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Derailment Safety Analysis of Optimum Speed of ROLA Transportation with Flat Cars (GD 400910) on the Extreme Curve of the PPI Madiun Track Using Universal Mechanism Software

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ABSTRACT

This study investigates the optimum operating speed of Rollende Landstraße (ROLA) transportation on a 42 m minimum curve radius at the PPI Madiun track. Dynamic simulations were conducted using Universal Mechanism (UM) to model the wheel–rail interaction of a GD 40 09 10 flat car carrying an Isuzu ELF NMR-L truck under full, half, and empty load conditions. Derailment risk was evaluated using the Nadal criterion (Y/Q), supported by ride index analysis. The results indicate that the maximum safe speeds are 14 km/h for full load, 17 km/h for half load, and 19 km/h for empty load. Exceeding these limits causes the Nadal value to surpass the allowable threshold duration, indicating a high potential for wheel climbing. These findings provide practical speed-limit references for safe ROLA operation on sharp curves.

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1. INTRODUCTION

The global shift toward sustainable freight transportation has increased interest in rail-based logistics solutions. One established approach is the Rollende Landstraße (ROLA) system, which transports complete road trucks on flat wagons. ROLA has been successfully implemented in several European corridors to reduce greenhouse gas emissions, road congestion, and infrastructure wear while maintaining operational flexibility.

Despite these advantages, ROLA operation generates complex dynamic interactions between the flat car, bogie, and transported truck, particularly on curved tracks. These interactions increase lateral wheel–rail forces and raise the risk of derailment, especially on sharp curves with small radii. Derailment on curves is commonly associated with wheel climbing phenomena, which are effectively evaluated using the Nadal criterion (Y/Q or L/V ratio), a widely accepted derailment safety indicator in railway engineering.

Previous studies employing multibody dynamic simulation tools, such as Universal Mechanism, have demonstrated the effectiveness of numerical modeling in assessing derailment risk and ride quality. However, most existing research focuses on standard railway infrastructure with curve radii greater than 250 m, while studies addressing extremely tight curves remain limited.

In many developing railway systems, curves with radii below 100 m are still common in depots and terminal areas. There is a lack of research evaluating ROLA operation on minimum-radius curves using time-based Nadal criteria under different truck loading conditions. In Indonesia, studies examining ROLA safety on a 42 m radius curve are currently unavailable.

To address this gap, this study investigates the dynamic behavior of a ROLA system using a GD 40 09 10 flat car transporting an Isuzu ELF NMR-L truck. Multibody simulations are conducted to analyze lateral forces, Nadal criteria, and ride index values for fully loaded, half-loaded, and unloaded conditions, with the objective of determining the maximum safe operating speed on a 42 m radius curve.

2. RESEARCH METHOD

2.1 Flowchart

The flow diagram that can be obtained from this study is as follows.

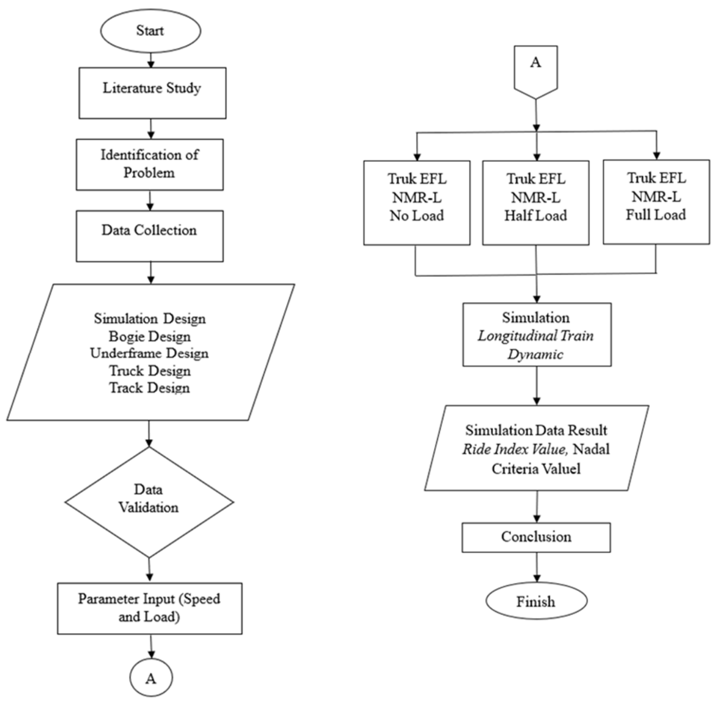

Figure 1. Flowchart

2.2 Data Collection Methods

Data collection methods are used to obtain research information, answer questions, test hypotheses, and evaluate results. Data is divided into two main types, namely primary data and secondary data.

