Extended Abstract

Measurement and assessment of vibrations induced on a railway track

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João Diogo Castanheira Cortês Damásio Geada

Department of Mechanical Engineering

Instituto Superior Técnico, Lisbon, Portugal

Abstract

Increased interest in renewable sources of energy was one reason behind the launch of the Helianto project. This project consists of designing and building a modified train carriage capable of operating with photovoltaic energy.

In order to better understand the characteristics and effects of the project, a scaled 1/8 model was built. This way it was possible to perform, cost-efficiently, a considerable portion of the tests that need to be done to the real train carriage on several types of railway sections (e.g. curves, straight, breaking, etc.). The work of this thesis analyses the railway/bogie interactions, retrieving the maximum amount of coherent experimental results with the aim of understanding the dynamic behaviour of the scaled model gathering all the necessary conclusions to better predict the real train's performance.

From the results achieved using all the data gathered through the model (railway/bogie acceleration and deformation) it was possible to conclude that the capacitive accelerometers have a measurement error of around 1.5% when comparing them to the piezoelectric ones, for the range of frequencies of 5-200 Hz, that the model suspension system attenuates around 66% of the vertical accelerations and that the rolling coefficient stands at 0.0052.

Furthermore, using this data it was also determined that the usage of accelerometers and strain gauges in the vehicle and rails, respectively, was effective and inexpensive, avoiding any damage to the rails, which could be costly to repair and prevent usage of the rail. Finally, the comfort and safety requirements of the model were assessed, presenting a track shift force of about 59 N, running stability of 31.9 N and 0.6 of derailment coefficient, all of which were within the acceptable ranges.

Key words:

Helianto project; Experimental results; Railway track; Train; Vibrations;Accelerations

List of programs: Matlab; Excel; Sigview

1.Introduction

The Helianto project [1] was designed to overcome the effectiveness problems of the photovoltaic powered type of railway vehicles.

The aim of the project is to build a lightweight train to operate on a pre-existent train line traditionally used by diesel powered beach trains operating in touristic beaches in Portugal’s coast.

A common solution for a photovoltaic train is to equip the train station ceiling with photovoltaic panels.

Furthermore, the quick deterioration of the railway track materials in the targeted environment reinforces the need for proper means of evaluating the interactions established between the bogie of the vehicle and the railway track.

The objective of this thesis is to set up a reliable vibrational assessment system and evaluate the case study. The testing operations of the developed assessment system were carried out using a prototype of the real carriage. The reason behind the usage of a prototype, instead of a real-size train carriage, is due to strict regulations for railway and train experimentation on vehicles that travel on those lines due to safety safeguards regarding passenger and infrastructure. Furthermore the prototype can be helpful in other types of tests (apart from the vibrational assessment) and there was no prototype available previously. This prototype allows further testing of several key features of the functioning of a train.

2. Standards and parameters

2.1 Vehicle parameters

Wheel-rail interaction is one significant part of the entire system of railway vehicle dynamics and is one of the most important safety-wise. The forces on the wheel-rail contact surface on each wheel must be assessed with precision when testing a new vehicle.

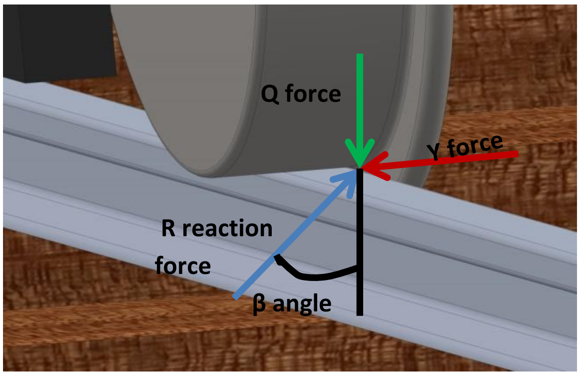

Figure 2.1 - Forces Q and Y directions [12]

2.2 Principles of railway vehicle testing

Linear accelerations are vital to the testing. As the International Union of Railways UIC's document shows, the linear acceleration felt by the wheel-set in the lateral direction is named $\ddot{y}^{+}$, and in the vertical axis $\ddot{z}^{+}$ and $\ddot{x}^{+}$ for acceleration on the railway axis. These accelerations, along with the accelerations felt on the body, above the bogie, are used to assess the average acceleration felt across the entire vehicle.

