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Design and Development of Beans (Phaseolus Vulgaris) Shelling Machine

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Abstract
The design and construction of a beans shelling machine, using locally available materials is presented. A detailed design of the various machine components was done, and the appropriate engineering materials were selected for various parts of the machine. A working drawing, detailing the machine components was done for the shop floor fabrication of the machine. The results of the performance evaluation carried out, showed that a shelling efficiency of 81.3% was obtained. The machine has a capacity of 100kg/hr of unshelled beans and a production cost of forty-one thousand, one hundred and fifty naira (₦41,150).
Keywords: Beans, Shelling Machine, Efficiency, Vibration.

1.0 Introduction

Shelling of beans is a post harvest operation carried out to remove beans grains from the pods. The need to reduce the stress and time associated with the manual method of processing {shelling} beans, gave rise to the conceptual design and construction of this machine. There are so many problems associated with the manual method of processing beans. These include breakages of the beans when shelling, and the introduction of foreign materials like stones and dirty particles. These can be eliminated when a motorized beans Sheller is used, since the beans pods will not be threshed on bare floor.

Ilori, et. al., studied about the effect of some ergonomic parameters namely; weight, age, height and arm length in relation to the resulting efficiencies: shelling efficiency, cleaning efficiency, mechanical damage and percentage loss of a hand powered Bean Sheller. Results from the study showed that the shelling efficiency increased with increases in weight of the operator and significantly with age and arm length.

Ashwin and Shaik, conducted a research on design, development and evaluation of a hand operated Bean Sheller. The Bean Sheller consisted of a cylinder and a concave. . It was observed by the author that for hand operated Bean Sheller at a moisture content of 12% w.b. and at a feed rate of 130kg/h, the shelling efficiency, unshelled percentage and visible damage was found to be 99.56%, 0.44% and 1.07%, respectively.

USDÀ agricultural engineers, designed an experimental Sheller that handles Bean gently during the shelling process. In laboratory tests with this device, Bean at 15% moisture content was shelled with no apparent damage to the kernels. However, the low durability of the belts, low capacity of the Sheller, and the decrease in shelling efficiency at high moisture are major problems of this shelling system.

Fox, designed another experimental Sheller based on the principle of rolling and squeezing. However, the rubber roller Sheller was reported to have feeding problems, and its shelling of unhusked ears was unsatisfactory. Fox did not report the shelling capacity of the rubber roller Sheller. A continuous type tamarind deseeder has been fabricated and evaluated by Karthickumar et al. The machine which was designed to have a capacity 75 kg·h-1, consisted of a seed separation unit and separate outlets for the seeds and deseeded fruits. Experimental evaluation was done with different operating conditions including different moisture content, varying wooden roller speed, feed rate and horizontal clearance. Results show that the machine has a maximum deseeding efficiency of 89.15% which is 93.34 % time saving and 74.9 % cost saving as compared to a manually operated machine.

Hiregoudar and Udhayakumar, studied the performance evaluation of traditional and mechanical methods of deseeding and reported that, maximum output was observed in the mechanical deseeder (16.07 kg/h) as compared to traditional method (2.31 kg/h) with higher mechanical damages to pulp and seed.

From previous works reviewed, it was observed that most of the earlier shelling machines developed where either not very efficient or viable enough to be commercialized. This article deals with the design and development of a more efficient locally fabricated beans shelling machine that could be commercialized.

2.0 Design Consideration

2.1 Design Consideration

The factors considered in the design of this machine include:

(i) Size of machine: length-1100mm, width- 500mm, height- 700mm.

(ii) Power requirement: source of power – electricity, prime mover – electric motor, power input- ≤ 3.0KW

(iii) Capacity of beans to be handled per unit time – 100kg/hr

(iv) Cost of production should be ≤ N50,000

2.2 Design Calculations for Beans Shelling Machine.

2.2.1 Mass of beans to be shelled per batch

Considering the capacity of the machine to be 100kg/hr and each batch of shelling expected to take an average time of 15 seconds, the mass of material to be processed (shelled) per batch is given by: (Hall et. al., 1980)

$$M_b = \frac{m_d \times T_b}{T_t} \quad \text{.. (1)}$$

Where,

m_d = mass of material to be processed in 1 hour = 100kg

T_b = Expected time of processing = 15 seconds

T_t = Total time required to process the expected quantity of the material = 1 hour = 3600 seconds.

