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Full Research Paper

Vibration Mode for Effective Mechanical Harvesting of Shengy Olive

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Abstract

The main aim of this study was to optimize the design parameters of the fruit shakers for efficient harvesting of Shengy olive. A single-degree-of-freedom spring-mass model was established to determine the natural frequency and damping coefficient of the limb. A tractor-mounted shaker that transmits vibration to limbs and fruits via a reciprocating mechanism was fabricated for field evaluation of the forced vibration modes. A 3×4 factorial experiment with a completely randomized design was conducted to investigate the effects of shaking amplitudes and frequencies on fruit removal. The shaking mode with a frequency of 10 Hz and amplitude of 80 mm transmitted the average power of 92 W to remove 95% of fruits in the field trial. This oscillation characteristic should be used to redesign the fruit shakers to pass human safety standards and efficient harvesting.

Keywords: Fruit harvester, Forced vibration, Limb shaker, Mathematical modeling, Natural frequency

Introduction

Fruit shakers shake the trunk or branches or have contact heads with rods that extend into the canopy (Giametta and Bernardi, 2010; Lavee, 2010; Ravetti and Robb, 2010; Sarri and Vieri, 2010; Tous *et al.*, 2010; Vieri and Sarri, 2010; Sola-Guirado *et al.*, 2014; Moreno *et al.*, 2015; Sola-Guirado *et al.*, 2016; Zhang *et al.*, 2016; Sola-Guirado *et al.*, 2018; Peça *et al.*, 2019). Frequency and amplitude are among the principal operating parameters of the shakers concerning humans, trees, and fruit (El Attar *et al.*, 2004; Blanco-Roldán *et al.*, 2009; Zhou *et al.*, 2013; He *et al.*, 2017a). Due to the damage of high frequency on humans, the manufacturers must declare the acceleration value in the machine instruction manual (Saraçoğlu *et al.*, 2011; Deboli *et al.*, 2014a). However, manufacturers of shakers do not often know which variety of their machines will be used on; therefore, they design shakers with fixed acceleration and frequency values leaving the operators the responsibility of choosing the suitable shaking mode (Costa *et al.*, 2013).

Modal testing is one of the best tools for the dynamic characterization of the fruit shakers (Crooke and Rand, 1969; Tsatsarelis, 1987; Amirante *et al.*, 2007; García *et al.*, 2007; Castro-García *et al.*, 2008; Torregrosa *et al.*, 2014). Recording accelerations of fruits or limbs in response to applied vibration modes by accelerometers (Torregrosa *et al.*, 2009; Du *et al.*, 2012; Bentaher *et al.*, 2013) and modeling of the vibration system by a mass-spring model (Gupta *et al.*, 2015; Du *et al.*, 2020; Chen *et al.*, 2021) were tried for analyzing vibration mode and redesigning of mechanical fruit shakers. Natural frequency, damping ratio, and transmitted power are the important oscillation factors for the redesign of the mechanical fruit harvesters.

The main objective of this study is to identify an effective oscillating mode of the Shengy olive limbs useful for the redesign of mechanical harvesters. Mathematical modeling of the single-degree-of-freedom system provides the vibration mode for olive shakers concerning fruit removal efficiency and human safety standard levels.

Materials and Methods

Mathematical modeling

Figure 1 shows a model of the limb which consists of a mass m, a spring with constant k, and a damper with a coefficient of viscous damping c.

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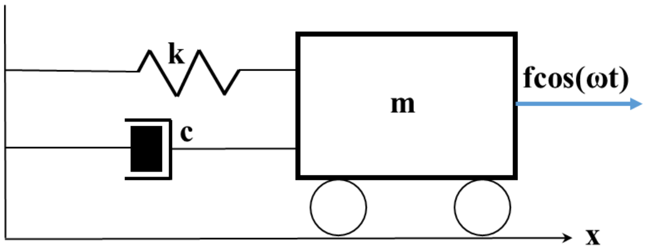

Fig.1. Schematic diagram of the olive fruit limb model represented by a spring (k), a damper (c), and a mass (m) with a driving force (f) acting in direction of (x).

