Modal Testing of Trunk Shakers Used in Olive Mechanical Harvesting

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Abstract
Nowadays, mechanical harvesting involved in olive-detachment through vibration is the most frequently used and extended technique. However, the mechanical processes lack in the development of efficient harvesting machinery. With the aim of obtaining an optimum design, a dynamic analysis of the harvesting machinery is needed, especially in those machines whose functioning depends on vibratory systems. In this sense, modal testing proves to be a useful method to identify the dynamic behaviour of the structure.

This research attempts at determining the dynamic behaviour of the vibrating heads used in the olive detachment process so as to achieve an optimum design that seeks for reduction of damage to the tree’s bark.

The results obtained from modal testing have established the most important natural frequencies in the shaker. Furthermore, the dynamic characteristics of the shaker have been identified according to its modal shapes and mode participation factors. The comparison of these results and those obtained in the spectral analysis have shown the problems in the design and use of the shaker from the point of view of both the damages caused to the bark and the vibration transmission.

INTRODUCTION
Over the last 30 years, the developing of harvesting techniques has been key to the mechanised cultivation of the olive tree and the growth of this sector. Those based on olive harvesting through vibration are the most frequently used techniques. The design of trunk shakers aims at transmitting maximum vibration to the tree in order to optimise fruit detachment while minimising vibration transmission to the transporting vehicle.

The achievement of these two objectives must avoid the excessive power used for vibration as well as possible damages to the olive tree that might reduce production (Horvath and Sitkei, 2001; Gil, J.A. et al., 2001). Nowadays, however, the mechanical processes lack in the development of efficient harvesting machinery and that is mainly due to the following reasons:

a) The olive tree features a characteristic fructification and dynamic behaviour. Its production is concentrated around last-year grown branches and it shows high damping values during the trunk shaking process (Castro et al., 2004)

b) The design of trunk shakers has traditionally had an empirical approach, featuring variations to suit all varieties of olive trees.
Although the variety of designs for vibrating heads are wide ranging, all vibrators currently used in olive-tree harvesting are inertial-mass and feature the same vibration system. The vibrating head remains both suspended from a variable number of fulcra located on the chassis of the vehicle and attached to the trunk through a grabbing device.

This research attempts at determining the dynamic behaviour of the vibrating heads used in the olive detachment process so as to achieve an optimum design that seeks for reduction of damage to the tree’s bark.

All harvesting through vibration techniques must avoid excessive displacements that may derive in tangential forces on the trunk (Abdel-Fattah, H.M. et al., 2003), especially when the grabbing device is close to the ground. With that aim, this research has taken into account only those situations where the stationary conditions of the vibrator have been established in advance.

As any other mechanical system, trunk shakers behave as physical oscillators where a hydraulic engine moves an unbalanced mass from within, thus generating a periodic propelling force. The vibrating heads may produce different responses in the tree in relation to the frequency value. When the angular frequency of the unbalanced mass (\(\omega\)) is close to any of the natural frequencies (\(\omega_0\)) of the vibrating head, resonance effects will occur in the vibrator. In those cases, close to resonance phenomena (\(\omega \approx \omega_0\)) there will appear the maximum displacements according to the predominant vibration mode and the established frequency.

We work on the assumption that the suspension of the vibrator from the chassis poses no restrictions to free rigid body vibrations of the system within a specific displacements range. Actually, systems complying with this characteristic are quite usual.

A multidegree-of-freedom system can feature up to six rigid body modes with the corresponding natural frequencies equal to zero. There can be three modes for rigid body translation (one for each of the Cartesian axes), and three modes for rigid body rotation, (one for each of the three Cartesian axes). Both vibration modes and natural frequencies can be determined by solving an equation of motion.

However, due to the attachment to the trunk, as well as to the support provided to the vibrating heads by the transporting vehicle, the system cannot be accurately called a free body. The forced vibration and the movement restrictions from the outside make both the system and the equation solving quite complex, for the number of degrees of freedom (\(n\)) becomes wide-ranging. Modal Testing proves to be one of the most common solutions to this problem. It works on the assumption of the expansion theorem, which states that the displacement of masses is expressed as a linear combination of the normal modes of the system.

Therefore, modal testing has been applied to the system integrated by the vibrating head and the trunk in order to determine its dynamic properties.
MATERIALS AND METHODS

A standard shaker, suitable for several kinds of olive tree has been used for research. The vibrating head is located at the front of a self-propelled vehicle. It weighs nearly 1000Kg and has a symmetric distribution of masses. A hydraulic engine moves the unbalanced mass located in the middle of the head. As figure 1 shows, the vibrating head is suspended from a telescopic arm through two soft suspension pads located at the back and through a chain at the front of the shaker. The grabbing device is made of two symmetric arms which stem from the same point located in the middle of the shaking head, behind the hydraulic engine. Both work simultaneously and at the same closing velocity, so that the trunk always remains in the shaker’s central axe (Y).

