Curving Analysis of Modified Designs of Passenger Railway Vehicle Trucks*

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In this paper, modified truck designs were studied with an objective to achieving better compatibility between high speed stability and curving behaviour compared to the conventional truck. The analysis has shown that USD truck can be designed to achieve better over all performance compared to other truck designs. The results of steady state curving program were validated using the commercial software package NUCARS. The comparison of steady state curving behaviour of different truck designs using NUCARS also showed that USD truck has the potential to achieve superior performance compared to other truck designs. The research reported in this paper has dealt with steady state curving behaviour. To look into the safety implications of the new designs, transient behaviour in curves must be studied. The response of the railway cars in entry and exit spirals should be evaluated. The effects of track irregularities on the response of these modified designs need to be analyzed.

Key Words : Truck Design, Railway Vehicle, Curving Analysis, Spirals, NUCARS, Steady State Curving, Angle of Attack, Creep Coefficients, Wheel/Rail Resistance

1. Introduction

There has been increased research effort devoted to the development of improved designs of railway vehicle trucks or bogies. One of the main objectives of new designs has been to resolve the inherent conflict in the truck performance between lateral stability on tangent track and its ability to steer through curves. Horak et al. compared the performance of radial and conventional rail vehicle trucks and developed optimum design parameters for stability and curving performance⁽¹⁾. Bell and Hedrick analyzed stability and curving mechanics of body steered rail vehicle trucks⁽²⁾. It was shown that the curve negotiation capability of forced steering trucks is significantly improved over conventional and self steering radial trucks. However, the forced steering concept could produce kinematic instability and severely reduce critical speeds for low conicities and creep

coefficients.

In Ref.(3) modified railway passenger truck designs were considered to improve compatibility between dynamic stability and the ability of the vehicle to steer around curves. The models analyzed for lateral stability were :

C Truck : The conventional truck that has symmetric plan view primary suspension and is provided with rigid wheel-axle sets. This is referred as C truck.

CD Truck : One method to improve the trade-off is by providing primary yaw dampers as shown in Fig. 1. The yaw dampers are fitted to the front axle of the leading truck and rear axle of the trailing truck. If the primary yaw dampers are used, the longitudinal restraint is removed for the low velocity steering motions; for higher frequency motions the damper resistance increases and the higher restraint helps to achieve desired critical speeds. Yaw dampers provided in the primary suspension helped achieve a better trade-off between stability and curving performance for the British Rail Class 37 locomotive as was reported by Pennington et al.⁽⁴⁾. This type of truck with damper in the primary suspension is referred as CD truck in this paper. The yaw damper is character-

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Fig. 1 USD truck model

Fig. 2 Comparison of C and CD trucks

ized by the yaw damper coefficient $C_{p\psi}$ (Ns/m) and the series damper stiffness $K_{d\psi}$ (N/m). The damper series stiffness comes from the damper end mounting and bulk modulus of the damper working medium.

USD Truck: An asymmetry is considered in the plan view primary suspension. This Unsymmetric Suspension configuration is referred as US truck. By choosing proper value of asymmetry index a_s perfect steering could be achieved in the US truck⁽⁵⁾. But the introduction of asymmetry reduces the critical speed of the US truck. To improve the lateral stability, a primary yaw damper is included in the US truck and is referred to as USD truck. Figure 1 shows a sketch of the USD truck.

UWSD Truck: The rear axle has an Independently Rotating Wheelset (IRW). This is an Unsymmetric Wheelset configuration and is referred as UW truck. The design of truck in which there is both Unsymmetric Wheelset and Unsymmetric Suspension and a primary yaw damper is referred as UWSD truck.

The lateral stability of the modified designs were analyzed using eigen value analysis in terms of critical speed and are detailed in Ref.(3). The stability improvement in terms of higher critical speed for CD truck as compared to C truck is shown in Fig. 2.

