Design of Sesame Mowing Machine

A thesis submitted to the Graduate Studies of Addis Ababa University in Partial Fulfillment of the requirements for the Degree of Master of Science in Mechanical Engineering (Mechanical Design)

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March, 2010

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ACKNOWLEDGMENT

I would like to deeply thank my thesis Advisor Dr.Ing.Tamrat Tesfaye for his unconditional support, guidance, teaching, understanding, and continuous support throughout my work and this really will be infinitely profitable for the rest of my life.

Special thanks go to office of Graduate studies and department of mechanical engineering AAU for their unyielding efforts in providing the necessary materials and required supports throughout my study.

I am indebted to my friends Yitagesu Birhanu and his wife Hanam Teshome for their motivating and sustaining supports during my study and all my best friends for their continuous upholding me and the brainstorming discussions for this work and I strongly believe that what I have learned from them during the study period is worthy to articulate for problem solving.

Abiyou Solomon

March, 2010

ABSTRCT

In Ethiopia, harvesting is practiced in primitive method of mowing trends while it is mechanized in most of the other parts of the globe showing that the future promises to be even more dynamic to widen the gap among the nations in the world requiring us to make our agriculture mechanized. Provided that there are problems regarding mechanizing the agriculture, designing appropriate equipments in a way that they can be manufactured inland is a step in mechanizing the farming to handle the associated problems. Therefore, in this work, the design of sesame mowing component for sesame and other similar cash crops is developed comprising of problem identification, product design of the mower, analytical mechanical designs, and orthographical representation of the mower.

In preparing the design of the mower, the problems are identified through professional survey of the plantation areas and literatures, and verbal information obtained from the owners of the plantations. Based on the problems that are identified by the gathered information, the basic preliminary specifications of the mower are set.

Once the specifications are set, before the analytical design of a product is started, the alternative ideas generation and concepts development for each of its components that have competitor alternatives need to be developed, and the best alternative concept that overweigh the other optional alternatives has to be select to get a more reliable and refined design of the product in consideration of product design aspects of any component. For this reason, the product design of the mower is included in this work starting from the development of the functional structure of the mower conceptual design up to the selection of the best alternative concepts for each of its components among their available competitors, and the analytical mechanical design details are also included for the product design at hand to determine the dimensional parameters of the mechanical parts of the mower.

Moreover, for the purpose of manufacturing aspect, design of a mechanical component needs to incorporate graphical representation, and the orthographical representations of the mower are prepared in assembled and parts drawing using the graphical application software AutoCAD2007.

Key words: Mowing Component, Cash Crops, Product Design, Mechanical Design, Orthographical Representation, and AutoCAD2007

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1. Introduction

Sesame, Sesame indicum is a flowering plant in the genus Sesamum. Numerous wild relatives occur in Africa and a smaller number in India. It is widely naturalized in tropical regions around the world and is cultivated for its edible seeds, which grow in pods.

It is an annual plant growing to 50 to 100 cm tall, with opposite leaves 4 to 14 cm long with an entire margin; they are broad lanceolate, to 5 cm broad, at the base of the plant, narrowing to just 1 cm broad on the flowering stem. The flowers are white to purple, tubular, 3 to 5 cm long, with a four-lobed. The fruit is an oblong, mucronate, pubescent capsule containing numerous small, oval, and yellow, white, red, brown, or black seeds .[1]

In agriculture, a cash crop is a crop which is grown for profit. The term cash crop is used to differentiate from subsistence crops, which are those fed to the producer's own livestock or grown as food for the producer's family. In earlier times cash crops were usually only a small (but vital) part of a farm's total yield, while today, especially in the developed countries, almost all crops are mainly grown for cash. In non-developed nations, cash crops are usually crops which attract demand in more developed nations, and hence have some export value. Therefore, as long as cash crops are grown solely for sale rather than for the farmer's own use, for example, coffee, cotton, sesame or sugar beet, many developing world countries grow these crops to meet their debt repayments rather than grow food for their own people.

In many tropical and subtropical areas, jute, coffee, cocoa, sugar cane, bananas, oranges, oil crops and cotton are common cash crops. In cooler areas, grain crops, oil-yielding crops and some vegetables predominate; an example of this is the United States, where corn, wheat, soybean are the predominant cash crops. Similarly, the most popular oil seed in Ethiopia is sesame, and almost the entire yearly product is exported to the developed countries as a cash crop. In general, sesame is one of the well known oil crops among the cash crops all over the world as it is grown in many parts of the world on over 20,000 km^2. The largest producer of the crop in 2007 was India, followed by China, Myanmar, Sudan, Ethiopia, Uganda and Nigeria, and by this year, sesame output is the largest in the eastern African country’s oil seed production mainly in Ethiopia.

With a rapidly growing export performance in recent years, destined for markets in China, Japan, Korea, Israel and Turkey, analysts say Ethiopia is poised to become one of the top two leading sesame exporting countries in the world, and currently occupying the fourth position on the global market after China, India, and Burma, respectively, the country exports almost all of its sesame production.

This year, Ethiopia aims to export "more than 720,000 tons of oilseeds, pulses and spices" which translates into some "$715 million in earnings, a 94 per cent increase in the previous year's revenue", the Ethiopian Oil Seeds, Pulses and Spices Exporters Association has said [2].

However, farming in Ethiopia is still practiced in a primitive ways of cultivation over wider areas of the country. This custom results in minimal productivity of the farmers that cannot at least balance the demand of the population and market, that doesn't attain a consummate foreign currency and decrease the initiation of the cultivators to farm up to exploiting the maximum production potential of the available land and water recourses, on account of the existing labor intensive and poor farming technology.

Hence, to ameliorate the problem associated with this custom, in the purpose of mechanizing the agriculture, designing and manufacturing alternative machineries and farm equipments such as mowing component, thresher, combiner, plowing machine, water pumps, etc. are tasks of utmost priority with provisions of appropriate mowing technology means.

Therefore, in this thesis, it is intended to design mowing component of sesame and of course other similar cereals with better qualities and features providing an exceedingly simple mechanism for cutting purpose so that the problems bulleted below connected with harvesting sesame will be ameliorated easily and the uses of sesame will be exploited to the lead of sesame production in the world and secure the self-sufficiency of the country in food. Hence, it is very evident and very reasonable to give due attention to the cultivation of sesame in Ethiopia. It is on account of these bases that it has been intended to design a sesame mower component.

2. Problem Formulation

2.1. Problems of Harvesting Sesame:

This session is to define the problems associated with sesame harvesting which are seeking for solution and the concern of this work:

- Sesame plant has to be harvested within 3 to 4 days before the first killing frost come about so that it is possible to obtain high quality sesame oil seeds.

- At maturity, the leaves and seeds will begin to fall off the plants so that it will be lost through the shading since sesame grows in the pods which burst at maturity to lose the seeds.

- Currently in Ethiopia, sesame is harvested manually by using too much labor power and labor cost, and it has been said verbally by an owner that about 80% of the total expense for the plantation, no matter how much wide the plantation is, is required for labor cost to mow a matured sesame plantation in a season, and the cost of daily laborers is highly significant based on the 80% figure about the expense of the plantation.

- There is scarcity of man power required to mow the sesame with the allowable time frame of mowing.

- For matured sesame plant, the seeds are sensitive to be shade on shaking during mowing or harvesting.

2.2. Goals of the work:

This work is to be done on the focus of the following goals:

- Enhancing the cultivators for better productivity.
- To avoid injuries associated with legendry sickle blades operated by hands and labor intensiveness of the commonly used type of mowing trends.
- To make low cost mowing component available on market.

- Facilitating the development goals of the country as far as agriculture is one of the economic sectors of the country to ensure the renascence of Ethiopia in the new millennium through enhancing the production of the **cash crops**.
- To give some sort of contribution to guarantee the self-sufficiency of the country in food.
- Semi-mechanizing sesame harvesting.

Therefore, the need to design this mower is recognized for it is one of the optional ways to meet the above goals and mitigate the problems associated with harvesting sesame.

2.3. Specification Scope

The product design specification acts as a control for the total design activity, because it sets the boundary conditions for the subsequent design. Based on the requirements needed by the end users of the mower, the goals to be meet and the problems to be mitigated associated with harvesting sesame, the following specifications are set:

- To prepare design of a sesame mowing component that can be attached to and driven by any suitable moving device
- Designing a mowing mechanism with the possible lesser cost of manufacturing.
- Designing a mowing component with lesser number of parts for minimal complexity and relatively compact in volume.
- Designing a mowing component for sesame and other similar cereals with good operability and maintainability.
- Designing a mower that can mitigate the problems associated with sesame harvesting problems.
- Designing a mower with a capacity of mowing to cover an area of 1.5m width at a time.
- Designing a mower with a power requirement of 7 hp.

However, the specification listed above cannot be fixed to be the final until:

- A prototype model is designed, manufactured, and tested
- Cost analysis is done and feedback is obtained after the equipment is put in to use.

3. Literature Review

3.1. Background Information

Sesame has diversity of uses such as food, industrial, nutraceutical, and pharmaceutical. Below are the common uses of sesame and genetic resource places all over the world.

3.1.1. Genetic Resources

Japan uses sesame seed as a health food and leads the world in sesame seed imports followed by Europe and the US. About 70% of the world's sesame seed is processed into oil and meal. White sesame seed is imported from Mexico, Guatemala, and El Salvador while black sesame seed comes from China and Thailand.

Total annual consumption is about 65% for oil extraction and 35% for food. The food segment includes about 42% roasted sesame, 12% ground sesame, 36% washed sesame, and 10% roasted sesame seed with salt. People generally consume more than twice as much white sesame as black sesame.

Sesame oil is also referred to as teel oil or benne oil and is a pale yellow, oily liquid, and almost odorless with a bland taste. The oil consists of glycerides with about 43% oleic and linoleic each, 9% palmitic, and 4% stearic fatty acids. Sesame genetic resources conserved at the USDA, Plant Genetic Resources Conservation Unit represents 869 available accessions from countries worldwide as it is shown in Table 3.1.

Table 3.1: **Sesame genetic resources conserved for research use at the PGRCU.**

Number of accession	Country of origin	Number of accessions	Country of origin	Number of accessions	Country of origin
15	Afghanistan	3	Italy	1	Somalia
5	Angola	16	Japan	1	South America
8	Argentina	2	Jordan	1	Sri Lanka

2	Cameroon	2	Korea	4	Sudan
52	China	1	Liberia	2	Syria
14	Egypt	3	Libya	5	Taiwan
2	Ethiopia	29	Mexico	1	Tanzania
51	Former Soviet Union	11	Mozambique	163	Turkey
14	Greece	9	Myanmar	119	United States
3	Guatemala	12	Nepal	28	Venezuela
154	India	1	Nicaragua	1	Virgin Islands
26	Iran	4	Nigeria	1	Yugoslavia
9	Iraq	17	Pakistan	5	Zaire
26	Israel	1	Philippines		

3.1.2. Food and industrial Uses

There are many foods in which sesame is an ingredient in Table 3.2. Europeans sometimes use it as a substitute for olive oil. Sesame oil is an excellent salad oil and is used by the Japanese for cooking fish. Aqua hulled sesame seeds undergo a special hulling process which produces a clear white seed. These seeds are then double washed, dried, and used on hamburger buns. This special process allows the seed to stick to the bun while maintaining a white color after baking. About one-third of the imported crop from Mexico is purchased by McDonalds for their sesame seed buns. The seeds are also used on bread and then eaten in Sicily. Sesame seed has a nutty taste when the seed is roasted. Bread, breadsticks, cookies, chocolate, and ice cream are ideal products for roasted natural sesame seed. In Greece seeds are used in cakes, while in Togo, Africa seeds are a main soup ingredient. Mechanically hulled sesame seed enriches bakery and candies plus it is also the basis for the creamy, sweet wholesome tahini. Tahini is rich in protein and a very good energy source. Sesame flour is an edible, creamy and light brown powder from sesame seeds.

Sesame flour has high protein, high levels of methionine and tryptophan, and 10% to 12% sesame oil. Sesame seeds contain three times more calcium than a comparable measure of milk.

Table 3.2:**Sesame food uses.**

Food	Country
Sesame cakes, wine, and brandy	Biblical Babylon
Bread stick, cracker	Worldwide
Salad and cooking oil	Worldwide
Roasted seed	India
Substitute for olive oil	Europe
On bread	Sicily
Cakes	Greece
Soup, spice, seed oil	Africa
Salad and fish oil	Japan
Confectionery	China
Sesame seed buns, chips	United States

Refined sesame oil is a fine oil because of its antioxidant properties allowing for greater shelf life plus improving its flavor and taste for use in the food industry. Roasted sesame oil resists rancidity due to the antioxidants formed during seed roasting and the particular roasted sesame flavor improves taste of fried products. African countries use the seeds as spice, seed oil, frying vegetables and meat, eaten raw or fried, and used in confections such as candy and baking. Sesame honey bits are made by combining hulled sesame seed with honey from a cactus flower in Mexico. Additional products sold in US grocery and health food stores with sesame seed as an ingredient include sesame crackers, honey puffed kashi seven whole grains and sesame cereal, sesame blues chips, unhulled sesame seed, and sesame seed candy. Many recipes contain sesame seeds as an ingredient such as sesame seed sprouts, sesame spread, tangerine and sesame, sesame seed cookies, hummus, sesame seed bagels, sesame granola, sesame broccoli rice, sesame mustard sauce, ginger sesame chicken, sesame pastry, sesame seed sauce, and sesame green beans. Sesame meal is excellent feed for poultry and livestock.

Several industrial uses of sesame have been identified in Table 3.3. African people have used sesame to prepare perfumes and cologne has been made from sesame flowers. Myristic acid is used as an ingredient in cosmetics. Sesamin has bactericide and insecticide activities plus an antioxidant which can inhibit the absorption of cholesterol and the production of cholesterol in the liver. Sesamolin also has insecticidal properties and is used for the production of insecticides. Sesame oil is used as a solvent, skin softener, and used in the manufacture of margarine and soap. Chlorosesamone obtained from roots of sesame has antifungal activity.

Table 3.3: **Sesame industrial, nutraceutical, and pharmaceutical uses.**

Use	Phytochemical
Industrial	
Antifungal	Chlorosesamone
Bactericide, insecticide	Sesamin, sesamolin
Cosmetics	Myristic acid
Solvent, soap	
Nutraceutical	
Antioxidant, hepatoprotective	Lecithin
Cancer preventive	Myristic acid
Cancer preventive, cardioprotective	Fiber
Fatty acid oxidation, antioxidant	Sesamin, sesamolin
Prevent heart disease	Sesame oil
Skin softener	
Pharmaceutical uses	

Drug vehicle and laxative	Sesame oil
Hypoglycemic activity	Flavonoids

3.1.3. Nutraceutical and Pharmacceutical Uses

Many nutraceutical uses, as in Table 3.3, have been discovered from sesame. Sesame lignans have antioxidant and health promoting activities. High amounts of both sesamin and sesamolin have been identified in sesame. Both sesamin and sesamolin were reported to increase both the hepatic mitochondrial and the peroxisomal fatty acid oxidation rate. Sesame seed consumption appears to increase plasma gamma-tocopherol and enhanced vitamin E activities which are believed to prevent cancer and heart disease. Sesamin remained at 90% of the original level after roasting.[3]

Historically, fiber is used as an antidiabetic, antitumor, antiulcer, cancer preventive, cardioprotective, and laxative. Fiber ranges from 27,100 ppm to 67,000 ppm in the seed with up to 166,000 ppm in the leaf. Sesame seed contains lecithin which has antioxidant and hepatoprotective activity and ranges from 58 ppm to 395 ppm. Lecithin is also likely effective for reducing hepatic steatosis in long term parenteral nutrition patients and a successful treatment for dermatitis and dry skin. Several pharmaceutical uses of sesame have been identified as shown in Table 3.3. Myristic acid also has cancer preventive capability and is found in sesame seed ranging from 328 to 1,728 ppm. [5]

Sesame oil is a pharmaceutic aid used as a solvent for intramuscular injections and has nutritive, demulcent, and emollient properties and has been used as a laxative.[9] The oil was used during the 4th century by the Chinese as a remedy for toothaches and gum disease. Sesame oil is known to reduce cholesterol due to the high polyunsaturated fat content in the oil. Other uses include the treatment of blurred vision, dizziness, and headaches. The Indians have used sesame oil as an antibacterial mouthwash, to relieve anxiety and insomnia.[4] A recent clinical trial proved that sesame oil was significantly more effective for treating nasal mucosa dryness due to a dry winter climate than isotonic sodium chloride solution. In addition, sesame oil contains large amounts of linoleate in triglyceride form which selectively inhibited malignant melanoma growth. Additional ethnobotanical uses of sesame are shown in Table 3.4. [6]

Table 3.4: **Sesame ethno botanical uses.**

Use	Country
Cancer	Germany
Cold	Dominican Republic
Colic	Haiti
Constipation, head cold, impotency, laxative, tonic, malaria, cold preventive, cancer, diarrhea, sore, venereal, wart	China
Cough	Venezuela
Dysentery, laxative	Turkey
Laxative	Mexico
Tonic	Malaya
Tumor	India

Generally, the uses of sesame are summarized as bulleted below:

- Sesame seeds, approximately 50% oil and 25% protein, are used in baking, candy making, and other food industries.
- Oil from the seed is used in cooking and salad oils and margarine, and contains about 47% oleic and 39% linoleic acid.
- Sesame oil and foods fried in sesame oil have a long shelf life because the oil contains an antioxidant called sesamol.
- The oil can be used in the manufacture of soaps, paints, perfumes, pharmaceuticals and insecticides. Sesame meal, left after the oil is pressed from the seed, is an excellent high-protein (34 to 50%) feed for poultry and livestock.

The following pictures are some of the products of sesame seeds:

Turkish bread with sesame seeds

A hamburger with a sesame seed roll

Tuna with toasted **sesame seed**

Sesame product of Olson Food Company

Sesame with vegetables and Spaghetti

Simple **sesame** cookies

Roasted **Sesame Seeds**

Black **sesame** powder

Sesame seed used for candy production

Black **Sesame Seed** Butter

Sesame Seed Oil

Chocolate of Sesame

Figure 3.1: **Products of Sesame seed**

3.2. Mower

A **mower** is a machine for cutting crops or plants that grow on the ground. A smaller mower used for lawns and sports grounds (playing fields) is called a **lawn mower** or grounds mower, which is often self-powered, or may also be small enough to be pushed by the operator. Grounds mowers have rotary or reel cutters. Larger mowers are used to cut hay or other crops and place the cut material into rows, which are referred to as windrows. Often, such mowers are called windrowers or mower-conditioners. Swathers are also used to cut hay and grain crops. Prior to the invention and adoption of mechanized movers, (and today in places where use of a mower is impractical or uneconomical), hay and grain was cut by hand using scythe or sickle.

3.3. Mower configurations

Larger mowers are usually ganged (equipped with a number or gang of similar cutting units), so they can adapt individually to ground contours. They may be powered and drawn by a tractor or draft animals. The cutting units can be mounted underneath the tractor between the front and rear wheels, mounted on the back with a three-point hitch or pulled behind the tractor as a trailer. There are also dedicated self propelled cutting machines, which often have the mower units mounted at the front and sides for easy visibility by the driver. **Boom** or **side-arm** mowers are mounted on long hydraulic arms, similar to a backhoe arm, which allows the tractor to mow steep banks or around objects while remaining on a safer surface. [14]

3.4. Mower types

The cutting mechanism in a mower may be one of the several types of different design as it follows:

3.4.1. Sickle mower

Sickle mowers, also called reciprocating mowers, bar mowers, or finger-bar mowers, have a long (typically six to seven and a half feet) bar on which are mounted fingers with stationary guard plates. In a channel on the bar there is a reciprocating sickle with very sharp sickle sections (triangular blades). The sickle bar is driven back and forth along the channel. The grass, or other plant matter, is cut between the sharp edges of the sickle sections and the finger-plates (this action can be likened to an electric hair clipper). The bar rides on the ground, supported on a skid at the inner end, and it can be tilted to adjust the height of the cut. A spring-loaded board at the outer end of the bar guides the cut hay away from the uncut hay. The so-formed channel, between cut and uncut material, allows the mower skid to ride in the channel and cut only uncut grass cleanly on the next swath. These were the first successful horse-drawn mowers on farms and the general principles still guide the design of modern mowers. [14]

Figure 3.2: Sickle bar mower

3.4.2. Rotary mower

Rotary mowers, also called **drum mowers**, have a rapidly rotating bar, or disks mounted on a bar, with sharpened edges that cut the crop. When these mowers are tractor-mounted they are easily capable of mowing grass at up to 20 miles per hour (32 km/h) in good conditions. Some models are designed to be mounted in double and triple sets on a tractor, one in the front and one at each side, thus able to cut up to 20 foot (6 meter) swaths. In rough cutting conditions the blades attached to the disks are swiveled to absorb blows from obstructions. Mostly these are rear-mounted units and in some countries are called **scrub cutters**. Self-powered mowers of this type are used for rougher grass in gardening and other land maintenance. [14]

Figure 3.3: Rotary cutters mounted on the Windrower shown above.

3.4.3. Reel mower

Reel mowers, also called **cylinder mowers** (familiar as the hand-pushed or self-powered cylinder lawn mower), have a horizontally rotating cylindrical reel composed of helical blades, each of which in turn runs past a horizontal cutter-bar, producing a continuous scissor action. The reel runs at a speed dependent on the forward movement speed of the machine, driven by wheels running on the ground (or in self-powered applications by a motor). The cut grass may be gathered in a collection bin. This type of mower is used to produce consistently short and even grass on bowling greens, lawns, parks and sports grounds. When pulled by a tractor (or formerly by a horse), these mowers are often ganged into sets of three, five or more, to form a **gang mower**. A well-designed reel mower can cut quite tangled and thick tall grass, but this type works best on fairly short, upright vegetation, as taller vegetation tends to be rolled flat rather than cut. [14]

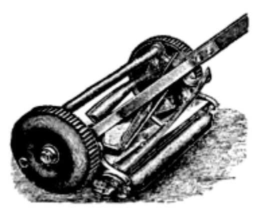
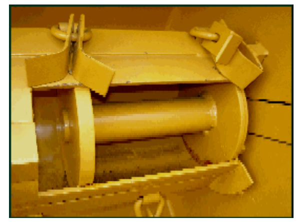


Figure 3.4: Reel Mowers

3.4.4. Flail mower

Flail mowers have a number of small blades on the end of chains attached to a horizontal axis. The cutting is carried out by the ax-like heads striking the grass at speed. These types are used on rough ground, where the blades may frequently be fouled by other objects, or on tougher vegetation than grass, such as brush or scrub. Due to the length of the chains and the higher weight of the blades, they are better at cutting thick brush than other mowers, because of the relatively high inertia of the blades. In some types the cut material may be gathered in a collection bin. As a reel boom mower of the above type, a flail mower may be used in an upright position for trimming the sides of hedges, when it is often called a **hedge-cutter.** [14]

Wide variety of knives & blades



Heavy duty cutter drums

Figure 3.5: Flail mower

4. Product Design

4.1. Introduction:

Product design may comprise of idea generation, concept development, testing and implementation of a physical objector product or service. Product Designers conceptualize and evaluate ideas, making them tangible through products in a more systematic approach. The role of a product designer encompasses many characteristics of the marketing manager, product manager, industrial designer and design engineer, and combines art, science and technology to create tangible three-dimensional goods. This evolving role could be facilitated by digital tools that allow designers to communicate, visualize and analyze ideas in a way that would have taken greater manpower in the past.

Product designers are equipped with the skills needed to bring products from conception to market. They should have the ability to manage design projects, and subcontract areas to other sectors of the design industry. Aesthetics is considered important in Product Design but designers also deal with important aspects including technology, ergonomics, usability, stress analysis and materials engineering.

As with most of the design fields the idea for the design of a product arises from a need and has a use. It follows a certain method and can sometimes be attributed to more complex factors such as association and telesis. Also used to describe a technically competent product designer or industrial designer is the term Industrial Design Engineer.

As an application of product design, some companies or individuals have particularly strong feel for developing new products than others. In the modern world these include especially technological companies. Many product designers are strategic assets to companies that need to maintain a competitive edge in innovation.

4.2. Conceptual Design

Conceptual design is part of a product design and used to determine the elements mechanisms, processes, configurations that in some combination or other result in a design that satisfies the need to make up the whole system. Conceptual design is the early stage of the design where the main decisions are made by means of design for manufacturing (DFM), as a rough idea is developed as to how a product will function and what it will look like and the best is selected based on decision theories for further design analysis.

4.3. Functional Modeling

Modular product architectures can reduce the number of parts in a product, reduce the time to manufacture and assemble the product, and streamline and simplify the conceptual design and embodiment design phases through the re-use of previous parts or ideas.

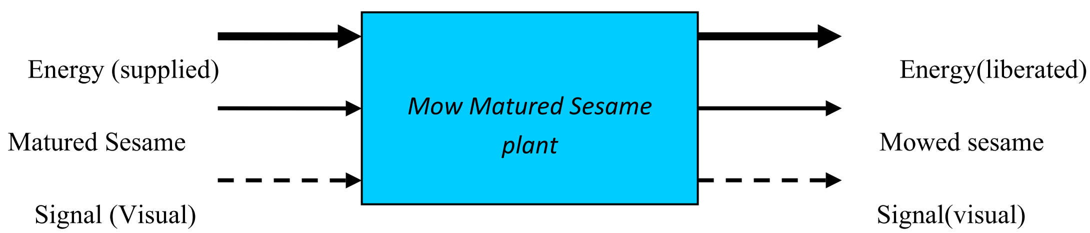


Figure 4.1: Overall Function of the mowing component

To completely describe product functionality, the overall production is decomposed into a set of sub functions. These sub functions provide a detailed description of what a product most do rather than what it is, which is called the functional structure of the design.

The figure below (**Fig.4.2**) is the functional structure of the design of the mowing component.

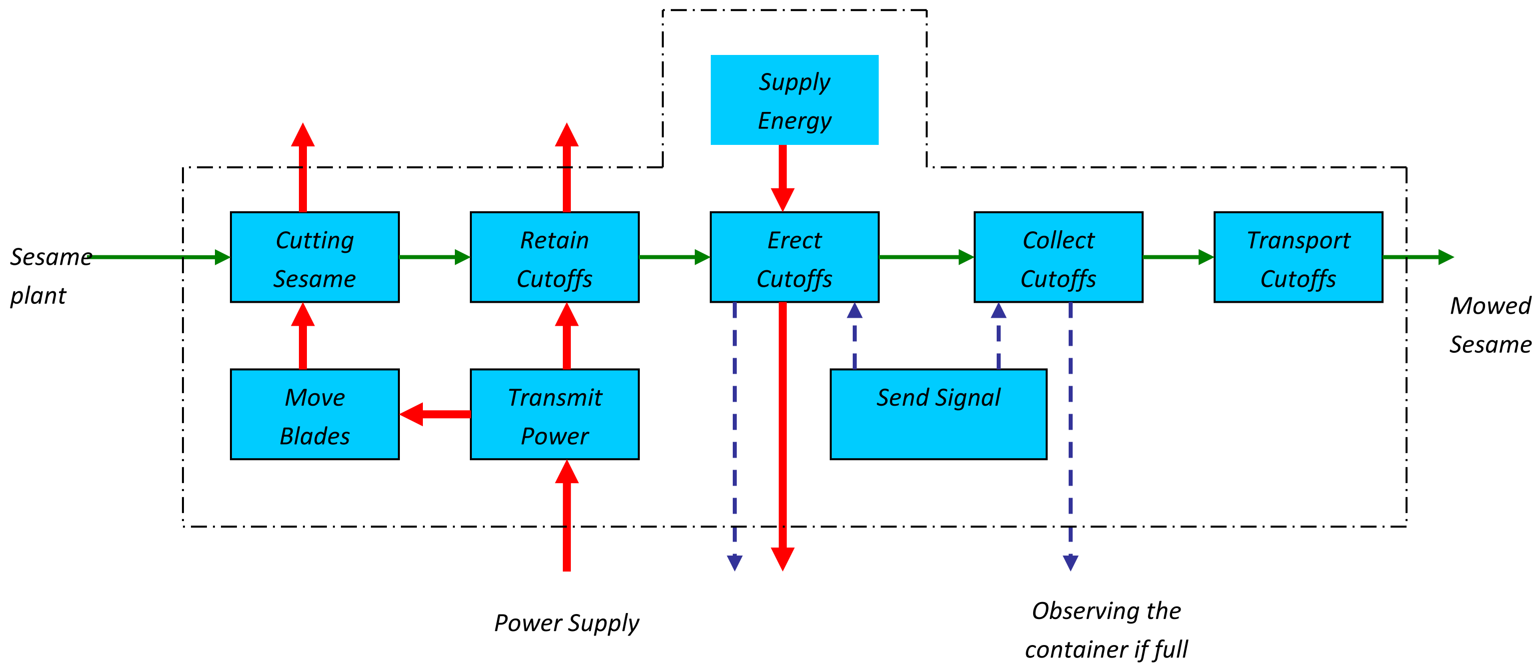


Figure 4.2: Functional Structure showing the sub functions

Key to Figure 2 and 3:

Energy Flow direction

Material Flow (Mowed sesame)

Signal flow (visual detection of the components of the machine)

System Boundary

Main Functions

Auxiliary Functions

4.4. Design Alternatives

There are a number of different design alternatives of mower that can fulfill the requirements of harvesting sesame. Below are presented four different design alternatives, which are going to be selected one as a best.

- Alternative A- Rotary Drum Mower
- Alternative B-Sickle-bar Mower
- Alternative C-Self-propelled reaper Harvesting Machine
- Alternative D- Sickle hand tool

➢ **Alternative A- Rotary** Drum **Mower**

The Drum Mower chops every type of grass herbs or different pasture plants with high speed. Because the discs which are very close to the ground, it increases the productivity. It chops the herbs in such a way that they dry easily and it is very easy to collect them afterwards. By the help of the knifes that are turned on to the mower, it gets the move from the P.T.O. shaft and makes so the chopping work. [12]

Figure 4.3: Alternative A- Rotary Drum Mower

- **Alternative B-Reciprocating Cutter Mower(sickle bar mower)**

It is a multi-purpose yard maintenance apparatus consisting of a reciprocating cutter mower, a sickle-bar hedge trimmer and a garden tiller. The apparatus comprises a gasoline engine and a handle assembly having a pair of handles. The apparatus further comprises a sickle-bar frame assembly having handle mounting means, sickle-bar hedge trimmer mounting means and a pair of wheels. Also included is a tiller housing assembly having handle mounting means, engine mounting means and a set of tines Finally, a sickle-bar hedge trimmer assembly having a housing, a engine mounting means, a pair of handles and a sickle-bar cutting assembly is provided to complete the components of the apparatus. In use the tiller housing assembly, the gasoline engine, and the handle assembly may be used separately as a tiller or the hedge trimmer assembly and the gasoline engine may be used separately as a hedge trimmer or the hedge trimmer assembly, the gasoline engine and the handle assembly may be used in combination with the sickle-bar frame assembly as a Reciprocating Cutter Mower.