A shaker-post system has been used to analyse the dynamic properties of the system, in order to standardise and to simplify research conditions for future repetitions with a different shaker. The dynamic parameters that have been identified are the following:

- Natural Frequencies. Each vibrating head features natural frequencies that may cause resonance. These have been analysed in a range between 10 and 100Hz.
- Vibration Modes. The rigid body mode and flexion mode that can appear during the resonance process are calculated for each natural frequency in order to determine how and when maximum displacements take place.

As shown in Figure 2, eleven points have been chosen to study the geometric characteristics of the shaker. The accelerations in each point have been registered by three 3D PCB 356A02 accelerometers in four series of measurements. This has produced a model of the structure with 33 degrees of freedom (DOFs) distributed over the vibrating head.

A 1.8 kg. impact hammer has been used to excite the system. It measures the force applied through an internal ICP-type load cell. To ensure the right excitation of all modes, impacts on the vertical and horizontal directions have been applied to the device. According to the weight of the vibrating head the impact surface of the hammer presents a high stiffness.

To have an accurate knowledge of the system, the frequency range of the shaker’s response has been studied from 10 to 168 Hz. An OROS-25 dynamic signal analyser (model PC-Pack II) has been used to identify the measurements registered by the sensors. The Frequency Response Functions (FRFs) will be subsequently used in the adjustment of the modal parameters, using as a reference the point of impact on every registration points. Figure 3 shows how tests are carried out on the shaking-post.

A MDOF (multi degree of freedom) technique has been employed through a MIMO (multi-input-multi-output) analysis on 11 different points of the machine within the frequency range concerned. The estimate of the stability poles for each modal frequency and damping ratio has also been carried out so as to identify the vibration modes. Software LMS Cada-PC v2.0 has been used for the curve fit of modal parameters.

The four modes and the data obtained in the curve fitting of modal parameters have been identified by means of a Stability Diagram. The stability poles for each mode within the frequency range have been established prior to the estimate of the complex residues related to each mode. This has enabled the identification of the four vibration modes. The modal frequencies and damping ratios are shown in Table 1.
The degree of independence between the vibration modes has been tested with the Auto MAC (Modal Assurance Criterion), where a set of mode shape vectors are self-correlated. It is expected that different modes are completely independent of each other. Usually, values in the MAC matrix close to 90% indicate similarity between modes.

Once the most important dynamic variables have been identified, it is necessary to test their influence on the real functioning of vibrators. Therefore, a second stage has focused on the study of the shaker applied to the trunk of an olive tree in normal working conditions. The aim is to identify the shaker’s functioning through the unbalanced mass rotation frequency ($\omega$) and its relation with the natural frequencies previously arranged ($\omega_0$), as well as through the results of the trunk vibration.

The harvesting tests have been carried out in an intensive-farming olive tree of the “picual” variety. Functioning under nominal conditions, the vibrating head has yielded a sample of 8 trees and an average vibration time of 15 seconds.

Two triaxial accelerometers have measured the accelerations produced in each tree. An accelerometer located on the vibrating head has measured accelerations derived from the shaker’s unbalanced mass, whereas accelerations in their transmission to the olive tree have been measured by another accelerometer attached to the trunk. As figure 4 shows, in both cases the sensors use the referential axes that were used in the modal testing applied to the shaking post.

RESULTS

Table 1 gives the natural frequencies and modal shapes of each frequency. The importance of each vibration mode in relation to the vibration results has been determined. The predominant rigid body modes appear at low frequencies, whereas the flexion modes results from higher frequencies. The AutoMAC matrix used in Table 2 shows the degree of independence of the modes.

Usually, the angular frequencies of the unbalanced mass may vary from 20 to 40 Hz. Though all the tests carried out, the shaker’s average frequency has been 37.3 Hz, as Table 3 shows. Therefore, the modal shapes fall into the frequency range that had been established for the modal testing. Since fruit detachment occurs at a low frequency, the rigid body vibration modes prove the most simple and interesting. A first mode (16.38 Hz) stands out with a modal participation close to 70%, followed by a second mode (41.41 Hz) whose frequency is very close to that of the shaker.

The vibration modes that have been identified (Figure 5) can be described as follows:

- **RBM 1.** (Rigid Body Mode). Displacement of the shaker’s plane (XY) on a horizontal rotation axe parallel to the X axe. It shows maximum displacements, when it is vertically attached to the trunk and both shaker’s arms work in the same direction.

- **RBM 2.** Displacement of the shaker’s plane on an vertical rotation axe parallel to the Z axe. The displacement of the shaker’s arms on the horizontal plane produces movements perpendicular to the trunk.

- **RBM 3.** Rotation of the shaker’s plane on the Y axe. It shows maximum displacements, when it is vertically attached to the trunk and both shaker’s arms work vertically but in different directions.

- **FBM.** (Flexion Mode) More complex mode shapes corresponding to higher frequency modes.
DISCUSSION

Several conclusions can be drawn from the results previously stated.

