Dynamic analysis of a new double deck passenger vehicle with bogie PW200

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Abstract

This paper dealt with the dynamic behavior of a new double deck passenger vehicle with bogie PW200, which was designed and produced by Puzhen Rolling Stock Works of China to meet the demands of high speed railway passenger transportation in China. A full vehicle model including all nonlinear factors was built in the mechanical software ADAMS/Rail. The results of simulation were compared with the measurement data of field test and another simulation results using random equivalent linearization method in frequency domain, to verify the model and determine the track irregularity level. Finally the major bogie suspension parameters were optimized, and the improvement of curve dynamic behavior was obtained using optimum parameters.

Key words

Dynamic analysis, simulation, ADAMS/Rail, high speed, railway vehicle

1. Introduction

Since the first quasi-high speed rail line, from Guangzhou to Shenzhen in China, came to use in 1994, the maximum travel speed of passenger train has reached to 160 km/h, and the travel time has decreased to 75 minutes from Guangzhou to Shenzhen. All this made the railway passenger transportation more compatible than ever in this region. In 1995 the passenger train's speed in main rail lines of China was increased to 140 km/h, to meet the increasing demands of railway passenger transportation. Three different new types of passenger vehicles were designed and produced by different rolling stock works last year in order to increase the passenger train speed to 200 km/h. A new double deck passenger vehicle with bogie PW200 was produced by Puzhen Rolling Stock Works.

The field dynamic test including these three passenger vehicles was finished last August, the maximum test speed reached to 240 km/h. All safety and comfort index were good while this vehicle run on the straight track, but the lateral ride comfort index in low floor and the wheel set lateral force in curve were somehow to big, and lateral comfort index of carbody in curve was greater than 2.5 while test speed exceeded 210 km/h

As all known, The good dynamic behavior of the vehicle on both straight and curve track could be obtained by designing the suspension system carefully, but the bogie suspension parameters have almost conflict effects on straight hunting stability and curve passing behavior. For the high speed bogie, we usually put more attention to the hunting stability and safety factors, this make the wheel set lateral force and carbody acceleration tendentiously big in curve track. With the development of computer simulation technology, now we can simulate the vehicle dynamic behavior and optimize suspension system early in design period

, or make some improvement of existing vehicle through parameter optimization.

In this paper a full vehicle model of double deck passenger car with bogie PW200 including all non-linear factors was built in the mechanical software ADAMS/Rail , Firstly the major work of model setup was to parameterize suspension systems, and to set up special nonlinear wheel-rail contact geometry of Chinese HLM profile wheel with R60 profile rail and other track parameters.

Secondary straight comfort analysis was finished. Vertical, lateral acceleration and comfort index of carbody were compared with the results of field test, so as to verify the model, and to determine the track irregularity level in simulation. At the same time, these results also compared with simulation results finished by author previously using random equivalent linearization method in frequency domain.

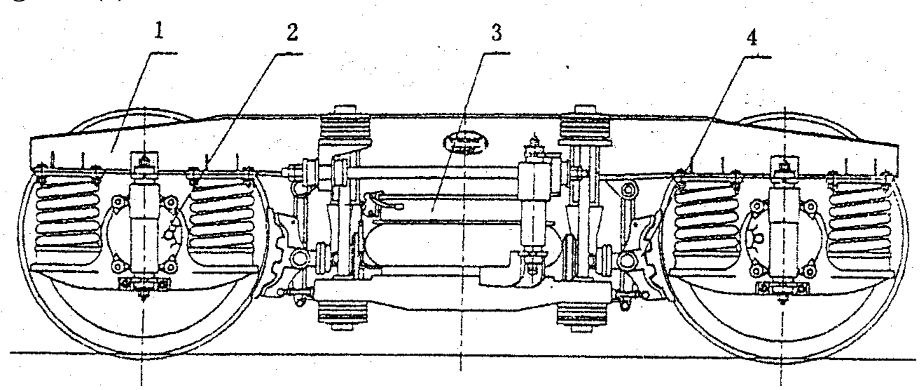
Thirdly hunting stability was studied with parameter variation. The suitable value of primary guide longitudinal stiffness and link bar stiffness between bogie and frame were chosen to give the sufficient high critical hunting speed. The next step was to study straight response and further optimize secondary suspension parameters include vertical and lateral damper coefficient. The dynamic curve behavior including comfort and safety index on both ideal track and real track with irregularities were simulated. Consequently the major bogie suspension parameters affecting curve behavior were optimized.

