8th European Workshop On Structural Health Monitoring (EWSHM 2016), 5-8 July 2016, Spain, Bilbao
www.ndt.net/app.EWSHM2016

A Novel Full Scale Roller Rig Test Bench for SHM Concepts of Railway Vehicles

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Abstract

Bogies are the core part of railway vehicles. The primary function is to carry the structure of the car body and to lead the railway vehicle on the track. Thus, it has a high impact on the dynamics of the whole train and due to its complex technical structure a huge number of failure cases can occur. Beside safety also economic motivations cause an increasing demand for reliable and robust monitoring systems to ensure the integrity of the main bogie parts. For the investigation of modern SHM systems a concept for a novel roller rig test bench is presented in this contribution. The roller rig test bench is designed for full scale bogies. With a maximum speed up to 300 km/h it is possible to investigate the dynamics of the bogie in a laboratory environment. It is also possible to analyze the behavior of the bogie on a curved track, since the design of the test bench allows a flexible configuration of the rail rollers. With this test bench, experimental simulation of different failure scenarios like broken spring or damper components can be conducted and new SHM concepts can be developed. In this paper, a SIMPACK MBS model is used to simulate different damage scenarios. For the development of a novel SHM concept different recently proposed strategies for damage detection and structural health monitoring using signals of acceleration sensors placed at different parts of the bogie are discussed. Based on the comparison of existing approaches a novel model-based method is introduced. Therefore a subspace-based identification algorithm is used to identify the eigenfrequencies of the system. To exemplify the applicability the front left spring is considered. The identified bogie pitch eigenfrequency can be used to distinguish between the healthy and the faulty state of the spring.

Keywords: modeling and simulation, railway, roller rig, defect characterization, subspace identification

1. INTRODUCTION

The derailment of a train is one of the most dangerous accidents for railway vehicles, due to its threat for the life of the passengers and the strong impact on the environment and infrastructure. Beside other causes, failures of bogie components can cause such a derailment. In the D-RAIL project, which focuses on the development of future rail freight systems and aims to reduce the risks and consequences of derailments, the main causes for derailment in

Europe were identified. Broken springs and failed suspensions are one of the ten most frequently occurring accident reasons [1]. Monitoring of critical components is one solution to prevent derailments and will be the main aspect of this publication. Suspension elements in bogies are undoubtedly important for providing both running safety and comfort. Resulting in uneven wheel loads their failure increases wear on wheels, flanges, rails and also the probability of a derailment. For the identification of failure often state estimation techniques based on a Kalman filter [2], a Rao-Blackwellized particle Filter [3], or banks of multiple Kalman filters [4] are used. These methods require an accurate mechanical model of the train system. Other approaches as surveyed in [5, 6] use data-driven methods to derive monitoring concepts solely on the output quantities of the system. An overview of various vibration-based damage diagnosis methods is also given in [5]. Beside these methods which rely only on the output quantities of the corresponding system in [7] a concept has been proposed that estimates an ARX model with the position of the wheel of the bogie as an additional input quantity. In our contribution a data-driven approach based on subspace identification method [8]-[10], similar to the CVA concept described in [6], will be employed to estimate the poles of the underlying mechanical system of the train. The obtained estimates of the poles will be used as features for the fault-detection strategy. Due to the diversity of engineering problems and the requirement of individual SHM applications, difficulties arise for authorities and governments to implement technical standards and regulations to increase the reliability of these systems. In the energy sector, insurance companies prescribe monitoring systems to the operators. This leads to an important step to launch SHM systems on that market. An opportunity to increase the reliance in powerful and modern SHM systems is the proof of these concepts on real railway parts in a secure environment. In case of a bogie a roller rig can be used. The aim of a roller rig is to test these concepts not only on a coupon or element level (like material testing) or on a parts level (like springs, frames, wheelsets) but on an assembly level, which links the different subparts to a whole module including all relevant parts. Therefore a roller rig is suitable to lift up SHM systems from the research labs to the market. This publication is divided in three parts. The first part covers the summary of a concept to implement a test infrastructure including a novel 1:1 full scale roller rig followed by the introduction of a virtual train model in the second part. In the third part we present a data based SHM suspension failure system and close this contribution with a short conclusion.

