



Impact of Longitudinal Train Dynamics on Train Operations: A Simulation-Based Study

Sunil Kumar Sharma¹ · Anil Kumar²

Received: 17 November 2015 / Revised: 25 January 2016 / Accepted: 27 August 2016 / Published online: 6 August 2018
© Krishtel eMaging Solutions Private Limited 2018

Abstract

Purpose The paper is an attempt to evaluate the impact of coupler forces in the train produced during braking. The braking signal generated from a locomotive control valve takes a few milliseconds to reach the adjoining car in coupled passenger trains. The brake comes into effect in the last car of the train after a few seconds due to the time lag. This delayed application of brakes results in pushing off the rear cars into the front cars, producing large compressive forces in the coupler. These compressive forces, mainly longitudinal in nature cause passenger discomfort and poor ride quality.

Methods Non-linear dynamic analysis has been conducted to represent the characteristic of coupling forces between the coaches in longitudinal train dynamics. The analysis involved the mathematical model of coupler force and braking force through experimental results. In addition, effects of various braking phases, i.e., auxiliary, service and emergency braking on the in-train forces, were also investigated using ‘universal mechanism—software of dynamics of mechanical systems’.

Results The train response due to different braking phases on the train longitudinal dynamics is thoroughly studied using multibody dynamics analysis. The results establish that the in-train forces were well within the limits prescribed by Research Design and Standard Organisation.

Conclusions The maximum compression force increased when the forward velocity of the train is reduced during the braking phase. And, this maximum compression occurs at the third quarter of the train.

Keywords Train dynamics · Brake delay time · Train longitudinal forces · Coupler connection

Introduction

Longitudinal train dynamics (LTD) is considered one of the important aspects in the design of trains. In common parlance, it is the movement of the locomotive and coaches of a train and the incidental in-train forces of the coaches with respect to the track. LTD depends on many factors such as braking dynamics (train brake lag time), coupler dynamic specifications, load distribution of coaches, operational parameters, and the train velocity. The study of train

longitudinal dynamics and stipulating out an optimal set of design parameters is a complex investigation. Any wrong design with respect to the above-mentioned factors will lead to a significant impact on the running stability, loading integrity, safety of the vehicle’s structure and passengers’ comfort [1, 2]. Due to this reason, in the preceding two decades, studies on LTD were conducted in the direction to reduce the longitudinal in-train forces and to augment the dynamic performance of trains [1, 3, 4].

When brakes are initiated, the train body is first exposed to an accentuated compression, trailed by bounce back and then with a run-over of oscillatory movements that determine a mechanical wave which traverses through the train numerous times. Subsequently, sundry tensile in-train forces and longitudinal compression forces develop, which generate stresses in couplers, affecting passengers’ comfort [5]. Therefore, the study of braking dynamics contributed significantly to predict the behaviour of train system and led to improvements in comfort and running safety, optimisation

✉ Sunil Kumar Sharma
sunilsharmaiitr@gmail.com

Anil Kumar
kumara.iitr@gmail.com

¹ Department of Mechanical Engineering, Amity School of Engineering and Technology, Amity University Uttar Pradesh, Noida, India

² Centre for Transportation Systems, Indian Institute of Technology Roorkee, Roorkee, India

of train structure, capacity growth, lower maintenance costs and lower damage risks.

Apart from braking dynamics, the in-train forces or coupler forces also play an important role in LTD. The in-train forces are the corollary of the sudden contrasts, i.e., tensile or compressive, of the longitudinal forces occurring between coaches. Immediate contrasts among forces that act on every coach determine the assembly of complex longitudinal responses, acting and being transmitted through the shock and traction apparatus. Wu et al. [6] presented a detailed review of modern dynamic models of draft gears. They concluded that draft gears modelling is one of the most important and difficult tasks in LTD. The ride quality and comfort of a vehicle depend on the velocity, displacement, acceleration, jerk of the vehicle along with other environmental factors, i.e., dust, noise, temperature and humidity [7, 8].

