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## Design and Development of Automated Coconut Dehusking and Crown Removal Machine

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### Abstract

This paper presents the design and analysis activities involved in developing an automated coconut de-husking and coconut crown removal machine. The main purpose of this machine is to eliminate the skilled operator involved in de-husking the coconut and to completely automate the dehusking and crown removing process. Although coconut dehusking machines have already been demonstrated in the work [1,3,4] and also in some small scale industries, the process is either manual or semiautomatic. A completely automated machine with manual loading and unloading of coconuts will yield productivity higher than the existing process. Because of that, the current work is mainly focused on an automated machine for dehusking and crown removing. Also, we can yield lot of useful and commercial products from coconut at various stages of its lifecycle [5]. One of the important products is Copra which is the dried meat or kernel of the coconut used to extract coconut oil. So, the machine aims at de-husking and removing the crown of the de-husked coconut of various sizes across the world. In order to get to know about the different sizes of the coconut, various places across India has been visited where exuberant yielding of coconuts are made. Also, dimensional data of coconuts have been collected in some of the other most eminent countries where prominent coconut cultivations are done.

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Based on the survey the maximum and minimum sizes of the coconut are determined. The machine is designed to accommodate different sizes of the coconut that are cultivated anywhere in the world. Also, various experiments have been conducted on both dry and mature coconuts in order to determine the force required to de-husk the coconut. The proposed experiments have been conducted on a standard Universal Testing Machine. Based on the diverse data collection, experiments conducted and by using machine design principles [10,11], a rigid machine is designed using Solid Works software so as to withstand the loads generated during the dehusking process. The critical elements of the machines like the main frame, drive shaft, rollers are analyzed for determining the Maximum Principal stress, Von Mises strain and Maximum Principal Strain using ANSYS Workbench software. While analysis the degree of freedom of the elements during the machine running time has been envisioned and taken care. The loads acting on each individual element has been calculated and involved them in the analysis. The results obtained for the various load conditions are discussed in detail in the paper. Conclusions are drawn based on the results obtained which serves to validate and improve the design technique.

**Keywords:** Automatic Coconut dehusking; Crown removal; Main frames; Toothed Rollers; Drive shaft; ANSYS Workbench.

## 1. Introduction

The machine proposed in this work basically does two processes. They are dehusking and crown removing. De-husking is the process of removing the fibrous portion (husk) from the nut. It has been a problem from time immemorial. Even though many types of equipment have been developed to de-husk coconuts [1,3,4], a majority of de-husking is carried out manually which goes to show that there are no superior machines developed to handle the coconuts. Coconut de-husking is the most fundamental issue in terms of finding labor and improving productivity. Few machines which are in current practice are described below. The tool in the figure 1 is most commonly used for de-husking coconuts and is even highly domesticated. It consists of a sharpened edge on which the coconut is placed by applying force. Then the hinged edge is opened with the help of the lever. This in turn, cuts and opens the husk of the coconut. This is done in various orientations and the remaining husk is pulled out till the coconut is completely de-husked. It is a manual process and requires a lot of effort from the worker. It is also dangerous as an in-experienced worker may hurt himself in the process.



Fig.1. Manual Dehusking tool

The machine shown in the figure 2 is called the husk peeling machine and is one of the modern and more innovative methods developed for de-husking coconuts. The innovation lies in the fact that the machine peels of the coconut husk as easily as one peels the outer skin of the banana fruit. It is a good solution in terms of quality of de-husking and completes the process of de-husking to the highest possible efficiency. Even though it requires no force to be exerted by the worker, it still remains non productive because it requires an operator to operate the machine at all times. It makes use of links which are actuated by pneumatics or toggle mechanisms to peel off the coconut husk. It also is a very time consuming process. One other disadvantage is that the husk after peeling is not as easily conducive as in other methods for coir processing.



Fig.2. Coconut Husk Peeling machine

The method in the figure 3 is by far, the best methods available for de-husking and is now being used in mass production. It has two horizontal rollers with blades. This is similar to the work [1]. The coconuts however have to be held in place by some clamping force. The coconut is held to the rotating rollers either by hand or through some leverage. The projected sharp edges do the husk removal process. This also requires continuous labor force. It does speed up the process of de-husking, even though the final productivity in terms of numbers largely depends on the speed and ability of the worker.



Fig.3. Horizontal roller dehusking machine

The crown of the coconut is also a part of the coconut that is processed in the proposed machine. Crown is the fibrous portion that is protruding from the coconut. It is usually seen covering the eyes of the coconut. It is very difficult to manually remove the crown from the nut. Crown removal is done for oil production and cooking after then the coconut is cut into two halves.



Fig.4. Coconut crown

## 2. Experiments and Data collection

The dimensions of the mature coconut are very much significant as far as the machine design is concerned. Various farms and sites were visited to comprehend the band of dimensions involved in the mature coconut. Some coconuts from Andaman and Nicobar Islands are very big when compared to the areas thrived with coconuts in India like Tamil Nadu and Kerala. The machine has been designed to overcome a large range of coconut sizes, with equal importance to productivity. Various places in and around Tamil Nadu and Kerala have been considered predominately to arrive at a conclusion on the size of coconuts. The dimensions have been measured using an external caliper. Majority of coconuts appear to have the dimensions such that the Y and Z direction dimensions are almost identical. However the critical and concerning dimension for this machine design is the dimension X.

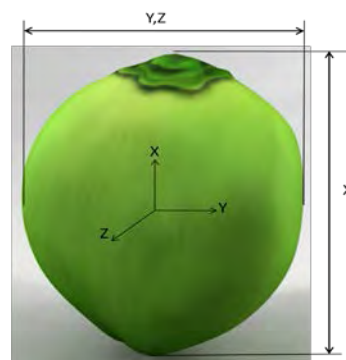


Fig.5. Coconut dimensional axes

The dimensions of the coconuts collected from various places have been coalesced and few dimensions are presented in ascending order in the below table. The dimensions in the first row represent the smallest coconut with husk and the dimensions in the last row represent the maximum size coconut with husk.

Table 1. Dimensions of the coconut with husk

No.	X-axis(mm)	Y-axis(mm)	Z-axis(mm)
1	121	113	109
2	157	146	142
3	182	161	165
4	246	218	214
5	276	230	227
6	287	269	263

In addition to the above data, the dimensions of the coconut after the husk being removed are required as they would play a greater role in the design of the crown removal station of the machine.

Table 2. Dimensions of the dehusked coconut

No.	X-axis(mm)	Y-axis(mm)	Z-axis(mm)
1	97	86	86
2	114	98	96
3	125	115	109
4	147	129	125
5	162	135	134
6	132	124	122

Based on the above data, the Stroke length of the Pneumatic Cylinders involved in the Dehusking Unit and Crown removal unit has been derived.

In addition to the above, some data on the loads acting on the coconut to dehusk it are required. The husk is removed from the Machine through the shear force exerted by the fixed toothed rollers in the machine. So, in relation to that the amount of shear load required to dehusk the coconuts has been determined. Both dry and mature coconuts of various sizes are tested experimentally in the Standard Universal Testing Machine (UTM). The mechanical properties of the coconut fiber are studied from the works [7,8] for this purpose.



Fig.6. Coconut specimen I in UTM




Fig.7. Coconut specimen II in UTM



Fig.8. Shear Force reading in the UTM


The Shear load required to dehusk the mature coconut of different sizes are listed below

Table 3. Shear Load required to dehusk mature coconuts

No.	Dimension in X-axis (mm)	Shear Load (kN)	Mature Coconut
1	128	0.32	
2	145	0.35	
3	178	0.39	
4	212	0.45	
5	248	0.52	

Similarly the Shear load required to dehusk the dry coconut of different sizes are listed below.

Table 4. Shear load required to dehusk dry coconuts

No.	Dimension in X-axis (mm)	Shear Load (kN)	Dry Coconut
1	133	0.24	
2	154	0.26	
3	174	0.29	
4	208	0.36	
5	252	0.42	

It is evident from the table that the load required for de-husking the dry coconut is higher than the load required for de-husking the mature coconut. Also, graphs are plotted for both dry and mature coconuts in such a way that X-Dimension of the coconut is taken as an independent parameter in X-axis and the shear load is taken along the Y-axis. This is due to the fact that the shear load varies based on the size of the coconut. The graphs are shown below.

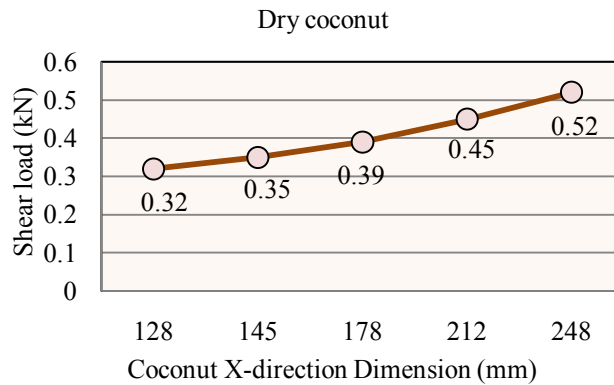


Fig.9 .Graph between X-direction dimensions of the dry coconut and shear load to de-husk the coconut.

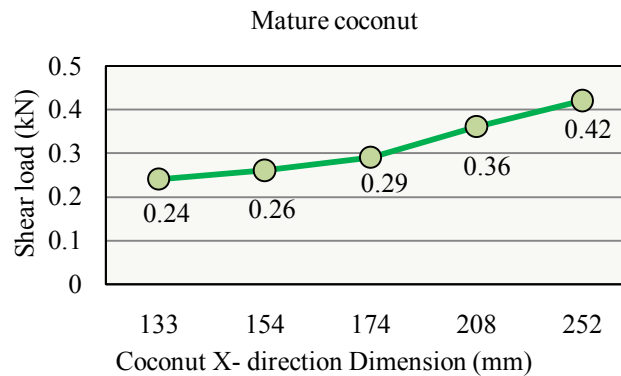


Fig.10 .Graph between X-direction dimensions of the mature coconut and shear load to de-husk the coconut

From the graphs it can be inferred that the shear load varies linearly with respect to the size of the coconut.

### 3. Design of the Dehusking unit

The de-husking unit is based on the rotating rollers, rotating in the opposite direction. The rollers are placed such that the line joining the centers is vertical. The two rollers are fixed using bearings to the two side frames. They rotate about their axes. The fixed roller has blades designed to penetrate the husk and pull it apart. There is a small clearance between the blades of the top and bottom rollers. There is a movable roller unit with three rollers attached to side frames using bearings and can rotate about their axes. The movable roller unit can move in and out from the fixed roller unit using guide ways, its motion controlled pneumatically. The rollers in the movable unit are fixed such that their centers are at an inclination. So, the three rollers are at different positions from the fixed roller unit. This is in order to facilitate de-husking of different sizes of coconuts. The fixed roller unit derives its power from a motor through a pulley and a set of gearing. The rollers in the movable unit are free to rotate. The fixed rollers rotate at about 160 rpm thus causing the process of de-husking to be accelerated. The coconuts can be dropped from the above. At the time of dropping, the fixed and movable roller units are apart in such a way that the smallest coconut and biggest coconut get caught in between and do not drop down. Once the

coconut is held in between the two units, the movable roller unit starts pushing the coconut thus aiding the de-husking process. A flywheel has been attached to the bottom roller of the fixed unit. As, there is time interval between two successive coconuts, the power required is not always the same. The flywheel takes up the extra energy and provides it to compensate impacts which may require more torque.

