Optimization and Design of a Railway Wheel Profile Based on Interval Uncertainty to Reduce Circular Wear

Yongjie Lu,1,2 Yun Yang,2 Jianxi Wang,2,3 and Bowen Zhu4

1State Key Laboratory of Mechanical Behavior and System Safety of Traffic Engineering Structures, Shijiazhuang 050043, China
2Shijiazhuang Tiedao University, Shijiazhuang 050043, China
3Key Laboratory of Road and Railway Engineering Safety Control (Shijiazhuang Tiedao University), Ministry of Education, Shijiazhuang 050043, China
4Taiyuan CRRC Times Rail Engineering Machinery Co., Ltd., Taiyuan, China

Correspondence should be addressed to Jianxi Wang; qianxi-2008@163.com

Received 15 June 2020; Revised 8 September 2020; Accepted 4 October 2020; Published 21 October 2020

Academic Editor: Mitsuhiro Okayasu

Copyright © 2020 Yongjie Lu et al. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

Wheel tread wear is a form of wheel damage that can seriously affect the performance of freight vehicles. A new numerical approach to optimizing wheel profiles can reduce circular wear on the LM wheel in the design cycle. This approach considers the influence of different line conditions and speed fluctuation on wheel wear, along with the performance of the wheel and the rail as the materials wear. In this approach, a nonlinear numerical optimization model for the wheel tread profile is built through a backpropagation (BP) neural network method. The multipoint Kik–Piotrowski (KP) contact mechanics model is applied to calculate the wheel/rail normal force, tangential creep force, the stick-slip area, and the size and shape of the contact patch. The optimal profile is obtained through the genetic algorithm (GA) method. In order to better reflect the random characteristics of wheel/rail matching and interval uncertainty, a random sampling technique is used to generate a random data sample at typical operating speeds.

1. Introduction

On high-speed and heavy-haul railways, wheel/rail wear problems have become serious, especially in terms of circular wheel wear and hollow-tread wear [1]. Severe wheel tread wear causes an increase in wheel flange height and running resistance; it can also affect vehicle stability and ultimately can lead to derailment [2]. Once a wheel is worn or otherwise damaged, it must be reprofiled to recover the standard profile of the wheel flange and tread. Reprofiling is expensive, and the amount of metal removed from the tread surface during the cutting process is often greater than the amount of metal lost to wear [3]. At present, Chinese heavy-haul railways use an LM wheel tread design in conjunction with 75 kg/m rail. However, the LM-type tread is designed to be matched with the 60 kg/m rail profile. The resulting mismatch of the wheel/rail profile is one of the causes of excessive wheel wear [4]. With the large-scale application of 75 kg/m rails on heavy-duty railways, rolling contact fatigue (RCF) between the outer rails and wheel is relatively serious in small radius curves. In order to reduce wheel/rail wear and RCF, extend the wheel service cycle, and reduce maintenance costs, it is necessary to design a wheel tread profile that is more compatible with the 75 kg/m rail profile.

Until the advent of computing technology, the effectiveness of the design and optimization of wheel profiles generally depended on the engineering experience. With the development of the computing technology and numerical optimization theory, numerical simulation has become the primary approach to multiobjective optimization of the wheel profile. It can be effectively used to design wheel profiles that better match the existing wheel/rail profiles, reduce wheel and rail wear, and improve the vehicle curving performance. Based on the nonlinear programming (NP) theory, Haque et al. [5] proposed a variable scale method for wheel profile design. The difference between the ideal
equivalent taper and the equivalent taper calculated in the optimization process, the wheel/rail contact patch, and contact stress are regarded as the constraint condition.

An effective wheel/rail profile design incorporates a tapered wheel tread to generate the rolling radius difference (RRD), which promotes steering in curves and has a significant influence on the dynamic vehicle performance, overall [6]. Shevtsov et al. optimized the wheel profile based on the wheel/rail RRD. The wear index and the lateral displacement were selected as indicators to verify the optimized profile [6]. Markine and Shevtsov [7] and Ignecci et al. [8, 9] identified the RRD function based on wheel/rail contact characteristics, which are also set as an optimization objective for the optimized wheel profile. Based on the geometric contact relationship between the wheel and rail, the wheel/rail contact point distribution and RRD are considered as an optimization objective by Gerlici and Lack [10]. And the iterative algorithm is used to calculate the optimized wheel profile.

The geometry of the wheel tread affects not only wheel/rail contact fatigue and rail damage but also has an impact on operational safety [11, 12]. According to conformal profile design concepts, Spangenberg et al. [13] reported that an optimized wheel/rail profile reduces wear and rolling contact fatigue.

The interval at which wheels are reprofiled plays a significant role in their long-term performance. Jiang et al. used a semi-Markov model to determine wheel repair time based on circumference wear, flange thickness, and repair capability [14].

