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Design and Construction a Walnut Peeler

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Abstract
Post-harvest operations have been used in order to increase the efficiency of product production. In the case of walnuts, one of the important processes for improving quality is peeling the walnut's shell. Because of the amount of walnuts produced in Iran and the necessity to reduce the loss of products, a device for quick removal of the walnut's shell is needed. This study examined and compared the different types of peeler machines, interior and exterior designs and also miscellaneous patents. Based on existing literature and to the needs of domestic consumers, we determined the important criteria and designed an improved machine for peeling walnuts.

Keywords: Green shell, Peeler, Walnut.

Introduction

Post-harvest processing of agricultural products, in addition to reducing the losses and costs related to transportation, not only supplies value-added products, but also provides a better export of them with increasing the quality. In case of walnuts, peeling the walnut's hulls immediately after harvest and drying are very important steps. The outermost surface of the walnut that just has been harvested is covered by a green layer. This layer is relatively thick and fleshy, and usually remains fresh and juicy for a long time. Compaction of walnuts may cause problems such as mustiness, crushing and blackening of the nuts. Therefore a machine that can peel the nuts quickly and easily seems to be necessary. One of the most high-yielding orchard fruits is walnut (*Juglans regia L.*). The Persian walnut is one of the most important nutritive nut crops (Ebrahimi *et al.,* 2009). According to FAO statistics in 2011, major producers in the world are China, Iran (Islamic Republic of), United States of America and Turkey (FAO 2011). Nowadays through the use of mechanized methods, walnut yields average 5.3 tons per hectare. Walnut has a variety of uses in industrial, pharmaceutical and cosmetics fields. Several studies in this area have shown that walnuts have a positive effect on cholesterol (Lavedrine *et al.,* 1999; Munoz *et al.,* 2001). It also has rich nutritional properties and beneficial effects on heart diseases (Chisholm *et al.,* 1998; Savage, 2001; Feldman, 2002; Çağlarırmak, 2003; Ozkan, 2005; Colaric *et al.,* 2006; Özcan, 2009). However, few studies exist on walnut post-harvest operations (Sen, 1986; Akca, 2001). Walnut harvest and post-harvest operations in Iran are still done manually (Ghafari *et al.,* 2011). Obviously, the traditional methods of peeling walnuts are not suitable for the high volume of walnuts that are produced. Therefore, the use of mechanical peelers is required. It is also clear that companies demand high-performing peelers (Dasso, 2012).

The purposes of designing this machine are as follows:

- To accelerate removal of walnut's hull after harvest and to improve its quality and value
- To prevent outbreaks of disease and pests and to improve sanitation and provide environmental protection (related to compaction and spoiling of unhulled walnuts)
- To decrease the number of processing operations by reduction of some activities such as whitening shells and decolorization.
- To allow longer storage of nuts
- To reduce the costs associated with damage to the skin of workers

History of researches done in the field of designing and manufacturing Walnut peeler machines

Barton (1956) managed to invent a nut stripping machine that used wire straps. This machine is used because of its compatibility with the rod-like unit of current systems such as Wizard and Hul-it. Also, this system can be completely replaced by those systems that mentioned. This machine operates by tapping and scratching the husks, by pulling the rod on the green shell. McFarland and colleagues (1977) have invented a peeling machine that is equipped with jagged rollers and washing nozzles, and Cacho (1978) has described a walnut peeling machine that has four main parts:

1. Container or encasement.
2. The rotating shaft that is placed inside and has a range of rotational speeds from 250 to 1400 rpm.

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3. A turbine-like component with a large number of blades attached radially to the aforementioned shaft.
4. An electric motor.

This system has the following advantages:

- It is used not only to remove the green husk but also to remove hard skin (shell).
- This machine has high efficiency and performance.
- Nuts of different sizes can be used.
- Does not demand any preliminary manual operations.

Volk and his colleagues (1981), invented a stripper that is equipped with a fluted cylinder and a plate which is fluted. This machine is designed to peel pistachios, but because it can be modified for a variety of sizes, it can also be used to peel walnuts. It takes very little power.

Trujillo (2003) invented a walnut peeler that is fitted with mixer fingers. Because of the high operation speed, this machine is especially suitable for pistachio. This device has a plate that is connected to the inner surface of the peeler and has a cover that is slightly raised and improves the peeling operation. This operation takes about 95 seconds for each walnut.

This instrument has the following advantages:

- The mixer is equipped with a stripper which rotates into the walnut tank and accelerates friction and separation of green husks.
- It has a low-cost and is affordable for companies and factories.
- Its increased efficiency reduces the need for repeat operations and other treatments.
- High speed
- Easy to operate
- Can be added to other systems to improve husking operations.
- Mixing fingers are replaceable if damaged.