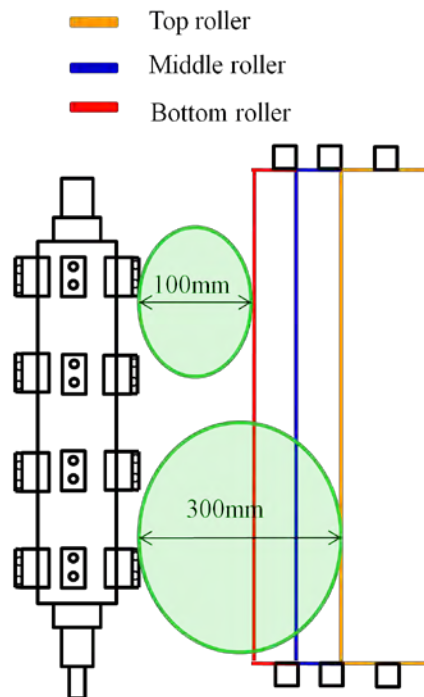


Fig.11. Explanatory diagram of the Machine's Top view

The de-husking unit consists of the following major parts

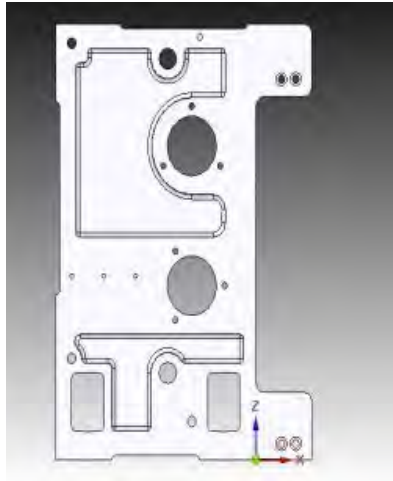
- Main Frame Left
- Main Frame Right
- Top roller
- Bottom roller
- Blades
- Drive shaft
- Bearings for the rollers
- Moving pressure roller unit
- Scrapper unit
- Flywheel

All the parts are designed using Solid Works Software and proper design calculations are done wherever required.



### 3.1 Left Main frame

The Left main frame has two holes to accommodate the bearings for the top roller, bottom roller. The frame is made using casting to withstand any vibrations if such occur during working of the machine. The frame has been created such that there is no stress concentration due to sharp edges, or abrupt area change.



3.2.1 Fig.12. Left side main frame - side view

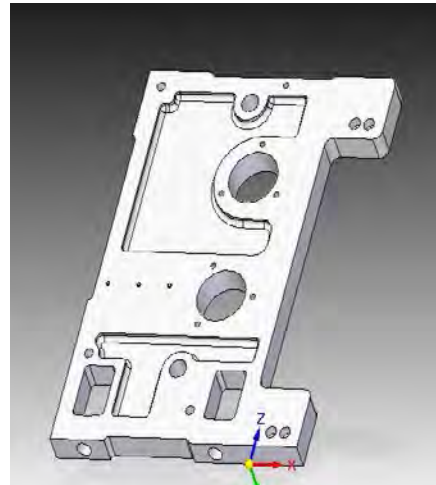


Fig.13. Left side main frame - 3D view

This is also similar to the Main Frame L except that it houses the drive shaft apart from the other two rollers. Tie bars have been added between the frames to give additional supports.

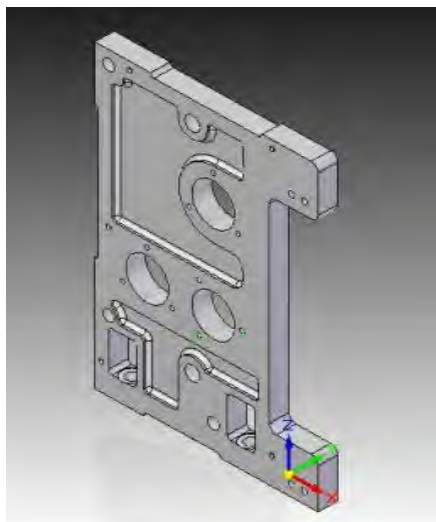


Fig.14. Right side main frame - 3D view

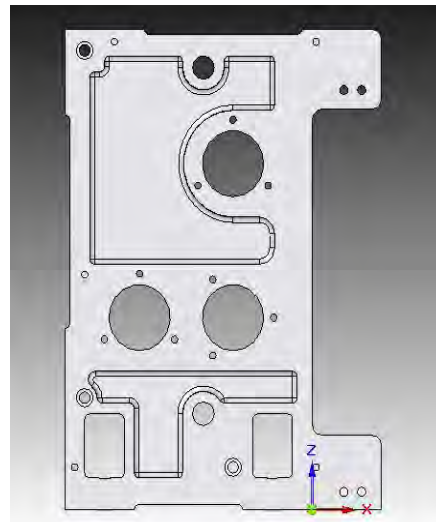


Fig.15. Right side main frame - side view

### 3.3 Top Roller unit

The top roller has been designed such that the length can accommodate one huge coconut for de-husking. It is a

solid roller with taps in frequent intervals to house the blades which will be fixed to the roller using screws. The top roller is holding a total of  $4 \times 6 = 24$  blades. The blades are subjected to heat treatment process to possess hardness value up to 60 HRC. The heat treatment process applied here is Nitriding. The blades have been designed in a profile which easily penetrates the husk and pulls it apart on rotation. The blades should not be very sharp as they may crack open the inner shell of the coconut which will lead to wastage. At the same time it should be able to penetrate the husk easily. It has 2 holes which are drilled and have counter bores. This is in order to secure the blade tightly on to the roller.

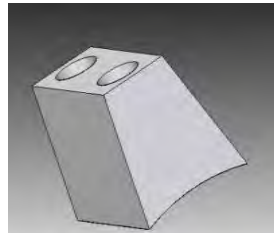


Fig.16. 3D view of the tooth used for dehusking the coconut

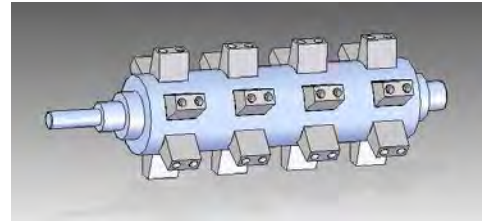


Fig.17. Top De-husking roller with teeth

The bearings have been selected for the bending loads acting on the top roller due to various factors including force generated by the coconut, self weight of the roller, etc. Proper bearings have been selected for the same and the effect due to bearing reaction forces on the frame has also been analyzed.

### ***3.3.1 Force analysis of the top fixed toothed roller assembly***

The method of Shear force evaluation is considered for the bearing force calculation [13].

The following Nomenclatures are used in all the bearing calculations presented.

C1,C2,C3 are the loads acting on the roller due to coconut

H1 and H2 are the load acting on the bearing in the Horizontal plane

V1 and V2 are the load acting on the bearing in the vertical plane

Pr is the radial load due to gear

Pt is the tangential load due to gear

G is the self weight of the gear

S is Self weight of the roller

Let  $C_1, C_4$  are equal to 250N and  $C_2, C_3$  are equal to 500N. Since  $C_2$  and  $C_3$  acts on the area where more load will act.

The figure 18 represents the forces acting on the fixed top roller unit in the horizontal plane.

All the dimensions are in mm.

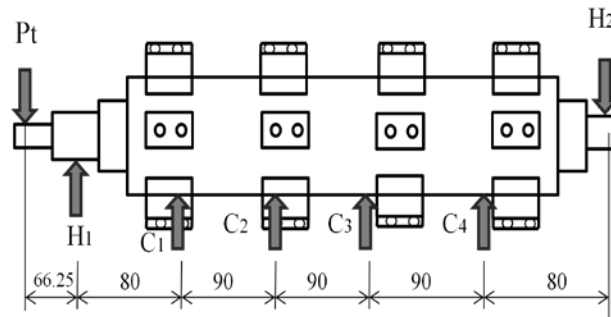


Fig.18. Forces acting on the top roller in the Horizontal plane

The figure 19 represents the forces acting on the force top roller unit in the vertical plane.

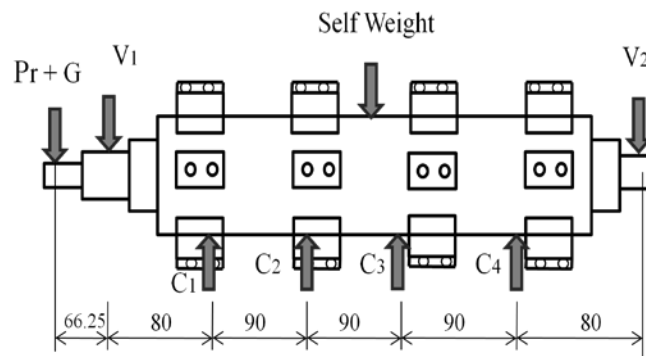


Fig.19. Forces acting on the top roller in the Vertical plane

Two 96 teeth spur gears are used for the power transmission from the bottom to the top roller. The radial and tangential force acting on the 96 teeth gear used in the top and bottom toothed rollers are calculated below.

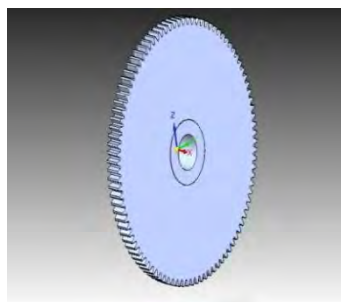


Fig.20. 96 teeth gear used in the top and bottom toothed roller

Both the gears forms a gear train which rotate both the rollers at the same rpm.

**3.3.1.1 Gear parameters of the 96 teeth gear**

Gear Teeth,  $Z = 96$  (1)

Module,  $m = 1.5$  (2)

Pitch circle Diameter =  $mZ = 144\text{mm}$  (3)

Speed of rotation of the gear  $N = 150$  rpm (4)

Power of the motor = 1 KW (5)

$\alpha =$ Pressure angle of the gear =  $20^\circ$  (6)

Self weight of the Roller =  $21.4\text{kg} = 210\text{N}$  (7)

Self weight of the gear =  $1.22\text{kg} = 12\text{N}$  (8)

Torque transmitted by the gear =  $M_t = 60 \times 10^6 \times \text{Power} / 2\pi N = 63694.26 \text{ Nmm}$  (9)

$P_t = 2M_t / \text{PCD} = 884.64 \text{ N}$  (10)

$P_r = P_t \tan \alpha = 321.98 \text{ N}$  (11)

The following calculation is meant for selecting the suitable bearing grade based on the load acting on the top toothed roller.

