Calculation and Analysis of Gauges for Suspension-type Monorail Vehicle

Shen Zhan*, Bingyan Chen, Jiancai Zou, Wenqiang Qu and Xiaojun Wang

CRRC Qingdao Sifang Co., Ltd, Qingdao, China
Email: zhanshen@cqsf.com

Abstract. The differences between two types of monorail vehicles are compared, and the configuration of suspension-type monorail vehicle is introduced. The vertical and lateral displacements of carbody under two running conditions are derived from the corresponding directional motions, which is the basis on calculating the vehicle gauge for suspension-type monorail vehicles. A program is written to calculate the displacement and then the results are transformed into vehicle gauges. The validity of the derivation formulas is verified.

Keywords. Suspension-type monorail vehicle, Carbody displacement, Vehicle gauge.

1. Introduction
With the continuous development of urban rail transit industry, the way people work and travel has changed dramatically [1]. As a new mode of transportation different from traditional bogie vehicles, monorail vehicles have various advantages including the small floor space, the low cost and noise, which becomes a hotspot in the transportation research field [2]. In terms of the types of monorail vehicles, they are divided into straddle-type and suspension-type according to their running modes [3]. The carbody of the straddle-type monorail vehicle is located on the track beam, where the left and right ends have guiding wheels and stabilizing wheels to achieve steering. Conversely, the bogie of suspension-type monorail vehicle is situated on the top of the carbody, where the running wheels and the guiding wheels are suspended in the box-shaped track beam, with the vehicle running under the track beam.

Bao et al. carried out researches concerned with operation safety and dynamic response when vehicle is running on bridge according to the vehicle-bridge coupling theory and finite element method, proving that the level of track irregularity can greatly affect the vibration safety of suspension-type monorail vehicles [4-5]. Cai et al. set up a 21-DOF multi-body dynamics model by ANSYS, and explored the dynamic response of vehicle system through simulation and filed test, reaching a conclusion that the lateral vibration is larger than the vertical vibration owing to the low lateral stiffness of the bridge [6]. He et al. conducted measurements for a typical suspension-type monorail vehicle based on field test, through which they analyzed and compared the carbody accelerations under different running speeds and tracks from the time and frequency domain, consequently, they proposed that some sensitive wavelengths of the track can influence the running safety of vehicles [7-8]. Lv et al. combined simulation with experiments considering the vehicle and track system, and studied the effects of wheel eccentricity on vertical acceleration of vehicles [9]. Jiang et al. introduced a brand-new suspended monorail vehicle model, and established a vehicle-bridge coupling system, illustrating that the bridges modes including constraint and free types are excited as the vehicle runs on the bridge [10].

It’s clear that plentiful work has been completed in terms of dynamic responses for suspension-type monorail vehicles considering vehicle-bridge coupling system. Nevertheless, there are few references
about the calculation methods of vehicle gauge, which deserves deeper study. Hence, this paper illustrated a detailed method for calculating the gauge of suspension-type monorail vehicles through the combination of theoretical derivation and simulation. Firstly, the three-dimensional model of suspension-type monorail vehicle was demonstrated. Then, both the vertical displacements upward and downward of carbody under different load conditions were derived according to Ref [11-12]. Additionally, a computational program was written to verify the formulas. The results suggested that the derivation formula can meet the requirement of the vehicle gauge.

2. Three-dimensional Model

Figure 1 shows a classical suspension-type monorail vehicle running on the track beam, which consists of carbody, bogie frame, rubber wheels and corresponding connection devices.

![Figure 1. Suspension-type monorail vehicle.](image)

Figure 2 illustrates a typical configuration of a straddle-type monorail vehicle bogie, where the frame, gearbox, damper, traction rod and guiding/running wheel are the main components.

![Figure 2. Structure of suspension-type monorail vehicle bogie.](image)

Considering that the motion attitude of the suspension-type monorail vehicle resembles in conventional railway vehicles, the relevant standards [11-12] can be references in calculating the vehicle gauges. However, it is different in the corresponding displacement calculation formula due to the distinguishing structure of the monorail vehicles.
3. Derivation of Carbody Displacement

3.1. Lateral and Roll Motion of Carbody in the Same Direction

While the suspension-type monorail vehicle runs on the track beam, the carbody can move slightly in the lateral and vertical motion. The lateral displacement of carbody consists of its lateral swing, yaw and roll motion, while the vertical displacement of carbody is made up of its vertical swing, pitch and roll motion. When the lateral swing is in the same direction with roll motion, then the lateral displacement of carbody can be expressed as equation (1).

