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(476)

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Date: 24/7/ 1998

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(330)

DESIGN AND CONSTRUCTION OF UNIVERSAL GRAIN THRESHER

BY

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ABSTRACT

Nigeria as an agricultural country has many farm products such as maize, millet, sorghum, rice, groundnut, cowpea and many others which are always threshed in traditional ways. Labour and time are wasted in the processing and recovering of the end products and therefore sometimes yields are low.

As most universal grain threshers in Nigeria are imported, their costs are high and many farmers cannot afford them.

The need to reduce cost, labour and time also to use locally available raw materials necessitated the design and construction of the universal grain threshing machine being presented in this paper.

The major components of themachine are the threshing unit, sieves, blower and the prime mover, each fabricated separately and assembled on a rigid frame.

The thresher is powered either by an electric motor or an internal combustion engine and standard consumable parts are incorporated for ease of maintenance.

The simple working principle of the machine is by threshing action, this is accomplished by shredding and

rubbing action of the rotating stub spike toothed drum on a stationary concave with matching tooth. The harvested and dried crops are fed into the threshing chamber through the hopper.

The threshed grains and the chaff escape through the perforated concave plates on to a reciprocating sieve housing. The grains are cleaned by a blast of air being issued by the blower. The threshed and cleaned grains are then collected through the outlet channel.

Standard evaluation were done by loading 20kg of sorghum head. The procedure was repeated for millet and maize.

The results showed that the machine has an overall efficiency of 85.5%.

The machine can thresh from 2.5 to 5 tonnes daily depending on the type of crops.

INTRODUCTION

The types of crop grown in many parts of Nigeria are maize, millet, sorghum, rice, wheat and grain legumes such as cowpea, bambara nut and groundnut.

Processing of such crop after the harvest is of utmost importance because they must be processed to some extent before being sold or stored. Most of the processing are done traditionally. Labour, time and expenses involved in this traditional process are so huge that either the prices of such commodities rise high in the open market or profits to the producer become low.

Conventionally, there are different types of threshing machine which are imported to the country. However their prices are very high, such as much as ₦110,000.

The needs to reduce costs, labour and time as well as to use locally available raw materials have necessitated the design and construction of universal grain threshing machine reported in this paper. This work will entail these following design procedures. While a few important design methods will be outlined in this paper. The details of the design methods are described elsewhere. [1]

2.0 DESIGN METHODOLOGY

Synthesis of various new and old ideas are employed in this design so as to produce an overall new idea.

2.1 Functional Requirements

The machine is designed to shell maize and groundnut as well as thresh and winnow grain such as millet, sorghum, wheat, rice, cowpea etc.

To satisfy these requirements the machine has the following components.

(i) Threshing drum (ii) Concave (iii) Blower (iv) Group of Sieves (v) Frame and various drives (vi) Hopper (vii) Source of power.

The machine derives its power from a five (5) horse power (3.725 kw) single phase electric motor.

See Fig. 1a, b, c for the isometric drawing of the machine. — No dimension, no title, no component drawing.

2.2 Design Feature

Some of the important factors considered in design of this machine are

(i) High efficiency (ii) Minimum cost of construction
(iii) Ease and Safety of use (iv) Ease of inspection
(v) incorporated standard consumable parts
(vi) Ease of maintenance (vii) Adaptability to a wide variety of crops and (viii) Durability.

2.3 Power Transmission

In the design of this machine, power is first transmitted from the prime mover to the blower shaft by a flat belt drive.

2.4 Analysis and Design of flat belt drive

Transmission of power from output shaft to blower shaft.

Power generated by motor (P_1) = 3.725 KW

Motor Speed (N_1) = 1440 rpm

Angular Speed of motor (W_1) $= \frac{2\pi N_1}{60} = \frac{2\pi \times 1440}{60}$

$= 150.79$ rad/sec.

Motor pulley diameter (d_1) = 180mm

Blower speed (N_2) = 1000 rpm

Angular speed of blower (W_2) $= \frac{2\pi N_2}{60} = \frac{2 \times \pi \times 100}{60} = 104.73$ rad/sec.