The suspension parameters in terms of general-

Table 1 Primary suspension parameters for various truck designs

Truck type	Kalker's Creep Value	Bending Stiffness Nm/rad		Remarks
		CV 80 m/s	CV 60 m/s	
C truck	50%	5.0×10^6	2.5×10^6	
	100%	4.5×10^6	3.0×10^6	
R truck	50%	1.75×10^6	1.25×10^6	
	100%	3.0×10^5	2.0×10^5	
CD truck	50%	1×10^6	3×10^5	
	100%	3.5×10^6	1.5×10^6	
USD truck	50%	$*1.49 \times 10^6$	5.5×10^5	* asymmetric ratio = 0.8 ** asymmetric ratio = 0.4
	100%	$**1.49 \times 10^6$	7.42×10^5	

ized bending stiffness K_b and generalized shear stiffness K_s for the different truck designs are given in Table 1 for critical speeds of 60 m/s and 80 m/s. The simulations were done for full creep and half creep Kalker coefficients.

The results of steady state curving performance of these designs are reported in this paper. A Steady State Curving (SSC) model incorporating non-linear wheel/rail profile and creep force saturation and linear suspension characteristics is used. The results of the model are compared with those from NUCARS (New and Untried Car Analytic Regime Simulation). NUCARS is a multi-body dynamic program for railway vehicle design and performance evaluation and is an industry wide standard in North America.

Nomenclature

- a : half of track gauge
- a_s : asymmetry index
- b : half of wheelbase
- $C_{p\psi}$: damping coefficient for the primary yaw damper
- d_b : half of lateral spacing between primary longitudinal springs
- d_s : half of lateral spacing between secondary vertical springs
- F_x : longitudinal creep force
- F_y : lateral creep force
- K_b : generalized bending stiffness
- K_s : generalized shear stiffness
- k_b : bending stiffness between wheelsets
- k_s : shear stiffness between wheelsets
- K_{x1}, K_{x2} : longitudinal stiffness of leading and trailing wheelsets in US truck
- $K_{d\psi}$: series stiffness coefficient for the yaw damper
- r_o : centered wheel radius
- r_l, r_r : radius at left and right wheels
- R : curve radius
- V : vehicle speed
- y : lateral displacement
- ψ : yaw rotation
- ϕ : roll angle
- δ_l : left wheel contact angle
- δ_r : right wheel contact angle

Fig. 3 Free body diagram of wheelset

δ_o : contact angle for centered wheelset

2. Steady State Curving Model

The development of analytical method for the non-linear model is based on the paper by Horak and Wormley⁽⁶⁾. To predict the wheel-rail forces in curving, non-linear wheel-rail geometry and non-linear creep forces have been taken into consideration.

Figure 3 shows the free body diagram of a single wheelset in the figure, δ_l and δ_r are the contact angles at the left and right wheels and ϕ is the roll angle of the wheelset. The equations of motion for wheelset lateral and yaw can be written from the free body diagram. The external forces and moments acting on the wheelset are:

F_s, M_s : primary suspension force and moment

$F_{ry}, F_{rz}, M_{rz}, F_{ly}, F_{lz}, M_{lz}$: creep forces and moments

N_r, N_l : normal wheel-rail forces

The wheelset force balance for the steady state analysis lead to the following equation.

Lateral:

$$F_{ry} + F_{ly} + N_r \sin(\delta_r - \phi) - N_l \sin(\delta_l + \phi) + F_s = 0 \quad (1)$$

Yaw:

$$a(F_{rx} - F_{lx}) + a(F_{ry} - F_{ly})\phi + (N_r \sin(\delta_r - \phi) + N_l \sin(\delta_l + \phi))a\phi + M_{lz} + M_{rz} + M_{susp} = 0 \quad (2)$$

The creep forces are non-linear functions of the creepages, normal wheel loads and wheel-rail geometry, mainly due to the dry friction saturation limit equal to μN where μ is the coefficient of friction and N is the normal wheel load. A heuristic non-linear creep model which combines Kalker's linear creep theory with a creep force saturation representation is used.

The contact plane creep forces and moments are transformed to the track coordinates by the following equations:

Longitudinal:

$$F_{lx} = F'_{lx} \quad F_{rx} = F'_{rx} \quad (3)$$

Lateral:

$$F_{ly} = F'_{ly} \cos(\delta_l + \phi) \quad F_{ry} = F'_{ry} \cos(\delta_r - \phi) \quad (4)$$

Vertical:

$$F_{lz} = F'_{ly} \sin(\delta_l + \phi) \quad F_{rz} = F'_{ry} \sin(\delta_r - \phi) \quad (5)$$

Once these forces are calculated, the displace-

ment vector is calculated by solving the following steady-state non-linear equation:

$$KY = F(y) \quad (6)$$

where K is the suspension stiffness matrix and $F(y)$ are the external forces and moments acting on the vehicle. The analytical technique is a multi-step iteration on the simultaneous equations of motion.

