研　究　速　報　　UDC 629.4.021.03

Influence of Traction/Brake Torque on Curving Dynamics of Trucks with Unsymmetric Suspension of Rail Vehicles

——2nd Report; Comparison with Symmetric Trucks——

鉄道車両用前後非対称支持台車における曲線旋回性能に与える駆動・制動トルクの影響
——第2報　対称台車との比較——

Juraj GRENCIK* and Yoshihiro SUDA**
ユライ　グレンチーク・須　田　義　大

1. INTRODUCTION

In the first report[1] of the study, the effect of traction/braking torque on curving behavior of rail vehicle has been examined. Curving performance of trucks with longitudinally unsymmetric truck design[2] was modeled. Here, both symmetric and unsymmetric trucks have been considered so that the curving performance of a vehicle with the conventional symmetric truck design could be examined and compared with the vehicle that differs only in longitudinal stiffness characteristics of the primary suspension. The vehicle modeled in the second report differs from that in the first report, so that to examine whether conclusions obtain from the previous report are valid more generally for other vehicle design.

Again, the A'GEM Rail Vehicles Dynamic Software Package[3] was used for simulation of full vehicle body dynamics with considering non-linear characteristics of the contact force between wheel and rail.

In the study, wheel lateral displacement and attack angle were chosen to describe the rail vehicle curving behavior. As in the first report, only the characteristics of leading axle have been used to show vehicle's curving behavior. Curving performance under various vehicle velocities and various curve radiuses has been calculated. Effect of vehicle's mass (load) has been considered, too.

*Assistant Professor of University of Transport and Communications in Zilina, Slovakia
**2nd dept. Institute of Industrial Science, University of Tokyo

2. CURVING PERFORMANCE SIMULATIONS

2.1. Vehicles and track parameters

Basic concept of the truck with longitudinally unsymmetric suspension characteristics has been described in the first report of this study. As well the forces acting on a truck in curve were shown there. Here, the proposed light weight vehicle (further "PLW") design for rapid passenger transport in North America[4] with standard gauge was modeled. Both the symmetric and unsymmetric trucks were used for the same vehicle. In both cases, car body mass (load) was a parameter, too.

The parameters of vehicles modeled in the study are written in Table 1. Similar simulation models as in the first report have been done. Vehicle models represent a 25 degree of freedom system, of which 4 degrees of freedom are the wheelsets rotation around their axles, to let the torque acting. Conical wheel profile with conicity 0.15 has been considered. Friction coefficients of wheel tread and flange were estimated to be 0.30 and 0.25 respectively.

Curves with 600 m, 800 m, 1400 m and 2000 m radii were chosen. Velocities of 27.7 m/s for the two smaller and 44.4 m/s for the two larger radii were considered. In all cases rail cant was 0.1 rad, so the different cant deficiencies (values of 0.03, −0.002, 0.043 and 0.0 respectively) resulted. For each curve radius the length of transitional curve was 50 m, total length of simulated track was 80 m. The flange clearance was always assumed to be 12 mm. Curve parameters were chosen in such a way so that there is no flange contact when the torque is zero.

The simulation calculations have been carried out in

Table 1 Vehicle parameters

	PLW unsym./	sym.
Wheelset Mass (kg)	500	500
Yaw inertia (kgm²)	250	250
Roll inertia (kgm²)	250	250
Pitch inertia (kgm²)	250	250
Wheel conicity (-)	0.15	0.15
Wheel diameter (m)	0.75	0.75
Truck Mass (kg)	1000	1000
Yaw inertia (kgm²)	500	500
Roll inertia (kgm²)	400	400
Height of center of gravity(m)	0.50	0.50
Wheel base (m)	2.50	2.50
Half distance between primary suspension (m)	0.90	0.90
Half distance between secondary suspension (m)	1.25	1.25
Car body Mass (kg) (empty/ loaded)	30000 45000	30000 45000
Yaw inertia (kgm²) (empty/ loaded)	100000 150000	100000 150000
Roll inertia (kgm²) (empty/ loaded)	1500000 2250000	1500000 2250000
Height of center of gravity(m)	1.50	1.50
Vehicle mass (kg) (empty/ loaded)	34000 49000	34000 49000
Primary suspension Longitudinal stiffness of leading wheelset (N/m)	1450000	5000000
Longitudinal stiffness of trailing wheelset (N/m)	12600000	5000000
Lateral stiffness (N/m)	5000000	5000000
Vertical stiffness (N/m)	1430000	1430000
Vertical damping (Ns/m)	3000	3000
Secondary suspension Longitudinal stiffness (N/m)	100000	100000
Longitudinal damping (Ns/m)	96000	96000
Lateral stiffness (N/m)	100000	100000
Lateral damping (Ns/m)	50000	50000
Vertical stiffness (N/m)	300000	300000
Vertical damping (Ns/m)	20000	20000
Track gauge (m)	1.435	1.435
Rail transverse radius (m)	0.30	0.30

