# DESIGN AND DEVELOPMENT OF A BUNCH SHAKER FOR VIBRATORY DATE DETACHMENT*.

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## ABSTRACT

The purpose of this research was to design and develop an experimental shaker for vibratory date detachment. The prototype machine was constructed and tested on Shahani dates fruit bunch. The bunch shaker was powered by a continuous variable speed motor. The rotational speed of the power unit could be continuously varied from 0 to 1400 rpm by means of a volume control knob on a digital board. The rotating output power of the motor was transmitted to a small flywheel through a V-belt drive system, where it was converted to a reciprocating motion by a slider-crank mechanism. The eccentricity of the crank mechanism was adjustable to provide stroke lengths of 20, 40, 80, 100 and 120 mm. The slider motion was transmitted to the shaker clamp by a 500 mm long boom made of 20mm steel tubing. The boom could reciprocate vertically and horizontally in a brass guide. By this shaker date fruit bunch can be oscillated in vertical, horizontal and hanging modes. At the vertical shaking mode, the resulting motion of slider-crank mechanism was transmitted to the fruit bunch clamped to the shaker frame, through a boom and a clamping device. The required average power for bunch shaking was estimated to be about 1kW by assuming constant-displacement limb shaker model. Result showed the bunch shaker was capable of removing ripe fruit from the bunch at 5-7 second without imparting rubbing and bruising damage to the fruits.

**Additional Key Words:** Date harvesting, Selective harvesting, Bunch shaker

## INTRODUCTION

Mechanization of date harvesting has been the interest of many growers and investigators in the past forty years. Variable maturity of fruit on a single palm is often a limiting factor in harvest mechanization. As all of the fruits in a bunch do not ripen at the same time, it is necessary to make several picking during the season. Selective hand-picking of the individual ripe fruits from each bunch is the most expensive cultural operation. The number of hand-picking gradually decreased due to the shortage of workers for harvesting and increasing their wages. Development of the mechanical harvesting began in 1961-62. Trials of several methods indicated that rapid removal of ripe fruit could be accomplished by shaking (6,7). Harvesting system has been developed using a variety of high-lifting personal platforms and bunch shakers. The use of these systems made the work easier and more attractive to the workers and resulted in reduction of labor requirement by 70% (3,5 ). Perkins and Brown in 1967 reported that the vertically shaking mode had the best effect on ripe fruit detachment. According to this result rigidly mounted and hand carried shakers were built. The hand carried shaker was hydraulically powered, weighed 12.5lb, and imparted 1.5in stroke to the bunch at about 1400cpm. It was developed for selective harvesting of ripe fruits from the uncut bunch. The rigidly mounted shaker was designed to impart a 3 1/4 in stroke at about 700 cpm to the bunch. Sarig *et al.* (1971) designed a portable tractor mounted date bunch shaker. It was hydraulically powered, and delivered a 9.5cm stroke vibration at 67 Hz (8). Later Sarig (1989) used an inertia type shaker for shaking the palm. They reported that shaking the palm at the lower quadrant of its height (about 2.5-3m) yielded the optimal rate of fruit removal (more than 90%). Furthermore they found that more fruits removal from the palms with upright fronds was more than those with hanging fronds (9).

**Description of the Experimental Shaker**

In order to study the vibratory mechanism of date harvesting system, it was necessary to design and develop a shaker capable of producing wide ranges of precise frequencies and strokes. It was realized that an experimental shaker powered by a continuous variable speed motor would be best suited for the purpose of this study. On this basis, a shaker was designed and fabricated (Fig.1). The machine consisted of the following main parts: main frame, power unit, power transmission, shaking mechanism, clamping devices and collecting unit (1).

**Main Frame**

The main frame was constructed by welding rectangular section steel tubes and angle iron beams together as shown in Fig.1 to support the other five parts.