The final dynamic simulation indicated that the improvement of curve passing behavior was obtained using optimum parameters.

2. Vehicle and bogie structure and model setting up

2.1 Bogie structure

The bogie PW200, see Figure 1, has primary and secondary suspension systems. There are two vertical helical springs and two elastic rubber guide cylinders inside of helical springs on every axle box in primary suspension. Only helical springs endure static empty weight of vehicle, both helical springs and rubber cylinders endure the dynamic force in loaded and running case. The wheel set connects elastically to bogie frame through elastic guide cylinders. The guide cylinders can supply different longitudinal and lateral wheel set guiding stiffness. There is an vertical hydraulic damper in every axle box. The wheel set guide device shows in figure 2(a) ,



1: Bogie frame 2: Wheel set and primary suspension 3: Bolster and secondary suspension 4: Braking system

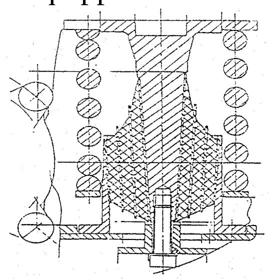
Figure 1 Bogie PW200

The secondary suspension includes bolster, airspring, vertical and lateral hydraulic dampers and lateral rubber bumpstop. The bolster connects with bogie frame through longitudinal link bar at its ends. The whole bolster swing link arrangement hangs to frame through bolster link. At each end of bolster link, there is a elastic rubber joint ,see figure 2(b), it can decrease the secondary lateral stiffness, the friction force and wear of relative revolve joins.

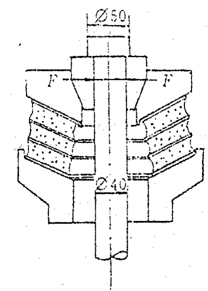
In vertical direction, the carbody is supported on the bolster through side bearing with low friction coefficient. In longitudinal direction, it connects with bogie through center tracing pivot.

An anti-roll bar is installed between bogie frame and carbody to decease carbody roll angle in curve track.

The bogie is equipped with both conventional block braking system and disk braking system.



(a) Wheel set elastic guide device



(b) Elastic linker rubber joint

Figure 2 Wheel set elastic guide device(a) and elastic link rubber joint (b)

2.2 Mathematics model setting up

2.2.1 Vehicle model

The full vehicle mathematics model included one carbody, two bogies frames, two bolsters, four wheel sets and eight axle boxes, all these parts were assumed as rigid bodies. Because the carbody was a double deck, it had a very strong structure, FEM analysis and mode test analysis indicated previously that the first vertical, lateral bend and torsional mode frequencies were more than 15 Hz, so this assumption could be acceptable for the comfort and stability analysis. Table 1 showed the basic data of vehicle and bogie.

Vehicle basic data Table 1

Wheelset base	2.4	m
Wheel diameter (new)	915	mm
Bogie base	18	m
Carbody mass	39000	kg
Height of center of gravity	2.346	m
Bogie mass	6900	kg

The primary elastic rubber guide cylinder was modeled as bushing element, with linear three direction stiffness and little structure damping. The vertical helical spring was modeled as linear general force vector , to give very exact translational and rotational characteristics.

The secondary suspension model was little complex, the swing bolster arrangement was in series with airspring. Because the whole lateral stiffness of bogie had been measured before field test, so the secondary vertical ,lateral and yaw connections between bolster and bogie frame were modeled as general force vector. In longitudinal direction, the link bar between bolster and frame was modeled as force vector.

