



# APPLIED RESEARCH AND INNOVATION

ISSN: 2993-8988 (ONLINE)

VOLUME 1 ISSUE 2 (2023)

PUBLISHED BY  
E-PALLI PUBLISHERS, DELAWARE, USA



## Design, and Construction of Power-Operated Garlic (*Allium Sativum* L.) Bulbs Breaker for Ethiopian Garlic

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### Article Information

**Received:** September 20, 2023

**Accepted:** October 13, 2023

**Published:** October 16, 2023

### Keywords

*Construction, Design, Economics, Garlic, Garlic-Bulb Breaker*

### ABSTRACT

Garlic bulb breaking is the unit operation through which cloves are separated to facilitate additional processing. This research work was carried out to design, and manufacture a prototype garlic bulb breaker for a Holeta local garlic variety grown in Ethiopia. The machine consisted of a feed hopper, a garlic separator (two rollers), a fan, outlets for cloves and trash (husk, root, and stem), the main frame, and a drive system. These components and machine assembling were designed using Solidworks software (Version S 2019). The economics of the constructed garlic bulb-breaker such as material and fabrication costs and cost of operation were evaluated with results of 9264.8 ETB and 76.89 ETB/hr, respectively. The cost saved by constructing a garlic bulb breaker over manual garlic bulb breaking by the labor force for planting and processing purposes was 83.42 and 86.63%, respectively. Therefore, the machine could be designed, and constructed for the best cost saved by constructing a garlic bulb breaker over manual garlic bulb breaking.

### INTRODUCTION

Garlic (*Allium Sativum* L.) is an important root vegetable that can be used as a spice in meals and has historically been used as a remedy for various ailments in Ethiopia (Abe Tullo, 2022). Allium is the most cultivated plant species in the family Alliaceae (Mnayer *et al.*, 2014) with seven hundred species worldwide (Tepe *et al.*, 2005). Garlic is the most used root vegetable after onions in Ethiopia. It has been used in many communities in Ethiopia as a flavoring agent in food and as a medicinal plant for various ailments (Addis and Abebaw, 2017).

Garlic (*Allium sativum* L.) is a fragrant herb eaten around the world in powder form and conventional treatment for many ailments. It has been reported to have various biological properties including anticancer, antioxidant, antidiabetic, protective, anti-weathering, antibacterial, antifungal, and antihypertensive activities in traditional medicine (Batiha *et al.*, 2020). Garlic is one of the most important root vegetables, has a pungent taste, and is used as a seasoning and flavoring all over the world. Organosulfur compounds such as allicin and DADS are the main components responsible for their pungent effect and pungent aroma. Garlic is known for its use in food preparation, especially dry food for preservation and some soups, and can be used both fresh and dried (T'esfaye and Mengesha, 2015).

### Statement of the Problem

Garlic bulb breaking is the unit operation by which the garlic cloves are separated to facilitate further processing. The garlic planting material is garlic cloves. Garlic clove separates from the whole bulbs and smashes them with wooden sticks. Usually, the garlic cloves are separated by rubbing the garlic bulb between the palms, as opposed to sacking or pounding with a wooden stick. These

techniques are tedious, time-consuming, and often lead to hand injuries (Ibrahim, 2013). Breaking garlic bulbs requires special care and skill with the characteristic physical properties and the presence of volatile essential oils in the epidermal cells that give garlic its characteristic aroma. Better seed production improves both yield and quality to fetch a higher market price (Channabasamma *et al.*, 2016). Seeds are the basic and important input materials for agricultural production. But quality also plays an important role, since the yield of the crop is directly dependent on the appearance and formation of the seedlings. The quality of the clove, whether for garlic bulb production or general cultivation, depends on some factors that affect the vegetable value of the clove (Channabasamma, 2014).

In Ethiopia, garlic bulbs, broken into cloves, are used as a spice in meals and historically used as a remedy for various ailments. The traditional methods of garlic cloves separation from the garlic bulb usually lead to contamination of the product with chaff, small fragments of cloves, and injured cloves. Garlic cloves are required for planting, grading, and meal processing. However, for plantation and grading functions and the food processing industry, the cloves are separated via way of means of the hand and beaten with a wood stick. This lead to poor physical condition and the quality of garlic cloves becomes eminent. These techniques also can be very exhausting and time-consuming. Garlic bulb primary processing usually improves garlic cloves' condition and quality and the process is a vital and necessary link between production, storage, and distribution. As a result, farmers are compelled to do additional work of separating garlic cloves from whole garlic bulbs and chaff that was reduce the quality and the value of the product prior to planting, and subsequent processing.

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The farmers of Arsi and West Arsi Zones of Oromia Regional State requested Asella Agricultural Engineering Research Center for a machine to separate cloves from garlic bulbs for planting due to the tedious and time-consuming the traditional method of garlic separation from a compound garlic bulb by hand, which results in hand injuries. There is a machine designed and developed for separating cloves from a garlic bulb in India and China. However, Those machines were very expensive and, designed and constructed for garlic varieties available in India and China. Therefore, Those machines need modification and additional costs due to big differences in the engineering properties of Ethiopian garlic varieties from those countries' garlic varieties. Considering the problems mentioned above, designing and constructing a low-cost and effective garlic bulb-breaking machine from row material available at Asella Agricultural Engineering Research Center is very important.

Considering the bulk garlic requirement for growing, grading, and processing, the design, and construction of an efficient garlic breaker is crucial for its separating the garlic bulbs into cloves to fill this gap. However, there is no research work on the design and construction of the garlic bulb-breaking machine for Ethiopian garlic (*Allium sativum* L.). Therefore, this research project is proposed to design and construct power operated garlic bulb breaker for garlic varieties grown in the Holeta area, Ethiopia.

## MATERIALS AND METHODS

### Experimental Location

Machine design, and fabrication, were carried out at the Asella Agricultural Engineering Research Center (AAERC).

### Machine Description

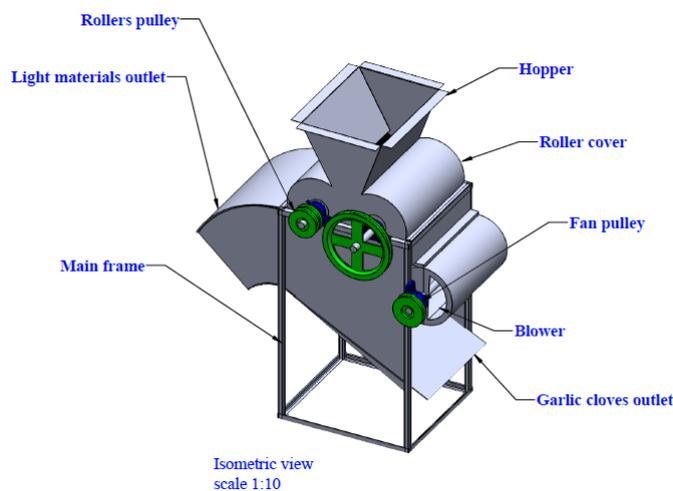
The separation of the garlic cloves from the whole bulbs

with such machines is accomplished by way of means of mixed shearing and impact forces (Channabasamma *et al.*, 2016). Based on this principle, a garlic clover machine was constructed. The machine consisted of, a feeding hopper, a double roller separating the cloves, a fan, an outlet to separate the garlic cloves and waste (shell, root, and stem parts), a power source, and a drive system (belt and pulley). A chute was provided above the inlet opening of the housing of the rollers for feeding the garlic bulbs. The rollers, which were covered with rubber, has a 208 mm diameter and 402 mm length and were rotating in opposite directions to achieve a shearing action on the bulbs to get the cloves.

