MULTIBODY DYNAMIC STABILITY ANALYSIS OF A DIESEL-HYDRAULIC LOCOMOTIVE

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Abstract

In the development stage of a rail vehicle, analyses to evaluate its dynamic stability are required. In this work, a newly designed Diesel-Hydraulic Locomotive is modelled as a multibody system consisting of several rigid bodies interconnected by elastic elements. Multibody dynamic analysis of the system is performed to obtain the dynamic response and stability evaluation. Stability on tangent and curve tracks as well as the Locomotive dynamic response is investigated. The stability on tangent track is limited by the locomotive critical speed, V_{cr}, evaluated for various wheel conicity and primary suspension stiffness. Operation beyond this critical speed will result in hunting which could lead to wheel-climb. Stability evaluation on the curve track is conducted through simulation of the model negotiating a curve with rail irregularity for various radii. The maximum and minimum velocities for negotiating the curve are evaluated. To evaluate the derailment safety on the curve track, the wheel-rail contact force ratio in lateral and vertical directions (L/V), and the loading-unloading ratio of the primary suspension in the vertical direction are computed, and are compared to limiting criteria. The results are found to meet the safety criteria. The guiding lateral force on the wheel entering a curve track for various primary suspension stiffnesses is also evaluated because its effect on wear rate of the wheel and rail. While lower stiffness value of the primary suspension results in favourable L/V and lower guiding force, it yields lower critical speed on tangent track. Hence, a parametric study of the primary suspension stiffness is conducted to obtain optimum value which yield acceptable critical speed and guiding force, yet still meet the safety criteria.

Keywords: *simulation of rail vehicle, locomotive, dynamic stability, critical speed, derailment*

1. Introduction

With increasing population and mobility in Indonesia, rail vehicle mass transportation should be considered as an effective solution to increasingly congested traffic. While the problem should be addressed comprehensively, including increasing national track length, the availability of train sets should play important role in alleviating the problem. To that end, the Indonesian Rollingstock Industry is developing a Diesel-Hydraulic Locomotive. This type of locomotive is deemed suitable for Indonesian railtrack condition.

The performances of railway vehicles are usually defined by safety and productivity. Safety relating to the vehicle-track system is usually associated with derailment. Performance indices employed are vehicle stability both in tangent- and curve-tracks as well as ride quality. While productivity is measured in terms of operating and maintenance costs that are affected by operating speed and wheel-rail forces, which influence the wear of vehicle components and degradation of track structure [1].

Above parameters could be obtained via dynamic analysis of the rail vehicle. Rail vehicle may be modelled as a multibody system to investigate its dynamic characteristics [2, 3]. Dynamic failure means wheel climb or derailment [4]. Knothe and Bohm [5] investigated hunting motion on tangent track resulting in instability. Wickens [6] discussed the effect of wheel-rail contact geometry on dynamic stability. Elastic wheel-rail contact model is employed to obtain a more accurate simulation results.

Stability analysis on curve tracks is more complex. Mohammadzadeh [4] proposed a new method to evaluate the derailment on curve track. Many researchers, including Ishida [7] used ratio of lateral to vertical force on a wheel (L/V), also known as Nadal criterion and loading-unloading ratio as a stability parameter on curve track in dynamic analysis of Shinkansen train. UIC 518 requires total lateral force as one of the parameters for assessing the operational safety [8]. Based on various findings, Gilchrist [9] concluded that higher stiffness of primary suspension and lower wheel conicity tends to improve lateral stability on tangent track, but would adversely affect the stability on curve track.

Research on rail vehicle stability in Indonesia was conducted during the development of several rolling stocks. Among others, Mahyuddin et al. [10] analyzed the stability and ride index of a railcar with NT-60 bogies. Wibisono [11] designed railcar for a speed of 120 km/h. Mahyuddin et al. [12] compare the dynamic performance of railcar on NT-60 and NT-11 bogies, including critical speed, ride index and wear rate. Effect of variation of suspension parameters on a bolsterless bogie dynamic performance was investigated [13]. However, no Indonesian study on locomotive dynamic stability has been conducted.