2. CONCEPT OF SUSTAINABLE MOBILITY - INTELLIGENT BOGIES FOR A RESOURCE-EFFICIENT RAIL TRANSPORTATION

The proposed roller rig will be part of a test infrastructure in the research framework SAMT. SAMT is an acronym for Sustainable Mobility and bases on four pillars (Fig. 1). The aims are to provide a human and ecologically friendly railway technology and a fast, convenient and reliable transportation of passengers and goods assuming an economical use of resources. Especially the noise and vibration reduction in inner cities is an important topic with respect to a relief of the individual transportation in metropolises. Therefore, different research fields are required and will be reflected by the four pillars: noise and vibration protection (I), new drive concepts (II), safety / reliability (III), lightweight design (IV). Measurement and optimization of sound transmission and sound emission is an important topic for an environmentally compatible design of railway vehicles and will be the core part of pillar I. Noise mechanisms have to be explored and correlated with vibrations patterns of bogie and wheels. Pillar II deals with new drive and braking concepts like the optimization of the bogie frame and frame-less bogies. Pillar III covers the typical SHM research field. Permanent monitoring

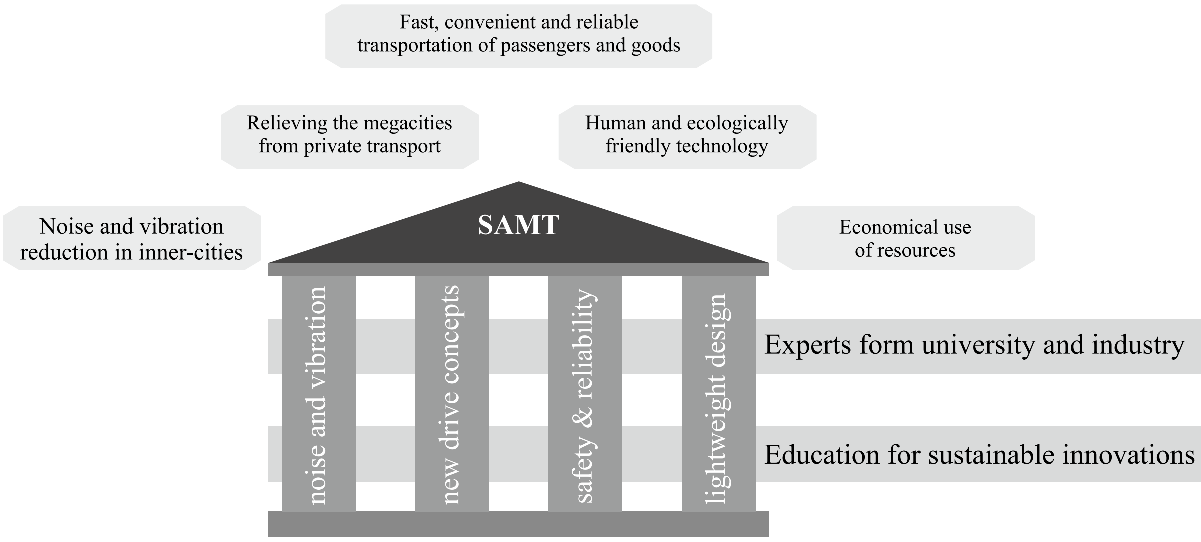


Figure 1: SAMT (Sustainable Mobility) - A four pillar concept

of safety relevant components of a bogie using data from integrated additional sensors helps to prevent unforeseen stops, unplanned maintenance or severe accidents in passenger and freight transportation. The last pillar deals with the investigation of new materials and the potential for replacement of traditional materials (predominantly steel) by other (lightweight) materials, e.g. by composites. Also the fatigue testing under very high cycle fatigue (VHCF) conditions is important with respect to the long operational life of trains. Besides the research topics which will be processed in a strong collaboration of universities, research institutes and experts from the industry it is important to care about a high quality education in the engineering disciplines.