M_b = Mass of unshelled beans on the machine at any time t, during machine operation.

From equation (1), mass of unshelled beans, M_b = 0.41667kg.

2.2.2 Mass of machine pulley on the shaft.

This is given by the relation: (Hall et. al., 1980)

$$M_p = \ell_p \times \mathrm{V_p} \quad \text{..(2)}$$

$$= \frac{\ell_p \times \overline{\wedge} \times d_p^2 \ \mathrm{t_p}}{4} \quad \text{..(3)}$$

Where,

ℓ_p = density of pulley = 6800kg/m^3

V_p = volume of pulley (mild steel)

d_p = diameter of pulley = 0.18m

t_p = thickness of pulley = 0.018m

From equation (2), the mass of pulley on the shaft is 3.115kg.

2.2.3 Mass of steel spikes

The mass of a spike tooth was measured with a scale to be 0.0011kg.Since there were 33 spike teeth welded to the shaft, the total mass of the spike teeth denoted by

M_s = 33 x 0.0011kg

M_s = 0.0363kg

2.2.4 Power requirement to drive the machine

The power requirement is a function of the total load on the shaft. (Hall et. al., 1980)

Thus, power is given by

$$P = F_T g V \quad \text{..(4)}$$

Where,

F_T = Total force on the shaft

P = Power required in watts

V = Velocity of the spike teeth for effective shelling.

g =acceleration due to gravity.

But,

$$V = \frac{\overline{\wedge}\,\mathrm{DN}}{60} \text{..(5)}$$

Where,

D = diameter of rotating spike teeth

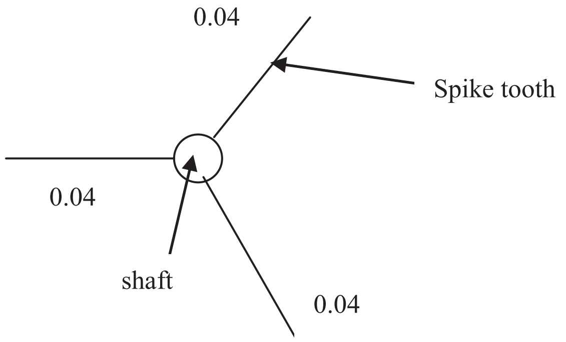

Fig.1: Schematic view of the shaft showing the spikes welded to it.
Total diameter = 0.1m
N = desired rpm of the spike teeth = 388.9 rpm
Thus,
$F_T = (M_b + M_p + M_s)$ ………………………………….(6)
Where,
M_b = mass of unshelled beans
= 0.41667kg
M_p = mass of pulley = 3.115kg
M_s = mass of spikes = 0.0363kg
Therefore,
Power = $F_T g V$ …………………………………………..(7)
From equation (3),the power requirement, P = 71.27watts
Using a factor of safety of 40%,
P = 71.27 x 1.4
= 99.78 watts
For design purpose, a 1-horse power motor with 1400 rpm was selected.

2.2.5 Size of the machine pulley

The desired size of the machine pulley is derived from the relation, (Hall et. al., 1980)
$N_1 D_1 = N_2 D_2$ ……………………………………………………………(8)
Where,
N_1 = rpm of electric motor pulley = 1400rpm
D_1 = diameter of electric motor pulley = 0.05m
N_2 = The desired speed of the machine pulley = 389.9rpm
D_2 = diameter of machine pulley
From equation (5), a pulley diameter of 180mm was selected.

2.2.6 Design of the central shaft

The mild steel shaft is transmitting 1 horse power (1hp) = 746 watts at 389 rpm carrying a load of 35.00179N that is, $(M_b + M_p + M_s)$ x 9.81= 3.56797 x 9.81 = 35.00179N, acting on a beam (shaft) of length 0.18m supported by two ball bearings and uniformly distributed loading.
Given that power, P = 746 watts, N = 389 rpm, w = 35.00179N, L = 0.18m,
Allowable shear stress of steel shaft f_s = 42N/mm^2, the maximum tensile or compressive stress of steel = 56N/mm^2
Let d be the diameter of the shaft
The torque transmitted by the shaft, (Khurmi and Gupta, 2005)

$$T = \frac{P \times 60}{2 \barwedge N} \qquad \text{……………………………………(9)}$$

From equation (6), the torque transmitted by the shaft, T = 18.31067N.m =18.3107 x 10^{2}N.m

The maximum bending moment of a simply supported beam with uniformly distributed loading is given by

$$M = \frac{wl^2}{8} \qquad \text{………………………………………………(10)}$$

Where,
w = weight on the shaft = 35.00179N
L = length of the shaft = 0.30m from measurement.