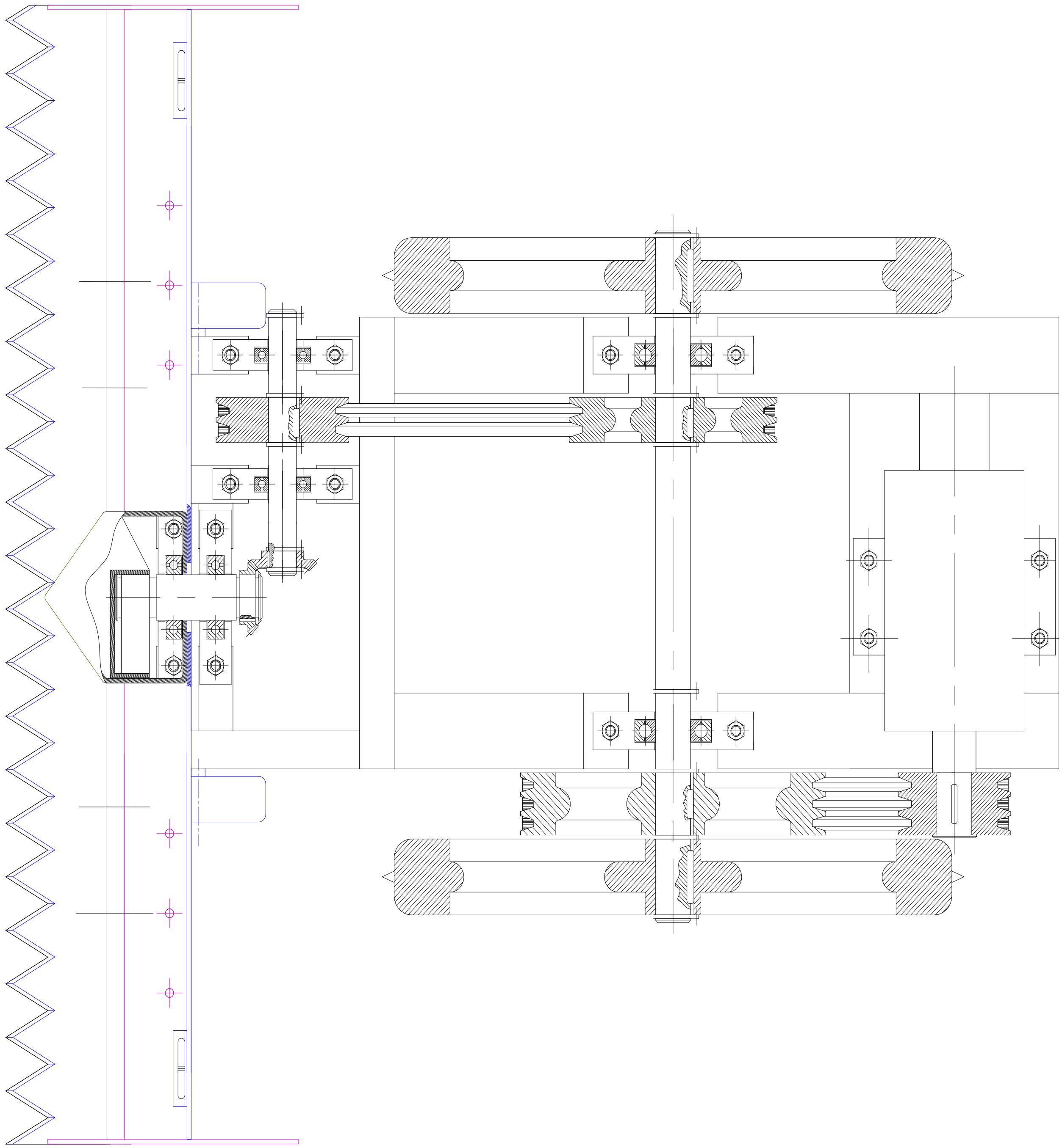


Figure 4.4: Alternative B- Reciprocating Cutter Mower (sickle bar mower) as viewed from top

- **Alternative C-Self-propelled sesame harvester**

There are several models of this machine different for dimensions, power sources and performances but all consist of a cutter bar, a crop dividers, a crop conveyor belt and of a series of star wheels: the first two elements are involved in the plant cutting while the belts and the star wheels are used to keep the crop standing upright as it is moved to the right to form a windrow. The plant stems are cut by the cutter bar near the soil surface placed in a windrow and then gathered. [13]

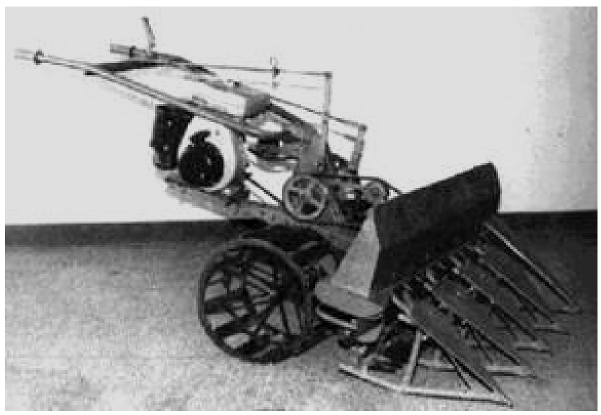


Figure 4.5: Alternative C-Self-propelled sesame harvester

- **Alternative D-Sickle hand tool**

A Sickle is one of the most ancient of harvesting tools, consisting of a metal blade, usually curved, attached to a short wooden handle.

It is a one hand-held agricultural tool with a curved blade typically used for harvesting grain crop or cutting grass for hay. The inside of the curve is sharp, so that the user can draw or swing the blade against the base of the crop, catching it in the curve and slicing it at the same time. The material to be cut may be held in a bunch in the other hand for example when reaping, held in place by a wooden stick, or left free. When held in a bunch, the sickle action is towards the user as left to right for a right-handed user, but when used free the sickle is usually swung the opposite way. Different types may be referred to as a grass hook, swap hook, rip-hook, slash-hook, reaping hook, brushing hook or bagging hook.

The blade of a sickle is often cranked to one side, to make it easier to keep the blade closer to the ground; this makes it right- or more rarely left-handed. Sickles used for reaping are usually serrated. Harvesting with a sickle is very slow, but because of its simplicity and low cost, it is still widely used over the world, especially to reap cereals such as wheat, rice and sesame and also as a gardening tool. [13]

Figure 4.6. Alternative D-sickle hand tool

4.5. Design Alternatives Selection

To select one of the four alternatives given above, there are steps to be followed

1. Setting criteria.
2. Calculating a weighting factor for each criterion by digital logic method (DLM).
3. Evaluating each design with respect to the selected criteria by using a decision matrix and finally,
4. Selecting the best design based on the result of the decision matrix

4.6. Criteria for Alternatives selection

The type and number of criteria are determined by individual judgment. There are no properly set rules for setting design criteria, since it depends on the type and application of design and its complication. The criteria for selection by considering the alternative designs and the requirements set by the end users of the mowing component are listed below:

1. Design Simplicity
2. Cost of Manufacturing
3. Adjustability of cutting height
4. Seeds Retaining Capacity

5. Mowing Cost/ Labor cost
6. Availability of the machine on market
7. Ease of Maintainability
8. Stability
9. Operability
10. Multi functionality/ Utility capacity
11. Labor Alleviating/ Lessening Labor Intensiveness

4.7. Weighing Factor Determination

When many design criteria are used to specify the degree of importance of each against the other, it may be difficult to re-establish weighting factors. One way to do so is to use a digital logic approach. Each criterion is compared with the other in every combination taken two at a time. To make the comparison, the criterion that is considered to be the more important of the two is given a '**1**'and the less important a '**0**'. The total number of possible comparison pairs under consideration is given by:

$N = n (n-1)/2$

Where N= the total number of possible comparison pairs

n= the total number of criteria under consideration, in this case n =11.

And the weighing factor, $W_i = m_i/N$

Where m_i= the total number positive decisions for the i^{th} criteria.

Therefore, N =11 (11-1)/2 =55. The above steps are shown in the following table 4.1.

Table 4.1: **Determination of weighting factor**

Criteria	1	2	3	4	5	6	7	8	9	10	11	m_i	W_i
1	-	0	0	0	0	0	1	1	0	1	0	3	0.055
2	1	-	0	0	0	1	1	1	1	0	1	6	0.109
3	1	1	-	0	1	1	1	1	1	1	1	9	0.164
4	1	1	1	-	1	1	1	1	1	1	1	10	0.182
5	1	1	0	0	-	1	1	1	1	1	1	8	0.145
6	1	0	0	0	0	-	0	1	0	0	0	2	0.036
7	0	0	0	0	0	1	-	1	1	1	0	4	0.073
8	0	0	0	0	0	0	0	-	0	1	0	1	0.018
9	1	0	0	0	0	1	0	1	-	1	0	4	0.073
10	0	1	0	0	0	1	0	0	0	-	0	2	0.036
11	1	0	0	0	0	1	1	1	1	1	-	6	0.109

4.8. Selecting the best design

In table 4.1, the weighting factor has been determined for each criterion. A decision matrix approach, which is a clear and simple way to arrive at a design decision, will be employed for the purpose of selecting the best design among the available alternatives by using the results of the weighting factors. The next step will be preparing decision matrix for selecting the best design alternatives.

The overall degree of satisfaction in achieving each design criteria is evaluated by the use of scale, shown in table 4.2, which is prepared to be conforming to the requirements of the mowing component.

To select the best design from the alternatives, the steps to be followed are:-

1. Taking one design from the alternatives and evaluating with respect to the criteria.
2. Giving percent satisfaction from table 4. 2.

3. Multiplying the percent satisfaction by the weighting factor of each criterion respectively.
4. Adding the result of each criterion separately for each design, which will give the overall satisfaction of the given design.
5. Taking the next design from the alternatives and repeating step 2 through and until all the alternative designs is evaluated.
6. Compare and contrast the overall satisfaction of the design alternatives.
7. Select the best designs i.e. the one with the greatest overall satisfaction.

The above steps are shown in table 4.3 below.

Table 4.2: **Satisfaction for achieving the criteria in Percentage (%)**

Satisfaction (%)	Description
100	**Excellent**, Complete satisfaction, objective satisfied in every aspect
90	**Very Good**, Extensive satisfaction, objective satisfied in all of important aspect
75	**Good**, Considerable satisfaction, objective satisfied in the majority of aspects
50	**Fair**, Moderate satisfaction, a middle point bin complete and no satisfaction
25	**Bad**, Minor satisfaction, objective satisfied in some but less than half of the aspect
10	**Failure**, Minimal satisfaction ,objective satisfied to very small extent
0	No satisfaction, objective is not satisfied in any aspect

Table 4.3: **Decision matrix**

Criteria / Alternatives		1	2	3	4	5	6	7	8	9	10	11	Overal Satisfy
W_i		0.055	0.109	0.164	0.182	0.145	0.036	0.073	0.018	0.073	0.036	0.109	
A	%	25	25	10	10	90	25	50	100	50	25	90	41.32
	%* W_i	1.375	2.725	1.64	1.82	13.05	0.9	3.65	1.8	3.65	0.9	9.81	
B	%	90	70	100	90	90	90	90	100	90	100	90	90
	%* W_i	4.95	7.63	16.4	16.38	13.05	3.24	6.57	1.8	6.57	3.6	9.81	
C	%	10	10	0	90	90	25	25	100	25	25	90	48.13
	%* W_i	0.55	1.09	0	16.38	13.05	0.9	1.825	1.8	1.825	0.9	9.81	
D	%	90	50	50	70	10	90	90	70	70	70	10	
	%* W_i	4.95	5.45	8.2	12.74	1.45	3.24	6.57	1.26	5.11	2.52	1.09	52.58

As it can be seen from the decision matrix, table 4.3, the design alternative B overall satisfaction is the greatest of all the others, i.e., **90**. Therefore, based on the result, design B is selected for further design analysis.

4.9. Determination of the power drive system

The drive system is the basic part of the proper functioning of the system, because it serves as the rotational power source of the transmission system. The engine power is transmitted through various means of power drives, in this case, these may be sprocket or gear or belt drives which are selected based on decision theories, to the wheels shaft, the first gear shaft, and the cam shaft for giving thrust force to the mowing machine and the reciprocating cutting mechanism.

In this design, there is a fuel consuming engine to be used as a prime mover. To select the type of power drive for each section of the power transmission system, it is customary to use decision matrix approach which has been used in the design alternative concept selection before this subtopic.

4.9.1. Power Drive Alternatives of the engine-wheels drive for the function 'Transmit power'

The function 'Transmit power' in the functional diagram of the mowing component can be fulfilled in variety of patterns of transmission paths. There are a number of different design alternatives for the engine-wheels shaft drive that can fulfill the requirements of power transmission. Below are presented three different power drive design alternatives from the prime mover, which are going to be selected one as a best.

- Alternative A- Belt Drive
- Alternative B- Gear Drive
- Alternative C- Chain and Sprocket drive

These alternative power drives are illustrated with schematic representation as follows.

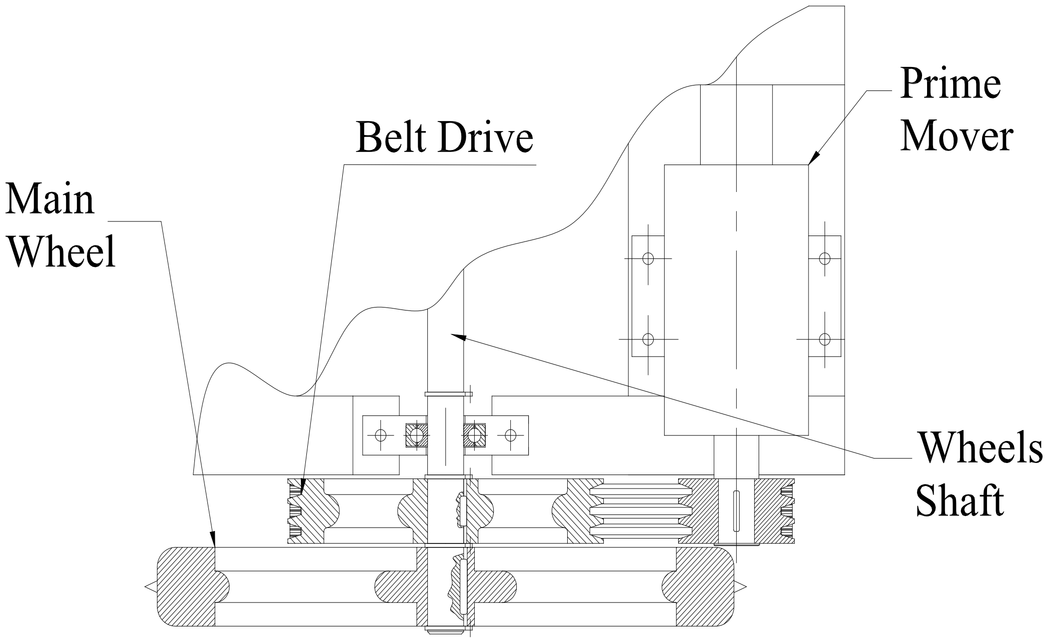


Figure 4.7: **Alternative A-Belt Drive**

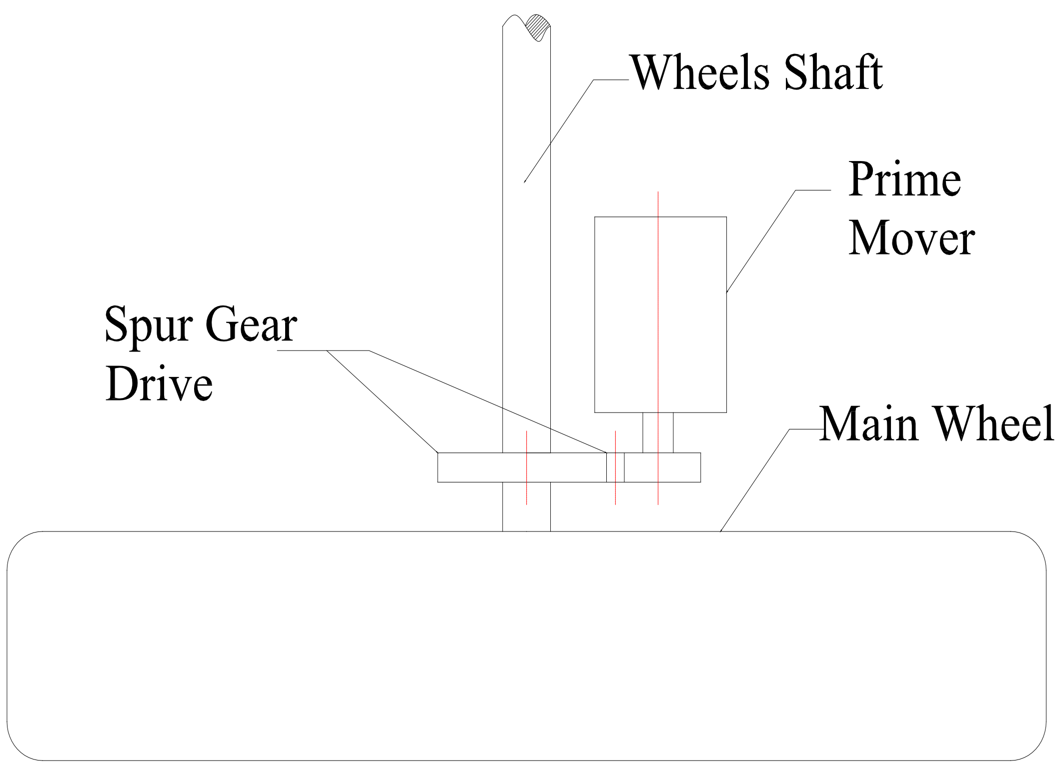


Figure 4.8: **Alternative B- Gear Drive**

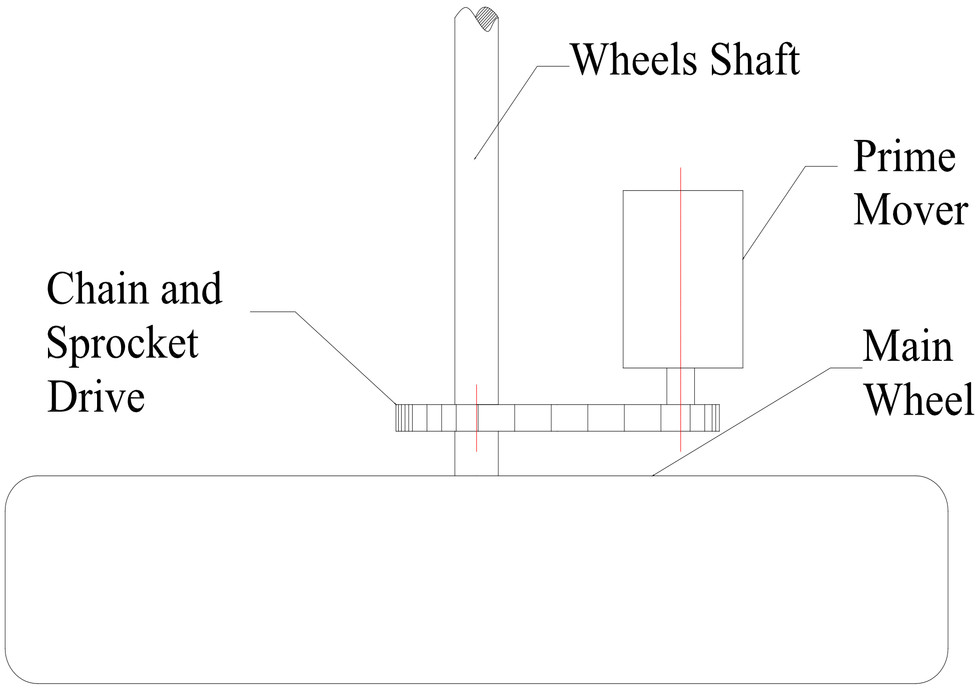


Figure 4.9: **Alternative C- Chain and Sprocket drive**

4.9.2. Power drive selection for the engine-wheels

The drive system will be selected by Johnson method, which is a more detailed type of decision matrix method that is appropriate for engineering design. The designer first decides on the attributes of the design that are most important to performance. One of the design alterative is selected as a reference and the attribute factors are normalized with respect to the reference. Usually the design for which there is the most prior experience is selected as the reference.

Attribute comparison rating may be either positive or negative in character. Wear rate or cost would be negative factors, but reliability or appearance would be classified as positive. As in the preceding decision matrix for the drive system, the design attributes are given weighting factors from 0 to 10. Johnson's weighting factors are given in table 4.4 and the design alternatives are evaluated by their features in table 4.5.

Table 4.4: Johnson's relative weighting scale for design attributes

Rating	Description
10	Extremely high importance
8-9	Very high importance
6-7	Above average importance
5	Average or moderate importance
3-4	Minor importance
1-2	Almost insignificant importance
0	No importance

Table 4.5: Decision matrix of the drive systems based on Johnson's technique

Criteria	Weighting factor	Attribute comparison rating for Design A	Attribute comparison rating for Design B	Attribute comparison rating for Design C
Compactness	6	Reference=0	1	-1
Ease of Manufacturing	8		-0.5	-2
Low cost	10		-2	-2
Simplicity of design	10		-2	-1
Efficiency	10		1	1
Durability	10		2	1
Ease of assembly	10		-1.5	-1
Ease of maintenanability	9		-2	-2
Reliability	9		2	1.5

Light weight	5		1	1
Overall Satisfaction	88		-9.5	-4
Weighting total		**0**	**-20**	**-43.5**

From the above table 4.5 it is evident that deign alternative A is the best solution among the others noting that the rating between -2 and 2 indicate that degree of fulfillment of the functionality of the alternatives with respect to the reference alternative A. Therefore, a belt drive is selected as a power drive from the engine to the wheels shaft.

4.9.3. Belt Type Selection for the engine-wheels shaft power transmission

Belt drives are used to transmit torque and angular velocity in a wide variety of applications. There is also a variety of belt types to choose from, such as v-belt, flat belt, timing belt or a combination of these. The selection of the suitable belt type for this design problem will depend on criteria tabulated as follows.

Table 4.6: **Decision matrix for the type of belt drives selection.**

Criteria	V-belt Rating	Flat Belt Rating	Timing belt Rating
Ease of Design	9	10	7
Compactness	10	3	10
Slip Reduction	9	5	10
Cost	10	4	3
Safety	10	4	10
Efficiency	8	5	10
Ease of assembly	9	8	9
Availability locally	10	8	6
Overall Satisfaction	**75**	**43**	**65**

Note that in this table the values are taken from table 4.4. Therefore, from the evaluation in the above table 4.6, v-belt is selected as it has got higher overall Satisfaction of 75 and it is determined to be designed.

4.9.4. Power Drive determination of the wheels-cutting system drive (the second power drive)

Once the rated power is transmitted to the wheels of the mowing component, the required amount of power for cutting system must be given to the other working components through appropriate means of power drives. In this mowing component, the power from the engine is to be used for two major functions, one for the propelling action to achieve the thrust force of the machine and the other for the cutting system of reciprocating mechanism. A belt drive is already selected to get the propelling action of the engine to transmit the rated power 7hp in to the wheels of the machine so that part of the power on the wheels will be used for the traction force of the machine and the rest to the cutting system and others like frictional losses.

Therefore, power drives need to be selected to transmit power from the wheels to the cutting system of the machine. The power on the wheels of the machine is rotational and this rotational motion has to be changed into linear motion of the cutting system to achieve a shearing action on the plant. To get this transformation from rotational to linear motion, a cam mechanism can be used, but for there is space limitation, a drive that can give the relief of space limitation better than cam mechanism to take the power first from the wheels has to be used. The possible alternative drives to take power first from the wheels are the belt, chain and gear drives. These alternative drives are illustrated by the following figures.

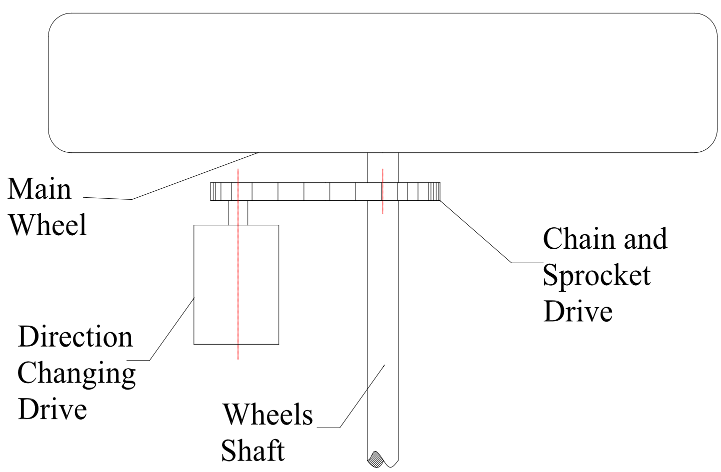


Figure 4.10: **Alternative A-Chain Drive**

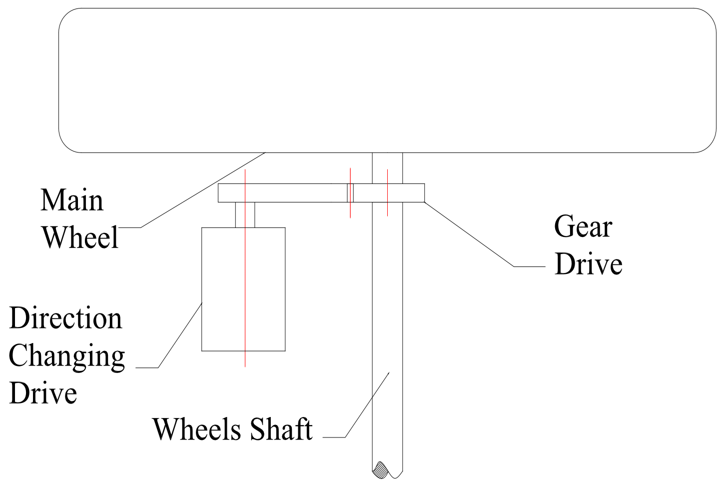


Figure 4.11: **Alternative B: Gear Drive**

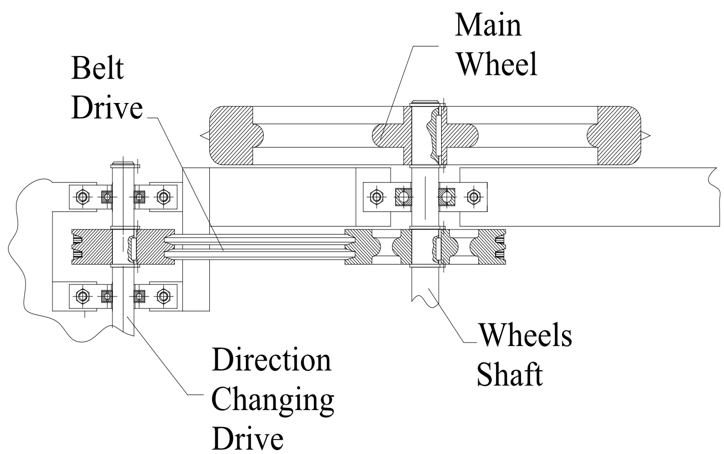


Figure 4.12: **Alternative C: Belt Drive**

4.9.5. Power drive selection for the wheels-cutting system drive

In a similar pattern to the selection of the power drive for the engine-wheels drive, the Johnson's decision matrix will be used under here, and the weighting factors for the given design alternatives are tabulated as follows.

Table 4.7: **Decision matrix of the wheels-cutting system drive based on Johnson's technique**

Criteria	Weighting factor	Attribute comparison rating for Design C	Attribute comparison rating for Design B	Attribute comparison rating for Design A
Compactness	7	Reference	-2	-1
Cost	10		-0.5	-1
Simplicity of design	10		-1.5	-2
Efficiency	9		1	1
Durability	9		1	1
Slip Reduction	9		2	2
Ease of maintenance	10		-2	-2
Ease of Assembly	9		-1	-1
Transmission Shaft Axis changeability	10		-2	-1.5
Availability on Market locally	10		-0.5	-1
Overall Satisfaction	93		-4	-4
Weighting total		**0**	**-60**	**-59.5**

Based on the result from the above table, the **alternative C** which is a belt drive is selected. Here, as long as there are a number of belt types, a further concept selection analysis for the type of the belt to select the best belt type that suits the requirements of the working conditions is required.

The optional belt drives are of V-belt, flat belt, and timing belt types among which one will be selected as the best of all. The selection criteria for the best belt type and the weighting of each belt types are taken to be similar to those of the engine-wheels belt drive selection, and are depicted by the following table.

Table 4.8: **Decision matrix of the wheels-cutting system drive for the belt type selection.**

Criteria	V-belt Rating	Flat Belt Rating	Timing belt Rating
Noise Reduction	7	10	7
Compactness	10	3	10
Slip Reduction	9	5	10
Cost	10	4	3
Safety	10	4	10
Efficiency	8	5	10
Availability locally	10	8	6
Overall Satisfaction	**64**	**39**	**56**

Then, the **V-belt type** is selected to be designed based on the criteria used and the relative weighting allotted to each of the alternatives.

4.9.6. Determination of the rotation axis changing power drive

The power of the cutting system gained from the propelling wheels of the machine is determined to be transmitted by V-belt drive in the above concept selection analysis, and hence the power received from the wheels is rotational as long as it conveyed by belt drive. Therefore, there is a need to change the direction of rotation axis of the transmission shafts by 90^0 in horizontal plane and for this reason, an appropriate power drive which is capable of fulfilling these power transmissions and direction changes of shafts rotation axes requirements should be chosen from the available means.

The available means as alternatives are gear, belt, chain drives, and cam mechanisms. But a chain drive and cam mechanisms are not suited to be selected for they are not used for axes changing purposes and ease of design view point, respectively. Therefore, it is insisted to select among the belt and gear drives. The selection approach is as the previous pattern using the Johnson's technique of concept selection.

The Johnson's decision matrix is given in the following table showing the weighting factors for the given design alternatives

Table 4.9: **Decision matrix of the rotation axis changing power drive selection**

Criteria	Weighting factor	Attribute comparison rating for Gear Drive	Attribute comparison rating for Belt Drive
Space Requirement	10	Reference	-2
Safety	7		-2
Slip Reduction	10		-2
Cost	6		1
Ease of design	8		1.5
Ease of maintainability	7		1
Efficiency	10		-1.5
Ease of Assembly	8		1
Availability on local market	10		0.5
Transmission Shaft Axis changeability	10		0
Over all Satisfaction	86		-2.5
Total Weighting		**0**	**-31**

Therefore, the reference alternative which is gear drive is selected to be the best as the above result dictates. There are a number of gear drives among which one can be selected as the best. These gear drives can be spur, bevel, and worm and wheel gear drives. The further selection procedure is shown in the following table.

Table 4.10: **Decision matrix of the gear type selection**:

Criteria	Bevel gear Rating	Worm and wheel gear Rating	Spur gear Rating
Space Requirement	10	8	7
Slip Reduction	10	10	10
Cost	9	8	9
Ease of design	10	8	10
Speed reduction ratio	10	6	9
Availability on local market	9	7	9
Transmission Shaft Axis changeability	10	9	0
Overall Satisfaction	**68**	**56**	**54**

Therefore, a bevel gear is selected to be designed on the basis of the result in the above table. This bevel gear will be designed and it can be used to change the axis of the shaft which gains rotational power from the wheels for the cutting system so that the power transmission task in the functional structure of the conceptual design of the machine is satisfied.

4.10. Determination of the cutting system

The cutting system is the part of the mowing component that is needed to accomplish the ultimate purpose of the machine, and there can be various number of available alternative cutting systems from them one could be selected as best of all.

To select the type of cutting system among the possible alternative systems, Johnson's decision matrix approach, which is a more detailed type of decision matrix method that is appropriate for engineering design, can be adopted under here for the cutting system does not have subsystems that need further concept selection analysis.

The widely used alternative mechanisms to accomplish the cutting function of the mowing component might be of rotary disc blades, reciprocating sickle bar blades similar to the small hair trimming machines, and knife like blades that can be used in reel or flail mowers. These alternative cutting blades are bulleted below with their designation

- Alternative A- Reciprocating sickle bar blades
- Alternative B- Rotary disc blades
- Alternative C- Knife like blades

and they are depicted in the figures below.

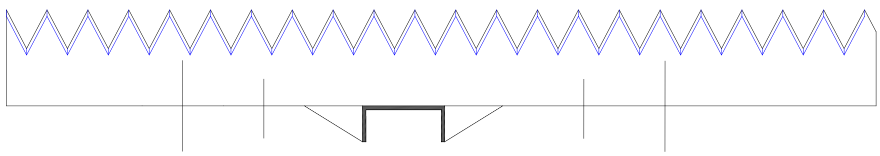


Figure 4.13: **Alternative A- Reciprocating Sickle Bar blade**

Figure 4.14: **Alternative B- commonly used Rotary disc blade types**

Figure 4.15: **Alternative C- Knife like blades shown on the blade drum of a flail mower**

4.10.1. Selection of Cutting systems

For one of the alternative systems to be raised as best, there must be criteria of selection based on which the best of all that conforms to the requirements of the mowing component will be chosen. The selection of the suitable cutting system type for this design problem will depend on criteria tabulated as follows.