2.1 Full Scale Roller Rig Test Bench

The depicted test bench in Fig. 2 is a concept of a novel 1:1 scale roller rig for railway bogies. It consists of six main modules: drive train (I), rail roller (II), loading unit (III), curve unit (IV), test specimen (V), adapter (VI). The design parameters are based on tram and regional train bogies which will be the focus in the future research, because there is a strong interest in improvements according to noise and vibration problems in inner cities and the surrounding country sides. Due to their special requirements caused by high velocities and

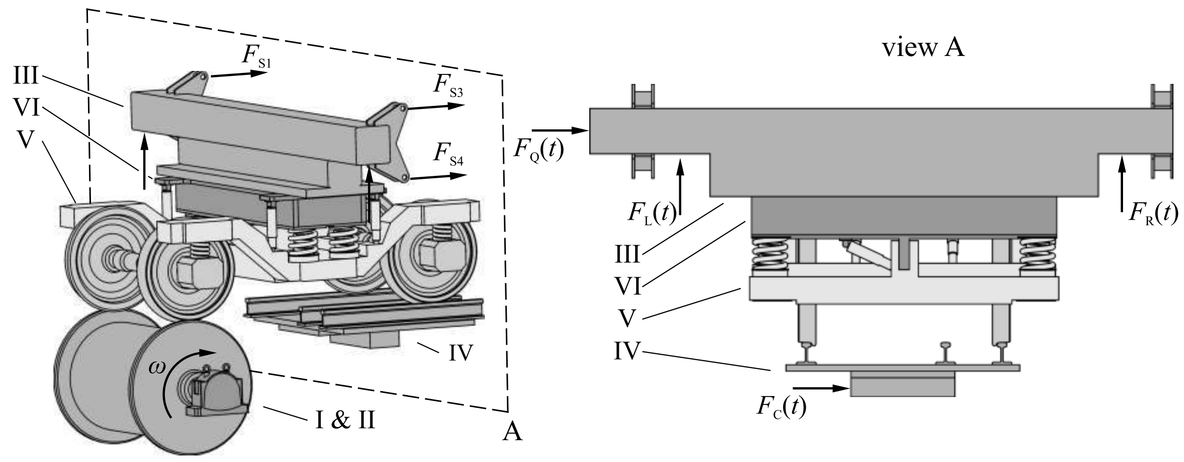


Figure 2: Concept of a 1:1 scale roller rig: drive train (I), rail roller (II), loading unit (III), curve unit (IV), test specimen (V), adapter (VI), vertical force (F_R, F_L), lateral force (F_Q), curving force (F_C), stabilizers (F_{S1} ... F_{S4})

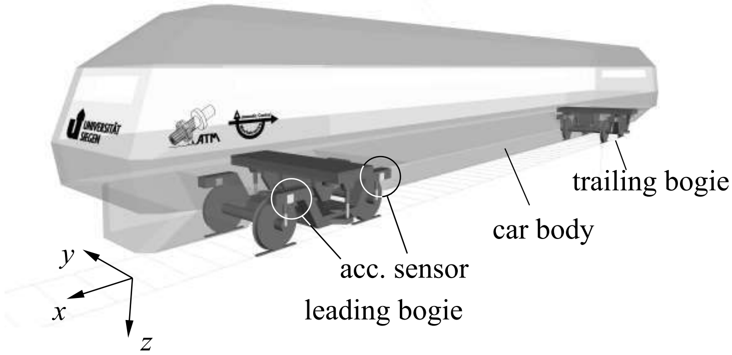
very high loads high-speed-train and freight bogies are not the main focus. The objective is not to design a test bench for the whole range of bogies, but design a test bench for the needs of trams and regional train bogies. The drive train (I) will be designed for a max. speed of 300 km/h. This fits the range for trams (0 - 80 km/h) and regional trains (0 - 160 km/h) but also an investigation of high-speed trains in the range about 250 km/h is possible in special cases. The rollers (II) simulate an infinite track with two track gauges (1000 mm or 1435 mm) including the possibility to change the rail profiles easily. To apply a load on the bogie (test specimen, IV) a loading unit (III) is connected via an adapter (VI). Vertical axle loads up to 21 t are dynamically applied by two hydraulic cylinders (F_R, F_L). A hydraulic cylinder (F_Q) with a maximum force of 16 t simulates a lateral load. In conjunction with the lateral movement of the curve unit (IV) it will be possible to investigate a curve ride.

3. MULTI-BODY SIMULATION - MODEL OF TRAIN AND FAILURE SCENARIOS

To study different failure scenarios and to generate acceleration signals a multibody model is used and implemented in the MBS software SIMPACK. The modeled test train is based on the general passenger vehicle 1 of the Manchester Benchmark [12], which is a simplified version of the ERRI B176 benchmark vehicle without yaw dampers. Also the test track and the ground model based on the Manchester Benchmark is used, here it is the track case 1. The physical and geometric parameters are easily available from the literature [13]. In the following sections the MBS structure will be briefly presented, followed by the model strategy of different failure modes and an eigenfrequencies analysis including a comparison between different failure scenarios.

