e-palli

# AMERICAN JOURNAL OF FOOD SCIENCE AND TECHNOLOGY (AJFST)

ISSN: 2834-0086 (ONLINE)

VOLUME 2 ISSUE 2 (2023)

PUBLISHED BY
E-PALLI PUBLISHERS, DELAWARE, USA

American Journal of Food Science and Technology (AJFST)

Volume 2 Issue 2, Year 2023
ISSN: 2834-0086 (Online)
DOI: https://doi.org/10.54536/ajfst.v2i2.1503
https://journals.e-palli.com/home/index.php/ajfst

# Design and Development of an Engine Driven Onion Grading Machine

Wabi Tafa[1*], Adesoji M. Olaniyan[2]

**Article Information**

**Received:** April 03, 2023

**Accepted:** May 30, 2023

**Published:** August 21, 2023

**Keywords**

*Design, Development, Onion, Onion Grader, Manufacturing Process*

**ABSTRACT**

Onion (*Allium cepa, L.*) is one of Ethiopia's highly cultivated vegetable crops. Farmers across the country sell their onions without grading them, which results in post-harvest losses during packing and transporting. As a result, modern technologies like engine-operated grading systems are essential. So, the design and development of an onion grading machine was undertaken at Asella Agricultural Engineering Research Center (AAERC). The physical and mechanical properties of the selected onion bulb variety were considered during the design. The developed grader consists of the mainframe, feeding hopper, grading unit, outlets, power source, and power transmission system. The cost of the grader was estimated to be 20,880.99 ETB and believed that it is simple in design, easy in operation, and found to be suitable for small and medium-scale farmers.

## INTRODUCTION

Onion (*Allium cepa L.*) is a member of the Allium genus of the Alliaceae family. It is thought to have originated in southwestern Asia, which was the center of domestication and variability. It is from this place that it spread worldwide and has been cultivated annually for bulb production for over 4700 years (Brewster, 2008). The introduction to Africa in general, and Ethiopia specifically, was not well-known. However, it has been suggested that as introduced by foreigners in the early 1970s and by now produced widely in many parts of the country than the traditionally grown shallot (Adgo, 2008). In Ethiopia, onion is one of the most important vegetable crops grown on a small scale. It is also one of the most economically important vegetables in the country. The area under onion is gradually rising, owing to its high profitability per unit area and ease of production, and the expansion of small-scale irrigation regions (Nigussie *et al.*, 2015). It ranked second in the production of all vegetable crops next to Tomato, which has been concentrated in the central rift valley of the country, particularly in the upper Awash and Lake Batu areas (Bossie, 2009). In Ethiopia, the crop is used extensively in food flavoring and everyday stews, as well as in various vegetable food preparation uses. Despite its year-round production settings, the crop's yield and productivity are much below (10.02 t/ha) the world average (19.7 t/ha) (FAO, 2012). Because of the lack of improved cultivars, the use of inappropriate agronomic practices such as pre-harvest and post-harvest management practices, and a lack of attention/awareness about the benefits of intensive production, the low yield results indicate that there is a huge gap in production and productivity in the country. In onion production, postharvest losses are widespread due to the perishable nature of the onion and the lack of proper postharvest practices and processing technology, which is a major production restriction that reduces productivity. The major causes, according to the findings, are primarily related to technical limitations in harvesting techniques, handling, conveying, packing, storage, and cooling facilities in difficult climatic conditions, infrastructure, and marketing systems (Gebru, 2015).

Similarly, several studies identified the most common causes of postharvest losses as a lack of sorting and grading to eliminate defects before storage. Demand for onion has increased over the years and so is its quality. After the harvest of onions, their quality is important to the final consumer. Damage-free post-harvest handling is regarded as a necessity for increasing the farmer's profit margin in in-freshly eaten commodities like an onion.Grading is the process of classifying materials into various homogeneous groups based on specified qualities such as size, shape, colour, and quality (Londhe *et al.*, 2018). It lowers handling losses during transport and saves time and energy in various processing procedures. Grading also facilitates packing, marketing, and other post-harvest procedures. Grading agricultural products, particularly fruits and vegetables, according to their size is an important value-adding technique that increases the market value required for cross-border trade (Charles and Etiese, 2020). According to the World Trade Organization (WTO), new marketing trends need high-quality graded items. Uniformity in size makes the product more attractive to consumers and also improves its processing qualities. In a developing country, most fruit and vegetable growers grade their products by hand. Grading by hand is expensive and the operation is hampered by labour shortages during high seasons. Human operations can be inefficient, inconsistent, and time-consuming. Farmers are hoping for suitable agricultural product grading equipment to help ease labour shortages, save time, and improve the quality of graded products (Mostafa, 2004).

[1] Oromia Agricultural Research Institute, Asella Agricultural Engineering Research Center, Asella, Oromia, Asella, Ethiopia
[2] Department of Agricultural and Bioresources Engineering, Faculty of Engineering Ikole-Ekiti Campus, Ikole-Ekiti 370001, Ekiti State, Nigeria
[*] Corresponding author's e-mail: wabitafa@gmail.com

In Ethiopia, onion growers are the major actors who perform most value chain functions from farm inputs preparation to post-harvest handling and marketing (Agidew, 2018). Farmers sell their onion produce without grading which leads to post-harvest losses during packing and transporting. Also, they receive a lower price for their produce. However, manual size grading of onion crops is practiced by whole-sellers and retailers thereby, they make a higher profit than farmers. Manual grading is labor-intensive, inconsistent, time-consuming, less efficient, and not standardized. To achieve a uniform size of onion bulbs proper grading mechanism is required, and with the aid of a grading machine, that goal can be achieved (Aher *et al.*, 2019). Different fruit and vegetable grading machines have been developed worldwide. Divergent roller type, expanding pitch type, inclined vibrating plate, and counter-rotating roller with inclination type graders are developed for grading fruit and vegetables depending on size basis(Charles and Etiese, 2020). In Ethiopia persons engaged in postharvest handling of crops have no chance to use imported high-cost size separation and grading techniques. Such high-grading equipment is not useful to small-scale farmers because of the high cost of the machine, high cost of operation, and more maintenance cost. Small-scale farmers and wholesalers are in crucial need of low-cost graders. Various types of onion grading machines suitable for small-scale farmers were developed using different design approaches.

However, these machines are with some limitations related to functional requirements, safety aspects, portability, and ease of operation which affects the performance of the machine. Mainly, the problem observed with those machines were a high percentage of onion bulb damage, low capacity, the rigidity of the design, and also the power source they used is not suitable for the farmers in our country who mostly have no access to electricity. In order to solve this, the present study work was carried out to design and develop an onion grading machine while taking into account the needs of the local market and the lack of suitable low-cost onion grading equipment for small and medium-scale farmers.

## MATERIALS AND METHODS

### Description of Experimental Site

To fulfill objectives, the designing and manufacturing of the onion grading machine prototype was conducted in the Asella Agricultural Engineering Research Center (AAERC) which is located in Oromia National Regional State, Ethiopia. It is located at 6° 59' to 8° 49' N latitudes and 38° 41' to 40° 44' E longitudes, having an elevation of 2430 meters above sea level.

### Design Principles and Considerations

To design and fabricate a suitable machine, machine design parameters were considered before the design and fabrication of the machine. Therefore, designs parameters consider to be important were as follows (Khurmi & Gupta, 2005);

The kinematics of the machine: The successful operation of any machine depends largely upon the simplest arrangement of the parts which give the motion required. Selection of materials: A designer must have a thorough knowledge of the properties of the materials and their behavior under working conditions. Some of the important characteristics of materials are strength, durability, flexibility, smoothness, weight, resistance to heat and corrosion, ability to be casted, welded, or hardened, machinability, electrical conductivity, etc.

### Design and Analysis of Machine Components

The machine was designed to grade onion bulbs into three distinct size ranges small (<40 mm), medium (40-60 mm), and large-sized (>60 mm) bulbs according to the size of onion bulbs with three rotating cylinders having parallel bars spacing of different dimensions. These categories of size were made based on international standards of the marketable size of onion bulbs suitable for commercial applications (Ashraf, 2007). The main components of the onion bulbs grader prototype designed and developed were: Feeding hopper, concentric cylindrical grading unit, pulleys and belts, bearings, power source, and supporting frame as shown below in Figure 1.