Primary data is obtained directly through measurement and observation. The primary data processed includes lateral and vertical force values, spring stiffness matrix, qriteria values, and ride index values.

Secondary data is data from other parties that serves as a reference and supplement to primary data to support research, including *bogie frame* dimensions*,* Isuzu ELF NMR-L truck dimensions, *wheelset* profiles and sizes*,* empty weight and *rolling stock* dimensions*,* rail road profiles, rail road geometry sizes, and rail road macrogeometry sizes.

2.3 Nadal Criteria (L/V or Y/Q)

The Nadal Criteria are a very important safety evaluation method in the world of railways to prevent *derail*ment. The Nadal Criteria analyze the wheel-rail interaction on curves to determine the safe limits of lateral forces, prevent wheel climbing, and improve train stability and safety. Contact occurs between the wheel and the rail, creating an angle γ with the track surface.

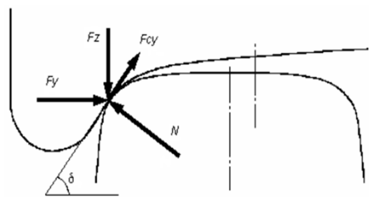

Figure 2. Contact Force on the Wheel

Therefore, the following formula is obtained:

$$\frac{Y}{Q} = q(\delta, \mu y) = \frac{tan\delta - \mu}{1 + \mu_y \tan\delta}$$

The Nadal criterion limit is 1.2, but for curve radii >250 m, it is set at 0.8 and must be maintained for 0.3 seconds to prevent wheel derailment.

2.4. Model Assumption and Limitation

. To simplify the dynamic analysis and focus on the dominant wheel–rail interaction mechanisms, several assumptions and limitations were applied in the ROLA model developed using Universal Mechanism software.

2.4.1 Model Assumption

The flat car, bogie components, wheelsets, and Isuzu ELF NMR-L truck were modeled as rigid bodies. Structural flexibility of the carbody, bogie frame, rail, and truck chassis was neglected, while elastic behavior was represented solely by suspension springs and friction elements.

The track was assumed to have ideal geometry, consisting of a constant curve radius of 42 m with uniform gauge and a rail inclination of 1:40. Track transition curves, vertical gradients, superelevation variation, and rail irregularities were not included. Micro-scale defects such as rail corrugation, surface roughness, and alignment imperfections were also neglected.

Wheel and rail profiles were assumed to be nominal and unworn, with a constant wheel–rail friction coefficient throughout the simulation. Vehicle operation was simulated under constant-speed conditions, without considering traction, braking, or longitudinal dynamic effects. The truck load was assumed to be rigidly secured to the flat car, with no internal load redistribution during motion.

2.4.2 Limitation

Due to the rigid-body assumption and ideal track conditions, the model may underestimate dynamic responses under real operational conditions where structural flexibility, track irregularities, and variable friction are present. Furthermore, the analysis was limited to a single minimum curve radius (42 m), and the results cannot be directly generalized to other track geometries. The absence of full-scale experimental validation also limits direct quantitative comparison with in-service performance.

Nevertheless, the model is considered adequate for evaluating relative derailment risk and determining safe operating speed limits based on the Nadal criterion under controlled conditions.

3. RESULTS AND DISCUSSION

3.1 Design Validation

Curving Bogies Formula Calculation

The initial step in calculating the resultant force between the truck and flat car during operation is zero, a condition known as dynamic equilibrium. The calculation is presented below.

ROLA with ELF NMR-L Truck Load at Full Capacity

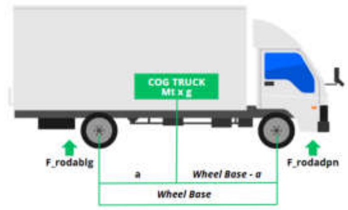

Figure 3. -*Center of Gravity*

a. Calculating the vertical force on the truck assuming equilibrium conditions

- Equilibrium condition $\sum \boldsymbol{Fy} = \mathbf{0}$

$F_{rodablkg} + F_{rodadpn} = M_{truk}\ x\ g$ ....(a)

