

EFFECTS OF SHAKING MODE, FREQUENCY AND AMPLITUDE ON 'SHAHANI' DATE FRUIT DETACHMENT. I: DESIGN AND DEVELOPMENT OF AN EXPERIMENTAL SHAKER

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(Received: November 19, 2000)

ABSTRACT

Trials of several methods indicated that selective removal of ripe date fruits could be accomplished by shaking the bunch. The purpose of this research was to design and develop an experimental shaker for investigating the effects of shaking mode, frequency and amplitude on ripe date fruit detachment. The experimental shaker was powered by a continuous variable speed electric motor. The rotational speed of the electric motor could be continuously varied from 0 to 1400 rpm by means of an electric volume control knob on a digital board. The rotating output power of the electric 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 20 mm steel tubing. The boom could reciprocate vertically and horizontally in a brass guide. Using this shaker, date fruit bunches can be oscillated in

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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. At the hanging shaking mode, the bunch stalk was clamped to the end of a rocking arm. The prototype machine was constructed and tested for 'Shahani' date fruit bunches in Jahrom, a date growing town in Fars province, Iran.

Key words : Bunch shaker, Dates, Fruit detachment, Selective harvesting.

تحقیقات کشاورزی ایران

(۱۳۸۰) ۱۲۸-۱۲۹:۲۰

تأثیر فرم، بسامد و دامنه ارتعاش بر جداسازی میوه خرماي

'شاهانی'. بخش اول : طراحی و ساخت یک دستگاه

ارتعاش دهنده آزمایشگاهی

محمد لغوی و محمد ابونجمی

به ترتیب دانشیار و دانشجوی سابق کارشناسی ارشد بخش مکانیک ماشینهای کشاورزی دانشکده کشاورزی دانشگاه شیراز، شیراز، جمهوری اسلامی ایران.

چکیده

بکارگیری روش های ارتعاشی در برداشت میوه نشان داده که برداشت خرماي رسیده به وسیله ارتعاش پنگ به خوبی امکان پذیر است. هدف این تحقیق، طراحی و ساخت یک دستگاه ارتعاش دهنده برای بررسی اثر فرم، بسامد و دامنه ارتعاش بر جداسازی میوه خرما از خوشه

بود. دستگاه ارتعاش دهنده قدرت مورد نیاز خود را از یک موتور الکتریکی ۲/۲ kW با دور متغیر پیوسته دریافت می کند. سرعت دورانی موتور الکتریکی به صورت پیوسته از صفر تا ۱۴۰۰ دور در دقیقه روی یک بورد دیجیتالی قابل تغییر است. توان خروجی موتور الکتریکی به وسیله سیستم رانش تسمه ای به یک فلاپویل کوچک انتقال پیدا می کند، سپس این حرکت دورانی به وسیله یک مکانیزم لنگ و لغزنده به حرکت ارتعاشی تبدیل می شود. مکانیزم لنگ، قابلیت تغییر دامنه نوسان در سطوح ۲۰، ۴۰، ۶۰، ۸۰، ۱۰۰ و ۱۲۰ میلی متر را دارد. حرکت ارتعاشی به وسیله یک بازوی ۵۰۰ میلی متری به گیره دستگاه ارتعاش دهنده منتقل می شود. این بازو می تواند در یک پوش برنزی حرکت رفت و برگشتی به صورت افقی یا عمودی انجام دهد. با این دستگاه، پنگ خرما را می توان در حالت های افقی، عمودی و آویزان به ارتعاش در آورد. در حالت ارتعاش عمودی، حرکت حاصل از مکانیزم لنگ و لغزنده به وسیله یک بازو و گیره متصل به آن، به پنگ خرما انتقال می یابد که انتهای ساقه آن به وسیله گیره متصل به شاسی دستگاه محکم نگه داشته شده است. در حالت ارتعاش آویزان، انتهای ساقه پنگ به گیره انتهای یک بازوی الکلنکی متصل می شود که حرکت ارتعاشی را از مکانیزم لنگ و لغزنده دریافت کند. این دستگاه ارتعاش دهنده، در یکی از نخلستان های شهرستان جهرم واقع در استان فارس آزمایش شد.