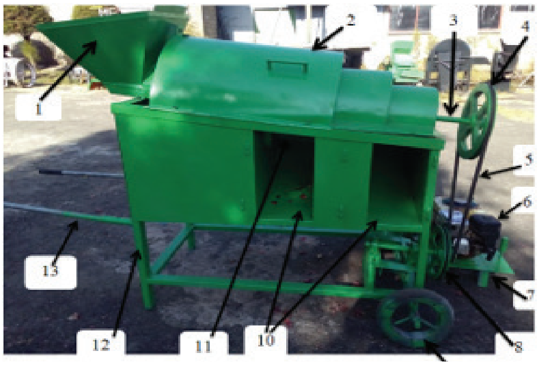
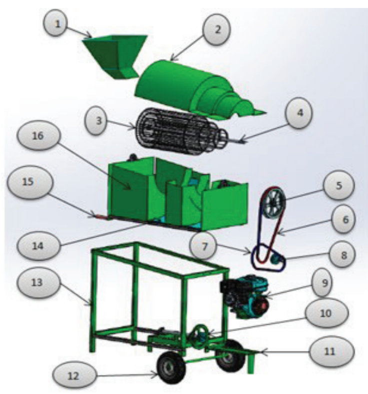



**Figure 1:** Pictorial view of the fabricated onion grading machine prototype.

1-Feeding rate; 2-Top cover; 3-Rotating shaft; 4-Pulley; 5-Belt; 6-Engine; 8-Speed reduction pulley; 9-Ground wheel; 10-Collection outlets; 11-Grading unit; 12-Mainframe; 13- Handle




| Item no | Part number |
|---|---|
| 1 | Feeding hopper |
| 2 | Top cover |
| 3 | Grading cylinders |
| 4 | Rotating shaft |
| 5 | Larger pulley |
| 6 | Belt-1 |
| 7 | Belt-2 |
| 8 | Smaller pulley |
| 9 | Engine |
| 10 | Speed reduction pulley |
| 11 | Engine set |
| 12 | Transportation wheel |
| 13 | Main-frame |
| 14 | Collection outlets |
| 15 | Handle |
| 16 | grading unit cover |

**Figure 2:** Exploded view of the various components of an onion grading machine.

**Main Frame**

The frame is the skeletal structure of the grader which forms the platform on which other components are mounted. The total weights carried by the frame are; the weight of the feed hopper, weight of the grading unit, weight of the collection tray, weight of the shaft, the weight of bearings, pulleys, and the belt. The two design factors considered in determining the material required for the frame were weight and strength. The machine frame was constructed from steel angles and rectangular pipe of 50×30×3 mm. Two rubber wheels were provided on one end to simplify the transportation of the machine. The frame and other parts of the grading machine were connected using appropriate sizes of bolts and nuts and the welding process.

**Design of Feeding Hopper**

The hopper was designed using a mild steel sheet with a 1.5mm thickness. The size of the hopper was determined on the basis of mass and volumetric capacity of the maximum feeding rate and average bulk density of ungraded onion bulbs. The inclination of the hopper was designed based on the angle of repose of onion bulbs (Ukey & Unde, 2010). A trapezoidal shape of the hopper was used for the machine due to consideration of the free flow of onion bulbs. The bottom of the hopper is slightly tilted downwards to enhance the flow of onion bulbs directly to the center of the rotating cylinder. The hopper has a sliding gate to control the feed rate. The following parameters were considered for the design of the hopper; Angle of repose = 46.30, Bulk density of onion bulbs = 549.7 kg/m$^3$. Then theoretically, it was assumed to hold at least 40 kg onion bulbs as the maximum feeding rate of the machine was 35 Kg/min and the volume of the hopper was calculated by the following formula according to Begum (2018).

$$V_h=1.1V_O \quad (1)$$

$$V_o=W_o/\rho_{bulk} \quad (2)$$

$$V_h=1.1((40\ kg)/(549.7\ kg/m^3))=0.073\ m^3$$

Where:

$V_h$ = volume of hopper, (m$^3$)

$V_o$= volume of onion bulbs, (m$^3$)

Wo= weight of onion bulbs, (kg)

ϱ_bulk= bulk density of onion bulbs, (kg)

Therefore, the required volume of the hopper is 0.073 m$^3$

Figure 3 show the shape of the hopper was designed in a trapezoidal shape and we can get the practical dimensions by assumptions. Hence, here also it was assumed that the hopper has a square in size with base width (a) =0.3 m, length of the hopper (b) =0.5 m, and height (h) =0.45 m, and its volume was calculated by Equation 3 (Charles and Etiese, 2020);

$$V_{dh}=1/2\ (a+b)\times h\times b \quad (3)$$

$$=1/2\ (0.3+0.5)\times 0.45\times 0.5=0.09\ m^3$$

Where:

$V_{dh}$ = the designed volume of the hopper, (m$^3$)

h = height of the hopper, (m)

a = base width (m)

b= Length of the hopper (m)

The designed volume of the hopper is 0.09 m$^3$ is greater than the theoretical volume (0.073 m$^3$) determined by equation 2. Therefore, the designed dimensions of the hopper are valid.

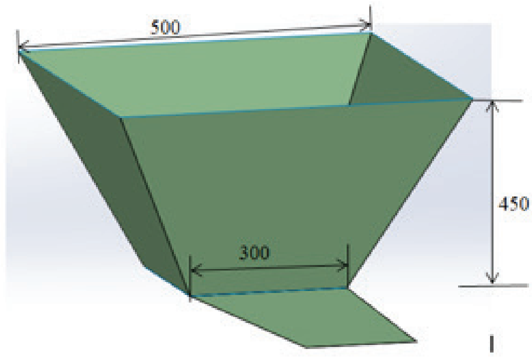



**Figure 3:** Isometric view of the hopper (all dimensions are in mm)

**Grading Unit**

The grading unit was designed using a mild steel rod, mild steel flat bar and mild steel shaft consisting of three cylindrical sieves having parallel bars cells. The rotating cylinders are arranged one inside the other according to the size (the larger spacing being internal, while the smaller one being external) and shown below in Figure 4. The ungraded onion bulbs are directed to the internal cylinder and the larger size bulbs are graded directly and collected at the end of the internal cylinder, while the medium and smaller size bulbs are dropped down to the second cylinder for further grading and the medium bulb size onion bulbs are left on this second cylinder collected at the end.

Finally, the bulbs with smaller sizes are going to the third sieve or cylinder and are collected at its end. The cylindrical sieves are fixed on the driveshaft and fixed with the machine frame. The volume of the rotating cylinders can be determined based on the flow of granular material through a circular shape. Many researchers suggested that the diameter of the rotating cylinder could be varied between 200 and 550 mm depending on the size range of the fruits and vegetables, volume, and desired capacity of the grading unit (Dereje, 2019; Charles and Etiese, 2020; Hegazy and Mady, 2018; Gunathilake *et al.*, 2016). For this design, the dimensions of the internal cylinder were 850 mm long and 250 mm in diameter with a spacing of 60 mm. The middle cylinder has 650 mm long and 350 mm in diameter with a spacing size of 40 mm, and the external cylinder has 450 mm in length and 450 mm in diameter with a spacing size of 20 mm.

The length of the rotating cylinders was decided by considering the previously developed rotating type graders and by assuming that the dimensions can provide enough path and time of retention for onion bulbs to be graded.

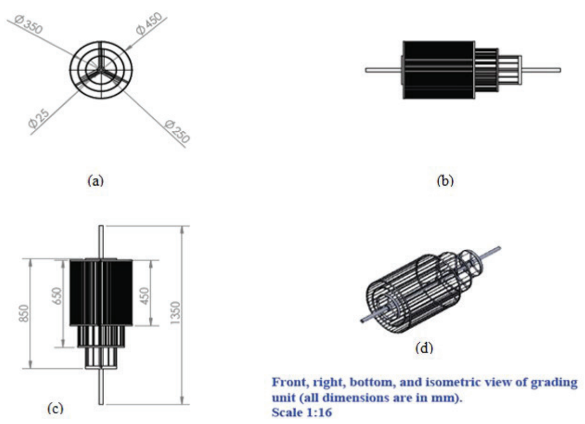



**Figure 4:** The grading unit: (a). Front; (b). Right; (c). Bottom; (d). Isometric view (all dimensions are in mm)

**Weight of Grading Unit**

The mass of the rotating cylindrical grading unit was determined from the volume and density of the mild steel rod and rolled mild steel flat bar materials used to manufacture the grading cylinder (Charles and Etiese, 2020). Based on previous work recommendations related to material selection to minimize bulb damage the mild steel rod having a diameter of 10 mm was used for all cylinders and arranged with different spacing and lengths as mentioned above. Also, the rolled flat bar with different diameters was welded with the steel rods and constitutes the overall mass of the rotating cylinders.

**Mass of Mild Steel Rod**

The mass of rods on the internal rotating cylinder was determined using Eq. (4) (Charles and Etiese, 2020).