Efficiency of flat belt drive

$$n = 0.96 \quad [1]$$

Power at the blower shaft (P_2) $= P_1 \times \eta$ [2]

$P_2 = 3.725 \times 10^3 \times 0.96 = 3576$ Watts

$P_2 = \underline{\underline{3.576 KW}}$

The diameter of the blower pulley (d_2) can thus be computed as follows:

$$d_2 = \frac{d_1 W_1}{W_2} = \frac{180 \times 150.79}{104.73} = 259.16mm$$

$$d_2 \simeq 250mm$$

To refine the angular speed of the blower shaft, assuming a relative speed loss factor E = 0.01, [1] the angular speed of the driven shaft is now

$$W_2 = \frac{d_1 W_1 (1-E)}{d_2} = \frac{180 \times 150.79(1-0.01)}{250} = 107.48 rad/sec.$$

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To determine the belt speed (V_1)

$$V_1 = \frac{W_1 d_1}{2} = \frac{150.79 \times 180}{2 \times 1000} = 13.57 \text{ m/s}$$

The belt speed lies within the recommended limit of 10 to 30 ms^{-1}.

To determine the centre distance (a_1) of the pulleys

$$a_1 \geqslant 2(d_1 + d_2) \qquad [3]$$

$$a_1 = 2(180 \text{x} 250) = 860\text{mm}$$

$$a_1 \approx \underline{900\text{mm}} \qquad [3]$$

To determine the required belt length (L_1)

$$L_1 = 2a_1 + (\pi/2)(d_1+d_2) + \frac{(d_2-d_1)^2}{4a_1} \qquad [3]$$

$$= 2(900)+(1.57)(180+250)+\frac{(250-180)}{4\text{x}900} = 2476.5\text{mm}$$

$$L_1 = 2.48\text{m}$$

The angle of lap of thebelt on the smallerpulley is given as thus, $\alpha_1 = 180^\circ - 57^\circ \frac{(d_2-d_1)}{a}$ 3

$$= 180-57 \frac{(250-180)}{900} = 175.56^\circ$$

The angle of lap for smaller pulley should not be less than 150° for an effectivedrive.

The angle of lap of the belt on the larger pulley is also given as thus: $\alpha a_1 = 180^\circ + 57^\circ \frac{d_2 d_1}{a}$ [3]

$$a_1 = 180^\circ + 57^\circ \frac{(250-180)}{900} = 184.44$$

By using the rubberished fabric belt, thethickness must meet its requirement.

The belt thickness $b_1 = \frac{d_1}{30} = \frac{180}{30} = 6.0mm$

The dimension for a ply of a rubberished belt including the interlayer (l_y) is 1.5mm [3] and should not exceed 7 plies. Therefore the number of plies is thus computed.

$$N_{ply} = \frac{b_1}{l_y} = \frac{6.0}{1.5} = 4 \text{ plies}$$

The belt width (b_o) can be computed from the following formular.

$$b_o = \frac{F_t}{[\sigma]\, b_1} \qquad [4]$$

Where F_t = Force, N

b_1 = Belt thickness

$[\sigma]$ = allowable effective stress

$$F_t = \frac{P_1}{V} = \frac{3.725 \times 10^3}{13.57} = 274.5N$$

$$[\sigma] = \sigma_{eff} \times C_o \times C_\alpha \times C_v \times C_s \qquad [3]$$

where eff = nominal effective stress determined under standard condition. For rubberished fabric belt with tensile pre-stress (σ_o) = 1.8 MPa. [3]. The nominal effective stress eff. is determined according to the ratio $\left(\frac{d_1}{b_1}\right)$.

$$\frac{d_1}{b_1} = \frac{180}{6} = 30$$

$$\sigma_{eff} = 2.17 \text{ MPa if } \frac{d_1}{\delta} = 30. \qquad [3]$$

The coefficient (C_o) takes care of the mutual arrangement for the shafts.

As reported by [1], when the drive from the prime mover to the blower shaft lies within the range of 80° to 90° to the horizontal, C_o = 0.8.

The coefficient (C_α) accounts for the effect for the angle of contact for the smaller pulley.