3.1 Model Structure

The whole model consists of the train, the ground and the track model. The test train in Fig. 3 consists of one car body, two bogies (leading and trailing) with two wheelsets per bogie. All bodies are assumed to be rigid with a total number of 41 degrees of freedom (DOF). The lateral car body joint is used as a rheonomic rail track joint. The coordinate system used in this study is shown in Fig 3. The x-axis corresponds to the longitudinal direction, the y-axis and z-axis correspond to the lateral resp. vertical direction. It is a symmetric vehicle with a simple primary suspension and linear viscous dampers (see Fig. 4). The car body is connected via the secondary suspension to the bogie. Each secondary suspension is composed of two lateral and two vertical dampers (${}^{L}c_{s,lat}$, ${}^{R}c_{s,lat}$, ${}^{L}c_{s,vert}$, ${}^{R}c_{s,lat}$), an anti-roll bar (k_{roll}) and a traction rod (k_{trod}), a bumpstop (k_{bump}) and two secondary springs (${}^{L}k_s$, ${}^{R}k_s$) considering a longitudinal and lateral shear stiffness as well a vertical and bending stiffness. The superscripts L and R represent the left and right bogie side. The wheelsets are mounted via two primary



Vehicle Dimensions	
Bogie wheelbase	2.56 m
Bogie pivot spacing	19.00 m
Wheel radius	0.46 m
Rigid Body	
Carbody	32 t
Bogie	2.615 t
Wheelset	1.813 t

Figure 3: University Siegen test train

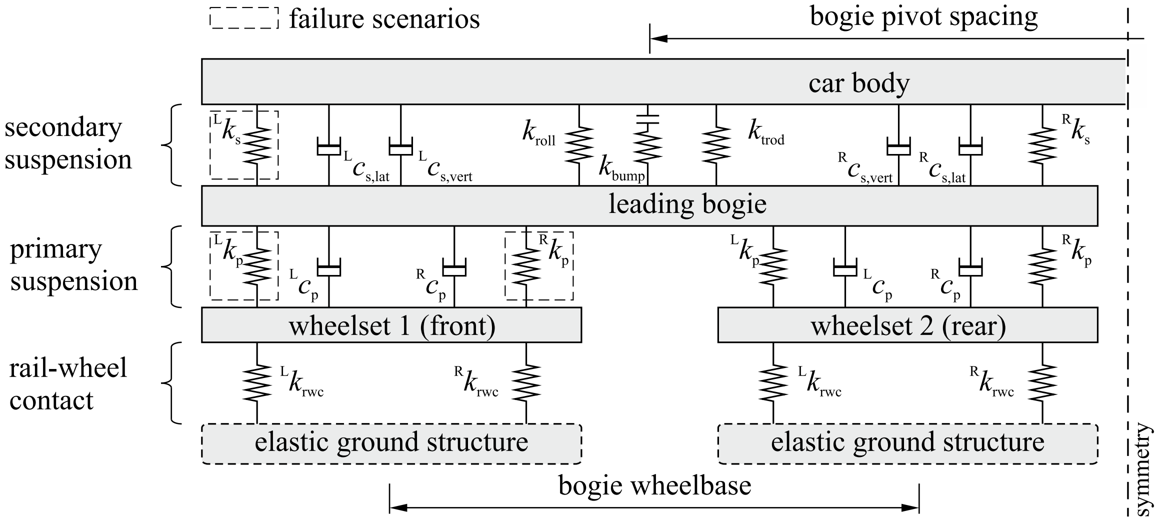
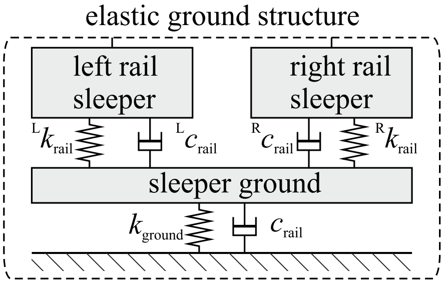