A limb consists of the limb, fruits, and leaves. The sinusoidal force applied on the spring-mass system of the limb by the shaker produces acceleration which depends on the spring rate and damper constant. Newton's Second Law for representing the limb behavior in response to an external harmonic force gives the equation of motion.

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = f\,cos(\omega t) \quad (1)$$

$$x(t) = (\frac{S}{2})\,cos(\omega t - \alpha) \quad (2)$$

$$\dot{x}(t) = (-\frac{S}{2})\,\omega sin(\omega t - \alpha) \quad (3)$$

$$\ddot{x}(t) = (-\frac{S}{2})\omega^2\,cos(\omega t - \alpha) \quad (4)$$

where m, f, c, k, ω, S, and α are the mass (kg), the applied force (N), the damping coefficient (kg s^{-1}), the spring rate (N m^{-1}), the circular frequency of the system (Rad s^{-1}), the amplitude (m), and the phase angle (Rad), respectively. The displacement of mass M at time t is x(t).

Replacing of (2) through (4) in (1) yields.

$$(\frac{S}{2})[(-m\omega^2 + k)\,cos(\omega t - \alpha) - c\omega\,sin(\omega t - \alpha)] = \underbrace{f\,cos\,\omega\,t}_{F(t)} \quad (5)$$

and by setting $t = 0$, one could re-write (5) as:

$$(\frac{S}{2})[(-m\omega^2 + k)\,cos(\alpha) + c\omega\,sin(\alpha)] = \boldsymbol{F} \quad (6)$$

The amplitude of the force vector **F** in (6) defines as:

$$(\frac{S}{2})[(-m\omega^2 + k)^2 + c^2\omega^2]^{\frac{1}{2}} = |\boldsymbol{F}| \quad (7)$$

Using the notation of **F** in (5), the power P can be calculated by:

$$P(t) = \underbrace{\boldsymbol{F}(t)}_{|\boldsymbol{F}|\,cos\,\omega t} \times \boldsymbol{V}(t) \quad (8)$$

Replacing (7) in (8) using the definition of (3) yields.

$$\left(-\frac{S^2}{4}\right)\omega[(k - m\omega^2)^2 + c^2\omega^2]^{\frac{1}{2}}\,cos\,\omega\,t\,sin(\omega t - \alpha) \quad (9)$$

By setting the derivative of (9), in respect to t, to zero, the maximum required power could be represented by:

$$\frac{dP}{dt} = \left(-\frac{S^2}{4}\right)\omega[(k - m\omega^2)^2 + c^2\omega^2]^{\frac{1}{2}}[-sin\,\omega\,t\,sin(\omega t - \alpha) + cos\,\omega\,t\,cos(\omega t - \alpha)] = 0 \quad (10)$$

Which implies:

$$cos\,\omega\,t\,cos(\omega t - \alpha) - sin\,\omega\,t\,sin(\omega t - \alpha) = 0 \rightarrow t = \frac{\frac{\pi}{4} + \frac{\alpha}{2}}{\omega} \quad (11)$$

By replacing (11) in (9), one could write the maximum power in the final form of:

$$P_{max} = \left(\frac{S^2}{8}\right)\omega[(k - m\omega^2)^2 + c^2\omega^2]^{\frac{1}{2}}\,(1 - sin\,\alpha) \quad (12)$$

Furthermore, the average power can be calculated as:

$$P_{av} = \sum\left(\frac{p.\Delta t}{T_f}\right) = \frac{1}{T_f}\int_0^T p.\,dt \quad (13)$$

and by replacing (9) in (13), one has that:

$$P_{av} = \left(\frac{S^2}{4}\right)\omega[(k - m\omega^2)^2 + c^2\omega^2]^{\frac{1}{2}}\,sin\,\alpha \quad (14)$$