In this work, the dynamic performance of the designed Diesel-Hydraulic Locomotive is evaluated by using a multibody model of the Locomotive. The performance on tangent track is limited by the locomotive critical speed, Vcr, above which speed hunting motion become unstable. This critical speed is evaluated for various wheel conicity and primary suspension stiffness. Stability evaluation on the curve track is conducted through simulation of the model negotiating a curve with rail irregularity for various radii. The maximum and minimum velocities for negotiating the curve are evaluated. To evaluate the derailment safety on the curve track, the wheel-rail contact force ratio in lateral and vertical directions (L/V), or Nadal criterion, and the loading-unloading ratio of the primary suspension in the vertical direction are computed, and are compared to limiting criteria. The guiding lateral force on the wheel entering a curve track for various primary suspension stiffnesses and wheel conicity is also evaluated because of its influence on wear rate of the wheel and rail. Parametric study of the primary suspension stiffness is conducted to obtain optimum value which yield acceptable critical speed and guiding force, yet still meet the safety criteria.

Analyses were performed with help of application software, Universal Mechanism 4.0 (UM), developed by Pogorelov [14]. This UM software consists of two modules, i.e. UM Input and UM Simulation. Modelling is performed with UM Input where bodies are inputted along with interconnection. The analysis on the multibody system is carried out by UM Simulation which could yield motion, forces on interconnection, as well as stability parameters of the Locomotive.

2. Diesel-Hydraulic Locomotive

The D-H Locomotive is designed for the Indonesian narrow-gage track with a maximum velocity of 120 km/h, could negotiate a 140 m radius curve safely, and a ride-index less than 2.5 at its operational speed. The structure consists of wheelsets, bogies and carbody interconnected by suspension system as shown in Fig. 1. The DH Locomotive is modelled as a multibody system to analyze its dynamic characteristics.

In this section the multibody model of the locomotive, wheel-rail contact effect, and suspension characteristics are discussed. The locomotive dynamic behaviour is investigated for various primary suspension system stiffness values.

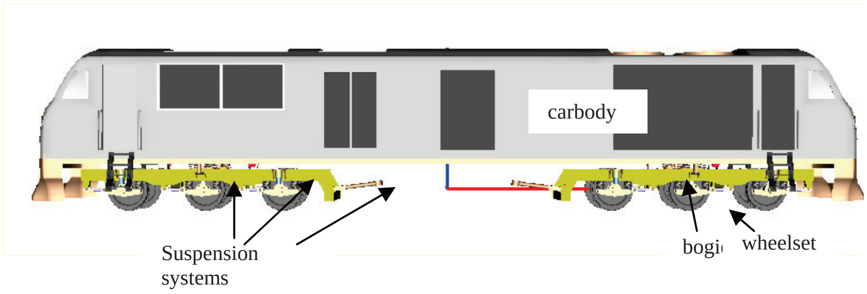


Fig. 1. DH Locomotive

2.1. Multibody model of DH locomotive

The multibody model of the DH locomotive is shown in Fig. 2. The system consists of a carbody, two bogies and 6 wheelsets for a total of 9 rigid bodies that are interconnected by suspension systems consisting of springs and dampers.

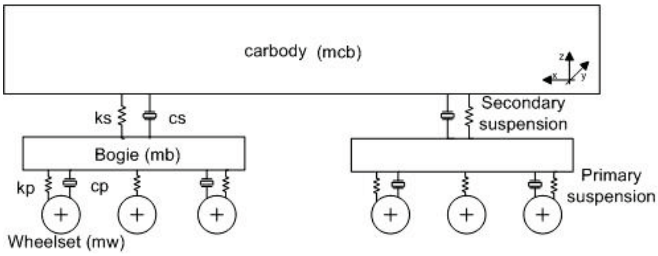


Fig. 2. Multibody model of DH Locomotive

Universal Mechanism (UM4.0) is employed to construct the multibody model by defining the bodies and interconnection as well as the wheel-rail contact model as input [15]. The following assumptions are taken: springs and dampers are linears; each body is rigid leading to a lumped mass model. Points of attachment or contact of an interconnection on a rigid body, known as a node, is located in terms local (body-fixed) coordinate system. The local reference point, usually the centre of mass of a body, are defined in term of moving global reference frame coordinate attached to the rail. The multibody model degrees-of-freedom are summarized in Tab. 1 as possible translational and rotational motions for each body. Constraints are enforced through interconnections and boundary conditions. Note that the motions in the longitudinal direction are relative with respect to the global reference frame. The mass and inertia of the DH Locomotive are obtained from the design and presented in Tab. 2.

