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Axle Gearbox Hinge as a Diagnostic Member for Rail Vehicle Drive

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Abstract. Railway transport offers high travel comfort, shipping speed and efficiency. Due to the high speed and large number of transported persons, it is necessary to ensure the safety in the best possible way. Apart from the legislation and regular service, the continuous monitoring during the operation is carried out. Two basic types of the drive are usually used for the public transport – partly and fully sprung drive. This contribution deals with the partly sprung drive, exactly with the possibilities of the usage of the axle gearbox hinge as a diagnostic member for the bogie group including the drive.

Introduction

Railway transport offers high travel comfort, shipping speed and efficiency. Due to the high speed and large number of transported persons, it is necessary to ensure the safety in the best possible way. Apart from the legislation and regular service, the continuous monitoring during the operation is carried out. Two basic types of the drive are usually used for the public transport – partly and fully sprung drive. This contribution deals with the partly sprung drive, exactly with the possibilities of the usage of the axle gearbox hinge as a diagnostic member for the bogie group including the drive, [1].

Partly sprung drive

Partly sprung drive consists of the traction engine (usually asynchronous motor) and transmission with axle gearbox. The engine is embedded on the bogie frame and can be taken as a suspended mass. The big gear of the axle gearbox is embedded directly on the wheelset axle. The reaction force caused by the input and output torque inequality is captured by means of the hinge, which is connected to the bogie frame. Thus, the axle gearbox becomes the partly suspended mass. Partly sprung drive is simpler and more economical than the fully sprung drive. On the other hand, it is less suitable on rail tracks with worse quality due to the larger amount of non-suspended mass. Thus, it is important to analyse the dynamic behaviour of such drive. Partly sprung drive is usually used in combination with one-stage gearbox, two-stage gearbox or the gearbox with inserted gear. One-stage gearbox has simple design and also small width, the schema can be seen in Fig. 1. It is used for example in Prague metro *M1*, Fig. 2.

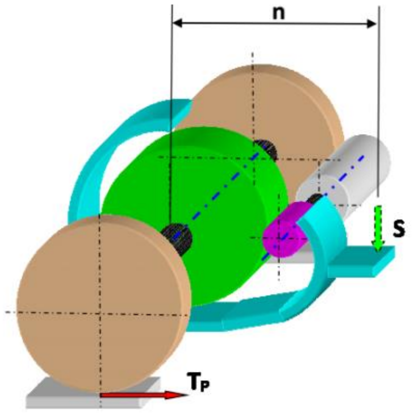

Fig.18 Schema of the partly sprung drive with one-stage gearbox, [1]

Fig.19 Prague metro M1 drive, [2]

The gearbox with inserted gear is used when larger axis distance is required due to the size of the engine. This type can be found in the high-speed train *AGV* drive, Fig. 3. For higher ratios two-stage gearbox is used, e.g. Prague tram *Škoda 14T*, Fig. 4.

Fig.20 AGV high-speed train, [3]

Fig.21 Škoda 14T tram, [4]

The gearbox hinge is very important and interesting part from the dynamic point of view. It is the link between the spatial and torsional dynamics. Previously, it was successfully used for the measurement of various quantities such as the input torque [1, 5, 6]. During these measurements the hinge served as a sensor and an idea to use the hinge as a diagnostic member for rail vehicles with partly sprung drive originated. This diagnostic member should monitor the behaviour of the bogie during the operation of the vehicle and the in-time signal processing should reveal the changes in the bogie group structure, such as damages of the suspension. The hinge could be used also for the predictive service of the vehicle drive. The reaction force in the hinge of the one-stage axle gearbox (Fig. 1) is defined as

$$S = \frac{M(i_c + 1) - J_p\ddot{\psi}}{n}, \qquad (1)$$

where M is the input torque, i_c is the gearbox ratio, J_p is the gearbox moment of inertia, ψ is the angle of the gearbox sway and n is the distance between the hinge and wheelset axle.

Dynamic simulation model

To prove and verify above mentioned proposals the dynamic simulation model of the partly sprung drive is assembled. It is a non-linear dynamic model of a single wheelset, which is mounted on the roller test rig [7, 8, 9, 10]. This test rig enables torque loading of the drive and transversal and vertical excitation of the wheelset. The parametric model describes both spatial and torsional dynamics of the system and includes eighteen degrees of freedom.