Materials and Methods

Machine Design Criteria

A peeling machine was made in the Department of Agrotechnology at the Abouraihan College and then evaluated and optimized (Fig. 1). The machine design criteria are:

- To be user-friendly and repairable by replacement of parts
- Uses the fewest possible parts
- Minimal size and weight for easy mobility
- Takes advantage of machines with different capacity and speed
- Good durability
- High quality in peeling operation (minimum percent breakage and damage)
- Has a reasonable capacity

Design and manufacturing processes

Design and construction phases of the machine were:

1. Study of designs and devices that are made in and outside of Iran in order to compare their features, strengths and weaknesses and then match a series of designs and inventions that meet the criteria mentioned in the previous section.
2. Design and production of a prototype device in order to evaluate its performance, capacity and power requirements.
3. Evaluate the performance quality and capacity of the machine.
4. Appraise the power requirements.
5. Design of machine components that are reliable under conditions of maximum power and then make necessary changes in the design of machine components.

In this procedure, resultant forces propel the walnut to the body of the grooved cylinder and also into each other, resulting in removal of walnut's hull. In addition, friction between walnut and body of grooved cylinder is effective in the peeling process. Therefore, linear velocity of abrasive wire brushes and the distance between inner surface plates and the grooved cylinder are important design parameters for this device.

The machine that resulted from the different designs consists of eight main parts:

1. Electric motor
2. Moving pulley
3. Set of stimulus (driver) pulleys
4. Shaft
5. Holder plate for the grooved cylinder along with a set of bearings
6. Abrasive wire brushes
7. Grooved cylinder
8. Chassis

Fig.1. Photograph of the walnut peeler

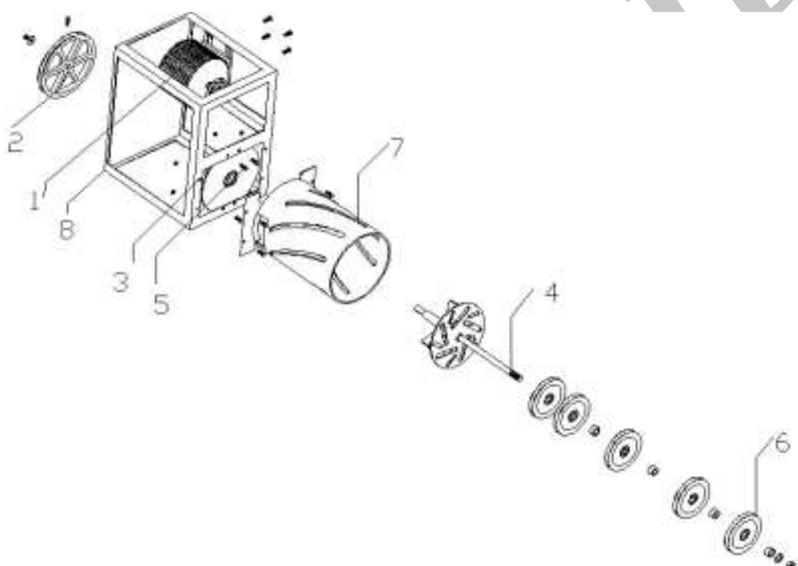

Fig.2. The exploded diagram of the machine: 1. Electric motor 2. Moving pulley 3. Set of stimulus pulleys 4. Shaft 5. Holder Plate for grooved cylinder with a set of bearings 6. Abrasive wire brushes 7. Grooved cylinder 8. Chassis

Results

Design of Machine Elements

An exploded diagram of the machine is shown in Fig. 2. In this figure different parts of the machine are numbered. Details of the design will be discussed in the following sections.

Selection of item number 1: the electric motor

Electrical power should be enough to provide required torque and speed of machine. Power is directly related to the torque and rotation. Relationship between the torque and power is expressed in Equation 1.

$$P = \frac{2\pi nT}{60000} \quad \text{Eq.1}$$

Where:

n = Rotational speed of the motor (rpm)

T = generated torque (Nm)

In the basic design, a single phase, 2 horsepower electrical motor is used at a rotational speed of 1420 rpm. During the testing and evaluation of the device, the heating rate performance and the quality of the initial launch was acceptable. According to equation 2, maximum amount of required current to operate the machine is extracted from its specified rotational speed. For two horsepower (equivalent to 1.5 kW) and 220 volt, the current is calculated to be 6.8 Amperes.

$$i = \frac{P}{V} \quad \text{Eq.2}$$

Where:
i = current (Ampere)
P = power (Watt)
V = voltage (Volt)

By measuring the input current of the machine, during testing and the initial evaluation, that is about 5.2 mA, in comparison with the maximum input current (6.8 amps), it found that with optimized performance of electrical motor it is possible to peel 25 walnuts in 60 seconds. It is worth noting that under these conditions, motor will not be overloaded.