From equation (7), maximum bending moment, M = 0.3938N.m
Equivalent twisting moment, (Khurmi and Gupta, 2005)

$$T_e = \sqrt{m^2 + T^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(11)$$

From equation (8), equivalent twisting moment, T_e = 1831.49N.m
Also,

$$T_e = \frac{\pi}{16} \times f_s \times d^3 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(12)$$

$$d^3 = \frac{16\,T_e}{\pi\, f_s} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(13)$$

$$d = \left[\frac{16T_e}{\pi\, f_s}\right]^{1/3} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(14)$$

$$d = \left[\frac{16 \times 1831.49}{\pi \times 42}\right]^{1/3} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(15)$$

d = 6.06mm
We also know that the equivalent bending moment, (Khurmi and Gupta, 2005)

$$M_e = \frac{1}{2}\left[M + \sqrt{m^2 + T^2}\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(16)$$

$$= \frac{1}{2}\left[m + T_e\right] \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(17)$$

since $\sqrt{m^2 + T^2} = T_e \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(18)$

$$M_e = \frac{1}{2}\left[0.3938 \times 10^2 + 1831.49\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(19)$$

M_e = 935.435 N.mm

$$M_e = \frac{\pi}{32} \times f_b \times d^3 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(20)$$

(source:…)

$$d^3 = \frac{32 m_e}{\pi\, f_b} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(21)$$

$$d = \left[\frac{32 \times m_e}{\pi \times f_s}\right]^{1/3} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(22)$$

$$d = \left[\frac{32 \times 935.435}{\pi \times 56}\right]^{1/3}$$

d = 5.54 mm
Taking the larger of the two values of diameter, we choose 6.06 mm; but to avoid untimely failure, a 20mm solid shaft was selected.
Hence, the diameter of shaft chosen was 20 mm.

2.2.7 Second polar moment of area of the shaft.

The second polar moment of area of the shaft is given by, (Khurmi and Gupta, 2005)

$$\mathrm{J} = \frac{\pi\, d_s^4}{32} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(23)$$

Where,
J = second polar moment of area,
d_s = diameter of shaft
Substituting values
From equation (10), J = 1.571 x 10^{-8}m^4

2.2.8 Torsional stress on shaft

The torsional stress on the shaft is given by, (Khurmi and Gupta, 2005)

$$T = \frac{M_t d_s}{2\ \mathrm{J}} \text{...(24)}$$

Where,

M_t = twisting moment and is given by

$$Mt = \frac{\overline{\Lambda} d_s^2 L_s \ell_s g}{8} \text{...(25)}$$

d_s = diameter of shaft = 0.02m

L_s = Length of shaft required for the machine =0.45m

J = second polar moment of area

= 1.571 x 10^{-8} m^4

λ_s = density of shaft = 6800kg/m^3

Substituting values, into equation (11), T = 3001470.822N/m^2

Since this value is less than allowable shear stress of steel without key way (55 x 10^6N/m^2), the diameter ofshaft selected was adequate.

2.2.9 Torsional rigidity

The amount of twist or radial deformation in the shaft is calculated as, (Khurmi and Gupta, 2005),

$$\phi = \frac{2TL}{Gd_s} \text{...(26)}$$

Where,

ϕ = Angle of twist in radians

L = Length of shaft

T = Torsional stress

G = Modulus of rigidity of shaft

= 8.0 x 10^{10}N/m^2

d_s = diameter of shaft = 0.02m

Substituting values into equation (12), ϕ =0.00168 radian

This value is quite too low to cause shaft deformation when compared to the size of the shaft and imposed load.

