

Study on longitudinal dynamics of 5000t heavy haul train on mountain railway

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Abstract. In order to analyze the feasibility of running 5000t heavy-haul trains on a railway in a mountainous area, a longitudinal dynamic model of heavy-haul trains is established, taking full account of the actual horizontal profile of the railway line in the mountainous area. The results of the model are compared and analyzed with those of China Academy of Railway Sciences through a numerical example, and the results are in good agreement, which verifies the accuracy of the model. Based on this model, the longitudinal dynamic performance of planned running marshalling under the conditions of ramp traction, restricted ramp braking, emergency braking and resistance speed control braking is analyzed. The results show that the longitudinal impulse of the marshalling under each condition has a large margin compared with the safety standard. The study demonstrates the feasibility of 5000t heavy-haul train operation in the mountainous area from the perspective of simulation, and provides a theoretical basis for the safe operation of 5000t railway in the mountainous area.

Keywords: longitudinal dynamics; coupler force; heavy haul train; draft gear; air brake; modelling and simulation

1. Introduction

China has a vast territory, but the mountainous area accounts for a large number of mountain lines, and a large number of mountain lines have been built. Most of these railways are built on mountains and are generally single-track railways. At the same time, there are large slope lines, many sections with small curves, low operating speed, and relatively backward infrastructure, which to a large extent limit the overall transportation capacity of the lines. According to the original transport mode, the transport capacity of these mountain routes is almost saturated, and measures such as model adjustment and route transformation are needed to alleviate the pressure of railway transport supply. Take a single non-electric railway in northwest Shanxi Province as an example, with a total length of 33.3km, 66 curves and a maximum curve radius of 2500m. The minimum curve radius is 300m, the radius of 800m and below 64 curves, including the radius of 300m curve 25, the total length of curves 18.5km, the existing line has 120 slope sections, the maximum slope of 20‰, the maximum slope length of 1200m, the minimum slope length of 100m, the design speed of the line is 40km/h. For a long time, two DF8B locomotives have been used to pull 27C80 trucks in the form of marshalling. In order to improve the capacity of the line, HXN3 diesel locomotive is planned to be added to the marshalling, and 2HXN3 locomotives+54C80 trucks+2DF8B locomotives (hereinafter referred to as C80 (2+2)) 5000t train marshalling will be opened.

With the large increase of train formation, the longitudinal impulse of train increases sharply, which poses a threat to train running safety. In view of the actual engineering demand of 5000t heavy haul train, this paper studies the dynamic performance of 5000t heavy haul train from the perspective of longitudinal dynamics and fully considers the actual line conditions.

2. Modeling and verification of longitudinal dynamics

2.1 Longitudinal dynamics model

The longitudinal dynamics model of heavy-haul train regards each car as a particle and only considers the longitudinal degree of freedom of each car. Through the force analysis of each particle, the corresponding longitudinal dynamics equation is established, and the longitudinal motion state of each particle is finally solved simultaneously^[1]. The stress situation of a certain vehicle is shown in Fig 1, and the differential equation of its longitudinal dynamics is:

$$m_i \ddot{x}_i = F_{ci-1} - F_{ci} - F_{wi} + F_{TEi} - F_{DBi} - F_{Bi} \quad (1)$$

Where i is the position of the vehicle in the train; m_i is the quality of the car i ; \ddot{x}_i is the acceleration of the car i ; F_{ci-1} is the front coupler force of the car i , when $i=0$, $F_{ci-1}=0$; F_{ci} is the rear coupler force of the car i , when $i=n$, $F_{ci}=0$; F_{wi} is the running resistance of car i , including basic running resistance, slope resistance, etc; F_{TEi} and F_{DBi} are the tractive and braking forces of the locomotive, and their values are zero for freight cars; F_{Bi} is the air braking force of car i ; α is the slope of the line profile where car i is located.