Design of parts number 2 and 3: driver and driven pulleys

For agricultural machinery, when it is not required to establish the precise rate of velocities, the pulley and belt system is used for power transmission. The advantages of this system are impact alleviation, no requirement for lubrication and fewer alignment problems compared with other transmissions. Thus, in addition to lower production costs, this sort of system also reduces maintenance costs. Other benefits of using a pulley and belt system to transfer power are ability to dampen high vibrations and simpler safety systems. It should be mentioned that both pulleys (driver and driven pulleys) in this machine are made from cast iron.

Important design parameters include:

- Selection the type of belt
- Determination of driver and driven pulleys according to the rotational speed of the machine
- Determination the distance between the centers of the pulleys due to limitations of space and location
- Determination of belt length required according to standards
- Determination the linear speed of the belt and specification of the number of belts needed for transmission
- Calculation the initial stretch of belts
- Determination belt's force on both sides of the pulley

Selection of belt type

According to the amount of power transfer needed and using a table of selection belts, the belt selected for this machine is type A. The minimum diameter of the driver pulley in belt Type A for transmission of 2 hp, is 7.2 cm (2.83 inches). Therefore for this machine, the minimum diameter was chosen as 7.2 cm. Using equation 3 and assuming 7.2 cm as the diameter of

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driver pulley, 1420 rpm for electric motor and 368 rpm for the driver pulley, the diameter of driven pulley is 27 cm.

$$\frac{D}{d} = \frac{n_2}{n_1} \qquad \text{Eq. 3}$$

Where:

D and n_1 are the diameter and rotational speed of driven pulley, respectively (connected to main shaft of machine), and d and n_2 are diameter and rotational speed of driver pulley, respectively, that are connected to the electric motor.

Determination of distance between centers of pulleys

Consisering geometric limitations and according to the following relationship, the distance between the centers of pulleys will be:

$$D \le C \le 2(D+d) \Rightarrow 27 \le C \le 2(27+7) \Rightarrow 27cm \le C \le 68cm \qquad \text{Eq. 4}$$

$$L = \frac{\pi}{2}(D+d) + 2C + \frac{(D-d)^2}{4C} \qquad \text{Eq. 5}$$

Where:
C = distance between two centers
D = diameter of bigger pulley
d = diameter of smaller pulley

Due to geometric limitations, the distance chosen was 28 cm.

The design features of driven and driver pulleys are shown in figures 4 and 5 respectively.

Calculation the length of belt

The belt length can be determined from the following equation:

Eq.5

Where:
L = length of the belt
C = distance between two centers
D = diameter of bigger pulley
d = diameter of smaller pulley

Therefore the calculated length of the belt is 113.2 cm (44.57 inches). Hence, based on lenths listed in existing tables in design references, a belt length of, 48 inches (121.9 cm) was considered

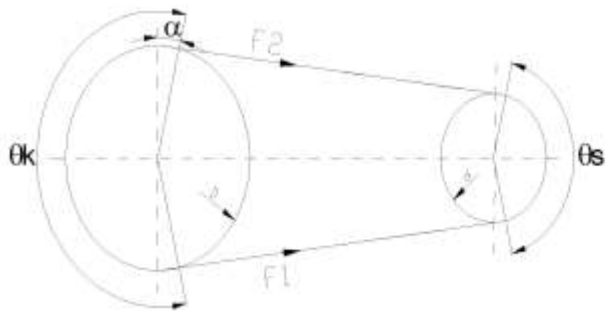

Fig.3. Applied forces on both sides of belt in a pulley-belt transmission system

Calculation of Peripheral velocity of belt

In order to calculate the peripheral speed of the belt, following equation is used:

$$V = R\omega = \frac{2\pi Rn}{60} = \frac{\pi \times 7 \times 1420}{60} = 520.2 cm/s \qquad \text{Eq.6}$$

Where:
V = peripheral speed
R = radius of driver pulley
ω = Angular velocity
n = rotational speed

Calculation the number of belts required for power transmission

According to machine design criteria, it is possible to transmit 0.66 hp power per belt. If it is necessary to transfer 2 horsepower, then three belts must be used.

$$N = \frac{2}{0.66} \cong 3 \qquad \text{Eq.7}$$

Where:
N is number of required belts for transmitting 2 horsepower.