If α_o = 175.56°, we can round it up to 180° which corresponds to the value of C = 1. [1]

The coefficient C_v accounts for the influence of the tension due to centrifugal force which tends to reduce traction. It depends on the periperal speed. When the speed V = 13.57 ms^{-1}, it can be rounded upto 15 ms^{-1} which corresponds to the value of C_v = 0.99 [1]

The coefficient C_s is the service factor which takes care of the type of driven machine and the daily times in operation.

The value is to be reduced by 0.1 in two shift operation and by 0.2 in three shift operation. [1]

$$C_s = 0.9 - 0.1 = 0.8$$

Thus $[\sigma]$ = 2.17x0.8x1x0.99x0.8=1.375 MPa

The belt width can thus be computed $b_o = \frac{F_t}{[\sigma] b_1} = \frac{274.5}{1.375 \times 6} = 33.27$mm

As reported by [3], the appropriate standard belt width is 32mm. However such dimension is not available in our market here. Therefore the standard size available is 50mm For this design, it was accepted.

Belt width = 50mm

and the pulley width (B) = 63mm

The contact surface of therun ofthe pulley is crowned to prevent the flat belt running off in operation. As a rule a crown is provided by an arc of a circle. The camber can be determined from the approximately standard table. [1]

Camber for the pulley is 0.8mm.

Determination of the pre-tension in the belt

$F_o = \delta_o b_o b_1 = 1.8 \times 50 \times 6 = 540N$ [4]

Where δ_o = the tensile pre-stress in the belt

To find the force acting on theshaft and the bearings.

$F_a = 2F_o \sin \frac{\alpha_1}{2}$ [4]

$F_a = 2 \times 540 \times \sin \frac{175.6}{2}$

$= 1079.2N$

To find the maximum force applied to the shaft due to belt tensioning. $F_{bmax} = 1.5F_o = 1.5 \times 540 = 810N$

2.5 Analysis and Design of Flat belt drive power transmission to the thresher drum

The thresher drum derives its power from the blower shaft, with belt drive inclination of 20^o to the horizontal.

Power available from the blower shaft (P_2) = 3.576KW

Flat belt drive efficiency η = 0.96

Power available at the drum shaft $(P_3) = P_2 \times \eta$ [2]

$P_3 = 3.576 \times 0.96 = 3.433$ KW

The drum is designed to run at a speed of not more than 600 rpm. Higher drum speeds damage grain in the threshing process.

Angular speed of drum (W_3) $= \frac{2\pi \times 600}{60}$

$$W_3 = 62.8 \text{ rad/sec.}$$

To determine the drum pulley diameter (d_3)

$$d_3 = W_2 d_2 / W_3 = \frac{107.48 \times 250}{62.8} = 427.86\text{mm}$$

Select standard value for the pulley diameter (d_3). [4]

The nearest standard size is 400mm.

$$d_3 = \underline{400\text{mm}}$$

To refine the angular speed of the drum shaft assuming a relative speed loss factor E = 0.01.

The angular speed of the driven shaft is

$$W_3 = \frac{d_2 W_2 (1-E)}{d_3} = \frac{250 \times 107.48(1-0.01)}{400} = 66.5 \text{ rad/sec.}$$

To determine the belt speed (V_2) $= \frac{W_2 d_2}{2}$

$$= \frac{250 \times 107.48}{2 \times 10^3} = 13.44 \text{ m/s}$$

The belt speed lies within the design limits of 10 to 30 m/s.

To determine the centre distance of the two pulleys

$$a_2 \geqslant 2(d_2+d_3) \qquad [3]$$

$$a_2 \geqslant 2(250 + 400) = \underline{\underline{1300\text{mm}}}$$

To determine the required belt length (L_2)

$$L_2 = 2a_2 + (\pi/2)\ (d_2+d_3) + \frac{(d_3-d_2)^2}{4a} \qquad [3]$$

$$= (2x1300)+(1.57)(250+400)+\frac{(400-250)^2}{4x1300}$$

$$= 3625mm$$

$$L_2 = 3.625m$$

Deu to the compactness of the machine this dimension is not feasible.

The angle of lap of the belt on the smaller pulley

$$\alpha_2 = 180^o - 57^o \frac{(d_3-d_2)}{a} \qquad [3]$$

$$= 180^o - 57^o\ \frac{(400-250)}{1300} = 173.4^o$$

The angleof lap on the larger pulley is

$$\alpha_{2a} = 180^o + 57^o\ \frac{400-250}{1300} = 186.6^o$$

Using the rubberished fabric belt, the thickness meet its requirement.