**Power Unit**

The shaker was powered by a 2.2 kw continuous variable speed electric motor. The rotational speed of the electrical motor could be

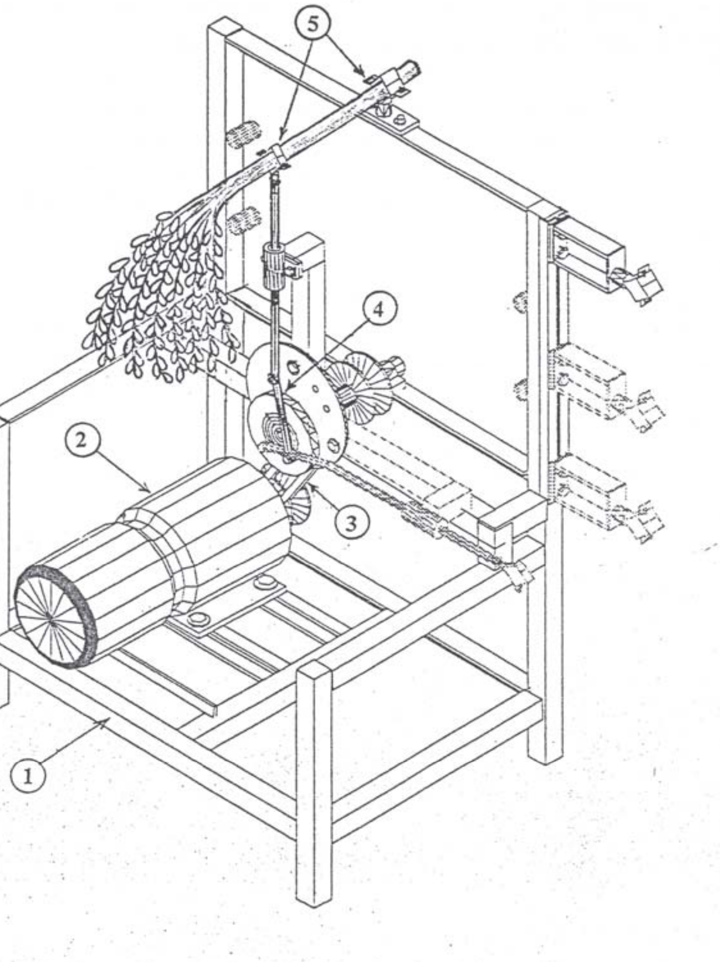


**Fig.1 -Schematic diagram of the experimental shaker**

1. Main frame 2. Power unit 3. Power transmission
4. shaking mechanism 5. Clamping device

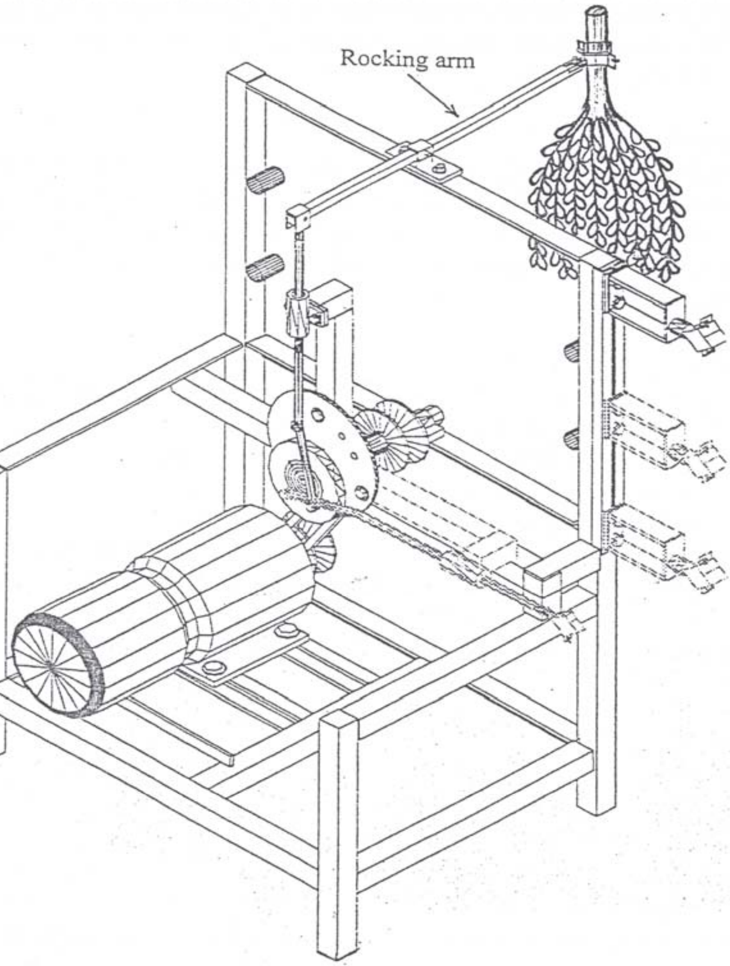


**Fig.2 -Schematic diagram of the experimental shaker equipped with a rocking arm to provide hanging mode vibration.**

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continuously varied from 0 to 1400rpm by means of an electric volume control on a digital board. By employing this digital board it was possible to gain any specific rpm almost immediately and then keeping the shaking frequency precisely constant. Shaking frequency was displayed constantly on the digital board monitor.

**Power Transmission**

The output power of the electrical motor was transmitted to a small flywheel through a V-belt drive system. Belt tension could be adjusted by changing the position of the driver pulley and no idler was needed. Flexible machine elements such as belts are elastic, and play an important role in absorbing shock loads.

**Shaking Mechanism**

A slider-crank mechanism was employed to generate shaking motion. The eccentricity of the crank-mechanism was designed adjustable to provide stroke lengths of 20, 40, 60, 80, 100 and 120mm. The driving shaft of the slider-crank mechanism was supported by two self-aligned ball bearings. The slider could reciprocate either vertically or horizontally in a brass guide. The slider motion was transmitted to the shaker clamp by a 500mm long boom made of 20 mm diameter steel tube. By using this shaker, the fruit bunch could be oscillated in any of the three possible shaking modes (vertical, horizontal and hanging) at any desired shaking amplitude and frequency.