To predict the wheel-rail geometric parameters like contact angle, roll angle, rolling radius for any particular value of lateral displacement, the wheel-rail characterization program written by Cooperrider and Heller was used⁽⁷⁾. The wheelset position can be described by its lateral position of its geometric center relative to the track center line and the yaw angle of the wheelset. The roll angle and vertical displacement of the wheelset are determined by the geometric constraints. This is because the railway wheelset moves in a manner constrained by the geometry of the wheel, rail and the track structure.

3. Analysis of Simulation Results

The performance of different truck designs are compared using the Steady State (SS) curving program based on the above methodology. The trucks considered are the Conventional Truck (C Truck), the Conventional Truck provided with Yaw Damper (CD Truck), the Unsymmetric Suspension Truck with Yaw Damper (USD truck), Radial Truck (R Truck) and the Unsymmetric Wheelset and Suspension Truck with Yaw Damper (UWSD Truck). The primary suspension parameters in terms of bending stiffness and shear stiffness are chosen for two critical speeds, 80 m/s and 60 m/s as given in Table 1. Other mass, inertia and geometric parameters of the truck and carbody used for the model are given in Ref.(3). Figures 4 and 5 present the equilibrium free body diagrams obtained from computer simulation results as the truck negotiates a 10 degree curve (175 m radius). The figures show all the creep and suspension forces and moments generated during steady state curving for the C Truck and USD Truck. A constant centre plate moment input of 7702 N-m is assumed for each case. The figures shown are for 50% Kalker creep coefficients. The trucks have a design critical speed of 80 m/s.

The centre plate yaw moment is reacted by wheel/rail lateral forces and wheelset yaw moments. The moment balance is given by the equation,

$$M_{cp} + 2f_w b + M_{w1} + M_{w2} = 0 \quad (7)$$

where

M_{cp} : centre plate yaw moment

M_{w1}, M_{w2} : wheelset yaw moments on axles 1 & 2

f_w : wheelset lateral force

b : truck wheelbase

Fig. 4 Steady state curving attitude of C truck

Fig. 5 Steady state curving attitude of USD truck

Referring to Fig. 4, for the C truck, 31 234 N of lateral force is generated on the leading, high rail wheel with an Angle of Attack (AOA) of 13 mrad. For the USD truck (Fig. 5), 9 269 N of lateral force is generated with an AOA of 0.6 mrad in the leading wheelset. It is seen that the better steering capability of the USD truck reduces the AOA and the resulting lateral force.

In the following sections, results of the simulation are given in terms of Angle of Attack (AOA), lateral force on lead high rail wheel and wear index. These parameters are analyzed for 100% and 50% Kalker creep coefficients. Creep coefficients in reality could vary over a wide range of values, due to contamina-

Fig. 6 Angle of attack for leading axle (100% Kalker creep coefficient)

Fig. 7 Angle of attack for leading axle (50% Kalker creep coefficient)

tion of the rail surface, work hardening etc. The 100% Kalker creep values would represent a clean, uncontaminated and dry surface.

3.1 Angle-of-attack of leading axle

Figure 6 compares the leading wheelset AOA for 100% Kalker creep values. The trucks are designed to achieve a critical speed of 80 m/s. For the C truck, the angle of attack rapidly increases as the curvature is increased. The angle of attack of 15.6 mrad is developed in the lead axle as the truck negotiates a 12 degree curve (146 m radius). The AOA is 13.1 mrad for the R truck, 14.3 mrad for the CD truck and 0.4 mrad for the USD truck.

The variation of angle of attack shown in Fig. 7 is for the case of 50% Kalker creep coefficient. On a 12 degree curve, the angle of attack is 16.4 mrad for the C truck, 8.7 mrad for the R truck, 3.7 mrad for the CD truck and 1.0 mrad for the USD truck. The performance of the CD and USD trucks is better compared to the other two truck designs, namely, the C truck and the R truck.