The loads acting in the Vertical plane are solved below.

Taking Moment about  $V_2$

$[(P_r + G) \times 496.25] + (V_1 \times 430) - (C_1 \times 350) - (C_2 \times 260) - (C_3 \times 170) - (C_4 \times 80) + (S \times 215) = 0$  (12)

$(333.98 \times 496.25) + (V_1 \times 430) - (250 \times 350) - (500 \times 260) - (500 \times 170) - (250 \times 80) + (210 \times 215) = 0$  (13)

$V_1 = 259.56\text{N} (\uparrow)$  (14)

Since the forces are in Equilibrium,

Sum of all the forces acting Downwards = Sum of the forces acting upwards (15)

$V_2 = 696.46\text{N} (\uparrow)$  (16)

The loads acting in the Horizontal plane are solved below.

Taking Moment about H<sub>2</sub>

$$(P_1 \times 496.25) + (H_1 \times 430) - (C_1 \times 350) - (C_2 \times 260) - (C_3 \times 170) - (C_4 \times 80) = 0 \quad (17)$$

$$(884.64 \times 496.25) + (H_1 \times 430) - (250 \times 350) - (500 \times 260) - (500 \times 170) - (250 \times 80) = 0 \quad (18)$$

$$H_1 = 270.94\text{N} (\downarrow) \quad (19)$$

Since the forces are in Equilibrium,

$$\text{Sum of all the forces acting Downwards} = \text{Sum of the forces acting upwards} \quad (20)$$

$$H_2 = 886.3\text{N} (\downarrow) \quad (21)$$

$$\text{Resultant Force on the Bearing 1} = B_1 = (V_1^2 + H_1^2)^{0.5} = (259.56^2 + 270.94^2)^{0.5} = 375.20\text{N}. \quad (22)$$

$$\text{Resultant Force on the Bearing 2} = B_2 = (V_2^2 + H_2^2)^{0.5} = (696.46^2 + 886.3^2)^{0.5} = 1127.20\text{N} \quad (23)$$

$$\text{Bearing Life} = (60 \times N \times \text{Operating Time}) / 10^6 \quad (24)$$

$$N = 150 \text{ rpm} \quad (25)$$

$$\text{Operating Period} = 3 \text{ years} = 8760 \text{ hours} \quad (26)$$

$$\text{Bearing Life} = (60 \times 150 \times 8760) / 100000 = 78.4 \text{ million revolutions} \quad (27)$$

$$\text{Load Carrying Capacity on Bearing 1} = 375.2 \times (78.4)^{0.33} \times 1.4 = 1947.20 \text{ N} \quad (28)$$

$$\text{Load Carrying Capacity on Bearing 2} = 1264.76 \times (78.4)^{0.33} \times 1.4 = 5849.99 \text{ N} \quad (29)$$

From SKP Bearing catalogue, Deep Groove Ball Bearing 6007 series with the following specifications has been selected.

Table 5. SKF DGBB 6007 Bearing specification for the top toothed roller

No.	Parameters	Results
1	Shaft Diameter	35mm
2	Tolerance	K6
3	Static Load carrying capacity	8800N
4	Dynamic Load carrying capacity	12500N
5	Maximum permissible speed	13000rpm

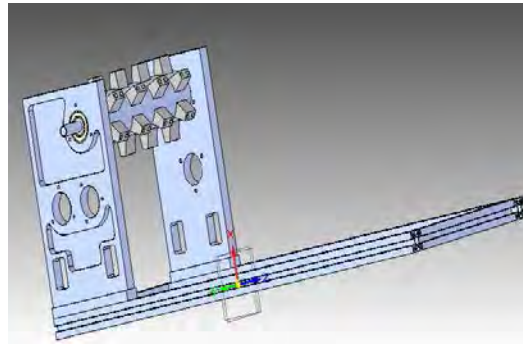


Fig.21. Top Roller Sub Assembly

### 3.4 Bottom fixed roller

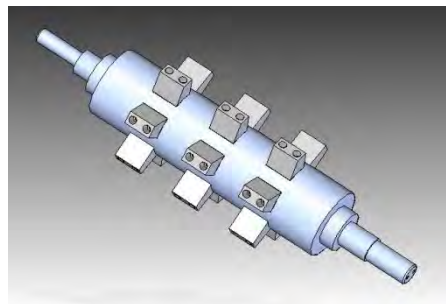


Fig.22. Bottom toothed dehusking roller

Even though the top and bottom toothed rollers share the same body dimensions, the teeth pattern are designed in such a way that the number of hardened teeth is more in top roller than the bottom roller. The top roller has 24 teeth throughout the circumference of the roller in such a way that 4 teeth are in a straight line. So, it will be on the 6 points on the circumference of the roller for every  $60^\circ$ . In contrary the bottom roller has 18 teeth throughout the circumference of the roller in such a way that 3 teeth are in a straight line. So, it will be on the 6 points on the circumference of the roller for every  $60^\circ$ . These 3 teeth fill the gap provided in between the teeth of the top roller. The pattern distinction is shown below.

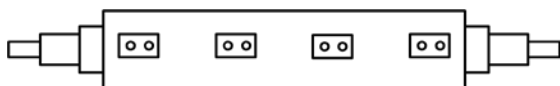


Fig.23. Teeth position in the Top roller



Fig.24. Teeth position in the Bottom roller

This pattern is designed in such a way that the coconut husk can be pierced effectively with minimum effort.

#### 3.4.1 Force analysis of the bottom fixed toothed roller assembly

The figure 25 represents the forces acting on the fixed bottom roller unit in the horizontal plane and the figure 26 represents the forces acting on the fixed bottom roller unit in the vertical plane

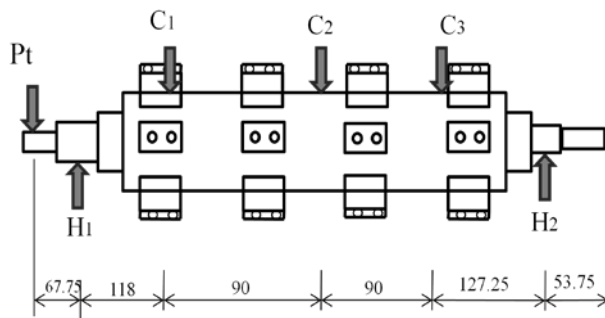


Fig.25. Forces acting on the bottom roller in the horizontal plane

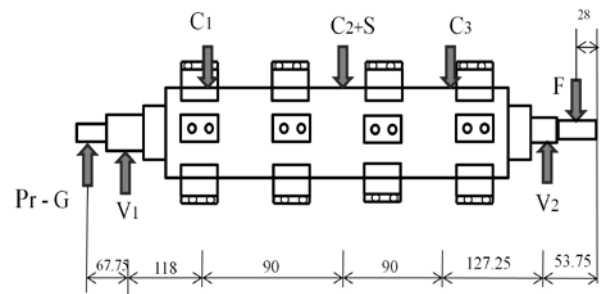


Fig.26. Forces acting on the bottom roller in the vertical plane

Since the 96 Gears used in both the top and bottom toothed rollers are sharing the same specification in all the aspects, the force calculation of the two gears are same.

Based on the following calculation suitable bearing grade for the bottom toothed roller has been selected.

The loads acting in the Vertical plane are solved below.

Taking Moment about V1

$$- [(Pr-G) \times 67.75] - (C_1 \times 118) - [(C_2 + S) \times 208] - (C_3 \times 298) + (V_2 \times 425.25) - (F \times 479) = 0 \quad (30)$$

$$- (309.98 \times 67.75) - (250 \times 118) - (710 \times 208) - (250 \times 298) + (V_2 \times 425.25) - (161.87 \times 479) = 0 \quad (31)$$

$$V_2 = 823.56 \text{ N } (\uparrow) \quad (32)$$

Since the forces are in Equilibrium,

$$\text{Sum of all the forces acting Downwards} = \text{Sum of the forces acting upwards} \quad (33)$$

$$V_1 = 238.33 \text{ N } (\uparrow) \quad (34)$$

The loads acting in the Horizontal plane are solved below.

Taking Moment about H1

$$[(Pt \times 67.75) - (C_1 \times 118) - (C_2 \times 208) - (C_3 \times 298) - (V_2 \times 425.25)] = 0 \quad (35)$$

$$(884.64 \times 67.75) - (250 \times 118) - (500 \times 208) - (250 \times 298) + (H_2 \times 425.25) = 0 \quad (36)$$

$$H_2 = 348.18 \text{ N } (\uparrow) \quad (37)$$

Since the forces are in Equilibrium,

$$\text{Sum of all the forces acting Downwards} = \text{Sum of the forces acting upwards} \quad (38)$$

$$H_1 = 1536.46 \text{ N } (\uparrow) \quad (39)$$

$$\text{Resultant Force on the Bearing 1} = B_1 = (V_1^2 + H_1^2)^{0.5} = [(1536.46^2 + 238.33^2)]^{0.5} = 1554.84 \text{ N} \quad (40)$$

$$\text{Resultant Force on the Bearing 2} = B_2 = (H_2^2 + V_2^2)^{0.5} = [(348.18^2 + 823.56^2)]^{0.5} = 894.14 \text{ N} \quad (41)$$

$$\text{Bearing Life} = (60 \times N \times \text{Operating Time}) / 10^6 \quad (42)$$

$$N = 150 \text{ rpm} \quad (43)$$

$$\text{Operating Period} = 3 \text{ years} = 8760 \text{ hours} \quad (44)$$

$$\text{Bearing Life} = (60 \times 150 \times 8760) / 10^6 = 78.4 \text{ million revolutions} \quad (45)$$

$$\text{Load Carrying Capacity on Bearing 1} = 1536.46 \times (78.4)^{0.33} \times 1.4 = 9184.95 \text{ N} \quad (46)$$

$$\text{Load Carrying Capacity on Bearing 2} = 894.14 \times (78.4)^{0.33} \times 1.4 = 5345.16 \text{ N} \quad (47)$$

From SKP Bearing catalogue, Deep Groove Ball Bearing 6007 series with the following specifications has been selected.

Table 6. SKF DGBB 6007 Bearing specification for the bottom toothed roller

No.	Parameters	Results
1	Shaft Diameter	35mm
2	Tolerance	K6
3	Static Load carrying capacity	8800N
4	Dynamic Load carrying capacity	12500N
5	Maximum permissible speed	13000rpm

It is evident from the above Bearing force calculations of both top and bottom toothed rollers, the desired parameters are identical.