Figure 4: Topology diagram of the test train

suspension units (one left and one right) to the bogie. Each primary suspension unit consists of a primary spring (k_p) considering a longitudinal, lateral and vertical stiffness and a primary damper (c_p). All stiffness and damping rates are assumed to be constant except for the bumpstop; here a non-linear stiffness is implemented. In Fig. 3 a brief summary of the vehicle dimension and weights is given. The complete set of design parameters can be found in [12]. Additionally the test train is equipped with eight three-dimensional accelerometers. Four are placed at the leading bogie and four at the trailing bogie. They are located above the primary spring connections. With a track gauge of 1435 mm, an inward inclination of 1:40, a double elastic ground is implemented (see Fig. 5). Each rail is fixed on a rail sleeper with two DOF (y, z) and connected via a simple spring damper system ($^{L}k_{rail}$, $^{L}c_{rail}$, resp. $^{R}k_{rail}$, $^{R}c_{rail}$) to the sleeper ground. The sleeper ground (k_{ground}, c_{ground}), or ballast mass, is a one DOF system in z-direction. In total 61 DOF are considered. The track line is a low speed track. The vehicle runs at a constant velocity of 4.4 m/s. The 198 m long track comprises a 50 m straight section of track followed by a 30 m linear transition into a curve of 150 m radius with 100 mm of cant. The steady curve lasts 60 m and is then followed by a run-off transition which includes a linear dip of 20 mm over 6 m in the outer rail, 21.56 m after the start of transition. In addition track irregularities will be applied. To calculate the contact forces in the rail-wheel-contact the SIMPACK in-house algorithms are used. The normal forces are calculated with the Hertzian method and can be found in [14, 15]. Tangential contact forces depend on normal forces, the friction coefficient and also the geometry of the contact patch and the stress distribution within the patch and will be calculated using the well-accepted FASTSIM procedure [14, 16].



		Rail sleeper	Sleeper Ground
	Mass	60 kg	500 kg
y	Lat. Stiffness	43.0 MN/m	37.0 MN/m
	Lat. Damping	0.24 MNs/m	0.24 MNs/m
z	Vert. Stiffness	50 MN/m	
	Vert. Damping	0.2 MNs/	

Figure 5: Topology diagram of the elastic ground

3.2 Modelling of Track Irregularities

To apply a random excitation to the dynamic vehicle, track irregularities are considered. A widely used approach in the studies of vehicle dynamics consists in the introduction of random data defined by the power spectral density (PSD) [17, 18]. Vertical and lateral track irregularities on the left and right rail are used. The analytical representation of the PSD is given in [19] and is defined as

$$S(\Omega) = \frac{b_0 + b_2\Omega^2}{a_0 + a_2\Omega^2 + a_4\Omega^4 + a_6\Omega^6} \tag{1}$$

with the wavenumber Ω. The coefficients a_0 to a_6 and b_0 to b_2 are chosen in this publication for a low vertical and horizontal (lateral) excitation.

	b_0	b_2	a_0	a_2	a_4	a_6
horizontal low	$1.440846 \cdot 10^{-7}$	0	0.00028855	0.6803895	1	0
vertical low	$2.741619 \cdot 10^{-7}$	0	0.00028855	0.6803895	1	0

Table 1: PSD coefficients for the track-irregularities [19]

According to the EN 13848-1 [20] the wavelength in the public transportation for typical track irregularities is chosen from $\lambda_1 = 3$ m to $\lambda_2 = 25$ m. The PSD function is given at 300 equidistant spaced discrete frequencies between λ_1 and λ_2.

3.3 Modelling of Failure Scenarios

To simulate a suspension failure different failure scenarios are investigated. The failure can occur as a single failure, that means only one component fails, or as a multiple failure, that means two or more components fail at the same time. Six of them (only single failures) are investigated in this publication. The vertical spring stiffness is reduced by 30% and 70%, each in the secondary suspension on the left and in the primary suspension on the left and right side of the front wheelset respectively.

3.4 Analysis of Eigenfrequencies

To get a first impression of the dynamics of the damaged vehicle an eigenfrequency analysis is performed with the aim to find detectable changes in the eigenfrequencies, corresponding to a failure scenario. Tab. 2 shows the typical eigenfrequencies of the healthy state. In Fig. 6 the changes of the eigenfrequencies with respect to the specified failure case are depicted. It can be clearly seen that a failure in the secondary spring has a big influence on the car body.