In general, the AOA is higher for 100% Kalker creep values compared to the case of 50% Kalker creep value. The performance of the CD truck deteriorates most rapidly as the Kalker creep value is increased from 50% to 100%. The performance of USD truck does not vary significantly as the creep coefficients change. The values of AOA for half creep

Table 2 Angle of Attack of front and rear wheelsets in mrad on a 12 degree curve

Truck type	Design Critical Speed 80 m/s				Design Critical Speed 60 m/s			
	Full Creep		Half Creep		Full Creep		Half Creep	
	Front	Rear	Front	Rear	Front	Rear	Front	Rear
C Truck	15.6	1.67	16.4	1.95	13.2	1.4	12.5	1.5
R Truck	13.1	1.3	8.66	1.39	1.6	0.54	4.25	1.17
CD Truck	14.3	1.54	3.7	1.0	4.85	0.83	1.8	1.6
USD Truck	0.4	1.0	1.0	2.12	0.41	0.9	1.12	2.12
UWSD Truck	-	-	6.54	-5.32	-	-	-	-

Fig. 8 Lateral force on lead high rail wheel (100% Kalker creep coefficient)

Fig. 9 Lateral force on lead high rail wheel (50% Kalker creep coefficient)

and full creep values are tabulated for two design critical speeds of 80 m/s and 60 m/s in Table 2. The angle of attack for the UWSD truck is 6.54 mrad for the critical speed of 80 m/s evaluated at half creep values. The UWSD truck configuration is not considered for further evaluation, since its curving performance is worse than the other two truck designs with primary yaw dampers. It is seen that USD truck achieves superior performance compared to other truck designs.

3.2 Lateral force on the leading axle

The lateral forces exerted on the high wheel are shown in Figs. 8 and 9 for the case of full and half creep values respectively. The direction of forces on the rails exerted by the wheels cause gauge widening

Table 3 Lead wheelset lateral forces in Newton on a 12 degree curve

Truck type	Design Critical Speed 80 m/s				Design Critical Speed 60 m/s			
	Full Creep		Half Creep		Full Creep		Half Creep	
	High	Low	High	Low	High	Low	High	Low
C Truck	-33483	22566	-33476	22629	-32928	22403	32105	22461
R Truck	-32966	22380	-30612	21708	-20134	12986	-24104	16744
CD Truck	-33203	22486	-21343	15783	-28163	20658	-13903	10324
USD Truck	-8309	4768	-11420	6799	-8303	4925	-11446	7591
UWSD Truck	-	-	-21693	25584	-	-	-	-

of the track. The unsymmetric truck configuration, USD truck, develop lower lateral forces compared to the symmetric configurations through out the range of curvatures. Table 3 compares the lateral force on the high rail wheel, on a 12 degree curve. The ratio of lateral force to vertical force on a wheel, more commonly referred as L/V ratio, gives an indication of the derailment potential of a truck. Typically, a value of L/V ratio larger than 1.0 is not desirable. The L/V ratio, when plotted, follows a trend similar to the lateral force in the steady state curving analysis considered here.

3.3 Truck wear index

One major objective in the evaluation of curving behaviour is to reduce wheel and rail wear in curves. The wear index suggested by Elkins and Allen is used for the analysis⁽⁸⁾. It is based on the premise that rates of wheel and rail wear can be related to the forces and creepages between the wheel and rail. Mathematically, the work done in the contact patch is equal to the dot product of the resultant creep force vector and the resultant creepage in the wheel-rail contact patch.

$$W = F_c \cdot \xi \quad (8)$$

In the above equation, W is the wear index, F_c is the resultant creep force and ξ is the resultant creepage. The wear index is not necessarily indicative of the actual wear taking place at the wheel-rail contact patch. But a comparison of wear index for different truck configurations could be done as a means of measuring the curving performance.

The variation of wear index for the different truck designs are shown in Figs. 10 and 11 for 50% and 100% Kalker creep values. The trucks are designed for a critical speed of 80 m/s. The wear index for the conventional truck increases sharply as the curvature increases. The wear index for the USD truck is less than other designs. Table 4 compares the wear index for different truck designs on a 12 degree curve.