The vibrating head chosen has a high weight and length and hangs from the chassis at one point located at the front and two at the back. The angular frequency of the shaker’s unbalanced mass varies between the frequency of the first and second vibration mode, although it is closer to the second and, therefore, to the resonance situations that are characteristic of that natural frequency with a maximum horizontal displacement of the shaker. Consequently, the displacements of the second vibration mode become more apparent during the transit period at the beginning and at the end of the vibration, when the force applied by the unbalanced mass has ceased and the system vibrates freely. At this stage, the excitation and natural frequencies are very close to each other, thus contributing to the movement and leading to maximum displacements of the vibrating head. The duration of the transit periods (when starting and stopping the vibration) is rather short (around 1 second) due to the shaker’s own mass and to the olive tree’s high damping values.

During the vibration’s stationary period, the higher vibration transfer values are registered in the closing direction of the shaker arms (X axe), whereas the lowest frequencies appear on the Y axe. As for the accelerations on the vertical axe (Z), low values appear ranging between 7.5 and 9.1 m/ s².

Finally, the second and third modes can be said to be less relevant to the torsion movement. This is due to a characteristic grabbing system of long arms and to the distance between the shaker’s centre of gravity and the trunk.

Literature Cited
Tables

Table 1. Modal Parameters for the tested shaker

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hz)</th>
<th>Damping Ratio (%)</th>
<th>Mode Participation (%)</th>
<th>Type Mode</th>
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<tbody>
<tr>
<td>1</td>
<td>16.38</td>
<td>10.10</td>
<td>67.8</td>
<td>RBM_1</td>
</tr>
<tr>
<td>2</td>
<td>41.41</td>
<td>7.96</td>
<td>10.8</td>
<td>RBM_2</td>
</tr>
<tr>
<td>3</td>
<td>65.97</td>
<td>6.11</td>
<td>12.6</td>
<td>RBM_3</td>
</tr>
<tr>
<td>4</td>
<td>97.03</td>
<td>5.85</td>
<td>8.9</td>
<td>FBM</td>
</tr>
</tbody>
</table>

Table 2. AutoMAC matrix for the established modes

<table>
<thead>
<tr>
<th>Mode</th>
<th>1</th>
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<th>4</th>
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<tbody>
<tr>
<td>1</td>
<td>100</td>
<td>0.2</td>
<td>1.5</td>
<td>0.1</td>
</tr>
<tr>
<td>2</td>
<td>0.2</td>
<td>100</td>
<td>7.2</td>
<td>4.6</td>
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<tr>
<td>3</td>
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<td>4</td>
<td>0.1</td>
<td>4.6</td>
<td>1.1</td>
<td>100</td>
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</table>

Table 3. Operational parameters for the tested shaker

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Transient Time (s)</th>
<th>Global Acceleration RMS (m/s²)</th>
<th>Transference Function</th>
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<tbody>
<tr>
<td>Mean</td>
<td>37.3</td>
<td>1.2</td>
<td>1.0</td>
</tr>
<tr>
<td>Stand Dev</td>
<td>1.1</td>
<td>0.1</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Figures

Figure 1. Tested Self-propelled trunk shaker

Figure 2. Trunk shaker’s geometry
Figure 3. Modal testing of the vibrating head employing a shaking-post

Figure 4. Cartesian Axes to register the measurements in the vibrating head and the tree’s trunk

Figure 5. Mode shapes of the fundamental (rigid and flexion) modes

RBM_1, 16.38 Hz, 10.10 %  
RBM_2, 41.41 Hz, 7.96 %  
RBM_3, 41.41 Hz, 6.11 %  
FBM, 41.41 Hz, 5.85 %
Analyse modale des vibreurs de tronc utilisés dans la récolte mécanique des olives.

Mots clés : Olivier, récolte mécanique, analyse modale, vibreurs de troncs, accélération, fréquence naturelle

Résumé

De nos jours, la technique de récolte mécanique des olives par vibration est la plus répandue et la plus souvent utilisée. Néanmoins, ce procédé mécanique souffre du développement de machines de récolte efficaces. Dans le but d'obtenir un design optimum, il est nécessaire de réaliser une analyse dynamique de ces machines, spécialement pour celles qui utilisent un système vibratoire. Dans cette optique, l'analyse modale est une méthode utile pour identifier le comportement dynamique de la structure.

L'objectif de cette recherche est de déterminer le comportement dynamique des têtes vibratoires utilisées dans le procédé de détachement de l'olive, dans le but de caractériser le design optimum réduisant les dégâts causés sur le tronc de l'olivier.

Les résultats de l'analyse modale ont permis d'établir la fréquence naturelle du vibreur. De plus, les caractéristiques dynamiques du vibreur ont été identifiées en accord avec ses formes modales et ses coefficients de participation modaux. La comparaison de ces résultats et de ceux obtenus par analyse spectrale a montré les problèmes liés au design et à l'utilisation du vibreur du point de vue des dégâts causés au tronc et de la transmission des vibrations.