### 3.5 Drive Shaft

The drive shaft carries a pulley and a 30 teeth spur gear with a bearing housing. It is used to transmit power from the pulley to the gearing. Suitable bearings have been selected for the same and its effect on the frame has also been analyzed.



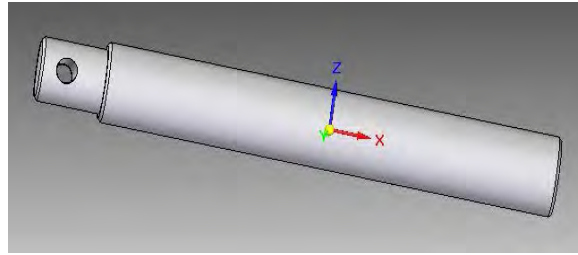


Fig.27. Drive Shaft

The following calculation is meant for selecting the suitable bearing grade based on the load acting on the drive shaft.

$$G\text{- Self weight of the gear} = 2.3\text{N} \quad (48)$$

$$\text{Self weight of the pulley} = 40\text{N} \quad (49)$$

$$\text{Self weight of the Housing} = 10\text{N} \quad (50)$$

### 3.5.1 Load calculation on the pulley

Let  $P_1$  and  $P_2$  are the tension force exerted by the pulleys

Suitable pulleys have been selected to transmit the required power from the motor to the fixed frame rollers.

The following are the specifications of the pulley

$$\text{Motor pulley diameter} = 78\text{mm} \quad (51)$$

$$\text{Drive shaft pulley diameter} = 232.4 \text{ mm} \quad (52)$$

The pulleys have been modeled according to existing pulley designs to prevent stress concentration, unbalance and to reduce weight at required places.

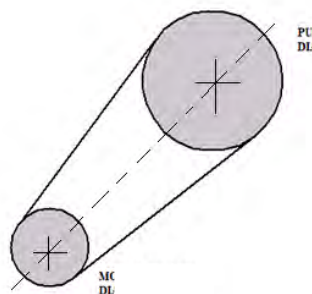


Fig.28. Line Diagram of pulleys in the main drive unit

$$\text{Diameter of the larger pulley, } D = 232\text{mm} \quad (53)$$

$$\text{Diameter of the smaller pulley} = 232 / 3 = 78\text{mm} \quad (54)$$

$$\text{Centre distance between the pulleys, } C = 400\text{mm} \quad (55)$$

$$\text{Surface velocity of the larger pulley} = \pi \times D \times N / 60000 = \pi \times 232 \times 1400 / 60000 = 17\text{m/s} \quad (56)$$

$$\text{Length of the belt} = L_B = (2C) + [\pi (D + d)/2] + [(D - d)^2/4C] = 2(400) + \pi (310)/2 + (154^2)/1600 \quad (57)$$

$$L_B = 1301.52\text{mm.}$$

$$\text{Input Power} = 0.74 \text{ kW} \quad (58)$$

$$\text{Correction factor } (F_a) = 1.2 \quad (59)$$

$$\text{Design Power} = 1.2 \times (0.74) = 0.9 \text{ kW} \quad (60)$$

From the Standard Design Data Book the followings are derived.

$$\text{Pitch Width} = 11\text{mm} \quad (61)$$

$$\text{Nominal Top Width} = 13\text{mm} \quad (62)$$

$$\text{Nominal Height} = 8\text{mm} \quad (63)$$

$$\text{Pitch Length} = L_p = [2C] + [\pi (D + d) / 2] + [(D - d)^2 / 4C] = 1430\text{mm} \quad [11] \quad (64)$$

$$1430 = 2C + \pi (310) / 2 + (154^2 / 4C) \quad (65)$$

$$C = 465.2\text{mm} \quad (66)$$

$$\text{New Correction Factor} = 0.96 \quad [12]$$

$$\text{Wrap Angle (or) Arc of Contact} = \alpha_s = 180 - [(D - d) / 2C] = 161^\circ \quad (67)$$

$$(P_1 - mv^2 / P_2 - mv^2) = e^{f\alpha / (\sin\theta/2)} \quad (68)$$

$$\text{Mass of the belt} = 0.5 \text{ kg/m} \quad (69)$$

$$\text{Groove angle for sheaves, } \theta = 34^\circ \quad (70)$$

$$\text{Coefficient of friction} = 0.2 \quad (71)$$

After calculation,  $P_1 - 6.82P_2 + 841 = 0$  (72)

We know that, Power =  $[(P_1 - P_2) \times v]/1000$  (73)

$P_1 - P_2 = 44.1$  (74)

On Solving (72) and (74),

$P_1 = 152.08 \text{ N}$  (75)

$P_2 = 196.18 \text{ N}$  (76)

Force exerted by the pulley in the Vertical Plane =  $\{(P_1 + P_2) \times \cos 30^\circ\} + \text{Self weight of the pulley} = 344.2 \text{ N}$  (77)

Force exerted by the pulley in the Horizontal plane =  $(P_1 + P_2) \times \sin 30^\circ = 174.13 \text{ N}$  (78)

The figure 29 represents the forces acting on the drive shaft unit in the horizontal plane and the figure 30 represents the forces acting on the drive shaft unit in the vertical plane

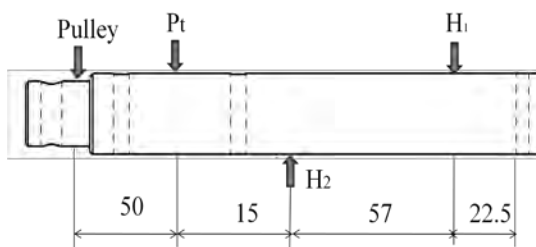


Fig.29. Forces acting on the drive shaft in the Horizontal plane

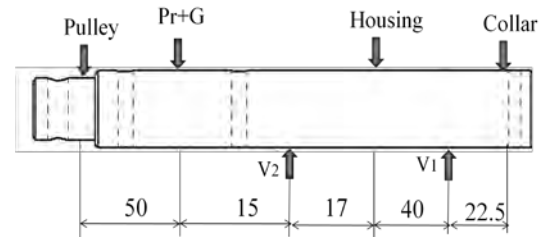


Fig.30. Forces acting on the drive shaft in the Vertical plane

**3.5.2 Radial and Tangential force of the 30 teeth gear**

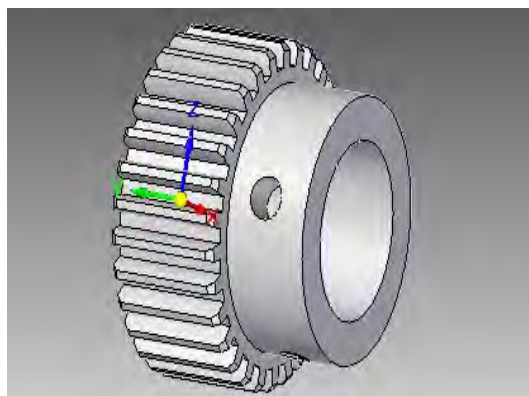


Fig.31. 30 teeth gear used in the drive shaft

A spur gear with 30 teeth is used, which transfers the drive from the drive shaft to the bottom toothed roller. The 30 teeth gear acts as drive pinion and the 96 teeth gear attached to the bottom roller acts as a driven gear.

The followings are the gear parameters of the 30 teeth spur gear.

$$\text{Gear Tooth, } Z=30 \quad (79)$$

$$\text{Module, } m=1.5 \quad (80)$$

$$\text{Pitch circle Diameter} = mZ = 45\text{mm} \quad (81)$$

$$\text{Speed of rotation of the gear } N= 480 \text{ rpm} \quad (82)$$

$$\text{Power of the motor} = 1 \text{ KW} \quad (83)$$

$$\alpha=\text{Pressure angle of the gear} = 20^\circ \quad (84)$$

$$\text{Torque transmitted by the gear} = M_t = 60 \times 10^6 \times \text{Power} / 2\pi N = 19904.45 \text{ Nmm} \quad (85)$$

$$P_t = 2M_t / \text{PCD} = 884.64 \text{ N} \quad (86)$$

$$P_r = P_t \tan \alpha = 321.98 \text{ N} \quad (87)$$

The following calculation is meant for selecting the suitable bearing grade based on the load acting on the drive shaft.

The loads acting in the Vertical plane are solved below.

Taking Moment about  $V_1$

$$(C_L \times 22.5) + (H_L \times 40) + (V_2 \times 57) + [(P_r + G) \times 72] + (P_v \times 122) \quad (88)$$

$$(-0.6 \times 22.5) + (10 \times 40) + (V_2 \times 57) + (324.3 \times 72) + (344.2 \times 122) = 0 \quad (89)$$

$$V_2 = 1153 \text{ N } (\uparrow) \quad (90)$$

Since the forces are in Equilibrium,

$$\text{Sum of all the forces acting Downwards} = \text{Sum of the forces acting upwards} \quad (91)$$

$$V_1 = 474.03 \text{ N } (\downarrow) \quad (92)$$

The loads acting in the Horizontal plane are solved below.

Taking moment about H<sub>1</sub>

$$-(H_2 \times 57) + (P_t \times 72) + (P_h \times 122) = 0 \quad (93)$$

$$-(H_2 \times 57) + (887 \times 72) + (174.13 \times 122) = 0 \quad (94)$$

$$H_2 = 1493.2 \text{ N } (\uparrow) \quad (95)$$

Since the forces are in Equilibrium,

$$\text{Sum of all the forces acting Downwards} = \text{Sum of the forces acting upwards} \quad (96)$$

$$H_1 = 571.1 \text{ N } (\downarrow) \quad (97)$$

$$\text{Resultant Force on the Bearing 1} = B_1 = (V_1^2 + H_1^2)^{0.5} = [(474.03^2 + 571.1^2)]^{0.5} = 742.2 \text{ N} \quad (98)$$

$$\text{Resultant Force on the Bearing 2} = B_2 = (H_2^2 + V_2^2)^{0.5} = [(1153^2 + 1493.2^2)]^{0.5} = 1886.5 \text{ N} \quad (99)$$

$$\text{Bearing Life} = (60 \times N \times \text{Operating Time}) / 10^6 \quad (100)$$

$$N = 480 \text{ rpm} \quad (101)$$

$$\text{Operating Period} = 4000 \text{ hours} \quad (102)$$

$$\text{Bearing Life} = (60 \times 480 \times 4000) / 10^6 = 115.2 \text{ million revolutions} \quad (103)$$

$$\text{Load Carrying Capacity on Bearing 1} = 742.2 (115.2)^{0.33} \times 1.2 = 4275.07 \text{ N} \quad (104)$$

$$\text{Load Carrying Capacity on Bearing 2} = 1886.5 (115.2)^{0.33} \times 1.2 = 10866.24 \text{ N} \quad (105)$$

From SKP Bearing catalogue Deep Groove Ball Bearing 6205 series with the following specifications has been selected.

