ADDIS ABABA UNIVERSITY

ADDIS ABABA UNIVERSITY INSTITUTE OF TECHNOLOGY SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Design and Development of Self-Prospered Riding Reapers Machine.

Thesis Submitted to the Addis Ababa University Institute of Technology School of Mechanical and Industrial and Engineering In Partial Fulfillment of the Requirements for the Degree Master of Science in Mechanical Engineering)

By

Henok Shiferaw

Advisor: Daniel Tilahun (PhD)

Co-Advisor: Siraj K. (PhD Candidate)

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Abstract

Harvesting is the process of collecting the mature crop from the field. Harvesting operation includes cutting, laying, gathering, transporting, stacking the cut crop. The mechanization of harvesting operation is essential to minimize the cost of harvesting, crop production cost, crop loss, turnaround time, weather risk, and to increase benefit by appropriate technology. In order to achieve the above goal, a manually controlled reaper machine is designed and developed which are support to the Farmers and it will reduce the cost of crop cutting and collecting in field and it will achievable highest performance under the given constraints.

Research methodology consists of series of actions or steps necessary to effectively carry out research and the desired sequencing of these steps. Identify Statement of the problems, Studying the present design of the reaper machine, Review the literature, Product development generic design process steps for reaper harvesters machine, planning, concept development, system-level design, detail design, testing and refinement and product ramp-up, in this step it's define the material properties according to the design. to model the reaper machine by using Catia software, Prototype development, testing for validation and Interpret and report.

Material are selected on the basis of strength requirement of various components of the Reaper machine. The machine is constructed from locally available materials. This self-propelled reaper is operated at forward speed of 2.54 m/s. It has 120 cm size of cutter bar and dropped bundling mechanism. The field capacity 0.306 ha/hr, respectively. The required labor for harvesting one hectare of wheat and rice reaper harvesting field operation are needs 2 man-hr/ha. The quantity of fuel required to fill the tank fully after harvesting the plot was measured to determine the quantity of fuel consumed for reaping the test plot and fuel consumption 1.17 l/h.

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CHAPTER ONE

1 Introduction:

1.1 Background and justification:

Today agriculture plays an important role in countries like Ethiopia. Wheat and Rice is one of the most important crop and staple food of millions of people which is grown in many countries of the world include our country.

Harvesting of crops is an important field operation, it is the process of collecting the mature rice and wheat crop from the field. Harvesting operation includes cutting, laying, gathering, transporting, stacking the cut crop. Harvesting methods being used are manual harvesting (traditional harvesting) and mechanical harvesting.

I. traditional harvesting: - is manual mowing method using sickles.

The harvesting of crops is traditionally done by manual methods. Harvesting of major cereals, pulse and oilseed crops are done by using sickle whereas tuber crops are harvested by country plough or spade. All these traditional methods involve drudgery and consume long time.

II. Mechanical harvesting: - is modern mowing method using machine.

Timeliness of harvest is of prime importance. During harvesting season, often rains and storms occur causing considerable damage to standing crops. Rapid harvest facilitates extra days for land preparation and earlier planting of the next crop. The use of machines can help to harvest at proper stage of crop maturity and reduce drudgery and operation time. Considering these, improved harvesting tools, equipment, combines are being accepted by the fanners.

Mechanized agriculture is the process of using agricultural machinery to mechanize the work of agriculture, greatly increasing farm worker productivity.

In modern times, powered machinery has replaced many jobs formerly carried out by manual labor or by working animals such as oxen, horses and mules. The history of agriculture contains many examples of tool use, such as the plough. Mechanization involves the use of an intermediate device between the power source and the work. This intermediate device usually transforms motion, such as rotary to linear, or provides some sort of mechanical advantage, such as speed increase or decrease or leverage. Current mechanized agriculture includes the use of tractors, trucks, combine harvesters, airplanes (crop dusters), helicopters, and other vehicles.

Until the nineteenth century, most grain was harvested by cutting with a sickle or scythe. During the nineteenth century, mechanical reapers were developed to cut and windrow grain for field drying. Reaper is a wheat and rice harvesting machine which reaps crops and gather Manually. Manually operated vertical conveyor reaper, Self-propelled vertical conveyor reaper, Tractor front mounted vertical conveyor reaper and Self-propelled binding reaper are available.

Reapers are used for harvesting of crops mostly at ground level. It consists of crop-row divider, cutter bar assembly, feeding and conveying devices. Reapers are classified on the basis of conveying of crops as given below:

1. Vertical conveying reaper windrower: - This type of machines cut the crops and conveys vertically to one end and windrows the crops on the ground uniformly. Self-propelled walking type, self-propelled riding type and tractor mounted type reaper-windrowers are available.
2. Horizontal conveying reapers: -This type of reapers is provided with crop dividers at the end, crop gathering reel, cutter bar and horizontal conveyor belt. They cut the crop, convey the crop horizontally to one end and drop it to the ground in head-tail fashion. It’s suitable for wheat, rice, soybean, and gram.
3. Bunch conveying reapers: -This type of reapers is similar to horizontal conveying reapers except that the cut crop is collected on a platform and is being released occasionally to the ground in the form of a bunch by actuating a hand lever.
4. Reaper binders: -The cutting unit of this type of reapers may be disc type or cutter bar type. After cutting, the crop is conveyed vertically to the binding mechanism and released to the ground in the form of bundles. Reaper binders are suitable for rice and wheat.

1.2 Statement of the problems:

In the field operation of most self- propelled reapers machine problems are, before operation it needs to prepare the field by using manually cropping on the four sides of field with width of 20cm and on four corners of field with square of 2 x 2 m and the machine can smoothly turn in field, when the reaper machine changes its direction during harvesting, which drives backward without function and turning round reaping, there is tiredness of farmers because during harvesting the farmer guides the machine by waking and handling .

To overcome these problems, in order to achieve the need for highest performance under the given constraints with reduced tiredness of crop gathering, the hardness of grains harvesting, manual efforts, fuel consumption and cost.

1.3 General objectives:

The general objective of this project is to design and develop Self-Prospered Riding Type Reapers Machine by adding binding mechanism.

1.4 Specific objectives:

- To use product development generic process
- To design and develop Self-Prospered Riding Reapers frame structure according to binding mechanism.
- To design and develop riding and finger conveyor mechanism.
- To design and Select Proper Self-Prospered Riding Reapers Machine component engine, power transmit ion, shaft, bearing and crop cutter knife
- Model 3D of Self-Prospered Riding Reapers Machine by using CATIA V5-R19 software.
- Prototype development
- Testing

1.5 Motivation

The motivation of this research is to minimize the technology gap from the developed countries and to reach the maximum by developing the Self-Prospered vertical conveyor Riding Type Reapers Machine.

1.6 Scope of the Research:

The ability to contribute the scope in design and develop significant to make it new understanding and conclusion. It can be the standard for the development of the material of Design and development Self-Prospered Riding Reapers. The study will only be limited on The scope will extend to cover the design and developed prototype and specification test Design and development Self-Prospered Riding Reapers.

1.7 Significance of the study:

The present study is to address specific problems occurred during crops seed harvesting. The study will be helpful to make harvesting crop seeds to be economical, qualitative and shares world market. It will change the traditional way of harvesting system to a technological way, our traditional harvesting system was by traditional equipment's, so this ways of harvesting system wastes time, man power and also money, so this thing affects the profitability of the farmer's. As we all know our economy bases are the farmers, so the profitability of these farmers directly affects the economic development of our country. But if we proceed to the new machine it will save man power, money and time, so that our general objectives will fulfill through time.

1.8 Thesis Outline

The thesis contents six chapters.

In Chapter 1 discussed introduction about Harvesting Machine, Statement of the problems, General objectives, Specific objectives and Scope of the Research.

In Chapter 2 discussed Literature review, to consider how past designs for reaper harvesters machine may be applicable to this project, examined databases such as, journals, farm books and engineering village. After finding the above literature reference, their associated reference number was further researched to gain more insight into the design. This was used to develop a list of equipment used previously in the reference. Using this reference and adapt some of them. To Use the data that obtained from the references to help design reaper machine. And this chapter contains literature summery.

Chapter 3 discuses Product development generic design process steps for reaper harvesters machine, planning, concept development, system-level design, detail design, testing and refinement and product ramp-up, in this step it's define the material properties according to the design. Embodiment design for each component and developed 3D Model and analysis of Self-Prospered Reapers Harvesting Machine by using Catia software

Chapter 4 puts the whole result from the Self-Prospered Reapers Harvesting Machine output and provides a detailed discussion on the results.

Chapter 5 finally summaries the results and give conclusion of this research work and recollects the main influences to the area of crop harvesting method and give recommendation and future work.

CHAPTER TWO

2 Literature review

This chapter deals with research work done in past by various investigation on the performance, the mechanization of harvesting operation is essential to minimize the cost of harvesting, crop production cost, crop loss, turnaround time, weather risk, and to increase benefit by appropriate technology. crop harvesting is the important part in agricultural mechanization, it is the process of collecting the mature crop from the field. Harvesting operation includes cutting, laying, gathering, transporting, stacking the cut crop. Harvesting methods being used traditional and mechanical harvesting. And also deals about Product development generic design process steps for reaper harvesters machine, planning, concept development, system-level design, detail design, testing and refinement and product ramp-up, in this step it’s define the material properties according to the design.

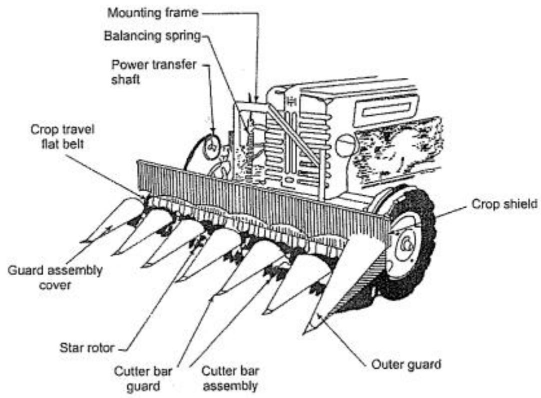
2.1 Study on design and develop a reaper harvesting

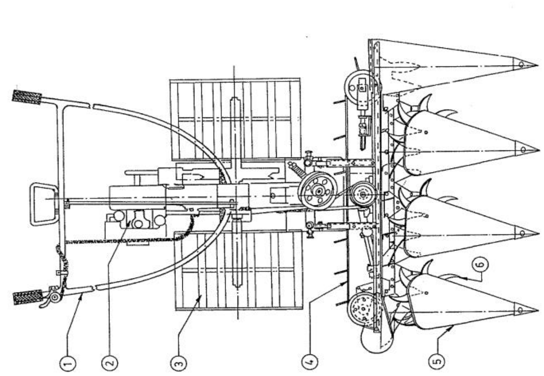
Devani and Pandey (1985), designed and developed a vertical conveyor belt windrower for harvesting wheat crop. They concluded that, field capacity achieved with 1.6 m wide unit was 0.269 ha/h and for 2.09 m wide unit was 0.337 ha/h. The costs of operation with tractor and power tiller models were low as compared to manual method by 20 to 30 %.

El-Sahrigi et.al. (1992), developed a front mounted reaper. The design features included a flat belt mechanism conveying the crop to the side of machine, improve cutter bar star wheel assembly to minimize clogging, a bevel gear drive for power transmission, a robust frame, a header provide design that will not dig in to the soil and provision to covert the flat belt conveyor drivers to chain without frame modification.

J Prasad et. al. (2001-2002) development of a self-propelled riding type reaper windrower powered by small diesel engine. The result of test conducted on 4 varieties of wheat crop, the mean values of plant height, effective field capacity, fuel consumption & total machine loss were 84.6 cm, 0.336 ha/h, 1.35 l/h & 0.610%, respectively.

D.N Sharma and S. mukesh (2010) they studied on the designing of self-propelled vertical conveyer reapers and tractor operated vertical conveyer reaper harvesting machines.





1. tractor operated vertical conveyer reaper b) self-propelled vertical conveyer reapers

Figure 1 vertical conveyer reapers

2.2 Study on comparative field performance of deferent reaper machines

El-Sharabasy (1997) compared four different machines (small combine harvester, power reaper, self-propelled mower and rear mounted mower) in wheat and rice fields. He found the followings: -

1. The minimum grain loss of 1.66 % was obtained under small combine harvester.
2. The highest efficiency of 78.04 % was obtained under self-propelled mower.
3. The highest field capacity of 1.64 fed/h was obtained under tractor power reaper.
4. Harvesting by self-propelled mower + threshing with stationary thresher recorded minimum energy of 25.38, wk./fed.

2.3 Study on tested and evaluated field performance of reaper machine

Ni Ni Aung, Win Pa Pa Myo and Zaw Moe Htet (2012), Field Performance Evaluation of vertical conveyor paddy reaper. The actual field capacity of the reaper was 0.24 ha/hr with a field efficiency of 92 % at an average operating speed of 2.18 km/h.

Mahrous (1995), tested and evaluated a horizontal flat conveyor belt for improving the efficiency of a rear mounted mower. The mower was operated at wheat crop under forward speeds of 3, 4, 5 and 6 km/h. The average field capacity, field efficiency, cutting efficiency and cost requirements were 1.18 fed/h, for the developed rear-mounted mower, respectively.

Habib et.al (2002) stated that the parameters affecting cutting process are related to the cutting tool, machine specifications and plant materials properties. They added that, the cutting energy consumed in harvesting process.

Sahar (1988), reported that, the use of a large scale machine is inappropriate for the following reasons: -it needs high technical experience for operation and maintenance, high capital requirements. Low field efficiency is in small holding and losses of straw are high on irregular furrowed soils. The use of small machines is appropriate for small holdings, low capital requirements and low technical operations and maintenance experience.

Prof. P.B. Chavan, Prof. D.K. Patil and Prof. D.S. Dhondge (2015) are identified that the Reaper are generally selected on the land holding of the farmer, greater the land holding, tractor operated vertical conveyer reaper is choosing, for medium land holding self-prospered vertical conveyer reapers are preferred and for small land holding Manually operated vertical conveyor reaper are preferred.

Manjunatha et. al. (2008) field performance evaluation of vertical conveyor paddy reaper. The actual field capacity of the power reaper was 0.3 ha/hr with a field efficiency of 73 % at an average operating speed of 3.2 km/h. The cost of cultivation of paddy crop could be reduced through mechanization of harvesting operations.

Ni Ni Aung, Win Pa Pa Myo and Zaw Moe Htet (2012) Field Performance Evaluation of a Power Reaper 4GL-120A for Rice Harvesting and stated the Preparation of in order to minimize the loss in reaping, before operation, it is necessary to prepare the field.

- Manually cropping on the four sides of field with width of 20cm to prevent the loss due to the fact that machine can’t reach the sides in reaping.
- Manually cropping on four corners of field with square of 2X 2 m, so that machine can smoothly turn in field as shown in Fig. 2

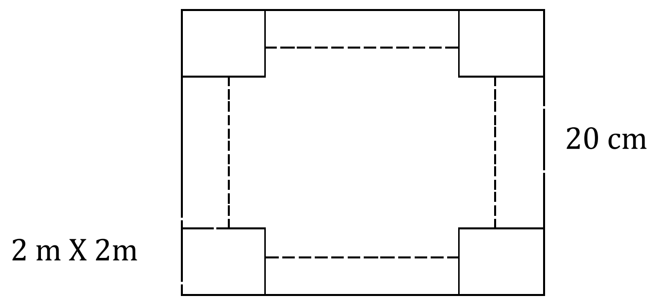


Figure 2 field area

Operation Method, when coming across with protruded part of field land where power reaper cannot operate, as field condition is different, it is necessary to field edge reaping method, which is adaptable to conditions.

