DESIGN AND CONSTRUCTION OF AN ANIMAL WASTE COLLECTOR

BY

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THESIS SUBMITTED TO

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DECLARATION

I declare that the work described in this thesis represents my original work and has not been submitted for any degree to any university or similar institution.

Gabriel Sunday Kolo.

CERTIFICATE OF APPROVAL

The undersigned hereby certify that this thesis work presented by the candidate be accepted as fulfilling part requirement for the award of degree of Masters in Agricultural Engineering, Federal University of Technology, Minna.

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ABSTRACT

This research work involves the development of an Animal waste collector affordable to the peasant farmers.

The material chosen for investigation is the cowdung, which can be composed into manure. Cowdung is readily available at Abattoirs and Fulani settlements.

The engineering properties of cowdung which included bulk density, angle of repose, co-efficient of friction and specific volume were used as design parameters for selection of different components that makes up the machine.

Results obtained included angle of repose (43^0), co-efficient of friction 0.93, Bulk density 801.9 kg/m^3, specific volume of 0.001246m^3/kg.

The machine was tested and the following results were obtained. work rate 9.6 tons/ha, field capacity 0.275 ha/hr, efficiency of collection as 84.5%.

The total length of the machine is 216cm, total height 110cm, transport width 214cm and working width of 66cm. The machine is operated from the tractor PTO drive of 540rpm and at a tractor speed of 1.505m/sec.

TABLE OF CONTENTS.

LIST OF FIGURES

LIST OF PLATES

LIST OF TABLES

LIST OF SYMBOLS

mm	millimeter
m	meter
Km	kilometer
Ha	Hectares
G	grammes
Kg	kilogrammes
ρ	Density
ρH	Bulk density (kg/m^3)
VH	specific volume
FH	co-efficient of friction
Ψ	Angle of repose
mc	moisture content (%)
P	power (kw)
rpm	revolutions per minute
Q	capacity (kg/m)
E	elastic modulus (GN/m^2)
I	moment of inertia (m^4)
π	Pi
Mt	maximum torque (Nm)
Mb	maximum bending moment (Nm)
S	allowable stress (MN/m2)
C	field capacity (ha/hr)
Pcr	critical load (N)
N	Newton
λ	Lambda

CHAPTER ONE

INTRODUCTION

Organic matter is by far the most important soil component. The content of organic matter, including humus, in arable layer of different soils varies widely. The organic portion of the soil represents a complex combination of various organic substances that can be divided into two groups namely: -

1) non humified organic substances of plants or animal origin.
2) Organic substances of a specific nature: humus or mull.

Organic matter is a major source of nutrients for plants. It contents almost all of soil nitrogen, a sizeable portion of phosphorus and sulphur, and small amounts of potassium, calcium, magnesium, and other nutrients.

Under constant cultivation, Nigerian soils are losing organic matter faster than it can be replaced. The total amounts on nutrients in the soil and the content of their available forms as well as the rate of nutrient transformation from the un-available to available state and vice-versa are the major factors of determining the nutrition, condition of plants and their requirement of fertilizers.

Organic and inorganic fertilizers enrich the soil with nitrogen and ash elements and also considerably intensify the mineralisation processes in it. Introduced with organic fertilizers are organic substances stimulating the activity of micro-organisms plus a highly diverse microflora (e.g. with manure) speeding up decomposition of the organic matter in the soil.

With the rapidly increasing cost of chemical fertilizers in Nigeria and new federal, state environmental laws has focused renewed interest in the old practice of adding animal manures to cultivated land. Given their low cost, simplicity of use and multiple soil benefits, the use of organics should be strongly encouraged especially for limited resources farmers. However, with current state of suitable organic technologies, there are many third world areas where it is still not possible to become totally reliant on organic fertilizers for all crops.

Waste disposal from cattle barns is among the most tiresome of drudgeries in any dairy operations. It is unthinkable even to consider using human labour for the handling of wastes from modern large cattle feedlots.

This is the reason, developing countries like Nigeria abandon the use of animal manure which is laying waste in many abattoirs in the major cities and Fulani settlements.

The selection of systems for the collection, storage and transport of cattle wastes is affected by the type of housing and by the concentration of animals. The concentration of animals determines the quantities of wastes which in turn determine the size and type of equipment to be used. The type of housing system determines if solid (mixture of excrements and bedding) or slurry handling equipment will be used.

The basic waste handling methods can be divided into stationary and mobile systems. Mobile systems have the advantage of servicing several barns and also of being used for other tasks while the stationary systems have advantages such as requiring little time to start, less maintenance and offering good possibility of automation and programming.

Local manure handling and application machinery must be developed in Nigeria. It is an effort to bridge this gap that the design and construction of a simple animal waste collector is being embarked upon.

1.1 Objective Of The Project

The following objectives are expected to be achieved with the design of the animal waste collector: -

a) To design a simple and affordable animal waste collector

b) To construct the machine using, locally available materials

c) To test the performance of the machine

1.2 Justification Of The Project

The rising need to be environmentally friendly and the increasing cost of chemical fertilizer has made it increasingly paramount to source for alternatives to inorganic fertilizers.

To justify, therefore, the development of a simple animal waste collector it should: -

a) Be easily affordable to peasant farmers

b) Reduce labour requirements involved in animal waste collection

c) Improve the sanitary conditions of our dairy farms and manure dumping sites such as abattoirs, Fulani settlements, cattle grazing ranches etc

CHAPTER TWO

2.0 LITERATURE REVIEW

2.1 Animal Manure

Organic manure is a well known source of supply of hums, playing an important part in the fertilization of the soil and in erosion control. There are four kinds: - green manure, compost, farmyard manure and slurry.

Farmyard manure is a bulky substance produced by animals and mixed with straw or other materials, the moisture content varies between 70 to 80%. Farmyard manure is very sensitive to climatic conditions and it may be possible that one third of nitrogen content is lost in 4 or 5 days after the manure has been spread on the field. Therefore, tillage has to be carried out within a short time after spreading.

The rapidly increasing cost of chemical fertilizer and new federal and state environmental laws have focused renewed interest in the centuries-old practice of adding animal manures to cultivated land. The most pragmatic means of disposing the massive quantity of manure is by additions to the soil, where its nutritive value can be utilized by growing plants. Cattle manure (dry basis) average 3 percent N, 0.8 percent P (1.8 percent P_{2O}), 2 percent K (2.4 percent K_2O), 25 percent organic carbon. plus varying amounts of other elements essential for plant growth. Although they contain a low percentage of nutrients, manures when added to soils in large quantities of many tons per hectare; supply considerable quantities of controlled release nitrogen, phosphorous and sulfur plus some chelates and micro nutrients. If widely used, manures are good source of fertilizer. (USDA, 1978)

2.2 Composition Of Animal Manures

Variations in composition of animal manures are the result of differences among kinds of animals and the kinds and amounts of feeds they consume. In comparison with chemical fertilizers, all manures supply relatively small quantities of plant nutrients per unit of dry weight. One comparison not usually made is the content of micronutrients in manures, which is usually higher in manures than in chemical fertilizers to which micro nutrient fertilizers have not been intentionally added.

Manufactured fertilizers have a high salt content (some are 100 percent soluble salt), as do manures (from 6 to 15 percent total soluble salt in dairy cattle manure). Salt (sodium chloride) is fed to most classes of livestock to increase appetite and to reduce kidney stones, much of it is voided in the manure.

2.3 Types And Physico-Mechanical Properties Of Organic Fertilizers.

Bosoi et al (1988) stated that manure density (ρh) and its specific volume (Vh) depend on the degree of its decomposition and moisture and are characterized by the following data: -

Manure:	Fresh	$\rho h T/m^3$	$Vh\ m^3/T$
	Loose	03 – 04	2.5 – 3.3
	Compact	0.5 – 0.7	1.4 – 2.0
	Semi-decomposed	0.7 – 0.8	1.2 - 1.4
	Humus	0.8	1.3

On an average, the co-efficient of friction of manure with steel, wood, and rubber is 0.9 – 1.0. Depending on the degree of manure decomposition, the angle of friction (repose) varies from $50^{o} - 38^{o}$.

Peat is used for bedding under cattle, for preparation of peat organic and mineral composts or directly as manure. The quantity of peat depends on its degree of decomposition; for top peat it varies between 20 and 40%. Peat density depends on its composition, degree of decomposition and moisture. Thus, density of top peat at a moisture of 40 – 86% varies between 0.30 and 0.98 kg/m^3 while the density of lowland peat at the same moisture is 0.27 – 1.02 kg/m^3.

2.4 Application Of Organic Manure

A good way to apply manure is to spread it evenly over the bed and work it thoroughly into the topsoil before planting or transplanting. Fresh manure should ideally be applied 1 – 2 weeks in advance of planting or transplanting and thoroughly mixed with the topsoil to avoid the possibility of plant injury. Fresh or composted manure applied at 6 – 12 litres/sq metre per crop planting should provide enough nutrients for good yields. This rate also supplies enough organic matter to achieve at least some improvement in soil physical condition. Meck et al (1974)

2.5 Environmentally Safe Applications

For centuries, manures have been applied purposely to soil to aid plant growth. The complications of modern developed civilizations are resulting in applications of manure to soil for a different reason just to get at rid of it. The question that needs to be answered is: how much can be applied to various soils without causing damage to plants growing in these soils, either from reduced yields or from toxic concentrations of various substances, most notably nitrates, salts and boron? Even if plants are not grown on soils that is used for manure or sewage sludge disposal, how much of the manures can microbes in the soil decompose, without large amounts of nutrients and salts leading into groundwater or surface water?

In spite of these potential hazards, relatively few cases of pollution occur. All the nitrogen in all manures is estimated to be only 2 percent of all current total nitrogen fertilizers used. Even if all sewage is included the total nitrogen is only about 4 percent of the amounts applied as manufactured fertilizer nutrients.

For greatest environmental integrity, manure should be spread daily on unfrozen or nearly flat land. Since this is frequently impossible, storage in lagoons (ponds) is common. The pond may become anaerobic when not artificially stirred but aeration reduces odour, hastens decomposition and reduces pathogens. Sutton (1974)

2.6 Importance Of Organic Manure

All the systems of organic manuring serve to put the soil in better shape for growth. The organic matter acts as a stabilizer of yield, protecting the plant against violent fluctuations of moisture content, structure or tilth, some degree against the effects of overdoses of inorganic fertilizers or changes in soil reaction. Dung is far from being a standard product, its composition varies widely according to the foods fed and management adopted, in fact any of the main constituents can vary in a ratio as wide as 1 : 2. under these circumstances average analytical figures are little more than a general guide. This raises the question of the effect of a dressing of farmyard manure on the returns to be expected from chemical fertilizers. It would be expected that farmyard manure would reduce the need for chemical fertilizers and this is certainly so in respect of potash; for dung is an excellent source of this constituent and it also probably holds true for phosphate.

According to Messimen (1992), there can be problems with over intensive use of artificial fertilizers. For example, on an intensive vegetal gardens growing a small range of plants, the use of only artificial fertilizers can result in soil sickness. This can result in the plant roots becoming less and less effective, as the fertilizer is increased in amounts.

The reasons for soil sickness are: -

a) Biological; with necroses and root galls being caused by fungi. nematodes, primary parasites and parasites which are usually unimportant but which become damaging due to their accumulation and the absence of organisms to compete and control them.

b) Rapid disappearance of organic matter, which the high rates of artificial fertilizers have helped to rapidly decompose.

c) Specific phenomena in certain soil types. Examples include excess acidification of ferrallitic soils caused by ammonium sulphates and low availability of phosphates in vertisols in which even soluble superphosphate are blocked by calcium ions.

2.7 Mechanisation Of Manure Handling

2.7.1 COLLECTION: Culpin (1975) stated that, until 1945 both mechanical loaders and spreaders were regarded as something of a luxury. Today, with moderately priced equipment available to increase a man's output about twenty fold while eliminating all the hard manual work, moving manure by hand fork should be regarded as a misuse of labour and unnecessary even on small farms. It does not follow, of course, that every small farmer can justify having his own loader and spreader.

Though there are other methods, such as use of a dozer blade on a tractor, the equipment now most widely used for handling farmyard manure is a hydraulic loader and a trailer type self emptying spreader. There are, however, many variations in types and capacities and there are circumstances where a hydraulic grab or rear loader rather than a front loader is chosen. Advantages of the more expensive hydraulic grab include less wear on the tractor clutch, gearbox and tyres and less cutting up of the ground surface.

Machines for the application of organic and organo-mineral fertilizers produced are intended for transportation and continous spreading for manure, peat, compost and other organic fertilizers.

Bosoi et al (1988) stated that the main types of spreaders are bunkers with a spreading device for fertilizers on the field. The hitch spreader main mechanisms consists of two spiral feed drums mounted on the body back rim, the transporter located on the body bottom with a reduction unit, a crank-and-connecting rod assembly together with a ratchet wheel gear.

The machine is driven from the power take off shaft through a cardon shaft and reduction unit. The lower spiral feed drum is put into a rotary motion by the reduction unit of the left shaft through a chain drive. The upper drum receives rotary motion from the lower drum shaft through a chain drive on the right side of the machine. Rotary motion is transmitted to the shaft of the transporter from the reduction unit right shaft through the crank and connecting rod assembly and the ratchet wheel gear. The speed of the transporter is determined on the basis of the fertilizer fed per second. The mass (kg) of fertilizer thrown on the field per second is:

	q	=	$U_{tr}Hly$
where	U_{tr}	=	speed of the transporter m/s
	H	=	fertilizer layer height m
	L	=	length of spiral feed drum m
	Y	=	volume mass of fertilizer kg/m^3

Vopyan (1982) stated that manure handling in developed countries is mechanized and in some instances automated in order to minimize human labour and drudgery. The basic waste handling methods can be divided into stationary and mobile systems. The most widespread stationary equipment for straw manure is the circulating endless chain cleaner equipped with paddles. Principally, it is a chain conveyor placed in the center or shallow concrete gutters, running the full length of the gutter behind the cattle stalls.

2.7.2 STATIONARY SOLID HANDLING SYSTEMS.

The most widespread stationary equipment for straw manure is the circulating endless chain cleaner equipped with paddles. Principally, it is a chain conveyor placed in the center of shallow concrete gutters, running the full length of the gutter, behind the cattle stalls. In one or two opposite corners there are gears with electric motors driving the chain. In the remaining corners or deflection points the chains are led by pulleys. At least one of them is sliding ad facilitates the tightening of the chains. The fall-through aperture is usually is usually situated in the corner near the drive unit. Manure falls through the aperture on to a conveyor or stacker which conveys the manure out of the building. The power requirements of such gutter cleaners are about 3h.p with a capacity of 3tons/ha. Velebil (1982)

The chain and padde cleaner is supplemented with a conveyor for loading manure on trailers or for stacking manure in layers on the farmsted manure heap. In large barns with minimal bedding, bars with flaps may be used to move manure out.

2.7.3 STATIONARY SLURRY HANDLING EQUIPMENT

The use of reciprocating barn cleaners to remove slurry from beneath slotted floors results in excessive corrosion and frequent malfunctioning. The so called towed cleaners are preferred. They consist of pushing bars or cables with hinged flaps on one or both sides. When in operation the flaps of the cleaner lean against the walls of the gutter, pushing the manure out.

The design of the pushing bar with flaps is simple and offers high operational reliability. Pushing bars with flaps have been founds satisfactory for gutters up to 1m in width. Improvements are being made which would permit the use of such cleaners in 3m wide gutters.

Pushing bars with flaps have been used in gutters under the slatted floors; economic advantages are proved when using them as surface units on dunging areas.

2.7.4 HYDRO MECHANICAL WASTE REMOVAL

Velebil (1982) stated that the first practical hydro mechanical systems of slurry removal have been the flushing systems. The excrements are helped up in the gutter for upto 3 days and are then flushed either by clean water or by recycling the supernatant from a sedimentation tank following the flushing operation.

All flushing systems require initial discharge of the head of the gutter of sufficient quantities of water to ensure an average velocity of at least 1m/sec in the channel. This is the minimum botton velocity necessary to scour manure solids on the floor. However, the flow in the channel should also be of sufficient depth, preferably 5 – 8 cm, so that the flow has the capacity to carry with it the waste load. A bottom velocity of 1m/sec is capable of moving gravel 30mm in diameter.

Flushing frequency is determined by the type of operation and by what is necessary to prevent high odour levels in the building. Flushing devices used in cattle operations are dosing siphons, tipping buckets or trap door tanks.

2.7.5 TRANSPORT OF WASTES

The method of transporting feedlot wastes from their point of generation or storage to disposal sites or to treatment units is determined by the solids content of the wastes. Generally, solid methods of transport are used for wastes with solid exceeding 20 percent.

a) Solid Waste Transport

The most common method of handling solid dairy manure containing straw is with the conventional type manure spreader. The usual capacity of large conventional spreaders is 10 – 15 m^3 or about 20 tons. They are either tractor drawn or mounted to lorries. Spreader mechanisms include paddles, flails and augers.

b) Liquid Manure Transport

Dairy manure usually contains sufficient quantities of fibrous materials which must be chopped before pumping in order to avoid pump clogging. Manure pumps are usually equipped with chopper attachments yet can maintain a pumping rate of 4 – 10 m^3/min when total dynamic head does not exceed 5 – 6m and manure slurries contain less than 20 percent total solids. These pumps are also equipped with a movable agitation pipe which can be swiveled in any desired direction so as to agitate the tank contents. The type of pumps used are mainly PTO driven submerged centrifugal pumps. Piston, helical rotor, and diaphragm pumps can also be used provided the solid contents are chopped or the solid contents is low.

Tank wagons for the transport of cattle slurries may vary in size from $2m^3$ to $40m^3$. the latter are used in beef cattle feedlots of 10,000 cattle and more capacity for dairy cattle operations, the size is in the range of 4 – 10 m^3. although vacuum – type wagons have been used for cattle wastes, the most commonly used tank wagons are pump loaded and gravity unloading. Some tank wagons operate agitating during transport for uniform field operation.

2.8 Soil Fertility

The total amount of nutrients in the soil and the content of their available forms as well as the rate of nutrient transformation from the unavailable to available state and vice-versa are the major factors determining the nutrition conditions of plants and their requirements for fertilizers. Once applied, fertilizers undergo diverse transformations in the soil with the result that the solubility of the nutrients they contain changes along with their mobility in the soil and availability to plants. Vopyan (1982)

Organic matter is a major source of nutrients for plants. It contains almost all of soil nitrogen in a sizeable portion and phosphorus, sulphur. small amounts of potassium, calcium, magnesium and other nutrients. Organic substances are involved in adsorption processes, improving soil structure, its moisture capacity, water and air permeability and thermal regime.