In which the phase angle (α) can be calculated by (Dimarogonas, 1976; Blevins, 2015):

$$\alpha = \tan^{-1} \frac{2\zeta\left(\frac{\omega}{\omega_n}\right)}{1-\zeta\left(\frac{\omega}{\omega_n}\right)^2} \quad (15)$$

Where,

$$\omega_n = \left(\frac{k}{m} - \frac{c^2}{4m^2}\right)^{\frac{1}{2}} \quad (16)$$

and

$$\zeta = \frac{c}{2m\omega_n} \quad (17)$$

By replacing (16) in (17), one could define the natural frequency, ω_n, as:

$$\omega_n^2 = \frac{k}{m}\frac{1}{(1+\zeta^2)} \quad (18)$$

In (15) ω, and ζ are the circular frequency of the system (Rad s^{-1}), and the damping ratio of the limb, respectively.

Experimental area and layout

A limb shaker was designed and fabricated to oscillate the limbs with different vibration modes (Fig.2a). The tractor-mounted fruit harvester received its power from the power take-off (PTO) shaft and transmitted it to the tree limb through a boom and a special clamping device (Fig.2b). A V-belt drive system changed the initial 540 revolutions per minute (rpm), supplied by the standard tractor PTO shaft, to 200-1200 rpm and corresponding frequency ranging from 3.2 to 20 Hz. Five levels of shaking amplitude (4, 6, 8, 10, 12 cm) were provided by a slider-crank mechanism, where rotating motion was converted to a reciprocating motion. The clamping device is a quick fastening type equipped with a soft pad as an intermediate media for the protection of the bark. The machine's overall dimensions are 140×80×100 cm. The machine seats on the ground utilizing a hydraulic system of the tractor.

a

b

Fig.2. a) Fabricated limb shaker during evaluation in the orchard, b) clamping device with soft pad attached to limb for vibration cushioning.

The experiments were performed in the Bash Garden in Shiraz (29° N latitude and 52° E longitude), Iran. A 3×4 factorial experiment with a completely randomized design with three replications was conducted to investigate the effect of shaking amplitude and frequency on fruit detachment of green (unripe) Shengy olives. Three levels of oscillating frequency (10, 14, and 20 Hz) and four levels of shaking amplitude (40, 60, 80, and 100 mm) were investigated. According to Rezaei *et al.* (2016), the duration of vibrations for trials was fixed at 5 s. The circular frequency was measured by a light tachometer on the slider-crank mechanism.

Before fruit harvesting, a sheet was spread under the tree. The mechanically removed fruits and fruits remaining on the branch (picked manually) were weighed separately to determine harvesting efficiency. The fruit removal percentage is:

$$e = \frac{w_a}{w_b + w_a} 100 \quad (19)$$

Where e is the fruit harvesting percentage (%), w_a is the weight of fruits harvested from each branch, w_b is the weight of fruits remained on each branch.

The damages to the tree were not considered here. The method of determining damages that comprised of the leaves which were detached from the tree and fallen on the ground, the bark injured by the clamping device, and broken small stems with some leaves presented by (Chen *et al.*, 2012; Gambella *et al.*, 2013; Kargarpour *et al.*, 2018; Memari *et al.*, 2019).

