Implications of brake pipe characteristics for in-train forces of heavy haul train

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Abstract

The longitudinal impact is a prominent problem for heavy haul trains. It can cause coupler fracture and parts fatigue. The brake cylinder action asynchronous on the cars in a heavy train is the main reason of the impact. It is very important to conduct research on the influence of the air brake system parameters on the longitudinal impact for heavy haul trains. An integrated air brake and longitudinal dynamics simulation system is used to quantitatively analyze the influence of air brake pipe, hose and connectors on longitudinal impact. The simulation results show that the brake pipe wall friction coefficient has an obvious effect on the speed of brake pipe pressure reduction on the train’s trailing vehicles, thus affecting the brake cylinder pressure rising speed and train’s longitudinal impact. The effect of the hoses and connectors is not as strong as to that of the brake pipe wall friction. The greater the resistance (produced by pipe wall friction or hose or angle cock connector etc.), the greater the brake cylinder pressure difference between the first and last vehicle in the train, the larger the longitudinal impact. Some guidelines and principles to follow in designing air brake system for heavy haul trains are proposed.

Keywords: Brake pipe; Heavy haul train; In-train forces

1. Introduction

The air brake system, the most widely used brake system in railway freight cars and relies on air to transmit brake signal and provide braking power. However as air transmits at a limited speed, it takes some time before brake propagation from the head end goes to the rear of train. This time lag will cause excessive longitudinal impulse, and thus lead to coupler fracture and fatigue damage of coupler related components. Such incidents have been witnessed on the Chinese dedicated heavy haul line (Daqin Line) on many occasions, which makes the damage of coupler and its related components one of current research topics for heavy haul trains.

In design of brake system for heavy haul train, the first idea is to improve the brake propagation speed. However as brake propagation speed is subject to air propagation velocity and sensitivity of the distribution valve, it is difficult to further increase. The reference [1] presents a new concept in design of distribution valve for heavy haul train, namely using the same brake propagation velocity but improving consistency of the brake cylinder pressure increase speed between cars in the front of train and at the rear of train. Such design has produced good results. The reference [2] makes new suggestions on how to reduce longitudinal impulse for heavy haul train by analyzing the influence of brake pipe geometry on performance of brake system and train longitudinal force. Reference [3] compared the effect of the train with difference brake cylinder charging speed on coupler force. There are few papers on research relating the brake system performance and heavy haul train’s coupler force.

The train air brake system is composed of pipes, connections, distribution valves and cylinders and reservoirs. For Chinese railways with more than 700,000 freight cars, any new design involving replacement of pipes or distribution valves has to be compatible with existing system. To maintain compatibility this paper only studied some parameters in analysis of their effects on train longitudinal impulse. These parameters are friction coefficient of train pipe inner wall, hose coupling resistance. The purpose of study is to provide fundamental principles for design of brake system for heavy haul trains.

In China most heavy haul trains are configured as 10,000 tons. Even trains of 30,000 tons or 20,000 tons are marshalled in 3 or 2 units of 10,000 tons. Therefore this paper only takes the 10,000-ton train into consideration. As service application with the maximum pressure reduction and emergency application can produce the largest coupler force, calculations are made in these two applications.

2. Train Longitudinal Dynamics Model

A train consisting in locomotives and cars connected by coupler and draft gear is shown in Fig. 1.
Each vehicle has following force equilibrium equation:

\[ m \ddot{x}_i(t) = F_{a,i}(t) - F_{g,i}(t) - F_{v,i}(t) - F_{w,i}(t) + F_{g,i}(t) - F_{v,i}(t) - F_{w,i}(t) \]  

(1)

For \( n \) vehicles in the train, there are \( n \) equations, For \( i = 1, F_{G1}(t) = 0; \ i = n, F_{Gn+1}(t) = 0 \)

Coupler force is calculated according to relative speed and displacement between two adjacent cars in following equation.

\[ F_{Gi}(t) = f(v_{i,j} - v_{i-1,j}, s_{i,j} - s_{i-1,j}) \]  

(2)

Accuracy of simulation results depend many factors among of which the major ones are draft gear features and brake system performance. In the work the draft gear features are obtained from actual impacting test of single car to other car. By assumption of draft gear feature relationship and use of test results, parameters of draft gear features are calculated. The draft gear feature relationship is assumed as follows:

\[ F_{Gi}(t) = K \Delta s_{i,j}(t) + C \Delta v_{i,j}(t) \]  

(3)

\[ \Delta s_{i,j}(t) = s_{i,j} - s_{i,j-1} \]  

(4)

\[ \Delta v_{i,j}(t) = v_{i,j} - v_{i,j-1} \]  

(5)

In numeric test it is found that \( K \) is not linear function of relative displacement, and \( C \) is not linear function of relative speed. For the typical draft gear in the test, at impacting speed of 3 and 5km/h, the relationship between coupler force and draft gear deflection is shown in Fig. 3.