**Clamping Devices**

Two clamping devices were required at each shaking mode. A fixed clamp for holding the bunch stalk fixed to the frame and the other at the end of the shaker boom for applying vibrational motion to the bunch stalk. The clamp holding the bunch stalk on the frame could be fixed at any position horizontally or vertically on frame's upper or right side members. With this arrangement, it was possible to apply shaking motion at any desired angle. These clamps consisted of a movable jaw which could be opened and closed easily by a fly nut. To clamp the bunch, the movable jaw could be opened to insert the bunch stalk in the clamping device. Because of the periodic impact of the shaker upon the bunch, the inner surfaces of the clamp jaws were covered with a rubber padding. This padding cushions vibrational impact forces and prevents imparting excessive injuries to the bunch stalk.

## Fruit Collecting Unit

The collecting unit consisted of a special nylon fabric curtain extended under the fruit bunch. This curtain was held by two 1.8m long horizontal bars pivotally connected to the shaker frame (Fig.3). The falling fruits after detachment were kept off the shaker parts by the collecting curtain, without being damaged.

## Mathematical Description of Bunch Motion

To describe the motion of bunch mathematically, which has a very complex and non uniform biological structure, many assumptions and generalizations were made. A structure such as a bunch has an infinite number of degrees of freedom. In many cases, however, such a member can be considered to be dynamically equivalent to one with finite degrees of freedom. The bunch system was analyzed as a single degree of freedom system undergoing base-excited sinusoidal motion. The bunch was considered as a stiffness member with internal damping.

In order to properly design this kind of machine, some estimation of its power consumption is needed.

## Apparent stiffness (K)

One method for describing the dynamic characteristics of a limb is to determine the ratio of the shaking force to the displacement at the point of shaker attachment. Fruit bunch was assumed as a cantilever beam with its strands concentrated at the free end (Fig.4).

L P

$\delta$

Fig. 4. Fruit bunch simplified model

The apparent stiffness can be calculated by using the following equation:

$$P = \delta K \qquad [1]$$

In order to determine K values, fruit bunches were randomly selected and their weight and dimensions were measured. A 50N spring scale, with 1N