As mentioned before, braking dynamics and coupling between coaches are affecting the ride quality and comfort. Therefore, these need to be considered while designing the trains. There is no literature available on the study of LTD considering braking dynamics and coupling between the coaches for Indian Railway (IR) systems to the best knowledge of the authors. Therefore, in the present article, LTD for a passenger train was investigated with braking dynamics and coupling dynamics.

Longitudinal Train Dynamics Modelling

Mathematical models were developed to represent the dynamics of various structures of train, i.e., train modelling, modelling of coupler or in-train forces, modelling of braking phenomena and rolling resistance.

Train Modelling

A typical train model for simulation of longitudinal dynamics, described in many papers is used in the presented research [5–8]. The train model is a damped elastic system that consists of n number of rigid bodies of masses, $m_1, m_2 \dots m_n$ representing the train coaches as exhibited in

Fig. 1. Only a longitudinal degree of freedom, connected by coupling devices along with the forces acting on each coach is considered. Lateral and vertical dynamics were not considered [13]. The various forces acting on each coach in the train body are inertia forces, $F_{i,1}, F_{i,2}, \dots F_{i,n}$; braking forces, $F_{brake,1}(t), F_{brake,2}(t), \dots F_{brake,n}(t)$; forces in the coupling devices, $F_1(\Delta x_1, \Delta \dot{x}_1), F_2(\Delta x_2, \Delta \dot{x}_2), \dots F_{n-1}(\Delta x_{n-1}, \Delta \dot{x}_{n-1})$; rolling resistances, $F_{rr,1}, F_{rr,2}, \dots F_{rr,n}$ and traction/braking effort.

The system of differential equations is shown in Eqs. (1–3):

$$m_1 \ddot{x}_1 - F(\dot{y}_1, y_1, \dot{y}_2, y_2) = F_{t/bd} - F_{brake,1}(t) - F_{rr,1} \tag{1}$$

For the i th coupling device (coach number i and $i + 1$), one can write it as

$$m_i \ddot{x}_i - F(\dot{y}_i, y_i, \dot{y}_{i-1}, y_{i-1}) = F_{t/bd} - F_{brake,i}(t) - F_{rr,i} \tag{2}$$

For the $(n - 1)$ th, the last coupling device (coach number $n - 1$ and n), one can write it as

$$m_n \ddot{x}_n - F(\dot{y}_n, y_n, \dot{y}_{n-1}, y_{n-1}) = F_{t/bd} - F_{brake,n}(t) - F_{rr,n} \tag{3}$$

The solution to the above-given equations is further obscured by the need to compute the system’s input forces, i.e., F_t, F_{brake} , and F_{rr} . Methodologies to model coach connections with every single forcing inputs are included and discussed in the subsequent subsections.

Coaches Connection System Modelling

In a train, the adjacent vehicle is connected mechanically using a device known as coupler which plays an important role in the analysis of LTD [9–13]. This technique is easier to apply than the AAR and has no significant error consequence unless used for very sharp curves (less than 0.1% at $R = 100$ m). In this study, AAR type H tight lock coupler (Fig. 2), with a low preload draft gear has been considered [14, 15].

The coupler system is a connection between two adjacent coaches in a train. It consists of coupler head with drawbar and its guide and draw and buffing gear in draft gear. The