Calculation of the initial stretch belts

To set up the machine, it is necessary to create an initial tension in the belt to prevent it from slipping on the pulley. Knowing this fact and that after belt the belt starts moving, the amount of tension increases in one side of the belt and is reduced on other side, it is necessary to calculate each force (shown in Fig. 3). The reason for measuring these forces and their importance is because when the belt generates a force, this force is transmitted to bearings and finally to the shaft and this force cause a bending moment on shaft. According to figure 3, for selecting bearings and designing of shaft, it is essential to calculate the generated forces on each side of the belt. To calculate the generated tensile forces on each sides of belt during operation, the following two equations (eq. 8 and eq. 9) are used:

$$\frac{F_1 + F_2}{2} = F_0 \qquad \text{Eq.8}$$

$$\frac{(F_1 - F_2)d}{2} = T \quad F_1 \geq F_2 \qquad \text{Eq.9}$$

$$T = \frac{P}{\omega} = \frac{1500 watt}{\frac{2 \times \pi \times 1420}{60}} \cong 10 n.m \qquad \text{Eq. 10}$$

Where:
F_0 = initial force in stretched belt (N)
F_1 and F_2 = Tensile forces that are created on both sides of the belt on the pulley during transmission (N)
D = diameter of driver pulley (m)
T = Maximum transmitted torque (N.m)
P = power (Watt)
ω = angular velocity (Rad.s^{-1})

It is worth noting that the initial stretch of the belt is calculated according to type of material the belt is made of and also to values of permissible stresses. The permissible value of tension in elastic belts should be less than the maximum stress. This tension in the belt is equal to 1.75 MPa; the cross-sectional area of the belt is equal to 104 mm^2. Hence:

$$\frac{F_0}{A} = 1.75 \qquad \text{Eq.11}$$

$$\Rightarrow F_0 = 1.75 \times 104 = 182N$$

$$A = 8 \times 13 = 104 mm^2$$

Therefore, the initial stretch of belt was chosen to be 150 N. If, durating operation, the machine needs more power, the position of engine can be changed by adjusting the position of the motor's chassis to achieve more initial tension because increasing the initial stretch of the belt, increases the power transmission capacity. Therefore, according to Equations 8 and 9:

$$\begin{cases} \dfrac{F_1+F_2}{2}=150 \\ \dfrac{(F_1-F_2)d}{2}=T=10 \end{cases} \Rightarrow F_1=1515N, F_2=1485N$$

And in the most difficult conditions (the maximum initial tension for maximum power transfer), we have:

$$\begin{cases} \dfrac{F_1+F_2}{2}=182 \\ \dfrac{(F_1-F_2)d}{2}=T=10 \end{cases} \Rightarrow F_1=1835N, F_2=1805N$$

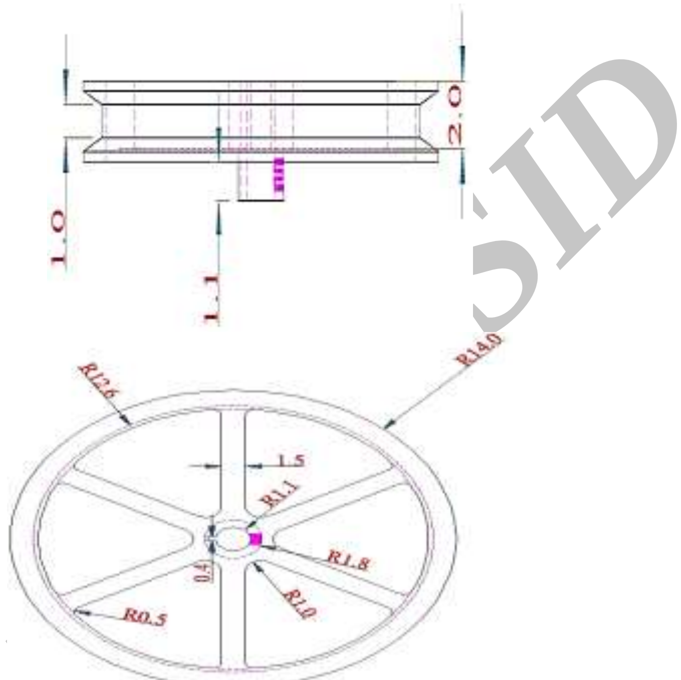

Fig.4. Driven pulley

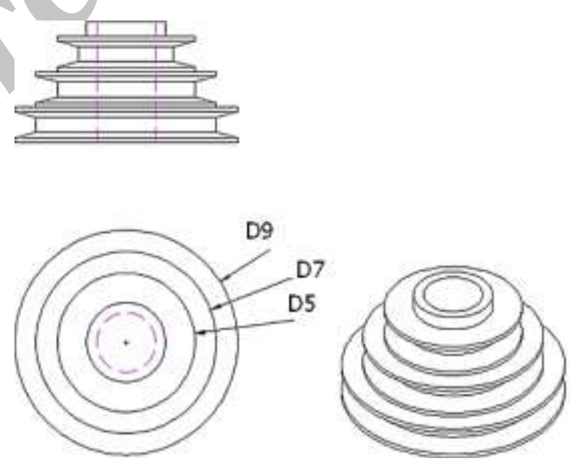

Fig.5. Driver pulley

Design of part number 4: shaft of machine

The role of the shaft is to transmit force to the abrasive brushes and also to stir the walnuts in the grooved cylinder. One end of the shaft is connected to the pulleys and its threaded end is free. To this shaft is welded an agitating plate and bearing holder; in this way these components become a single piece. Important design parameters for the shaft are:

