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Fault detection of railway vehicle suspension through on-board condition monitoring

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Abstract

Laboratory tests in order to acquire data of the selected wagon components failure (i.e. suspension helical springs) to be monitored have been carried out. In particular, the present work considers possibilities for the condition monitoring of helical springs based on measuring axle box accelerations.
Undamaged and damaged springs are studied of a Y25 bogie, which have been exposed to a theoretical and experimental analysis. For analytical study, the dynamic stiffness method is going to be applied, which through its derivation, the dynamic behaviour of the helical spring is described. The experimental tests are conducted on an INSTRON MHF25 testing machine. Two types of dynamic tests have been performed: force-displacement relationship and FRF of the accelerometers. A good agreement between experimental and theoretical results has been observed. A positive aspect is that the results obtained with accelerometers are more satisfactory than those obtained through the representation of the evolution of dynamic stiffness. However, after analysing the FRF distribution of the damaged springs, a great dispersion has been observed between them. There is no clear trend in frequency response depending on the damage type of the spring. Despite this, satisfactory results have been obtained in order to identify damaged springs in service based on condition monitoring data defective. Through this analysis, significant correlation between helical spring indicator and defect type has been found, indicating promising chances of detecting a defect before there is a risk of high failure high, therefore ensuring the safe running of the vehicle.

Keywords: Helical spring, Rail vehicle suspension, Fault detection, Condition monitoring, Freight locomotive, Dynamic stiffness method.

1 Introduction

A good bogie design of freight locomotive ensures ride comfort and lateral stability [1]. Moreover, the reliability of the railway suspension system is of critical importance to the safety of the vehicle. A freight rail vehicle has the frequent failure of primary suspension [2] and this is considered one of the most critical failures that trigger wagon maintenance. In order to guarantee safety, the inspections to be carried out are mainly visual. Machine test of the springs are carried out to ensure the compliant characteristics of maintained springs. This is time and resources consuming since all the components of the wheelset must be removed. Given this scenario, it is evident that a real-time monitoring for the suspension system can be highly beneficial for the railway maintenance service in terms of savings (time and money) and reliability, from reducing inspections, and at the same time enhancing the safety of the component during its life.

Several works aimed at detecting suspension failures of railway vehicles using signal analysis monitoring have been conducted during the last years [3–5]. Specifically, in these papers [6,7] it is shown that on-board condition monitoring system can efficiently detect suspension failures in order to improve failure mode detection in freight rail operations. In short, state of the art technology can capture physical variables to monitor their performance; however, there are not cost-effective solutions connected with reference values capable of providing recommendation and alarms to trigger maintenance action. One of the bottlenecks is the development of algorithms adapted to specific sensors and the integration of the data within a maintenance information system.

In this work, research in this area is proposed. Laboratory tests in order to acquire data of the selected wagon components failure (i.e. suspension helical springs) to be monitored have been carried out. This will indicate which output can be expected from testing in the field and will generate knowledge about the lifecycle of the components, type and amount on data needed. In particular, the present work considers possibilities for the condition monitoring of helical springs based on measuring axle box accelerations. In order to validate the model, the following strategy is applied: the assessment of the vertical stiffness against the experimental results and cross-validation of dynamic stiffness method [8,9].

2 Methods

The springs used in the bogie suspension of freight railway vehicles are compression springs [10]. Therefore, the test object is the internal helical spring of a primary suspension, which belong to Y25 bogie (Figure 1 above). There are two types of springs: eight inner and eight outer. Each wheel has two outer and two inner springs, as shown in the drawing of Figure 1 (bottom).

Undamaged and damaged springs are studied. The latter have been catalogued according to the damage type (Table 1 and Table 2), which have been provided by DB Cargo. Since no nominal spring of the Y25 bogie is available, another undamaged spring model has been used.