### **3.6 Pressure Roller Unit**

It has two frames fixed to the machine skeleton. There is guide shaft connected between these frames and the main frames of the fixed roller unit. They are connected using fixed blocks. The pressure roller unit also has a

movable unit with two side plates holding three pressure rollers between them. They are held in such a way that the centers of the pressure rollers forms a slanting line such that at a given point of time each pressure roller is at different positions from the fixed rollers. The side plates have two blocks with phosphor bronze bushes to facilitate the sliding of the movable pressure roller unit. The guide rods are made of steel and have a diameter of 19 mm and are chamfered at the ends.

Table 7. SKF DGBB 6205 Bearing specification for the drive shaft

No.	Parameters	Results
1	Shaft Diameter	25mm
2	Tolerance	K6
3	Static Load carrying capacity	7100N
4	Dynamic Load carrying capacity	12000N
5	Maximum permissible speed	13000rpm

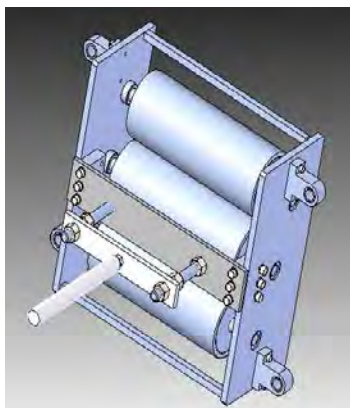


Fig.32. Pressure roller unit

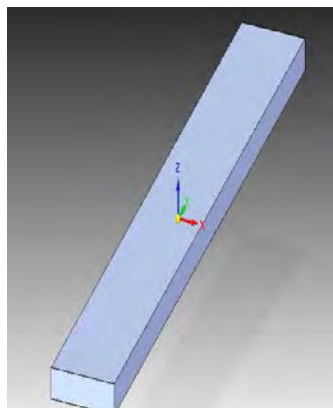


Fig.33. Support plate in the pressure roller unit

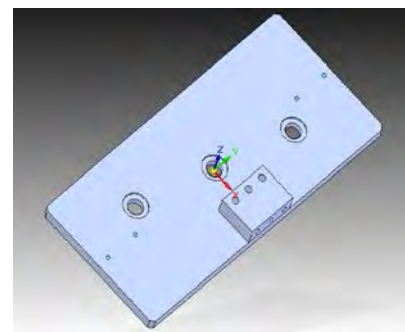


Fig.34. Supporting frame of the Pressure Roller unit

### 3.7 Flywheel

The flywheel is not a necessary component in the machine as the motor itself produces sufficient torque to de-husk the coconuts. Even otherwise, the flywheel has been added to the bottom roller to provide the extra energy whenever required. As, there is time interval between two successive coconuts, the power required is not always the same. The flywheel takes up the extra energy and provides it to compensate impacts which may require more torque.

The specification of the flywheel selected for this purpose are listed below

Flywheel's solid, mass = 16.5 kg

(106)

Diameter = 240 mm

(107)

Radius of Gyration = 68.22 mm

(108)

Mass Moment of Inertia = 0.075 kgmm<sup>2</sup>

(109)

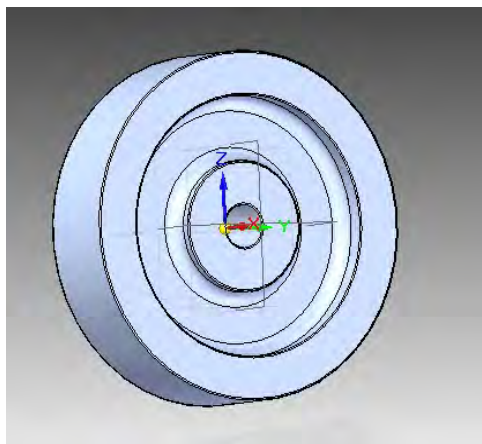


Fig.35. Flywheel

### 3.8 Drive calculation of the dehusking unit

From the experimental values Torque required for de-husking has been determined.

From that torque the drive reduction between the motor and the rollers has been determined.

Torque required to de husk a coconut = Force required shearing the coconut x Perpendicular distance

$$= 450 \times 80 = 36000 \text{ Nmm} = 36 \text{ Nm} \quad (110)$$

A factor of safety of 1.25 has been selected.

$$\text{So the Torque} = 45 \text{ Nm.} \quad (111)$$

$$\text{Power of the motor (P)} = 1 \text{ HP (0.74 KW)} \quad (112)$$

$$\text{Speed of the motor (N)} = 1400 \text{ rpm} \quad (113)$$

$$\text{Torque developed by the motor} = (60 \times P) / (2 \times 3.14 \times N) = (60 \times 0.75) / (2 \times 3.14 \times 1400) = 5.05 \text{ Nm} \quad (114)$$

$$\text{Torque required to de-husk the coconut} / \text{Torque supplied by the motor} = 45 / 5.05 = 9. \quad (115)$$

$$\text{Reduction ratio} = 9 \quad (116)$$

$$\text{Speed of the Fixed roller unit} = 1400 / 9 = 160 \text{ rpm (approx.)} \quad (117)$$

So, the de-husking rollers are designed to rotate at 160 rpm

#### **4. Design of the Crown removal unit**

The crown unit is a technology which has been developed to remove the crown of the de-husked coconut in case the coconut is required for oil production and cooking. It has two similar units on either side of the base frame skeleton. It is capable of moving to and fro in guide ways which are specifically provided for it. The strokes are adjusted in such a way that they will reach the smallest and biggest coconuts and also will not damage the coconuts or cause breakage as the pressure will be set in such a way that the pneumatic cylinders will act as springs. This unit will aid in removing the crown from the coconut and on completion, the coconut will be ejected out using another cylinder.

##### **4.1 V-block**

A V-block has been used to collect the de-husked coconut. The drop of the coconut is roughly around 100 to 150 mm. Different experiments have been tried dropping the de-husked coconut of different sizes onto the V-block in different orientations and have found out that it falls exactly on the V-block. In our case, the de-husked coconut will fall in only either of 2 orientations shown in the figures 41 and 42.



Fig.36. V-Block

##### **4.2 Motor and drive of the crown removal unit**

A motor attached with a gear box is used as the source of the drive. The output from the shaft is 90rpm. The



output spindle coming from the gearbox is attached to a pulley of diameter 40mm. A belt drive connects the pulley to another pulley of same diameter on a shaft which also carries the blades used to remove the crown of the coconut. The two shafts carrying the crown blades are connected by meshing spur gears which are having 35 teeth. There are two such crown removal units on both sides, as the de-husked coconut may fall in either of these two orientations. The crown removal unit has a frame on which it rests and guides on which it moves forward and backwards.

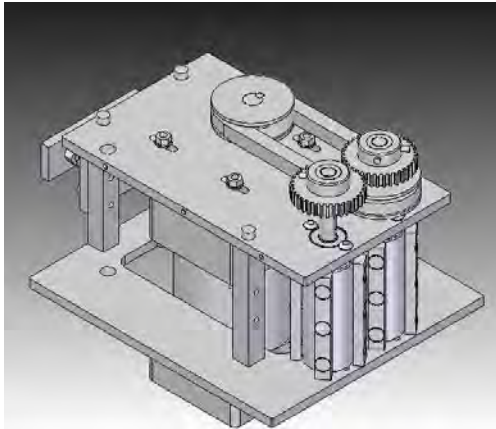


Fig.37. Crown removal unit drive system

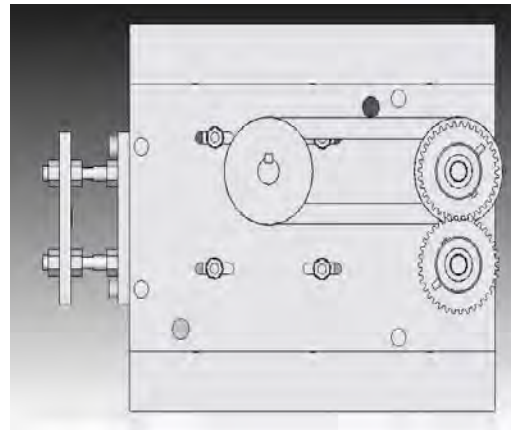


Fig.38. Crown removal unit drive system – Top view

The cutting edges are provided by the straight bar which are bolted to the plain roller as shown below in the figure 39 and standard GGM motor used for the drive is shown in the figure 40.

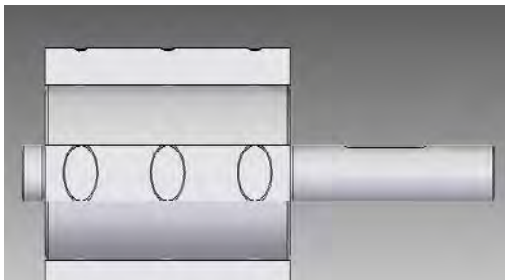


Fig.39. Crown removal unit Roller

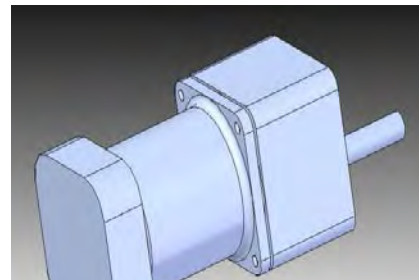


Fig.40. GGM motor used in crown removal unit

The two possible orientations of the falling coconut are shown below.

The crown can be removed by the two rotating identical rollers and the removed crown is collected separately at the bottom.



Fig.41. Coconut position on the crown removal unit when the crown is on the left side

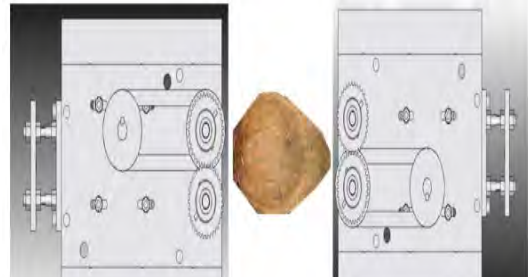


Fig.42. Coconut position on the crown removal unit when the crown is on the right side

The rollers are actuated in the linear direction by double acting Pneumatic cylinders of stroke length 150mm and rod diameter 40mm. Solenoid actuated direction control valves are used for flow control.

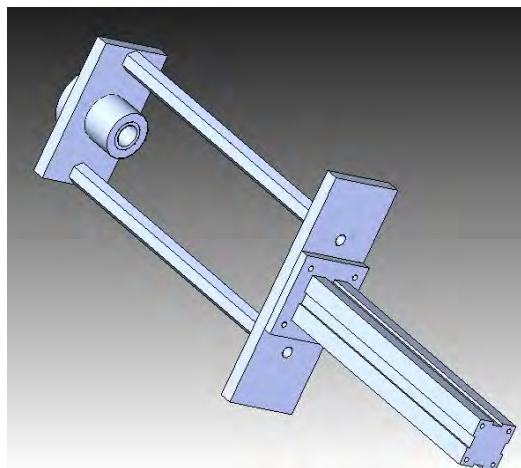


Fig.43. Pneumatic Cylinder holding assembly

The stroke length of the Pneumatic cylinder has been derived from the dehusked coconut dimensions collected from various sources as shown in the table 2. Based on those data the position of the pneumatic cylinder is fixed in the machine. The frames made up of Cast iron are used for the entire crown removal unit assembly. It has been designed to move to and fro within the guide ways.