- Reaping method of left turn is the field standard operation method as shown in Fig.3

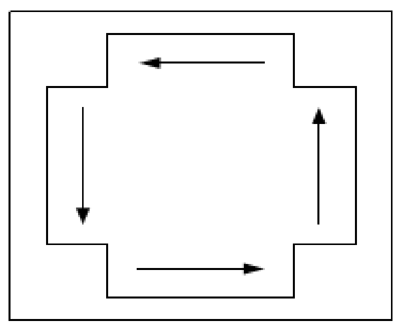


Figure 3 Reaping method of left turn

- Reciprocating reaping method is suited to right angled quadrilateral long and narrow field as shown in Fig.3

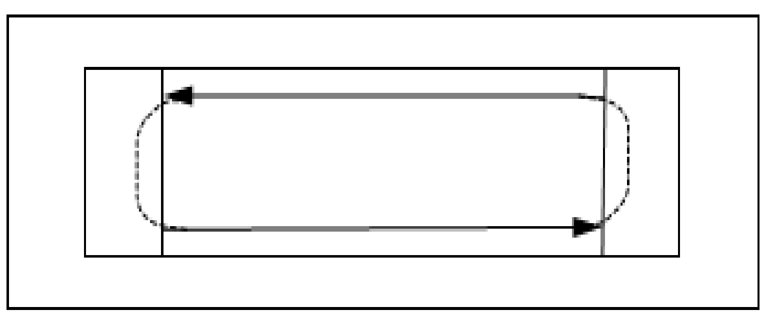


Figure 4 Reciprocating reaping method

Field Performance Evaluation of a Power Reaper for Rice Harvesting Fig.3. When changing the direction of power reaper in small sized field, driving backward and turning round reaping method can be used, and gear shifting and through "driving forward"," reversing". Changing the direction using "driving forward" method is a good method. The turning round reaping method needs no gear shifting, but field edge area is larger than that of using the "reversing". In the wet field, the turning round reaping method is more effective, however, the field edge is wider as shown in Fig.5

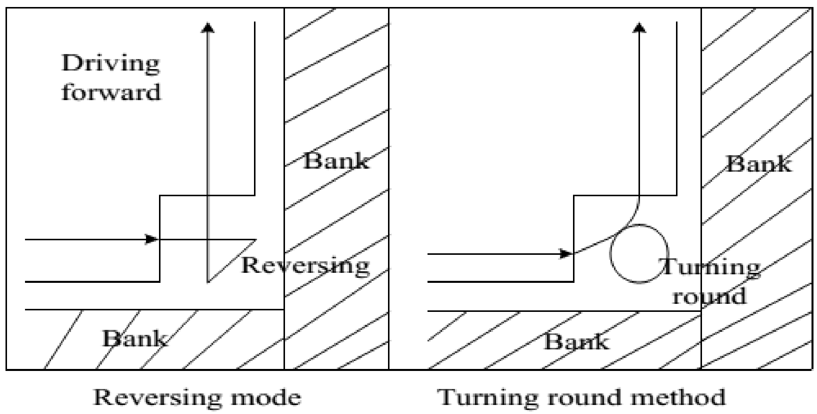


Figure 5 field edge

In the process of harvesting by using self-prospered reaper machine Before it needs to prepare the field or manually cropping which can perform the following operations, Easy harvesting of grains, Less manual efforts, Low cost and less maintenance.

In the field operation of most self- prospered reapers machine problems are: -

1. Before operation it needs to prepare the field, manually cropping on the four sides of field with width of 20cm and on four corners of field with square of 2 x 2 m, so that machine can smoothly turn in field.
2. During harvesting, when the reaper machine changes its direction, which drives backward without function and turning round reaping.
3. There is tiredness of farmers, because during harvesting he guides the machine by waking and handling.

2.4 Product development

Karl Ulrich and Steven D. Eppinger (2016). are identified that the six phases of the generic development process are:

1. Planning (Phase 0)
2. concept development (phase 1)
3. system-level design (phase 2)
4. detail design (phase 3)
5. testing & refinement (phase 4)
6. production ramp-up (phase 5)

Roozenburg, N. and Eekels, J. (1998). are stated that the weighted objectives method, this method is an evaluation method for comparing design concepts based on an overall value per design concept.

R**.S. Khumi and J.K. Gupta (2010).** are stated that manufacturing process steps through which raw materials are transformed into a final product. The manufacturing process begins with the product design, and materials specification from which the product is made.

Generally, From the literature survey is studied to recommend the appropriate system for medium land holding self-prospered vertical conveyer reapers harvesting machine, the system was evaluated according to the technical parameters: knife speed, operating speed, actual field capacity, theoretical field capacity, field efficiency, cutting efficiency, cost economics and percentage of grain losses. To consider how past designs for crop harvesting reaper machine may be applicable to this project and the Literature review is done to solve the above problems which are identified in studying the present machine.

CHAPTER THREE

3 Product development

Product development is the process of creating a new product to be sold by a business or enterprise to its customers. In the document title, Design refers to those activities involved in creating the styling, look and feel of the product, deciding on the product's mechanical architecture, selecting materials and processes, and engineering the various components necessary to make the product work. Development refers collectively to the entire process of identifying a market opportunity, creating a product to appeal to the identified market, and finally, testing, modifying and refining the product until it is ready for production.

Generic Design Process

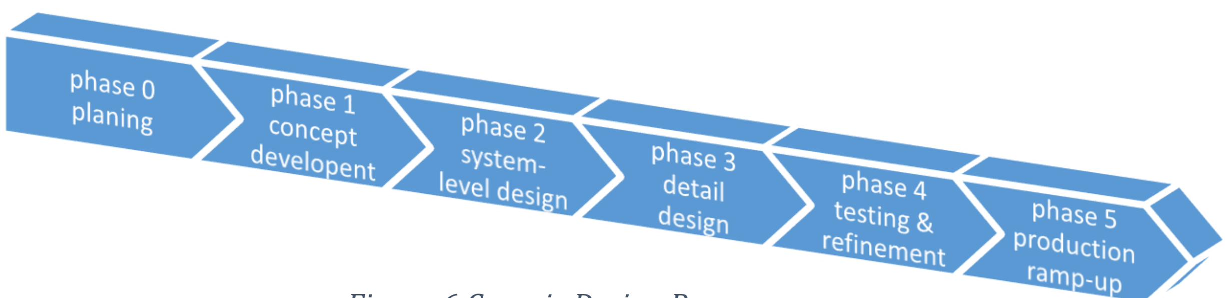


Figure 6 Generic Design Process

3.1 Planning

- ❖ Planning: -is a systematic search for, and selection and development of product ideas but is not a part of production design process but it is pushes the process
  - ✓ focus area
  - ✓ To make mission statement
  - ✓ To get approval for management
  - ✓ Rough idea about the product

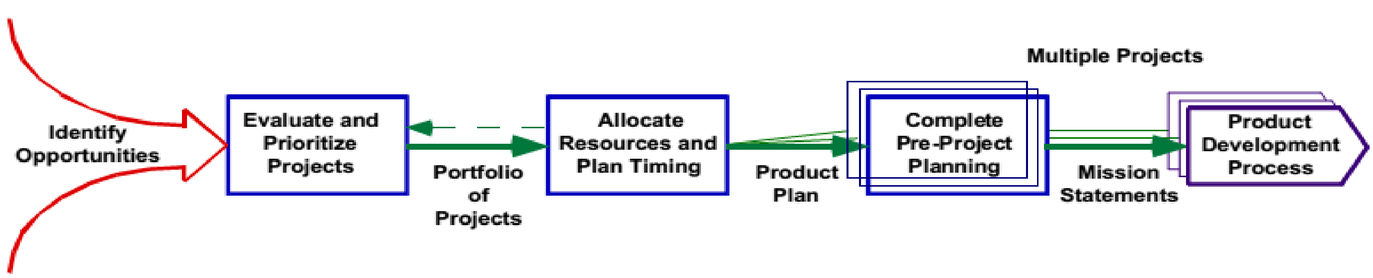


Figure 7 Planning process

Identify opportunities: - is called as opportunities funnel, the source of identifies are mention in below

- ➢ Marketing and sales personal
- ➢ Resource and technology development organization
- ➢ Current production development team
- ➢ Manufacturing and operations organizations
- ➢ Current or potential customers
- ➢ Third parties such as supplies, mentors and business

Evaluate and prioritize projects: - On the basis of four strategies you will evaluate the project

1. Competitive strategy
2. Market segmentation
3. Technological trajectory
4. Produce plate form planning

Complete pre-project planning

1 Technical leader ship
2 Cost-leader ship
3 Customer focus
4 Imitative

3.1.1 Focus area

agricultural on crop harvesting

3.1.2 What is the mission of this machine?

The mission of this thesis is to develop crop harvesting machine for wheat and rice. the purpose of this machine is to harvest and binding the cut crops without tiredness and by minimum cost. The mission of this harvesting machine is to solve the following problems.

This problem is

- ➢ difficult to harvesting and collecting the crops
- ➢ tiredness of farmers
- ➢ high west production
- ➢ more time taken
- ➢ high cost of operation

Primary Market

- Do-it-yourself consumer
  Secondary Markets
- Casual consumer
- Light-duty professional

Stakeholders

- User
- Retailer
- Sales force
- Service center
- Production
- Legal department

3.2 Concept Design

Designers and engineers develop a number of product concepts to illustrate what types of products are both technically feasible and would best meets the requirements of the target specifications. A product concept is a general description of the working principal and form of the product. It is usually expressed as a sketch or as a rough three dimensional model with some brief description The concept generation process begins with a set of customer needs and target specifications and results in a set of product concepts from which the final concept is chosen. The structured approach to concept generation consists of the following five steps:

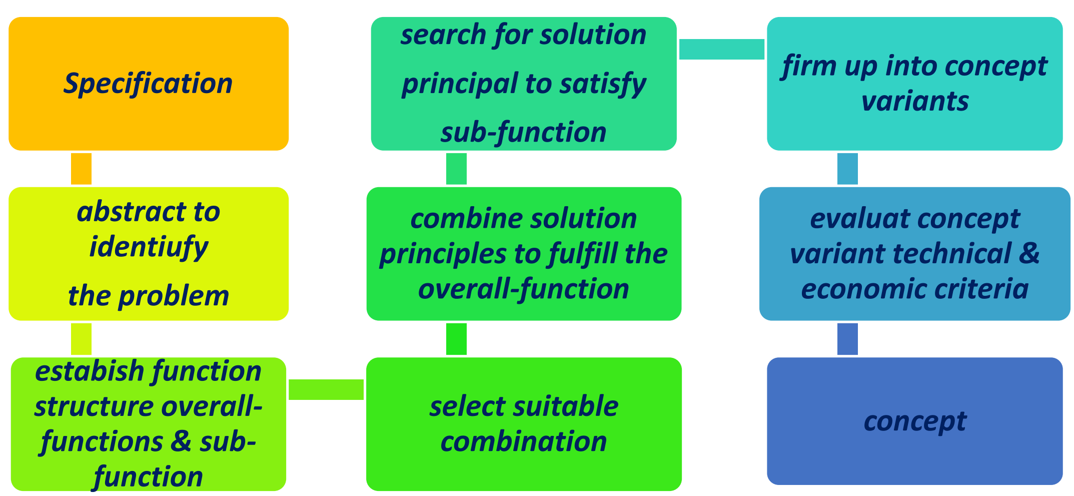


Figure 8 concept generation structured

This is the stage where the feature of the crop harvesting is preset as a stepping stone for the rest of the design process. Here, many alternatives and different features are discussed, compared and specified based on geometrical consideration and expected mechanical, ergonomic and aesthetic requirements.

3.2.1 Overall function

objective of harvesting operation is to harvesting and binding. for different agro-climatic conditions to achieve optimum yields.

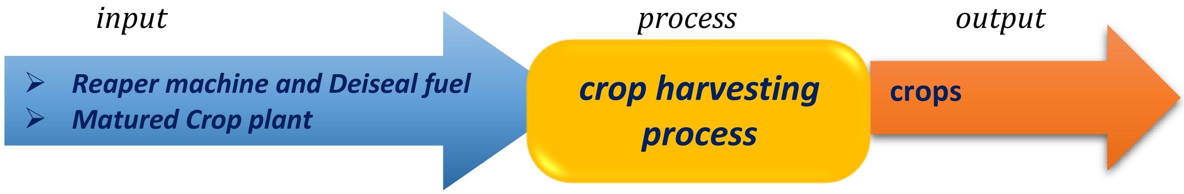


Figure 9 overall function

Definitely the main concern of the design of harvest is to find a simple, with easy manufacturability and relative cost method of metering root seed harvesting so that to meet a specific requirement. Hence the following are some of the competitive mechanisms anticipated to meet the quality and feature needed. After completing the task clarification phase, the conceptual design phase determines the principle solution. This is achieved by abstracting the essential problems, establishing function structures, searching for suitable working principles and then combining those principles into a working structure.

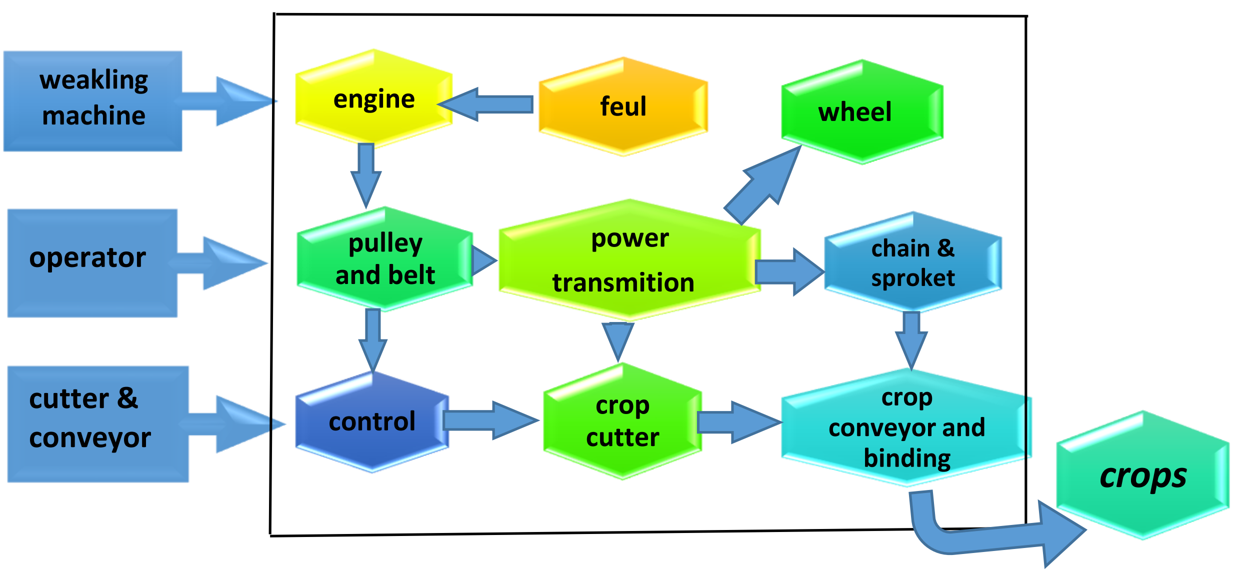


Figure 10 working structure

3.2.2 Concept generation

- **Concept one crop harvesting conveyor and collector by belt leg digger**

- ✓ The purpose of this simple machine is crop harvesting and drop on surface of the ground by belt leg conveyor mechanism.
- ✓ This type of machines cut the crops and conveys vertically to two side end and windrows the crops on the ground uniformly in both direction according to forward reaper direction.
- ✓ It has crop cutter knife, star wheel and belt leg conveyor
- ✓ complex mechanism
- ✓ Used 4 horse power gasoil engine.