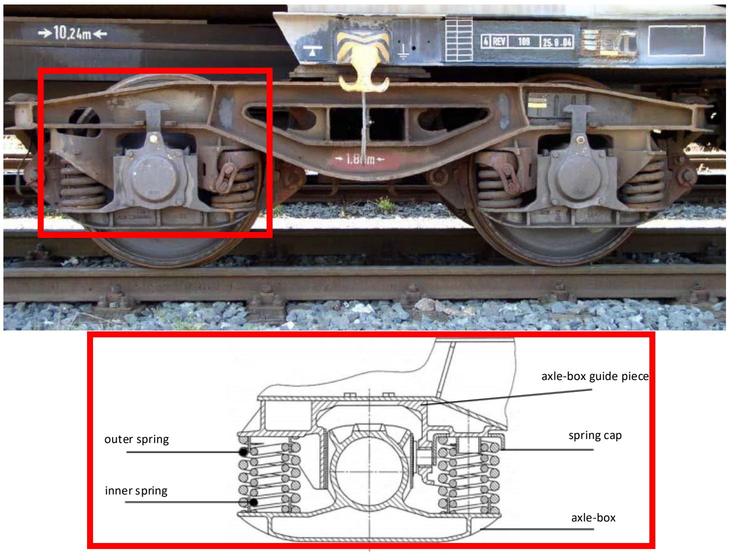
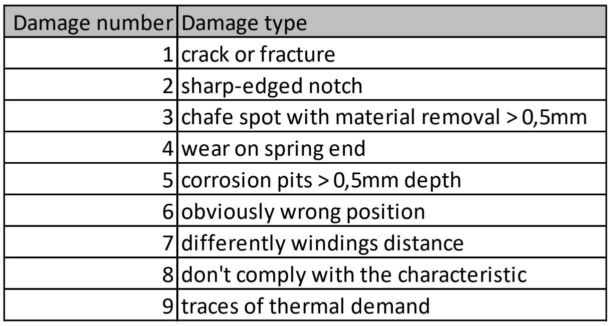


Figure 1. The Y25 bogie, photo (above) and scheme of the springs (bottom) [11]

	Free length	Outside diameter	Inside diameter	Diameter of the wire	Pitch	Total number of coils	Number of active coils	Damage type
	L0 (mm)	De (mm)	Di (mm)	d (mm)	P (mm)	Nt	Na	
inner theoretical	234	114,4	65,6	24,4	35,5	7,4	5,9	
inner nº1	234	115	66,5	24,25	35,3	7	5,5	1 / 3
inner nº2	234	114,5	66,5	24	v	7	5,5	1 / 5
inner nº3	231	114,7	66	24,35	v	7,5	6	8
inner nº4	235	113	65	24	34,6	7,5	6	5
inner nº5	233	112,5	66	23,25	42	7	5,5	3 / 4 / 7
inner nº6	237	114,4	66	24,2	37	7,5	6	8
inner nº7	233	113	67	23	47	6,5	5	3 / 4 / 5 / 7
inner nº8	236	113	66	23,5	36,5	7,5	6	4

Table 1. Physical parameters and damage type of internal springs; v means variable pitch between the coils.



Damage number	Damage type
1	crack or fracture
2	sharp-edged notch
3	chafe spot with material removal > 0,5mm
4	wear on spring end
5	corrosion pits > 0,5mm depth
6	obviously wrong position
7	differently windings distance
8	don't comply with the characteristic
9	traces of thermal demand

Table 2. Damage type classification provided by DB Cargo.

For analytical study, the dynamic stiffness method is going to be applied, which through its derivation, the dynamic behaviour of the helical spring is described. The dynamic stiffness method based on Timoshenko beam theory was presented by Lee and Thompson [8], from which the dynamic stiffness matrix is obtained. It links the force/moments vector with the displacement/rotation vector at the six degrees of freedom of each end of the spring, providing an efficient tool for steady-state forced vibration calculations.

The experimental tests are conducted on an INSTRON MHF25 testing machine (Figure 2). In Table 3 the sensors used are described.

Figure 2. Spring and accelerometer assembly on Instron MHF25.

Channel 1	Model: PCB 333 B31 Serial: 18042 Sensitivity: 0,1V/g
Channel 2	Model: PCB 333 B51 Serial: 27288 Sensitivity: 1V/g

Table 3. Accelerometers model used in the experimental tests.

Two types of dynamic tests have been performed:

- The spring is excited with a frequency sweep through the Instron MHF25. Two types of signals are analysed: on the one hand, the Instron MHF25 has a sophisticated program that provides the value of dynamic stiffness and the loss angle from the resulting hysteresis cycle for each frequency and amplitude. On the other hand, the root mean square rms value of each signal is calculated and the evolution of the rms(C2)/rms(C1) ratio is plotted with the frequency.
- The spring is excited after the introduction of a noise through the LMS Test Lab commercial software. This software postprocesses the signals by providing the Frequency Response Function FRF (C1/C2).

