Vibration Analysis of an Olives Mechanical Harvesting System

Paolo Amirante[1], Pasquale Catalano[2], Ferruccio Giametta[2], Alessandro Leone[3], Gian Luca Montel[3]

[1] University of Bari, Progesa Dept., Via Amendola 165/A, 70100 Bari, Italy

[2] University of Molise, Sava Dept., Via De Sanctis, 86100 Campobasso, Italy

[3] University of Foggia, Prime Dept., Via Napoli 25, 7100 Foggia, Italy

paolo.amirante@agr.uniba.it, catalano@unimol.it, ferruccio.giametta@unimol.it, a.leone@unifg.it, g.montel@unifg.it

ABSTRACT

Fruit removal is the main aim of vibratory fruit-harvesting machines used on fruit-bearing trees. it is well known that fruits suddenly fall, especially within the transient and when frequency span technique is used, periodically accelerating and slowing down the shaker. The aim of this paper is to give some new and not yet investigated results about these machines, in particular clamp-trunk coupling and its effect on vibration transmission. A clamp-trunk coupling model was established for the separation of the different phenomena occurring when using shakers with two independently driven eccentric masses. First of all the acceleration can be broken up into two multiplicative factors representing the first one the main vibration mode, the second one the precession motion of the trajectory of the acceleration vector in XY plane. The second phenomenon highlighted in this paper is typical of clamp-trunk coupling: the phase modulation of the acceleration measured on the trunk.

Keywords: Vibration, olives, transmission, accelerations, shaker

1. INTRODUCTION

Fruit removal is the main aim of vibratory fruit-harvesting machines used on fruit-bearing trees. Trunk shakers are mostly used combined with convenient intercepting system on the soil or around the trunk, typically constituted by nets or by a reverse umbrella.

Many researchers have investigated on shaker design and optimisation both, analysing fruit removal efficiency and tree damage Adrian and Fridley (1965), presented fundamental vibration theory and design criteria for different type of tree shakers. Parameswarakumar and Gupta (1991) showed that, to obtain maximum fruit removal with minimum tree damage, the shaker should be operated in the range of 76–102 mm amplitude and frequencies of 11–13 Hz for 4 s. Horvath and Sıtkei (2001) proposed a tree model analysing different kinds of trunk motion. Based on acceleration measurements in the soil body, a new mass component was included, in addition to the common mass components. More recently (Horvath and Sıtkei - 2005) investigated on energy requirements showing that energy calculations often give lower values than true ones. H.M. Abdel-Fattah (2003), uses a vibration shaker managed by a variable speed electrical motor to reproduce and control vibration level along a single axis and introduce wavelet filtering to better estimate the displacement by integration and main frequencies of the acquired signals. Erdogan *et al*. (2003) studied harvesting of apricots by mean of an inertia type limb shaker. They analysed fruit damage and removal efficiency protecting fruit mechanical properties and harvesting parameters. L.M. Mateev and G.D. Kostadinov (2004) presented a probabilistic model of the fruit removal in which the quantity of the non-removed fruit from the tree and the probability for an insufficient harvest duration decrease exponentially when the vibratory impact duration upon the tree increases, despite the differences in harvesting conditions. Sessiz and Ozcan (2006) carried out harvesting of olive by pneumatic branch shaker and abscission chemical. The maximum harvesting efficiency (96%) was achieved at 24 Hz and 6.25 ml/l of chemical abscission concentration. Lang (2006) describes mathematically the power consumption, generated amplitude and specific power for each tested trunk using a tree structure model which comprise both trunk and main roots. Finally, Sanders (2005) presents an interesting review on mechanical harvesting methods used for orange trees but applicable also to other type of trees.

On these basis it is well known that fruits suddenly fall, especially within the transient and when frequency span technique is used, periodically accelerating and slowing down the shaker. Particularly, olives harvesting is characterised by scalar ripening therefore those drupes not immediately detaching, hardly fall shaking on and on. In some cases better results are obtained stopping shaking, opening the claws, and moving the shaker in another position with respect to the trunk in order to shake it along a different direction. Another solution consists in shaking directly the branches which, for particular geometric shape and physical properties, are not sufficiently affected by the vibration. In order to avoid multiple shaking, multidirectional shakers are used. In particular in a previous paper (Catalano et al., 2006) the precession motion of the main mode was investigated and it resulted too slow requiring a long shaking time to complete the rotation.