A top surface of the rubber padding was applied to prevent any damage to the cloves. A straight-blade type blower with a blade length of 460 mm, a width of 95 mm, and a diameter of 191 mm were used for generating air current for cleaning the material passing between the rollers. A petrol engine was used as the prime mover for operating various units of the machine. Power was transmitted by the belt and pulley arrangement to the roller and blower units. All the above components were fitted with the necessary supports, and, fittings on the mainframe. The overall dimensions of the machine were 505 mm in length, 463 mm in width, and 830 mm in height. Four metal wheels with a diameter of 200 mm were attached to the four legs to facilitate the movement of the machine constructed.

### Description of the Machine Components

The machine consisted of a feed hopper, a garlic separator (two rollers), a fan, outlets for cloves and trash (husk, root, and stem), the main frame, and a drive system. These components and the assembling of the machine were designed using Solidworks software (Version S 2019).



**Figure 1:** Components of designed garlic bulb breaker

### Feed Hopper

The hopper was designed based on the angle of repose, bulk density, and weight of garlic bulbs. The incline of the feed hopper was set at an angle of  $77.27^\circ$ , which was larger than the angle of repose of the garlic bulb

( $49.70^\circ$ ) so that the bulbs would flow freely to the bottom of the hopper. The hopper was in the shape of a trapezoidal prism and was made of a 1.5 mm thick mild steel sheet. It has a lid, with a handle at the top to facilitate opening.

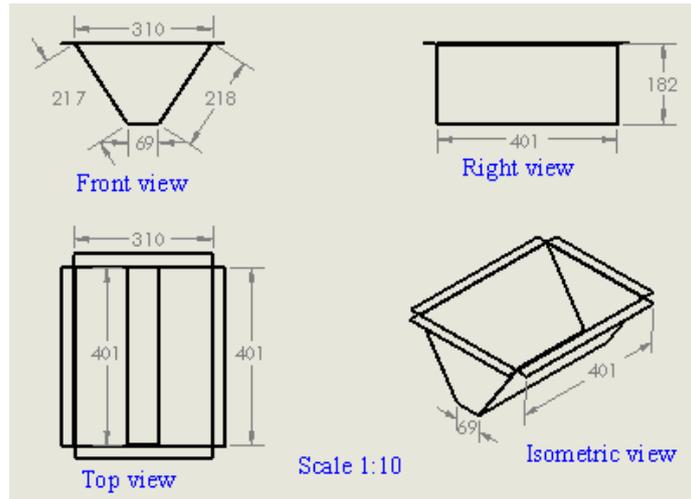


Figure 2: Diagram showing the dimensions of the hopper in mm

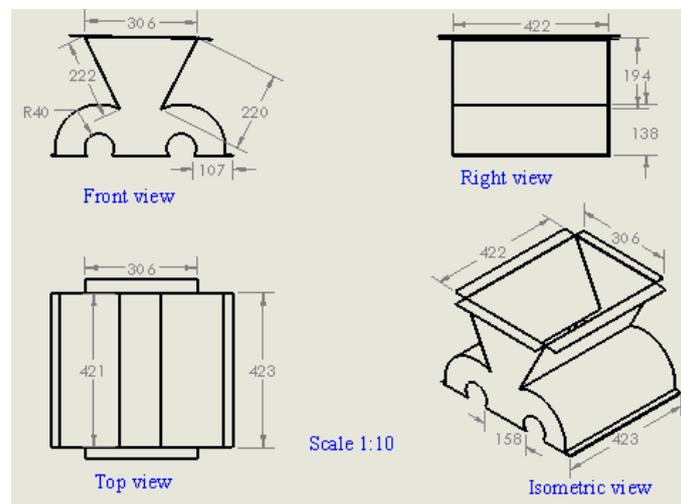


Figure 3: Diagram showing the top half of the machine cover (all dimensions are in mm)

### Bulb Breaking Unit

The garlic separator consists of twin rollers and a casing to contain the materials to be processed. The distance between the rollers was selected according to the size of the garlic clove which was 19 mm which was the same as the one reported by Lee (2007). The rollers were fabricated with an outer diameter of 200 mm diameter and 402 mm length covered with a mild steel sheet thickness

of 2.5 mm. The rollers were mounted on the main frame with the necessary bearings and supports. The rollers are mounted on the mainframe with the necessary bearings and brackets. The surface of the roller was covered with 4mm thick rubber to prevent damage to the garlic when breaking. The roller housing was made of steel sheet metal 1.5 mm thick. It has been securely fixed to protect the rollers and prevent the bulb/clove from spillage.

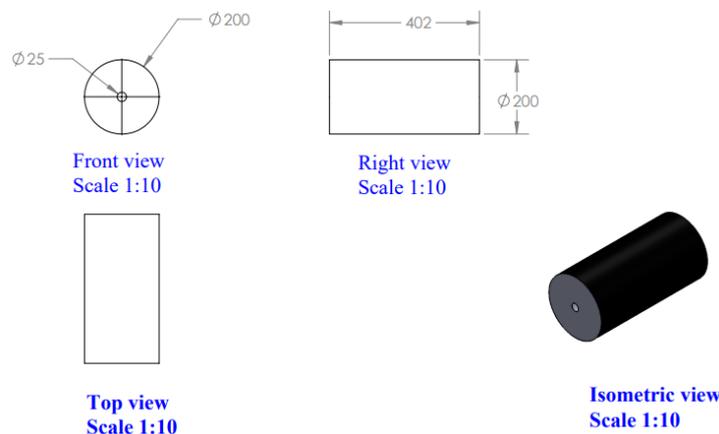


Figure 4: Diagram showing the dimensions (mm) of the roller

### Outlets of the Machine

Two outlets had been provided below the rollers for the gathering of cloves and light material. An outlet for the cloves was designed as a chute such that the surface onto which the materials slide out of the machine was designed to have a downward angle of 45° with the horizontal, which is greater than the 37.560 angles of repose of garlic clove.

The outlet for the debris was positioned on the opposite side of the air blower so that the air current conveys the light waste materials out of the machine. The rectangular shape of the outlet for the debris is oriented upward at the inner end to receive the material with the conveying air and assumes a curved length orienting its outlet end downward for directing the discharge to the ground.

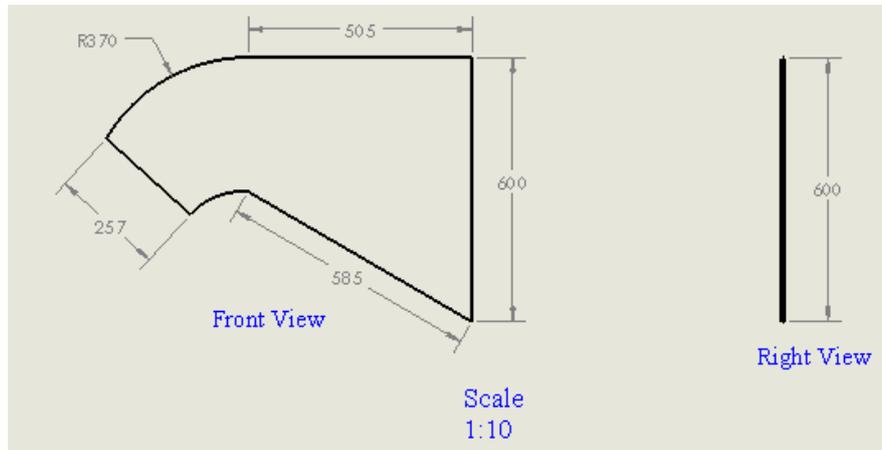


Figure 5: Outlet chutes with chamber cover (all dimensions are in mm)

### Machine Mainframe

The machine's mainframe was made of mild steel angle of 40×40×6 mm. The overall dimensions of the frame were 505 mm long, 465 mm wide, and 830 mm high. Four support wheels with a diameter of 200 mm were attached

to the four supports to increase the mobility of the machine. The feed hopper, bulb-breaking roller, blower, outlet, and motor were mounted on the mainframe. The frame was fixed to mount and support the other parts of the machine.