3 Results

The undamaged spring has been exposed to a theoretical and experimental analysis. The theoretical results are found using the dynamic stiffness method [8]. Figure 3 shows a good agreement between both methods.

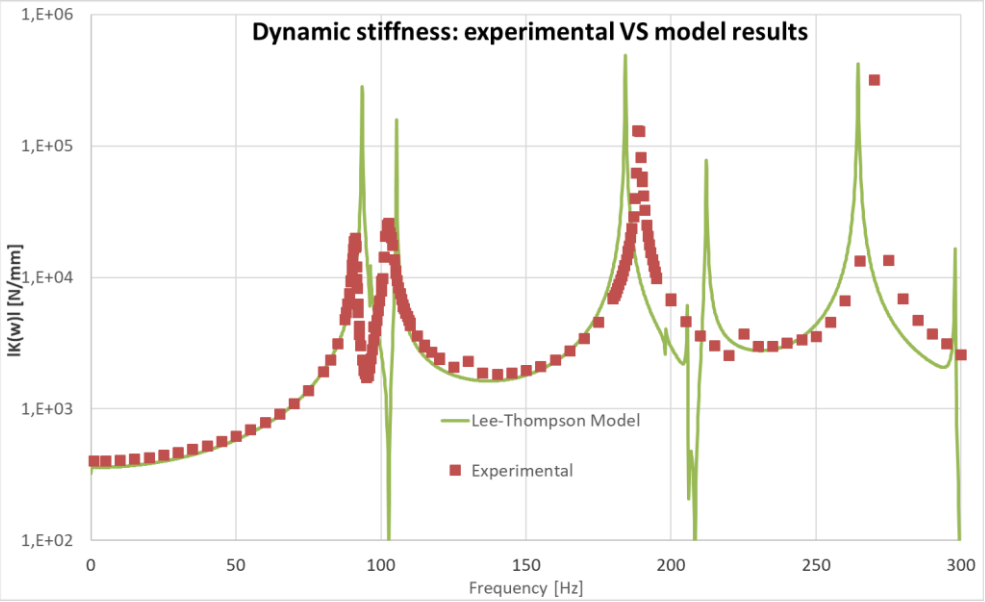


Figure 3. Theoretical (Lee-Thompson in green) vs experimental (red) dynamic rigidity module.

For experimental analysis, two types of tests have been performed (Figure 4). Obtained results through the force-displacement relationship and through the FRF of the accelerometers are similar.

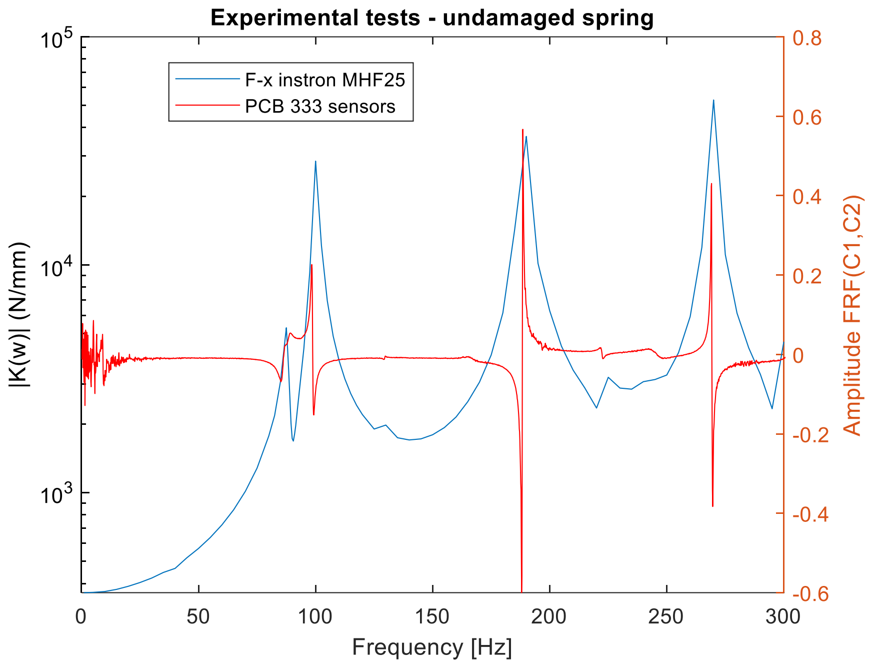


Figure 4. Experimental contrast of the dynamic stiffness module (blue) with the relationship between the accelerometer response (red).

