DESIGN, FABRICATION AND PERFORMANCE EVALUATION OF A RICE THRESHER

BY

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A PROJECT IN THE DEPARTMENT OF AGRICULTURAL ENGINEERING, SUBMITTED TO THE SCHOOL OF ENGINEERING AND ENGINEERING TECHNOLOGY, IN PARTIAL FULFILMENT OF THE REQUIREMENT FOR THE AWARD OF MASTERS OF ENGINEERING.

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DEDICATION

This project is dedicated to the Glory of the Almighty God for seeing me through all the hindrances and difficulties encountered in the course of carrying out this work. And also to the memory of my late uncle and his wife Dr. & Mrs. Wankari Musa

CERTIFICATION

This is to certify that this project work was carried out by Dauda, Solomon Musa in the Department of Agricultural Engineering, Federal University of Technology Minna.

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Abstract

A rice thresher was designed and fabricated to solve threshing problem in Nigeria. For objective evaluation of the thresher, performance tests were carried out. Test results at 13.83% average grain moisture content (wet basis) and machine threshing cylinder speed of 560 rpm show that, the thresher has threshing efficiency of 98.01%, cleaning efficiency of 99.32%, input capacity of 490Kg/hr., output capacity of 267.9Kg/hr., grain recovery range (GRR) of 95.22%, capacity utilization (CU) of 54.67%, threshing index (TIX) of 51%, threshing intensity (TIN) of 0.029Kw/Kg and a power requirement of 7.9Kw (10hp). The thresher is efficient with minimal grain loss of 4.78%.

The physical properties of Faro 51 Rice variety and its grains such as length of ears, length and breadth of grain, grain/straw ratio and moisture content of grain were also measured.

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Interdependence is a higher value than independence.

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LIST OF FIGURES.

LIST OF TABLES

CHAPTER ONE

1.0 INTRODUCTION

Threshing is the first and most important post-harvest operation of grain crops. It involves the detachment of grain kernels from the stalk heads (Nkama, 1992). The traditional method of grain threshing in Nigeria is by hand beating with stick or hitting the grain stalk or pods on the floor. This method is not only inefficient but also very laborious and the output is low resulting in delay in handling large volumes of grain harvest and subsequently leading to losses. Mechanical or powered threshers have been introduced to overcome these difficulties, but local acceptability have been very low due to high cost. The use of engine powered machines and equipment in Nigerian Agriculture dates back to the early 1960s following the establishment of farm settlements in the eastern and western regions of Nigeria (Chukwu, 1994). Also, there is now a general awareness in Nigeria and other developing countries that the rapid development of agriculture depends on a large extent on the successful introduction of modern indigenous agricultural machinery. Most of the farm equipment presently used in Niger State are imported from several countries as shown in Table 1. This has resulted in assortment of different machines with problems ranging from non availability of spare parts, inadequate provision of power drive units among others. Based on the above reasons, there is need to design, develop and evaluate the performance of a rice thresher. The thresher should be capable of reducing drudgery, grain damage and losses. The physical properties of rice crop and its grains such as length of ears of rice crop, grain length and breadth, grain/straw ratio are important properties that affect the design of the machine. Parameters such as cylinder peripheral speed, type of beaters concave clearance among others were selected and designed (The Food Agency, 1995, Hem, 1981).

TABLE 1: INVENTORY OF RICE THRESHERS BY ESTABLISHMENT AND BY MAKE IN NIGER STATE IN 1993.

lishment	No. of makes	Votex	FATE	Vicon	Akshat	Embee	Cecoco	Kubota	Alvan Blanch	TNA*
', MINNA	ONE	-	-	1	-	-	-	-	-	1
', BIDA	SIX	9	2	1	2	1	1	-	-	16
., MINNA	ONE	-	-	-	-	-	-	3	-	3
', KUTA	ONE	2	-	-	-	-	-	-	-	2
₹, MINNA	TWO	2	-	1	-	-	-	-	-	3
JR,	NILL	-	-	-	-	-	-	-	-	0
:L FARMS	ONE	6	-	-	-	-	-	-	-	6
FARMS	ONE	3	-	-	-	-	-	-	-	3
FARMS	TWO	2	-	-	2	-	-	-	-	4
INS'T	ONE	1	-	-	-	-	-	-	-	1
A										
BADEGGI	TWO	2	-	-	-	-	-	-	2	4
-		27	2	5	2	1	1	3	2	43
TOTAL BY		62.79	4.65	11.63	4.65	2.33	2.33	6.97	4.65	

TNA* = Total Number Available

Table 1 shows that most of threshers in use in Niger State were imported

Source: Chukwu, 1994

1.2 PROJECT OBJECTIVES

The main objective of this project is to develop a rice thresher from locally available materials and to evaluate its performance.

I. The machine should be suitable for threshing the commonly available rice varieties.

II. The machine should be simple, easy to operate and maintain.

1.3 PROJECT JUSTIFICATION

The enormous rice production capability of Nigeria has not been fully exploited due principally to lack of appropriate indigenous processing technology as obtained in other developing Asian countries. Therefore, there is the need to develop an efficient and affordable rice threshing technology, which can be adopted by people. It is also envisaged that manufacturers and research institutions can adopt the developed rice thresher.

CHAPTER TWO

2.0 LITERATURE REVIEW

2.1 TAXONOMY AND RICE ORIGIN

Rice like wheat, corn, rye, oats and barley belongs to gramineae or grass family. The genus oryza to which cultivated rice belongs, has twenty one wild and two cultivated species. Nine of the wild species are tetraploid and the remaining are diploid. Harland and De Wet (1971) proposed classifying the wild relatives of a crop species into three categories on the basis of isolation barriers and the case of gene transfer to the cultivated species. This is a useful concept for breeders. In recent years, efforts have been made to introduce useful genes from wild species to cultivated rice through interspecific hybridization (Brar et al 1996; Jena and Khush 1990; Multani et al 1994). On the basis of ease of gene transfer, the primary gene pool comprises the wild species.

Historically, the rice improvement programme in Nigeria started with the British colonial administration with the establishment of the Federal Department of Agriculture at Moor Plantation, Ibadan in 1899. In 1953, a rice research station was established at Badeggi, under the supervision of the Federal Department of Agriculture, as the main rice research station. By 1975, the National Cereals Research Institute (NCRI) came into being and was assigned the national mandate for rice research in the country (Misari et al., 2001). Fifty-one varieties were released to farmers as the Federal Agricultural Research Oryza (FARO) series. Agronomic characteristics of released rice in Nigeria are attached as appendix A.

2.2 BOTANICAL CHARACTERISTICS

Rice is the only major crop specie that is aquatic; rice plants can grow successfully in standing water. Rice has a shallow root system and is therefore very susceptible to water stress. Rice has a loose, freely branched panicle type inflorescence. Each branch of the panicle bears several spikelets containing a single floret. Unlike other cereals, each rice floret has six (rather than three) stamens and two, long styles (Chapman and Carter, 1976). When rice is threshed, the lemma and palea generally remain fused to the grain as hulls. Different types of rice are recognized by the length of the rice grain: short grain rice has kernels about 6.6mm long; long-grain rice has kernels about 7 or 8mm long (Chapman and Carter, 1976). Short-grain rice has shorter straw and is less prone to lodging than other types. Rice has short stems and the number of nodes varies. Late maturing varieties have more nodes and are therefore usually taller than early maturing varieties. Depending on the variety and the growing condition, rice plants vary in height from 60-180cm (Chapman and Carter, 1976).

2.3 HARVESTING

It is essential that rice crop be harvested on time to reduce losses due to feeding rats, birds, insects, and from shattering and lodging. The best time to harvest is 32-42 days after heading in wet season and between 28-30 days after heading in dry season depending on variety under cultivation. The crop should be ready for harvest when about 80% of the panicle and straw are colored and the grains in lower portion are in hard dough stage (Dauda et al. 1999).

2.4 RICE THRESHING

For ease of processing, the paddy should be threshed. Threshing methods are usually classified as manual (hand or threading) or mechanical (by using rotating drums with spikes or rasp bars). Each of these methods involves impact forces (Nkama, 1992). It is the separation of the paddy rice kernels from the spikelets on the panicle. Paddy rice is the rice kernel (caryopsis) which is enclosed by the lemma and palea, which together form a relatively tight hull or husk. Problems usually associated with traditional methods of rice threshing are: low capacity of threshing, excessive breakage of the grains when pounding is employed, grain loss due to incomplete threshing and contamination by foreign matter such as stones, mud and pebbles. Olunade (1990), reported that "hand beating" method of threshing gives a threshing capacity of 30-40kg/hr, with losses of 5-10% due to un-threshed grain, and grain shattering losses. These indicated losses are quite high and reflect a huge financial loss especially when large hectares of rice farms are involved. In an attempt to solve the above problems, modern mechanical threshers were developed and imported into the country, which vary in design, operation, efficiency and complexity. There are "throw-in" types in which the rice ears are put into the thresher and there are the "hold-on" types in which the rice panicles are hand-fed into the thresher.

Porlicarpio et al. (1978) reported that "throw-in" types provide higher output per man-hour than the "hold-on" types. Most small threshers are designed to use the "hold-on" method of feeding whereas the votex ricefan (the most widely used thresher in the country (Chukwu, 1994)) uses the "throw-in" method of feeding. A comparative performance of the two methods is presented in Table 2.

Mechanical threshers could be manually operated or motorized. Generally, mechanical rice threshers have threshing cylinders or drums with different arrangements and configurations of the threshing teeth. Commonly available threshing cylinders are the spike-tooth, the syndicator and the beater types.

Beater type threshers were the first to be introduced in Nigeria (Policarpio et al., 1978). According to Saxena et al., (1971), the beater type threshers lost their popularity mainly due to their high power requirements. Both spike-tooth type and syndicator type threshers are very popular in Nigeria. Although, the spike-tooth type threshers have advantage of simple and compact design with low grain breakage, and can handle very dry crops with grain moisture content below 10% (Saxena et al., 1971). Whenever there is rain, much time is lost in drying the crop to the desired level of moisture content for threshing.

The syndicator type thresher has the advantages of low energy requirement, production of uniform size straw and capability to handle crops of high moisture content above 15%. Banga (1981) reported higher grain breakage for syndicator type threshers compared to spike-tooth threshers. Verma and Bhatia, (1980) found that the number of accidents reported using syndicator type threshers were more than those reported for other types of threshers.

Many mechanical threshers have been developed in Nigeria but most of the designs have not been made adequately suitable for the poor rural farmer who has no supply of electricity. An example of a locally developed thresher is the IAR Samaru multicrop thresher, which was reported to thresh rice effectively (Olunade, 1990). Also, GUSAC Ltd. A firm of engineering consultants, Agro-industrialists and manufacturers based in Owerri, Imo State, manufactures rice threshers as the G101 series (Dalrymple, 1986).

Table 2: COMPARATIVE PERFORMANCE OF "HOLD-ON" AND "THROW-IN" METHODS IN RICE THRESHING.

Items of comparison	HOLD-ON METHOD	THROW-IN METHOD
Capacity	30-40kg pr man/hr	Feeding rate up to 1,500kg/hr and Thresher outputs up to 1000kg grain per hour.
Losses	5-10% Un-Threshed grain.	0.5 – 2% Un-threshed grain.
Grain damage	2-3% dehusked grains in case of dry crops (gram M.C.< 15% dehusked grains incase of Threshing with M.C. 12%.	2-3% dehusked grains in case of dry crops (gram M.C.< 15% dehusked grains incase of Threshing with M.C. 12%.

Source: Policarpio et-al. (1978)

2.5 DIFFERENT MACHINE – CROP VARIABLES THAT AFFECT THRESHING

In developing a thresher such parameters as cylinder peripheral speed, type of beaters, concave clearance, range of moisture content of grain to be handled, and grain – straw ratio have to be studied. Tandon et al., (1988) studied the interaction of different machine and crop variables. In order to identify the contribution of each variable they developed prediction equation using step-wise regression analysis.

Tandon et al., (1988) used the stepwise multiple regression technique to study the relationship among different independent variables, namely, cylinder types, concave clearance, cylinder peripheral speed and moisture content in relation to the two dependent variables, namely, Threshing efficiency and invisible grain damage on grains. The moisture content was found to have a significant effect on Threshing efficiency and invisible grain damage at 1% level. The effect of concave clearance and cylinder peripheral speed, though numerically small was significant at 5% level (Tandon et al, 1988).

In summary, Tandon et al. (1988) indicated that:

(i) Threshing efficiency increased with cylinder peripheral speed.

(ii) As concave clearance increased, the grain damage decreased.

(iii) The grain damage increased with an increase in the cylinder speed and the moisture content.

(iv) The extent of variation in the threshing efficiency explained by moisture content alone was 63% while the extent of variation in invisible grain damage explained by moisture content alone was 41%.

(v) The inclusion of cylinder concave clearance in addition to grain moisture content accounts for 66% and 48% of variation in threshing efficiency and invisible grain damage respectively as against 63% and 41% explained by moisture content alone.

(vi) The inclusion of concave clearance and cylinder speed along with moisture content accounted for 68% and 52% variation in Threshing efficiency and invisible grain damage, respectively, as against 66% and 45% that is explained by moisture content and concave clearance.

(vii) The inclusion of cylinder beater did not account for an increase greater than 1%.

Based on the findings of Tandon et al. (1988), it may be concluded that the moisture content has a high significant effect on threshing efficiency and invisible grain damage. The effect of concave clearance and cylinder speed was also significant.

Yunus (1987) identified grain moisture content, grain number, length of grain, weight of 1000 grains, speed of Thresher drum, and size of sieve aperture as some of the crop machine properties that affect grain losses at harvesting and Threshing of paddy in Turkey.

He indicates that grain losses occur when paddy is threshed at unsuitable grain moisture contents and at unsuitable speed of thresher drum. The grain number (which gives a measure of grain – straw ratio) as well as the weight of 1000 grains are different for different rice varieties and affect the output capacity of a thresher.

From the above discussion it can be stated that several machine-crop parameters influence the performance of a thresher. Some of these parameters are grain moisture content, grain number, grain length, cylinder type, cylinder peripheral speed, concave clearance and size of sieve aperture.

2.6 TERMINOLOGY FOR GRAIN THRESHERS

For the purpose of this work, the following definitions shall apply, however, only the major and unique terms are defined here given, the more complete list in appendix B

2.7 History of Thresher Parameters

A good amount of work has been reported on the threshers and combines to determine the effect of different parameters on the threshing of different crops. The test reports of some of the existing threshers give a broad idea about the design parameters of threshers (Hem, 1981).

A comparative performance of wire loop, peg tooth and rasp bar cylinders on paddy was studied at IRRI (1969). The effect of hold-on feeding and throw-in feeding on thresher performance and damage was investigated. The tests showed that high cylinder velocities gave less semi-threshed paddy. Except for the wire loop cylinders without concave, the throw-in feeding gave poorer performance than hold-on feeding at low velocities. It was accepted that the throw-in types are better suited for high output than the hold-on type.

At IRRI (1963), a cone-type thresher was designed, fabricated and tested and it indicated that the principle of the cone-type thresher was found adequate and gave a maximum output of 300kg/hr using a 6-hp engine. A power take-off driven model (IRRI, 1964) with more power and screen area was built. A 36-hp tractor stalled when the thresher was fed with straw and grain at the rate of 60kg/min. The centrifugal nature of the cone thresher resulted in increased friction on the screen and drag on the drum, creating a high power requirement. Complete threshing was achieved, but even a 4.5m^2 screen did not fully separate the grain from the straw. Efforts were made to improve the cone thresher (IRRI, 1972, IRRI, 73, IRRI, 74). It gave high threshing output but the grain cleaning performance was not satisfactory.

Harrington (1970) designed a multi-crop thresher having spike-tooth cylinder and fixed concave on the basis of acceptance of Japanese paddy threshers and American wheat threshers, which used spike-tooth cylinders. The clearance was kept at 2.5cm. The losses in paddy and wheat were 3%. The thresher was not suitable for straw making as it normally reduced wheat straw to 25-50% of original length depending upon cylinder speed and moisture condition. The cylinder tip speed ranged between 1,200 and1,400m/min resulting in less than 0.1% un-threshed grain and between 800 and 1,000m/min resulting in less than 0.1 % visible damage. The un-threshed grains were less than 1.0%.

Ramos (1975) tested five types of threshers, namely; (I) single drum-no-blower-no separator; (ii) double drum-single blower-no separator type; (iii) double drum-single blower with oscillating screen type; (iv) axial flow thresher; and (v) single drum-single blower with oscillating screen type. He found that the capacity of these threshers were 704, 250, 1,148, 936 and 2,449kg/hr, respectively, for paddy crop. In these tests, the drums used differed from one thresher to another. The paddy was wet and the grain and straw moisture contents ranged from 20-23% and 80%, respectively, on wet basis. The length of material used in the test was 30-40cm with the grain-straw ratio ranging from 0.46-0.65. The second, third and fifth type threshers had 23-38% blower loss. Unthreshed loss was high in thresher types (I), (ii), and (v).