Tab. 1. Multibody model degrees-of-freedom

No	Body	Type of Motion (degree-of-freedom)					
		Longitudinal	Lateral	Vertical	Roll	Pitch	Yaw
1	Wheelset (*i = 1,2, … ,6*)	-	Y_{Wi}	Z_{Wi}	ϕ_{Wi}	-	Ψ_{Wi}
2	Bogie (*j=1,2*)	X_{Bj}	Y_{Bj}	Z_{Bj}	ϕ_{Bj}	θ_{Bj}	Ψ_{Bj}
3	Carbody	X_C	Y_C	Z_C	ϕ_C	θ_C	Ψ_C

Tab. 2. Mass and Inertia

No	Body	Mass (kg)	Inertia (kgm^2)		
			I_{xx}	I_{yy}	I_{zz}
1	Wheelset	2.1×10^3	1.0×10^3	60	1.0×10^3
2	Bogie	4.0×10^3	1.83×10^3	1.07×10^4	9.4×10^3
3	Carbody	6.94×10^4	2.0×10^5	2.5×10^6	2.5×10^6

Rail irregularity acts as the source of based-excitation at the wheel-rail contact. The suspension systems isolate the carbody from the vibration excitation and maintain the wheelset on the rail. The excitation from the wheelset to the bogie is dampened by primary suspension, and from the bogie to the carbody by secondary suspension. The primary suspension system consists of 24 primary springs, 8 vertical dampers, and 12 trailing arms. The secondary suspension system is made of 8 secondary springs, 4 vertical and lateral dampers and two traction rods.

The designed stiffness and damping parameters for the DH Locomotive are as shown in Tab. 3. The design assumes that the lateral stiffness of the primary and secondary springs is one-half of its axial stiffness.

Tab. 3. Suspension stiffness and damping constants

No	Component	Stiffness (N/m)		
		x	y	Z
1	Primary suspension, k_p	5.0×10^5	5.0×10^5	1.0×10^6
2	Trailing arm, k_{ta}	9.81×10^6	6.38×10^6	9.81×10^6
3	Secondary suspension, k_s	5.0×10^5	5.0×10^5	1.0×10^6
4	Traction rod, k_{tr}	1.1×10^7	$1,1 \times 10^7$	5.0×10^6
	Damper	Damping (Ns/m)		
5	Primary suspension, C_p	-	-	6.0×10^4
6	Trailing arm, C_{ta}	1.5×10^4	5000	2.0×10^4
7	Secondary vertical suspension, C_{sv}	-	-	5.0×10^4
8	Secondary lateral suspension, C_{sl}	-	6.5×10^4	-

The suspension systems components act as interconnection attached to nodes at connecting bodies. The model shown in Fig. 1 has 62 interconnections. As an example, in Fig. 3, 24 nodes of a bogie is shown, and nodes 1-4 and 5-8 (shown in insert) are the connecting points for the wheelset shown in Fig. 4.

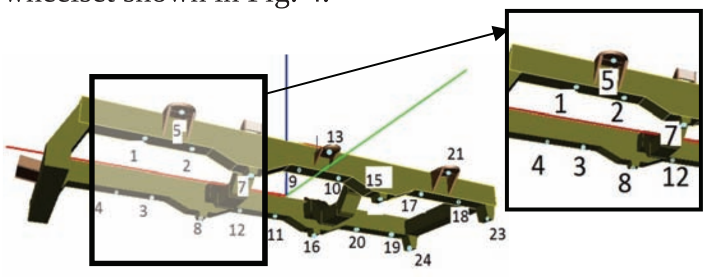


Fig. 3. Bogie nodes

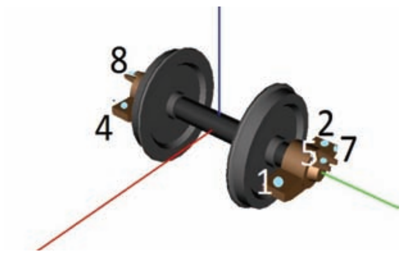


Fig. 4. Wheelset nodes

The data in Tab. 2 and 3 as well as the geometry and interconnection types are used as input to UM Input that constructs the multibody model and its governing equations of motion.

