BOB MOVEMENT AND VEHICLE ACCELERATION RUNNING A SUPPORT OR COMPRESSION TOWER

Pierre JAUSSAUD and Nadine MARTIN[1]

1 Introduction

Each country applies strong rules to ensure safe transport to passengers on a rope line. Among these rules, many concern the strains in vehicles, and the lifetime of the installation (towers and vehicles). Up to now, most of these rules are checked after the rope line has been build. This may have very strong economical consequences, as it is difficult to forecast the experimental results, and sometimes, the conception has to be modified when vehicles or towers have been build. Most of the necessary characteristics could be calculated from finite elements models, provided one knows the dynamic forces that the installation endures. Some research has already been done on that topic, for compression towers, which give a signal comparison between theory and experiment that need be improved [1], [2] [3]. To our knowledge, no paper has been published on support towers. We have been looking for a method that would give the dynamic forces applied to towers and vehicles. We want to show here how these forces can be modeled from line calculation and characteristics, both for support and compression towers, although with different mechanisms.

The first problem is to make the right assumptions about the mechanical origin of the forces and see what the model gives. We then present some experimental results of our models, compared to theoretical calculation.

2 Common Physical features.

Compression and support towers have common features : in both cases, the bob is designed to rock around its main axis. In both cases, the bob is also a resonant mechanical system. The number of resonance frequencies equals n-1, where n is the bob wheel number.

Both rocking and vibration phenomena are always present. The relative amplitude of these phenomena depends on the tower type. Rocking is induced by the travel of the vehicle all over the tower. Resonant frequencies are excited by shocks or periodical forces. The frequency analysis of vehicle acceleration on compression or support towers gives a spectrum that is rather large, independently of the rope induced vibrations. This indicates a predominance of shock excitation. On the other hand, support tower observation suggests that rocking is the principal mechanism.

Finally, we must keep in mind that, mechanically speaking, the low frequencies are more dangerous than the high frequencies : for a given acceleration, the amplitude movement and hence deflections vary as the inverse of squared frequency.

3 Support tower case

Compared to compression towers, support towers are usually smoother. However, on lines that have a small rope tension and heavy vehicles, we could observe strong

[1] LIS/ENSIEG, BP.46, F-38402-ST MARTIN D'HERES CEDEX
@Mail : pierre.jaussaud @lis.inpg.fr or nadine.martin@lis.inpg.fr

dynamic effects. On this type of lines, the bob rock is quite visible. This is often the case for pulsed gondolas, mainly if the hydraulic jack is under-dimensioned. This situation can be suspected to be one of the causes of some accidents. A usual characteristic of support towers is that the rocking movement is more important than the shocks which come from contacts between the grip jaw. We shall thus focus our interest to this rocking movement.

To study the bob movement we have to solve the differential equation that the system satisfies. This equation stands as : $J\frac{d^2\theta}{dt^2} + R\frac{d\theta}{dt} = \sum External\ Torques$. θ is the bob angle with the equilibrium position and R the damping constant. In the external torques acting on the bob we find the torque due to the rope and the torque due to the vehicle weight. In the inertia J we count the rotation bob inertia, the rotation vehicle and the rope inertias. Obviously, the vehicle rotation inertia varies when the vehicle goes on. We assume initial conditions on angular position, speed and acceleration for the bob. These will be taken as the angular equilibrium position when the installation starts, and zero for initial bob angular speed and acceleration. The neighboring towers are represented by their fixed drawing point.
2-wheels bob sub structures are not taken into account : this is equivalent to neglect the bob curvature and seems to be reasonable as we are not interested in the wheels load evolution.

The equation can only be solved numerically, step by step. We have done that in our program **CARMEN** for rope lines calculations. Speed at step n is calculated from positions at steps n-1 and n. In the same way, acceleration at step n is calculated from speed at steps n-2, n-1 ,and n.
When the bob rotation acceleration is known, it is very simple to get the vehicle acceleration.

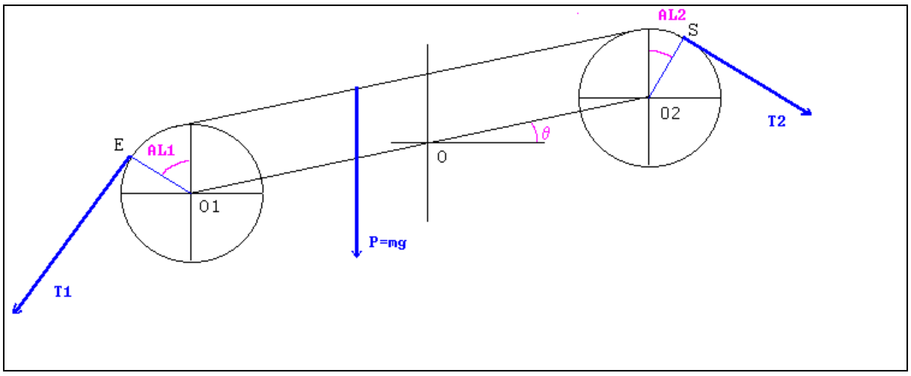


Figure 1 : BOB MODEL

Fig.1 shows the parameters that are involved in the calculation to find the bob position θ, the speed and the acceleration : rope tensions, input and output rope points E and S. P is the vehicle weight.
The vehicles are moved step by step, and we shall have to check the influence of the step length. We first tested the coherency of our model with a simple symmetric rope line, represented in fig. 2. This allowed us to check that symmetry properties were respected.