The belt thickness (b_2)

$$b_2 = d_{2/30} = {}^{250}/30 = 8.33mm$$

The dimension for a ply of a rubberished belt including the inner layer is 1.5mm and should not exceed 7 plies. Therefore the number of plies of the belt can be determined

$$N_{ply} = \frac{8.33}{1.5} = 5.5$$

$$N_{ply} \simeq 6 \text{ plies}$$

The belt width can be computed from the following formular

$$b_o = F_t/([\sigma]\ xb1) \qquad [4]$$

The terms involved had been defined earlier.

$$F_{t_1} = P_2/V_2 = \frac{3.433 \times 10^3}{13.44} = 255.4N.$$

$$= \delta eff \times Co \times C_\alpha \times Cv \times Cs \qquad [3]$$

The above terms are defined earlier and their corresponding values can be obtained from [1].

$$d_2/b_2 = \frac{250}{8.33} = 30.012$$

Then $\delta_{eff} = 2.17$ MPa

$C_o = 1$

$C = 0.97$

$C_v = 0.99$

$C_s = 0.8$

Thus $[\sigma] = 2.17 \times 1 \times 0.97 \times 0.99 \times 0.8 = 1.67$MPa

The belt width can thus be computed

$$b_o = F_t/[\sigma]b_1 = \frac{255.4}{1.67 \times 8.33} = 18.36 \text{ mm}$$

This dimension of the belt is not realistic compared to its duty and moreover such width for a flat belt is not available around. In this case the belt width to be assumed is 50mm.

Therefore belt width $(b_o) = 50$mm

To determine the pretension on the belt

$$F_o = \delta obob1 = 1.8 \times 50 \times 8.33 = 750N$$

To find the force acting on the shaft and bearings

$$F_b = 2F_o \sin \frac{\alpha_2}{2} \qquad [4]$$

$$F_b = 2(750) \sin \frac{17.3.4}{2} = 1499.5N$$

To find the max force applied to the shaft due to belt tensioning.

$$F_{bmax} = 1.5F_o = 1.5 \times 750 = 112.5N$$

Diameter of pulley = 400 mm.

Corresponding camber for run width for upto 125 mm is given 1.0mm. [2]

3.0 DESIGN AND ANALYSIS OF POWER TRANSMISSION SHAFT

3.1 Determination of Blower Shaft Diameter

The blower shaft diameter (ds) can thus be determined as follows:

$$d_2^3 = \left\{\left(\frac{16}{\delta s}(K_t\ M_t)^2 + (K_b\ M_b)^2\right)\right\}^{1/2}$$

Where K_t and K_b = combine shock and fatique factors.

δs = allowable stress for shafts having keyways

= 40 MN/m^2 [4]

M_t and M_b = the torsional and resuctact bending moments respectively

$$M_t = \frac{9500 x kw}{rpm};\ N_m \quad [4]$$

M_b = 73.38 Nm from the bending moment diagrams [1]

$$d_s^3 = \frac{16}{40x10^6}\left\{(1x34.15)^2+(1.5x73.03)^2\right\}^{1/2} \quad [4]$$

d_s = 0.024m

d_s = 24.45mm selecting a factor of safety 1.23 [1]

d_s = 24.45x1.23 = 30.06m

The thresher shaft diameter (d_T) can be computed as follows

$$d_T^3 = \frac{16}{\delta s}\left\{(K_tM_t)^2 + (K_bM_b)^2\right\}^{1/2} \quad [4]$$

$$d_T^3 = \frac{16}{x40x10^6}\left\{(1x54.64)^2+(1.5x101.18)^2\right\}^{1/2}$$

d_T = 0.0274m

d_T = 27.4mm

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Now selecting a power of safety of 1.28

$$d_T = 27.4 \times 1.28 = 35.05 \text{mm}$$

The permissible angle of twist for the shaft can be determined as follows

$$\theta = \frac{584\ M_t L}{G d_s^{44}} \qquad [4]$$

Where θ = angle of twist (degree)