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 fruits in a bunch do not ripen at the same time, it is necessary to make several pickings during the harvest season. Selective hand-picking of the ripe fruits from each bunch is the most expensive cultural operation. The number of hand-picking has gradually decreased due to shortage of labor for harvesting and increasing their wages. Development of the date mechanical

harvesting began in 1961-62. Trials of several methods indicated that rapid removal of ripe fruit could be accomplished by shaking (4, 5). Harvesting systems has been developed using a variety of high-lifting personal platforms and bunch shakers. The use of these systems have made the work easier and more attractive to the workers and resulted in reduction of labor requirement by 70% (2, 3). Perkins and Brown (4) reported that the vertically shaking mode had the best effect on ripe fruits detachment. According to this result rigidly mounted and hand carried shakers were built. The hand carried shaker was hydraulically powered, weighed 12.5 lb, and imparted 1.5 in stroke to the bunch at about 1400 cpm. It was developed for selective harvesting of ripe fruits from the uncut bunch. The rigidly mounted shaker was designed to impart a 3¼ in stroke at about 700 cpm to the bunch. Sarig *et al.* (6) designed a portable tractor mounted date bunch shaker. It was hydraulically powered, and delivered a 9.5 cm stroke vibration at 67 Hz. Later, Sarig (7) used an inertia type shaker for shaking the palm. He reported that shaking the palm at the lower quadrant of its height (about 2.5-3 m) yielded the optimal rate of fruit removal (more than 90%). Furthermore, he found that fruit removal from the palms with upright fronds was more than those with hanging fronds (7).

MATERIALS AND METHODS

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.

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 electric motor could be continuously varied from 0 to 1400 rpm 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 keep the shaking frequency precisely constant. Shaking frequency was displayed continuously on the digital board monitor.

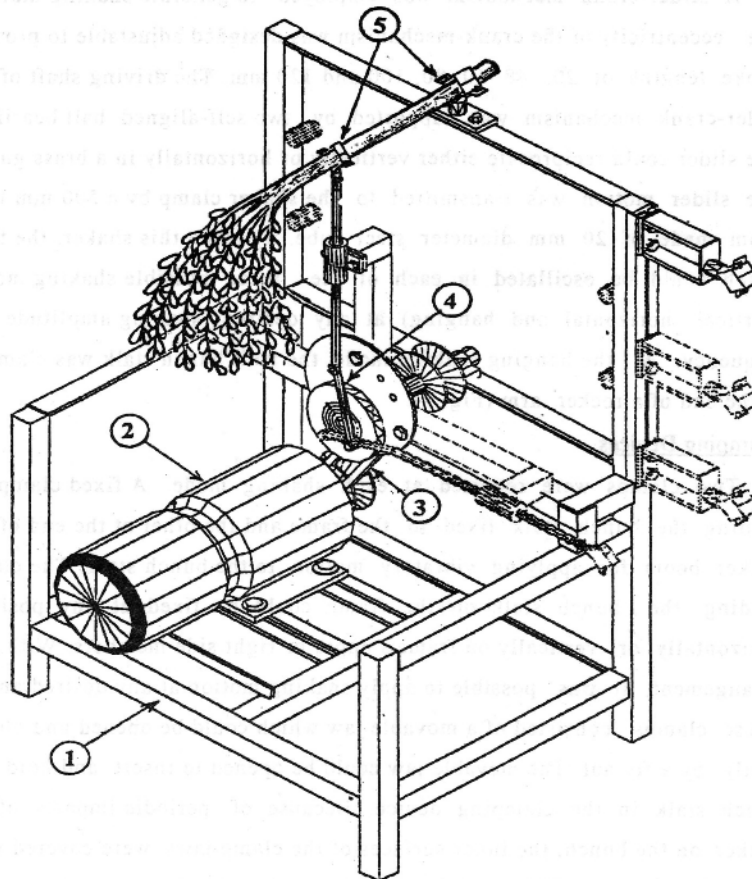


Fig. 1. Schematic diagram of the experimental shaker.
1. Main frame. 2. Power unit. 3. Power transmission.
4. Shaking mechanism. 5. Clamping device.