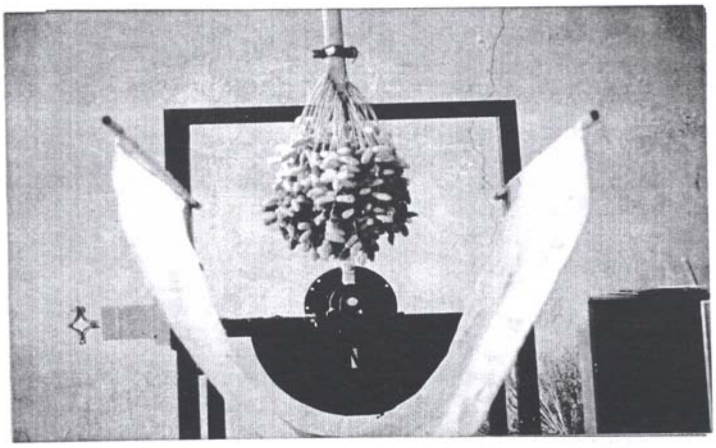

Fig. 3. Fruit Collecting Unit



divisions was used to apply a lateral force to the bunch. Experiment results showed that the average K values in two loading modes (horizontal and vertical) were 4.100 and 2.400 kN/m respectively, with an overall average of K=3.250 kN/m

**Damping Ratio**

The method of free vibration decay was used to measure the internal damping ratio. It can be expressed and measured by the logarithm of two successive oscillation amplitudes (4).

damping ratio $$\varepsilon = \frac{1}{2\pi(n-1)} \ln \frac{X_1}{X_n} \qquad [2]$$

At first, fresh fruit bunches were selected and then each bunch was clamped to a massive steel support to eliminate any energy dissipation at the support. Then the bunch was manually displaced and released. This caused the bunch to vibrate at its natural frequency. By recording the change of oscillation amplitude, and averaging the values of damping ratio it was found $\varepsilon$ =0.16

**Natural Frequency**

An important stage in this study was to determine bunch natural frequency which could be found by the following formula (4):

$$\omega^2{}_n = \frac{k}{m}(1+\varepsilon^2) = \frac{3250}{10}\left(1+(0.16)^2\right) = 333.32 \quad ,$$

$$\omega_n = 18.25\text{Hz}$$

**Phase Angle($\alpha$ )**

$$\alpha = \tan^{-1}\frac{2\varepsilon(\frac{\omega}{\omega_n})}{(1-\varepsilon(\frac{\omega}{\omega_n})^2} \ ,$$

$\omega$ =12.5Hz, maximum applied frequency

$$\alpha = \tan^{-1}\frac{2(0.16)(\frac{12.5}{18.25})}{1-(0.16)(\frac{12.5}{18.25})^2} = 0.23\text{ rad}$$

$\alpha$ = 0.23 rad

F COS $\omega t$

According to Fig.5 the bunch was considered as a stiffness member with internal damping and we have:

F=ma or (spring +damping +applied force) = inertia force

$$-kx - c\frac{dx}{dt} - m\frac{d^2r}{d^2t}\cos\omega t = M\frac{d^2x}{dt^2}$$

$$M\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = mr\omega^2\cos\omega t \qquad [3]$$

The solution of equation is of the form

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

$$\frac{dx}{dt} = -\frac{S}{2}\omega Sin(\omega t - \alpha)$$

$$\frac{d^2x}{dt^2} = -\frac{S}{2}\omega^2 Cos(\omega t - \alpha)$$

By substituing these values into eq.[3] it can be shown that:

$$S \cong \frac{2mr}{M} \qquad [4]$$

Also it can be shown that the internal force is:

$$F = m\omega^2\left[\frac{S}{2}\cos(\omega t - \alpha) + r\cos\omega t\right] \qquad [5]$$

By differentiation to determine the maximum, the design force is found to be:

$$F_d = mr\omega^2\left[(S/2r)^2 + 1 + \frac{S}{r}\cos\alpha\right]^{1/2} \qquad [6]$$

The power required to vibrate the system can be expressed as force times velocity and then we have:

$$P_{in} = \left[m r \omega^2 \cos \omega t\right]\left[-\frac{S}{2}\omega \sin(\omega t - \alpha)\right] \quad [7]$$

$$P_{ave} = \frac{\Sigma(p\delta t)}{T} = \frac{1}{T}\int_0^T P dt = \frac{1}{T}\int_0^T \left[m r \omega^2 \cos\omega t\right]\left[-\frac{S}{2}\omega \sin(\omega t - \alpha)\right] dt$$

$$= -\frac{S^2}{4}\omega\left[\left(k - m\omega^2\right)^2 + c^2\omega^2\right]^{1/2} \sin\alpha \quad [8]$$

$$P_{av} = \frac{(0.06)^2}{4}(2)(3.14)(12.5)\{\ (3250 - 10((2)(3.14)(12.5))^2 +$$
$$(52.8)^2((2)(3.14)(12.5))^2\ \}^{1/2} \sin(13) \approx 1kw$$

