

Study of fatigue crack initiation location of wheel and rail under rolling contact using finite element method

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Abstract The rail transit system is widely used for freight and passenger transportation. Due to the fact that its economic worthiness and high safety mode. Maintenance and damage prevention of wheel and rail are important factors affecting the safety of the system. The previous studies show that the most damage of wheel and rail is fatigue cracking, which is caused by the contact stress resulting from wheel and rail interaction. This article presents the study of the fatigue crack initiation location of wheel and rail under rolling contact at the wheel speed of 80 km/h using Finite Element Method (FEM). The three dimensional finite element models were created using the UIC60E1 wheel profile and BS100 rail profile. The Dang Van criteria was applied to analyse the fatigue crack initiation location in case of the wheel's position was changed along the rail lateral direction while the rail inclination angle was also varied at 0, 1/40, 1/30 and 1/20, respectively. The analysing results show that the fatigue crack initiation, determined by the Dang Van stress ratio, tends to increase when the wheel is moved from gauge side to field side. Additionally, the fatigue crack damage is likely to decrease when the rail inclination increases up to the inclination of 1/30 and the fatigue crack initiation locations were found underneath the wheel and rail surfaces. The obtained result can be a primary guideline for maintenance planning.

1 Introduction

Rolling contact fatigue is a damage generally appearing on contact surfaces between wheel and rail. This fatigue occurs from the repeated load on a surface by millions of intensive wheel and rail contact cycles. However, the previous studies [1] reported that the several accidents of train derailment resulted from damages of wheel/rail, such as, spalling, fatigue crack, shelling, rolling fatigue and thermal crack [2]. Rolling fatigue crack is very common to all kind of trains and can be found in both wheel and rail. Thus, maintenance of wheel/rail must be the prime concern and necessary to be steadily improve. Toumi et. al. [3] employed finite element method to study rolling contact stress between wheel and rail with increased spin effect. The FEM solution is compared to the solution of the CONTACT software [4]. This paper reported that the creep force characteristic from FEM solution is slightly lower than the result obtained by the CONTACT software. Martin Pletz et al. [5] purposed a full scale test rig of rolling contact fatigue and performed the simulation of wheel and rail loading of test-rig by using FEM combined with CONTACT software. From this research, it was found that there are two contacts, one of them is near on the top and the other is on the gauge corner of its rail. The contact area on the top of the rail is bigger than the other

at the gauge corner. This might be due to the lower lateral load.

However, this article aims to employ finite element method and Dang Van criterion to predict the rolling fatigue crack initiation in the wheel/rail when the contact area is varied from field side to gauge side and the wheel is rolling under constant axle load and constant speed. Additionally, the effect of rail inclination angle will be numerically investigated.

2. Dang Van high cycle fatigue criterion

In 1993, Dang Van proposed a multiaxial fatigue criterion to predict crack initiation when the two solids are in a rolling contact. This criterion is based on the elastic shakedown principle at the mesoscopic scale. The Dang Van criterion predicts that the fatigue crack of the material will occur on the grain boundary. In the macroscopic scale, the Dang Van criterion considers this to happen when the stress inside the material reaches critical value and can be expressed by

$$\tau_{\max}(t) + \alpha_{DV} \sigma_H(t) = \tau_w \quad (1)$$

When α_{DV} is a constant to be determined, τ_w is the fatigue limit of pure torsion. $\sigma_H(t)$ is the instantaneous hydrostatic component of stress tensor and $\tau_{\max}(t)$ is the

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value of the Tresca shear stress which can be calculated by[6] :

$$\tau_{\max}(t) = \left(\frac{\sigma_1(t) - \sigma_3(t)}{2} \right) \quad (2)$$

When $\sigma_1(t)$ and $\sigma_3(t)$ are maximum and minimum principal microscopic deviatoric stress at each instant t . The constant of α_{DV} is evaluated by [7];

$$\alpha_{DV} = 3 \left(\frac{\tau_w}{\sigma_w} \frac{1}{2} \right) \quad (3)$$

In the relationship of ratio from maximum shear stress $\tau_{\max}(t)$ and fatigue limit of bending σ_W and fatigue limit of pure torsion τ_w and hydrostatic pressure. Therefore, in order to predict crack initiation, the maximum Dan Van stress ratio at any time t can be defined as

$$DV = \text{Max} \left[\frac{\tau(t)}{\tau_w - \alpha_{DV} \sigma_H(t)} \right] \quad (4)$$

In Eq.(4), if DV value is more than one, the fatigue failure will be expected to occur and the position that has the maximum DV is prone to damage faster.