Evaluation of how effectively wheel and rail profiles are optimized is currently based on wheel and rail wear and the dynamic vehicle performance based on the as-designed profiles, without taking the effects of wear into account, when in reality, the wheel and rail shapes and the actual wheel/rail contact are constantly changing with wear [15]. Wheel profiles exhibit some nonuniform wear after a given service cycle, which changes the contact relationship between the wheel and rail. This causes wheel and rail wear to increase rapidly and requires frequent reprofiling of wheels [16]. Moreover, the typical wheel profile design method uses specific curve parameters and does not take varying operating speeds and line parameters into account.

Therefore, a new approach to wheel profile design that takes normal wear characteristics into account is proposed. In this approach, simulation and analysis are based on actual line and speed-related operating conditions. The typical wheel tread wear rate is factored into the optimal design. The wheel profile is composed of a multisection curve. The parameter of the wheel profile arc is set as an independent variable.

The wheel profile shape satisfies the geometric characteristic, and the vehicles’ dynamic performance satisfies operational safety requirements. These are defined as the constraint condition. The wheel profile optimization model is established by a neural network method, and the optimal profile is calculated using a genetic algorithm (GA) [17, 18].

2. Modelling of Wheel Tread Profile Optimization

2.1. Modelling of Wheel Wear

2.1.1. Modelling of Vehicle-Rail Dynamics. Although freight vehicles run in various groups on heavy-haul lines, research has shown that a grouping of three vehicles can reflect the basic dynamic characteristics of any group of vehicles within a train [19]. So, a three-vehicle grouping model is built for simulation through the UM (Universal Mechanism) software (with vehicle parameters referenced from the literature) [20].

The lumped parameters are used to simulate the rigidity and damping coefficient of the subrail foundation (the track of this model is made of 75 kg/m rail, type III sleepers, type II elastic fasteners, and the thickness of the track bed is 35 cm). A multipoint Kik–Piotrowski (KP) contact mechanics model [21] is applied (with the computing method shown in the literature) [22]. The contact parameters are shown in Table 1.

2.1.2. Calculation of Wear Loss. The Archard wear model is used to calculate wheel wear loss. The expression of this computing model is written in the following equation:

\[ V_{\text{wear}} = K \frac{F_n d}{H}, \]

where \( V_{\text{wear}} \) represents the wear volume (m\(^3\)) of the wheel surface material; \( d \) is the sliding distance (m); \( F_n \) means the normal force (N) of wheel/rail contact surface; \( H \) stands for the wheel material hardness (N/m\(^2\)); and \( K \) represents the wear coefficient.

The total wear depth of wheel \((e)\) is represented by wheel wear depth per unit of mileage. \( \alpha \) and \( \beta \) are used to represent the weights of curve \( W_{\text{left}} \) and speed \( V_{\text{right}} \), respectively. The wear depth per unit of mileage can be expressed by the following equation:

\[ e_k = \sum_{s=1}^{S} \alpha_s \sum_{j=1}^{K} \beta_{ij} \cdot e_k(x), \quad k = l, r, \]

where \( x \) is the lateral coordinate across the wheel profile, and \( e_l \) and \( e_r \) are the wear depth of the left and right wheels of the unit mileage, respectively. The maximum wear depth per unit of mileage \( e_{\text{max}} \) of left and right rail pairs is calculated in the following equation:

\[ e_{\text{max}} = \max_k \{ \max \{ e_{l,i}(x) \}, \max \{ e_{r,i}(x) \} \}. \]

Every increment of wear depth 0.1 mm \((\delta = 0.1 \text{ mm})\) is taken as the profile update condition. When every increment of wear depth reaches 0.1 mm, the line mileage is expressed by the following equation:

\[ S = \frac{\delta}{e_{\text{max}}} (\delta = 0.1 \text{ mm}). \]

The wear loss of wheel section in x-axis direction is given in the following equation:
Table 1: Contact parameters of wheel/rail.

| Parameters             | Values               |
|------------------------|----------------------|
| Rail type              | 75 kg/m              |
| Wheel/rail material density | 7,800 kg/m$^3$          |
| Friction coefficient   | 0.25                 |
| Young’s modulus        | 2.1 × 10$^{11}$ Pa   |
| Poisson’s ratio        | 0.27                 |
| Rail cant              | 1/40                 |
| Number of units in the contact spot | 10             |
| Number of strips in the contact spot | 20           |
| Critical creep speed   | 0.1 m/s              |
| Shear modulus          | 8.077 × 10$^{10}$ Pa |

$E(x) = e_k(x) \cdot S$. (5)

2.1.3. Analysis of Factors Affecting Wheel Wear Loss. Rail surface irregularity is a key factor affecting wheel/rail wear loss. The state of special heavy-haul railway lines in China is similar to the class 5 track spectrum of the United States [23]. So, the U.S. class 5 spectrum is used as random irregular excitation for simulation analysis. Furthermore, railway line parameters, such as curvature, grade, and superelevation, have a direct impact on the wheel/rail contact relationship. In order to better reflect the random characteristics of wheel/rail matching, data are randomly sampled to generate sample vehicle speeds. The weight coefficient is assigned to a wear loss calculation model based on the probable distributions between operations at varying degrees of curvature and speed.