Considerable amount of thickness has been incorporated to withstand the load exerted on it by the various components such as GGM motor, Gear box, pulleys, Rollers and as well as other external agents.

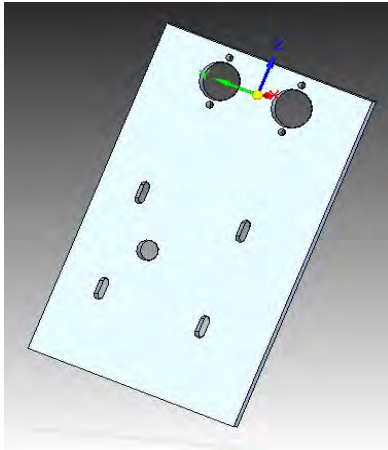


Fig.44. Top frame in the crown unit

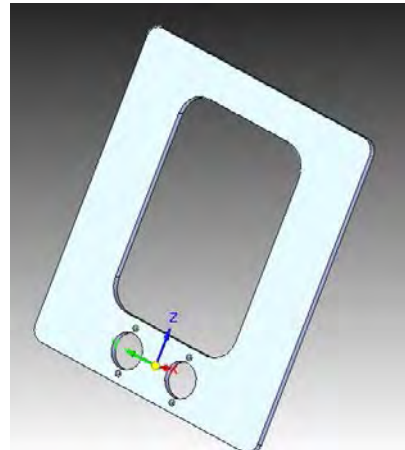


Fig.45. Bottom Frame in the crown unit

#### 4.3 Base Frame skeleton

The skeleton of the machine and the base has been contrived as a framed construction. The aluminium extrusion bars have been chosen for the frame . It has high strength to weight ratio and will not deflect or deform under heavy loads. The cross section is shown in figure 46.

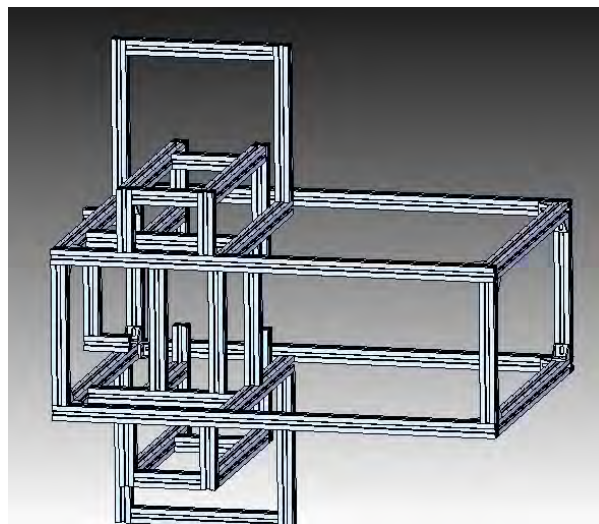


Fig.46. Skeleton of the frame unit

### 5. Assembled View of the machine

The entire parts of the machine are assembled with standard assembly tolerance values. The CAD views of the assembled machine are shown below.

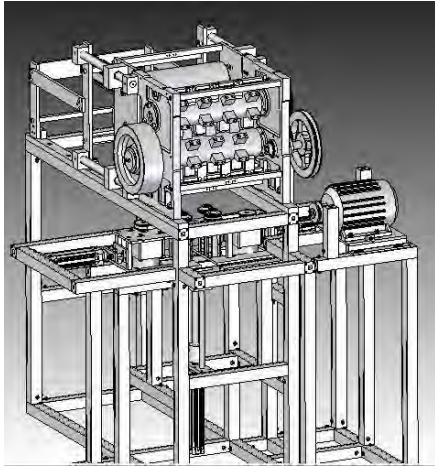


Fig.47. Solid works 3D view of the machine

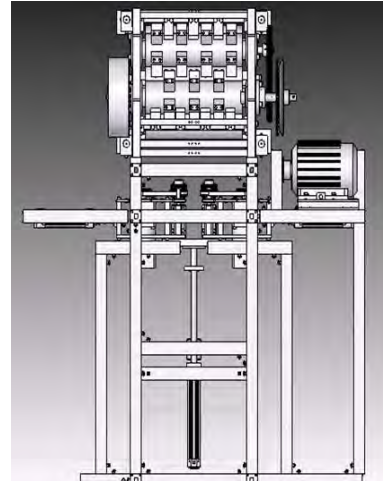


Fig.48. Side view of the machine

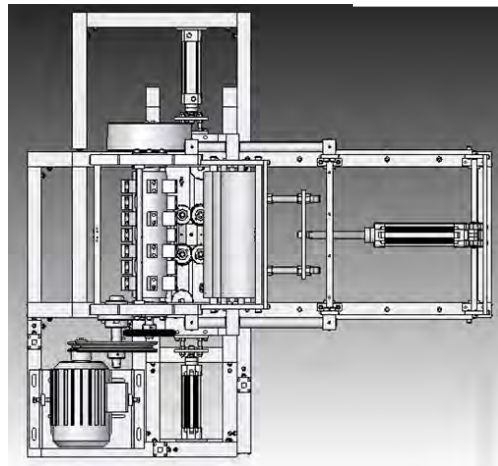


Fig.49. Top view of the machine

## 6. Structural analysis of the machine components

### 6.1 Introduction

The analysis of components of the machine elements has been done using ANSYS WORKBENCH v12.0. The static structural analysis has been done for the critical elements in the machine namely, the main frames which hold the rollers and also the drive shaft. The loads have been estimated from the experiments conducted and those values have been used to analyze the components.

Proper analysis procedure is followed by adding appropriate loads, arresting the degrees of freedom at required places and meshing using tetragonal mesh. The software has generated a report which suggests that the elements which have been analyzed are safe.

Table 8. Material specification of the machine components.

Component	Material	Density (kg/m <sup>3</sup> )	Strength (MPa)
Right main frame	Cast Iron	7200	35
Left main frame	Cast Iron	7200	35
Drive Shaft	Structural Steel	7850	90

### 6.2 Procedures adopted in the analysis

- Importing the component CAD drawing in the ANSYS workbench
- Incorporating appropriate meshing for each component. The meshing is auto generated by the ANSYS workbench.
- Pressure application on the machine components wherever applicable. The pressure value has been deduced based on the area of application and the force value derived from the bearing calculations.
- Application of displacement points on the component surface wherever applicable.
- Arresting the degree of freedom based on the real time scenario in the machine.
- The software has generated the stress distribution results for the component based on the pressure and constraint given.

### 6.3 Theories considered in the analysis

The following theories have been considered for acquiring the parameters involved in the analysis [10], [11].

At any point in a strained material, there are three planes perpendicular to each other and in those the plane which have zero shear stress is called Principal plane and the direct stress acting on that plane is called Principal stress. Here the shaft and the main frames are subjected to combined stresses due to the simultaneous action of tensile load or compressive load and shear stresses. So, it is subjected to simultaneous bending and torsion. In such cases, the maximum principal stress due to the combination of tensile or compressive stresses and shear stresses should be obtained. Since the main frames are made of a brittle material the following failure theory is applicable for the analysis.

#### 6.3.1 Maximum Principal Stress Theory

According to this theory, the failure occurs at a point in a member when the maximum principal or normal stress in a biaxial stress system reaches the ultimate strength of the material in a simple tension test.

#### 6.3.2 Maximum Principal Strain Theory

According to this theory, the failure occurs at a point in a member when the maximum principal or normal strain in a bi-axial stress system reaches the limiting value of strain of the material in a simple tension test.

### 6.3.3 Maximum Shear stress Theory

According to this theory the yielding occurs at a point in a member when the maximum shear stress in a bi-axial stress system reaches a value equal to the shear stress at yield point in a simple tension test.

### 6.3.4 Maximum Distortion Theory (Von Mises Theory)

According to this theory the yielding occurs at a point in a member when the distortion strain energy per unit volume in a bi-axial stress system reaches the limiting distortion energy (distortion energy at the yield point) per unit volume as determined from a simple tension test.