$m=\varrho\times v$ (4)

Where:

m=Mass of mild steel rod (kg);

$\varrho$ = density of mild steel rod (kg/m$^3$)

= 7850 kg/m$^3$; and

v =volume of mild steel rod (m$^3$)

But,

$v=\pi r^2 h$ (5)

$v=\pi r^2 h=\pi(0.005)^2\times 0.85=0.000066758$ m$^3$

$m\_sr=\varrho\times v=7850\times 0.000066758=0.524$ kg

Where:

v = volume of the rotating cylinder (m$^3$)

r= Radius of the rod (m) = 0.005 m

h = length of the internal cylinder (m) = 0.85m

Also, the number of rods required for each cylinder was estimated by using equation (6)(Kankal, 2013);

$Ns=(\pi Drc)/S$ (6)

Where:

Ns = numbers of mild steel rods per segment

$D_{rc}$= Diameter of the internal rotating cylinder (0.25 m)

S = spacing between rod on the cylinder (0.06+0.01= 0.07)

Number of rods required for the cylinder, $N=(\pi(0.25))/0.07= 11$

Therefore, the mass of rods on the cylinder, m=11×0.524 kg=5.76 kg

By using the same procedure we have used above we can compute the mass of rods on the middle cylinder;

Where:

The density of the rod, $\varrho$=7850 kg/m$^3$,

The radius of a rod, r = 0.005 m,

Length of the medium cylinder, h = 0.65 m,

The diameter of the middle cylinder, $D_{mc}$=0.35 mm,

Spacing between rod on the cylinder, S=0.05 m, and

Number of rods required = 22

$m=22(\varrho\times v)=23(\varrho\times\pi r^2 h)$

$=22(7850\times\pi(0.005)^2\times 0.65)$

$=22\times 0.40075$ kg=8.813 kg

Mass of rods on the external cylinder,

Where: $\varrho$=7850 kg/m$^3$,

The radius of a rod, r = 0.005 m,

Length of the external cylinder, h = 0.45 m,

The diameter of the external cylinder, $D_{ec}$=0.45 mm,

Spacing between rod on the cylinder, S = 0.03 m, and

Number of rods required = 50

$m=47(\varrho\times v)=47(\varrho\times\pi r^2 h)$

$=47(7850\times\pi(0.005)^2\times 0.45)$

$=47\times 0.2774$ kg=13.04 kg

**Mass of Rolled Flat Bar**

According to Charles and Etiese, (2020) the mass of rolled flat bar used for each rotating cylinder was calculated;

$m_{fb}=\varrho\times v$ (7)

Where: $m_{fb}$= mass of rolled flat bar (kg),

$\varrho$ = density of the flat bar (kg/m$^3$)

=7850kg/m$^3$, and

v = volume of rolled flat bar (m$^3$).

But, $v=\pi r^2 h$

Where: v = volume of the rolled flat bar (m$^3$),

r= radius of the rolled flat bars used for each rotating cylinder (0.125 m, 0.175 m, and 0.225 m for the internal, middle, and external cylinder respectively), and

h = thickness of the flat metal bar (m)

= 3 mm=0.003 m

$v=\pi r^2 h$

$=2(\pi\times 0.125^2\times 0.003)+2(\pi\times 0.175^2\times 0.003)+2(\pi\times 0.225^2\times 0.003)$

$=0.001826$ m$^3$

$m_{fb}=\varrho\times v=(7850\text{ kg})/\text{m}^3\times 0.001826\text{ m}^3=14.33$ kg

Also, the mass of the bushing and round bar used to attach the cylinder with a rotating shaft is 1.467 kg. So, the total mass of the rotating cylinder ($m_{rc}$), was estimated as;

$m_{rc}=m_{sr}+m_{fb}=5.76+8.813+13.04+14.33+1.467$

$=43.41$ kg

And the weight (W) of the grading unit is also determined from Eqn. (8).

$W=m_{rc}\,g$ (8)

= 43.41 kg x 9.81m/s = 425.85 N

Where: W = weight of the grading unit (N),

$m_{rc}$ = mass of rotating grading cylinder (kg), and

g = acceleration due to gravity (9.81m/s$^2$).

**Theoretical Analysis of the Grading Unit**

To determine the power required to move onion bulbs through the rotating grading unit it is important to consider the process by which ungraded onion bulbs are delivered from one end of the inclined rotating cylinder and try to reach the other end of the cylinder. During this process, onion bulbs at the lower portion of the cylinder are lifted upward by the screen surface, after which they are again lifted along with it and slide down. They gradually move toward the opposite end of the cylinder. The bulbs are in contact with only a part of the cylindrical surface with no relative movement with respect to it during their lifting. The nature of the motion of onion bulbs over the surface of the cylindrical sieve depends upon the coefficient of friction on the given surface. The kinematics operating condition is governed by centripetal acceleration ($r\omega^2$), the initial conditions of motion of the bulbs, the point at which they are delivered onto the grading surface (grading cylinder), and their initial velocity. Depending upon the relationship between the above factors, the bulbs in the cylinder may slide along with it, separated from its surface, and perform a free flight or may move with the surface being at rest relative to it. In the last case, the bulbs are not graded (Bosoi *et al.* 1991). Neglecting the sliding motion of the onion bulbs inside the cylinder, the release of onion bulbs through the cylindrical grader depends upon the relative velocity and the forces acting on each bulb.

The weight of onion bulbs is directed downward and the centrifugal force is given below (Abd El-Tawwab *et al.*, 2012).

$W_o = mg$ (9)

$Fc = m\, r\, \omega^2$ (10)

Where:

$W_o$ = weight of the onion bulbs (N),

m = mass of the onion bulbs (kg),

$F_c$ = the centrifugal force (N),

g = acceleration due to gravity ($9.81 m/s^2$),

r = the radius of the internal cylinder to which ungraded onion bulbs were applied first (0.125 m),

ω = the angular speed of the rotating cylinder ($s^{-1}$).

The motion of the bulb on the cylinder surface is not determined by the tangential forces alone. But, if the resultant of the normal forces ($N_f$) is not directed towards the cylinder surface, the bulb will lose contact with the cylinder.

To find the equation that describes the motion of the onion bulbs through the rotating cylinder (Figure 5), it can be found:

$N_f = mr\omega^2 + mg\cos\dot{\alpha}$…… in the normal direction (at $\bar{y}\ \bar{y}$) (11)

$mg \sin\dot{\alpha} = \mu N_f$ ………… in the tangential direction (at $\dot{x}\ \dot{x}$) (12)

Where:

$N_f$ = normal force (N),

$\dot{\alpha}$ = is the angular position of the onion bulb on the cylinder surface measured from the horizontal axis in the direction of rotation (degree),

μ= coefficient of friction between the onion bulb and the cylindrical surface,

By substituting ($N_f$) from equation (11) into equation (12) according to Abd and Magda (2011) then:

$mg \sin\dot{\alpha} = \mu(mr\omega^2 + mg\cos\dot{\alpha})$ (13)

$(mg \sin\dot{\alpha})/\mu = m(r\omega^2 + mg\cos\dot{\alpha})$ (14)

$\mu r\omega^2 = g\sin\dot{\alpha} - \mu\, g\cos\dot{\alpha}$ (15)

$\omega^2 = (g\sin\dot{\alpha} - \mu\, g\cos\dot{\alpha})/\mu r$ (16)

$\omega^2 = g/\mu r\, (\sin\dot{\alpha} - \mu\cos\dot{\alpha})$ (17)

$\omega = \sqrt{(g/\mu r\, (\sin\dot{\alpha} - \mu\cos\dot{\alpha})}$ (18)

But, $\omega = (2\pi N^c)/60$ (19)

Where ($N_c$) is the number of revolutions of the rotating cylinder. From equation (18) and equation (17) we get:

$N_c = 60/2\pi\, \sqrt{(g/\mu r\, (\sin\dot{\alpha} - \mu\cos\dot{\alpha})}$ (20)

At, $\alpha' = 90^0$

$N_c = 60/2\pi\, \sqrt{(g/r\mu)}$ or $N_c = 30/\pi\, \sqrt{(g/r\mu)}$ (21)