Singh and Joshi (1977) reported that the Japanese type thresher of 40cm drum diameter and 90cm width rotating at 400rpm by a 3-hp electric motor gave an output of 145kg/hr. A rasp thresher of 28.5cm drum diameter and 32cm width gave an output of 139kg/hr of clean paddy. The energy needed per quintal of thresher and cleaning was 0.36-hp/hr of manual and 1.64-hp/hr of electrical energy. For through feed rasp-bar thresher, it was 0.44-hp/hr of manual and 0.72hp/hr of electrical energy.

As reported by Chabbra (1975), the adequate drum length for an axial flow thresher was 203.2cm at IRRI and any further reduction in the length caused separation losses. The concave area was increased from 0.73m^2 by using two concaves, and improvement in separation efficiency was observed. A threshing output of up to 660kg,hr with 1% separating and threshing loss was obtained when a 35.6 diameter and 127 cm long wire loop type threshing drum was used. Cleaning efficiency was 95-98%. The length of the cylinder can be reduced by increasing the concave area to obtain desired separating and cleaning efficiency. The studies made a varying cylinder-concave clearance and pitch of wire loops indicated a 1.9cm clearance and 10.2cm pitch as optimum. A throw-in type multi-crop axial flow thresher was developed at IRRI (1972). It had wire loop threshing drum of 35cm diameter and 127cm length with a concave screen enclosing the full length of the cylinder. A 6.5-hp air-cooled engine was used as prime mover. A peg-tooth cylinder with bolts of 1.1cm in diameter as pegs in an axial flow thresher gave better threshing performance and longer life comparison with wire loop threshing cylinder. The capacity of the thresher was increased by replacing the perforated sheet metal concave with a concave of steel rods.

Chabbra (1975) developed and tested an axial flow thresher on the basis of drawing procured from IRRI. He found that it can thresh paddy and wheat quite efficiently. In paddy threshing at 500rpm (794m/min peripheral speed), gave a threshing efficiency was 100% and the feed rate was 710kg/hr which resulted into an output of 213kg clean grain per hour. To achieve 99% cleaning efficiency the blower losses were 9%.

The above-described machines are mostly sophisticated and very costly. A typical example is the ASI rice thresher developed by four organizations, The Senegalese River Valley Development Agency, known as SAED, Senegal, The Senegalese Institute for Agricultural Research (ISRA), Senegal and The West African Rice Development Association (WARDA, known as ADRAO in French), Cote d'Ivoire costs about US$5,000. Also the operator must have a sound technical knowledge to be able to operate and maintain them. In the same vain, these machines are not in the reach of the financial ability of an average Nigerian farmer. In view of these problems mentioned above, there is need for a rice thresher, simple in design, easy to maintain with little technical knowledge to operate, and affordable within the financial reach of an average Nigerian farmer.

CHAPTER THREE

3.0 METHODOLOGY

3.1 DETERMINATION OF PHYSICAL & ENGINEERING PROPERTIES OF RICE

The following properties of rice, pertinent to the design of the thresher were determined; length of rice grain, grain breadth, grain/straw ratio and length of ears of rice crop. The length and breadth of grain was used to select sieve hole diameters, length of ears of rice crop was used to select hopper size and the grain/straw ratio was to determine the theoretical capacity of the thresher.

The grain sizes were determined by measuring the length, breadth and length of ears of rice crop with a micrometer screw gauge.

Grain moisture content was determined by oven dry method at a temperature of 104°C for 24hours (A.O.A.C 1984) all measurements were taken in the laboratory at a room temperature of 28°C.

The following design parameters were established after measurement of the physical properties of the Rice grain and review of available literatures (Hem, 1981, Ahuja and Sharma, 1989, The Food Agency, 1995). 560rpm was selected for cylinder speed, 4.46m/s fan air velocity, 40mm concave clearance, 8mm and 3mm sieve perforation diameters for top and bottom sieves respectively and a power requirement of 7.9kw.

3.2 DESIGN APPROACH

The principles of beating stick are used for threshing rice. The threshing drum is made of mild steel and provided with sideboard at both ends. A shaft is installed which is supported by bearings. Iron rods 50mm in length are welded to the threshing drum. There is a cover over the threshing drum to prevent grain scattering. This cover together with the concave section under the drum forms the threshing space. The cover is fitted with colliding plates to assist threshing.

The concave is provided under the threshing drum to separate grains and assist threshing simultaneously. It is made of smooth iron rods. Grains that have passed through the concave drop along a grain flow plate. On the way, they are separated into whole grains, chaff and straw by airflow generated by the blowing fan.

3.3 DESIGN PARAMETERS AND ANALYSIS

3.4 CAM DESIGN FOR RECIPROCATING FLOW PLATE

The reciprocating flow tray has a stroke length of 80mm. The forward speed of a machine which is 1.3 – 1.4mls (Devani and Pandey, 1985). A value of 1.4mls was chosen (Appendix D 1).

3.5 BELT DRIVE DESIGN

Engine pulley diameter = 12cm

Drum pulley diameter = 30cm

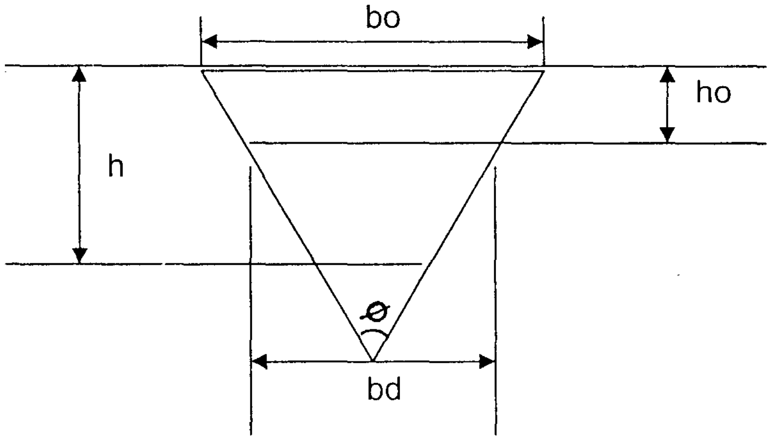

Fig. 1: Belt Drive Section

From Standard table, assume bd = 8.5, bo = 10, h = b, ho = 2.1 (Reshetov 1978).

The peripheral velocities (mls) on the pulleys (fig. 2 Below):

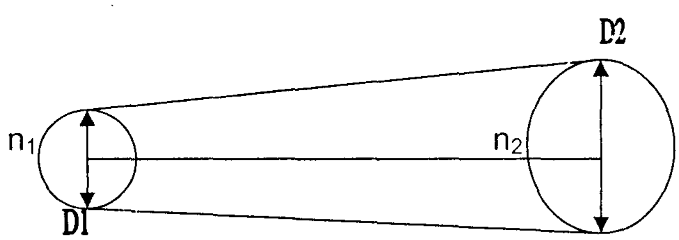

Fig. 2: The Peripheral Velocities on pulleys

3.6 MACHINE PARTS DESIGN

Each component of the threshing machine is designed to fulfill a specific task each involves, threshing and cleaning. The orders in which the parts are designed are in accordance with maintenance and assembly sequence.

3.7 BELT SELECTION

V-belt is chosen because of its advantage over the flat Belt, some of the advantages are:

- V-belt is used with smaller sheave

- Its center distance is smaller
- It is more durable and less costly
- V-belt has no risk of falling off the pulley.

3.8 MOTOR-CYLINDER BELT

Using the speed rate formular, the speed of the cylinder can be determined from the prime mover (appendix D2).

3.9 ANGULAR VELOCITY OF MOTOR – CYLINDER BELT

Let the angular velocity of the motor pulley and the cylinder pulley be ω_1 and ω_2 respectively, n_1 and n_2 speed of the motor and cylinder pulleys in rpm respectively (Appendix D 3).

3.10 ANGLE OF RAP OF MOTOR – CYLINDER PULLEYS

This is the external angle that the point of contact of the belt on each of the pulleys makes with the center of the pulley (Appendix D 4).

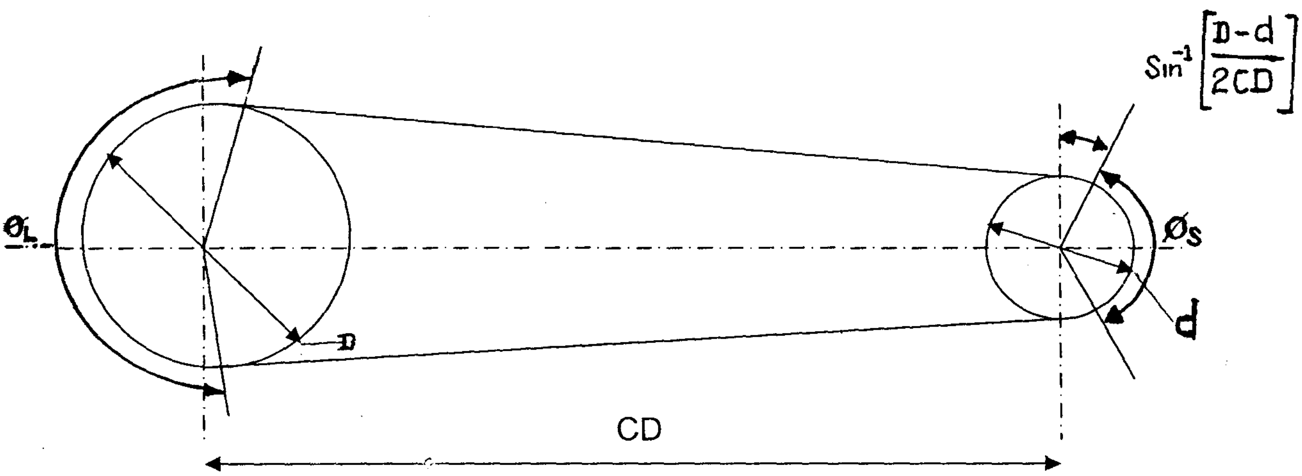

Fig. 3: Showing angles of rap for motor - cylinder pulleys

3.11 CYLINDER – BLOWER BELT

Ogunlowo and Oladapo (1990) observed the terminal velocity of paddy r[illegible] be 5.37mls

However, for adequate separation of grain from trash and other foreign ma[illegible]s, air velocity must be less than the terminal velocity.

Speed of cylinder shaft = 560rpm

Radius of blower = 0.10m

Dimension chosen by designer (Appendix D 5).

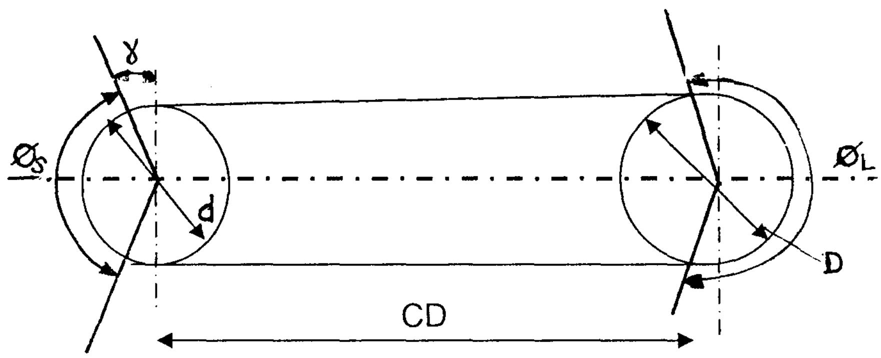

Fig. 4: Showing angles of rap for blower and cylinder pulleys.

3.12 BLOWER-FLOW PLATE BELT (WINNOWER)

Using the speed rate formular, the speed of the winnower can be determined from the blower pulley (appendix D 6).

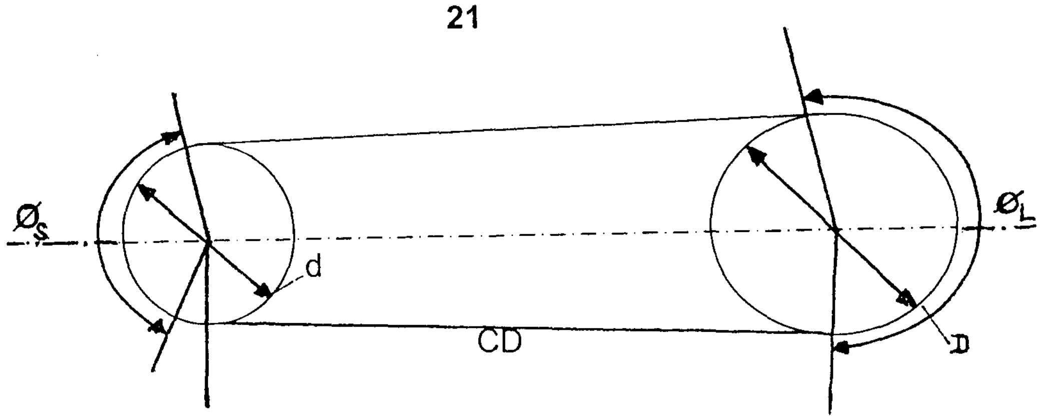

Fig. 5: Showing angles of rap for winnower and blower pulleys

3.13 FAN (BLOWER)

A blower is a device which propels air continuously against the pressure loss of a closed circuit system, in which the ambient atmosphere may be a component. A blower circuit consist of:

(i) Rotor or rotating member

(ii) shaft through which power flows to the rotor; and

(iii) casing or housing to control and direct the generated air flow.

The vanes of the rotor impact energy to the fluid by virtue of pressure forces in their surfaces, which are undergoing a displacement as rotation takes place. In selecting fans, a few considerations are made viz:

(i) volume flow rate and whether any variation is required.

(ii) type of fan required

(iii) density of air to be moved

(iv) type of drive and height of the drive from the cylinder shaft

The straight bladed type of radial flow fans sometimes called the paddle wheel type of fan is used in this design. It is operating in an environment containing dust particles such as occurs during threshing (Appendix D 7).

3.14 DESIGN OF SHAFT

Components mounted or integrated with shafts cause various stresses in the shaft. These stresses could be axial (tensile or compressive), bending stresses, shear stresses (torsion). These stresses can also be either static, cyclic (variable) or shock loaded.

The design analysis is to obtain shaft diameter that will ensure failure – free operation of the shaft under these loading conditions. Here, bending and torsional stresses are most active (Appendix D 8).