(1) Effect of Different Curves on Wheel Wear. In order to determine the optimal wheel profile in actual line operation, parameters of a heavy-haul railway, including right and left hand curves, curve radius, and superelevation are included. For simulation purposes, curves on the railway lines (49 curves in total) are classified into 12 categories. The percentage of tangent, spirals, and curve body associated with each curve is calculated and shown in Table 2.

The middle vehicle of a three-vehicle set can better reflect the dynamic response of any group of vehicles within a train, and the wear law of the middle vehicle’s guide wheel is more representative. So, the distribution of wear on the wheels of the middle vehicle of a three-vehicle set over 10,000 km of operation is calculated based on the distribution shown in Figure 1. Since wear loss and distribution vary significantly with curvature, actual line conditions must be taken into account.

(2) Effect of Speed on Wheel Wear. The range of operating speeds in curves is calculated based on the curve parameters shown in Table 2 and the speed distribution of vehicles in actual operation on a heavy-haul railway. According to the 3σ rule of normal distribution [24], the speeds for No. 1 and No. 7 curves meet normal distribution (50, 3.332), while other curves meet normal distribution (60, 3.332).

Figure 2 shows the wear distribution of the leading wheel at different velocities for curve No. 3. It indicates that the worn area moves gradually toward the flange as speed increases. The circular wear rate of wheel tread decreases gradually with speed. Hence, it is important to take differences in speed into account when calculating wheel tread wear.

(3) Weight Coefficient Calculation of the Influencing Factor. Random sampling is performed to generate a random sample of operating speeds. The weight coefficients $a_i$ and $b_j$ of curve $W_m$ and speed $V_n$ in equation (2) are assigned to wear loss calculations based on the relationship between curves and velocities. In No. 3 curve, for example, the speed meets normal distribution (60, 3.332). Using the normal distribution of operating speed at the curve, the Latin hypercube sampling method can be applied to calculate the weight coefficients at intervals of 4 km/h [25]. The speed sampling result is shown in Figure 3.

Weighting coefficients account for the probability of the wheel encountering curves of a given radius and certain velocities. Curve No. 1 and No. 7 have the same radius and superelevation and, therefore, have the same probability of occurring at 0.0204 (Table 2). In this case, they also have the same speed profiles. Table 3 shows the weighting coefficients of different velocities for the No. 1 and No. 7 curves.

Tables 4 and 5 show the weight coefficients of other curves and different velocities, respectively. The simulation condition of each sample wheel profile is based on the combinations of 12 curves and 6 velocities (72 operating conditions in total).

2.2. Selection of an Independent Variable Based on Arc Parameter. The initial profile of the LM wheel may have three arcs in contact with the 75 kg/m rail. So, the optimization range is chosen based on these three arcs. The optimization area consists of three arcs between points $A$ and $D$, where the interval length $l = 58.596$ mm, as shown in Figure 4. The LM wheel profile is optimized by adjusting the lengths and radii of arcs $AB$, $BC$, and $CD$.

A coordinate system is established with the nominal rolling circle of the wheel as the origin of the coordinate $O$. Arcs $AB$ and $BC$ are internally tangent with each other at point $B$ ($x_B, y_B$), while arcs $BC$ and $CD$ are internally tangent with each other at point $C(x_C, y_C)$. The fixed points $A$’s coordinates are ($-20.194, 0.886$), and point $D$’s coordinates are ($38.402, -12.987$). The circle center coordinates of these 3-section arcs are $O_i(x_{R_i}, y_{R_i})$, and their radii are $R_i$ ($i = 1, 2, 3$). The equations for these 3-section arcs can be established based on the fact that the arcs composing the wheel profile are convex. The curve $A D$ in the optimization area can be described if $R_2$, $R_3$, $x_B$, and $x_C$ are given. The solving process is given below:

Since the slope at point $D$ is $\tan 110^\circ$, the circle center ($x_{R_3}, y_{R_3}$) of arc $CD$ can be determined by the following equation:

$$\begin{cases}
x_{R_3} = x_D - R_3 \sin 70^\circ, \\
y_{R_3} = y_D - R_3 \cos 70^\circ.
\end{cases}$$ (6)
Since $x_C$ is the horizontal coordinate of point of tangency between arcs $CD$ and $BC$, $y_C$ can be determined using the following equation:

$$y = y_{Ri} + \sqrt{R_i^2 - (x - x_{Ri})^2}, \quad (i = 1, 2, 3). \quad (7)$$

Because arcs $CD$ and $BC$ are tangent to each other at point $C$ while the circle center of arc $BC$ passes through straight line $O_2O_3$, the circle center $(x_{Ri}, y_{Ri})$ of arc $BC$ can be determined. Equation (8) is obtained as

$$\begin{cases} y_C = y_{Ri} + \sqrt{R_i^2 - (x_C - x_{Ri})^2}, \\ y_{Ri} = k_{23}x_{Ri} + b_{23}, \end{cases} \quad (8)$$

where $k_{23} = (y_{C} - y_{Ri}/x_C - x_{Ri})$; $b_{23} = -k_{23}x_{Ri} + y_{Ri}$. A prior condition for the above-noted equations to have solutions is $b^2 - 4ac \geq 0$, where $a = k_{23}^2 + 1$, $b = 2x_C - 2a_kk_{23}$, $c = a_0 + x_C^2 - R_i^2$, and $a_0 = y_C - b_{23}$.

---

**Table 2: Curve parameters analysis of a heavy-haul railway.**

| Track curvature | Radius (m) | Superelevation (mm) | Length of tangent (m) | Length of spiral (m) | Length of curve (m) | Number | Right and left curves | Percentage |
|-----------------|------------|---------------------|-----------------------|----------------------|---------------------|--------|-----------------------|------------|
| 1               | 400        | 85                  | 300                   | 120                  | 145                 | 1      | Left                  | 2.04       |
| 2               | 400        | 110                 | 300                   | 90                   | 162                 | 2      | Left                  | 4.08       |
| 3               | 500        | 90                  | 300                   | 110                  | 219                 | 8      | Left                  | 16.33      |
| 4               | 600        | 75                  | 300                   | 140                  | 223                 | 9      | Left                  | 18.37      |
| 5               | 800        | 55                  | 300                   | 130                  | 250                 | 1      | Left                  | 2.04       |
| 6               | 1000       | 45                  | 300                   | 100                  | 211                 | 2      | Left                  | 4.08       |
| 7               | 400        | 85                  | 300                   | 120                  | 145                 | 1      | Right                 | 2.04       |
| 8               | 400        | 110                 | 300                   | 90                   | 162                 | 1      | Right                 | 2.04       |
| 9               | 500        | 90                  | 300                   | 110                  | 219                 | 15     | Right                 | 30.62      |
| 10              | 600        | 75                  | 300                   | 140                  | 223                 | 3      | Right                 | 6.12       |
| 11              | 800        | 55                  | 300                   | 130                  | 250                 | 4      | Right                 | 8.16       |
| 12              | 1000       | 45                  | 300                   | 100                  | 211                 | 2      | Right                 | 4.08       |

**Table 3: Weight coefficients of different velocities (No. 1 and No. 7 curves).**

| Speed (unit: km/h) | 40 | 44 | 48 | 52 | 56 | 60 |
|--------------------|----|----|----|----|----|----|
| Weight coefficients| 0.02| 0.10| 0.26| 0.35| 0.20| 0.07|

---

**Figure 1:** Distribution of wheel wear under different curve parameters ($x$ is the lateral coordinate across the wheel profile).

**Figure 2:** Distribution of wheel wear at different velocities (No. 3 curve) ($x$ is the lateral coordinate across the wheel profile).

**Figure 3:** Normal distribution of velocities (No. 3 curve).
2.3.1. Design Cycle of Wear. The overall life of the wheel can be extended if the service life is extended through profile optimization at each design cycle. According to statistical analysis [15], wheel tread wear accounted for a high proportion of the overall wheel wear per design cycle. For example, after one design cycle, the tread wear of wheels on K5- and K6-type bogies represented about 31% and 29.4%, respectively, of the allowable 5 mm limit. The circular wear rate of the LM wheel tread was also high in the initial stage [1, 2]. Figure 5 shows the LM wheel with a worn wheel tread.

The literature notes that effectively reprofiling wheels significantly reduces overall maintenance costs [3]. This paper proposes an approach that optimizes the wheel profile during each design cycle, an approach that reduces the amount of metal removal that is required over the life of the wheel. In this paper, we define one cycle to be the time it takes for a wheel to circular wear depth $d = 0.45$ mm at the tapping line. We also call this the design cycle.

