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

——1st Report; Effects of Track and Operating Conditions——

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1報 軌道および走行条件の影響——

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1. INTRODUCTION

A question arose, how is the curve negotiation affected when the traction or braking torque is applied. In an adhesion trains operation, that means trains with traction/braking force being transmitted through contact force between vehicle wheels and rail, this situation is indispensable since every train has to accelerate and decelerate its movement. Moreover, running uphill or downhill results in longer traction or braking torque actions. Although many studies describing curving behavior of rail vehicles have been done, torque has usually not been considered. The question of traction/braking torque influence on curving performance was important also from the point, whether the improved curving performance of the recently suggested unsymmetric trucks[1)] will not be badly deteriorated by acting torque.

Usually, only steering force under simplified conditions used to be calculated. The A'GEM Rail Vehicles Dynamic Software Package[2)] enables simulation of full vehicle body dynamics with considering non-linear characteristics of the contact force between wheel and rail. It also enables to calculate the effects of torque applied on axles which produces a longitudinal wheel/rail force.

To describe the rail vehicle curving behavior, wheel lateral displacement and attack angle were chosen as the typical characteristics. Since a leading axle in a leading truck, in general, has the biggest lateral displacement and attack angle, only the characteristics of leading axle were used to demonstrate the curving behavior under acting torque.

In this study, the Japanese narrow gauge railway vehicle having truck with longitudinally unsymmetric stiffness characteristics of the primary suspension has been modeled. Curving performance under various vehicle velocities and various curve radii has been calculated. Influence of vehicle's mass (load) was in question, too.

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2. OBJECTS STUDIED AND MODELS

2.1. Vehicle in curve

Tapered (conical) railway wheels rolling on the pair of rails are seemingly simple mechanical couples. But wheel and rail contact geometry and acting contact forces with nonlinear creep friction characteristics make this case rather complicated.

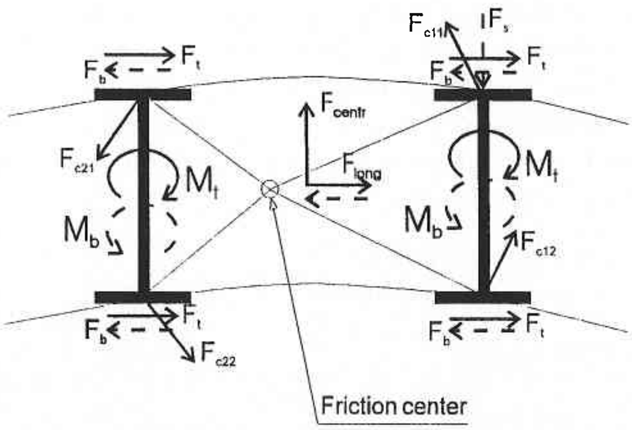

Fig. 1 External forces acting on a truck in curve

Schematically, the forces acting on a truck in curve are shown in the Fig. 1. When running in curve, creep forces (F_{c11}, F_{c12}, ...) and flange forces F_s can be observed. If the truck is rigid, axles remain parallel and there exists a point about which the moment due to wheel-rail forces and external forces sums to zero[3]. Although in the study an unsymmetric trucks with self steering leading axle are examined, for clarity, only the case with rigid axles is taken to show directions of creep forces. Creep forces depend on longitudinal and transversal creepages, while this dependence is nonlinear and differs for longitudinal and transversal creepages. Flange force obviously occurs in sharper curves with smaller curve radii. Here, only a flange force on a leading axle is shown although case with flange force on a trailing axle (not typically) can occur, too. In curve dynamics calculations, usually only creep and flange forces (F_c, F_s) and centrifugal forces (F_{centr}) have been considered. In our case, traction torque M_t and braking torque M_b are considered, too. Torque actions result in longitudinal forces F_{long}, acting between truck and carbody, and tangential contact forces F_t, F_b, acting between wheels and rails.

In the study, truck behavior during entering the curve from the straight track was examined. Wheelsets from their initial centered position are moving aside. Curves with large radius may be negotiated without a flange contact. In sharper curves, the flange contact normally can't be avoided. However, this is very undesirable case causing wear of wheel tires and rail heads, not to mention noise being emitted from the flange contact.