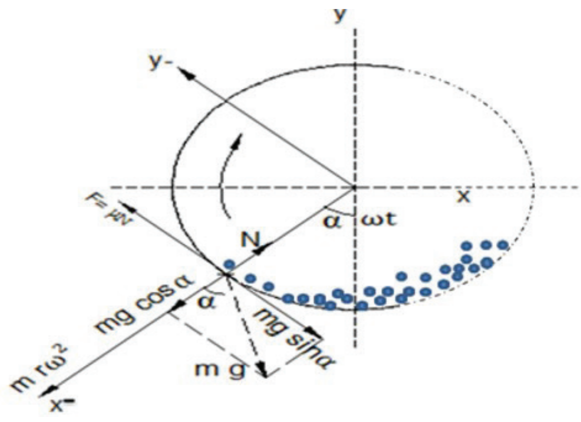



**Figure 5:** The forces acting on the onion bulbs situated on the internal cylinder surface.

The behavior of an onion bulb depends on the cylinder's angular speed at a given radius with other factors such as onion bulb layers and internal friction that affect the performance of a grading machine. Let us determine the above factors as a certain value of (K) that ranged from 0.33 to 0.40 according to Abd El-Tawwab *et al.*, (2012). Then equation (21) may be equal to;

$N_c = 30/\pi\, K\sqrt{(g/\mu r)}$ (22)

Given that, $g = 9.81\ m/s^2$, r = 0.125 m, μ = 0.29 (the measured static friction angle was 16.25 degree), and let k=0.40 (the maximum value from above range) and by substituting in equation (22).

$$N_c = \frac{30}{\pi} K \sqrt{\frac{g}{\mu r}} = \frac{30}{\pi}(0.4)\sqrt{\frac{9.81}{0.289 \times 0.125}}$$

$$= 62.95 \text{ rpm}$$

The number of rotations per minute ($N_c$) found above is the critical speed at which falling ends and onion bulb remains in constant grading cylinder contact, held there by centrifugal force. The rotating cylinders are set inclined to the horizontal plane, to improve the internal pressure forces during the rotation of the mass of the onion bulbs. In this study, the slope angle of the rotating cylinders on the horizontal plane (6°) was selected according to Charles & Etiese (2020).

Also, the normal force, torque, angular speed, and power required to move onion bulbs in rotating cylindrical surfaces were found by using Eqns. (11), (23), (24), and (25) (Khurmi and Gupta, 2013).

$T=N_f\ r$ (23)

$\omega=(2\pi N^c)/60$ (24)

$=(2\pi\times62.95)/60=6.60$ rad⁄s , so

$T=541.03\times0.125=67.63$ N.m

$P_O=T\omega$ (25)

$=67.63\times6.6=446.35$ Watts

Where:- $P_O$ = power required to move the onion bulbs through the rotating cylinder (Watts),

T = torque required to drive grading cylinder (Nm),

ω = angular velocity (rad/s),

$N_c$ = rotational speed of the grading cylinder (62.95 rpm), and

r = radius of grading cylinder (0.125 m).

**Rotational Torque of the Rotating Cylinder**

The torque developed by a rotational body was estimated by multiplying the Force (Fc) causing the motion by the radius of rotation (r) of the rotating cylinder (Hannah and Stephens, 2006).

$T=F_c\times r$ (26)

Where:

$F_c$ = centrifugal force required to produce acceleration (N) and r = radius of rotation (radius of the rotating cylinder). But,

$F_c=ma=m\omega^2 r$ (27)

Where:

m = mass of the rotating cylinder (kg),

ω = angular speed (rad/sec),

$N_c$ = rotational speed of the rotating cylinder (62.95 rpm) based on the value from equation (21), and

r = radius of rotational cylinder = 0.125 m.

$\omega=(2\pi N_c)/60=2\pi\times62.95/60=6.60$ rad/sec

So, $F_c=m\omega^2\ r=43.41\times(6.60)^2\times0.125=235.80$ N

The torque on the rotating cylinder was;

$T=F_c\times r=235.80\times0.125=29.48\ N_m$

The power required by the rotating cylinder was determined using the equation below as developed by Khurmi and Gupta, (2008).

$P_C=(2\pi N_c\ T)/60$ (28)

$=(2\pi\times62.95\times29.48)/60=194.30$ Watts

The power required to overcome air resistance was determined by Eqn. (29) (Sale *et al.*, 2016).

$P_{AR}=K_f\times F_r\times V_t^2$ (29)

Where:

$P_{AR}$= power required due to air resistance, (Watts),

$K_f$= constant which is equal to 0.06 (Sale *et al.*, 2016),

$F_r$= feed rate, 35kg/min = 0.5833 kg/s,

$V_t^2$= peripheral or tangential velocity of the grading mechanism (m/s)

$N_c$ = rotational speed of the grading cylinder (62.95 rpm), and

r = radius of cylinder (0.125mm).

$V_t=N_c\times\pi/30\times r$ (30)

$V_t=62.95\times\pi/30\times0.125$ m

$V_t$= 0.83 m/s

Then, the power requirement due to air resistance $P_{AR}$ will be;

$P_{AR}=0.06\times0.5833\text{kg/s}\times(0.83\text{ m/s})^2$

$P_{AR}$= 0.0238 kW = 23.80 Watts

**Power Transmission and Drive System**

**Selection of Pulley and Speed Reduction**

The pulleys used in the drive system were groove-type pulleys made of cast iron. Pulley diameters were selected based on the need to reduce the engine speed to the required one. The power transmission system was designed to reduce the engine output shaft speed from 1400 rpm to 87.5 rpm of the grading unit rotating shaft. During the test, the engine throttle valve was used to adjust the rotating speed to an optimum level. The power reduction was designed in

2 stages and finally, the pulleys shown in Figure 6 were selected. Two-speed reduction stages are given below: The speed ratio was calculated by using Eqn. (46) (Khurmi and Gupta, 2005).

Speed ratio= $N_1/N_2=D_2/D_1$ (31)

Where:

$D_1$= diameter of the driving pulley on the engine shaft(m),

$D_2$= diameter of the driven pulley on the speed reduction shaft(m),

$N_1$= speed of the engine pulley (1400 rpm), and

$N_2$= speed of the driven pulley on the speed reduction shaft (rpm)

**First Stage Speed Reduction**

In the first stage of reduction, the driving and driven pulley has been selected. Assume the diameter of the driving pulley with a speed ratio of 3.33: 1. From Eqn. (31) the diameter and speed of the driven pulley were computed as follows.

$D_2/D_1=3.33=D_2/0.075=\ D_2=0.249$ m, an available pulley of 250 mm diameter was selected. So, the speed of the counter pulley on the speed reduction system was calculated using the following formula (Khurmi & Gupta, 2005).

$N_1/N_2=D_2/D_1=N_1\ D_1=N_2\ D_2,\ N_2=N_{1*}D_1/D_2=(1400*0.075)/0.25=420$ rpm, hence, in the first reduction stage engine speed was reduced from 1400 rpm to 420 rpm.

**Second Stage Speed Reduction**

In the second stage, speed was reduced by using a small size pulley mounted on the freely rotating shaft with a speed ratio of 4.8:1. Then, the diameter and speed of the pulley mounted on the rotating grading cylinder shaft were calculated as follows (Khurmi and Gupta, 2005).

$D_4/D_3=4.8=D_4/0.075=D_4=0.36$ m = 360 mm.

Also, the speed of the driven pulley was calculated as follows.

$N_3/N_4=D_4/D_3=N_3\ D_3=N_4\ D_4,N_4=N_{3*}D_3/D_4=(420*0.075)/0.36=87.5$ rpm, In the second stage of speed reduction speed was reduced from 420 rpm to 87.5

rpm.
Where:
$D_3$= diameter of the driving pulley on the speed reduction shaft (m),
$D_4$= diameter of the driven pulley on the grading unit rotating shaft (m),
$N_3$= speed of the driving pulley on the speed reduction shaft (420 rpm), and
$N_4$= speed of the driven pulley on the grading unit rotating shaft (rpm).

Therefore, in power transmission, engine speed was reduced by two stages from 1400 rpm to 87.5 rpm for grading operation. Further material selection and design calculation related to the driving pulley on the speed reduction shaft and driven pulley mounted on the grading unit rotating shaft used for the grader power transmission system was computed as follows.