- $\sum M_{rodadpn} = 0$

$$M_{truk}\ x\ g\ x\ (Wheelbase\ truck - a) = F_{rodablkg}\ x\ Wheelbase\ truck$$

$$F_{rodablkg} = \frac{M_{truk}\ x\ g\ x\ (Wheelbase\ truck - a)}{Wheelbase\ truck}$$

$$F_{rodablkg} = \frac{8.250\ \ x\ 9{,}81\ x\ (4.175 - 1.316)}{4.175}$$

$F_{rodablkg}$ = 55,421.80 N..........(b)

- Substitute equation (b) into (a)

$$F_{rodablkg} + F_{rodadpn} = M_{truk}\ x\ g$$

$$F_{rodadpn} = (M_{truk}\ x\ g) - F_{rodablkg}$$

$$F_{rodadpn} = (15.000\ \ x\ 9{,}81\ ) - 55.421{,}80$$

$F_{rodadpn}$ = 25,510.70 N

b. Calculate the vertical force on a flat car assuming equilibrium conditions vertical force on a flat car assuming equilibrium conditions

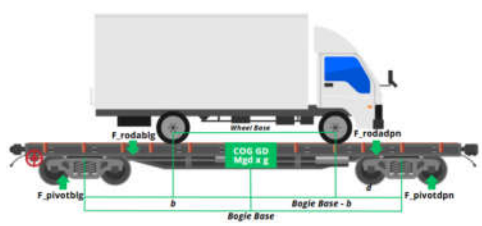

Figure 4. Center Of Gravity ROLA

- Equilibrium condition $\sum Fy = 0$

$$F_{pivotblkg} + F_{pivotdpn} = F_{rodablkg} + F_{rodadpn} + M_{gerbongdatar} \cdot g$$

- $\sum M_{rodadpn} = 0$

$$\left(F_{rodablkg} \cdot c\right) + \left(M_{gerbongdatar} \cdot g \cdot (Bogie\ Base - b)\right) + \left(F_{rodadpn} \cdot d\right) = F_{pivotblkg} \cdot Bogie\ Base$$

$$F_{pivotblkg} = \frac{\left(F_{rodablkg} \cdot c\right) + \left(M_{gerbongdatar} \cdot g \cdot (Bogie\ Base - b)\right) + \left(F_{rodadpn} \cdot d\right)}{Bogie\ Base}$$

$$= \frac{(55.421{,}8006 \cdot\ 4.175) + \left(15.000 \cdot 9{,}81 \cdot (9{,}8 - 4{,}3435)\right) + \ (15.510{,}6994 \cdot\ 0{,}931)}{9{,}8}$$

= 107,965.35 N

- Substitute equation (b) into (a)

$$F_{pivotblkg} + F_{pivotdpn} = F_{rodablkg} + F_{rodadpn} + (M_{gerbongdatar} \cdot g)$$

$$F_{pivotdpn} = F_{rodablkg} + F_{rodadpn} + (M_{gerbongdatar} \cdot g) - F_{pivotblkg}$$

$$F_{pivotdpn} = 55.421{,}8006 + 25.510{,}6994 + (15.000 \cdot 9{,}81) - 107.965{,}3524$$

$$F_{pivotdpn} = 120.117{,}15\ N$$

a. *Calculate the vertical force on each bogie received by the four wheels of the flat car.*

- Calculate the load on the front bogie axle (Q1l, Q1r, Q2l, and Q2r)

$F_{bogiedpn} = (M_{bogie} . g) + F_{pivotdpn}$
$F_{bogiedpn} = (6.630 .\ 9{,}81) + 120.117{,}15$
$F_{bogiedpn} = 185{,}157.45$ N
$Q = \frac{F_{bogieblkg}}{Jumlah\ wheel}$
$Q = \frac{185.157{,}45}{4}$
$Q = 46{,}289.36$ N / 4.72 tons

- Calculating the load on the rear bogie axles (Q3l, Q3r, Q4l, and Q4r)

$F_{bogieblkg} = (M_{bogie} . g) + F_{pivotblkg}$
$F_{bogieblkg} = (6.630 .\ 9{,}81) + 107.965{,}35$
$F_{bogieblkg} = 173{,}005.65$ N
$Q = \frac{F_{bogieblkg}}{Jumlah\ wheel}$
$Q = \frac{173.005{,}65}{4}$
$Q = 43{,}251.41$ N / 4.41 tons

- Calculating the load on the rear bogie axle and front bogie axle (Q3l, Q3r, Q2l, and Q2r)

$F_{gandartengah} = \frac{F_{bogiedpn} + F_{bogieblkg}}{2}$
$F_{gandartengah} = \frac{46.289{,}36 + 43.251{,}41}{2}$
$F_{gandartengah} = 44{,}770.39$ N / 4.57 tons

The study validated the design by comparing the results of the Universal Mechanism simulation and manual calculations to ensure that the lateral force of the left front wheel was in accordance with formula [9].

$$H_2 - H_1 = T_{\eta 2L} + T_{\eta 2R} - T_{\eta 1L} - T_{\eta 1R} - (Q_{1L} + Q_{1R} + Q_{2L} + Q_{2R}) \frac{ay}{g}$$

1) Calculating the total lateral force values for the right and left rear wheels.