Also on this model, it was easy to check some important points:

1. Bob angular acceleration an vehicle vertical acceleration should decrease when rope tension increases.
2. These accelerations should increase with the cabin weight.
3. Dynamical effects increase with rope speed.

The first calculation that was done is named “reference calculation” . Only one parameter was changed at the time to check the here above mentioned points.
To simplify the understanding of the results, line characteristics are such that there is only one vehicle on each span of the line. The bob inertia was calculated from a standard 8 wheels bob. Damping coefficient is taken from our observations on bob oscillations and from our signal analysis.
This reference calculation had the following parameters :

Rope speed : 5m/s
Passenger flow :1800 p/h
Cabin weight : 600 DaN
Rope tension on return pulley :25000 DaN
Vehicle separation : 100 m
Up-hill Load : 10 passengers
Down-hill load : 0
Bob length : 4.3 m
Wheel radius : 0.3 m
Wheel weight :30 DaN
Bob rotation inertia : 3900 kg.m²
Calculation step : 0.25 m
Damping coefficient : 4000 N.sec

Test line is thus as follows :

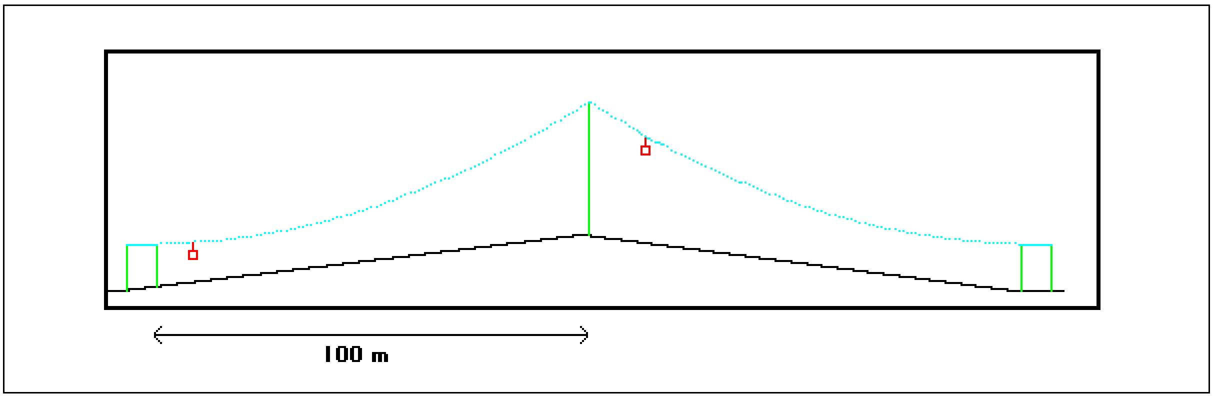


FIGURE 2 : TEST LINE

Using the above parameters, we find the results presented in fig.3 for up-hill side. For the other side, we get similar results.

Upper part of figure 3 presents the bob angle, and lower part presents the vertical component of acceleration on the vehicle.