An important part of longitudinal dynamics modeling is the modeling of buffer, whose accuracy is directly related to the accuracy of longitudinal dynamics simulation results. At present, the buffer model used by researchers is basically the table-lookup model and the simplified model of inclined wedge-spring^[2-3]. Table lookup method is based on the results of drop hammer test or impact test to fit the impedance characteristic curve of the buffer, in which factors such as coupler clearance, initial pressure and rigid impact can be considered at the same time. The main difference between the table lookup model established by researchers is reflected in the processing method of the transition stage. In this paper, the buffer model is modeled using table lookup method, and the drop weight test curve of MT-2 buffer is shown in Fig 2.

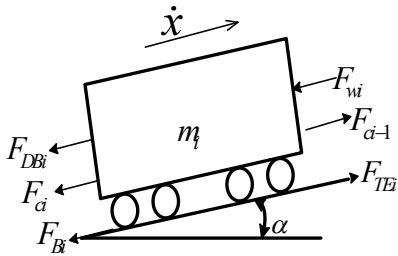


Fig. 1 Force analysis of single car

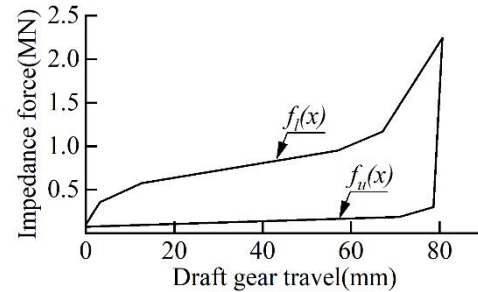


Fig. 2 MT-2 drop weight test curve

In Fig 2, $f_l(x)$ and $f_u(x)$ are the loading and unloading characteristic curves of MT-2 buffer respectively. When the buffer is in the loading stage, interpolation is calculated on $f_l(x)$ through the relative workshop displacement; when the buffer is in the unloading stage, interpolation is calculated on $f_u(x)$. When the buffer is converted between loading and unloading phases, there will be discontinuity points. In this paper, static friction analogy method is used to deal with the transition phase^[4-5], and its mathematical model is shown in Formula 2.

$$F = \begin{cases} f_l(\Delta x) & \Delta v > 0 \\ f_l(\Delta x) + \frac{\Delta v}{ev} (f_l(\Delta x) - f_u(\Delta x)) & -ev \leq \Delta v \leq 0 \\ f_u(\Delta x) & \Delta v < -ev \end{cases} \quad (2)$$

Where Δx is the difference between the displacement of two adjacent vehicles; Δv is the difference in speed between two adjacent cars; ev is the conversion speed;

In this paper, a multi-parameter mathematical model is used to simulate the change characteristics of the air pressure of the brake cylinder^[6], which mainly involves the

propagation speed characteristics of the brake wave, the inflation characteristics of the brake cylinder, the braking conditions and other factors. The change of the air pressure of the brake cylinder is shown in Formula 3 to 5.

$$P_i(t) = \begin{cases} 0 & t < t_{d,i} \\ P_{\max} \left(\frac{t - t_{d,i}}{t_1 + t_{\Delta,i}} \right)^{\frac{\lambda}{t - t_{d,i}} + \lambda} & t_{d,i} \leq t < t_{d,i} + t_1 + t_{\Delta,i} \\ P_{\max} & t \geq t_{d,i} + t_1 + t_{\Delta,i} \end{cases} \quad (3)$$

$$t_{d,i} = t_1 + \frac{t_N - t_1}{(N-1)^\varphi} (i-1)^\varphi \quad (4)$$

$$t_{\Delta,i} = \frac{T_N - T_1}{(N-1)^\kappa} (i-1)^\kappa \quad (5)$$

Where $t_{d,i}$ is the time when the brake cylinder of vehicle i starts to inflate, s; $t_{\Delta,i}$ is the time difference between the braking cylinder inflation of vehicle i and vehicle 1, s; λ is the characteristic parameter of the brake cylinder control valve; P_{\max} is the maximum value of brake cylinder pressure rise, kPa; φ is the characteristic parameter of braking wave propagation speed; κ is the charging characteristic parameter of brake wave; N is the total number of train vehicles; t_N and t_1 are the time when the brake cylinder starts to inflate for the first and N vehicles respectively, s; T_N and T_1 represent the braking cylinder inflation time of the first and N vehicles respectively, s;