The width of the pulley (face width) "B" is usually considered to be 25% greater than the width of belt (b) and Eqn. (32) (Richard and Nisbett, 2011) can be used to estimate the width of the pulley. According to IS:2494-1974 top width of an "A" type v-belt for a 75 mm pulley diameter is 13 mm. So,

$B=b+0.25b=1.25b$
$=13+0.25\times13=16.25$ mm

The thickness (t) of the driving and driven pulley rim for a single v-belt can be determined using Eqn. (33 or 34) (Richard and Nisbett, 2011).

$$t_1=D_3/300+2\text{mm} \quad (33)$$
$=75/300+2\text{mm}=2.25$ mm$=0.00225$ m
$$t_2=D_4/200+3\text{mm} \quad (34)$$
$=360/200+3\text{mm}=4.8$ mm$=0.0048$ m

Where:
B = width of pulley (mm),
b = width of belt (mm),
$t_1$ = rim thickness of the driving pulley on the speed reduction rotating shaft (mm),
$t_2$ = rim thickness of the driven pulley on the grading unit rotating shaft (mm),
$D_3$ = diameter of driving pulley on the speed reduction rotating shaft (m) = 0.075 mm, and
$D_4$ = diameter of driven pulley on the grading unit rotating shaft (m) = 0.360 mm

The weight of the pulley was computed using the following equations (Thet *et al*., 2019).

$$W_p=m\times g \quad (35)$$
$$m=\varrho\times V \quad (36)$$
$$V=A_p\times t \quad (37)$$

Where:
$W_p$= Weight of the pulley (N),
m = mass of pulley (kg),
g = acceleration due to gravity (9.81 $m/s^2$),
$\varrho$ = density of the cast Iron pulley ($kg/m^3$) = 7,200 $kg/m^3$,
v = volume of the flat bar ($m^3$),
$A_p$= Area of pulley material ($m^2$) = ($\pi d^2/4$),
d= Diameter of pulley (m), and
t= rim thickness of the pulley (m).

So, for a driving pulley;
$V=(\pi d^2/4)\times t=(\pi\times0.075^2)/4\times0.00225=9.94\times10^{-6}\ m^3$
$m=\varrho\times V=7200\times9.94\times10$^(-6)$=0.0716$ kg
$W_p=m\times g=0.0716\times9.81=0.7021$ N
Also for the driven pulley, we have;
$V=(\pi d^2/4)\times t=(\pi\times0.36^2)/4\times0.0048=4.88\times10^{-4}\ m^3$
$m=\varrho\times V=7200\times4.88\times10^{(-4)}=3.5$ kg
$W_p=m\times g=3.5\times9.81=34.5$ N

The velocity ratio between two pulleys transmitting torque is given as:

$$\omega_3/\omega_4=N_3/N_4=D_4/D_3 \quad (38)$$

Where:
$\omega_3$= angular velocity of the driving pulley,
$\omega_4$ = angular velocity of the driven pulley,
$N_3$= rpm of a driving pulley (420 rpm),
$N_4$ = rpm of the driven pulley,
$D_3$ = diameter of a driving pulley (75 mm), and
$D_4$ = diameter of driven pulley (360 mm).
$N_3/N_4=D_4/D_3$ ,$N_4=(N_3\times D_3)/D_4$ = 87.5 rpm
$\omega_3/\omega_4=N_3/N_4$ , $\omega_3=(2\pi N_3)/60=(2\times\pi\times420)/60=43.98$ rad/s
$\omega_4=(\omega_3\times N_4)/N_3=(43.98\times87.5)/420=9.16$ rad/s

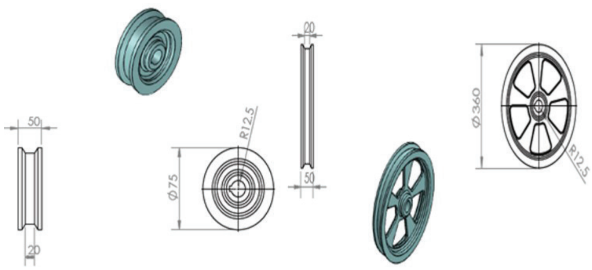



**Figure 6:** Detail dimension and isometric view of smaller and larger pulley.

### Selection of the Belt

Classical V-belt and groove-type pulley arrangements were used in this work to transmit the power required by the grading machine (Figure 7). The main reasons for using the v-belt drive are its flexibility, simplicity, and low maintenance costs. Additionally, the v- belt can absorb shocks thereby mitigating the effect of vibratory forces (Khurmi and Gupta, 2005).

### Determination of Belt Contact Angle

The belt contact angle($\varphi$) is given by the following equation(Khurmi and Gupta, 2005).

$$\varphi=\sin^{-1}((D_4-D_3)/2x) \quad (39)$$

Where: $\varphi$ = belt contact angle (rad),
$D_3$ = diameter of driving pulley on the speed reduction rotating shaft (m) = 0.075 m,
$D_4$ = diameter of driven pulley on the grading unit rotating shaft (m) = 0.36 m, and
x = distance between the pulleys from center to center.

The distance between the pulleys from center to center was determined by using Eqn. (40) (Khurmi and Gupta, 2005).

$$x=((D_4+D_3)/2)+D_3 \quad (40)$$

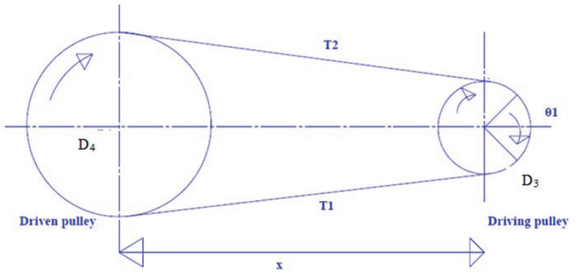



**Figure 7:** Geometry of V-belt drive

$=((0.36+0.075)/2)+0.075=0.2925$ m

Hence, the belt contact angle (φ) is computed as;

$\varphi=\sin^{-1}(D_4-D_3)/2x)$

$=\sin^{-1}(0.36-0.075)/(2\times0.2925)=29.15^0=29.15\times\pi/180=0.5$ rad

The wrap angle on the smaller pulley is determined using Eqn. (56) while that of the larger pulley was determined by using Eqn. (41) (Akintunde *et al.*, 2005).

$\theta_1=180-2\sin^{-1}(D_4-D_3)/2x)$ (41)

$\theta_1=180-2\sin^{-1}(0.36-0.075)/(2\times0.2925))=121.69^o$

$=121.69\times\pi/180=2.12$ rad

$\theta_2=180+2\sin^{-1})(D_4-D_3)/2x)$ (42)

$\theta_2=180+2\sin^{-1}(0.36-0.075)/(2\times0.2925))=238.3^o$

$=238.3\times\pi/180=4.16$ rad

Where: $\theta_1$= angle of wrap of a belt on driving pulley (rad) and

$\theta_2$ = angle of wrap of a belt on driven pulley (rad)

**Determination of Belt length**

The nominal pitch length of the belt from the speed reduction shaft to the grading cylinder was determined to know the actual belt size needed to transmit power the reduced power to the grading cylinder. Therefore, according to Khurmi and Gupta (2005)

the nominal pitch length (L) was determined using Eqn. (43).

$L=\pi/2\,(D_3+D_4)+2x+(D_4-D_3)^2/4x$ (43)

$L=\pi/2(0.075+0.36)+2\times0.2925+(0.36-0.075)^2/(4\times0.2925)=1.330$ m

Hence, according to IS: 2494-1974, the belt of the nearest standard pitch length (1331 mm) was selected.

Where:

$D_3$ = diameter of a driving pulley (mm),

$D_4$ = diameter of driven pulley (mm), and

x = center distance between the motor pulley and rotating cylinder shaft pulley (mm)

**Determination of Belt Tensions**

The driving pulley was considered for the calculation of tensions in the V-belt because the pulley of a smaller diameter governs the design. According to IS: 2494-1974 dimensions of "A" type v-belt for 75 mm pulley diameter is; top width, b = 0.013 m, thickness, t = 0.008 m, bottom width, w = 0.011 m, length of the belt, L=1.331 m, the density of rubber belt, $\varrho$=1140 kg/m$^3$, coefficient of friction, μ = 0.25 and the allowable stress, $\sigma_a$= 2.5 MPa (Khurmi and Gupta, 2005).

The peripheral velocity, V of the belt on the driving pulley is:

$V=r\times\omega_3$ (44)

$V=0.0375\times43.98=1.65$ m/s

Where:

V= the peripheral velocity of the belt on the driving pulley (m/s),

r = radius of the driving pulley, and

$\omega_3$= angular velocity of a belt on the speed reduction driving pulley (rad/s).

The mass of the belt, m was given by;

m=cross sectional area ×length×density (45)

$m=b\times t\times L\times\varrho$ (46)

$m=0.013\text{ m}\times0.008\text{ m}\times1.331\text{ m}\times1140\text{ kg/m}^3=0.1578$ kg

Where; m is the mass of the belt.