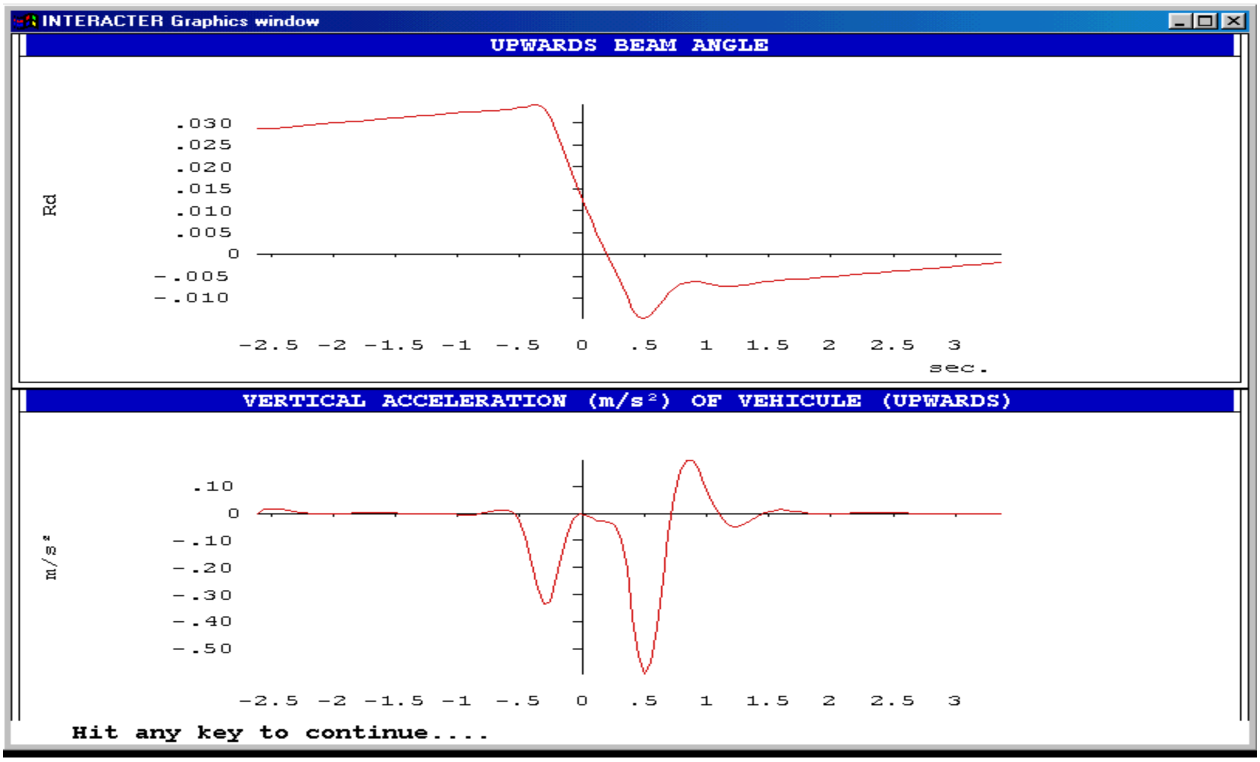
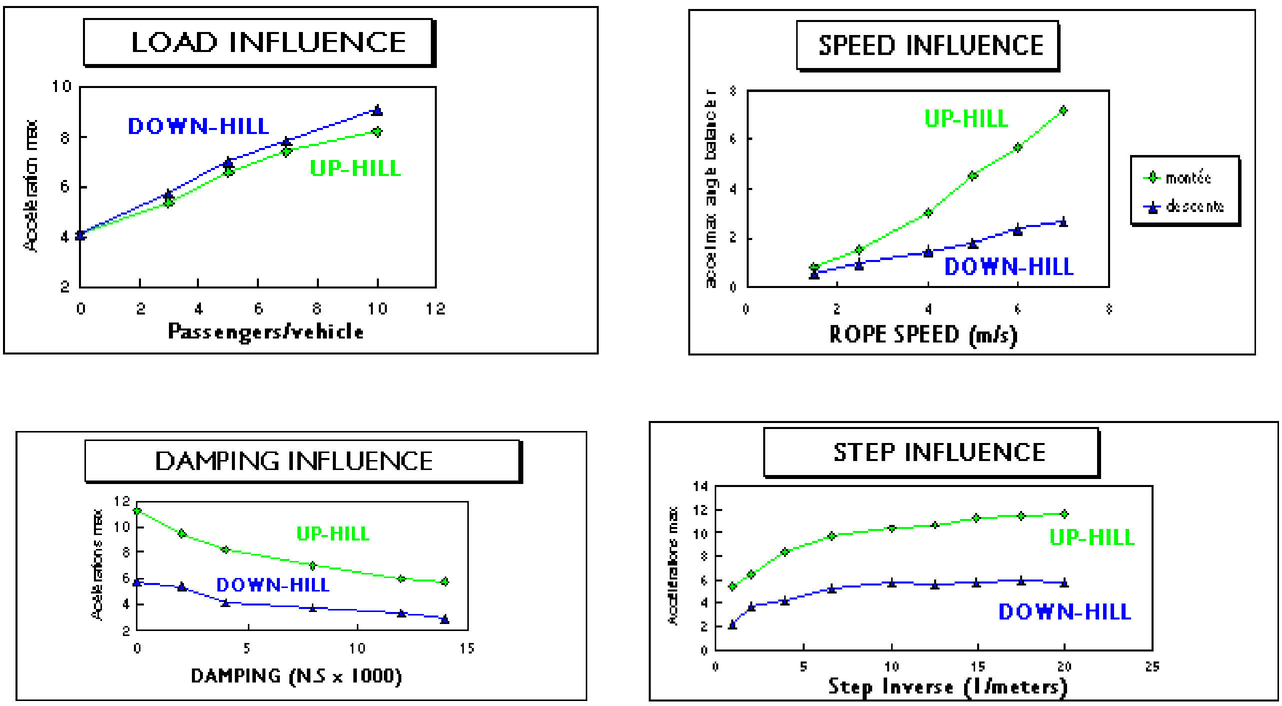