Figure 11 Concept one

- ***Concept two crop harvesting conveyor and binding***

- ✓ *The purpose of this simple machine is crop harvesting and conveys vertically to one end and windrows the crops on the ground by binding mechanism.*
- ✓ *It has crop cutter knife, star wheel, belt leg conveyor and binding mechanism devise in to right side windrow.*
- ✓ *complex mechanism*
- ✓ *Used 5 horse power gasoil engine.*

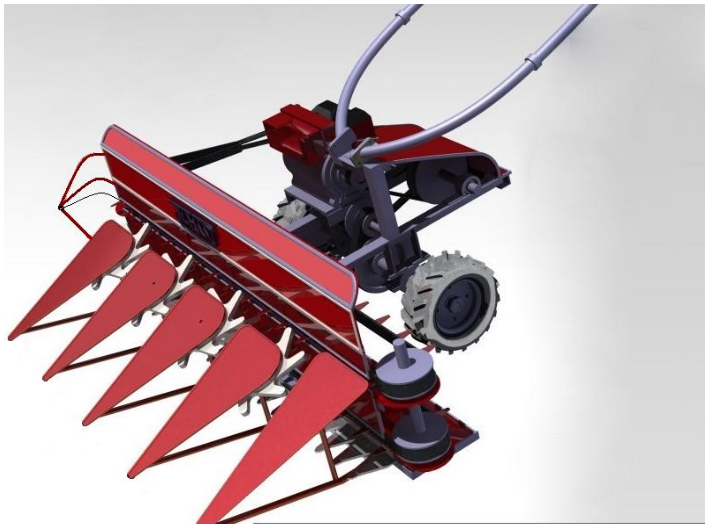


Figure 12concept two

- **Concept three crop harvesting by simple reaper bundle machine**

- ✓ this types of reaper machine cutting the crop and release the cut crop in the middle of the
  machine at ground.
- ✓ Self-waking riding machine.
- ✓ Simple operation.
- ✓ Used 5 horse power gasoil engine.

Figure 13 concept three

Concept four crop harvesting by tractor attached binding reaper machine

- ✓ tractor attached reaper and it has two small wheel.
- ✓ This machine harvest wheat and rice crop.
- ✓ The mechanism of crop conveyor and binding.

- ✓ Used tractor transmit ion power.

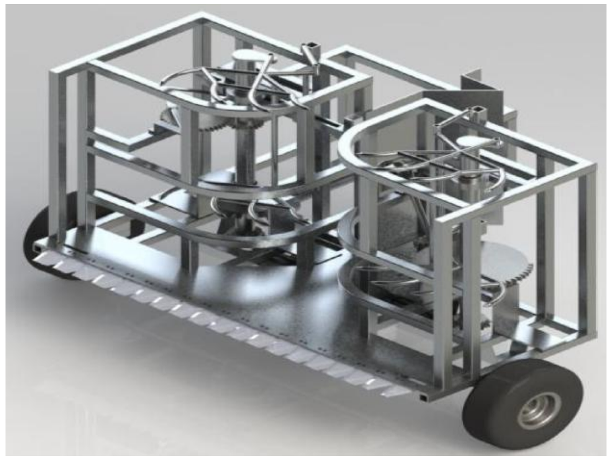


Figure 14 concept four

- ➢ **Concept five Self-Prospered Riding Reapers with binding mechanism**

- ✓ the purposes of this machine is to harvest and binding crop like, wheat and rice crops in modern why.
- ✓ three-wheel driving mechanism Used 5 horse power diesel engine and the speed of the machine is 2.54 km/h.
- ✓ In harvesting process of Self-prospered riding reaper machines is cut the crops and collects crop from on both left and right side and pushes towards the middle position of machine for dropping to the ground between in the form of bundles.

Figure 15 concept five

- **Concept six harvesting by Self-Prospered Riding Reapers Machine**
  - ✓ The purpose of this harvesting machine is crop cutting and binding the cutter crops and drop on surface of the ground by binding mechanism devise.
  - ✓ Over all activity finished one all in all.
  - ✓ This machine powered by 5 horse power.
  - ✓ complex mechanism

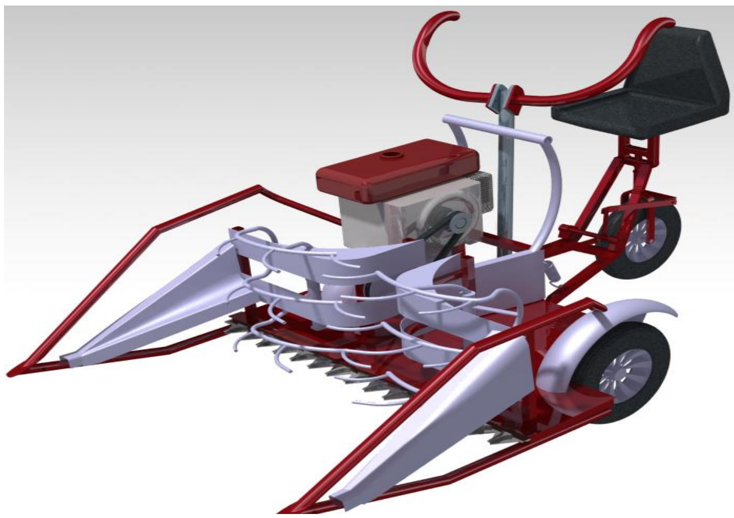


Figure 16 concept six

3.2.3 Concept selection

Through the process of evaluation and tradeoffs between attributes, a final concept is selected. The selection process may be confined to the team and key executives within the company, or customers may be polled for their input. Candidate appearance models are often used for additional market research; to obtain feedback from certain key customers, or as a centerpiece of focus groups. The concept selection method in this section is built around the use of decision matrix for evaluating each concept for toilet seat cleaner with respect to a set of selection criteria.

3.2.3.1 Evaluating and selecting alternatives

The procedure is, therefore, first setting criteria for each sub functions, then comparing the given possible solutions for each sub functions with a reference based on the sated criteria finally selecting the best alternative from the given based on their result. The following are the steps to be used in the selection process.

3.2.3.2 Concept Selection Process

- Prepare the Matrix
  - Criteria
  - Rate of Concept
  - Weightings Rate Concepts
  - Rank the concepts
  - Sum Weighted Scores Combine and Improve
  - Remove Bad Features
  - Combine Good Qualities Select Best Concept
  - Reflect on results and process

3.2.3.3 Generating Alternatives Design

To select one of the superlative choices given for each component, there are steps to be followed.

1. Setting Criteria

2. Calculating a weighting factor for each criterion

3. Evaluating each design with respect to the selected criteria by using a decision matrix.

4. Select the preeminent design based on the decision matrix.

The type and number of criteria are determined by individual judgment. There are no proper set of rules for setting design criteria, since it depends on the type and application of design and its complication.

Concept selection for root seed harvesting machine Concept Scoring Matrix. The generic steps involved in concept selection using concept scoring matrix are:

1. Develop evaluation criteria

2. Assign importance rate to each criterion

3. Rate each concept with respect to each evaluation criterion

4. Rank the concepts

5. Combine and improve the concepts

6. Select one or more concepts

7. Reflect on the results and process

3.2 Concept Selection Criteria

The following selection criteria for root seed harvesting machine was finalized by taking into consideration the customer needs:

1. Performance
2. Inexpensive
3. Easy to operate
4. Durability
5. Ergonomic
6. Simple Design
7. Easy of Manufacture
8. Simple Assembly
9. Ease of Maintenance

The next task was assigning weights to each selection criteria in the scoring matrix. Several different schemes can be used to weight the criteria such as assigning the importance value from concepts for root seed harvesting with respect to selection criteria was done on a scale of 1-6 with following definitions for each scale.

3.2.3.4 The weighted objective method

The weighted objective method for crop harvesting machine is shown in Table. The method uses a weighted sum of the ratings to determine concept ranking. The reference points for each criterion are signified by bold rating values.

The weighted objectives method is an evaluation method for comparing design concepts based on an overall value predesign concept and it involves assigning weights to the different criteria. The weighted objectives method is best used when a decision has to be between a select number of design alternatives, design concepts or principle solution.

Table 1 the weighted Concept selection objective method

Customer requirements	*weight*	*All concept*											
		Concept one		*Concept two*		*Concept three*		*Concept four*		*Concept five*		*Concept six*	
		rating	*Weight score*	*rating*	*Weight score*	*rating*	*Weight score*	*rating*	*Weight score*	*rating*	*Weight score*	*rating*	*Weight score*
performance	*15*	*1*	*15*	*2*	*30*	*2*	*30*	*3*	*45*	*5*	*75*	*7*	*105*
cost	*15*	*5*	*75*	*5*	*75*	*4*	*60*	*4*	*60*	*4*	*60*	*2*	*30*
Easy to OPERATION	*10*	*3*	*30*	*2*	*20*	*3*	*30*	*3*	*30*	*4*	*40*	*5*	*50*
Simple Design	*10*	*5*	*50*	*4.5*	*45*	*4*	*40*	*3*	*30*	*3*	*30*	*2*	*20*
Easy of Manufacture	*12*	*4*	*48*	*4*	*60*	*4*	*48*	*4*	*48*	*3*	*36*	*2*	*24*
Simple Assembly	*8*	*4*	*32*	*4*	*32*	*3.5*	*28*	*3*	*24*	*3*	*24*	*2*	*16*
Ease of Maintenance	*10*	*6*	*60*	*4*	*40*	*5.5*	*55*	*5*	*50*	*4*	*40*	*3*	*30*
Fuel consumption	*5*	*3*	*30*	*4*	*40*	*5*	*50*	*4.5*	*45*	*4*	*40*	*2*	*20*
Life time	*4*	*3*	*12*	*3*	*12*	*2.5*	*10*	*2.5*	*10*	*4*	*16*	*10*	*40*
Total score		*353*		*354*		*351*		*342*		*361*		*335*	
rank		*3*		*2*		*4*		*5*		*1*		*6*	
Sum, total score		*no*		*no*		*no*		*no*		*yes*		*no*	

The weighted objective method indicates that Concept five in which "Self-Prospered Riding Reapers Machine" received the highest score of 361. All other concepts remained below the score of concept number five. Therefore, Concept five was finally chosen as the best design for crop harvesting modern reaper harvesting.

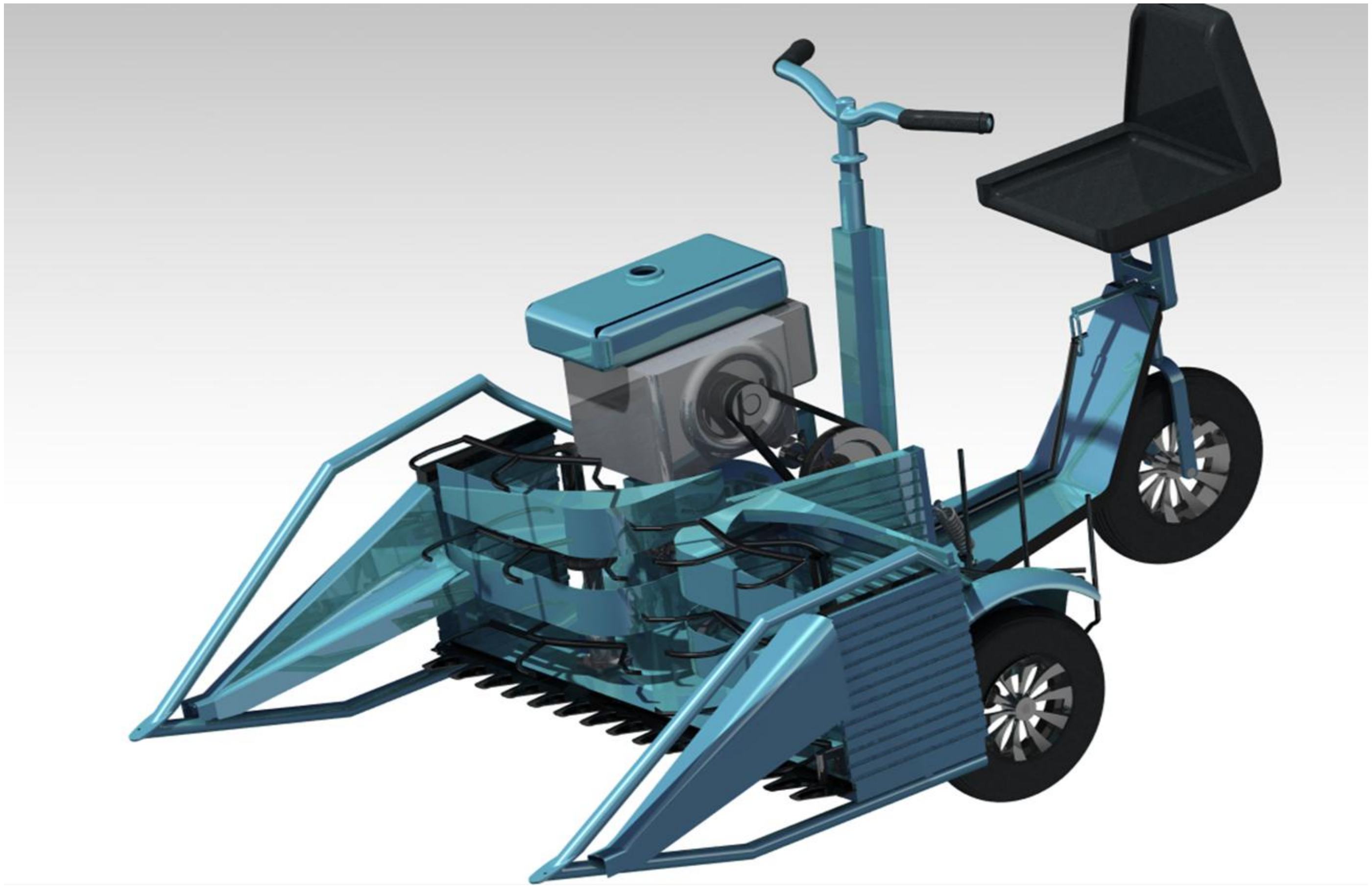


Figure 17 self-prospered riding reaper machine

3.3 System Design

System-level design, or the task of designing the architecture of the product, is the subject of this stage. In prior stages, the designer was focused on the core product idea, and the prospective design was largely based on overviews rather than in-depth design and engineering. The goal of a system approach to reaper machine design (system design) is to define the technical specification, so each component in such a way that the machine as a whole performs its functions according to assigned procedures and objectives.

Technical specifications mean a set of physical measurements that define each part completely, without the use of detailed drawings; sometimes technical specifications are a complement to a conventional detailed drawing, but can be used as a substitute.