(a) SHAFT LOADING IN THE VERTICAL DIRECTION

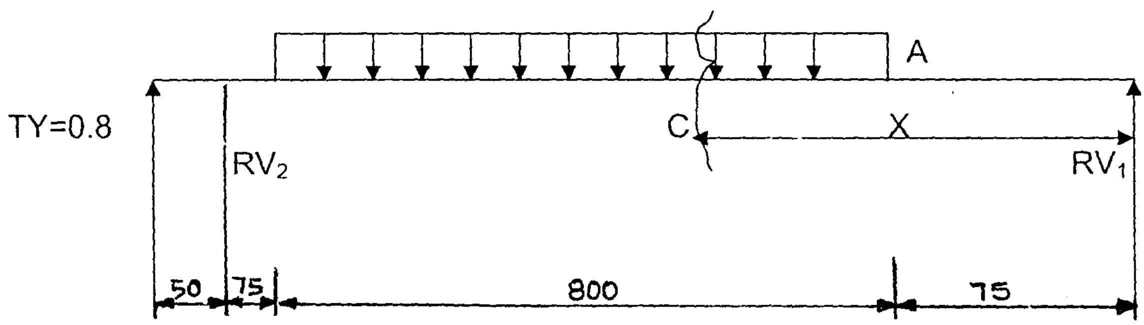

Fig: 6: Shaft Load Distribution in the Vertical Direction

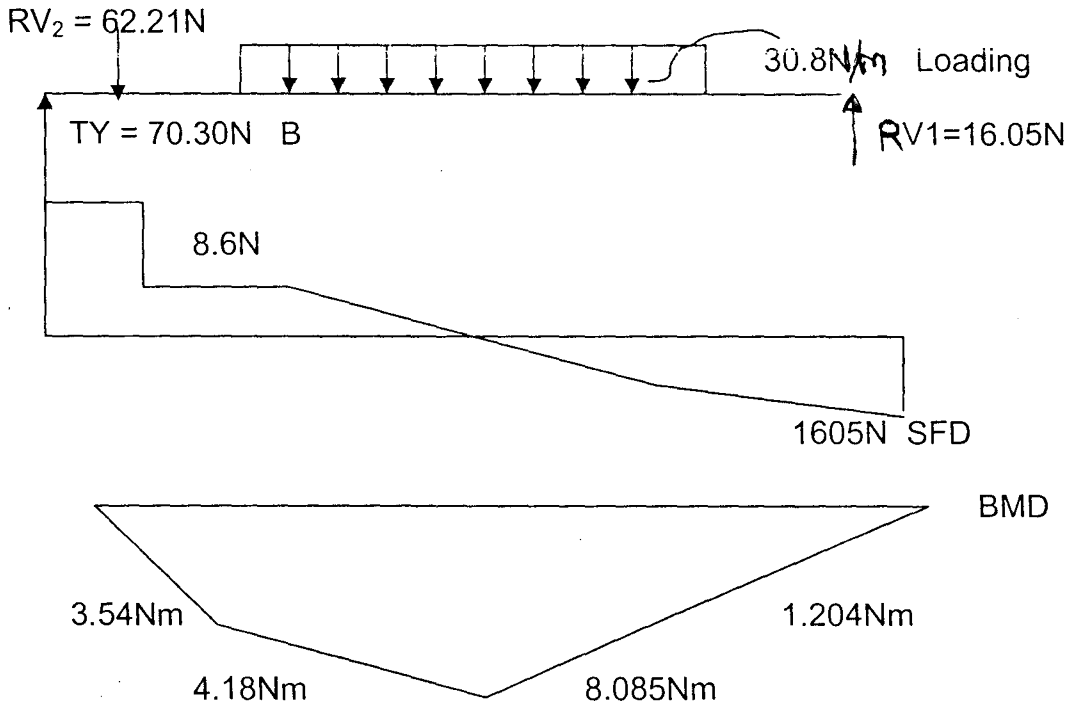

Fig. 7: Showing shaft loading, shear force diagram and bending moment diagram in vertical direction

(b) **SHAFT LOADING IN THE HORIZONTAL DIRECTION**

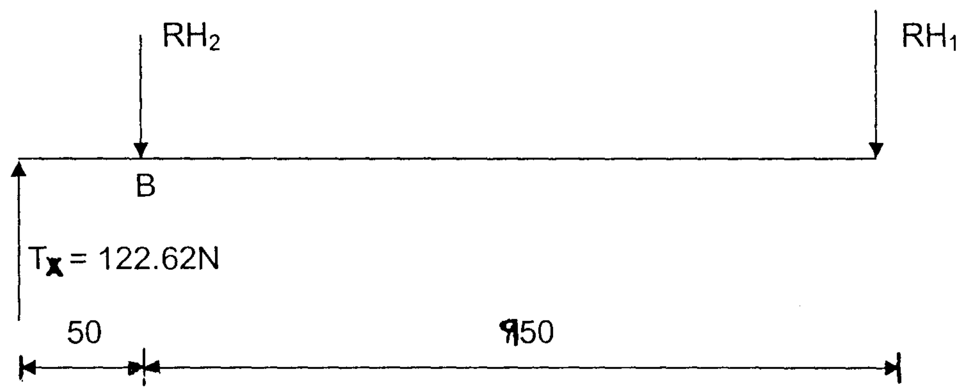

Fig. 8: Shaft Load Distribution in the Horizontal Direction

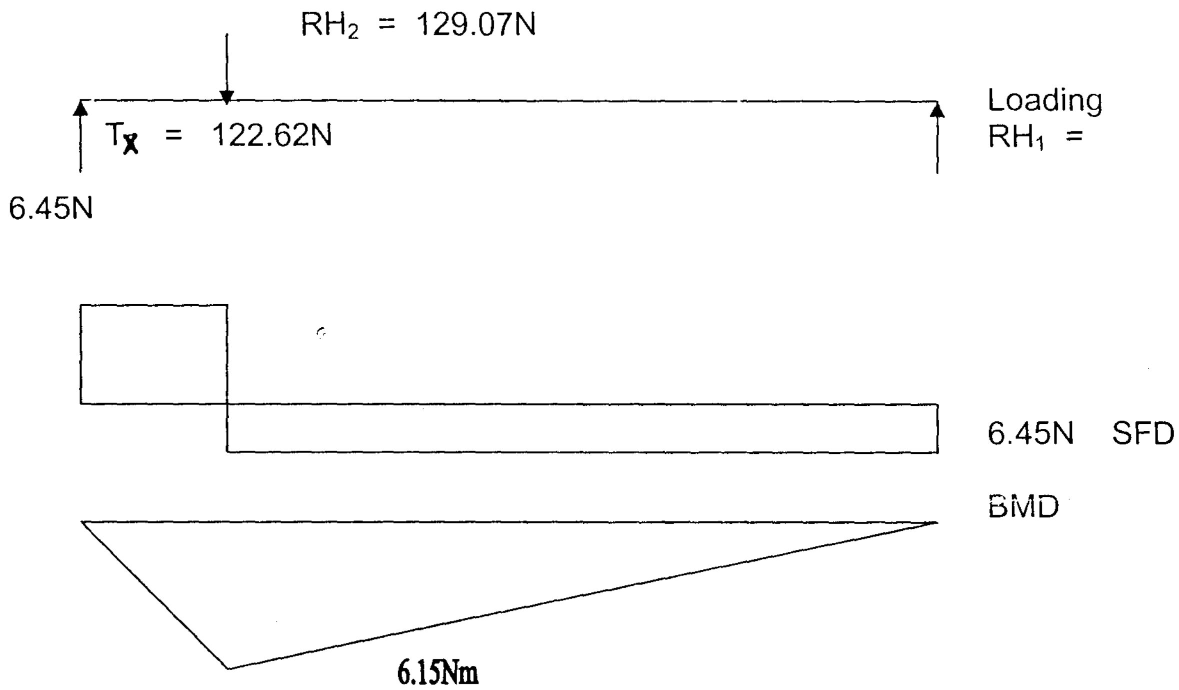

Fig. 9: Shaft Loading condition, shear free diagram and bending moment diagram in
horizontal condition

3.15 CYLINDER SHAFT DESIGN

Optimum cylinder peripheral velocity for threshing paddy on impact force is 9.42m/s (Ahuja 1989). Cylinder of diameter 300mm rotating at a speed of 560rpm satisfies this threshing speed.

3.16 ELEMENTS ON THE SHAFT

The shaft has some elements on it, which contributes in the threshing action of the machine. The elements include cylinder and cylinder end plates, steel spikes. On the steel cylinder support, the weight of each is calculated (Appendix D 9).

3.17 CYLINDER

Housing mild steel sheet

Diameter = 300mm

Thickness = 1mm

Length = 600mm

(Appendix D 9).

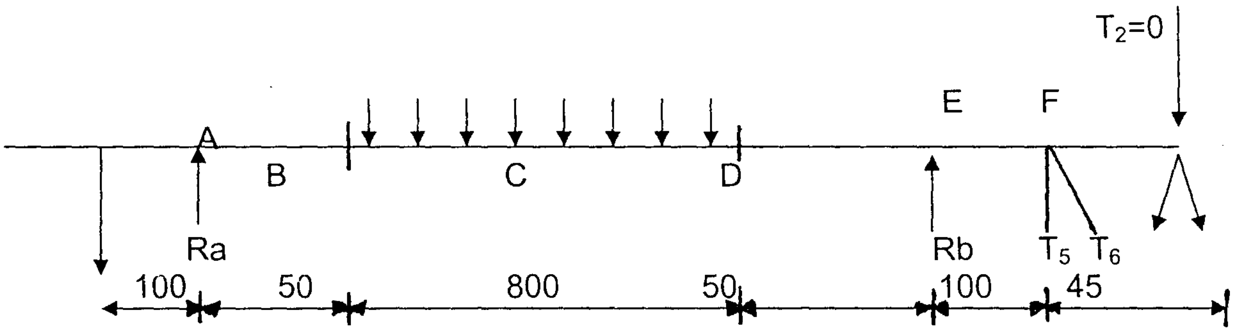

Fig. 10: Loading on Cylinder Shaft

3.18 CYLINDER SHAFT LOADING IN VERTICAL DIRECTION

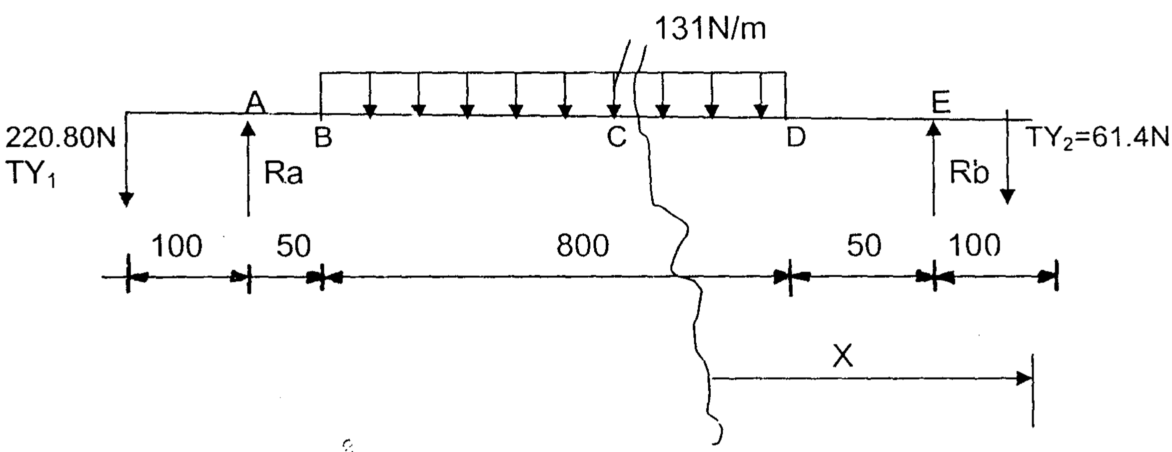

Fig. 11: Vertical Load distribution

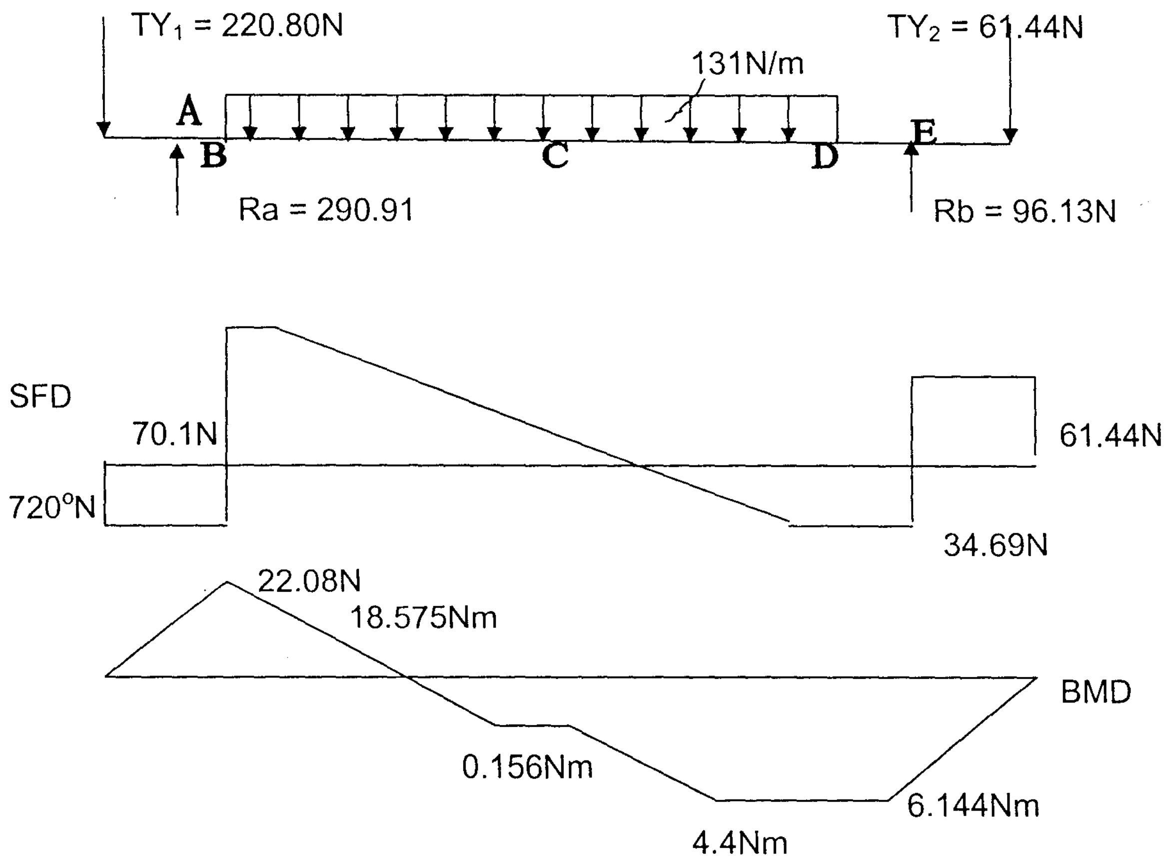

Fig. 12: Showing the loading, shear force and bending moment diagram in vertical direction for cylinder shaft.

3.19 CYLINDER SHAFT LOADING IN HORIZONTAL DIRECTION

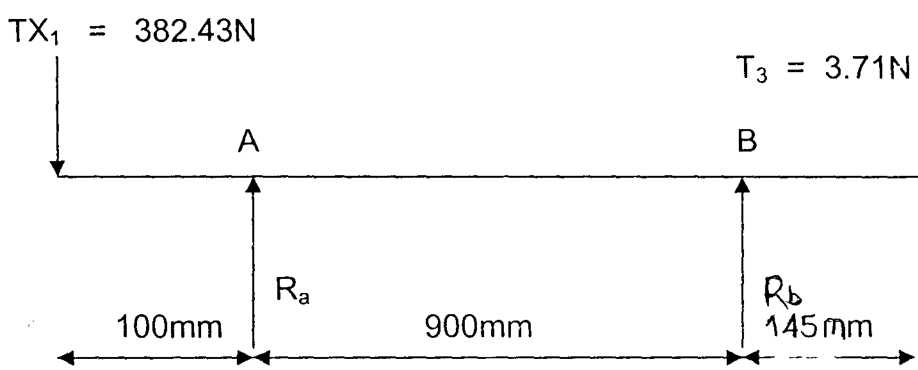

Fig. 13: Loading of Cylinder Shaft in Horizontal direction

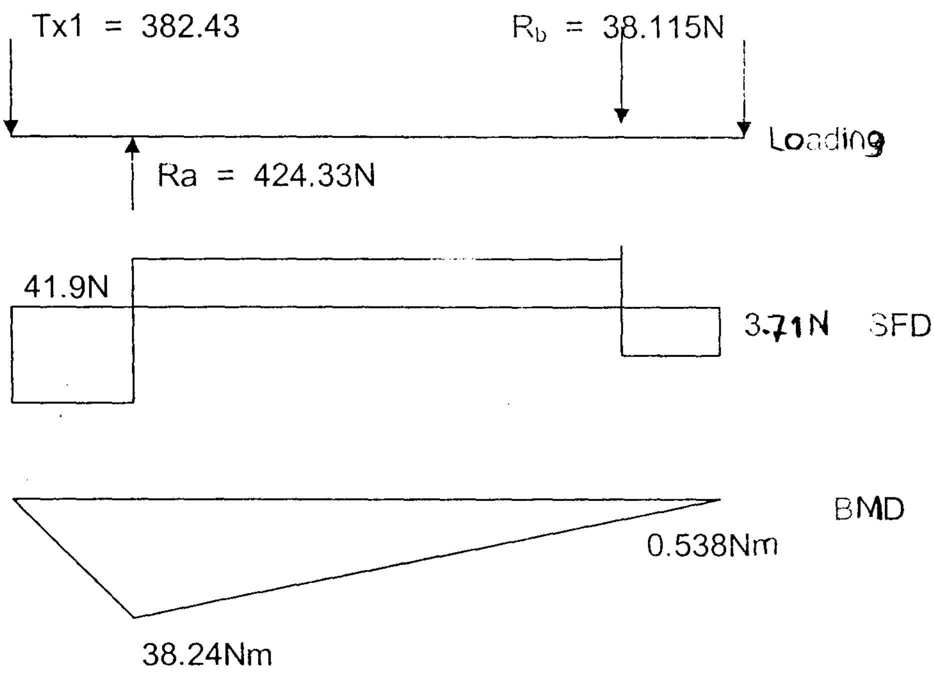

Fig. 14: Cylinder Shaft Loading Condition, Shear Force Diagram

3.20 BEARING DESIGN AND SELECTION

Fig. 15: Bearing Reaction
(Appendix D 10)

3.21 FRAME

The frame supports the total weights of the machine components. Those weight include:

(i) Weight of cylinder, blower and cam shafts combined = 9.68kg

(ii) Weight of cylinder, cylinder plates, cylinder supports and cylinder spikes combined = 10.684kg

(iii) Weight of four pulleys combined = 9.5kg

(iv) Weight of housing, assumed to be 0.5 times the sum of weight from i – iii = 14.932kg

(v) Weight of hopper = 8.5kg

(vi) Weight of concave = 10.80kg

(vii) Weight of cleaner = 11.90kg

3.22 FRAME DESIGN

The frame supports the total weight of the machine components. For axially and laterally loaded frame (Appendix D 11).