2.2 Model Verification

In order to verify the calculation accuracy of the proposed model under braking conditions, the comprehensive test of 10,000 t heavy-duty combined train running on Daqin Line was taken as an example to verify the proposed model^[7-8]. The test condition was that common full braking was carried out on the down slope with a gradient of 12‰, and the initial braking speed was 80km/h. The simulation calculated the maximum coupler force of each coupler under the same conditions, as shown in Fig. 3. The maximum coupler force of both simulation and test appeared near No. 50 coupler, and the maximum was 670kN.

By comparing the results of the comprehensive test of ten thousand t train on Daqin Line with the results of simulation calculation in this paper, the accuracy of the longitudinal dynamic model established in this paper is good, which verifies the accuracy of the longitudinal dynamic model in this paper.

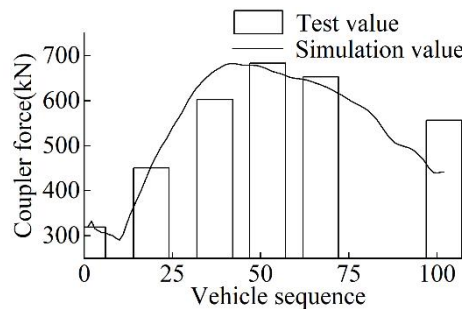


Fig 3. Comparison of maximum coupler force during braking of 10,000 t combination train

3. Calculation working condition and result analysis

In this paper, the factors of railway profile in a mountain area are fully considered, and the simulation of slope starting, emergency braking, general braking and speed regulating braking conditions are carried out respectively. The object of simulation analysis is C80(2+2), and the dynamic mode is pull forward and push back.

3.1 Ramp starting traction condition

The maximum slope of the railway ramp in this mountain area is 20‰, and the longitudinal dynamic performance of the whole train when starting at 16‰, 18‰ and 20‰ uphill slopes is analyzed by simulation. According to the actual operation of the line, the two locomotives at the head start at the same time, and the two locomotives at the tail start with a delay of 3s relative to the head locomotive, and reach the maximum after 32s traction gear. Initial state Each coupler is in free state.

Fig. 4 and Fig. 5 show the distribution diagram of maximum coupler force and maximum longitudinal acceleration of each car respectively, where x axis represents the rear coupler force of each car, y axis represents the maximum coupler force, the retractor force is positive, and the compression force is negative. As can be seen from Fig. 4, the distribution of the maximum coupler force of each car is basically the same under different slope conditions. The maximum retractor force appears in the rear coupler of the second locomotive, and the maximum crimping force appears in the rear coupler of the 54th truck, both of which are lower than the maximum retractor force and the maximum crimping force limit of 2000kN and -2250kN.

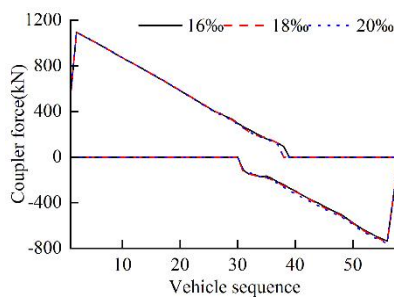


Fig. 4 Maximum coupler force distribution per car

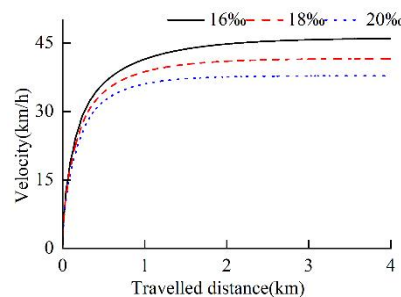


Fig. 5 Train speed changes with mileage

Fig. 5 shows the change of train speed with running mileage, from which it can be seen that the greater the slope of the line, the worse the speed improvement performance of the train in the initial stage of starting, and the smaller the train running speed in the later stage of starting. Through the above analysis, it can be seen that the increase of the slope of the uphill slope has a great effect on the speed raising performance of the train during the starting process, but has no obvious effect on the longitudinal impulse of the train.