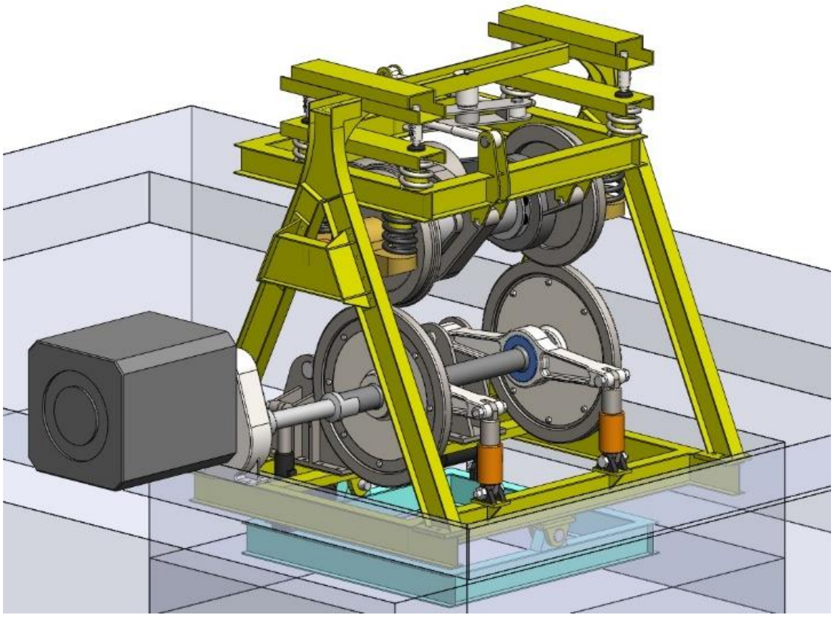

Fig.22 Roller test rig, [1]

Spatial Dynamics. Spatial dynamics of the wheelset and the bogie frame (see Fig. 6) is modeled with consideration of all degrees of freedom and geometrically non-linear. Free body diagrams are defined in general positions, thus all the contributions of forces and moments are included in the equations and higher order expressions are not neglected. Detail description was introduced in [11].

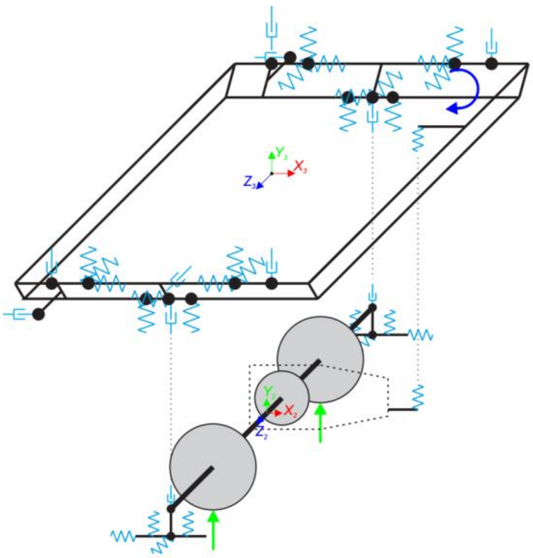

Fig.23 Spatial dynamics – schema [1]

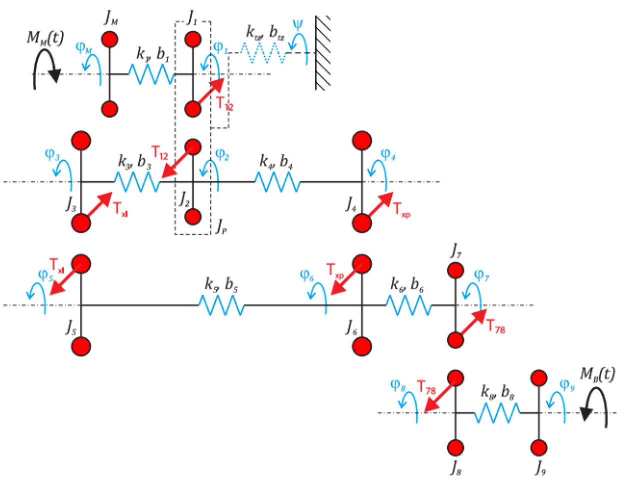

Fig.24 Torsional dynamics – schema, [1]

Torsional Dynamics. The torsional system (see Fig. 7) is consisted of the asynchronous motor, gear coupling and one-stage axle gearbox. The gear ratio is realised by the pinion and big gear, that is mounted to the wheelset. The gearbox is not in the geometrical centre of the wheelset so the stiffness of the left and right part of the axis is different. The driving forces are transmitted from the wheels to the roller shaft, which is coupled through the propeller shaft to the one-stage reduction gearbox and loading motor. The axle gearbox is connected with the bogie frame by means of the hinge, see Fig. 8. Thus the hinge is influenced both by the spatial and torsional dynamics.

