CONDITION MONITORING AND FAULT DETECTION OF SUSPENSION COMPONENTS IN FREIGHT WAGONS USING ACCELERATION MEASUREMENTS

Stefano ALFI, Bin FU, Stefano BRUNI

Department of Mechanical Engineering, Politecnico di Milano
Via La Masa 1, 20156 Milano, Italy
e-mail stefano.alfi@.polimi.it
ph. +39 02 2399 8495

ABSTRACT

Condition-based maintenance is seen as a major means to improve the business case for railway freight operators, and to improve the reliability and availability of freight trains. This paper reports on investigations performed aimed at developing methods for the condition monitoring of suspension components of Y25 bogies for freight wagons, enabling the implementation of CBM strategies for the wagon.

Keywords: Rail vehicle dynamics, monitoring of suspension components, freight wagons.

1. INTRODUCTION

Implementing condition-based maintenance (CBM) strategies for the railway running gear is a key priority for railway operators as it offers the opportunity of significantly reducing the total life cycle costs for railway vehicles, at the same time increasing safety, reliability and availability of operation. Particularly for railway freight vehicles, CBM can be a key factor to increase their attractivity compared to competing transportation means, thereby contributing to a shift of freight traffic from roads and air to the rail. This paper reports on investigations performed within project INNOWAG, funded by Shift2Rail, aimed at developing methods for the condition monitoring of suspension components of Y25 bogies for freight wagons.

In the work, the dynamic behaviour of a freight vehicle equipped with two Y25 bogies with suspensions in healthy and faulty conditions is simulated by means of a fully non-linear multi-body model and the results obtained are used as "virtual" measurements to be processed by the diagnostic algorithms. In this way, it is possible to assess the effectiveness of the algorithms in detecting different types and levels of faults in the suspensions. Given the highly non-linear behaviour of the suspension in the Y25 bogie, the use of data-driven fault detection methods is preferred compared to model-based ones and the work focusses on the analysis of the cross-correlation of bounce, pitch and roll accelerations of the bogie frame, derived from vertical acceleration signals measured at three different locations in the bogie frame.

2. SIMULATION OF DIFFERENT SUSPENSION FAULT MODES USING MULTI-BODY MODELS

A multi-body model of a wagon with faulty primary suspension was developed using software ADTreS, which is POLIMI's in-house software for the simulation of railway vehicle dynamics [1]. ADTreS is the Italian acronym for "dynamic analysis of train-track interaction".

The model of the wagon is based on the use of rigid bodies to represent the wagon body, two bogie frames and four wheelsets. The primary suspensions and the connection of the bogies to the wagon body are represented by means of non-linear springs and dry friction dampers. Wheel/rail contact forces are considered using a fully non-linear wheel/rail contact module taking into account the actual shape of the wheel and rail profiles and the non-linear creepage-creep force relationship. More details on the modelling of railway vehicles in this software are provided in reference [1].

Two main types of fault were modelled in this work, see Figure 1:

- Type A: failure of one coil spring in one primary suspension;
- Type B: failure of the dry friction damper in one primary suspension.