Table 4.11: **Decision matrix for the type of cutting system selection.**

Criteria	Alternative A Rating	Alternative B Rating	Alternative C Rating
Ease of assembly	10	9	8
Arrangement suitability	10	3	5
Ease of design	10	3	6
Manufacturing Cost	10	4	5
Ease of Maintainability locally	10	4	6
Durability	9	7	7
Maintenance cost	10	5	7
Ease of Manufacturing	10	3	5
Raw material Availability locally	7	3	3
Overall Satisfaction	**86**	**41**	**52**

Therefore, the result in table 4.11 dictates that the best cutting system is that of the reciprocating sickle bar blades with an overall wightage of 86.

4.10.2. Mechanism Selection for Motion of Sickle bar blades

The sickle bar blades are used to accomplish the cutting task through linear reciprocating motion, and appropriate mechanisms can transmit the cutting power and generate the required linear motion of the cutting bar along the direction of cutting should be selected best among the possible alternative mechanisms that can generate a linear reciprocating motion of the cutter from the input rotational power source.

The most suitable power drive alternatives for the power transmission from the wheels shaft to the driving gear shaft are taken to be similar to those of the engine-wheels shaft indicated in section 4.9.1 for the performance criteria are almost same, and the criteria to select one best of the given alternatives are determined to be similar to those of the engine-wheels shaft power requirements.
The most applicable mechanism that can change rotational motion of a mechanical component into its linear motion is a cam mechanism, and there can be variety of cam mechanisms for this purpose. The alternative cam mechanisms for the purpose of changing rotational motion from the power source into reciprocating linear motion of the cutter bar are listed below.

- Alternative A- Yoke Cam Mechanism
- Alternative B- Inverse Cam Mechanism

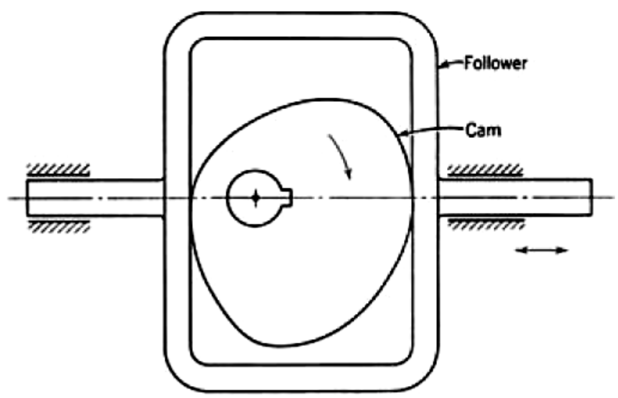


Figure 4.16: **Alternative A-Yoke Cam Mechanism (19)**

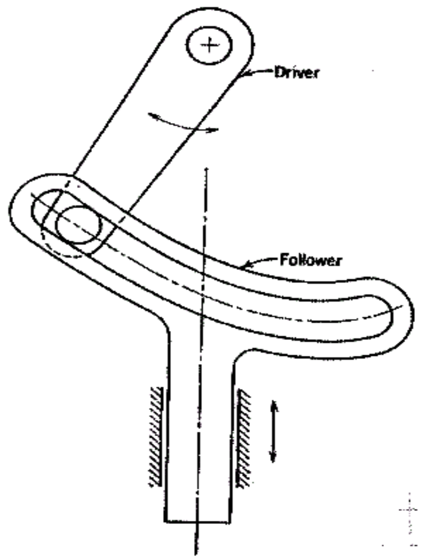


Figure 4.17: Alternative B-Inverse Cam Mechanism (19)

In these mechanisms, the motion of the follower is linear motion and that of the driver is rotational as it is shown in the above figures. The decision matrix showing the selection criteria and the weight of each alternative is tabulated as below:

Table 4.12: **Decision matrix for cam mechanism selection.**

Criteria	Alternative A Rating	Alternative B Rating
Ease of Assembly	10	8
Number of parts	10	7
Ease of design	10	8
Manufacturing Cost	7	6
Ease of Maintainability locally	10	7
Durability	9	6
Maintenance cost	10	5
Ease of Manufacturing	8	4

Raw material Availability locally	7	7
Overall Satisfaction	**81**	**58**

From the above decision matrix, the yoke cam weighs more than the other alternative, and it is selected to be the better one for this design.

4.11. Determination of the wheel type

Vehicles and self-propelling machines use wheels to achieve the forward motion. There are various types of wheels to be used for different applications. For instance, agricultural equipments which are self-propelling may use either cast steel wheel with tyre or only cast steel wheel based on environmental factors and their requirements for which they are to be used, and hence wheels for this agricultural purpose are to be selected for design.

The worst environmental condition for this machine are the probability of existence of marsh farm land on which the machine may be working and the probable difficulties of the farm land irregularities and depressions which may cause frequent wheel failure in addition to the weight of the machine that is supported by the rear wheels. Therefore, giving due concern to the rear wheels which carry the majority of the machine weight, selection of the type of the wheels with high durability and sufficient traction resistance will be carried out. Based on the following criteria and alternatives tabulated below, the wheel type will be selected.

Table 4.13: **Decision matrix for wheel type selection.**

Criteria	Rating of cast steel wheel with tyre	Rating of Cast steel wheel with spike
Ease of assembly	7	10
Ease of design	8	10
Manufacturing Cost	6	9
Ease of Maintainability locally	10	10
Durability	7	10

Maintenance cost	5	10
Ease of Manufacturing locally	9	9
Failure frequency	7	10
Traction resistance on marsh farm lands	7	10
Overall Satisfaction	**66**	**88**

Therefore, a cast steel wheels with spikes are selected to be designed for this machine based on the result of the above concept selection analysis.

4.12. Determination of the Prime Mover

In the design of this mowing component, there is a need of power source from which a mechanical energy for propelling the machine is gained. This power source may be either of electrical motors or of mechanical engines as the only alternatives from which one of them will be selected as the better source. In the agricultural application, electrical motors cannot be used as a power source because of the difficulties of the power cables extension and the requirements of high technology for electrical power sources to be used for self-propelling machines and vehicles.

Therefore, the only option to be used as a power source is the mechanical energy sources that are fuel consuming to generate the power required. These fuel consuming mechanical energy sources are engines utilizing either petrol or diesel fuel. Having these engines as the only alternatives for power source of the machine, the selection will be carried out by the following table.

Table 4.14: **Decision matrix for engine selection.**

Criteria	Rating of Diesel Engine	Rating of Petrol Engine
Availability on Market locally	10	9
Running Cost of Fuel Consumption	10	6
Availability of fuel	9	7
Cost of the engine	9	10

Compression Ratio	10	6
Maintenance cost	10	9
Ease of Operability	9	9
Overall Satisfaction	**67**	**54**

Therefore, a diesel engine is selected based on its greater overall satisfaction figure 67 as compared with the petrol engine.

To close the work of the conceptual design of the machine, the flow chart of the functional elements for the selected design of mowing machine is shown by the following figure.

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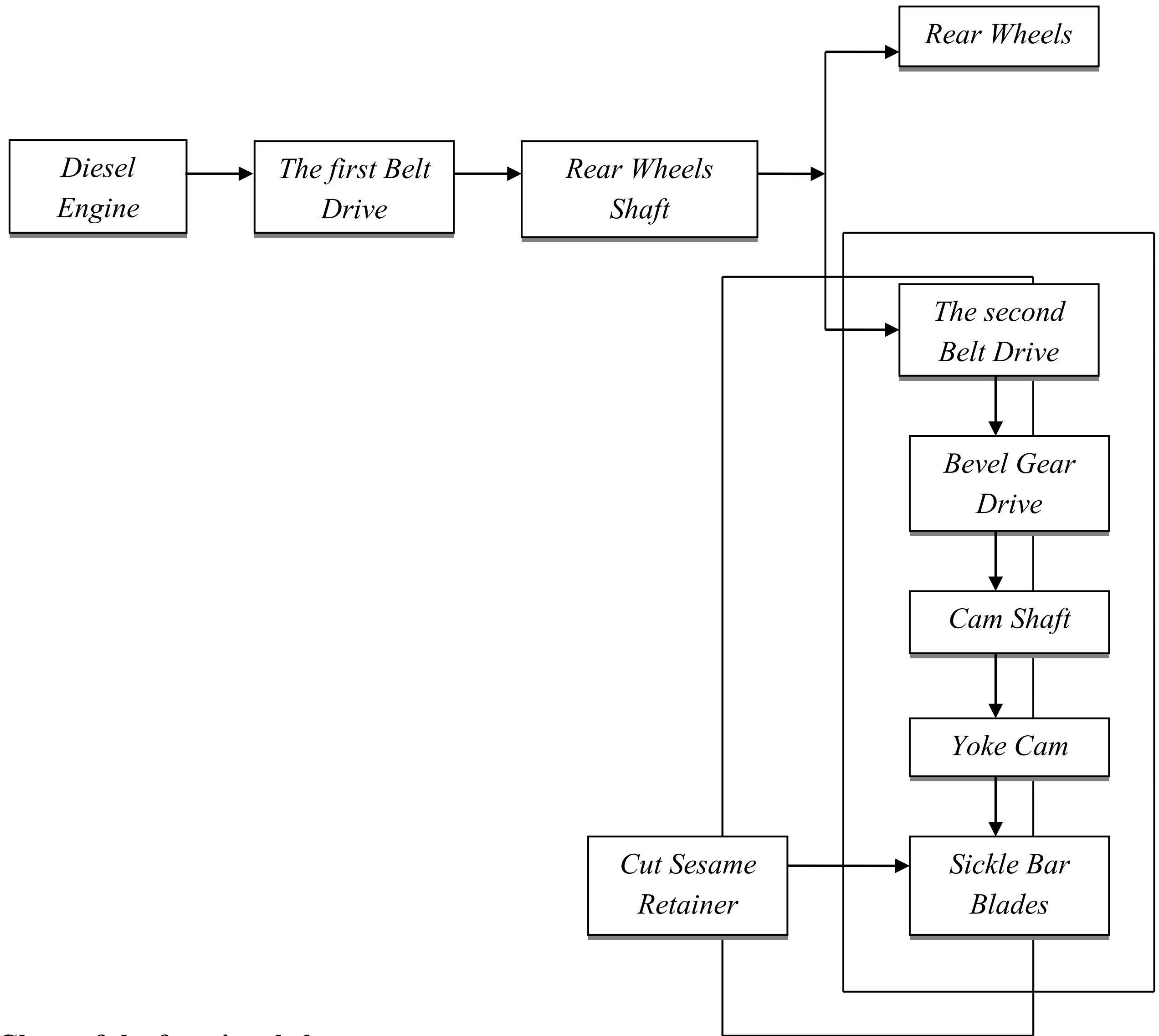

Figure 4.18: **Flow Chart of the functional elements**

5. Design of Components

5.1. Introduction

The product design of the mower dictates the mower to consist of various functions and hence there is corresponding number of mechanical components to be designed so that each function is fulfilled. The major mechanical components to be designed are bulleted below:

- A v-belt drive to transmit power from the prime mover
- A shaft on the wheels which receives power from the prime mover through the first v-belt drive.
- The second v-belt drive which used to transmit power from the wheels shaft in to the mowing mechanism.
- A gear shaft which transmits power from the wheels shaft to the gear drive through the second v-belt drive.
- A bevel gear drive used to change the direction of power transmission.
- A cam mechanism to achieve the reciprocating motion of the mowing bar.
- Keys and circlips used to mount pulleys, gears, and the cam on the shafts.
- Bearings and their housings.
- Engine bed frame and Seat of other components of the machine.
- The sheet metals covers of the mower component and the handle of the operator
- Belt tensioner use to control the speed of the machine.

5.2. V-belt Design for the first belt drive

This power drive is to be designed for transmitting power from the prime mover to the other working parts of the mower. In the mower, a 7-hp reciprocating engine running t 1175 rpm is to be used as power source to drive the mower wheels shafts which is expected to work 18 hrs per day. For this purpose of power transmission, the v-belt will be designed with the following design specification and design decisions.

Initial Design Specifications:

- Rated Power From the engine, P = 7-hp
- Smaller pulley speed, N_1= 1175 rpm
- Larger pulley speed, N_2= 400 rpm
- For the economy of space, maximum center distance is needed to be 500 mm; i.e., C = 500 mm.
- Operation Capacity of 18 hrs per day is expected.
- Larger pulley diameter, D = 420 mm.

Design Decision:

- A service factor of 1.5 corresponding to heavy duty services as such as in agricultural machines.
- If single v-belt is not sufficient, multiple v-belts can be used.
- Coefficient of friction between the pulley and belt materials [cast iron for pulleys and rubber fabric for belts], μ = 0.3 and the pulleys groove angle, $\beta = 38^0$ are taken.

1. Calculation of Design Power(DP):

Design Power DP is given by:

$$DP = (RatedPower)(ServiceFactor) \quad (1)$$

$\Rightarrow DP = 7hp * 1.5 = 10.5hp$. This is the design power by which the belt drive is to be designed.

Using this design power, the belt section can be selected for the corresponding smaller pulley speed in rpm. The smaller pulley speed is N_1= 1175 rpm and the design power, DP = 10.5 hp, and for these parameters, the belt section type is read from standards in IS 2494, 1964 as B-section belt type.

2. Calculation of belt speed(V):

The belt speed (V) is computed by: $V = \frac{\pi d N_1}{60}$ (2)

where d = smaller pulley pitch diameter, m

N_1= smaller pulley speed, rpm

$$\Rightarrow V = \frac{\pi * 1175}{60}\left[\frac{400 * 0.42}{1175}\right] m/s$$

$\Rightarrow V = 8.8 m/s$. This is the belt speed in m/s.

or $V = 1731.59\, ft/\min$

3. Calculation of contact angles (θ_S and θ_L):

The contact angles for the smaller and larger pulleys θ_S and θ_L are respectively determined by:

$$\theta_S = 180^0 - 2\sin^{-1}\left[\frac{D-d}{2C}\right] \text{ and } \theta_L = 180^0 + 2\sin^{-1}\left[\frac{D-d}{2C}\right] \quad (3)$$

Where D = larger pulley pitch diameter.

d = smaller pulley pitch diameter.

C = the center distance of the belt drive.

The center distance of the belt drive C is to be recommended to be in the range

$D \le C \le 3(D+d)$

This implies that

$420mm \le C \le 3(420+143)mm$

Then, the center distance is taken to be $C_{\max} = 500mm$.

Hence, $\theta_S = 180^0 - 2\sin^{-1}\left[\frac{420-143}{2*500}\right] = 147.84^0 = 2.58rad$

$\theta_L = 180^0 + 2\sin^{-1}\left[\frac{420-143}{2*500}\right] = 212.16^0 = 3.7rad$

4. Determination of belt length(L):

For the orientation of an open belt arrangement, the belt length for a belt drive is given by:

$$L = \sqrt{4C^2 - (D-d)^2} + \frac{[D\theta_L + d\theta_S]}{2} \qquad (4)$$

Where C = center distance of the drive

D = pitch diameter of the larger pulley

d = pitch diameter of the smaller pulley

θ_S = the contact angle of the larger pulley in rad and

θ_L = the contact angle of the smaller pulley in rad.

Then, $L = \sqrt{4(0.5)^2 - (0.42-0.143)^2} + \frac{[0.42*3.7+0.143*2.58]}{2} = 1.924m = 75.75in$

5. Determination of the horse power rating per belt(hp):

The horse power rating for the section B belt corresponding to the next higher speed of the smaller pulley, 1400 rpm, and the recommended smaller pulley pitch diameter can be read from the standard tables in IS 2494, 1964.

Hence, N_1= 1400rpm is the next higher rpm to the smaller pulley speed, and

$d = \frac{DN_2}{N_1} = \frac{420*400}{1175} = 143mm = 5.63in$

Then, by interpolation, for 1400rpm and $d = 5.63in$,the horse power rating per belt is read from standards charts in IS 2494, 1964 to be 4,661-hp.But this horse power rating per belt has to be corrected for the contact angle of the smaller pulley and the belt length with factors K_1 and K_2 respectively. From chart of an angle of contact versus correction factor K_1 in charts of standards IS 2494, 1964, K_1= 0.92, and from table of belt-length correction factor K_2 for the belt length $L = 75.75in$ and belt type B, K_2=0.96.

Therefore, the horse power rating per belt will be:

hp = (4.66*0.92*0.96) hp = 4.12hp/belt. This is the design power per belt.

6. Determination of the number of belts(n) required :

The number of belts (n) required is given by

$$n = \frac{DP}{hp} \Leftrightarrow n = \frac{10.5hp}{4.12hp/belt} = 2.55belts$$

Therefore, taking the next higher number of belts, $n = 3belts$ of B-section types are to be used.

7. Belt Tension calculation(Maximum Tensions):

The maximum tensions on the tight and slack sides of the belt F_1 and F_2, respectively can be computed by using the design power, DP, the belt speed, V, the friction coefficient, μ, the pulley groove angle, and the smaller pulley contact angle, θ_S.

Then, the relation is given by:

$$F_1 - F_2 = \frac{1000DP}{V} \text{ and } \frac{F_1}{F_2} = e^{\left[\frac{\mu\theta_S}{\sin\left(\frac{\beta}{2}\right)}\right]} \quad (5)$$

$$\text{Then, } F_1 - F_2 = \frac{1000 * 7.833}{8.8} N = 890.11N \quad (6)$$

$$\text{and } F_1 = F_2 * e^{\left[\frac{\mu\theta_S}{\sin\left(\frac{\beta}{2}\right)}\right]} = F_2 * e^{\left[\frac{0.3*2.58}{\sin e19^0}\right]} = 10.78F_2 \quad (7)$$

Therefore, solving for F_1 and F_2 simultaneously,

F_1= 91.04 N and **F_2= 981.46 N**. These are the belt tensions of the first belt drive.

8. Profile Dimensions of the pulleys:

The standard sectional view of the rim of the B section V-belt pulleys is shown in the figure below.

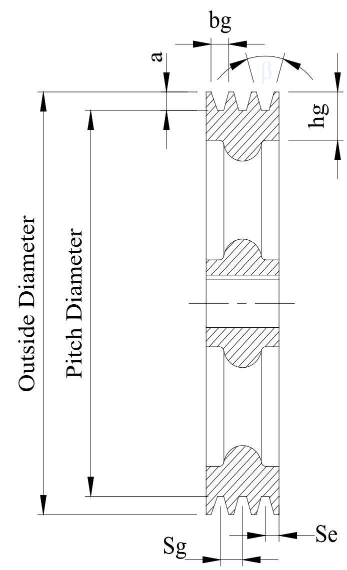

Figure 5.1: **V-Belt sectional view for the profile dimensions indicated.**

Therefore, the profile dimensions of the B-section deep grooved v-belt pulley are read from standard table in IS 2494, 1964 to be:

- Groove angle, $\beta \pm 0.38^0 = 38^0$
- $b_g = 0.774in = 19.66mm \approx 20mm$
- $h_g = 0.73in = 18.542mm \approx 19mm$
- $2a = 0.71in = 18.034mm \approx 18mm$
- $S_g \pm 0.025 = 0.875in = 22.86mm \approx 23mm$

- $S_e = 0.0.562^{+0.120}_{-0.065} in = 17.323mm \approx 18mm$

9. Pulleys Weight Calculation:

For the worst loading condition of the shafts carrying pulleys, it can be assumed that the pulleys on the shaft as a solid cylindrical disk so that it would be convenient to determine the contribution of their weight to the loads on the shafts.

Hence, based on the assumption of solid disk pulleys, the weight of the pulleys can be given as:

Weight of the pulley, $W_p = V\rho g$ (8)

Where V= volume of the pulley

ρ = Density of the pulley material = 7200 Kg/m^3 for Cast Iron Pulleys.

g = Gravitational Acceleration = 10m/s^2

- For the larger Pulley, $V = \frac{\pi D_1^2 f_1}{4}$

Where f_1= face width of the pulley=82mm which is constant for both pulleys of the drive.

D_1= Outer pulley diameter

and it is given by:

$D_1 = D + 2a = 420 + 18 = 438mm = 0.438m$

Hence, $V = \frac{\pi * 0.438^2 * 0.082}{4} m^3 = 0.012355m^3$. This is the volume of the larger pulley of the prime mover belt drive.

Then, the weight of the larger pulley W_{PL} of the first belt drive is

$W_{PL} = (0.012355 * 7200 * 10)N = 889.56N$ .This weight will be used for the design of the wheels shaft of the mower.

- For the smaller Pulley, $V = \frac{\pi d_1^2 f_1}{4}$ (9)

Where d_1 = outer diameter of the smaller pulley and it is given by:

$$d_1 = d + 2a = 143 + 18 = 161mm = 0.161m$$

Hence, $V = \frac{\pi * 0.161^2 * 0.082}{4} m^3 = 0.001669 m^3$. This is the volume of the smaller pulley of the belt drive.

Then, the weight of the smaller pulley, w_{ps} will be given by:

$w_{pl} = (0.001669 * 7200 * 10)N = 120.2N$. This is the weight of the pulley to be used in the design of the gear shaft.

5.3. V-belt Design for the second belt drive

Once the prime mover delivers power to the mower wheels shaft, other power drive must be devised to transmit power for the mower mechanisms from the wheels shaft. For this purpose, a v-belt drive which is capable of transmitting an amount of power required for mowing operations of the reciprocating cutter bars will be designed. Hence, design of the belt drive will be pursued based on the following initial design specifications and decisions.

Initial design Specifications:

- Rated Power to be transmitted, P = 3hp, based on the recommended approximate reference values of power requirement for small lawn mower.
- Larger pulley Diameter, D = 280 mm for there is space limitations.
- Larger pulley speed, n_1 =400rpm
- Required Speed ratio, VR=2
- Maximum center distance is not to exceed 500mm.
- 18 hrs per day operation is expected.

Deign Decisions:

- An overload service factor of 1.4 corresponding to a heavy duty service as of agricultural machineries.
- If single belt is not sufficient, multiple v-belts can be used.
- For cast iron pulleys and rubber fabric belt, friction coefficient, $\mu = 0.3$ and the pulleys groove angle, $\beta = 38^0$ are taken.

Belt Drive materials;

- Rubber fabric is the material to be used for the belt for is low cost and availability on market.
- Cast iron is selected to be used for the pulley for it is one of optional pulley material, low cost and availability on market.

1. Calculation of The design power(DP):

The design power DP is computed as:

$$DP = (RatedPower)(ServiceFactor)$$

$$\Rightarrow DP = 3hp * 1.4 = 4.2hp = 3.1332KW \ .$$

Using this design power DP = 4.2hp required to be transmitted; the type of belt section for the corresponding smaller pulley speed can be selected on a design power versus speed of the smaller pulley chart in standards IS 2494, 1964.

Then, $\frac{n_2}{n_1} = 2 \Leftrightarrow n_2 = 2n_1 = 2 * 400rpm$

$\Rightarrow n_2 = 800rpm$. This is the speed of the smaller pulley.

Hence, for $DP = 4.2hp$ and $n_2 = 800rpm$, the belt cross section type B is selected.

2. Calculation of belt speed(V):

The belt speed in (m/s) can be given by:

$$V = \frac{\pi d n_1}{60}$$

Where d = smaller pulley pitch diameter,

n_1= smaller pulley speed, rpm

But, $\frac{n_2}{n_1} = 2 = \frac{D}{d} = \frac{280}{d} \Leftrightarrow d = 140mm$

and $V = \frac{\pi * 0.14 * 800}{60} = 5.86 m/s$

or $V = 1154.32 fpm$. This is the belt speed of the belt drive.

3. Determination of the horsepower rating per belt(hp):

The horsepower rating for the b-section V-belt selected corresponding to the next higher speed of the smaller pulley and the smaller pulley recommended pitch diameter is found to be 3.5136hp from standard tables of IS 2494, 1964. But this horsepower rating has to be corrected for the angle of contact and belt length correction factors K_1 and K_2 respectively.

- Calculation of angle of contact(θ_S)

The angle of contact for the smaller pulley is given by:

$$\theta_S = 180^0 - 2\sin^{-1}\left[\frac{D-d}{2C}\right] \quad (10)$$

Where D = larger pulley pitch diameter.

d = smaller pulley pitch diameter.

C = the center distance of the belt drive.

- The center distance, C, is recommended to be in the range $D \le C \le 3(D+d)$.

Then, this implies:

$$280 \le C \le 3(280+140)$$

$$\Rightarrow 280 \le C \le 1260mm$$

Therefore, for the center distance is not to exceed 500mm as the design decision, the center distance is determined to be C=500mm, the maximum permissible dimension.

Then, $\theta_S = 180^0 - 2\sin^{-1}\left[\frac{280-140}{2*500}\right] = 163.9^0 = 2.861rad$

4. Determining the belt length(L):

For the orientation of an open belt arrangement, the belt length is computed by:

$$L = \sqrt{4C^2 - (D-d)^2} + \frac{[D\theta_L + d\theta_S]}{2} \quad (11)$$

Where C = center distance of the second drive

D = pitch diameter of the larger pulley

d = pitch diameter of the smaller pulley

θ_S = the contact angle of the larger pulley in rad and

θ_L = the contact angle of the smaller pulley in rad.[19]

- Determining the larger pulley contact angle θ_L:

The contact angle θ_L is known by:

$$\theta_L = 180^0 + 2\sin^{-1}\left[\frac{D-d}{2C}\right] = 180^0 + 2\sin^{-1}\left[\frac{280-140}{2*500}\right] = 196.1^0 = 3.423rad$$

Then, $L = \sqrt{4(0.5)^2 - (0.28-0.14)^2} + \frac{[0.28*3.423 + 0.14*2.861]}{2} = 1.6675m = 65.73in$. This is the length of the belt as far as the diameters used in the equation are pitch diameters. Then the correction factor for angle of contact K_1 and the belt length K_2 can be read corresponding to the belt angle of contact and length respectively.

Hence, K_1=0.96 for $\theta_S = 2.861 rad$, and

K_2=0.931 for $L = 65.73 in$ and belt cross section B. therefore the horsepower rating per belt is calculated as:

hp =3.5136hp*0.96*0.931

$\Rightarrow$ hp =3.142hp/belt. This is the design power per belt.

5. Calculation of Number of Belts required(n):

The number of belts required (n) is given by:

$$n = \frac{DP}{hp} \text{ Where DP = Design Power} \quad (12)$$

hp =Design horsepower per belt

$$\Rightarrow n = \frac{4.2hp}{3.142hp/belt} = 1.3367 belts$$

So, the next higher number of belts will be n=2.

Therefore, two belts with B-cross section are required for this drive.

6. Belt tension Calculation (Maximum Tensions):

The maximum tension on the tight and slack sides of the belt F_1 and F_2 respectively can be calculated by using the design power DP and the belt speed V as:

$$F_1 - F_2 = \frac{1000DP}{V} = \frac{1000 * 3.1332}{5.964} N = 534.31N \text{ and}$$

$$F_1 = F_2 * e^{\left[\frac{\mu\theta_S}{\sin\left(\frac{\beta}{2}\right)}\right]} = F_2 * e^{\left[\frac{0.3*2.861}{\sin e19^0}\right]} = 13.962F_2 \quad (13)$$

Then, Solving for F_1 and F_2,

F_1=41.22 N and F_2= 575.55 N,

7. Profile Dimensions of the pulleys:

The sectional view of the rim of the standard B-section V-belt pulley is shown below in the figure below.

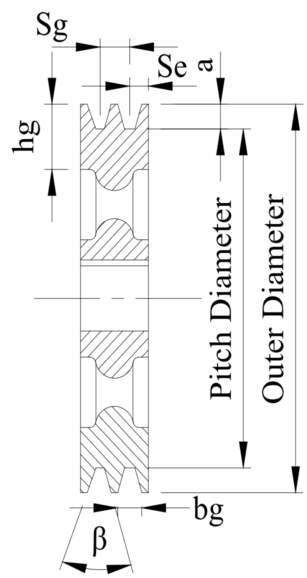

Figure 5.2: **V-Belt sectional view for the profile dimensions**

Therefore, the profile dimensions of the pulleys groove for the belt design of B-section as read from standard tables of IS 2494, 1964 for B-section are:

- Groove angle, $\beta \pm 0.38^0 = 38^0$
- $b_g = 0.774in = 19.66mm \approx 20mm$
- $h_g = 0.73in = 18.542mm \approx 19mm$
- $2a = 0.71in = 18.034mm \approx 18mm$
- $S_g \pm 0.025 = 0.875in = 22.86mm \approx 23mm$

- $S_e = 0.0.562^{+0.120}_{-0.065} in = 17.323mm \approx 18mm$

8. Pulley weights Calculation(W_p):

In the drive, the worst loading condition assumption of solid cylindrical pulley is employed as of the first belt drive.

Hence, the weight of the pulley is computed by:

Weight of the pulley, $W_p = V\rho g$ (14)

Where V= volume of the pulley

ρ = Density of the pulley material = 7200 Kg/m^3 for Cast Iron Pulleys.

g = Gravitational Acceleration = 10m/s^2

- For the larger Pulley, $V = \frac{\pi D_2^2 f_2}{4}$ (15)

Where f_2= face width of the pulley=0.059m which is constant for both pulleys of the drive.

D_2= Outer pulley diameter

and it is given by:

$D_2 = D + 2a = 280 + 18 = 298mm = 0.298m$

Hence, $V = \frac{\pi * 0.298^2 * 0.059}{4} = 0.004115m^3$ .this is the volume of the larger pulley.

Then, the weight of the larger pulley W_{PL} of the second drive is:

$W_{PL} = V\rho g = (0.004115 * 7200 * 10)N = 296.3N$. This is the weight of the larger pulley to be used for the design of the wheels shaft of the mower.

- For the smaller Pulley, $V = \frac{\pi d_2^2 f_2}{4}$

Where f_2= face width of the pulley=0.059m which is constant for both pulleys of the drive.

d_2= Outer diameter of the smaller pulley

and it is given by:

$$d_2 = d + 2a = 140 + 18 = 158mm = 0.158m$$

Then, $V = \frac{\pi * 0.158^2 * 0.059}{4} = 0.0011566m^3$. This is the volume of the smaller pulley, and then the weight of the pulley w_{PL} $w_{PL} = V\rho g = (0.0011566 * 7200 * 10)N = 83.3N$. This is the weight of the pulley to be used for the design of the shaft transmitting power to mowing mechanism.

5.4. Wheels shaft Design

A shaft is a mechanical device for transferring power from the engine or motor to the point where useful work is applied. Most engines or motors deliver power as torque through rotary motion: this is extracted from the linear motion of pistons in a reciprocating engine; water driving a water wheel; or forced gas or water in a turbine. From the point of delivery, the components of power transmission form the drive train.

Shafts are carriers of torque: they are subject to torsion and shear stress, which represents the difference between the input force and the load. They thus need to be strong enough to bear the stress, without imposing too great an additional inertia by virtue of the weight of the shaft. Therefore, this shaft is a part used to transmit the rated power 7hp from the engine in to other accessories of the mower requiring power. The design of this shaft will be done for the following design specifications, design consideration and material of the shaft.

Design specifications:

- Power to be transmitted, P = 7hp
- Speed of the shaft, N_s=400rpm.

Design decision:

- Maximum shaft Length, L_s=1.032m
- The shaft is to be supported at its ends by the wheels.

Material of the shaft:

Plain Carbon steel is used for components subjected to repeatedly applied impact loads as for leaf springs. Hence this material in annealed condition and heat treated after forming process can be used for this shaft application. Average mechanical properties of the plain carbon steel from design data book [18] are:

- Density, $\rho = 7830 \frac{kg}{m^3}$
- Ultimate tensile strength, $\sigma_u = 380MPa$
- Elastic limit or Yield Point:
  - Tensile Yield strength, $\sigma_y = 217MPa$
  - Compressive yield strength, $\sigma_c = 217MPa$
  - Shearing Yield strength, $\tau = 140MPa$

Static Analysis:

The following figure shows the free body diagram of the shaft loading, as an indeterminate beam fixed at both ends.