The accelerations suffered by the body are named $\ddot{x}^{*}$, if they act in the direction of movement (railway axis), $\ddot{y}^{*}$ if the acceleration pulls towards the side of the vehicle, and $\ddot{z}^{*}$ if the acceleration is felt in the vertical axis. Finally, the average acceleration felt on the body above the bogie will be named $\ddot{x}_m$, $\ddot{y}_m$ and $\ddot{z}_m$ and can be approximately determined using the accelerations named above.

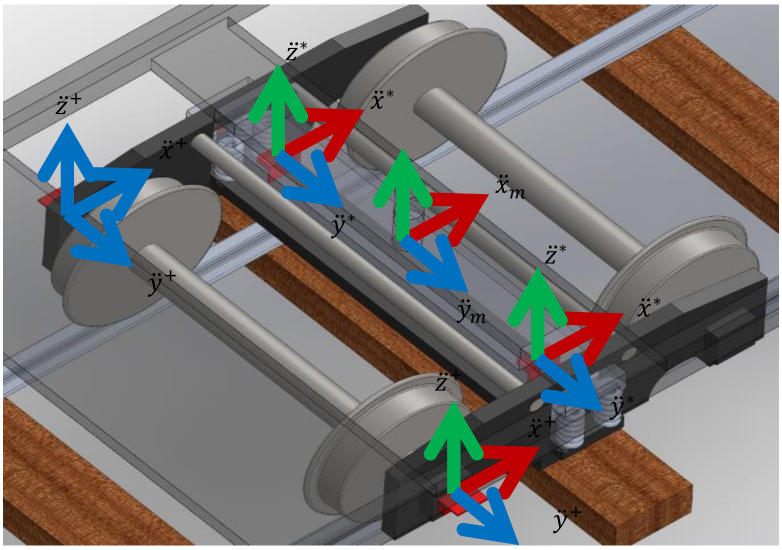

Figure 2.2 - Accelerations felt in vehicle bogie and bottom of body.

2.3 The prototype

In the context of the Helianto project, the buiding of the prototype provided several advantages, when comparing with the alternative of using a real carriage:

- Much lower testing cost than using a real train carriage in an actual railway to assess vibrations.
- Time needed to repeat or prepare a testing series was greatly reduced, while also taking less effort to assemble all the instruments.
- The most important reason is that it is actually generally forbidden to assemble devices on either the carriages and, above all, on public railway tracks. The clearance to do so is extremely difficult to obtain, since the consequences of any kind of damage on the rails could have serious consequences and, high repair costs.

3. Measuring devices and data aquisition

3.1 Measuring devices

Since the core of the work was to measure the accelerations mentioned, the basic device needed was an accelerometer, in this case a three axial accelerometer with the aim of measuring all the accelerations at the same time. There are several types of accelerometers, and the type of accelerometer that best suited the testing was assumed to be the capacitive accelerometer. However, it was necessary to make sure it would be effective, even when compared with, for instance, a piezoelectric accelerometer.

It was concluded that, although the piezoelectric accelerometers had much wider dynamic and frequency ranges and lower noise level than the capacitive accelerometers, these characteristics would not be critical in terms of the results for the frequency range needed.

3.2 Capacitive accelerometers

Both accelerometers have capabilities for the tests intended, however the capacitive ones are less costly. The piezoelectric and piezo-resistive are too expensive for the experiment hence the most adequate accelerometers were indeed the capacitive ones.

In conclusion, the capacitive accelerometers were the ones chosen after applying a cost-benefit analysis. However, further testing had to be conducted in order to assess if these accelerometers produced results close enough to the piezoelectric ones, in this testing environment. After all that testing was done, by measuring and comparing the first 22 vibration modes of a railway track with both the capacitive and piezoelectric accelerometers, the average relative error for the former, when comparing with the latter was of 1.5%, with a maximum error (of the accelerometers actually used) of 0.1g if only one accelerometer is considered on its most erroneous axis. This magnitude of errors is perfectly acceptable.