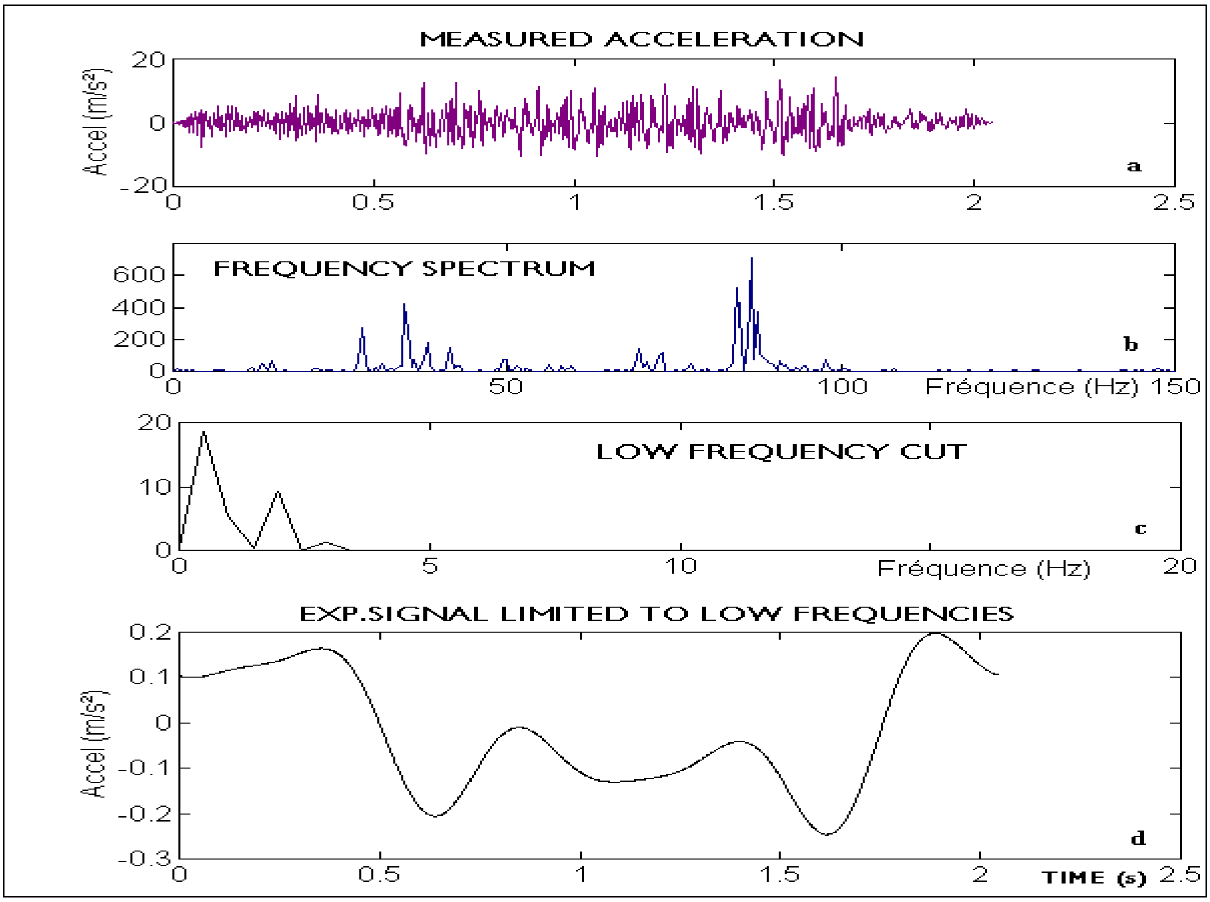
FIGURE 3 : CALCULATED SIGNAL ON TEST LINE

Figure 4 : INFLUENCE OF SOME PARAMETERS



4 COMPARISON MODEL/EXPERIMENT

We performed comparisons on various types of lines : fixed chairlift, detachable chairlifts and double rope installation. To make short, we only present here the comparison for a detachable chairlift, situated in VAL D'ISERE(France). The signal measured is the vertical component of the acceleration. The accelerometer is fixed to the head of the vehicle suspension). The accelerometer is a strain-gage accelerometer. Its pass band extends from 0 to 15000 Hz. The sampling rate was 300Hz. An example of the measured signal is given by fig.5a



.