Early agriculturalists and present 'organic farmers' regard the return of organic matter to soil as a basis of permanent soil fertility. Conventional farmers also consider the use of organic residues as wise management. Both groups recognize the enhancement of soil as growth medium when increased organic levels are maintained by regular moderate additions of crop residues and other suitable organic materials. Organic farmers emphasize the value of 'natural' and 'organic' materials and warn against the use of systhetics or chemicals produced by modern science. Oclhof, (1978)

Urea can be made commercially but the same urea exists naturally in animal excreta and is recommended for use. The urea from either source will be changed by the soil enzyme urease to NH_4^+ and can then be absorbed by plants. In either case, the organic nitrogen form excreta or the chemical nitrogen both end up as inorganic NH_4^+ before the plants absorb them.

Perhaps the major advantages of soil organic matter as a nutrient source are its conservation of nutrients against leaching losses and its continual release of nutrients. Because only small portions of organic matter decompose in a few days, heavy rainfall or irrigation is able to leach small amounts of solubilized nutrients.

2.9 Possible Advantages Of Organic Fertilizers

d) Organic like compost and manure are generally free or very low cost for most farmers.

e) Organic fertilizers take relatively little skill to use properly

f) Plant or animal derived organics like manure not only supply plant nutrients but also organic matter which improves soil physical condition, stimulates beneficial soil micro-organisms.

g) Much of the nitrogen and phosphorous in organics is in a slow-release, organic form. This is a plus for nitrogen which is susceptible to leaching when supplied by chemical fertilizers.

h) The phosphorous in organic fertilizers is less prone to soil tie-up than that from chemical fertilizers, making it more available to plants.

Possible disadvantages of organic fertilizers include most plant derived organics like compost and manures are low-strength fertilizers; this means very large amounts must be applied to supply enough nutrients for crop growth and to add enough humus to benefit soil physical condition.

The exact nutrient content of most organic like compost or manure varies a lot and also takes a good deal of labour to apply most organics or to make compost because of the large amounts needed. Leonard (1986)

Overuse of chemical nitrogen fertilizer over the years have been linked to nitrate contamination of rivers, lakes and wells due to leaching and runoff of the excess nitrogen. Excessive nitrate levels in leaf vegetables like spinach, tomatoes are converted to nitrites which are toxic in themselves but can also further be changed into nitrosamines which are strongly linked to stomach cancer. Compost and well rotted manure release their nitrogen slowly enough to avoid this problem.

Leonard (1986) stated that analysis has shown that overly high rates of chemical nitrogen fertilizers markedly lower the vitamin C content of leafy vegetables like lettuce and Chinese cabbage. Heavy applications of compost and well-rotted manure don't have this effect because they release nitrogen slowly.

Given their low cost, simplicity of use and multiple soil benefits, the use of organic manure should be strongly encouraged especially for limited resource farmers. However, since many small farmers are not likely to have enough organic fertilizer to cover all their land, they are better off using what is available on their smaller plots which are usually used for vegetables. This will enable them to apply a high enough rate to supply a beneficial amount of nutrients and organic matter. If enough is available, it can also be applied to the larger fields which are typically devoted to staple cereals and pulses such as maize, sorghum and cowpeas.

Messimen (1992) revealed that there are protestations from ecologists over the heavy dosage of chemical fertilizers:

a) Nitrate pollution of the ground water

b) Wastage of energy for industrial nitrogen fixation

c) The solubilisation of phosphates

d) Purifying of potassium salts.

The intensive use of chemical pesticides may appear even more disturbing than that of artificial fertilizers. The most important danger is in the effects that they have on the organisms or biological processes that are not their main target.

Various soils differ in composition of the inorganic portion, content and composition of the organic matter. Therefore, the content of basic nutrients in various soils differ too.

The total content of nitrogen in soils is in direct proportion to the amount of humus; there is also more phosphorous in soils if they are rich in organic matter, while the potassium content is determined primarily by the texture of the inorganic portion of the soil. In most soils, the overall content of nitrogen, phosphorous and potassium is rather high, tens and hundreds of times exceeding the amount of these nutrients removed by a crop. However, the major part of nutrients is present in the soil in the form of compounds completely or almost unavailable to plants. Nitrogen is present mainly in the form of complex organic compounds (humus substances, proteins etc.) most of phosphorous forms part of difficulty soluble inorganic compounds and organic substances and most of potassium is found in insoluble aluminosilicates.

The overall content of nutrients in the soil merely characterizes its potential fertility. To evaluate the effective fertility, that is the actual capacity of the soil to ensure high yields of farm crops, it is extremely important to take into account the content of available nutrients. Plants can take up the nutrients that are present in the soil in the form of compounds soluble in water and weak acids as well as in the exchangeable state. The mobilization of nutrients in various soils proceeds at different rates, depending on their nature, climatic conditions, soil properties and the agrotechnical level. If no fertilizers are applied, the soil often lacks the available forms of nutrients, necessary for high yields, that emerge in it during the vegetation period. Therefore, to enhance the effective fertility of soils and crop yields, application of organic and inorganic fertilizers is of paramount importance.

The content of available forms of nutrients as a function of type of soil, its cultural state and other factors, may differ not only from one farm to another, but also from field to field at the same farm. This is why agro chemical soil analysis carried out to determine the content of mobile forms of nitrogen, phosphorous and potassium in combination with field tests are extremely important for correct application of fertilizers.

2.10 Engineering Properties Of Manure

The following properties of manure were considered as they are necessary for the design of the manure collector.

a) Angle of repose

Mohsenin (1978) defined angle of repose as the angle with the horizontal at which granular material piled. This angle is influenced by size, shape, moisture content and orientation of particles.

Bosoi et al (1988) stated that the angle of repose for manure ranges between 38^{o} to 50^{o} depending on the degree of decomposition.

b) Co-efficient of friction.

Co-efficient of friction of a material as given by Mohsenin (1978) is equal to the tangent of the angle of repose and according to Bosoi et al (1988) the co-efficient of friction of manure ranges from 0.9 to 1.0.

c) Bulk density.

Bulk density of a material is the total weight of that material including the weight of air per unit volume (Sidney 1963).

The bulk density of manure ranges between $0.0013kg/m^3$ to $800kg/m^3$ depending on the moisture content. (Bosoi et al, 1988)

d) Moisture content.

The moisture content of a material is defined as the ratio of the weight of water to the weight of the dry solid in the same volume expressed in percentage basis (Sidney, 1963).

a) To obtain the angle of repose for steel, Mohsenin (1978) method was adopted.

Materials: Manure sample, tilting table, steel plate protractor, compass.

Method: A sample of the material (manure) was placed in a heap on a tilting table of a steel plate. This table was then tilted until the material starts to glide, the angle of tilt is then measured. This process is then repeated to obtain a number of readings.

Result: Average angle of repose obtained = 43^{o}

b) Co-efficient of friction is given as the tangent of the angle of repose of the material. For this material (manure) the angle of repose (ψ) is 43^{o}.

$$\text{Co-efficient of friction } (\rho H) = \tan 43$$
$$= 0.93$$

Angle of repose = 43^{o} and co-efficient of friction = 0.93 both of these results are within the range recommended.

c) Bulk density: Using a platform scale method the bulk density of the material (manure) was determined as presented by Mohsenin (1978)

$$\text{Bulk density}_{(PH)} = {}^{W}/_{V} \qquad \text{where } w = \text{weight}$$
$$V = \text{volume}$$

Readings recorded shows that

$$\text{Bulk density}_{(PH)} = \frac{0.5046}{6.292 \times 10^{-4}}$$

$$\rho H = 801.970 kg/m^{3}.$$

$$\text{Specific volume } V_{H} = \frac{6.292 \times 10^{-4}}{0.5046} = 0.001246 m^{3}/kg$$

$$\text{Moisture content } (MC\%) = \frac{\text{weight of water}}{\text{Weight of dry sample}} \times 100\%$$

Weight of sample (moist) = 0.500kg

Weight of oven dry sample = 0360kg

Weight of water removed = 0.140kg

$$\therefore MC\% = \frac{Ww}{Ws} \times 100$$

$$= \frac{0.140}{0.360} \times 100 = 38\%.$$

2.11 Animal Waste Collector

A variety of machines are available for both the loading of manure and the spreading of it. Generally, loading is carried out by the use of a fore-end loader which is hydraulically operated, but other methods of loading can be used. For example, engine-driven elevators used in conjunction with a mechanically moved fork, and there is also a rear mounted hydraulically operated loader. In most modern farms, the collection of animal waste (especially from dairy farms) has been fully automated using chains and slats to scrap these waste. Shippen et al (1973).

In developing countries, such as Nigeria there is a gap between the type of housing used for our animals and the technology of manure collection; the idea to develop the animal waste collector is to fill this gap. From Appendix A, there is a good distribution of animals in Nigeria that makes the development of this machine a worthwhile undertaking.

CHAPTER THREE

3.0 METHODOLOGY

3.1 Design Consideration.

The main factor that was considered in the adoption of the present concept is the properties of manure. It was revealed that manure could best be handled when the decomposing process must have commenced. Under this condition, the manure can easily flow under gravity. The following points were kept in mind while designing the machine:

a) The machine should match the power available to the farmer.

b) It should be simple to operate and maintain even by the farmers who do not have formal education.

c) It should be meant for farmers who finds it difficult to meet the cost of chemical nitrogenous fertilizer.

d) It should be able to reduce the labour requirement and drudgery involved in traditional method of manure collection.

3.2 Material Selection

The strength and durability of farm equipment or machines depend largely upon the kind and quality of materials used in building it.

According to Bosoi et al (1988) the choice of materials for a machine determines its reliability which is the property of a machine to fulfill the given function, preserving its operation indices within given limits during the required time interval.

Basic engineering design considerations such as power requirements, sweeper speed, machine capacity, the physical and engineering properties of manure were combined with the cost of the machine to achieve the desired objective of the development.

The animal waste collector has the following components such as power transmission unit, a sweeping unit, a conveying unit, a hopper a towing bar and transport wheels.

The materials selected for the construction of the machine are mild steel, rubber, cast iron and fibre brush.

3.3 Bevel Gear Selection

Bevel gears are the most common means of transmitting power and motion between intersecting shafts.

In solving any gear design problem, the usual practice is to make a rough preliminary draft design before proceeding to finalise the design data consisting of the finer aspects of the art of gear design. The design involves such considerations as the types and magnitudes of stresses which the gear is likely to be subjected to, an estimate of the approximate gear size keeping the space and weight restrictions in mind, the power rating to meet the requirements and other operational parameters.

3.3.1 Material

Selection of proper material depend mainly on the requirements of space and weight and overall price of the gear drive. Cast iron is good enough for ordinary purposes but steel offer better strength. Cast iron is chosen.

3.3.2 Number Of Teeth.

The number of teeth of the pinion and the gear are to be so chosen that a minimum value of 1.1 for contact ratio is assured. Guidelines for the selection of minimum number of teeth of pinion are given below: -

Table 3.1 Minimum number of pinion teeth.

Type of service	Minimum number of teeth
Heavy duty and high speed	16
Medium speed	12
Light duty and low speed	10

3.3.3 Transmission Ratio

If the transmission ratio is high, a multi stage gear set is used to avoid unnecessarily big gears.

In general, the transmission ratio per stage is given by:

i	$\leq$	7	for general purpose drive
	=	10	for maximum value in special cases
	=	4	for maximum value for change gear sets.

Transmission ratio selected = 4.

Design Requirements

a) Speed ratio = 3.1:1

b) Speed of driver gear = 540rpm

c) Horse power transmitted = 29.1kw

d) Gear material is cast steel having minimum tensile strength of 55 kg/cm^2

e) Tensile strength for steel is 20,000 kg/cm^2

f) Modulus of elasticity of steel is 2.06×10^6 kg/cm^2

g) Ratio of cone distance to face width = 3

h) Designed tensile strength for cast steel is 5500kg/cm^2

i) Gear module = 8

j) Torque transmitted $= \dfrac{kw \times 9550}{rpm}$

$= 514.64Nm.$

3.3.4 Pitch Circle Diameter (PCD)

The final value of pitch circle diameter will depend on many different factors. Only broad outlines can be given for initial calculations.

Pitch Circle Diameter = $d_1 \geq 1.5$ x shaft diameter of pinion shaft i.e. when the Pinion and the shaft are made from one stock.

Shaft diameter = $d(mm) = 160 \sqrt[3]{\frac{P_1(kw)}{n_1(rpm)}}$ .... (3.1)

$= 160 \times 0.3777$

$= 60.434$

$= 60.4$ mm.

$\therefore$ PCD $= d_1 \geq 1.5 \times 60.0$ mm

≥ 90.65 mm

3.3.5 Module

Small modules permit noiseless running. The limiting factors comprise strength of tooth root, quality of bearing and manufacturing constraints and the PCD and number of teeth.

PCD = $d_1 = z_1 m$ .... (3.2)

$m = \frac{d_1}{z_1}$ .... (3.3)

where d_1 = pitch center diameter

z_1 = number of teeth of pinion

m = module.

Given that $d_1 = 90.65$mm and $z_1 = 12$

$m = 90.65/12 = 7.554 \simeq 8.$

$m = 8mm$

$PCD = zm = 12 \times 8 = 96mm$

Pitch cone angle (shaft angle is equal to 90^o)

$$\text{Tan } \delta_1 = \frac{d_1}{d_2} \qquad \ldots\ldots\ldots\ldots (3.4)$$

Given $d_1 = 96mm$, $d_2 = 384mm$

$$\text{Tan } \delta_1 = \frac{96}{284} = 0.25$$

$$\delta_1 = 14^o$$

Tip circle diameter $= d_{tc} = d_1 + 2m \cos \delta_1$ .... (3.5)

$d_1 = 96mm$, $m = 8mm$, $\delta_1 = 14^o$

$$d_{tc} = 96 + 2 \times 8 \times \cos 14$$

$$= 96 + 15.526$$

$$= 111.52mm$$

cone distance $= R = \dfrac{d_1}{2 \sin \delta_1}$ .... (3.6)

$d_1 = 96mm$, $\delta_1 = 14^o$

$$R = \frac{96}{2 \sin 14} = \frac{96}{0.4838}$$

$$= 198.42mm$$

face width $= b_{max} = {}^{R}/_{3}$ ....(3.7)

$$= \frac{198.42}{3} = 66.14mm$$

whole depth $= h = 2m + 0.2m$ .... (3.8)

$= 2.2m$

$m = 8mm$

$= 2.2 \times 8 = 17.6mm$

Addendum $= h_{ad1} = h_{ad2} = m$ .... (3.9)

$m = 8mm$

Dedendum $= h_{dd1} = h_{dd2} = (1.2)m$ .... (3.10)

$= \quad 1.2 \times 8 = 9.6mm.$

Addendum angle $= \quad \tan\theta_{ha} \quad = \quad m/R$ .... (3.11)

$m \quad = \quad 8mm, R = 198.42mm$

$$\tan\theta_{ha} \quad \frac{8}{198.42} \quad = \quad 0.0403$$

$$\theta_{ha} \quad = \quad 2^{o}.30'$$

Dedendum angle $= \quad \tan\theta_{hd} \quad = \quad 1.2\ m/R$ .... (3.12)

$$= \quad 1.2 \times \frac{8}{198.42}$$

$$= \quad 0.04838$$

$$= \quad 2^{o}.76' \quad \simeq \quad 3^{o}$$

Crown height $= \quad CH \quad \frac{d_1}{2} - m \sin\delta_1$ .... (3.13)

$d_1 = 96mm, m = 8mm, \delta_1 = 14^{o}$

$$CH \quad = \quad \frac{96}{2} - 8 \times \sin 14^{o}$$

$$= \quad 48 - 1.935$$

$$= \quad 46.065mm$$

Back cone distance $= R_{cd} = R \tan\delta_1$ .... (3.14)

$R = 198.42, \delta_1 = 14^{o}$

$$R_{cd} \quad = \quad 198.42 \tan 14^{o}$$

$$= \quad 49.47mm$$

3.3.6 Force Analysis For Bevel Gears

For force analysis of a pair of mating bevel gears, it is assumed that the total force (fN) acts on the pitch point P at the middle of the tooth width.

The mean tooth force (fN) is resolved into three mutually perpendicular components – the tangential force or transmitted load ft, the radial force fr and the axial force, fa.

Tangential force = Fn $\cos\alpha$ .... (3.15)

This force is calculated from the torque and is given by

$$Ft = \frac{2T_1}{dm_1} = \frac{T_1}{rm_1} \quad \text{.......................... (3.16)}$$

where T_1 = pinion torque in Nm = $9550\ P_1/N_1$..... (3.17)

P_1 = pinion power in kw

N_1 = pinion speed in rpm

Radial force $Fr_1 = F_t \tan\alpha \cos\delta_1$ .... (3.18)

$Fr2 = F_t \tan\alpha \cos\delta_2$ .... (3.19)

Axial force $Fa_1 = F_t \tan\alpha \sin\delta_1$ .... (3.20)

$Fa_2 = F_t \tan\alpha \sin\delta_2$ .... (3.21)

If the shaft angle is 90^o

Then $Fa_1 = Fr_2$ .... (3.22)

$Fa_2 = Fr_1$ .... (3.23)

3.3.7 Bevel Gear Bearing Loads

For the determination of bearing loads in case of bevel gears in mesh, it is convenient to analyse first the forces acting n the bearings in two mutually perpendicular planes and then add the partial bearing loads vectorially to arrive at the resultant load on each bearing.

Three forces which act on a bevel gear tooth are tangential force F_t, radial force f_r, and axial force F_a.

We have in X – Z plane.

$F_{BLX}L = F_t (L_1 + L)$ whence $F_{BLX} = F_t\ ^{(L1 + L)}/_L$ .... (3.24)

$F_{BLX} = F_t^{L1}/_L$ .... (3.25)

Similarly in Y – Z plane, we have

$$F_{BLY} = \frac{F_r(L_1 + L) - F_a rm}{L} \qquad \ldots\ldots\ldots\ldots (3.26)$$

$$F_{BLY} = \frac{F_r L_1 - F_a rm}{L} \qquad \ldots\ldots\ldots\ldots (3.27)$$

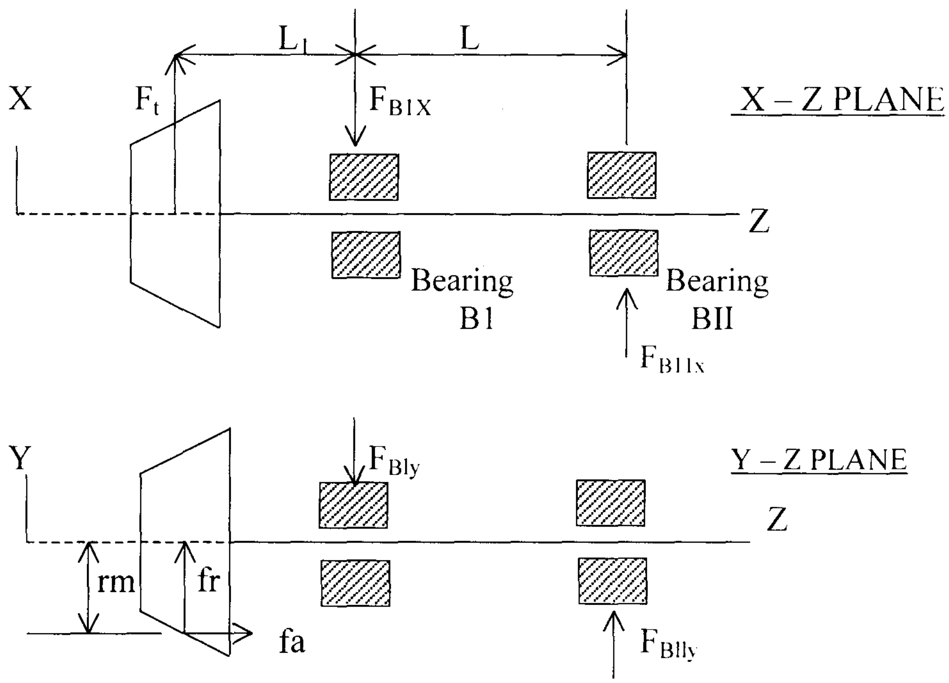

Fig 3.1 Bevel gear bearing loads.