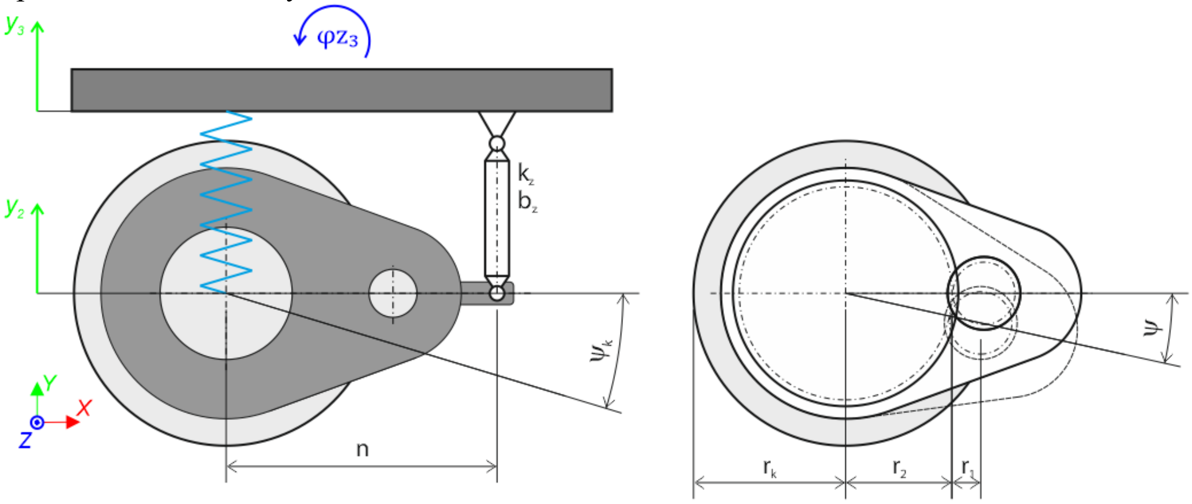

Fig.25 The schematics for the gearbox sway, [1]

The total angle of sway ψ is given as a combination of the sway due to the input torque ψ_m and the sway due to the kinematic excitation ψ_k (see Fig. 8)

$$\psi = \psi_m + \psi_k = \frac{\varphi_1 - \varphi_2 i_c}{1 + i_c} + \frac{y_2 - y_3}{n} - \varphi_{z3}, \qquad (2)$$

where φ_1, φ_2 and φ_{z3} are angles of rotation of the pinion, big gear and the bogie frame respectively, y_2 and y_3 are the vertical deflections of the wheelset and the bogie frame. This movement also influences the forces in gear meshing, the tangential force T_{12} can be derived from the dynamic equation of equilibrium for the gearbox

$$T_{12} = \frac{J_P\ddot{\psi} + b_z n^2 \dot{\psi} + k_z n^2 \psi}{r_1 + r_2}, \qquad (3)$$

where J_p is the moment of inertia of the axle gearbox, ψ is the angle of rotation, k_z, b_z is the stiffness and the damping of the hinge, r_1, r_2 are radii of the pinion and big gear, n is the distance between the hinge and the axle.

System of equations. The final set of motion equations was assembled in program *Maple* and exported into program *Matlab*. Eighteen second-order differential equations are solved there by means of *Matlab* function for solving moderately stiff ordinary differential equations *ode23t*. The dynamic model is fully parametric and can be used for testing the dynamic response with various parameters of stiffness, damping, dimensions etc.