Detailed drawings are needed solely to explain how apart must be manufactured, while technical specifications explain the function of apart, how its performance can be measured together with the acceptable values for this performance. Clearly, coherence between detailed drawings and technical specifications is necessary, but only the latter guarantee that the performance targets can be reached.

design and develop Self-Prospered Riding Reapers Machine processing tree

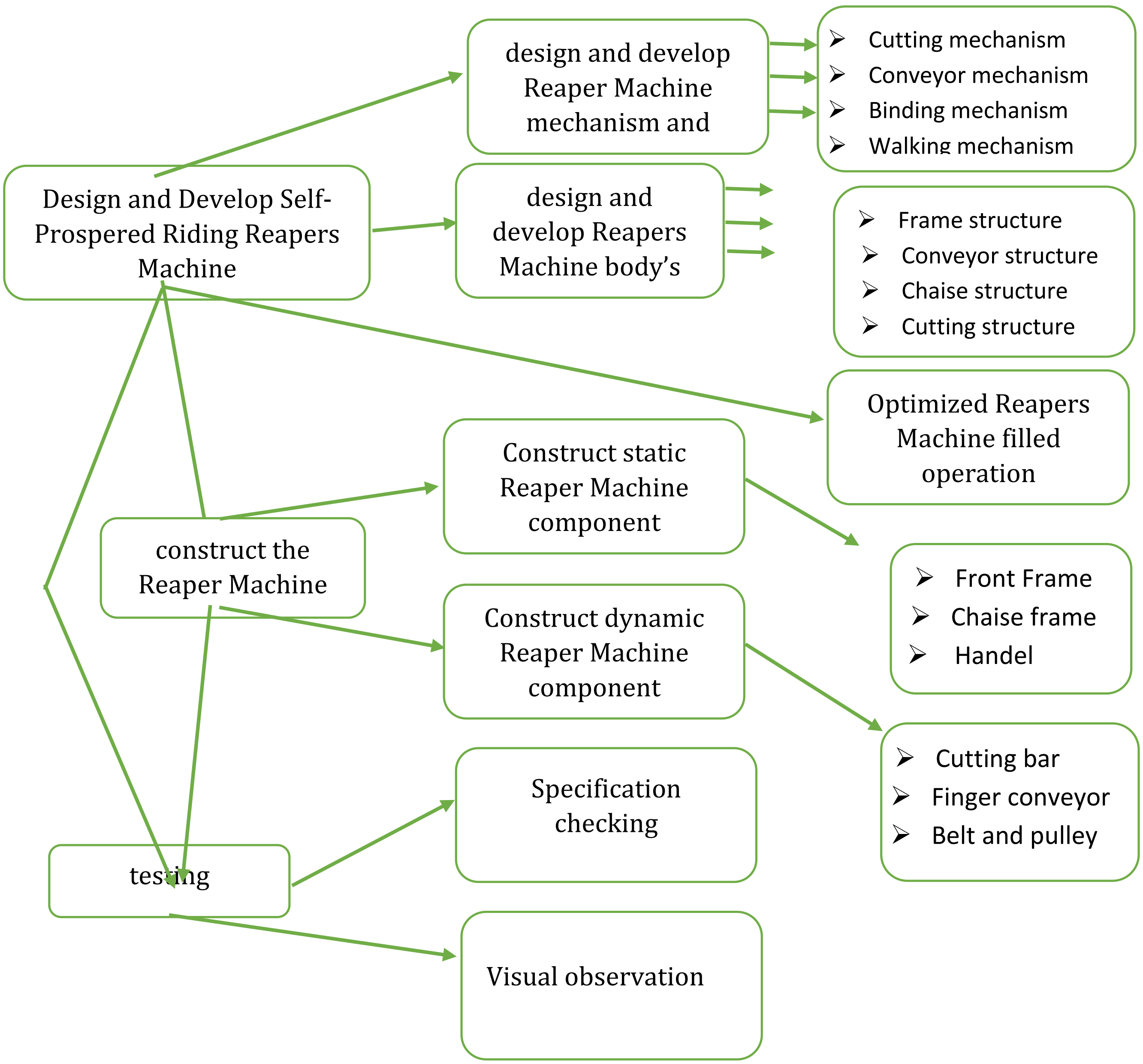


Figure 18 processing tree

3.4 Embodiment design

3.4.1 Definition

Embodiment design is that part of the design process in which, starting from the principle solution or concept of a technical product, the design is developed in accordance with technical and economic criteria and in the light of further information, to the point where subsequent detail design can lead directly to production.

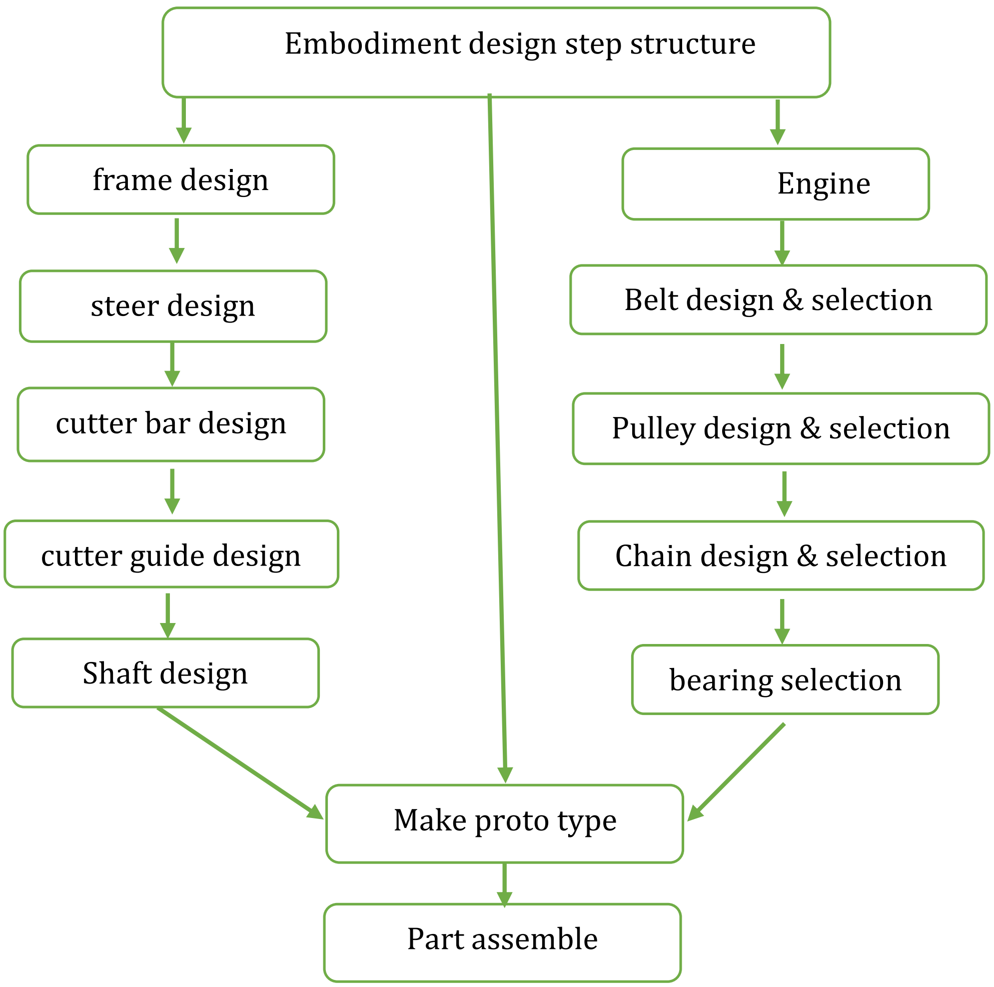


Figure 19 Embodiment design step structure

3.5 Framework of Methodology

This thesis Research methodology consists of series of actions or steps necessary to effectively carry out research and the desired sequencing of these steps. The first part of the research is a frame of reference in a literature study. The second part of the study focuses on mathematical and theoretical method The three part of the study focuses on model and prototype of Self-Prospered Reaper Harvesting Machine by using CATIA software. which 3D model are made using part modeling and assembly design module generated in CATIA software.

3.5.1 Materials:

In choosing equipment and materials for the design harvesting reaper machine, certain factors/elements needed to be considered in order to ensure that this machine will be successfully fabricate.

3.5.2 Material used for various components

Material as selected on the basis of strength requirement of various components of the mechanism. The machine is expected to be constructed from locally available materials;

The component of the machine will be: -

1. Engine with support frame
2. Machine body
3. Cutting bar and cutting lip.
4. Crop divider
5. Ground wheel
6. Frame structure
7. Steering and handle
8. finger conveyor
9. Power transmission belt
10. Bolt and nut

The materials include standard parts and manufactured parts used in the above component the machine. The reaper machine structure is made with steel beams of rectangular hollow section (RHS), and angle iron with different size. The sizes of RHS used are RHS 40X40X4 mm, 30X30X2.5 mm, 25X25X2 mm, 20X20X1.5 mm and angle iron 40X40X4mm and 30x30x3mm.

Table 2 Material property of RHS and angle iron steel grade C450L0/C350L0 [DURACAL(2003)].

No.	Material property	Structural steel
1.	Tensile yield strength	350 MPa
2.	Compressive yield strength	350 MPa
3.	Tensile ultimate strength	430 MPa
4.	Density	7850 kg/m3
5.	Young's modulus	200 GPa
6.	Elongation in degree	26^0

Table 3 material and component

NO	Component(part of the machine)	DESCRIPTION	Material used
1.	Engine with support	5 hp engine	gasoil engine
		Square Bar 4*40mm	RHS st.37.2
		Angle bar 4*40mm	Angle iron St. 37.2
2.	Frame	Square Bar 4*40mm	RHS st.37.2
		Square Bar 3*40*40mm	<<
3.	Machine body	0.8 mm thk sheet metal	<<
		angle Bar 25*30mm	Angle iron St. 37.2
		Square bar 20*20*1mm and 3*30*3mm	RHS st.37.2
4.	Cutting bar and cutting lip.	Flat bar 100*5mm	St. 37.2
		Flat bar 100*3mm	St. C 45
		Angle bar 4*40mm	Angle iron St. 37.2
		angle Bar 3*30mm	<<
		Square Bar 2*20mm	RHS st.37.2
		1mm sheet metal	<<
5.	painting	Painting spray anti-reset and silver	Shin spray
6.	Ground wheel	35mm Ø Round bar shaft	St. C 45
		Apollo 165/80 tire	rubber
		Apollo 165/80 chercke	Mild steel
		Deep groove ball brg, (30X62X16)	standard
7.	Steering wheel	Apollo 165/80 chercke	standard
		Apollo 165/80 tire	rubber

		Hex. Head bolt, m30øx3x200mm long	standard
		Breaking and fuel injecting control wire	Rope wire
		Bicycle breaking and fuel injecting control	standard
8.	*Finger conveyor*	*'8 mm round bar*	St. 37.2
		Pulley	*Cast iron*
		Double bearing holder (20x42x14mm)	*standard*
9.	*Power transmit ion belt*	*v-Belt for 1st stage (A 39)*	*rubber*
		v-Belt for 3rd stage (A 24)	<<
		v-Belt for 4th stage (A 52)	<<
10.	*Power transmit ion chain and sprocket*	*Chain 2nd stage (pitch=12.5,center to center=400 mm)*	*standard*
		sprocket 2nd stage driver, (ID. 20 mm, OD. 60 mm). driven (ID. 22 mm, OD. 150mm)	<<
		Chain 3rd stage (pitch=12.5,center to center=450 mm)	<<
		sprocket 3rd stage driver, (ID. 35 mm, OD. 60 mm). driven (ID. 22 mm, OD. 150mm)	<<
11.	*Bolt and nut*	*Hexagonal head M5,m8,m10,m12*	<<

3.5.3 Frame structure design

The Frame structure design according to binding mechanism. frame element is formulated to model a straight bar of an arbitrary cross-section, which can deform not only in the axial direction but also in the directions perpendicular to the axis of the bar. The bar is capable of carrying both axial and transverse forces, as well as moments. Therefore, a frame element is seen to possess the properties of both truss and beam elements. In fact, the frame structure can be found in most of our real world structural problems, for there are not many structures that deform and carry loadings purely in axial directions nor purely in transverse directions. [Lorenzo Morello, Lorenzo Rosti Rossini, Giuseppe Pia and Andrea Tonoli (2011)].

3.5.4 Design Chassis Frame

Determine the bending moment and shear force diagrams for a simply supported beam with an overhang as shown in fig.20 the beam is 150cm. long with a 10 cm. overhang. It is subjected to a standard human weight 65 kg along the center 10 cm of its span.

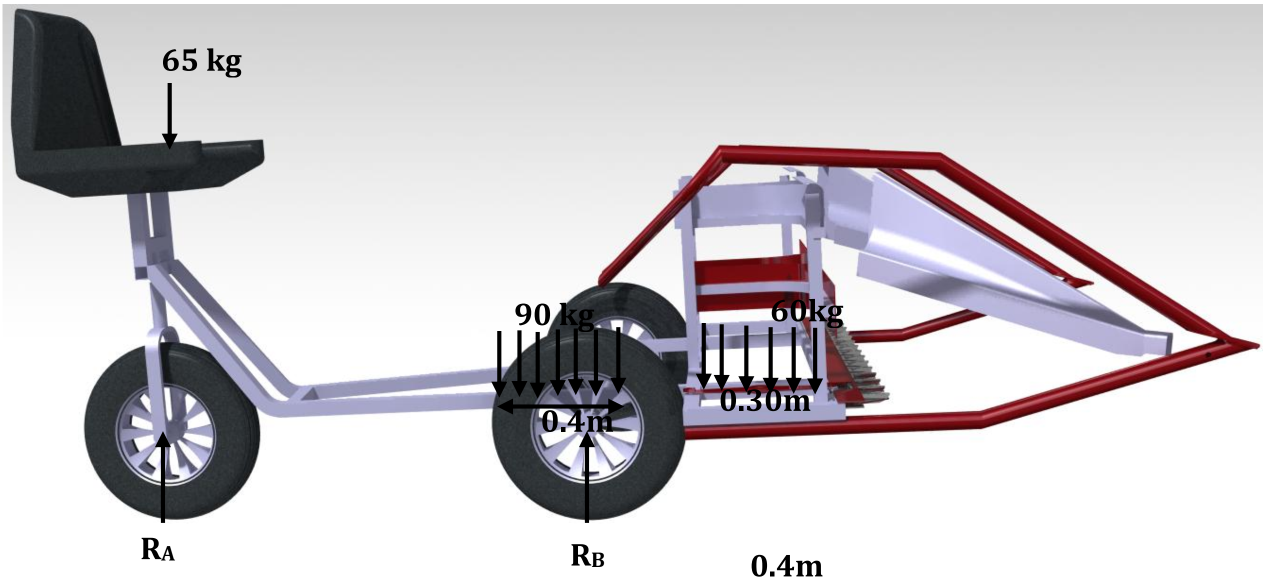


Figure 20 frame and chaise structure

$$P_{load\ 1} = m_{middle} * g \quad (3.1)$$

$$P_{load\ 1} = 90\ kg * 9.81\ m/s^2$$

$$P_{load\ 1} = 882.9\ N$$

$$P_{load\ 2} = m_{front} * g \quad (3.2)$$

$$P_{load\ 2} = 60\ kg * 9.81\ m/s^2$$

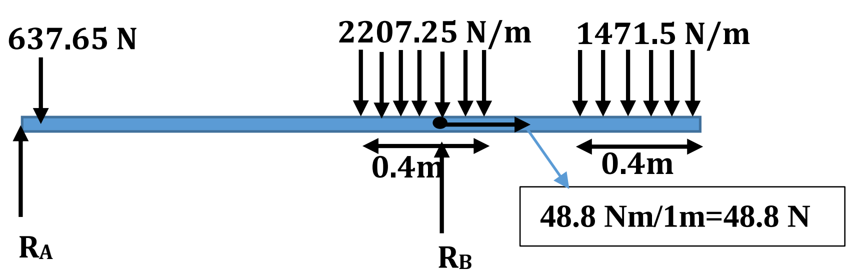
$$P_{load\ 2} = 588.6\ N$$

$$P_{total\ load} = 882.9\ N + 588.6\ N$$

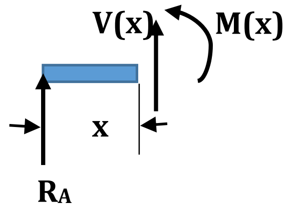
$$P_3 = 65\ kg * 9.81\ m/s^2$$

$$P_{3} = 637.65\ N$$

$$P_{total} = 2109.15\ N$$



(a)overhung beam



(b)Support reactions.