The side bearing was modeled as nonlinear single force complement. The magnitude and direction of output force depended on the relative velocity of carbody and bolster at its location. Lateral bumpstop was modeled as nonlinear single force complement, its output force was depend on the relative lateral displacement between carbody and bolster in its location. Anti-roll bar was modeled as simple linear torsion spring.

The full vehicle model in ADAMS/Rail was shown in figure 3.

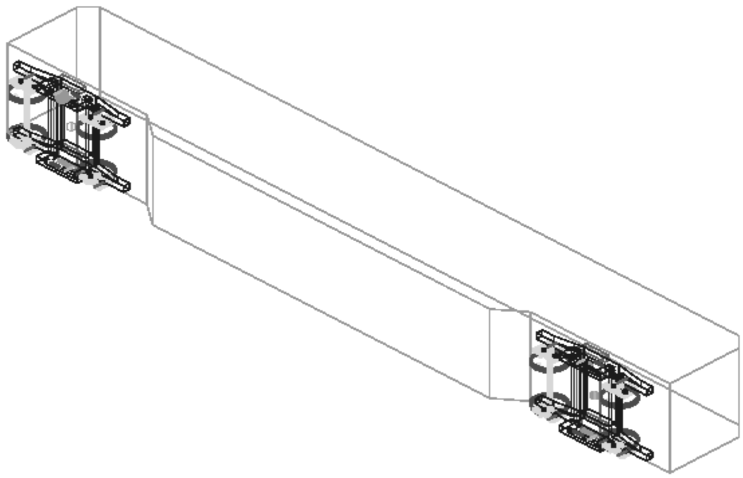


Figure 3 Full vehicle model in ADAMS/Rail

2.2.2 Wheel-rail contact model

As description previously, The wheel has the HLM profile, the normal rolling diameter is 915mm. The rail has the R60 profile. The rail inclination is 1:40 in China. The wheel-rail contact model is shown in figure 4. The new wheel and rail profile were used to calculate the contact geometry and contact table, the linear contact model level 2a in ADAMS/Rail was used in stability parameter study, the non-linear contact model level 3 was used in dynamic stability, straight and curve dynamic response studies.

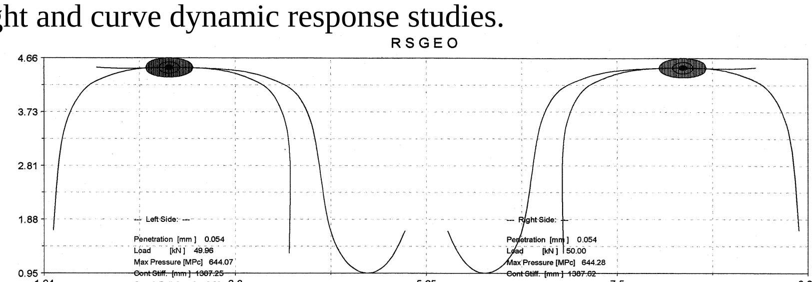


Figure 4 Wheel rail contact profile: HLM/R60

2.2.3 Track model

To determine track irregularities is always difficult because there are many factors affecting the track geometry and statues. The following formulas are power spectral densities(PSD) of track irregularities suggested in China .

Alignment irregularity:

$$S_A = A_A \cdot \frac{\Omega_C^2}{(\Omega^2 + \Omega_R^2) \cdot (\Omega^2 + \Omega_C^2)} \tag{1}$$

Cross level irregularity:

$$S_C = A_V \cdot \frac{\Omega_C^2 \cdot \Omega^2}{(\Omega^2 + \Omega_C^2)\cdot(\Omega^2 + \Omega_S^2)\cdot(\Omega^2 + \Omega_R^2)} \qquad (2)$$

Profile irregularity:

$$S_V = A_V \cdot \frac{\Omega_C^2}{(\Omega^2 + \Omega_R^2)\cdot(\Omega^2 + \Omega_C^2)} \qquad (3)$$

Where: Ω_C=0.8246, Ω_R=0.0206, Ω_S=0.4380 rad/m

With low level perturbation: A_A=2.119x10^{-7}, A_V=4.032x10^{-7} m.rad

With high level perturbation: A_A=6.125x10^{-7}, A_V=10.8x10^{-7} m.rad

Because we can not get the real measurement data of test track irregularities, all the simulation are finished using IVPSD function in ADAMS/Rail, These PSD spectrum could be transferred to track exciting input in time domain. The final track irregularity level used in model would depend on the comparison and agreement between dynamic simulation and field test.