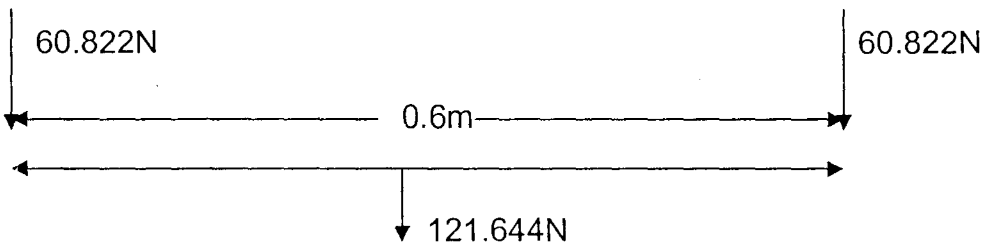

Fig. 16: Loading on the frame

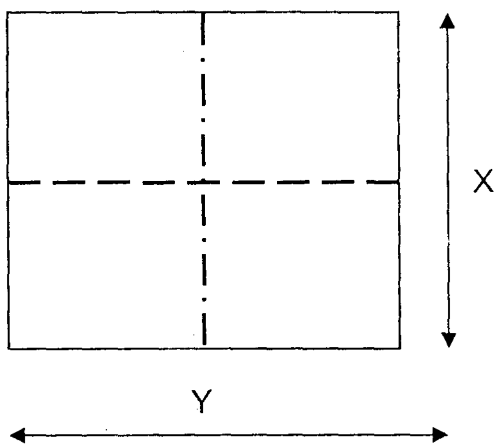

Fig. 17: Frame arrangement

CHAPTER FOUR

4.0 MATERIAL SELECTION, CONSTRUCTION AND PRINCIPLES OF OPERATION

Material selection is an important consideration when constructing the components of a machine. Tensile strength (ultimate strength) is in some cases the greatest stress experienced by the materials. The primary goal of designing machine components is that the strength of the parts should have a factor of safety greater than one. Nevertheless material selection must also match with the cost of the components being used for the machine. The materials used are presented in table 3.

Table 3: Component parts materials for construction

Component Parts	Material for Construction
THRESHING UNIT	
Cylinder shaft	Mild steel shaft
Cylinder	Mild steel flat bar 37.5mm
Spikes	Mild steel iron rod Ø8mm
Cylinder cover	Mild steel sheet (gauge 16)
Concave	Mild steel rod Ø8mm
Bearing	Ball bearing FS P206
BLOWER UNIT	
Blower shaft	Mild steel shaft Ø22mm
Blower blades	Mild steel metal sheet g.16
Blower housing	Mild steel metal sheet g.16
Bearing	Ball bearing FS P205
CLEANING UNIT	
Top screen	Ø8.0mm(r)
Middle screen	Ø8.0mm(r)
Bottom screen	Ø3.0mm(r)

Note: r=round

4.1 MACHINE FRAME

The frame supports the machine and it is constructed with 50 x 50mm angle-iron made of mild steel.

4.2 MOTOR MOUNTING BRACKET

Angle iron of 50 x 50mm is used for the construction of the motor mounting bracket.

4.3 PULLEY DRIVE SYSTEM

The threshing unit requires a 300mm-diameter pulley, which takes drive from the prime mover, and a 190mm-diameter pulley, which sends drive to the blower pulley.

The blower pulley is double grooved with a diameter of 120mm which sends drive to the cleaning unit. The cleaning unit pulley is of 100mm diameter. The pulleys are made up of Alluminium material to be able to withstand vibration, friction, slip etc.

4.4 FEEDING UNIT

The material for the construction of the feeding unit is made of mild steel metal sheet gauge 16.

4.5 BELT DRIVE

V-Belts are used and the material for the construction is made of leather.

Table 4: MISCELLANEOUS COMPONENTS

Features	Material for Construction
Welds	Electric arc welding making use of gauge 10 and 12 electrodes. These were employed for joining parts and cutting of metal sheet where necessary.
Fasteners	13mm, 17mm and 19mm bolts, nuts and washers.
Paint	Oil Paint (gloss)

4.6 COSTING

This is the cost of producing the machine in terms of materials and labour

Table 5: MATERIAL COST

S/N	Material & Specification	Quantity	Unit Cost (₦)	Amount (₦)
1	Gauge 16 mild steel sheet	3	1,500	4,500
2	Mild steel shaft 30mm Ø by 1,200mm	1	1,200	1,200
3	Mild steel shaft 22mm Ø by 1,000mm	2	1,000	2,000
4	Mild Steel 6mm Ø rods by 6,096mm	1	950	950
5	Angled bars 50 x 50mm x 6mm			
		4	1,000	4,000
	Flat bar 37.5mm x 6mm			
6				
	Flat bar 25mm x 6mm	2	600	1,200
7				
	$^5/_8$" iron rod	1	350	350
8				
	BALL BEARINGS	2	200	1,480
9	FSP 205			
	FSP 206			
	6203Z	4	1,000	4,000
		2	1,500	3,000
	PULLEYS	4	100	400
10				
	300mm alluminium single groove			
	190mm alluminium single groove	1	2,000	2,000
	120mm alluminium double groove	1	1,500	1,500
	100mm alluminium single groove	1	1,500	1,500
		1	1,000	1,000

Table 5: cont'd

S/N	Material & Specification	Quantity	Unit Cost (₦)	Amount (₦)
11	Leather V-belts	3	3,000	900
12	Gauge10 electrodes	1	650	650
13	Bolts and nuts, 17, 13, 19mm	30	10	300
14	Ø3mm screen 57x100mm	1	1,500	1,500
15	Ø8mm screen 57x70mm	1	1,800	1,800
16	Ø8mm screen 57x120mm	1	2,000	2,000
17	gloss paint	1	1,000	1,000
18	petrol engine 10.5hp	1	20,000	20,000
19	Miscellaneous			3,000
	Sub-total Add 10% inflation for possible increase in Price			58,350
	Sub-Total			₦64,185

LABOUR COSTS

The following assumption were made for adequate labour costing with respect to the labour situation in the country as at the time of construction of this machine. Assume 8hrs/day.

i. Engineers fee = ₦640/day = ₦80/hr

ii. Machinists fee = ₦320/day = ₦40/hr

iii. Technician/Welder = ₦240/day = ₦30/hr

Table 6: Labour costing

S/N	Operation	Time(min)	No of Personnel	Rate ₦/hr	Amount ₦
A.	**THRESHING UNIT**				
1.	Cylinder				
	Marking	30	1 Technician	30/hr	15.00
	Cutting	30	-do-	-do-	15.00
	Rolling	120	"	"	60.00
	Welding	60	"	"	30.00
2	**Spikes**				
	Cutting	120	1 Technician	30	60.00
	Welding	180	1 Welder	30	90.00
3	**End Plates**				
	Marking	30	1 Technician	30	15.00
	Cutting	30	-do-	30	15.00
	Drilling	30	"	30	15.00
	Welding	60	1 Welder	30	30.00
4	**Shaft**				
	Machining	60	1 Machinist	40/hr	40.00
5	**FEEDING UNIT**				
	Marking	60	1 Technician	30	30.00
	Cutting	30	-do-	30	15.00
	Bending	30	"	30	15.00
	Welding	120	1 Welder	30	60.00
6.	**Concave**				
	Cutting	60	1 Technician	30	30.00
	Welding	120	1 Welder	30	60.00
7	**Cylinder Cover**				
	Marking	60	1 Technician	30	30.00
	Bending	60	-do-	30	30.00
	Welding	90	1 Welder	30	45.00

Table 6: cont'd

S/N	Operation	Time(min)	No of Personnel	Rate ₦/hr	Amount ₦
B	**BLOWER UNIT**				
1	**Blower (Fan)**				
	Marking	60	1 Technician	30	30.00
	Cutting	30	-do-	30	15.00
	Welding	90	1 Welder	30	45.00
2	**Shaft**				
	Cutting	30	1 Technician	30	15.00
	Machining	60	1 Machinist	40	40.00
3	**Housing**				
	Marking	30	1 Technician	30	15.00
	Cutting	30	-do-	30	15.00
	Rolling	30	"	30	15.00
	Bending	30	"	30	15.00
	Welding	60	1 Welder	30	30.00
4	**Housing Cover**				
	Marking	30	1 Technician	30	15.00
	Cutting	30	-do-	30	15.00
	Welding	30	"	30	15.00
C	**CLEANING UNIT**				
	Marking	30	1 Technician	30	15.00
	Bending	30	-do-	30	15.00
	Welding	30	1 Welder	30	15.00
D	**POWER UNIT**				
1.	**Motor Mounting**				
	Cutting	30	1 Technician	30	15.00
	Drilling	60	-do-	30	30.00
	Welding	60	1 Welder	30	30.00
2.	**Motor Installation**				
	Bolting	30	1 Technician	30	15.00

	Pulley assembly	30	-do-	30	15.00
	Belt tensioning	30	"	30	15.00
E	**BEARING INSTALLATION**				
	Installation	90	1 Machinist	40	60.00
	Welding	30	1 Welder	30	15.00
F	**Machine Support Frame**				
	Marking	60	1 Technician	30	30.00
	Cutting	120	-do-	30	60.00
	Welding	180	1 Welder	30	90.00
G	**Machine Assembly**				
	Threshing Unit	60	1 Engineer	80	80.00
	Blower Unit	60	-do-		
	Cleaning Unit	120	-do-		
	Power Unit	30	-do-		
	Painting				
	Painting	120	1painter		
	Sub-total	1,820			
	Production Cost = Material cost + Labour cost = 64,185 + 1820 Add 10% Contingency Allowance to Production Cost				₦6,600.50
	Total Production Cost = N66,005 + 6,600.5 Plus 15% contingency to cover designers fee and supervisor's fee.				₦72,605.5

NOTE: All the labour was supplied by the student under the supervision of the project adviser and the National Cereals Research Institute, Badeggi Engineering Workshop Staff.

The labour costs shown are therefore opportunity costs.

4.11 CONSTRUCTION AND OPERATION OF THE THRESHER

4.12 CONSTRUCTION PROCEDURE

The construction of the rice-thresher-cleaner was carried out in stages. The procedures followed are outlined below.

STAGE 1: THE FRAME

Material: 50 x 50mm mild steel angle iron

a. Cut the specified pieces of angle iron

b. Structure out the main frame, forming a 90° angle on each end, and tack all joints in position. The structure must not be given permanent welding. This gives room for necessary adjustments.

STAGE 2: CONCAVE

Materials: Ø8mm mild steel rod

37.5mm flat bar

a. Cut out 3 pieces 510mm flat bar

b. Form it into semi-circle

a. Line the concave longitudinally with Ø8mm mild steel rod with 30mm spacing

b. Tack it to the frame

STAGE 3: CYLINDER

Materiais: Ø30mm mild steel shafting material

37.5mm mild steel flat bar

Gauge 16 M.S. Flat plate

a. Cut 1050mm length of 30mm mild steel shaft

b. Machine each end to fit bearing diameter. These are meant to accommodate bearings and pulleys

c. Mark and mill out 130mm length of key-way on each end carrying pulley

d. Cut out 48 pieces of Ø8mm mild steel rod 50mm in length to form the threshing spikes.

e. cut out 3 pieces of 37.5mm of steel flat bar 940mm in length.

f. roll and form them into rings

g. cut out 8 of 37.5mm mild steel flat bar 380mm in length.

h. arrange the 8 bars at equidistant round the ring

i. weld the spikes at 500mm spacing along the length of the rotating cylinder

j. Cut out gauge 16 m.s. Flat plate 300mm diameter and cut out at the center 30mm holes to allow shaft pass.

k. Insert the shaft through the cylinder and attach bearings at both ends.

l. Weld the shaft to the drum

STAGE 4: THRESHING SPACE AND HOPPER

Materials: Gauge 16 mild steel sheet

a. Cut out 378.5mm x 706mm mild steel sheet to form the upper cover of the drum

b. Measure and cut out 260 x 330mm rectangular portion to form the hopper.

c. Mark out and bend the sheet at 230, 300 and 330mm.

d. Cut out 300 x 300mm square portion to pave way for the insertion of the hopper

e. Mark out four pieces of 200 x 200mm mild steel sheet bend to form the lower part of the hopper

f. Weld the lower part to the frame to cover the concave.

STAGE 5: BLOWER UNIT

Materials: Gauge 16 mild steel sheet

Ø22mm mild steel shafting material

a. Cut 940mm length of Ø22mm mild steel rod to form the blower shaft

b. Machine the two wends of the rod to the specified lengths and diameter for the insertion of bearings and pulleys

c. Mark and mill out the length of the key-way at one end of the shaft.

d. Cut out 3 of 160 x 680mm gauge 16 mild steel sheet to form the fan blades

e. Form involutes on mild steel sheet and cut out the shape, duplicate the involutes

f. cut out a circle of diameter 100mm at the center of each involute to form the air inlet into the blower unit.

g. Cut out the blower housing to the given specification

h. Tack the involute with the cover to hold the blower casing in position

STAGE 6: WINNOWING UNIT

Materials: Gauge 16 mild steel sheet

Ø8mm sieve plate

Ø3mm sieve plate

¾ " pipe

½ " angle iron

a. Mark and cut gauge 16 mild steel sheet 1290 x 1200mm and bend to 360 x 570 x 360mm into U-shape.

b. Cut the Ø8mm sieve plate 1200 x 570mm for the first sieve, 700 x 570mm for the second sieve.

c. Cut the Ø3mm sieve plate 1000 x 570mm for the third sieve

d. Cut ¾" pipe for the formation of the width of the sieve tray

e. Cut 2 each of ½" angle iron 1200, 700 and 1000mm to form the lengths of the sieve trays

f. Tack the pipes and angle iron according to specification at 90° and place the sieves on them.

g. Arrange the sieve in the U-shaped winnowing unit and mount it on the bearings

STAGE 7: THE DRIVE

The machine is equipped with V-belt pulley drive mechanism. The pulleys are made of Alluminium. Alluminium scraps, motor cycle engine blocks were purchased and melted in the foundry.

Pulley Dimensions:
(i) motor pulley - Ø120mm
(ii) Motor - cylinder pulley - Ø300mm
(iii) Cylinder – blower pulley - Ø190mm
(iv) Blower - cam pulley - Ø120mm
(v) Cam pulley - Ø100mm

a. Machine the Alluminium casts into the desired diameters

b. Make a V-groove of specified dimension on each of the pulleys

c. Drill out holes at the center of each pulley specified to each shaft diameter for the purpose of tightening the pulleys on the shaft.

FINAL ASSEMBLY

a. The cylinder was positioned over the concave screen with holes corresponding to those on the main frame through the slots on the bearing housing and bolt it using 19mm bolts, nuts and washers

b. The hopper and the upper cover of the cylinder were bolted to the main frame using 13mm bolt and nuts.

c. The main frame was given a permanent weld

d. The cover sheets (sides, front and back) were welded to the main frame.

e. The clean grain outlet was welded

f. The blower shaft with attached blades was inserted into the blower housing

g. The bearings and housing were inserted to hold the blower shaft in position horizontally to the support frame

h. It was ensure that the blades are free to weld the second involute to put the blade arrangement in free position.

i. The winnowing unit was placed under the concave on the bearings and bolted to the cam.

j. The pulleys were fitted to the drive shafts and keyed.

k. The belts were fitted and ready to start.

4.13 MAINTENANCE

The maintenance of the machine is very easy and this is achieved through the following provisions:

i. Bolting of the upper cover of the threshing chamber

ii. Bolting of the bearing housings on the main frame. This gives room for changing damaged bearings.

The recommended maintenance schedules for the machine is:

1. **Before Daily Operation**
   a. Check all the bolts and nuts and tighten loose ones
   b. Check all the bearings for damage
   c. Run the machine for five minutes to remove the entire trapped dirt Particles left over from the previous days operation

2. **After 4 hours continuous usage**

   Stop the machine and check if the bolted components are intact. If not, tighten the loose bolts and nuts and make other necessary adjustments and or replacement.

3. **After daily operation**
   a. Open the threshing chamber and clean all the trapped straws.
   b. Clean all the other parts of the machine

4. **After Seasonal Operation**
   a. Clean properly all the parts of the machine
   b. Lubricate all the bolts and bearings

Stationery concave

Concave clearance: 40mm

Separating Drum

Length:	680mm
Diameter:	300mm
Speed for threshing:	560rpm
Number of sieves:	3
Dimension of sieve	Ø8mm and Ø3mm round

It is possible to have the machine driven by an electric motor or diesel engine.