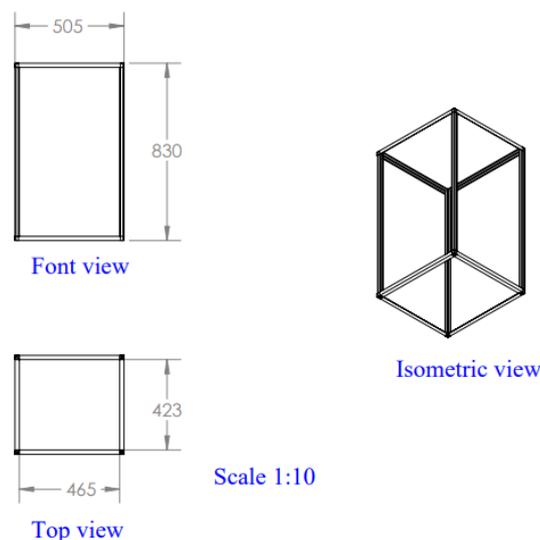


Figure 6: Schematic of the designed frame (all dimensions are in mm)

### Power Transmission System

The garlic bulb separating rollers were equipped with a 5 hp petrol engine mounted on a separate frame attached to the left side of the main frame. The drive is transmitted to the two rollers and the blower through 'V' belts and pulleys assembly. Provision was made to provide the required belt tension.

was fitted as part of the machine and covered by a mild steel sheet of a thickness of 1 mm to form a house. This housing had openings on both ends to allow air in and an outlet for the air on the inner side to direct the pressurized air toward the downward-flowing material just below the rollers. The blower is driven at a speed of 1200 rpm via a pulley and belt arrangement linked to the pulleys of the rollers.

### Air Blower

An air blower consisting of 4 blades mounted on a shaft

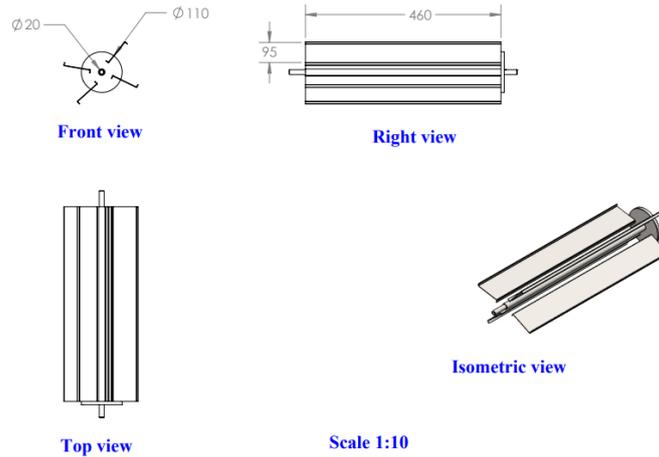


Figure 7: Diagram showing the parts and dimensions (in mm) of the air-blower unit

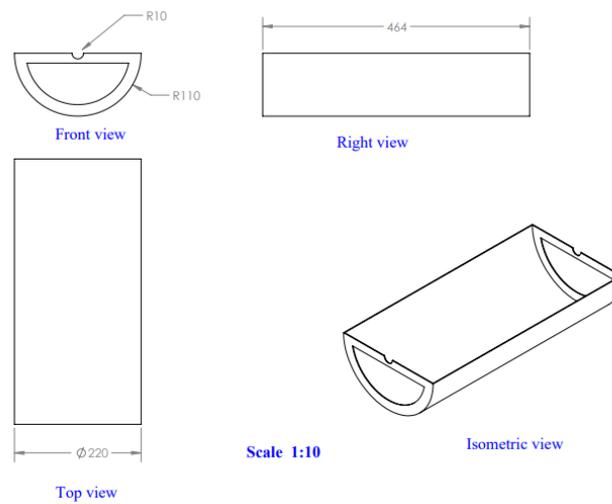


Figure 8: Diagram showing blower cover with an air outlet (all dimensions are in mm)

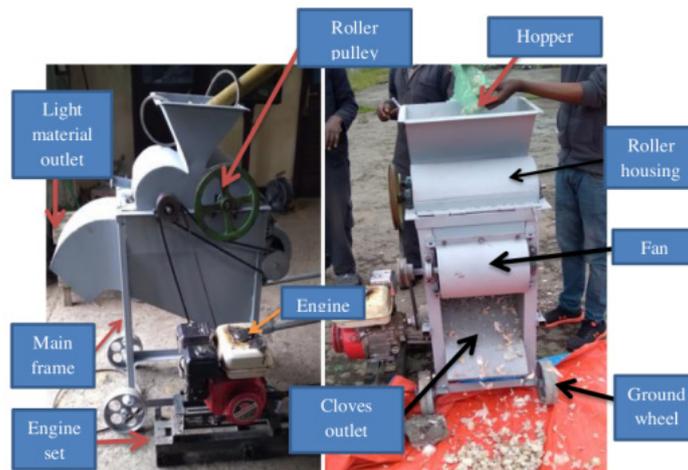


Figure 9: Constructed garlic bulb breaker prototype

### Design Analysis and Calculation

Design calculation of the major components of the machine includes the shaft, bearing, belt and pulley, casing, and engine.

### Selection and Determination of Pulley Diameter

The machine required two pulleys; one driving pulley

was mounted on the crankshaft of the engine and the driven pulleys were mounted on the rollers shaft. Pulleys made from cast iron with 0.075 m diameter for the driving pulley and 0.27 m diameter for the driven pulley were selected based on their availability, low cost, and high performance. Based on the required revolution per minute, the diameter of the driven pulley was determined

according to Spolt (1988) and Khurmi and Gupta (2017) by equation 1.

$$D_1/D_2 = N_2/N_1 \tag{1}$$

$$D_2 = (D_1 N_1)/N_2$$

$$D_2 = (0.075 \text{ m} \times 1400 \text{ rpm}) / (376 \text{ rpm})$$

$$D_2 = 0.27 \text{ m} = 270 \text{ mm}$$

Where;

$$N_1 = \text{speed of the driver} = 1400 \text{ rpm}$$

$$N_2 = \text{speed of the driven} = 376 \text{ rpm}$$

$$D_1 = \text{diameter of the driver} = 0.075 \text{ m}$$

$$D_2 = \text{diameter of the driven} = 0.27 \text{ m}$$

### Belt Selection and Determination of Its Length and Center Distance

V-belt and pulley arrangements were adopted in this work to transmit power from the engine to the shaft of rollers. The main reasons for adopting the v- belt drive was 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). The length of the open belt was calculated as given below in Equation 2:

$$L_p = 2C + \frac{\pi}{2}(D_p + d_p) + \left(\frac{D_p - d_p}{4C}\right)^2 \tag{2}$$

$$L_p = 2 \times 0.4 \text{ m} + \pi/2 (0.075 \text{ m} + 0.27 \text{ m}) + ((0.075 \text{ m} - 0.27 \text{ m}) / (4 \times 0.4 \text{ m}))^2$$

$$L_p = 1.39 \text{ m}$$

The center distance minimum,  $C_{min}$  was calculated using Equation 3:

$$C_{min} = (D_1 + D_2) / 2 + D_1 \tag{3}$$

$$C_{min} = (0.075 \text{ m} + 0.27 \text{ m}) / 2 + 0.075 \text{ m}$$

$$C_{min} = 0.25 \text{ m}$$

Maximum center distance,  $C_{max}$  was calculated using Equations 4 and 5:

$$C_{max} = 2(D_1 + D_2) \tag{4}$$

$$C_{max} = 2(0.075 \text{ m} + 0.27 \text{ m})$$

$$C_{max} = 0.69 \text{ m}$$

Where;

$$L_p = \text{effective length of the belt, mm,}$$

$$C = \text{Center distance, mm,}$$

$$(D_1 + D_2) / 2 + D_1 \leq C \leq 2(D_1 + D_2) \tag{5}$$

$$0.25 \text{ m} \leq C \leq 0.69 \text{ m}$$

The closest standard length of the belt was selected from the standard table and found to be 1.43 m (A 49 V-Belt) and  $C = 0.4 \text{ m}$  taken value.