Figure 5 : 4 SEATS DETACHABLE CHAIRLIFT

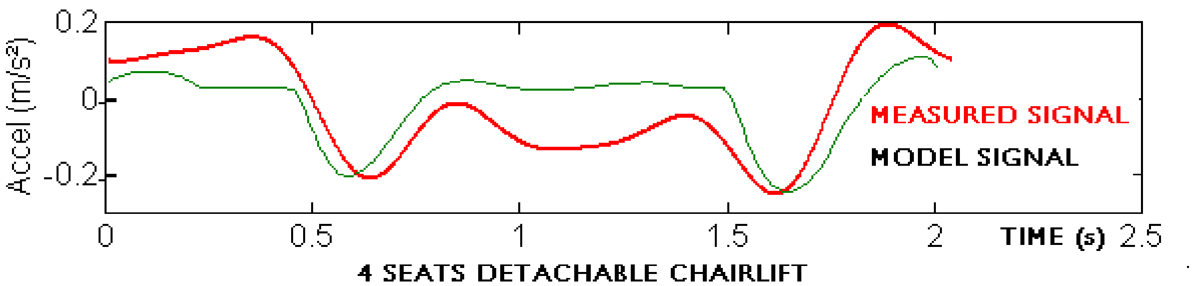


FIGURE 6 : COMPARISON MODEL/EXPERIMENT

The signal in Fig.5a may be compared to the calculated signal for that line. The comparison was performed in the following way : First of all, the experimental signal was filtered so that only low frequencies (<5Hz) were kept (Fig.5c). These frequencies are then recombined to give the signal in Fig.5d. Then this figure and the model signal are plotted on the same figure, namely figure 6. If we had not performed this filtering, the signal would have kept high frequencies due to rope vibrations and shocks between the grip and the wheels. These frequencies have nothing to do with the bob movement and are not generated by our program. As figure 6 shows, the agreement is fairly good.

5 COMPRESSION TOWERS

As mentioned earlier, shocks seem to be the most important contribution to dynamic effects on compression towers. In earlier works, [1 to 6] it was tried to use all frequencies of vibrations. Some of them [3, 5] even included tower resonance frequencies. Although the overall shape of the model signal had some similarities with the experimental signal, the agreement was not satisfactory. We came to the conclusion that the model should be modified, taking into account only one frequency, the resonance frequency of the two-wheels bobs that make the 4 and 8 wheels bobs. The reason for that is that the grip is always in contact with a 2-wheels bob, and all these have the same frequency. The coupling between two-wheels bobs is not very strong, and therefore the elementary resonance frequencies should not be widely separated : in other words, all frequencies are nearly equal.
So we assume that one resonance frequency is excited by the shocks. The bob response is given by a damped sine under the form of the real part of

$$\gamma(t) = Ae^{j(2\pi\nu t-\varphi)}e^{-\alpha t} .$$

In this formula we have 4 parameters to determine : amplitude A, phase φ, frequency ν and damping coefficient α. Frequency can be calculated from rope tension and bob inertia. It is commonly about 30 Hz. We found 29 Hz. This value, as well as α can be checked on the experimental signal, at the end of the signal, when the last shock has occurred leaving then the signal gently decrease on a few periods.
Amplitude is not calculable because friction and shocks amplitude are probably one of the least known fields in Physics. However, we could observe that for a given manufacturer type, the first shock always has the same amplitude. Phase is usually π or 0, depending if the shock comes from extremities of the grip or from the transition between grip slope and flat part.
When a grip has to pass a compression tower, it must thus insert itself between the wheels and the rope. The result is a shock (a shock is defined by a very rapid momentum variation). Due to the usual shape of the grips, shocks occurs 4 times for each wheel (Fig.7) at : grip contact (shock N°1), transition from grip slope to grip flat (N°2), transition from grip flat to grip slope (N°3) and wheel contact loose (N°4).
The shocks N° 1 and 4 are likely to be identical, as the momentum change is identical, and so are N° 2 and 3. But 1 and 2 are assumed to be different in amplitude. So we shall assume an amplitude ratio that has to be determined, as well as the damping coefficient. The phase of the signal is determined by initial conditions. From experimental signal we shall assume in a first step that it is π for shocks 1 and 4, and 0 for shocks 2 and 3, as we assume that the initial situation for vertical speed and movement is zero. One of the aims of the signal processing last part of this paper is to validate these assumptions.

Finally, to the previous signal we must add the vibrations due to the rope shape which is not a perfect cylinder. The frequency of these vibrations may easily be calculated from the rope cabling step and speed and usually stands between 50 and 90 Hz for a speed of 4.25 m/s.