L = Length of shaft m

M_t = Torsional moment, Nm

G = Torsional modulus of elasticity, N/m^2

d_s = Shaft diameter (blower shaft)

$$\theta \ \frac{584 \times 34.15 \times 0.72}{80 \times 10^9 \times 0.03^4} = 0.22^o/m$$

It can be seen that the shafts are torsionally rigid. The bending stress on theblower shaft can be computed as follows

$$\delta_b = \frac{32\ M_b}{d_s^3} \qquad [4]$$

$$\delta_b = \frac{32 \times 73.03}{0.03^3} = 27.55 \times 10^6 \quad N/m^2$$

The bending stress of the threshing drum shaft is

$$\delta_T = \frac{32\ M_t}{d_T^3} = \frac{32 \times 101.18}{0.035^3} = 24.03 \times 10^6 \ N/m^2$$

The combined torsional stress can also be determined as follows

$$\tau_{xy} = \frac{16 M_t}{d^3} \qquad [3]$$

The torsional stress in blower shaft is (τ)

The torsional stress for the threshing drum shaft (τ_{xy})

$$\tau^{t}_{xy} = \frac{16 \times 91.62}{0.035^3} = 10.88 \times 10^6 \ N/m^2$$

There are no axial loads on the shafts. The calculated stress are well below the yield strength of steel.

3.3 <u>Grain Threshing</u>

The theoretical principle applicable to the threshing processes are: [5]

(i) Aerodynamic separation based on the terminal velocities.

(ii) Movement of the crop through the sieves

(iii) Escape of the grain through the opening of the sieves.

Aerodynamic separation is based on the pneumatic conveying of chaft and straw in turn depends upon the terminal velocities and the drag coefficient of the different components in the crop oscillating conveyors. [6]

Grain motion, through the chaft and straw mat is due to the gravity and the resistive force caused by the straw [4]. The escape of grain through the sieve opening is based on the theory of probability. [5]

The following assumptions are made:

(i) The drag coefficient is independent of the air velocity.

(ii) The particles (grains) are accelerated as free bodies and not as a mat.

(iii) The velocity of air through the sieve housing is constant.

(iv) Air flow above the upper screen is streamlined and parallel to the orientation.

Result and Discussion

After the design and construction of the machine to ascertain whether the machine is functioning accordingly.

The standard evaluation were done by loading 20kg of sorghum head. Few heads were fed into the machine at a time and it is noticed that the machine is functioning accordingly.

The following apparatus were adopted during the process.

(a) Universal threshing machine

(b) Weighing scale

(c) Stop watch

(d) sorghum heads

See Table 1 for the result.

TABLE 1: Results obtained from Sorghum Trial

Crop Type	Qty before threshing	Weight of threshed	Weight of husks/ chaffs	Time taken	Loss of grain/ husks	Power consumed
Sorghum	1kg	0.7kg	0.28kg	15sec	0.02kg	3.048KW
"	2kg	1.5kg	0.45kg	20sec	0.05kg	3.075KW
"	3kg	1.0kg	0.96kg	26sec	0.04kg	3.10KW
"	4kg	2.7kg	1.02kg	32sec	0.28kg	3.127KW
"	5kg	3.3kg	1.5kg	38sec	0.20kg	3.154KW

The procedure followed in the sorghum trial is repeated on this experiment for millet and maize and the following results are also obtained.

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Table 2: Result obtained from Millet Trial

Crop Type	Qty before threshing	Weight of threshed	Weight of husks/ chaffs	Time taken	Loss of grain/ husks	Power consumed
Millet	1kg	0.6kg	0.35kg	12sec	0.05kg	2.28KW
"	2kg	1.3kg	0.6kg	21sec	0.1kg	2.306KW
"	3kg	1.8kg	1.0kg	28sec	0.2kg	2.328kW
"	4kg	2.2kg	1.5kg	30sec	0.3kg	2.344KW
"	5kg	2.7kg	2.1kg	36sec	0.2kg	2.36KW