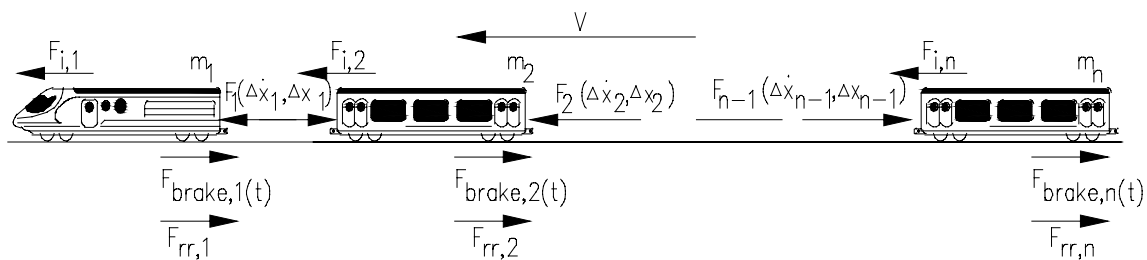


Fig. 1 Forces acting in a train

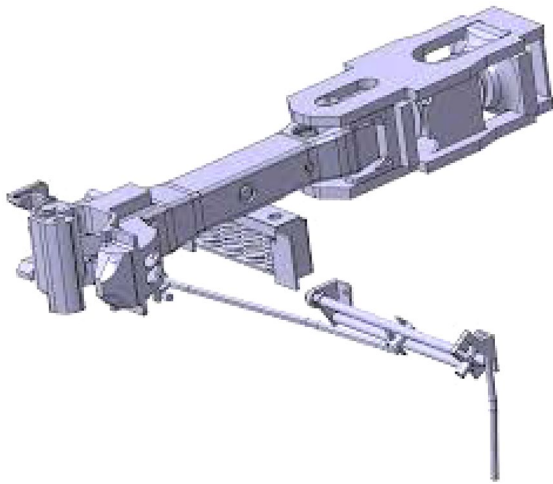


Fig. 2 AAR type H tight lock coupler [15]

coupler system is usually shortened into single-element models, so every two draft gears are exhibited in series as a single unit. A unit model of coupler system has to incorporate characteristics that can simulate under both draft and buff conditions. The final model must also consider coupler slack as well as the limiting stiffness that appears after springs are fully compressed. When installed, draft gears are usually preloaded, which should also be incorporated.

To sum up, a desirable coupler system model should include the above-discussed elements: slack, limiting stiffness, preload, velocity dependent friction, and transitional characteristics. In Eq. (4), k_a is an elastic constant and k_{fr} is a frictional spring constant [15, 16]. The coupler system consists of preload for holding the adjacent coupler head which is here denoted by P . Where, y is the relative displacement of draft gear stroke and \dot{y} is its relative speed. The conditions of coupler assemble in buff mode, preload tensile and draw gear are $y < 0$, $y = 0$, and $y > 0$. For performing simulations, the main parameters used for the buffers in buff mode are: $P = 25$ kN, $k_{ab} = 11,785$ kN/m, $k_{frb} = 5457$ kN/m; and $k_{ad} = 9430$ kN/m and $k_{frd} = 4365$ kN/m [16] for the draw gear. Thus, coupler force was evaluated by the following equation and shown in Fig. 3.

$$F_i(\dot{y}, y) = \begin{cases} k_{ab}y_i + k_{frb}|y_i| \tanh(u \cdot \dot{y}) + P & \text{for } y < 0, \\ P & \text{for } y = 0, \\ k_{ad}y_i + k_{frd}|y_i| \tanh(u \cdot \dot{y}) + P & \text{for } y > 0 \end{cases} \quad (4)$$

Air Brake System Modelling

The passenger train under study is fitted with automatic air brake system. The peculiar air brake system begins with the reduction of air pressure in the brake hose. This is achieved

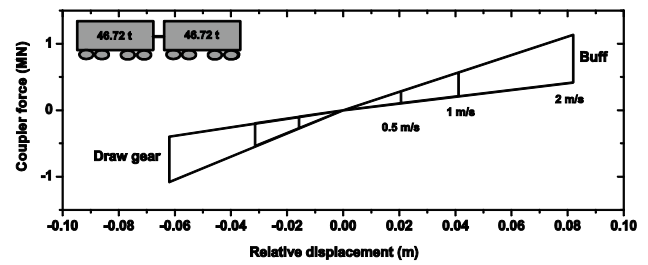


Fig. 3 Force-relative displacement characteristics of the coupler at different initial velocities for 0.5, 1.0 and 2 m/s