Air brake force is calculated by air brake system simulation which based on air flow theory. The basic principle is to calculate real-time gas flow in the brake system so as to achieve air brake system features. The simulation principle and accuracy of calculation results for air brake system are demonstrated in the reference [4].

Based on the geometry of typical pipes, the air flows in the pipes are assumed to move in one dimension. At the same time, given that there may be heat exchange, friction from the pipe walls, variable cross-sections of piping, and changing air flow entropy, the following equations are developed on the basis of the principles of continuity of air flows, conservation of momentum, and conservation of energy. The equations for air flow in pipe as \( (6), (7) \).

\[ \frac{\partial p}{\partial t} + \frac{\partial u}{\partial x} + u \frac{\partial p}{\partial x} + \rho F \frac{\partial u}{\partial x} = 0 \]  

(6)

\[ \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + \frac{1}{\rho} \frac{\partial p}{\partial x} + G = 0 \]  

\[ \frac{\partial p}{\partial t} + \frac{\partial u}{\partial x} \left( a^2 \frac{\partial p}{\partial x} + u \frac{\partial p}{\partial x} \right) - (k-1) \rho (q + uG) = 0 \]  

(7)

\[ G = \frac{4}{2} \frac{u^2}{|D|} \]

The equations are partial differential equations. They can be converted into ordinary differential equations by the method of characteristics and solved.

For the boundary, such as closed-end boundary condition, partial open-end boundary condition, branch pipes boundary condition etc. are described in [5]. A vehicle 120 control valve (popular control valve in China) model is show in Fig. 4.
In the model, there are two parts of brake pipe, one branch pipe, 3 connect pipes between valve and reservoir (or cylinder, chamber), and 6 chambers (or reservoir, cylinder). \( \varphi_1 \sim \varphi_2 \) are orifices areas of air flow which is dependent on the state of valve’s movable parts. In the simulation of air brake system chambers instantaneous pressure are calculated at any time, and brake cylinder instantaneous pressure was used to calculate brake force according to the (8), (9). Then apply the brake force to longitudinal dynamics simulation system. The brake force of one brake shoe \( F_{b1}(t) \) and total brake force of one car \( F_B(t) \) are calculated in following formula:

\[
F_{b1}(t) = \frac{\pi r^2 p_{a1}(t) n_{12} \gamma n_2}{n_k \times 10^6} \varphi_{hi}
\]

\[F_B(t) = n_k F_{b1}(t)\]  

(8)  

(9)

Examples for train air brake system and longitudinal dynamic principles and calculation validity based on the above theories is given in reference paper [6] and [7].

3. Effect of Brake System Parameters on Train Longitudinal Impulse

3.1. Effect of Friction Coefficient of Train Pipe

When air flows in the pipes, it encounters resistance from pipe inner surface. Here such flow resistance is described in “friction coefficient”. This resistance exists in brake pipe, branch pipes, connecting pipes between distribution valve and each chamber (cylinder). Two comparative scenarios are considered: the common pipe with resistance, and the pipes without resistance (non-friction pipe). Figs. 5 and 6 showed the maximum coupler force distributions along the train length for common pipe and non-friction pipe respectively in emergency application and service application.

It can be seen from the figures that if pipe has smaller friction coefficient, the maximum coupler force occurred is smaller. Compared with common pipe, the non-friction pipe will reduce the maximum coupler force by 5.8% in emergency application and 21% in service application for the 10000-ton train. It seems that pipe friction coefficient has a greater effect on the service application than the emergency application in terms of reduction of train longitudinal impulse.

To identify factors in changing coupler force, brake cylinder pressures are also analyzed. Figs. 7 and 8 are brake cylinder pressure for the first car and the last car respectively in emergency application and service application.

Fig. 7 shows that the first car brake cylinder pressure of common pipe are basically overlapped with that for non-friction pipe. It means the charging times and speeds for the first car cylinder are same for common pipe and non-friction pipe. However for the last car brake cylinder pressure for common pipe are far different from that for non-friction pipe. For non-friction pipe, time for brake cylinders of the last car to produce effective pressure is 7.60 seconds, while for common pipe, time is 8.85 seconds. The 14.1% earlier in brake application for non-friction pipe can be understood as larger brake propagation velocity, which can be translated into less train longitudinal impulse.