Results and Discussions

The effects of shaking frequency and amplitude were both significant on fruit detachment, while no interacting effect was observed (Table 1). Figure 3 shows the comparison between mean values of detached fruits (%) from limb for different amplitudes and frequencies at 5% significance level. The oscillation mode with a shaking frequency of 10 Hz and amplitude of 80 mm, which caused 95% fruit detachment, passes the standard level of daily vibration dose of 11.4 Hz recommended by Aiello *et al.* (2019) where the damaging effect on the hand is reached. Oscillation of olive trees with a frequency of 24 Hz and amplitude of 60 mm (Sessiz and Özcan, 2006), 16 Hz and 100 mm (Rezai *et al.*, 2015), 23 Hz and 100 mm (Babanatsas *et al.*, 2018), 22-27 Hz (Leone *et al.*, 2015), 35 Hz and 25 mm, and 25 Hz and 25 mm (Alzoheiry *et al.*, 2020), 41 Hz and 25 mm, and 17 Hz and 100 mm (Ferguson *et al.*, 2010) have significant differences with the recommended frequency in this research. The low ratio of the weight of the fabricated machine (120 kg) to the limb weight (8 kg) couple them as an oscillating system. However, in tree shakers *i.e.*, (Pellenc, 2019; Solano, 2020) the large mass of the machine avoids coupling the machine with the tree.

Table 1- Analysis of variance of the percentage of detached fruits for different configurations of amplitude and frequency

Source of variation	Degrees of freedom	Sum of squares	Mean square	FS	P
Replication	2	0.172	0.08	0.0044	
Amplitude	3	9459.80	3153.26	162.71	0.01
Frequency	2	480.62	240.31	12.35	0.01
Amplitude × frequency	6	245.41	40.90	2.10	ns
Error	22	427.75	19.44		

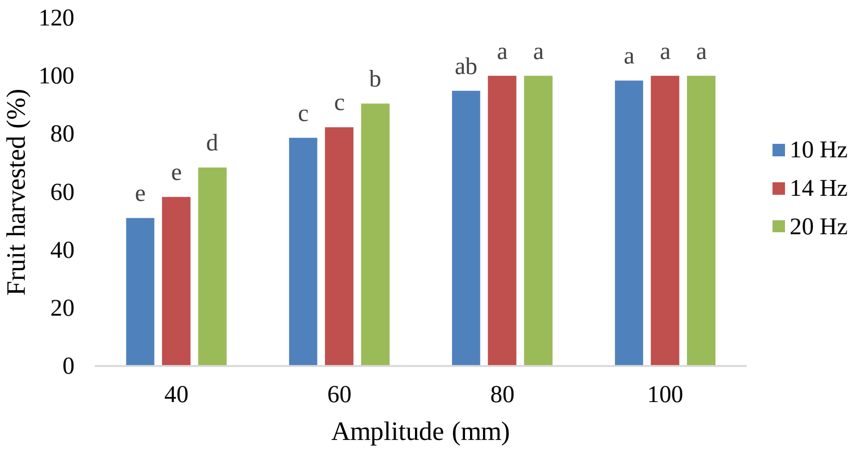

Fig.3. Percentage of detached fruits for different configurations of amplitude and frequency. The same letters do not have a significant difference with a probability of 5%.

By replacing the mean value of 13675N m^{-2} of elasticity and 8 kg for the mass of a limb (Golpira, 1998), the damping ratio of 0.1 for olive limbs (Adrian *et al.*, 1965; Hoshyarmanesh *et al.*, 2017) in (18) the limb's natural frequency (Hz).

$$\omega_n^2 = \frac{k}{m}(1+\delta^2) = \frac{13675}{8}(1+0.1^2) = 41.31 \quad (20)$$

According to (17), c (kg s^{-1}) is

$$c = 2 \times 8 \times 41.31 \times 0.1 = 66.15 \quad (21)$$

By replacing the frequency of 10 Hz in (15) the phase angle is:

$$\alpha = \tan^{-1}\frac{2\times 0.1\left(\frac{10}{41.31}\right)}{1-0.1\left(\frac{10}{41.31}\right)^2} = 0.05 \quad (22)$$

The same reasoning to introduce the concept of electromechanical modes can be extended to interpret (20). Any torque imbalance may cause several electro-mechanical modes with different frequencies (Golpîra *et al.*, 2021). The exciting vibration of 10 Hz to the limb may produce a natural frequency of 41.31 Hz in olive fruits. Natural frequencies of 20.2 and 37.7 Hz (García *et al.*, 2007), and 33.9, 31.9, and 28.0 Hz were calculated for olive limbs (Alzoheiry *et al.*, 2020).