3.2 Limit the braking conditions commonly used on ramps

The C80(2+2) uses the 120 brake, the train tube constant pressure is 600kPa, the train tube decompression is 170kPa when the common full brake is implemented, and the brake cylinder pressure is 450kPa. At the same time, the maximum downward slope of the line in the direction of heavy vehicles is -19.6‰, and the normal operating speed of trains on this line is 29~35km/h. Therefore, when analyzing the working conditions of common full braking on restricted slopes, all vehicles are located on slopes with slopes of -16‰, -18‰ and -20‰ respectively, and the trains implement common full braking at an initial speed of 35km/h. According to the actual operation of a mountain railway, when air braking is implemented, the rear locomotive accesses the train air duct, and only the front locomotive sends the air braking instruction.

Fig. 6 and Fig. 7 show the distribution diagram of the maximum coupler force of each car. It can be seen that under the condition of full braking with common gradient, the coupler is subjected to little pull force, the coupler is mainly under pressure, and the maximum pressure hook force all appears in the middle of the train. This is because the air braking command is transmitted from front to back along the train tube through air waves, resulting in delay in braking of the front and rear vehicles. Fig. 7 shows the change of train speed with running mileage. It can be seen that the train speed increases slightly in the early braking stage, because the brake cylinder pressure is still rising in the early braking stage, and the air braking force is not enough to offset the running resistance. In

addition, with the increase of the slope, the speed increases more. It is already very close to the designed speed limit of 40km/h for a mountain railway.

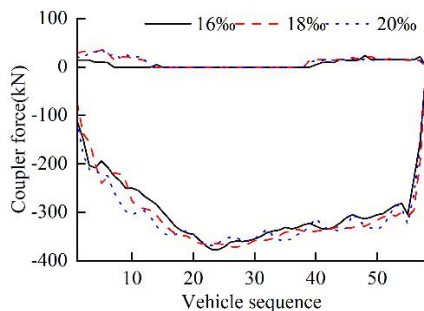


Fig. 6 Maximum coupler force distribution per car

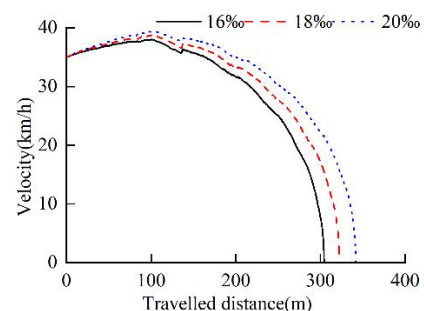


Fig. 7 Train speed changes with mileage

Through the above analysis, it can be seen that under common braking conditions, the increase of downhill slope has little impact on the longitudinal impulse of the train and a greater impact on the braking performance of the train. Therefore, when the train is about to pass through the downhill slope with a large gradient, it is recommended to reduce the speed in advance to avoid overspeed during braking.

3.3 Emergency braking condition

The train tube constant pressure of C80(2+2) is 600kPa, and the maximum brake cylinder pressure of type 120 brake engine is 430kPa when implementing emergency braking under heavy vehicle conditions. According to the relevant provisions in literature, emergency braking of freight trains is implemented at the initial speed of 80km/h on a straight line, and the braking distance shall not exceed 800 meters. The simulation calculated the emergency braking at the initial speed of 80km/h on the straight track. Fig. 8 shows the change of the train running speed with the running mileage. It can be seen that the braking distance of the train running at the initial speed of 80km/h on the straight track is 371.99m, which meets the safety requirement of less than 800m.

Fig. 9 shows the distribution diagram of the maximum coupler force of each parking space in which the train performs emergency braking on the straight lane at an initial speed of 80km/h. It can be seen that the maximum coupler force appears in the rear of the train and the maximum retractor force appears in the head of the train, and the longitudinal impulse of the train is within the safety standard range.