\[
\Delta X_{BP} = \eta_{w0} + \eta_{w1} + \Delta e + \left| Y \left( \frac{\Delta h_{x2}}{l} f_f \right) + \left( \frac{l - d}{2} \right) a_n + \left( \Delta q_1 + \Delta q_2 + \Delta w_1 + \Delta w_2 \right) a_n \right| + 100 m_f g_f \left( \frac{Y - h_{xP}}{K_{gP}} + \frac{Y - h_{xS}}{K_{gS}} + \frac{Y - h_{xR}}{K_{gR}} \right) + Y - h_{rol} \theta
\]

(1)

where

\[
C_h = \left( Y - h_{xP} \right) \frac{h_{wP} - h_{xP}}{K_{gP}} + \left( Y - h_{xS} \right) \frac{h_{wS} - h_{xS}}{K_{gS}} + \left( Y - h_{xR} \right) \frac{h_{wR} - h_{xR}}{K_{gR}} \]

(2)

\[
C'_h = \left( Y - h_{xP} \right) \frac{h_{wP} - h_{xP}}{K_{gP}} + \left( Y - h_{xS} \right) \frac{h_{wS} - h_{xS}}{K_{gS}} + \left( Y - h_{xR} \right) \frac{h_{wR} - h_{xR}}{K_{gR}} \]

(3)

\[
f_f = \frac{1}{m_f g \left( \frac{h_{wP} - h_{xP}}{K_{gP}} + \frac{h_{wS} - h_{xS}}{K_{gS}} + \frac{h_{wR} - h_{xR}}{K_{gR}} \right) + 1} \]

(4)

\[
K_{gP} = 0.5 n_p c_P b_P^2 + 4 K_{N1} \]

(5)

\[
K_{gS} = 0.5 n_s c_S b_S^2 + 2 K_{N2} \]

(6)

\[
h_{rol} = \left( \frac{\Delta h_{xP}}{K_{gP}} + \frac{\Delta h_{xS}}{K_{gS}} + \frac{\Delta h_{xR}}{K_{gR}} \right) \]

(7)

For the vertical upward displacement of carbody, it can be calculated in equation (8).
\[ \Delta Y_{BU} = \Delta m_9 + X \left( \frac{\Delta h_{z2}}{t} f_F \right) + 100 m_2 g f_F \left( \frac{1}{K_{\phi P}} + \frac{1}{K_{\phi S}} + \frac{1}{K_N} \right) X + X \vartheta \]

\[
\left[ \frac{\Delta f_P a_n}{h_{eq}} \right]^2 + \left( \frac{\Delta f_{SS} a_n}{h_{eq}} \right)^2 + \Delta m_{16}^2 + \Delta m_{17}^2 + \Delta m_{116}^2 + \Delta m_{117}^2 + \delta_c^2 \\
+ \left( \frac{\Delta h_{z1} f_F}{h_{eq}} \right)^2 + \left( \frac{\Delta h_{z1} f_F}{t} \right)^2 \]

\[
\left[ A_w P_n f_F X \left( \frac{h_{sw} - h_{sw}}{K_{\phi P}} + \frac{h_{sw} - h_{sw}}{K_{\phi S}} + \frac{h_{sw} - h_{sw}}{K_N} \right) \right]^2 \\
+ \left[ m_o a_o f_F X \left( \frac{h_{se} - h_{se}}{K_{\phi P}} + \frac{h_{se} - h_{se}}{K_{\phi S}} + \frac{h_{se} - h_{se}}{K_N} \right) \right]^2 \]

For the vertical downward displacement of carbody, it can be calculated in equation (9).

\[
\Delta Y_{BD} = f_{01} + f_1 + f_{02} + f_2 + \Delta m_9 D + \delta_c + \delta_w + \delta_e + FR \\
+ \sqrt{\left( \frac{\Delta f_P a_n}{h_{eq}} \right)^2 + \left( \frac{\Delta f_{SS} a_n}{h_{eq}} \right)^2 + \Delta m_{16}^2 + \Delta m_{17}^2 + \Delta m_{116}^2 + \Delta m_{117}^2 + \delta_c^2} \\
- X \vartheta \left[ \frac{\Delta h_{z2} f_F}{t} \right] - 100 m_2 g f_F X \left( \frac{1}{K_{\phi P}} + \frac{1}{K_{\phi S}} + \frac{1}{K_N} \right) X \\
+ \left( \frac{\Delta h_{z1} f_F}{h_{eq}} \right)^2 + \left( \frac{\Delta h_{z1} f_F}{t} \right)^2 \]