The aim of this paper is to give some new and not yet investigated results about these machines, in particular clamp-trunk coupling and its effect on vibration transmission.

2. MATERIAL AND METHODS

Tests were carried out in Monopoli (BA) - Italy - to determine how the oscillating force is transmitted from the shaker to the trunk. The harvester is made by:

- a tractor;
- a hydraulic power pack, divided into two parts: the pump and the tank usually on the rear of the tractor, and the hoses connections with directional valves on the front;
- a telescopic extensible arm fixed on the tractor front, with a rotating chassis on its upper end, that carries beneath it the vibrating head, suspend by three chains;
- the shaker, in which a sinusoidal oscillation is generated by means of a couple of eccentric rotating masses, moved by two hydraulic motors;
- the gripping clamp, that are used to grab and transmit the vibration to the trunk.

In particular the shaker has two slightly different eccentric masses rotating in opposite directions at almost the same speed, developing multidirectional shake.

Accelerations were measured using two piezoelectric accelerometers, the first one located on the shaker, and the second one on the trunk near the clamping point.

The first accelerometer was bolted onto an steel plate 2x8cm fastened onto the trunk by means of two wood screws, and the other placed onto the shaker case by means of a magnet.

Triaxial, piezometric accelerometers were chosen for these tests because of their light weight (11g) and because of their high frequency span (flat response up to 5 kHz). They have to be fed with 4mA DC excitation current, and their sensitivity and full scale are 100mV/g and 50g respectively.

The measurement system was made up by a personal computer with the data acquisition card PCI6052E (National Instruments), the LabVIEW® measurement environment, two triaxial PCB accelerometers Piezotronics and a SCXI 1531 (National Instruments) signal conditioning module equipped with an analogical band-pass filter with passing band 0,2 Hz - 500Hz. The sampling frequency was set to 1000 Hz.

There have been a lot of test sessions on young olive trees (about 10 years old). Each one was carried out varying the frequency of the vibration in the range 10 – 30 Hz so to analyse clamp – trunk coupling in true operative conditions. Main results following are shown.

3. RESULTS AND DISCUSSION

In figure 1 the shaker used for the tests is shown; in particular the cartesian coordinate system adopted to represents the measured accelerations is pointed out.

Figure 1 – Shaking equipment and Cartesian coordinate system used in the following developments

In figures 2a-d X and Y component of the accelerations measured on the shaker and on the trunk are shown during the first test. In particular the whole X component was plotted to highlight amplitude modulation effect and a zoomed version of the Y component was shown to point out frequency (period) variation vs. time.

