Nagurka, M.L., Hedrick, J.K., and Wormley, D.N., "Curving Performance of Rail Transit Trucks, " Vehicle System Dynamics, Vol. 12, No. 1-3, July 1983, pp. 18-23.

# Curving Performance of Rail Transit Trucks

M. L. NAGURKA, J. K. HEDRICK, and D. N. WORMLEY*

On curved track the wheelsets of a conventional rail vehicle are prevented from aligning radially by the stiffness of suspension elements needed to achieve dynamic stability. Angular misalignments of the wheelsets increase wheel/rail forces and result in increased wear, fuel consumption, and risk of derailment. Advanced rail truck designs have been proposed as a means of achieving wheelset radial alignment during curve negotiation while

* Department of Mechanical Engineering M.I.T. Cambridge, Mass. 02139, USA



maintaining stability. These designs include self-steered trucks with direct interconnections between the wheelsets and forced-steered trucks with both direct interconnections and passive linkages between the carbody and axles.

This paper investigates the steady-state curving performance of rail vehicles that employ conventional and advanced truck designs. A detailed nonlinear model is developed that accounts for single-point and two-point wheel/rail contact.

*Model Development*

The generic truck model, shown in Fig. 1, can represent conventional, self-steered, and forced-steered truck configurations. The model represents a variety of truck designs, since it accommodates arbitrary wheelset-truck-carbody interconnections. The effects of linkages between the wheelsets, truck, and carbody are reflected in geometric offsets. Yaw offsets, $\Delta\psi_1$ and $\Delta\psi_2$, are connected in series with linkage bending stiffnesses; lateral offsets, $\Delta y_1$ and $\Delta y_2$, are connected in series with linkage shear stiffnesses. The geometric offsets are controlled by the following general steering laws:

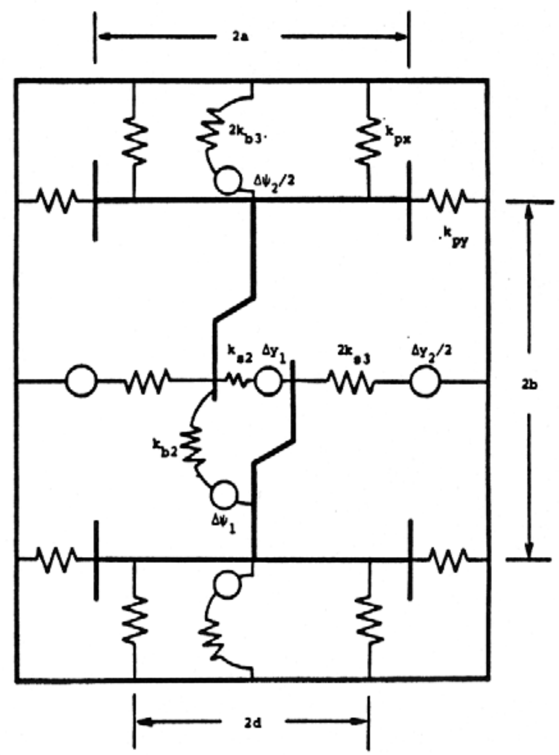


Fig. 1. Generic Forced Steering Truck Model

$$\Delta\psi_1 = \Delta\psi_2 = \pm 2G_1\left(\frac{\psi_1+\psi_2}{2} - \psi_c\right) \pm 2G_2\left(\frac{y_1-y_2}{2b} - \psi_c\right) + 2G_3\left(\frac{y_1+y_2}{2} - y_T\right)$$

$$\pm 2G_4(\psi_T - \psi_c) \pm 2G_5\left(\frac{\psi_1+\psi_2}{2} - \psi_T\right) \pm 2G_6\left(\frac{y_1-y_2}{2b} - \psi_T\right) \quad (1)$$

$$\Delta y_1 = \Delta y_2 = 2H_1\left(\frac{y_1+y_2}{2} - y_c\right) + 2H_2(y_T - y_c) + 2H_3\left(\frac{y_1+y_2}{2} - y_T\right) \quad (2)$$

where the $G$'s and $H$'s are the steering gains; $y_1$, $y_2$, $y_T$, and $y_c$ ($\psi_1$, $\psi_2$, $\psi_T$ and $\psi_c$) are the lateral displacements (yaw angles) of the leading wheelset, trailing wheelset, truck, and carbody, respectively; and the notation $\pm$ implies $+$ for the front truck and $-$ for the rear truck. The generic truck model is reduced to particular truck configurations by assigning appropriate stiffness and steering gain values.