### Determination of Belt Contact Angle

The belt contact angle was given by the following equation according to Hall *et al.* (2004) in the following equation 6.

$$\sin^{(-1)} \beta = (R - r) / c \tag{6}$$

$$\sin^{(-1)} \beta = (0.135 \text{ m} - 0.0375 \text{ m}) / (0.4 \text{ m})$$

$$\sin^{(-1)} \beta = 0.25$$

$$\beta = 14.48^\circ$$

Where;

$$R = \text{radius of the larger pulley, mm}$$

$$r = \text{radius of the smaller pulley, mm}$$

According to Hall *et al.* (2004), the wrap angle was determined using equations 7 and 8 given below.

$$\alpha_1 = 180 - 2\sin^{(-1)} ((R - r) / c) \tag{7}$$

$$\alpha_1 = 180 - 2\sin^{(-1)} ((0.135 \text{ m} - 0.0375 \text{ m}) / (0.4 \text{ m}))$$

$$\alpha_1 = 180 - 2\sin^{(-1)} (0.25)$$

$$\alpha_1 = 151.04^\circ$$

$$\alpha_1 = 151.04^\circ \times \pi / 180$$

$$\alpha_1 = 2.64 \text{ rad}$$

$$\alpha_2 = 180 + 2\sin^{(-1)} ((R - r) / c) \tag{8}$$

$$\alpha_2 = 180 + 2\sin^{(-1)} ((0.135 \text{ m} - 0.0375 \text{ m}) / (0.4 \text{ m}))$$

$$\alpha_2 = 180 + 2\sin^{(-1)} (0.25)$$

$$\alpha_2 = 208.96^\circ$$

$$\alpha_2 = 208.96^\circ \times \pi / 180$$

$$\alpha_2 = 3.65 \text{ rad}$$

Where;

$$\alpha_1 = \text{angle of wrap for the smaller pulley, rad.}$$

$$\alpha_2 = \text{angle of wrap for the larger pulley, rad.}$$

$$C = \text{center-to-center distance between small and large pulley, mm}$$

### Determination of the Weight of Constructed Garlic Bulb Breaker

The load on every part of the constructed machine was estimated. Estimating the weight of all parts was used to determine the weight of constructed garlic bulb breaker. Accordingly, the weights of the hopper, frame, rollers, and air blower with covering devices were estimated using appropriate equations.

**Table 1:** Mass of each component of the constructed garlic bulb breaker

S/no	Components	Quantity	Surface area (m <sup>2</sup> )	Volume (m <sup>3</sup> )	Density (kg/m <sup>3</sup> )	Unit mass (kg)	Total mass (kg)
1	Rollers	2	0.692	0.00039	7850	3.11	6.22
2	Hopper with cover	1	1.405	0.00018	7850	1.47	1.47
3	Bottom cover	1	0.689	0.00043	7850	3.32	3.32
4	Chamber cover with outlet chute	1	0.602	0.00037	7850	2.89	2.89
5	Chamber floor cover	1	0.671	0.00041	7850	3.23	3.23
6	Roller 1 pulley	1	0.11	0.00015	7850	1.23	1.23
7	Roller 2 pulley	1	0.0465	0.00009	7850	0.78	0.78
8	Fan blower with shaft and pulley	1	0.573	0.00023	7850	1.79	1.79
9	Roller 1 shaft	1	0.0457	0.00014	7850	1.1	1.1
10	Roller 2 shaft	1	0.0481	0.00015	7850	1.15	1.15

11	Frame	1	0.734	0.00028	7850	2.25	2.25
12	Blower cover	1	0.64	0.00012	7850	1.86	1.8
<b>Total mass</b>						<b>24.18</b>	<b>27.54</b>
<b>A mass of parts lay on the roller and fan shaft including 2% for bolts, nuts, and other</b>						<b>28.1</b>	
<b>Mass of garlic</b>						<b>5.2</b>	

**Determination of Power Requirement**

The required power of the rotating roller was determined by Khurmi and Gupta (2008) using Equation 9 below.

$$P = (2\pi NT) / 60 \tag{9}$$

The amount of torque generated by a rotating body was given by the product of the force causing the motion (Fc) and the turning radius (r) given in equations (10 and 11) according to Hannah and Stephens (2006).

$$T = F_c \times r \tag{10}$$

F<sub>c</sub> = centrifugal force required to generate acceleration (N); r = turning radius (radius of rolling roller) = 0.1m where F<sub>c</sub> = ma = mω<sup>2</sup> r

Where m = mass of the rotating roller (kg) = 3.11 kg; ω = angular velocity (rad/s) = ((2πN))/60;

N = Rotational roller speed (rpm) at 376 rpm.

$$\omega = (2\pi \times 376 \text{ rpm}) / 60 = 39.46 \text{ rad/s}$$

$$F_c = m\omega^2 r$$

$$F_c = 3.11 \text{ kg} \times (39.46 \text{ rad/s})^2 \times 0.1 \text{ m} = 484.26 \text{ N}$$

$$T = F_c \times r = 484.26 \text{ N} \times 0.1 \text{ m} = 48.4 \text{ Nm}$$

$$P = (2\pi NT) / 60 = (2\pi \times 376 \text{ rpm} \times 48.4 \text{ Nm}) / 60 = 1910.04 \text{ W}$$

$$P = 2.56 \text{ hp}$$

determined according to Barber (2003) using equations 14 and 15.

$$2.3 \log T_1 / T_2 = (\mu \alpha) / (\sin(\theta/2)) \tag{14}$$

$$2.3 \log T_1 / T_2 = (0.42 \times 2.64) / (\sin(40/2))$$

$$\log T_1 / T_2 = 1.41$$

$$T_1 / T_2 = e^{1.41} = 4.1$$

$$T_1 = T_2 \times 4.1$$

$$P = 4.1 T_2 - T_2$$

$$358.52 = 3.1 T_2$$

$$T_2 = 115.65 \text{ N}$$

$$T_1 = T_2 \times 4.1 = 474.17 \text{ N}$$

$$T_r = R(T_1 - T_2)$$

$$T_r = 0.135(474.17 \text{ N} - 115.65 \text{ N}) = 48.4 \text{ N}$$

Where;

T<sub>r</sub> = resultant torque = Mt

T<sub>1</sub> = tension on the fixed side of the belt, N

T<sub>2</sub> = tension on the loose side of the belt, N

θ = groove angle = 40°

μ = 0.42 (coefficient of friction between rubber belt and pulley, Khurmi and Gupta (2005)

R = larger pulley radius, mm

**Determining the Power Transmitted to the Shaft**

The power transmitted to the shaft by the belt was determined according to Barber (2003) using equations 12 and 13.

$$V = (\pi DN) / 60 \tag{12}$$

$$V = (\pi \times 0.27 \text{ m} \times 376 \text{ rpm}) / 60$$

$$V = 5.33 \text{ m/s}$$

$$P = (T_1 - T_2) \times V \tag{13}$$

$$1910.89 \text{ W} = (T_1 - T_2) \times 5.33 \text{ m/s}$$

$$358.52 \text{ N} = (T_1 - T_2)$$

**Determining Roller and Fan Shaft Diameter**

The shaft design was based on a combination of impact fatigue, bending, and torque. The diameter of a solid shaft with little or no axial load was calculated according to Khurmi and Gupta (2008). The tight and slack side tensions in the belt were 474.17 N, 115.65 N and weight of pulley 1.85 kg x 9.81 m/s<sup>2</sup> = 18.15 N, respectively and (T<sub>1</sub> + T<sub>2</sub> + W<sub>p</sub>) = (474.17 + 115.65 + 18.15) N = 607.97 N, for roller load (W<sub>ro</sub>) = (3.11 kg x 9.81 m/s<sup>2</sup>) = 30.51 N, for garlic bulb load (W<sub>g</sub>) = (5.2 kg x 9.81 m/s<sup>2</sup>) = 51.01 N and (W<sub>ro</sub> + W<sub>g</sub>) = (30.51 + 51.01) N/m = 81.52 N/m = 34.24 N.