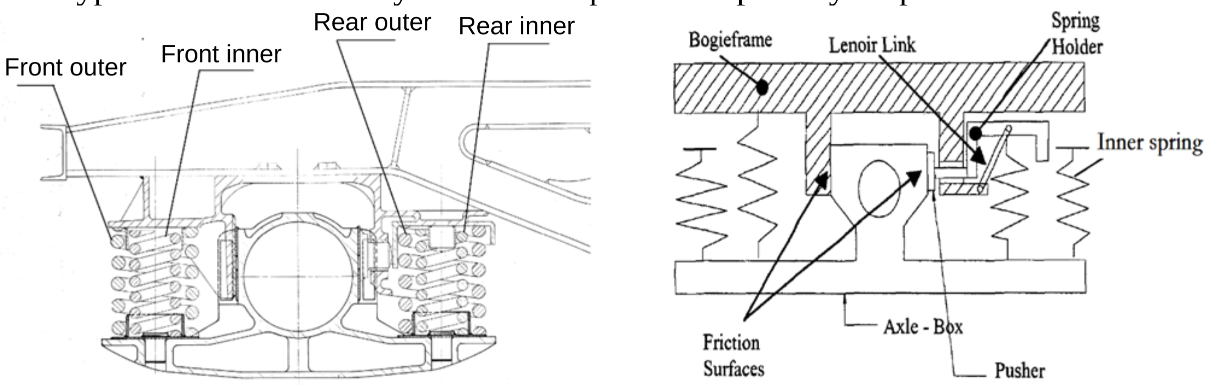

Figure 1 - The primary suspension of the Y25 bogie (left) and a functional scheme of the suspension (right)

2.1 Modelling of fault type A

The failure of one coil spring in the primary suspension is modelled as a change in the vertical stiffness of the spring element connecting the axle box to the bogie frame in one corner of the bogie. Given that the change in stiffness implies a re-distribution of the static forces transmitted by the coil springs, the threshold force at which sliding takes place in the friction element is also affected.

The changes in the parameters of the suspension are different depending on whether the failure takes place in the “front outer”, “front inner”, “rear outer” or “rear inner” coil spring, see Figure 1 left. The parameters of the suspension also depend on the axle load. In this work, two extreme conditions are considered: the vehicle in full-load condition (carbody mass 80.60t) and in tare condition (carbody mass 6.86t).

Table 1 shows the values of the stiffness (K_z) and sliding force (F_s) parameters for the vehicle in full-load condition when the fault takes place at each one of the coil springs shown in Figure 1 and for different types of fault, i.e.:

- broken spring, the stiffness of the single spring is set to 0;
- spring stiffness reduced to 50%;
- spring stiffness increased to 150%;
- spring stiffness increased to 200%;

Fault cases consisting of an increased value of spring stiffness are intended to represent failure modes leading to an improper increase of the suspension stiffness, e.g. the spring or slider getting stuck to some extent.

	0% (broken)	50%	150%	200%
Front outer	K_z=2.06MN/m F_s = 22.33kN	K_z=2.31 MN/m F_s = 19.10kN	K_z=2.81 MN/m F_s = 14.38kN	K_z=3.06 MN/m F_s = 12.60kN
Front inner	K_z=1.78 MN/m F_s =21.65kN	K_z=2.17 MN/m F_s = 18.62kN	K_z=2.95 MN/m F_s =14.96kN	K_z= 3.34 MN/m F_s =13.77kN
Rear outer	K_z=2.06 MN/m F_s = 10.69kN	K_z=2.31 MN/m F_s = 13.92kN	K_z=2.81MN/m F_s = 18.64kN	K_z=3.06 MN/m F_s = 20.42kN
Rear inner	K_z=1.78MN/m F_s = 11.37kN	K_z=2.17 MN/m F_s = 14.40kN	K_z=2.95 MN/m F_s = 18.06kN	K_z=3.34 MN/m F_s = 19.25kN

Table 1 - Parameters of the primary suspension for different Type A faults for the full-load condition (values in healthy condition are K_z = 2.56 MN/m and F_s = 16.51 kN).

Table 2 shows the values of the stiffness (K_z) and sliding force (F_s) parameters for the vehicle in tare condition for the same types of fault considered in Table 1. As shown in Figure **1**, the inner springs are mounted with a gap between the top of the spring and the bogie frame and enter in contact with the bogie frame only when the vertical load on the suspension exceeds a threshold and compresses the outer spring sufficiently. In tare condition, the load on the suspension is too low and the two inner springs do not make contact with the bogie frame and therefore do not contribute to the total stiffness of the suspension. In this condition, the presence of a fault in the inner springs does not affect the behaviour of the vehicle and therefore cannot be detected.