3.3 Data acquisition

Arduino board

The choice of a platform to receive and interpret the signal went to the Arduino UNO acquisition board. The Arduino UNO board is capable of I2C communication. The advantage of this acquisition board is the fact that, both the hardware, and software required to work with it, are totally open source, giving the user all the freedom to alter the board for its specific purposes. The following figure 3.1 shows the chain of data acquisition.

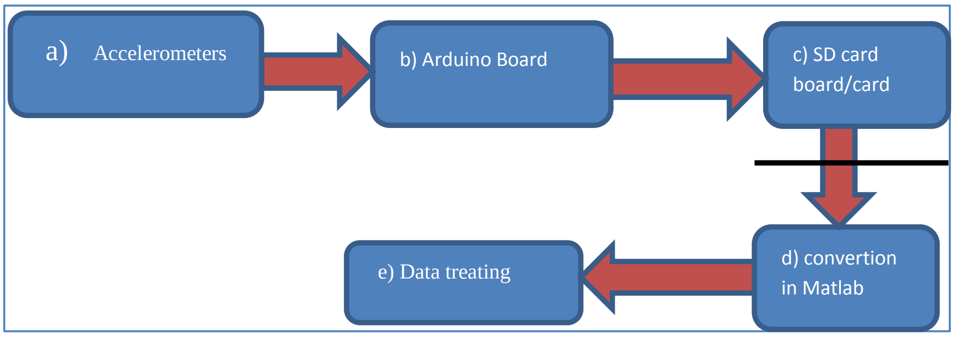

Figure 3.1 - Data collection diagram

Strain gauge acquisition board

This board is from National Instruments which uses acquisition software Labview alongside with a strain gauge input module which technical specifications. This board has 8 simultaneously sampled analog input channels, ranging between 0 and 10V and 1 to 10000, a programmed 4-pole Butterworth and NI-DAQmxX.

4. Vehicle/bogie instrumentation

4.1 Device positioning and instrumentation

Accelerometer positioning

The image presented below shows the number and positioning of all of the accelerometers and the direction of their axis:

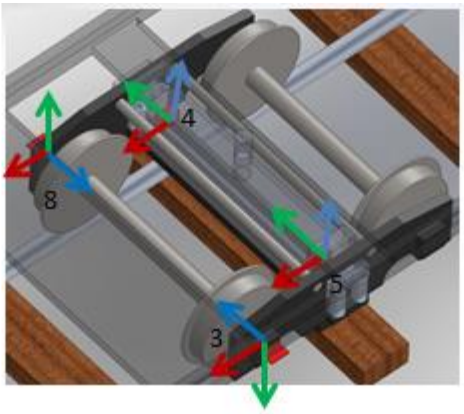

Figure 4.1 - Accelerometer positions

Prototype instrumentation

The following information refers to all the instrumentation used in the prototype:

- The maximum voltage an accelerometer can withstand (while fed with DC) is much lower than an Arduino board thus the Arduino boards were fed with a voltage of 12V (DC), and they, consequently, fed the accelerometers with 3.3V of voltage (also DC);
  In order to feed the accelerometers remotely a lead acid battery of 12V was used;
- All the equipment needed to be secured in place while the bogie was being moved across the railway. Duct tape was used to fasten all the moving objects (cables mostly);
- A type of cyanoacrylate was used to firmly tie the accelerometers' base to the metallic material;

- A micro SD memory card was used to store all the information. This card was placed in the Arduino SD board inside the carriage.
- The accelerometers' positioning is described on figure 4.
- The possibility of having all the accelerometers starting collecting data at the same time made data analysing and comparing easier. Thus there was a need to add some kind of switch that could control (start and stop recording information) all the accelerometers at the time. This problem was solved by using a three stage switch which worked flawlessly.
- All electronic components had to be isolated from metallic parts to avoid errors and they had to be wedged firmly to the carriage/bogie (figure 4.2)