2.2. Unsymmetric truck concept

To improve curving performance, vehicles with longitudinally unsymmetric bogies having different primary longitudinal stiffness of leading and trailing axle both in leading and trailing trucks have been recently suggested[1]. These trucks have theoretically (under simplified conditions) perfect curving ability. If values of primary longitudinal stiffnesses satisfy the following relation, then curve is negotiated with zero attack angle:

$$\frac{a_s}{a} = \frac{k_{x2}.b_x^2.r}{2.b.a.f_{11}.\lambda + k_{x2}.b_x^2.r}$$

where:

$$a_s = -a.\frac{k_{x1} - k_x}{k_{x1} + k_{x2}}$$

a -half of wheel base (m)
a_s -unsymmetric stiffness index (m)
b -half of wheelset/rail contact distance (m)
b_x -half of primary longitudinal suspension distance (m)
f -friction coefficient between wheel tread and rail (−)
f_{11} -creep coefficient (N)
g -gravity acceleration (ms^{-2})
k_{x1} -primary longitudinal stiffness of leading axle (Nm^{-1})
k_{x2} -primary longitudinal stiffness of trailing axle (Nm^{-1})
r -centered wheel rolling radius (m)
λ -wheel equivalent conicity (−)

In the Fig. 2 a basic concept of the truck with longitudinally unsymmetric suspension is presented. Here the truck with a switched yaw damper of primary longitudinal suspension or with actuator are shown. Switched damper or actuator are used to enable bidirectional operation, so that leading wheelset has soft and trailing wheelset has hard primary longitudinal suspension stiffness.

2.3. Vehicles and track parameters

A Japanese narrow gauge vehicle (further "JNG") with unsymmetric truck was modeled. The basic parameters of the vehicle are as follows: wheelset mass −1540 kg; wheel diameter −0.86 m; truck mass −3661 kg; wheel base −2.20 m; longitudinal stiffness of leading wheelset −2.94.10^6 N/m; longitudinal stiffness of trailing wheelset −17.6.10^6 N/m. Car body mass was considered a parameter, that is why "empty" and "loaded" car of mass 22 645 kg, resp. 45 290 kg were considered. Total mass of "empty" vehicle was 36 127 kg and 58 772 kg of "loaded" vehicle.

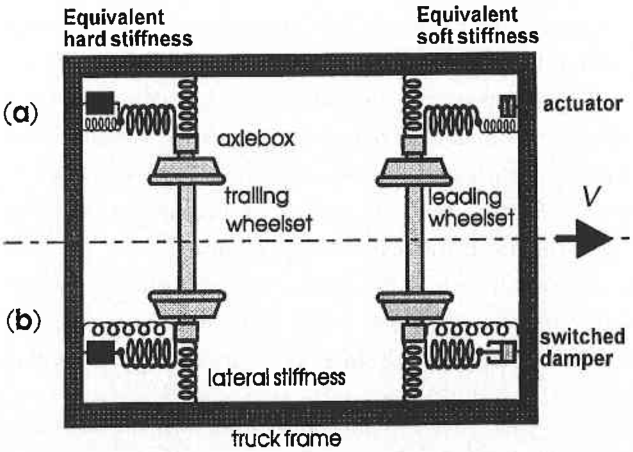

Fig. 2 Basic concept of the truck with longitudinally unsymmetric suspension

Simulation model of 21 degree of freedom was used, plus 4 degrees of freedom were added to allow wheelsets rotate around their axles so that torque could be modeled. In this study, conical wheel profile with conicity 0.15 was considered. Friction coefficients of wheel tread and flange were estimated to be 0.30 and 0.25 respectively.

Two curve radiuses of 800m and 1400m were chosen, both with the rail cant of 0.062 rad. Train velocities were 20 m/s and 30 m/s respectively. In the first case the centrifugal forces are over balanced (negative cant deficiency of −0.011), in the second case centrifugal forces are slightly under balanced (cant deficiency of 0.005). Length of transitional curve from straight track to curve with constant radius was set to be 25 m and total length of simulated track was 40 m.