	0% (broken)	50%	150%	200%
Front outer	$\boldsymbol{K_z = 0.50\ MN/m}$ $\boldsymbol{F_s = 4.23\ kN}$	$K_z = 0.75\ MN/m$ $F_s = 2.82\ kN$	$K_z = 1.25\ MN/m$ $F_s = 1.69\ kN$	$K_z = 1.50\ MN/m$ $F_s = 1.41\ kN$
Rear outer	$\boldsymbol{K_z = 0.50 MN/m}$ $\boldsymbol{F_s = 0.\ kN}$	$K_z = 0.75\ MN/m$ $F_s = 1.41 kN$	$K_z = 1.25 MN/m$ $F_s = 2.54\ kN$	$K_z = 1.50\ MN/m$ $F_s = 2.82\ kN$

Table 2 - Parameters of the primary suspension for different Type A faults for the tare condition (values in the healthy condition are K_z = 1.0 MN/m and F_s = 2.12 kN).

2.2 Modelling of fault type B

The failure of the dry friction damper in one corner of the bogie is modelled as a change in the threshold force F_s at which sliding takes place in the friction element, with no effect on the stiffness of the suspension. Table 3 shows the values of the F_s parameter for different types of fault, with friction ranging from 50% to 150% of the nominal value.

-	Full load (Car-body mass: 80.60t)			
	50%	75%	125%	150%
Threshold force F_s	$F_s = 11.6\ kN$	$F_s = 16.74kN$	$F_s = 27.91kN$	$F_s = 33.49kN$
	Tare load (Car-body mass: 6.86t)			
Threshold force F_s	$F_s = 1.06\ kN$	$F_s = 1.59kN$	$F_s = 2.65kN$	$F_s = 3.18kN$

Table 3 – Values of the maximum sliding force for different Type B faults and for the wagon in full-load and tare condition (values in the healthy condition are 22.33 kN in full-load condition and 2.12 kN in tare load condition).

3. FAULT DETECTION ALGORITHM

Considering the relatively low economic value of one freight wagon, fault detection has to be based on a simple algorithm, requiring a simple measuring and data processing system. Considering different methods proposed in the scientific literature (see [2] for a recent review of the State-of-Art), the use of cross-correlation between acceleration signals measured at different locations in the bogie frame [3] appears as the best suited fault detection method in terms of achieving a good compromise between the measuring / processing effort required and the performance in terms of accuracy and reliability of fault detection.

The principle of the method is that the modes of vibration of a bogie with healthy suspensions are symmetric, due to uniform behaviour of the primary suspensions at the four corners of the bogie. Therefore, if the vibration of the body is excited by uncorrelated random irregularity from the left and right rails, the correlation between different components of bogie frame acceleration, e.g. bounce vs. roll or pitch vs. roll will be low. When a failure occurs at one corner, the symmetry of the modes of vibration is perturbed, resulting in a coupling of bounce, roll and pitch components of motion which can be detected from an increase of the cross-correlation between the acceleration signals for these components of bogie vibration.

A schematic representation of the method is shown in Figure 2. Three mono-axial accelerometers (numbered from 1 to 3 in the figure) are mounted at three corners of the bogie frame, to measure the vertical acceleration of the bogie frame over the axle-boxes.

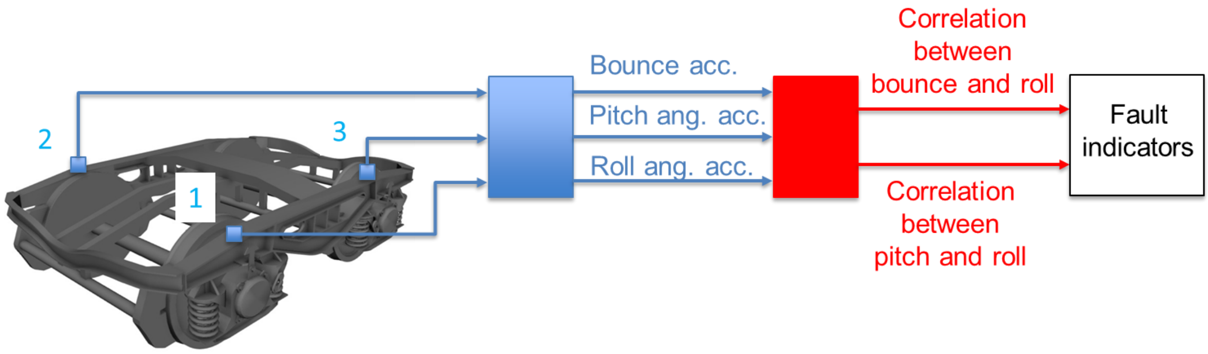

Figure 2– Schematic representation of the signal processing technique for the detection of faults in the primary suspensions