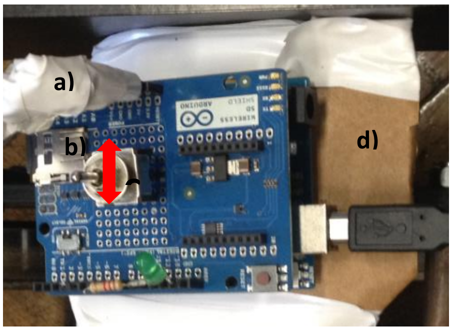

Figure 4.2 - Arduino board.

a) Data cable linked to the Arduino board end;
b) Micro SD card. Device where all the data about the acceleration variation is stored;
c) Three stage switch, although only the upper and bottom stages were used, meaning switched on (start collecting and saving data) and switched off (stop saving data);
d) Cardboard used to isolate the base of the Arduino board from the metallic parts of the vehicle.

4.2 Testing characteristics (straight railway track)

There were eight tests made with the prototype (last 4 with extra weight and the first 4 without), on a straight railway track.

The tests done without added weight consisted of the debug test, made to test the equipment, the deceleration test, the acceleration and deceleration tests, made to assess vibration with either the accelerometers and the extensometers, and the stopping by braking the wheels. The other tests were the same but with extra payload, with the exception of the debug test, which was replaced with a running test (a test which consisted of letting the prototype run the track as many times as possible, in both ways).

5. Results

5.1 Spring-damper effect

The vibrations on the carriage body caused by a change in rail had the characteristic configuration of a spring-damper system which could be analysed more deeply to assess if the vibrations are uncomfortable for the passengers and what are the spring and damper constants and how much energy is dissipated by the damper, among others.

Furthermore, all the variables of the spring damper system were assessed out of the data reading from the accelerometers, giving the following results $\delta = 0.586,\ \xi = 0.0928,\ \omega_d = 120.83\, rad/s\ \ and\ \omega_n = 121.35 Hz$ (logarithmic decrement, damping ratio, natural damped frequency and natural frequency).

To ensure that the values of frequency were correct (since they were calculated using the integrals of the data obtained), they were compared with the ones determined through the oscillatory values of acceleration, instead of displacement. This comparison resulted in $\omega_n = 121.35 Hz$, and Tacceleration = Tdisplacement = 0.052s, so there was no difference. Hence, using the correct frequency, $m = 23.3\ Kg$, $8c = 524.7\ Ns/m$, $8k = m\omega_n{}^2 = 343.11\ kN/m$.

Hence the final differential equation that describes the movement of the vehicle is:

$$23.3.\frac{d^2z(t)}{dt^2} + 524.7.\frac{dz(t)}{dt} + 343110.z(t) = 0 \qquad (5.1)$$

Another interesting result was the percentage of acceleration attenuation. To obtain an appropriate estimate of the acceleration difference between the wheel-set level and the passenger level, the mean acceleration attenuation (measured at the spot of peak acceleration felt) throughout all of the testing was determined. This resulted in 2.6g on the wheel-set and 0,88g on the body floor which represents a 66% acceleration attenuation.

Strain - Tests 1 to 3 (deceleration, acceleration and braking)

From the tests made it can easily be assessed what effect did the weight increase had in the strain measured. In the case with no added weight, the extension suffered on the xy (horizontal) and xz (vertical) planes was of 89.1 and 68.2 μƐ respectively, while in the case with 50% extra weight, both planes suffered extensions of 111.8 and 82.1 μƐ, which is an increase of 25.5% and 20.4% in extension.

5.2 Velocity and friction results

The velocity graphs/data were obtained by integrating the acceleration data collected with the sole purpose of assessing what velocities were achieved during the testing. Hence, to each and every test conducted corresponds one of this time-velocity charts. Two examples of this velocity data are presented in the figure below (figure 5.1).