The resultant values of the bearing loads on bearings BI and BII are:

$$F_{BI} = \sqrt{(F_{BLX})^2 + (F_{BLY})^2} \qquad \ldots\ldots\ldots\ldots (3.28)$$

$$F_{BII} = \sqrt{(F_{BIIX})^2 + (F_{BIIY})^2} \qquad \ldots\ldots\ldots\ldots (3.29)$$

3.3.8 Bending Moments

The shaft will be subjected to bending besides torsion and the axial thrust. F_a by which the bearing will be loaded. Normally, F_a, can be neglected without any significant effect in final calculation. The bending moments (B) are calculated for the points P and mid point of BL.

They are given by

$$B_P = F_a rm \quad \ldots\ldots\ldots\ldots. (3.30)$$

$$B_{BL} = F_{BII}L \quad \ldots\ldots\ldots\ldots. (3.31)$$

Given that p = 29.10kw, n = 12, m = 8mm

$\Sigma = 90^o$, $a = 20^o$ $\delta_I = 14^o$, distance between bearings L = 100mm

overhung distance = 50mm.

To find bearing loads and bending moments

$$T = 9550\,{}^{P}/_{n} = 9550 \text{ X } \frac{29.10}{540} = 514.6\text{Nmm}$$

$$d = mz = 8 \text{ x } 12 = 96\text{mm}$$

$$R = {}^{d}/_{2\sin\delta} = 198.42\text{mm}$$

$$b = {}^{R}/_{3} = 66.14\text{mm}$$

$$dm = d - b \sin \delta_I = 80\text{mm}$$

$$F_t = {}^{T}/_{rm} = \frac{514.6 \text{ x } 1000}{31} = 16600\text{N}$$

$$F_r = F_t \tan 20^o \cos 14^o = 5861.3\text{N}$$

$$F_a = F_t \tan 20^o \sin 14^o = 1461.25\text{N}$$

$$F_{BIx} = \frac{16600(50 + 100)}{100} = 24900\text{N}$$

$$F_{BIIx} = \frac{16600 \text{ x } 50}{100} = 8300\text{N}$$

$$F_{BIy} = \frac{5861.0\,(50 + 100) - 1461 \text{ x } 31}{100} = 8338.59\text{N}$$

$$F_{BIIy} = \frac{5861 \text{ x } 50 - 1461 \text{ x } 31}{100} = 2477.59\text{N}$$

using Eqs 3.28 and 3.29

$$F_{BI} = \sqrt{(24900)^2 + (8338.59)^2}$$
$$= 26259\text{N}$$

$$F_{BII} = \sqrt{(8300)^2 + (2477.59)^2}$$

$$= 8661.89N = 8662N.$$

Bending moments at P and for the mid-point BI at

$$P = Bp = F_a rm \quad \ldots\ldots\ldots\ldots\ldots(3.32)$$

$$= 1461 \times 31/1000$$

$$= 45.29Nm.$$

$$\text{at mid BI} = B_{BL} = F_{BII}L \quad \ldots\ldots\ldots\ldots\ (3.33)$$

$$= 8662 \times 100/1000$$

$$= 966.2\ Nm.$$

To determine the output shaft diameter the formula below is used.

$$d^3 = {}^{16}/_{\pi SS} \sqrt{(k_m M)^2 + (k_r T)^2} \ldots\ldots\ldots\ldots\ (3.34)$$

where d = diameter of shaft (mm)

k_m = combine shock and fatique factor applied to bending moment

k_T = combine shock and fatique factor applied to torsional moment

M = maximum bending moment

T =. Maximu torsional moment

SS = allowable stress for steel = $40MN/m^2$

$$\therefore d^3 = {}^{16}/_{3.142 \times 40 \times 10^6} \sqrt{(1.5 \times 514.639)^2 + (1. \times 866.2)^2}$$

$$= 0.000000127 \sqrt{292919.92 + 750302.44}$$

$$= 0.000000127 \text{ X } 1160.268$$

$$d = \sqrt[3]{0.000147354}$$

$$= 0.0528m = 52.8mm.$$

3.4 Belt Selection

A V-belt consists of an endless flexible belt that transmits power by contacting and gripping the sheaves which are keyed to the shafts of the driving and driven mechanisms. The primary reasons for using a V – belt drive are its simplicity and low maintenance costs. In addition the elastometric V – belt is able to absorb moderately high shock loads and miligate the effects of vibrating forces. Krutz et al, (1984)

Because of its obvious advantages, V – belt was selected for power transmission in the machine under design. The following method was used for belt selection.

a) Determining the designed horse power hp (kw)

V – belt manufacturers publish service factors for belts used in various applications. Table below shows service factors for V – belt drives.

Table 3.2: Service Factors For V – belt drives.

Typical machines	Type of service	Service factor
Domestic washing machines, ironers, small fans and blowers etc.	Light	1.0 – 1.2
Fans and blowers (heavy rotors) centrifugal pumps, workshop machines	Medium	1.2 – 1.4
Reciprocating pumps and compressors, drill presses and grinders, machines for industrial use	Heavy	1.4 – 1.6

Design horsepower for the belt is found by taking the product of the rated horsepower of the device driven by the belt and the service factor

Power from tractor	=	29.1kw
Service factor	=	1.4
Designed power	=	29.1 x 14
	=	40.74kw

b) Determining the belt speed

The belt speed of agricultural V – belt drives ranges from 0.5m/s to more than 30m/s. generally, the drive is designed so that the belt speed is in the range of 5m/s to 20m/s.

The expression belt speed is

$$V = \frac{rpm \times PD}{19100} \qquad \ldots\ldots\ldots\ldots\ldots (3.35)$$

Where V = belt speed (m./s)

rpm = shaft speed (rpm)

PD = pitch diameter of sheave (mm)

With shaft speed at 1710 rpm and pitch diameter of sheave 320 mm

$$\therefore V = \frac{1710 \times 320}{19100} = 28.6$$

$$= 29 \text{ m/sec.}$$

b) Determining the length of the belt

The length of an open belt drive is calculated by

$$L = 2C + \pi \frac{(PD_1 + PD_2)}{2} + \frac{(PD_1 + PD_2)^2}{2}$$

where L = length of belt (mm)

PD_1 = pitch diameter of larger sheave (mm)

PD_2 = pitch diameter of smaller sheave (mm)

C = center to center distance between the two

Shafts (mm)

With $C = 680mm$, $PD_1 = 30mm$ and $PD_2 = 250mm$

$$L = 2 \times 680 + 3.142 \frac{(320 + 250)}{2} + \frac{(320 - 250)^2}{4 \times 680}$$

$$= 1360 + 895.35 + 1.801$$

$$= 2257.15mm$$

the length of closed belt drive is calculated by

$$L = 2C + \frac{\pi}{2}(PD_1 + PD_2) + \frac{(PD_1 + PD_2)^2}{4C}$$

with $C = 520mm$, $PD_1 = 30mm$, $PD = 150mm$

$$L = 2 X 520 + 1.57 (320 + 105) + \frac{(320 + 150)^2}{4 \times 520}$$

$$= 1040 + 737.9 + 106.20$$

$$= 1884.1mm.$$

d) Determining the arc of contact

the arc of contact is given by

$$B^o = 180 - 60 \frac{(d_1 - d_2)}{C} \qquad \text{...........} (3.37)$$

Where B^o = arc of contact (degrees)

d_1 = large pitch diameter (mm)

d_2 = small pitch diameter (mm)

C = distance between shaft centers (mm)

$$\therefore B^o = 180 - \frac{60 (320 - 150)}{520}$$

$$= 180 - 19.615$$

$$= 160.38^o$$

From Appendix I, the calculated length of belt (2257.15mm) is nearest to the standard belt recommended 2266mm for open belt drive. For closed belt drive, the calculated length of 1884.1mm is nearest to 2012mm.

Therefore, 2266mm and 2012mm are chosen for a HC cross section.

3.5 Sweeping Mechanism Shaft Design

In the design of the sweeping mechanism, several options were considered. They range from fibre brush by belt drive to wire brush by chain and sprocket. Since a sweeping action is desired with minimum lifting of the topsoil; fibre brush was selected to be mounted on flat steel bars. The total mass of the sweeper unit is found to be 18.98kg that is a weight of 186.19N.

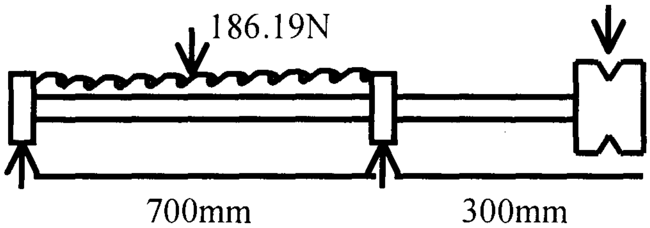

Fig 3.2: sweeping mechanism bearing design.

To determine the radial load acting on a shaft, the power transmitting by the drive is:

$$F = \frac{kw \times 19.1 \times 10^6 \times k}{PD \times RPM} \quad \text{.............. (3.38)}$$

Where F = radial force on the shaft, N

Kw = power transmitted, kw

PD = pitch diameter of sheave, mm

rpm = speed of shaft, rpm

k = drive tension factor; 1 = chain drive and 1.5 for belts.

The radial force on the shaft from the V – belt is

$$F_{vbelt} = \frac{29.1 \times 19.1 \times 10^6 \times 1.5}{250 \times 346}$$

$$= 9638.3N.$$

load from the V – belt at bearing B

$$F_B = \frac{9638.3 \times 300}{700} = 430.7N$$

load from the V – belt pull at bearing A

$$F_A = \frac{9638.3 \times 1000}{700} = 13769\ N$$

load from the sweeper unit at bearing A

$$F_{SA} = \frac{186.19 \times 350 \times 700}{350} = 130333N$$

load from the sweeper unit at bearing B

$$F_{SB} = \frac{186.19 \times 350 \times 700}{300} = 152055\ N$$

combining the two loads at each point =

$$F_{SA} - F_A = 130333 - 13769$$

$$= 116546N$$

$$F_{SB} + F_B = 152055 + 4130.7$$

$$= 156185.7\ N$$

Therefore ,the load at Bearing B would be used to select the appropriate –sized bearing because it is greater than the load at bearing A. From Appendix F ,Bearing of basic design no SKF/FAG 6234 was selected.

Determination of Reaction at A and B.

Taking moments about A

$\Sigma M_{RA=} 0;$

RAXO+ 186.19X .7X..35-4130.7X..7+9638.3X1

$= 45.59+9638.3-2891$

$= 6792.89N$

Summation of vertical forces

$\sum_{fy}=0,$

$RA + RB - W_1 - W_2 = 0$

$RA + RB\ W_1 + W_2$

$RB = W_1 + w_2 - RA$

$=180.19+9638.3 - 6792.89$

$= 3031.6N.$

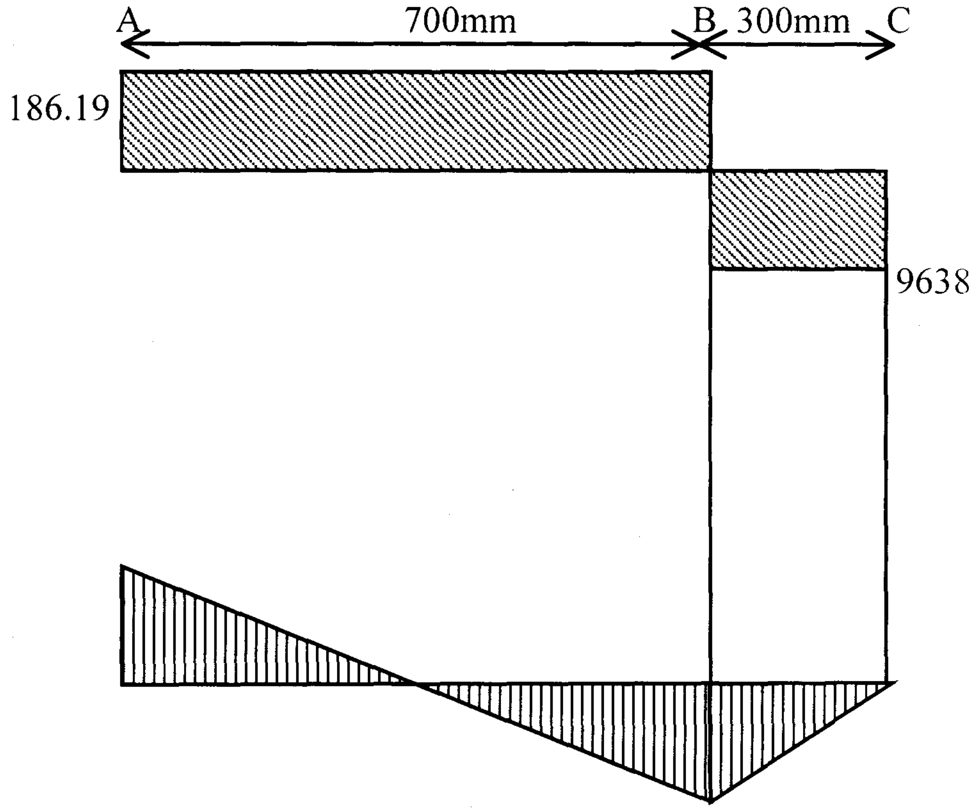

Fig 3.3: Shearing force and Bearing moment diagrams.

3.6 Flight Conveyors

Flights conveyors are used for moving granular, lumpy or pulverized materials along a horizontal path or an incline. A conveyor steeply inclined should have closely spaced flights so that the material will not avalanche over the tops of the flights. The capacity of a given conveyor diminishes as the angle of slope increases.

Spirakovsy (1985) said the type of pick-up member is the principal element of these conveyors and depends on the properties of the transported materials (lump size, abraisiveness etc.) and the required through put capacity of the machine.

Table 3.3 Capacity and size of lumps of flight conveyors.

Flight width and depth	Quantity of material	Approx capacity at 0.5m/s with 800kg/m^{3}metri
Mm	M^3/min	Tonne/ha.
300 x 150	0.037	54
380 x 150	0.06	66
460 x 150	0.052	76
610 x 200	0.108	158
76 x 250	0.149	218
910 x 300	0.223	327

Spivakovsky et al (1985)

The capacity of an inclined conveyor is found by multiplying the capacities in table 3.3 by the following factors.

Angle of inclination (deg)	20	25	30	35	and above
Factors	0.9	0.8	0.7	0.6	

Determining the load rating for the two bearings supporting the shaft of the conveyor.

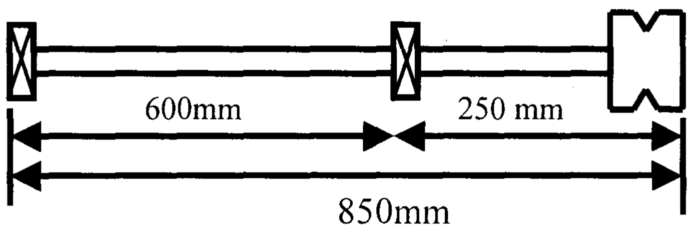

Fig. 3.4 Bearing Design for conveyor

The radial force on the shaft from the belt drive is calculated using equation 3.38

$$F_V \text{ belt} = \frac{29.10 \times 19.1 \times 10^6 \times 1.5}{270 \times 600}$$

$$= 5146.38\text{N}$$

At bearing A

Load from V – belt pull

$$= F_A = \frac{F_{belt} \times b}{a} = \frac{516.38 \times 250}{500}$$

$$= 2144.32\text{N}$$

At bearing B

Load form V – belt pull

$$= F_B = \frac{F_{belt} \times C}{a} = \frac{5146.38 \times 850}{600}$$

$$= 7290.70\text{N}$$

The loading at bearing B which is greater will be used to determine the size of bearings.

3.7 The Hopper

The hopper is the container that holds whatever is swept and conveyed. This hopper is constructed using a 2mm (gauge 14) thick metal sheet. It is made up of rectangular and trapezoid shapes joined together by welding.