A nonlinear model has been developed to predict the steady-state curving behavior of a single wheelset [1]. The model distinguishes between wheel/rail contact in which: (1) single-point contact occurs at both wheels of the wheelset; (2) simultaneous tread and flange contact, i.e., two-point contact, occurs at the outer wheel and single-point tread contact occurs at the inner wheel of the wheelset. Case (2) occurs for many wheel/rail profiles, especially those with steep flanges on small radius curves. The wheelset model also accounts for nonlinear wheel/rail profile geometry (large contact angles), saturation of wheel/rail friction (creep) forces and laterally flexible rails.

The nonlinear wheelset model is coupled to the generic truck model by means of suspension elements. The primary suspension system is modelled as a system of piecewise linear, hardening springs in the longitudinal and lateral directions. The secondary suspension system consists of linear springs in the lateral and vertical directions, and a nonlinear torsional spring in the yaw direction which saturates at the centerplate breakaway torque.

The vehicle steady-state curving equations are conditions of simultaneous force and moment equilibrium of the wheelsets, trucks, and carbody. The equilibrium equations are written as:

$$\boldsymbol{K}(\boldsymbol{X}) \cdot \boldsymbol{X} = \boldsymbol{B}(\boldsymbol{X}) \quad (3)$$

where the matrix product $\boldsymbol{K}(\boldsymbol{X}) \cdot \boldsymbol{X}$ represents a vector of internal suspension forces and moments and $\boldsymbol{B}(\boldsymbol{X})$ represents a vector of all external forces and moments due to track curvature, cant deficiency (lateral unbalance), and forced-steering (from carbody yaw). The matrix product is composed of a geometry state vector $\boldsymbol{X}$ and a nonlinear stiffnes matrix $\boldsymbol{K}(\boldsymbol{X})$ due to nonlinear primary and secondary suspension components.

The equilibrium conditions are a set of simultaneous nonlinear algebraic equations which are solved using a combined Newton-raphson and steepest descent method [2]. To reduce numerical computations, a single truck/half-carbody model is used for initial numerical studies.

*Performance Studies*

Parametric studies to determine the effects of wheel/rail profile, suspension designs, and track curvature on the curving performance of conventional, self-steered and several forced-steered truck designs have been conducted [3].

The combined influence of primary longitudinal stiffness, $k_{px}$, and track curvature on the



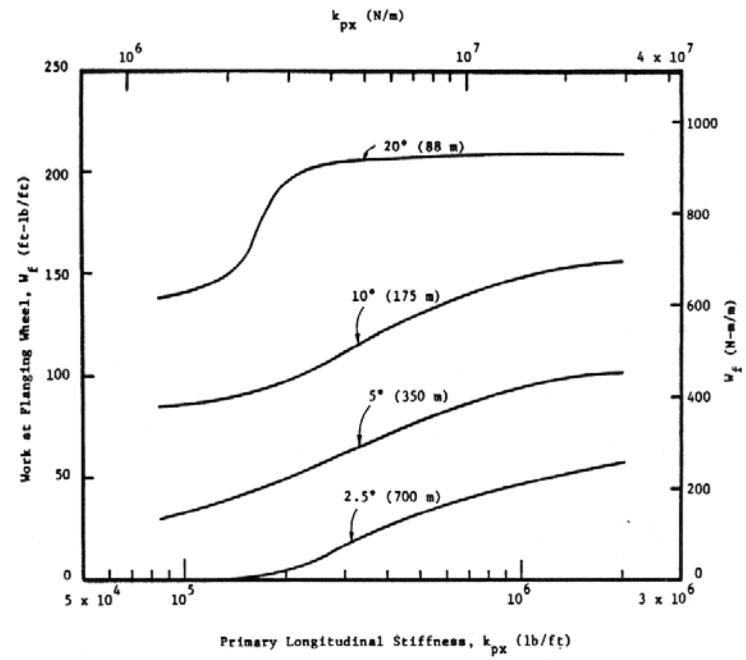


Fig. 2. Work at Flanging Wheel vs. Primary Longitudinal Stiffness of a Conventional Truck with New Wheels Negotiating Four Track Curvatures.

curving behaviour of a conventional truck with new AAR 1 in 20 wheels on worn rails is shown in Fig. 2. The curving performance is measured in terms of the work expended at the flanging wheel. As $k_{px}$ is increased, the work at the flanging wheel also increases. For the tight 88m (20°) curve, the work rises sharply and then asymptotically reaches a constant as $k_{px}$ is increased, due to saturation of the creep forces. Figure 2 also demonstrates that the work index increases with larger degree curves, due to the larger creepages and creep forces.

The wheel/rail profile strongly influences the curving behavior. Parametric results similar to Fig. 2 show that substantially less work is expended for a conventional truck with single-point contact Heumann wheels than for a truck of identical stiffness with two-point contact profiles. The Heumann wheel profile maintains single-point contact at all lateral excursions, and in comparison to two-point contact profiles provides a larger restoring moment aligning the wheelsets radially and improving curving performance.