3. Model verify and comparison with field test

Firstly the quasistatic curve model was used to verify the model, the simulation results showed that the carbody accelerations corresponded correctly to the theoretic ones given by speed, curve radius and cant.

Then the original bogie and vehicle parameters in field test were used to simulate the straight response of vehicle. The calculated lateral and vertical acceleration and ride comfort index on the floor of carbody above bogie were compared with test data correspondingly in figure 5 and figure 6.

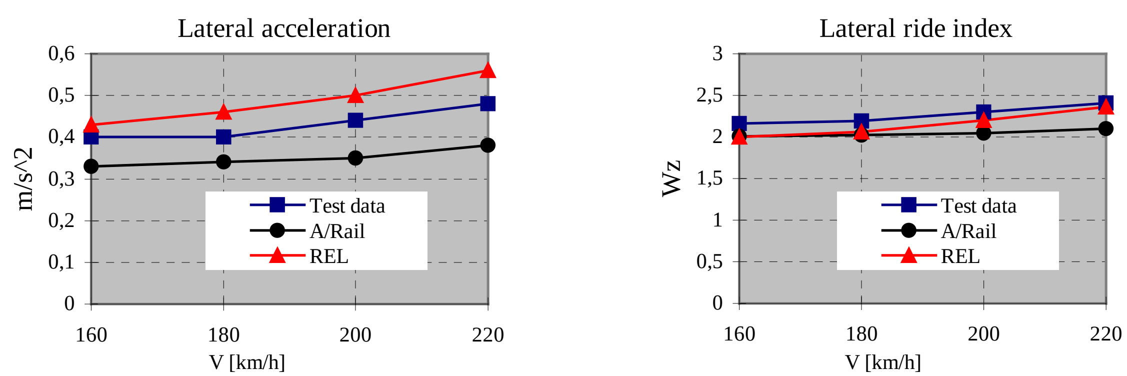


Figure 5 Lateral acceleration and ride comfort index comparison

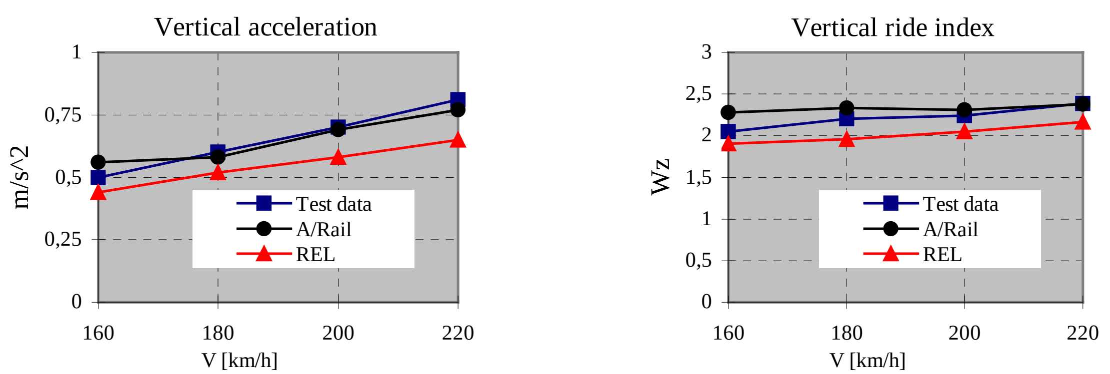


Figure 6 Vertical acceleration and ride comfort index comparison

At the same time, these results were also compared with dynamic simulation finished by author previously using random equivalent linearization method(REL) in frequency domain, the REL values of lateral and vertical acceleration are the three times of standard deviation(3 σ) in figure 5 and 6. All simulations were finished under the low track irregularity level from 160 to 220 km/h.