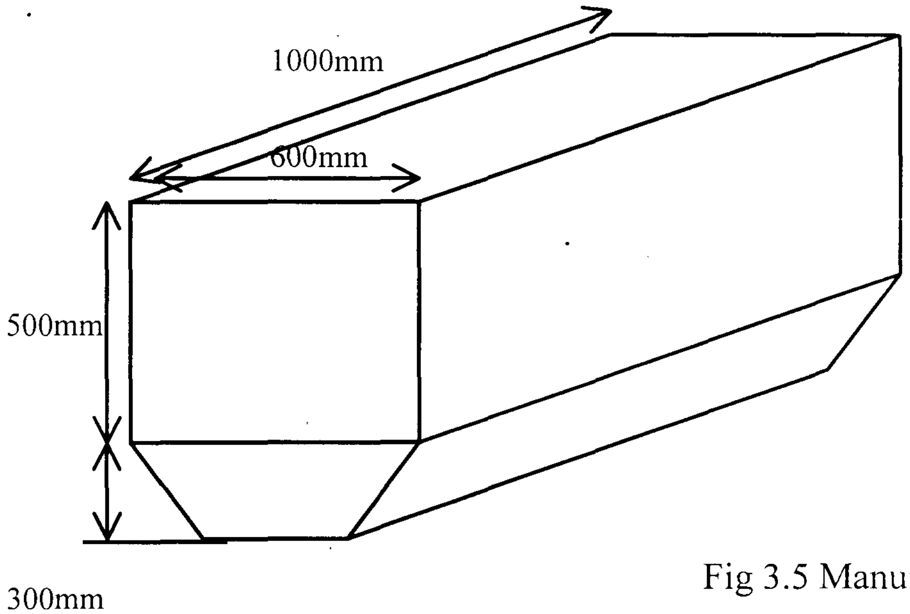

Fig 3.5 Manure collection hopper

	Thickness of metal sheet	=	2mm (gauge14)
	Density of mild steel	=	$7850 kg/m^3$
a)	Area of the rectangular box	=	length x breadth.. (3.43)
		=	1000 x 600
		=	$600000 mm^2$
		=	$0.6 m^2$

Volume of material for the rectangular box

= Area x thickness ....(3.44)

= 0.6 x 0.002

= $0.0012 m^3$

Mass of the material = volume x density ......(3.45)

= $0.0012 m^3 \times 7850 kg/m^3$

= 9.42kg

b) for the trapezoided shape

Area of side A	=	$\frac{1}{2}(a + b)h$	.... (3.46)
	=	$\frac{1}{2}(1000 + 600)\,300$	
	=	$240000mm^2$	
Both sides	=	$0.24m^2 \times 2 = 0.48m^2$	
Volume of material	=	Area x thickness	
	=	0.48×0.002	
	=	$0.00096m^3$	
mass of material	=	$0.00096 \times 7850kg/m^3$	
Area of side B	=	$\frac{1}{2}(a + b)h$	
	=	$\frac{1}{2}(600 + 200)\,300$	
	=	$120000mm^2$	= $0.12m^2$
Both sides	=	$0.12m^2 \times 2 = 0.24m^2$	
Volume material	=	Area x thickness	
	=	$0.24m^2 \times 0.002m$	
	=	$0.00048m^3$	
Mass of material	=	$0.00048m^3 \times 7850kg/m^3$	
	=	3.768kg	

c)

Area of hopper bottom	=	length x breadth
	=	600mm x 200mm
	=	120000mm
	=	$0.12m^2$
Volume of material	=	$0.12m^2 \times 0.002m$
	=	$0.00024m^3$
Mass of material	=	$0.00024m^3 \times 7850kg/m^3$
	=	1.884kg.

d) Total mass of hopper material =

mass of rectangular section = 9.42kg

mass of trapezloided section = 7.536 + 3.768kg

= 11.304kg

mass of hopper bottom = 1.884kg.

∴ Total mass of hopper = 9.42 + 11.304 + 1.884

= 22.608kg

Volume of hopper (rectangular section) = l x b x h

= 1000 x 600 x 500

= 300,000,000mm^3

= 0.30m^3

Volume of hopper (trapezoidal section) = ½ (a + b)h x L

= ½ (600 + 200) 300 x 1000

= 120000mm^2 x 1000mm

= 0.12m^3

Total volume of hopper = volume of rectangular + volume of trapezoidal

= 0.30m^3 + 0.12m^3

= 0.42m^3

Mass of manure = Bulk density of manure x Volume of hopper

= 800kg/m^3 x 0.42m^3

= 336kg.

Weight of hopper and material =

Weight of hopper material + weight of manure

= 22.608kg + 336kg

= 356.608kg.

3.8 Hopper Frame

The hopper frame is constructed using a square hollow steel section of 80m x 80m with a cross sectional area of 14.9 x $10^{-4}m^2$ with 5mm thickness. It was joined together by fillet welding. Since the hopper frame will be acting as a load relieve plate for the hopper, the strength of the frame will be such that it would be able to withstand the compressive force due to the weight of the hopper when fully loaded.

3.8.1 Analysis Of Deflection Of Main Frame.

The main frame has a length of 1000mm and a width of 700mm constructed by 80 x 80 x 5 hollow steel section.

In analyzing the frame for failure, it is assumed that the frame is fixed to its attachment points. In this case the frame might fail by deflection over the years. It is also assumed that the load acting on the frame is uniformly distributed. The allowable deflection for the fixed frame is given as:

$$\delta_{all} = \frac{\text{Span}}{360}$$

where span = length of the beam in mm.

The maximum deflection (δmax) which can occur along the length of the frame is thus given by:

$$\delta max = \frac{5WL^2}{384EI} \qquad \text{(Singh, 1982)}$$

where W = uniformly distributed load (KN/m)

L = span length of frame (m)

E = elastic modulus of material (GN/m^2)

I = moment of inertia (m^4)

$$\delta max = \frac{5WL^2}{384EI}$$

$$= \frac{5 \times 4440 \times 1^2 m^2}{384 \times 207 \times 10^9 \times 139 \times 10^{-8}}$$

$$= \frac{22200}{11048832} = 0.002$$

$$= 2mm$$

The available deflection for the 1000mm span main frame length is:

$$\delta_{all} = \frac{1000}{360} = 2.78m$$

Since δmax is less than δ_{all}, the main frame is considered safe.

3.8.2 Analysis Of Buckling For Conveyor Support Bars.

The vertical members of the conveyor frame are of equal length. They may fail due to the weight of the conveyor and weight of the manure. To ensure safety, the critical load is determined using the Euler's formular as:

$$P_{cr} = \frac{4\pi^2 EI}{L^2} \quad \text{................ (3.47)}$$

Where P_{cr} = critical load (N)

E = elastic modulus of the material (GN/m^2)

I = moment of inertia of the material (m^4)

L = axial length of the material (m)

$$P_{cr} = \frac{4 \times 3.142^2 \times 207 \times 10^9 \times 7.86 \times 10^{-8}}{0.400^2}$$

$$= 4.014 \text{ MN}$$

The calculated load is much more than the load supported by the bars and so the design is safe.

3.9 Universal Joints

Krutz et al (1984) stated that the primary use of the universal joint is to connect intersecting shafts to transmit power and motion, the kinematics of motion of a joint is unusual. When the joint operates at an angle, the motion of the yoke does not follow the motion of the input yoke in angular displacement, velocity and acceleration. The variation between the input and output motions depends on the operating angle between the two shafts.

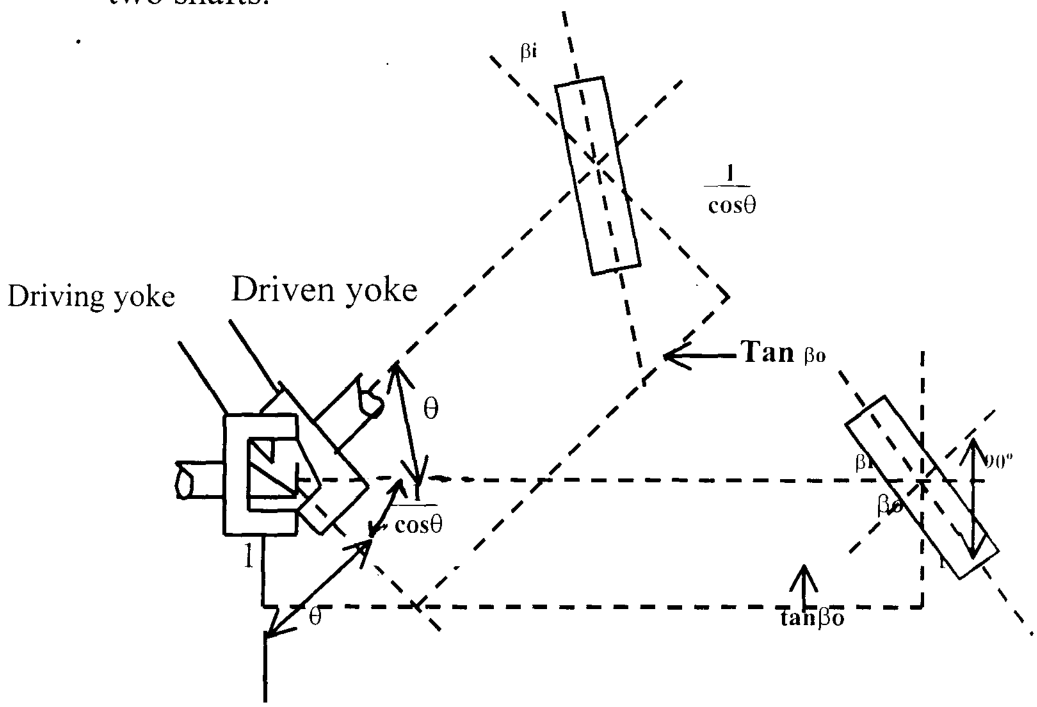

Fig. 3.6 Joint operating angle relationship

The operating angle is assumed constant with time and should not exceed 60°. The maximum angular velocity ratio is

$$w_m = \frac{\cos\theta}{1-\sin^2\theta} = \frac{1}{\cos\theta} \qquad \text{........... (3.48)}$$

By differentiating above equation wrt time, the angular acceleration is

$$a_w = \frac{w^2\cos\theta\,\sin^2\theta\sin 2\beta_1}{(1-\sin^2\theta\,\sin^2\beta_1)^2} \qquad \text{.......... (3.49)}$$

Maximum acceleration is

$$(a_r)m = w^2\sin^2\beta_1$$

The relative rotational positioning of the yokes on the connecting shaft so that the output motion is nearly uniform with the input motion is termed phasing.

(b) Phasing.

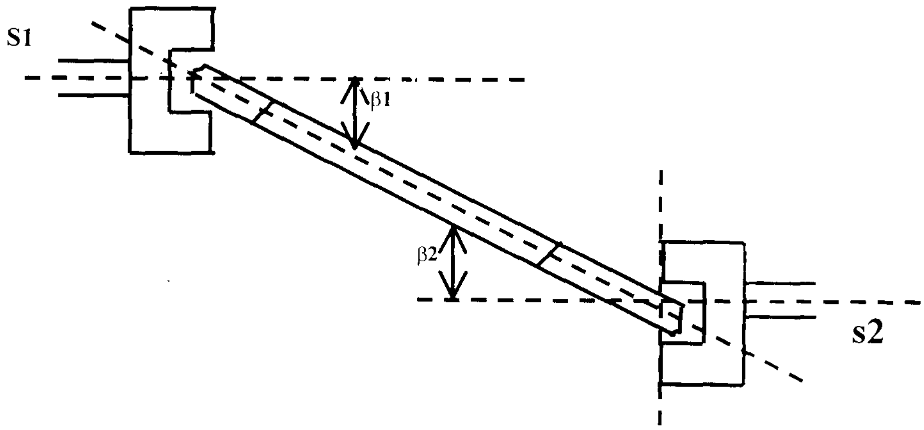

Fig 3.7: Driveline arrangement with all shafts in common plane.

To determine the torque of shaft S_2 shown above, given that the torque of input is 514.6Nmm, $\theta = 20^o$, $\beta = 90^o$

$$T_{s1}w_1 = T_{S2}w_2 \qquad (3.50)$$

$$\text{Or} \quad T_{S1}w_1 = T_{S2}w_1\cos\theta$$

$$\therefore \quad TS_2 = \frac{514.6}{\cos 20}$$

$$T_{S2} = 547.625\text{Nsmm.}$$

To avoid the tractor going into the manure material the machine needs to be phased or offset to the tractor.

3.10 Tool Bar Design.

The tool bar is used for the coupling of the machine to the tractor draw bar .The length of the tool bar is guided by the standard recommended by the ASAE. The location of the drawbar hitch and the implement input connection shaft shall be in relationship as stated in ASAE S203, S204 S217. It is recommended that the regular drawbar position is 350mm from the tractor PTO shaft while the implement input connection shaft is further 356mm away from the hitch point. This distance however increases per every 5^o degrees increase of implement input connection point above the drawbar .The total length of tool bar should not exceed 2184mm.

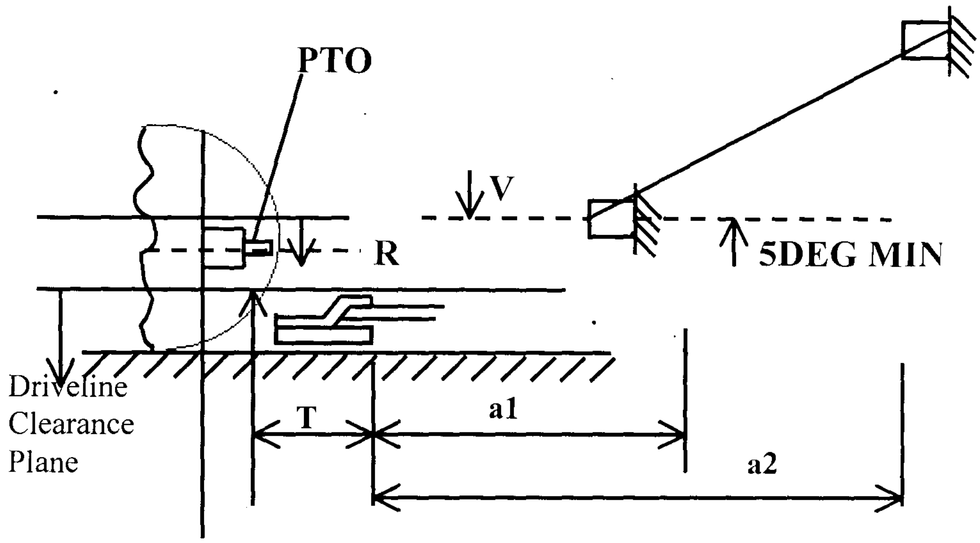

Fig 3:8 POSITION OF I.I.C FOR TRAILING IMPLEMENTS

Table 3.4 Relation of Implement Input Connection for Trailed Implement.

PTO Type	T Regular position Drawbar (mm)	a_1 +100 mm	R	V
1	350	356	SEE ASAE 5482	Equal to Or greater Than R
2	400	400		
3	500	500		

ASAE S203

The tool bar normally subjected to tensile force by the tractor pull at one end and the weight of the implement resisting the pull from the other end.

The ratio of unit stress to the unit strain of the material in tension is called Young's Modulus (E). for many common engineering materials, the Young's Modulus in compression is very nearly equal to that found in tension.

Using Euler's equation the critical load on the bar can be determined .The bar is a solid steel bar of 80mm X 30mm nominal size and length of 950mm.

Euler's formula = $Pcr = \frac{\pi^2 E I}{L^2}$

Where Pcr = Critical tool (N)

E = Young's modulus (GN/m^2)

I = Moment of the inertia (m^4)

L = Length of the member (m)

$$I = \frac{b^3 h,}{12} \qquad I = \frac{0.08^3 \text{ X } 0.03}{12} = 1.28 \text{ X } 10^{-6} m^4$$

$$E = 207 \text{ X } 10^9 \text{ GN/m}^2$$

$$L = 0.95m.$$

$$Pcr = \frac{3.142 \text{ X } 207 \text{ X}10^9 \text{ N X } 1.28 \text{ X } 10^{-6} \text{ m}^4}{0.95^2 \text{ m}^2 \text{ X m}^2}$$

$$= 9.22MN.$$

3.11 Tyre Selection.

Generally, tyres used on the farms fall into three categories namely traction , steering and implement tyres. Implement tyres in addition to ease mobility, also are designed to support the weight of the machine. Implement tyre are this selected for this machine .

Tyre size are designated by its cross sectional diameters and the diameter of the rim (Smith and Wilkes. 1980) For example tyre size designated as 13.6-36 means that the tyre has a cross sectional diameter of 13.6cm and a rim diameter of 38cm.

The total weight of a machine to be carried on the wheels determines the size of the wheels to be used.

Bosoi et al (1988) gave the formular for determining the service (total) mass for such machines as follows:

$M_{Sr} = m_m + m_n + m_c + m_b + m_i + m_f$

Where M_{sr} = service (total) mass of the machine

M_m, m_n, m_c, m_b, m_i, m_f are the masses of machine, fuel, water, lubricants, instruments filling materials such as seeds, seedlings, fertilizers etc.

For this machine, the formular is modified as:

$M_{sr} = m_{rn} + m_{sw} + m_{cu} + m_t$

m_{sr} = total mass of machine

m_{rn} = mass of hopper and manure

m_{sw} = mass of sweeper unit

m_{cu} = mass of conveying unit

m_t = mass of drive transmission unit

The weight are computed as: $m_m = 356.608$, $m_{sw} = 18kg$, $m_{cu} = 38kg$, $m_t = 40kg$.

$\therefore M_{sr} = 356.608 + 18 + 38 + 40 = 452.608kg$

Force due to this load $= 452.608 \times 9.81 m/s^2$

$= 4440N$

The total weight to be supported by both wheels is 452kg, which means each tyre size will support a weight of 226kg. ASAE (1998) recommends that tyre size 6.00 X 16 be used for tire load of 465kg maximum ,for inflation pressure of 170kpa

3.12 Design Principles And Theory Of The Sweeper

The Animal waste collector is designed on the principle of the rotary tiller. It is to be powered by a small internal combustion engine (tractor) and only a small earthing up enough to remove sticky waste materials will be done .This is to keep the draft requirement at a minimum as the design is intended for small/medium sized farms .

Fig 3.9 is the sketch of a rotary sweeper with a set of brushes denoted as A_1, A_2, A_N, A_Z .The sketch shows the mechanism of operation of the brushes. The radius of brush, R ,and the depth of sweep is denoted by a. Let V_m and ω be the forward speed of the machine and the angular velocity of the sweeping brush respectively . So the implement moves forward a distance of V_mT while the brushes rotates through an angle ωt to assume the position A'_1, A'_2, A'_N, A'_Z . The co-ordinates of A'_1 is hereby referred to as x and y-axes and given by:

$$X = V_m t + R \cos \omega t \quad \dots\dots\dots\dots\dots\dots (3.51)$$

$$Y = R (1 - \sin \omega t) \quad \dots\dots\dots\dots\dots\dots (3.52)$$

3.12.1 Condition For Sweeping Off The Waste Material

The equation given above 3.51 and 3.52 describe the absolute motion of the brush to trace the locus of $A'1$ over time as sketched .Odigboh et al (1979)

Let us say, A_1 .O denote the position of A_1 .as it begins to sweep through the soil .The depth of sweep, a, the angle through which A_1 rotates to get to A1.Ois given by θo which equal to w_{to} such that.

$$\sin \theta_o = 1 - a/R \quad \dots\dots\dots\dots\dots (3.53)$$

As A1start to penetrate the soil at A_1 .O its direction of motion is vertical down. Afterward its motion will be opposite to the direction of implement forward travel as it sweeps off the waste material. This means that there is a point of inflexion in the locus trace by A_1, at which the absolute velocity of the brush in the x direction is zero and given by equation.

$$\frac{dx}{dt} = V_m - R\omega \sin\theta = 0 \text{ ------3.54}$$

and $\sin\theta o = Vm/R\omega = 1/\lambda$ -------3.55

Which defines λ as

$$\lambda = R\omega/V_m$$

from equation 3.53, λ could be given as

$$\sin\theta^{o} = \lambda = 1 - a/R = R/(R-a) \text{ ----3.56}$$

$$= 1/(1-m)$$

Where m is the ratio of a/R

The expression of 3.54 and 3.55 give the condition for a good sweep of the soil .If the relative magnitudes of the peripheral velocity of the brushes , $R\omega$, and the machine forward speed Vm are such that dx/dt has the same direction as the implement travel then the brushes withdraws from the soil without scooping the materials . This condition is further illustrated by possible paths of the brush through the soil for various values of λ (fig 3.11) It shows that scooping most of the animal waste material will be possible only when λ is greater than unity . Therefore, for any good sweeping to occur with the animal waste collector the ratio $R\omega/Vm$ must be greater than unity.