4.15 DESCRIPTION AND OPERATION OF THE MACHINE

The main features of the rice thresher are: The hopper, the transmission unit, reciprocating cleaning unit, straw outlet, grain outlet and the supporting frame. The drawings, and dimensions of the thresher are shown in Figs 18 - 20.

The hopper is trapezoidal in shape. It forms the feeding chute through which rice heads are fed into the threshing unit. The material of construction was gauge 16 mild steel sheet with all sides slanting inwards.

The transmission unit consists of five pulleys, bearings, shafts and v-belts. Two pulleys are mounted on the cylinder shaft, one on each side.

The threshing unit consists of a cylinder, beaters and a concave made of mild steel iron rods, formed into semi circle with 8mm spaced in between. The cylinder is placed above the concave, it is made of 37.5mm mild steel flat bar rolled into circle of 300mm and connected with eight rows of 37.5mm flat bars.

This arrangement and orientation is to aid the conveyance of the straw to the straw outlet. The beaters are made from mild steel iron rods, in each row, the beaters are spaced at 100mm from each other, and the clearance between the free end of beaters and concave is maintained at 40mm. The threshing unit is covered with steel plate (gauge 16) to prevent loss of rice grains through scattering.

A reciprocating sieve arrangement unit made of mild steel sheet arranged at a determined angle for free flow of paddy resting on four bearings is positioned 100mm below the concave. Below the sieve unit is the clean grain outlet. The fan is stationed below the first sieve to give a cleaning effect. It is an axial flow type with three blades made of gauge 16 mild steel sheet welded to a shaft.

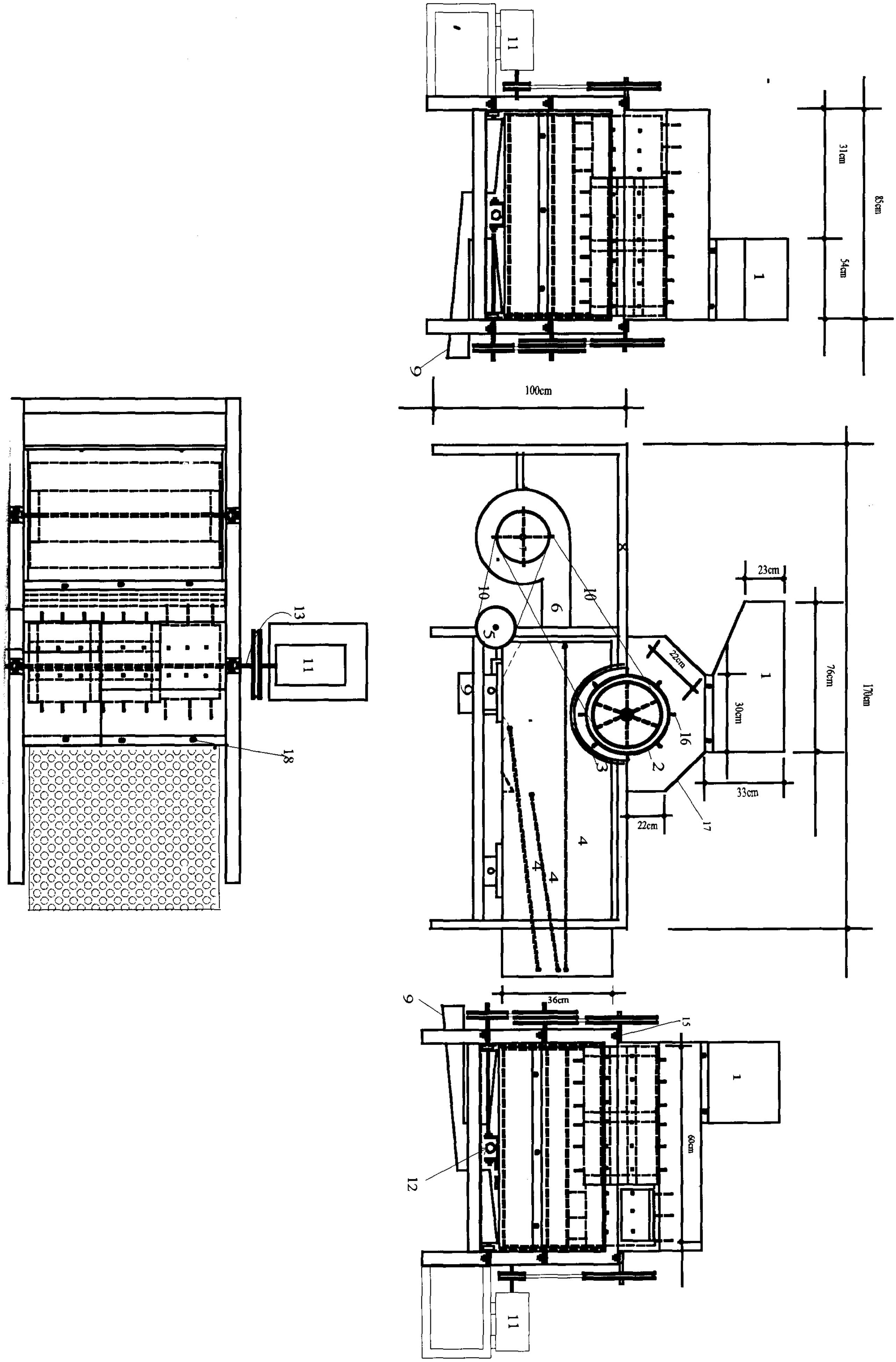
11
1
31cm
85cm
54cm
9
100cm
8
10
6
5
23cm
22cm
76cm
170cm
30cm
16
2
3
33cm
17
4
36cm
15
60cm
12
13
18

Table 7: Legends for Fig. 18

No. of item	Description and specification	Qty	Material
1	Trapezoidal feeding hopper (540x760x330mm)	1	Gauge 16 ms sheet
2	Threshing cylinder 2300x680mm	1	37.5mm flat bar
3	Threshing concave semi circular 2460x380mm	1	iron rods
4	Sieve ∅8x1200x570mm, ∅8x700x570mm and ∅3x1000x570mm.	3	gauge 16 ms sheet
5	Pulleys ∅120, ∅300, ∅190, ∅120 and ∅100mm.	5	Alluminium cast
6	Fan housing ∅450x530mm	1	Gauge 16 ms sheet
7	Fan blade ∅80x196x500mm	3	Gauge 16 ms sheet
8	Frame 50x50mm angle bar	1	Ms angle bar
9	Grain outlet ∅300x150mm	1	Gauge 16 ms sheet
10	v-belts AA 85, AA55 and AA38	3	Leather
11	Prime mover gasoline engine 10.5 hp	1	
12	Cam	1	
13	Shaft	1	
14	Straw outlet	1	
15	Bearings, FSP206, FSP205	6	FSP, 206, 205
16	Threshing teeth (beaters) 26x50mm	48	Ms iron rod
17	Cover plate (trapezoidal) 220x300x220x600mm	1	Gauge 16 ms sheet
18	Bolts and nuts, 13mm, 17mm, 19mm		

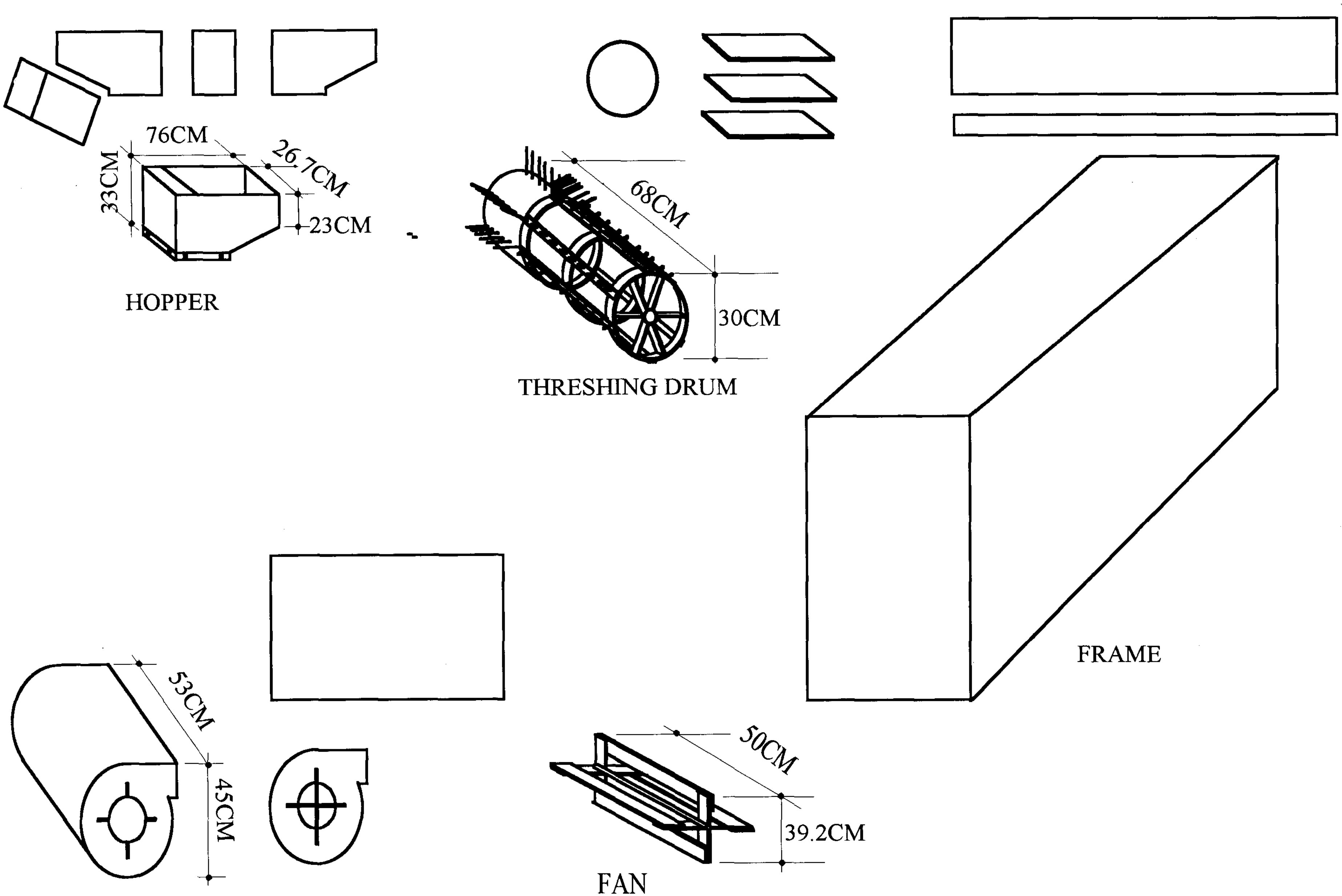

Fig. 19: Parts Drawing of Thresher

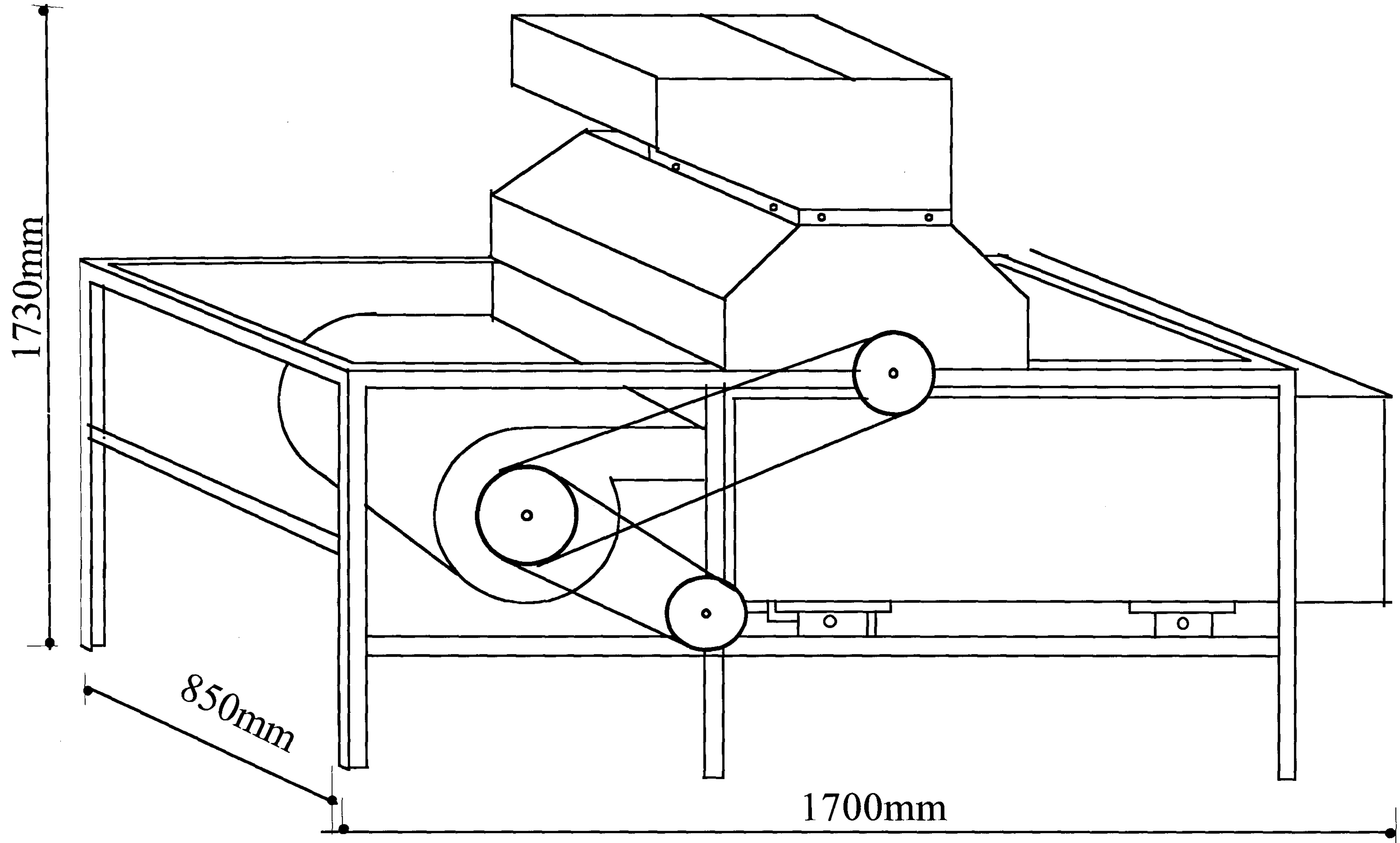

Fig. 20: OBLIQUE PROJECTION OF THRESHER

The structural frame forms the mounting support of all other units of the thresher and is made of 50mm angle iron. The overall dimension is 1700x1730x850mm. The Rice thresher works on principles of impact. Rice heads are fed uniformly into the hopper. The ear/heads fall by gravity on the rotating cylinder and are threshed by impact of the beaters and are whirled round between the concave and the rotating cylinder. The grains and little chaff fall through the concave openings onto the reciprocating sieve plate. Just before falling into the collection chute/grain outlet, the blower air stream blows off the chaff over the second screen, leaving behind clean grains. The sieve is automatically agitated to further grade the grain before collection at the grain outlet by the cam, see thresher picture in fig 21.

Fig. 21: Picture of the Rice Thresher

4.13 MECHANISM OF OPERATION

To operate the machine effectively, the operator should be able to identify the major component parts and understand the basic working principles of each component.

4.16 THE HOPPER

The hopper forms the opening through which dry head rice is fed into the threshing unit. It is rectangular in shape and the head rice falls into the threshing under the effect of gravity.

4.17 Threshing Unit

The threshing unit consists of the threshing spikes, threshing drum and the concave. There is a cover over the threshing drum to prevent grain scattering. This cover together with the concave section forms the threshing space. The threshing drum is cylindrical in nature and is made of 37.5mm flat bar rolled into circles and are connected together with 37.5mm flat bars, it is also provided with side boards made of gauge 16 metal sheet at both ends. A shaft drive 30mm in diameter is installed, which is supported by bearings. The threshing teeth made of hard steel bars are fitted to the threshing drum. The threshing teeth are placed along equi-distant lines on the drum. The rotating track interval of the threshing teeth is 40mm. The concave is provided under the threshing drum to separate grains and assist threshing simultaneously. It is made of steel bars 8mm diameter arranged 30mm in between.

4.18 Winnowing Unit

This consists of three sieves arranged at intervals to promote excellent grain separation. Grain that have passed through the concave drop along the first sieve, the second and third. On the way they are separated into whole grains, chaff and premature grains. The whole unit reciprocates on four bearing beneath the flow tray via a cam.