through brake control valve. Simultaneously, the distributor valve of every compartment comes into operation, due to which air flows from auxiliary air reservoir to the brake cylinder. This ultimately causes the cylinder thrust to push into moving wheels of each individual compartment. The ratio of the transmission is a function of parameters such as the size of the cylinder, dynamic efficiency of rigging and cylinder, rigging ratio, friction coefficient of disc brake along with other operational conditions. Hence, to calculate the braking force Eq. (5) is used [16, 17]. Table 1 gives the value of brake modelling parameter:

$$F_{\text{brake}(n)} = \left(\left(\frac{\pi d_{bc}^2}{4} \right) \eta_{\text{cyl}} p_{\text{max}(n)} - F_{\text{sr}} \right) i_r n_{bc} \frac{2r_m}{D_0} \mu_d \eta_{br}. \quad (5)$$

Rolling Resistance Modelling

The sum of air resistance and rolling resistance is known as propulsion resistance of the vehicle. Its effect is largely dependent upon the design and shape of the vehicle and aerodynamic drag complexity [18–20]. The reckoning of the summation process is still a function of the empirical formulae. In the study, a relation similar to that of Davis relation was used for predicting propulsion resistance [21] as given in (6):

$$\begin{aligned} \text{Locomotive(WAP5)} &= 1.34819 + .02153v + 0.00008358v^2 \\ \text{Passenger coaches(LHB)} &= 0.69873 + .021533v + 0.0000835v^2. \end{aligned} \quad (6)$$

Table 1 Brake modelling parameters

Parameters	Value
d_{bc} , diameter of brake cylinder	254.8 mm
F_{sr} , release spring force	1.3 kN
Amplification ratio	1.954
D_0 diameter of wheel	1092 mm
r_m medium friction radius	588 mm
η_{bc} number of brake cylinder	8
μ_d coefficient of friction between pads and discs	0.35
η_{br} brake rigging mechanical efficiency	95%

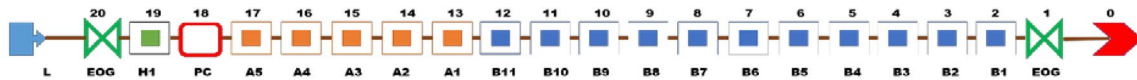


Fig. 4 Arrangement of coaches in Rajdhani express (LHB) Rake

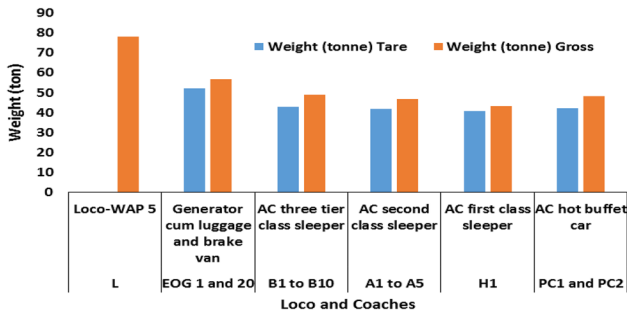


Fig. 5 Types of coaches and their weight [15]

Running resistance is given by an empirical formula, $F_{rr} = a + bv + cv^2$, where a , b and c are dependency constants, v is speed in km/h and F_{rr} is train resistance in kg/ton.

Case Study and Software Modelling

Case Study Information

In the case study, the India’s fastest passenger train, i.e., Rajdhani Express is considered which runs at 40 m/s. The arrangement of various types of coaches is shown in Fig. 4. The various types of coaches and their respective weight for tare and the gross condition are shown in Fig. 5. The length of railway track considered from Palwal to Mathura is 87 km.

Locomotive Design Aspects

The WAP-5 locomotive is considered for the study. For the typical configuration, the design characteristics and specifications are given in Table 2.