series with torque being the principal parameter. Torque was varied by step of 1000 Nm per axle starting from zero. Maximum value of torque (moment) was approximately set by condition that the tangential longitudinal force between wheel and rail, produced by the acting torque, should be smaller or equal to the maximum friction force.

$$M_{max} \leq \frac{m_v.g.f}{n}.r$$

where:

f -friction coefficient between wheel tread and rail (−)
g -gravity acceleration (ms^{-2})
M_{max} -maximum moment acting on axle (Nm)
mv -vehicle's mass (kg)
n -number of axles (−)
r -centered wheel rolling radius (m)

When substituting, following values of maximum torque are obtained:

PLW	empty (36 t)	9 380 Nm
	loaded (49 t)	13 520 Nm

2.2. Simulation results

The typical results of simulation calculations are shown in the following figures. For better clarity, 3D graphs have been used. Similar basic dependencies, as in case of the JNG vehicle (defined in the first report), can be observed. Up to the certain value of acting torque, the increase of wheel lateral displacement with torque is only slight. Then a steep increase of wheel lateral displacement can be observed. Traction and braking torque always influenced the curving behavior very similarly, so only results with positive (traction) torque are presented.

In Fig. 1 wheel lateral displacement of vehicles with and unsymmetric longitudinal suspension characteristics (a) and symmetric (conventional) (b) are shown. When no torque is acting, the wheel lateral displacement of the unsymmetric truck is only about half of the value for the symmetric truck. Flange contact is reached with torque of 5000 Nm per axle in the case of symmetric truck, while it is 8000 Nm in the case of unsymmetric truck. Better curving performance of the vehicle with unsymmetric truck (a) is apparent.

From Fig. 2, effect of the vehicle mass can be observed. In the same curve, flange contact is reached with higher value of torque in case of "loaded" vehicle (b). The ratio of torque at which the flange contacts are reached and the ratio of vehicles masses are approximately same (7:11 ≅ 34:49). Similar conclusion was found in the first report.

In Fig. 3 almost zero attack angle within wide range of torque can be observed for the unsymmetric truck (a), while