**Determination of the Belt Tension**

The belt tension developed in the slack side was

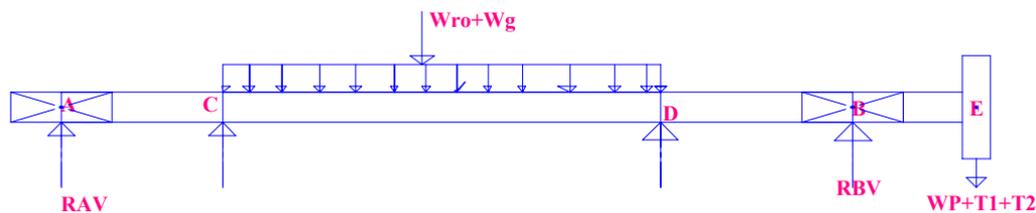


Figure 10: Vertical forces acting on the roller shaft

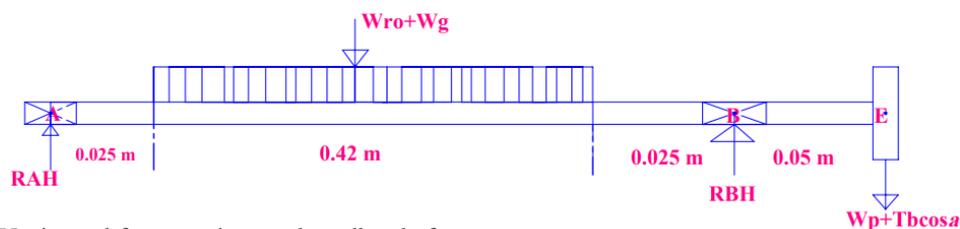
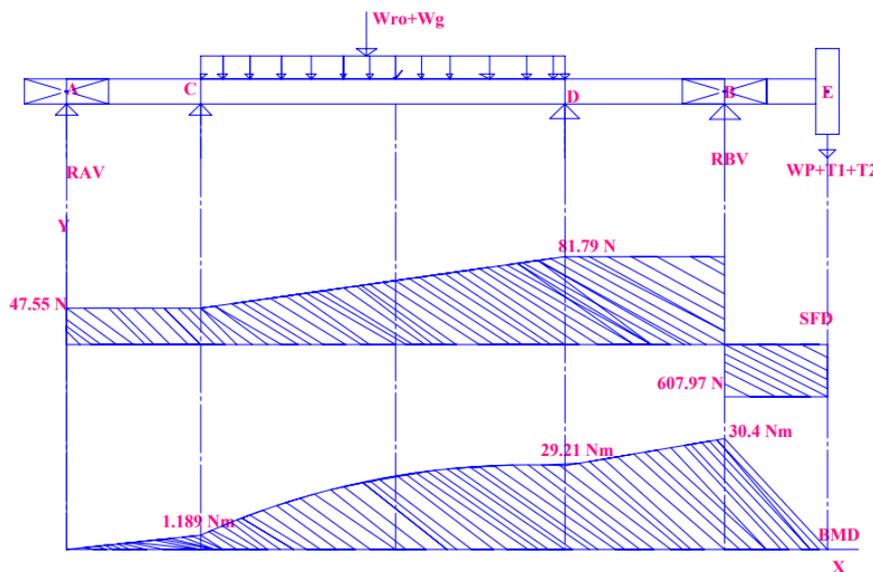


Figure 11: Horizontal forces acting on the roller shaft

Where RAV and RBV = reactions at the vertical support  
 RAH and RBH = reactions at the horizontal support  
 Wp = weight of the pulley  
 Wro = weight of the roller  
 Wg = weight of garlic bulb feed between two rollers  
 The bending moment at point A = 0 for vertical forces acting on the roller shaft,  
 $\sum BMA = 0$   
 $-607.97 \text{ N} \times 0.52 \text{ m} + RBV \times 0.47 \text{ m} - 34.24 \text{ N} \times 0.235 \text{ m} = 0$   
 $RBV (0.47 \text{ m}) = 316.14 \text{ Nm} + 8.046 \text{ Nm}$   
 $RBV = 689.76 \text{ N}$   
 $\sum Fv = 0$   
 $RAV + RBV = 607.97 \text{ N} + 34.24 \text{ N} = 642.2 \text{ N}$   
 $RAV = 642.2 \text{ N} - 689.76 \text{ N} = - (47.55 \text{ N})$  down ward  
 To draw a shear force diagram, all segment points of vertical forces acting on the roller shaft were determined:

Shear force at point A = 47.55 N  
 SF at point C = 47.55 N  
 SF at point D = 47.55 N + 34.24 N = 81.79 N  
 SF at point B = 81.79 N - 689.76 N = -607.97 N  
 SF at point E = -607.97 N - (-607.97 N) = 0 N  
 Where SF = shear force reacting at the support.  
 To draw the bending moment diagram, all segment points of vertical forces acting on the roller shaft were determined:  
 $BMA = 0 \text{ Nm}$   
 $BMC = 47.55 \text{ N} \times 0.025 \text{ m} = 1.189 \text{ Nm}$   
 $BMD = 47.55 \text{ N} \times 0.445 \text{ m} + 34.24 \text{ N} \times 0.235 \text{ m} = 29.21 \text{ Nm}$   
 $BMB = 30.4 \text{ Nm}$   
 $BME = 0 \text{ Nm}$   
 Where BM = bending moment reacting at the support.



**Figure 12:** Shear force and bending moment diagrams on the roller shaft due to vertical force

The bending moment at point A = 0 for horizontal forces acting on the roller shaft,  
 $\sum MA = 0$   
 $-Wp \times 0.52 \text{ m} + RBH \times 0.47 \text{ m} - 34.24 \text{ N} \times 0.235 \text{ m} = 0$ , where,  $Wp = 18.5$   
 $RBH (0.47 \text{ m}) = 9.438 \text{ Nm} + 8.046 \text{ Nm}$   
 $RBH = 37.2 \text{ N}$   
 $\sum FH = 0$   
 $RAH + RBH = 18.15 \text{ N} + 34.24 \text{ N} = 52.39 \text{ N}$   
 $RAH = 52.39 \text{ N} - 37.2 \text{ N} = 15.19 \text{ N}$   
 To draw a shear force diagram, all segment points of horizontal forces acting on the roller shaft were determined:  
 Shear force at point A = 15.19 N  
 SF at point C = 15.19 N  
 SF at point D = 15.19 N - 34.24 N = -19.05 N  
 SF at point B = -19.05 N + 37.2 N = 18.15 N  
 SF at point E = 18.15 N - 18.15 N = 0 N  
 Where SF = shear force reacting at the support.  
 To draw the bending moment diagram, all segment points of horizontal forces acting on the roller shaft were determined:

$BM_A = 0 \text{ Nm}$   
 $BM_C = 15.19 \text{ N} \times 0.025 \text{ m} = 0.38 \text{ Nm}$   
 $BM_D = -15.19 \text{ N} \times 0.445 \text{ m} + 34.24 \text{ N} \times 0.235 \text{ m} = 1.286 \text{ Nm}$   
 $BM_B = 18.15 \text{ N} \times 0.05 \text{ m} = 0.91 \text{ Nm}$   
 $= -15.19 \text{ N} \times 0.235 \text{ m} + 34.24 \text{ N} \times 0.21 \text{ m} = 3.665 \text{ Nm}$   
 $BM_E = 0 \text{ Nm}$   
 Where BM = bending moment reacting at the support.  
 The ASME code equation for a solid shaft having little or no axial loading was calculated according to Hannah (2004):  

$$ds^3 = \frac{16}{\pi \sigma s} \sqrt{(K_b M_b)^2 + (K_t M_t)^2} \quad (16)$$
  
 The total resultant components of horizontal and vertical bending moments on the roller shaft can be obtained as follows:  
 $M_b = (M_v^2 + M_h^2)^{1/2}$   
 $M_b = [(30.4 \text{ Nm})^2 + (3.67)^2]^{1/2} = 30.62 \text{ Nm}$   
 $ds^3 = (16) / (\pi \times 40 \times 10^6) \sqrt{((2 \times 30.62)^2 + (1.5 \times 48.4)^2)}$   
 $ds = 0.02295 \text{ m} = 22.95 \text{ mm}$   
 Therefore, the standard size of 25 mm shaft diameter was used.