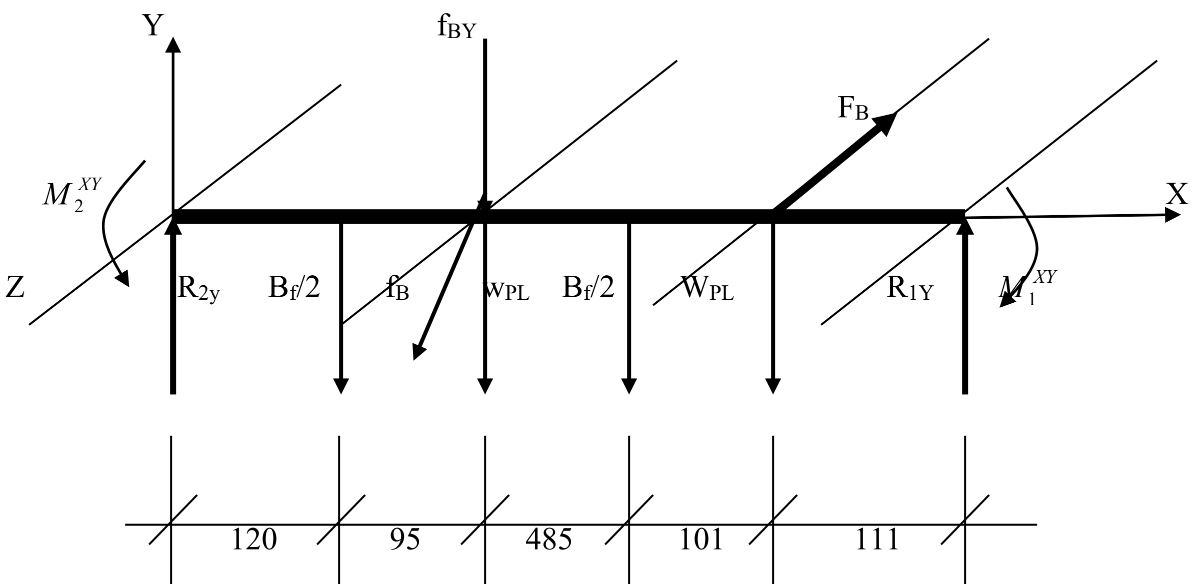

Figure 5.3: **Free body diagram of the wheel shaft (Dimensions are in mm)**

Where T_1= the input torque from the engine

F_B= belt force of the first belt drive directed in the negative Z direction.

W_{PL}= weight of the pulley of the first belt drive on the shaft

B_F=load on the bearing

T_{EW}=Torque due to the engine weight.

T_2= the torque induced on the shaft due to the second belt drive

w_{PL}= weight of the pulley of the second belt drive on the wheels shaft.

f_B= belt force of the second belt drive (inclined at 30^0 from the horizontal as shown).

T = Reaction torque of the wheels traction forces considered to be equal on both Wheels

R_2=Reaction force on the left end of the wheel shaft having the components R_{2Z} and R_{2Y}

R_1=Reaction force on the right end of the wheel shaft having the components R_{1Z} and

R_{1Y}

M_1 and M_2= Reaction Moments at the ends of the shaft (both having the components

$M_1^{XY}, M_1^{XZ}, M_2^{XY}$ and M_2^{XY})

5.4.1. *Calculation of the external loads*

- Belt forces for the first belt drive:

For the worst loading condition of the shaft, the force F_B has to be the sum of the first belt tensions F_1 and F_2.

Hence, $F_B = F_1 + F_2 = (91.04 + 981.46) = 1072.5N$

- The input torque from the engine(T_1):

The maximum torque to be transmitted by the shaft can be computed from the power torque relation:

$$P = T_1 * V \tag{16}$$

Where V = speed of the first belt

P = Power rated to be transmitted.

Hence, $T_1 = \frac{P}{V} = \frac{7*0.746}{8.8} = 593.41KNm$

- Weight of the pulley of the first belt drive on the shaft(W_{PL})

From the design of the belt drive, the weight W_{PL} is computed for the worst condition of the shaft loading, to be:

$W_{PL} = 889.56N$

- Load on the bearings(B_F):

For the calculation of this load, based on the geometric appearance of the mower structure, a structure made up of 6m U-Channel structural steel with 4”x1.72” standard dimension and 6m angle iron with standard dimension 2”x2”x(1/8)” are supported by the bearings. Moreover, the weight of the engine is also supported by the bearings. Therefore, these loads have to be considered for the worst condition of the shaft loading.

Mass of the U-Channel per unit length in (Kg/m) is read from standard table to be 12Kg/m, and that of the angle iron to be 2.5 kg/m. the mass of the 7hp reciprocating engine available on market is 35 kg. Therefore, the total load supported bearing B_F is

$$B_F = [(6x12) + (6x2.5) + 35] = 1220N$$

- Torque due to the engine weight(T_{EW}):

This torque is induced due to the weight of the engine at an arm length of the center distance of the first belt drive, C.

Hence, $T_{EW} = m_e gC$ (17)

Where m_e = engine mass= 35kg

g = Gravitational Acceleration = 10 m/s^2

C = The center distance of the first belt drive = 0.5 m.

Hence, $T_{EW} = m_e gC = 35x10x0.5 = 175Nm$

- The torque induced on the shaft due to the second belt drive(T_2):

By using the pulley radius as the arm length of the torque, r = d/2 =70mm and belt tensions $F_1 = 41.22N$ and $F_2 = 575.55N$ of the second belt, it would be possible to compute the torque.

Hence, $T_1 = (F_2 - F_1) * r = (575.55 - 41.22) * 0.07 = 37.40Nm$

- Weight of the pulley of the second belt drive on the wheel shaft(W_{PL}):

This pulley weight is already calculated in the design of the belt drive as the weight of the large pulley of the drive. Hence, it is found to be: $w_{PL} = 296.3N$

- Belt force of the second belt(f_B):

This force is the contribution of the belt tensions of the second belt drive. For the worst condition of loading, it is calculated as the sun of the belt tensions as the maximum contribution to the shaft loading with the condition of the belt to be inclined at 30^0 angular orientations as shown in the figure below if the shaft axis is perpendicular to this page.

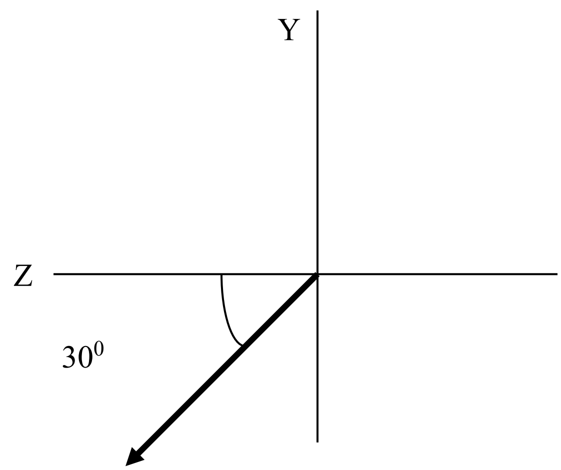

Figure 5.4: Angular orientation of the second belt drive

Hence, from the figure showing the orientation of the resultant belt tensions,

$f_B = F_1 + F_2 = 575.55 + 41.22 = 616.77N$.

This is the belt force of the second belt. This force can be resolved in to the vertical and the horizontal reference axes shown in the above figure.

Then, $f_{BY} = f_B \sin 30^0 = 616.77 * \sin 30^0 = 308.385N$ and

$$f_{BZ} = f_B \cos 30^0 = 616.77 * \cos 30^0 = 534.14N$$

5.4.2. *Calculation of the reaction loads*

The reaction loads to be calculated are in the form of reaction torques in the yz plane and the reaction forces in the xy and xz planes.

Hence, these reaction loads are to be determined in the planes xz, xy, and it respectively.

- Reaction forces on xy plane (R_{2Y} , R_{1Y} M_1^{XY} , and M_2^{XY})

Using the following free body diagram and the singularity function method, the bending moment at the sections of the shaft shown, M_{xy} is given by:

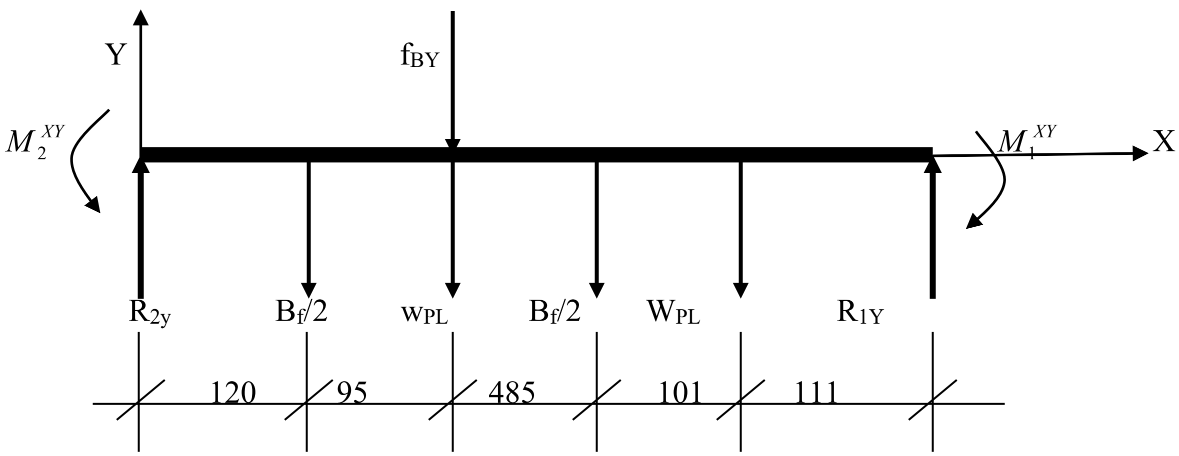

Figure 5.5: Free body diagram of the Wheel shaft in the XY plane (dimensions are all in mm)

$$M_{XY} = -M_2^{xy}(x)^0 + R_{2y}(x)^1 - \frac{B_F(x-0.12)^1}{2} - f_{BY}(x-0.215)^1 - w_{PL}(x-0.215)^1 - \frac{B_F(x-0.7)^1}{2}$$
$$-W_{PL}(x-0.801)^1 + R_{1y}(x-0.912)^1 + M_1^{xy}(x-0.912)^0$$

The equation of flexure for the beam is known from strength of materials to be

$$EI\frac{d^2y}{dx^2} = M_{xy} \tag{18}$$

Where E = Young's Modulus of the shaft material

I = Mass moment of inertia of the shaft cross-section.

M_{xy} = Bending moment represented by singularity function.

Integrating the differential equation as an indefinite integral,

$$EI\frac{dy}{dx} = \int [-M_2^{xy}(x)^0 + R_{2y}(x)^1 - \frac{B_F(x-0.12)^1}{2} - f_{BY}(x-0.215)^1 - w_{PL}(x-0.215)^1 - \frac{B_F(x-0.7)^1}{2}$$
$$-W_{PL}(x-0.801)^1 + R_{1y}(x-0.912)^1 + M_1^{xy}(x-0.912)^0]dx$$
$$\Rightarrow EI\frac{dy}{dx} = -M_2^{xy}(x)^1 + \frac{1}{2}R_{2y}(x)^2 - \frac{B_F(x-0.12)^2}{4} - \frac{1}{2}f_{BY}(x-0.215)^2 - \frac{1}{2}w_{PL}(x-0.215)^2 - \frac{B_F(x-0.7)^2}{4}$$
$$-\frac{1}{2}W_{PL}(x-0.801)^2 + \frac{1}{2}R_{1y}(x-0.912)^2 + M_1^{xy}(x-0.912)^1 + C_1$$

Where C_1= Integration constant to be determined from the boundary conditions of both ends fixed and intermediate loads of the shaft.

<u>Boundary conditions:</u> when x = 0, $\frac{dy}{dx} = 0$. Hence C_1= 0.

Then,

$$EI\frac{dy}{dx} = -M_2^{xy}(x)^1 + \frac{1}{2}R_{2y}(x)^2 - \frac{B_F(x-0.12)^2}{4} - \frac{1}{2}f_{BY}(x-0.215)^2 - \frac{1}{2}w_{PL}(x-0.215)^2 - \frac{B_F(x-0.7)^2}{4}$$
$$-\frac{1}{2}W_{PL}(x-0.801)^2 + \frac{1}{2}R_{1y}(x-0.912)^2 + M_1^{xy}(x-0.912)^1$$

integrating for the deflection y of the shaft,

$$EIy = -\frac{1}{2}M_2^{xy}(x)^2 + \frac{1}{6}R_{2y}(x)^3 - \frac{B_F(x-0.12)^3}{12} - \frac{1}{6}f_{BY}(x-0.215)^3 - \frac{1}{6}w_{PL}(x-0.215)^3 - \frac{B_F(x-0.7)^3}{12}$$
$$-\frac{1}{6}W_{PL}(x-0.801)^3 + \frac{1}{6}R_{1y}(x-0.912)^3 + \frac{1}{2}M_1^{xy}(x-0.912)^2 + C_2$$

Where C_2=integration constant similar to C_1.

<u>Boundary conditions:</u> y = 0 at x = 0, hence C_2 = 0

Then,

$$EIy = -\frac{1}{2}M_2^{xy}(x)^2 + \frac{1}{6}R_{2y}(x)^3 - \frac{B_F(x-0.12)^3}{12} - \frac{1}{6}f_{BY}(x-0.215)^3 - \frac{1}{6}w_{PL}(x-0.215)^3 - \frac{B_F(x-0.7)^3}{12}$$
$$-\frac{1}{6}W_{PL}(x-0.801)^3 + \frac{1}{6}R_{1y}(x-0.912)^3 + \frac{1}{2}M_1^{xy}(x-0.912)^2$$

Imposing the boundary conditions $\frac{dy}{dx} = 0$ at x = L and y = 0 at x = L to yield two equations to be solved simultaneously for M_2^{xy} and R_{2y}. Hence, for x = L = 0.912m,

-0.912 M_2^{xy} +0.416 R_{2y}=357.41

$-0.416 M_2^{xy} + 0.126\, R_{2y} = 85.813$

Hence, $M_2^{xy} = 160.58$ Nm and $R_{2y} = 1211.20$ N.

Once M_2^{xy} and R_{2y} are determined, from equilibrium of the loads, the4 sum of the vertical forces must vanish.

Hence, $\sum F_{Vertical} = 0$ (19)

$$\Rightarrow R_{1y} + R_{2y} = B_F + f_{BY} + w_{PL} + W_{PL}$$

$$\Rightarrow R_{1y} = 1503.1N$$

And for M_1^{xy} to be determined, at x = 0, y = 0.this implies that

$$0 = -\frac{1220*(-0.12)^3}{12} - \frac{1}{6}(604.7)(-0.215)^3 - \frac{1220*(-0.7)^3}{12} - \frac{1}{6}(889.6)(-0.801)^3$$
$$+\frac{1}{6}(1503.1)(-0.912)^3 + \frac{1}{2}M_1^{xy}(-0.912)^2$$

$$\Rightarrow M_1^{xy} = 186.977Nm$$

- Reactions on the XZ plane(R_{2Z}, R_{1Z}, M_1^{XZ}, and M_2^{XZ})

Using the following FBD, it is possible to develop the singularity function for the bending moment M_{XZ}:

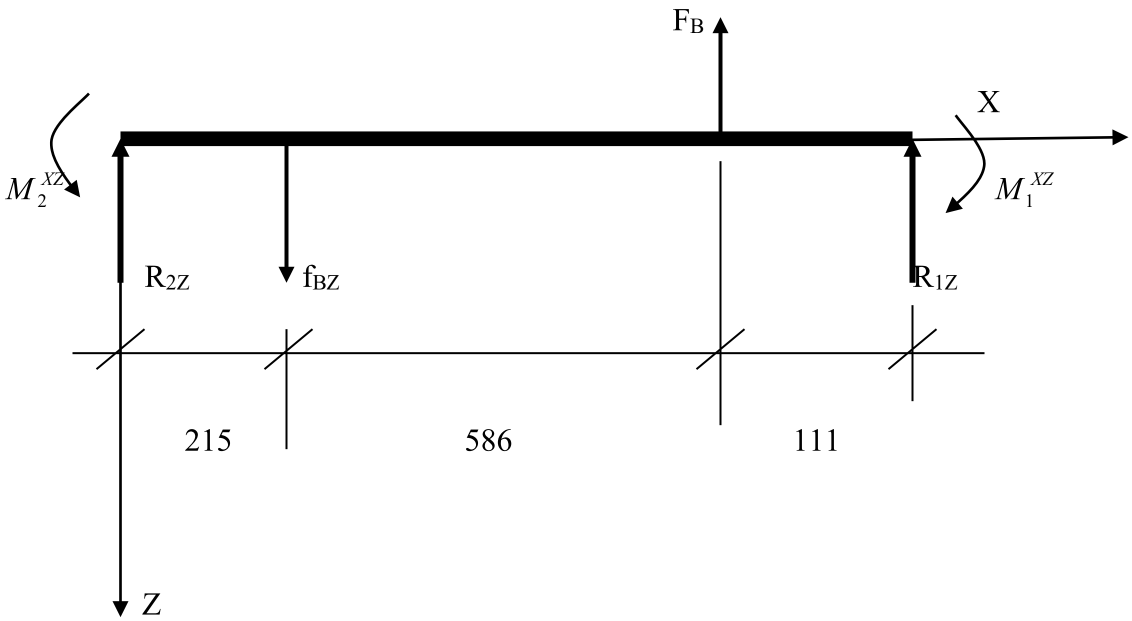

Figure 5.6: **Free body diagram of the Wheel shaft in the XZ plane (dimensions are all in mm)**

$$M_{XZ} = -M_2^{xz}(x)^0 + R_{2Z}(x)^1 - f_{BZ}(x-0.215)^1 + F_B(x-0.801)^1 + R_{1Z}(x-0.912)^1 + M_1^{xz}(x-0.912)^0$$

Applying the equation of flexure for the shaft,

$$EI\frac{d^2y}{dx^2} = M_{xz}$$

Integration yields the slope of the shaft as:

$$EI\frac{dy}{dx} = -M_2^{xz}(x)^1 + \frac{1}{2}R_{2Z}(x)^2 - \frac{1}{2}f_{BZ}(x-0.215)^2 + \frac{1}{2}F_B(x-0.801)^2$$
$$+\frac{1}{2}R_{1Z}(x-0.912)^2 + M_1^{xz}(x-0.912)^1 + C_1$$

Where C_1= integration constant.

Boundary conditions: when x = 0, $\frac{dy}{dx} = 0$. Hence C_1= 0.

Then,

$$EI\frac{dy}{dx} = -M_2^{xz}(x)^1 + \frac{1}{2}R_{2Z}(x)^2 - \frac{1}{2}f_{BZ}(x-0.215)^2 + \frac{1}{2}F_B(x-0.801)^2 + \frac{1}{2}R_{1Z}(x-0.912)^2 + M_1^{xz}(x-0.912)^1$$

Again integration yields the deflection as:

$$EIy = -\frac{1}{2}M_2^{xz}(x)^2 + \frac{1}{6}R_{2Z}(x)^3 - \frac{1}{6}f_{BZ}(x-0.215)^3 + \frac{1}{6}F_B(x-0.801)^3 + \frac{1}{6}R_{1Z}(x-0.912)^3 + \frac{1}{2}M_1^{xz}(x-0.912)^2 + C_2$$

Where C_2 = integration constant.

<u>Boundary conditions:</u> y = 0 at x = 0, hence $C_2 = 0$

Then,

$$EIy = -\frac{1}{2}M_2^{xz}(x)^2 + \frac{1}{6}R_{2Z}(x)^3 - \frac{1}{6}f_{BZ}(x-0.215)^3 + \frac{1}{6}F_B(x-0.801)^3 + \frac{1}{6}R_{1Z}(x-0.912)^3 + \frac{1}{2}M_1^{xz}(x-0.912)^2$$

And, imposing the boundary conditions $\frac{dy}{dx} = 0$ at x = L and y = 0 at x = L, the following simultaneous equations are found:

$-0.912 M_2^{xz} + 0.416 R_{2z} = 123.143$

$-0.416 M_2^{xz} + 0.126 R_{2z} = 29.896$ (20)

From these equations, M_2^{xz} =53.034 Nm and R_{2z}=412.37 N.

Once M_2^{xz} and R_{2z} are determined, from the static's of the shafts, the sum of the horizontal forces must vanish.

Then,

$$\sum F_{Horizontal} = 0 \quad (21)$$

$\Rightarrow R_{1z} + R_{2z} + F_B - f_{BZ} = 0$

$\Rightarrow R_{1z} = -950.73N$.The negative sign shows that the direction is opposite to the assumed direction of R_{1z}.

Hence, $R_{1z} = 950.73N$ in the positive z-direction.

And for M_1^{xz} to be determined, at x = 0, y = 0. This implies that

$$0 = -\frac{f_{BZ} * (-0.215)^3}{6} + \frac{1}{6} F_B (-0.801)^3 + \frac{R_{1z} * (-0.912)^3}{6} + \frac{1}{6} M_1^{xz} (-0.912)^2$$

Substitution of the known values results in $M_1^{xz} = -70.22Nm$. The negative sign shows that its direction is opposite to the assumed one.

Therefore, the reaction loads are found to be

for the XZ plane, $M_1^{xz} = -70.22Nm$ and $R_{1z} = -950.73N$

M_2^{xz} =53.034 Nm and R_{2z}=412.37 N

and for the XY plane, $M_1^{xy} = 186.977Nm$ and $R_{1y} = 1503.1N$

M_2^{xy} =160.58 Nm and R_{2y}=1211.20 N

- Reaction torques(moments) on the yz plane(T):

These reaction torques are induced due to the traction forces on the wheels intended to have equal load shares. As far as all the torque are acting on the yz-plane, taking sum of torques on the shaft from its static condition,

$$\sum T = 0 \qquad (22)$$

$$\Rightarrow 2T + T_2 + T_{EW} - T_1 = 0$$

$$\Rightarrow 2T = T_1 - T_{EW} - T_2$$

$$\Rightarrow T = 190.51Nm$$

5.4.3. *Determination of Maximum Bending Moments*

In determining the maximum bending moment, the bending moments in each section of the shaft corresponding to the loads in xy and xz panes will be computed and the maximum of the results in each plane will be taken for the determination of the maximum bending moment.

- Maximum bending moment in the xy plane($M_{\max}^{xy}$)

From the singularity function of the bending moment in the xy plane,

$$M_{xy} = -M_2^{xy}(x)^0 + R_{2y}(x)^1 - \frac{B_F(x-0.12)^1}{2} - f_{BY}(x-0.215)^1 - w_{PL}(x-0.215)^1 - \frac{B_F(x-0.7)^1}{2}$$
$$-W_{PL}(x-0.801)^1 + R_{1y}(x-0.912)^1 + M_1^{xy}(x-0.912)^0$$

There are five loading sections of the shaft in the xy plane and bending moment will be reckoned for each section.

- For $0 \le x \le 0.12$,

$$M_{xy} = -M_2^{xy}(x)^0 + R_{2y}(x)^1 = (-160.58) + (1211.2 * 0.12) = -15.24Nm$$

- For $0.12 \le x \le 0.215$,

$$M_{xy} = -M_2^{xy}(x)^0 + R_{2y}(x)^1 - \frac{B_F(x-0.12)^1}{2}$$
$$= -160.58 + (1211.2 * 0.215) - (610 * 0.095) = 41.878Nm$$

- For $0.215 \le x \le 0.7$,

$$M_{xy} = -M_2^{xy}(x)^0 + R_{2y}(x)^1 - \frac{B_F(x-0.12)^1}{2} - f_{BY}(x-0.215)^1 - w_{PL}(x-0.215)^1 = 40.181Nm$$

- For $0.7 \le x \le 0.801$,

$$M_{xy} = -M_2^{xy}(x)^0 + R_{2y}(x)^1 - \frac{B_F(x-0.12)^1}{2} - f_{BY}(x-0.215)^1$$
$$-w_{PL}(x-0.215)^1 - \frac{B_F(x-0.7)^1}{2} = -21.783Nm$$

- For $0.801 \le x \le 0.912$,

$$M_{xy} = -M_2^{xy}(x)^0 + R_{2y}(x)^1 - \frac{B_F(x-0.12)^1}{2} - f_{BY}(x-0.215)^1 - w_{PL}(x-0.215)^1 - \frac{B_F(x-0.7)^1}{2}$$
$$-W_{PL}(x-0.801)^1 + R_{1y}(x-0.912)^1 + M_1^{xy}(x-0.912)^0 = -1.65Nm$$

From the above five intervals, the maximum bending moment in the xy plane occurs in the interval $0.12 \le x \le 0.215m$.

Hence, $M_{\max}^{xy} = 41.878Nm$.

- Maximum bending moment in the xz plane($M_{\max}^{xz}$):

In a similar pattern, using the singularity function for the bending moment in xz plane is given by:

$$M_{xz} = -M_2^{xz}(x)^0 + R_{2Z}(x)^1 - f_{BZ}(x-0.215)^1 + F_B(x-0.801)^1 + R_{1Z}(x-0.912)^1 + M_1^{xz}(x-0.912)^0$$

In the plane xz, there are there loading sections of the shaft, and the bending moment will be computed for each section as follows:

- For $0 \le x \le 0.215$, $M_{xz} = -M_2^{xz}(x)^0 + R_{2Z}(x)^1 = 35.63Nm$
- For $0.215 \le x \le 0.801$, $M_{xz} = -M_2^{xz}(x)^0 + R_{2Z}(x)^1 - f_{BZ}(x-0.215)^1 = -35.73Nm$
- For $0.801 \le x \le 0.912$,
  $M_{xz} = -M_2^{xz}(x)^0 + R_{2Z}(x)^1 - f_{BZ}(x-0.215)^1 + F_B(x-0.801)^1 + M_1^{xz}(x-0.912)^0 = -0.42Nm$

From the results, the maximum bending moment occurs in the interval of length $0.215 \le x \le 0.801$ m, which is greater than in the other intervals.

Hence, $M_{\max}^{xz} = -35.73Nm$

Therefore, the maximum bending moment M_{max} on the shaft to be used for the design is:

$$M_{\max} = \sqrt{\left(M_{\max}^{xy}\right)^2 + \left(M_{\max}^{xz}\right)^2} \tag{23}$$

$$\Rightarrow M_{\max} = \sqrt{(-36)^2 + (42)^2} = 55.142Nm$$

5.4.4. *Determination of Maximum Torsional Moments*

The following free body diagram shows the acting torsional moments.

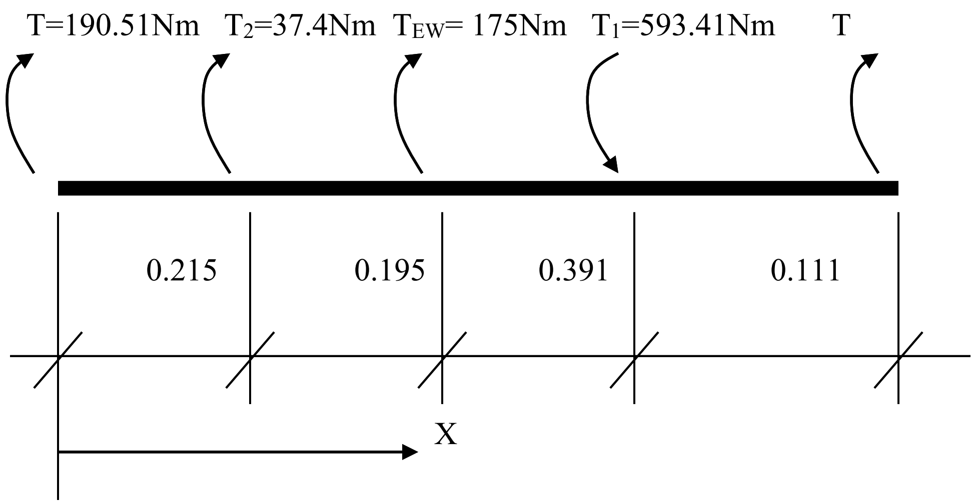

Figure 5.7: **Free body diagram of the Wheel shaft for torsion (dimensions are all in mm)**

- For $0 \le x \le 0.215$, $T^* = 227.91Nm$
- For $0.215 \le x \le 0.41$, $T^* = 402.91Nm$
- For $0.41 \le x \le 0.801$, $T^* = -190.51Nm$
- For $0.801 \le x \le 0.912$, $T^* = 0$

From the intervals, the maximum torsional moment is

$T^*_{\max} = 402.91Nm$. This is going to be used for design of the shaft in combination with the maximum bending moment $M_{\max} = 55.142Nm$.

5.4.5. *Determination of the shaft diameter (d)*

By using the maximum shear stress theory, the shaft diameter, is given as:

$$d = \left[\frac{32}{\pi \sigma_{all}} \left[(k_b M_{\max})^2 + (k_s M_{\max})^2 \right]^{\frac{1}{2}} \right]^{\frac{1}{3}} \tag{24}$$

Where $\sigma_{all} = 0.18\sigma_u or 0.36\sigma_y$ = allowable stress of the shaft material

σ_u = Ultimate strength of the shaft material

σ_y = Yield strength of the shaft material.

$k_b = 1.5$, bending shock factor for suddenly applied steady loads with moderate shocks.

$k_s = 1.5$, torsional shock factor for suddenly applied steady loads with moderate shocks. [19]

For the consideration of keyways and snap rings to mount pulleys, the allowable stress needs to be modified as

$\sigma_{all} = 0.75\sigma_{all}$

Using the properties of shaft material selected to be suitable for the working conditions,

$\sigma_u = 380MPa$ and $\sigma_y = 217MPa$.

Then, the minimum allowable strength modified for the design of the shaft is

$\sigma_{all} = 51.3MPa$

Therefore, the size of the shaft is determined $d = 5cm$ is the minimum diameter of the shaft for its static design.

Fatigue Analysis:

By using the maximum shear stress theory, the shaft diameter considering fatigue loading is given as:

$$d = \left[\frac{32(FS)}{\pi}\left[\left(\frac{M_{\max}}{\sigma_e}\right)^2 + \frac{3}{4}\left(\frac{T^*_{\max}}{\sigma_y}\right)^2\right]^{\frac{1}{2}}\right]^{\frac{1}{3}} \quad (25)$$

Where FS= Factor of Safety.

σ_e = Modified endurance limit of the shaft material

σ_y = Yield strength of the shaft material= 217MPa.

- Factor of safety Determination

For continuously acting loads, the safety factor FS for strength is expressed as:

$$FS = \frac{\sigma_u}{\sigma_y} R \tag{26}$$

Where σ_u = Ultimate strength of the shaft material

σ_y = Yield strength of the shaft material.

R = reliability =2 when predictable loads and medium weight is important.

Hence, $FS = \frac{380}{217} * 2 = 3.5$

- Determination of the modified endurance limit(σ_e)

The modified endurance limit (fatigue strength) is expressed as

$$\sigma_e = \frac{K_a K_b K_c K_d}{K_f} \sigma_e^* \tag{27}$$

Where K_a=stress concentration factor for surface finish

K_b= stress concentration factor for size

K_c= stress concentration factor for reliability

K_d= stress concentration factor for Impact

K_f= stress concentration factor for geometric change=1+q(K_t-1)

q= Notch sensitivity

K_t= theoretical stress concentration factor

σ_e^* = Endurance limit=170MPa

And therefore, K_a=0.8 for $\sigma_u = 380MPa$ read from charts.

K_b=1.85$d^{-0.19}$, for preliminary design let d=60mm

= 0.85

K_c= 0.868 for 0.95 reliability

K_d= 0.8 for moderate load type (impact)

K_f= 1.3 for K_t=1.3 from charts and q=1 for full notch sensitivity. [19]

Therefore, by the substitution of the above values in to the endurance limit-safety factor empirical relation

$$\sigma_e = \frac{K_a K_b K_c K_d}{K_f}\sigma_e^* = 61.748MPa$$

Therefore, using the equation of maximum shear stress theory for in fatigue loading, the shaft size is determined to be:

d= 50mm is the minimum shaft diameter in fatigue loading condition which is approximately equal to the diameter achieved in the static design of the shaft. Therefore, the shaft size is d=5cm.

5.5. Gear Drive Design

This power drive is to be designed in the purpose of direction change of the power transmission through an angle of 90^0, and for this working condition, bevel gear is highly suitable. The design consideration of the gear are the application requirements, gear selection, recommended gear and pinion sizes, recommended number of teeth, recommended face width, diametral pitch of the gear and the pressure angle. Once these design considerations are at hand, the design can be said started.

Gear Design considerations are the application requirements, gear selection, recommended gear sizes, pinion pitch diameter, recommended number of teeth, face width, diametral pitch, pressure angle, circular pitch, pitch angle, cutter selection, center distance, addendum and dedendum of the gear, dynamic load, wear, and gear shaft load.