- Determination of the transmitted torque and rotational speed of the shaft
- Selection of appropriate material
- Definition of geometric limitations in the shaft and specification of the location of bearings and pulleys
- Specification of forces, shear torque and bending moment, and eventual estimation of the minimum required diameter of the shaft

The rotational speed of the shaft must be equivalent to the optimized rotational speed for hulling action (368 rpm). The transferred power is assumed to be 2 hp, so according to equation 10, the transmitted torque is 10 N.m. The shaft is made according to the standard table DIN17200 from C45 steel with yield strength of 335 to 430 MPa. Three dimensional shaft diagrams are shown in Fig.6 and Fig. 7.

Fig.6. Three dimensional shaft design

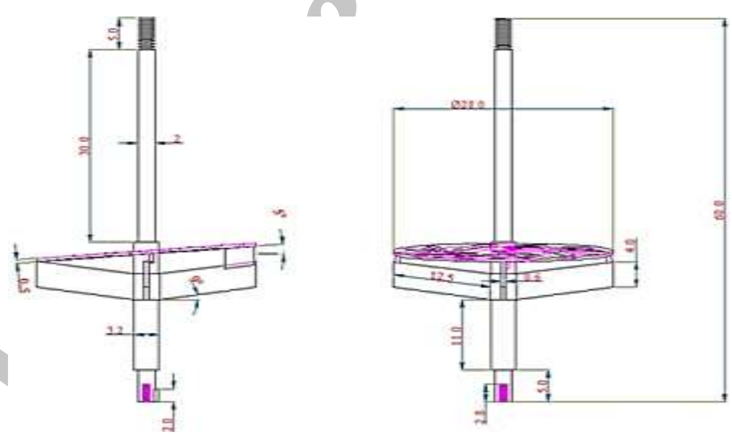

Fig.7. Location of pulleys, bearings and abrasive brushes on the shaft

The next step is to determine the maximum shear stress in the assembly, to identify critical points and also to determine diameter of the shaft at various places with a safety factor 5. In order to determine the critical points, two points 1 and 2 are considered as shown in Fig.8. In point 1, which is simply the base of shaft, the belt forces (F_1 and F_2) that are determined in the section (above) on pulley design, because bending moments? These moments have maximum values on the right side of the base of the shaft. The two mentioned forces are multiplied to 5 cm (distance from the beginning of shaft where the driven pulleys are located), to obtain the maximum bending moment. The generated anchor force causes the bending moment to remain constant on the left side of the base.

So in comparison with critical point number 1, number 2 has an equal bending moment load, but because of sudden changes in thickness and the diameter of the small shaft at this point, the critical point number 2 is investigated. In Fig.9, the bending moment diagram, the torque of the shaft and its critical points are shown.

According to ASME regulations and the following equation (Eq. 12) we have:

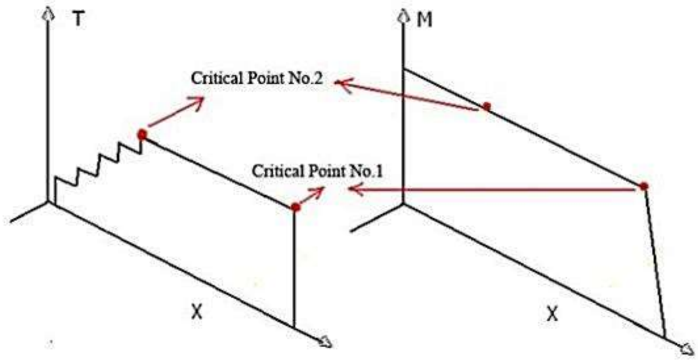

Fig.8. Schematic diagram of applied forces and moments to the shaft. Points No.1 and No.2 are detected as critical points

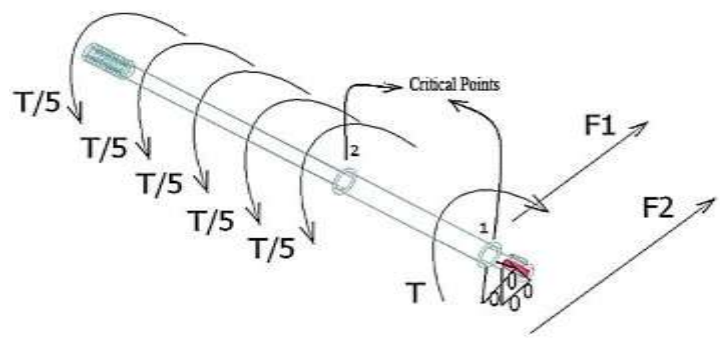

Fig.9. Diagram of bending moment of shaft and it's torque

$$d = \left\{ \left(\frac{32n}{\pi \sigma_{yp}} \right) \left[(C_m M)^2 + (C_t T)^2 \right]^{1/2} \right\}^{1/3} \qquad \text{Eq.12}$$

$$M = (F_1 + F_2) \times (L_1) = (2F_0) \times (L_1) = 364 \times 0.05 = 18.2 N.m$$

$$T = 10 N.m \qquad \text{Eq.13}$$

$$C_m = 2$$

$$C_t = 1.5$$

Where:

d= diameter of shaft (m)

n = safety factor

σ_{yp} = yield stress (N.m)