These attributes and specifications were utilized to model locomotive performance curves for each standard unit for their implementation in the train dynamics studies, so as to describe the capability of the locomotive and dynamic braking. Manufacturers measure tractive effort (TE) and braking effort (BE) as a function of speed [24, 25]. The TE and BE of WAP-5 electric locomotive are shown in Fig. 6. These

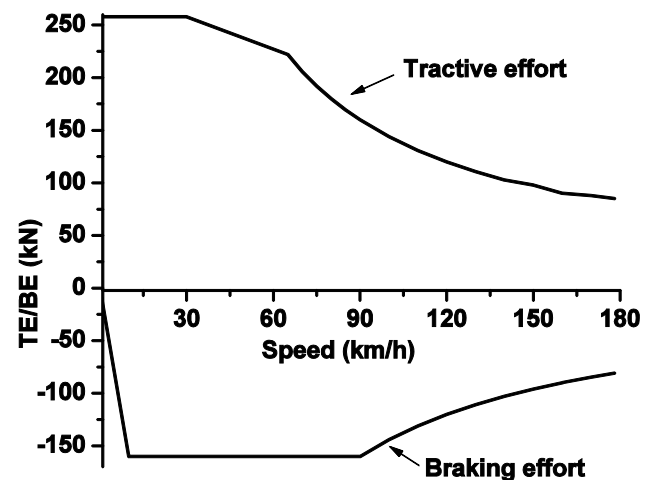


Fig. 6 Tractive effort and braking effort performance curve for WAP-5

Table 2 Characteristics and specifications of WAP5 locomotive [22, 23]

Characteristics	Locomotive specifications
Electric locomotive	WAP-5
Traction motors	3-phase squirrel cage induction motors ABB’s 6FXA 7059, forced-air ventilation weight 2050 kg, 96% efficiency, partially suspended, torque 6930/10,000 Nm
Wheel diameter	New: 1092 mm, full-worn: 1016 mm
Axle load (ton)	19.5
Power	Max: 6000 hp (4474 kW)
Tractive effort	Max: 258 kN
Gear ratio	3-stage gears (67:35:17)
Bogies	Bo–Bo, Henschel Flexi float
Locomotive weight (ton)	78

graphs were obtained experimentally by RDSO [25, 26]. In the graphs, the maximum tractive effort is 258 kN and braking effort is 160 kN [26–28].

Train Model in Universal Mechanism Software

A study on heavy haul train and car dumper model were carried out using universal mechanism (UM) to analyze train longitudinal dynamics during dumping by Kovalev et al. [13]. Hence, for the analysis of LTD, the 3D dynamic train model was developed in UM software using the information provided in the previous sections. The train model is shown in Fig. 7.

Condition for Simulation

The train was moving at a speed of 40 m/s. The braking process that began in different modes at different times in the simulation are given in Fig. 8. It has been assumed that the speed of propagation of air in the brake line (the so-called wave speed braking) is 220 m/s. The final pressure in brake cylinder of locomotive is 53,200 Pa.