The centrifugal tension, $T_c$ between the contact pulley face and belt is given by;

$T_c=mv^2$ (47)

$=0.1578\text{ kg}\times(1.65\text{ m/s})^2=0.43$ N

The maximum allowable tension in the belt was determined as;

$T_{max}=\sigma_a\times A$ (48)

$=2.5\text{ N/mm}^2\times104\text{ mm}^2=260$ N

Therefore, the tension in the tight side of the belt, $T_1$ was computed as;

$T_1=T_{max}-T_c$ (49)

$T_1=(260-0.43)\text{N}=259.57$ N

Where:

$T_1$ = tension in the tight side (N),

$T_c$ and $T_{max}$ = the centrifugal and maximum tension of the belt (N) respectively,

A = cross-sectional area of a belt (mm$^2$),

m = mass per unit length of a belt (kg/m), and

σ = maximum allowable stress of belt (MPa).

Tension in the slack side of the belt, $T_2$ was calculated as follows (Khurmi and Gupta, 2005).

$\ln(T_1/T_2)=\mu\times\theta_1\times\text{Cosec}\beta$ (50)

Where:

$T_1$ = tension of the belt on the tight side (N),

$T_2$ = tension of the belt on the slack side (N),

μ = coefficient of friction between the belt and the pulley (0.25),

$\theta_1$ = lap angle on driving pulley (2.12 rad), and

β = groove angle (18o).

$\ln(T_1/T_2)=0.25\times2.12\times\text{Cosec}18$

$(259.57\text{ N})/T_2)=e^{1.715}$

$((259.57\text{ N})/T_2)=5.557$

$5.557\,T_2=25$

$T_2=46.77$ N

**Design of Grading unit Driving Shaft**

The shaft transmitting power under various operating and loading conditions can be designed based on the strength or rigidity principle (Richard and Nisbett, 2011). The shaft would be subjected to fluctuating torque and bending moments during operation under load conditions. Shock and fatigue factors were therefore taken into consideration. To determine the diameter of the grading cylinder shaft; all forces acting on the shaft (weight of grading cylinders, weight of maximum feeding rate of onion, tight side and slack side tensions of the belt, and weight of pulley) were considered. From Eqns. (8, 9, 35, 49, and 50), the weight of grading cylinders, the weight maximum feeding rate of onion bulbs, the weight of the pulley, and the tight side and slack side tensions of the belt were determined as 425.85 N, 343.35 N, 259.57 N, 46.77 N, and 34.5 N, respectively. Shear force and bending moment diagrams on the vertical and horizontal planes were determined by considering the inclination of the rotating grading unit shaft to the horizontal plane. For rotating shafts, where the load is suddenly applied (minor shock) the combined shock and fatigue factor applied to the bending moment is given as 1.20 to 2.00 while the combined shock and fatigue factor applied to the torsional moment is given as 1.00 to 1.50 (ASME, 1995). The diameter of the rotating shaft was determined by computing the force acting on the shaft in a vertical and horizontal direction as shown below in Figure 8.

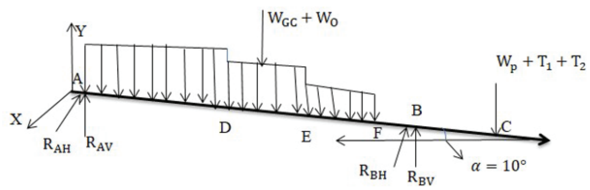



**Figure 8:** Free body diagram of all forces acting on the shaft.

Where:

$R_{AH}$ = horizontal bearing reaction force at A (N),
$R_{AV}$ = vertical bearing reaction force at A (N),
$R_{BH}$ = horizontal bearing reaction force at B (N),
$R_{BV}$ = vertical bearing reaction force at B (N),
$W_p$ = weight of pulley (N),
$W_{GC}$ = weight of grading cylinder (N),
$W_O$= weight of onion bulbs (N), and
$T_b$ = total belt tension (N).

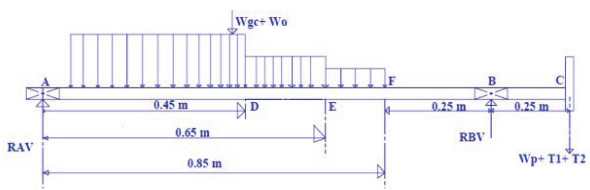



**Figure 9:** Free body diagram forces on the shaft on the vertical plane.

**Forces Acting on Grading unit Driving Shaft on Vertical (YZ) Plane**

Analysis of vertical forces acting on the shaft as shown in Figure 9below.

The reactions, R_AV and R_BV were determined by taking moment about A;

$\sum M_A=0$

$1.35\times340.8N_1.1\times R_{BV}+0.425\times168.67N+0.325\times162.6 N+0.225\times144.3\ N=0$

$1.1\times R_{BV}=1.35\times340.84N+0.425\times168.67N+0.325\times162.6 N+0.225\times144.3\ N$

$R_{BV}=617.13/1.1=561.03\ N$

Also, the summation of all vertical forces is;

$\sum F_V=0$

$R_{AV}$-167.75 N-161.71 N-143.51 N+$R_{BV}$-338.97 N=0

$R_{AV}+R_{BV}$= 167.75 N+161.71 N+143.51 N+338.97 N

$R_{AV}$+R_BV=811.94 N, but, $R_{BV}$=561.03 N

$R_{AV}$=(811.94 -561.03)N=250.91 N

The shear force and bending moment diagram on the main shaft due to loading is shown in Figure 10.

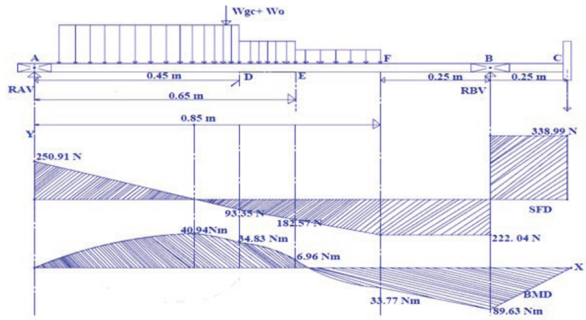



**Figure 10:** Shear force and bending moment diagram in the vertical plane.

**Forces Acting on Grading unit Driving shaft on Horizontal (XZ) Plane**

Analysis of horizontal forces acting on the shaft as shown in Figure 11.

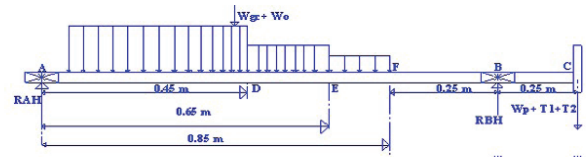



**Figure11:** Free body diagram forces on the shaft on the horizontal plane.

The reactions, $R_{AH}$ and $R_{BH}$ were determined by taking moment about A;

$\sum M_A=0$

$1.35\times340.84\ N-1.1\times R_{BH}+0.425\times197.34\ N+0.325\times208.58\ N+0.225\times173.61\ N=0$

$1.1\times R_{BH}=1.35\times340.84\ N+0.425\times197.34\ N+0.325\times208.58\ N+0.225\times173.61N$

$R_{BH}=650.85/1.1=591.68\ N$

Also, the summation of all horizontal forces is;

$\sum F_H=0R_{AH}$-20.74 N-26.15 N-33.52 N+$R_{BV}$-35.63 N=0

$R_{AH}+R_{BH}$= 20.74 N+26.15 N+33.52 N+35.63 N

$R_{AH}+R_{BH}$=116.04 N, but, R_BH=591.68 N

$R_{AH}$=(116.04 -591.68)N=475.64 N

Downward

Based on the magnitude and location of all horizontal forces acting on the shaft of the rotating grading unit, the shear force and bending moment diagram on the horizontal plane (Figure 12) was drawn.

The resultant bending moments on the shaft were determined using Eqn. (51).

$$M_b=\sqrt{((M_V)^2+(M_H)^2} \quad (51)$$

$=\sqrt{((89.63)^2+(8.91)^2} =90.07\ N_m$

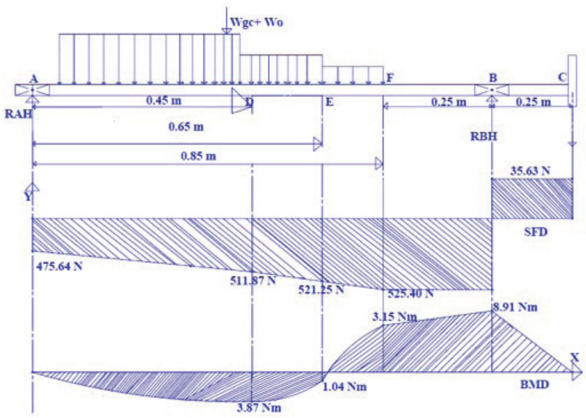



Figure 12: Shear force and bending moment diagram in the horizontal plane.

Where:

$M_b$ = resultant bending moments ($N_m$),

$M_V$ = maximum bending moments on a vertical plane ((89.63) $N_m$), and

MH = maximum bending moments on a horizontal plane (8.91 Nm)

According to Khurmi and Gupta (2005), torque on a shaft can be calculated using Eqn. (52).