In service application for common pipe, the brake cylinder of the first car began charging a little earlier. Brake propagation for non-friction pipe is a little quicker than that for common pipe, but such difference in service application is not as big as in emergency application. However brake cylinder pressure curves in service application showed that brake cylinder charging time difference between first and last car was quite different. For non-friction pipe, time difference for cylinder charging to 95% of pressure between the front car and the last car is 7.0 seconds, while for common pipe, the time difference is 32.7 seconds. The reason for such large time difference can be found from train pipe pressure curves shown in Fig. 9. For the first car train pipe pressure curve for common pipe is basically at the same level with that for non-friction pipe, except that curve for non-friction pipe lags little behind. However for the last car train pipe pressure curve for common pipe is quite different from that for non-friction pipe. For common pipe time for train pipe to achieve 95% of final reduction is 73.0 seconds, while for non-friction pipe time is 44.9 seconds. It thus can be concluded that for common train pipe, pressure was reduced at a slower speed for rear cars, especially the last car. Such slower reduction limits the response of the brake cylinder in the rear cars, causing different behaviours and larger longitudinal impulse. In short, pipe with smaller friction can reduce train longitudinal impulse.

Non-friction pipe is of course an extreme condition and not attainable in real world. But pipe with smaller friction can be attainable by for example use of stainless steel pipe instead of carbon steel pipe or by putting copper lining inside carbon steel pipe. To ascertain effect of smaller friction on train longitudinal impulse, another calculation is made when friction
The coefficient of pipe is reduced by half. The calculation results showed that the maximum coupler force was reduced by 3.0% at emergency application and by 14.8% at service application. Therefore in design of brake system, pipe with smooth surface and a dust collector with smallest resistance should be preferred.

3.2 Effect of Hose Coupling Resistance

The hose coupling is connection between train pipes of cars. The hose coupling resistance referred to in this paper are resistance from angle cock, hose and hose coupling modelled in hose coupling resistance model. The basic resistance of hose coupling currently used in China is determined through test results. Calculation and comparison are made on train longitudinal impulse for the original hose coupling resistance and half of original hose coupling resistance. The calculation results showed that in the emergency application the reduction of hose coupling resistance barely changed coupler force, but in service application the reduction caused decrease of coupler force by 1%. However as diameter of hose coupling currently used in China was not so much different from that of original pipework, such reduction of resistance had little effect on coupler force.

![Figure 5. Effect of Friction of Pipe on the Coupler Force in Emergency Application](image1)

![Figure 6. Effect of Friction of Pipe on the Coupler Force in Service Application](image2)

![Figure 7. Effect of Friction of Pipe on the Coupler Force in Service Application](image3)

![Figure 8. Effect of Pipe Friction on Brake Cylinder Pressure in Service Application](image4)
4. Conclusion

To address the outstanding excessive train longitudinal impulse, new requirements are suggested for design of brake system for heavy haul trains. Train air brake and longitudinal dynamics integrated simulation system was used to analyse effect of brake system parameters on longitudinal impulse and following conclusions are made:

1) Brake pipe resistance has a significant effect on train longitudinal impulse. The greater the resistance is, the larger the longitudinal coupler force is.
2) Hose coupling resistance has a little effect on train longitudinal impulse.
3) For designing the brake system of heavy haul train, pipes with lower friction coefficients are suggested.

**NOTATION**

\( a \) : Speed of sound;

\( C \) : Draft gear damping;

\( D \) : Diameter of pipe;

\( f \) : Friction coefficient of pipe inner wall

\( F \) : Area of cross section of pipe;

\( F_{G_i}(t) \) : Coupler force for \( i^{th} \) vehicle;

\( F_{A_i}(t) \) : running resistance for \( i^{th} \) vehicle;

\( F_{B_i}(t) \) : Air brake force for \( i^{th} \) vehicle;

\( F_{L_i}(t) \) : Traction force or dynamic brake force for \( i^{th} \) vehicle, traction force is positive and brake force is negative;

\( F_{W_i}(t) \) : Grade resistance for \( i^{th} \) vehicle;

\( F_{C_i}(t) \) : curve resistance for \( i^{th} \) vehicle;

\( k \) : Ratio of specific heats;

\( K \) : Draft gear stiffness;

\( m_i \cdot x_i(t) \) : Inertial force for \( i^{th} \) vehicle;

\( n \) : Total number of locomotive and vehicles in the train;

\( n_k \) : Number of brake shoes for one car.
$n_z$ : Number of brake cylinders;
$p$ : Pressure of the compressed air;
$p_{zi}(t)$ : brake cylinder instantaneous pressure;
$q$ : Heat transfer rate per unit time;
$r$ : Brake cylinder radius;
$s_{i,j}$ : Displacement of the $i^{th}$ vehicle at $t$ time;
$t$ : Time;
$u$ : Velocity of the compressed air;
$v_{i,j}$ : Speed of the $i^{th}$ vehicle at $t$ time;
$x$ : Distance;
$\rho$ : Density of compressed air;
$\eta_z$ : Transmission efficiency of brake rigging calculated;
$\gamma_z$ : Brake leverage magnification;
$\phi_{ki}$ : Friction coefficient of brake shoes;

References