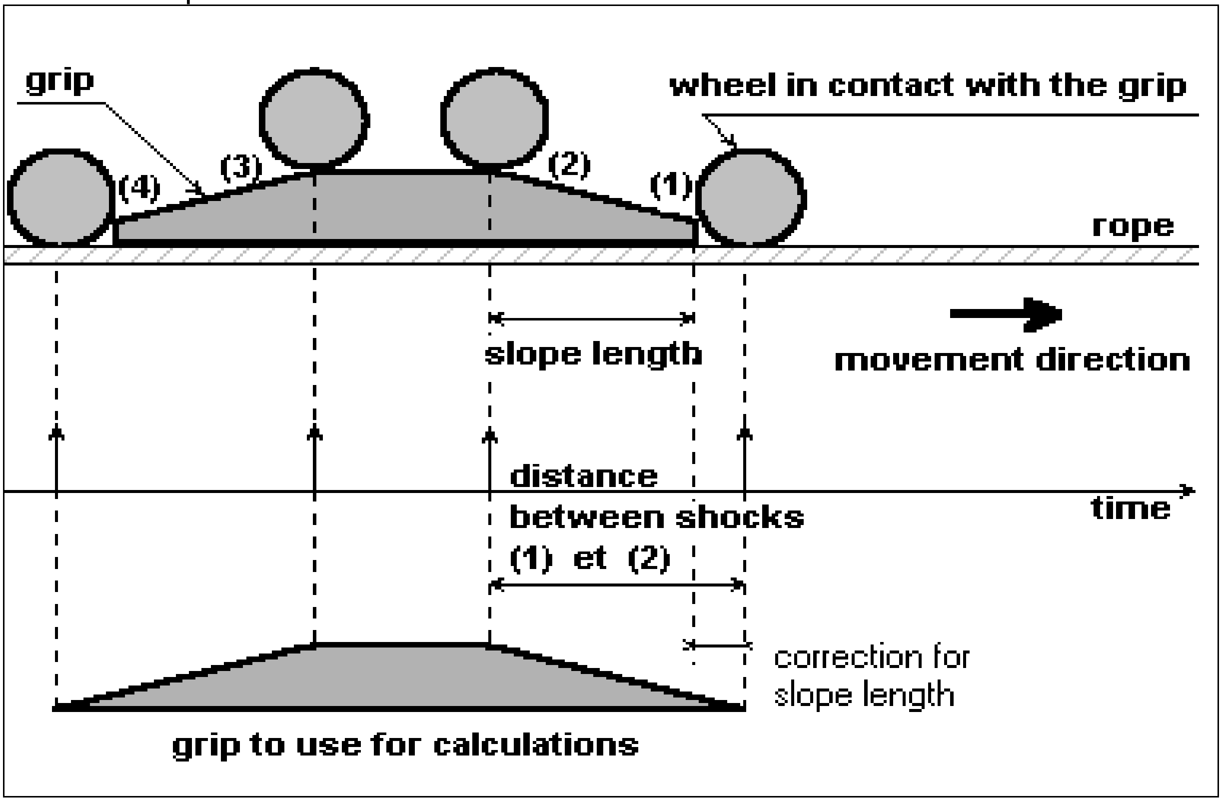


Figure 7 : GRIP DESCRIPTION

6 RESULTING SIGNAL MODEL

The complete signal will be obtained by superposing the damped sines with time shifts as expected from the wheel distribution. Another shift factor must be introduced due to the wheel and grip shapes, as shown on fig.7, which presents the correction to contact points. As explained before, the shock N°i generates an acceleration signal γ_i in the form of a damped sine :

γ_i (t)= $A_i \exp(-\alpha t) \sin(2\pi\nu t - \varphi_i - \delta_i)$ (1)

In this expression δ_i is the time shift between shocks and is given by $\delta_i = 2\pi\nu d_i/\mathbf{V}$, where d_i is the distance between wheel N° i and wheel N° 1 and **V** is the rope speed.

The resulting acceleration is the sum for all shocks (4 x number of wheels) given by the former expressions, so that

$\gamma(t) = \Sigma_i \gamma_i(t)$

Parameters for shocks 1 and 4 were determined above. Let us now look at shocks 2 and 3. For geometrical reasons the phase is opposite to that for shocks 1 and 4, and thus equals 0. However we are not able to estimate the amplitude of the shocks 2 and 3. This is the only parameter still unknown. We are only sure that this amplitude is less than the former amplitude for shocks 1 and 4. The ratio of this amplitude to the former one is a parameter to adjust once for all for each type of rubber on the wheels. We shall have to find it from a comparison between the experimental signal

and the model. We used for that a signal measured in ISOLA 2000 (France) on a twelve wheels tower. A few iterations on this value and comparison experiment/model leads us to an amplitude value of 12 m/s². The rest of the paper will deal all along with a chairlift in VAL D'ISERE, although we found a good agreement for the other plants.
To the accuracy of the speedmeter, the speed was about 4.25 m/s. This is a critical parameter for the model, as a small variation of speed is equivalent to an error on sampling, and will thus lead to disagreement between experimental and calculated signal.
Using the method described above, we could model the signal for the vehicle : the experimental signal is presented on fig. 8 (red), superposed to the model (blue). Although some peaks have a different amplitude, most of the peaks are well estimated, and their position is fairly accurate. The same comparison is presented in fig.9 for Power Spectral Density calculated with Fourier transform. The sampling rate of measurements is 300 Hz. The sampling period is therefore 0.33 ms , and, when speed is 4.25 m/s, a sampling distance of 14 mm, which is the order of magnitude of the slope length correction (fig. 7). Thus we are likely to be limited by the precision fixed by our measurement equipment. It can actually be observed that most of calculated maxima are higher than the experimental maxima (Fig.8). This result that may be understood from obvious sampling considerations : the samples are not necessarily taken on the maxima.