2.3.2. Sampling of the Wheel Tread Profile. The selection of representative wheel tread profile samples is performed through Latin hypercube stratified sampling [25]. The process consists of three steps:

1. The optimized profile is realized through reprofiling based on the standard profile; the maximum reprofiling depth of the wheel tread profile is 1 mm.
2. 81 wheel profiles were selected. The profiles that do not meet the constraint condition are removed.
3. The design variables for wheel profile design include $R_3, R_2, x_B$, and $x_C$, which are represented by $s_1, s_2, s_3$, and $s_4$, respectively.

The values used for each factor in the optimized profile design are based on the practical geometrical characteristics of wheel profiles. The proposed values for each factor are listed in Table 6.

2.3.3. Evaluation Method of Profile Optimization. The wheel tread wear rate—specifically, the ratio of the circular wheel tread wear depth to the total mileage within the design wear cycle (in unit of mm/km)—is the parameter used to determine an index for wheel profile optimization. It is the ratio of the circular wheel tread wear depth to the total mileage within the design wear cycle in unit of mm/km. The wear rate $r$ is calculated in the following equation:

\[
\begin{align*}
\text{Wear rate } r &= \frac{\text{Circular wear depth}}{\text{Total mileage}} \\
&= \frac{d}{d_{\text{total}}}
\end{align*}
\]
2.5. Modelling of Wheel Profile Optimization. The independent variable \( s = (s_1, s_2, s_3, s_4) \) and dependent variable \( r \) are determined through the analysis of various aspects of the wheel tread profile optimization model. A series of constraints need to be imposed on the optimization model, including the value range of independent variables and requirements relating to vehicle operation, safety, and stability. Thus, a nonlinear mathematical model with a single-objective, multivariable, and multiconstraint is established in the following equation:

\[
\begin{align*}
\min & \quad r(s), \\
\text{s.t.} & \quad C(s),
\end{align*}
\]

where the upper and lower limits of independent variable are \( s_1 \in [14, 14.2], \quad s_2 \in [100, 110], \quad s_3 \in [9, 10], \) and \( s_4 \in [29, 29.2]. \) The vehicle operation, safety, and stability are not taken into account in the building and solving optimization model, which must be checked to ensure that it complies with operating requirements after the profile is optimized.

The optimization process includes a backpropagation (BP) neural network that incorporates a training algorithm with error backpropagation as a feed-forward network. This algorithm has unique characteristics of nonlinearity, adaptability, and fault tolerance. This is so it can be used to build a nonlinear model between the rail profile and optimization objective [26]. Of 81 sample wheel tread profiles, 69 were chosen randomly as training samples; the remaining 12 samples are used as a test set. The Levenberg–Marquardt (L-M) learning algorithm of the neural network is a combination of the gradient drop method and Gaussian–Newton method, which uses similar second-order derivative information, is insensitive to over-parametrical problems, and the chance of cost functions falling into local minimum values is greatly reduced. In terms of training times and accuracy, the L-M algorithm is significantly superior to the adaptively adjusted learning rate BP algorithm, so the Levenberg–Marquardt (L-M) algorithm is selected when training the neural network. The number of the nodes in the hidden layer is 11 in the backpropagation neural network. The nonlinear model is used to predict the wheel tread wear rate of the test sample and is then compared to the actual wear rate. As shown in Figure 7, the overall relative error of the wear rate is within 5%, which is a clear indication that the wheel tread profile optimization model based on a BP neural network is a good predictor of performance.

### Table 6: Design level of each factor (unit: mm).

| Independent variable | Design level |
|----------------------|--------------|
| \( s_1 (R_1) \)      | 14, 14.1, 14.2 |
| \( s_2 (R_2) \)      | 100, 105, 110 |
| \( s_3 (x_0) \)      | 9, 9.5, 10    |
| \( s_4 (x_c) \)      | 29, 29.1, 29.2 |

\[
\begin{align*}
\sum_{j=1}^{m} d_j
\end{align*}
\]

where \( m \) represents the number of profile updates and \( d_j \) and \( q_j \) represent the wear depth and mileage when the profile is updated, respectively.

Figure 6 shows the wear distribution comparison of the 4 wheels (2 wheelsets) on the first bogie of the middle vehicle in the three-vehicle grouping model. It is apparent that right and left wheels on the leading wheelset have worn at a higher rate than those on the trailing wheelset. Wear on the left wheel is in the range of 5–30 mm, while wear on the right wheel is in the range of −30–−5 mm. There is also a diameter difference of 0.1 mm between left and right wheels. Based on the tread surface of the right leading wheel of the middle vehicle, the design cycle of the wheel profile is defined as 0.45 mm.

The circular wear rate of 81 wheel profile samples was obtained through dynamic simulation and data processing, based on the coefficients of various curves and velocities.
algorithm is used to obtain the LM optimization model shown in Section 2.5. The GA method is employed to solve the wheel profile optimization model [27]. The wear calculation results are listed in Table 7.