Based on these analysis and comparisons, the acceptable choose of track irregularity is low level. Through this could affect the simulation accuracy, it is useful to optimize the suspension parameters and foresee the improvement of dynamic behavior using different parameters.

4. Dynamic simulation and parameter study

4.1 Stability analysis and parameter study

The first step is to study linear stability using linear wheel-rail contact model, to get the general idea how suspension parameters affect on the vehicle hunting stability.

Figure 7 and 8 show how the critical hunting speed of the vehicle changes with wheel set guide longitudinal stiffness K_{PX} and lateral stiffness K_{PY}.

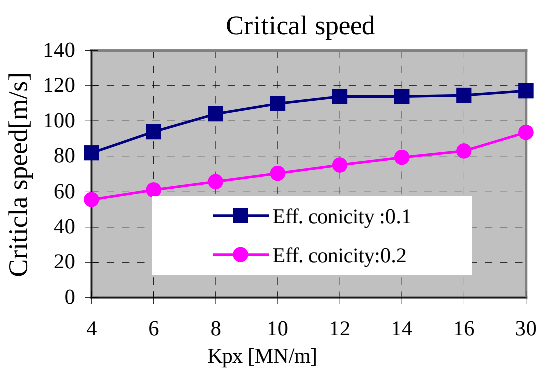


Figure 7 Linear critical speed & K_{PX}

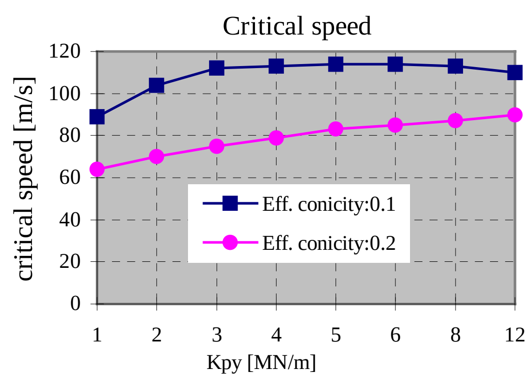


Figure 8 Linear critical speed & K_{PY}

In order to obtain sufficient high stability , we chose the wheel set guide longitudinal stiffness K_{PX}>12 MN/m, lateral stiffness K_{PY}>4 MN/m.

To determine the effect of the nonlinear side bearing friction coefficient μ_{SB} in series with secondary link bar stiffness K_{SX}, we use nonlinear dynamic stability model, and nonlinear wheel-rail contact profile new HLM/R60, the vehicle excited by special track model. The final critical hunting speed of full vehicle is about 350 km/h, this is sufficient for this vehicle with the maximum running speed 200 km/h.

4.2 Straight response analysis and parameter study

The straight response analysis is finished to optimize secondary suspension parameters, such as secondary suspension lateral stiffness K_{SY}, lateral and vertical damper coefficient C_{SY} and C_{SZ}, side bearing friction coefficient μ_{SB}. Figure 9 to figure 12 show how these parameters affect on the vehicle ride comfort. All the simulation speed is 60 m/s, the accelerations which are used to calculate lateral or vertical ride index are measured on the middle floor of carbody above lead and trail bogie.