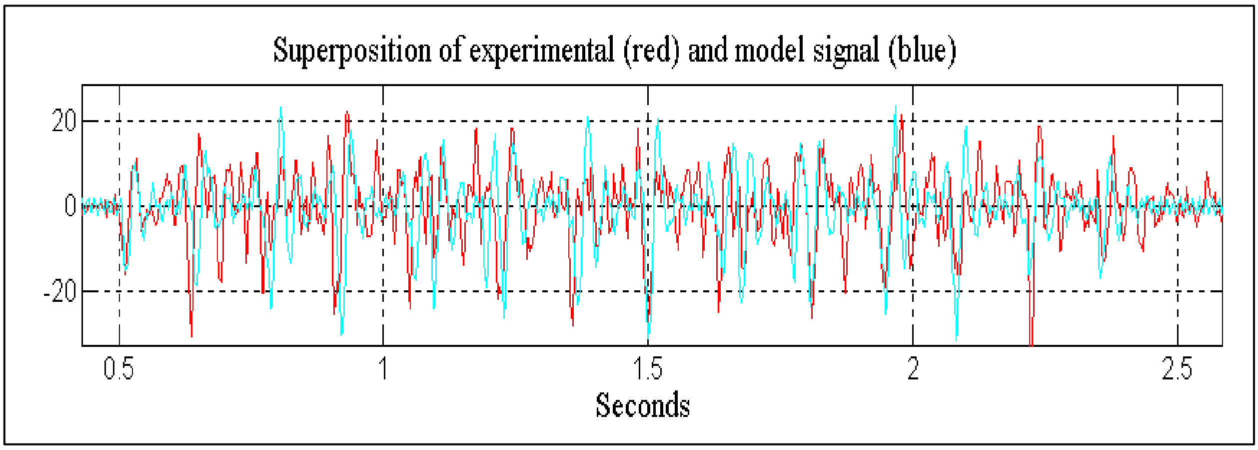


Figure 8 : Experimental and model signal

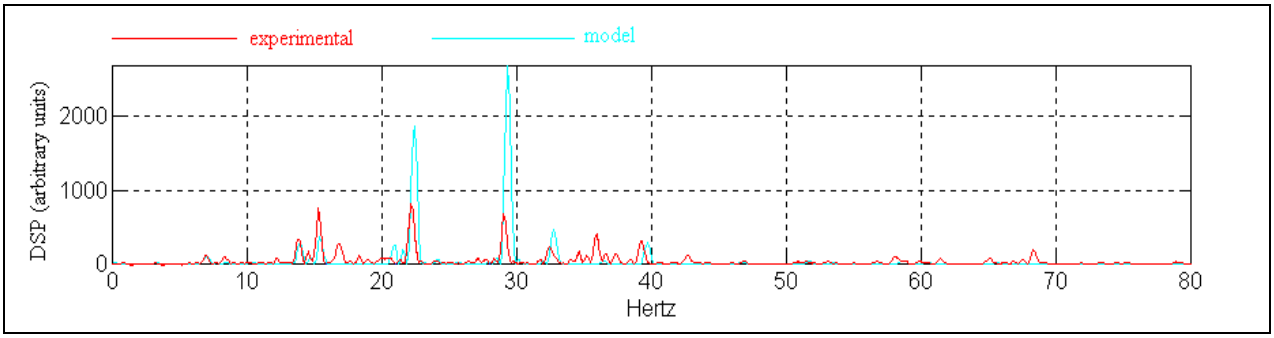


Figure 9 : Squared Fourier PSD of signals in fig. 8

7-<u>IMPROVING THE ANALYSIS WITH SIGNAL PROCESSING</u>

Up to now, we assumed that all parameters where conserved for all corresponding shocks. We have no real proof of this point. So we are now interested in a better knowledge of these parameters. We will thus use a method that we developed and that is based on successive PRONY analysis [7], as Prony analysis is well fitted to that type of problem since it deals with damped sines.
A classical application of this method cannot separate transients when they are close to each other. The analysis window has to start at the same time as the shock. To remove this difficulty, our analysis holds 3 successive gliding PRONY analysis. The first two are devoted to detect the starting point of the shocks. This will allow us to define contiguous windows from the shocks instants detection. The third analysis is performed to get the value of the 4 parameters described in equation 1.
Let us present these three steps:
1-The first PRONY step is devoted to shock instant detection. It involves a point by point gliding window larger than the separation expected for two successive shocks. The Prony estimated amplitude is maximum when the window opens at a shock instant.
2-The second PRONY analysis is performed with contiguous windows the width of which is equal to the time separation between two successive shocks as estimated from the first step. These windows are thus smaller than the former windows and have a small number of points. We apply to these windows the first step of the PRONY analysis to get an estimate of the frequencies and damping factors.
3-The third PRONY analysis is performed with contiguous windows defined from the shocks instants that we got in the former step. This leads to the final values of all four parameters of equation 1 for each shock.

All these results can now be used to reconstruct the model of the signal. The result may then be compared to the experimental signal (Fig.10). From that figure, we can observe a better agreement between the amplitude of the experimental and calculated signals. One of the major causes of small differences probably come from the signal sampling in experiments.