3.4 Summary of simulation results

The comparison of steady state curving behaviour is carried out with the objective to obtain a better compatibility between high speed stability and curv-

Fig. 10 Wear index per truck (100% Kalker creep coefficients)

Fig. 11 Wear index per truck (50% Kalker creep coefficients)

ing behaviour compared to the conventional truck. The analysis has shown that USD truck can be designed to achieve better curving performance for a given critical speed compared to other truck designs. In the next section, the results obtained here will be validated using a commercially available software program.

4. Curving Analysis Using NUCARS

NUCARS (New and Untried Car Analytic Regime Simulation) is a general purpose, multi-body dynamic program used in the modeling and simulation of railway vehicles⁽⁹⁾. NUCARS models a rail vehicle system as a collection of interconnected lumped masses. This program was developed by Association of American Railroads (AAR). This program has undergone extensive validation since its development and is an industry-wide standard in North America.

In the following sections, curving performance of different truck designs are evaluated using NUCARS. The objective is to compare the results of the steady state curving analysis obtained earlier with results from NUCARS. Thus an indirect validation of the steady state curving analysis program has been carried out.

4.1 Angle of attack

The leading axle angles of attack for C truck, CD

Table 4 Wear Index per truck in Nm/m on a 12 degree curve

Truck type	Design Critical Speed 80 m/s		Design Critical Speed 60 m/s	
	Full Creep	Half Creep	Full Creep	Half Creep
C Truck	1079	1137	920	857
R Truck	976	695	577	437
CD Truck	996	322	462	177
USD Truck	215	154	216	161
UWSD Truck	-	813	-	-

Fig. 12 Angle of attack from NUCARS, critical speed 60 m/s (K_b corresponds to half creep)

Fig. 13 Angle of attack from NUCARS, critical speed = 60 m/s (K_b corresponds to full creep)

truck and USD truck predicted by NUCARS are shown in Figs. 12 and 13. As seen from Table 1, the CD truck requires a bending stiffness in the range of 3×10^5 N/m to 1.5×10^6 N/m, depending on the creep coefficient considered, to achieve a critical speed of 60 m/s. For half Kalker creep values, the CD truck has a softer primary suspension compared to the USD truck. As the creep coefficient is increased, much higher primary bending stiffness is required for CD truck to maintain the same critical speed compared to the USD truck. The result is improved steering up to a curvature of 12 degrees for the lead axle of the USD truck as seen in Fig. 10. Figures 14 and 15 show a similar trend when the trucks are designed for a critical speed of 80 m/s.

4.2 Lateral force on the leading axle

The lateral force on the high rail wheel obtained from NUCARS simulations are plotted in Figs. (16)

Fig. 14 AOA predicted by NUCARS, critical speed = 80 m/s (K_b correspond to 1/2 creep)

Fig. 15 AOA predicted by NUCARS, critical speed = 80 m/s (K_b correspond to full creep)

Fig. 16 Lateral force on the high rail wheel, critical speed 80 m/s (K_b correspond to 1/2 creep)

Fig. 17 Lateral force on the high rail wheel, critical speed 80 m/s (K_b correspond to full creep)

Fig. 18 Total wheel/rail resistance predicted by NUCARS (K_b and a_s values correspond to half creep)

Fig. 19 Total wheel/rail resistance predicted by NUCARS (K_b and a_s values correspond to full creep)

and (17). The design critical speed is 80 m/s. The trend is similar to the plots in Figs. 8 and 9 obtained from the steady state curving program. The absolute values of lateral force obtained from NUCARS simulation for the USD truck are higher compared to the steady state program. Still the USD truck achieves the lowest lateral force compared to the C truck and CD truck.

4.3 Wheel/rail resistance

The wear index developed in NUCARS is similar to the one defined earlier by Eq.(7). It relates the energy dissipated in the contact patch to the wheel wear. By summing the wear indices for all wheel/rail contact points on a vehicle the total rolling resistance of the vehicle due to wheel on rail contact is obtained in NUCARS. This is a measure of the drag induced in the wheel/rail contact area. The rolling resistance obtained from NUCARS is different from the wear index used earlier in the steady state curving analysis in that it takes into account wheel/rail wear of two trucks.