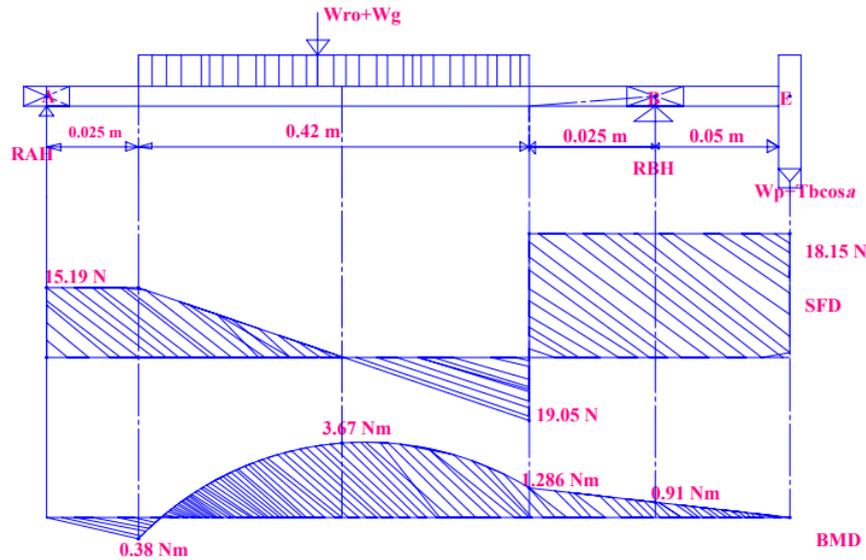


Figure 13: Diagram of shear force and bending moment on roller shaft due to vertical force

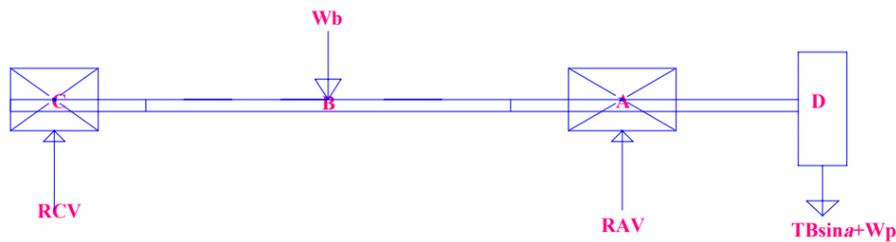


Figure 14: Vertical forces acting on the fan shaft

where,  $d_s$  = shaft diameter, m;  
 $M_b$  = bending moment = 30.62 Nm  
 $M_t$  = torque = 48.4 Nm  
 $K_b$  = impact and fatigue coefficient for bending moments and  
 $K_t$  = impact and fatigue coefficient for torque  
 $\tau$  = permissible shear stress of the shaft material, MN/m<sup>2</sup>  
 The values of  $K_b$  and  $K_t$  were taken as 2 and 1.5, respectively, for the load suddenly applied to the rotating shaft and the allowable axial shear stress ( $\tau$ ) of 40 MN/m<sup>2</sup> based on the ASME code of the American Society of Mechanical Engineers.  
 The shear force and bending moment acting on the fan shaft can be calculated as follows:  
 The moment at A = 0 for vertical forces acting on the fan shaft,  
 $T_b \sin \alpha + W_p = 314.14 \text{ N}$ , where,  $\alpha = 320$   
 $\sum M_A = 0$   
 $(T_b \sin \alpha + W_p) \times 0.04 \text{ m} - (W_f \times 0.175 \text{ m}) + R_{CV} \times 0.335 \text{ m} = 0$   
 $314.14 \text{ N} \times 0.04 \text{ m} - (0.46 \text{ N} \times 0.175 \text{ m}) + R_{CV} \times 0.335 \text{ m} = 0$   
 $R_{CV} (0.335 \text{ m}) = 12.48 \text{ Nm}$   
 $R_{CV} = 37.27 \text{ N}$   
 $\sum F_v = 0$   
 $T_b \sin \alpha + R_{AV} - W_f + R_{CV} = 0$   
 $314.14 \text{ N} - 0.46 \text{ N} + 37.27 \text{ N} + R_{AV} = 0$   
 $R_{AV} = 277.33 \text{ N}$   
 To draw a shear force diagram, all segment points of vertical forces acting on the fan shaft were determined:

Shear force at point D = 314.14 N  
 SF at point A = 314.14 N - 277.33 N = 36.81 N  
 SF at point B = 36.81 N + 0.46 N = 37.27 N  
 SF at point C = 37.27 N  
 Where SF = shear force reacting at the support.  
 To draw the bending moment diagram, all segment points of vertical forces acting on the fan shaft were determined:  
 $BM_D = 0 \text{ Nm}$   
 $BM_A = -314.14 \text{ N} \times 0.04 \text{ m} = -12.57 \text{ Nm}$   
 $BM_B = -314.14 \text{ N} \times 0.215 \text{ m} + 277.33 \text{ N} \times 0.175 \text{ m} - 2.3 \text{ N} \times 0.1 \times 0.12 \text{ m}$   
 $BM_B = -19.017 \text{ Nm}$   
 $BM_A = 0 \text{ Nm}$   
 Where BM = bending moment reacting at the support.  
 The moment at A = 0 for horizontal forces acting on the fan shaft  
 Where  $R_{AV}$  and  $R_{CV}$  = reactions at the vertical support  
 $R_{AH}$  and  $R_{CH}$  = reactions at the horizontal support  
 $W_p$  = weight of the pulley  
 $W_b$  = weight of fan blade blower  
 $T_b \cos \alpha + W_p = 509.025 \text{ N}$ , where,  $\alpha = 320$   
 $\sum M_A = 0$   
 $-509.025 \text{ N} \times 0.04 \text{ m} + R_{CH} \times 0.335 \text{ m} = 0$   
 $R_{CH} (0.335 \text{ m}) = 20.361 \text{ Nm}$   
 $R_{CH} = 60.78 \text{ N}$   
 $\sum F_H = 0$   
 $-(T_b \sin \alpha + W_p) + R_{AH} + R_{CH} = 0$   
 $-509.025 \text{ N} + R_{AH} + 60.78 \text{ N} = 0$   
 $R_{AH} = 448.24 \text{ N}$

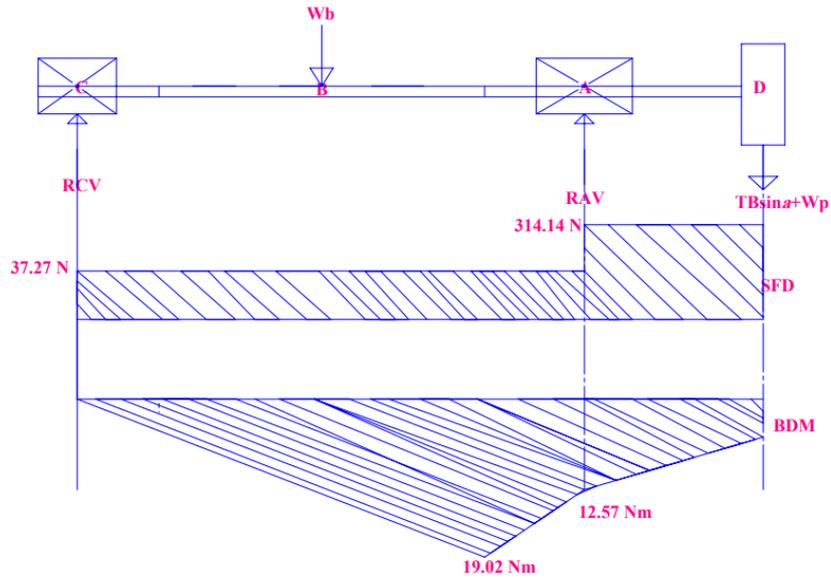


Figure 15: Diagram of shear force and bending moment on the fan shaft due to vertical force