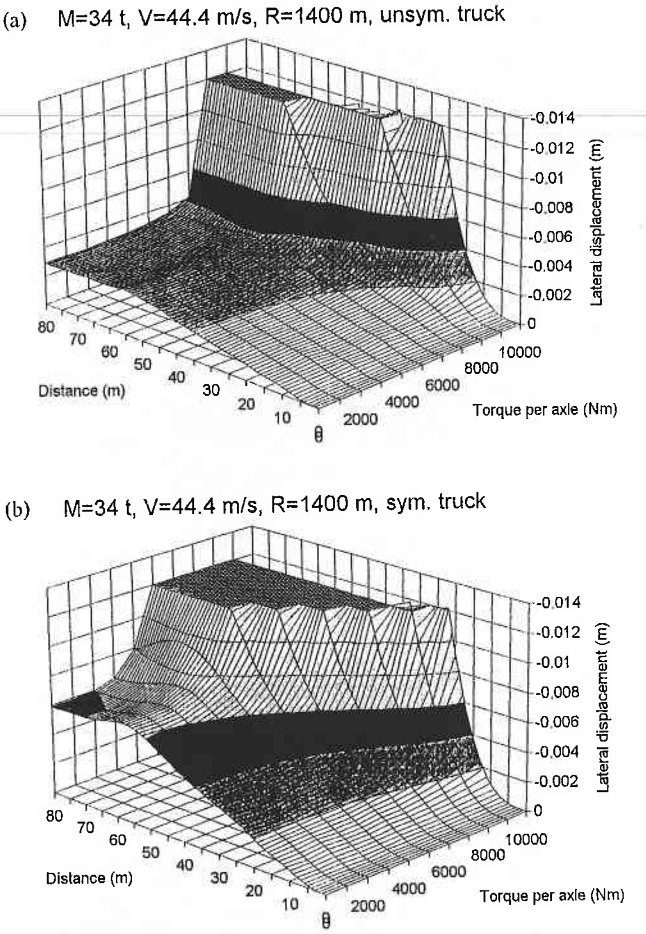


Fig. 1 Simulation results, PLW, wheel lateral displacement of vehicles with symmetric and unsymmetric trucks.

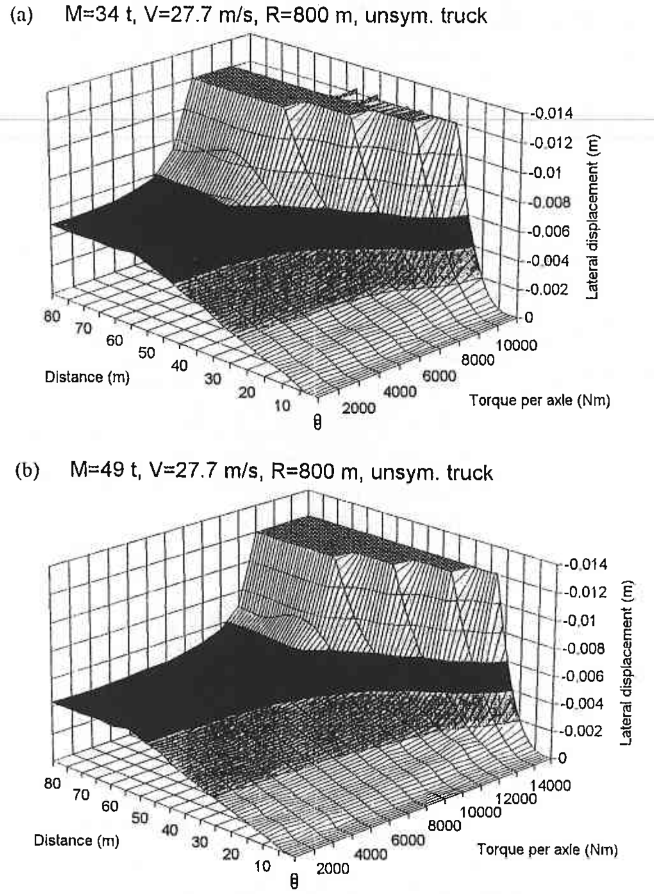


Fig. 2 Simulation results, PLW, wheel lateral displacement; effect of vehicle mass.

the symmetric truck (b) has considerably higher values of attack angle and more rapid growth of attack angle.

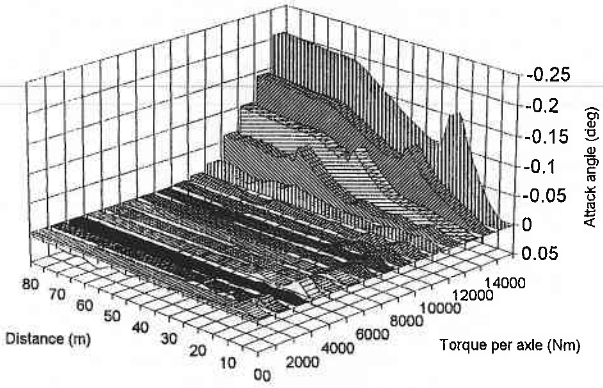
In Fig. 4 curving behavior of the vehicle with unsymmetric trucks in another two curves with 600 m (a) and 2000 m (b) radii are shown. One can compare also with Fig. 1 (a) and Fig. 2 (a) to see the truck behavior in wider range of curves. With smaller curve radius, the wheel lateral displacement is lerger and flange contact is reached earlier.

3. CONCLUSIONS

Influence of torque on rail vehicles curving performance has been studied. The A'GEM Rail Vehicle Dynamics Software Package has been used for simulations. Vehicle with unsymmetric truck design has been compared with the conventional one. The basic conclusions from comparison are following: The new unsymmetric truck design proved its enhanced curving properties; wheel attack angle was almost zero within wide range of torque; wheel lateral displacement was considerably smaller than that of symmetric truck.