4.19 Blower Unit

The blower unit consists of the fan made of 3 blades, housing, and a shaft supported by bearings. The airflow is directed in between the first and second sieves this help separate grain effectively. Adjusting the volume of exhaust air regulates the airflow rate.

4.20 The Transmission Unit

The transmission unit is the drive mechanism, which transmits power from the engine to the threshing drum. The unit consists of pulleys, belts, bearings and the cam. Four different pulleys were used in the design of the thresher-cleaner. One pulley attached to the engine shaft 120mm in diameter, two to the drum shaft 300mm and 190mm diameters one double grooved pulley attached to the blower shaft 120mm diameter and the fifth attached to the cam shaft 100mm diameter. The three V-belt were properly fixed to connect the driver pulley from the engine A85 to the drum, the drum to the blower A55 and the blower to the cam A38. Two bearings of series FSP 206 were mounted on the drum shaft; two each of series FSP205 were mounted on the blower and camshafts respectively.

4.21 The Frame

The structural frame forms the mounting support for all the units of the thresher-cleaner. It is rectangular in shape and made of 50 x 50mm angle iron. Four pieces of the angle iron 850 x 1700mm were cut to form the rectangular frame. Arc welding was used to join the pieces using gauge 10 and 12 electrodes.

4.22 CLEAN GRAIN EJECTING UNIT

Having passed through the air stream from the blower, and the sieve arrangement, the clean grain are collected bellow through the conveniently positioned outlet.

CHAPTER FIVE

5.1 Performance Evaluation

Technical evaluation is a scientific method of ascertaining the technical condition of the various components of a system with a view to establishing how the components contribute to the overall efficiency of the system. For an objective evaluation of machine such as threshers, performance tests are to be carried out. Type tests are tests carried out on a thresher to prove the conformity with the requirements of relevant standards (or design specifications of the threshers). Type tests carried out include general tests, test at no load and test at load.

5.2 General Test

I. Checking of specifications

II. Checking of material

III. Visual observation and provision for adjustments

5.3 Test at no load

This is running the thresher at no load for at least half an hour at the specified revolutions of the threshing unit using an electric motor or appropriate power and recording the readings of the energy meter at intervals of five minutes (NCAM,19[illegible]). The difference between two consecutive readings gives power consumption for five minutes. This is used in calculating the power consumption at no load for one hour. During and after completing power consumption, the following observations are made and recorded.

I. Presence of any marked oscillation during operation

II. Presence of undue knocking or rattling sound

III. Frequent slippage of belts

IV. Other observations(if any)

5.4 Test at load

Test at load is of two types.

a. Short run test and

b. Long run test

Short run test enables the tester to get the following data.

I. Total losses

II. Threshing efficiency

III. Cleaning efficiency

IV. Power consumption

V. Input capacity

VI. Output capacity

The final sample obtained is analyzed for cracked and broken grains, refractions, unthreshed grains and clean grains. Analysis for cracked and broken grains is made only from the samples taken at specified grain outlets. Grain losses, threshing and cleaning efficiencies are also determined.

5.5 DETERMINATION OF TOTAL LOSSES

a. Percentage unthreshed grain

$$= \left(\frac{\text{quantity of unthreshed grain obtained from straws in kg}}{\text{total grain received at grain outlet in kg}} \right) \times 100$$

$$= \frac{5.33}{267.9} \times 100 = 1.99\%$$

b. Percentage of cracked and broken grain

$$= \left(\frac{\text{cracked and broken grain from grain outlet in kg}}{\text{Total grain received at grain outlet in kg}} \right) \times 100$$

From the sample taken, no broken or cracked grain seen. Therefore nothing was recorded.

$$= \frac{0}{267.9} \times 100 = 0\%$$

c. Percentage of blown grain

$$= \left(\frac{\text{quantity of clean grain obtained at straw outlet in kg}}{\text{Total grain input in kg}} \right) \times 100$$

There was no clean grain obtained at the straw outlet, hence there was nothing recorded.

$$= \frac{0}{267.9} \times 100 = 0\%$$

d. Percentage of sieve loss

$$= \left(\frac{\text{clean grain at sieve overflow + sieve underflow + stuck grain in kg}}{\text{Total grain input in kg}} \right) \times 100$$

The major component of the sieve loss is the sieve underflow. The loose grain under the sieve was found by spreading a mat under the machine to collect and check the losses. The grain obtained was weighed. The sieve overflow and stuck grain were also weighed.

Clean grain obtained at Sieve overflow = 2.0kg

Clean grain obtained at sieve underflow = 3.27kg

Stuck grain = 2.2kg

Total sieve loss = 7.47 kg

Percent sieve loss = $\frac{7.47}{267.9} \times 100 = 2.788\%$

e. **Total losses** = Sum of losses obtained at (a), (b), (c) and (d) above.

= 1.99 + 0 + 0 + 2.79 = 4.78%

5.6 DETERMINATION OF EFFICIENCIES

a. **Threshing Efficiency**: This is the ratio of total weight of grains threshed to the total weight of grains fed into the thresher for threshing expressed as a percentage. It is also the difference between 100% and the percentage of unthreshed grain.

Threshing efficiency (TE) = 100 - percent of unthreshed grain

= 100 – 1.99% = 98.01%

b. **Cleaning Efficiency**: The ratio of the weight of clean grains that pass through the cleaning unit to the total weight of grains at the outlet of the grain retainer expressed as a percentage.

Cleaning efficiency (CE) =

$$\left(\frac{\text{clean grain received at grain outlet in kg}}{\text{total grain received at grain outlet in kg}}\right) \times 100$$

$= \frac{266.0}{267.9} \times 100 = 99.32\%$

5.7 DETERMINATION OF OUTPUT CAPACITY

To determine the output capacity, the weight of threshed grain received at specified grain outlet is taken and recorded.

Capacity Utilization (CU) = $\frac{\text{output capacity}}{\text{Input capacity}} \times 100\%$

$$= \frac{267.9}{490} \times 100\% = 54.67\%$$

Threshing Index (TIX) = GRR x CU x TE, decimal

$$= 0.9522 \times 0.5467 \times 0.9801 = 0.510 \text{ (or } 51.0\%\text{)}$$

Threshing Intensity (TIN) = $\frac{\text{power consumed by thresher}}{\text{Output capacity of thresher}}$

$$= \frac{7.9KW}{267.9Kg} = 0.029KW/Kg$$

TABLE 8: TEST DATA SHEET

s/n	Date	Starting time	Stopping time	Duration of operation	Speed (rev/min)	Feed rate (kg/hr)	Power required (kw) or fuel consumed (lit/hr)	No. of primary sample	Quantity (kg) of primary samples from			Total quantity of grain train outlets	Total quantity at sieve underflow (kg)	Total quantity of material stuck in thresher (kg)
									Grain outlet(s)	Straw outlet				
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
1	*5/6/2002*	*4.30pm*	*5.30pm*	*1hour*	*560rpm*	*485*	*7.9kw*	i	*265.32*	*219.68*			*3.1*	*2.1*
2	*6/6/2002*	*3.30pm*	*4.30pm*	*1hour*	*560rpm*	*495*	*7.9kw*	ii	*268.50*	*226.50*			*2.9*	*2.3*
3	*7/6/2002*	*10.10am*	*11.10am*	*1hour*	*560rpm*	*490*	*7.9kw*	iii	*270.00*	*220.00*			*3.3*	*2.2*

TABLE 9: DATA SHEET FOR ANALYSIS OF FINAL SAMPLE

s/n	Feed (kg)	Threshed (kg)	Sample from	Weight of			
				Un-threshed grain (kg)	Cracked and broken grain (kg)	Clean grain (kg)	Registration
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
1	*485*	*265.32*	(i) grain outlets	-	-	*261.43*	-
2	*495*	*268.50*	(ii) straw outlet	-	-	-	-
3	*490*	*270.00*	(iii) sieve overflow	*3.03*	-	*2.0*	-
			(iv) sieve underflow	-	-	*3.27*	-
			(v) material stuck in thresher	*2.3*	-	*2.2*	-

CHAPTER S!X

RESULTS AND DISCUSSION

Preliminary studies on rice plant pertinent to the design process is presented in table 10. The parameters were necessary and hence used in the rice thresher design

Table 10: Some Physical and Engineering Properties of Rice

S/no.	Length of ears of Rice crop(cm)	Length of Rice Grain (mm)	Breadth of Rice grain (mm)	Grain/Straw ratio
1.	90	6.30	3.36	1.22
2.	92	7.56	3.05	1.25
3.	80	6.76.	2.85	1.20
4.	70	7.84	3.14	1.23
5.	89	6.49	3.76	1.30
6.	78	7.14	3.82	1.18
7.	93	6.50	3.46	1.18
8.	56	7.56	3.22	1.22
9.	50	6.82	3.44	1.24
10.	91	7.00	3.55	1.20
Mean	78.9	6.99	3.36	1.22
SD	14.77	0.49	0.29	0.03

The summary result of the performance evaluation of the rice thresher is presented in table 11.

Table 11: Results of performance evaluation of Rice Thresher

I.	Variety of rice handled:	FARO 51
II.	Grain/Straw ratio:	1:1.22 (average)
III.	Moisture content of rice handled:	13.83% (average) wet basis
IV.	Power requirement:	7.9KW (10.5hp) gasoline engine
V.	Losses:	
	a) Unthreshed grain:	1.99%
	b) Cracked and broken grain:	0%
	c) Sieve loss:	2.79%
	d) Blown grain	0%
VI.	Threshing efficiency:	98.01%
VII.	Cleaning efficiency:	99.32%
IX.	Input capacity:	490kg/hr.
X.	Output capacity:	267.9kg/hr.
XI.	Grain Recovery Range (GRR)	95.22%
XII.	Capacity Utilization (CU)	54.67%
XIII.	Threshing Index (TIX)	51.0%
XIV.	Threshing Intensity (TIN)	0.029%
XV.	Observations affecting performance:	
	a) Moisture content of rice crop and straw	
	b) Length of rice crop exceeding 93cm	
	c) Number of workers required is three	
	d) Machine requires continuous feeding	

The thresher is the hand fed type developed mainly for threshing rice, but has potentials of threshing other grain crops. The results presented in table .... represents the average performance evaluation of the rice thresher. The grain condition and environmental conditions were essentially uniform since threshing in Niger State is mainly carried out between December and January and grain conditions are averagely the same for FARO 51 rice variety.

Threshing intensity has been defined as the horsepower or kilowatt consumed by a thresher per unit output of rice paddy threshed. It is expressed in KW/Kg. Any value of threshing intensity much higher than 0.075KW/Kg (0.1hp) is unattractive as threshing could be more economically carried out manually. Threshing intensity of 0.029KW/Kg was computed for the thresher. Therefore the thresher is efficient in power utilization. The capacity utilization was calculated as 54.67%. capacity Utilization (CU) is defined as the ratio of the output to input capacities of a thresher expressed in percentage (NCAM, 1990). Grain Recovery Range (GRR) is expressed as the difference between 100% and percent total losses. GRR of 95.22% was computed for the thresher. Threshing index is the product of grain recovery range GRR, capacity utilization CU and threshing efficiency TE. A value of 0.51 (51.0%) was computed for the thresher.

CHAPTER SEVEN

SUMMARY CONCLUSION AND RECOMMENDATION

A rice thresher has been designed and developed to thresh the commonly available rice varieties. It will be very useful to both small and medium scale farmers.

The following could be concluded from the experimental results, the maximum output of the thresher was 267.9kg/hr at a design cylinder speed of 560rpm, the power required for operating the thresher was 7.9kw. The threshing efficiency was 98.01%, cleaning efficiency, 99.32% and the total percentage losses was 4.78%. The condition of the paddy rice head such as moisture content, grain/straw ratio, length of rice crop, grain length, cylinder speed, concave clearance and size of sieve aperture are machine-crop parameters that affect the performance of the thresher. Thus the use of this rice thresher can reduce the demand of labour during threshing and increase output at threshing period.

Based on the overall performance indices of threshing intensity, power requirement, capacity utilization, grain recovery range, threshing efficiency, cleaning efficiency, percent total losses and the attendants required, the thresher is recommended for small and medium scale rice farmers in Nigeria. We also recommend that government should grant revolving loans for local fabricators for the mass production of this thresher.

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APPENDIX A

Agronomic characteristics of Released Rice Varieties in Nigeria*

Variety	Cultivar Source	Ecology	Year of Release	Growth Duration	Grain Type	Reaction to Blast
FARO 1	BG-79	SS	1955	133-174	B	S
FARO 2	D-114	S	1958	135-176	B	S
FARO 3	Agbede	Upland	1958	95-120	B	S
FARO 4	KAV-12	DS	1959	189-220	B	MR
FARO 5	Makalioka 823	SS	1960	135-154	B	S
FARO 6	I.C.B.	SS	1961	176-198	B	MR
FARO 7	Mailong	DS	1962	160-217	A	MR
FARO 8	MAS 2401	SS	1963	155-60	A	S
FARO 9	SIAM-29	SS	1963	180-220	A	S
FARO 10	SINDANO	SS	1963	115-145	A	MR
FARO 11	OS-6	Upland	1968	115-120	B	R
FARO 12	SML-140/10	SS	1969	145-155	A	MR
FARO 13	IR-8	SS	1970	135-140	B	S
FARO 14	FRRS-43	DS	1971	170-198	B	MR
FARO 15	FRRS-162-B	SS	1974	145-160	B	MR
FARO 16	FRRS-168-11-2-B	SS	1974	140-160	B	MR
FARO 17	FRRS-148	SS	1974	145-160	B	MR
FARO 18	Tjina	SS	1974	167-179	B	R
FARO 19	IR-20	SS	1974	135-140	B	MR
FARO 20	BPA-76 (BICOL)	SS	1974	125-130	B	MR
FARO 21	Taichung Native 1	SS	1974	90-110	C	R
FARO 22	IR 627-131-3-37	IS & SS	1974	145-150	B	MR
FARO 23	IR 5-47-2	IS & SS	1974	145-150	B	MR
FARO 24	DeGaulle	IS & SS	1974	135-145	A	S
FARO 25	FAROX 56/30	Upland	1976	115-120	B	MR
FARO 26	TOS-78	IS & SS	1982	130-135	B	MR
FARO 27	TOS 103	IS & SS	1982	110-115	A	MR
FARO 28	FAROX-118A	IS & SS	1982	135-140	A	MR
FARO 29	BG90-2	IS & SS	1984	125-135	B	S
FARO 30	FAROX 228-2-1-1	IS	1986	110-115	B	R
FARO 31	FAROX 228-3-1-1	IS	1986	110-115	B	R
FARO 32	FAROX 228-4-1-1	IS	1986	110-115	B	R
FARO 33	FAROX 233-1-1-1	IS	1986	110-115	A	MR
FARO 34	FAROX 239-2-1-1	IS & SS	1986	105-115	B	MR
FARO 35	ITA 212	IS	1986	105-115	B	R
FARO 36	ITA 222	IS	1986	120-135	B	R
FARO 37	ITA 306	IS	1986	120-135	A	R
FARO 38	IRAT 133	Upland	1986	125-140	C	R
FARO 39	IRAT 144	Upland	1986	100-105	C	R
FARO 40	IRAT 299	Upland	1986	100-105	B	R
FARO 41	IRAT 170	Upland	1986	115-120	B	R
FARO 42	ART 12	Upland	1986	115-120	B	R
FARO 43	ITA 128	Upland	1986	115-120	B	R
FARO 44	SIPI 692033	IS	1992	100-115	A	R
FARO 45	ITA 257	Upland	1992	90-100	B	r
FARO 46	ITA 150	Upland	1992	100-105	B	
FARO 47	ITA 117	Upland	1992	110-115	A	
FARO 48	ITA 301	Upland	1992	110-11[illegible]	B	[illegible]
FARO 49	ITA 315	Upland	1992	115-[illegible]	B	R
FARO 50	ITA230	SS	1992	13[illegible]	B	R
FARO 51	CISADANE	IS & SS	1998	145-150		MR

*SS = Shallow Swamp; DS = Deep Swamp; IS = Irrigated Swamp

A = Long grain type; B = Medium grain type; C = Short grain type

R = Resistant; MR = Moderately Resistant; S = Susceptible

APPENDIX B

i. ASAE Standard: ASAE S343.1 (1981)- Terminology for combines and grain harvesting. ASAE, St. Joseph, MI, USA.

ii. NCAM test code for power maize Sheller (1990)*, Draft Nigerian Industrial Standard. Prepared by National Center for Agricultural Mechanization (NCAM), Ilorin for consideration and finalization by the NCAM-NSAE Technical Co-ordinating Committee on Agricultural Engineering Standards.