3.12.2 Forward Travel By Scoop

The distance denoted XZ (FIG 3.9) between two consecutive point of contact of two adjacent set of brushes ,A1 and A2 with the soil surface at $A'1.O$ and $A'2.O$ respectively is called the forward travel per scoop.

Since the time of one revolution of the sweeper is $2\pi/\omega$ the forward travel per revolution of sweeper given by;

$$X_{rev} = V_m(2\Pi/\omega) = (2\Pi R)/\lambda \quad \text{.................. (3.57)}$$

Taking the number of brushes on the sweeper as Z and the forward travel per scoop is given by ;

$$X_Z = 2\dot{\Pi}R/\lambda z \quad \text{.................. (3.58)}$$

In fig 3.10, the thickness of the slice is designated by omax and may computed as;

$$d_{max} = A'_1{,}0\ A'_2 0\ COS\phi_o = X_Z Cos\phi_o \quad \text{...........(3.59)}$$

From above it can be deduced that apart from the effect of the number of brushes the soil slice can be decreased by increasing the radius of the brush, R, and decreasing the depth of sweep, a.

3.12.3 Determining The Value Of λ.

The waveform of the surface of swept by the animal waste collector can be traced as shown in fig 3.9 The value of the amptitude of that wave denoted as h is influenced by the value of λ. Therefore choosing the value is desirable in order to keep the finished surface to a desired form. To determine the λ as a function of h, the co-ordinate of the wave peak at B is given as

$$X_h = V_m t_1 + R\cos\omega t, \quad \text{.............3.60}$$

$$Y_h = h = R(1 - \sin\omega t_1) \quad \text{...........(3.61)}$$

Replacing it by ϕ_o/ω and wt by ϕ_1.

$$X_h = (R/\lambda)\phi_1 + R\cos\phi_1 \quad \text{..........(3.62)}$$

$$Y_h \quad = \quad h \quad = \quad R(1 - \sin\phi_1) \qquad \ldots\ldots\ldots\ldots(3.63)$$

From fig 3.10

$$Xh = x_1 + X_2/2$$

Since x_1 is the x – co-ordinate of A_1 when it has rotated through ωt equal to $\pi/2$; x_1 may be computed as

$$X_1 = V_m(\pi/2\omega) \quad = \quad (R\pi)/(2\lambda).$$

Recalling that X_2 equal $(2\pi R)/\lambda_2$, X_h may be re-written as

$$X_h = (\pi R/\lambda)\,({}^1/_2 + {}^1/_Z)$$

But from equations 3.62 and 3.64 it can be deduced that

$$\pi\,({}^1/_2 + {}^1/_Z) \quad = \quad \phi_1 + \lambda\cos\phi_1 \qquad \ldots\ldots\ldots\ldots\ldots (3.65).$$

While from equation 3.63

$$\operatorname{Sin}\phi_1 = 1 - {}^h/_R$$

$$\phi_1 = \sin^{-1}(1 - {}^h/_R)$$

$$\operatorname{Cos}\phi_1 = ({}^1/_R)\,(2Rh - h^2)^{1/2}$$

Substituting the value of $\theta 1$ and $\cos\theta 1$ into (3.65) yields an expression of λ as a function of h thus;

$$\lambda = \frac{[\pi({}^1/_2 + {}^1/_Z) - \sin^{-1}(1 - {}^h/_R)]}{[({}^1/_R)(2Rh - h^2)^{1/2}]} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (3.66)$$

Expression (3.66) is useful in determining the value of λ which will give a desired value of h and therefore desired finished surface of worked area.

Since the machine is pto –driven it is necessary that the drive train arrangement ensures that the rpm of the sweeper satisfies the condition, $R\omega/V_m > 1$.The drive line goes through a gear box which consist of 38 teeth crown gear and a 12 teeth pinion gear. The out put shaft from the gearbox continues to a drive pulley which in turns drive the rotary sweeper. The value of λ will depend on the forward speed of operation and the rotational speed of the sweeper through the driveline.

Therefore the ratio of forward speed /pto rpm gearbox ratio and the drive pulley ratios are to be selected to give the desired value of λ.

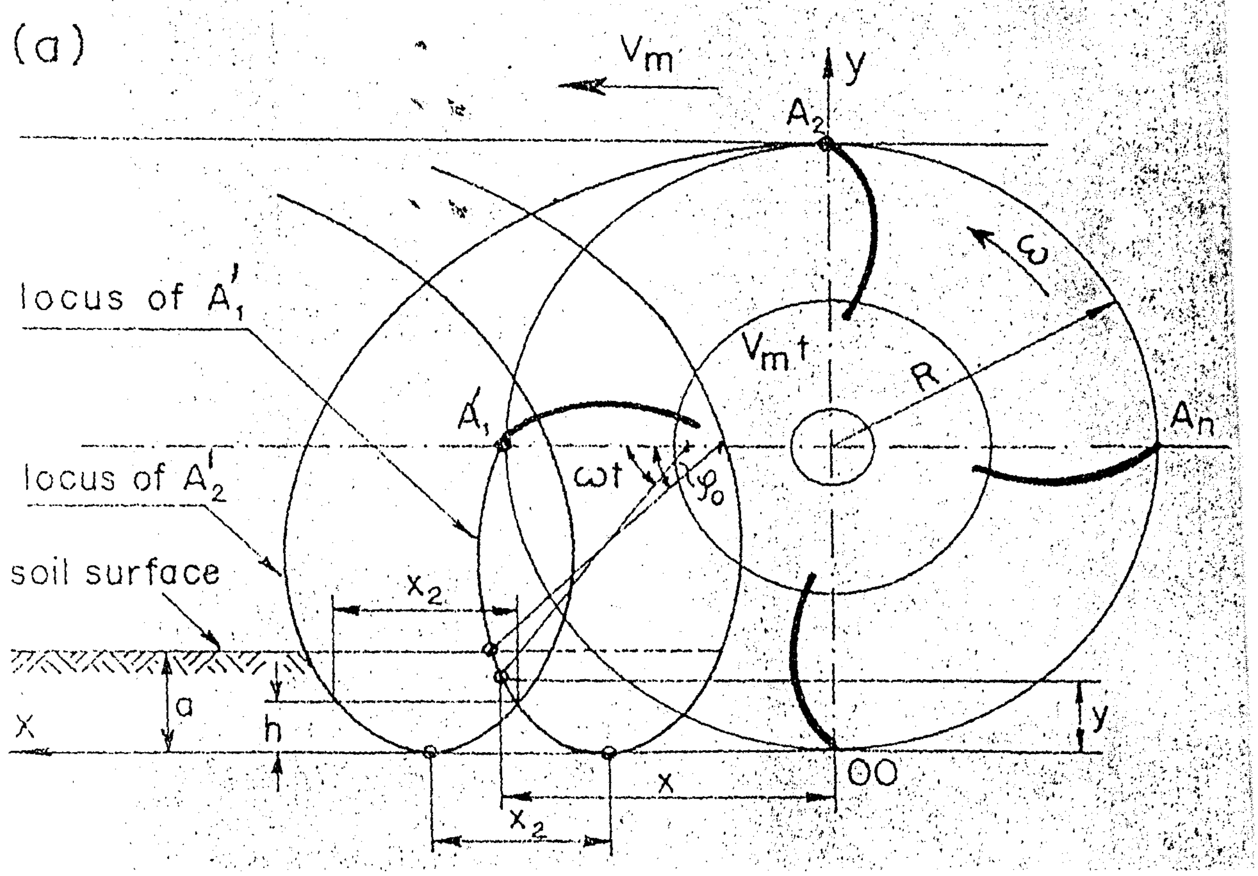

Fig 3.9: Mechanism of Operation of Sweeping brush.

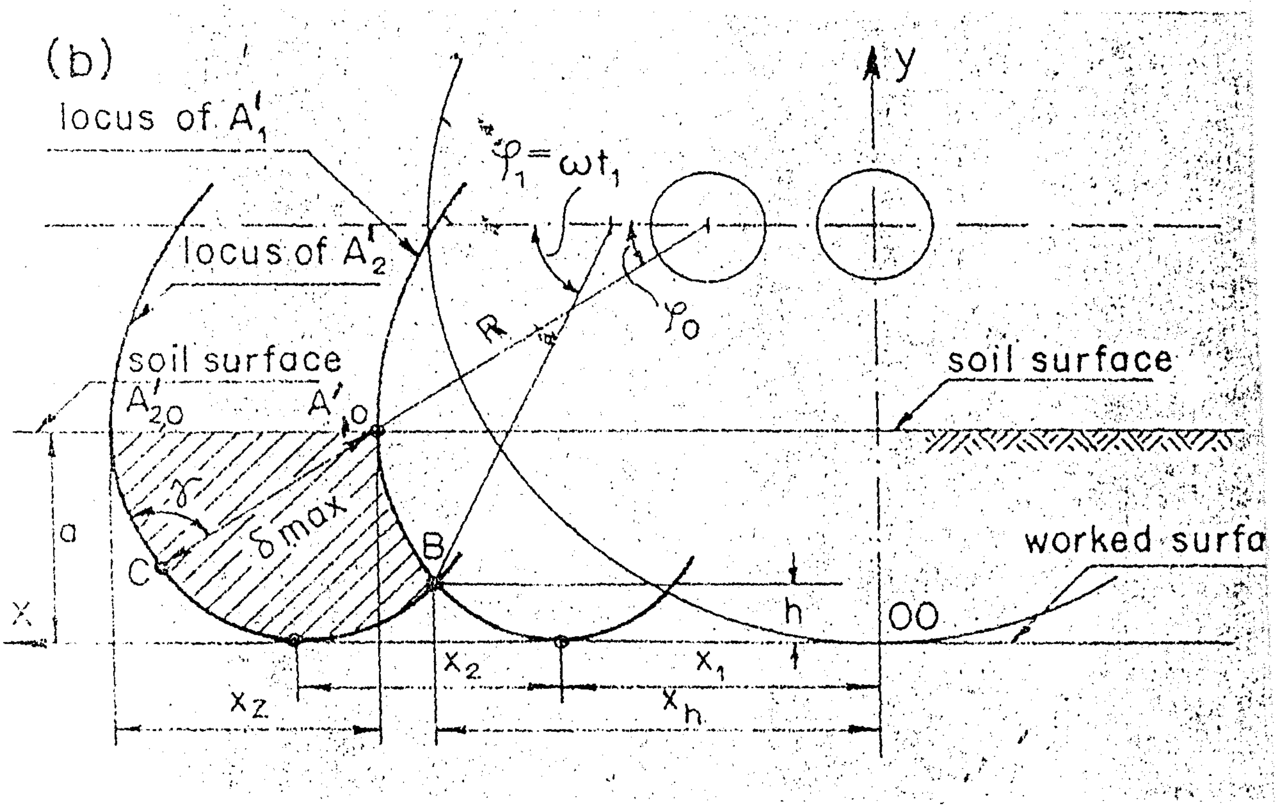

Fig 3.10: **Parameters of Soil Slice Scoop.**

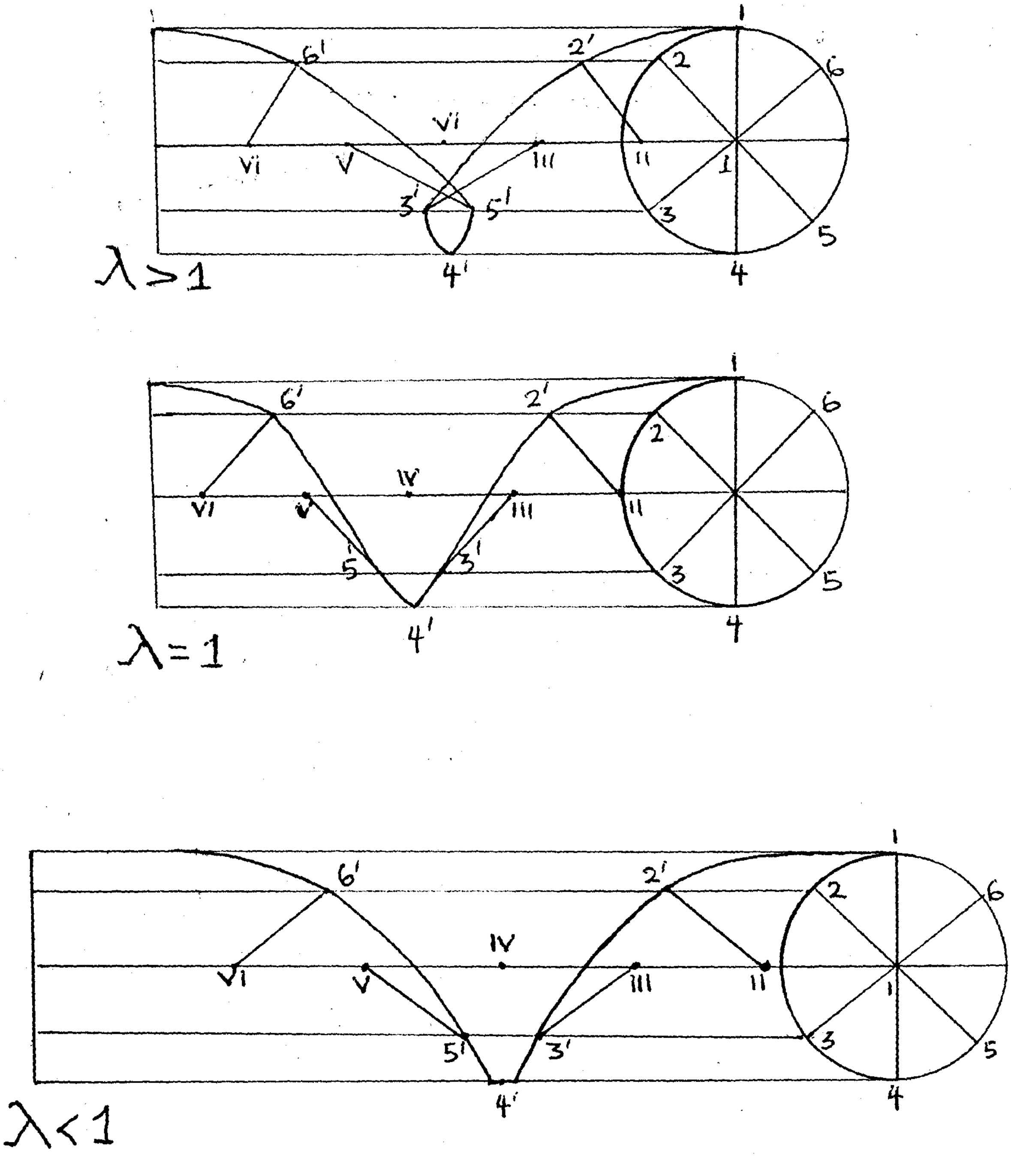

Fig 3.11: Locus of brush at different values of

CHAPTER FOUR

4.0 MATERIAL SELECTION, CONSTRUCTION AND PRINCIPLE OF OPERATION.

The production of this machine has its cost elements in terms of material cost, labour cost and miscellaneous cost.

4.1 Material Cost

The material costs are the amount spent in purchasing the materials termed as standard. The table below shows the cost of materials.

Table 4.1: Costing of Materials for construction.

No	Material	Specification	Quantity	Rate	Amount
1	Metal steel	Gauge 14	2 sheets	2200	4400
2	Hollow steels section	80 x 80 x5	1	3000	3000
3	Angle iron bar	50 x 50 x 3	1	1200	1200
4	Flat steel section	60 x 80 x 3	1	800	800
5	Bicycle wheel rim	40cm Dia	2	400	800
6	Fibre brushes	250 x 50	12	230	2760
7	Solid shaft	100 x 250	1	800	800
8	Gear box	Standard	1	9000	9000
9	Belt pulley	Φ 320	1	950	950
10	Belt pulley	Φ 250	1	600	600
11	Belt pulley	Φ 150	1	450	450
12	Conveyor belt	Standard	1	8000	8000
13	'U' shaped steel section	80 x 60 x 3	1	850	850
14	PTO shaft	Standard (6splines)	1	7500	7500
15	Ram cylinder	100 x 40	1	5000	5000
16	Hydraulic hose	Standard (10m)	1	2500	2500
17	Tyre	6.00 x 12	2	1500	3000
18	Tube	6.00 x 12	2	750	1500
19	Bearings	FAG 6234	4	350	1400
20	V – belt	HC 2266	1	850	850
21	V – belt	HC 2012	1	550	550
TOTAL				**=N=55,910.00**	

4.1.1 Labour Cost

This is the cost of the labour put into the production of the machine. It is recommended that 30% cost of materials should be considered as labour cost. The total cost of material was found to be =N=55,910.00 therefore labour cost is

$$\frac{30}{100} \times 55,910.00$$

=N=16,773.00

4.1.2 Miscellaneous Cost

This is the cost incurred which does not fall into either material or lab our cost. Such costs includes transportation; purchase of electrodes, Hacksaws photographing etc.

Table 4.2 Additional cost of production.

S/No.	Item	Amount
1	Purchase of electrodes (2 pkts)	1,500.00
2	Purchase of hacksaw blades, grinding stones and emery paper	1,400.00
3	Payment for cow dung and transportation	1,800.00
4	Photographs of machine construction and testing.	1,600.00
TOTAL		**=N=6,300.00**

The total cost of producing the machine is the sum of material, labour and miscellaneous costs.