3 Simulation of wheel rolling on rail using Finite Element Method

In this article, the wheel and rail geometry are based on the data of purple line project, Bangsue to Bangyai, Bangkok, Thailand. The wheel diameter is 860 mm and the rail is UIC60E1[1]. The material properties of wheel and rail are shown in Table 1.

Table 1. Material properties of wheel and rail

Properties	Wheel	Rail
Material type	SSW-Q3S	R260
Standard	JIS E 5402	EN 13674
Young's modulus, E	210×10^3 MPa	210×10^3 MPa
Yield stress, σ_y	500 MPa	528 MPa
Poisson's ratio, ν	0.25	0.25

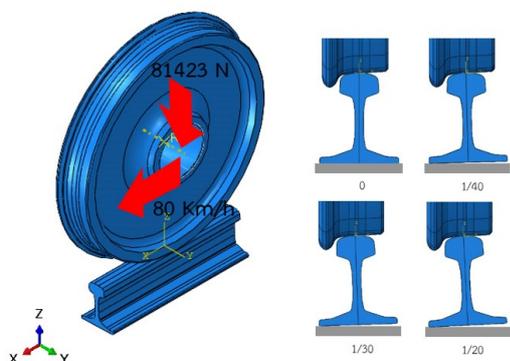


Fig. 1 Three-dimensional solid model of wheel and rail

Fig. 1 shows that the axle loading of 81,420 N is applied to the wheel and the wheel speed is 80 km/h.

Also, rail inclination is varied at 0, 1/20, 1/30 and 1/40, respectively. Contact condition between wheel and rail is frictional and the friction coefficient is assumed to be 0.3. The bottom surface of rail foot is fixed in all directions.

Fig. 2 shows the assembly FEM model of wheel and rail. In FEM model, the hexahedron and tetrahedron 3D elements are employed and the model consists of 210,541 elements. Additionally, in the contact zone, the hexahedron element size of 1 mm is employed to capture the correct contact stress distribution.

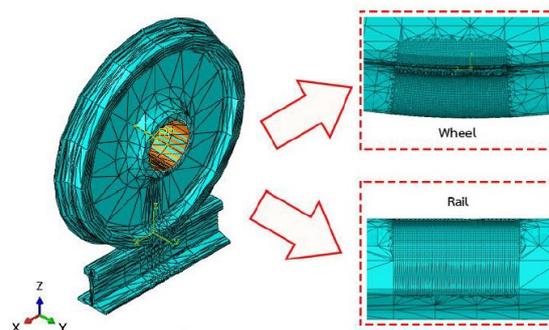


Fig. 2 Finite element model of wheel and rail

4 Results and discussions

4.1 Validation of finite element model

In order to confirm that the FEM model of Fig. 2 is valid, the Hertz's contact theory[8,9] is employed. Fig. 3 shows the stress variation underneath the contact area along z-axis obtained from FEM and Hertz's contact theory. The results obtained from both methods are in good agreement and the maximum discrepancy is approximately 1%.

Moreover, the comparison of contact dimensions and the peak contact pressure were also investigated. Table 2 shows the comparison of contact dimensions and the peak contact pressure. It was found that the finite element results are reasonable in a good agreement. Therefore, the finite element model of wheel and rail developed in this article can be accepted.

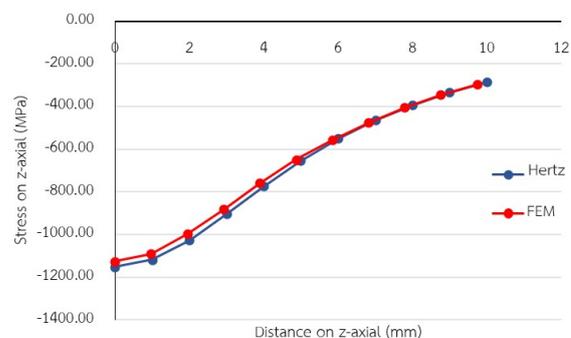


Fig. 3 Stress underneath contact area distribution along z-direction

Table 2. Comparison of the maximum contact pressure and contact dimensions

	Hertz	FEM	Discrepancy
Half of width contact a	6.53 mm	6.09 mm	6.81 %
Half of width contact b	5.13 mm	5.11 mm	0.30 %
Peak contact pressure P_0	1152.53 MPa	1153.55 MPa	0.09 %

4.2 Peak contact pressure and Von-Mises stress at rail inclination of 1/40

In this article, the peak contact pressure and area occurring between wheel and rail with the rail inclination of 1/40 were firstly investigated when a wheel lateral displacement is varied from -4 mm to 4 mm. The wheel nominal position is at 0 mm displacement. If the wheel displacement is less than 0 mm, it is considered as gauge side of rail. On the other hand, the field side of rail is a wheel displacement that is from 0 mm to 4 mm. Fig.4 shows the variation of peak contact pressure. According to the graph, it shows that the peak contact pressure is gradually increased from gauge side to field side. Due to the fact that the contact area is reduced, as shown in Fig.5. The maximum contact area is 142.31 mm² as the wheel lateral displacement is -4 mm and, at this position, the peak contact pressure is minimum, 920.55 MPa. On the other hand, with the wheel lateral displacement of 4 mm, the contact area is minimum, 88.73 mm², and the peak contact pressure is maximum, 1408.78 MPa.