Bogie		
lateral	out of phase	2.46 Hz
	in phase	2.31 Hz
yaw	out of phase	2.60 Hz
	in phase	2.60 Hz
longitudinal	out of	4.60 Hz
	in phase	4.63 Hz
bounce	out of phase	7.42 Hz
	in phase	7.44 Hz
roll	out of phase	9.69 Hz
	in phase	9.69 Hz
pitch	out of phase	11.80 Hz
	in phase	11.80 Hz

Body	
lower sway	0.53 Hz
yaw	0.80 Hz
bounce	1.07 Hz
upper sway	1.34 Hz
pitch	1.29 Hz

Wheelset		
lateral	out of phase	16.92 Hz
	in phase	16.89 Hz
bounce	out of phase	33.04 Hz
	in phase	33.04 Hz

Table 2: Modeshapes and eigenfrequencies of the train model in the healthy state

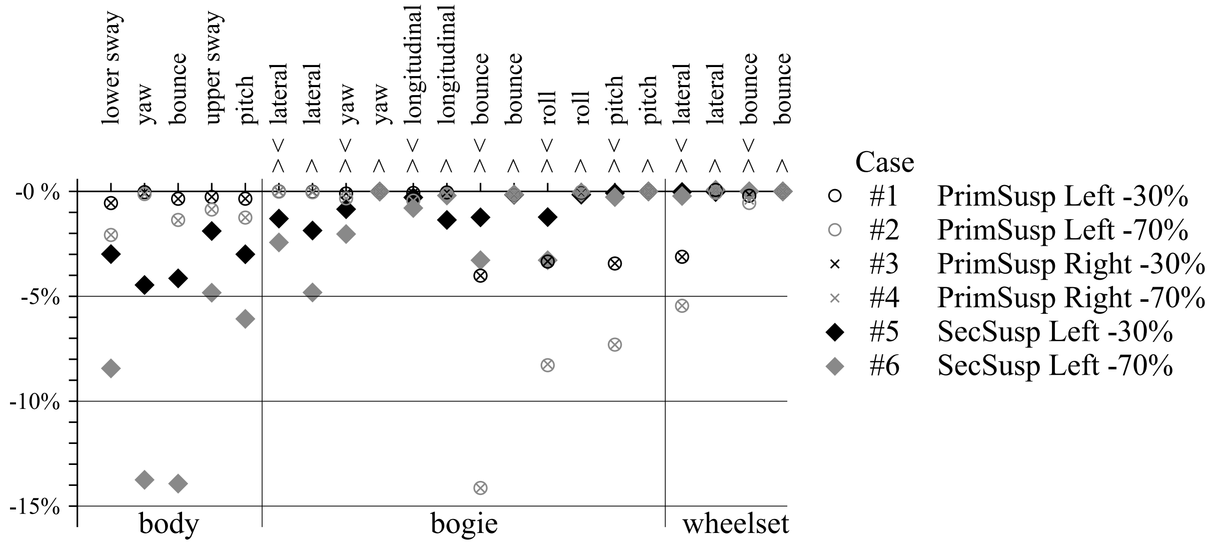


Figure 6: Changes of eigenfrequencies in different failure scenarios (>< out of phase; > in phase)

Changes at around -3 to -5% in the low damage level case and up to -14% in the high damage level case can be observed. A stiffness reduction in the primary spring can be observed in the bogie eigenfrequencies. Especially in the bounce, roll and pitch mode they are in a range between -3 to -4 % in the low damage level case and -6 to -14% in the high damage level case. This information can be used to design an SHM monitoring system. Special attention should be paid to frequency changes at around 7 to 12 Hz to detect a primary suspension failure and from 0.5 to 1.5 Hz to detect failures in the secondary spring.

4. CONCEPT FOR MONITORING OF MECHANICAL BOGIE PARAMETERS

In this section the subspace identification based structural health monitoring concept is reviewed. To prove the applicability of the proposed monitoring concept only one failure case is considered, i.e. the front left spring failure. In the first paragraph the feature generation method based on a subspace method is introduced, while in the second paragraph the results for the obtained failure cases are presented.