$T_{\eta 3L} = T_{\eta 3R} = \mu Q_{3L} . \cos \xi_{3L}$
$\cos \xi_{3L} = \frac{2b - l_{RM}}{q_{3L}}$
$q_{3L} = \sqrt{0{,}565^2 + 3{,}90152623^2}$
$q_{3L} = 3{,}94$ ton
$\cos \xi_{3L} = \frac{8{,}1326 - 3{,}94}{3{,}94} = 0{,}99\ ton$

$$T_{\eta 3L} = 0{,}36\ x\ 4{,}565195\ x\ 0{,}99 = 1{,}627\ ton$$

2) Calculate the total lateral force on the right and left front wheels.

$\cos \xi_{2L} = \frac{l_{RM}}{q_{3L}}$
$q_{2L} = \sqrt{0{,}565^2 + 4{,}22^2}$
$q_{2L} = 4{,}26\ ton$
$\cos \xi_{2L} = \frac{4{,}22}{4{,}26} = 0{,}99\ ton$
$T_{\eta 2L} = 0{,}36\ x\ 4{,}57\ x\ 0{,}99 = 1{,}629\ ton$

3) Calculate the vertical force and centripetal force.

$(Q_{2L}+Q_{2R}+Q_{3L}+\ Q_{3R})=4{,}57x\ 4=18{,}28\ ton$

$\frac{a_y}{g}=\frac{V^2/R}{g}$

$\frac{a_y}{g}=\frac{1{,}3889^2/42}{9{,}81}=0.00469$

4) Calculating the total lateral force value.

$H_3-H_2=\ T_{\eta 3L}+T_{\eta 3R}-T_{\eta 2L}-T_{\eta 2R}\ -(Q_{2L}+Q_{2R}+Q_{3L}+Q_{3R})\frac{a_y}{g}$

$H_3-H_2=\ 1{,}627+1{,}627-1{,}629-1{,}629-(18{,}28\ )\ x\ 0{,}00469$

$H_3-H_2=\ 2{,}84\ ton\ (V\ 5\ km/jam)$

$H_3-H_2=\ 3{,}10\ ton\ (V\ 10\ km/jam)$

$H_3-H_2=\ 3{,}54\ ton\ (V\ 15\ km/jam)$

$H_3-H_2=\ 4{,}16\ ton\ (V\ 20\ km/jam)$

$H_3-H_2=\ 4{,}95\ ton\ (V\ 25\ km/jam)$

$H_3-H_2=\ 5{,}93\ ton\ (V\ 30\ km/jam)$

$H_3-H_2=\ 7{,}08\ ton\ (V\ 35\ km/jam)$

$H_3-H_2=\ 8{,}40\ ton\ (V\ 40\ km/jam)$

Prud'Homme Analysis is performed to determine the rail's resistance limit to lateral force and estimate the potential for derailment due to rail and bearing displacement.

$H_p=10kN+\frac{V}{3}$

$H_p=10kN+\left(\frac{11{,}625}{3}\right)x\ 9.81$

$H_p=48{,}01\ kN$ (Flat car with a fully loaded Isuzu ELF NMR-L truck)

The UM simulation produced a lateral force of 28,337.7 kgf at a radius of 42 m with a full load of ELF NMR-L trucks, validated by the Curving Bogies Formula calculation of 28,351.77 kgf (±2.8 tons) with the error difference calculated from the formula.

$\%error=\frac{Hasil\ Simulasi-Hasil\ Analitik}{Hasil\ Simulasi}x\ 100\%$

$\%error=\frac{28.337{,}7\ -28.351{,}7682}{28.337{,}7}x\ 100\%=0{,}04962\%$

The analytical and simulation comparisons are valid with an error of <10%, according to the criteria of Maharani et al. (2023).