a. Segment AF

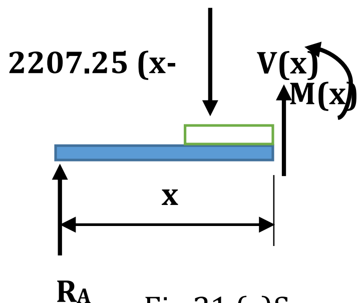


Fig 21 (c)Segment AF

b. Segment FG

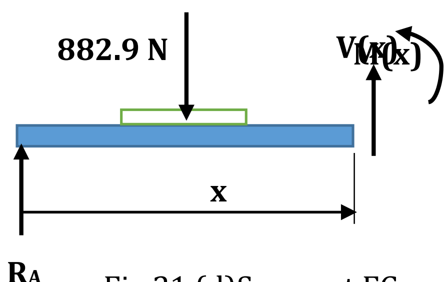


Fig 21 (d)Segment FG

Figure 21(a)overhung beam , (b)Support reactions. (c, d) Segment AF

Step 1 calculation of reactions

The two reaction (R_A and R_B) of the beam are obtained from a transverse EE and a rotational EE written at E, which is the midpoint of the distributed load p.

$$\sum_y M: \quad R_A + R_B - P_{y_{total}} = 0 \tag{3.3}$$

$$R_A + R_B - P_1 - P_{dis\ load\ 1} - P_{dis\ load\ 2} - P_{\ 3} = 0 \tag{3.4}$$

$R_A + R_B - 2109.15\ N = 0$

$$\sum_a M: \quad 0 * MR_A + 1.1MR_B - 0.01mP_1 - 1mP_{dis\ load\ 1} - 1.4M * P_{dis\ load\ 2} = 0$$

$$0 + 1.1MR_B - 0.01m * 637.65\ \ N - 1M * 882.9\ N - 1.4M * 588.6\ \ N = 0$$

$1.1MR_B - 6.3765\ NM - 882.9\ \ NM - 824.04\ NM = 0$

$1.1MR_B = 1713.3165NM$

$R_B = 1713.3165NM/1.1M$

$R_B = 1557.56\ N$

AND

$R_A + R_B - 2109.15\ N = 0$

$R_A + 1557.56\ N - 2109.15\ N = 0$

$R_A = 551.59\ N$

The accuracy of the reaction can be checked by writing the moment EE at any convenient location.select locations at A and B because the moment is 0 at these points ($M_A = M_B = 0$).

$$\sum_a M: \quad 0 * MR_A + 1.1MR_B - 0.01mP_1 - 1mP_{dis\ load\ 1} - 1.4M * P_{dis\ load\ 2} = 0$$

$$0 + 1.1MR_B - 0.01m * 637.65\ \ N - 1M * 882.9\ N - 1.4M * 588.6\ \ N = 0$$

$$1.1MR_B - 6.3765\ NM - 882.9\ \ NM - 824.04\ NM = 0$$

$$1713.316 - 6.3765\ NM - 882.9\ \ NM - 824.04\ NM = 0$$

$$0 = 0$$

AND

$$\sum_b M: \quad 1.1MR_A + 0 * MR_B - 1.09mP_1 - 0.1 * mP_{dis\ load\ 1} + 0.4P_{dis\ load\ 2} = 0$$

$$1.1 * 551.59\ N - 1.09m * 637.65\ \ N - 0.1M * 882.9\ \ N + 0.4M * 588.6\ N = 0$$

$$0 = 0$$

Step 2-bending mmoent and shear force diagrams

Because of the nature of load disrtibution, the BM and SF diagrams have to be constructed separately for four segment: AF, FG, GB, and BC as shown in fig (c_1) to (c_4). The BM and SF for each segment follow.

Segment AF $(0 \leq x \leq 1M)$

$V(x) = -R_A = -551.59\ \ N$

$M(x) = 551.59\ \ N * X$

Segment FG

$V(x) = -551.59\ \ N + \mathbf{2207.25}(x - 0.4M)$

$V(x) = -551.59N - \mathbf{2207.25} * x - 882.9\ NM)$

$$M(x) = 551.59N * X - \frac{\mathbf{2207.25}}{2}(x - 1m)^2 = 551.59X - 1103.6(x - 1m)^2)$$

Segment GB

$V(x) = -551.59N + 588.6N = 37.01\ N$

$V(x) = -551.59\ x - 588.6N(x - 1.4)$

$V(x) = -1140.2\ N * x + 824.04\ N$

Segment GB: for simplicity, the calculation for, this section uses the loads in the overhang. $V(x)\ and\ V(x)$ are positive as marked in Fig 9-1

$V(x) = -1557.56\ \ N$

$M(x) = -1557.56\ \ N(0.4 - X)$

The SF and BM are verified for few locations in bbeam:

Location F: The SF and BM callculated at F(x=1m) from segments AF and FG must be in agreement.

Segment AF

$V_F = -R_A = -551.59\ N$

$M_F = 551.59\ N * 1\ m$

$M_F = 551.59\ N.m$

Segment FG

$V_F = V(x = 0.4m) = -551.59\ N + 2207.25(x - 0.4M) = -551.59\ N$

$M_F = V(x = 0.4m) = 2207.25\ N * x$

$M_F = 2207.25N * 0.40M$

$M_F = 882.9\ N.m$

Segment G

$V_G = V(x = 1.4m) = -551.59\ \ N + 882.9N = 331.31\ N$

$M_G = 551.59\ x - 882.9N(x - 1.4)$

$M_G = 551.59\ x\ N - 882.9x + 1236.06\ N$

$M_G = 1236.06\ N - 331.31\ \ N.m * x$

Segment B

$V_B = V(x = 0.4m) = V(x) = -588.6\ N$

$M_B = -588.6\ N(0.4 - X)$

$M_B = 235.44N - 588.6\ N * x$

$235.44\ N - 588.6\ N * 0.4$

$M_B = 0$

$V_B = V(x = 0) = V(x) = -1557.56\ N + 588.6\ N = -968.96\ N$

$M_B = -588.6\ N(0.4 - X)$

$M_B = -235.44\ N$

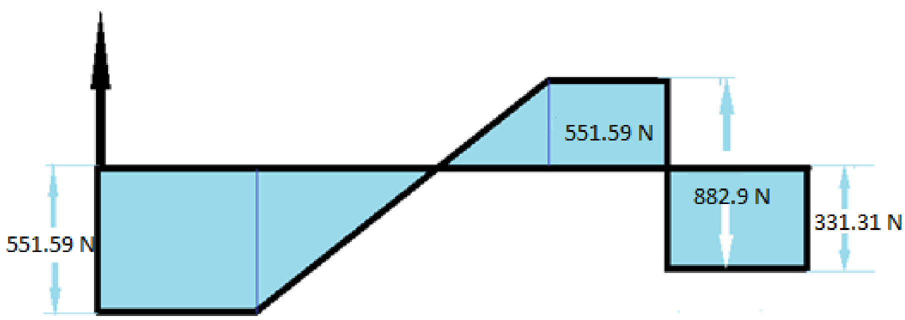


Figure 22 shear stress

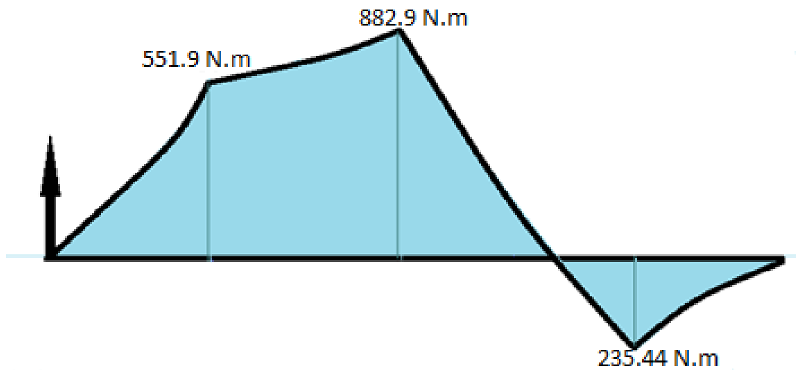


Figure 23 beandin g moment

shear stress on the middle rectangular mild steel

Determine the shear stress along the z-plane of the edges of the 40*40*4 rectangular cross section and the load apply on the four rectangular RHS is 882.9 N.

given: h = 40 mm, b = 40 mm , t =4 mm , $V_d = 882.9\ \mathrm{N}/4 = 220.725\mathrm{N}$

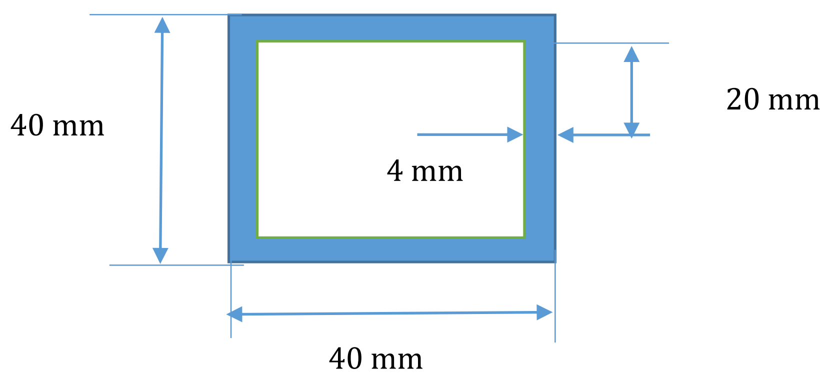


Figure 24 shear stress

$$I_Z = \frac{b * h^3}{12} - \frac{(b - t) * (h}{12} \qquad (3.5)$$

$$I_Z = \frac{40 * 40^3}{12} - \frac{36 * 36^3}{12} = \frac{40^4}{12} - \frac{36^4}{12} = \frac{2560000 - 1679616}{12}$$

$$I_Z = 73365.33\ mm^4$$

$$\bar{y} = 20\ mm$$

$$Q = \Delta A * \bar{y} = 4 * 4 * 20 = 320\ mm^3$$

$$q = \frac{VQ}{I} = \frac{220.725\text{N } N * 320 \ mm^3}{16278.67 \ mm^4} = 0.963 \ N/mm$$

q is the total shear flow acting on the particle with two cutting planes)

$Shear \ Flow \ per \ unit \ length \ (\tau)$

$$\tau = \frac{1}{2}\frac{q}{t} = \frac{1}{2} * \frac{0.963N/mm}{4 \ mm} = 0.12 \ N/mm^2$$

Determine the shear stress along the z-plane of the edges of the 30*30*2 rectangular cross section and the load apply on the four rectangular RHS is 882.9 N.

given: h = 30 mm, b = 30 mm , t =2.5 mm , $V_d = 882.9 \text{ N}/4 = 220.725\text{N}$

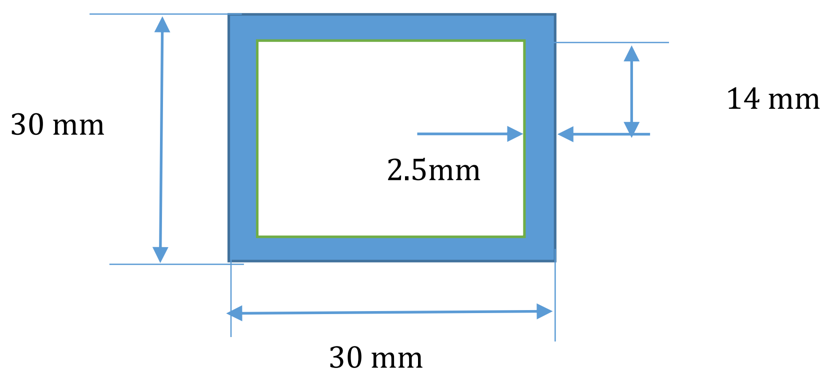


Figure 25 shear stress

$$I_Z = \frac{b * h^3}{12} - \frac{(b-t)*(h-t)^3}{12} \qquad (3.6)$$

$$I_Z = \frac{30 * 30^3}{12} - \frac{28 * 28^3}{12} = \frac{30^4}{12} - \frac{28^3}{12} = \frac{810000 - 614656}{12}$$

$I_Z = 16278.67 \ mm^4$

$\bar{y} = 14 \ mm$

$Q = \Delta A * \bar{y} = 2 * 2 * 14 = 56 \ mm^3$

$$q = \frac{VQ}{I} = \frac{220.725\text{N}\ N * 56\ mm^3}{16278.67\ mm^4} = 0.76\ N/mm$$

q is the total shear flow acting on the particle with two cutting planes)

Shear Flow per unit length (τ)

$$\tau = \frac{1}{2}\frac{q}{t} = \frac{1}{2} * \frac{0.76N/mm}{2\ mm} = 0.19\ N/mm^2$$

shear stress on the chair supporter RHS

Determine the shear stress along the z-plane of the edges of the 30*30*2 rectangular cross section and the load apply on the two rectangular mild steel is 637.65 N.

given: h = 30 mm, b = 30 mm , t =2.5 mm , V_d = 882.9 N/4 = 318.825N

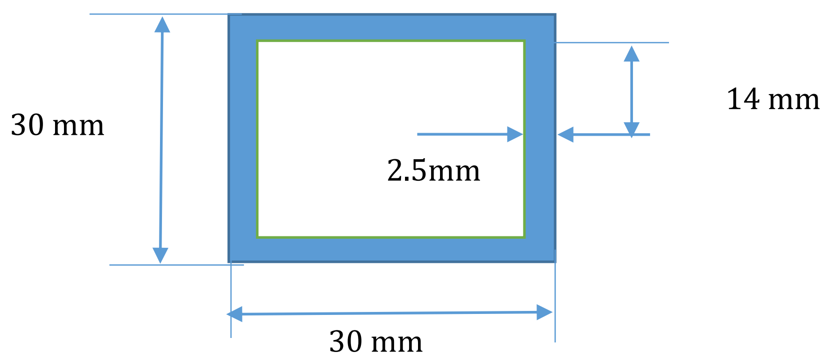


Figure 26 shear stress

$$I_Z = \frac{b * h^3}{12} - \frac{(b-t) * (h-t)^3}{12} \quad (3.7)$$

$$I_Z = \frac{30 * 30^3}{12} - \frac{28 * 28^3}{12} = \frac{30^4}{12} - \frac{28^3}{12} = \frac{810000 - 614656}{12}$$

$$I_Z = 16278.67\ mm^4$$

$$\bar{y} = 14\ mm$$

$$Q = \Delta A * \bar{y} = 2 * 2 * 14 = 56\ mm^3$$

$$q = \frac{VQ}{I} = \frac{318.825\ N * 56\ mm^3}{16278.67\ mm^4} = 1\ N/mm$$

q is the total shear flow acting on the particle with two cutting planes)

$Shear\ Flow\ per\ unit\ length\ (\tau)$

$$\tau = \frac{1}{2}\frac{q}{t} = \frac{1}{2} * \frac{1N/mm}{2\ mm} = 0.25\ N/mm^2$$

shear stress on the conveyor supporter rectangular mild steel

Determine the shear stress along the z-plane of the edges of the 25*25*2 rectangular cross section. and the load apply on the four rectangular RHS is 588.6 N.

given: h = 25 mm, b = 25 mm , t =2 mm , $V_d = 588.6\frac{\text{N}}{4} = 147.15\text{N}$

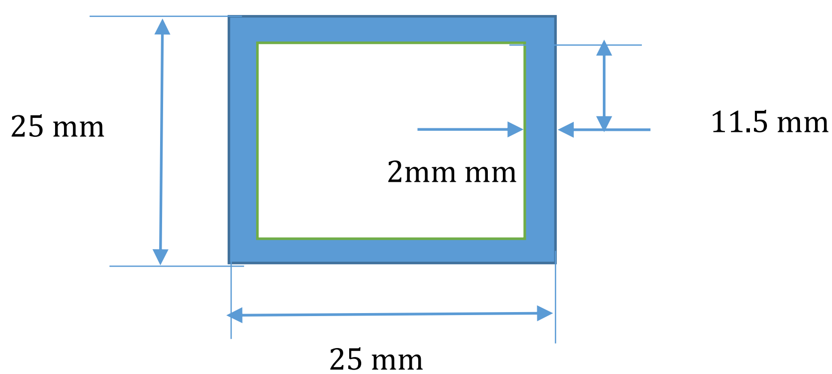


Figure 27 shear stress

$$I_Z = \frac{b * h^3}{12} - \frac{(b-t) * (h-t)^3}{12} \tag{3.8}$$

$$I_Z = \frac{25 * 25^3}{12} - \frac{23 * 23^3}{12} = \frac{25^4}{12} - \frac{23^3}{12} = \frac{390625 - 279841}{12}$$

$$I_Z = 9232\ mm^4$$

$$\bar{y} = 11.5\ mm$$

$$Q = \Delta A * \bar{y} = 2 * 2 * 11.5\ = 46\ mm^3$$

$$q = \frac{VQ}{I} = \frac{147.15\ N * 46\ mm^3}{9232\ mm^4} = 0.73\ N/mm$$

q is the total shear flow acting on the particle with two cutting planes)