TABLE 3: Result obtained from Maize Trial

Crop Type	Qty before threshing	Weight of Grain	Weight of husks/ Chaff	Time taken	Loss of grain/ Chaff	Power consumed
Maize	1kg	0.6kg	0.3kg	13sec	0.1kg	3.354KW
"	2kg	1.3kg	0.6kg	18sec	0.1kg	3.382KW
"	3kg	2.1kg	0.8kg	26sec	0.1kg	3.411KW
"	4kg	2.6kg	1.2kg	31sec	0.2kg	3.406KW
"	5kg	3.0	1.8	37sec	0.2kg	3.422KW

Machine Efficiency

Overall machine efficiency

$$\text{Efficiency}(\eta) = \frac{\text{Average Power Output(Consumed)}}{\text{Average Power Input}}, (\%)$$

$$\text{Output power (average)} = \frac{(3.101+2.325+3.375)}{3}$$

$$= 2.933\text{KW}$$

Average power input = 3.433KW

$$\therefore \eta = \frac{2.933}{3.433} \times 100 = 85.5\%$$

$$\text{Average machine output/sec} = \frac{9.6+8.6+10.2}{125+127+131}$$

$$= 0.07415\text{kg/sec.}$$

Average output/hour = 0.7415x3600 = 266.95kg

Average output/day at 9 hrs operation period =

= 267x9 = 2.4 Tonnes.

CONCLUSION

The imported threshing machine at present market value cost (₦110,000) while the designed and constructed universal grain thresher using locally available material is estimated at (₦51,074).

The standard evaluation was done by loading 20kg of sorghum heads and the procedure was repeated for millet and maize.

The result showed that the machine has an overall efficiency of 85.5%.

The machine can thresh from 2.5 to 5 tonnes daily depending on the type of crops.

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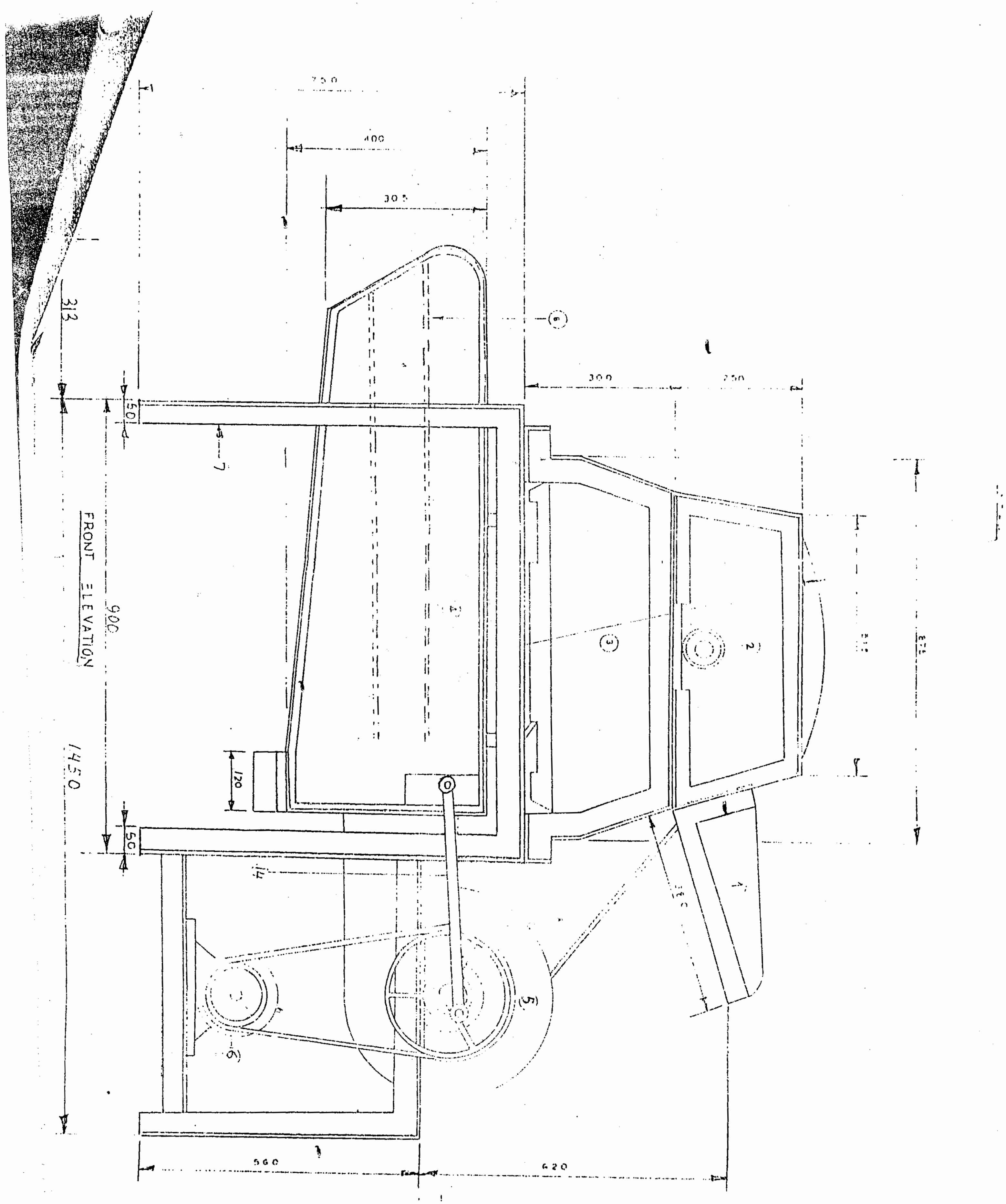