Figures 18 and 19 compare the total wheel/rail resistance values for the different truck designs for the critical speed of 80 m/s. The trend is similar to the wear index plot shown in Figs. 10 and 11.

Table 5 NUCARS simulation results on a 12 degree curve, critical speed = 80 m/s

Truck type	AOA, mrad		Lateral Force, N		Rolling Resistance, Nm/m
	Front	Rear	High	Low	
C Truck	15.4	0.7	-34936	22231	2257
CD Truck, $K_b = 1 \times 10^5$	6.7	2.2	-32899	25095	1449
CD Truck, $K_b = 3.5 \times 10^5$	14.2	1.7	-37786	24993	2344
USD Truck, $a_{\phi} = 0.8$	2.5	2.1	-29140	16766	1270
USD Truck, $a_{\phi} = 0.4$	2.6	3.8	-25416	17279	1217

Table 6 NUCARS simulation results on a 12 degree curve, critical speed = 60 m/s

Truck type	AOA, mrad		Lateral Force, N		Rolling Resistance, Nm/m
	Front	Rear	Front	Rear	
C Truck	13.4	1.7	-34028	22306	1927
CD Truck, $K_b = 3 \times 10^5$	5	3.7	-29265	24703	1418
CD Truck, $K_b = 1 \times 10^6$	7.7	1.6	-35445	24756	1505
USD Truck, $K_b = 5.57 \times 10^5$	3.7	4.3	-24079	20422	1298
USD Truck, $K_b = 7.43 \times 10^5$	3.5	4.5	23721	20066	1266

4.4 Comparison of NUCARS simulations with the SS curving model

Tables 5 and 6 summarize the values predicted by NUCARS for a car on a 12 degree curve for two cases of design critical speeds, 80 m/s and 60 m/s. The values can be compared with those predicted by SS curving model given in Tables 2 to 4. The steady state curving attitude of the USD truck on a 10 degree curve is shown in the Fig. 20. The forces and moments shown in the figure are values predicted by NUCARS. This can be compared to the results of the steady state curving analysis shown in Fig. 5.

The values of angle of attack for the C Truck on a 12 degree curve compare reasonably well as seen from Tables 2 and 5. The predicted AOA by the steady state curving program is 15.6 mrad and 1.67 mrad for the front and rear wheelsets. The corresponding values predicted by NUCARS is 15.4 mrad and 0.7 mrad for the front and rear wheelsets respectively. The lateral forces for the high rail wheel are 33 476 N and 34 936 N for the steady state curving program and NUCARS respectively.

Results of CD truck simulations using NUCARS also match reasonably with results of steady state curving simulations. The performance of CD truck is much superior to the C truck as predicted by NUCARS for the truck with stiffness parameters designed for half creep. The CD truck designed for the critical speed of 80 m/s with full creep coefficients develop a lateral force of 37 786 N as predicted by NUCARS. This is in comparison to the value of 33 203 N predicted by the SS curving model. The

Fig. 20 Steady state curving attitude of USD truck predicted by NUCARS

values of AOA of the lead wheelset predicted by NUCARS for an USD truck on a 12 degree curve is 2.5 mrad and the lateral force is approximately 29 140 N. Thus the USD truck achieves the best curving performance compared to other truck designs. The trend predicted by NUCARS is similar to the SS curving model. One of the main differences between the two programs is the computation of wheel/rail interaction forces. In NUCARS, forces are computed using Kalker's non-linear creep theory. The steady state curving model computes the wheel/rail forces using Kalker's linear theory combined with the creep force saturation model as explained in section 2.

5. Summary

In this paper, modified truck designs were studied with an objective to achieving better compatibility between high speed stability and curving behaviour compared to the conventional truck. The analysis has shown that USD truck can be designed to achieve better over all performance compared to other truck designs. The results of steady state curving program were validated using the commercial software package NUCARS. The comparison of steady state curving behaviour of different truck designs using NUCARS also showed that USD truck has the potential to achieve superior performance compared to other truck designs.

The research reported in this paper has dealt with steady state curving behaviour. To look into the safety implications of the new designs, transient

behaviour in curves must be studied. The response of the railway cars in entry and exit spirals should be evaluated. The effects of track irregularities on the response of these modified designs need to be analyzed.

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