$T=(T_1-T_2)\times D_4/2$ (52)

$=(259.57\ N-46.77N)\times 0.36/2=38.30\ Nm$

The power required to rotate the grading cylinder shaft will be determined from the equation below.

$P_{SH}=2\pi NT/60$ (53)

$P_{SH}=(2\pi\times 62.5\times 38.30)/60=252.48$ Watts

Where: $P_{SH}$= power required to rotate the grading unit shaft (Watts)

T = torque on the shaft (Nm),

$T_1$ = tight side tension of a belt (259.57 N),

$T_2$ = slack side tension of a belt (46.77 N),

$D_4$ = diameter of the driven pulley on the grading unit rotating shaft (0.36 m), and

N= rotational speed of rotating shaft (62.95 rpm).

For a solid shaft with little or no axial load, the diameter of the shaft that can withstand the applied loads can be determined using Eqn. (53) (ASME, 1995; Khurmi and Gupta, 2005).

$d_3=16/\pi\tau\ \sqrt{((K_b\times M_b)^2+(Kt\times Mt)^2}$ (53)

$=16/(\pi\times 45)\quad \sqrt{((1.3\times 90.07\times 1000)^2+(1.2\times 38.30\times 1000)^2}$

$d_3=14{,}236.29\ mm^3$

d=24.24 mm

Therefore, the diameter of the driving shaft was taken as 25 mm.

Where:

d = diameter of the shaft (mm),

$M_b$ = bending moment (90.07Nm),

$M_t$ = torsional moment (38.30 Nm), and the values of combined shock and fatigue factor applied to bending moment($K_b$) and combined shock and fatigue factors applied to the torsional moment($K_t$) were taken as 1.3 and 1.2 respectively for the gradually applied load on the rotating shaft and the allowable shear stress of the shaft ($\tau$) as 45 N/mm² based on the ASME code.

The amount of twist permissible depends on the type of load application and varies between 2.5 to 3.00 degrees per meter for line shafting. Therefore, the angle of twist (for the solid shaft) can be determined using Eqn. (54) Khurmi and Gupta, (2005).

$\theta=(T\times L)/(J\times G)$ (54)

$=(38.30\times 1.35)/(3.8\times 10^{-8}\times 80\times 10^9)=$ 0.01685 rad or 0.96o

The maximum twist on the shaft was found to be 0.96o/m. The calculated angle of twist was less than the permissible angle of twist (3°/m). So, the shaft design based on the torsional rigidity is safe.

Where:

θ= torsional deflection, or angle of twist ( radians or o/m),

L = length of the shaft (1.35 m),

J = polar moment of inertia for a cross-sectional area about the axis of rotation (3.8×10-8 m⁴),

T = torsional moment (38.30 Nm),

G = torsional modulus rigidity (80 x 109 N/m²), and

d = diameter of the shaft (0.025 m).

2.3.6. Bearing selection

A bearing is a machine element that supports another moving machine element (known as a journal). It permits a relative motion between the contact surfaces of the members while carrying the load (Khurmi and Gupta, 2005). Bearing size can be selected by determining the maximum resultant force on it, bore size, and desire maximum lifespan.

$R=(\sqrt{(R_{BH}^2+R_{BV}^2}\,)$ (55)

$=(\sqrt{(591.68)^2+(561.03)^2}$

=0.82 kN

The design load of a bearing can be calculated using the following formula (Robert *et al.*, 2018).

$P_d=VR$ (72)

$=1\times 0.82=0.82$ kN

Where: $P_d$= load of the bearing (N), V= rotation factor 1.0 if the inner race of the bearing rotates and 1.2 if the outer race rotates, and R = radial load (N)

For a specified design life in hours and a known speed of rotation in rpm, the number of design revolutions, L_d for the bearing would be estimated using Eqn. (56) (Robert *et al.*, 2018).

$L_d = H_m\times N\times(60\ min)/hr$ (56)

$=(6{,}000\ hr)\times(62.95\ rpm)(60min/hr)=2.3\times 10^7$ rev.

Dynamic load rate can be determined by using Eqn. (57) (Richard and Kelth, 2011).

$C=P_d\,(L_d/10^6)^{(1/k)}$ (57)

=0.82((2.3×10^7)/10^6 )^(1/3) =1.08 kN

Where:

C= basic dynamic load rating,

RBH = maximum resultant force acting on a bearing at point B on the horizontal plane (kN), $R_{BV}$ = maximum resultant force acting on a bearing at point B on the vertical plane (kN),

R = maximum resultant reaction force on the bearings at point B (kN),

$N$ = rotational speed of the rotating the shaft (rpm),
$L_d$= the number of design revolutions,
$H_m$ = according to Robert, (2018), the desired maximum life span of the bearings for agricultural equipment was 6,000 hours, and
K= 3 for ball bearings and 10/3 for roller bearings.
Since the diameter of the shaft has been determined to be 25 mm, the bearing number 205 by the bore of 25 mm with a width of 15 mm was selected according to Khurmi & Gupta, (2005)(Figure 13). Consequently, the basic dynamic load rating of the bearing was estimated using Eqn. (57) and found to be 1.08 kN. Since the basic dynamic load rating of the bearing selected is less than the dynamic load rating by the manufacturers (i.e. 1,080N < 11000 N), the bearing selected was safe.

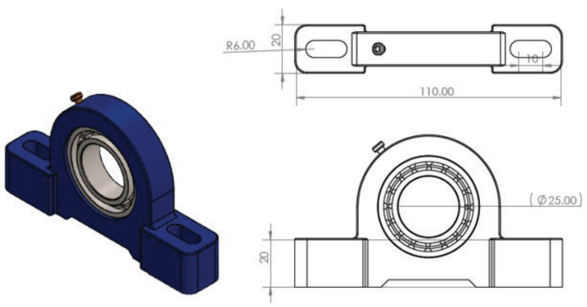



**Figure 13:** Isometric view and detail dimensions of bearing.

**Estimation of Power Required to Operate the Grading Machine**

The power required to operate the grading machine is the total sum of the power required to move the onion bulb inside the cylinder ($P_O$), the power required to drive the grading unit (rotating cylinder) (Pc), the power required to overcome air resistance ($P_{AR}$), and the power required to rotate the grading unit shaft ($P_{SH}$), based on equations (25), (28), (29), and (53). The total power ($P_{TO}$) required for grading and overcoming friction was determined using Eqn.58 (Nduka *et al.*, 2012).

$$P_T=P_O+P_C+P_{AR}+P_{SH} \quad (58)$$

$$=446.35+194.30+23.08+252.48=916.21 \text{ Watts}$$

$$P_{TO}=P_T+0.10P_T \quad \text{(10\% is possible power loss due to friction drive)} \quad (59)$$

$$=916.21+0.10(916.21)= 1007.83 \text{ Watts}$$

$$= 1.35 \text{ hp}$$

Where:
$P_T$ = power required by the machine (Watts),
$P_{TO}$= Total power required for grading machine and overcome loss due to friction, (watts)
$P_C$ = power required to rotate grading cylinder (Watts),
$P_{SH}$= power required rotating the grading cylinder shaft (Watts)
$P_O$= power required to move the onion bulb inside the cylinder (Watts), and
$P_{AR}$ = power required to over-come air resistance (Watts).
Therefore, a 3 hp Honda diesel engine with the desired crank speed of 1400 rpm was selected to drive the grading machine during the performance evaluation of the prototype.

**Manufacturing Process of the Machine Components Constructional Materials**

The selection of a proper material for a machine part or structural member is one of the most important decisions the designer is called on. The main factors considered while selecting materials were the availability of the materials, suitability of the materials for the working conditions in service, and the cost of the materials. The important properties, which determine the utility of the material, are physical, chemical, and mechanical properties (Khurmi and Gupta, 2005).
This chapter will cover a detailed description of the manufacturing process of each machine part of the grading machine Manufacturing can be defined in two ways; technologically and economically. Manufacturing from the technology point of view can be defined as the application of physical and/or chemical processes to alter the geometry, properties, and/or appearance of a given starting material to make parts or products. Then these multiple parts may be assembled to make finished products. Economically manufacturing is the transformation of materials into items of greater value by means of one or more processing and /or assembly operations, i.e. manufacturing adds value to a starting material by changing its shape or properties, or by combining it with other materials that have been similarly altered. The material has been made more valuable through the manufacturing operations performed on it (Groover, 2020). The step of the manufacturing process helps to produce the machine locally using available material. Required tools and machines are used as per necessity. The processes to accomplish manufacturing of the machine involve a combination of machinery, tools, power, and labor to give the finished part as summarized in Table 1below.
Machines used in the manufacturing of the grader components are:

1. Lath machines
2. Welding machine
3. Bending machine
4. Manual rolling machine
5. Grinding machine
6. Hand drilling machines

Measuring, cutting, holding and other tools used:

1. Turning tool and drill bits
2. Measuring tape and Caliper
3. Steel rule, try square, screwdrivers
4. Grinder discs and cutter discs
5. Metal sand paper, Scriber (pencil), and Center punch
6. Vice gripper, bench vice, and c-clamp
7. Hammer (plastic and metal), compass, and protractor
8. Paints and paint brushes.