## 6.4 Analysis of the right dehusking frame

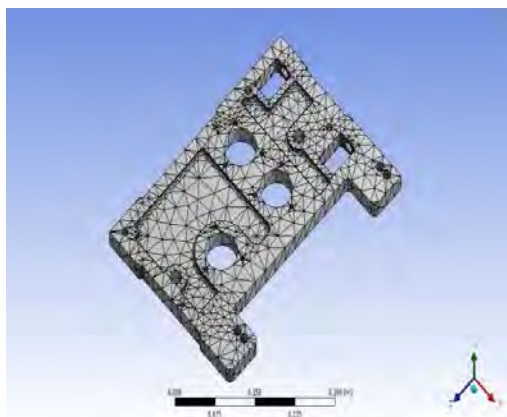


Fig.50. Meshing of the Right Frame



Fig.51. Pressure application on the right frame

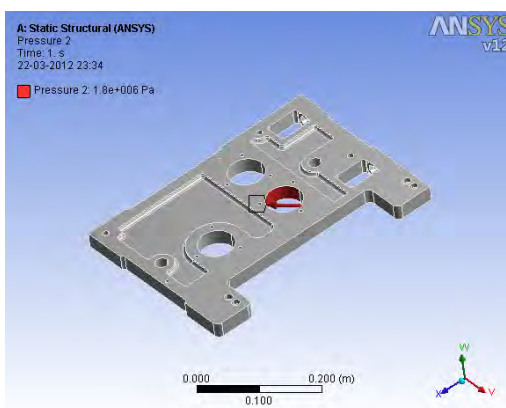


Fig.52. Pressure application on the bottom roller bearing area

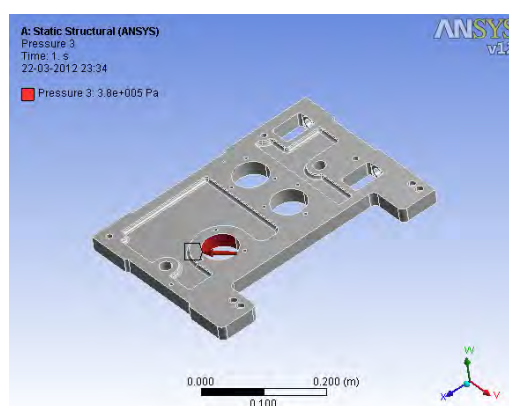


Fig.53. Pressure application on the top roller bearing area

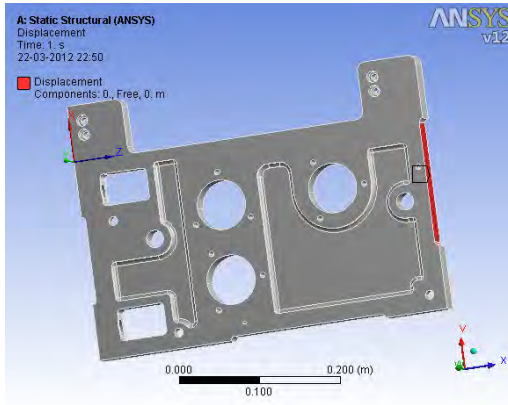


Fig.54. Displacement 1 on the right frame

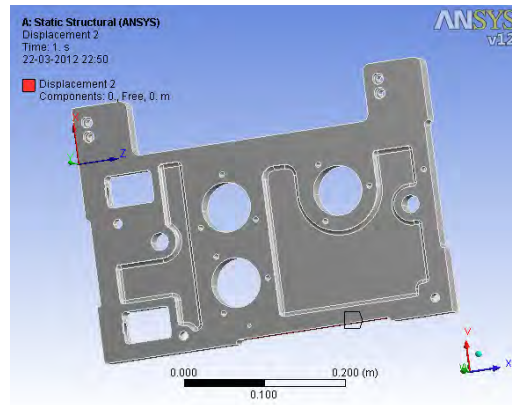


Fig.55. Displacement 2 on the right frame

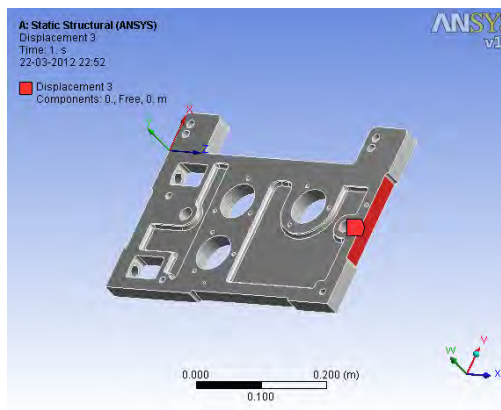


Fig.56. Displacement 3 on the right frame

According to ASME code for the design of Cast iron parts, the maximum permissible working stress in tension and compression is taken as 35 MPa [10].

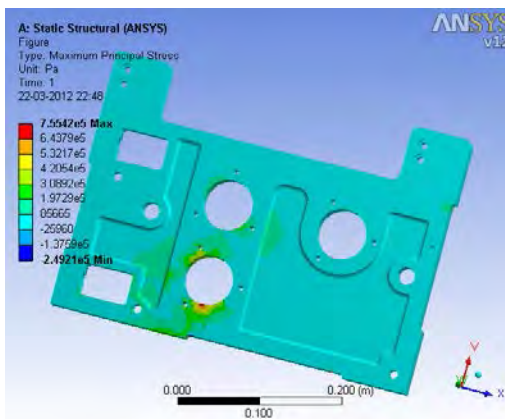


Fig.57. Maximum Principal Stress in the right frame

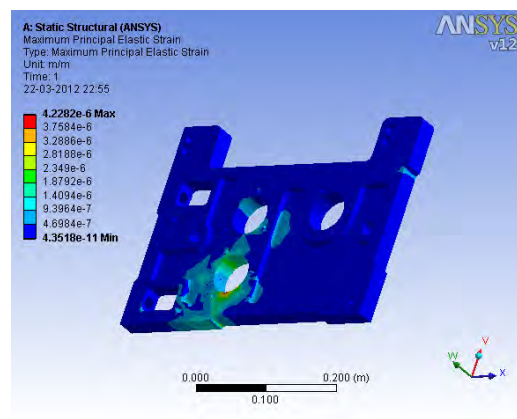


Fig.58. Maximum Principal Strain in the right frame

From the result the maximum Principal stress for the right main frame is 0.755 MPa which is well within in the safety limit when compared to the working strength of 35 MPa.

The maximum principal elastic strain value is  $4.23 \times 10^{-6}$  m which is almost negligible. So based on the stress stain analysis the design of the main right frame is safe.

### 6.5 Analysis of the left dehusking frame

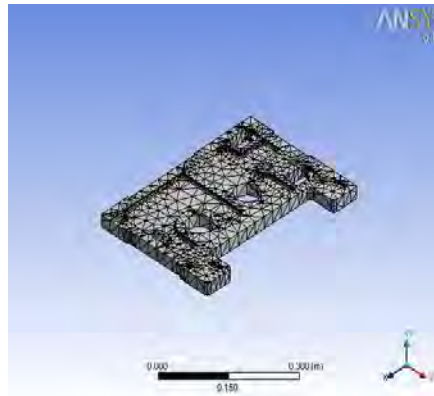


Fig.59. Meshing on the Left frame

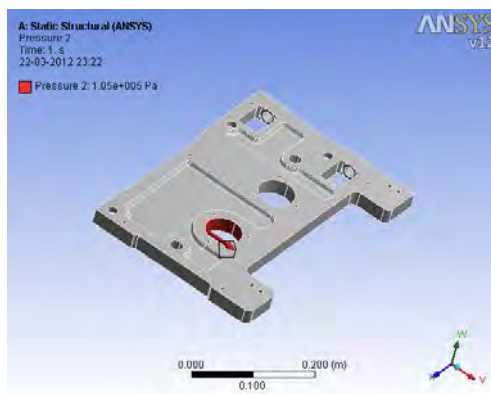


Fig.60. Pressure application on the top roller bearing area

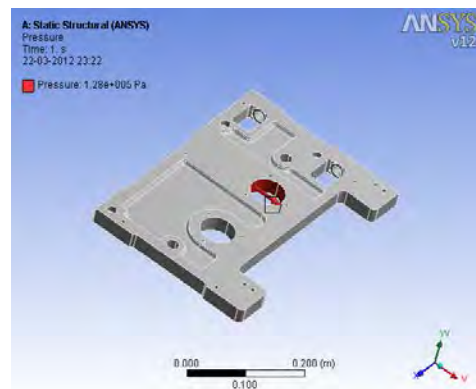


Fig.61. Pressure application on the bottom roller bearing area

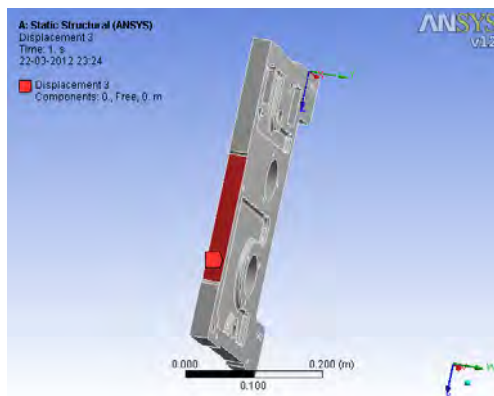


Fig.62. Displacement 1 on the Left frame

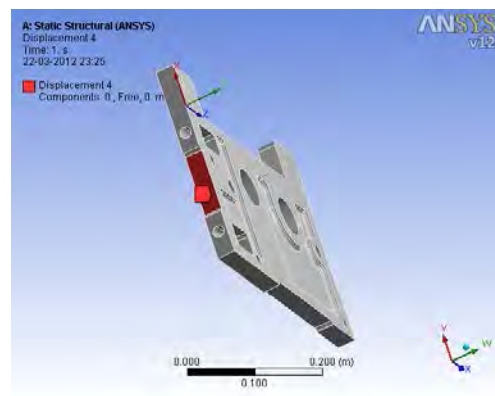


Fig.63. Displacement 2 on the Left frame



According to ASME code for the design of Cast iron parts, the maximum permissible working stress in tension and compression is taken as 35 MPa.

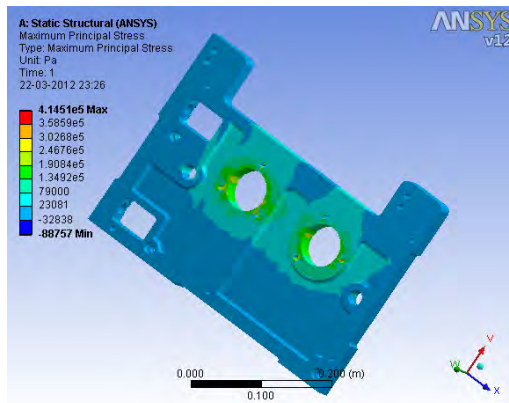


Fig.64. Max Principal stress in the left frame

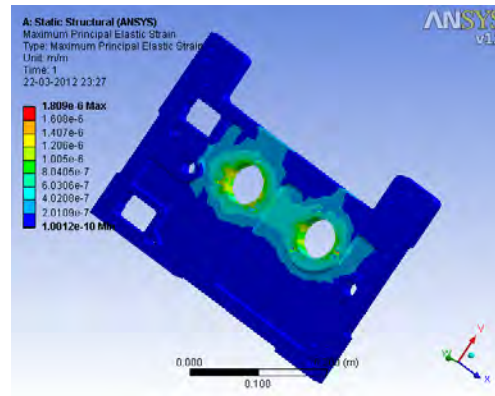


Fig.65. Max Principal Strain in the left frame

From the result the maximum Principal stress for the left main frame is 0.4145 MPa which is well within in the safety limit when compared to the working strength of 35 MPa.

The maximum principal elastic strain value is  $1.809 \times 10^{-6}$  m which is almost negligible. So based on the stress stain analysis the design of the main left frame is safe.

### 6.6 Analysis of the Drive shaft

Since the drive shaft is a made of ductile material maximum shear stress theory and maximum distortion theory are applicable in analysis in addition to the maximum Principal stress and maximum Principal strain theory.

According to ASME code for the design of power transmission shafts, the maximum permissible working stresses in tension and compression is be taken as 112 MPa and the maximum permissible shear stress is taken as 56 MPa [10].