The vertical acceleration signals are converted into the bounce, pitch and roll acceleration components for the bogie frame (a_B, a_P and a_R respectively) according to the kinematics of a rigid body:

$$a_B = \frac{a_2 + a_3}{2}$$

$$a_P = \frac{a_1 - a_3}{2l}$$

$$a_R = \frac{a_1 - a_2}{2b}$$

where $2l$ is the bogie wheelbase and $2b$ is the axle box gauge. Finally, the cross-correlation between bounce and roll acceleration C_{BR} and between pitch and roll acceleration C_{PR} are evaluated:

$$C_{BR}(\tau) = \frac{1}{T}\int_{-\frac{T}{2}}^{\frac{T}{2}} a_B(t)a_R(t-\tau)dt$$

$$C_{PR}(\tau) = \frac{1}{T}\int_{-\frac{T}{2}}^{\frac{T}{2}} a_P(t)a_R(t-\tau)dt$$

To diagnose faults in the suspensions, four fault indicators are computed.

The first two indicators are based on the standard deviation of the cross-correlation between the acceleration signals while the other two indicators are based on the instantaneous value of the cross-correlation between pitch and roll at zero-time delay.

- Indicator 1: Standard deviation of the cross-correlation between bounce and roll accelerations for time delay -10 s $\leq \tau \leq$ +10 s;
- Indicator 2: Standard deviation of the cross-correlation between pitch and roll accelerations for time delay -10 s $\leq \tau \leq$ +10 s.
- Indicator 3: Value at time delay τ=0 of the cross-correlation between pitch and roll accelerations low pass filtered at 10 Hz;
- Indicator 4: Value at time delay τ=0 of the cross-correlation between pitch and roll accelerations without low pass filter.

4. RESULTS OF NUMERICAL EXPERIMENTS

Numerical experiments were run to consider fault types A and B for full load and tare load condition of the wagon. The results are described below for the different cases considered.

Figure 3 shows the value of the four fault indicators for different degrees of severity of a fault of type A occurring in the rear outer spring of the primary suspension, when the vehicle is in the full-load condition. It is confirmed that all four indicators have a minimum absolute value when the suspension is in the nominal condition (stiffness is 100% of the nominal value) whilst the absolute values of all indicators increase when a deviation of the spring stiffness is applied in the numerical experiment, either decreasing or increasing the stiffness parameter. In principle, all 4 indicators appear suitable for the detection of the fault considered, but indicators 2 and 4 show more clearly the presence of a fault.

Similar numerical experiments were performed considering a fault happening in the rear inner, front outer and front inner springs of the primary suspensions, leading to the same conclusions as in the case presented below. The results of these additional cases are not shown for the sake of brevity but are considered in a summary table (Table 4) reported at the end of this section.