2.2.10 Strain energy in the shaft

The strain energy in the shaft due to torsion is, (Khurmi and Gupta, 2005)

$$E_s = \frac{1}{2} T\phi \text{...(27)}$$

Where,

E_s = strain energy in joules

T = Torsion in shaft = 3.00147 x 10^6N/m^2

ϕ = radial deformation in shaft

= 0.001688 radian

Therefore, from equation (27), $E_s = \frac{3.00147 \times 10^6 \times 0.001688}{2}$ ....(28)

= 2533.24 Joules

= 2.53324 kJ

2.2.11 Size and length of V-belt required for power transmission

Since a 1 hp (horse power) motor is to be transmitted by the shaft, a standard v-belt with the following nominal cross-section in accordance with BS 1440: 1971 will be adequate.

Cross section = A

Nominal width = 13mm

Nominal thickness = 8mm

Minimum sheave diameter = 76mm

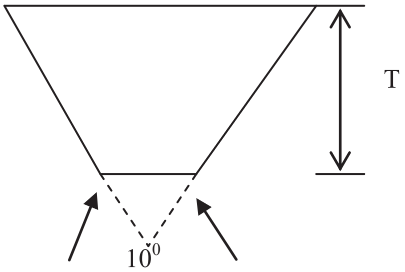

Fig. 2: Cross-section of a v-belt

2.2.12 The length of a v-belt required is calculated using the formular: (Khurmi and Gupta, 2005),

$$L_T = \pi \frac{(D_2 + D_1)}{2} + 2c + \frac{(D_2 - D_1)^2}{4} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (29)$$

Where,
L_T = Length of belt required
D_1 = Diameter of motor pulley = 0.05m
D_2 = Diameter of machine pulley = 0.18m
c = center to center distance of the machine and electric motor pulleys = 0.36

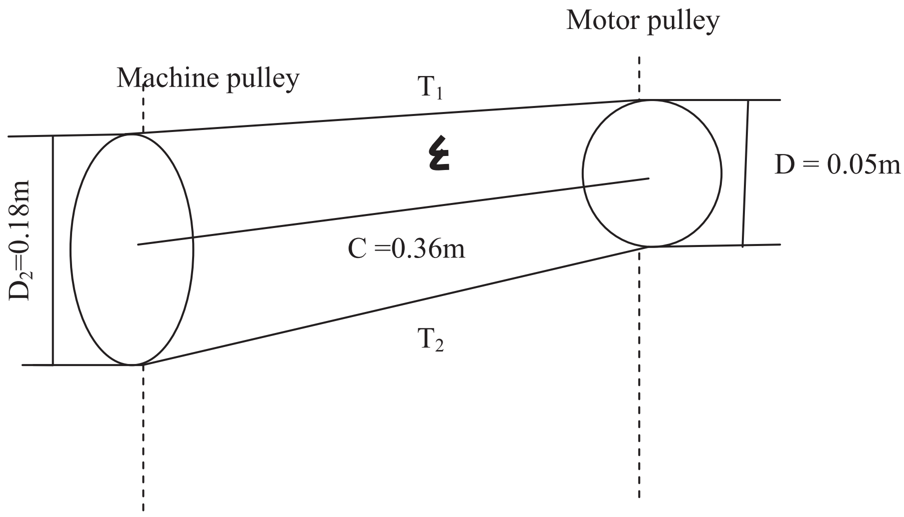

Fig. 3: sketch of power transmission system using a v-belt

Substituting values into equation (14), L_T = 1.085m

Due to size of belt available in the market and in order to make provision for belt adjustment, a 1.00m length of belt was selected.

2.2.13 Angle of contact between pulleys and the belt

The angle of contact between the belt drive and the driven pulleys is calculated as shown below:

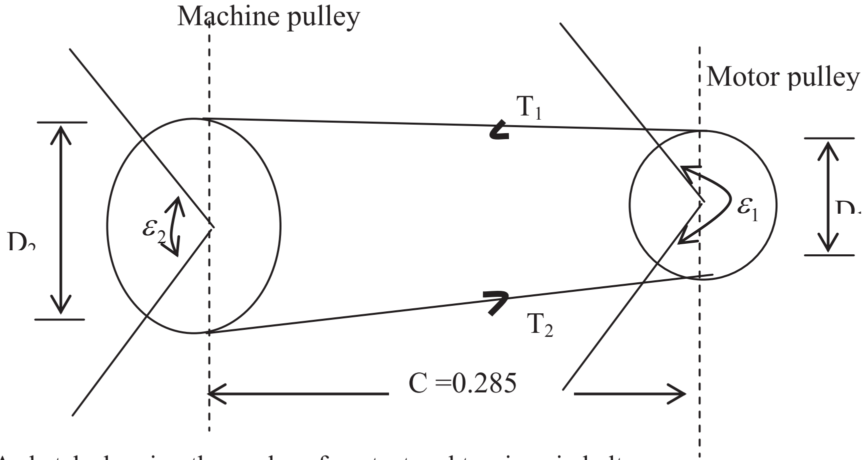

Fig. 4: A sketch showing the angles of contact and tensions in belt

$\varepsilon_1 = 180^0 - 2\beta$ ……………………………………(30)

From equation (15), $\varepsilon_1 = 153.6^0$

$\varepsilon_2 = 180^0 + 2\beta$ …………………………………… (31)

From equation (16), $\varepsilon_2 = 206.4^0$

2.2.14. Tension in belt

The tension (fig. 4) on both sides of the belt could be obtained from the following expressions: (Khurmi and Gupta, 2005)

$$\frac{T_1 - T_c}{T_2 - T_c} = e^{f\varepsilon} \text{..(32)}$$

Where,

T_1 = centrifugal tension which tends to cause the belt to leave the pulley and reduces the power that may be transmitted.

This is given by, mv^2

Neglecting centrifugal tension, T_c as compared with the tensions in the belt, equation (32) reduces to

$$\frac{T_1}{T_2} = e^{f\varepsilon} \text{..(33)}$$

Also,

$P = (T_1\text{-}T_2)v$ …………………………(34)

$$V = \frac{\overline{\Lambda} DN}{60} \text{…………………………...(35)}$$

Where,

D = diameter of smaller pulley

N = rpm of motor pulley

T_1 = effective tension on the tight side, N.

P = power transmitted = 746watts (1 hp).

T_2 = effective tension on the slack side, N.

$T_c = mv^2$ = centrifugal tension

m = mass of belt per unit length

m = (btp), kg/m^3

v = velocity of belt (m/s)

t = thickness of belt, m

f = coefficient of friction between belt and pulley = 0.03

ε = angle of wrap on smaller pulley in radian

The pulley with the smaller value of

$e^{f\varepsilon}$ governs the design.

That is, for the bigger pulley,

$$e^{f\varepsilon} = e^{0.3x\left[\frac{2064\overline{\Lambda}}{180}\right]} \text{..(36)}$$

$$e^{f\varepsilon} = 2.946$$

For the smaller pulley with $\varepsilon = 153.6$,

$$e^{f\varepsilon} = e^{0.3x\left[\frac{1536x\overline{\Lambda}}{180}\right]} \text{..(37)}$$

$$e^{f\varepsilon} = 2.235$$

Since this is the smaller value, this governs the design.

2.2.15. Velocity of belt

$$v = \frac{\pi DN}{60} \text{………………………………….. (38)}$$

Where,

D = diameter of smaller pulley = 0.05m

N = rpm of motor pulley = 1400rpm

Therefore, $v = \frac{\overline{\Lambda}x0.05x1400}{60}$ ....(37) m/s

v = 3.665m/s

Substituting values into equation (18), we obtain,

$T_1 - T_2 = 203.5$ …………………………………………….. (38)

From equation (33)

$$\frac{T_1}{T_2} = 2.235 \, (i.e. \text{ smaller value of } e^{f\varepsilon})$$

$T_1 = 2.235T_2$……………………………… (39)

Substituting equation (22) into (21), T_2 =164.8N

From equation (22), T_1 = 368.4

Therefore, pull on pulley and on shaft and bearing,

$T_p = T_1+T_2$, T_p = 533.2N

2.2.16 Selection of bearing

In the selection of bearings, the following conditions were considered;

I. Effect of high starting torque.
II. Ability of bearings to withstand or absorb combination of radial and thrust loads.
III. Ability of bearings to carry high overload for short period.

Based on the above conditions, two suitable medium ball bearings of bore diameter 20mm were selected to bear the load on the shaft that transmits power to the spike teeth.