As shown in Table 7, the circular wear rate of the wheel tread within the design wear cycle of the wheel profile obtained through GA is lower than that of the sample profile. The simulation values of high rail wear rates corresponding to various profiles: \( r \) (BZ) \( > r \) (optimal sample) \( > r \) (optimal profile). The lowest predicted optimal profile tread wear rate is 62.54% lower than the wear rate of the standard profile, a clear indication that the use of the optimization algorithms can lead to reduced wear.

### 3.1. Wear Analysis of the Optimized Wheel Profile

In order to apply the optimized wheel profile to actual lines, the amount of metal that must be cut from the optimal profile is determined through calculation, as shown in Table 8.

Wear comparisons are performed between an optimized wheel profile and the standard LM profile. The evolution of the optimized profile within the designed wear cycle is shown in Figures 8 and 9.

Figure 9 indicates that tread and flange wear on the optimized wheel profile are within the range \(-17\sim10\) mm. The wear is more uniform and less than that of the standard profile. Within the design wear cycle, circular wear of the optimal profile is 62.54% less than the standard LM profile. When mileage increased to 426.44 million kilometers from 68.151 million kilometers, the tread wear was 0.45 mm, and wear is no longer concentrated at the nominal rolling circle, which helps to reduce hollow-tread wear.

### 3.2. Performance Analysis of the Optimized Wheel

Lateral stability is very important for the safe operation of a train. Once vehicle hunting instability occurs, the relationship between the wheel and rail deteriorates. This condition can lead to derailment. In this paper, the limit cycle method is used to analyze the lateral stability of the vehicle system with an optimized wheel profile. The vehicle system first runs a distance on a random irregular track, and then runs on an ideal smooth track. If lateral oscillation of the wheelset is no longer attenuated with time, but tends to stabilize the limit cycle, the speed at this point is the critical speed of the vehicle system [28]. Through calculation, the vehicle critical speed with the optimized wheel profile is 249 km/h under full loading, while the critical speed using the standard LM profile is 160 km/h. For empty vehicles, the critical speed with the optimized wheel profile is 100 km/h, and the critical speed using the standard LM profile is 83 km/h. Therefore, the vehicles’ critical speed with the optimized wheel profile is higher than that of the standard LM wheel profile, which improves the lateral stability of the vehicle.

Indicators such as the derailment coefficient, wheel load reduction rate, and wheel/rail lateral forces are used to evaluate the dynamic performance [29]. Using the maximum and mean values for left and right wheels and calculating the dynamic simulation at 72 km/h under the same line conditions as previously referenced, the computed results of optimized and standard profiles provide a good indication of the vehicles’ safety and stability (Table 9).

As shown in Table 9, the derailment coefficients, all the dynamic wheel load reduction rates, and all maximum values of wheel/rail lateral force fall within the allowable range of the specification. The table shows that the optimized wheel profile is compliant with safety and stability requirements. In addition, the dynamic performance of the optimal profile is superior to that of the standard profile.

Wheel/rail contact stress has a great influence on wear and contact fatigue damage. In this paper, a three-dimensional elastoplastic wheel/rail contact finite element model is established, and the wheel/rail contact stress of the optimized wheel profile is analyzed. The wheel material is CL60 steel, the elastic modulus is 210 GPa, Poisson’s ratio is 0.3, and the yield strength is 590 MPa. The rail material U78Cr steel has a modulus of elasticity of 210 GPa, Poisson’s ratio of 0.3, and a yield strength of 850 MPa. Wheel and rail materials are both considered elastoplastic materials, using a bilinear hardening model. According to the standard UIC 510-5 and the literature [30], in wheel/rail contact analysis, a vertical force of 187.5 kN, a lateral force is 90 kN, and a longitudinal force is 30 kN are applied. Contact stresses under different vehicle profile conditions are analyzed for conditions in which the wheel pair has no angle of attack. This is shown in Table 10 (where 0 mm is the center-to-center position of the wheel and rail, and the direction of the wheel flange and the rail gauge is positive).

It can be seen from Table 10 that when the lateral displacement is zero, the wheel/rail contact stress of the optimized wheel profile is reduced from 1763 MPa to 804 MPa, a reduction of 54.4%. This shows that the optimized wheel profile can greatly reduce the wheel/rail contact stress, which will help reduce wear and fatigue damage, when the lateral displacement is zero. Under other lateral displacements, the wheel/rail contact stress changes little. Table 10 also indicates that when the lateral displacement is \(-5\) mm, the wheel/rail contact stresses are greater with the optimized profile than with the standard LM profile. The optimized profile will
Table 7: Comparison of wheel profile parameters’ effect on wear.