Total cost = 55,910 + 16,773 + 6,300

= =N= 78,983.00

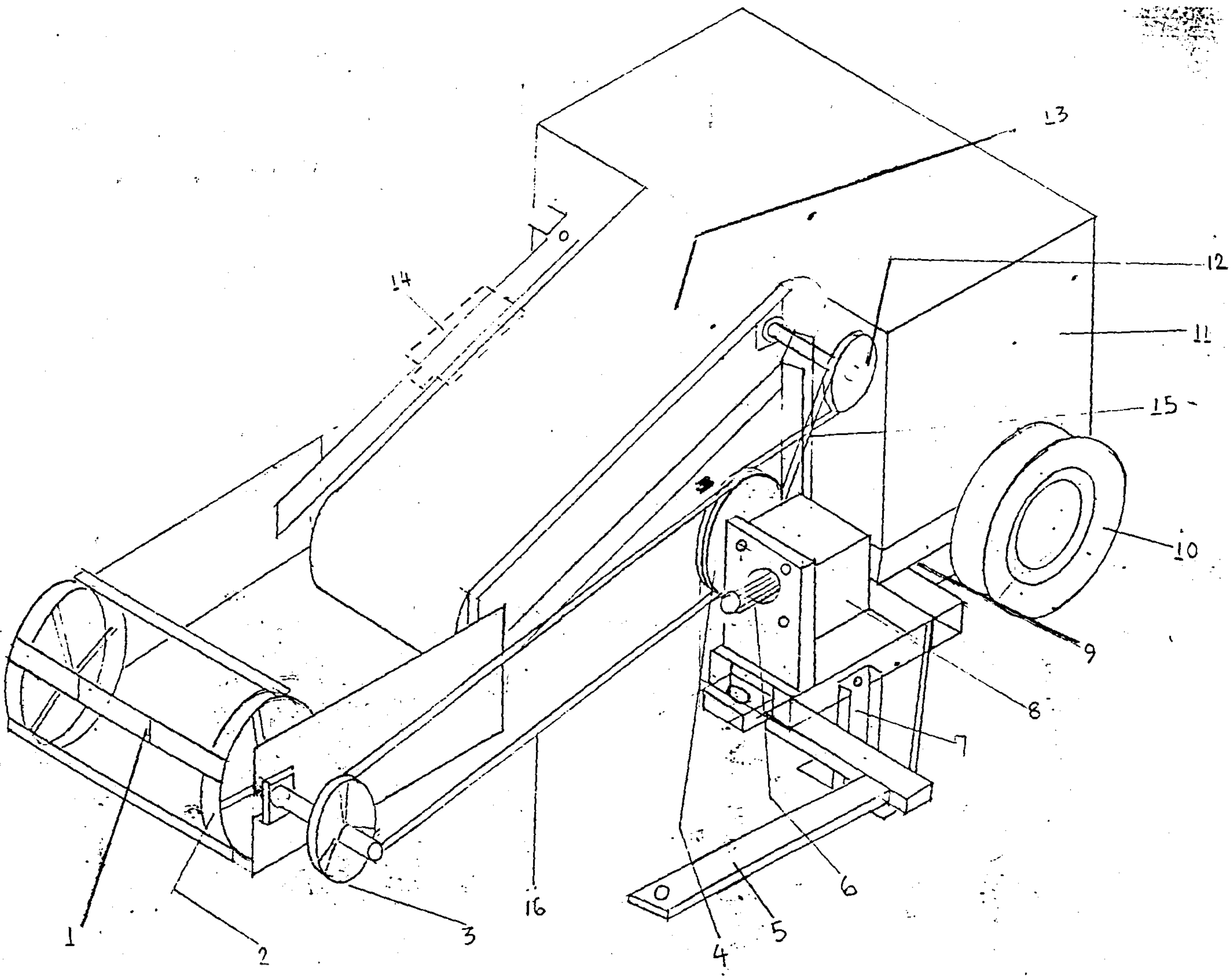

Fig 3.12: The Assembled Machine.

1.	Sweeping brush
2.	Sweeper Wheel
3.	Pulley Ø 250
4.	Pulley Ø 320
5.	Draw bar
6.	PTO Shaft
7.	Stand
8.	Gear box
9.	Main frame
10.	Land wheel
11.	Hopper
12.	Pulley Ø 150
13.	Convey or Unit
14.	Ram Cylinder
15.	Belt
16.	Belt

4.2 Construction Of Components

With the design of all components completed and the materials for the construction sourced, the fabrication of the machine was done in a standard workshop. Some of the components were of a standard specifications and such components were purchased directly from the market. Table below (Table 4.3) gives the breakdown of components, materials, and operations involved in the construction of the machine.

Table 4.3: Materials and Components of Construction.

Components	Materials	Operation carried out
Hopper	Mild steel Metal sheet Gauge 14 (2mm)	The metal sheet was marked out and welded into shape 1000mm x 600mm x 700mm
Main frame	Square hollow steel Section-mild steel 80 x 80mm x 5mm	The section was cut and drilled, Welded into a rectangular Mainframe of 1000mm x 610mm
Conveyor support Bars	Flat steel section Mild steel. 50mm x 50mm x 3mm	The iron bar was cut into equal Halves of 450mm of length each. A hole was drilled into one side in which the conveyor is hinged.
Collecting unit	Flat steel section 80mm x 3mm Fibre brushes 250mm length. Bicycle rims 2 No	The section was marked and cut Into sizes of 600mm in length with rods welded into the rims to form wheel spokes. Brushes were then connected unto the bar to form a sweeping mechanisms
Collection wheel Spindle	Solid shaft 25cm diameter	This shaft was machined to the Desired diameter and length of 25mm x 650mm. It will carry the drive sprocket from the gearbox to the collecting unit.

4.3 Description Of The Machine

The animal waste collector was built to the specifications outlined. The wheel of the sweeper is made up of two bicycle wheel rims (400mm dia) joined together by a common shaft, with the bicycle spokes replaced with 6mm diameter mild steel rods for greater strength. Flat bars of 80 x 60 x 3 were welded on the rims to connect the two wheels as a unit. (Table 4.3)

Fibre brushes of 250 x 50 were then attached unto the flat bars by use of bolts and nuts (17mm). The bicycle wheel now revolves round carrying the brushes round its circumference. The axle of the sweeper wheel also provides attachment for the drive pulley.

Sweeping takes place as the wheel carries the brushes round its circumference.

The machine has an endless conveyor belt of 400mm wide, which was converted from a disused YRSA-matic pneumatic fertilizer spreader. The conveyor unit is carried on two angle iron bars which is hinged on two support bars to allow the whole unit to be lifted in transport position. These angle iron bars are set at an angle of 50° inclination.

Vertical slats of 70mm high are attached unto the conveyor belt by use of bolt and nuts. These slats are provided to prevent the avalanche of the material as they are being conveyed.

A ram cylinder is attached to the head section (sweeper and conveying unit) of the machine which raises the section during transport and/or when a heap of materials is to be worked upon.

A hopper, which collects all materials swept, was provided at the back of the machine. The conveyor belt discharges all materials conveyed into the hopper. The power line is taken from the tractor PTO at 540 rpm through a gearbox to the various components. The drive to the sweeper unit is by an open belt drive at a belt speed of 29m/sec with a power rating of 3.24kw. (Appendix H)

The machine is trailed on two transport wheels of 6.00 x 16. the wheels are provided mainly to take up the weight of the machine and also transport the machine.

4.4 Principle Of Operation

The sweeping wheel is rotated at a speed of 346 rpm carrying fibre brushes on its circumference. The brushes sweeps any material on its path onto a collection platform which forms the base of the sweeping unit. From this platform, the material is picked up by the revolving conveyor belt (with the help of flights provided) and carried backwards to be discharged into the collection hopper.

The materials are off-loaded by gravity through a duct provided at the back of the hopper.

During transport, the whole of sweeping unit is lifted by the use of a ram cylinder provided.

The drive train consist of tractor PTO drive (540rpm) coupled directly to a 3.1:1 speed reducer which in turn drives a two stage belt arrangement to further reduce the speed. The second stage belt drive is a close type used to drive the conveying unit.