3.2 Nadal Criteria Simulation

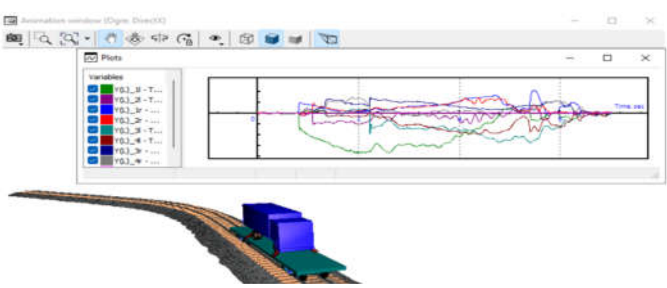

Figure 11. Nadal Criteria Simulation

3.2.1 Dynamic ROLA test with a full-loaded ELF truck

Table 1. Maximum Nadal Values at a Radius of 42 m with Full Truck Load

Wheel Position	Speed									
	5	10	14	15	20	30	35	40	41	42
1l	0.81	0.84	0.84	0.84	0.82	0.84	0.84	0.84	0.85	0.85
1r	0.39	0.39	0.40	0.40	0.45	0.52	0.50	0.50	0.47	0.38
2l	0.38	0.38	0.37	0.35	0.35	0.36	0.38	0.38	0.39	0.39
2r	0.38	0.38	0.33	0.28	0.32	0.37	0.60	0.60	0.64	0.65
3l	0.53	0.60	0.58	0.57	0.54	0.58	0.64	0.64	0.68	0.87
3r	0.40	0.41	0.41	0.41	0.41	0.41	0.42	0.42	0.43	0.45
4l	0.35	0.35	0.39	0.39	0.36	0.42	0.68	0.68	0.72	0.76
4r	0.45	0.44	0.43	0.41	0.37	0.35	0.55	0.55	0.58	0.67

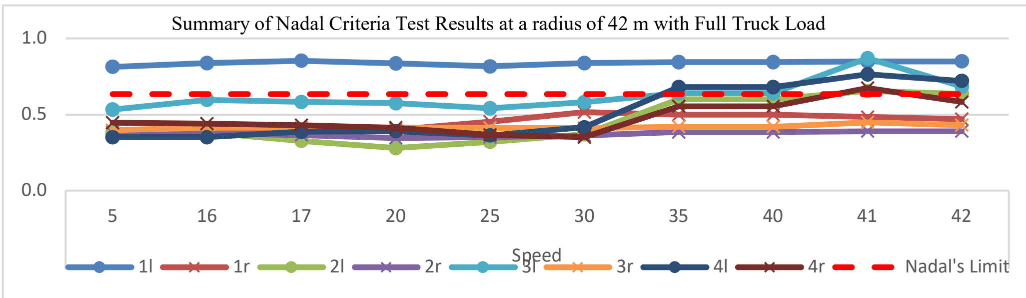

Figure 12. Graph of Nadal Limit Value Duration and Distance with Full Load

Dynamic testing of the Nadal criterion showed high Nadal values on the front left and rear right wheels due to direct contact with the rails when turning right. The test results recorded that at a low speed of 5 km/h, the Nadal value had already exceeded the threshold of 0.830. However, on tracks with a radius greater than 250 meters, the Nadal criterion value limit is less than 0.8 for 0.3 seconds. The maximum safe speed is set at 14 km/h with a Nadal value of 0.837 lasting 0.2793 seconds and a distance of 0.743 m. Derailment occurs at a speed of 42 km/h with a Nadal value of 0.850, lasting 0.694 seconds, and a distance of 1.9315 m.

3.2.2 Dynamic ROLA test with a half-loaded ELF truck

Table 2. Maximum Nadal Values for Half-Loaded Trucks

Wheel Position	Speed									
	7	10	15	16	17	18	30	40	43	44
1l	0.63	0.77	0.75	0.72	0.73	0.78	0.80	0.81	0.83	0.85
1r	0.39	0.39	0.40	0.40	0.45	0.51	0.52	0.54	0.69	0.67
2l	0.38	0.38	0.37	0.37	0.36	0.37	0.44	0.55	0.59	0.59
2r	0.42	0.42	0.35	0.32	0.34	0.36	0.37	0.40	0.62	0.63
3l	0.54	0.58	0.58	0.56	0.55	0.57	0.59	0.56	0.79	0.76
3r	0.40	0.41	0.42	0.41	0.41	0.41	0.40	0.44	0.43	0.44
4l	0.36	0.37	0.39	0.41	0.37	0.43	0.64	0.64	0.66	0.67
4r	0.39	0.38	0.37	0.40	0.39	0.38	0.39	0.63	0.68	0.73