$Shear\ Flow\ per\ unit\ length\ (\tau)$

$$\tau = \frac{1}{2}\frac{q}{t} = \frac{1}{2} * \frac{0.73N/mm}{2\ mm} = 0.18\ N/mm^2$$

Determine the shear stress along the z-plane of the edges of the 20*20*1.5 mm rectangular cross section.

given: h = 20 mm, b = 20 mm , t =1.5 mm , $V_d = 250$ N

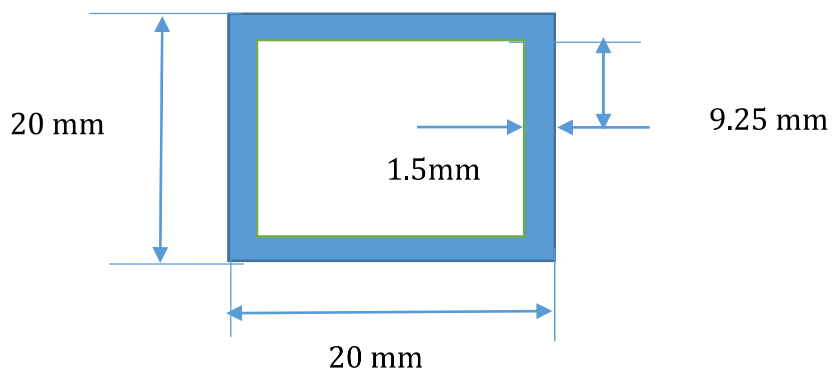


Figure 28 shear stress

$$I_z = \frac{20 * 20^3}{12} - \frac{18.5 * 18.5^3}{12} = \frac{20^4}{12} - \frac{18.5^3}{12} = \frac{160000 - 279841}{12}$$

$I_z = 3572\ mm^4$

$\bar{y} = 11.5\ mm$

$Q = \Delta A * \bar{y} = 2 * 2 * 9.25 = 37\ mm^3$

$$q = \frac{VQ}{I} = \frac{250\ N * 37\ mm^3}{16278.67\ mm^4} = 0.56\ N/mm$$

q is the total shear flow acting on the particle with two cutting planes)

$Shear\ Flow\ per\ unit\ length\ (\tau)$

$$\tau = \frac{1}{2}\frac{q}{t} = \frac{1}{2} * \frac{0.56N/mm}{2\,mm} = 0.14\ N/mm^2$$

3.5.5 Self-Prospered Riding Reapers Machine structure

3.5.5.1 internal and External Loads

A Reapers Harvesting Machine is stressed by loads which typically are determined by the dynamic interaction between the excitation and the Reaper. During the initial stages of design, the data available regarding the Reaper and its subsystems may not be enough to model such dynamic interactions. In this phase an evaluation of the loads is necessary never the less in order to identify an initial design. In a following stage a dynamic condition check is performed generating a more refined design. On basis of these considerations, the objective of the following paragraph is to illustrate some load conditions useful for the initial design of the components of the structural body. Loads in the following conditions will be considered: [Lorenzo Morello, Lorenzo Rosti Rossini, Giuseppe Pia and Andrea Tonoli (2011)].

Parking on flat road

the Reapers Harvesting Machine parked on a flat road, with the maximum payload possible, is not in an extreme load condition. Never the less flat road parking is interesting because the other conditions can be determined by calculating the variations with respect to This case. Given the mass and the position of the center of gravity, the vertical Loads F_{za} and F_{zP} acting on the front and rear axles are simply: machine mass $(m) = 250\ kg, p = 2m, a = 0.35m, b = 0.65m$

$$F_{za} = \text{mg b/p} \tag{3.9}$$

$$F_{za} = 250 \text{ kg} * 9.81 \text{ m/s} * 0.65\text{m} /1.1 = 1450 \text{ N}$$

$$F_{zP} = \text{mg} - F_{za} \tag{3.10}$$

$$F_{zP} = 250\ kg * 9.81\ m/s - 1450 = 1000\ N$$

$$\text{car weighing} = F_{za} + F_{zP} = 145 \text{ N} + 100.25\ N = 2452.5 \text{ KN}$$

Parking with a wheel on a curb

Referring to Fig. 28 it is assumed that the car has a wheel on a curb high enough to cause one of the other three wheels to lift off the ground. If the height of the curb is negligible with respect to the wheelbase, with one wheel lifted, the car is supported in astatically determined way by the ground. Since the wheel contact points are not symmetric with respect to the middle plane (on which the center of gravity is assumed toile) the structure becomes twisted. To understand which wheel leaves the ground, I tis sufficient to consider the maximum torque that can be balanced by the front or rear axle. The sketch of Fig. 29 shows one axle isolated from the rest of the vehicle. Applying a torque Mt determines a vertical load transfer between the two wheels with track:

$$\Delta F_z = \mathrm{Mt/t} \qquad (3.11)$$

Since the wheel-ground contact is mono lateral, the maximum torque which Can be balanced by the axle is that which completely unloads the wheel:

$$M_{tmax} = \frac{F_{zi}}{2} t \qquad (3.12)$$

Where Fzi represents the load on the axle and the subscript i=a, p denotes the Front and rear axle respectively. Higher torque values cannot be balanced and Would cause capsizing. When the car in Fig. 27 rises on to the curb, the front Axle applies a torque to the rear one.

When one of the four wheels leaves the ground (in the figure the front right or the rear left) the torque cannot increase further. Therefore, the maximum torque

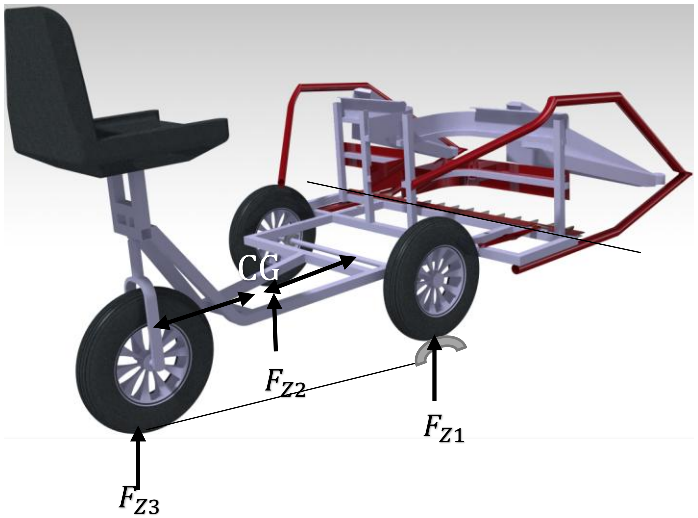


Figure 29 Parking with a wheel on a curb high enough to lift one wheel off the ground

so that the car is supported by three wheels only.

During parking with a wheel on a curb is the minimum between the moments

that can be balanced by the two axles:

$$M_{tmax} = M_{min} \frac{F_{za}}{2} t_a, \frac{F_{xa}}{2} t_p \qquad 3.5$$

$$M_{tmax} = M_{min} \frac{F_{za}}{2} t_a, \frac{F_{xa}}{2} t_p$$

If the torque transmissible by the front axle is lower than the rear axle,

$$\frac{F_{za}}{2} t_a < \frac{F_{xa}}{2} t_p \qquad (3.13)$$

the first wheel to rise is a front one;

vice versa if:

$$\frac{F_{za}}{2} t_a > \frac{F_{xa}}{2} t_p \qquad (3.14)$$

the first wheel to rise is a rear one.

For the sole purpose of evaluating the order of magnitude of the static moment, consider a car weighing FF=2452.5 N, with the center of gravity located at half

of the wheel base a=b=p/2. In this case F_{za} =1450 N, $F_{zp} = 1002.5$ N. If the two front wheel rack are equal t_a=1m, the result is:

$$M_{tmax} = \frac{1450\ N}{2} * 1m = 725\ Nm$$

The considerable magnitude of the torque requires a design intended to avoid the material yielding, the welds pots to break (static or fatigue failure) and, above all, excessive deformations

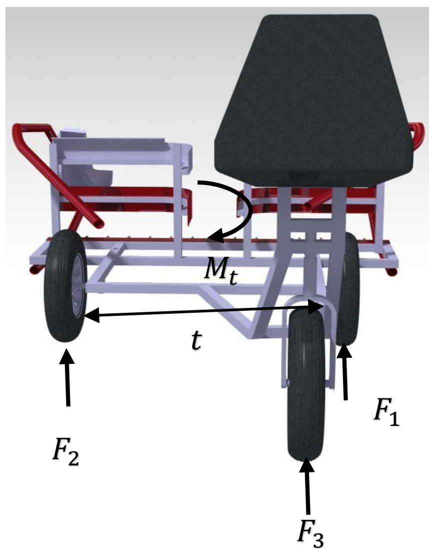


Figure 30 reaper lateral force

Load transfer due to a torque Mt applied to an axle isolated from the reaper lateral force (F_{ymax}) When a car maneuvers a constant radius corner, the maximum lateral force(F_{ymax})on the vehicle is limited by two conditions:

1)reaching the tires maximum lateral adhesion, or

2)the incipient capsizing.

The condition of tires maximum lateral adhesion is given by:

$$F_{ymax}\mu = \mu_{y\ max} F_z \quad (3.14)$$

Where F_z is the total weight and $\mu_{y\,max}$ is the maximum lateral adhesion coefficient that can be produced by the whole vehicle. Its value does not only depend On the tires features and on the road surface characteristics (dry surface, wet, snow), but is also influenced by vehicle dynamics issues, such as the features of the suspension system, its stiffness, the presence of antiroll bars, the center of gravity position, just to list the principal factors. The incipient capsizing condition is reached when the Resultant of the inertia force and the weight force (both applied on the center of gravity) crosses the contact patch of the external wheel. In this case, the lateral load transfer cancels the vertical force on the wheels inside the corner:

$$\Delta F_z = F_{y\,max} R \frac{h}{t} = \frac{F_z}{2} \qquad (3.15)$$

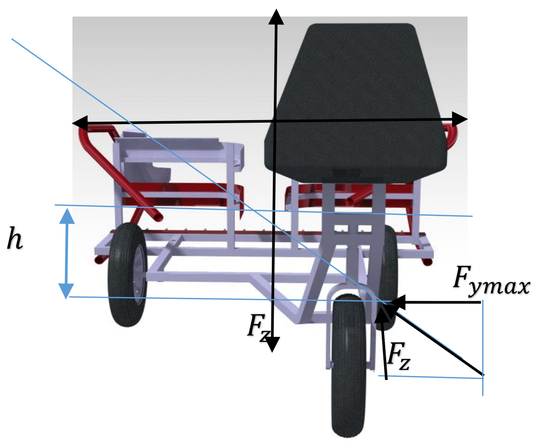


Figure 31 lateral force known grip characteristics

So a maximum lateral grip force $F_{y\,max}R$ is obtained which corresponds to the Incipient capsizing:

$$F_{y\,max} R = \frac{t * F_z}{h * 2} \qquad (3.16)$$

$$F_{y\,max} R = \frac{1 * 2452.5\,N}{0.25 * 2} = 4905$$

Maximum lateral force due to the incipient capsizing for a reaper travelling on a road with known grip characteristics, the maximum lateral force is given by the lowest of the limit loads.

$$F_{y\,max}=\min(\mu_{y\,max}F_z, \frac{1}{2}\frac{t}{h}F_z) \tag{3.17}$$

In the case of a road with good grip conditions $\mu_{ymax} \approx 0.8 \div 0.9$, depending On the ratio 2h/t between the height h of the center of gravity and the semi track (t/2), the limit of the lateral force is given by the tires or by the incipient capsizing. Considering a vehicle with track t =1.2 relatively low center of gravity: h=0.5m, so the ratio t/(2h)=1.2> $\mu y\, max$. The maximum lateral force is then imposed by $\mu_{y\,max}$. (capsizing):car weighing= F_z =2452.5 N For a small commercial vehicle with the same track and a higher center of gravity h=0.8mt/2h=0.75, the maximum lateral force is imposed by the capsizing condition.

$$F_{y\,max} = \frac{1}{2}\frac{t}{h}\,F_z = \frac{1}{2*0.25} * 2452.5\,N = 4905\,N$$

Longitudinal limit

The maximum longitudinal force conditions can be reached during braking or Accelerating to the limit of tires adhesion. For common vehicles the maximum torque produced by the brakes is much higher than the torque given by transmission, so the braking condition features higher longitudinal forces; if Fz is the total weight then:

$$F_{x\,max}\,\mu=\mu_{x\,max}F_z, \tag{3.18}$$

Where$\mu_{x\,max}$ is the maximum longitudinal adhesion coefficient that can be produced by the whole vehicle.

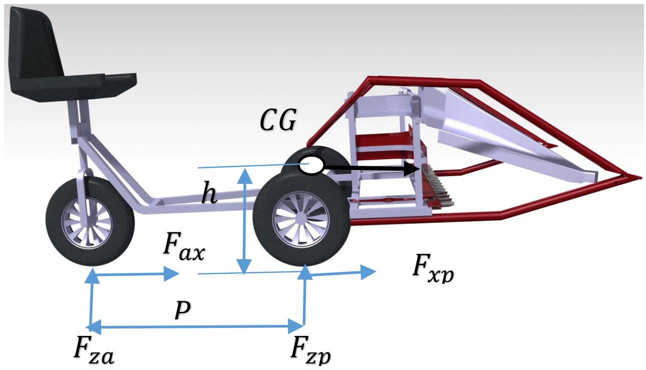


Figure 32 Load transfer between the rear and front axle during braking

Similarly, to the case of limit transversal forces, the incipient capsizing condition can be taken in to account. The load transfer in the braking condition is:

$$\Delta F_z = \frac{h}{p} F_{x\,max} R \qquad (3.19)$$

The incipient capsizing is reached when the load transfer cancels the vertical load On the rear axle (Fig. 32):

$$\Delta F_z = F_{zp} = \frac{a}{h} F_z \qquad (3.20)$$

In this case the whole car is completely weighing on the front axle. Therefore:

$$F_{x\,max} R = \frac{a}{h} F_z \qquad (3.21)$$

$$F_{x\,max} R = \frac{1m}{0.25\;m} * 2452.5\;N$$

$$F_{x\,max} R = 9810\;N$$

The maximum longitudinal force is given by the lowest of the loads defined by equations 3.11 and 3.14:

$$F_{x\,max}\;\mu{=}min\left(\mu_{x\,max}, \frac{a}{h} F_z\right) \qquad (3.22)$$

Since usually a>h the maximum longitudinal force is commonly given by Reaching the limit of tires adhesion.