Fig.1a

Title ?

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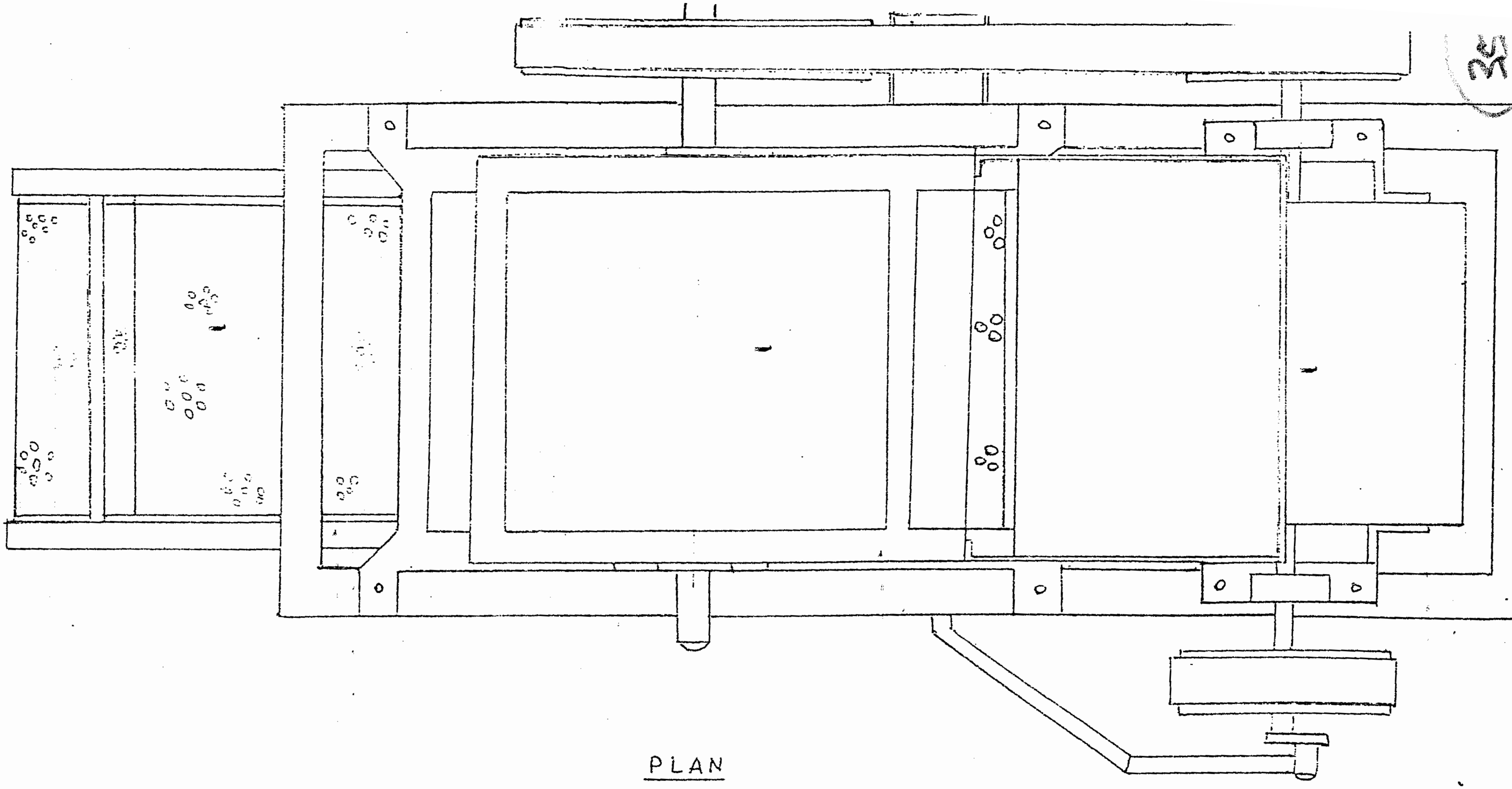
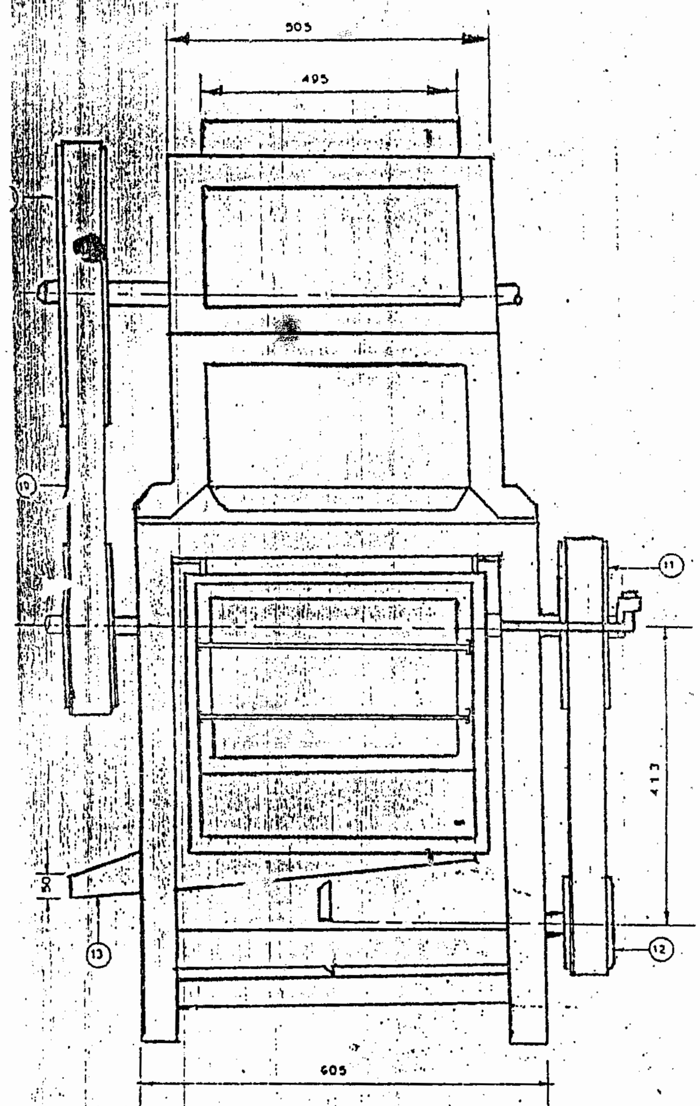


PLAN

Fig. 1b



END ELEVATION

NOTES:

S/No	DESCRIPTIONS OF PARTS	QUANTITY
1	HOPPER	1
2	HOOD	1
3	THRESHING CHAMBER	1
4	SIEVE HOUSING	1
5	BLOWER	1
6	MOTOR	1
7	FRAME	1
8	SIEVE	2
9	THRESH. CYL. PULLEY	1
10	FLAT BELT	2
11	BLOWER PULLEY	2
12	MOTOR PULLEY	1
13	GRAIN OUTLET	1
14	CON. ROD	1

SCALE: 1:6 ALL DIM. IN MM

PROJECTION

Fig. 1c