rail and inner rail is about 5 sleeper spacing, which is the same as the range of rail dynamic deflection of ordinary ballast track in China. This means the overall stiffness of the turnout is suitable.
- Under the quasi-static wheel load, rail vertical displacement is about 1.2~1.6 mm. From the perspective of track structure design, if the rail vertical displacement is controlled around 1.5 mm, the structure dynamic stress can be reduced and the service life of turnout components can be extended under the premise of ensuring the safety and comfort of the train, so it is a reasonable value to balance the train's running performance and track stress. It is obvious that the track deformation of the diamond-shaped double slip turnout is suitable.
- When the train passes the turnout from the main line (straight line), the longitudinal variation of track stiffness under the stock rail and inner rail is similar and the maximums both appear near the sleeper No.10. The longitudinal change of track stiffness is even and the maximal longitudinal change rate of track stiffness is only 1.42.
- When the train passes the turnout from the branch line, the longitudinal variation of track stiffness under the stock rail and inner rail is the same in range of sleeper No.1~12; in range of No.13~23, their difference is big. But both maximum track stiffness appear near sleeper No.10 and the maximal longitudinal change rate is only 1.34.
- In the whole, the longitudinal change rates of track stiffness in the turnout are all below 1.5; when the track geometry state is in good condition and the train's speed is at or below 120 km/h, it meets the requirement that the longitudinal change rates of the track stiffness is smaller than 1.5.

Table 4: The extreme values of results

Rails		Straight stock rail	Inner rail of straight line	Curved stock rail	Inner rail of branch line
Track stiffness (kN/mm)	Maximum	196.26	186.89	187.11	205.21
	Minimum	137.05	148.66	140.55	152.33
Maximum longitudinal change rate of track stiffness		1.42	1.25	1.34	1.28

CONCLUSION

- The deeper the groove on rubber pad is, the bigger the equivalent stresses of iron plate and the pad under iron plate become; the smaller the distance between two groove centers, the bigger the equivalent stresses of iron plate and the pad under iron plate become.
- Under pressure of 85.52 kN on sleeper, the maximal equivalent stresses of iron plate and the pad under iron plate are 72.959 and 2.141 MPa respectively, both smaller than their allowable value; both have big intensity storage and their intensity both satisfy the requirements.
- The maximal longitudinal variation rates of entire track stiffness are 1.42 and 1.34 respectively; the stiffness uniformity of the diamond-shaped double slip track turnout can meet the requirement that the train runs at or below 120 km/h. All these achievements can be used as the theoretical guidance for performance optimization of double slip turnout and improving the running stability of heavy haul train as well as extending service life of the turnout.

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