3.5.6 Engine selection(standard)

Engine, machine for converting energy into motion or mechanical work. the internal combustion engines (briefly written as I. C. engines) are those engines in which the combustion of fuel takes place inside the engine cylinder. The I.C. engines use either petrol or diesel as their fuel. The operating cycle of an I.C. engine may be completed either by the two strokes or four strokes of the piston. Four stroke diesel engine is suitable to this machine according to the propose. Diesel engines are more efficient than gasoline-powered engines and do not require spark plugs. Relatively to the previous harvesting machine select 5 kw single cylinder 4 stroke, air cooled, Diesel engines and the speed of engine shaft is 970 rpm (102.4 rad/s) an engine delivers 5 kilowatts; express this power in horsepower. *D. N Sharma; S. Mukesh (2008 and 2010)*]

P = 5 kw x 1.36 = 6.8 hp.

calculate the engine torque

$$T = P/\omega \qquad (3.23)$$

P: engine power, expressed in n W

T: engine torque, expressed in Nm

ω: angular velocity, expressed in rad/s

T = 5000 w /102.4 rad/s = <u>48.8 Nm</u>

Figure 33 Engine [D. N Sharma; S.mukesh (2008 and 2010)]

3.5.7 design and select Belt

The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds. The amount of power transmitted depends upon the following factors:

1. The velocity of the belt.
2. The tension under which the belt is placed on the pulleys.
3. The arc of contact between the belt and the smaller pulley.
4. The conditions under which the belt is used.

3.5.7.1 Selection of a Belt Drive

Following are the various important factors upon which the selection of a belt drive depends: [16].

1. Speed of the driving and driven shafts,
2. Speed reduction ratio,
3. Power to be transmitted,
4. Centre distance between the shafts,
5. Positive drive requirements,
6. Shafts layout,
7. Space available, and
8. Service conditions

3.5.7.2 Types of Belts

Though there are many types of belts used these days, yet the following are important from the subject point of view [*D. N Sharma; S. Mukesh (2008 and 2010)*].

1. Flat belt. is mostly used in the factories and workshops, where a moderate amount of power is to be transmitted, from one pulley to another when the two pulleys are not more than 8 meters apart.
2. V-belt. is mostly used in the factories and workshops, where a great amount of power is to be transmitted, from one pulley to another, when the two pulleys are very near to each other.

3. Circular belt or rope. is mostly used in the factories and workshops, where a great amount of power is to be transmitted, from one pulley to another, when the two pulleys are more than 8 meters apart.

3.5.7.2.1 Types of Belt Drives

The belt drives are usually classified into the following three groups: [*D. N Sharma; S. Mukesh (2008 and 2010)*].

1. Light drives. These are used to transmit small powers at belt speeds up to about 10 m/s as in agricultural machines and small machine tools.
2. Medium drives. These are used to transmit medium powers at belt speeds over 10 m/s but up to 22 m/s, as in machine tools.
3. Heavy drives. These are used to transmit large powers at belt speeds above 22 m/s as in compressors and generators.

3.5.7.3 Material used for flat and v-belts

The material used for belts and ropes must be strong, flexible, and durable. It must have a high coefficient of friction. The belts, according to the material used, are classified as follows:

1. Leather belts.
2. Cotton or fabric belts.
3. Rubber belt.
4. Balata belts.

select rubber belt because simple to get and not expensive. The rubber belts are made of layers of fabric impregnated with rubber composition and have a thin layer of rubber on the faces. These belts are very flexible but are quickly destroyed if allowed to come into contact with heat, oil or grease. One of the principle advantages of these belts is that they may be easily made endless. These belts are found suitable for saw mills, paper mills where they are exposed to moisture. Mass density in kg / m3 is 1140 Belt Speed. A little consideration will show that when the speed of belt increases, the centrifugal force also increases which tries to pull the belt away from the pulley. This will result in the decrease of power transmitted by the belt. It has been found that for the efficient transmission of power, the belt speed 20 m/s.

3.5.7.4 Standard Belt Thicknesses and Widths

The standard flat belt thicknesses are 5, 6.5, 8, 10 and 12 mm. The preferred values of thicknesses are as follows:

1. 5 mm for nominal belt widths of 35 to 63 mm
2. 6.5 mm for nominal belt widths of 50 to 140 mm,
3. 8 mm for nominal belt widths of 90 to 224 mm,
4. 10 mm for nominal belt widths of 125 to 400 mm, and (e) 12 mm for nominal belt widths of 250 to 600 mm.

The standard values of nominal belt widths are in R10 series, starting from 25 mm up to 63 mm and in R 20 series starting from 71 mm up to 600 mm. Thus, the standard widths will be 25, 32, 40,50, 63, 71, 80, 90, 100, 112, 125, 140, 160, 180, 200, 224, 250, 280, 315, 355, 400, 450, 500, 560 and 600 mm. [R.S. KHURMI and J.K. GUPTA (2005)].

3.5.7.5 Types of Flat Belt Drives for this machine

The power from one pulley to another may be transmitted by any of the following types of belt drives.

1. Open belt drive: - is used with shafts arranged parallel and rotating in the same direction.
2. Crossed or twist belt drive: - is used with shafts arranged parallel and rotating in the opposite directions.
3. Quarter turn belt drive (also known as right angle belt drive): - is used with shafts arranged at right angles and rotating in one definite direction. In order to prevent the belt from leaving the pulley, the width of the face of the pulley should be greater or equal to 1.4 b, where b is width of belt.
4. Belt drive with idler pulleys (also known as jockey pulley drive): - is used with shafts arranged parallel and when an open belt drive cannot be used due to small angle of contact on the smaller pulley.
5. Compound belt drive: - is used when power is transmitted from one shaft to another through a number of pulleys.
6. Stepped or cone pulley drive. is used for changing the speed of the driven shaft while the main or driving shaft runs at constant speed.

7. Fast and loose pulley drive. is used when the driven or machine shaft is to be started or stopped whenever desired without interfering with the driving shaft.

Select v-belt and Open belt drive V-belts are the most common means of driving light loads between short range pulleys. V-belts are small in cross-sectional to reduce friction and heating. They have a greater load capacity at high speeds than do flat belts.

3.5.7.6 Calculate length and thickness Frist stage belt

- calculate belt length

; - assume the distance between driver pulley and driven pulley, 400mm according to frame structure.

Center - to - center distance (C) between driven pulleys is given by:

$C = 60/2 + 400 + 200/2 = 530$ mm

The length of belt is given by

$L = 2C + \pi (D + d)/2 + (D - d)^2/4C$

$L = 2 \times 530 + \pi (200 + 60)/2 + (200 - 60)^2/4C$

$L = 1468.4$ mm

- calculate belt tension and width

Design a rubber v-belt to drive a gasoil engine 5 kW at 970 r.p.m. and fitted with a pulley 60 mm diameter.

Allowable stress for belt = 2.1 MPa

Density of rubber = 1000 kg / m3

Angle of contact driver pulley = 165°

Coefficient of friction between belt and pulley = 0.3

Given: $P = 5$ kW $= 5 \times 10^3$

W; N= 970 r.p.m.; d= 60 mm = 0.06 m; η, d = 85% = 0.85; $\sigma = 2.1$ MPa $= 2.1 \times 10^6$ N/m2

ρ= 1000 kg/m^3; θ= 165° = 165 × π/180= 2.88 rad; μ= 0.3

T_1= Tension in the tight side of the belt, and

T_2= Tension in the slack side of the belt.

We know that velocity of the belt,

$$V = 1 + \frac{\pi.d.N}{60} = 1 + \frac{\pi X0.06X970}{60} = 4.05\frac{M}{S}$$

And power transmitted (P)

$$10 \times 10^3 = (T_1 - T_2)\,v$$

$$10 \times 10^3 = (T_1 - T_2)X4.05$$

$$10 \times 10^3 = 4.05\,(T_1 - T_2)$$

$$\therefore\ T_1 - T_2 = 5\,x10^3/4.05 = 1234.57\text{ N} \qquad (3.24)$$

We know that

$$2.3\log\frac{T_1}{T_2} = \mu.\theta = 0.3\text{ x }2.88 = 0.864$$

$$\log\frac{T_1}{T_2} = \frac{0.864}{2.3} = 0.3756$$

$$\frac{T_1}{T_2} = 2.375 \qquad (3.25)$$

... (Taking antilog of 0.3756)

From equations (3.24) and (3.25), we find that

T_1= 2132.435 N; and T_2= 897.87 N

Let b= Width of the belt in meters, and

t = Thickness of the belt in meters.

Assuming thickness of the belt, t= 10 mm = 0.01 m, we have

Cross-sectional area of the belt= $b \times t = b \times 0.01 = 0.01$ bm2

We know that mass of the belt per meter length,

m= Area x length x density = $= 0.01$ b x 1 x 1000 $= 10$ bkg / m

∴Centrifugal tension,

$T_C = m.v^2 = 10\ b(3.05)^2 = 283.726\ bN$

We know that maximum tension in the belt,

$T = \sigma.b.t = 2.1 \times 10^6 \times b \times 0.01 = 21\ 000\ bN$

And tension in the tight side of belt (T_2),

897.87 N = T– $T_C = 21\ 000\ b - 283.726\ b = 20716.274b$∴

b= 897.87 N / 20716.274 = 0.0435 m = 43.5 mm

The standard width of the belt (b) is 50 mm. Ans.

3.5.7.7 Calculate length and thickness 3rd stage belt

- calculate belt length

; - assume the distance between driver pulley and driven pulley, 200 mm according to frame structure.

Center - to - center distance (C) between driven pulleys is given by:

C = 60/2 + 200 + 70/2 =270 mm

The length of belt is given by

$L = 2C + \pi\ (D + d)/2 + (D - d)^2/4C$

$L = 2 \times 270 + \pi\ (60 + 70)/2 + (60 + 70)^2/4C$

L = 540 + 204.2 + 15.64

L = 759.5~760 mm

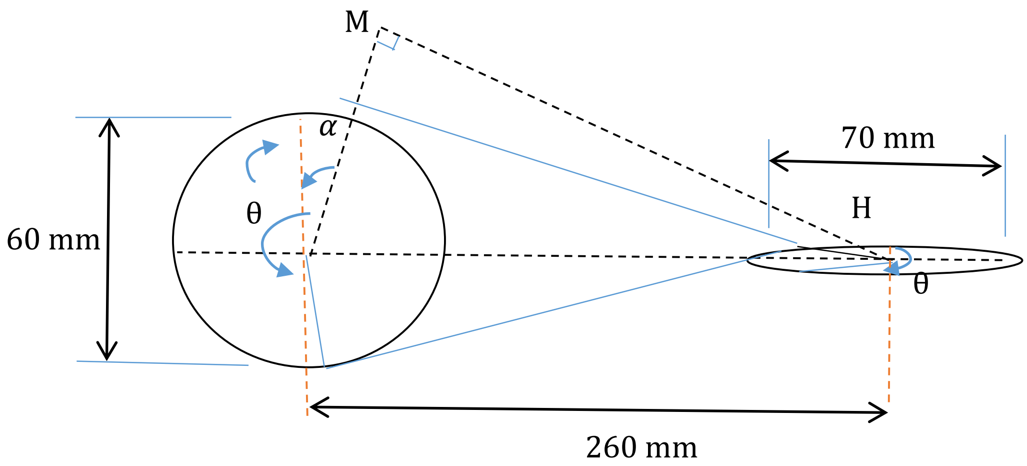


Figure 34 Angle of contact between the belt and each pulley

- Angle of contact between the belt and each pulley

Let θ =Angle of contact between the belt and each pulley.

We know that for a crossed belt drive,

$$\sin\alpha = \frac{r_1 + r_2}{c} = \frac{30 + 35}{240} = 0.24$$

$$\alpha = 8.83^0$$

$$\theta = 180^o + 2\alpha = 180^o + 2(8.83^0) = 188.83^0$$

$$\theta = 188.83\frac{\pi}{180} = 3.3\,rad$$

- calculate belt tension and width

Design a rubber v-belt to drive a gasoil engine 5 kW at 1/3 x 970 r.p.m. and fitted with a pulley 60 mm diameter.

Allowable stress for belt = 2.1 MPa

Density of rubber = 1000 kg / m3

pulley angle of contact = 188.83^0

Coefficient of friction between belt and pulley = 0.3

Given: P= 5 kW = 5×10^3

W; N= 323.33 r.p.m.; d= 60 mm = 0.06 m; η= 85% = 0.85; σ= 2.1 MPa = 2.1×10^6N/m2

ρ= 1000 kg/m^3; θ= 188.83^0= 188.83 × π/180= 3.3 rad; μ= 0.3

T_1= Tension in the tight side of the belt, and

T_2= Tension in the slack side of the belt.

We know that velocity of the belt,

$$V = 1 + \frac{\pi.\mathrm{d.N}}{60} = 1 + \frac{\pi X0.06X1/3 \ x970}{60} = 1.35\frac{M}{S}$$

And power transmitted (P)

$$10 \times 10^3 = (T_1 - T_2)\ v$$

$$= (T_1 - T_2)X1.35$$

$$= 1.35\ (T_1 - T_2)$$

$$\therefore\ T_1 - T_2 = 5\ x10^3/1.35 = 3707.7\ N \qquad (3.26)$$

We know that

$$2.3 \log\frac{T_1}{T_2} = \mu.\theta = 0.3\ x\ 3.3 = 0.9$$

$$\log\frac{T_1}{T_2} = \frac{0.9}{2.3} = 0.39$$

$$\frac{T_1}{T_2} = 2.375 \qquad (3.27)$$

... (Taking antilog of 0.39)

From equations (3.26) and (3.27), we find that

T_1= 13579.1 N; and T_2= 9871.4 N

Let b= Width of the belt in meters, and

t = Thickness of the belt in meters.

Assuming thickness of the belt, t= 10 mm = 0.01 m, we have

Cross-sectional area of the belt= $b \times t = b \times 0.01 = 0.01\ bm^2$

We know that mass of the belt per meter length,

m= Area x length x density = $= 0.01\ b \times 1 \times 1000 = 10\ bkg / m$

∴Centrifugal tension,

$T_C = m.v^2 = 10\ b(1.35)^2 = 18.225\ bN$

We know that maximum tension in the belt,

$T = \sigma.b.t = 2.1 \times 10^6 \times b \times 0.01 = 21\ 000\ bN$

And tension in the tight side of belt (T_2),

9871.4 N = $T - T_C = 21\ 000\ b - 18.225\ b = 20716.274b$∴

b= 9871.4 N / 20981.775 = 0.047 m = 47mm

The standard width of the belt (b) is 50 mm. Ans.

3.5.7.8 Calculate length and thickness 4th stage belt

- calculate belt length
  ; - assume the distance between driver pulley and driven pulley, 600mm according to frame structure.
  Center - to - center distance (C) between driven pulleys is given by:

  C = 60/2 + 600 + 60/2 =660 mm

  The length of belt is given by

  $L = 2C + \pi (D + d)/2 + (D - d)^2/4C$

  $L = 2 X 660 + \pi (60 + 60)/2 + (60 + 60)^2/4C$

  L = 1122 mm

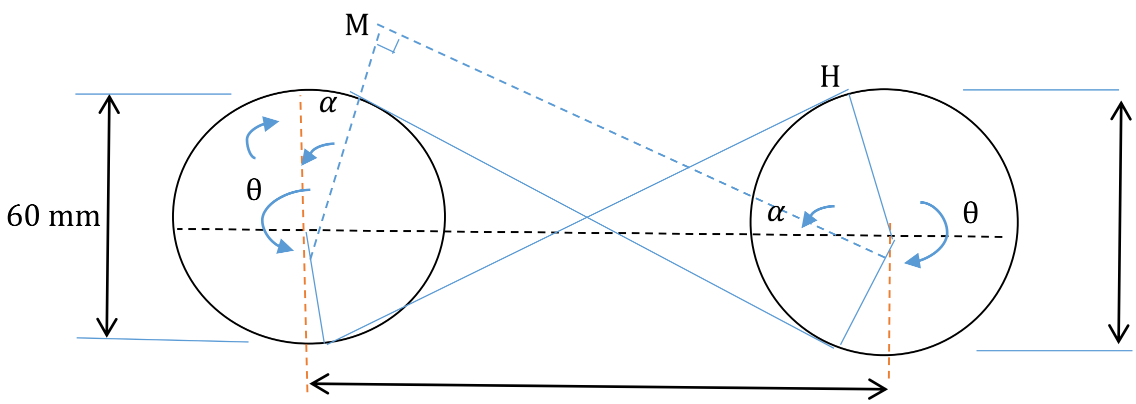


Figure 35 Angle of contact between the belt and each pulley

- Angle of contact between the belt and each pulley

Let θ =Angle of contact between the belt and each pulley.