3. RESULTS OF CALCULATIONS

Series of simulation calculations have been carried. The torque, being a basic parameter, was varied by steps of 2000 Nm per axle, starting from zero toque. Maximum value of torque (moment) was 12000 Nm per axle in case of empty vehicle and 16000 Nm per axle in case of loaded vehicle. These values were approximately set by condition that the tangential force between wheel and rail, produced by the acting torque, should be smaller or equal to the maximum friction force.

Some typical results of JNG simulations are shown in Fig. 3. Wheel lateral displacement is displayed in the distance step of 8 m. In all cases wheel after entering the curve moves aside from its initial centered position (zero lateral displacement). Influence of torque on wheel lateral displacement is clearly presented in all cases. From (a) and (b) influence of traction and braking torque can be compared. In the first half of the torque range, the values of lateral displacement are almost identical. Only with higher torque, in case of braking (negative torque) flange contact is reached earlier than in case of traction (positive torque). This result was typical for each case that has been calculated, so further only cases with positive torque are shown. In (c) the same curve but "empty " vehicle is shown. Flange contact is reached with torque of 10 kNm, compared to value of 16 kNm in previous cases. One can see that the torque ratio when flange contact is reached and the ratio of vehicles masses are approximately same (10:16 ≅ 36:59). In (d) the lateral displacement in curve with smaller radius and lower vehicle velocity is shown. Here, the lateral displacement without torque is bigger and flange contact is reached with smaller torque.

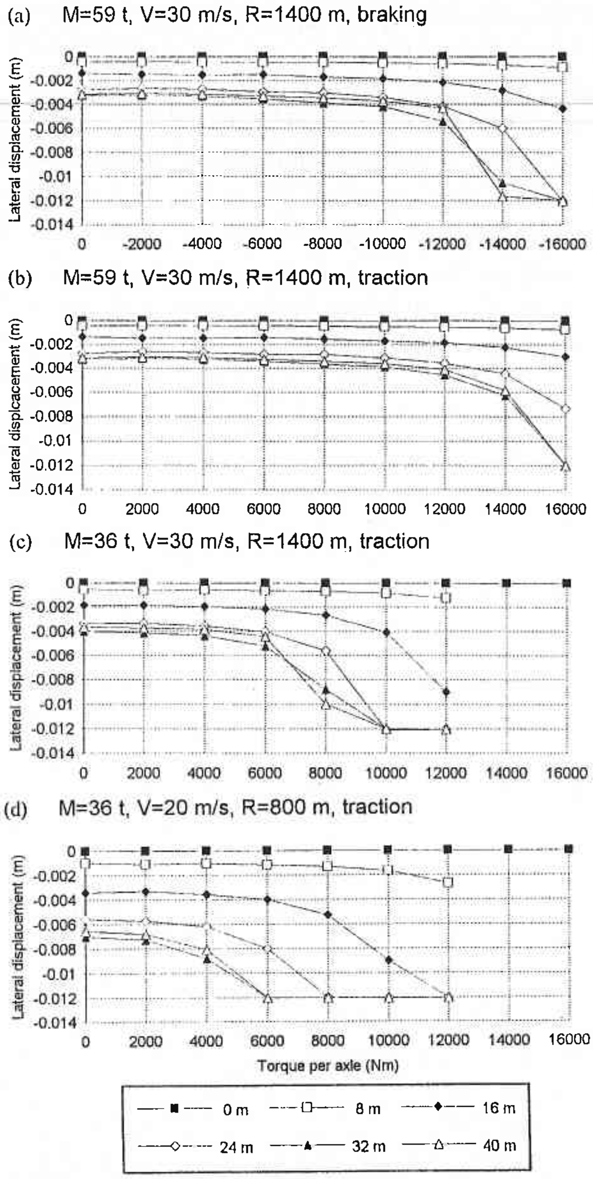

Fig. 3 Simulation results, JNG wheel lateral displacement

4. 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 modeled. The basic conclusions are following: Torque acting on wheelsets deteriorate vehicle's curving performance. Flange contact is reached with increasing torque. The increase of wheel lateral displacement is not linear with increasing torque. In the beginning the growth is small and only for higher values of torque rapid increase of displacement is observed. Under the same curve conditions, this increase depends on the vehicle's mass. Traction and braking torque affect the curving performance almost identically, but the rapid increase of wheel lateral displacement starts with lower torque in case of braking.

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