In addition to the curving analyses, stability studies have been conducted to determine the speed at which lateral instability or hunting occurs on tangent track. This speed, called the critical speed, is determined by computing the eigenvalues of a linearized model. The combined results of the curving performance and stability studies of conventional, self-steered, and forced-steered truck configurations demonstrate an inherent design tradeoff.

Principal design parameters, such as suspension stiffnesses and wheel profiles, which improve the curving performance (i.e., decrease the work generated) degrade the stability properties (i.e., lower the critical speed).

The curving properties of truck designs with the same critical speed have been compared in order to identify designs with improved curving performances. Figure 3 shows the work at the flanging wheel as a function of curvature for conventional, self-steered, and two forced-steered trucks with new wheels, all designed for a critical speed of 193 km/hr (120 mph). The two forced-steered radial truck designs are FSR I with a soft primary

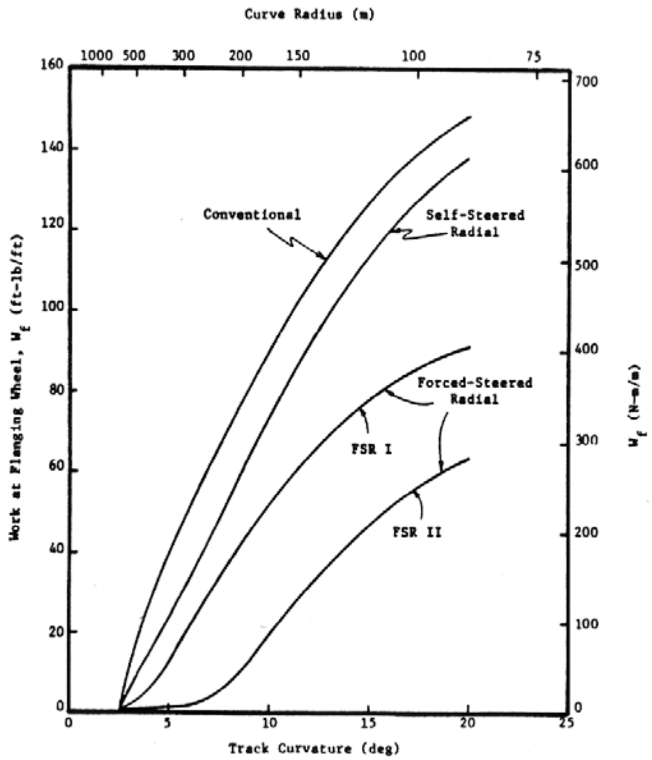


Fig. 3. Work at Flanging Wheel vs. Track Curvature for Four Truck Designs with New Wheels and 120 mph (193 km/hr) Critical Speeds.

longitudinal stiffness ($1.0 \times 10^6$ N/m or $7.0 \times 10^4$ lb/ft) and FSR II with a very soft primary longitudinal stiffness ($1.5 \times 10^4$ N/m or $1.0 \times 10^3$ lg/ft). Figure 3 demonstrates that a 20 percent improvement in curving performance in terms of decreased flanging wheel work is gained by employing a self-steered truck rather than a conventional truck for negotiating a 175m (10°) curve. Further, the forced-steered trucks, FSR I and FSR II, offer a 42 and 79



percent reduction in flanging wheel work, respectively, in comparison to a conventional truck for negotiating the same radius curve.

*Conclusions*

The results of these studies indicate that forced-steered trucks potentially offer substantial performance improvements in comparison to conventional trucks for negotiating moderate and high degree curves. However, the increased complexity associated with implementation and maintenance of forced-steering linkages must be assessed. The results also show that the work generated during curving negotiation can be reduced by using single-point contact Heumann profiles in comparison to two-point contact profiles.

REFERENCES

1. Nagurka, M. L., Bell, C. E., Hedrick, J. K., and Wormley, D. N., "Computational Methods for Rail Vehicle Steady-State Curving Analysis," *Computational Methods in Ground Transportation Vehicles*, ed. Kamal, M. and Wolf, J., A.S.M.E. AMD-Vol. 50, 1982, pp. 153-179.
2. Rabinowitz, P., *Numerical Methods for Nonlinear Algebraic Equations*, Gordon and Breach Science Publishers, London, 1970.
3. Wormley, D. N., Hedrick, J. K., and Nagurka, M. L., "Stability and Curving Performance of Conventional and Advanced Rail Transit Vehicles," U.S.D.O.T., Interim Report, DTRS 57-80C-00152, 1982.