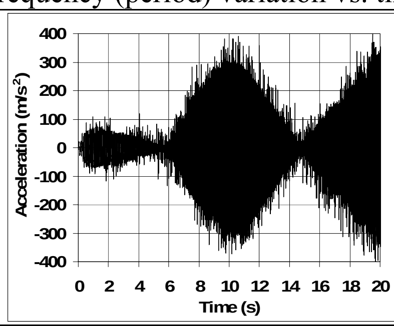
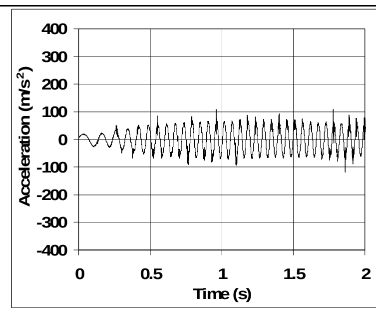
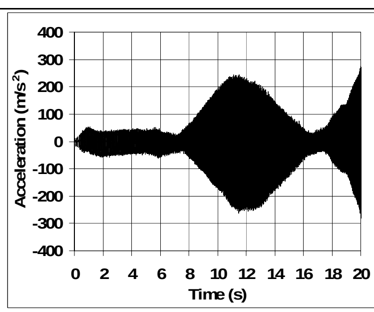
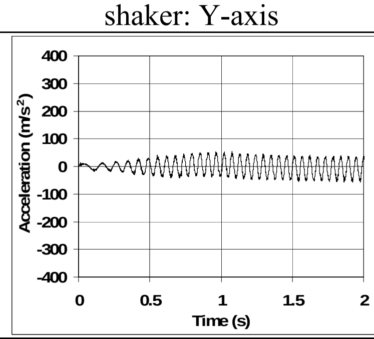
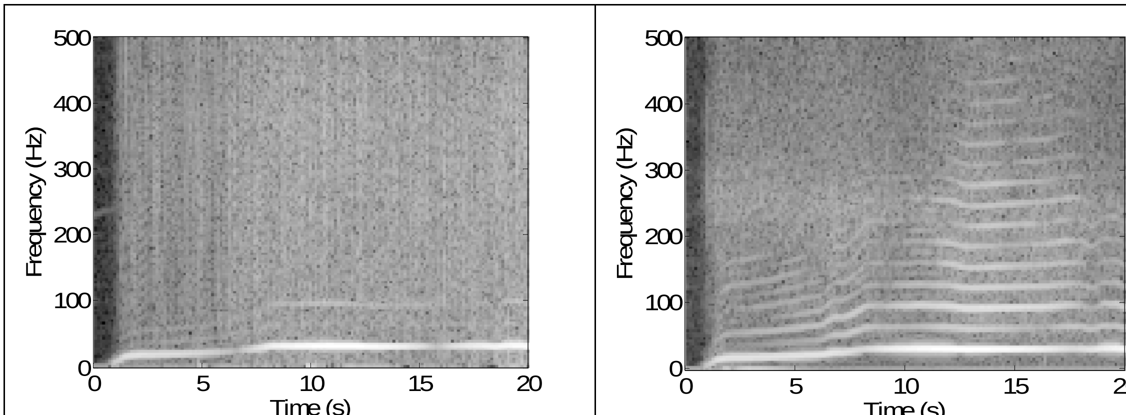
	
a) Acceleration measured on the shaker: X-axis	b) Acceleration measured on the shaker: Y-axis
	
c) Acceleration measured on the trunk: X-axis	d) Acceleration measured on the trunk: Y-axis

Fig. 2 - Acceleration measured on the X and Y axis on the shaker (a-b) and on the trunk (c-d)

From these plots we can point out the amplitude modulation effect due both to its proportionality to rotational speed squared and acceleration vector rotation in xy plane.

Time varying characteristic – and therefore frequency modulation effect - can be easily highlighted using the spectrogram plot of each discrete acceleration signal. A spectrogram is the magnitude of the Short Fast Fourier Transform (SFFT) of a signal. In particular each signal was divided into 256 sub-sequences, an FFT was computed over each resulting segment, and the obtained spectrum was assigned to the middle time of each segment (Figs 3a-b). In these figures white lines represent the time evolution of each vibration mode: the frequency of the main mode of every signal varies from 0 to 20 Hz during the first 7 s and from 20 to 33 Hz in the last period.

However, more interesting is the presence of harmonics of the main mode: this effect is most evident in trunk measures, especially during the last period when the acceleration amplitude modulation increases a lot compared with the initial sequence (7 s). These harmonics are clearly caused by non-linearity behaviour of the vibrating system; particularly, by the sliding between clamps and trunk due to the presence of a gummed strip. The effect of this behaviour on the acceleration measured on the trunk is a phase modulation of the signal pointed out by the harmonics of the main signal. This effect is very limited when observing the acceleration measured on the shaker because of the very little sliding between case and the eccentric mass system.



a) Spectrogram of the acceleration measured on the shaker along X-axis	b) Spectrogram of the acceleration measured on the trunk along X-axis

Fig. 3 – Spectrogram of the acceleration measured on the shaker (a) and on the trunk (b)

In figure 4 the envelope of the acceleration vector on the XY plane, measured on the shaker case (driving force), is shown. It is clear that the acceleration describes on XY plane an ellipse because of the different initial position and opposite rotation of the two masses. Moreover, this ellipse has a slow clockwise precession motion caused by a very little difference between the angular speeds of the two eccentric masses.