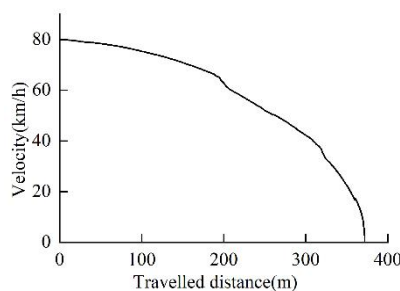


Fig. 8 Changes of train speed with mileage

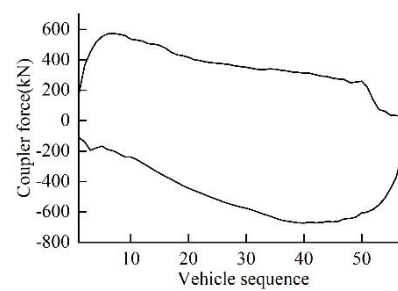


Fig. 9 Distribution of maximum coupler force per car

3.4 Resistance braking speed regulation condition

According to the requirements of relevant operation specifications, the locomotive equipped with power braking device should adopt the method of power braking as the main, air braking as the auxiliary and mutual use when the train speed regulation. There is a continuous downhill in K28+100~K29+900 on the mountain railway. The resistance braking performance of the train when it passes through is studied to analyze whether the train can meet the speed regulation requirement when only resistance braking is used.

At the initial time of calculation, the first locomotive is located at K28+110, the speed is 35km/h, and the time for the braking force to increase from 0 to the rated value is 32s, and then the rated value is maintained until the train is out of this section. Fig. 10 shows the speed mileage variation diagram of the head car when the maximum dynamic braking force is maintained, from which it can be seen that the limit speed of the planned running marshalling can be well increased only through resistance braking. However, if the maximum resistance braking force is maintained through the downhill slope, the train will stop on the downhill slope in advance. Therefore, it is planned to adjust the resistance braking force to maintain the train speed above 30km/h through the downhill ramp. Fig. 11 shows the resistance gear position and the variation of head car speed with running mileage for maintaining a high speed through the downslope. It can be seen that the resistance braking speed regulation performance of the planned running group on the downslope is sufficient.

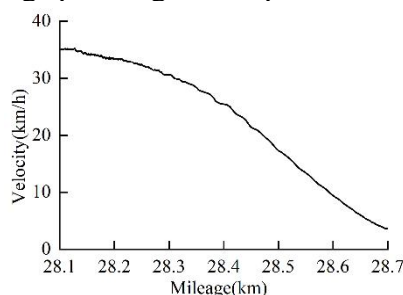


Fig. 10 Speed changes with mileage

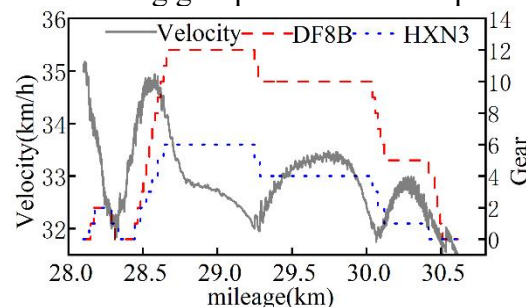


Fig. 11 Speed changes with mileage

4 Summary

(1) By comparing the simulation results in this paper with the comprehensive test results of the 10,000-ton heavy load combined train conducted by China Academy of Railway Sciences, the simulation results of the longitudinal dynamics model established in this paper are in good agreement with the test results, indicating that the model established in this paper can calculate the longitudinal dynamics performance of the heavy load train more accurately.

(2) In the case of fully considering the profile factors of a mountain railway, the simulation calculation of four working conditions of traction ramp starting, emergency braking, common full braking and speed control braking was carried out respectively. The results show that the couplings force of C80(2+2) has a large margin compared with the safety standard. The change of slope slope has little effect on the longitudinal impulse of the train, but has great effect on the acceleration/braking performance of the train. The emergency braking performance and resistance braking speed regulation of C80 (2+2) running on this line can well meet the standard specifications.

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