(Manuscript received, March 15, 1994)

(a) - M=49 t, V=27.7 m/s, R=800, unsym. truck



(b) M=49 t, V=27.7 m/s, R=800, sym. truck

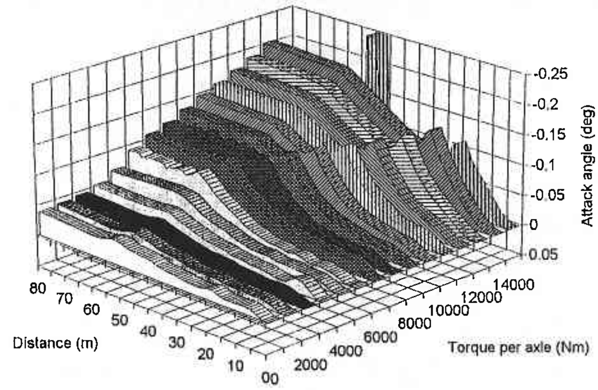
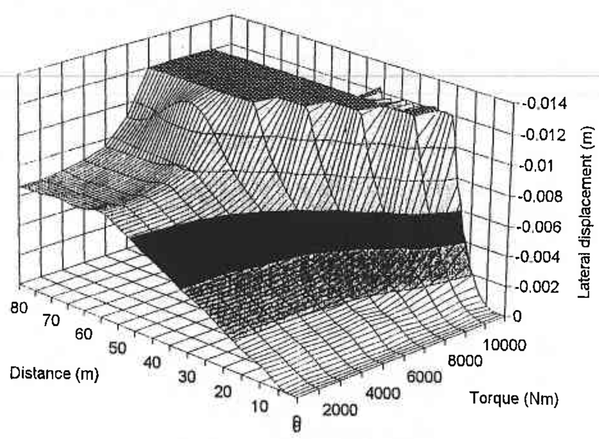


Fig. 3 Simulation results, PLW, wheel attack angle.

(a) M=34 t, V=27.7 m/s, R=600 m, unsym. truck



(b) M=34 t, V=44.4 m/s, R=2000 m, unsym. truck

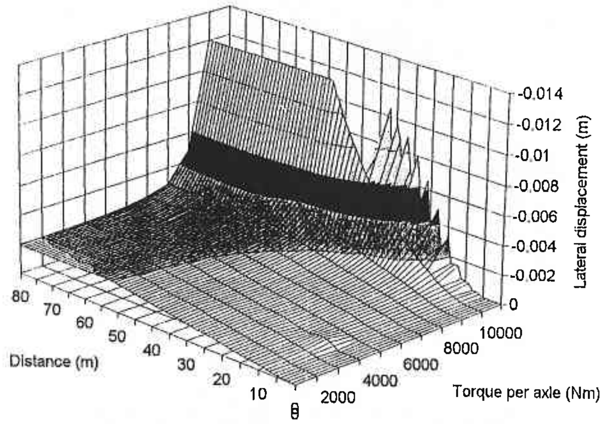


Fig. 4 Simulation results, PLW, wheel lateral displacement; effect of curve radius.

References

1) Grencik, J., Suda, Y., "Influence of Traction/Brake Torque on Curving Dynamics of Trucks with Unsymmetric Suspension of Rail Vehicles (1st Report, Effects of Track and Operating Conditions)", Seisankenkyu 1994 46–6, No 536, pp. 63–66.

2) Suda, Y., Anderson, R. J., "Dynamic Behavior of Longitudinally Unsymmetric Trucks for Rail Vehicles in Curve Negotiation", CSME Forum "Transport 1992+", Concordia Univ., Montreal, pp. 688–693.

3) Anderson, R. J., Fortin, C., "The A'GEM Rail Vehicle Dynamics Software Package User's Manual," Dynamics Laboratory Report No. DL/RJA/90/1, Dept. of Mech. Eng., Queen's Univ., 1990.

4) Smith, R. E., Anderson, R. J., "The Case for Design and Development of a New High Speed Train Specifically Suited for North America", ASME WAM 91, RTD-Vol. 4, (1991) pp. 137–141.