Power Transmission

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

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 120 mm. 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 500 mm long boom made of 20 mm diameter steel tube. By using this shaker, the fruit bunch could be oscillated in each of the three possible shaking modes (vertical, horizontal and hanging) at any desired shaking amplitude and frequency. At the hanging shaking mode, the fruit bunch stalk was clamped to the end of a rocker arm (Fig. 2).

Clamping Devices

Two clamps 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 vibratory 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. The movable jaw could be opened to insert and hold the bunch stalk in the clamping device. Because of periodic impacts of the shaker on 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.

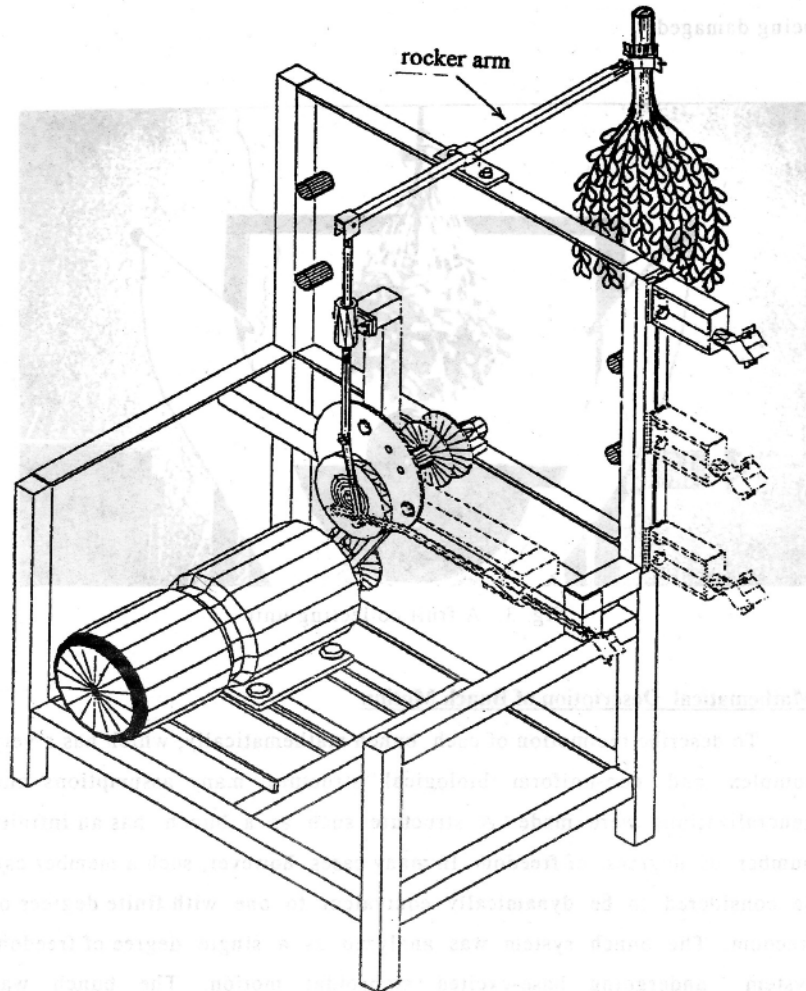


Fig. 2. Schematic diagram of the experimental shaker equipped with a rocker arm to provide hanging mode vibration.

Fruit Collecting Unit

The collecting unit consisted of a special nylon fabric curtain extended under the fruit bunch. This curtain was held by two 1.8 m long horizontal bars pivotly 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.



Fig. 3. A fruit collecting unit.

Mathematical Description of Bunch Motion

To describe the motion of each 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 was 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. A fruit bunch was assumed as a cantilever beam with its strands concentrated at the free end (Fig. 4).