5.5.1. Application Requirements

These considerations comprise of specifications or purpose of the drive, the load as the power rating to be transmitted from the prime mover, the required speed in rpm, the special operating conditions have to be clearly defined as the first step.

Hence, the gear drive is required:

- To transmitte 3hp as power rating from the prime mover
- To have 800 rpm on the gear
- To have a gear ratio of 1.
- The pinion and gear shafts of the drive have to be intersecting at 90^0 shaft angle.

5.5.2. Selection of the gear

The bevel gear can be in straight, spiral or hypoid form. All of these types can be used for 90^0 direction change of power transmission, but from manufacturing cost view point, bevel gears of straight type are cheaper than the others, and straight bevel gears are recommended for peripheral speeds of gears up to 5.08 m/s where maximum smoothness and quietness are not of prime importance. [19]

Therefore, since the peripheral speed required is less than 5.08 m/s, a straight bevel gear type is selected to be designed.

5.5.3. Recommended Gear size (Pitch diameter PD_g)

This parameter is determined based on the torque to be transmitted and the desired gear ratio. The gear ratio is determined to be 1 and the torque T to be transmitted can be computed as:

$P = T * \omega$ Where $\omega =$ the angular speed o the gear=83.8 rad/sec. (28)

$P =$ The power rating= 3hp=2238 Watts

Then, $T = \frac{2238}{83.8} = 26.71 Nm$

Therefore, the recommended gear pitch diameter PD corresponding to the torque $T = 26.71 Nm$ and gear ration of 1 is found from charts to be PD= 2in = 5.08cm. This gear pitch diameter can be used for all type of bevel gears by applying correction factors based on the gear material, gear type either it is straight, spiral, or hypoid bevel gear, the surface durability required, the strength, and the working conditions as such as for statically loaded gears subjected to vibration.

Hence, this pitch diameter desires modification by the application of factors as follows:

a. The material for bevel gear is optionally selected to be heat-treated steel with Brinell hardness number ranging from 210 to 245 HB for the pinion of the drive. For this gear material, the material factor C_M is found to be 1.65 from standard table for the surface durability and strength charts for materials with class and requirements of ASTM class 30,SAE 111.[20]
b. Among the types of bevel gear, a straight bevel gear is selected for the variety of reasons of cost and speed limitations, and the factor for the gear type selected is found to be recommended as $C_1 = 1.2$.
c. For statically loaded straight-bevel gears subjected to vibration, the recommended multiplying factor is found to be $C_2 = 0.7$.

Therefore, the corrected pitch diameter of the gear can be determined by multiplying it by these factors as:

PD_g = 0.7*1.2*1.65* 5.08cm = 7.05cm. This is the corrected gear pitch diameter to be used for the gear drive design.

5.5.4. Determination of the pinion pitch diameter (PD_p)

The pinion pitch diameter can be estimated by the relation:

$$PD_P = \frac{PD_g n}{N} \quad \text{Where } PD_P = \text{gear pitch diameter} \tag{29}$$

$$\frac{n}{N} = \text{The reciprocal of gear ratio} = 1 \text{ because n=N.}$$

Hence, PD_P=7.05cm

5.5.5. Recommended number of gear teeth(N)

The recommended number of gear teeth corresponding to its pitch diameter is presented in charts by different design hand books. From standard charts of recommended number of gear teeth against gear pitch diameter, the number of gear teeth is found to be N=24. [21]

5.5.6. Determination of face width(F)

For all types of bevel gears, it is recommended that the face width of a gear should not exceed 30% of its cone distance, and on this basis, charts are presented showing recommended face width for its corresponding pitch diameter.

Hence, from standard charts, the face width F corresponding to the pitch diameter PD=7.05cm=2.78in is read as:

F=0.6in=1.524cm.

5.5.7. Determination of diameteral pitch (pitch p)

Diametral pitch is the number of teeth on the gear per unit length of the pitch diameter. It is defined by the following relation:

$$P = \frac{N}{PD_g} = \frac{24teeth}{7.05cm} = 3.40\frac{teeth}{cm} \text{ of the pitch diameter.} \qquad (30)$$

5.5.8. Determination of circular pitch (P)

The circular pitch is the sum of the tooth free space and tooth width measured along on the pitch circle. It is defined as:

$$p = \frac{\pi(PD_g)}{N} = \frac{\pi * 7.05}{24} = 0.9228cm. \qquad (31)$$

5.5.9. Pressure Angle

Practically, the commonly used pressure angle for all bevel gears designed today is 20^0, and hence this value will be adopted in this design of gear drive.

5.5.10. Determination of pitch angle (Center angles α_1 and α_2)

For the shaft angle of 90^0, the pitch angle α_1 and α_2 are given in terms of the number of teeth of the gear and pinion as:

$$\tan\alpha_1 = \frac{N_1}{N_2} \text{ and } \tan\alpha_2 = \frac{N_2}{N_1} \qquad (32)$$

where N_2=gear number of teeth

N_1=pinion number of teeth

But $N_1 = N_2 = N$, this implies that $\alpha_1 = \tan^{-1}(1) = 45^0$ and $\alpha_2 = \tan^{-1}(1) = 45^0$

And hence, $\alpha_1 = \alpha_2 = \alpha = 45^0$

5.5.11. Cutter Selection and its corresponding virtual number of teeth (N)

The teeth can be cast, milled, or generated. But for the purpose of milling or generating, the gear tooth cutter has to be selected based on the virtual number of teeth N'. For the purpose of determining the virtual number of teeth corresponding to the cutter, the Tredgold's approximation can be employed, and the virtual number of teeth in the imaginary gear to be used for generating or milling the tooth defined as:

$$N' = \frac{2\pi(PD_g)}{2p\cos\alpha} \quad \text{(33)}$$

Where PD_g= pitch diameter of the gear

$p =$ Circular pitch. [20]

$$N' = \frac{2 * \pi * 7.05}{2 * 0.9228 * \cos 45^0} = 33.94 \approx 34 \text{ Teeth.}$$

Therefore, the tooth cutter will be selected based on the virtual number of teeth of the imaginary generating gear:

N'=34teeth.

5.5.12. Determination of Center distance (L)

From analysis of beam strength for straight-tooth bevel gears, Lewis equation for dynamic loads can be adopted for the determination of the cone distance. The Lewis equation for dynamic loads is given by:

$$F_{\max} = \frac{F k_2 Y C_v \sigma_e}{k_f k_1 P}\left(1 - \frac{F}{L}\right) \quad \text{(34)}$$

Where F_{max}=Maximum tangential load on the tooth a radius r from the axis of gear rotation.

C_v=velocity factor

F= Face width

K_2= form factor coefficient= 1.7 for bevel gears.

Y=Lewis Form factor based on the virtual number of teeth

$\sigma_e =$ Endurance limit

K_f= fatigue stress concentration factor

K_1= service factor =1.5 for shock loads.

P= diametral pitch.

L= Cone distance

Then,

- $F_{\max} = \frac{T_{\max}}{r} = \frac{2T_{\max}}{PD_g} = \frac{2*26.71}{0.0705} = 757.73N$
- Face width, F=0.01524m
- Lewis Form factor, Y=0.371 corresponding to the virtual number of teeth N'=34 and 20^0 pressure angle.
- Velocity factor, C_v=0.81 for velocity at pitch radius V=2.954m/s=581.5fpm
- Endurance limit, σ_e = 50,000Psi corresponding to the smallest brinell hardness number of the gear material (steel) for one-way bending as far as the rotation is not reversed. Hence, σ_e =344.75 MPa.
- Fatigue stress concentration factor, k_f=1.5 corresponding to the diametral pitch P=3.4 teeth/cm=8.636 teeth/in.
- Diametral Pitch, $P = \frac{3.4}{0.01} = 340\frac{teeth}{m}$

Therefore, by using the above relation of the center distance and the other parameters, the center distance can be determined as L=1.943cm.

Therefore, the cone distance taking the next larger value is L= 2cm= 20mm.

5.5.13. Calculation of Addendum, **a** *and Dedendum,* **b**

Addendum **a** is defined as the reciprocal of diametral pitch P,

i.e., $a = \frac{1}{P} = \frac{PD_g}{N}$ (35)

Then, $a = \frac{1}{P} = \frac{PD_g}{N} = \frac{1}{3.4} = 2.94mm/teeth$

And dedendum, **b**, is also defined as:

$b = 1.25(\frac{1}{P}) = 1.25(\frac{PD_g}{N}) = \frac{1.25}{3.4} = 3.6765mm/teeth$

Then, the uniform clearance is b-a = 0.7365mm.

5.5.14. Dynamic load and Wear

- The approximate Buckingham equation for dynamic load can be used for bevel gears as: $W_d = W + \dfrac{0.05VF_2}{0.05V + \sqrt{F_2}}$ (36)

Where W_d= dynamic tooth load at mean radius, lb

W=equivalent transmitted tangential tooth load at mean radius of gear, lb

V=pitch line velocity at average pitch radius, fpm.

F_2= force required to deform teeth amount of effective error, lb

= FC+W

F=face width, in.

C= deformation factor obtained from tables.

The above parameters can be computed as:

- $W = \dfrac{T}{r_{average}} = \dfrac{T}{\frac{PD_g}{2} - \frac{F\cos 45^0}{2}} = \dfrac{26.71}{0.03525 - \frac{0.01524\cos 45^0}{2}} = 894.45N = 201.08lb$
- $V = \omega r_{average} = \left(\dfrac{800*2\pi}{60}\right)\left(\dfrac{PD_g}{2} - \dfrac{F\cos 45^0}{2}\right)$ = 2.5m/s = 492.125fpm.
- $F_2 = FC + W = ((0.6*3440) + 201.08) = 2265.08lb$, for error in action 0.002 in, C=3440.

Then, the dynamic load, W_d is

$$W_d = W + \frac{0.05VF_2}{0.05V + \sqrt{F_2}} = 201.08 + \frac{0.05*492.125*2265.08}{(0.05*492.125) + \sqrt{2265.08}} = 973.044lb = 4328.3N$$

- Limiting load for wear of the tooth(W_w):

A wear equation can be adopted for use with bevel gears by using the virtual spur gear of generation taken at the mean radius of the bevel gear. The equation for wear load W_w is:

$$W_w = D'_{average} f_{eq} kQ \quad (37)$$

Where $D'_{average}$ = diameter of the back cone circle obtained from average bevel-pinion pitch diameter.

f_{eq} = Equivalent face width, 0.75F for outboard pinion and F for inboard pinion, in.

$k =$ Load-stress factor

$Q = \frac{2N_1^{'}}{N_1^{'} + N_2^{'}} = \frac{2N^{'}}{2N^{'}} = 1$, ratio factor.

And $N_1^{'} =$ Virtual number of teeth of the pinion

$N_2^{'} =$ Virtual number of teeth of the gear.

- $$D_{average}^{'} = 2\left(\frac{PD_g}{2} - \frac{F\cos 45^0}{2}\right) \quad (38)$$

  $\Rightarrow D_{average}^{'} = 0.05972m$

- $f_{eq} = F = 1.524cm = 0.01524m$
- $k = 106Psi = 1.3514MPa$, for steel-steel with Brinell hardness number 280BHn and 20^0 pressure angle bevel gear.

Then, the limiting load for wear, W_w, is

$W_w = D_{average}^{'} f_{eq} kQ = 0.05972 * 0.01524 * 1.3514 * 10^6 = 1229.93N$.

This is the load at which and beyond which wear of the tooth occurs, but as long as the loads on the gears are far below this load, wear of the tooth is much less probable to occur. The same is true for the dynamic load.

5.5.15. Gear Shaft Loads

In bevel gearing, there are loads induced due to the presence of tangential load of the torque from the power source. These loads are the radial and thrust components which have prime importance in the design of the gear shafts and bearings. For the determination of the radial and thrust loads, the tangential load, W, needs to be computed as:

$$W = \frac{T}{r_{average}} \quad (39)$$

Where T= the torque transmitted and

$r_{average}$= the pitch radius of the gear at the midpoint of the tooth.

Then, $W = \frac{T}{r_{average}} = \frac{T}{\frac{PD_g}{2} - \frac{F\cos 45^0}{2}} = \frac{26.71}{0.03525 - \frac{0.01524\cos 45^0}{2}} = 894.45N = 201.08lb$

And from the trigonometry of the gear drive, the radial and thrust components are given by:

$W_R = W \tan\phi \cos\alpha$

$W_T = W \tan\phi \sin\alpha$ (40)

Where W_R = Radial component.

W_T = Thrust component.

ϕ = Pressure angle=20^0.

α = Pitch angle=45^0, the same for both meshing gear and pinion.

W = The tangential load. [20]

Hence, $W_R = 894.45 \tan 20^0 \cos 45^0 = 230.2N$

and $W_T = 894.45 \tan 20^0 \sin 45^0 = 230.2N$

These are the loads to be used for the design of the gear shafts and their bearings.

5.6. Driving Gear Shaft Design

The shaft to be designed is apart used to transmit the 3hp of the rated power of the engine in to the mowing mechanism of the cam shaft through the gear it supports. This shaft contains the smaller pulley of the second belt drive, one of the bevel gear drive, and two supporting bearings, and hence it is subjected to transverse loads, torsional loads, and axial loads due to the weight of the pulley, the belt tensions, and the bevel gear radial and thrust loading, respectively.

The free body diagram of the shaft is shown below:

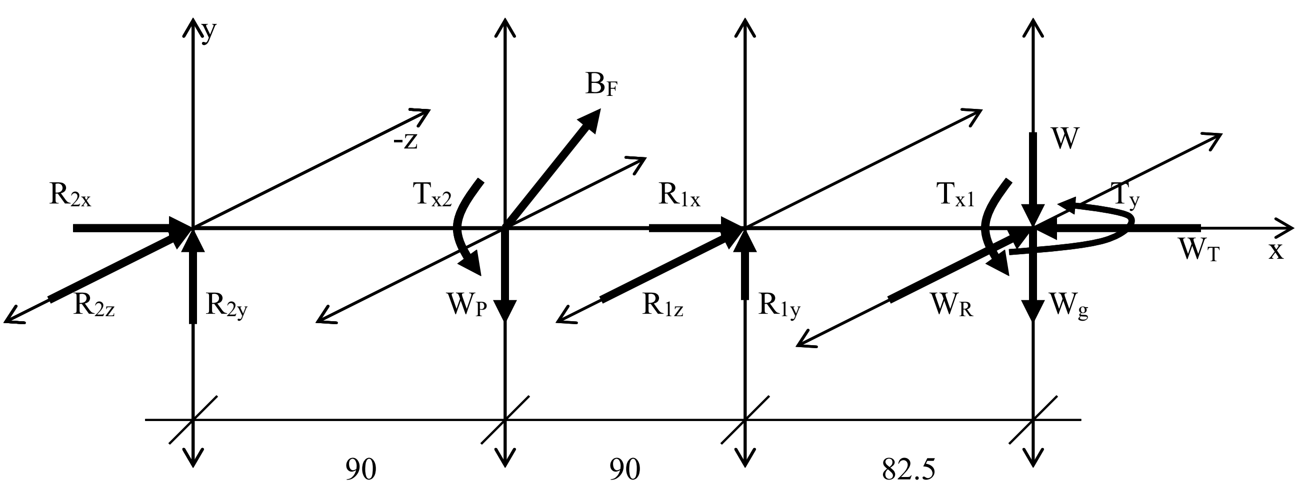

Figure 5.8: **Free Body Diagram of the gear shaft**

The dimensions are taken preliminarily for initial design specification in mm.

Where R_{1z}, R_{1y}, R_{2z}, R_{2y}, R_{1x}, R_{2x}= bearing reactions at the points shown

W_P= weight of the smaller pulley of the second belt drive.

B_F= Resultant belt tension on the shaft for the belt drive orientation.

T_{x2}=Torsion due to the belt tensions about x-axis

W_R= Radial load due to the gear

W_g= Equivalent weight of the gear

W_T=Thrust load due to the gear

T_{x1}=Torsion due to the gear tangential load about x-axis

W=Tangential load of the driven gear

T_y= Bending moment due to the thrust load about the y axis

The design of the shaft will be carried out for both static and dynamic conditions with the following design specifications, decisions and shaft material.

Design Specifications:

- Power to be transmitted, P=3hp.
- Speed of the shaft, N_S=800rpm.

Design Decisions:

- Maximum Shaft length, L_s=400mm=0.4m
- The shaft is modeled as a simply supported beam based on the loading conditions as shown in the free body diagram.

Shaft Material:

- The material for this shaft is taken to be similar to that of the wheels shaft material and the mechanical properties can be used from the design of the wheels shaft.

Static Analysis

5.6.1. Determination of the external loads

The applied loads on the shaft that are shown on the free body diagram are W_P, B_F, T_{x2}, W_R, W_g, W_T, T_y, W, and T_{x1}. The magnitude of these loads in the indicated respective directions will be computed under here.

- The weight of smaller pulley of the second belt drive, W_P, is computed in the design of the second belt drive to be: $W_P = 83.3N$

- The resultant belt tension load on the shaft, B_F, is computed based on the worst load contribution of the tensions on the shaft for its design and for this reason, the load B_F is the sum of the belt tensions.

Hence, $B_F = F_1 + F_2$ where $F_1 = 575.55N$ and $F_2 = 41.22N$

$\Rightarrow B_F = 616.77N$

- The torsion due to the belt tension of the second belt drive, T_{x2}, is calculated by using the belt tensions at the arm length of the smaller pulley pitch radius.

Then, $$T_{x2} = \frac{(F_1 - F_2)d}{2} \quad (41)$$

Where F_1 and F_2 =belt tensions on the tight and slack side of the belt.

d= pitch diameter of the smaller pulley of he second belt drive=140mm.

Hence, $T_{x2} = \frac{(575.55 - 41.22) * 0.14}{2} = 37.4Nm$

- The gear radial load, W_R, is determined in the gear design of the shaft loads to be: $W_R = 230.2N$.

- The equivalent weight of the gear, W_g, can be computed on the basis of the equivalent weight of a cylindrical gear blank with base radius of the pitch circle for the worst condition of loading of the shaft. Then, the gear weight is given by:

$$W_g = V\rho g \quad (42)$$

Where V = volume of the cylindrical blank

ρ = Density of the gear material

g = Gravitational acceleration= 10m/s

For the gear material of steel with Brinell hardness number of 280BH, the density is 7770kg/m^3, the pitch diameter of the gear is taken from the gear design to be 7.05cm, and the height of the gear blank, h is determined empirically to be 3.233cm.

Hence, $$W_g = \frac{\pi (PD_g)^2}{4} h\rho g \quad (43)$$

$$W_g = \frac{\pi (0.14)^2}{4} 0.03233 * 7770 * 10 = 9.81N$$

- The gear thrust, W_T, is determined in the design of he gear drive to be $W_T = 230.2N$.
- The torsion due to the gear tangential load on the shaft, T_{x1}, is computed as

$$T_{X1} = \frac{W(PD_g)}{2} \quad (44)$$

where W=tangential load on the gear from the gear design.

PD_g=pitch diameter of the gear from the gear design.

Then, $T_{X1} = \frac{894.45 * 0.0705}{2} = 31.53Nm$.

- The tangential load of the gear, W is computed in the gear design to be W=894.45N.
- The bending moment due to the thrust on the gear, T_y, is the couple of the thrust load on the gear at arm length of the gear pitch radius.

Hence, $T_y = \frac{W_T(PD_g)}{2} = \frac{230.2 * 0.0705}{2} = 8.115Nm$

5.6.2. Determination of the reaction loads (R_{1z}, R_{1y}, R_{2z}, R_{2y})

These reaction loads are acting at the bearing in the x, y, and z axes as shown in the free body diagram, and they are going to be determined for their magnitude in the directions shown.

- Reactions in the y axis(R_{1y} and R_{2y}):

Taking the sum of forces in the xy plane and y direction for the static equilibrium of the shaft,

$$\sum F_y = 0 \qquad (45)$$

$$\Rightarrow R_{2y} + R_{1y} + B_F \sin 30^0 - W_P - W_g - W = 0$$

$$\Rightarrow R_{2y} + R_{1y} = 679.18N$$

And considering the sum of bending moments about one end of the shaft in the xy plane,

$$\sum M_{xy} = 0 \qquad (46)$$

$$\Rightarrow 0.18R_{1y} + 0.09 * 616.77 * \sin 30^0 - 0.09 * 83.3 - 0.2625 * 9.81 - 894.45 * 0.2625 = 0$$

$$\Rightarrow R_{1y} = 1206.2N \text{ and } R_{2y} = -526.99N$$

- Reactions in the z-axis (R_{1z} and R_{2z})

For equilibrium of the forces on the shaft in the z-direction,

$$\sum F_z = 0$$

$$\Rightarrow R_{2z} + R_{1z} + B_F \cos 30^0 + W_R = 0$$

$$\Rightarrow R_{2z} + R_{1z} = -764.34N$$

And from equilibrium of moments at the right hand end of the shaft,

$$\sum M_y = 0 \qquad (47)$$

$$\Rightarrow 0.0825R_{1z} + 0.2625R_{2z} + 0.1725B_F \cos 30^0 - T_y = 0 \quad \Rightarrow 0.0825R_{1z} + 0.2625R_{2z} = -84.024$$

Solving simultaneously,

$$R_{1z} = -647.84N \text{ and } R_{2z} = -116.5N$$

- Reactions in the x-direction(R_{1x} and R_{2x})

These reactions are of prime importance for the bearing selecting, and hence it is enough to determine only their sum. Hence, from equilibrium of forces in the x-direction,

$$\sum F_x = 0 \quad (48)$$

$$\Rightarrow R_{2x} + R_{1x} - W_T = 0$$

$$\Rightarrow R_{2x} + R_{1x} = 230.2N$$

5.6.3. Determination of the maximum bending moment ($M_{\max}^{xy}$ and $M_{\max}^{xz}$)

In determining the maximum bending moment, the bending moments in each section of the shaft corresponding to the loads in the xy and xz planes will be determined and the resultant of the maximum bending moments of each plane will be the maximum bending moment of the shaft to be used for its design.

- Maximum bending moment in the xy plane($M_{\max}^{xy}$)

Using singularity function of the bending moment in the xy plane,

$$M_{xy} = R_{2y}(x)^1 + (B_F \sin 30^0)(x-0.09)^1 + R_{1y}(x-0.18)^1 - W_P(x-0.09)^1 + W_g(x-0.2625)^1 - W(x-0.2625)^1$$

For the three sections of the shaft, the bending moment will be computed as follows:

For $0 \le x \le 0.09$, $M_{xy} = R_{2y}(x)^1 = -526.99 * 0.09 = -47.43Nm$

For $0.09 \le x \le 0.18$, $M_{xy} = R_{2y}(x)^1 + (B_F \sin 30^0)(x-0.09)^1 - W_P(x-0.09)^1 = -74.601Nm$

For $0.18 \le x \le 0.2625$,

$$M_{xy} = R_{2y}(x)^1 + (B_F \sin 30^0)(x-0.09)^1 + R_{1y}(x-0.18)^1 - W_P(x-0.09)^1 = 0.0038Nm$$

Then, the maximum bending moment in the xy plane, $M_{\max}^{xy}$ is found to be $M_{\max}^{xy} = 74.601Nm$

- Maximum bending moment in the xz plane($M_{\max}^{xz}$)

Setting the singularity function of bending moment in the xz-plane,

$$M_{xz} = R_{2z}(x)^1 + (B_F \cos 30^0)(x-0.09)^1 + R_{1z}(x-0.18)^1 + W_R(x-0.2625)^1 + T_y(x)^0$$

For $0 \le x \le 0.09$, $M_{xz} = R_{2z}(x)^1 = -116.5 * 0.09 = -10.5Nm$

For $0.09 \le x \le 0.18$, $M_{xz} = R_{2z}(x)^1 + (B_F \cos 30^0)(x-0.09)^1 = 27.1025Nm$

For $0.18 \le x \le 0.2625$,

$$M_{xz} = R_{2z}(x)^1 + (B_F \cos 30^0)(x-0.09)^1 + R_{1z}(x-0.18)^1 + W_R(x-0.2625)^1 + T_y(x)^0 = 16.23Nm$$

Then, the maximum bending moment in the xz plane, $M_{\max}^{xz}$ is found to be:

$$M_{\max}^{xz} = 27.1025Nm$$

Therefore, the resultant maximum bending moment is then,

$$M_{\max} = \sqrt{\left(M_{\max}^{xy}\right)^2 + \left(M_{\max}^{xz}\right)^2} \qquad (49)$$

$$\Rightarrow M_{\max} = 79.372Nm$$

5.6.4. Determination of the maximum torsional moment($T_{\max}$)

The torsional moment will be computed for the three sections of the shaft as follows:

For $0 \le x \le 0.09$, $T = T_{x2} = 37.4Nm$

For $0.09 \le x \le 0.18$, $T = T_{x2} = 37.4Nm$

For $0.18 \le x \le 0.2625$, $T = T_{x2} - T_{x1} = 5.87Nm$

Therefore, the maximum torsional moment among the three sections is $T_{\max} = 37.4Nm$

5.6.5. Calculation of the shaft diameter(d)

Using the maximum shear stress theory, the shaft diameter, is given by:

$$d = \left[\frac{32}{\pi\sigma_{all}}\left[\left(k_b M_{\max}\right)^2 + \left(k_s M_{\max}\right)^2\right]^{\frac{1}{2}}\right]^{\frac{1}{3}} \text{and } \sigma_{all} = 0.18\sigma_u \text{ or } \sigma_{all} = 0.36\sigma_y \qquad (50)$$

Where σ_{all} = allowable stress of the shaft

σ_u = Ultimate strength of the shaft material

σ_y = Yield strength of the shaft material

$k_b = 1.5 =$ Bending shock factor for suddenly applied steady loads with moderate shocks

$k_s = 1.5 =$ Torsional shock factor for suddenly applied steady loads with moderate shocks

Considering keyways and snap rings grooves to mount the pulley and the gear, the allowable stress is modified as: $\sigma_{all} = 0.75\sigma_{all}$

Selecting the shaft material to be similar to the wheels shaft material

$\sigma_{all} = 51.3MPa$

Then, the size of the shaft through the substitution of the above values will be $d = 3cm$ taking the larger standard shaft size.

Fatigue Analysis

Using the maximum shear stress theory, the shaft size for fatigue loading is give as:

$$d = \left[\frac{32(FS)}{\pi} \left[\left(\frac{M_{\max}}{\sigma_e} \right)^2 + \frac{3}{4} \left(\frac{T^*_{\max}}{\sigma_y} \right)^2 \right]^{\frac{1}{2}} \right]^{\frac{1}{3}} \qquad (51)$$

Where FS= Factor of Safety.

σ_e =Modified endurance limit of the shaft material

σ_y = Yield strength of the shaft material= 217MPa.

- The safety factor for the shaft material similar to the wheels shaft one and equal reliability, SF is $FS = 3.5$.
- And the modified endurance limit, σ_e similar to the wheels shaft, is $\sigma_e = 61.748MPa$

Therefore, the size of the shaft for the fatigue loading is determined to be as: $d = 4cm$ taking the larger standard shaft size.

Hence, when the shaft is designed both for static and fatigue loading, the shaft size for the fatigue is greater.

Therefore, the shaft diameter will be $d = 4cm$ so that the shaft will not fail both for the static and fatigue loadings.

5.7. Cam Design

A cam is an element of the cam-follower mechanical system that compels the movement of the follower by direct contact. The motion of the follower is the result of a program, just as a computer is programmed, so is a cam. Thus, the system can be thought of as a mechanical information device. Accordingly, the design is done as to build a program, establish the locus of the contact points between the cam and follower, and produce the cam profile coordinates system.

Cam-follower mechanisms are found in almost all mechanical devices and machines, i.e., agriculture, transportation equipment, textiles, packaging, machine tools, printing presses, automobile internal combustion engines, food processing machines, switches, and control systems, and more recently in micro machines such as microelectromechanical systems [MEMS].

5.7.1. Cam-follower System Criteria

The cam-follower system may be designed for path, motion, or function generation. In the mower design, the cam and follower mechanism is treated almost totally as a function generator in which the output of the follower is a function of the cam input. [21] The cam moves the cutter bar in such a way that the bar mows the harvest as the function generated. Hence, the cam design is carried out for the function generation criterion.

5.7.2. Follower types

Cam follower systems are classified by referring to the follower or the cam or both. Considering the follower first, the follower movement is translation, oscillation, or indexing. The follower surface may be knife-edge, flat, curved, or roller. The follower restraint to the cam is positive-driven by the use of rollers in the cam groove or multiple conjugate cams, is spring-loaded, or occurs by gravity. Also, the translating follower line of motion with reference to the cam center may be radial or offset.

The **knife-edge, or point**, follower (Fig.5.9.a), is, as the name implies, a sharp edge in contact with the cam. Although simple in construction, this type of follower is not practical because it results in excessive wear of the contact point. It is employed in design as the center of the roller follower.

For proper performance, the follower is constrained to the cam at all speeds. The **roller follower**, Fig.5.9.b, is the most popular design for accomplishing this criterion. Commercially available roller followers use ball or needle bearings supported by a stem. **Positive drive action** is accomplished by a roller follower internally guided in a cam groove or track, by followers on the opposite side of a single cam (Fig.5.10.Yoke Cam), or by conjugate dual disk cams (Fig.5.11). The roller follower has a low coefficient of friction when compared to the other followers and is most frequently used in production machinery and some automotive engines.

In contact with the cam surface the follower roller action at low speeds can be that of pure rolling; however, at higher speeds significant sliding is evident. In groove cams the fluctuation in roller speeds is the result of the driven rotational acceleration of the roller as it rides on different radial surfaces of the cam. Experience has shown that the grooved cam roller follower does not provide exact positive-driven action because of the necessary clearance (backlash) between the roller sides and its groove. Sliding, wear, noise, vibration, and shock may be induced at high speeds. Furthermore, high speeds and high load may necessitate the use of crowned or conical rollers to accommodate the potential misalignment of contacting surfaces to avoid excessive surface stresses and wear.

Figures 5.9.c and 5.9.d show **mushroom** followers in which the contacting surface is either flat or spherical. The spherical face of the Figure 5.9.d mushroom follower has a large radius that compensates for detrimental deflection or misalignment that may occur with the flat mushroom follower.

The **radial follower** is one in which the follower translates along an axis passing through the cam center of rotation. This type, shown in Fig.5.9, is the most common.