| Profile number | Standard profile | Optimal sample | Optimal profile |
|----------------|------------------|----------------|-----------------|
| $s_1$          | 14.00            | 14.20          | 14.20           |
| $s_2$          | 100.00           | 110.00         | 110.00          |
| $s_3$          | 9.56             | 10.00          | 9.23            |
| $s_4$          | 29.15            | 29.20          | 29.20           |
| Travelled mileage (km) | Predicted value | Simulation value | 69652 |
|                  |                  |                | 42544           |
|                  |                  |                | 68151           |
| Wear rate (mm/km) | Predicted value | Simulation value | $6.46 \times 10^{-6}$ |
|                  |                  |                | $1.06 \times 10^{-5}$ |
|                  |                  |                | $6.60 \times 10^{-6}$ |
| Operation ratio (%) |                | 60.19          | 62.54           |

Table 8: Cut amount for the optimal profile.

| Horizontal coordinate (mm) | Cut amount (mm) |
|----------------------------|-----------------|
| −20                        | 0.00            |
| −16                        | 0.26            |
| −10                        | 0.50            |
| −5                         | 0.61            |
| 0                          | 0.65            |
| 5                          | 0.62            |
| 9                          | 0.56            |
| 15                         | 0.41            |
| 20                         | 0.32            |
| 26                         | 0.22            |
| 31                         | 0.05            |
| 38                         | 0.00            |

Figure 8: Profile comparison between the LM wheel and optimized wheel.

Figure 9: Wear loss comparison between the LM wheel and optimized wheel (44,000 km mileage).
require further improvement to mitigate contact stresses in this lateral displacement zone in the future.

4. Conclusions

(1) A vehicle/rail dynamics model is built to study the effect of actual railway line conditions and vehicle operating velocities on wheel wear. The stochastic properties of freight vehicles must be taken into account to provide a realistic and reliable basis for optimizing the wheel profile.

(2) A BP neural network is used to develop a nonlinear numerical optimization model for the wheel tread profile. The arc parameters of the wheel are defined as the independent variable, and the circular wear rate of the wheel tread is the dependent variable. The relative error between actual and predicted wheel wear rates is less than 5%, indicating that the proposed wheel profile optimization model has good prediction characteristics.

(3) A GA method is used to enable the circular rate of wheel tread wear to meet the minimum optimized profile. The circular wear rate of the optimal profile is 62.54% less than that of the standard LM profile. This increases the mileage interval from 42,544 km to 69,151 km before circular wear reaches 0.45 mm. It also reduces the diameter difference between left and right wheels from 0.059 mm to 0.037 mm, thereby attenuating wheel tread wear and extending service life.

This paper is based on the optimization of the wheel profile. The wheel-rail is a contact pair, and it should be a comprehensive optimization of the wheel-rail profile. However, there are still many problems in multiobjective comprehensive optimization. This is the future of the research.

Data Availability

The data used to support the findings of this study have not been made available because the data belong to the third party, and the authors did not get the permission to share it from the third party.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Acknowledgments

This work was supported by the National Natural Science Foundation of China (grant nos. 11572207, 11472180, and 11872255), the Natural Science Foundation of Hebei Province (grant nos. A2020210039 and E2020210092), Program for Top 100 Innovative Talents in Colleges and Universities of Hebei Province (III), China (grant no. SLRC2019036), and the Service Center for Experts and Scholars of Hebei Province.

References

[1] X. Feng, “Cause analysis and improvement measures of the wheel wear and wheel flange wear of railway freight vehicle wheel tread,” SME Management and Technology, vol. 7, pp. 86–88, 2016.
[2] Q. Yuan, Railway Freight Vehicle Construction and Maintenance, China Railway Publishing House, Beijing, China, 2008.
[3] X. Sun, “Analysis on the economics of the tread repair of the metro wheel,” Urban Mass Transit Research, vol. 19, no. 5, pp. 91–97, 2016.
[4] H. Zhong, W. Wang, and Q. Liu, “Influence of rail type on the matching relationship of heavy-haul wheel/rails,” Mechanical Design & Manufacture, vol. 4, pp. 61–64, 2014.

[5] I. Haque, D. A. Latimer, and E. H. Law, “Computer-aided wheel profile design for railway vehicles,” ASME Journal of Engineering for Industry, vol. 111, pp. 288–291, 1989.

[6] I. Y. Shevtsov, V. L. Markine, and C. Esveld, “Optimal design of wheel profile for railway vehicles,” Wear, vol. 258, no. 7-8, pp. 1022–1030, 2005.

[7] V. L. Markine and I. Y. Shevtsov, “Optimization of a wheel profile accounting for design robustness,” Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, vol. 225, no. 5, pp. 433–442, 2011.