The maximum torque is found by dividing maximum power by angular velocity. The maximum power requirement is found by differentiation of equation [7] :

$$P_{max} = \frac{m r \omega^3 S}{4}(\pm 1 - \sin\alpha) \quad [9]$$

$$= \frac{10(0.06)^2((2)(3.14)(12.5)^3(1 - \sin(13))}{4} = 3.37kw$$

$$T_{max} = \frac{m r \omega^2 S}{4}(\pm 1 - \sin\alpha) \quad [10]$$

$$= \frac{10(0.06)^2((2)(3.14)(12.5))^2(1 - \sin(13))}{4} = 43.35N.m$$

**Design of V-belt Drive:**

Flexible machine elements, such as belts are elastic, and play an important role in absorbing shock loads and in damping out the effects of vibrating forces. These advantages are important as far as life of the driving machine is concerned. Furthermore the cost of the driving unit is an important factor in the selection of power transmission system. The specifications of the V- belt drive required for power transmission from the electric motor to the slider-crank mechanism was selected according to the design manual published by the Gates rubber company (2).

**Design Power:**

Design power = (service factor) (calculated power)
= 1.5 (3.37) ≅ 5kW

**Belt Type and Cross Section**

According to the table of design power and maximum drive speed (1400 cpm) an agricultural HB section V-belt was selected.

**Belt Length**

The length of v-belt is obtained by the formula:

$L_p = 2C + 1.57(D+d) + (D-d)^2/4C$

where:

$L_p =$ Pitch or effective length of belt

C = 55 = center distance (cm)

D =14 = pitch diameter of large sheave (cm)

d = 14 = pitch diameter of small sheave (cm)

$L_p = 2(50) + 1.57(2)(14) = 143.96$ cm = 56.6 in

**Angle of Contact of Belt and Pulley**

$$\theta = \pi + 2\,Sin^{-1}\left(\frac{D - d}{2C}\right)$$

$\theta = \pi$

**Belt speed**

$$V = \frac{\pi d n}{60}$$

where:

V= belt speed (m/s)

n = 750 (rpm)

d = 0.14 = diameter of pulley(m)

V=3.14(0.14)(750)/60 =5.49 m/s

**Effective Pull**

$$R_{a\theta} = \frac{T_1}{T_2} = e^{k\theta}$$

Where:

k = Coefficient of friction = 0.512 (10),

$\theta$ = Angle of contact.

$T_1$ = Tight side tension (N)

$T_2$ = Slack- side tension (N)

$$\frac{T_1}{T_2} = e^{0.512(3.14)} = 4.995 \approx 5$$

$$(T_1 - T_2) = 1000P / V$$

Where:

k = Coefficient of friction = 0.512 (9),
$\theta$ = Angle of contact.
$T_1$ = Tight side tension (N)
$T_2$ = Slack- side tension (N)

$$\frac{T_1}{T_2} = e^{0.512(3.14)} = 4.995 \approx 5$$

$$(T_1 - T_2) = 1000P / V$$

Where:

$P$ = design power (kW)
$(T_1 - T_2)$ = 1000 (5)/5.49 =920 N
$4T_2$ = 920 N    $T_2$ = 230 N, $T_1$ = 1150 N

**Fatigue Rate**

According to the Gate V-belt design manual (1) the corresponding fatigue rate is 5.5.
Belt life = length of belt(in)100/fatigue rate
= 56.6(100)/5.5 = 1030 hr
The expected belt life for tree shaker is between 400-1000 hour(1) , so the calculated value is acceptable.