2.2. Wheel-Rail contact model

Simplified method is employed to evaluate dynamic behaviour due to wheel/rail wear level, while the creep force is evaluated by Muller method. The parameters are wheel conicity and equivalent angle of contact [15]. The contact force is modelled as friction independent of wheel spin. Wheel equivalent conicity is determined by the rolling radius difference due to lateral wheelset displacement. Hence, for worn wheel the equivalent conicity increases [16].

3. Dynamic stability analysis

The dynamic analysis of the locomotive at tangent and curve tracks are conducted for values of primary suspension axial stiffness, k_{pz}, in the range of 0.5 – 1.5 kN/mm.

3.1. Stability on tangent track

The stability on the tangent track is evaluated for locomotive model travelling on tangent track with a sinusoidal wave lateral irregularity at constant speed, followed by smooth track to evaluate the suspension ability to suppress hunting motion [2]. Lateral motion of the wheelset is observed for hunting motion. Speed is increased at 2 km/h interval until critical speed, V_{cr}, is reached as indicated by the absence of damping on the hunting motion. Effect of wheel conicity is also investigated, from new wheel with a conicity of 0.05 up to worn wheel with a conicity of 0.35. The results are tabulated in Tab. 4 for various combinations of primary suspension stiffness and conicity. The shaded areas are values of critical speed below that of the design speed of 140 km/h.

Tab. 4. Critical speed for various primary suspension stiffness and wheel conicity

	Primary suspension stiffness, k_{pz} (kN/mm)				
Conicity	0.50	0.75	1.00	1.25	1.50
0.05	262 km/h	288 km/h	312 km/h	336 km/h	358 km/h
0.125	158 km/h	176 km/h	190 km/h	208 km/h	220 km/h
0.2	124 km/h	138 km/h	148 km/h	158 km/h	170 km/h
0.275	104 km/h	114 km/h	126 km/h	130 km/h	144 km/h
0.35	92 km/h	104 km/h	112 km/h	118 km/h	128 km/h

It may be seen that the critical speed, V_{cr}, decrease with the increase of conicity and increase with increase suspension stiffness. For k_{pz} = 1.0 kN/mm, for conicity up to 0.275, the critical speed is higher than the operational speed of 120 km/h. But, for conicity of 0.275 the critical speed of 126 km/h is lower than the design speed of 140 km/h.