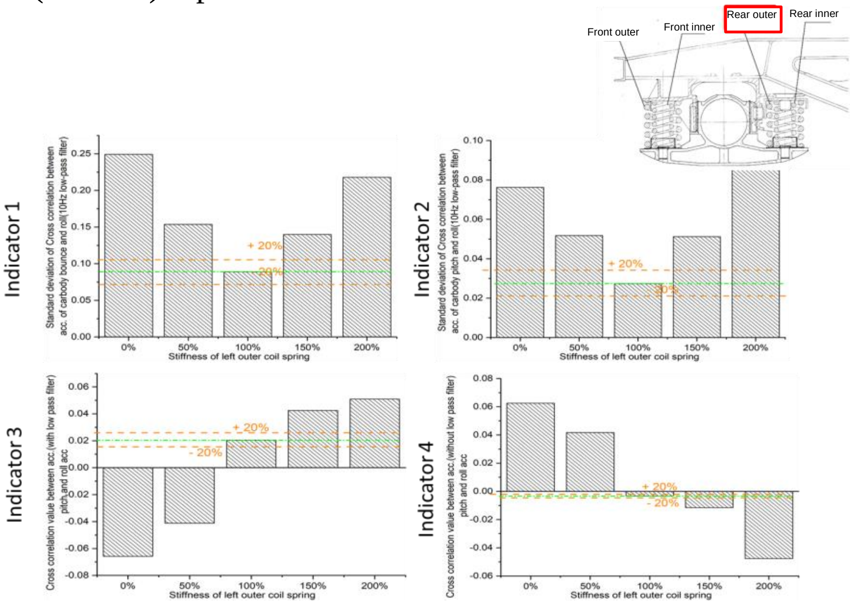

Figure 3 - Fault indicators 1 to 4 for a fault in the rear outer coil spring (type A) and vehicle in full load condition

Figure 4 shows the value of the four fault indicators for different degrees of severity of a type A fault occurring in the rear outer spring of the primary suspension, when the vehicle is in the tare load condition. In this case, the indication is less clear than in the case of the full load condition, probably because the effect of spring fault on the dry friction damper is lower. The only indicator that shows a clear trend with the change of stiffness in the faulty spring is Indicator 3. Similar results are obtained when the fault is simulated in the front outer spring, again with the wagon in the tare load condition.

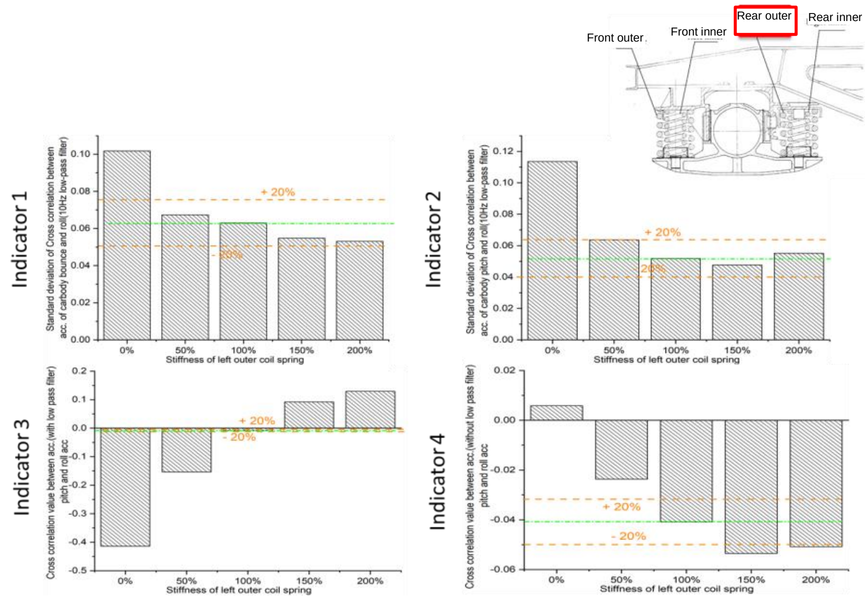

Figure 4 - Fault indicators 1 to 4 for a fault in the rear outer coil spring (type A) and vehicle in tare condition

Figure 5 shows the value of the four fault indicators for a fault occurring in the friction element of the primary suspension (fault type B), with the vehicle in the full load condition. In this case, a clear indication of the fault is obtained from all indicators except indicator 3. When the same fault type is considered for the vehicle in tare condition (not shown due to space restrictions) the only indicator that provides a clear trend with the occurrence of the fault is Indicator 3.