Fig.6 shows the variation of maximum Von-Mises stress underneath the wheel and rail contact surfaces when the wheel lateral position is varied. It is found that the Von-Mises stress is higher in the rail. The Von-Mises stress tends to increase when the wheel lateral position is moved from gauge side to the field side, in the same manner as the peak contact pressure. Moreover, the maximum Von-Mises stresses of wheel and rail are 518.18 MPa and 549.45 MPa, respectively, which are higher than the material yield stress. Therefore, it is expected that plastic deformation will occur.

4.3 Effect of rail inclination on peak contact pressure and contact area

In this section, the rail inclination effect on peak pressure and contact area had been studied. Fig.7 shows the variation of peak contact pressure as the rail inclination angles are 0, 1/40, 1/30 and 1/20. The maximum contact pressure is found as rail inclination angle is zero. The peak contact pressure tends to increase if the wheel lateral position is varied from gauge side to field side. However, this trend is opposite to the case of 1/20 rail inclination. On the other hand, as depicted in Fig.8, the minimum contact area is found as rail inclination is zero.

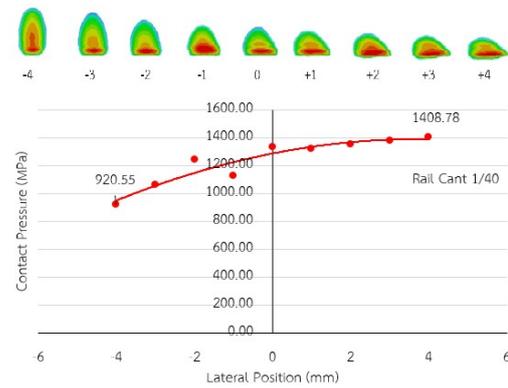


Fig. 4 Contact pressure at rail inclination of 1/40

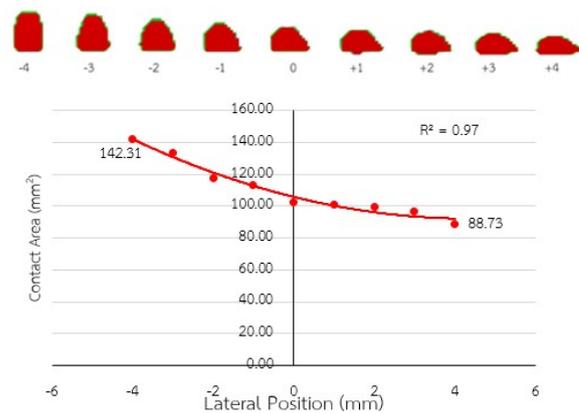


Fig. 5 Contact area at rail inclination of 1/40

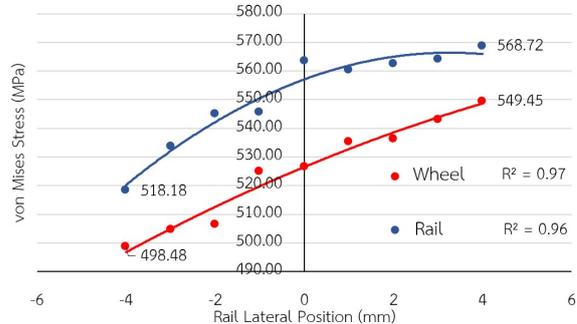


Fig. 6 Maximum Von-Mises stress variation of wheel and rail at rail inclination of 1/40