The **offset follower** is one in which the axis of the follower movement is displaced from the cam center of rotation. Offsetting often improves action by reducing force, stress, and also the cam's size. The eccentricity should be in the direction that improves force components tending to jam the translating follower in its bearing guide. Figure 5.12.a shows a follower on a radial cam with an offset shown. Figure 5.12.b shows the same relationship for a translating cam. In both cams, the follower path is not the profile displacement of the cam. [21]

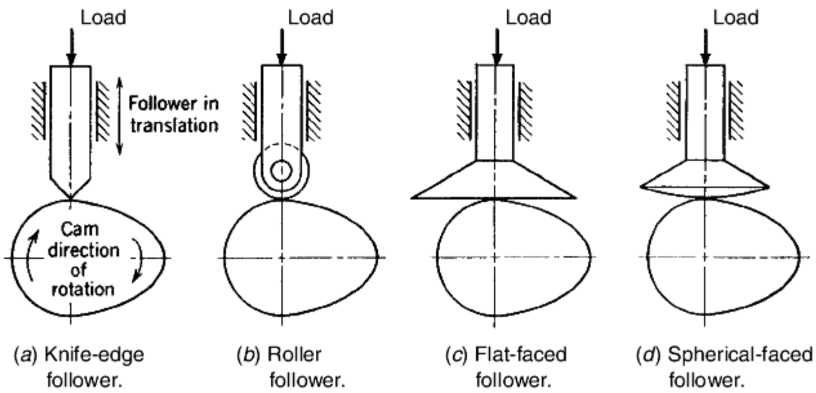

Figure 5.9: **Types of follower contact surfaces. [21]**

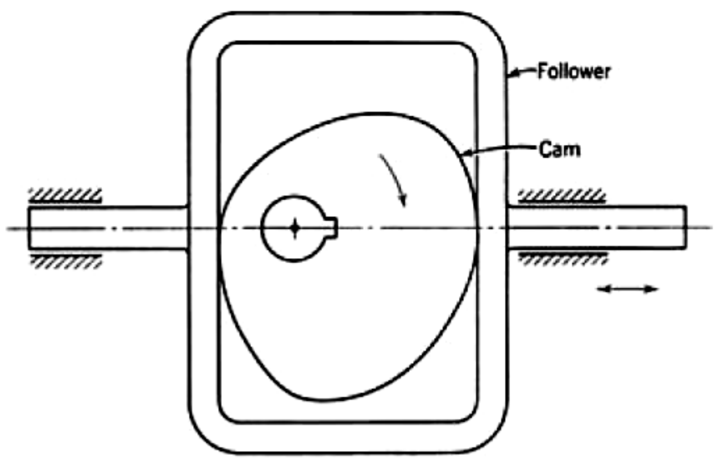

Figure 5.10: **Yoke Cam [21]**

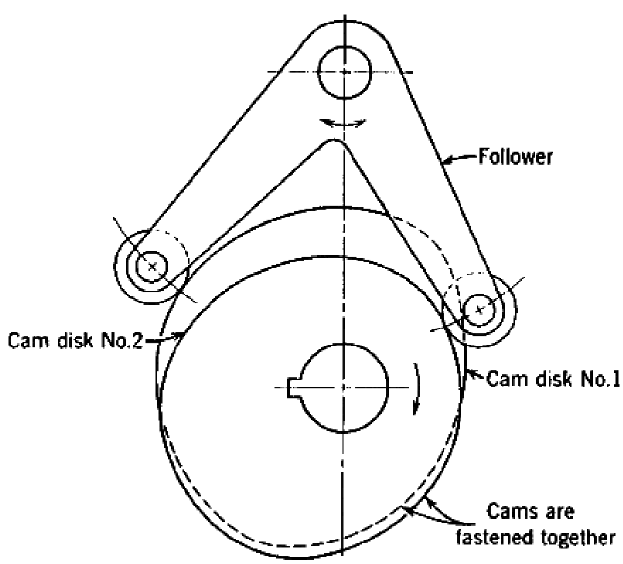

Figure 5.11: **Conjugate cam-oscillating follower (dual rollers and cams) [21]**

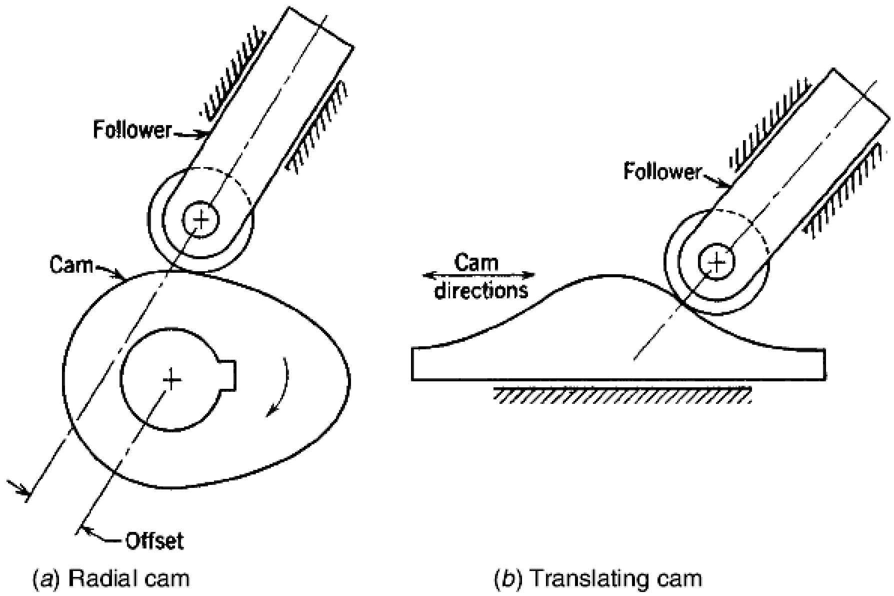

Figure 5.12: **Offset followers [21]**

5.7.3. Cam classifications

Cams are classified in three ways:

- In terms of their shape, such as wedge, radial, cylindrical, globoidal, conical, spherical, or three-dimensional;(Fig.5.14)
- In terms of the follower motion, such as dwell-rise-dwell (DRD), dwell-rise-return dwell (DRRD), or rise-return-rise (RRR);(Fig.5.13)
- In terms of the follower constraint, which is accomplished by either positive drive or spring load as mentioned previously. [21]

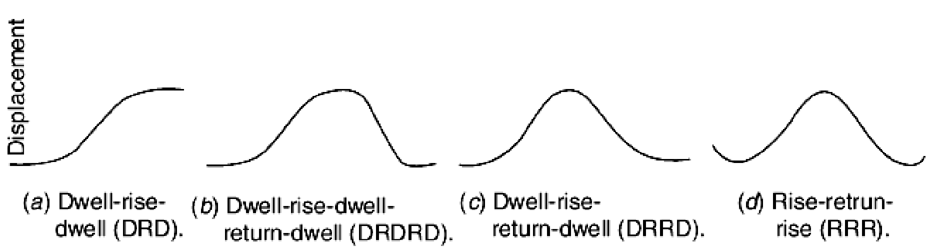

Figure 5.13: **Types of cams in terms of follower motion [21]**

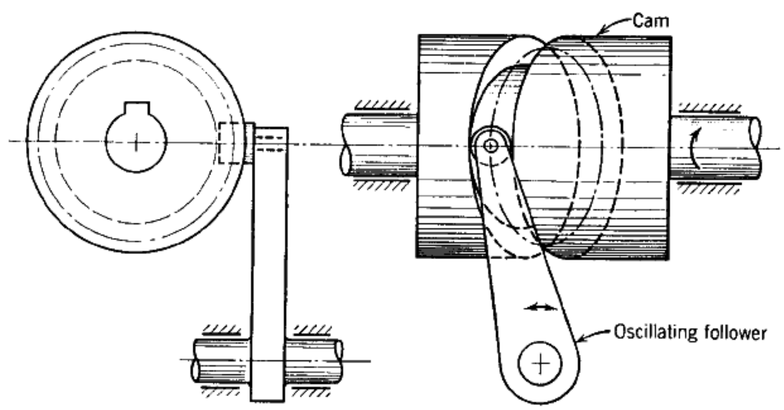

Figure 5.14:**Cylindrical cam-oscillating roller follower (a)[21]**

5.7.4. Basic Curves

For designing a cam, it is necessary to provide an accurate mathematical information for the cam characteristics of displacement velocity, acceleration, and sometimes the jerk. In so doing one can interpret and control the ultimate design performance. Also, the higher the cam speed, the more critical is the investigation. This is especially true of the acceleration data which is the important determining factor of the dynamic loads on cam-follower systems. Cams can be designed for any acceptable curve or shape. They include the ellipse, parabola, hyperbola, logarithmic spiral, and involutes of a circle. Alternative combinations of these curves combined with the circle and straight line can also be used. Generally, this subtopic presents mathematically established basic curves of the cam to be designed which are the first selections to establish the follower action. They are easy to analyze and manipulate.

5.7.5. Follower characteristics

A cam can be considered similar to a wedge having a cyclical rise and fall which establishes the motion of the follower. In all cams, the displacement of the follower is given by the mathematical relationship

$$y = f(\theta) \tag{52}$$

Where $\theta =$ cam angle rotation in radians. However, since the cam rotates at a constant angular velocity, the displacement can also be written as

$$y = g(t) \tag{53}$$

and $\theta = \omega t$ (54)

Where t = time for cam to rotate through angle θ, sec

$\omega =$ Cam angular velocity, rad/sec

By the use of Eq. (52) the follower characteristics can be normalized (dimensionless) as follows:

The cam profile is usually given as a function of the angle θ. Thus

$$y = \text{Follower displacement} \quad (55)$$

The instantaneous angular rate of change of displacement

$$y' = \frac{dy}{d\theta} = \text{Follower Velocity.} \quad (56)$$

The instantaneous angular rate of change of velocity

$$y'' = \frac{d^2 y}{d\theta^2} = \text{Follower Acceleration.} \quad (57)$$

The instantaneous cam angle rate of change of acceleration

$$y''' = \frac{d^3 y}{d\theta^3} = \text{Follower Jerk} \quad (58)$$

By the use of Eq. (53) the follower characteristics can be expressed as direct time dependent as follows:

The follower velocity can be written as

$$\dot{y} = \frac{dy}{dt} = \left(\frac{d\theta}{dt}\right)\left(\frac{dy}{d\theta}\right) = \omega\left(\frac{dy}{d\theta}\right) = \omega y' \quad (59)$$

The follower acceleration

$$\ddot{y} = \frac{d^2 y}{d^2 t} = \frac{d}{dt}\left(\left(\frac{d\theta}{dt}\right)\left(\frac{dy}{d\theta}\right)\right) = \omega \frac{d}{d\theta}\frac{dy}{d\theta}\frac{d\theta}{dt} = \omega^2\left(\frac{d^2 y}{d\theta^2}\right) = \omega^2 y'' \quad (60)$$

And the follower Jerk

$$\dddot{y} = \omega^3\left(\frac{d^3 y}{d\theta^3}\right) = \omega^3 y''' \quad (61)$$

These equations facilitate converting from one family of dimensional units to another. By having the cam profiles in the mathematical form of Eq. (52) or Eq. (53) it is also easily possible to find the other characteristics by differentiation.

From the foregoing equations, it can be seen that each successive derivative can be determined from the slopes of any point on the previous curve. Thus, at any point the slope of the displacement curve yields the velocity of the follower. The slope of the velocity curve is the acceleration of the follower, and the slope of the acceleration curve is the jerk of the follower. This procedure is called the graphical-slope differentiating method. It is presented for a quick appraisal of the displacement, velocity, acceleration, and jerk curves. Utilizing the graphical-slope method, the velocity curve is reasonably accurate and the acceleration curve is at best only an approximation.

Some concepts to be discussed in this section are:

- The follower displacement motion y refers to the knife-edge follower or the pitch curve center of the roller follower if employed.
- The follower velocity y' or $\dot{y}$ is an indication of both the cam pressure angle and the cam torque.
- The follower acceleration y'' or $\ddot{y}$ is critically significant for design considerations.
- Acceleration values are related to the system inertia forces in which the maximum acceleration establishes the maximum inertia forces.
- The shape of the follower acceleration curve is of critical concern for the design of moderate- to high-speed machines. From it, analysis can be made for cam-follower system shock, noise, wear, and vibration.
- The acceleration curve shall have no discontinuities where it blends with the dwell action of the motion.
- Acceleration values at each point are related to the radius of curvature of the cam profile, which in turn relates to the surface stresses and wear.
- Over the complete rise-fall motion, the following is true: $\int y'' d\theta = 0$ and $\int \ddot{y} dt = 0$

Therefore, the area of the acceleration curve for any cam has an area of positive action (+) equal to the area of the negative action (-). This observation could aid the designer in developing the optimum cam acceleration curve shape.

5.7.6. Basic Curve Classification

In this subtopic, some basic curves of the DRD type and their kinematic relationships will be presented. In design, the first step is to sketch a time chart. Then the basic cam curve may be chosen to satisfy the cam-follower requirements. The basic curves of the rise-fall displacement diagram are primarily of two families: the simple polynomial and the trigonometric.

- **Simple polynomial curves** The displacement equations of simple polynomial curves are of the form

$$y = C\theta^2 \tag{62}$$

Where n = any number

C = a constant

In this polynomial family, we have the following popular curves with integer powers: straight line, n = 1; parabolic or constant acceleration, n = 2; cubic or constant jerk, n = 3. High degree polynomial curves are shown in Chapter 3.

- **Trigonometric curves** The curves of trigonometric form are: simple harmonic motion (SHM) or crank curve, which has a cosine acceleration curve; cycloidal, which has sine acceleration curve; double harmonic; and elliptical.

- **Other curves** In addition to these two families are the miscellaneous, little-used curves as modified straight-line circular arc and the circular arc curves. These are employed primarily as an improvement over the characteristics of the straight line curve and for special design requirements.

5.7.7. Comparison of basic curves

The basic curves considered can be applied for low to moderate speeds as a first design selection. For optimum cam follower performance, combinations of these basic curves and other mathematical curves compatible for relevant cam profile with high speed follower are also present.

Comparing the curves most often used, the trigonometric ones (simple harmonic motion, cycloidal, and related to them) give better overall performance than those in the basic polynomial family (straight-line, parabolic, and cubic curves). The advantages are smaller cams and reduced translating follower ride thrust. The cycloidal curve, for most machine requirements, is the first choice since it has no abrupt change in acceleration; it gives the lowest vibrations, wear, stress, noise, and shock. It is easy to start, requires small springs, and induces low follower side thrust. However, the necessary accuracy of fabrication is higher than for low-speed curves. It has a slightly larger maximum acceleration than some others. Figure 5.15 and 5.16 show the comparison of the basic curve characteristics of displacement, velocity, and acceleration for the follower rises 1-1/2 inches in 150 degrees of dwell-rise-dwell cam rotation at 300rpm.

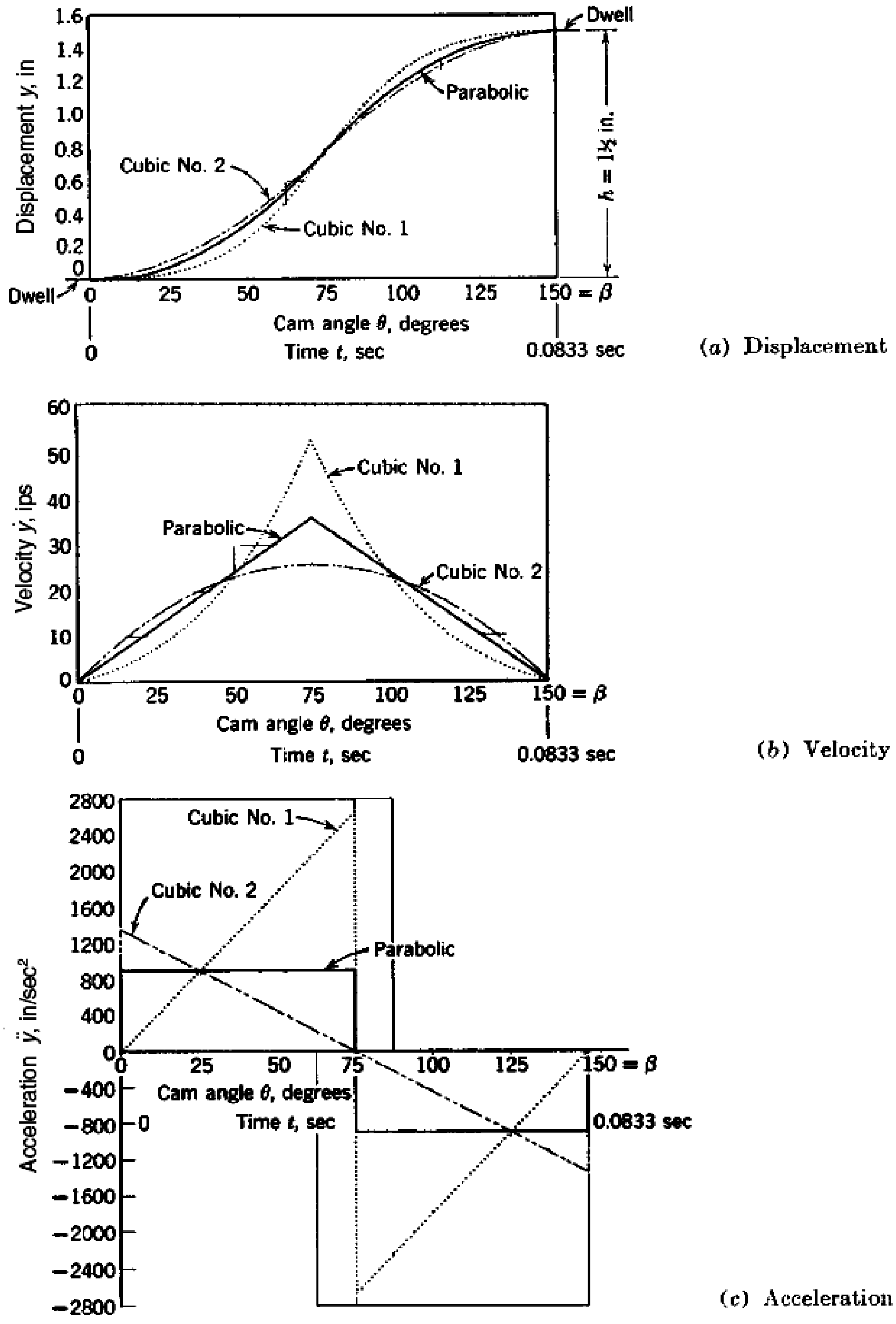

Figure 5.15: Comparison of basic curves—DRD cam. (Follower has 1-1/2-in. rise in 1500 of cam rotation at 30rpm.)

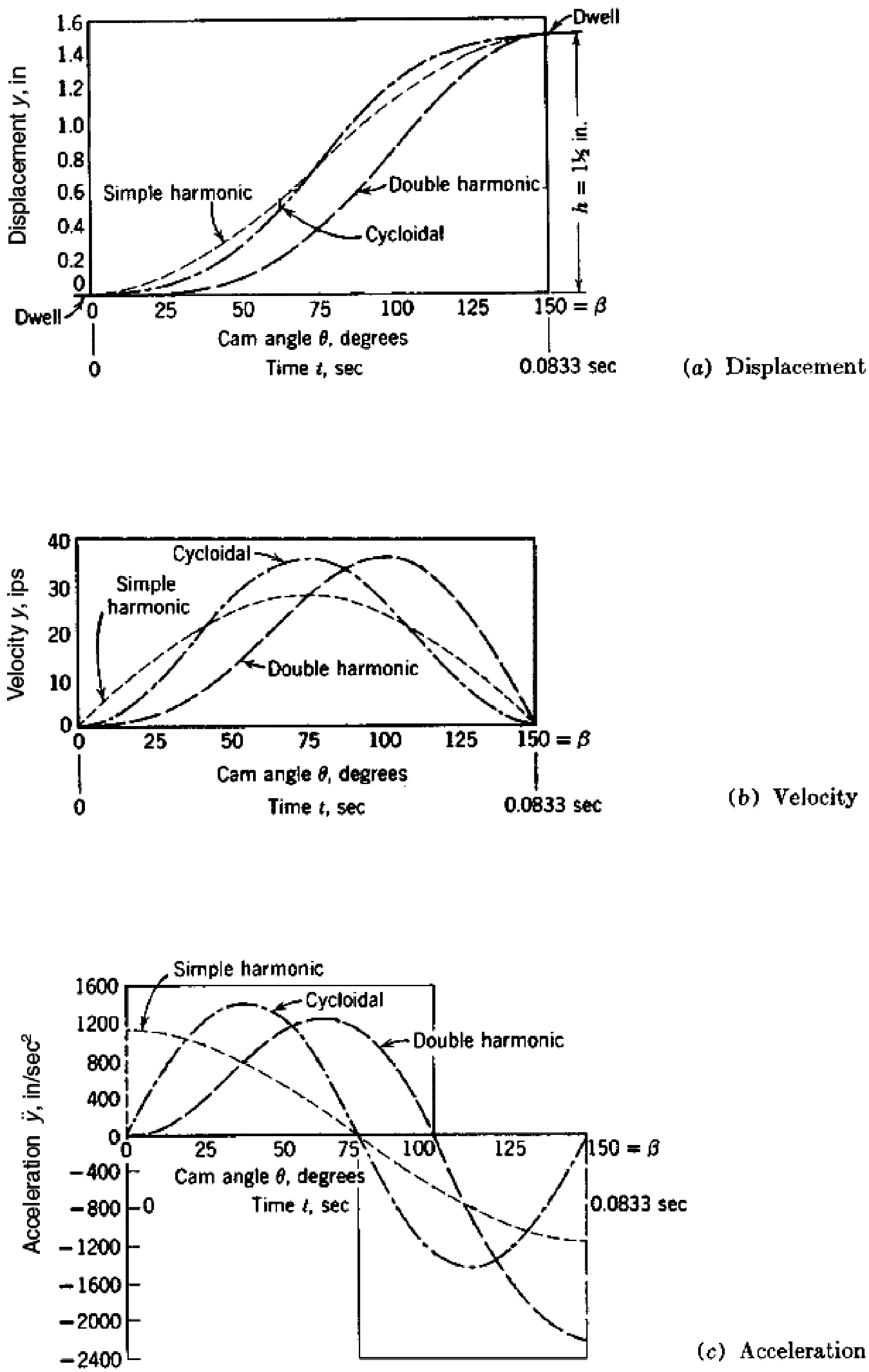

Figure 5.16: **Continued from figure 5.15**

The cycloidal motion curve, as the name implies, is generated from a cycloid and has a sine acceleration curve. A cycloid is the locus of a point on a circle that is rolled on a straight line. The circumference of the circle is equal to the follower rise h. The cycloid is the first dwell-rise-dwell curve discussed that does not have any discontinuities in the acceleration curve. Therefore, it can be applied to higher speeds even though its maximum acceleration is higher in some cases.

The dwell-rise-dwell (DRD) and dwell-rise-return-dwell (DRRD) curves are analyzed in almost the same way. It can be noted that the DRD cam curve is a portion of the total action which could be a part of the dwell-rise-dwell-return-dwell (DRDRD) cam. Figure 5.17 shows the complete cycle of blended DRD cycloidal curve producing a DRDRD cam. The period of rise is smaller than the period of fall, producing higher rise maximum acceleration than the maximum fall acceleration.

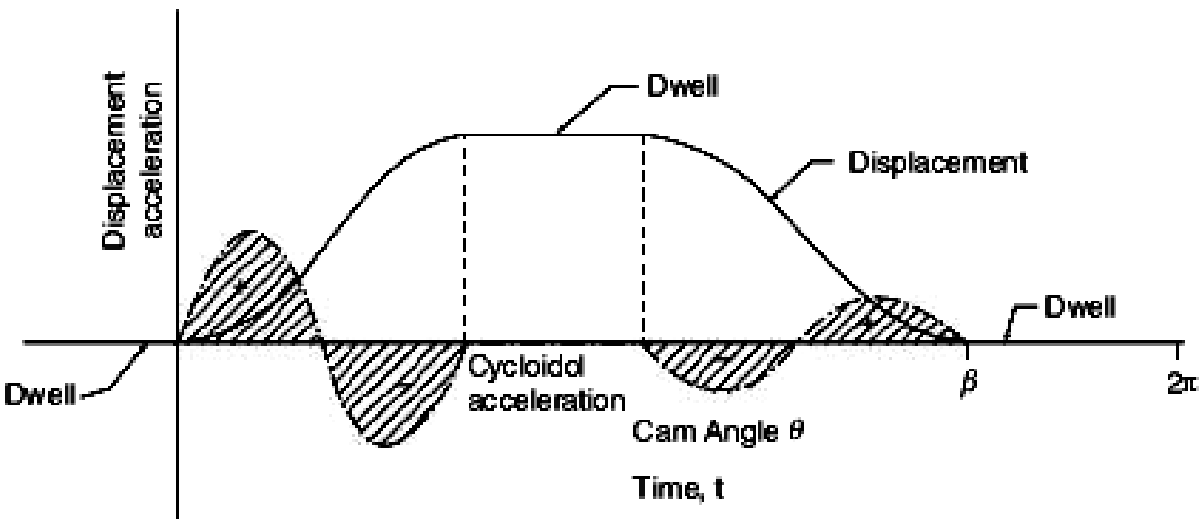

Figure 5.17: **Dwell-rise-dwell-return-dwell cam with cycloidal acceleration curves.**

Because of their simplicity of mathematical analysis and ease of construction, cycloidal curves are employed for many machine performance requirements.

Therefore, in this mowing component, the curve of trigonometric form specifically cycloidal form is suitable for the required effect of the follower (the reciprocating cutter bar) motion. Then, the design of the cam will be done with the following design specifications and decisions.

Initial design Specifications:

- Total rise of the follower required is 70mm.
- The base circle of the cam is to have the diameter of 60mm.

- The rotational speed of the cam is 800rpm.
- The cam angle for the effect of the follower total rise is 90^0 for motion of the follower (the reciprocating cutter bar).
- The dwell angle of the cam is 90^0 for motion of the follower (the reciprocating cutter bar).
- For steel-steel cam and follower materials contact surfaces, friction coefficient,

  $\mu = 0.3$.

Design Decisions:

- A cam type of dwell-rise-dwell-return-dwell (DRDRD) class of yoke cam for motion of the follower (the reciprocating cutter bar).
- The follower type suitable for the required working configuration is the flat faced type of follower as of in the yoke cam.
- For the required effect of the cam, cycloidal form curve is to be employed.

5.7.8. Cycloidal Motion Curves

The cycloidal motion curve, as the name implies, is generated from a cycloid and has a sine acceleration curve. A cycloid is the locus of a point on a circle that is rolled on a straight line. The circumference of the circle is equal to the follower rise h. The cycloid is the first dwell-rise-dwell curve known for it does not have any discontinuities in the acceleration curve. Therefore, it can be applied to higher speeds.

Designate point A (**Fig.5.18**) as the center and draw a circle with the radius equal to $\frac{h}{2\pi}$. Divide this circle into the same number of parts used for the horizontal axis. From the projection of these points on the ordinate, draw lines that are parallel to the diagonal AD that intersect the vertical projections of the divisions of this abscissa. These intersections locate the necessary points for the displacement curve.

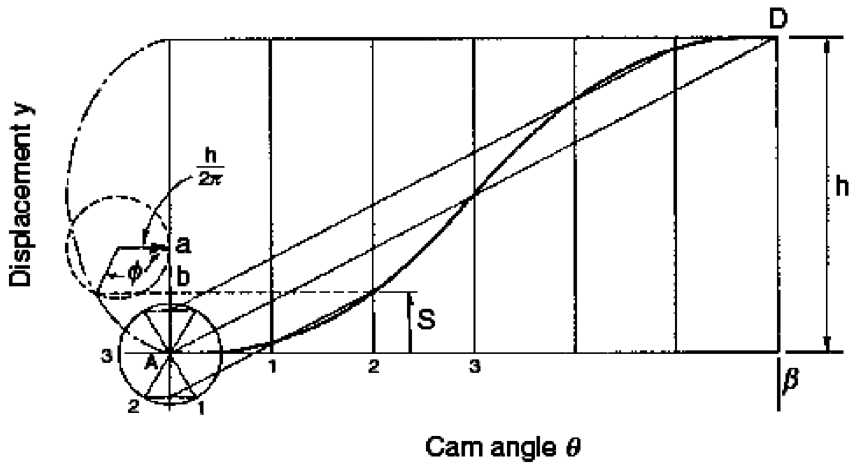

Figure 5.18: **Cycloidal motion curve construction [21]**

From Fig.5.18, the radius of the circle by which the cycloide is to be developed is $\frac{h}{2\pi}$ in which displacement y is

$$y = \frac{h}{2\pi}\phi - \frac{h}{2\pi}\sin\phi \tag{63}$$

Where $\phi =$ angle of circle rotation, rad.

It can be seen that

$$\frac{\theta}{\beta} = \frac{\phi}{2\pi} \tag{64}$$

Substituting in equation (64) yields

$$\text{Displacement } y = h\left(\frac{\theta}{\beta} - \frac{1}{2\pi}\sin\frac{2\pi\theta}{\beta}\right) \tag{65}$$

In Fig.5.18, It can be seen that the first term of the equation, $\frac{h\theta}{\beta}$, is the sloping line and the second term, $\frac{h}{2\pi}\sin\frac{2\pi\theta}{\beta}$, is the subtracted harmonic displacement. Thus

Velocity $y' = \frac{h}{\beta}(1 - \cos\frac{2\pi\theta}{\beta})$

Acceleration $y'' = \frac{2h\pi}{\beta^2}\sin\frac{2\pi\theta}{\beta}$

Jerk $y''' = \frac{4h\pi^2}{\beta^3}\cos\frac{2\pi\theta}{\beta}$ (66)

The plot of the above equations 66 are known to have the pattern shown in Fig. 5.19. Note that this curve has no sudden change in acceleration for the complete cycle and thus has high-speed applications.

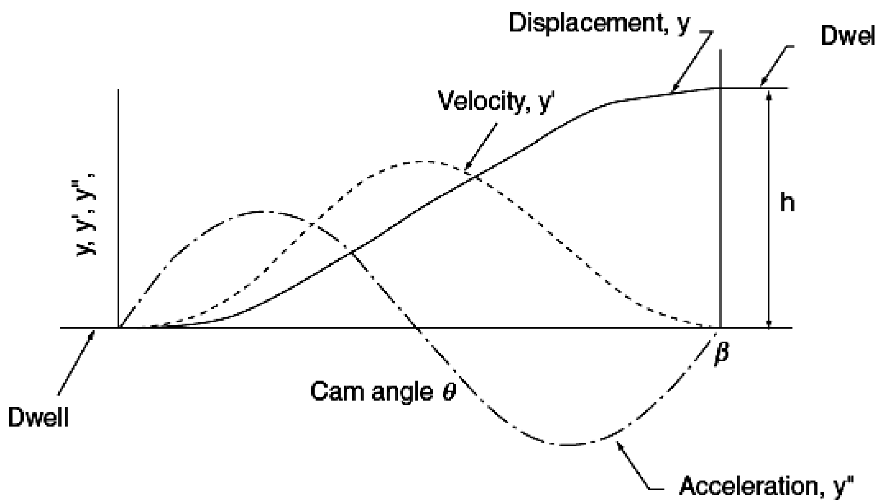

Figure 5.19: **Cycloidal motion curve—DRD cam.**

Inserting the required design parameters of the cam in to equations 65 and 66 results in

- The total rise of the follower, $h = 70mm = 0.07m$, and
- The cam angle for the effect of the follower total rise, $\beta = 90^0 = \frac{\pi}{2}rad$
- The range of the cam angle is taken to be in $0 \le \theta \le 2\pi$.

To yield the follower characteristics as:

Displacement $y = h(\frac{\theta}{\beta} - \frac{1}{2\pi}\sin\frac{2\pi\theta}{\beta}) = 0.07(\frac{2\theta}{\pi} - \frac{1}{2\pi}\sin 4\theta)$ (67)

Velocity $y' = \frac{h}{\beta}(1 - \cos\frac{2\pi\theta}{\beta}) = \frac{0.14}{\pi}(1 - \cos 4\theta)$ (68)

Acceleration $y'' = \frac{2h\pi}{\beta^2}\sin\frac{2\pi\theta}{\beta} = \frac{0.56}{\pi}\sin 4\theta$ (69)

Jerk $y''' = \frac{4h\pi^2}{\beta^3}\cos\frac{2\pi\theta}{\beta} = \frac{2.24}{\pi}\cos 4\theta$ (70)

Equations 67 to 70 will be used to generate the characteristics curve equations of the follower (the reciprocating cutter bar).

5.8. Cam Shaft Design

This shaft, containing the cam and the driven gear, is a part used to transmit the 3hp of the rated power of the engine in to the reciprocating cutting bar. Since the shaft contains the cam at one end, a driven gear at the other end, and two supporting bearings, it is subjected to transverse loads, torsional loads, and axial loads due to the weight of the gear, the weight of cam, and the bevel gear radial, axial and thrust loading, respectively.