3.2 Lateral and Roll Motion of Carbody in the Reverse Direction

When the lateral swing is in the reverse direction with roll motion, similarly, the lateral displacement can be in the form of equation (10).
\[ \Delta X_{BP} = \eta_{w0} + \eta_{w1} + \Delta e + \left( \frac{1-d}{2} \right) a_n + (\Delta q_1 + \Delta q_2 + \Delta w_1 + \Delta w_2) a_n \]

\[ + \sqrt{\left( \frac{\Delta d}{2} a_n \right)^2 + \left( \frac{\Delta w_3}{2} a_n \right)^2 + \Delta c^2 + \Delta m_{12}^2 + \Delta m_{14}^2 + \Delta m_{16}^2 + \Delta m_{17}^2} \]

\[ - 100m_c g f_F \left( \frac{Y - h_{cp}}{K_{\phi p}} + \frac{Y - h_{cr}}{K_{\phi t}} \right) \]

For the vertical downward displacement of carbody, the calculation formula can be derived as equation (12).
\[ \Delta Y_{BD} = f_0 + f_1 + f_2 + f_3 + f_4 + f_5 + f_6 + f_7 + f_8 + f_9 + f_0 + f_1 + f_2 + \Delta m_{23} + \Delta \delta_0 + \Delta \delta_1 + XR + X \sin \theta \]

\[ + X \left( \frac{\Delta \theta_2 f_F}{t} \right) + 100m_g f_F X \left( \frac{1}{K_{\phi_P}} + \frac{1}{K_{\phi_S}} + \frac{1}{K_{\phi_R}} \right) \]

\[ \left( \Delta f_2 a_n \right)^2 + \left( \Delta f_3 a_n \right)^2 + \Delta m_{23}^2 + \Delta m_{17}^2 + \Delta m_{117}^2 + \delta_r^2 \]

\[ + \left( \frac{\Delta x_{bd} X}{h_{eq}} \right)^2 + \left( \frac{\Delta \theta_2 f_F}{t} \right)^2 \]

\[ + \left( A_P f_F X \left( \frac{h_{w} - h_{eq}}{K_{\phi_P}} + \frac{h_{w} - h_{eq}}{K_{\phi_S}} + \frac{h_{w} - h_{eq}}{K_N} \right) \right)^2 \]

\[ + \left( m_d f_F X \left( \frac{h_{w} - h_{eq}}{K_{\phi_P}} + \frac{h_{w} - h_{eq}}{K_{\phi_S}} + \frac{h_{w} - h_{eq}}{K_N} \right) \right)^2 \]

(12)

4. Computational Verification

According to the above displacement calculation formula of carbody under different working conditions, a program is written to acquire the vertical and lateral displacement of carbody, then the results are transformed to its correlated vehicle gauge schematic diagram.

Figure 3 presents a gauge calculation result of a suspension-type monorail vehicle which is running on the straight track beam. It could be seen that the construction gauge and device gauge are in the Ref [11], while the vehicle gauge including lateral and roll motion of carbody in the same/reverse direction is in the limitation of the corresponding device gauge, which can satisfy the requirement of the gauge standard.

Figure 3. Gauge calculation result on straight track.
5. Conclusion
In this paper, the calculation of the suspension-type vehicle gauge under different running conditions was carried out in detail with the help of the standards. At the same time, for the derivation formula, the corresponding calculation program was written to verify the validity of the formula. The schematic diagram of the vehicle gauge obtained from the calculation results shows that the calculation formula meets the requirements of the vehicle gauge regardless of the direction of the lateral and roll motion.