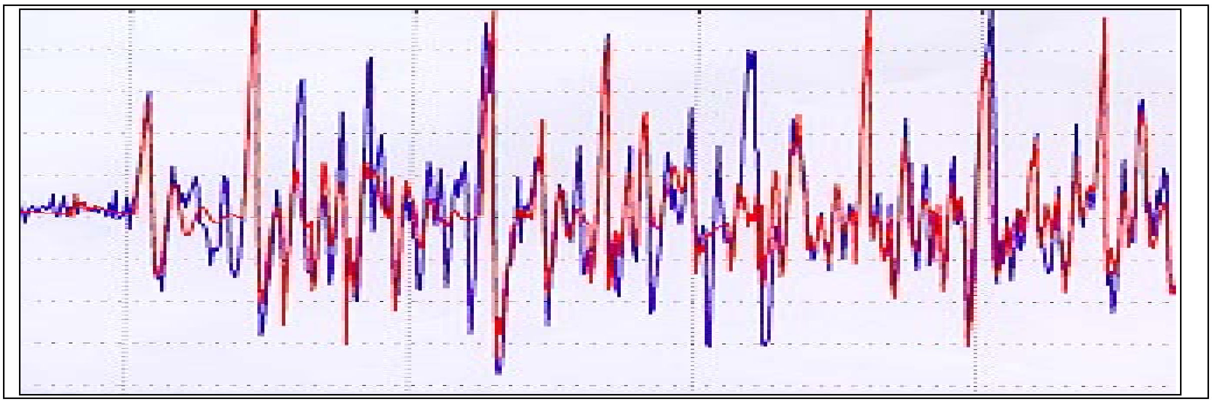


Figure 10 : reconstructed signal by the proposed method (acceleration(±20m/s²) /time. Blue : Measure. Red : Model

8 CONCLUSION

Through this paper we reached to aims : present a way to model the acceleration of a vehicle passing both support and compression tower on a rope line, and using the same type of signal, show that multiple gliding PRONY analysis can be used to extract the shock instants and parameters of grip to wheels.

The model reaches a rather good accuracy and may be used to estimate the acceleration to apply in fatigue calculations of vehicles or towers. It gives us the possibility to find the real dynamic safety factors used in fatigue calculation. This point is very important, as an increase from 1 to 1.2 decreases the expected life time by a factor 2.5.

REFERENCES

[1] CONSONNI R., DUBUISSON J., et JAUSSAUD P. : “Origine des sollicitations des véhicules des téléphériques monocâbles au passege d’un pylône compression”, 3° congrès ISR , Vienne, 22 et 23/04/1985.

[2] DUBUISSON J., Etude et Modellisation du fonctionnement d’un balancier de compression d’un télésiège, Thesis, I.N.P.G.,1986

[3] PADOVESE L.R., Etudes de phénomènes dynamiques dans les pylônes compression des téléphériques monocables, Thesis, I.N.P.G.,1992.

[4] DUBUISSON J., TERRIEZ J.M., Simulation of the movement given to carriers of ropeways installations when passing over towers, CSME, Mechanical Eng., TORONTO, 1990.

[5] PADOVESE L.R. : “Étude des phénomènes dynamiques dans les pylônes compression des téléphériques monocâbles”, Thèse, Université Joseph Fourier, Grenoble. 1992.

[6] PADOVESE L.R., TERRIEZ J-M, MARTIN N. : “Étude des phénomènes dynamiques dans les pylônes compression des téléphériques monocâbles”, Mécanique Industrielle et Matériaux - Vol. 48 - N°5 - Mars 1995.

[7] JAUSSAUD P. and MARTIN N., PSIP Symphosium, January 18-19, 1999, PARIS : Detection and modeling of a vehicle acceleration on a compression tower by a triple Prony processing.