By substituting the amplitude of 0.08 (m) in (12) the maximum power is

$$P_{max} = -\left(\frac{0.08^2}{8}\right)10 \times 2 \times 3.14[(13675 - 8 \times (10 \times 2 \times 3.14)^2)^2 + 66.15 \times 2 \times 3.14 \times 10]^{\frac{1}{2}}(1 - sin0.05) = 1756 \quad (23)$$

By replacing the values of parameters in (14) the average power is

$$P_{av} = \left(\frac{0.08^2}{4}\right)10[(13675 - 8 \times 10^2)^2 + 66.15 \times 2 \times 3.14 \times 10]^{\frac{1}{2}}(sin0.05) = 92 \quad (24)$$

The lower amount of energy regards to the lower shaking frequency and amplitude transmitted to the system could result in the higher fruit collection (He *et al.*, 2017b), lower tree damages (Gupta *et al.*, 2016), and lower human damages (Çakmak *et al.*, 2011; Deboli *et al.*, 2014b).

Conclusions

This paper merged field experimental data of the olive limb shaking mode with the dynamic model of the spring-mass system to redesign the fruit harvesters. Mathematical modeling of the olive limb identified the natural frequency of 41.31 Hz with a damping coefficient of 66 kg s^{-1}. The single-degree-of-freedom model showed the maximum and average power of 1756 and 92 W for limb oscillation of Shengy olives. The free exposure to harmful hand-arm vibration mode with a working frequency of 10 Hz and an amplitude of 80 mm detached 95% of olives in the field trial. Modeling the coupling system of the limb and the machine with two unequal masses and springs can further improve the results.

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Nomenclature

c: Damping coefficient (kg s^{-1})
f: Force (N)
F: Force vector
k: Spring constant (N m^{-1})
m: Mass (kg)
$P(t)$: Power (W)
S: Amplitude (m)
$V(t)$: Velocity (m s^{-1})
t: Time of oscillation (s)
x: Displacement of excitation (m)
$\dot{x}$: First derivative of the x (m s^{-1})
$\ddot{x}$: Second derivative of the x (m s^{-2})
ω: Circular frequency (Rad s^{-1})
ω_n: Natural frequency (Hz)
ζ: Damping ratio
α: Phase angle (Rad)

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مقاله پژوهشی

مد ارتعاشی موثر برای برداشت مکانیکی زیتون رقم شنگی

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چکیده

هدف از انجام این تحقیق بهینه‌سازی عوامل طراحی ماشین‌های برداشت میوه برای برداشت موثر زیتون رقم شنگی بود. مدل جرم-فنر با یک درجه آزادی برای تعیین بسامد طبیعی و ضریب میرایی شاخه زیتون استفاده شد. یک شاخه‌تکان سوار که در آن ارتعاش از راه بازوی میل‌لنگی به شاخه و میوه انتقال می‌یافت برای ارزیابی مدهای ارتعاشی ساخته شد. به‌منظور تعیین تاثیر دامنه و بسامد ارتعاش بر میزان جداسازی میوه از شاخه یک آزمایش فاکتوریل (۳×۴) در قالب یک طرح کاملاً تصادفی اجرا شد. مد ارتعاشی با بسامد ۱۰ هرتز و دامنه ۸۰ میلی‌متر توان متوسط ۹۲ وات را برای برداشت ۹۵ درصد از میوه‌ها انتقال داد. برای برداشت موثر و منطبق بر استانداردهای سلامتی می‌توان از این مد ارتعاشی برای بازطراحی درخت تکان‌ها استفاده کرد.

واژه‌های کلیدی: ارتعاش، بسامد طبیعی، شاخه تکان، ماشین‌های برداشت میوه، مدل ریاضی

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