4.1 Subspace Method for Systems without Inputs

The application of subspace identification techniques for the identification of linear time invariant systems is a mature technique, see e.g. [9, 10, 11, 21]. The application of these identification schemes for input-free vibration monitoring has been derived in [8], while the notation introduced in this paragraph is taken from [21]. The following autonomous system is considered

$$\begin{aligned} \mathbf{x}(k+1) &= \mathbf{A}\mathbf{x}(k) + \mathbf{w}(k) \\ \mathbf{y}(k) &= \mathbf{C}\mathbf{x}(k) + \mathbf{v}(k). \end{aligned} \tag{2}$$

Here $\mathbf{y}(k)$ is the measured acceleration signal, k is the discrete time, $\mathbf{A}$ and $\mathbf{C}$ denote the system matrices and $\mathbf{w}(k)$ and $\mathbf{v}(k)$ are independent and identically distributed. Gaussian noise with the covariance matrix $\mathbf{R}$ and $\mathbf{Q}$ respectively. If the state $\mathbf{x}(k)$ of the system is known the ahead predictors of $\mathbf{y}(k)$ can be written as

$$\begin{pmatrix} \mathbf{y}(k|k-1) \\ \mathbf{y}(k+1|k-1) \\ \mathbf{y}(k+2|k-1) \\ \vdots \\ \mathbf{y}(k+r-1|k-1) \end{pmatrix} = \underbrace{\begin{pmatrix} \mathbf{C} \\ \mathbf{CA} \\ \mathbf{CA}^2 \\ \vdots \\ \mathbf{CA}^{r-1} \end{pmatrix}}_{\mathcal{O}_r} \mathbf{x}(k) \tag{3}$$

with the observability matrix $\mathcal{O}_r$ where r is the prediction horizon. Under the assumption that the system is completely observable the rank of the observability matrix is equal to the number of states n. The measured values of $\mathbf{y}(k)$ can be seen as a noise disturbed measurements of the optimal ahead predictors. To deal with the noise in the measurements of each ahead predictor a model based on the previous outputs is identified and at timestep k can be written as

$$\begin{pmatrix} \mathbf{y}(k|k-1) \\ \mathbf{y}(k+1|k-1) \\ \vdots \\ \mathbf{y}(k+r-1|k-1) \end{pmatrix} = \underbrace{\begin{pmatrix} \theta_{11} & \theta_{12} & \dots & \theta_{1s} \\ \theta_{21} & \theta_{22} & \dots & \theta_{2s} \\ \vdots & \vdots & \ddots & \vdots \\ \theta_{r1} & \theta_{r2} & \dots & \theta_{rs} \end{pmatrix}}_{\Theta} \begin{pmatrix} \mathbf{y}(k-1) \\ \mathbf{y}(k-2) \\ \vdots \\ \mathbf{y}(k-s) \end{pmatrix}. \tag{4}$$

It is important to notice that the first row of the parameter matrix is equal to the parameters of an AR-model of order s. Introducing the block Hankel matrices

$$\mathbf{Y}_r = \begin{pmatrix} \mathbf{y}(s+1) & \mathbf{y}(s+2) & \dots & \mathbf{y}(N-r+1) \\ \mathbf{y}(s+2) & \mathbf{y}(s+3) & \dots & \mathbf{y}(N-r+2) \\ \vdots & \vdots & \ddots & \vdots \\ \mathbf{y}(s+r) & \mathbf{y}(s+r+1) & \dots & \mathbf{y}(N) \end{pmatrix} \tag{5}$$

and

$$\Phi = \begin{pmatrix} \mathbf{y}(s) & \mathbf{y}(s+1) & \dots & \mathbf{y}(N-r-1) \\ \mathbf{y}(s-1) & \mathbf{y}(s) & \dots & \mathbf{y}(N-r-2) \\ \vdots & \vdots & \ddots & \vdots \\ \mathbf{y}(1) & \mathbf{y}(2) & \dots & \mathbf{y}(N-s-r) \end{pmatrix} \tag{6}$$

the problem of estimating Θ can be written as a Least-Squares problem

$$\mathbf{Y}_r = \Theta\Phi, \tag{7}$$

with the solution

$$\hat{\Theta} = \mathbf{Y}_r\Phi\left(\Phi^T\Phi\right)^{-1}. \tag{8}$$

It is now possible to estimate the r-step ahead predicted values of y by right side multiplying (8) with Φ and obtaining

$$\hat{\mathbf{Y}}_r = \mathbf{Y}_r\Phi\left(\Phi^T\Phi\right)^{-1}\Phi \tag{9}$$