6. Acknowledgments
This work was supported by the National Key R&D Program (Grant No.2018YFE0201400).

7. Appendix. Vehicle Parameters

| Parameters                                           | Values          |
|------------------------------------------------------|-----------------|
| Length between bogie centers                         | \( a=5800 \text{ mm} \) |
| Wheelbase between guiding wheel                      | \( p=1600 \text{ mm} \) |
| Weight of carbody                                     | \( m_b=25000 \text{ kg} \) |
| Lateral manufacturing tolerance of guiding wheel      | \( \Delta d=1.5 \text{ mm} \) |
| Static lateral displacement between frame and track   | \( \Delta X_{db}=2 \text{ mm} \) |
| Wears and deformations of guiding wheel               | \( \Delta q=10 \text{ mm} \) |
| Vertical elastic deformations of running wheel        | \( f_{01}=5 \text{ mm} \) |
| Deflection of running wheel                          | \( f_i=10 \text{ mm} \) |
| Installation tolerance between the secondary suspension| \( f_{02}=5 \text{ mm} \) |
| Deflection of the secondary suspension                | \( f_2=3 \text{ mm} \) |
| Dynamic deflection of running wheel                   | \( \Delta f_p=5 \text{ mm} \) |
| Lateral deformation of the secondary spring(static)   | \( \Delta w_0=10 \text{ mm} \) |
| Lateral deformation of the secondary spring(dynamic)  | \( \Delta w_1=15 \text{ mm} \) |
| Wears of center pivot                                | \( \Delta w=1 \text{ mm} \) |
| Installation tolerance of center pivot                | \( \Delta m_0=1 \text{ mm} \) |
| Dynamic deflection of secondary suspension upward     | \( \Delta f_c=15 \text{ mm} \) |
| Dynamic deflection of secondary suspension downward   | \( \Delta f_c=15 \text{ mm} \) |
| Distance between the secondary spring and rail        | \( h_{c}=709 \text{ mm} \) |
| Vertical stiffness of individual secondary spring      | \( C_s=160 \text{ N/mm} \) |
| Distance between the secondary spring                 | \( b_s=490 \text{ mm} \) |
| Height of side wall of carbody                        | \( h_{cw}=2767 \text{ mm} \) |
| Area of wind area of carbody                         | \( a_w=33.5 \text{ m}^2 \) |
| Pressure of crosswind                                 | \( p_w=400 \text{ N/m}^2 \) |
| Distance between bottom of carbody and rail           | \( h_{wa}=736 \text{ mm} \) |
| Gradient of the carbody                               | \( \Delta x_{qq}=10 \text{ mm} \) |
| Lateral acceleration                                 | \( a_0=0.5 \text{ m/s}^2 \) |
| Installation tolerance of the carbody surface devices | \( \Delta m_{10}=1 \text{ mm} \) |
| Installation tolerance of the carbody upward          | \( \Delta m_{15}=5 \text{ mm} \) |
| Installation tolerance of the carbody downward        | \( \Delta m_{20}=2 \text{ mm} \) |
| Tolerance of the floor                                | \( \Delta m_{2}=10 \text{ mm} \) |
| Lateral tolerance of track beam centerline            | \( \Delta r=5 \text{ mm} \) |
| Vertical tolerance of track beam centerline           | \( \sigma_r=5 \text{ mm} \) |
| Lateral deformation of track beam                     | \( \Delta r=1 \text{ mm} \) |
| Vertical deformation of track beam                    | \( \sigma_r=2 \text{ mm} \) |
References
[1] Zhai W M and Zhao C F 2016 Frontiers and challenges of sciences and technologies in modern railway engineering Journal of Southwest Jiaotong University 51 (2) 209-226 (in Chinese)
[2] Li Z J, Lin H S and Wu B W 2020 Analysis of curbe negotiation performance of suspended monorail train Railway Standard Design 64 (4) 42-47 (in Chinese)
[3] Li F, Xu W C and An Q 2014 Development and current status of suspended monorail vehicle Electric Drive for Locomotives 2 (1) 16-21 (in Chinese)
[4] Bao Y L, Li Y L and Ding J J 2016 A case study of dynamic response analysis and safety assessment for a suspended monorail system International Journal of Environmental Research and Public Health 13 (11) 1121-1138
[5] Bao Y L, Xiang H Y and Li Y L 2020 A dynamic analysis scheme for the suspended monorail vehicle-curved bridge coupling system Advances in Structural Engineering 23 (8) 1728-1738
[6] Cai C B, He Q L and Zhu S Y 2019 Dynamic interaction of suspension-type monorail vehicle and bridge: Numerical simulation and experiment Mechanical Systems and Signal Processing 118 (1) 388-407
[7] He Q L, Cai C B and Zhu S Y 2019 Field measurement of the dynamic responses of a suspended monorail train-bridge system[J] Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit 234 (10)
[8] He Q L, Cai C B and Zhu S Y 2020 Key parameters selection if suspended monorail system based on vehicle-bridge dynamical interaction analysis Vehicle System Dynamics 58 (3) 339-356
[9] Lv K K, Wang K Y and Chen Z H 2017 Influence of wheel eccentricity on vertical vibration of suspended monorail vehicle: experiment and simulation Shock and Vibration 1367683
[10] Jiang Y Z, Wu P B and Zeng J 2019 Researches on the resonance of a new type of suspended monorail vehicle-bridge coupling system based on modal analysis and rigid-flexible coupling dynamics Vehicle System Dynamics 58 (9) 1-21
[11] Ministry of Housing and Urban-Rural Development of the People's Republic of China 2018 Standard for Metro Gauges (Beijing: China Construction Industry Press) (in Chinese)
[12] Ministry of Housing and Urban-Rural Development of the People's Republic of China 2008 Code for Straddle Monorail Transit (Beijing: China Construction Industry Press) (in Chinese)