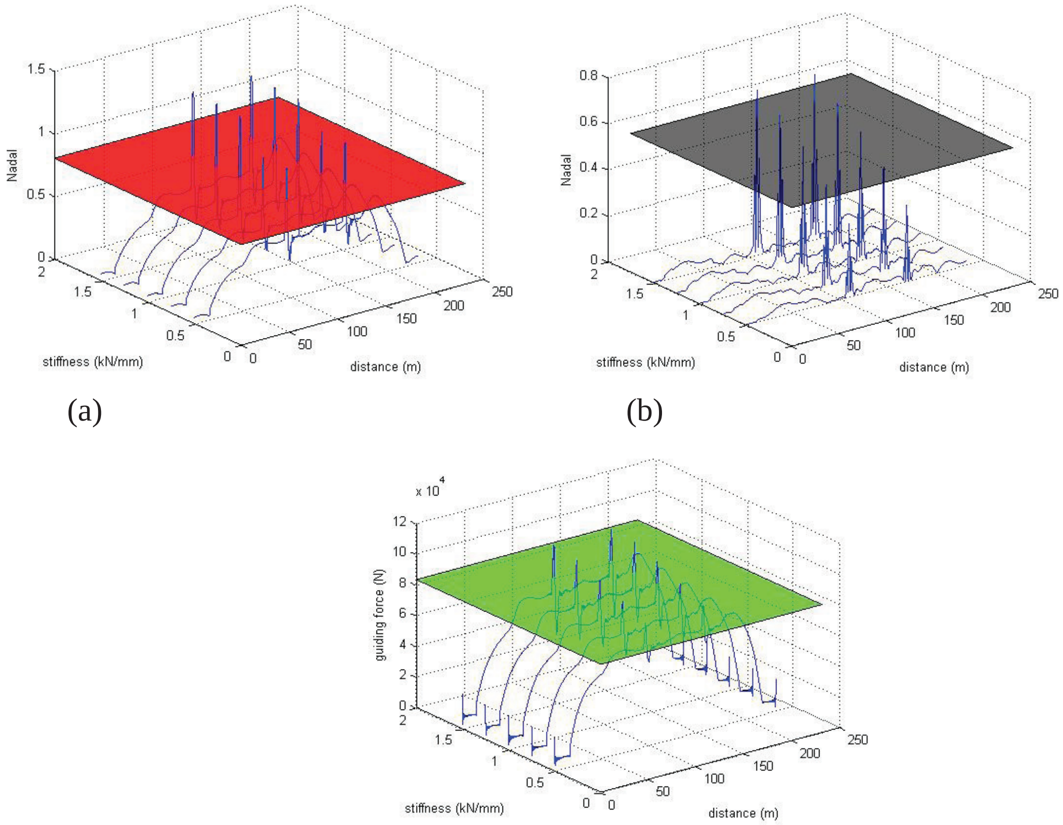
3.2. Stability and wear on curve track

For stability analysis on curve track, the model is evaluated for track radii, R = 140, 300, and 600 m, where R = 140 m is the smallest radius for Indonesian rail tracks. The curve track start with a 10 m tangent track, followed by 50 m transition, before reaching steady curve with a constant radius.

When vehicle negotiates curve track, it experiences significant lateral force due to the wheel-rail interaction force arises from the kinematics of the profiled wheels and imbalance between gravitational and centrifugal forces. To avoid excessive lateral movement, the outer rail of curve track is raised, or super elevated. Super elevation of the rail is based on the Indonesia Railways regulation, and in this study, a maximum value of 100 mm is taken. In addition, outside track has irregularity in the form of vertical-V with a gradient of 1:88 to introduce twist. The positions of the twists (vertical-V) at steady curve are at 100 m and 160 m from the starting point.

Wheel conicity of 0.2, representing a worn wheel at operational condition, is selected. Simulation is conducted for various curve negotiating speeds and the stability parameters are checked against the safety criteria, i.e. Nadal criterion, loading-unloading ratio. The guiding force indicating the sum of lateral force at the wheel-rail contact for an axle is also evaluated. Nadal criterion represents the safety with regard to wheel-climb. Loading-unloading is the ratio of vertical dynamic force with respect to static forces on the primary suspension. Guiding force is the lateral force at the wheel-rail contact and represents the resistance occurring as the wheelset negotiate a curve. Higher guiding force means larger resistance during curve negotiating and results in higher wheel wear rate. Nadal criterion and guiding force is observed for the front wheelset, where maximum values occur. The loading-unlading ratio is based on the front wheelset of the rear bogie, where maximum value occurs.

The limit for Nadal criterion is 0.8, and the guiding force is 83 kN [8], while ratio of loading-unloading should be less than 0.6 [9]. Additionally, Nadal value above 0.8 could be tolerated as long as its duration is not more than 0.3 s [1]. The guiding force is filtered for high frequency content with a 10 Hz low-pass filter [8]. As an example, Fig. 5 presents the values of (a) Nadal, (b) loading-unloading ratio, and (c) guiding force for locomotive model, as it negotiates curve track with R = 140 m, at a speed of 40 km/h for values of k_{pz} = 0.5, 0.75, 1.0, 1.25, and 1.5 kN/mm. The values are presented along distance-axis where increases in parameters values are observed at the locations of twists. The planes represent the safe limit. Thus, a Nadal curve that intersects a plane indicates unsafe operation. By observing Fig. 5, safe curve negotiating speeds where there are no values above the limits, may be obtained.



(a) (b)

(c)

Fig. 5. (a) Nadal (b) Loading-unloading (c) Guiding force

As an illustration of stiffness value variation, the average and maximum values of dynamic performance criteria of the simulation are presented in Tab. 5.