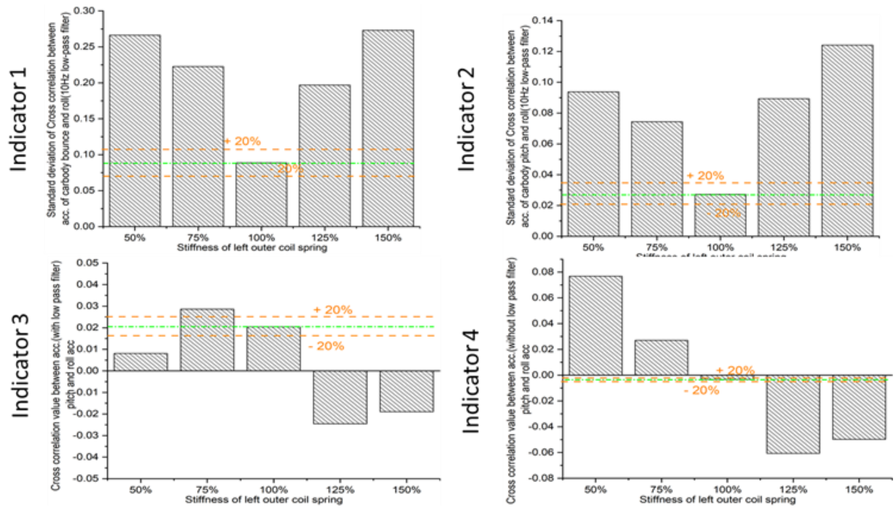

Figure 5 - Fault indicators 1 to 4 for a fault in the dry friction element (type B) and vehicle in full load condition

A summary of results from numerical experiments performed to assess the proposed method is provided in Table 4, with symbols "✓" and "✗"denoting respectively successful and non-successful identification of the fault. It is confirmed by the results shown in the table that faults in suspensions components are more easily detectable when the vehicle is in full-load condition.

			Indicator1	Indicator2	Indicator3	Indicator4
Fault Case A: coil springs	Full load	Front outer	✓	✓	✓	✓
		Front inner	X	✓	✓	✓
		Rear outer	✓	✓	✓	✓
		Rear inner	✓	✓	✓	✓
	Tare load	Front outer	X	X	✓	X
		Rear inner	X	X	✓	X
Fault Case B: Friction components	Full load	\	✓	✓	X	✓
	Tare load	\	X	X	✓	X

Table 4– Summary of the results of numerical experiments for identification of faults in primary suspensions.

5. CONCLUSIONS

In this paper, a method to detect faults in the suspension of a freight wagon with Y25 bogie is proposed, based on the examination of the cross-correlation of bounce, pitch and roll vibration. The method is assessed based on numerical experiments performed using a multi-body model of the wagon. It is shown that different fault indicators can be defined that could prospectively allow the detection of faults in coil springs and friction dampers.

Future extension of this work is foreseen, particularly to identify and classify faults based on the use of suitable outlier detection / artificial intelligence methods. Furthermore, the application of the method to measurements from on-track tests is envisaged.

ACKNOWLEDGEMENTS

The work presented in this paper has received funding from the Shift2Rail Joint Undertaking under the European Union's Horizon 2020 research and innovation programme under grant agreement no. 730863 – INNOWAG. The contents of this publication only reflect the authors' view and the Joint Undertaking is not responsible for any use that may be made of the information contained in this paper.

6. REFERENCES

1. **Di Gialleonardo E., Braghin F. and Bruni S.:** The influence of track modelling options on the simulation of rail vehicle dynamics. Journal of Sound and Vibration, vol. 331, p. 4246-4258, 2012.
2. **C. Li, S. Luo, C. Cole and M. Spiryagin:** An overview: modern techniques for railway vehicle on-board health monitoring systems. Vehicle System Dynamics, vol. 55(7), pp. 1045-1070, 2017.
3. **Mei T. X. and Ding X. J.**: Condition monitoring of rail vehicle suspensions based on changes in system dynamic interactions. Vehicle System Dynamics, vol. 47(9), pp. 1167-1181, 2009.