Damaged inner spring num6 has been exposed to experimental analysis. Dynamic stiffness evolution provided by the test machine and evolution of the rms(C2)/rms(C1) ratio of each test are plotted as a function of frequency in Figure 5. From the postprocessing of the force/displacement values only a unique mode is obtained, which coincides with that calculated by the rms(C2)/rms(C1) ratio at 190Hz; the accelerometers indicate something more at 63Hz, 97Hz and 126Hz. They could be very attenuated modes.

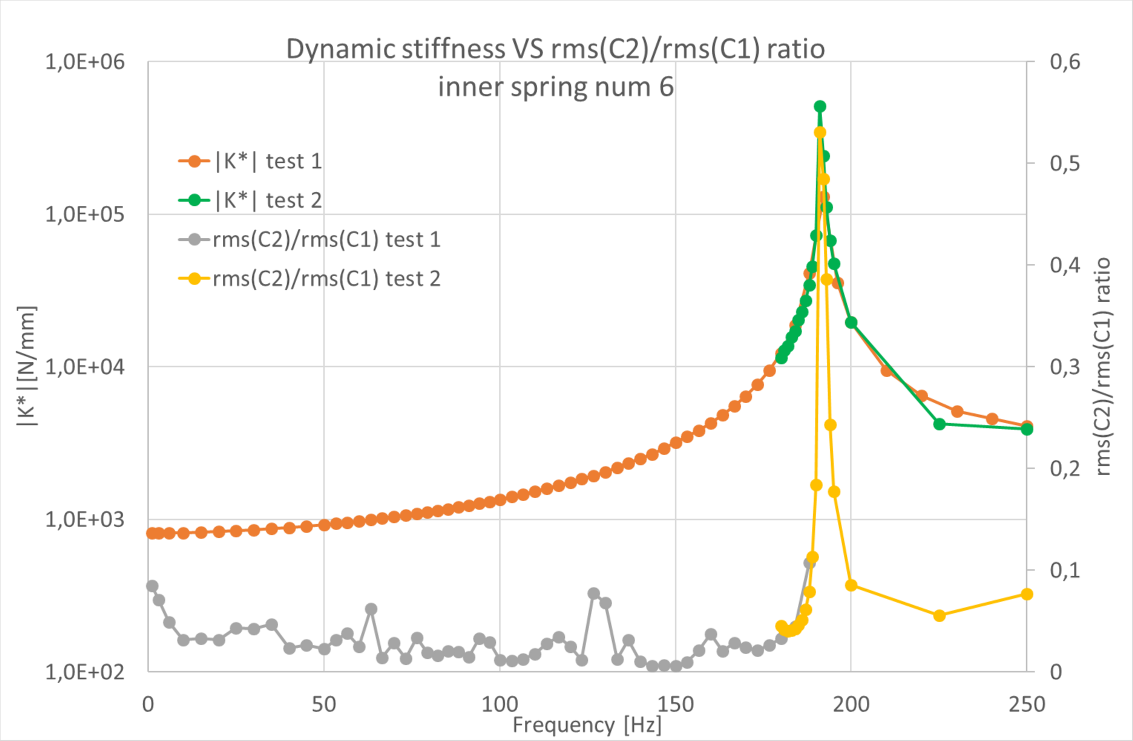


Figure 5. Dynamic stiffness evolution and the rms(C2)/rms(C1) ratio depending on the frequency.

An excitation has been introduced through external noise and 8 inner springs have been tested. Figure 6 shows FRF(C1/C2) of the 8 inner springs, individually potted for better visualization. It is appreciated that, in all cases, at low frequencies, there is a noise that is suspected may be due to the structure's own noise. Looking at the results at higher frequencies, there is a great dispersion between the springs. However, the following can be observed:

- Spring num3 and num6 have the same type of damage num8, that is, do not meet the characteristics, height out of tolerance; the num3 by default and the num6 by excess. The mode around 190Hz is similar in both springs, in addition to the amplitude (Figure 7). However, at lower frequencies it is observed that the resonance is opposite; at similar frequency values but opposite.
- Springs num5 and num7 have similar damages. The behaviour around 150Hz is similar; however, the mode from 200Hz is different (Figure 8). The difference in the amplitude value may be due to damage type 5, which has only the spring num7.