Testing and results

The dynamic model was tested in various test regimes – torsional loading of the drive, vertical oscillations of the wheelset (simulation of the drive on the rough rail track) and the lateral movement of the wheelset, [1, 11]. For testing diagnostic abilities of the hinge, various test simulations are performed and its response – axial reaction force behaviour is analysed. In this contribution, the changes/damages in primary and secondary suspension are carried out. Exactly, the stiffness of one primary or secondary spring is reduced. The first test case is performed during the drive on the rough rail track. The roughness is simulated by the sum of sine functions

$$h_i(t) = A_1 \sin 2\pi f_1 t + A_2 \sin 2\pi f_2 t + A_3 \sin 2\pi f_3 t, \tag{4}$$

where the amplitudes A_i are 2 mm, 3 mm and 1 mm, the frequencies are dependent on the driving speed corresponding to the wave lengths 50 m, 20 m and 5 m and they are listed in the Table 1.

Tab.1 Frequencies of the rail track roughness for the wheelset excitation

v [km/h]	f_1 [Hz]	f_2 [Hz]	f_3 [Hz]
54	0.3	0.75	3
90	0.5	1.25	5

During the following test, one of four primary springs is taken as damaged, it is simulated with the 10% stiffness. The test is performed with drive speed 54 km/h and 90 km/h. To compare obtained results it is convenient to transform the time behaviour of the hinge force to the frequency domain. The results for both drive speeds can be seen in Fig. 9 and Fig. 10.

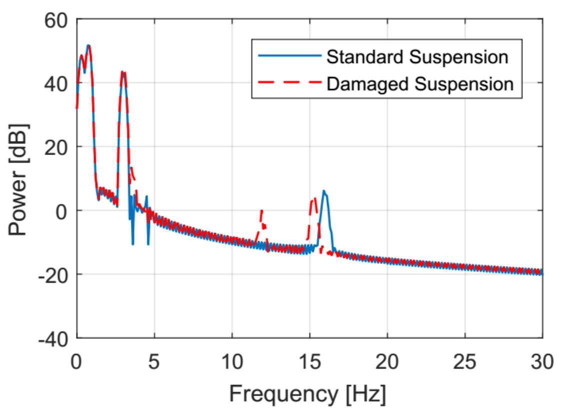

Fig.26 Signal comparison of standard and damaged primary suspension during drive on rough rail track, v = 54 km/h

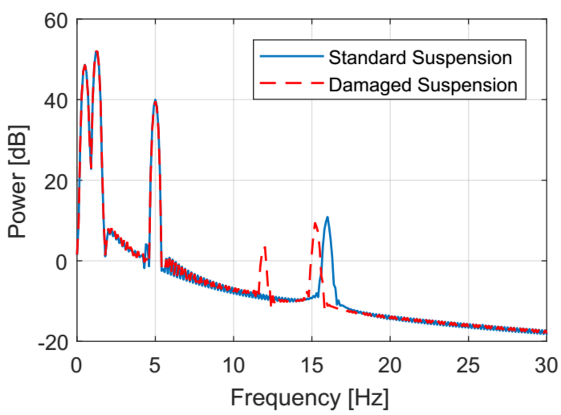

Fig.27 Signal comparison of standard and damaged primary suspension during drive on rough rail track, v = 90 km/h

There is appreciable difference between the spectrum with standard and damaged spring. In both cases, there can be seen three excitation frequencies from the roughness of the rail track. If the spring is damaged, there can be seen the change in the frequency spectrum – one frequency round 16 Hz moves to the lower frequencies and furthermore new one appears at frequency round 12 Hz. Both frequencies correspond to the natural frequencies of the bogie frame. The frequency round 16 Hz is related to the vertical oscillations of the bogie frame and it could be simplified calculated by means of following formula

$$f = \frac{1}{2\pi}\sqrt{\frac{k_p + k_s}{m}}, \tag{5}$$

where k_p and k_s is the stiffness of the primary and secondary suspension respectively and m is the mass of the bogie frame. Similar settings are set during the test with damaged secondary suspension, which is again simulated with 10% stiffness. Results for both drive speeds can be seen in Fig. 11 and Fig. 12. There can be seen equivalent behaviour of the hinge response in frequency domain, even the changes are not so significant.