This rotation is surely useful to cover all possible directions in XY plane and it is a good substitute to the stop-reposition-repeat process still widely used, but it is, in this particular case, too slow as it needs about 10 s to complete a full rotation. Probably the oil flow in the two hydraulic motors, separately managing the two eccentric masses, should be better controlled to achieve a full rotation within 5-6 s.

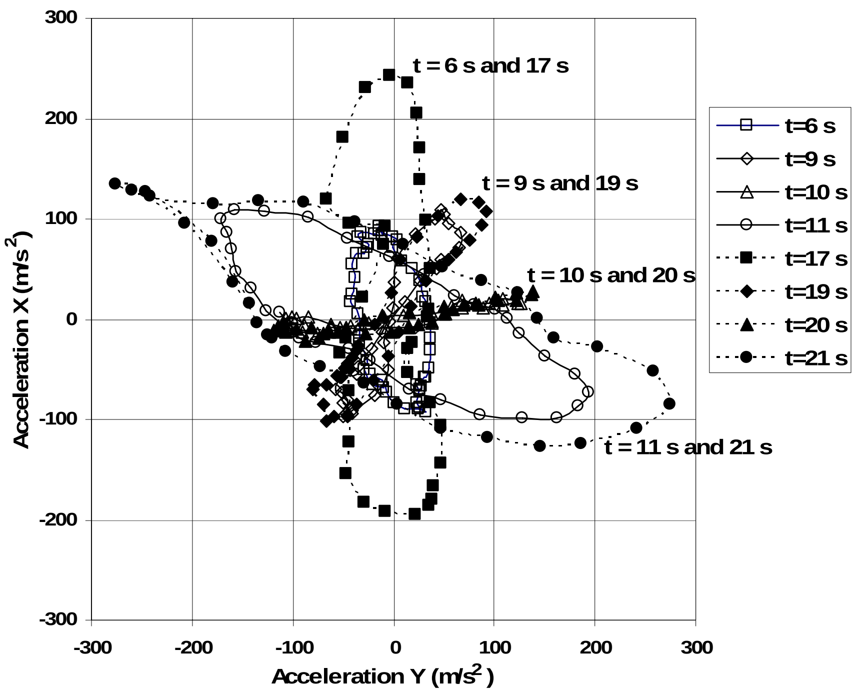


Fig. 4: variation of the envelop of the acceleration vector in XY plane during vibration test

All these considerations require a deeper analysis of clamp-trunk coupling. To do this we represent acceleration vector a_{xy} in the XY plane as a complex number where its real part is the acceleration a_x on the X axis and its imaginary (j) part is the acceleration a_y on Y axis:

$$a_{xy} = a_x + ja_y \tag{1}$$

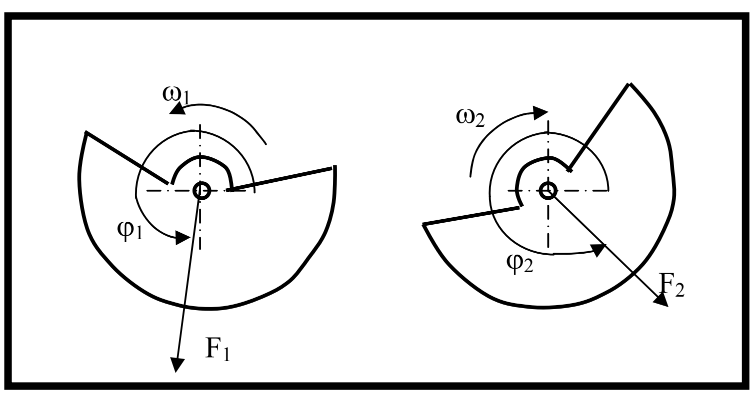


Fig. 5: Scheme of the two eccentric masses: the main mechanical parameters are shown

In figure 5 the scheme of the two eccentric masses - managed by two different hydraulic motors - is shown with the indication of the forces F_1 and F_2 produced by centrifugal acceleration $r_1\omega_1^2$ and $r_2\omega_2^2$ of two equivalent masses m_1 and m_2 placed in the centre of gravity of each eccentric mass:

$$F_k(t) = m_k a_{xy}^{(k)} = m_k r_k \omega_k^2 e^{(-1)^{k-1} j(\omega_k t + \varphi_k)} = I_k \omega_k^2 e^{(-1)^{k-1} j(\omega_k t + \varphi_k)} \qquad k = 1, 2 \tag{2}$$

where I_1=m_1r_1 and I_2=m_2r_2 are the equivalent momentum of inertia of the two masses. Both ω_1 and ω_2 are positive quantities with $0 \le \phi_k \le 2\pi$, k =1, 2.