2.2.17 Expected life of a selected bearing

The life of a ball bearing is determined from the formular, (Hall et. al., 1980)

$$L = \left(\frac{c}{p}\right)^3 \times \frac{10^6}{60 \times rev/\min} \text{...(40)}$$

Where,

L = expected life in hours = 15,000 hours

c = basic load rating = 13.5kN

p = equivalent load rating, N

Making P the subject of the relation,

$$L = \left(\frac{c}{p}\right)^3 x \frac{10^6}{60xrev/\min} \text{...(41)}$$

p^3 x L x 60 x rev/min = c^3 x 10^6

$$p = \left[\frac{c^3x10^6}{Lx60x\,rev/\min}\right]^{1/3} \text{...(42)}$$

p = 1917.1N

Since this load is greater than the actual load/pull on the bearings,

T_1+T_2 = 533.2N, the bearings selected will be able to carry the imposed load and be able to last the expected life of 15, 000hours.

3.0 Construction

The beans shelling machine has a rectangular frame made of angle iron bar welded together (see table 2). It has a length of 1100mm, width of 500mm and a height of 700mm.

A shaft was machined to 20mm diameter and cut with a power saw to a length of 300mm and mounted on two ball bearings. Spike teeth were then welded to the shaft in three rows and to the concave at intervals of 15mm. A 180mm diameter pulley was then bolted to the shaft at one end. A platform for mounting the electric motor of 1 Hp, 1400rpm was welded to the frame and a pulley of 50mm diameter was bolted to the shaft of the motor. A V-belt was now connected to both pulleys for power transmission.

On top of the shaft was placed the hopper made of steel sheets welded together. This can be screwed to the base with bolts and nuts. Just below the shaft, a slanting sheet was fixed so that the shelled beans can roll to the outlet for collection as they leave the threshing unit. All materials used and their costs are given in Table 1.

4.0 Operation
The dried beans should be put in the hopper while the machine is switched on. As the beans pods are hit by the spike teeth on the shaft against the ones on the concave, they are shelled. And roll down to the outlet where they will finally be collected with a container.

5.0 Performance Evaluation
In order to evaluate the performance of this machine, a known quantity of dried harvested beans were fed into the machine through the hopper. The machine shelling efficiency on each shelling operation was computed as:

Shelling Efficiency= $\frac{\text{Total Melon Shelled by Machine}}{\text{Total Melon fed in Machine}} \times 100$ ………………………… (43)

The efficiency was calculated to be 81.3%.

6.0 Conclusion and Recommendation
A locally made standard beans Sheller has been produced. The machine components include spike teeth, electric motor, bearings, shaft and hopper. The machine has an efficiency of 81.3 % with a 100 kg/hr capacity. The machine is affordable and can be commercially produced at a cost of N41, 150.00.
It is recommended that a blower be incorporated in future designs.

TABLE 1: Bill of materials on the design and construction of a beans shelling machine

Item	Description	Qty.	Unit price ₦	Amount ₦
1.	Angle iron bar (30mm x 30mm x 6100mm)	3	1100	3300
2.	Metal sheet (Gauge 1.2 x 1220mm x 2400mm)	1	2500	2500
3.	Shaft (300mm length and 20mm diameter)	1	3000	3000
4.	Ball bearings (20mm Diameter)	2	700	1400
5.	Pulleys (50mm diameter)	1	750	750
6.	Pulleys (180mm diameter)	1	1000	1000
7.	Steel electrodes	1 packet	1000	1000
8.	Electric motor (200/220v, 1Hp, 1400rpm)	1	8500	8500
9.	3-pin Plug (15Amps)	1	250	250
10.	Square pipe rod 20mm x 20mm x 6100mm	1	550	550
11.	V-belt (8mm x 13mm x 1000mm)	1	150	150
12.	Paint	1tin	150	150
13.	Labour	-	-	15, 000
14.	Bolts and nuts	8	75	600
15.	Miscellaneous	-	-	3000
Grand total				**₦41,150**

TABLE 2: Parts list of a beans shelling machine

PARTS	DESCRIPTION	QUANTITY REQUIRED
	Frame	1
	Shaft	1
	Hopper	1
	V-belt(8mm x 13mm x 1000mm)	1
	Outlet	1
	Electric motor (220/220v, 1Hp, 1400rpm)	1
	Bolts and nuts (M10,50mm)	8

Materials of construction are all mild steel unless stated otherwise.