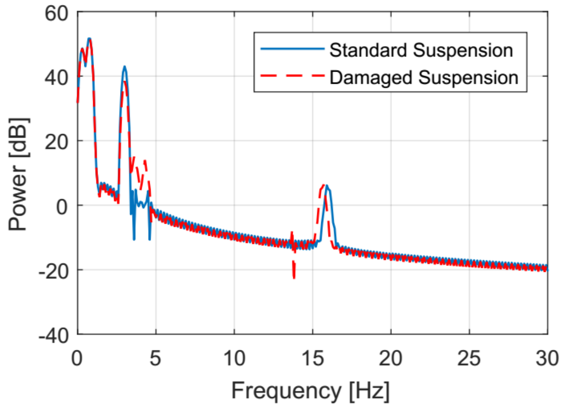

Fig.28 Signal comparison of standard and damaged secondary suspension during drive on rough rail track, v = 54 km/h

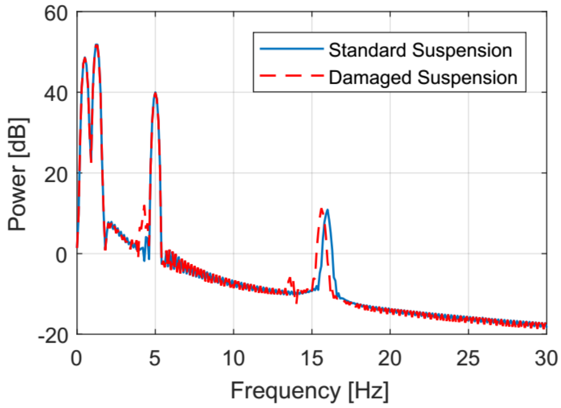

Fig.29 Signal comparison of standard and damaged secondary suspension during drive on rough rail track, v = 90 km/h

In the third test, there is investigated the response during lateral movement of the wheelset during the drive speed 90 km/h. Both primary suspension and secondary suspension spring stiffness is reduced to the 90%. The changes in the stiffness is much smaller than in previous cases. In figures Fig. 13 and Fig. 14, there can be seen significant changes in the response spectra. Both in the frequency and in the power magnitude.

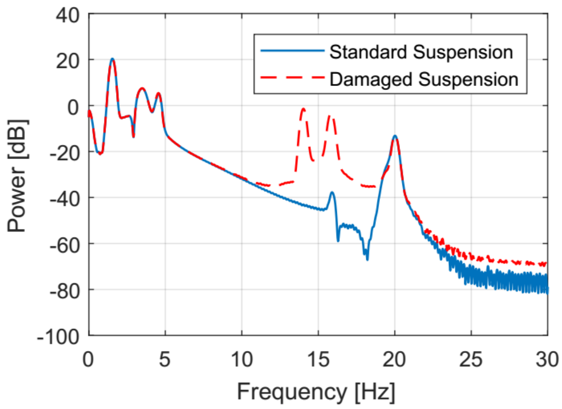

Fig.30 Signal comparison of standard and damaged primary suspension during lateral movement of the wheelset, v = 90 km/h

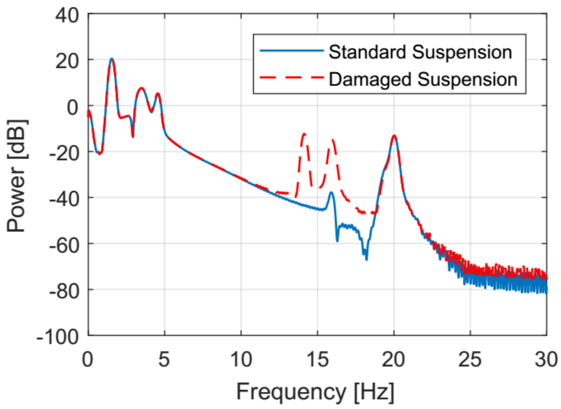

Fig.31 Signal comparison of standard and damaged secondary suspension during lateral movement of the wheelset, v = 90 km/h

Conclusions

In this contribution, there was introduced the problematics of drives especially the partly sprung drive with axle gearbox hinge. The specific partly sprung drive was described by means of the dynamic model. It was used for testing of the drive in various test regimes with different parameters of the suspension. That should simulate possible damages of the suspension springs during the operation of the vehicle and their in-time detection. From above mentioned facts and results follows that the axle gearbox hinge in various partly sprung drives can be conveniently used as and diagnostic component. It is also important aspect that the hinge can be used for diagnostics without considerable changes of its design. The diagnostic hinge could be used also for data acquisition, which is helpful in designing new drives and bogies. The hinge as a diagnostic component should serve also for predictive service. That can significantly help to prevent damages of drives and bogies by means of early warning.

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