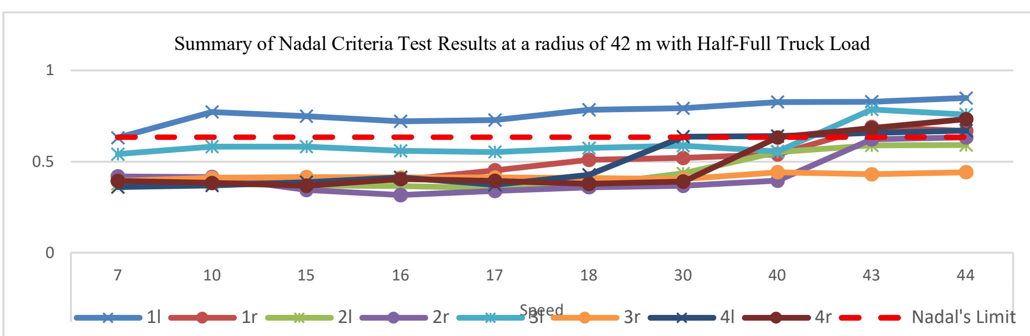

Figure 13. Nadal Criteria Graph for a 42 m Radius with Load

Based on the half-load test, high Nadal criteria values were observed on the left front and right rear wheels when turning right. At a turning radius of 42 meters, a speed of 7 km/h resulted in a Nadal value of 0.631. Although still below the safety limit, derailment potential is considered present if the Nadal value exceeds 0.8 for 0.3 seconds. The maximum recommended safe speed is 17 km/h, while a derailment occurs at 44 km/h, where the Nadal value reaches 0.850

3.2.3 Dynamic ROLA test with a empty-loaded ELF truck

Table 3. Maximum Nadal Value for Empty Load

Wheel Position	Speed									
	5	**13**	**14**	**19**	**20**	**30**	**35**	**40**	**44**	**45**
1l	0.63	0.62	0.75	0.75	0.77	0.76	0.77	0.80	0.80	0.82
1r	0.39	0.39	0.39	0.39	0.43	0.47	0.53	0.51	0.48	0.54
2l	0.39	0.34	0.32	0.32	0.30	0.30	0.33	0.35	0.50	0.62
2r	0.38	0.37	0.34	0.36	0.36	0.36	0.37	0.37	0.43	0.39
3l	0.54	0.56	0.54	0.54	0.52	0.55	0.57	0.57	0.64	0.67
3r	0.40	0.41	0.42	0.42	0.42	0.41	0.41	0.41	0.41	0.44
4l	0.34	0.36	0.38	0.38	0.38	0.35	0.38	0.56	0.66	0.69
4r	0.38	0.38	0.37	0.37	0.37	0.37	0.37	0.38	0.46	0.49

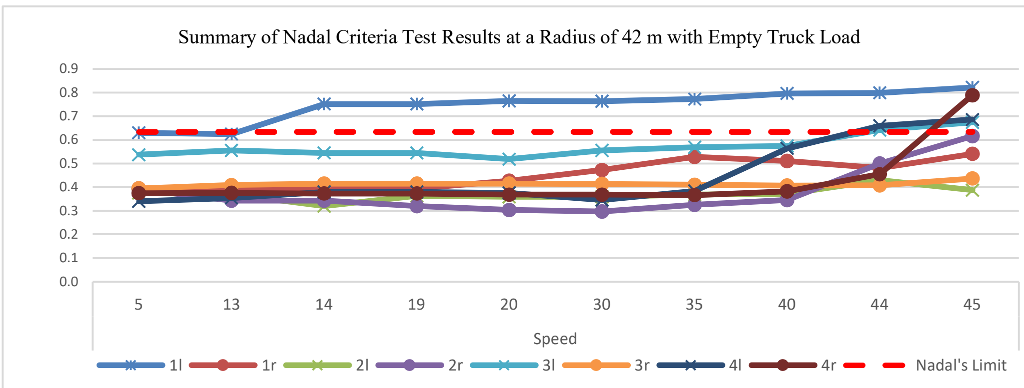

Figure 14. Nadal Criteria Graph for a 42 m Radius with an Empty Truck Load

Based on the results of the dynamic test without load, high Nadal criteria values were recorded on the left front and right rear wheels when turning right at a radius of 42 meters. At a speed of 13 km/h, the Nadal value recorded was 0.624, which is still below the safety limit. This value has not exceeded the threshold, but for radii less than 250 meters, values above 0.8 for 0.3 seconds are already classified as at risk of derailment. The maximum recommended safe speed is 19 km/h, while derailment was detected at a speed of 45 km/h with a Nadal value of 0.822.