The free body diagram of the cam shaft is shown below:

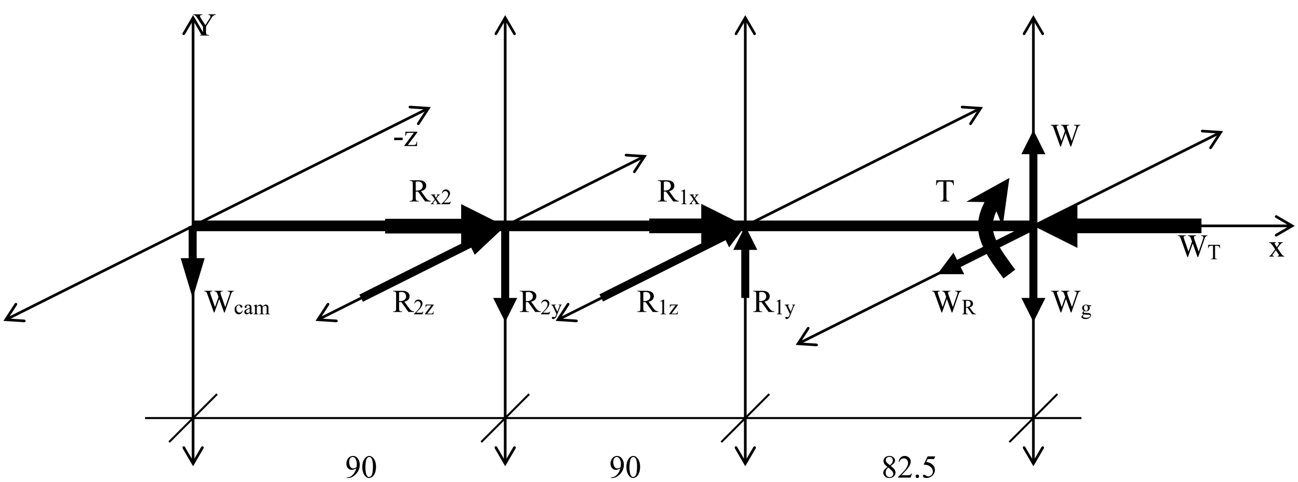

Figure 5.20: **Free-body diagram of the cam shaft**

The dimensions are taken preliminarily for initial design specification in mm.

Where R_{1z}, R_{1y}, R_{2z}, R_{2y}, R_{1x}, R_{2x}= bearing reactions at the points shown

W_{cam}= weight of the smaller pulley of the second belt drive.

W_R= Radial load due to the gear

W_g= Equivalent weight of the gear

W_T=Thrust load due to the gear

T=Torsion due to the gear tangential load about x-axis

W=Tangential load of the driven gear.

The design of the cam shaft will be carried out for both static and dynamic conditions with the following design specifications, decisions and shaft material.

Design Specifications:

- Power to be transmitted, P=3hp.
- Speed of the cam shaft, N_S=800rpm.

Design Decisions:

- Maximum Shaft length, L_s=400mm=0.4m
- The cam shaft is modeled as a simply supported beam based on the loading conditions as shown in the free body diagram.

Shaft Material:

- The material for this shaft is taken to be similar to that of the wheels shaft material since the loading conditions are in the same situation as of the gear shaft and the mechanical properties can be used from the design of the wheels shaft.

Static Analysis:

5.8.1. Determination of the applied loads

The applied loads on the shaft which are shown on the free body diagram are W_{cam}, W_R, W_g, W_T, T, and W. The magnitude of these loads in the indicated respective directions will be computed under here.

- The gear radial load, W_R, is determined from the gear design is found to be: $W_R = 230.2N$.

- The equivalent weight of the gear, W_g, can be computed on the basis of the equivalent weight of a cylindrical gear blank with base radius of the pitch circle for the worst condition of loading of the shaft. Then, the gear weight is given by:

$W_g = V\rho g$

Where $V =$ volume of the cylindrical blank

$\rho =$ Density of the gear material

$g =$ Gravitational acceleration= 10m/s

For the gear material of steel with Brinell Hardness number of 280BH, the density is 7770kg/m^3, the pitch diameter of the gear is taken from the gear design to be 7.05cm, and the height of the gear blank, h is determined empirically to be 3.233cm.

Hence, $$W_g = \frac{\pi (PD_g)^2}{4} h\rho g \quad (71)$$

$$W_g = \frac{\pi (0.14)^2}{4} 0.03233 * 7770 * 10 = 9.81N$$

- The gear thrust, W_T, is determined in the design of the gear drive is found to be $W_T = 230.2N$.

- The torsion due to the gear tangential load on the shaft, T, is computed as $T = \frac{W(PD_g)}{2}$

Where W=tangential load on the gear from the gear design.

PD_g=pitch diameter of the gear from the gear design.

Then , $T = \frac{894.45 * 0.0705}{2} = 31.53Nm$.

- The tangential load of the gear, W is computed from the gear design to be W=894.45N.

- The equivalent weight of the cam is determined on the basis of worst condition of the shaft loading. For this reason, the shape of the cam is assumed to be a cylinder with the maximum radius equal to its rise plus the diameter of its base circle, and height of 40mm, i.e, d_c = 2(rise+base circle diameter)=2(70+60)=260mm, and h_c = 40mm. Then, the weight of cam $W_{cam} = \rho V$ Where $\rho =$ cam material density $V =$ volume of the cam, and it is given by $V = \frac{\pi}{4} d_c^2 h_c = \frac{\pi}{4} * 0.26^2 * 0.04 = 2.124 * 10^{-3} m^3$.

The cam material has to be Selected based on the application requirement of wear resistance, and hence a case hardened carbon steel with a density of $\rho = 7700 \text{kg/m}^3$ from Indian standard IS:1570(Part II/sec 1)-1979(Reaffirmed 1991) is selected.

Then, $W_{cam} = \rho V = 7700 * 2.124 * 10^{-3} = 16.355N$

5.8.2. Determination of reaction loads (R_{1x}, $R_{,2x}$, R_{1y}, R_{2y}, R_{1z}, R_{2z})

Considering loads in the xy plane and applying Newton's 2nd law,

$$\sum F_y = 0 \tag{72}$$

$$\Rightarrow R_{1y} + W - W_{cam} - R_{2y} - W_g = 0$$

$$\Rightarrow R_{1y} - R_{2y} = W_g - W + W_{cam} = 16.355 + 9.81 - 894.45 = -868.29N$$

Taking moments about the points at R_{1y}

$$\sum M_{xy} = 0 \tag{73}$$

$$\Rightarrow -0.0825W - 0.18W_{cam} - 0.09R_{2y} + 0.0825W_g = 0$$

This implies

$R_{2y} = -843.63N$ and $R_{1y} = -1711.92N$.

The negative signs indicate that the directions of the reactions R_{1y} and R_{2y} are opposite to the indicated directions in the free body diagram.

Similarly, in the xz plane

$$\sum F_z = 0 \tag{74}$$

$$\Rightarrow R_{1z} + R_{2z} - W_R = 0$$

$\Rightarrow R_{1z} + R_{2z} = 230.2N$, and taking moments about the point at R_{2z},

$$\sum M_{xz} = 0 \tag{75}$$

$$\Rightarrow 0.09R_{1z} - 0.1725W_R = 0$$

$$\Rightarrow R_{1z} = 441.22N \text{ and } R_{2z} = -211.02N$$

And in the x-direction,

$$\sum F_x = 0 \tag{76}$$

$\Rightarrow R_{1x} + R_{2x} = 230.2N$, for this condition, the system is found to be indeterminate, and hence the worst design condition would be assumed that

$R_{1x} = 230.2N$ and $R_{2x} = 230.2N$

5.8.3. Determination of Maximum bending moment($M_{\max}^{xy}, M_{\max}^{xz}$)

- Maximum bending moment in the xy plane ($M_{\max}^{xy}$)

Using singularity function of the bending moment in the xy plane

$$M_{\max}^{xy} = -W_{cam}(x)^1 - R_{2y}(x-0.09)^1 + R_{1y}(x-0.18)^1 - W_g(x-0.2625)^1 + W(x-0.2625)^1$$

Then,

for $0 \le x \le 0.09$, $M_{\max}^{xy} = -W_{cam}(x)^1 = -16.355 * 0.09 = -1.472Nm$

For $0.09 \le x \le 0.18$,

$M_{\max}^{xy} = -W_{cam}(x)^1 - R_{2y}(x-0.09)^1 = -16.355*0.18-(-843.63)(0.09) = 72.983Nm$

For $0.18 \le x \le 0.2625$,

$M_{\max}^{xy} = -W_{cam}(x)^1 - R_{2y}(x-0.09)^1 + R_{1y}(x-0.18)^1$

$\Rightarrow M_{\max}^{xy} = (-16.355)(0.2625)^1 - (-843.63)(0.1725)^1 + (-1711.92)(0.0825)^1$

$\Rightarrow M_{\max}^{xy} = 282.47Nm$

Therefore, the maximum bending moment in the xy plane is

$M_{\max}^{xy} = 282.47Nm$

- Maximum bending moment in the xz plane ($M_{\max}^{xz}$)

In the same pattern, the bending moment in the xz plane

$M_{\max}^{xz} = R_{2z}(x-0.09)^1 + R_{1z}(x-0.18)^1 - W_R(x-0.2625)^1$

Then,

for $0 \le x \le 0.09$, $M_{\max}^{xz} = 0$

For $0.09 \le x \le 0.18$, $M_{\max}^{xz} = -211.02*(0.09)^1 = -18.992Nm$

For $0.18 \le x \le 0.2625$, $M_{\max}^{xz} = -211.02*(0.1725)^1 + 441.22*(0.0825)^1 = 0.0$

Hence, the maximum bending moment in the xz plane is

$M_{\max}^{xz} = -18.992Nm$

Therefore, the maximum bending moment will be determined by

$$M_{\max} = \sqrt{(M_{\max}^{xy})^2 + (M_{\max}^{xz})^2} = \sqrt{(18.992)^2 + (282.47)^2} = 283.11Nm, \quad (77)$$

and then the maximum torsional moment on the shaft is

$T = T_{\max} = 31.53Nm$

5.8.4. Determination of the shaft diameter (d)

The maximum shear stress theory dictates the shaft diameter

$$d = \left[\frac{32}{\pi \sigma_{all}} \sqrt{(k_b M_{\max})^2 + (k_t T_{\max})^2} \right]^{\frac{1}{3}} \quad (78)$$

Selecting the shaft material to be the same as that of the wheels shaft,

$$d = \left[\frac{32}{\pi * 51.3 * 10^6} \sqrt{(1.5 * 283.11)^2 + (1.5 * 31.53)^2} \right]^{\frac{1}{3}} \cong 45m$$

Design for fatigue

The size of the shaft for fatigue loading using the maximum shear stress theory is

$$d = \left[\frac{32(FS)}{\pi} \sqrt{\left(\frac{M_{\max}}{\sigma_e} \right)^2 + \frac{3}{4} \left(\frac{T_{\max}}{\sigma_y} \right)^2} \right]^{\frac{1}{3}} \quad (79)$$

Where σ_e =modified endurance limit of the shaft material=61.7MPa

σ_y =yield strength of the shaft material=217MPa

FS =safety factor=3.5, the same as that of the wheels shaft for they are of same material.

Then, $d = \left[\frac{33.52 *}{\pi * 10^6} \sqrt{\left(\frac{283.11}{61.75} \right)^2 + \frac{3}{4} \left(\frac{31.53}{217} \right)^2} \right]^{\frac{1}{3}} \cong 60mm$

Therefore, the size of the cam shaft will be 60mm so that it would be safe for both fatigue and static loading condition.

5.9. Design of Keys

A key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them. It is always positioned parallel to the axis of the shaft. Keys are used as temporary fastenings and are subjected to considerable crushing and shearing stresses. A keyway is a slot or recess in a shaft and hub of the pulley to accommodate a key.

Among the existing types of keys, a rectangular type of sunk key is selected to be used for this mowing component based on its ease of assembly, maintainability, and design simplicity.

5.9.1. Key Design for the wheels shaft mountings

Keys are subjected considerably to shearing and crushing loads, and they tend to fail by shearing or crushing. Therefore, they are designed for shearing and crushing.

- Determination of key cross-sectional dimensions

Design data: Diameter of the wheels shaft, d=50mm.

The width, w, and the thickness of the key, t, are given in terms of the shaft diameter as:

$W = \frac{d}{4}$ and $t = \frac{d}{6}$ where d= diameter of the wheels shaft.

This implies

$W = 12.5mm$ and $t = 8.33mm$

In order to determine the length of the key, strength of material approach for shearing and crushing with the assumption of the same material for the shaft and key for the key maximum length will be employed.

- For shearing strength:

  It is known that for shear, the maximum torsional torque is given by

  $$T^*_{max} = l * W * \tau_{all} * \frac{d}{2}$$

  Where T^*_{max} = the maximum torque to be transmitted by the shaft

  l = Length of the key

  W = Width of the key

  τ_{all} = Allowable design shear stress of the key and shaft material

  d = The shaft diameter

And the torsional shear strength of the shaft is known to be

$$T^*_{\max} = \frac{\pi d^3}{16}\tau_{all}$$

which implies

$$l * W * \tau_{all} * \frac{d}{2} = \frac{\pi d^3}{16}\tau_{all}$$

$l = \frac{\pi d}{2} = \frac{\pi}{2} * 50mm = 78.54mm$. This is the maximum length of the key if the same material is to be used for both the shaft and the key.

- For crushing strength:

The torque to be transmitted by the shaft can be given as

$T^*_{\max} = l * \frac{t}{2} * \sigma_C * \frac{d}{2}$ and $\sigma_C = \frac{\sigma_{yt}}{FS}$

Where $t =$ key thickness

$\sigma_C =$ crushing strength of the key

$\sigma_{yt} =$ yield strength of the key and shaft material = 217MPa.

$FS =$ Factor of safety=3.5

Then,

$l = \frac{24T^*_{\max}(FS)}{d^2\sigma_{yt}}$ and using the values given above yields

$l = 62.4mm$

Therefore, with same material, the length of the key would be between 6.24 and 7.85cm so that the key would fail by shear before the shaft fails.

Hence, the maximum key length for the wheels shaft is determined to be

$l = 8cm$ which is approximated to the next standard key length.

5.9.2. Determination of Gear shaft key cross sectional dimensions

Keys of the gear shaft are subjected considerably to shearing and crushing loads, and they tend to fail by shearing or crushing. Therefore, they are designed for both shearing and crushing.

- Determination of key cross-sectional dimensions

Design data: Diameter of the gear shaft, d=40mm.

The width, w, and the thickness of the key, t, are given in terms of the shaft diameter as:

$$l * W * \tau_{all} * \frac{d}{2} = \frac{\pi d^3}{16} \tau_{all}$$

$W = \frac{d}{4}$ and $t = \frac{d}{6}$ where d= diameter of the gear shaft. (80)

This implies

$W = 10mm$ and $t = 6.67mm$

In order to determine the length of the key, strength of material approach for shearing and crushing with the assumption of the same material for the shaft and key as of the wheels shaft key material for the gear shaft key maximum length will be employed.

- For shearing strength:

It is known that for shear, the maximum torsional torque is given by

$$T^*_{\max} = l * W * \tau_{all} * \frac{d}{2}. \quad (81)$$

Where $T^*_{\max} =$ the maximum torque to be transmitted by the gear shaft in other words the maximum torsional moment among the three sections of the gear shaft, $T_{\max} = 37.4Nm$

$l =$ Length of the key

$W =$ Width of the key

$\tau_{all} =$ Allowable design shear stress of the key and shaft material

$d =$ The shaft diameter

And the torsional shear strength of the shaft is known to be

$$T^*_{\max} = \frac{\pi d^3}{16} \tau_{all} \quad . \quad (82)$$

This implies

$l = \frac{\pi d}{2} = \frac{\pi}{2} * 40mm = 62.832mm$. This is the maximum length of the key if the same material is to be used for both the shaft and the key.

- For crushing strength:

The torque to be transmitted by the shaft can be given as

$T^*_{\max} = l * \frac{t}{2} * \sigma_C * \frac{d}{2}$ and $\sigma_C = \frac{\sigma_{yt}}{FS}$

Where $t =$ key thickness

$\sigma_C =$ crushing strength of the key

$\sigma_{yt} =$ yield strength of the key and shaft material = 217MPa.

$FS =$ Factor of safety=3.5

Then,

$$l = \frac{24T^*_{\max}(FS)}{d^2\sigma_{yt}}$$ and using the values given above yields (83)

$l = 9.121mm$

Therefore, with same material, the length of the key would be between 6.28 and 0.912 cm so that the key would fail by crushing before the shaft fails.

Hence, the maximum key length for the wheels shaft is determined to be

$l = 7cm$ approximated to the next standard key length.

5.9.3. Key Design for the cam shaft mountings

The keys for the cam shaft are designed in the similar procedure as in the previous sections of key design. Therefore, the key dimensions for the cam shaft are to be determined as follows.

➢ Determination of key cross-sectional dimensions

Design data: Diameter of the cam shaft, d=60mm.

The width, w, and the thickness of the key, t, are given in terms of the shaft diameter as:

$W = \frac{d}{4}$ and $t = \frac{d}{6}$ where d= diameter of the cam shaft.

This implies

$W = 15mm$ and $t = 10mm$

And hence strength of materials approach will be employed for the same key material as the cam shaft material.

- For shearing strength:

  It is known that for shear, the maximum torsional torque is given by

$$T^*_{\max} = l * W * \tau_{all} * \frac{d}{2}. \tag{84}$$

Where $T^*_{\max} =$ the maximum torque to be transmitted by the shaft

$l =$ Length of the key

$W =$ Width of the key

$\tau_{all} =$ Allowable design shear stress of the key and shaft material

$d =$ The shaft diameter

And the torsional shear strength of the shaft is known to be

$$T^*_{max} = \frac{\pi d^3}{16}\tau_{all} \quad (85)$$

which implies

$$l * W * \tau_{all} * \frac{d}{2} = \frac{\pi d^3}{16}\tau_{all}$$

$l = \frac{\pi d}{2} = \frac{\pi}{2} * 60mm = 94.25mm$. This is the maximum length of the key if the same material is to be used for both the shaft and the key.

- For crushing strength:

The torque to be transmitted by the shaft can be given as

$$T^*_{max} = l * \frac{t}{2} * \sigma_C * \frac{d}{2} \text{ and } \sigma_C = \frac{\sigma_{yt}}{FS}$$

Where $t =$ key thickness

$\sigma_C =$ crushing strength of the key

$\sigma_{yt} =$ yield strength of the key and shaft material = 217MPa.

$FS =$ Factor of safety=3.5

Then,

$l = \frac{24 T^*_{max}(FS)}{d^2 \sigma_{yt}}$ and using the values given above yield (86)

For the maximum torsional moment on the shaft is

$$T^*_{max} = 31.53Nm,\ l = 33.90mm$$

Therefore, with same material, the length of the key would be between $94.25mm$ and $33.90mm$ for both design for crushing and shearing so that the key would fail by crushing before the shaft fails.

Hence, the maximum key length for the wheels shaft is determined to be $l = 10cm$ approximated to the next standard key length.

5.10. Design of cutting height adjuster Screw

The lifting screw is subjected to a compressive load of the frontal wheels traction forces. For the worst condition of loading of the screw, the torque transmitted by the rare wheels is taken for design purpose. This torque at the rare wheels shaft is determined as:

$$P = T * \omega \tag{87}$$

Where $P =$ the power for the traction force=4hp

$T =$ Torque at the rare wheels shaft

$\omega =$ Angular speed of the wheels shaft in rad/ sec.=4hp

Then, $T = \frac{P}{\omega}$ and substitution of the values result in

$T = 71.24Nm$

So this torque would be used for the design of the screw at the worst condition.

Considering moments at the front wheels link about the point through the screw and the link axes intersection, taking the same amount of torque for design is to be transmitted for the traction of the wheels for worst condition,

$$T = F * l \tag{88}$$

Where $T =$ the torque due to the front wheels=71.24Nm

$F =$ the compressive force on the screw

$l =$ Offset distance by which the compressive force acts on the screw=0.025m.

This implies

$F = \frac{T}{l}$ and inserting the known values results in

$F = 2849.6N$

Then, using this compressive force, it would be possible to determine the core diameter of the screw by the relation

$$F = \frac{\pi d_C^2}{4} \sigma_C \tag{89}$$

Where $d_c =$ screw core diameter

$\sigma_C =$ Permissible compressive stress of the screw material=51.3 MPa for the same material as of the wheels shaft determined earlier.

Then, using the values, the core diameter is determined to be

$d_c = 8.4mm$.

The next higher standard screw core diameter is 12mm, and therefore an M12 screw thread is selected.

5.11. Bearing Selection

A bearing is a connector that permits the connected members to rotate or to move relative to one another. Often one of the members is fixed, and the bearing acts as a support for the moving member. Most bearings support rotating shafts against either transverse (radial) or thrust (axial) forces. To minimize friction, the contacting surfaces in a bearing may be partially or completely separated by a film of liquid (usually oil) or gas. These are sliding bearings, and the part of the shaft that turns in the bearing is the journal. Under certain combinations of force, speed, fluid viscosity, and bearing geometry, a fluid film forms and separates the contacting surfaces in a sliding bearing, and this is known as a hydrodynamic film. An oil film can also be developed with a separate pumping unit that supplies pressurized oil to the bearing, and this is known as a hydrostatic film.

The surfaces in a bearing may also be separated by balls, rollers, or needles; these are known as rolling bearings. Because shaft speed is required for the development of a hydrodynamic film, the starting friction in hydrodynamic bearings is higher than in rolling bearings. To minimize friction when metal-to-metal contact occurs, low-friction bearing materials have been developed, such as bronze alloys and other metal alloys.

The principal advantage of rolling bearings is the ability to operate at friction levels considerably lower at startup, the friction coefficient having the values $\mu = 0.001 \pm 0.003$. Also, they have the following advantages over bearings with sliding contact:

- They maintain accurate shaft alignment for long periods of time.
- They can carry heavy momentary overloads without failure.
- Their lubrication is very easy and requires little attention and

- They are easily replaced in case of failure.

The important parts of rolling bearings are outer ring, inner ring, rolling element and separator (retainer). The role of the separator is to maintain an equal distance between the rolling elements. The races are the outer ring or the inner ring of a bearing. The raceway is the path of the rolling element on either ring of the bearing. Rolling bearings may be classified using the following criteria:

- The rolling element share: ball bearings, roller bearings (cylinder, cone, and barrel), and needle bearings.
- The direction of the principal force: radial bearings, thrust bearings, radial-thrust bearings, or thrust-radial bearings.
- The number of rolling bearing rows: rolling bearings with one row, with two or more rows.

Based on their advantage over sliding contact bearing, rolling bearings are selected for the application required for this mowing component. Among the rolling types of bearings, ball bearing is to be used for ease of availability on market.

5.11.1. Bearings for wheels shaft

In order to select the most suitable radial bearings, the basic dynamic radial load should be determined prior to other calculations. It is then multiplied by the service factor K_s to get the design basic dynamic radial load capacity. After determining the design basic radial load capacity, the selection of bearing is made from the catalogue of a manufacturer.

Design Decision:

- Life of bearings required for this mowing component having agricultural purpose, L_H is recommended to be 20,000-30,000 hours.
- The sservice factor for moderate shock loads is K_s =2.0.

1. Determination of equivalent dynamic load(W):

The dynamic equivalent radial load,W, for radial ball bearing under combined constant radial load, W_R, and axial load, W_A, is given by

$W = XVW_R + YW_A$ (90)

Where V = rotational factor=1, for inner race is rotating.

X = Radial load factor.

Y = Axial load factor.

The ratio of the axial to radial laods is

$\frac{W_A}{W_R} = 0$, and when $\frac{W_A}{W_R} = 0 \leq e$ for $0.22 \leq e \leq 0.44$, the values of load factors are

$X = 1$ and $Y = 0$.

And the bearing radial load $W_R = B_F = 1220N$ from the design of wheels shaft, and therefore, the dynamic equivalent radial load

$W = XVW_R = 1*1*1220 = 1220N$

2. Determination of bearing life in revolutions(L):

The relationship between the life in revolutions, L, and the life in working hours, L_H is given by

$L = 60*N*L_H$ (91)

Where N = the wheels shaft speed in rpm=400rpm

L_H = the life of bearing in working hours.

Then, $L = 60*N*L_H = 60*400*30{,}000 = 720*10^6$ revolutions, taking the maximum possible recommended life in hours.

3. Determination of the basic dynamic radial load(C):

The basic dynamic radial load rating, C, is given by

$C = W\left(\frac{L}{10^6}\right)^{\frac{1}{K}}$ in N. (92)

Where K= 3, for ball bearings.

Then, $C = 1220 * \left(\frac{720 * 10^6}{10^6} \right)^{\frac{1}{3}} N = 10.935 kN$. This is the basic dynamic radial load rating.

4. Determination of the design basic dynamic radial load rating(C^*):

The design basic dynamic radial load

$$C^* = K_s C \quad (93)$$

Where K_s = service factor=2.0

C = basic dynamic radial load=10.935kN.

Then, $C^* = K_s C = 2.0 * 10.935 \cong 22 kN$

5. Selection of the bearing basic number:

The bearing basic number is the number corresponding to the bearing type selected specifically in accordance with the basic design dynamic radial load.

Hence, the bearing basic number corresponding to $C^* = 22kN$ is 310 with the bore diameter of 50mm equal to the wheels shaft diameter and outer diameter of 110mm. The other dimensions of the bearing 310 are

Width, w=27mm

Corner fillet radius, r=2.03mm

Pitch diameter of inner race, d_{pi} =64.3mm

Pitch diameter of outer race, d_{po} =96.5mm.

5.11.2. Bearings for driving gear shaft

1. Determination of equivalent load(W):

The equivalent dynamic load

$$W = XVW_R + YW_A \quad (94)$$

Where V = rotational factor=1, for inner race is rotating.

X = radial load factor.

Y = axial load factor.

W_A = the axial load on the bearing=230.2N, from the design of wheels shaft

W_R = the radial load on the bearing=1369.17N, from the design of wheels shaft.

The ratio of the axial to radial laods is

$\frac{W_A}{W_R} = \frac{230.2}{1369.17} = 0.168$, and when $\frac{W_A}{W_R} = 0 \leq e$ for $0.22 \leq e \leq 0.44$, the values of load factors are

$X = 0.56$ and $Y = 2.0$.

therfore, the dynamic equivalent radial load

$$W = XVW_R + YW_A = 0.56*1*1369.17 + 2*230.2 = 1227.14N$$

2. Determination of the basic radial laod(C):

The basic dynamic radial load

$$C = W\left(\frac{L}{10^6}\right)^{\frac{1}{K}} \text{ in N.} \tag{95}$$

Where K= 3, for ball bearings. [20]

For the same service life,

$$C = 1227.14*\left(\frac{720*10^6}{10^6}\right)^{\frac{1}{3}} = 10.999kN \cong 11kN$$

3. Design basic dynamic radial load(C^*):

The basic design dynamic radial load

$$C^* = K_s * C = 2 * 11kN = 22kN \tag{96}$$

4. Determination of the bearing basic number:

The bearing basic number corresponding to the design basic dynamic radial load $C^* = 22kN$ is 208 with bore diameter of 40mm and outer diameter of 80mm. The other dimensions of this bearing are listed below

Bearing width, w=18mm

Bearing inner race pitch diameter, d_{pi} = 48mm

Bearing outer race pitch diameter, d_{po} = 72.4mm

Corner fillet radius, r=1.02mm.

5.11.3. Bearings for the Cam shaft

The cam shaft is supported at two points between the driven gear and the cam. The supporting machine element is selected to be a ball bearing for the cam shaft of diameter 60mm, and the ball bearings for the supporting points will be selected under here.

Design decision:

- The life of bearings to be selected for agricultural machineries is recommended to be 20,000-30,000 hours, i.e, $L_H = 30{,}000 hours$ for the possible maximum service life.
- The service factor K_s for moderate shock loads is 2.0.
- Two ball bearings are required.

1. Determination of equivalent dynamic load(W)

The dynamic equivalent radial load, W, for radial ball bearing under combined constant radial load, W_R and axial load, W_A is given as

$$W = XVW_R + YW_A \tag{97}$$

Where V = rotational factor=1, for inner race is rotating.

X = radial load factor.

Y = axial load factor.

The resultant radial load

$$W_R = \sqrt{R_{1x}^2 + R_{1y}^2 + R_{1z}^2} \text{ or } W_R = \sqrt{R_{2x}^2 + R_{2y}^2 + R_{2z}^2} \quad (98)$$

This results in

$W_R = 1782.8N$ or $W_R = 899.6N$

Then, the maximum design resultant radial load is $W_R = 1782.8N$, and the maximum design resultant axial load from gear design is

$W_A = W_T = 230.2N$

The ratio of the resultant axial to radial loads

$\frac{W_A}{W_R} = \frac{230.2}{1782.8} = 0.129$, and when $\frac{W_A}{W_R} = 0 \leq e$ for $0.22 \leq e \leq 0.44$, the values of load factors are

$X = 1$ and $Y = 0$.

Therfore, the dynamic equivalent radial load

$W = XVW_R + YW_A = 1*1*1782.8 + 0*230.2 = 1782.8N$

2. Determination of the basic dynamic radial load(C)

The basic dynamic radial load for the same service life of the wheels shaft bearing

$$C = W\left(\frac{L}{10^6}\right)^{\frac{1}{K}} \text{ in N.} \quad (99)$$

Where K= 3, for ball bearings.

$$C = 1782.8 * \left(\frac{720 * 10^6}{10^6} \right)^{\frac{1}{3}} N \cong 16kN$$

3. Design basic dynamic radial load(C^*)

The design basic dynamic radial load $C^* = K_s * C = 2 * 16kN = 32kN$

Therefore, the bearing basic number of designation for 32kN design basic dynamic radial load corresponding to the cam shaft diameter 60mm as bore diameter of the bearing is 212 which is capable of supporting up to 40.5kN basic dynamic radial load that is greater than the design basic dynamic radial load.

The geometric parameters of a 212 single row ball bearing are listed below based on the international standards of bearing.

Bearing bore, d=60mm

Bearing Outer diameter, OD=110mm

Bearing width, w=22mm

Bearing inner race pitch diameter, d_{pi} = 70.6mm

Bearing outer race pitch diameter, d_{po} = 99.3mm

Corner fillet radius, r=1.52mm. [20]

5.12. Selection of Snap rings (Circlips)

A **circlip** or **snap ring** is a type of fastener consisting of a semi-flexible metal ring with open ends which can be snapped into place, into a machined groove on a dowel pin or other part to permit rotation but to prevent relative longitudinal movement between moving parts. There are two basic types as internal and external, referring to whether they are fitted into a tube or over a shaft. Circlips are often used to secure pinned or keyed connections.

Circlips which are fitted may be removed with a pair of needle-nosed pliers or a special snap ring tool if the circlip is designed to include entry points for the pliers or tool. These general types of fasteners are sized to provide interference fit onto or into, in the case of an internal fastener, a groove or land when in use, such that they must be elastically deformed in order to install or remove them.

All the circlips used in this machine are external types with expanded eyes for ease of assembly and disassembly, and only three different circlips for the three shafts corresponding to their diameters are selected based on the standard designation of heavy series circlip in type A for shaft diameter *d* as: Circlip, Heavy A *d* IS: 3075, 1965.

Therefore, the circlips for the each shaft are designated as

- For wheels shaft with diameter of 50mm: Circlip, Heavy A50 IS: 3075, 1965.
- For gear shaft with diameter of 40mm: Circlip, Heavy A40 IS: 3075, 1965.
- For cam shaft with diameter of 60mm: Circlip, Heavy A60 IS: 3075, 1965.

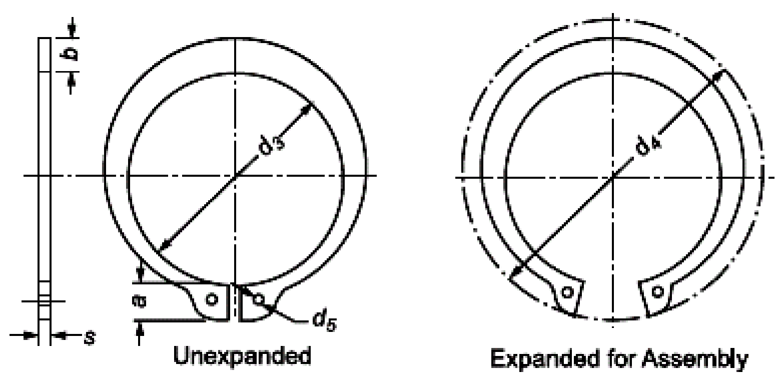

Figure 5.21: **External circlips by the designation: Circlip, Heavy A *d* IS: 3075, 1965**

Therefore, the dimensions of each circlips required for each corresponding shafts are:

- For wheels shaft: $d_3 = 45.8mm$, $d_4 = 68mm$, $d_5 = 5mm$, $a = 8mm$, $b = 6mm$, $s = 5mm$.
- For gear shaft: $d_3 = 36.5mm$, $d_4 = 55.5mm$, $d_5 = 5mm$, $a = 7mm$, $b = 6mm$, $s = 5mm$.
- For cam shaft: $d_3 = 55.8mm$, $d_4 = 79mm$, $d_5 = 5mm$, $a = 9mm$, $b = 6mm$, $s = 5mm$.