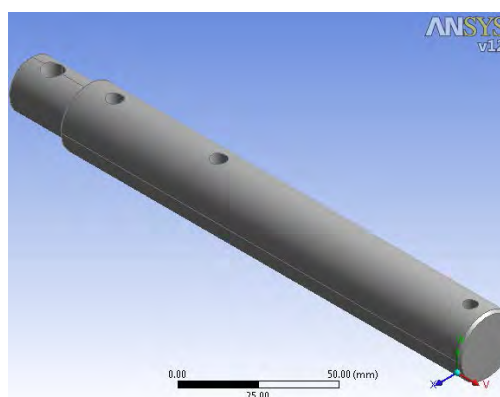


Fig.66. Drive Shaft

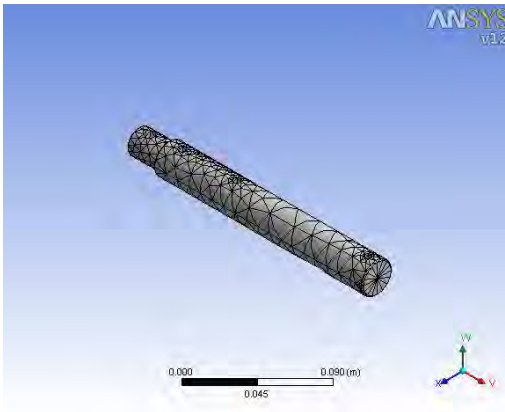


Fig.67. Meshing on the drive shaft

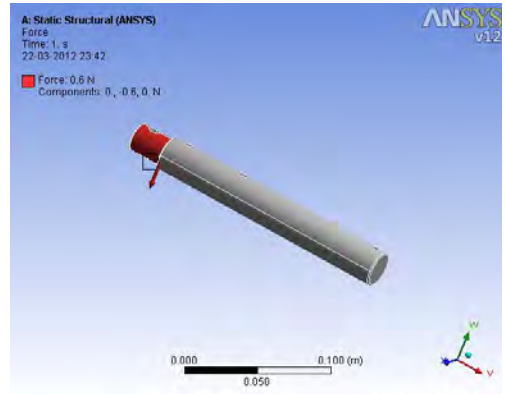


Fig.68. Force 1 on the shaft

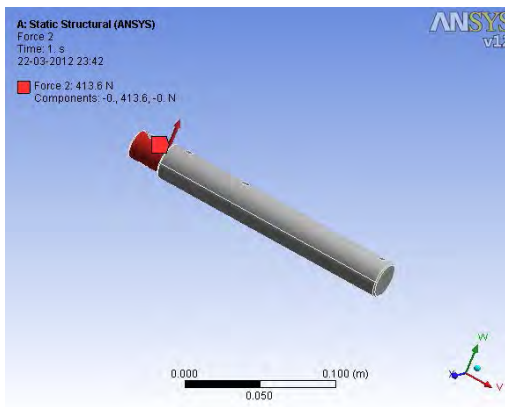


Fig.69. Force 2 on the shaft

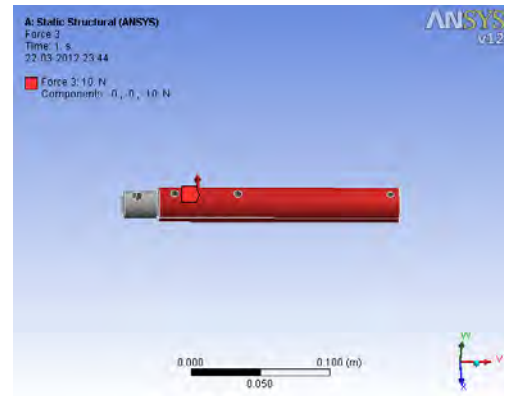


Fig.70. Force 3 on the shaft

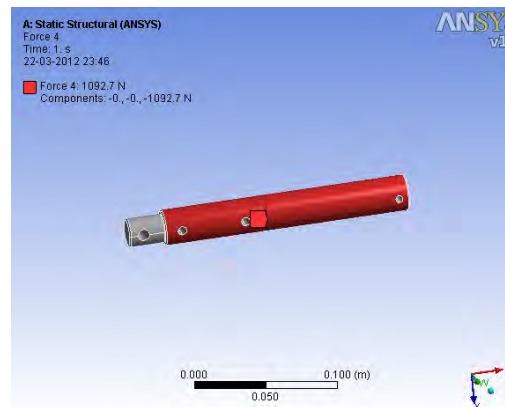


Fig.71. Force 4 on the shaft

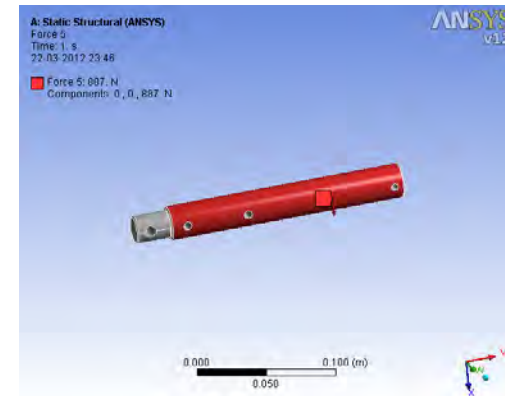


Fig.72. Force 5 on the shaft

Similarly forces are applied wherever required and the following structural results are obtained.

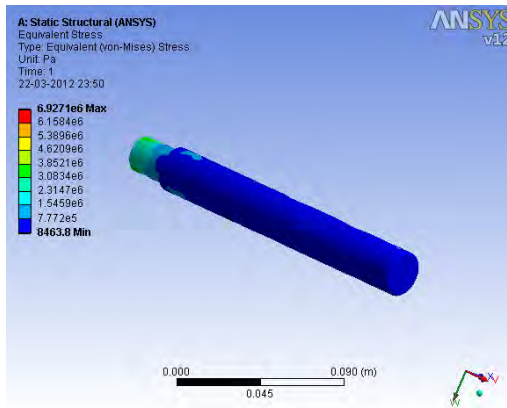


Fig.73. Von Mises stress in the shaft

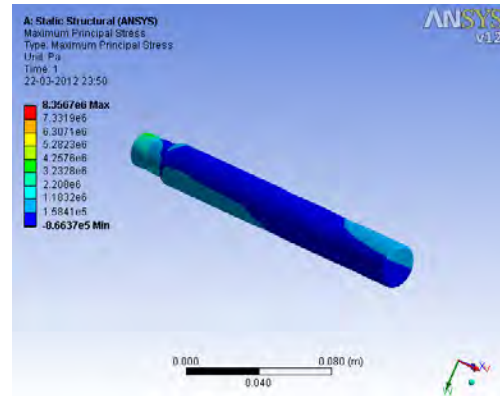


Fig.74. Max Principal Stress in the shaft

From the result the maximum Von-Mises stress for the drive shaft is 6.927 MPa which is well within in the safety limit when compared to the working strength of 112 MPa.

Also, from the result the maximum Principal stress for the drive shaft is 8.356 MPa which is well within in the safety limit when compared to the working strength of 112 MPa.

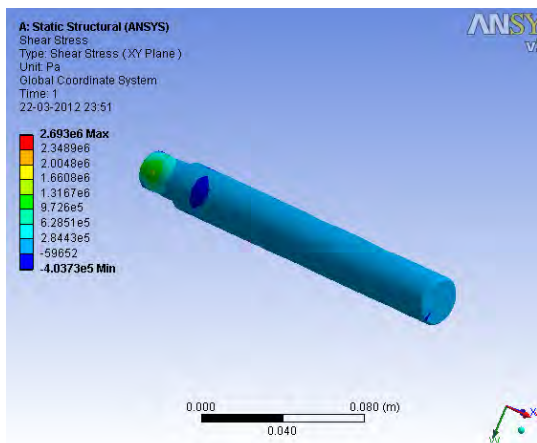


Fig.75. Shear Stress in the shaft

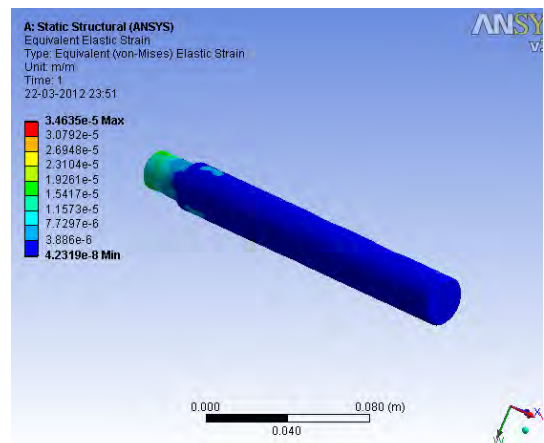


Fig.76. Von Mises Elastic Strain in the shaft

From the result the maximum shear stress for the drive shaft is 2.693 MPa which is well within in the safety limit when compared to the working strength of 56 MPa.

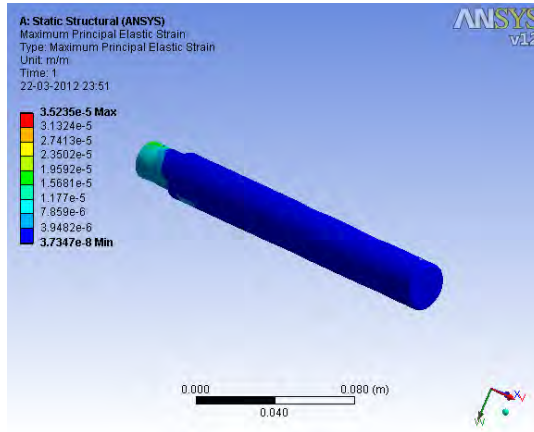


Fig.77. Max Principal Strain in the shaft

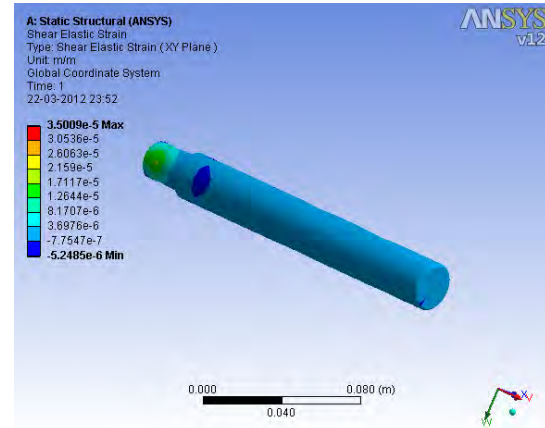


Fig.78. Shear Strain in the shaft

From the strain results all the strain values are negligible. Since, all the stresses are within the limits the design is considered as safe

### 7. Results and Discussion

From the analysis done using ANSYS, design of all the machine components are considered safe as far as the process is concerned. In order to validate the project scope, the dehusking capacity along with the crown removing capability of the machine has been evaluated. The experiment has been conducted in such a way that the machine has been operated by a different operator and the total number of the process time consumed has been measured using a stop watch. The capacity of the developed machine ranged from 190 to 220 nuts per hour depending on the operator and the size of the coconut, however on an average 210 nuts per hour.

Table 9. Results of the Capacity Evaluation of the machine

S/No.	Machine Output per hour	Cycle time of the machine (seconds)
1	194	18.56
2	207	17.39
3	216	16.67
4	212	16.98
5	221	16.28

### 8. Conclusion

An automated machine for coconut dehusking and crown removal has been developed for the small scale farm holders in the agricultural and rural areas. The operation of the machine is simple and the maintenance of the

machine is also not expensive. The machine can produce an average of 210 nuts per hour. Introducing this machine in the farm areas can reduce the risk involved in the use of spikes in dehusking the coconut and also eliminates the skilled manpower required for dehusking the coconuts. The machine can also be integrated along with the further processing steps of the nuts such as the production of copra.

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