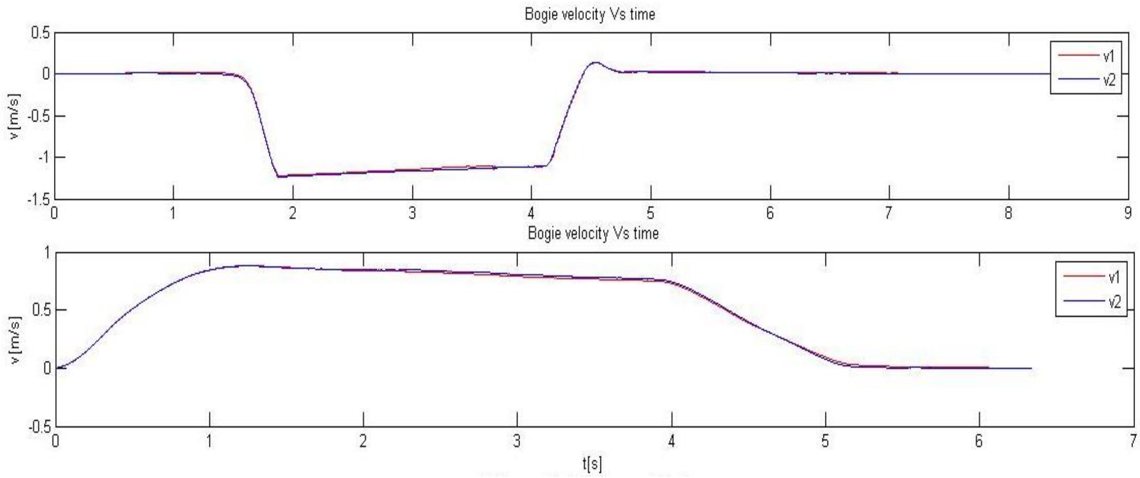

Figure 5.1 - Sample graph of the bogie's accelerometer horizontal velocity variation without and with extra load – tests 3 and 4.

Although the top speed may vary from test to test, in these two different tests, it is safe to say that the maximum velocity was about 1.2m/s for test 3 and 1m/s for test 4.

These results are consistent with the maximum kinetic energies of both systems (kinetic energy before friction forces start to affect the system) which show that $v3 = 1.21m/s$ $v4 = 0.96m/s$ and $m3 = 40kg$, $m4 = 62.5kg$ hence $Ec3 = 29.3\,J$ and $Ec4 = 28.8\ J$ (measured using the vehicle velocities in both tests and the standard kinetic energy formula). Thus the error between the two tests in terms of energy is of about 0.5J or 1.7% of relative error which was probably caused by differences in the initial impulse, hence the integration was done properly and all the results seem correct.

The slight slope in the graphs is proportional to the friction force acting on the vehicle axle while it passes over the railway. Using that slope the global friction coefficient felt by the vehicle parts can also be measured, which turned out to be 0.0052 (around an average of 2N for the case without weight and 3,3N with extra payload).

5.3 Curve tests

Tests in curve are of particular interest not only due to the standards' rules, which make mandatory to make curving tests to the new vehicle, but also because the results of curving tests allow the calculation of more variables related to the vehicle/prototype's safety and comfort.

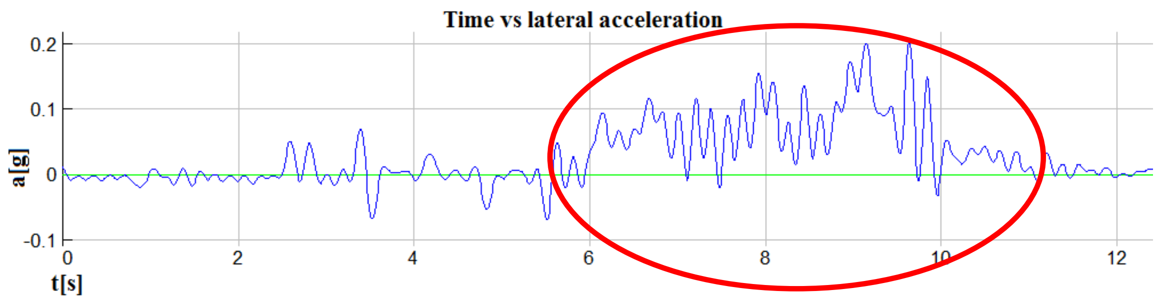

Figure 5.2 - Lateral acceleration variation with time from test in a curve (image filtered at 6Hz low-pass filter).