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

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

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

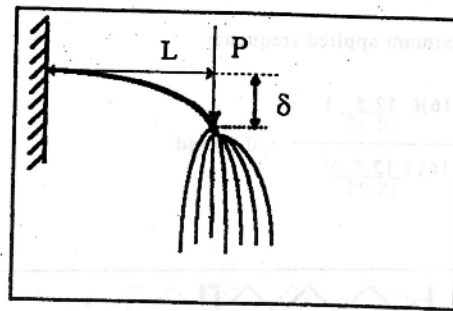


Fig. 4. A fruit bunch simplified model.

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 (9)

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

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 changes of oscillation amplitude, and averaging the values of damping ratio it was found $\epsilon=0.16$.

Natural Frequency

An important stage in this study was to determine fruit bunch natural frequency which could be found by the following formula (3):

$$\omega_n^2 = \frac{k}{m} (1 + \epsilon^2) = \frac{3250}{10} (1 + (0.16)^2) = 333.32, \quad \omega_n = 18.25 \text{ Hz}$$

Phase Angle (α)

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

$\omega = 12.5 \text{ Hz}$, maximum applied frequency

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

$$\alpha = 0.23 \text{ rad}$$

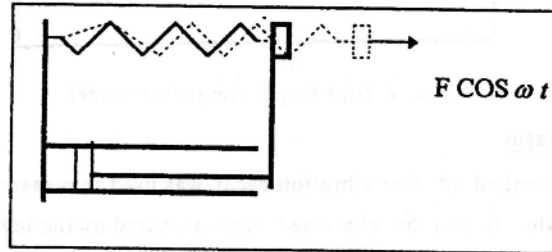


Fig. 5. The bunch spring-mass model.

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

$$F = ma \quad \text{or} \quad (\text{spring} + \text{damping} + \text{applied force}) = \text{inertia force}$$

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

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

The solution of equation is of the form

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

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

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

By substituting these values into Eq.[3] it can be shown that:

$$S \equiv \frac{2 m r}{M} \quad [4]$$

Also it can be shown that the internal force is:

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

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

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

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

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

$$\begin{aligned} P_{ave} &= \frac{\int_0^T (p \delta t)}{T} = \frac{1}{T} \int_0^T P dt = \frac{1}{T} \int_0^T [m r \omega^2 \cos \omega t] \left[-\frac{S}{2} \omega \sin(\omega t - \alpha) \right] dt \\ &= -\frac{S^2}{4} \omega \left[(k - m \omega^2)^2 + c^2 \omega^2 \right]^{1/2} \sin \alpha \end{aligned} \quad [8]$$

$$\begin{aligned} P_{av} &= \frac{(0.06)^2}{4} (2)(3.14)(12.5) \left\{ (3250 - 10((2)(3.14)(12.5))^2 + \right. \\ &\left. (52.8)^2 ((2)(3.14)(12.5))^2 \right\}^{1/2} \sin(13) \approx 1 \text{kw} \end{aligned}$$

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

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

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

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

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

Design of V-belt Drive:

Flexible machine elements, such as belts are elastic, and play an important role in absorbing shocks 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 (1).

Design Power:

$$\text{Design power} = (\text{service factor}) (\text{calculated power}) = 1.5 (3.37) \cong 5 \text{ kW}$$

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)

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$$L_p = 2(50) + 1.57(2)(14) = 144 \text{ cm} = 56.6 \text{ in}$$

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 = pitch diameter of small sheave

$$V = 3.14(0.14)(750)/60 = 5.49 \text{ m/s}$$

Effective Pull

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

where:

K = efficient of friction = 0.512 (8)

θ = 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)

$$(T_1 - T_2) = 1000 (5) / 5.49 = 920 \text{ N}$$

$$4T_2 = 920 \text{ N} \quad T_2 = 230 \text{ N}, \quad T_1 = 1150 \text{ N}$$

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 hr (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.