M = the bending moment transmitted by the shaft (N.m)

T = torque transmitted by shaft (N.m)

C_m and C_t = coefficients that are related to the fatigue phenomenon.

Considering snap load conditions, the coefficients C_m and C_t are 1.5 and 2, respectively (from related tables). Thus, with confidence factor of 5, the the diameter of shaft is calculated to be 182 mm, and the optimized diameter, 200 mm.

Design of part number 5: holding plate and its bearing

The most important design factor for this device is the type of bearing. When considering bering selection, parameters such as load toleration capacity and geometric characteristics have to be determined. Bearings are chosen when the minimum diameter of the shaft has been determined and also when radial loads and location of bearings relative to other components have been specified. The dimensional features and also design parameters of the holding plate are shown in figures 10 and 11, respectively

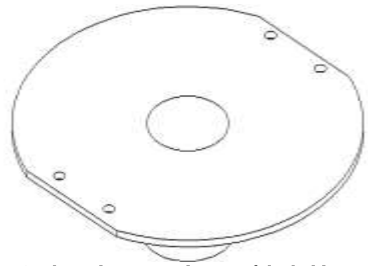

Fig.10. Three-dimensional view of the holding plate

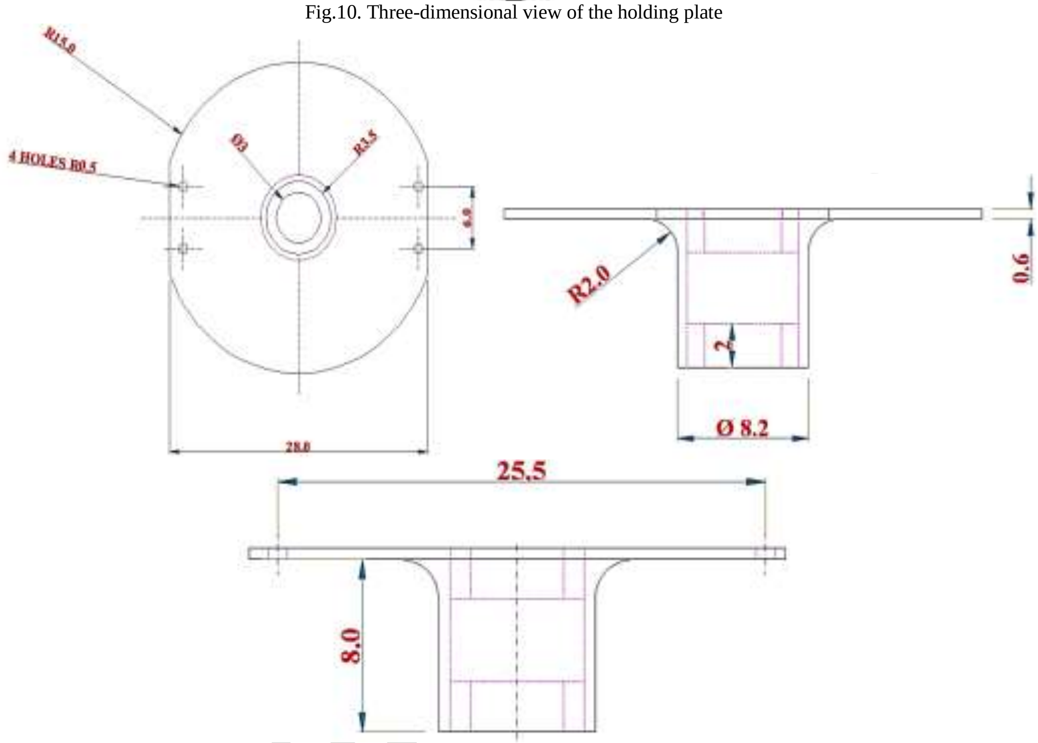

Fig.11. Blue print of holding plate

It should be mentioned that if the shaft or its base has a deviation in direction and or is deformed, resulting in a central deviation, self-adjusting (servo) bearings are used. These kinds of bearings act like a simple fulcrum and do not cause resisting moment on shaft. Thus, because shaft rotates vertically in this machine and in order to avoid distortion of shaft's axis from the geometrical axis of bearing, self-adjusting bearings are used. When radial loads and lateral loads are applied simultaneously, design of bearings generally is done based on equivalent load. For radial bearings and under combined loads (radial and lateral loads), the magnitude of static load (P) or equivalent load is obtained from following equation:

$$P_0 = X_0 F_r + Y_0 F_a \qquad \text{Eq.14}$$

Where:

X_0 = radial load coefficient

Y_0 = lateral load coefficient

F_r = radial load

F_a = lateral or axial load

X_0 and Y_0 values are obtained from the related tables in design books. Radial load on the shaft is due to belt tension. This tension can cause reactions at the bearings (Fig.7). Because locations of fulcrums are very close together, it can be said that both of these forces are equal to:

$$(F_1 + F_2) = (2F_0) = 364N \qquad \text{Eq.15}$$

In addition to the static load that results from belt tension, the axial load (due to the weight of rotating plate, shaft and pulleys) and the centrifugal load (resulting from rotation of axes) are applied on each bearing. The centrifugal force due to the symmetry is neutral, so the axial load is due only to weight. The total amount of these forces is calculated and then is divided by two to obtain the axial force acting on each bearing. The total weight applied to the bearing includes: the weight of the shaft and abrasive brushes (170 N) and the weight of 25 walnuts (equivalent to 30 N), that is, 200 N in total. Since radial force is bigger on the right side of primary bearing the design is focused on this bearing (Fig.8). X_0 and Y_0 values are extracted from the related tables and are:

$$X_0 = 1$$

$$Y_0 = 0.44 \cot \alpha$$

The angle α is between 1 and 2 degrees on the servo (self-adjusting) bearings that we consider the mean value. Thus the value of Y_0 obtained is equal to 16.8; therefore for the equivalent load, we have:

$$P_0 = 1 \times 364 + 16.8 \times 200 = 3724N = 3.7kN$$

The inner diameter of the bearing was calculated based on acceptable minimum diameter (200 mm) of shaft determined above. From bearing tables in design references, a bearing with a 20-mm inner diameter is selected, because it can sustain a 9.45-kN load, which is bigger than the calculated load of 3.7 kN. A suitable bearing is 41280 NBR.

Design of part number 6: abrasive brushes and filler bushing

Five abrasive brushes and six filler bushes are used to create friction between the grooved cylinder and the walnuts. In industry, these brushes have various uses, such as eliminating rust from metals and scouring. During testing and assessment, the optimal distance between the grooved cylinder and brushes was determined to be 16 cm. In Fig. 12, schematic diagrams of the brush with the filler bush are shown. It should be mentioned that, filler bushes do not experience too much load, so they are made of ordinary steel

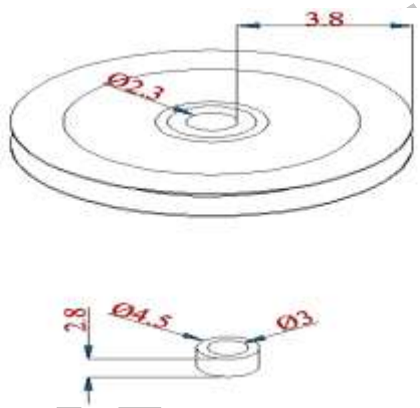

Fig.12. schematic diagram of abrasive brush with dimension of filler bush

Design of part number 7: grooved cylinder

The grooved cylinder acts as a container in which the operations of abrasion and peeling can occur. It is composed of three parts: the base (for connection to the frame by screws), the main body and the outlet valve. Fig. 13 shows a three dimensional view of grooved cylinder and its different parts; fig. 14 shows the blueprint of grooved cylinder. The main frame has several diagonal grooves with sharp edges that play more roles in abrasion and peeling actions. The thickness of grooved cylinder should not be so low that walnuts colliding with body of cylinder cause dents, and it should not be so high that it makes the machine overly heavy. It was determined that for manufacture of the cylinder, a steel sheet of 3 mm thickness is suitable. Walnuts are added at the top of cylinder and, after peeling, are ejected from the lateral outlet valve by application of shaft and brush forces.

Design of part number 8: chassis

All parts of the machine are assembled on the chassis. In particular, the electric motor, the holding plate and the grooved cylinder. Figure 15 shows a three dimensional scheme of the chassis, as well as a detailed plan for it.