Fig. 7 Model of the train in universal mechanism software

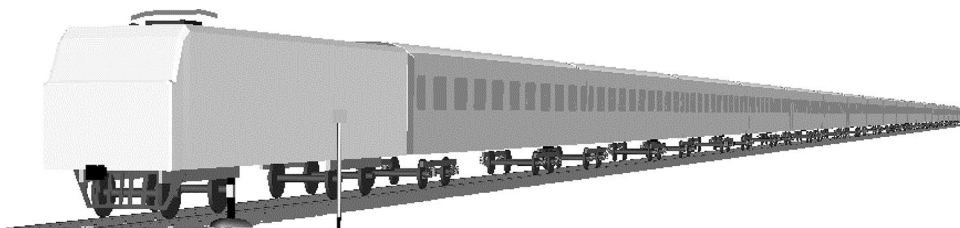
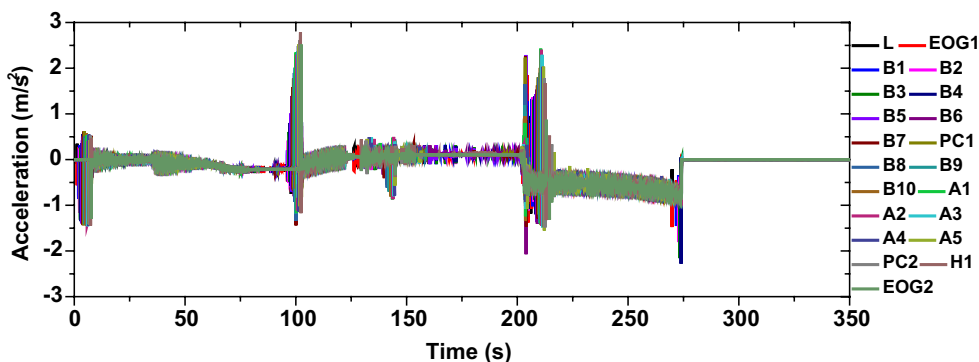


Fig. 8 Different modes at different times in the simulation



Fig. 9 Coaches acceleration due to braking phases



Results and Discussion

The various braking phases considered for analysis have been shown in Fig. 8. In this analysis, application of the dynamic brake system was applied solely to the locomotive. However, these brake phases lead to change in the acceleration and in-train forces, which could be observed in Figs. 9 and 10.

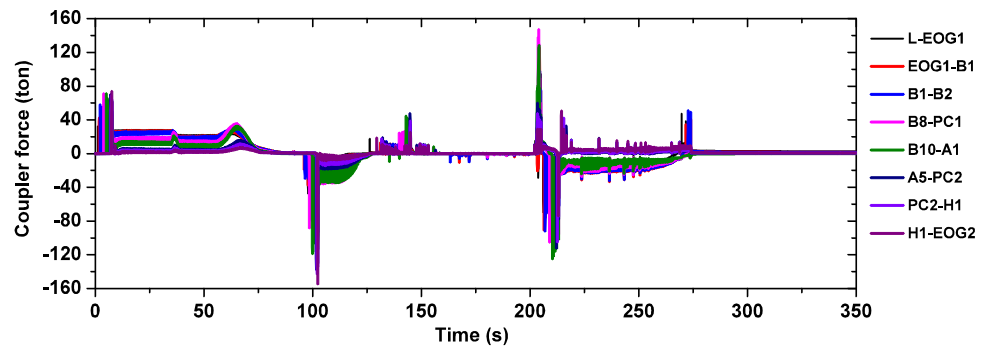
Phase 1: Auxiliary Brake (Service Braking)

Phase 1 ($0 \leq t \leq 25$ s) was a type of auxiliary braking in service braking mode. In this phase, the initial velocity drops from 40 to 39.6 m/s and acceleration lies between $+ 0.54$ to $- 1.375 \text{ m/s}^2$. The maximum in-train force occurred in the last three coaches. The maximum in-train force was about 63, 62 and 68 tons, respectively.

Phase 2: Brake Pipe (Service Braking)

In this phase ($25 \leq t \leq 80$ s), the service brake was applied to the train which caused more reduction in the train velocity up to 38.3 m/s and acceleration varied between $+ 0.0098$ to $- 0.11 \text{ m/s}^2$. The brake application

Fig. 10 In-train forces due to braking phases



time was observed as 20.41 s for full-service application on the last coach. That resulted in a sequential brake lag time of 0.97 s among the adjoining cars. In contrast to the preceding phase, at the preliminary stage of this phase, a high magnitude in-train compressive force occurred, at first three coaches of train, i.e., 24.47, 24.21 and 24.37 tons for first three coaches, respectively.