The same equations should be used to exam the accelerations measured on the trunk.

However, in this case the momentum of inertia cannot be so precisely modelled due to many factors such as the non central position of the shaker with respect to the centre of rotation of the trunk and the quasi-elliptic shape of the trunk cross section (14 x 17 mm). This leads to the time dependence of I_k and, of course of ω_k and ϕ_k.

Therefore in this case it is more correct to adopt a semi-empirical model that uses specific momentum of inertia (per mass unit) i_1 and i_2 instead of absolute ones I_1 and I_2, thus introducing equivalent masses considering the whole clamp-trunk system.

$$a_{xy}^{(k)}(t) = i_k(t)\omega_k^2(t) e^{(-1)^{k-1} j\left[\int_0^t \omega_k(\tau)d\tau + \varphi_k(t)\right]} = F_k(t)/m_k \tag{3}$$

where in equation (3) time dependence of $i_k(t)$ and $\omega_k(t)$ has been pointed out and that of $\varphi_k(t)$ (phase modulation) caused by clamp-trunk sliding as described above in the paper. It is important to highlight that $\omega_k(t)$ are very slowly varying signals (first white line in figure 3) instead of $\varphi_k(t)$ which vary rapidly.

The acceleration measured on the trunk is obtained by summation:

$$a_{xy}(t) = \sum_{k=1}^{2} a_{xy}^{(k)}(t) = \sum_{k=1}^{2} i_k(t)\omega_k^2(t) e^{(-1)^{k-1} j\left[\int_0^t \omega_k(\tau)d\tau + \varphi_k(t)\right]} . \tag{4}$$

where the time varying parameters are influenced by the clamp-trunk coupling.

Equation (4) clearly allows the separation of positive frequencies component ($a_{xy}^{(1)}$) from negative one ($a_{xy}^{(2)}$) using Fast Fourier Transform (FFT). Time dependence of ω_1 and ω_2 can be obtained by:

- computing the time derivative of the phase of each component,
- filtering the resulting signals with a low pass filter (0 - 1 Hz).

Once $\omega_1(t)$ and $\omega_2(t)$ are known, $i_1(t)$ and $i_2(t)$ can be easily computed from the module of each component $i_1(t)\omega_1^2(t)$ and $i_2\omega_2^2(t)$.

Moreover a_{xy} can be broken up into two factors considering the difference between ω_1 and ω_2 so separating useful vibration from the precession motion:

$$a_{xy}(t) = \left\{ \sum_{k=1}^{2} i_k(t)\omega_k^2(t) e^{(-1)^{k-1} j\left[\int_0^t \omega(\tau)d\tau + \varphi(t)\right]} \right\} e^{-j\left[\int_0^t \Delta\omega(\tau)d\tau + \Delta\varphi(\tau)\right]} \tag{5}$$

where $\omega(t) = \dfrac{\omega_2(t) + \omega_1(t)}{2}$ is the mean frequency of the main mode and $\varphi(t) = \dfrac{\varphi_2(t) + \varphi_1(t)}{2}$ its the mean phase deviation at time t; $\Delta\omega(t) = \dfrac{\omega_2(t) - \omega_1(t)}{2}$ is the frequency of the precession motion and $\Delta\varphi(t) = \dfrac{\varphi_2(t) - \varphi_1(t)}{2}$ its phase deviation at time t.