Tab. 5. Parameters values at R = 140 m; V = 40 km/h

Criteria		Primary suspension stiffness, k_{pz} (kN/mm)				
		0.5	0.75	1.0	1.25	1.5
Nadal	*Average*	0.38	0.39	0.41	0.43	0.44
	Maximum	0.83	1.12	1.28	1.29	1.30
Loading-unloading	*Average*	0.031	0.043	0.049	0.056	0.064
	Maximum	0.33	0.47	0.57	0.64	0.71
Guiding force (kN)	*Average*	52.4	53.9	55.4	56.90	58.30
	Maximum	82.7	88.6	96.6	102.1	107.0

It may be seen that while the average values meet the criteria limits, the maximum values are much larger and in most cases are beyond the allowable limit. Evaluation for safe speeds should be conducted based on the values of the three criteria. As an example, in Fig. 5(a), the Nadal maximum value for k_{pz} = 0.5 kN/mm, is higher than 0.8 for a distance of 2.5 m, corresponding to time less than 0.3 s. Hence, the speed of 40 km/h for k_{pz} = 0.5 kN/mm, is considered safe because the loading-unloading and guiding-force maximum values are below the limit values.

Similar evaluations are conducted for curve tracks with R = 300 and 600 m. The results are summarized as safe speeds (in km/h) in Tabs. 6 – 8, for R = 140, 300 and 600 m, respectively. The unsafe speeds, for various primary suspension stiffness values, are indicated by the shaded areas.

Tab. 6. Safe speed for a radius 140 m curve track

Speed (km/h)	Primary suspension stiffness (kN/mm)				
	0.50	0.75	1.00	1.25	1.50
20	unsafe	Unsafe	unsafe	unsafe	Unsafe
30	unsafe	Unsafe	unsafe	unsafe	Unsafe
40	Safe	unsafe	unsafe	unsafe	Unsafe
50	Safe	Safe	safe	unsafe	Unsafe
60	unsafe	Unsafe	unsafe	unsafe	Unsafe

Tab. 7. Safe speed for a radius 300 m curve track

Speed (km/h)	Primary suspension stiffness (kN/mm)				
	0.50	0.75	1.00	1.25	1.50
30	Unsafe	unsafe	Unsafe	unsafe	Unsafe
40	Unsafe	Unsafe	Unsafe	unsafe	Unsafe
50	Safe	safe	Unsafe	unsafe	Unsafe
60	Safe	safe	Safe	unsafe	unsafe
70	Safe	safe	Safe	safe	Unsafe
80	Safe	safe	Unsafe	unsafe	Unsafe
90	Unsafe	unsafe	Unsafe	unsafe	Unsafe

Tab. 8. Safe speed for a radius 600 m curve track

Speed (km/h)	Primary suspension stiffness (kN/mm)				
	0.50	0.75	1.00	1.25	1.50
50	safe	unsafe	unsafe	unsafe	Unsafe
60	safe	Safe	unsafe	unsafe	Unsafe
70	safe	Safe	unsafe	unsafe	Unsafe
80	safe	Safe	Safe	unsafe	Unsafe
90	safe	Safe	Safe	unsafe	Unsafe
100	safe	Safe	unsafe	unsafe	Unsafe
110	safe	unsafe	unsafe	unsafe	Unsafe
120	unsafe	unsafe	unsafe	unsafe	Unsafe

The simulation results on curve tracks show that higher stiffness of primary suspension result in narrower spectrum of safe speeds. This is due to higher Nadal value, loading-unloading ratio, and guiding force for a given speed. As expected, larger radius yields lower parameters values. At lower speeds, Nadal criterion and guiding force tend to be larger than the safe limit criteria. The safe curve negotiating speeds have lower and upper bounds. Super elevation affects the safe speed spectrum since the stability on curve track is governed by the centrifugal and gravitational forces. At low speed the locomotive may derail due to low centrifugal force that is insufficient to counter the moment created by the gravitational force. On the other hand, higher speed would results in high centrifugal force that may cause the locomotive wheel to climb the rail. Suspension stiffness affects how wheelset positioned itself with respect to the track. Higher stiffness would decrease the ability of the wheelset to move on the rail.