Phase 3: Auxiliary Brake (Running Repeater)

The running repeater mode braking was applied in Phase 3 ($80 \text{ s} \leq t \leq 155 \text{ s}$) of braking operation. In this phase, the velocity of the train was initially decreased from 38.3 to 32.94 m/s and then it started to increase eventually to reach to 28.5 m/s at the end of the phase. This variation of reduction and increment in the velocity was due to the track elevation as the train goes on uphill and downhill at the end of this phase. The average acceleration was -0.11 m/s^2 . Moreover, the train experiences buff mode on the uphill and the maximum in-train compressive force was of considerable magnitude. In the later stage of braking, the train gets stretched as it starts going downhill, and the amount of in-train tension forces increased rapidly. The maximum in-train force was about -124 , -127 and -141 ton for the last three coaches, respectively, as shown in Fig. 10.

Phase 4: Brake Pipe (Release)

In this phase, the release of service brake was considered ($155 \text{ s} \leq t \leq 210 \text{ s}$). The train velocity increased and reached to 35 m/s and acceleration lied between 0.44 and 0.025 m/s^2 . In contrast to the previous phases, the extent of in-train tension force among the coaches was reduced. Moreover, the train was making up and the train dynamic motion appeared to be very complex with multi-mode behaviour. The maximum in-train tension force value lied between 27.17 and -17.8 tons for the middle coaches, i.e., between 10th and 18th coach.

Phase 5: Brake Pipe (Emergency Braking)

In this phase, the emergency brake was applied to stop the train. The train velocity changed from 35 to 0 m/s and the time lag, which was defined as the time duration in which the pressure signal initiated from the locomotive reached the end of the train was measured to be 5.3 s. That amounted to a sequential brake lag time among the succeeding cars of 0.12 s. The maximum in-train forces lied between 143 and -120 ton.

Conclusions

The results of different braking phases on the train longitudinal dynamics are thoroughly studied using multibody dynamics analysis. The train consists of a locomotive and 20 coaches fitted with AAR type *H* tight lock coupler, with a low preload draft gear running at maximum speed. A mathematical model is formulated to analyse the coupler characteristics and air brake dynamics. The brake delay times are considered to be 0.12 and 0.97 s for emergency and service brakes. This period of time paved the way for inter coaches' states to slowly move from bunched to stretch or vice versa, avoiding large in-train forces. The simulation results suggested that for all braking phases considered the maximum compression force increased when the forward velocity of the train is reduced. It is observed that compression with maximum magnitude is noticed at the third quarter of the train. The maximum compression force observed is around 143 tons.

References

- Sharma SK, Chaturvedi S (2016) Jerk analysis in rail vehicle dynamics. *Perspect Sci* 8:648–650. <https://doi.org/10.1016/j.pisc.2016.06.047>