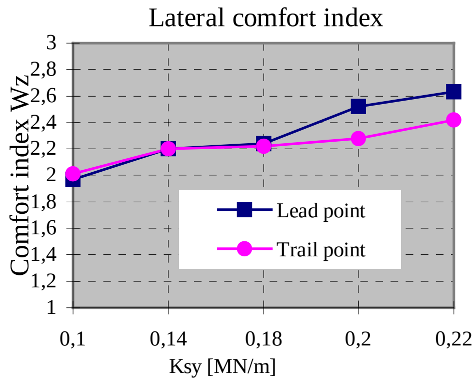


Figure 9 Lateral comfort index & K_{SY}

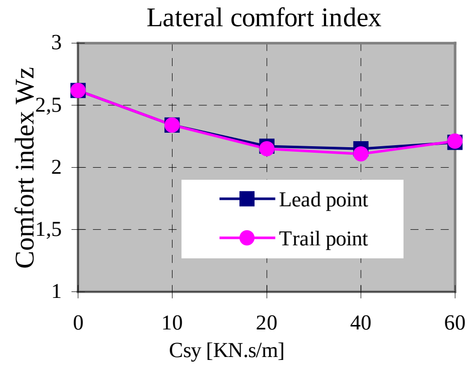


Figure 10 Lateral comfort index & C_{SY}

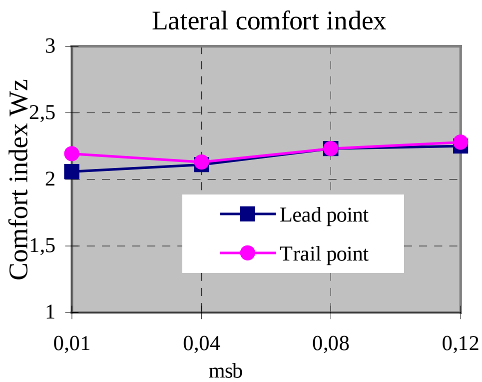


Figure 11 Lateral comfort index & μ_{SB}

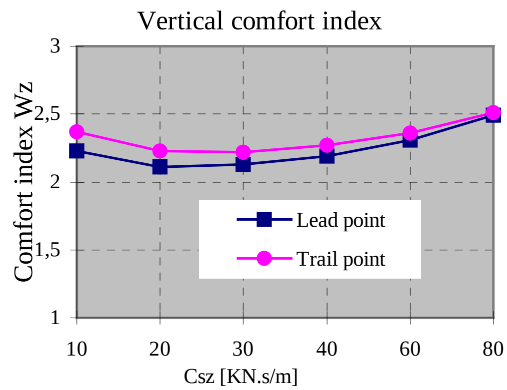


Figure 12 Lateral comfort index & C_{SZ}

4.3 Dynamic curve analysis and parameter study

The dynamic curve analysis is finished to optimize bumper stop free clearance and contact stiffness, to see how the side bearing yaw moment affects on the lateral force and yaw angle of wheel set. All simulations is under the same track condition: curve radius is 1500 m, transition curve length is 80 m, track initial cant is 80 mm, cant deficiency is 150 mm(curve passing speed is about 170 km/h). Track irregularities are also considered to calculate the ride comfort index in the circular part.

Figure 13 and 14 show the effect of bumpstop characteristic, figure 15 show how side bearing friction coefficient μ_{SB} affects on the wheel set lateral force(no track irregularities in this case).

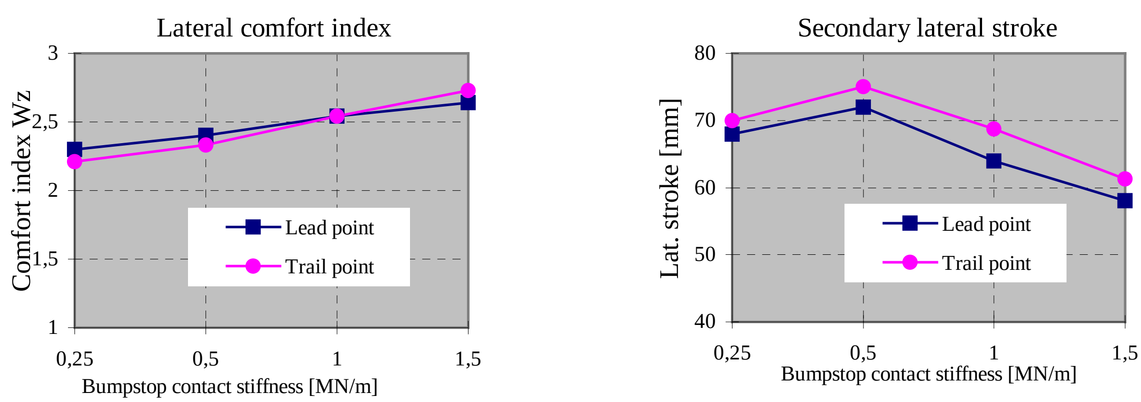


Figure 13 Bumpstop contact stiffness & comfort index in curve (Free clearance 40 mm)