5.4 Safety and comfort results

Track shift force

The track shift force is determined by the net difference between the lateral forces acting on the outer and inner wheels of the vehicle. These forces are generated by the inertial centrifugal force and the friction force on the contact surface of the wheel, respectively.

After filtering the data of the accelerations measured and using the track shift force equation in accordance with the European standardization [2], the force was determined to be of 59,5N. However, since the equation for the track shift force safety condition possesses a constant a, with the value of 10 kN, the results hereby consider the formula using a proportionally smaller constant value in accordance with the smaller area (64 times smaller) and yield stress of the rails used [4], when comparing the real rails [3] with the ones used to test the prototype. This leads to a constant a = 93,3N and a maximum allowed track shift force for these rails of 126,6N > 59,5N which is higher than the maximum track shift force exerted by the vehicle.

Running stability

Another safety standard [2] is related to running stability. Running stability is measured by calculating the root mean square of the variation of the lateral force or the maximum of the lateral force value. The limit for running stability is given by thefollowing equation::

$$\frac{K_1\left(10+\frac{2Qo}{3}\right)}{2} = 63.3N \tag{5.2}$$

Also, once again, the objective function applied in this equation is the maximum of the root mean square lateral force between all of the axles (shown below), even though in the case of this work only one axle was measured; however the accelerations felt on the front axles and rear axles are similar. The objective function was determined to be $\Gamma rs = 31{,}9N < 63{,}3N$ which is less than the maximum allowed force for the prototype.

Risk of derailment

Since this is a dimensionless coefficient one can apply it directly to the model safety, given the fact that the geometry of the rails used in the model is equal to the geometry of the rails used for a real size carriage. With this being said the equation is presented below as well as the results of such coefficient, using the values of the forces measured, once again using the accelerations felt on the vehicle.

$$max\left(\left(\frac{Y}{Q}\right)_{20Hz,99.85\%}\right) = \frac{59.5N}{\frac{40.7*9.806}{4}} = 0.596 \leq 0.8 \tag{5.3}$$

Once more, the target value used corresponds to the maximum of the proportion of Y to Q; However Q can be considered to be constant since there are no bumps on the railway track and, as was explained in the assumptions, the slope of the vehicle body is negligible, thus, the maximum of the combination of variables corresponds to the maximum of lateral force (or acceleration).

5.5 Real vehicle extrapolations

One of the most fundamental relations to be considered while working with models is the geometric relations between the real and the model objects. Geometry is important since it defines the directions and, thus, the relations between the forces.

In this case the model tests were made on a 7 m wide curve, hence, in order to maintain the same geometric positions of the vehicle wheels both in the model and in the full-size vehicle, it would be required to consider an hypothetical curve 8 times wider (since the real vehicle is 8 times larger). With that being said, what is left to determine the estimate of the centrifuge acceleration is the speed at which the vehicle travels as well as its mass. This velocity is stated to be of 30 Km/h and the fully loaded vehicle weight is 5300 Kg [1]. This makes it possible to have some estimates of the track shift force and risk of derailment results.

track shift force

In this case, one must start by determining the centrifuge force, which ended up being $Fc = 1.24\, m/s^2$ and then the track shift force itself can be calculated, resulting in a track shift force of 1643 N. The maximum allowed track shift force is of 14.33 kN, which is much higher than the 1643N exerted by the real train on the real axels.

Risk of derailment

The derailment coefficient for the risk of derailment assessment is the most direct to obtain since it is a un-dimensional value, hence it only depends on geometric properties and not on other physical properties which means that the maximum allowed coefficient is the same for the prototype and the real train. Hence the derailment coefficient was determined to be 0.123, which is below the 0,8 accepted.

6. References

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