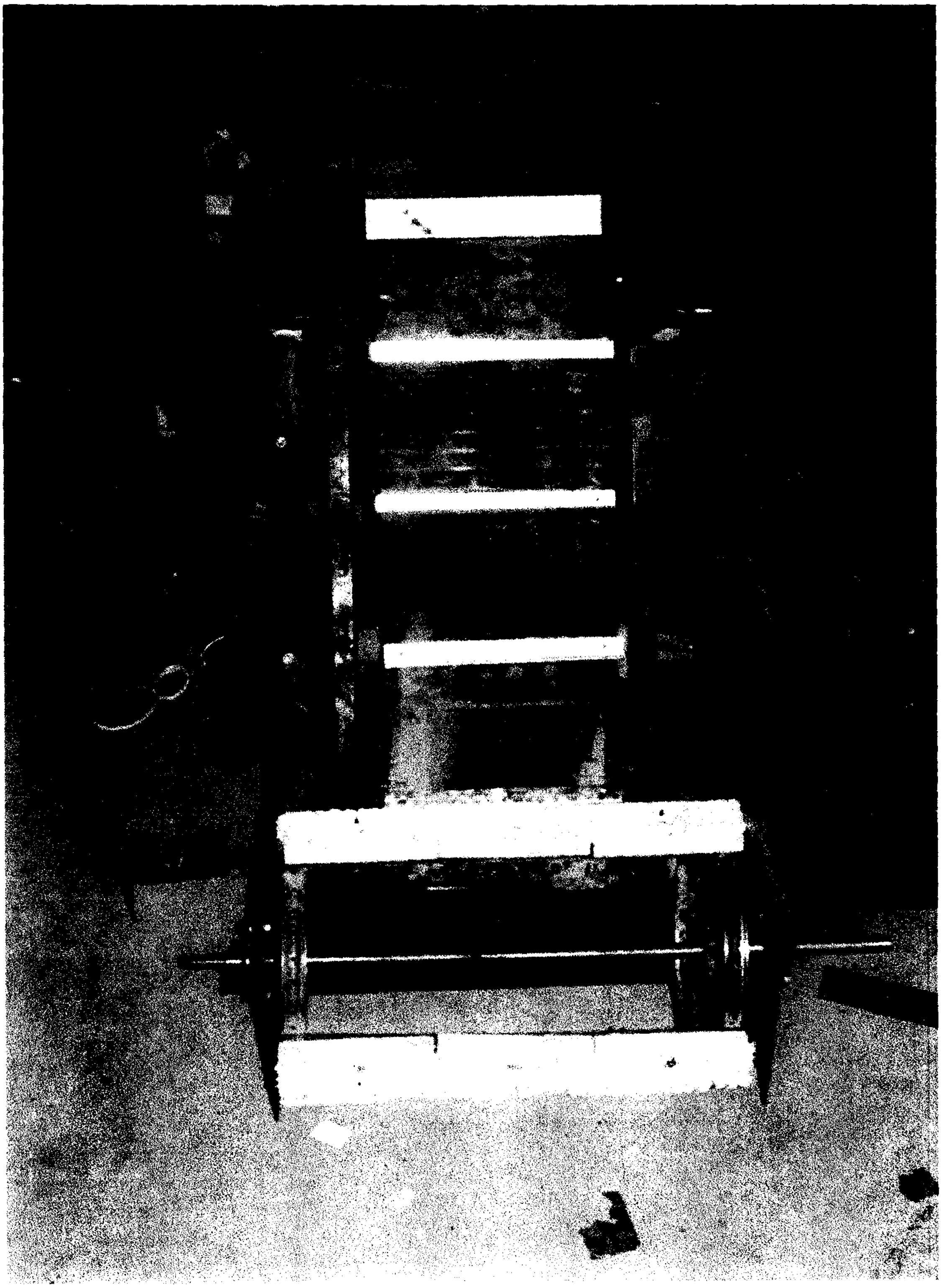

PLATE 1: TOP VIEW OF THE MACHINE AT CONSTRUCTION

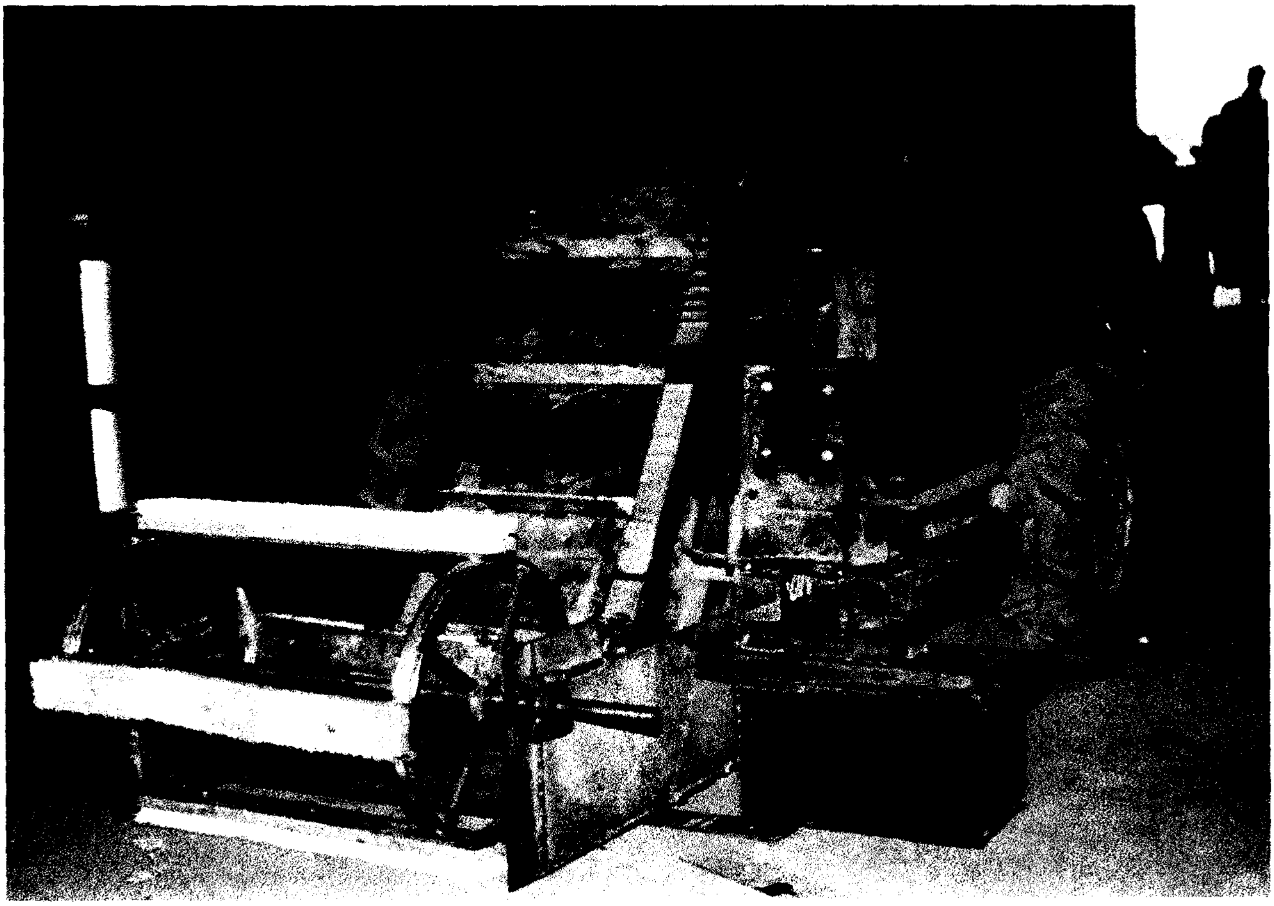

PLATE 2: SIDE VIEW OF MACHINE AT CONSTRUCTION

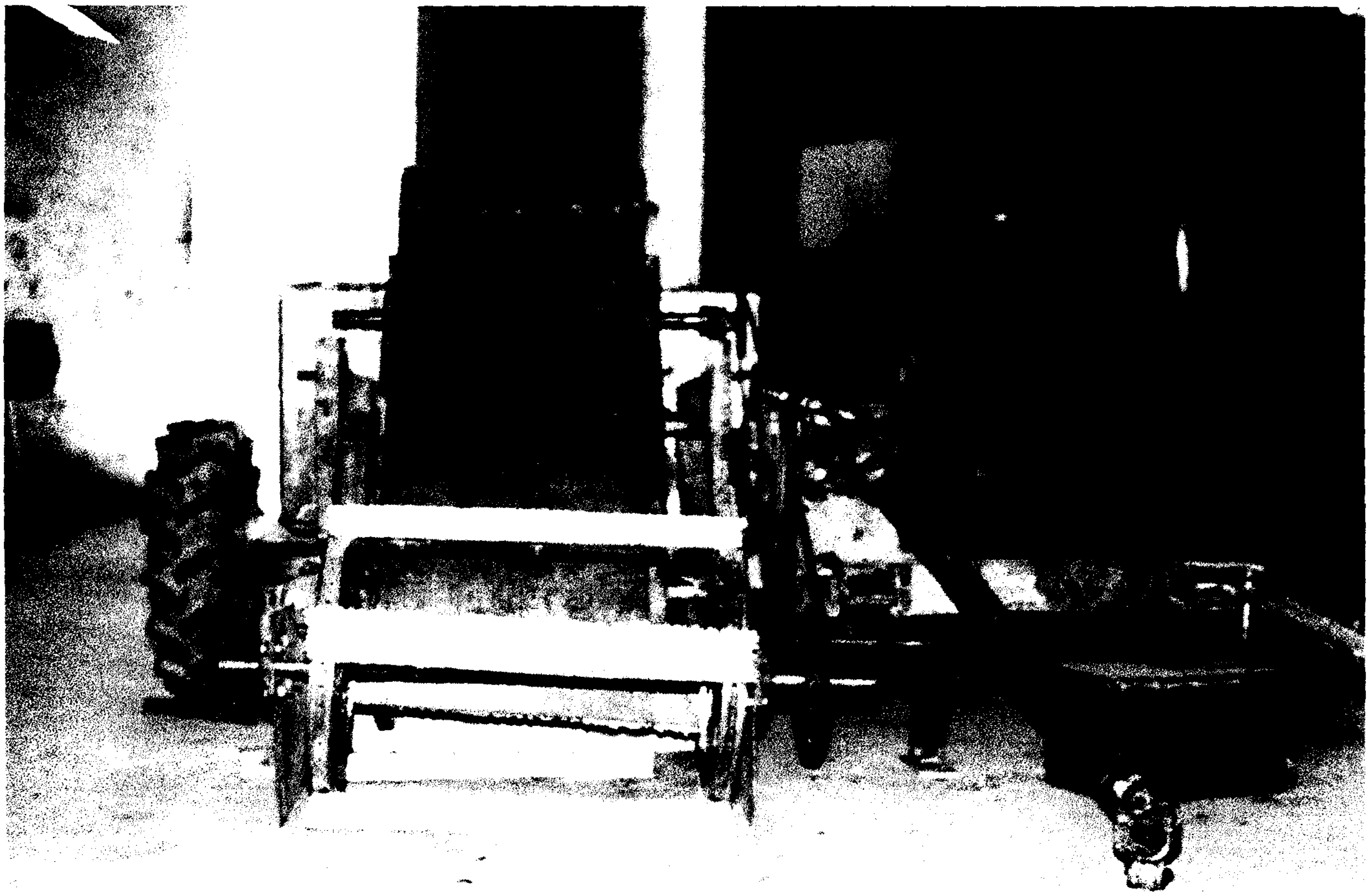

PLATE 3: THE ASSEMBLED MACHINE

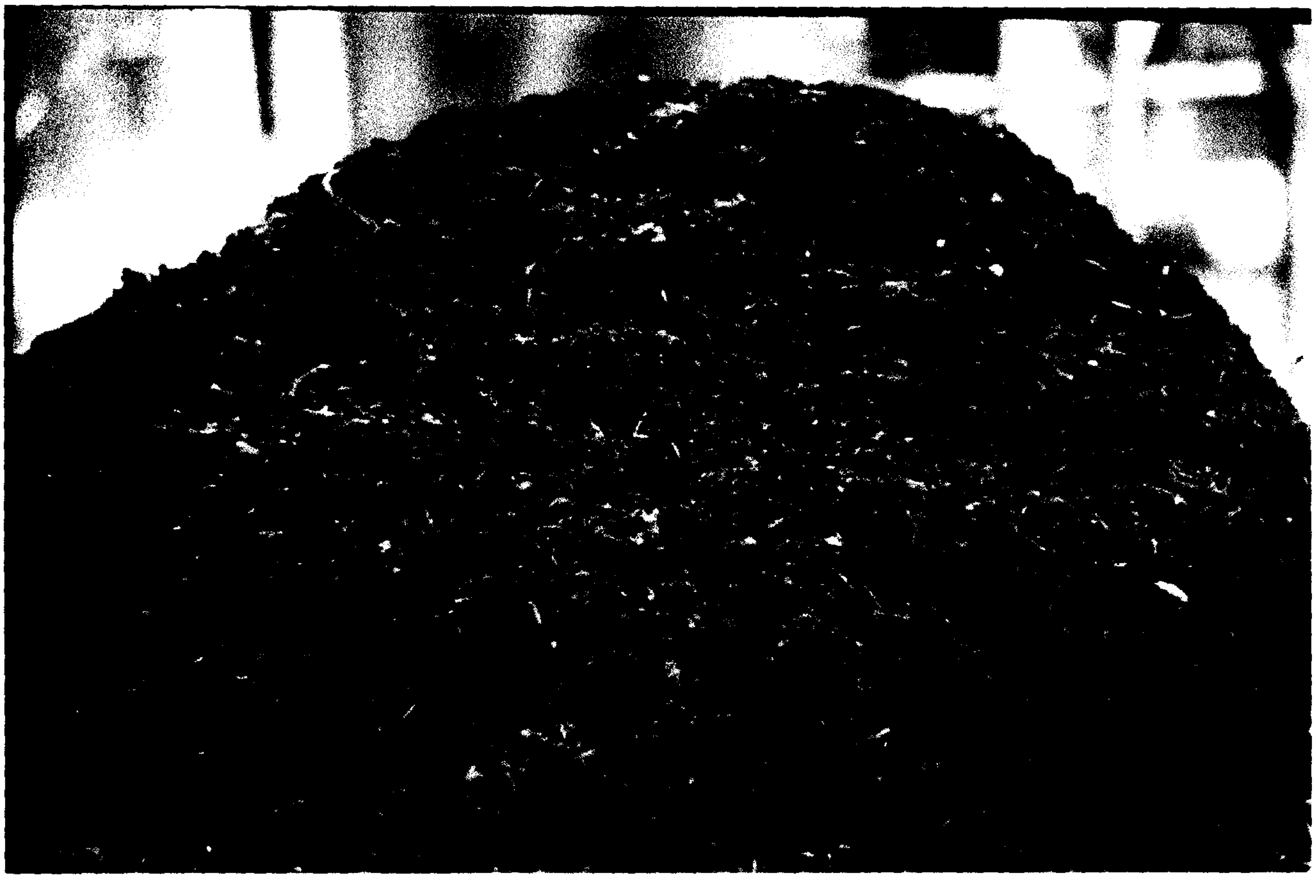

PLATE 4: MATERIALS OF INVESTIGATION (MANURE)

CHAPTER FIVE

5.0 TESTS, RESULTS AND DISCUSSION

Two main tests were carried out on the machines namely mechanical performance test and field performance test.

5.1 Mechanical Performance Test

After assembling the machine and coupling to a MF tractor Pto, it was then made to run for 30 mins to test the performance of the various components that makes up the machine. During the test a couple of faults was observed.

a) The conveyor belt was running in the wrong direction.

b) The flights attached on the conveyor was flapping the hopper

The anomalies above resulted in the conveyor belt being severed into two during the first test run. These faults were, however, corrected and smooth operation of the machine was achieved.

5.2 Field Performance Test

Field efficiency is the ratio between the productivity of a machine under field conditions and theoretical maximum productivity. Field efficiency/performance accounts for failure to utilize the theoretical operating width of the machine, time lost because of operator capability and habits and field characteristics. Field efficiency is not a constant for a particular machine but varies according to the size and shape of the field, material yield and crop/material conditions.

PROCEDURE FOR TESTING

1. 75kg of animal waste was obtained and applied over a distance of 100 meters
2. The machine was then operated over the test plot
3. Five runs was made over the distance.
4. The materials gathered was collected and weighed the following results were obtained.

Table 5.1: Field Test Result

Runs	Distance traveled (m)	Time taken (sec)	Quantity collected (kg)
1	100	66.45	63.0
2	100	66.5	65.5
3	100	66.41	64
4	100	66.36	62
5	100	66.41	63
Average		**66.43**	**63.5**

Work Rate $= \frac{QK}{LW}$ (ASAE S341)

Where
Q = weight of material collected (kg)
K = constant (10,000)
W = effective width of sweep (m)
L = distance traveled (m)

$\therefore$ Work Rate $= \frac{63.5 \times 10{,}000}{0\ 0.66 \times 100}$

$= 9621.2$ kg/ha

$= 9.6$ tons/ha.

The field capacity of a machine is the rate at which it will cover an area performing its intended function, expressed in ha/hr.

$$C = S \times W \quad \text{(Hunt, 1972)}$$

Where C = field capacity (ha/hr)

S = speed of operation (km/hr)

W = width of operation (m)

$$\text{Speed} = \frac{\text{distance traveled}}{\text{Distance taken}} = \frac{100}{66.43} \text{ m/sec}$$

$$= 1.505\text{m/sec}$$

width of operation = 66cm =0.66m

$$C = \left(\frac{1.505 \times 10000}{60 \times 60}\right) \text{X} \left(\frac{0.66}{10}\right)$$

$$= 4.18 \times 0.0066 = 0.027588\text{ha/hr.}$$

Sweeping effect test

This test tries to find out the degree of sweeping uniformity by the machine. The distance traveled (100m) was segmented into 5equal plots. The test material was the applied over each of the plots and the test carried out. The material gathered was collected and weighed.

Table 5.2: Sweeping Test Result.

No	Xi	$\overline{X}$	Xi - $\overline{X}$	(Xi – $\overline{X}$)^2
1	12.0	12.58	-0.58	0.3364
2	13.0	12.58	0.42	0.1764
3	12.4	12.58	0.18	0.0324
4	12.9	12.58	0.32	0.1024
5	12.6	12.58	0.02	0.0004

$$\text{Standard deviation} = \sqrt{\frac{\Sigma(Xi - \overline{X})^2}{N - 1}}$$

$$= \sqrt{\frac{0.9396}{4}}$$

$$= 0.4846$$

$$CV. = \frac{SD \times 100}{X}$$

$$= \frac{0.4846 \times 100}{12.58} = 3.85$$

Sweeping efficiency

The efficiency of the machine was determined by collecting all unswept materials while operating on the test plots.

The quantity of materiel applied on plots = 75kg

The quantity of material collected = 63.5kg

$$\text{Efficiency} = \frac{\text{the quantity of material collected}}{\text{The quantity of material applied}} \times \frac{100}{1}$$

$$= \frac{63.5}{75} \times \frac{100}{1} = 84.66$$

$$= 84.6\%$$

The four parameters determined during the testing are as follows: -

i)	Work rate	9.6 tons/ha
ii)	Field capacity	0.275 ha/hr
iii)	Uniformity of sweep	3.85
iv)	Efficiency	84.6%

TECHNICAL DETAILS

Working width	-	66cm
Transport width	-	214cm
Total length	-	216cm
Total height	-	110cm
Work rate	-	9612 kg/ha
Drive	-	PTO 540 rpm
Hopper capacity	-	336 kg
Speed ratio (gear box)	-	3.1 – 1
The time taken to fill the hopper	-	4 mins.
Distance travel to fill the hopper	-	550 m.

5.3 Discussion Of Results.

During the construction of this machine various options of conveying methods was considered. The belt type was chosen due to the fact that it is recommended as best when lumpy materials is being handled. Secondly, chemical reaction of the manure material on rubber is very negligible as compared with chain and slats.

The total length of the conveyor belt determines the angle of inclination and this should not exceed 70^0 when handling lumpy materials. One of the major handicap of this project is the short length of the conveyor used. This increased the angle of inclination to 50^0 and its resultant effect is that some quantity of manure material is left unconveyed on the collection platform. During mechanical performance test, the conveyor belt was severed into two halves. Reducing the volume of the hopper by cutting away some part of the hopper solved the problem. This should be an area for improvement in future development.

The work rate of the machine is 9.6 tons/ha, achieved at a speed of 1.505 m/sec. This work rate can be affected with a change in tractor forward speed and also the degree of fibrous content in the manure material.

Two types of sweeper were used depending on the situation. Fibre brushes was used when the material is dried, powdery and/or fresh. Angle iron sweepers are also fixed on the sweeping unit when a very fibrous material was to be collected in a heaped situation. Angle iron sweepers are better suited for material with high fibrous content.

The efficiency of the machine of 84.6% falls within the recommended efficiencies for agricultural implements. There is, however, room for improvement to achieve a smoother operation of the machine. Chain and sprocket drive line for the conveyor unit should also be tried in comparison to the belt drive used.

CHAPTER SIX

6.0 CONCLUSION AND RECOMMENDATION.

6.1 Conclusion

The objective of developing an animal waste collector locally has been achieved in this research. The machine was tested and the results of field capacity of 0.275 ha/hr, rate of sweeping of 9612 kg/ha and efficiency of 84.6% is encouraging. There is, however, room for improvement. The machine is simple to operate and affordable to our farmers at =N=78,983.00, considering the elimination of an additional tractor needed for the same job.

6.2 Recommendation

1) Further work is recommended to improve the performance of the machine such as changing the drive system from belt to chain and sprocket, to avoid belt slippage when conveying huge capacity.
2) The off loading of material is by gravity and this can be improve upon by incorporating an agitator or an auger especially when fibrous material is handled.
3) A bigger working width of operation should be tried, using a longer conveying unit. This will reduce the angle of inclination thereby improving the conveying efficiency.

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APPENDIX A

DISTRIBUTION OF ANIMALS IN NIGERIA

No	State	Cattle	Sheep	Goat	Donkeys	Horses	Carmels	Pigs
1.	Ak/Ibom	7000	576000	816000	-	-	-	89000
2.	Anambra	64000	426000	1,4670000	-	-	-	62000
3.	Bauchi	1,732000	2,811000	3,465000	96000	13000	-	536000
4.	Benue	146000	864000	2,432000	-	-	-	703000
5.	Bendel	47000	737000	1,248000	-	-	-	180,000
6.	Borno	2,727000	2,424000	3,188000	181,000	88000	27000	76000
7.	C/River	10,000	117000	351000	-	-	-	68000
8.	Gongola	1,503000	1,324000	1,97000	50,000	10,000	-	476000
9.	Imo	13000	495000	1,281000	-	-	-	8,000
10.	Kaduna	998000	441000	866000	15000	2000	-	229000
11.	Kano	999000	2,059000	2,490000	106000	23000	7000	-
12.	Katsina	625000	1,553000	2,009000	153000	23000	7000	-
13.	Kwara	563000	843000	1,152000	26000	1000	-	80,600
14.	Lagos	3000	57000	158000	-	-	-	25000
15.	Niger	1,165000	732000	969000	26000	3000	-	81000
16.	Ogun	27000	340,000	905000	-	-	-	150,000
17.	Ondo	9000	589000	1,747000	-	-	-	291,000
18.	Oyo	296000	863000	1,859000	1000	-	-	178,000
19.	Plateau	1,054000	904000	1,865000	28000	3000	-	536,000
20.	Rivers	3100	509000	67000	-	-	43000	66000
21.	Sokoto	1,769000	2,546000	2,449000	247000	24000	87000	21000
	Total	13761000	21,230000	33867000	929000	200000		336700

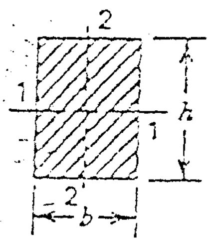
Source : NLPD Kaduna 1992

B

COMMONLY USED CROSS SECTIONAL PROPERTIES

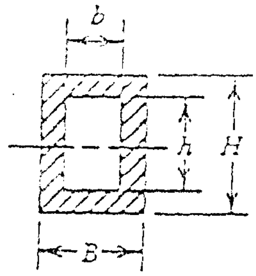
Section of beam $\quad I$ = moment of inertia

$$Z = \text{section modulus} = \frac{I}{C}$$



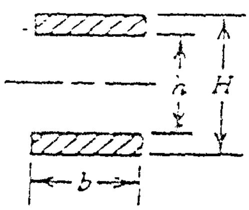
$$I_{1-1} = \frac{bh^3}{12} \qquad I_{2-2} = \frac{b^3h}{12}$$

$$Z_{1-1} = \frac{bh^2}{6} \qquad Z_{2-2} = \frac{b^2h}{6}$$



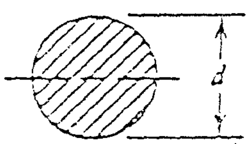
$$I = \frac{BH^3 - bh^3}{12}$$

$$Z = \frac{BH^3 - bh^3}{6H}$$



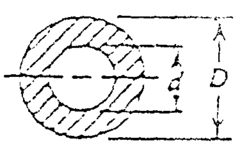
$$I = \frac{b}{12}(H^3 - h^3)$$

$$Z = \frac{b}{6H}(H^3 - h^3)$$



$$I = \frac{\pi d^4}{64} = 0.0491 d^4$$

$$Z = \frac{\pi d^3}{32} = 0.0982 d^3$$

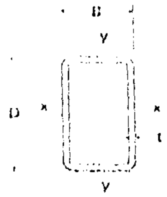


$$I = \frac{\pi}{64}(D^4 - d^4) = 0.0491(D^4 - d^4)$$

$$Z = \frac{\pi}{32D}(D^4 - d^4) = 0.0982\left(\frac{D^4 - d^4}{D}\right)$$

TABLE 8.6 PROPERTIES OF METALS (Average values)

Material	Condition	Ultimate Stresses (MN/m²)			Tensile Elastic Limit (MN/m²)	Elongation (per cent)	Young's Modulus E (GN/m²)	Modulus of Rigidity G (GN/m²)	Density (kg/m³)
		Tensile	Compressive	Shear					
Wrought Iron	—	340	275	275	185	25 to 45	186	83	7800
Cast Iron (Grey)	—	138 to 330	650	—	—	—	95.5	41	7200
Cast Iron (Malleable)	—	310 to 520	—	—	—	5 to 15	172	—	
Steel (Mild) 0·05 to 0·3% C	Normalised	430	430	340	275	25	207	80	7850
Steel (Medium) 0·3 to 0·5% C	Normalised	480	480	330	310	20	207	80	7850
Steel (High Carbon) 0·5 to 0·9% C	—	620 to 820	—	—	—	—	207	80	7850
Aluminium	Cast	100	83	83	45	—	70	—	2700
Aluminium	Forged	165	—	—	83	—	70	—	2700
Duralumin	—	410	340	250	—	10 to 20	70	—	2760
Copper	Annealed	226	—	—	—	—	124	—	8940
Copper	Cold Worked	275	—	—	—	—	124	—	8940
Brass	—	310 to 535	—	—	—	—	103	38	8250
Phosphor Bronze	—	310 to 620	—	—	—	—	96	—	8900
Magnesium Alloys	Cast	150 to 280	—	—	—	—	45	17	1800
Magnesium Alloys	Rolled	186 to 340	—	—	—	—	45	17	1800
Beryllium	—	300 to 600	—	—	—	—	300	—	1800
Titanium Alloys	—	750+	—	—	—	—	100	—	4500
Monel	Annealed (Cold Rolled)	460 to 540	—	—	—	—	175	—	8800
M[illegible]	Wire annealed (Cold Drawn)	450 to 500	—	—	—	—	175	—	8800



RECTANGULAR HOLLOW SECTIONS

DIMENSIONS AND PROPERTIES

Designation		Mass per metre	Area of section	Moment of inertia		Radius of gyration		Elastic modulus		Plastic modulus		Torsional constants	
Size D × B	Thickness t			Axis x–x	Axis y–y	Axis x–x	Axis y–y	Axis x–x	Axis y–y	Axis x–x	Axis y–y	J	C
mm	mm	kg	cm²	cm⁴	cm⁴	cm	cm	cm³	cm³	cm³	cm³	cm⁴	cm³
50 × 30	2.6	3.03	3.86	12.4	5.45	1.79	1.19	4.96	3.63	6.21	4.30	12.1	5.90
	3.2	3.66	4.66	14.6	6.31	1.77	1.16	5.82	4.21	7.39	5.08	14.2	6.81
60 × 40	3.2	4.66	5.94	28.3	14.8	2.18	1.58	9.44	7.38	11.7	8.76	30.8	11.8
	4.0	5.72	7.28	33.6	17.3	2.15	1.54	11.2	8.67	14.1	10.6	36.6	13.7
80 × 40	3.2	5.67	7.22	58.1	19.1	2.84	1.63	14.5	9.56	18.3	11.1	46.1	16.1
	4.0	6.97	8.88	69.6	22.6	2.80	1.59	17.4	11.3	22.2	13.4	56.1	18.9
90 × 50	3.6	7.46	9.50	99.8	39.1	3.24	2.03	22.2	15.6	27.6	18.1	89.3	26.9
	5.0	10.1	12.9	130	50.0	3.18	1.97	28.8	20.0	36.6	23.9	116	32.9
100 × 50	3.2	7.18	9.14	117	39.1	3.58	2.07	23.5	15.6	29.2	17.9	93.3	26.4
	4.0	8.86	11.3	142	46.7	3.55	2.03	28.4	18.7	35.7	21.7	113	31.4
	5.0	10.9	13.9	170	55.1	3.50	1.99	34.0	22.0	43.3	26.1	135	37.0
100 ×	3.6	8.59	10.9	147	65.4	3.68	2.45	29.3	21.8	36.0	25.1	142	36.0
	5.0	11.7	14.9	192	84.7	3.60	2.39	38.5	28.2	48.1	33.3	187	46.9
	6.3	14.4	1[illegible]	230	99.9	3.54	2.33	46.0	33.3	58.4	40.2	224	55.0
120 × 60	3.6	9.72	12.4	230	76.9	4.31	2.49	38.3	25.6	47.8	29.2	183	43.3
	5.0	13.3	16.9	304	99.9	4.24	2.43	50.7	33.3	63.9	38.8	242	56.0
	6.3	16.4	20.9	366	118	4.18	2.38	61.0	39.4	78.0	46.9	290	66.0
120 ×	5.0	14.8	18.9	370	196	4.43	3.21	61.7	48.8	75.4	56.7	401	77.9
	6.3	18.4	23.4	447	234	4.37	3.16	74.5	58.4	92.3	69.1	486	93.0
	8.0	22.9	29.1	537	278	4.29	3.09	89.5	69.4	113	83.9	586	110
	10.0	27.9	35.5	628	320	4.20	3.00	105	80.0	134	99.4	688	126
150 × 100	5.0	18.7	23.9	747	396	5.59	4.07	99.5	79.1	121	90.8	806	127
	6.3	23.3	29.7	910	479	5.53	4.02	121	95.9	148	111	986	153
	8.0	29.1	37.1	1106	577	5.46	3.94	147	115	183	137	1202	184
	10.0	35.7	45.5	1312	678	5.37	3.86	175	136	220	164	1431	216