*The NCAM Test code was based on the following National and International (Regional) standards.

iii. IS 7052-(1973) Indian Standard test code for power maize Sheller, Bureau of Indian Standards, N. Delhi. India.

iv. RNAM test code and procedure for farm machinery (1983)- Test Code and procedures of power grain threshers. Published by economic and social commission for Asia and pacific, regional network for Agricultural Machinery, Philippines technical series no.12.

The following terminology apply to grain threshers (ASAE, 1981).

Cleaning: The isolating of desired seed or grain from chaff, small debris and un-hreshed material.

Concave: A concave-shaped stationary element adjacent to a threshing cylinder or rotor, fitted primarily to promote threshing.

Concave Clearance: The maximum gap or clearance between threshing cylinder and concave.

Grain Damage: For the purpose of this work, grain damage refers only to that attributable to the machine. It shall be expressed as the percentage by weight of damaged kernels in the sample.

a. **Visible Grain Damage:** Kernel damage where the seed coat appears broken to the naked eyes.

b. **Invisible Grain Damage:** Kernel damage, which requires instrumentation or special procedures for determination.

Material –other-than-Grain (MOG) to Grain ratio: The total weight of material-other-than-grain divided by the weight of grain in the sample.

Moisture Content: Moisture content of the crop shall be expressed on wet basis, it is the ratio of weight of water in product to total weight of product, expressed in percentage.

Rice Thresher: A machine operated by a prime mover to separate the rice grains from the spikiest on the panicle and also remove the straw from grains.

Threshing: The detaching of seed or grain from the head or spikiest.

Threshing Cylinder: A rotating element, which in conjunction with a stationary element adjacent to it, is fitted primarily to promote threshing. The crop being threshed is contained between rotating and stationary elements for less than 360%.

Threshing Efficiency: Percentage by weight of threshed grains from all outlets of the thresher with respect to total grain input.

Threshing Index: This is the product of grain recovery range (GRR) in percent, capacity Utilization (CU) expressed in percent and threshing efficiency (TE) in percent.

Notes: **(i) Grain Recovery Range =100 – percent total losses.**

(ii) Capacity Utilization = $\frac{\text{output Capacity x100\%}}{\text{Input capacity.}}$

Threshing Intensity: This is the house power (or kw) consumed by a thresher per unit out put of rice paddy thresher expressed in hp/kg or kw/kg.

Winnowing: Pneumatic cleaning of threshed grains.

The following terminology's apply to grain threshing (NCAM, 1990)

Cleaning Efficiency: Clean grain received at the specified grain out lets with respect to total grain received at grain out lets expressed as percentage by weight.

Feed Rate: The weight of the grains and parried feed into the thresher per unit time.

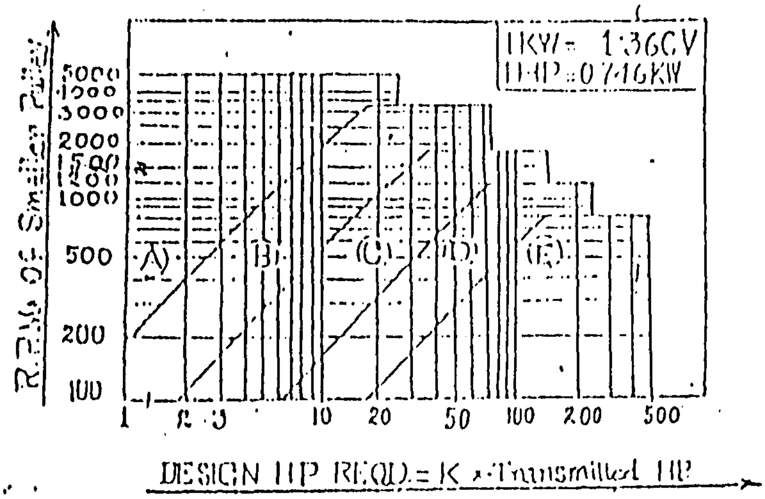
Input Capacity: The Maximum feed rate at which the power requirement is minimum and total losses and efficiencies are within the specified limits.

Output Capacity: The weight of grains received at the specified grain outlet when collected at input capacity.

Prime Mover: An electric motor, engine or tractor raised for turning a thresher.

Routine Test: Test carried out on a machine to check the requirements, which are likely to vary during production.

APPENDIX C
FACTORS FOR HORSE POWER RATINGS FOR V-BELTS



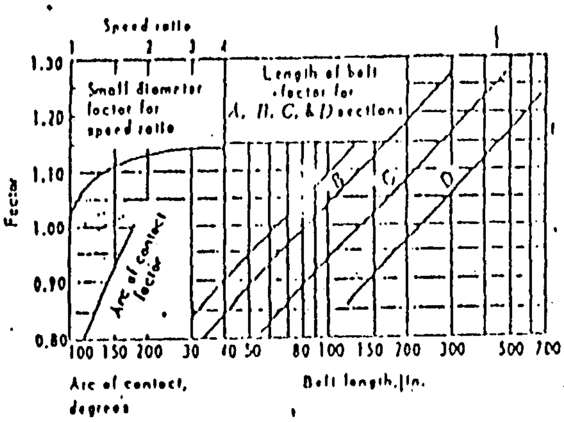


Fig. 4 II Factors for horsepower ratings of V belts.

URCE: Chakrabarty, N.K., Data book for machine designers)

APPENDIX D

DESIGN CALCULATIONS

1. CAM DESIGN CALCULATIONS

L = Length of cam follower

S = Stroke length (mm)

r = Cam radius

$Vc/Vm = 1.4 m/s$ ------------------- (1)

$Vc = 1.4 m/s \times Vm$ ---------- (2)

Where: Vc = flow tray speed, mls

Vm = forward speed of machine, mls

Flow tray speed is calculated from:

$$Vc = \frac{S \times Nc}{30} \qquad \text{\_\_\_\_\_\_\_\_\_\_} \quad (3)$$

(Devami and Pandey, 1985)

Where: Nc = Revolution per min. of the Cam

S = Stroke length

$$Nc = \frac{Vc \times 30}{0.080} \qquad \text{\_\_\_\_\_\_\_\_\_\_} \quad (4)$$

$$= \frac{1.4 \times 30}{0.080} = 525rpm$$

The speed of the Cam is:

$Nc = 1400 \times 1/1.16 = 1207rpm$

Assume belt slippage of 10%

$Nc = 1207 \times 0.9 = 1086rpm$

2. MOTOR-CYLINDER BELT DESIGN CALCULATIONS

$$\frac{PD_R}{PD_N} = \frac{RPM_N}{RPM_R}$$

Where: PD_R = Pitch diameter of driver (Prime mover) Pulley = 12cm

PD_N = Pitch diameter of the driven (Cylinder) pulley = 30cm

RPM_N = Speed of cylinder pulley

RPM_R = Speed of Prime mover (1400rpm)

Speed of cylinder $RPM_N = \frac{12 \times 1400}{30} = 560rpm$

If the diameter of the cylinder is 30cm

Then speed of cylinder $= \frac{\pi \times 0.300 \times 560}{60} = 8.79m/s$

3. ANGULAR VELOCITY OF MOTOR-CYLINDER BELT

$\omega_1 = \frac{2 \times \pi \times 1400}{60} = 146.6rad/sec.$

$\omega_2 = \frac{2 \times \pi \times 560}{60} = 58.64rad/sec.$

POWER ON MOTOR – CYLINDER BELT

Power = Torque x angular velocity

$= T\omega$

Torque on prime mover pulley to accelerate the cylinder = tm and tm $= \omega_1 r_1$

r_1 = radius of motor pulley

Hence, power = tm $\omega_1 = \omega_1^2 r_1$

Therefore, power delivered by the prime mover.

$Pm = (146.6)^2 \times \frac{0.12}{2} = 1289.5watts$

For efficiency of 95%, Pm = 1289.5 x 0.95 = 1225watts

Power required to drive the cylinder, $Pc = \omega^2 r_2$

LENGTH OF MOTOR – CYLINDER BELT

The entire length of belt is calculated from the formular

$$L = 2CD + \frac{(D + d)}{2} + \frac{(D - d)^2}{4CD}$$

$$L = 2 \times 219 + \frac{(300 + 120)}{2} + \frac{(300 - 120)^2}{4 \times 219}$$

$L = 684.9mm$

The length correction factor, K_L is obtained from table to be 0.84

Therefore, the corrected belt length $= K_L \times 683.9$

$= 0.84 \times 683.9$

$= 575.4mm$

5. CYLINDER-BLOWER BELT DESIGN CALCULATIONS

Blower velocity $V_b = \frac{\pi Dn}{60} = \frac{\pi \times 0.20 \times 560}{60} = 5.9mls$

The expression for the speed ratio is

$$\frac{PD_R}{PD_N} = \frac{RPM_N}{RPM_R}$$

Where: PD_R = Pitch diameter of the driver pulley (cylinder)

PD_N = pitch diameter of the driven pulley (blower)

RPM_N = speed of the blower pulley

RPM_R = speed of the cylinder pulley

Let $PD_R = 0.190 = 2r_3$

But $RPM_R = 560rpm$

Substituting into the above equation, RPM_N is evaluated thus:

$$\frac{0.190}{0.120} = \frac{RPM_N}{560}$$

$$RPM_N = \frac{0.190 \times 560}{0.120} = 886rpm$$

ANGULAR VELOCITY OF CYLINDER – BLOWER PULLEYS

Let the angular velocity of cylinder pulley and the blower pulley be ω_3 and ω_4 respectively.

$$\omega_3 = \frac{2\pi N_3}{60} = \frac{2\pi 560}{60} = 58.6rad/sec.$$

$$\omega_4 = \frac{2\pi N_4}{60} = \frac{2\pi 886}{60} = 92.78rad/sec.$$

POWER ON CYLINDER – BLOWER PULLEYS

Torque on cylinder pulley to accelerate the blower

$Tc = \omega_3 r_3$

Power $= Tc\omega_3 = \omega^2{}_3 r_3$

Therefore power delivered by the cylinder pulley

$$Pc = (58.6)^2 \times \frac{0.120}{2} = 206W$$

At efficiency of 85%, Pc = 206 x 0.85 = 175W

Power required to operate the blower

$$= 92.78 \times \frac{0.120}{2} = 5.5668W$$

CENTRE DISTANCE OF CYLINDER – BLOWER PULLEYS

Center distance

CD = max (2R, 3r + R)

Where:

r = radius of blower pulley $= \frac{0.120}{2}$

R = radius of cylinder pulley $= \frac{0.190}{2}$

hence $CD = \max\left(2 \times \frac{0.190}{2},\ 3 \times \frac{0.120}{2} + \frac{0.190}{2}\right)$

$= (0.190, 0.275)$

$= 275mm$

Since CD is not less than the diameter of the larger pulley it can be used.

$\O_R$ = angle of rap of cylinder pulley

$= \pi + 2\sin^{-1} \frac{(190 - 120)}{2 \times 275} = 10.5^o$

$\O_R$ = angle of rap of blower pulley

$\pi - \sin^{-1} \frac{(190 - 120)}{2 \times 275} = -4.17^o$

LENGTH OF CYLINDER - BLOWER BELT

The expression for the length of open V-belt is given as:

$$L = 2CD + \frac{(D + d)}{2} + \frac{(D - d)^2}{4CD}$$

Substituting into the equation

$$L = 2 \times 275 + \frac{(190 + 120)}{2} + \frac{(190 - 120)^2}{4 \times 275}$$

$= 709.45mm$

From Appendix C.

Length correction factor (K_L) = 0.84

Corrected belt length = 0.84 x 709.45

= 595.94mm

SURFACE SPEED OF CYLINDER-BLOWER BELT

From the large pulley,

Circumference of large pulley $= 2\pi r_3$

$$= \frac{2\pi (0.190)}{2}$$

$$= 0.597m$$

But speed = 560rpm = 9.33rps

Length covered per second of rotation is $8.83 \times 0.190 \times \pi$

= 5.3mls

From the small pulley:

Circumference $= 2\pi r_4 = 2\pi \frac{(0.120)}{2} = 0.377m$

Length covered per second of pulley rotation is

$10.8 \times 0.120 \times \pi = 4.07m/s$

CYLINDER-BLOWER BELT CONFIGURATION

Blower operating for a maximum of 12 hours/day falls under light duty load and service factor, K = 0.85

$$\text{Design horse power} = \frac{\text{Horse power}}{\text{Service factor}} = \frac{3.156}{0.85} = 3.713KW$$

$$= 4.23hp$$

The values of design horse power and that of the speed of the blower, place the belt in "A" section

$K_L = 0.84$
$K\emptyset_3 = 1.00$

Therefore corrected horse power $= K\emptyset_3 \times K_L \times$ design hp

$= 1 \times 0.84 \times 4.23$

$= 3.6hp$

$= 2.7KW$

One belt is used.

6. BLOWER-WINNOWER BELT DESIGN CALCULATIONS

$$\frac{PD_R}{PD_N} = \frac{RPM_N}{RPM_R}$$

Where: PD_R = Pitch diameter of blower pulley (driver)

PD_N = pitch diameter of winnower pulley (driven)

RPM_N = speed of the winnower pulley

RPM_R = speed of blower pulley

Substituting into the above equation

$$\frac{120}{100} = \frac{RPM_N}{\text{~~560~~ } 886}$$

$$= \quad RPM_N = \frac{120 \times 886}{100} = 1063rpm$$

ANGULAR VELOCITY OF BLOWER – WINNOWER PULLEYS

Let the angular velocities of the blower and winnower pulleys be ω_5 and ω_6

$$\omega_5 = \frac{2\pi N_5}{60} = \frac{2\pi 886}{60} = 92.8rad/sec$$

POWER ON BLOWER – WINNOWER PULLEYS

Torque on blower pulley to accelerate the winnower

$$TC = \omega_5 r_5$$

$$Power = Tc\,\omega_5 = \omega_5^2 r_5$$

Therefore power delivered by blower pulley is:

$$PC = (92.8)^2 \times \frac{0.120}{2} = 516watts$$

at 85% efficiency, $PC = 516 \times \frac{85}{100}$

$= 493W \quad = 0.175kw$

CENTER DISTANCE OF THE BLOWER – WINNOWER PULLEYS

Center distance

$CD = \max(2R, 3r \pm R)$

Where:

r = radius of winnower pulley $= r_6 = \frac{0.100}{2} = 0.05m$

R = radius of blower pulley $= r_5 = \frac{0.120}{2} = 0.06m$

$CD = \max(2 \times 0.06, 3 \times 0.05 + 0.06)$

$= \text{Max}(0.12, 0.21)$

$= 0.23m$

ANGLE OF RAP OF BLOWER-WINNOWER BELT

$Ø_L$ = Angle of rap of winnower pulley

$= \pi + 2Sin^{-1} \frac{(120 - 100)}{2 \times 230} = 8.13^0$

$Ø_S = \pi - 2Sin^{-1} \frac{(120 - 100)}{2 \times 230} = -1.84^0$

LENGTH OFBLOWER - WINNOWER BELT

$L = 2CD + \frac{(D + d)}{2} + \frac{(D - d)^2}{4CD}$

$= 2 X 230 + \frac{(120 + 100)}{2} + \frac{(120 - 100)^2}{4 \times 230}$

$= 570.43mm$

Appendix C,

$K_L = 0.91$

Corrected length $= 0.91 \times 570.43$

$= 519.1mm$

BELT SURFACE SPEED

At the large pulley

Circumference of the larger pulley

$$= 2\pi \frac{120}{2} = 376.99mm$$

Speed = 688rpm = 11.2rps

The belt moves at the same speed on the pulley at its pitch diameter.

Length covered per second of pulley rotation $= 0.120\,\Pi \times 11.2 = 4.2m/s$

From the small pulley

$$\text{Circumference} = 2 \times \pi \frac{100}{2} = 314.16mm$$

Length covered per second of pulley rotation is:

$0.100 \times \pi \times 11.2 = 3.52m/s$

BELT CONFIGURATION

Winnower operating for a maximum of 12hrs/day falls under heavy duty load,

$K = 1.3$

Design horse power $= 0.233 \times 1.3 = 0.30hp$

But $KØs = 0.95$, $K_L = 1$

So corrected horse power = KØs x KL x Design horse power

$= 0.95 \times 1 \times 0.30$

$= 0.285hp$

$$\text{Number of belts} = \frac{\text{Design horse power}}{\text{Corrected horse power}} = \frac{0.30}{0.285} = 1.053$$

One belt of the configuration is used.