5.13. Design of the machine wheels

In the product design of this machine, it is determined that the wheels for the mower are to be made from cast steel. Since these cast steel wheels are rotating, they can be designed by considering them as a rotating disc on which various loads are applied on its peripheral and inner surfaces. The loads that act on the peripheral surfaces of the wheels are the traction force induced from the ground, the weight of the machine at its center by the wheels shaft, and the reaction of weight of the machine as normal force by the ground. For the design of the wheels, the traction force and the normal force due to the reaction of the machine weight by the ground are considered to act on the peripheral surface of the wheels and their total is taken to be the external load on the rotating discs (the wheels. In a similar fashion, the weight of the machine by the wheels shaft will be taken as the inner surface load of the same rotating discs.

Design Decisions:

- The outer diameter of the wheels considered as rotating discs is required to be R_2=800mm.
- The maximum rotational speed of the wheels required is 400rpm.
- The rated power to be transmitted at the required rotational speed is up to 4hp.
- The face width of the rim of the cast steel wheels is 100mm which is considered for increasing the surface area required for the probability of sinking of the wheels in marsh farm lands.
- The material for the wheels is determined to be cast steel which is a type of ferrous material. On this basis, the best ferrous material suited for typical applications of flywheels, brake drums, and clutch plates which need design requirements of continuous frictional interactions with mating surfaces is gray cast iron. Therefore, gray cast iron is selected to be the wheels material with the mechanical properties as per the class and requirements of ASTM class 30,SAE 111:
  - Ultimate Strengthes-Tensile, σ_{ult}=214MPa
    -Compressive, σ_{ulc}=752MPa
    -Shear, τ_{uls}= 303MPa
  - Yield Strength, σ_y=220MPa
  - Torsional Shear Strength, τ=276MPa

- Endurance Limit, σ_e=97MPa
- Brinell Hardness,H_B=210
- Density,ρ=7100kg/m^3
- Modulus of Elasticity-Tension,E=113GPa

  -Compression,E=172GPa

  - Shear,G=36GPa

5.13.1. Determination of the internal and external surface loads on the wheels

The loads to be determined for the design of the wheels are the traction load and the normal reaction load due to the weight of the machine which both summed up to give the external surface force on the peripheral surface of the rotating disc to be used as the wheels, and the load due to the weight of the machine applied by the shaft which in turn gives the inner surface load.

Under here, these forces will be determined analytically.

➢ Determination of external surface loads:

The traction force F_t that the wheels should resist will be determined by

$$F_t = \frac{2P}{d\omega} \tag{100}$$

Where P= the rated power to be transmitted by the wheel in watts.

d= the outer diameter of the wheel in meters.

ω = the rotational speed of the wheel in radians per second.

Hence, $F_t = \dfrac{2*(4*746)}{0.8*\left(400*\dfrac{2\pi}{60}\right)} = 178.1N$. This is the traction force on the wheel for worst condition of loading without considering the efficiency of transmission which lessens the force.

And the total weight of the machine W is determined for the worst condition of loading as the sum of the weight of components:

$$W = 1.5*(W_g + W_{cam} + W_P + W_{Cutter} + W_{engine}) \tag{101}$$

Where W_g = weight of the gears=2*83.5=167N

W_{cam} = Weight of the cam=16.4N

W_P = Weight of the pulleys=889.56+120.2+296.3+83.3=1389.4N

W_{Cutter} = Weight of the cutting system=350N

W_{engine} = Weight of the engine=350N

Then, $W = 1.5 * (167 + 16.4 + 1389.4 + 350 + 350) = 3409.2N$. This is the weight of the mowing machine for the worst condition of the wheels loading.

Therefore, the external surface F load is the sum of the traction force and the weight of the mower, that is:

$$F = F_t + W = 178.1 + 3409.2 = 3587.3N \quad (102)$$

- Determination of internal surface loads:

The following figure shows the reaction force due to the weight of the machine that contributes for the internal surface loads of the wheels.

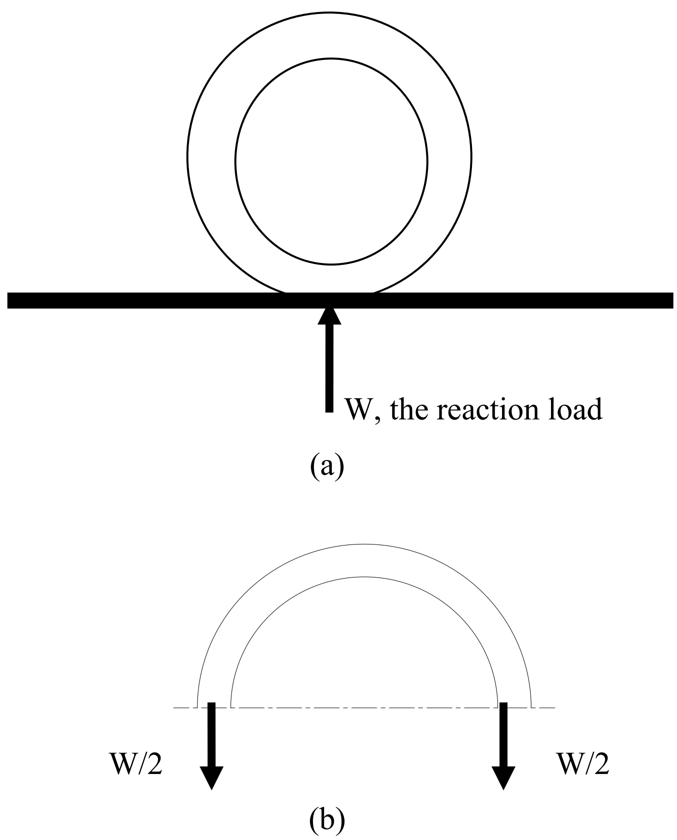

Figure 5.22: Wheel loading

The stresses at the internal surface of the disc shown in the figure 5.21 will be equal to the net force of the loads shown in figure 5.21(b) multiplied by the projected area of the disc internal diameter.

Therefore, the net force that is countable to the internal stress load of the disc is the reaction load due to the weight of the machine.

That is the internal surface load on the wheel is therefore

$W = 3409.2N$

5.13.2. Determination of internal diameter of the wheels

The inner diameter of the wheels can be determined by applying approach used in rotating discs. For rotating discs, the general equation for the radial stresses on rotating discs is given by:

$$\sigma_r = A - \frac{B}{r^2} - \frac{(3+\nu)\rho\omega^2 r^2}{8} \quad (103)$$

Where A and B are constants that are to be determined using boundary conditions

r = radius of the wheel

ω = rotational speed of the wheel

ρ= density of the wheel material

ν = poison's ratio[21]

To determine the stresses conditions at the outer and inner surfaces which are to be used as the boundary conditions, the radial stresses at the inner and outer radius need to be calculated as:

- Stress at the outer radius,

$$\sigma_r = -\frac{F}{2R_2 f} \quad (104)$$

Where F= external surface force

$R_2 =$ Outer radius of the wheel

$f =$ Face width of the wheel

Then, $\sigma_r = -\frac{3587.3}{2*0.4*0.1} = -4.5*10^4 \, Pa$ which is to be used as one of the boundary conditions.

- Stress at the inner radius,

$$\sigma_r = -\frac{W}{2R_1 f} \quad (105)$$

Where W= the internal surface force

$R_1 =$ Inner radius of the wheel

$f =$ Face width of the wheel

Then, $\sigma_r = -\frac{3409.2}{2R_1 * 0.1} = -\frac{17046}{R_1}$ which is the other boundary condition.

Imposing the boundary conditions in to the general equation of the radial stresses, the constants A and B are determined as:

$$A = \left(\frac{17046}{R_1} - 45000\right)\frac{R_1^2}{\left(R_2^2 - R_1^2\right)} + \frac{(3+\nu)\rho\omega^2}{8}\left(R_2^2 + R_1^2\right) - 45000 \quad (106)$$

$$B = \left(\frac{17046}{R_1} - 45000\right)\frac{R_1^2 R_2^2}{\left(R_2^2 - R_1^2\right)} + \frac{(3+\nu)\rho\omega^2}{8}\left(R_2^2 R_1^2\right) \quad (107)$$

Therefore, the inner radius of the wheel can be determined using the second boundary condition as: the stress at the inner surface after simplifying the radial stress equation of the wheel is

$$\sigma_r = -\frac{17046}{R_1} = -4.5 * 10^4 \, Pa$$

This implies that the inner radius

$$R_1 \approx 0.37m$$

Therefore, from this result, the rim thickness of the wheel is determined to be 30mm.

5.14. Design of Fasteners

The whole parts of this mowing machine are interconnected to each other functionally. This functional relation among them requires mechanical joints of various forms. The method of achieving the joints is using mechanical fasteners which may be either permanent or temporary. The permanent fasteners are such as welding and riveting whereas the temporary fasteners are the threaded fasteners used to make screwed joints. In this work, there are metallic components to be joined together for proper functioning of the machine. For the components which may be disassembled for maintenance and which are not integrated when they are fabricated, the threaded fasteners are selected for the purpose of joining, and for those which are to be fabricated integrated and to be joined once for all of their service life, a permanent type of fastener specifically welding is selected to be designed.

5.14.1. Design of Bolted joints

In this machine, the temporary joints are achieved by using threaded fasteners, and all of the joints that require fastening can be made using a bolt and nut joint. The design of these bolted joints is accomplished on the basis of worst condition of loading, and hence the joint that is subjected to maximum external load causing prying in the connection is the joint between the beam on which the engine is mounted and the wheels shaft bearing. Therefore, once a bolt and nut joint is designed for this joint, the same bolts and nuts can be used for the other bolted joints in the machine.

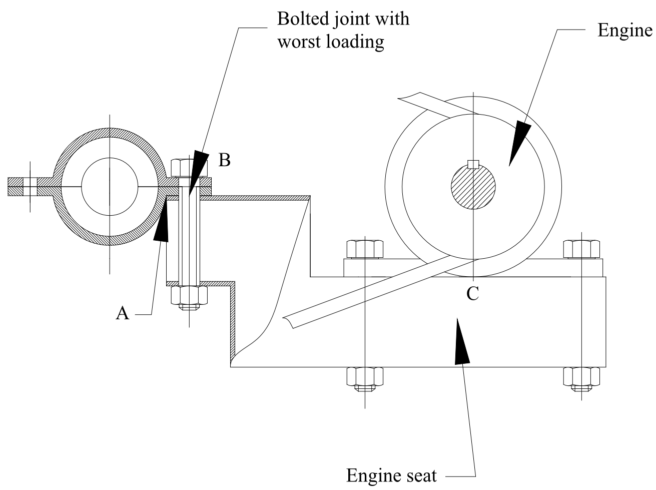

Figure 5.23: ***Bolted Joint with worst loading***

Once the bolt and nut joint for the indicated assembly is determined, then the bolt and nut designed for this joint can be used for the other joint of bolt and nut since the loading is less intensive as compared with this joint.

As it is shown in the figure, the weight of the engine at point C loads the bolts at point B to form prying which causes a tensile force leading to separation phenomenon. For this loading condition, the engine seat is subjected to a reaction load at point A in the down ward direction, the bolt tension at point B to the upward, and the weight of the engine to the down ward at point C.

The force on the bolt F_b due to the external engine weight loading is given by

$$F_b = \frac{(AC)}{(AB)} W_{engine} \tag{108}$$

Where W_{engine} = the engine weight =350N

AB = the length between points A and B=26mm

AC = the length between points A and C =347mm

Then, $F_b = \frac{0.347}{0.026} * 350 = 4671.2N$

This force is the external load on the bolt if single bolt is to be used, but since at least two are used the tensile load on each bolt will be half of the above calculated value. Therefore, the tension on the bolts due to the weight of the engine is 2325.6N.

Bolted Joints are initially stressed during tightening and hence there will be initial tightening force. This initial load is the additional force applied on the bolts. Therefore, the bolts has to withstand both the initial tightening load and the external load due to the weight of the engine.

To determine the initial tightening load on the bolts, the following empirical relation between the load and the nominal diameter of the bolt can be used.

$$F_i = 2840d \tag{109}$$

Where F_i =initial load in N.

d =Nominal diameter of the bolt in mm.

Therefore, the total load on one of the bolts is then

$$F = F_i + 0.5F_b = 2840d + 2325.6 \tag{110}$$

and the core diameter of the bolt can be given according to the standard bolt and nut profile as

$$d_c = d - 1.2268p \tag{111}$$

Where p = pitch of the bolt thread

d = nominal diameter of the bolt.

d_c =core diameter of the bolt.

Substitution of equation 111 into 110 gives the equation of the load on the bolt in terms of the core diameter of the bolt as

$$F == 2840d_c + 3484.1p + 2325.6 \tag{112}$$

Where p = pitch of the bolt thread in mm.

d_c =core diameter of the bolt in mm.

For fine thread of pitch 1.5mm, equation 112 will become

$$F = 2840d_c + 7551.8 \quad (113)$$

The total load on the bolt can also be given by

$$F = \frac{\pi}{4} d_c^2 \sigma_{all} * 10^{-6} \quad (114)$$

Where σ_{all} = allowable stress of the bolt material

F =the total load on one of the bolts to be used for the joint

Taking the material for the bolts to be a carbon steel of designation AISI/4320, UNS/G43200 as per the international standards from IS 1570, 1979 with a material yield strength, σ_y=464.0MPa.[22]

And the relation between the allowable stress and yield stress of the bolt material is given by the following empirical relation as

$$\sigma_{all} = 0.36\sigma_y = 0.36 * 464 = 167.1MPa$$

Therefore, using this allowable tensile stress of the bolt material, the core diameter of the bolt will be determined as

$$2840d_c + 7551.8 = \frac{\pi}{4} d_c^2 \sigma_{all} * 10^{-6} \quad (115)$$

Substitution of the value for the allowable strength of the bolt yields a quadratic equation on the core diameter of the bolt as

$$131.3d_c^2 - 2840d_c - 7551.8 = 0 \quad (116)$$

The solution of equation 116 is therefore $d_c \approx 24mm$.

This implies that the corresponding bolt nominal diameter is 27mm, and M27 is the standard bolt and Nut designation as per IS: 4218(part III) 1976(Reaffirmed1996). Therefore, for mounting the engine M27 bolt and nut will be used but for other bolted joints M15 can be used because there are no loads on the other bolted joints and hence only initial tightening load will be accommodated by the bolts and nuts.

5.14.2. Design of welded joints

The segment of the mower which is subjected to the worst loading condition at its weld is shown by the following figure.

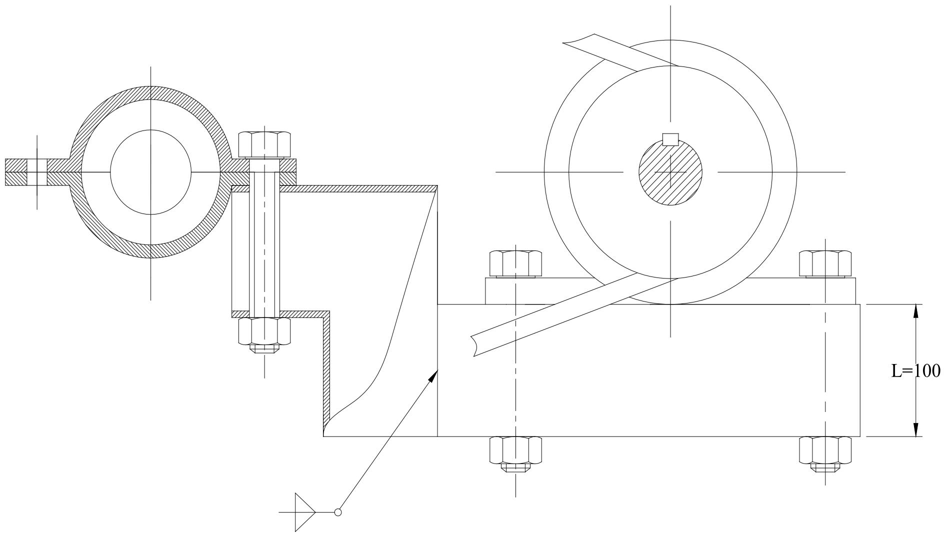

Figure 5.24: **Fillet Weld indicated by the symbol**

The fillet weld at the indicated place is all round weld around the square cross section of the engine seat cantilever on which the weight of the engine causes an eccentric load on the weld by which it would be subjected to bending and shearing load condition.

Therefore, the weld will be designed by applying the maximum shear stress theory. The material of the parts to be welded is determined to be a high strength steel with ultimate tensile strength of 435MPa and yield strength of 290MPa in ASTM specification A242 and type 1. The length of the weld is, therefore, equal to the side of the square as it is indicated in figure 5.24.

The combined normal stress at the root of the weld σ_n is given by

$$\sigma_n = \frac{W_{engine}}{0.37wl} \tag{117}$$

Where W_{engine} = Engine weight = 700N

w = Weld leg dimension

l = Effective length of the weld =0.1m

And the shear stress is also given by

$$\tau = \frac{W_{engine}}{0.707wl} \tag{118}$$

The maximum shear stress theory is represented by

$$\tau_{\max} = \frac{FS}{2}\sqrt{\sigma_n^2 + 4\tau^2} \tag{119}$$

The stress given by equation (119) is the maximum stress which is expressed in terms of the yield stress or the ultimate strength as

$$\tau_{\max} = 0.36\sigma_y \text{ and } \tau_{\max} = 0.18\sigma_u \tag{120}$$

Where $\sigma_y = 290MPa$

Then, $\tau_{\max} = 0.36*290 = 104.5MPa$ or $\tau_{\max} = 0.18*435 = 78.3MPa$

For continuously acting loads, the safety factor FS to be accounted for strength is expressed as:

$$FS = \frac{\sigma_u}{\sigma_y}R \tag{121}$$

Where σ_u = Ultimate strength of the welded materials

σ_y = Yield strength of the welded materials

R = reliability =2 when predictable loads and medium weight is important.

Hence, $FS = \frac{435}{290}*2 = 3$

and hence using equation (119),

$$78.3*10^6 = \frac{3}{2}\sqrt{\left(\frac{2.71*700}{0.1w}\right)^2 + 4\left(\frac{700}{0.0707w}\right)^2}$$

which results in the weld leg dimension $w \approx 10mm$

5.15. Sheet Metal parts and handle for the operator

Design of sheet metal products can be a complex and elaborate process. However, many important aspects of a sheet metal product are determined at the early stages of the design. Sheet metal design differs from the traditional mechanical design in several ways, requires the implementation of specific methods and concepts. The design of sheet metal products should be developed by sketching, based on principles of early incorporation of mechanical drawing and imprecise analysis. Using this approach, various preliminary aspects of a sheet metal part design, such as manufacturability, optimal flat pattern, and fold pattern can be estimated automatically based on only a rough freehand sketch of a product, without requiring accurate details.

Therefore, it needs the generation of the flat pattern or the fold pattern to represent the part in its unfold and fold state respectively. To generate the flat or bent pattern, one has to consider the type and thickness of the sheet metal, the grain direction and the machine setup accounting for how the part will be manufactured in order to develop standard bend allowance that are used in calculating the pattern.

There are optional methods of sheet metal part design, and designers can often use the two common sheet metal part design methods, folded, which is how the part appears after manufacturing, and flat, where the part's flat pattern is first developed and then folded to represent in its real appearance. Each design method has its benefits, but the folded state is easier approach over the flat state, and hence, the folded state method will be used in this work.

In this subtopic, there are parts in this mowing component to be designed and manufactured from sheet metals. These are the harvest collector and covers of the machine both are having the purpose of retaining the cut sesame and covering the internal parts of the mowing components respectively.

5.15.1. Harvest Collector

The harvest collector is designed for the purpose of retaining the cut sesame plant maintaining its upright orientation provided with expansion provision of retaining space so that it can accommodate a large bundle of harvest at a time. It is to be made from sheet metal with dimensions that conform to the geometry of the other adjacent parts of the mower. The following figure shows the component of harvest collector.

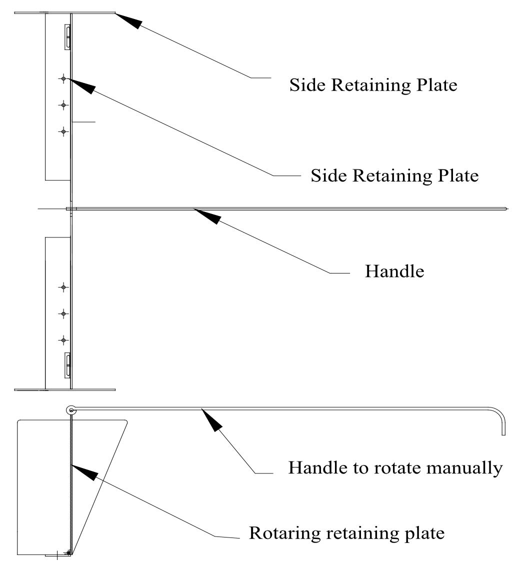

Figure 5.25: **Harvest Collector**

The harvest collector will play the role of accumulating the cut sesame as of buckets in flour mills. It has three parts to be assembled together which are the left and the right side retaining plates, the rotating retaining plate, and the handle used by the operator to rotate the rotating retaining plate as shown in the figure 5.23 above. Among these parts, only the side and the rotating retaining plates are to be manufactured from sheet metal while the handle for angular movement of the rotating retaining plate will be made from steel rod.

Therefore, the material type and the thickness of the sheet metal for the side and rotating retaining plates is required to be stainless steel and 6mm thick for the purpose of corrosion prevention and rigidity requirement respectively, and the dimensions of these parts are determined on the basis of the other adjacent parts of the mowing machine which are shown in the multi-view drawings of the mower.

Based on these specifications, the developed length of the flat state, L, to be used in determining the size of the required flat state sheet metal, which is enough to develop the necessary folded pattern of the part, will be determined by the following relation

$$L = L_1 + L_2 + BA \qquad (122)$$

Where BA = the bend allowance

L_1 = length of sheet metal on one side of the bend

L_2 =length of the sheet metal on the other side of the bend

But the bend allowance depends on the inner and outer bend radii, and as a recommendation of sheet metal forming, the inside bend radius should be equal to material thickness. When the radius is less than recommended, this can cause material flow problems in soft material and fracturing in hard material. Hence, the inner bend radius is taken to be

R_1 =6mm.

All the bending of these parts need 90^0 bend, and for this 90^0bend, using the inner radius, the bend allowance can be determined as

$$BA = \frac{\pi}{2}\left(R_1 + \frac{t}{2}\right) \tag{123}$$

Where t = the sheet metal thickness

R_1 =the inner bend radius

Hence, $BA = \frac{\pi}{2}\left(6 + \frac{6}{2}\right) = 14.14mm$

Therefore, the bends of these parts will be formed by inner radius of 6mm and the bend allowance of 14.14mm.

For the manufacturing aspects, there are shearing, punching, and bending operations, and corresponding to these operations, the shear, punch press, and press brake machines will be used respectively.

5.15.2. Cover plate of the mower

The cover plate is the outermost cover of the internal parts of the mower with the purpose of only covering the body of the machine, and the design of this part will be in the same pattern as in the above subtopic.

Because of the only covering role of this part, its thickness is taken to be 2mm, and the geometric shape of the cover is traced be the dimensions of the internal parts to be covered with, and the two dimensional drawing showing its details is included in the part drawings in the deliverables.

5.15.3. Operator Handle

The operator handle is the part by which the operator will control the direction of the machine motion acting as a handle to walk with the machine from behind. It can be easily fabricated from 40mm diameter steel pipe.

6. Performance Determination of the mower

The performance of the mower can be measured using various means of parameters such as the capacity in hectars per day, the power-to-weight ratio of the mower, the labor requirement, and the cutting height, and these parameters are determined below.

6.1. Capacity Determination

The capacity of the mower is the area of the plantation that the equipment can mow per day. It can be determined from the mowing speed of the equipment and the cutting width that the mower cutters can attain at a time.

The maximum speed of the mower V by which it can move forward during operation is computed using its larger wheels rotational speed ω in radian per seconds and their radii R in meters as

$$V = \omega R \tag{124}$$

For $\omega = 13.3\pi \dfrac{rad}{\sec}$ and $R = 0.21m$ from the design of the first belt drive,

$V = 13.3\pi \, {rad}/{\sec} * 0.21m = 8.77 \, {m}/{\sec}$ which is too large for the operator to walk behind the mower during operation. But this speed can be adjusted by the engine accelerator to synchronize it with the operator walking speed.

And the cutting width that can be covered by the cutter bars at a time is

w =1.5m from the initial design requirement of the cutting width of the mower.

Hence, the maximum capacity of the mower C in $\dfrac{m^2}{\sec}$ for 8 working hours per day

$$C = wV = 1.5m * 8.77\frac{m}{\sec} = 13.2\frac{m^2}{\sec} \approx 38\frac{hectars}{day}$$

For a walking operator with comfort at an approximate speed $V = 1\frac{m}{s}$, which is obtained from a test of comfortable walking on 16.26m long walk way in 18 seconds recorded using tape rule and stopwatch, the capacity of the mower for the normally waking operator

$$C = wV = 1.5m * 1\frac{m}{\sec} = 1.5\frac{m^2}{\sec} = 1.5 * 10^{-4}\frac{hectars}{\sec}$$

And for 8 working hours per day, the capacity of the mower operated by a normally walking operator

$C = 4.32\frac{hectars}{day}$ which is much better that the available alternative harvesting tools and machines with the majority of them are having 2.5 hectars per day in DIN system. [24]

6.2. Power-to-Weight Ratio Determination

Power-to-weight ratio (or specific power or power-to-mass ratio) is a calculation commonly applied to engines and mobile power sources to enable the comparison of the best alternative design to other available alternatives. Power-to-weight ratio is a measurement of actual performance of any engine or power sources. It is also used a measure of performance of a vehicle as a whole, with the engine's power output being divided by the weight (or mass) of the vehicle, to give a metric that is independent of the vehicle's size. Examples of high power-to-weight ratios can often be found in motor bicycles and turbines. This is because of their ability to operate at very high speeds.

Power-to-weight ratios for vehicles and vehicle like movable machineries are usually calculated using curb weight for cars or wet weight for motorcycles, in other words, excluding weight of the driver and any cargo attached to them. [25]

Therefore, the power that the engine of the mower can generate is determined to be 7hp which is taken from the initial design specifications of the first belt drive, and the crub weight of the mower is 227.28kg which is taken from the design of the main wheels.

Hence, the power-to-weight ratio

$$PW = \frac{P}{W} \tag{125}$$

Where $P =$ Power

$W =$ Weight

This implies that the power-to-weight ratio

$PW = \frac{7hp * 0.746 \frac{kW}{hp}}{227.28kg} = 0.03 \frac{kW}{kg}$ which is a better value that majority of similar purpose machineries can have that is about less than $0.02 \frac{kW}{kg}$ in DIN system. [24]

6.3. Labor Requirement Determination

The labor requirement is the number of the operators of the mowing machine. It depends on the number of the operation to be executed during the mowing task, and the main tasks are driving the machine and picking the harvest from the collecting bin when it is full. In this context, the number of operaters is two with the one is the main operator who will be driving by simply walking behind the mower and the other who will collect the harvest when the bin of the machine is full. Therefore, the labor requirement is two men.

6.4. Cutting Height Determination

The cutting height is determined based on the height variation of the harvest which the machine is to be used for. As far as sesame is concerned to the first priority and it is an annual plant growing to 50 to 100 cm tall, the minimum cutting height is determined to be 130mm from the ground and the maximum cutting height is 360mm which are determined from the geometry of the assembly drawing. These values of cutting height of the machine can give the cutting height adjustment allowance of 230mm.

Therefore, the values obtained are compatible with respect to the plant growing height and the designed cutting height achieved.

7. Deliverables

The deliverables are the assembly and part drawings of the mowing machine. These are attached here with this report of the work.

Conclusions and Recommendations

Eventually, from the mechanical design analysis results obtained, it can be inferred that

- The capacity of the sesame mowing machine is 4.32 hectars for 8 working hours per day which is a considerable result as compared with similar alternative harvesting machines.
- The power-to-weight ratio of the machine is $0.03\frac{kW}{kg}$ that is satisfactory for the intended applications, and hence the rated power selected is reliable.
- The labor power required during the harvesting operation is two men, and hence the recurrent problem of very large number of labor power requirement is also sufficiently mitigated.
- The cutting height adjustability is almost solved for the cutting height is determined to be from 130mm to 360mm with an adjustment allowance of 230mm.
- The capacity of the mower dictates that the labor intensiveness of harvesting activity is almost alleviated for 4.32 hectares of a mature cash crop plantation such as of sesame in Ethiopia cannot be harvested by two men in a day with the usual harvesting trends that are being used currently.

Based on the conclusions made, it is recommended that

- The mower can be used to harvest not only cash crops like sesame but also other oil seeds and similar cereals.
- A number of this mowing machine can be in operation at a time for large scale plantations of cash crops such as sesame and the likes since it covers at least 4.32 hectares per day so that production of cash crops, oil seeds, and similar cereals is enhanced and the question in food self-sufficiency of Ethiopia will be on the safe side of the answer.
- This harvesting machine has to be tested after manufacturing in the working environments in which it is going to be used to assure about its functionality.

- Designing such machineries will contribute its share in mechanizing harvesting activities and enhancing the farmers in the country for better productivity of such cash crops.

Bibliography

[1] Felter and Lloyd 1898; McCormick Sesame Seed 2001

[2] The **AFRIK.**COM and **REUTERS** AFRICA Presses

[3] Abe, S., Y. Hirakawa, S. Takagi. 2001. Roasting effects on fatty acid distributions of triacylglycerols and phospholipids in sesame (Sesamum indicum) seeds. J. Sci. Food Agr. 81:620–626.

[4] Annussek, G. 2001. Sesame oil. In: Gale encyclopedia of alternative medicine. Gale Group and Looksmart.

[5] Beckstrom-Sternberg, S.M., and J.A. Duke. "The phytochemical database." http://ars-genome.cornell.edu. (Data version July 1994a).

[6] Beckstrom-Sternberg, S.M., J.A. Duke, and K.K. Wain. "The ethnobotany database." http://ars-genome.cornell.edu. (Data version July 1994b).

[7] Begum, S., T. Furumoto, and H. Fukui. 2000. A new chlorinated red naphthoquinone from roots of Sesamum indicum. Biosci. Biotech. Biochem. 64:873–874.

[8] Cooney, R.V., L.J. Custer, L. Okinaka, and A.A. Franke. 2001. Effects of dietary sesame seeds on plasma tocopherol levels. Nutr. Cancer 39:66–71.

[9] Dark, Graham. On-line medical dictionary. http//www.graylab.ac.uk. (Data version December 12, 1998).

[10] Dipasa. Sesame honey bits. www.dipasa.nl

[11] Food, Industrial, Nutraceutical, and Pharmaceutical Uses of Sesame Genetic Resources, http://www.hort.purdue.edu

[12] http://www.freepatentsonline.com

[13] Tools for Harvesting: http://www.fao.org

[14] Mower- Wikipedia, the free encyclopedia: http://en.wikipedia.org

[15] Flail Mower Work r| Reference.com: http://en.wikipedia.org

[16] Hessayon, D.G. (2007) The Lawn Expert. Transworld Publishers, London. ISBN 9780903505482. P.28/33

[17] Push mowers are reportedly favorites of "eco-friendly landscapers" in metropolitan areas.

[18] Machine Design data book, downloaded from Digital Engineering Library at McGraw-Hill (www.digitalengineeringlibrary.com)

[19] Machine Elements, Section 8, HEARD K. BAUMEISTER, ANTONIO F. BALDO, GEORGE W.MICHALEC, 1987.

[20] Machine Design, Joseph Shigley.

[21] Cam design Handbook, Harold A. Rothbart, McGraw-Hill Companies, Inc., 2004

[22] ASM Metals Handbook, American Society for Metals. Metals Park, Ohio. 1988

[23] Tresemer, David Ward, The scythe book, Green River Tools, 1982.

[24] Introduction by Ian Carruthers, TOOLS FOR AGRICULTURE: A buyer's guide to appropriate equipment, Intermediate Technology Development Group (ITDG), 3rd edition, 1985.

[25] General Motors 2009 Data Book, http://eogld.ecomm.gm.com/, September 5, 2008.