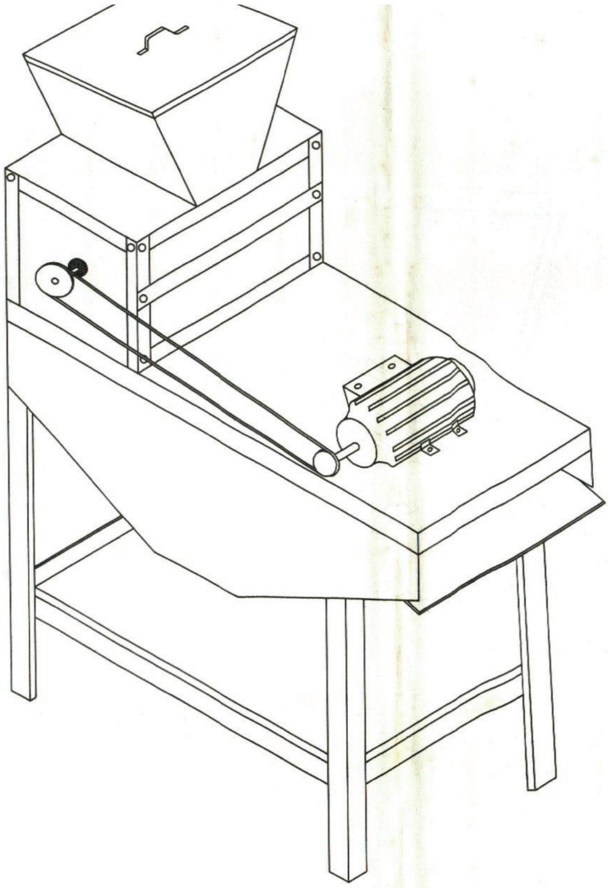

Figure 5: Isometric view of the beans shelling machine

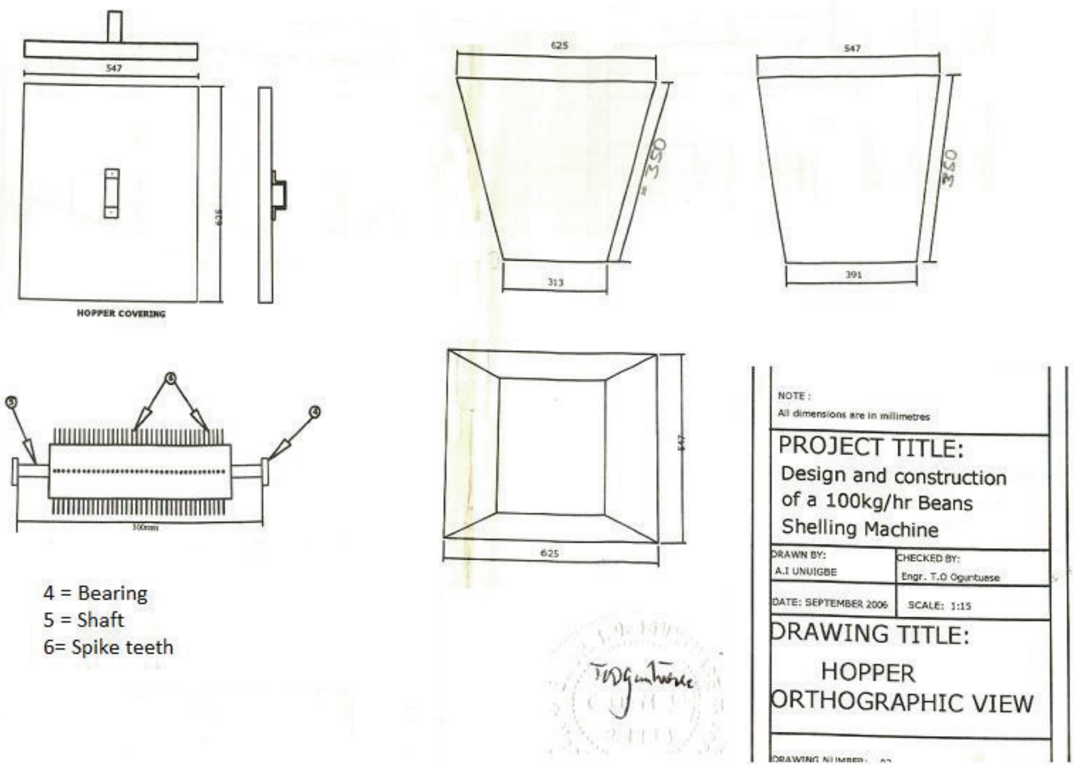