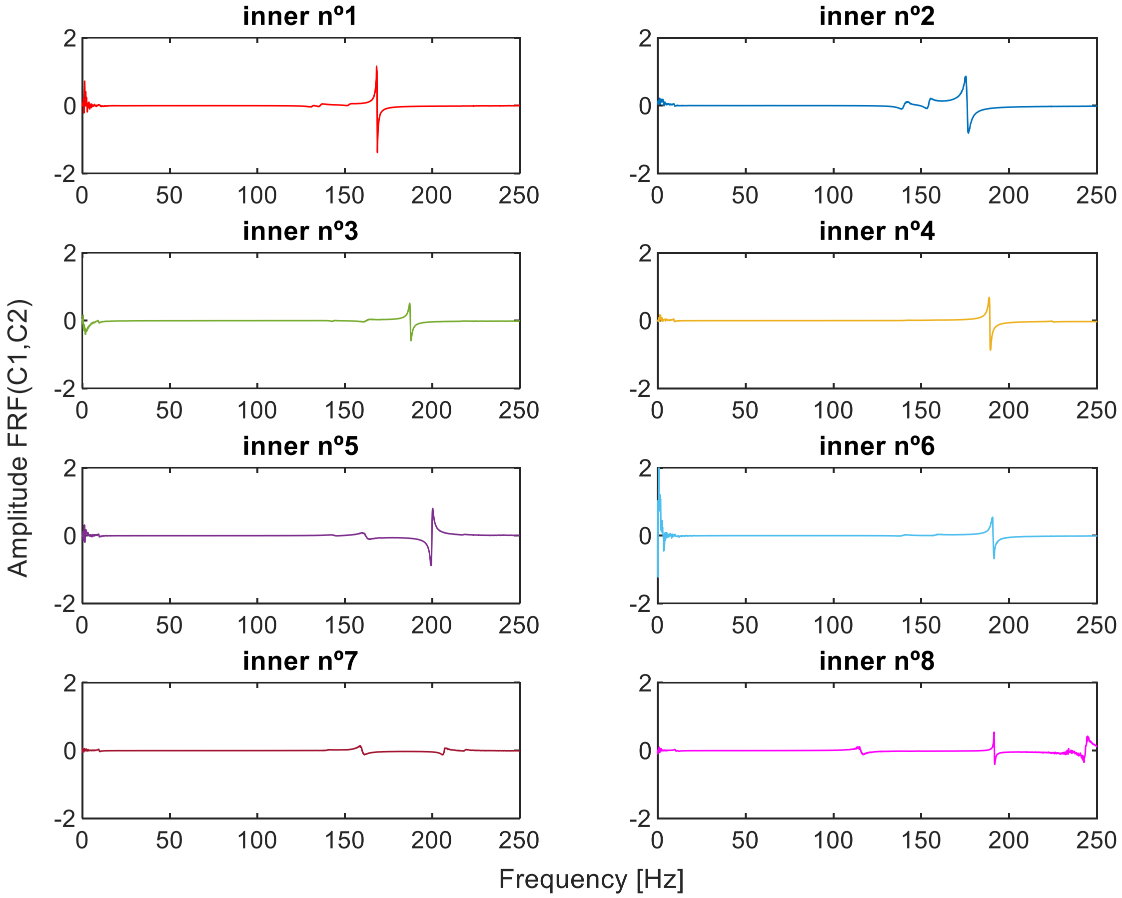


Figure 6. FRF(C1/C2) of the 8 inner springs.

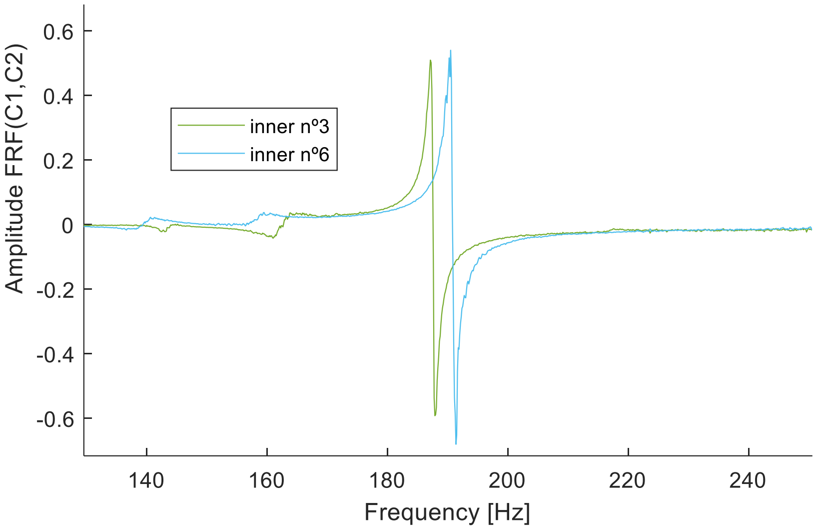


Figure 7. FRF(C1/C2) of the springs num3 and num6.