In figures 6,7,8 the trends of $i_1(t)$ and $i_2(t)$, $\omega(t)$ and $\varphi(t)$, $\Delta\omega(t)$ and $\Delta\varphi(t)$ respectively are shown (measured on the trunk):

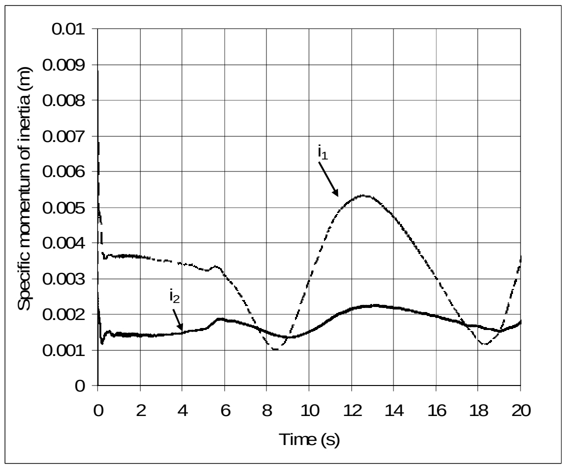


Fig. 6 – Trend of the momentum of inertia of the two equivalent masses.

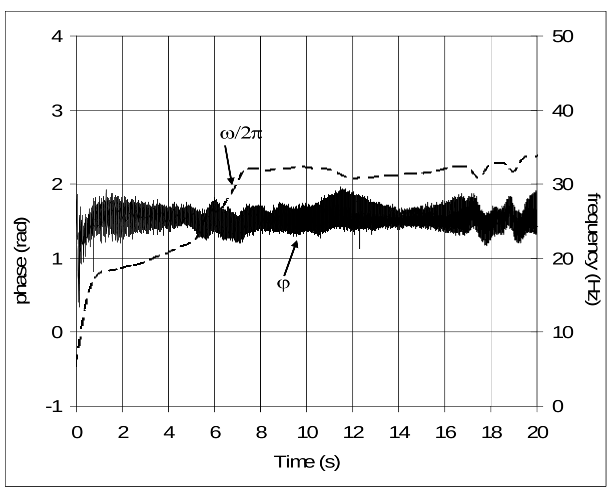


Fig. 7 – Trend of main mode frequency and phase during the test.

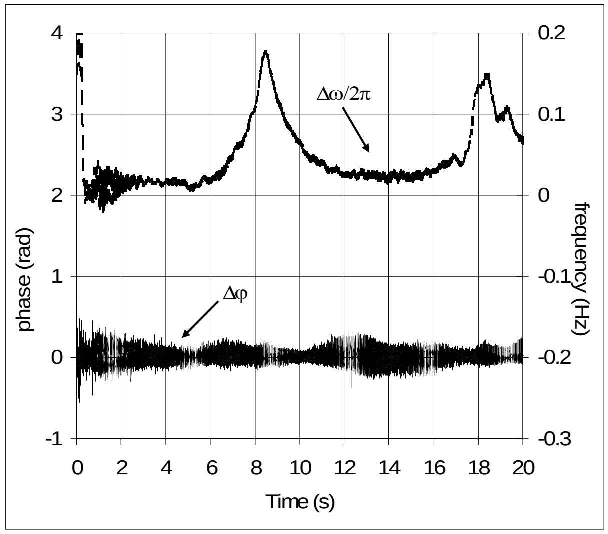


Fig. 8 – Frequency and phase trend of the precession motion due to the little difference in the angular velocity of the two hydraulic motors.

Analysing figure 6 the periodic character of the two equivalent momentum of inertia introduced in eq. 3 can be emphasised. It is caused by the precession motion of the source force and the elliptical shape of the trunk cross section (many variation could only be visible if a perfectly cylindrical trunk is used, due at most to a non perfect balance of the shaker case).

In figure 7 both the frequency span due to the acceleration of the shaker (see also figure 2c) and the phase modulation due to the clamp-trunk sliding can be pointed out. The first effect corresponds to the different speed of the hydraulic motors as imposed by the tractor driver to improve harvesting efficiency varying both detachment force and frequency. The second effect is mainly due to clamp-trunk sliding and introduces some harmonics of the main mode (as already seen in figure 3b).

Finally, as already stated with the graphs shown in figure 4, in figure 8 the precession motion can be also pointed out by the non-zero value of $\Delta\omega$. But, in this case, a deepen analysis could be carried out: the precession motion is not at all stationary. In fact it rapidly varies during the transition occurring at the beginning of the test and when the peak value of trunk acceleration increases.