3. Conclusion

The dynamic characteristics of a DH Locomotive were investigated by multibody model using UM4.0 software. Based on the simulation, it was found that higher stiffness value results in higher V_{cr}, but decrease the dynamic performance and productivity on the curve track. Higher stiffness value also results in narrower range of safe curve negotiating speeds as summarized in Tabs. 6 – 8. The design stiffness of k_{pz} = 1.0 kN/mm yields a critical speeds higher than the operational speed, and results in a relatively large range of safe curve negotiating speeds. It may be concluded that k_{pz} = 1.0 kN/mm is a good compromise value, where both lateral dynamic stability and curve negotiating performance meet the design criteria.

References

[1] Dukkipati, R. V., Amyot, J. R., *Computer-Aided Simulation in Railway Dynamics*, Dekker, New York 1988.
[2] Bosso, N., Gugliotta, A., Soma, A., *Simulation of narrow gauge railway vehicles and experimental validation by mean of scaled tests on roller rig*, Meccanica, Springer Science+Business Media B.V., Vol. 43, pp 211-223, 2008.
[3] Schiehlen,W., *Dynamical Analysis of Vehicle Systems*, University of Stuttgart, Lecture Courses, Springer, Wien - NewYork 2007.
[4] Mohammadzadeh, S., Sangtarashha, M., Molatefi, H., *A novel method to estimate derailment probability due to track geometric irregularities using reliability techniques and advanced simulation methods*, Arch Application Mechanics, Springer-Verlag, 2011.
[5] Knothe, K., Bohm, F., *History of Stability of Railway and Road Vehicles*, Vehicle System Dynamics, Vol. 31, pp. 283-323, Swets & Zeitlinger,1999.
[6] Wickens. A. H., *Fundamentals of Rail Vehicle Dynamics*, Swets and Zeitlinger, 2003.
[7] Ishida, H., Matsuo,M., *Safety Criteria for Evaluation of Railway Vehicle Derailment*, QR of RTRI, Vol. 40, No. 1, 1999.
[8] UIC 518, *Testing and approval of railway vehicles from the point of view of their dynamic behaviour -Safety - Track fatigue - Ride quality*, 2005.
[9] Gilchrist, A. O., *The Long Road to Solution of the Railway Hunting and Curving Problem*, Proceedings of the Institution of Mechanical Engineers, ProQuest Science Journals, Vol. 212, No. 3, pp. 219-226, 1998.
[10] Mahyuddin, A. I., Adrian, E., Suharto, D., Wibisono, I., *Analisis Stabilitas Dinamik dan Tingkat Kenyamanan Bogi NT-60 pada kereta JS-950* (in Indonesian), Proceedings of Experimental and Theoretical Mechanics, Institut Teknologi Bandung, 1997.
[11] Wibisono, I., Handoko, Y. A., *Analisis Karakteristik Dinamik Prototype Bogi untuk Peningkatan Kecepatan dengan Perangkat Lunak Medyna* (in Indonesian), Proceedings of Experimental and Theoretical Mechanics, Institut Teknologi Bandung, 1997.
[12] Mahyuddin, A. I., Komara, A. A., Suharto, D., *A Comparative Study of the Dynamic Behaviour and Performance of NT-11 and NT-60 Bogies*, ECM&SP, Proc. Institut Teknologi Bandung, 1999.
[13] Satria, I., Mahyuddin, A. I., *Analisis Kestabilan dan Respons Dinamik Kereta Penumpang dengan Bogi Bolsterless* (in Indonesian), Proceedings Experimental and Theoretical Mechanics, Institut Teknologi Bandung, , 2001.
[14] Kovalev, R., et al., *Freight car models and their computer-aided dynamic analysis, Multibody System Dynamics*, Springer Science+Business Media, Vol. 22, pp. 399-423, 2009.
[15] Universal Mechanism, *Users' Manual*, Bryansk State Technical University, Russia 2009.
[16] Polach, O., *Wheel profile design for target conicity and wide tread wear spreading*, Wear, Vol. 271, pp. 195 – 202, Elsevier, 2011.