7. FAN-BLOWER DESIGN CALCULATIONS

Area of each paddle $= 0.8 \times 0.1 = 0.08 m^2$

Sheet thickness $= 1.0mm$

Volume $= 0.08 \times 1 \times 10^{-3} = 8.0 \times 10^{-5} m^3$

Total Volume $= 3 \times 8.0 \times 10^{-5} = 2.4 \times 10^{-4} m^3$

Mild steel of density $7550 kg/m^2$

Max $= 7850 \times 2.4 \times 10^{-4} = 1.884 kg$

Weight of blower paddles $= 1.884 \times 9.81 = 18.48N$

0.8m Length, load $= \frac{18.48}{0.8} = 23.1 Nm^{-1}$

$T3 = 104.31$

$T4 = 37.28$

$W = 23.1 Nm^{-1} \times K$ where $K = 1.5$ = factor of safety

$= 23.1 \times 1.5 = 34.65 Nm^{-1}$

8. SHAFT DESIGN CALCULATIONS

$Ty = T_3 \sin 30^\circ + T_4 \sin 30^\circ$

$= 104.31 \sin 30^\circ + 37.28 \sin 30^\circ$

$= 70.80N$

$Tx = T_3 \cos 30^\circ + 37.28 \cos 30^\circ$

$= 104.51 \cos 30^\circ + 37.28 \cos 30^\circ$

$= 122.62N$

SHAFT LOADING IN THE VERTICAL DIRECTION CALCULATIONS

$30.8 (0.8) = RV_1 + RV_2 + 70.8$

$-46.16 = RV_1 = RV_2$ -- ------ (6)

Taking moment about RV_2

$RV_1 = (0.95) - 30.8 (0.8) (0.475) - 70.8 (0.05) = O$

$RV_1 = 16.05N$

From (6)

$RV_1 = -62.21N$

Showing that RV_2 is acting in opposite direction

Taking moment about A.

$BMA = 16.05 (0.075) = 1.204Nm$

Taking moment about B,

$BM_B = 70.8 (0.125) - 62.21 (0.075) = 4.184Nm$

For maximum bending moment at C

$RV_1 - W(x - 0.075) = 0$

$16.05 - 30.8x + 2.81 = 0$

$x = 0.596$

bending moment at x = 0.596m

$$BMc = RV1 (x) - \frac{w}{2} (x - 0.075)^2$$

$$= 16.05 (0.596) - \frac{30.8}{2} (0.596 - 0.075)^2$$

$= 5.39Nm = Bmax V \times 1.5 = 8.085Nm$

Maximum bending moment in the vertical direction is 8.085Nm

SHAFT LOADING IN THE HORIZONTAL DIRECTION

$RH_1 + RH_2 = 122.62N$ -------------- (7)

Taking moment about RH_1

$-RH_2 (0.950) + 122.62 (1.0) = O$, $RH_2 = 129.07N$

Substituting into equation -------------- (7)

$RH1 = - 6.45N$

Indicating that RH1 acts in the opposite direction, maximum bending moment occurs at point B.

$BM_B = 122.62\,(0.05) = 6.13Nm = Bmax\,h$

RESULTANT BENDING MOMENT

Maximum bending moment $= \sqrt{Bmax\,h^2 + Bmax\,V^2}$

$= \sqrt{5.39^2 + 6.13^2} = 8.163Nm$

(D) **DIAMETER OF SHAFT**

The material selected for the shaft is smooth mild steel iron rod

Design bending moment = Bmax x 2

= 8.163 x 2 = 16.33Nm

Design Tonque = 2 x Tmax

= 2 x 3.183 = 6.366Nm

The material has a maximum shear stress

$= 40 \times 10^6\ N/m^2$

$$d^3 = \frac{16}{\pi \times Tmax} (Fbm)^2 + (KLT)^2$$

$$d = \left(\frac{16}{\pi \times 40 \times 10^6} \sqrt{(1.5 \times 16.33)^2 + (1.5 \times 6.366)^2} \right)^{1/3}$$

d = 0.022Cm

= 22mm

HEIGHT OF FALL

Total area covered by paddle

$= 3 \times 0.1 = 0.3m^2$

for a blower velocity of 6.8m/s, discharge or volume of air produced

$= 0.3 \times 6.8 = 2.04 m^3/s$

Terminal velocity of rice = 5.37m/s

Initial velocity = 0

But $V^2 = U^2 + 2gh$

V = terminal velocity

U = initial velocity

h = height of fall

g = acceleration due to gravity

that is:

$5.37^2 = 0 + 2 \times 9.81 \times h$

h = 1.47m

9. CYLINDER SHAFT DESIGN CALCULATIONS

Volume = Circumference x length x thickness

$= 2\pi (155) \times 600 \times 1.0$

$= 7.79 \times 10^5 mm^3$

Since density of steel is $7800 kg/m^3$

Mass $= 7800 \times 7.79 \times 10^5 = 6.077 kg/m^3$

CYLINDER END PLATES

Two end plates made of mild steel sheet were used

Diameter = 300mm

Thickness = 1mm

Volume $= \pi r^2 (\text{thickness}) = \pi \left(\frac{300}{2}\right)^2 \times 1.0mm = 7.5 \times 10^4 mm^3$

But a bore of diameter 30mm was opened at the center of each plate. This hole is where the shaft passes before being welded to the plates.

Therefore, the volume of each of these plates is the volume less the bole.

i.e.

$$\text{Volume of hole} = \pi r^2 = \pi \frac{(30)^2}{2} \times 1.0 = 706.858mm^3$$

Therefore Volume of each cylinder end plate

= Volume of plate - Volume of hole

$= 7.5 \times 10^4 - 706.858 = 7.4 \times 110^4 mm^3$

Mass of the cylinder plates

= 2(Volume x density)

$= 2(7.4 \times 10^4 \times 7800)$

= 1.15kg

SPIKES ON THE CYLINDER

Ahuja and Sharma, 1989 established spike spacing for his manually operated thresher at 30 – 50mm. Most existing threshers have one legged spike. In this design, one-legged spike of 80mm x 120mm spacing is used.

The spike is made of mild steel with height = h = 55.Sin 60 = 47.63mm

Diameter = 6mm

$$\text{Volume of each spike} = \pi r^2 \text{ (Length)} = \frac{(6)^2}{2} \times 55$$

$$= 1.555 \times 10^3 mm^3$$

mass of each spike

= Volume x density

$= 1.555 \times 10^3 \times 10^{-9} m^3 \times 7800$

$= 1.213 \times 10^{-2} kg$

LOADING ON CYLINDER SHAFT

$TY_1 = T_1 \, Sin30 + T_2 \, Sin30 + T_3 Sin30 + T_4 Sin30$

$= 212 \, Sin30 + 88 \, Sin \, 30 + 104.31 \, Sin \, 30 + 37.28 \, Sin \, 30$

$= 106 + 44 + 52 \; 155 + 18.64 = 220.80N$

$TX_1 = T_1 \, COS \, 30 + T_2 \, COS \, 30 + T_3 \, COS \, 30 + T_4 \, COS \, 30$

$= 212 \, COS \, 30 + 88 \, COS \, 30 + 104.31 \, COS \, 30 + 37.28 COS \, 30$

$= 382.43N$

$TY_2 = T_5 \, Sin \, 90 + T_6 \, Sin \, 90$

$= 43.35 \, Sin \, 90 + 18.09 \, Sin \, 90$

$= 43.35 + 18.09$

$= 61.44N$

$TX_2 = O$

CYLINDER SHAFT LOADING IN VERTICAL DIRECTION

$220.80 + 131.0 \, (0.8) + 61.44 = Ra + Rb = 387.04$ ------------ (8)

Taking moment about Ra

$TY_2 \, (1.000) - R_6 \, (0.900) + 131 \, (0.800) (0.450) - TY_1 (0.100) = O$

$61.44 \, (1.0) - Rb \, (0.9) + 131 \, (0.8) (0.45) - 220.80 \, (0.1) = O$

$61.44 - Rb \, (0.9) + 47.16 - 22.80 = O$

$Rb = 96.13N$

Substituting into equation (8)

$Ra = 290.91N$

Taking moment about A,

$BM = -220.80 (0.1) = -22.08NM$

Taking moment about B,

$BM = - 220.80 (0.15) - 290.91 (0.05) = -18.575NM$

Taking moment about D,

$BH = 61.44 (0.15) – 96.13 (0.05) = 4.4Nm$

Taking moment about E

$BM = 61.44 \times 0.1 = 6.144Nm$

Bending moment at C,

$Ra - TY_1 - W (x - O.15) = O$

$X = O.685m$

For Bending moment at x = O.O685m

$$BMC = TY1 \times Ra (x – 0.1) - \frac{W}{2} (x - O.15)^2$$

$$= -220.8 (0.665) + 290.91 (0.685 – 0.1) - \frac{131}{2}$$

$$(0.665 - 0.15)^2 = - 151.248 + 170.182 - 18.78$$

$$= 0.186Nm$$

Maximum bending moment in the vertical direction is 22.08Nm

CYLINDER SHAFT LOADING IN HORIZONTAL DIRECTION

$Ra + Rb = 382.43.71 = 386.14N$ ------------- (9)

Taking moment about Ra

$3.21 (1.045) – rb (0.9) - 382.43 (0.1) = 0$

$Rb = -38.185N$

Showing that Rb acts in the opposite direction

Substituting into eqn. (9)

Ra = 424.33N

Taking moment about A

Bm = 382.43 (0.1) = 38.24Nm = Bmaxh

Bm = 3.71 x 0.145 = 0.538Nm

Maximum bending moment occurs at point A

RESULTANT BENDING MOMENT

Maximum bending moment $= \sqrt{B^2max\ h + B^2max\ V} =$

$\sqrt{(38.24)^2 + (22.08)^2}$

= 44.16Nm = Bmax

TORISIONAL LOAD

Torque, T $= \frac{\text{Power}}{\text{Angular velocity}}$

$= \frac{575 \times 60}{211 \times 530} = 10.36Nm$

Twisting moment = 10.36Nm = Tmax = 10.36 x 1.5

= 15.54Nm

DIAMETER OF SHAFT

Light - Shock factor for bending, K_b = 1.5

Light - Shock factor for torsion, K_c = 1.5

Smooth round steel bar was used for the shaft

Design bending moment = Bmax x 2 = 44.16 x 2 = 88.32Nm

Design (torque = (Tmax) x 2 = 10.36 x 2 = 20.72Nm

The material has a maximum shear stress as Tmax $= 40 \times 10^6$ N/m^2

From $d3 = \frac{16}{Øx\ 40 \times 10^6} \quad (K6m)^2 + (K + T)^2$

$$d = \left[\frac{16}{Ø \times 40 \times 10^6} \quad (1.5 \times 88.32)^2 + (1.5 \times 20.70)^2 \right] 1/3$$

$d = 0.0259m = 26mm$

from standard shaft, 25mm is selected

DESIGN FOR TORSIONAL RIGIDITY

$Ø = \frac{TL}{GJ}$ ------------ (10)

But $Ø_{all} = \frac{0.88}{300} \times : = \frac{0.08}{300} \times 1145 = 0.3053^o = 0.005rad$

Substituting into (10)

$$Ø = \frac{10.36 \times 1.145}{80 \times 10^9 \times J}$$

$$J = \frac{10.36 \times 1.145}{81 \times 10^9 \times} = \frac{d^4\pi}{32}$$

Using $Ø = Ø_{all}$

$$d^4 = \frac{32 \times 10.36 \times 1.145}{\pi \times 80 \times 10^9 \times 0.005}$$

$= 0.0234m = 23mm$

from standard shaft, 25mm is selected

10. SHAFT BEARING CALCULATIONS

Loading carried at point A has a resultant

$$Fra = \sqrt{R^2av + R^2ah}$$

$$Fra = \sqrt{290.91^2 + 424.33^2} = 514.48N$$

At Point B

$$Frb = \sqrt{Rb^2 \times 96.13^2} = 103.44N$$

Finding the equivalent load Pe, one use

Pe = XVFr x Yfa

Pe = equivalent radial load

Fr = applied radial load

Fa = applied thrust load

V = rotation factor

X = a radial factor

Y = a thrust factor

The rotation factor V is to correct for various rotating ring conditions. For a rotating ring V = 1- this is the situation on this design. The X and Y factor depend upon the geometry of the bearing including the number of balls and ball diameter. For this design, thrust load are not applied hence, Fa = O and Y = 0, X = 1.

At Point A

Pe = 1 x 1 x 1 x 514.48 = 514.48N (52.44kgf)

At Point B

Pe 1 x 1 x 103.44 = 103.44N (10.54kgf)

Number of revolution per (N) minute of shaft in bearing = 560rpm

Let the bearing have a reliability of 97%

Since the system operates for 48 hours per week (having six working days), 50 weeks per year, and for a probable life of 15 years. Life in hours = 48 x 50 x 15 = 36,000 hours

Using Kleibull expression

$$R = e^{-\left(\frac{L}{6.64L_{10}}\right)^{1.17}} \quad \text{----------- (11)}$$

Where:

R = Reliability (97%)

L = bearing life in hours (36,000 hrs)

L_{10} = rated life = Life of bearing for 97% survival at 1 million revolutions

From equation (11)

$$\ln R = -\left(\frac{L}{6.84\,L_{10}}\right)^{1.17}$$

$$L_{10}hr = \frac{36{,}000}{-(\ln R)\,1.17\,(6.84)}$$

$$= \frac{36{,}000}{-\ \ln(0.97)\ \text{x}\ 6.84} = 104{,}043.13 \text{ hrs.}$$

L_{10} hr = 104,043.13 hours

L_{10} revs = 104,043.13 x 560 x 60 = L_{10}hr x N x 60

= 3.308571527×10^9 revs

= 3309 million revolutions

= 3.309×10^6 revolutions

Since Fra > frb

Fra is used in this design

The dynamic load capacity is given by

$$C = \left(\frac{L_{10}\,rev}{L_{10}}\right)^{1/K} \times Fra + K_s$$

Where:

Fra = Load or bearing

K_s = service factor =1.5 for rotary machine

K = exponent = 3 for small bearings

$$C = \left(\frac{3309 \times 10^6\,rev}{10^6 rev}\right)^{1/3} \times 514.48 \times 1.5$$

= 11499.85N (1172.26kgF)

Basic dynamic capacity = 1021.13N

FS bearing P206 was selected from the series.

D = 42mm

B = 14mm

A = 25mm

BLOWER SHAFT BEARING

$$Fra = \sqrt{R^2v + R^2h}$$

$$= \sqrt{16.05^2 \approx 6.45^2}$$

= 17.30N

$$Frb = \sqrt{R^2v + R^2h}$$

$$= \sqrt{62.212 + 129.072}$$

= 143.28N

Number of revolutions for minute (N) of shaft in bearing = 648rpm. Let the bearing reliability be 97% rated life, $L_{10}hr$ = 104,043.13 hours

L_{10} revs = 104,043.13 x 648 x 60

= 4.045.96894 x 10^9 revs

= 4045 x 10^6 revs

Frb = 143.28N is used in the design

The dynamic load capacity = 3424N

FS bearing P205 was selected for the blower

D = 20mm

B = 12mm

A = 22mm

CAMSHAFT BEARING

Fra $= \sqrt{R^2v + R^2h}$

$= \sqrt{16.052 + 6.45}$

= 17.30N

Frb $= \sqrt{R^2v + R^2h}$

$= \sqrt{62.21^2 + 129.07^2}$

= 143.28N

Number of revolution per minute of shaft in bearing = 672rpm. Let the bearing reliability be 97%.

Rated life = L_{10} hr. = 104,043.13hrs.

L10 rev. = 104,043.13 x 672 x 60

= 4.238717116 x 10^9 rev

$= 4238 \times 10^6$ rev

Frb = 143.28N is used hr

Fs bearing P205 was selected for the cam shaft

D = 20mm

B = 12mm

d = 22mm

11. FRAME DESIGN

$$\frac{Fc}{Pc} + \frac{Fbc}{Pbc} < 1 \text{ --------------- } (12)$$

Where:

Fc = actual direct axial stress

Fbc = actual direct bending stress

Pbc = allowable bending stress

Pc = allowable axial stress

But $Fc = \frac{F}{A}$ -------------- (13)

Where: F = axial load

A = cross sectional area of the section

$$Fbc = \frac{M}{Z} \text{ --------------- } (14)$$

Whr: M = Moment

Z = Sectional modulus

From $Fbc = \frac{M}{Z}$

Assume frame to be rectangular

$Y = 850mm$

$X = 1700mm$

$M_{xx} = 121.64 \times 0.3 = 36.49Nm$

$M_{yy} = 121.64 \times 0.225 = 27.37Nm$

$Z_{xx} = \frac{bd^2}{6}$ ------------------ (15)

$Z_{yy} = \frac{db^2}{6}$ ----------------- (16)

Where:

B = width, d = depth

$Z_{xx} = \frac{0.85 \times 1.70^2}{6} = 0.409m^3$

$Zyy = \frac{1.70 \times 0.85^2}{6} = 0.205m^3$

Therefore Fbc = 36.49 = 892.18N/m²

$\underline{Pc} + \underline{Fbc} < 1$