2. Jesussek M, Ellermann K (2015) Fault detection and isolation for a nonlinear railway vehicle suspension system. *Adv Vib Eng* 3(6):743–758
3. Sharma SK, Sharma RC, Kumar A, Palli S (2015) Challenges in rail vehicle-track modeling and simulation. *Int J Veh Struct Syst* 7(1):1–9
4. Sharma SK, Kumar A (2016) Dynamics analysis of wheel rail contact using FEA. *Procedia Eng* 144:1119–1128. <https://doi.org/10.1016/j.proeng.2016.05.076>
5. Pugi L, Malvezzi M, Papini S, Vettori G (2013) Design and preliminary validation of a tool for the simulation of train braking performance. *J Mod Transp* 21:247–257
6. Wu Q, Luo S, Cole C (2014) Longitudinal dynamics and energy analysis for heavy haul trains. *J Modern Transp* 22:127–136
7. Kumar H, Sujatha C (2010) Analytical and experimental studies on prediction of the lateral dynamic behavior of rail bogie used in Indian railways. *J Vib Eng Technol* 9(2):191–205
8. Sharma RC, Palli S, Sharma SK, Roy M (2017) Modernization of railway track with composite sleepers. *Int J Veh Struct Syst* 9(5):321–329
9. CrĂciun C, Mazilu T (2014) Simulation of the longitudinal dynamic forces developed in the body of passenger trains. *Ann Fac Eng Hunedoara Int J Eng* 12:19–26
10. Sharma SK, Kumar A (2017) Impact of electric locomotive traction of the passenger vehicle ride quality in longitudinal train dynamics in the context of Indian railways. *Mech Ind* 18(2):222. <https://doi.org/10.1051/meca/2016047>
11. Sharma SK, Kumar A (2017) Ride performance of a high speed rail vehicle using controlled semi active suspension system. *Smart Mater Struct* 26(5):055026. <https://doi.org/10.1088/1361-665X/aa68f7>
12. Tamboli JA (2006) Measurement of ride comfort of a vehicle: a case study. *Adv Vib Eng* 5(3):201–205
13. Kovalev R, Sakalo A, Yazykov V, Shamdani A, Bowey R, Wakeing C (2016) Simulation of longitudinal dynamics of a freight train operating through a car dumper. *Veh Syst Dyn* 54:707–722
14. Sharma SK, Kumar A (2018) Ride comfort of a higher speed rail vehicle using a magnetorheological suspension system. *Proc Inst Mech Eng Part K J Multi-body Dyn* 232(1):32–48. <https://doi.org/10.1177/1464419317706873>
15. Chandra K (2002) Maintenance manual for BG coaches of LHB design. Ministry of Railways, India
16. Swaroop A (2011) A technical note in continuation of reasoned document of EOI of AAR 'H'type tight-lock couplers having balanced type draft gear. Research Designs and Standards Organisation, Lucknow
17. Sharma SK, Kumar A (2018) Disturbance rejection and force-tracking controller of nonlinear lateral vibrations in passenger rail vehicle using magnetorheological fluid damper. *J Intell Mater Syst Struct* 29(2):279–297. <https://doi.org/10.1177/1045389X17721051>
18. Palli S, Koona R, Sharma SK, Sharma RC (2018) Advances in dynamic analysis of rail vehicle coach. *Int J Veh Struct Syst* 10:3
19. Sharma SK, Sharma RC, Palli S (2018) Rail vehicle modeling and simulation using Lagrangian method. *Int J Veh Struct Syst* 10:3
20. Sharma SK, Sharma RC (2018) Simulation of quarter-car model with magnetorheological dampers for improving ride quality. *Int J Veh Struct Syst* 10:3
21. Jain MK (2013) Train, grade, curve and acceleration resistance, rail electric. <https://www.railelectrica.com/traction-mechanics/train-grade-curve-and-acceleration-resistance-2/>. Accessed 15 July 2015
22. 12951/Mumbai Central-New Delhi Rajdhani Express (2015). <http://indiarailinfo.com/>. Accessed 25 Aug 2015
23. Indian locomotive class WAP-5 (2015) Wikipedia. https://en.wikipedia.org/wiki/Indian_locomotive_class_WAP-5/. Accessed 06 Aug 2015
24. Sharma SK, Kumar A (2016) The impact of a rigid-flexible system on the ride quality of passenger Bogies using a flexible carbody. In: Pombo J (ed) Proceedings of the third international conference on railway technology: research development and maintenance Stirlingshire UK 5–8 April 2016 Cagliari Sardinia Italy. Civil-Comp Press, Stirlingshire, p 87. <https://doi.org/10.4203/ccp.110.87>
25. WAP-5 and WAP-7 (2001) Tractive effort/Braking effort in Technical Data and Notes, Indian Railways, India. <http://www.irfca.org/docs/index.html#tech>. Accessed 12 Aug 2015
26. (2001) Tractive effort/Braking effort vs. Speed, Research Designs and Standards Organisation, Lucknow, India
27. Sharma SK, Kumar A (2014) A comparative study of Indian and Worldwide railways. *Int J Mech Eng Robot Res* 1(1):114–120
28. Kumar H, Sujatha C (2010) Ride comfort analysis of a railway vehicle moving on straight track and subjected to average vertical profile (AVP) excitation. *Adv Vib Eng* 9(2):119–130