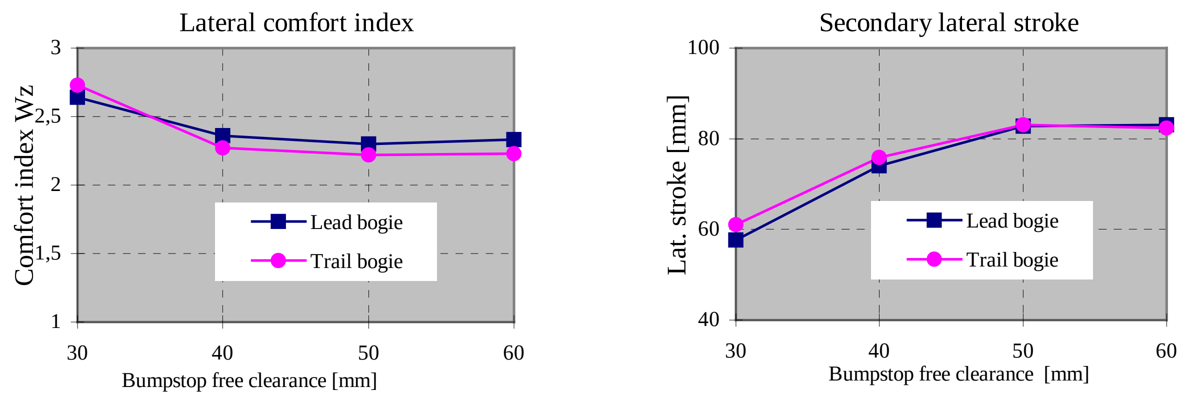


Figure 14 Bumpstop free clearance & comfort index in curve (Contact stiffness 0.4MN/m)

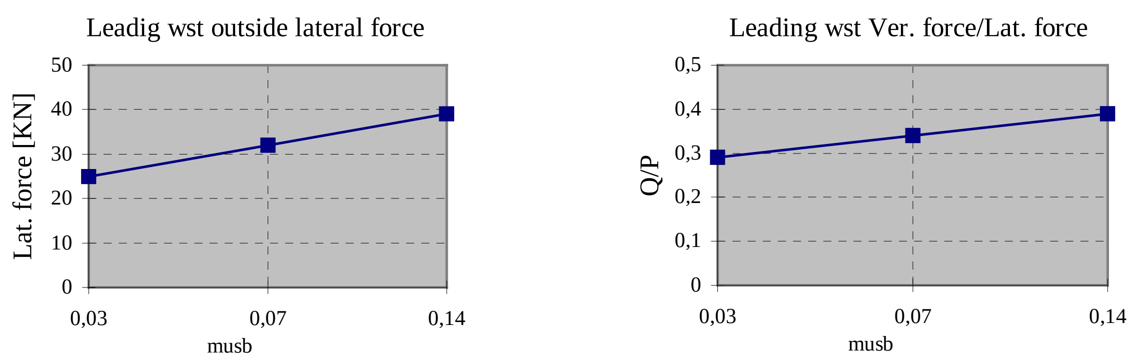


Figure 15 Side bearing friction coefficient μ_{SB} & wheel-rail lateral force

4.4 Dynamic behavior improvement through parameter optimum

Based on above dynamic analysis and parameter study, we find that original major suspension parameters of bogie PW200 are suitable and very close to optimum values, such as wheel set guide longitudinal and lateral stiffness, secondary link bar stiffness, vertical damper coefficient, but secondary lateral damper coefficient, bumpstop free clearance and contact stiffness could be optimized further to give improvement of curve passing behavior.