[8] M. Ignesti, A. Innocenti, L. Marini, E. Meli, and A. Rindi, “Development of a wear model for the wheel profile optimisation on railway vehicles,” Vehicle System Dynamics, vol. 51, no. 9, pp. 1363–1402, 2013.

[9] M. Ignesti, A. Innocenti, L. Marini, E. Meli, and A. Rindi, “A numerical procedure for the wheel profile optimisation on railway vehicles,” Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, vol. 228, no. 2, pp. 206–222, 2014.

[10] J. Gerlci and T. Lack, “Railway wheel and rail head profiles development based on the geometric characteristics shapes,” Wear, vol. 271, no. 1-2, pp. 246–258, 2011.

[11] G. F. M. D. Santos, L. A. S. Lopes, E. J. Kina et al., “The Influence of wheel profile on the safety index,” Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail & Rapid Transit, vol. 1, no. 5, pp. 1–6, 2010.

[12] U. Spangenberg, R. D. Fröhling, and P. S. Els, “Influence of wheel and rail profile shape on the initiation of rolling contact fatigue cracks at high axle loads,” Vehicle System Dynamics, vol. 54, no. 5, pp. 638–652, 2016.

[13] U. Spangenberg, R. D. Fröhling, and P. S. Els, “Long-term wear and rolling contact fatigue behaviour of a conformal wheel profile designed for large radius curves,” Vehicle System Dynamics, vol. 4, pp. 1–20, 2018.

[14] Z. Jiang, D. Banjevic, M. E., and B. Li, “Optimizing the re-profiling policy regarding metropolitan train wheels based on a semi-Markov decision process,” Proceedings of the Institution of Mechanical Engineers, Part O: Journal of Risk and Reliability, vol. 231, no. 5, pp. 495–507, 2017.

[15] D. Zou, Z. Cheng, R. Jiang et al., “Analysis of wheel wear evolution and running performance of railway wagons,” Railway Vehicles, vol. 55, no. 10, pp. 6–9, 2017.

[16] D. Cui, Research on Design Method of High-Speed Wheel Tread, Southwest Jiaotong University, Chengdu, China, 2013.

[17] J. Wang, S. Chen, X. Li, and Y. Wu, “Optimal rail profile design for a curved segment of a heavy haul railway using a response surface approach,” Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, vol. 230, no. 6, pp. 1496–1508, 2016.

[18] I. Persson, R. Nilsson, U. Bäk, M. Lundgren, and S. Iwinski, “Use of a genetic algorithm to improve the rail profile on Stockholm underground,” Vehicle System Dynamics, vol. 48, no. 1, pp. 89–104, 2010.

[19] Z. Ren, Optimization of Asymmetrical Profile of Rail Section in Heavy-Haul Railway, Shijiazhuang Tiedao University, Shijiazhuang, China, 2016.

[20] G. Tian, Research on Dynamics of Heavy-Haul Train System, Southwest Jiaotong University, Chengdu, China, 2009.

[21] J. Piotrowski and W. Kik, “A simplified model of wheel/rail contact mechanics for non-Hertzian problems and its application in rail vehicle dynamic simulations,” Vehicle System Dynamics, vol. 46, no. 1-2, pp. 27–48, 2008.

[22] J. Piotrowski and H. Chollet, “Wheel–rail contact models for vehicle system dynamics including multi-point contact,” Vehicle System Dynamics, vol. 43, no. 6-7, pp. 455–483, 2005.

[23] J. Xue, Wheel/rail Relationship at Shuohuang Heavy-Haul railway, China Railway Publishing House, Beijing, China, 2013.

[24] P. Wu, “Problem 178: how to understand the 3ψ principle of normal distribution,” Mathematical Communication, vol. 12, p. 28, 2009.

[25] M. Stein, “Large sample properties of simulations using Latin hypercube sampling,” Technometrics, vol. 29, no. 2, pp. 143–151, 1987.

[26] H. Zulin, W. Qian, and Li Gu, “Application of improved LM-BP neural network in water quality evaluation,” Water Resources Protection, vol. 30, no. 4, pp. 22–25, 2008.

[27] X. Wang and L. Cao, Genetic Algorithm--Theory, Application and Software Implementation, Xi’an Jiaotong University Press, Xi’an, China, 2004.

[28] J. Lu, J. Zeng, and M. Chi, “Numerical simulation research on judgment of lateral motion stability of the vehicle system,” Railway Vehicle, vol. 49, no. 9, pp. 1–6, 2011.

[29] Ministry of Railways of the People’s Republic of China, GB5599-85 Railway Vehicle Dynamics Performance Evaluation and Test Identification Specification, China Planning Press, Beijing, China, 1985.

[30] J. L. Cuperus and G. Venter, “Numerical simulation and parameterisation of rail-wheel normal contact,” Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, vol. 231, no. 4, pp. 419–430, 2017.