4. CONCLUSIONS

A clamp-trunk coupling model was established for the separation of the different phenomena occurring when using shakers with two independently driven eccentric masses. First of all the acceleration can be broken up into two multiplicative factors representing

- the first one the main vibration mode,
- the second one the precession motion of the trajectory of the acceleration vector in XY plane.

This behaviour is clearly due to the very little differences between the two independent hydraulic motors driving the eccentric masses. It allows a true multidirectional action avoiding some defects of traditional shakers where a narrow elliptic trajectory is drawn by acceleration so limiting a lot the multidirectional behaviour. On the other hand, according with literature, optimal shaking duration is requested to be less than 10 s. Therefore the precession motion is too long in the shaker used in the experimental tests and a better regulation of the flow rate in the two hydraulic motors is surely needed.

The second phenomenon highlighted in this paper is typical of clamp-trunk coupling: the phase modulation of the acceleration measured on the trunk. This effect is clearly due to the sliding between clamps and trunk and it is unavoidable as it preserves trunks from scraping. Anyway if well controlled, this effect could be useful for fruit removal efficiency. In fact, some harmonics of the main mode are introduced in the range of useful frequencies, behaving as a partial instantaneous frequency span and allowing a better fruit detachment effect.

The authors have contributed to the same extent to the present study

5. ACKNOWLEDGMENTS

Authors thanks Barnaba farm & Verdegiglio firm, both in Monopoli (BA -Italy-), for place field and machinery at test disposal and mr. Franco Marzano (PROGESA dept - technical staff) for his help in experimental tests arrangement.

6. REFERENCES

Abdel-Fattah H.M., Shackel K.A., Slaughter D.C. 2003. Methodology for determining Almond Shaker Displacement and Frequency. *Applied Engineering in agriculture*, 19 (2), 141-144.

Adrian, P. A., & Fridley, R. B. 1965. Dynamics and design criteria of inertia type tree shakers. *Transactions of the ASAE*, 8(1), 12–14.

Catalano P., Giametta F., Leone A., Montel GL. 2006. Vibration analysis of an olive mechanical harvesting system. In *proc*. *World Congress on Agricultural engineering for a better world*, Bonn, Germany, 3-7 Sept. CDROM printed. ISBN 3-18-091958-2.

Erdogan D., Guner M., Dursun E., Gezer I. Mechanical Harvesting of Apricots. *Biosystems Engineering* (85) 19–28

Horvath E., Sitkei G 2001. Energy consumption of selected tree shakers under different operational conditions. *Journal of Agricultural Engineering Research*, 80(2), 191–199.

Horvath E., Sitkei G. 2005. Damping properties of plum trees shaken at their trunks. *Transaction of the ASAE*. (48) 19 – 25.

Lang Z., 2006. Dynamic Modelling Structure of a fruit tree for inertial shaker System design. *Biosystems Engineering*, 93 (1), 35–44.

Mateev L.M., Kostadinov G.D. 2004. Probabilistic model of fruit removal during vibratory Morello harvesting. *Biosystems Engineering* (87) 425–435.

Parameswarakumar M., Gupta C.P. 1991. Design parameters for vibratory mango harvesting system. *Transactions of the ASAE*, 34(1), 14–20.

Polat R., Gezer I., Guner M., Dursun E., Erdogan D., Bilim H. C. 2007. Mechanical harvesting of pistachio nuts. *Journal of Food Engineering* (79) 1131–1135.

Sanders K.F. Orange Harvesting Systems Review 2005. *Biosystems Engineering* (90) 115–125.

Sessiz A., Ozcan M.T. 2006. Olive removal with pneumatic branch shaker and abscission chemical. *Journal of Food Engineering*, (76) 148 – 153.

LIST OF SYMBOLS

a	acceleration (m/s^2)
F	force (N)
I	momentum of inertia (kg m)
$i=I/m$	specific momentum of inertia (m)
$j=\sqrt{-1}$	*imaginary unit*
m	mass (kg)
r	distance of the centre of gravity of each eccentric mass from the rotation axis (m)
φ	phase angle of the centrifugal force vector **F** (rad)
ω	angular velocity (rad/s)
Subscripts:	
x,y	Cartesian axes
1	eccentric mass (counterclockwise rotation)
2	eccentric mass (clockwise rotation)