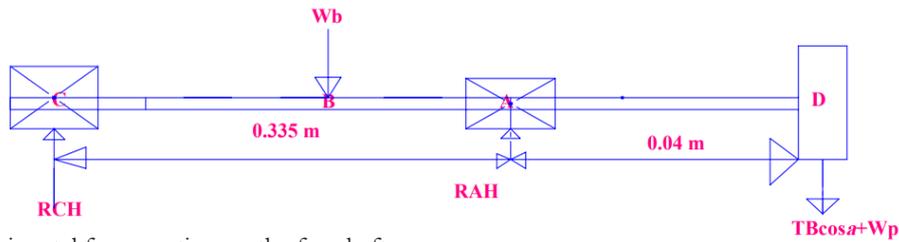


Figure 16: Horizontal forces acting on the fan shaft

To draw a shear force diagram, all segment points of horizontal forces acting on the fan shaft were determined:  
 Shear force at point D = 509.025 N  
 SF at point A = 509.025 N - 448.24 N = 60.78 N  
 SF at point B = 60.78 N - 0.46 N = 60.78 N  
 SF at point C = 60.78 N  
 Where SF = shear force reacting at the support.  
 To draw the bending moment diagram, all segment points of horizontal forces acting on the fan shaft were determined:

$BM_D = 0 \text{ Nm}$   
 $BM_A = - 509.025 \text{ N} \times 0.04 \text{ m} = - 20.361 \text{ Nm}$   
 $BM_C = 0 \text{ Nm}$   
 Where BM = bending moment reacting at the support.  
 The total resultant components of horizontal and vertical bending moments on the fan shaft can be obtained as follows:  
 $M_b = (M_v^2 + M_h^2)^{1/2}$   
 $M_b = [(19.02 \text{ Nm})^2 + (20.361)^2]^{1/2} = 27.86 \text{ Nm}$   
 $ds^3 = (16) / (\pi \times 40 \times 10^6) \sqrt{((2 \times 27.86)^2 + (1.5 \times 48.4)^2)}$

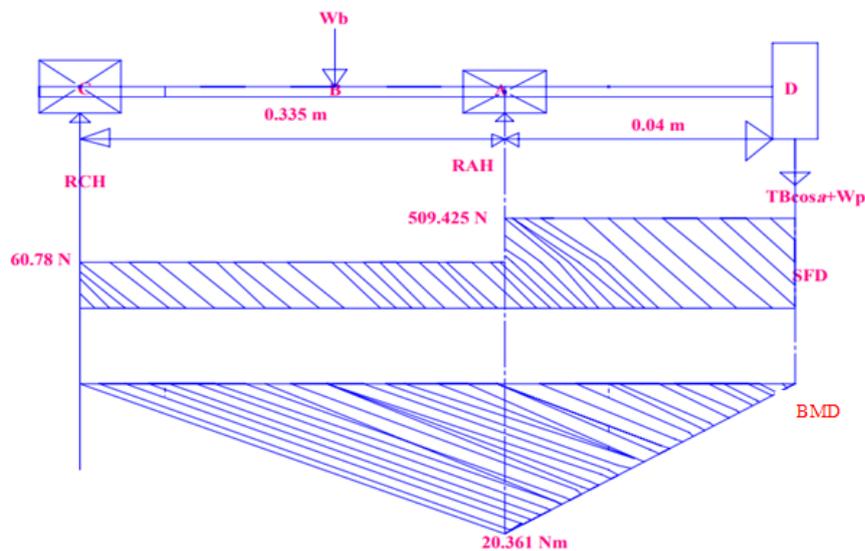


Figure 17: Diagram of shear force and bending moment on the fan shaft due to horizontal force

$d_s = 0.02267 \text{ m} = 22.67 \text{ mm}$

Therefore, the standard size of 25 mm shaft diameter was used.

where,  $D_s$  = diameter of the shaft, m;

$M_b$  = bending moment = 27.86 Nm

$M_t$  = Torque = 48.4 Nm

$K_b$  = impact and fatigue coefficient for bending moment and

$K_t$  = impact and fatigue coefficient for torque

$\tau$  = permissible shear stress of the shaft material, MN/m<sup>2</sup>

Based on the American Society's mechanical engineers. ASME code, when the load was suddenly applied to the rotating shaft and the allowable axial shear stress ( $\tau$ ) was 40MN/m<sup>2</sup>, the values of  $K_b$  and  $K_t$  were taken as 2 and 1.5, respectively.

### Bearing Selection

The size of the bearing used depends on the size of the shaft required and the space available. The bearings also had a high enough load capacity to provide an acceptable combination of service life and reliability. The bearing size was determined using the maximum resultant force acting on the bearing and the maximum desired life (Khurmi and Gupta, 2005). Equations (17 and 18) were used to analyze and select bearings.

The bearing pressure  $P_b$  on the edge of the shaft is given by:

$$P_b = R/A \tag{17}$$

$$P_b = 49.92/0.314 = 158.97 \text{ N/mm}^2$$

$$R = (\sqrt{RA_H^2 + RA_V^2}) \tag{18}$$

$$= (\sqrt{15.19^2 + 47.55^2})$$

$$R = 49.92 \text{ N}$$

Where,

$R$  = Maximum resultant force acting on a bearing, N (See Figure 11)

$A$  = Area of contact,  $(\pi d^2/4 = \pi \times (20\text{mm})^2/4 = 0.314 \text{ mm}^2)$

$RA_V$  = maximum resultant force acting on a bearing at point A on the vertical plane (N),

$RA_H$  = maximum resultant force acting on a bearing at point A on the horizontal plane (N),

The allowable bearing capacity  $P_{bp}$  is given by equation (19) (ACME, 1987).

$$P_{bp} = P_0 \times C_1 \times C_2 \tag{19}$$

$$= 1542 \text{ N/mm}^2 \times 0.3 \times 0.72 = 333.1 \text{ N/mm}^2$$

Where,  $P_0$  = Allowable bearing pressure (1542 N/mm<sup>2</sup>)

$C_1$  = Correction factor (0.3) (ACME, 1987)

$C_2$  = Correction factor (0.72) (ACME, 1987)

The condition to be satisfied for selecting the bearing is

$$P_b < P_{bp}$$

Therefore, the bearing is selected as the condition for selecting the bearing is satisfied

$$P_{(b)} (158.97) < P_{bp} (333.1)$$

Dynamic load rating can be determined using equation (20), (Khurmi and Gupta, 2005)

$$C = ((L \times f R (N \times H_m) / 10^6)^{1/k}) \tag{20}$$

$$C = ((3 \times 0.05 \text{ KN} (376 \text{ rpm} \times 20,000 \text{ hr}) / 10^6)^{1/3}) = 1.04 \text{ KN}$$

Where,  $C$  = basic dynamic load rating, (KN)

$N$  = revolution of the shaft in rpm

$H_m$  = desired maximum lifespan of the bearings (hr)

$L \times f$  = Load application factor or service factor

$k$  = exponent for the life equation (3.0 for ball bearings, 10/3 for roller bearings)

The bearing size was selected using the maximum resulting reaction (0.05 KN) acting on the bearing and the dynamic load capacity (1.04 KN) was determined using the formulas (18) and (20). The application element acts as a safety factor; increases the design load to account for overload and dynamic loads. For moderate impactor applications, the application load factor ( $L \times f$ ) ranges from 1.5 to 3.0. The maximum value of the application factor 3.0 was used for bearing selection. The desired maximum bearing life has been selected based on the application of the bearing. Machines use 8 hours per day; therefore, the desired maximum bearing life ranges from 12,000 to 20,000 hours. Therefore, a maximum life value of 20,000 hours was chosen to determine the base engine load. Bearing No. 204 with a bore of 20 mm and a width of 14 mm was chosen because the minimum shaft diameter was determined to be 20 mm. Therefore, the six bearings used (i.e. two for the blower shaft and four for the two-roller shaft of the garlic breaker) were selected based on common criteria for bearing selection.