The above parts of an onion grading machine are those that can be manufactured with the tools and machines available at Asella Agricultural Engineering Research

**Table 1:** Selected manufacturing process of the grading machine's main components

| Serial No | Grader Components | Required material | Required tools and activity/process |
|---|---|---|---|
| 1 | Handle | Galvanized Iron pipe | measuring, cutting, rolling, welding, grinding, drilling, and painting |
| 2 | Top cover | Mild steel sheet metal | measuring, cutting, rolling, welding, grinding, drilling, and painting |
| 3 | Frame | Mild steel angles and rectangular pipe | measuring, cutting, drilling, welding, and grinding |
| 4 | Driving Shaft | Mild Steel | turning, facing, cutting, and milling |
| 5 | Hopper | Mild steel sheet metal | measuring, cutting, bending, welding, grinding, drilling, and painting |
| 6 | Rotating Cylinder | Mild steel flat Iron and round bar | measuring, cutting, rolling, welding, grinding, and painting |
| 7 | Outlet chutes | Mild steel sheet metal | measuring, cutting, bending, and welding |

Center (AAERC) workshops and tools that can be purchased easily from local markets. However, bearings, belts, and pulleys are directly purchased from the market based on the design requirements and dimensions.

## CONCLUSION AND RECOMMENDATION

By utilizing materials found on the local market, an onion grading machine's design and construction were completed. The machine, which has wheels for transporting it from one location to another, weights 125 kg. The developed grading device had a 1488.43 kg/hr grading capacity and a 96.04% grading effectiveness. By adding additional feeding mechanisms and using manufacturing materials appropriate for grading other crop types, the design can still be enhanced to increase the capacity and efficiency of a grading operation. In areas with access to electricity, using a variable speed motor for straightforward control of the grading unit cylinder speed is recommended. It is advised to investigate the possibility of enhancing the design and fabrication in order to put the product into mass production, which can also open up the possibility of income generation.

## REFERENCES


Abd El-Tawwab, I.M., Owies, T.R. and Kaddour, U.A., (2012). Design and performance evaluation of multi-germ beet-seeds grading machine. *Journal of Soil Sciences and Agricultural Engineering, 3*(4), 519-536.

Adgo, T.M., (2008). Farmers' evaluation and adoption of improved onion production package in Fogera District, South Gondar, Ethiopia. MSc thesis (Agricultural Communication and Innovation), Haramaya University, Haramaya, Ethiopia.

Agidew A., (2018). Review on onion value chain analysis in Ethiopia. *Nutrition and food science international journal, 6*(5), 555698.

Aher, D., Dabhade, A., Desai, P., Waghmare, G., and Suryawashi, P. K., (2019). *The Onion Grading Machine*, 624–631.

Akintunde, B. O., Oyawale, F. A., and Tunde-Akintunde, T.Y., (2005). Design and fabrication of a cassava peeling machine. *Nigerian Food Journal,* (23), 1-8.

Ashraf, M., Muhammad, S.S., Manzoor, A., and Muhammad, Y., (2007). Design, development, and performance evaluation of fruit and vegetable grader. *Pakistan Journal of Agricultural Science., 44*(4).

ASME., (1995). American Society of Mechanical Engineers, "Design of Transmission Shafting", ASME paper No. B106.1M.

Begum, F., (2018). Development and Testing of a Gherkin Grader. Master of Technology in Agricultural Engineering (Processing And Food Engineering), Acharya NG Ranga Agricultural University), 79.

Bosoi, E.S., Verniaev, O.V., Smirnov, I.I., and Sultan-Shakh, E.G., (1991). Theory construction and calculations of agricultural machines, *Mashinostroenie Publishers*, 2.

Bossie, M., Tilahun, K. and Hordofa, T., (2009). Crop coefficient and evapotranspiration of onion at Awash Melkassa, Central Rift Valley of Ethiopia. *Irrigation and Drainage Systems, 23*(1), 1-10.

Brewster, J.L., (2008). Physiology of crop growth, development, and yield. Onions and other vegetable alliums, (Ed. 2), 85-170.

Charles, K. D., and Etiese, I., (2020). Development and performance evaluation of manually operated onion grading machine. *Journal of Agriculture and Food Research*, 3(2), 100070.

Dereje A.A., (2019). Design, Manufacturing, and Performance Evaluation of Potato Grading Machine. MSc thesis (Agricultural Machinery Engineering), Haramaya University, Haramaya, *Ethiopia.*

FAO, (2012). Major Food and Agricultural Commodities and Producers -Countries by Commodity". Food and Agriculture Organization of the United Nations Www.fao.org.

Gebru, H., (2015). A review of the comparative advantages of intercropping to the mono-cropping system. *Journal of Biology, Agriculture and Healthcare, 5*(9), 1-13.

Groover, M.P., (2020). Fundamentals of modern manufacturing: materials, processes, and systems. *John*

*Wiley & Sons.*
Gunathilake, D.M.C.C., Wasala, W.M.C.B. and Palipane, K.B., (2016). Design, development, and evaluation of a size grading machine for onion. *Procedia food science, 6*, 103-107.
Hannah, J., and Stephens, R.C., (2006). Mechanics of Machines (Advance Theory and Examples).New Delhi: Second Edition, Published by Vinod Vasishtha for Viva Books Private Limited, New Delhi 110002, India.
Hegazy, K.E.S. and Mady, M.A.A., (2018). Development and Performance Evaluation of Olive Grader. Misr *Journal of Agricultural Engineering, 35*(1), 19-38.
Kankal, U.S., (2013). Design and development of self-propelled weeder for field crops. *International Journal of Agricultural Engineering, 6*(2), 304-310.
Khurmi, R. S., and Gupta, J. K., (2005). A Textbook of Machine Design. S.I. Units. Eurasis Publishing House (PVT) Limited, New Delhi. (I).
hurmi, R.S., and Gupta, J.K., (2008). A Textbook of Machine Design, New Delhi.
Khurmi, R.S. and Gupta, J.K. (2013). Design of machine elements. Chand and Company Pvt. Ltd., New Delhi. 509-557.
Londhe, D.H., Nalawade, S.M., Pawar, G.S., Atkari, V.T. and Wandkar, S.V., (2013). Grader: A review of different methods of grading for fruits and vegetables. A*gricultural Engineering International: CIGR Journal, 15*(3), 217-230.
Mostafa, H.M. (2004). Developing an appropriate system for onion grading. M.Sc. thesis. Agricultural Engineering Department, At Moshtohor, Zagazig University\ Benha branch, Egypt.
Nduka, N.B., Okay, O.A. and Jonah, A.C., (2012). Design, fabrication, and evaluation of palm nut-pulp separator. *Journal of Emerging Trends in Engineering and Applied Sciences, 3*(1), 144-151.
Nigussie, A., Kuma, Y., Adisu, A., Alemu, T. and Desalegn, K., (2015). Onion production for income generation in small-scale irrigation users agropastoral households of Ethiopia. *Journal of Horticulture, 2*(3), 1-5.
Richard G.B., and Nisbiett J. K., (2011). Shigley's Mechanical Engineering Design, Mcgraw Hill 9th edition, ISBN: 978-0-07-352928-8.
Robert L. Mott, Edward M. Vavrek, Jyhwen Wang., (2018). Machine elements in mechanical design, Sixth Edition, Pearson, NY.
Sale, N.A., Muhammed, U.S., Dalha, I.B. and Idris, S.I., (2016). An improved IAR sorghum thresher. Agricultural Engineering International: *International Commission of Agricultural and Biosystems Engineering (CIGR) Journal, 18*(3), 119-126.
Thet, W.W., Kyaw, S. and Sein, M.M., (2019). Design of Threshing Cylinder for a Combine Paddy Harvester (KUBOTA). *Iconic Research and Engineering Journals*, 1701716.
Ukey, P.D., and Unde, P.A., (2010). Design and development of sapota fruit grader. *International Journal of Agricultural Engineering, 3*(1), 35-39.