3.3 Ride Index Simulation

The first step in this process is to calculate the primary suspension springs (inner, outer, friction spring) on the GD 40 10 09 flat car based on the dimensions from the 2D drawing.

Table 4. Stiffness Matrix Calculation Results

	Inner Spring **(N/mm)**	***Outer Spring*** **(N/mm)**	***Friction Spring*** **(N/mm)**
C_l	316.223,14	910.757,98	172.787,73
C_a	416,75	3.499,81	227,72
C_s	100.687,77	616.457,40	36.503,59
C_φ	1.321,14	9.820,72	746,66

After the coefficients are entered into the inspection car model, a dynamic simulation is performed with varying speeds and specific results are obtained.

Table 5. Results of the Ride Index for ROLA Transportation with Full Load

Speed (km/h)	*Ride Index*				Description
	Rear Bogie Lateral *Ride Index*	**Rear Bogie Vertical *Ride Index***	**Rear Bogie Lateral *Ride Index***	**Front Bogie Vertical *Ride Index***	
5	0.65849	1.32742	0.65364	0.84409	Special
10	1.59806	2.52193	1.2794	1.51959	Special
14	1.69738	3.34192	1.41797	1.57956	Special
20	2.01063	3.63891	2.01063	2.11427	Almost Exceptional
30	2.13775	3.75214	2.0654	2.22465	Almost Exceptional
35	2.32471	4.75286	2.24364	2.79181	Almost Exceptional
40	2.41202	4.77325	2.29168	2.81713	Almost Exceptional
41	2.43473	4.7748	2.33307	2.83121	Almost Exceptional
42	*Not Calculated*	*Not Calculated*	*Not Calculated*	*Not Calculated*	≥ Dangerous

Table 6. Results of Ride Index for ROLA Transportation with Half Full

Speed	*Ride Index*				
	Rear Bogie Lateral *Ride Index*	**Rear Bogie Vertical *Ride Index***	**Front Bogie Lateral *Ride Index***	**Front Bogie Vertical *Ride Index***	**Description**
7	0.864697	1.8394497	0.85171679	1.427556	Special
10	1.47052	2.0458535	1.1906084	1.7511543	Special
15	1.51650	2.2877138	1.27773611	1.76087	Special
17	1.58473	2.9849734	1.4115314	1.7930157	Special
18	1.72483	3.5424798	1.6639723	1.8844013	Special
30	1.75641	3.74972	1.6912407	1.9431762	Almost Exceptional
35	2.14182	3.89395	2.0325938	2.4859618	Almost Exceptional
43	2.25133	4.01748	2.1177146	2.587734	Almost Exceptional
44	*Not Calculated*	*Not Calculated*	*Not Calculated*	*Not Calculated*	≥Hazardous

Table 7. Results of Ride Index for ROLA Transportation with No Load

Speed	*Ride Index*				Description
	Rear Bogie Lateral *Ride Index*	**Rear Bogie Vertical *Ride Index***	**Front Bogie Lateral *Ride Index***	**Front Bogie Vertical *Ride Index***	
5	0.72520957	1.4123853	0.72195469	0.89486177	Special
13	1.3178353	1.676222	1.0251106	1.08017654	Special
14	1.5876006	2.3964427	1.376396	1.7724905	Special
19	1.6913763	3.1989041	1.6058806	1.8569092	Special
20	2.158725	3.7414991	2.0909126	2.2211435	Almost Exceptional
30	2.3170334	4.7452768	2.2367668	2.785715	Almost Exceptional
35	2.4060062	4.7743914	2.2823487	2.8192081	Almost Exceptional
44	2.4153317	4.8204775	3.00099054	3.1210766	Almost Exceptional
45	*Not Calculated*	*Not Calculated*	*Not Calculated*	*Not Calculated*	≥Hazardous

3.4 Validation Model

To confirm the Verification & Validation process, the following table presents a quantitative comparison between the results of the Universal Mechanism simulation and analytical calculations under ROLA conditions with a fully loaded ELF NMR-L truck at a radius of 42 m.