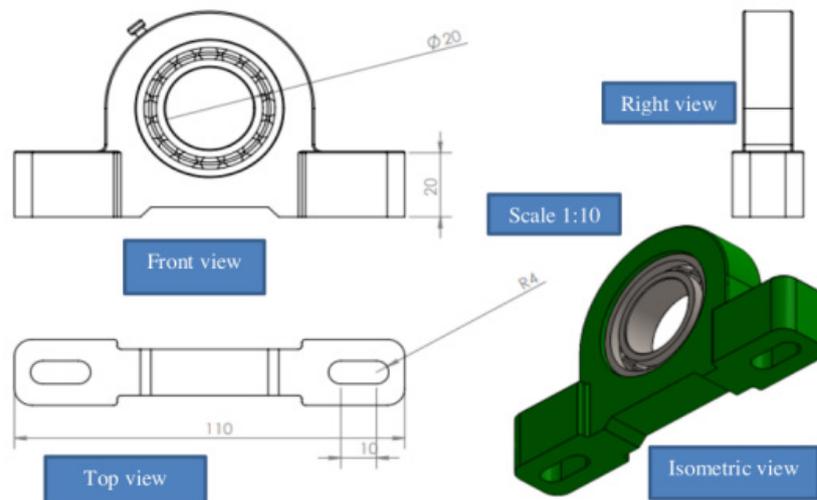


Figure 18: Diagram showing bearing in mm

**Economics Evaluation**

The economics of the constructed machine was evaluated in terms of material and fabrication cost, interest on capital, cost of repairs and maintenance, cost of fuel consumed, labor cost, and depreciation cost. The cost of the constructed machine was estimated in Ethiopian Birr (ETB) according to Channabasamma (2014). The economic life of 10 years and the constructed machine was expected to work for 548 hours per year.

**Fixed Cost**

Material cost + Fabrication cost (C) + Machine cost (C) = 9264.8 ETB

where:-Material cost = cost of material used to construct the machine,

Fabrication cost = cost of labor for fabrication,

Machine cost= cost of machine work

Salvage value (S) @ 10% of total cost machine = 926.48 ETB

**Operating Cost**

- a. Annual Usage (U) (Estimated Uptime) = 548 hours
- b. Life expectancy (L) = 10 years

**Fixed Cost**

- i. Depreciation (D) by equation 21.

$$D = ((C-S))/UL = ((9264.8 \text{ ETB} - 926.48 \text{ ETB}) / ((548 \text{ h} \times 10))) = 1.52 \text{ ETB/h} \quad (21)$$

- ii. Interest on capital investment @ 12% per annum on average price (I) by equation 22.

$$I = ((C+S))/2U \times 0.1 = ((9264.8 \text{ ETB} + 926.48 \text{ ETB}) / ((2 \times 548 \text{ h}) \times 0.12)) = 1.12 \text{ ETB/h} \quad (22)$$

- iii. Repair/Maintenance cost @ 2%(R) was given below by equation 23.

$$R = C/UL \times 0.02 = ((9264.8 \text{ ETB}) / ((548 \text{ hr} \times 10))) \times 0.02 = 0.03 \text{ ETB/hr} \quad (23)$$

Total Fixed Cost (D + I + R) = (1.52+1.12+0.03) ETB/hour = 2.67 ETB/hour

**Operational Cost/Variable Cost by Equations 24 and 25**

- i. Fuel consumption (P) = 1.67 liter/hr

Cost per unit of liter fuel = 16 ETB

Therefore, cost of fuel consumed = 1.67 x 16 = 26.72 ETB/hr

- ii. Labor cost (b) of 380 ETB/day (8 hours) per person or 47.5 ETB/h

Total variable cost (P+b) = 26.72+47.5 = 74.22 ETB/h (24)

Total Operating Cost (A) = Total Fixed Cost + Total Variable Cost (25)

$$= 2.67 \text{ ETB/h} + 74.22 \text{ ETB/h} = 76.89 \text{ ETB/h}$$

The breaking capacity for the seeding and processing of the machines were 245.91 kg/h and 306.34 kg/h, respectively when the machine was built with maximum extraction of unique clove with high separating efficiency and minimal damage. Therefore, the cost of breaking garlic bulbs for planting and processing was 0.31 and 0.25 ETB/kg, respectively.

The cost of the machine to break 1 kg of garlic bulbs by Equation 26.

$$(M) = ((\text{total operating cost}) / ((\text{breaking capacity}))) \quad (26)$$

For planting = (76.89 ETB/hr)/(245.91 kg/hr)=0.31 ETB/kg

For processing = (76.89 ETB/hr)/(306.34 kg/hr)=0.25 ETB/kg

**Manual Operational Cost**

Labour cost (75 x 3 persons) = 225 ETB per day (8 hours) per person or 28.12 ETB/h.

The maximum manual breaking potential of garlic bulbs – is one hundred twenty kg/day (eight hours).

When performing operations with a high degree of separation with maximum recovery of bulbs, the average labor productivity was 15 kg/h. Therefore, the cost of manually separating garlic bulbs (L) was 1.87 ETB/kg.

Compared to hand-separating garlic bulbs for seeding and processing, the cost savings of the garlic bulb-breaking machine was 83.42% and 86.63%, respectively.

The cost saved by the bulb-breaking machine was given below using equation 27.

$$\text{The cost saved by bulb breaking machine} = (L-M) / L \times 100 \quad (27)$$

For planting = (1.87-0.31)/1.87×100=83.42%

For processing = (1.87-0.25)/1.87×100=86.63%

Where L = 1 kilogram of manual garlic bulb breaking (labor force) cost,

M = 1 kilogram of garlic bulb-breaking machine cost.

**Table 2:** The summarized cost of constructed garlic bulb breaker

No.	Variable	Cost (ETB)
1	Raw material	6363.21
2	Materials Wastage = 2.5% of 1	159.08
3	Machine cost	383.51
4	Labor cost	380
5	Overhead = 5% of (3+4)	38.17
6	Profit = 10 % of (1+2+3+4)	732.39
7	Sell tax =15% of (1+2+3+4+5)	1208.45
8	Selling price = (1+2+3+4+5+6)	9264.81



**Figure 19:** Garlic cloves (after breaking)

**CONCLUSIONS**

Breaking garlic bulbs is a unit operation where garlic bulbs are separated into cloves for ease of processing. Breaking the garlic bulbs requires special care and skill due to their typical physical properties and the presence of volatile essential oils in the epidermal cells that impart the

characteristic garlic aroma. For that purpose, the physical properties of Holeta's local variety of garlic bulbs and cloves were determined to generate data that was used for designing the prototype machine. This research work has concluded that garlic bulb-breaker uses low fuel consumption and reduces the time of garlic cloves separation from the garlic bulb. The breaking capacity for the seeding and processing of the machines were 245.91 kg/h and 306.34 kg/h, respectively. The constructed garlic bulb breaker saved cost over manual garlic bulb breaking was 83.42% and 86.63% for planting and processing purposes, respectively. Thus, the machine is very acceptable. On the other hand, the machine has large clumps of cloves and needs an increment of breaking capacity. This could be an assignment for researchers to additionally improve the machine.

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## Appendix Figures





Figure 20: Prototype construction of garlic bub breaker