**Determination of the Shaft Loads and Moments**

Assuming F, as the maximum lateral force exerted by fruit bunch and shaker boom on the shaft.

$$T_{max} = F \times r, \qquad F = \frac{T_{max}}{r} = \frac{43.35}{0.06} = 717N$$

Flywheel mass = 10kg

$$W = mg = 10 \times 9.81 = 98.1N$$
$$F_1 = F + W = 717 + 98.1 = 815.1N$$
$$F_2 = T_1 + T_2 = 1380\ N$$

Free body diagram of shaft is:

A B C D

F4 F2 F3 F1

F1 and F2 are the loads exerted by boom and pulley and F3 and F4 are the loads on the shaft which are supported by the ball bearings. These bearing reaction are calculated by solution of the equilibrium equations as:
F3= 1776.67 N
F4= 418.33 N
The maximum bending moment on the shaft occurs at bearing C and it is easily calculated as:

Mmax = 815.1 $\times$0.1= 81.51 N-m
shaft diameter is obtained by the formula (9):

$$d^3 = (\frac{16}{\pi\tau})\left[(C_m \times M)^2 + (C_T \times T)^2\right]^{1/2}$$

Where:

$M$ =maximum bending moment
$C_m$ =1.5-2
$C_t$ =1-1.5
$\tau$ =41.38$\times$10$^6$ (N/m$^2$)

By substituting upper values in this formula we have:

$d^3$ = (16/($\pi$ $\times$41.38$\times$10$^6$ ))[(281.513)$^2$+(1.543.02)$^2$]1/2
$d$ = 27mm

A 30 mm standard size commercially steel shaft was used for this purpose.

**Bearings Selection**

Ball bearing are usually operated with some combination of radial and thrust loads. The equation for equivalent radial load for ball bearing can be found by the formula: (9)

$Fe = XVFr + YFa$

The X and Y factors in the equation depend upon the geometry of the bearing, including the number of balls and ball diameter. There are two values of X and Y. The set of values giving the largest equivalent load should be used.

$X1=1 \qquad X2=0.5$

$Y1=0 \qquad Y2=1.4$

According to the preceding calculations, the maximum radial force is 1776.1 N and thrust force is zero.

Fe 1=111776.1+0 = 1.776 kN

Fe 2=0.511776.1+0 = 0.888 kN

Standard tables of bearing selection (9) shows that the rating load of a plain ball bearing with 10 mm inner bore, is 3.58 kN. But the calculated shaft diameter is 27mm. So a standard 30mm shaft and housing shoulder diameter was used to secure adequate support for the bearing and to resist the maximum loads.

## RESULTS

The prototype of the shaking machine was constructed in Farm Machinery department at Shiraz University . The shaker was used in an extensive field experiment conducted at the end of September 1999 in Jahrom, a date growing region in Fars province. In that study the effects of shaking mode, frequency and amplitude on date fruit detachment was investigated. Results revealed that at 300cpm and 60mm amplitude the most effective of Shahani ripe date fruits with minimum unripe fruit removal has occurred, and no fruit damaged was encountered.

## ACKNOWLEDGMENT

The authors are grateful to Mr. Maharlui, Mr. Hejazi and Mr. Sedaghat for their assistance in the manufacture of the experimental shaking machine. .

## NOMENCLATURES

$\delta$ = Beam deflection (m)

P = Applied force(N)

$x$ = Instantaneous displacement from equilibrium position,(m)

$k$ = Spring stiffness, (N/m)
$c$ = Coefficient of viscous damping, (N/m-s)
$r$ = Eccentricity, (m)
$m$ = Mass of unbalance, (kg)
$M$ = Total mass of the system including m, (kg)
$t$ = Time, (s)
$\omega$ = Exciting frequency, (rad/s)
$S$ = Limb displacement, (m)
$\alpha$ = Phase angle, amount the displacement lags impressed force, (degree)
$F_e$ = Equivalent radial load (N)
$F_r$ =Applied radial load (N)
$F_a$ =Applied thrust load (N)
$V$ = A rotation factor (=1 for rotating inner ring)
$X$ =A radial factor
$Y$ = A thrust factor
$\frac{\delta}{2\pi} = \varepsilon$ = Damping ratio
n = Number of oscillation
$x_1, x_n$ = Maximal of two oscillation amplitudes, n periods aparts

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