Here we simulate the dynamic curve behavior under the field test condition: curve radius is 3500 m, transition curve length is 40 m, track initial cant is 60 mm, curve passing speed is 220 km/h.

The simulation results show in figure 16 to figure 18.

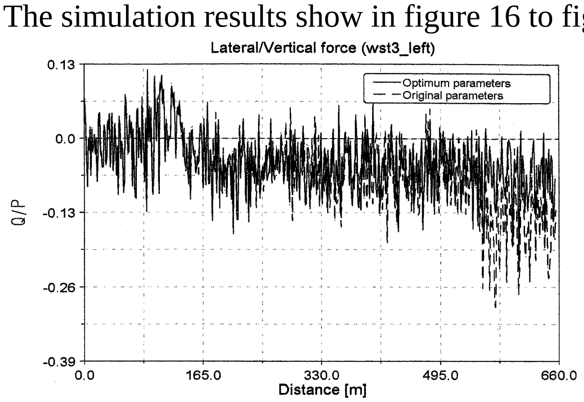


Figure 16 Lat./ Ver. force of wst3 left side

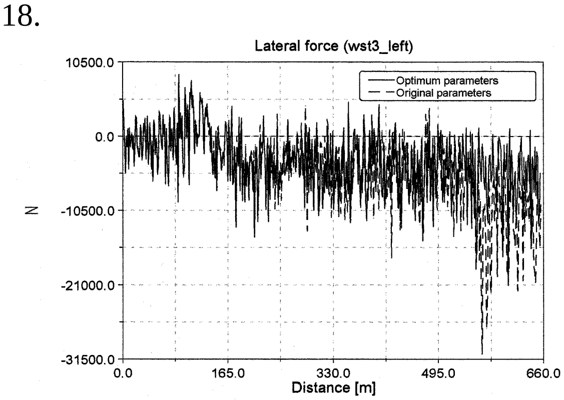


Figure 17 Lateral force of wst3 left side

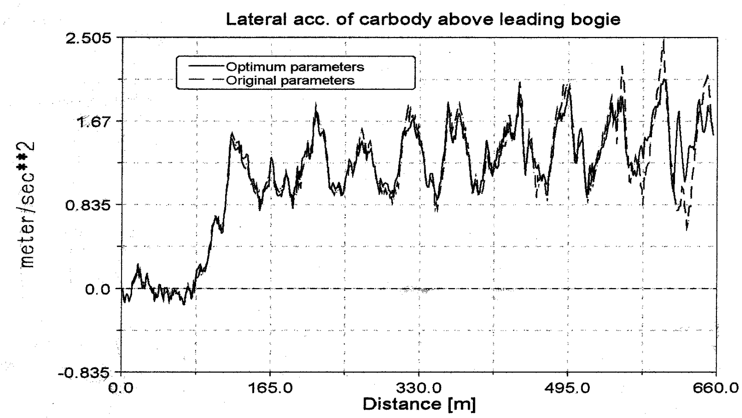


Fig 18 Lateral acceleration on the floor of cardody above leading bogie

5. Conclusion

With the help of ADAMS/Rail, the dynamic analysis of a new double deck passenger vehicle with bogie PW200 has been finished, and the major suspension parameters are optimized. The main conclusions can be made from this study :

This mathematics model of the full vehicle with bogie PW200 gives general acceptable agreement with test data, and is suitable for dynamic simulation and parameter optimum. The parameters study shows that major suspension parameters of bogie PW200 are suitable and very closed to optimum values.

The track irregularity has important effect on the carbody acceleration and ride comfort in both straight and curve track case. the further measurements of the track irregularity are useful to simulate the behavior of vehicle accurately.

The effect of suspension positions on dynamic behavior could be studied using this model, to decease roll motion of the carbody and improve lateral ride comfort of low floor of the carbody.