E

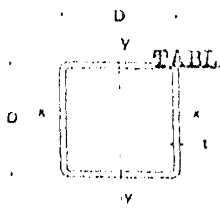


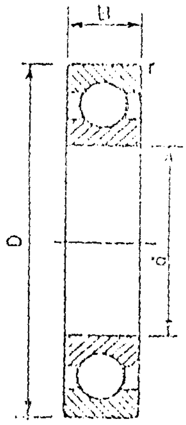
TABLE 9: SQUARE HOLLOW SECTIONS

DIMENSIONS AND PROPERTIES

Designation: Size D × D	Designation: Thickness t	Mass per metre	Area of section	Moment of inertia	Radius of gyration	Elastic modulus	Plastic modulus	Torsional constants: J	Torsional constants: C
mm	mm	kg	cm²	cm⁴	cm	cm³	cm³	cm⁴	cm³
20 × 20	2.0	1.12	1.42	0.76	0.73	0.76	0.96	1.22	1.07
	2.6	1.39	1.78	0.88	0.70	0.88	1.16	1.44	1.23
30 × 30	2.6	2.21	2.82	3.49	1.11	2.33	2.88	5.80	3.30
	3.2	2.66	3.38	4.00	1.09	2.67	3.37	6.45	3.76
40 × 40	2.6	3.03	3.86	8.94	1.52	4.47	5.38	14.0	6.41
	3.2	3.66	4.66	10.4	1.50	5.22	6.40	16.5	7.43
	4.0	4.46	5.68	12.1	1.46	6.07	7.61	19.5	8.60
50 × 50	3.2	4.66	5.94	21.0	1.91	8.02	10.4	33.8	12.4
	4.0	5.72	7.28	25.5	1.87	10.2	12.6	40.4	14.5
	5.0	6.97	8.88	29.6	1.83	11.9	14.9	47.6	16.7
60 × 60	3.2	5.67	7.22	38.7	2.31	12.9	15.3	60.1	18.6
	4.0	6.97	8.88	46.1	2.28	15.4	18.6	72.4	22.1
	5.0	8.54	10.9	54.4	2.24	18.1	22.3	86.3	25.8
70 × 70	3.6	7.46	9.50	69.5	2.70	19.9	23.8	108	28.7
	5.0	10.1	12.9	90.1	2.64	25.7	31.2	142	36.8
80 × 80	3.6	8.59	10.9	106	3.11	26.5	31.3	164	38.6
	5.0	11.7	14.9	139	3.06	34.7	41.7	217	49.8
	6.3	14.4	18.4	166	3.00	41.3	50.6	261	58.8
90 × 90	3.6	9.72	12.4	154	3.52	34.1	40.0	237	49.7
	5.0	13.3	16.9	202	3.40	45.0	53.6	310	64.9
	6.3	16.4	20.9	242	3.41	53.9	65.3	381	77.1
100 × 100	4.0	12.0	15.3	243	3.91	48.6	54.9	381	68.2
	5.0	14.8	18.9	283	3.87	56.6	67.1	459	81.9
	6.3	18.4	23.4	341	3.81	68.2	82.0	533	97.9
	8.0	22.9	29.1	408	3.74	81.6	99.9	646	116
	10.0	27.9	35.6	474	3.65	94.9	118	761	134
120 × 120	5.0	18.0	22.9	503	4.69	83.8	98.4	776	122
	6.3	22.3	28.5	610	4.63	102	121	949	147
	8.0	27.9	35.5	738	4.56	123	149	1159	176
	10.0	34.2	43.5	870	4.47	145	178	1381	206

F

Table 13.11. Deep Groove Ball Bearing (Series 62)



ISI No.	Bearing of basic design No. (SKF/FAG)	d mm	D_1 min	D mm	D_2 max	B mm	r ≈ min	r_1 mm	Basic Capacity, kgf: Static Co	Basic Capacity, kgf: Dynamic C	Max. permissible speed rpm
10 BC 02	6200	10	14	30	26	9	1	0·6	224	400	20000
12 BC 02	6201	12	16	32	28	10	1	0·6	300	540	20000
15 BC 02	6202	15	19	35	31	11	1	0·6	355	610	16000
17 BC 02	6203	17	21	40	36	12	1	0·6	440	750	16000
20 BC 02	6204	20	26	47	41	14	1·5	1·0	655	1000	16000
25 BC 02	6205	25	31	52	46	15	1·5	1·0	710	1100	13000
30 BC 02	6206	30	36	62	56	16	1·5	1·0	1000	1530	13000
35 BC 02	6207	35	42	72	65	17	2	1·0	1370	2000	10000
40 BC 02	6208	40	47	80	73	18	2	1·0	1600	2280	10000
45 BC 02	6209	45	52	85	78	19	2	1·0	1830	2550	8000
50 BC 02	6210	50	57	90	83	20	2	1·0	2120	2750	8000
55 BC 02	6211	55	64	100	91	21	2·5	1·5	2600	3400	8000
60 BC 02	6212	60	69	110	101	22	2·5	1·5	3200	4050	6000
65 BC 02	6213	65	74	120	111	23	2·5	1·5	3550	4400	6000
70 BC 02	6214	70	79	125	116	24	2·5	1·5	3900	8400	5000
75 BC 02	6215	75	84	130	121	25	2·5	1·5	4250	5200	5000
80 BC 02	6216	80	91	140	129	26	3	2·0	4550	5700	5000
85 BC 02	6217	85	96	150	139	28	3	2·0	5500	6550	4000
90 BC 02	6218	90	101	160	149	30	3	2·0	6300	7500	4000
95 BC 02	6219	95	107	170	158	32	3·5	2·0	7200	8500	4000
100 BC 02	6220	100	112	180	168	34	3·5	2·0	8150	9650	3000
105 BC 02	6221	105	117	190	178	36	3·5	2·0	9300	10400	3000
110 BC 02	6222	110	122	200	188	38	3·5	2·0	10400	11200	3000
120 BC 02	6224	120	132	215	203	40	3·5	2·0	10400	11400	3000
	6226	130	144	230	216	40	4	2·5	11600	12200	2500
	6228	140	154	250	236	42	4	2·5	12900	12900	2500
	6230	150	164	270	256	45	4	2·5	14300	137000	2500
	6232	160	174	290	276	48	4	2·5	15600	143000	2000
	6234	170	187	310	293	52	5	3·0	19000	166000	2000
	6236	180	197	320	303	52	5	3·0	20400	176000	1600
	6238	190	207	340	323	55	5	3·0	24000	200000	1600
	6240	200	217	360	343	58	5	3·0	26500	212000	1600

D_1, abutment diam. on shaft. D_2, abutment diam. on housing.
r_1, corner radii on shaft and housing.

Tire type nomenclature	
Code	Tire type
I-1	Rib tread
I-2	Moderate traction
I-3	Traction tread
I-6	Smooth tread

– Diagonal (bias) ply agricultural implement tires (SI metric units)

Basic tire loads for speeds 40 km/h and under (see footnote 2)

Tire size designation		170	190	220	250	280	300	330	360	390	410
		Basic tire loads (kg) at various cold inflation pressures (kPa)									
00-9	SL#	165	180	200	220	230	250(4)				
00-12	SL	205	225	245	270	285	307(4)				
00-15	SL	240	270	290	315	335	355(4)				
00-18	SL	265(2)	295	320	350	375	400(4)				
00-15	SL	330	365	405	437(4)						
.50-16	SL	410	455	495	530(4)						
.90-15	SL	385	430	470	515(4)						
00-16	SL	465	515	560(4)	610	655	690(6)				
.40-15	SL	435	480	530(4)	570	610	650	690(6)			
.50-16	SL	520	580	640	690	740	775(6)				
.70-15	SL	480	535	580(4)	635	680	730(6)				
.50-10	SL	505	560	610	670(6)						
.50-14	SL	555	615(4)								
7.50-16	SL	670	750(4)	820	885	950	1005	1065	1120	1175	1215(10)
7.50-18	SL	700	780	855	925(6)						
7.50-20	SL	720	800(4)	880	950(6)						
7.50-24	SL	760	850(4)								
7.60-15	SL	565	630(4)	695	750	800(6)	855	905	950(8)		
9.00-10	SL	670(4)	755	820	890	955	1010	1070	1120(10)		
*9.0-16	SL	890	990	1080	1170	1250	1320(8)	1405	1500(10)		
9.00-24	SL	1150	1285(6)	1400	1515	1600(8)					
10.00-15	SL	1030	1150	1255	1355	1450(8)	1545	1650(10)	1725	1800(12)	
11.25-24	SL	1495	1665	1850(8)							
11.25-28	SL	1550	1730	1890	2040	2190	2325	2430(12)			
13.50-16.1	SL	1600(6)	1770	1950(8)	2090	2240(10)	2360(12)				
Low section height											
8.5L-14	SL	635	710	775	850(6)						
*9.5L-14	SL	710(4)	800	875(6)	945	1010	1090([illegible])				
*9.5L-15	SL	750	[illegible]	900(6)	990	1055	1120(8)				1430@440(12)
*11L-14	SL	840	925(6)								
*11L-15	SL	875	975(6)	1065	1150(8)	1240	1320(10)	1390	1450(12)		
*11L-16	SL	910	1030(6)	1110	1215(8)	1290	1360(10)				
*12.5L-15	SL	1035	1150(6)	1260	1360(8)	1460	1550(10)	1640	1750(12)		
*12.5L-16	SL	1080	1200	1310	1400(8)	1520	1615	1710	1800(12)	1900(14)	
14L-16.1	SL	1400(6)	1590	1750(8)	1900(10)	2010	2120(12)				
*16.5L-16.1	SL	1800(6)	2000(8)	2205	2360(10)	2575(12)	2715	2900(14)			
19L-16.1	SL	2265	2515	2725(10)							
*21.5L-16.1	SL	2725(8)	3000(10)	3250(12)	3550(14)						

#SL—service limited to agricultural usage
*Indicates that this is an industry-wide, high production volume size and should be considered as a preferred size for new design. Tire sizes listed are not necessarily available in all tire types for all ply ratings and some additional ply ratings may be available. Consult your tire supplier for availability information.

NOTES

1 Figures in parentheses denote ply rating. All loads to the left of ply rating denote maximum load for indicated inflation pressure.
2 For speeds not exceeding 15 km/h, above loads may be increased by 15% with no change in inflation pressure.
3 For implement tires used in free-rolling steering service on self-propelled equipment, use loads from table 4. If the size required is not listed in table 4, use loads from this table reduced by 33%. Steering tires on towed equipment do not require reduced loads.
4 Shipping inflation pressure shall not exceed the maximum pressure for the ply shown.

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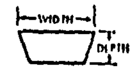
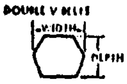
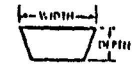
Table 16.28. Ratings for V-Belts in Kilowatt
Cross-Section A

Belt speed (m/s)	*Equivalent pitch diameter d_0 in mm*					
	80	*90*	*100*	*110*	*120*	*125 and ove*
0·5	0·13	0·13	0·14	0·14	0·15	0·15
1	0·22	0·24	0·25	0·27	0·28	0·29
2	0·37	0·40	0·43	0·46	0·49	0·51
3	0·51	0·58	0·64	0·68	0·72	0·74
4	0·58	0·74	0·81	0·88	0·93	0·96
5	0·74	0·85	0·95	1·04	1·13	1·18
6	0·81	0·94	1·05	1·15	1·25	1·32
7	0·88	1·08	1·25	1·39	1·50	1·54
8	0·96	1·17	1·35	1·50	1·62	1·69
9	1·03	1·32	1·54	1·69	1·77	1·84
10	1·10	1·40	1·62	1·77	1·91	1·99
11	1·18	1·47	1·69	1·91	2·06	2·13
12	1·25	1·54	1·84	2·06	2·21	2·28
13	1·32	1·62	1·91	2·13	2·35	2·43
14	1·32	1·69	1·99	2·28	2·50	2·50
15	1·32	1·77	2·06	2·35	2·57	2·65
16	1·40	1·84	2·13	2·43	2·65	2·79
17	1·40	1·84	2·21	2·50	2·79	2·87
18	1·40	1·84	2·28	2·57	2·87	2·94
19	1·40	1·84	2·28	2·65	2·94	3·02
20	1·32	1·91	2·35	2·72	3·02	3·09
21	1·32	1·91	2·35	2·72	3·02	3·16
22	1·25	1·91	2·35	2·72	3·09	3·24
23	1·25	1·84	2·35	2·79	3·09	3·24
24	1·18	1·84	2·35	2·79	3·16	3·31
25	1·10	1·77	2·28	2·79	3·16	3·31
26	1·03	1·69	2·28	2·72	3·16	3·31
27	0·88	1·62	2·20	2·72	3·09	3·31
28	0·81	1·54	2·13	2·65	3·09	3·24
29	0·66	1·47	2·06	2·57	3·02	3·24
30	0·51	1·32	1·99	2·50	2·94	3·16

I

[illegible]

Cross Section		in. Width	in. Depth	mm Width	mm Depth
V BELTS 	HA	0.50	0.31	12.7	7.9
	HB	0.66	0.41	16.7	10.3
	HC	0.88	0.53	22.2	13.5
	HD	1.25	0.75	31.8	19.0
	HE	1.50	0.91	38.1	23.0
DOUBLE V BELTS 	HAA	0.50	0.41	12.7	10.3
	HBB	0.66	0.53	16.7	13.5
	HCC	0.88	0.69	22.2	17.5
	HDD	1.25	1.00	31.8	25.4
ADJUSTABLE SPEED BELTS 	HI	1.00	0.56	25.4	12.7
	HJ	1.25	0.59	31.8	15.0
	HK	1.50	0.69	38.1	17.5
	HL	1.75	0.78	44.4	19.8
	HM	2.00	0.88	50.8	22.2

—EFFECTIVE LENGTHS OF AGRICULTURAL ADJUSTABLE SPEED BELTS

All Cross Sections HI, HJ, HK, HL, & HM in.	mm	Length Tolerance in.		mm	
60	1524				
64	1626				
68	1727	+0.4	−0.9	+10	−23
72	1829				
76	1930				
80	2032				
84	2134				
88	2235	+0.5	−1.0	+13	−25
92	2337				
96	2438				
104	2642	+0.5	−1.0	+13	−25
112	2845	+0.6	−1.1	+15	−28
120	3048	+0.6	−1.2	+15	−30
128	3251	+0.6	−1.3	+15	−33
136	3454	+0.6	−1.3	+18	−36
144	3658	+0.7	−1.5	+18	−38

TABLE 2—EFFECTIVE LENGTHS OF AGRICULTURAL V-BELTS AND DOUBLE V-BELTS

V-Belt Cross Sections										Double V-Belt Cross Sections							
HA		HB		HC		HD		HE		HAA		HBB		HCC		HDD	
in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	mm
28.1	714																
33.1	841																
35.1	892																
37.1	942	37.9	963														
40.1	1018	40.9	1039														
44.1	1120	44.9	1140														
48.1	1222	48.9	1242														
50.1	1272	50.9	1293														
53.1	1349	53.9	1369	55.2	1402					53.1	1349	53.9	1369	55.2	1402		
55.1	1399	55.9	1420														
57.1	1450	57.9	1471									57.9	1471				
62.1	1577	62.9	1598	64.2	1631					62.1	1577	62.9	1598	64.2	1631		
64.1	1628	64.9	1648														
66.1	1679	66.9	1699														
68.1	1730	68.9	1750														
70.1	1780	70.9	1801	72.2	1834					70.1	1780	70.9	1801	72.2	1834		
73.1	1857	73.9	1877														
77.1	1958	77.9	1979	79.2	2012					77.1	1958	77.9	1979	79.2	2012		
80.1	2034	80.9	2055														
82.1	2085									82.1	2085						
		83.9	2131									83.9	2131	85.2	2164		
		85.9	2182	85.2	2164												
87.1	2212	87.9	2233	89.2	2266					87.1	2212	87.9	2233	89.2	2266		
92.1	2339	92.9	2360	94.2	2393					92.1	2339	92.9	2360	94.2	2393		
98.1	2492			100.2	2545					98.1	2492			100.2	2545		
		99.9	2537									99.9	2537				
07.1	2720	107.9	2741	109.2	2774					107.1	2720	107.9	2741	109.2	2774		
14.1	2898	114.9	2918	116.2	2951					114.1	2898	114.9	2918	116.2	2951		
22.1	3101	122.9	3122	124.2	3155	125.2	3180			122.1	3101	122.9	3122	124.2	3155	125.2	3180
30.1	3304	130.9	3325	132.2	3358	133.2	3383			130.1	3304	130.9	3325	132.2	3358	133.2	3383
		138.9	3528	140.2	3561												
		146.9	3731	148.2	3764	149.2	3790					146.9	3731	148.2	3764	149.2	3790
		160.9	4087	162.2	4120	163.2	4145					160.9	4087	162.2	4120	163.2	4145
				166.2	4221	167.2	4247							166.2	4221	167.2	4247
		175.9	4468	177.2	4501	178.2	4526					175.9	4468	177.2	4501	178.2	4526
		182.9	4646	184.2	4679	185.2	4704	187.0	4750			182.9	4646	184.2	4679	185.2	4704
		197.9	5027	199.2	5060	200.2	5085	202.2	5131			197.9	5027	199.2	5060	200.2	5085
		212.9	5408	214.2	5441	215.2	5466	217.0	5512			212.9	5408	214.2	5441	215.2	5466
		241.4	6132	242.2	6152	242.7	6165	243.5	6185			241.4	6132	242.2	6152	242.7	6165
		271.4	6894	272.2	6914	272.7	6927	273.5	6947			271.4	6894	272.2	6914	272.7	6927
		301.4	7656	302.22	7676	302.7	7689	303.5	7709			301.4	7656	302.2	7676	302.7	7689
				332.2	8438	332.7	8451	333.5	8471					332.2	8438	332.7	8451
				362.2	9200	362.7	9213	363.5	9233					362.2	9200	362.7	9213