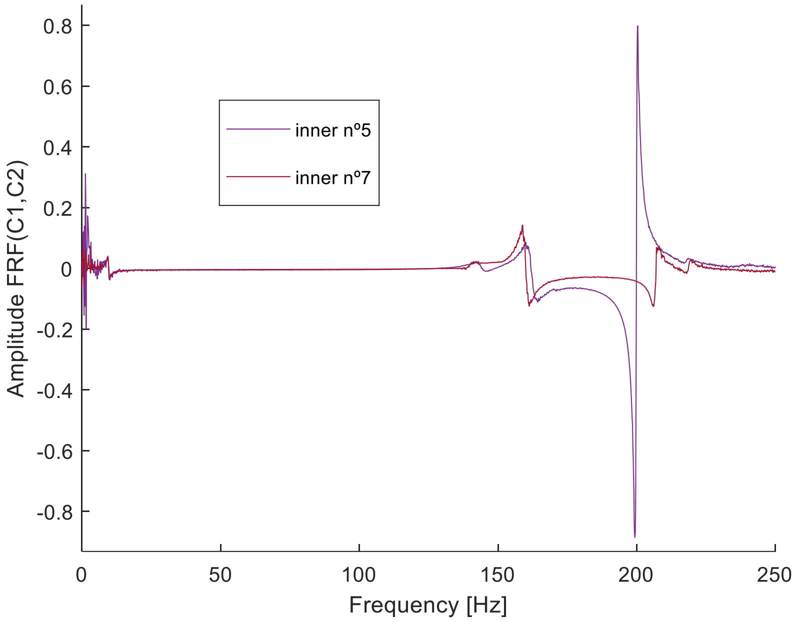


Figure 8. FRF(C1/C2) of the springs num5 and num7.

4 Conclusions and Contributions

Some internal undamaged and damaged helical springs of a primary suspension have been studied theoretically and experimentally to know their structural and dynamic response, and with the aim of identifying the degradation type. The following conclusions can be drawn:

For undamaged spring:

- This spring has been exposed to a theoretical and experimental analysis. A good agreement between experimental and theoretical results has been obtained. The minor discrepancies are due to uncertainties in the material properties and the absence of structural damping in the mathematical model. One of the most relevant results is that the higher frequency of the dynamic response is around 270Hz, which sets a lower limit for the sampling frequency of the sensors to be used in the on-board system of 540Hz.
- This has been the basis for a more detailed testing campaign, in which different sets of degraded springs have been characterized and their experimental response have been compared against the model, in order to implement a failure classifier.
- For the experimental analysis, two types of tests have been performed. The obtained results through the force-displacement relationship and the FRF of the accelerometers are similar. Both types of results provide equivalent frequency values.
- Therefore, it can concluded that for undamaged springs similar results are obtained with the exposed three analysis methods.

For damaged springs:

- Evolution of the dynamic stiffness provided by the test machine and evolution of the rms(C2)/rms(C1) ratio of each test provide similar results. However, from the postprocessing of the force/displacement values only a unique mode is obtained, the accelerometers indicate other attenuated modes.
- A positive aspect is that the results obtained with accelerometers are more satisfactory than those obtained through the representation of the evolution of dynamic stiffness.
- After analysing the FRF(C1/C2) distribution of the 8 damaged springs, a great dispersion has been observed between them. There is no clear trend in frequency response depending on the damage type of the spring. The fact of not having undamaged springs for the moment makes this distinction difficult.
- Despite this, satisfactory results have been obtained in order to identify damaged springs in service based on condition monitoring data defective. Through this analysis, significant correlation between helical spring indicator and defect type has been found, indicating promising chances of detecting a defect before there is a risk of high failure high, therefore ensuring the safe running of the vehicle.

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