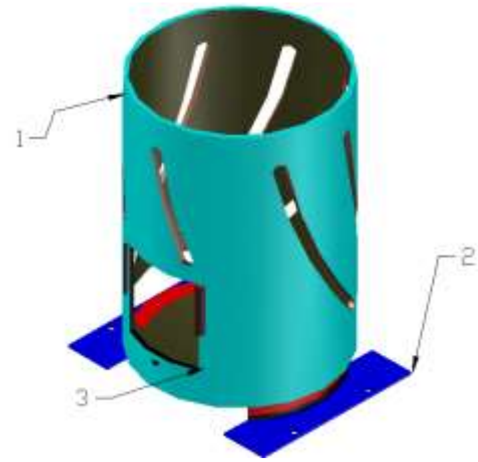

Fig.13. Three-dimensional view of the grooved cylinder and its parts, which includes:

1- main body 2- Core 3- valve outlet

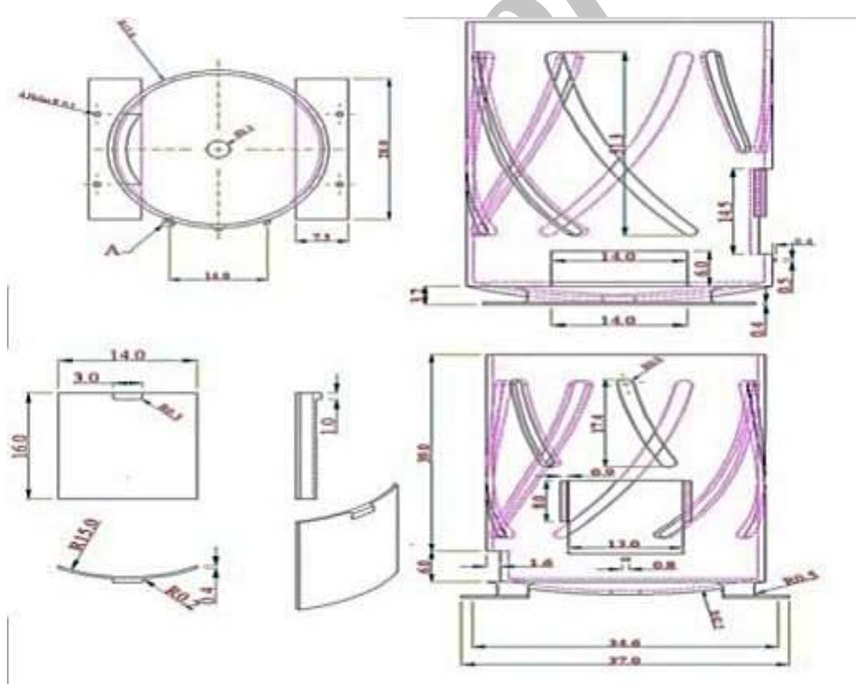

Fig.14. blueprint of grooved cylinder

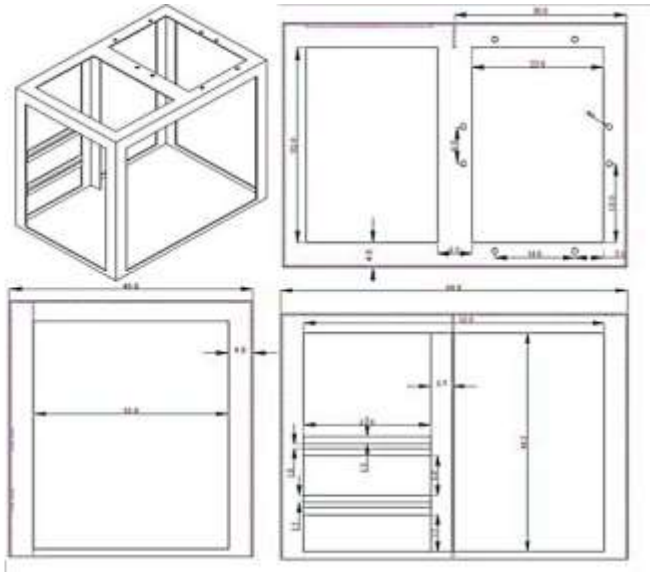

Fig.15. Three dimensional view and blue print of chassis

Discussion

Walnut is one of the most productive horticultural and orchard products. However, because in most orchards throughout the world, post-harvest operations are not are not done in a timely way, a large percentage of this product is lost. Therefore, it is desirable to further automate the post-harvest processing of walnuts. This paper describes the design and construction of a machine to remove green husk of walnut in a convenient manner so as to prevent walnut and walnut meat from decay. In order to achieve this goal, we used a mechanism that simultaneously utilizes friction and rotation to remove the green husk. The peeler designed utilizes an electric motor, a moving pulley, a set of stimulus (driver) pulleys, a shaft, a holder plate to which is attached a grooved cylinder and set of bearings, abrasive wire brushes, a grooved cylinder and a chassis. For each part optimal values are calculated. A single phase, 2 horsepower electrical motor rotating at 1420 rpm is used. Due to the geometric limitations, the distance between centers of pulleys and the length of belt was chosen 121.9 cm. For selection of the bearing, parameters such as load toleration capacity and geometric characteristics were investigated. Finally, it is suggested that the future versions of this machine incorporate the following:

- A bearing on the other end of the shaft to reduce vibrations,
- An entrance door or a tank,
- Devices, such as an automatic shutoff to enhance operating safety.

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