Figure 6: Hopper Orthographic View

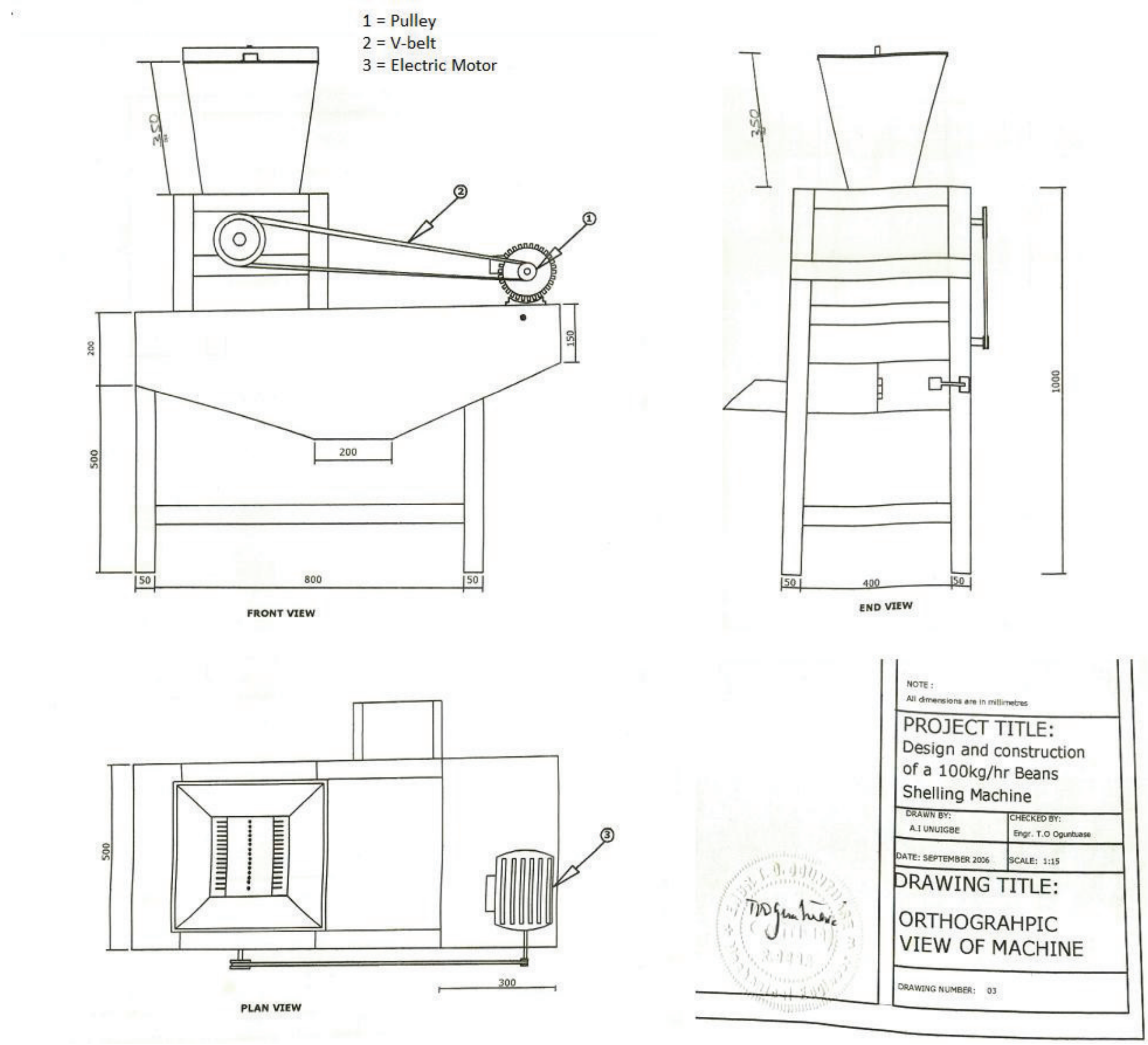

Figure 7: Orthographic view of Machine

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