The following relationship between the r-step ahead predictors, the augmented state matrix and the observability matrix holds

$$\hat{\mathbf{Y}}_r = \hat{\mathcal{O}}_r\hat{\mathbf{X}} = \mathbf{USV}^T \tag{10}$$

whereby the last step is the singular value decomposition of the estimated ahead step predictors. In the ideal noise free case the singular values of $\hat{\mathbf{Y}}_r$ are different from zero up to the true order of the system. In the noisy case for the estimation of the observability matrix the n highest singular values are used and the estimate is formed by truncating the $\mathbf{U}$ matrix. The estimated $\hat{\mathcal{O}}_r$ matrix can now be used to obtain estimates for $\hat{\mathbf{A}}$ by applying the relation

$$\hat{\mathcal{O}}_r(n_y : n, :) = \hat{\mathcal{O}}_r(1 : n - n_y, :)\hat{\mathbf{A}} \tag{11}$$

using a Matlab like indices notation with n_y denoting the number of outputs measured. The solution of (11) can be obtained by applying a Least-Squares like approach. Afterwards the equivalent undamped eigenfrequencies and the modal damping coefficient can be obtained. Usually the estimated observability matrix is calculated for different orders and for each of them the eigenfrequencies are estimated based on the system matrices. The results are then presented for different orders in so-called stabilization diagrams.

4.2 Results for the Subspace Method

The method described in the previous paragraph is now used to obtain estimates of the system eigenfrequencies. For the damage detection only the vertical acceleration signal of the front left bogie sensor is used. The derived results are shown in Fig. 7, where the stabilization diagram of the considered signal is depicted. It can be observed that in the frequency range between 10 Hz and 12 Hz, where the pitch mode of the bogie is present, the subspace identification algorithm identifies the eigenfrequency of the bogie pitch mode accordingly. There are two observations that can be made, when applying the subspace identification method. One is that the results clearly depend on the irregularities of the track, the other is that the identified stability diagram is sensitive with respect to the prediction horizon r selected. Based on these observations a roller rig test stand as described in the second paragraph of this contribution is necessary to develop realistic choices even for the tuning parameters of the algorithms and to validate the damage detection approach.

5. CONCLUSION

The concept of a novel roller rig test bench has been presented and the SAMT concept has been introduced. A MBS model for the simulation of SHM relevant scenarios of passenger trains has been created and different failure scenarios have been simulated. The application of a subspace SHM method has been exemplified for a faulty front left spring of the bogie. It has been shown that identified deviations in the corresponding pitch eigenfrequency of the bogie can be utilized to detect the fault. In the future the erection of a test stand for the evaluation of the corresponding and other SHM methods is an important step to increase the reliance in the methods for modern train maintenance systems.

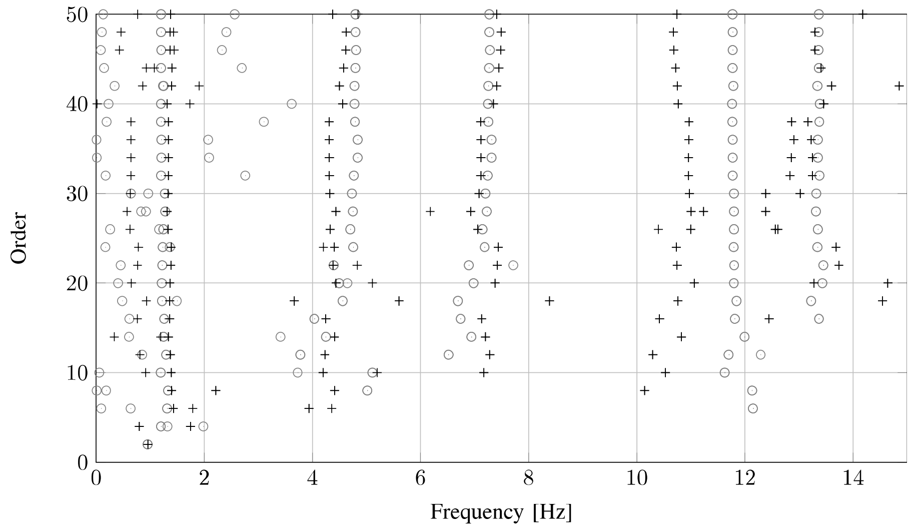


Figure 7: Results of the subspace algorithm applied to the test case (+ indicates the result for the faulty system, while o corresponds to the healthy state)

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