Parameter	Analytical Calculation	UM Simulation	Difference	Error (%)
Maximum lateral force (kgf)	28,351.77	28,337.70	14.07	0.05
Wheel vertical force (N)	46,289.36	46,100 – 46,500	±200	<1
Nadal maximum ratio (Y/Q)	≈ 0.84	0.837	0.003	0.36
Safety status	safe	safe	-	-

The derailment analysis in this study employs the Nadal criterion (Y/Q), which is widely accepted as a conservative indicator of wheel climb derailment risk in railway vehicle dynamics. Iwnicki (2006) and Shabana (2012) report that Y/Q values approaching 0.8–1.0 become critical when sustained over a finite time or distance, particularly on tight-radius curves. In the present study, the fully loaded ROLA configuration shows a maximum safe speed of 14 km/h with a Nadal value of approximately 0.84 sustained below the critical 0.3 s duration, while half-loaded and unloaded conditions allow higher safe speeds of 17 km/h and 19 km/h with Nadal values remaining below the derailment threshold. Although absolute safe speed values vary due to vehicle and track characteristics, the observed Nadal–speed relationship is consistent with international multibody simulation studies, supporting the validity of the proposed model.

4. CONCLUSION

This study demonstrates that the safe operation of Rollende Landstraße (ROLA) transportation on industrial tracks with minimum curve radii is primarily governed by the interaction between loading conditions, operating speed, and wheel–rail force distribution. Using multibody dynamic simulation in Universal Mechanism, the combined application of Nadal criteria and ride index is shown to provide a robust framework for evaluating derailment risk and ride stability on sharp curves. The findings underline the necessity of load-dependent speed regulation and appropriate bogie suspension design, rather than uniform operational limits. From a policy perspective, the results support the incorporation of simulation-based safety assessment into regulatory standards, operational guidelines, and infrastructure acceptance criteria, enabling evidence-based decision-making for operators, vehicle designers, and industrial track authorities in the deployment of ROLA systems.

REFERENCES

[1] PT Kereta Api Indonesia (Persero). Annual and Sustainability Report, 2023. Available: https://kalogistics.co.id/annual, 2023
[2] J. Korzeb and A. Kostrzewski, Rail Vehicle Dynamics. Cham, Switzerland: Springer International Publishing, 2015.
[3] S. Iwnicki, Handbook of Railway Vehicle Dynamics. Boca Raton, FL: CRC Press, Available: https://doi.org/10.1201/9781420004892, 2006
[4] A. A. Shabana, "Nadal's formula and high speed rail derailments," J. Comput. Nonlinear Dyn., vol. 7, no. 4, pp. 1–8, Available: https://doi.org/10.1115/1.4006730, 2012
[5] I. S. Utomo, D. S. Atmaja, and B. E. Febrilian, "Nadal Critical And Ride Index Characteristics Of Inspection Train Using Universal Mechanism Software On Different Back To Back," Sainstech Nusantara, vol. 1, no. 4, pp. 10–20, (2024). Available: https://doi.org/10.71225/jstn.v1i4.66
[6] Dahlan and S. B. Soemantri, "Uji coba profil aus untuk mengurangi keausan flens roda gerbong KKBW KA Babaranjang," Teknik Mesin, vol. 17, no. 1, pp. 1–8, 2017.
[7] Renault Trucks Deliver, Heavy Construction Range K, 2014.
[8] Universal Mechanism, "Simulation of rail vehicle dynamics," Railway Technical Report, vol. 1, no. 1, pp. 107–119, 2005.
[9] M. Zahro and N. I. Sutantra, "Analisis Pengaruh Posisi Center Of Gravity Terhadap Stabilitas Arah Bus Double Deck," Jurnal Teknik ITS, vol. 7, no. 2, 2018.
[10] S. Iwnicki, "Simulation of wheel–rail contact forces," Vehicle System Dynamics, vol. 37, no. 6, pp. 391–415, 2002.
[11] J. Pombo and J. Ambrósio, "An alternative method to include track irregularities in railway vehicle dynamic analyses," Vehicle System Dynamics, vol. 51, no. 6, pp. 907–930, 2013.
[12] R. M. Anderson and R. L. Johnson, "Rail vehicle derailment mechanisms and prevention," Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, vol. 223, no. 1, pp. 1–14, 2009.
[13] J. Kalousek and K. L. Johnson, "An investigation of wheel–rail contact stress and derailment criteria," Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, vol. 215, no. 1, pp. 1–14, 2001.
[14] X. Zhao, Z. Li, and R. Dollevoet, "Influence of wheel–rail interaction on derailment safety under curving conditions," Vehicle System Dynamics, vol. 55, no. 12, pp. 1865–1886, 2017.
[15] Direktorat Jenderal Perkeretaapian, Spesifikasi Teknis Gerbong Datar Pendukung Crane, 2014.
[16] Menteri Perhubungan, Peraturan Menteri Perhubungan Nomor: Km. 43 Tahun 2010 tentang Standar Spesifikasi Teknis Gerbong, 2010.