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

Flywheel mass = 10 kg

$$W = mg = 10 \times 9.81 = 98.1 \text{ N}$$

$$F_1 = F + W = 717 + 98.1 = 815.1 \text{ N}$$

$$F1 = T1 + T2 = 1380 \text{ N}$$

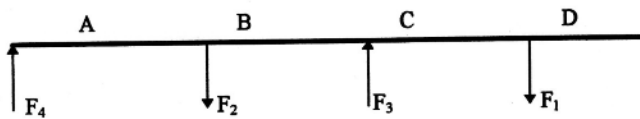


Fig. 6. Free - body - diagram of the power transmission shaft.

On the free-body-diagram of power transmission shaft (Fig . 6), F₁ and F₂ are the loads exerted by boom and pulley and F₃ and F₄ are the bearings loads on the shaft. These bearing reaction are calculated by solution of the equilibrium equations as:

$$F_3 = 1776.67 \text{ N}$$

$$F_4 = 418.33 \text{ N}$$

The maximum bending moment on the shaft occurs at bearing C and it is easily calculated as:

$$M_{max} = 815.1 \times 0.1 = 81.51 \text{ N-m}$$

shaft diameter is obtained by the formula [9] :

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

where:

M = maximum bending moment

$$C_m = 1.5 - 2$$

$$C_t = 1 - 1.5$$

$$\tau = 41.38 \times 10^6 \text{ (Nm}^{-2}\text{)}$$

By substituting upper values in this formula we have:

$$d^3 = (16 / (\pi \times 41.38 \times 10^6)) [(2 \times 81.51)^2 + (1.5 \times 43.35)^2]^{1/2}$$

$$d = 27 \text{ mm}$$

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

Bearings Selection

Ball bearings 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]:

$$F_e = X V F_r + Y F_a$$

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.

$$X_1 = 1$$

$$X_2 = 0.5$$

$$Y_1 = 0$$

$$Y_2 = 1.4$$

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

$$F_{e1} = 1 \times 1 \times 1776.1 + 0 = 1.776 \text{ kN}$$

$$F_{e2} = 0.5 \times 1 \times 1776.1 + 0 = 0.888 \text{ kN}$$

Standard tables of bearing selection (8) 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 27 mm. So a standard 30 mm shaft and housing shoulder diameter was used to secure adequate support for the bearing and to resist

the maximum loads. Calculated parameters pertaining to the design of the experimental shaker are given in Table 1 .

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 this study the effects of shaking mode, frequency and amplitude on date fruit detachment was investigated. The results of this research will be published in a separate article.

Table 1. Calculated parameters pertaining to the design of the experimental shaker.

Fruit bunch static apparent stiffness, k (kN m^{-1})		
Horizontal plane = 4.1	Vertical plane = 2.4	Ave. = 3.25
Fruit bunch damping ratio $\varepsilon = 0.16$		
Fruit bunch natural frequency, $\omega_n = 18.25$ Hz		
Drying force - displacement phase angle, $\alpha = 0.23$ rad		
Shaking power requirement, P(kW)		
Max. instantaneous = 3.37	Ave. = 1	
Max. driving torque, $T_{\text{max}} = 43.35$ N.m.		
Belt selection design power = 5 kW		
Belt type and length, HB-V-belt, $L = 144$ cm		
Belt expected life = 1030 hr		
Driving shaft and bearing diameter, $d = 30$ mm		

ACKNOWLEDGEMENT

The authors are grateful to Mr. Maharlui, Mr. Hejazi and Mr. Sedaghat for their assistance in the manufacture of the experimental shaking machine. This research was conducted at Shiraz University as a part of the project No.513 funded by the National Research Council.

NOMENCLATURES

δ = Beam deflection (m)

P = Applied force (N)

X = Instantaneous displacement from equilibrium position, (m)

K= Apparent 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)

ω = Exciting frequency, (Hz)

S = Limb displacement, (m)

α = Phase angle, amount the displacement lags impressed force, (rad)

Fe= Equivalent radial load (N)

Fr= Applied radial load (N)

Fa= 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 oscillations

X₁, X₂= Maxima of two oscillation amplitudes, n periods apart

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