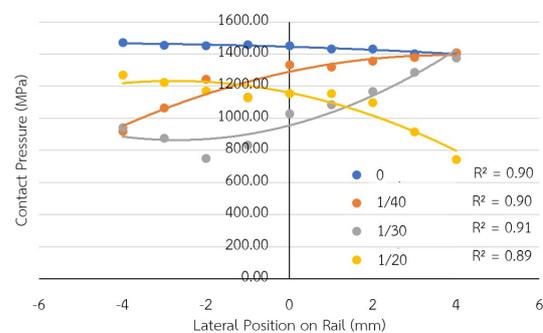


Fig. 7 Peak contact pressure at rail inclination of 0, 1/40, 1/30, 1/20

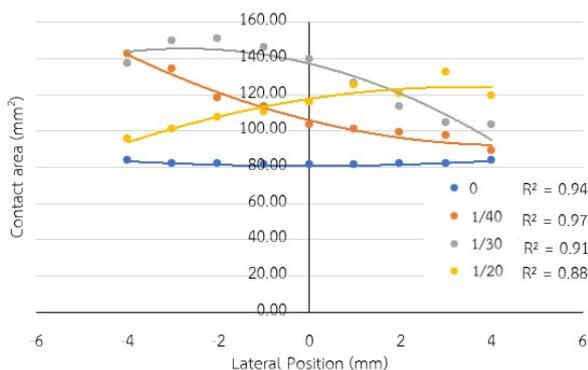


Fig. 8 Contact area at rail inclination of 0, 1/40, 1/30, 1/20

Also, the contact area is not much affected by the wheel lateral position. In the case of the rail inclination of 1/40 and 1/30, the contact area is gradually decreased if the wheel lateral position is varied from gauge side to field side. This posture is completely different from the rail inclination, which is 1/20, as shown in Fig.8.

4.4 Rail inclination effect on Dang Van Stress Ratio

Fig. 9 and 10 show the Dang-Van(DV) stress ratio, as defined in eq.(4), distribution on the wheel and rail. Maximum DV stress ratio is found as rail inclination is zero and its value is higher on the rail. Therefore, fatigue crack initiation on the rail is expected to happen faster. The DV stress ratio tends to decrease as the wheel lateral position is moved from gauge side to field side in all cases. However, this is not the case for 1/20 rail inclination. Therefore, the fatigue crack initiation is potential to happen on the field side of the rail.

From the FE results, the maximum Dang-Van stress ratio of wheel and rail were found at the depth of 2.45 mm and 5.65 mm from wheel and rail surfaces, respectively.

5 Conclusions

In this article, the numerical simulations of wheel and rail under rolling contact using finite element method were carried out by varying the rail inclination angles and the wheel lateral position, from gauge side to field side. The major findings can be summarized as follows:

- (1) The peak contact pressure is potential to increase as the rail inclination increases up to 1/30 and the peak contact pressure gradually increases if the wheel lateral position is moved from gauge side to field side. However, for the 1/20 rail inclination, this posture is completely opposite.
- (2) The contact area tends to decrease if the wheel lateral position is varied from gauge side to field side in most cases. However, this is not valid for the 1/20 rail inclination.
- (3) The Dang Van stress ratios of wheel and rail were influenced by the rail inclination and the wheel lateral position. Its value is potential to

be higher if the rail inclination increases up to 1/30 and the wheel lateral position is on the field side. Therefore, the fatigue crack initiation location is prone to occur earlier on the field side. However, for the 1/20 rail inclination, this posture is completely different. Moreover, the FE results reveal that the maximum Dang Van stress ratio was found below the wheel and rail surfaces, at the depth of 2.45 mm and 5.65 mm, respectively.

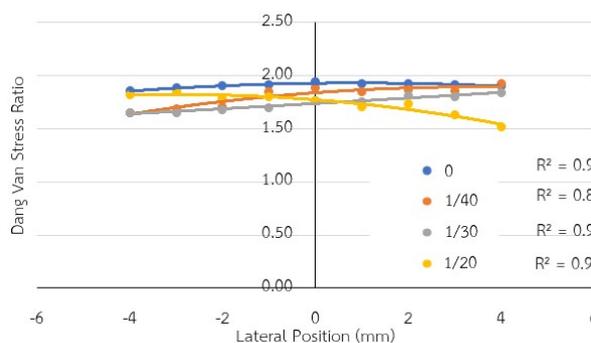


Fig. 9 Dang-Van stress ratio of wheel on various rail inclinations

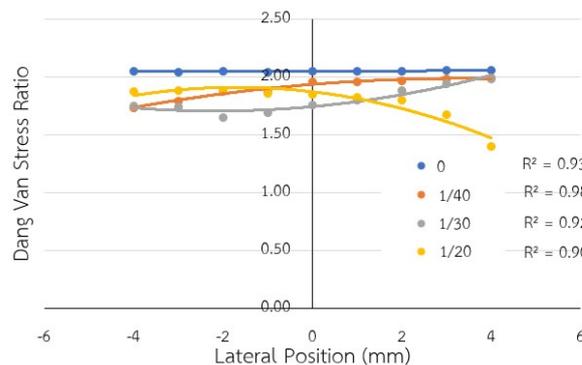


Fig. 10 Dang-Van stress ratio of rail on various rail inclinations

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