We know that for a crossed belt drive,

$$\sin\alpha = \frac{r_1 + r_2}{c} = \frac{30 + 30}{660} = 0.091$$

$$\alpha = 5.216^0$$

$$\theta = 180^o + 2\alpha = 180^o + 2(5.216^0) = 185.216^0$$

$$\theta = 193.3\frac{\pi}{180} = 3.23\, rad$$

- calculate belt tension and width

Design a rubber v-belt to drive a gasoil engine 5 kW at 1/3 x 6/7 x970 r.p.m. and fitted with a pulley 60 mm diameter.

Allowable stress for belt = 2.1 MPa

Density of rubber = 1000 kg / m3

pulley angle of contact = 193.3^0

Coefficient of friction between belt and pulley = 0.3

Given: P= 5 kW = 5 × 10^3

W; N= 277.14 r.p.m.; d= 60 mm = 0.06 m; η= 85% = 0.85; σ= 2.1 MPa = 2.1 × 10^6N/m2

ρ= 1000 kg/m^3; θ= 185.216^0= 185.216× π/180= 3.23 rad; μ= 0.3

T_1= Tension in the tight side of the belt, and

T_2= Tension in the slack side of the belt.

We know that velocity of the belt,

$$\mathrm{V} = 1 + \frac{\pi.\mathrm{d.N}}{60} = 1 + \frac{\pi \mathrm{X} 0.06 \mathrm{X} 1/3\ \mathrm{x} 6/7 \mathrm{x} 970}{60} = 1.157\,\frac{\mathrm{M}}{\mathrm{S}}$$

And power transmitted (P)

$$10 \times 10^3 = (\mathrm{T}_1 - \mathrm{T}_2)\ \mathrm{v}$$

$$= (\mathrm{T}_1 - \mathrm{T}_2)\mathrm{X}1.157$$

$$= 1.35\ (\mathrm{T}_1 - \mathrm{T}_2)$$

$$\therefore\ \mathrm{T}_1 - \mathrm{T}_2 = 5\,\mathrm{x}10^3/1.157 = 4321\ \mathrm{N} \qquad (3.28)$$

We know that

$$2.3 \log\frac{\mathrm{T}_1}{\mathrm{T}_2} = \mu.\theta = 0.3\ \mathrm{x}\ 3.23 = 0.97$$

$$\log\frac{\mathrm{T}_1}{\mathrm{T}_2} = \frac{1}{2.3} = 0.39$$

$$\frac{\mathrm{T}_1}{\mathrm{T}_2} = 2.375 \qquad (3.29)$$

... (Taking antilog of 0.39)

From equations (3.28) and (3.29), we find that

T_1= 7463.54N; and T_2= 3142.54 N

Let b= Width of the belt in meters, and

t = Thickness of the belt in meters.

Assuming thickness of the belt, t= 10 mm = 0.01 m, we have

Cross-sectional area of the belt= b x t = b x 0.01 = 0.01 bm^2

We know that mass of the belt per meter length,

m= Area x length x density = = 0.01 b x 1 x 1000 = 10 bkg / m

∴Centrifugal tension,

$T_C = m.v^2 = 10\,b(1.157)^2 = 13.38\text{ bN}$

We know that maximum tension in the belt,

$T = \sigma.b.t = 2.1 \times 10^6 \times b \times 0.01 = 21\,000\text{ bN}$

And tension in the tight side of belt (T_2),

9871.4 N = T− T_C = 21 000 b− 13.38 b = 20986.62b∴

b= 7463.54N / 20986.62b = 0.0355 m = 35.5 mm

The standard width of the belt (b) is 40 mm. Ans.

3.5.8 design and select Pulley

The pulleys are used to transmit power from one shaft to another by means of flat belts, V-belts or ropes. Since the velocity ratio is the inverse ratio of the diameters of driving and driven pulleys, therefore the pulley diameters should be carefully selected in order to have a desired velocity ratio. The pulleys must be in perfect alignment in order to allow the belt to travel in a line normal to the pulley faces. The pulleys may be made of cast iron, cast steel or pressed steel, wood and paper. The cast materials should have good friction and wear characteristics. The pulleys made of pressed steel are lighter than cast pulleys, but in many cases they have lower friction and may produce excessive wear. [*D. N Sharma; S. Mukesh (2008 and 2010)*].

3.5.8.1 types of pulleys for v-belt and flat-belt
Following are the various types of pulleys for flat belts:

1. Cast iron pulleys
2. Steel pulleys
3. Wooden pulleys
4. Paper pulleys, and
5. Fast and loose pulleys.

3.5.8.1.1 Cast iron pulleys
The pulleys are generally made of *cast iron, because of their low cost. The rim is held in place by web from the central boss or by arms or spokes.

3.5.8.1.2 Steel Pulleys
Steel pulleys are made from pressed steel sheets and have great strength and durability. These pulleys are lighter in weight (about 40 to 60% less) than cast iron pulleys of the same capacity and are designed to run at high speeds. They present a coefficient of friction with leather belting which is at least equal to that obtained by cast iron pulleys

3.5.8.1.2.1 Wooden pulleys
Wooden pulleys are lighter and possess higher coefficient of friction than cast iron or steel pulleys. These pulleys have 2/3rd of the weight of cast iron pulleys of similar size. They are generally made from selected maple which is laid in segments and glued together under heavy pressure

3.5.8.1.3 Paper pulleys
Paper pulleys are made from compressed paper fiber and are formed with a metal in the center.

These pulleys are usually used for belt transmission from electric motors, when the center to center shaft distance is small.

3.5.8.2 FAST AND LOOSE PULLEYS
A fast and loose pulley, used on shafts enables machine to be started or stopped at will. A fast pulley is keyed to the machine shaft while the loose pulley runs freely. The belt runs over the fast pulley to transmit power by the machine and it is shifted to the loose pulley

when the machine is not required to transmit power. By this way, stopping of one machine does not interfere with the other machines which run by the same line shaft.

According to the strength, cost of pulley, and also the availability of pulley in local market we select cast iron.

3.5.8.3 Calculate of Cast Iron Pulleys

The following procedure may be adopted for the design of cast iron pulleys.

Dimensions of pulley

The diameter of the pulley (D) may be obtained either from velocity ratio consideration or centrifugal stress consideration. We know that the centrifugal stress induced in the rim of the pulley, $\sigma_t = \rho r v^2$

Where ρ = Density of the rim material= 7200 kg/m3 for cast iron, V = Velocity of the rim = πDN/ 60, D being the diameter of pulley and Nis speed of the pulley.

The following are the diameter of pulleys in mm for flat and V-belts.

20, 22, 25, 28, 32, 36, 40, 45, 50, 56, 63, 71, 80, 90, 100, 112, 125, 140, 160, 180, 200, 224, 250, 280, 315, 355, 400, 450, 500, 560, 630, 710, 800, 900, 1000, 1120, 1250, 1400, 1600, 1800, 2000, 2240, 2500, 2800, 3150, 3550, 4000, 5000, 5400. The first six sizes (20 to 36 mm) are used for V-belts only. [*D. N Sharma; S. Mukesh (2008 and 2010)*].

If the width of the belt is known, then width of the pulley or face of the pulley (B) is taken 25% greater than the width of belt.

∴ B = 1.25 b; where b= Width of belt.

According to Indian Standards, IS: 2122 (Part I) – 1973 (Reaffirmed 1990), the width of pulley is fixed as given in the Standard width of pulley table III in appendix.

3.5.8.3.1 Frist stage drive pulley

A cast iron pulley transmits 5 kW at 1:1 of 970 r.p.m. The diameter of pulley is 60 mm and has smooth straight arms of elliptical straight arms of cross-section in which the major axis is twice the minor axis. Find the dimensions of the arm if the allowable bending stress is 15 MPa. Mention the plane in which the major axis of the arm should lie.

Let b_1 = Minor axis, and

b_1 = Major axis = $2b_1$

We know that the torque transmitted by the pulley,

$$T = \frac{PX60}{2X\pi XN} = \frac{5x10^3X60}{2X\pi X970} = \frac{30x10^4}{6091.6} = 49.25\ N.M$$

Maximum bending moment at the hub end,

$$M = \frac{2T}{n} = \frac{2x49.25\ N.M}{1} = 98.5\ N.M$$

And section modulus,

$$Z = \frac{\pi}{32} Xb_1X(a_1)^2 = \frac{\pi}{32} Xb_1X(2b_1)^2 = \frac{\pi(b_1)^3}{8} \tag{3.30}$$

We know that the bending stress σ_b

$$15 = \frac{M}{Z} = \frac{98.5X8X10^3}{\pi(b_1)^3} = \frac{788X10^3}{\pi(b_1)^3}$$

$$(b_1)^3 = \frac{788X10^3}{15X\pi} = 25.57\ mm$$

$a_{1=}2b_1 = 2x25.57\ mm = 51.14\ mm$

3.5.8.3.2 Frist stage driven pulley

A cast iron pulley transmits 5 kW at 1/3 of 970 r.p.m. The diameter of pulley is 200 mm and has full arms of elliptical straight arms of cross-section in which the major axis is twice the minor axis. Find the dimensions of the arm if the allowable bending stress is 15 MPa. Mention the plane in which the major axis of the arm should lie.

$$\sigma_b = 15\text{mpa} = 15\text{N/mm}^2$$

Let b_1= Minor axis, and

b_1= Major axis = $2b_1$

We know that the torque transmitted by the pulley,

$$T = \frac{PX200}{2X\pi XN} = \frac{5x10^3X200}{2X\pi X323.33} = \frac{10x10^5}{2031.35} = 492.3\ \text{N.M}$$

Maximum bending moment per arm at the hub end,

$$M = \frac{2T}{n} = \frac{492.3\ \text{N.M}}{1} = 984.6\ \text{N.M}$$

And section modulus,

$$Z = \frac{\pi}{32}Xb_1X(a_1)^2 = \frac{\pi}{32}Xb_1X(2b_1)^2 = \frac{\pi(b_1)^3}{8}$$

We know that the bending stress σ_b

$$15 = \frac{M}{Z} = \frac{984.6\ \text{N.MX}8X10^3}{\pi(b_1)^3} = \frac{7876.8X10^3}{\pi(b_1)^3}$$

$$(b_1)^3 = \frac{7876.8X10^3}{15X\pi} = 55\ \text{mm}$$

$$a_1 = 2b_1 = 2x55\ \text{mm} = 110\ \text{mm}$$

3.5.8.3.3 Second stage drive pulley (double sprocket pinion)

A steel sprocket pinion transmits 5 kW at 1/3 of 970 r.p.m. The diameter of sprocket pinion is 60 mm and has full arms of elliptical straight arms of cross-section in which the major axis is twice the minor axis. Find the dimensions of the arm if the allowable bending stress is 35 MPa. Mention the plane in which the major axis of the arm should lie.

$$\sigma_b = 35\text{mpa} = 35\text{N/mm}^2$$

Let b_1= Minor axis, and

b_1= Major axis = $2b_1$

We know that the torque transmitted by the pulley,

$$T = \frac{PX60}{2X\pi XN} = \frac{5x10^3X60}{2X\pi X323.33} = \frac{30x10^4}{2031.35} = 147.69\ N.M$$

Maximum bending moment at the hub end,

$$M = \frac{2T}{n} = \frac{147.69\ N.M}{1} = 295.37\ N.M$$

And section modulus,

$$Z = \frac{\pi}{32}Xb_1X(a_1)^2 = \frac{\pi}{32}Xb_1X(2b_1)^2 = \frac{\pi(b_1)^3}{8} \tag{3.31}$$

We know that the bending stress σ_b

$$25 = \frac{M}{Z} = \frac{295.37\ N.M\ X8X10^3}{\pi(b_1)^3} = \frac{2362.96X10^3}{\pi(b_1)^3}$$

$$(b_1)^3 = \frac{2362.96X10^3}{35X\pi} = 27.8\ mm$$

$a_{1=}2b_1 = 2x27.8\ mm = 55.6mm$

3.5.8.3.4 Second stage driven pulley (sprocket gear)

A steel sprocket pinion transmits 5 kW at1/3 x 1/3 of 970 r.p.m. The diameter of sprocket pinion is 120 mm and has three arms of elliptical straight arms of cross-section in which the major axis is twice the minor axis. Find the dimensions of the arm if the allowable bending stress is 15 MPa. Mention the plane in which the major axis of the arm should lie.

Let b_1= Minor axis, and

b_1= Major axis = $2b_1$

We know that the torque transmitted by the pulley,

$$T = \frac{PX120}{2X\pi XN} = \frac{5x10^3X120}{2X\pi X107.78} = \frac{12x10^5}{677.2} = 886\ N.M$$

Maximum bending moment at the hub end,

$$M = \frac{2T}{3} = \frac{2 * 886\ \mathrm{N.M}}{3} = 590.67\ \mathrm{N.M}$$

And section modulus,

$$Z = \frac{\pi}{32} X b_1 X (a_1)^2 = \frac{\pi}{32} X b_1 X (2b_1)^2 = \frac{\pi (b_1)^3}{8} \qquad (3.32)$$

We know that the bending stress σ_b

$$15 = \frac{M}{Z} = \frac{590.67\ \mathrm{N.M}\ X8X10^3}{\pi (b_1)^3} = \frac{47725.33X10^3}{\pi (b_1)^3}$$

$$(b_1)^3 = \frac{47725.33X10^3}{15X\pi} = 46.46\mathrm{mm}$$

$$a_{1=} 2b_1 = 2x46.46\ \mathrm{mm} = 92.9\ \mathrm{mm}$$

3.5.8.3.5 Second stage driven pulley (sprocket gear)

A steel sprocket pinion transmits 5 kW at1/3 x 6/7 of 970 r.p.m. The diameter of sprocket pinion is 70 mm and has full arms of elliptical straight arms of cross-section in which the major axis is twice the minor axis. Find the dimensions of the arm if the allowable bending stress is 15 MPa. Mention the plane in which the major axis of the arm should lie.

Let b_1= Minor axis, and

b_1= Major axis = $2b_1$

We know that the torque transmitted by the pulley,

$$T = \frac{PX70}{2X\pi XN} = \frac{5x10^3 X70}{2X\pi X277.14} = \frac{35x10^4}{1741.34} = 201\ \mathrm{N.M}$$

Maximum bending moment at the hub end,

$$M = \frac{2T}{1} = \frac{2 * 201\ \mathrm{N.M}}{1} = 402\mathrm{N.M}$$

And section modulus,

$$Z = \frac{\pi}{32} X b_1 X (a_1)^2 = \frac{\pi}{32} X b_1 X (2b_1)^2 = \frac{\pi (b_1)^3}{8} \quad (3.33)$$

We know that the bending stress σ_b

$$15 = \frac{M}{Z} = \frac{402 \text{ N.M } X8X10^3}{\pi (b_1)^3} = \frac{3215.9X10^3}{\pi (b_1)^3}$$

$$(b_1)^3 = \frac{3215.9X10^3}{15X\pi} = 29.47\text{mm}$$

$$a_{1=}2b_1 = 2x29.47 \text{ mm} = 58.9 \text{ mm}$$

3.5.8.3.6 Third stage drive pulley

A cast iron pulley transmits 5 kW at 1/3 x 1/2 of 970 r.p.m. The diameter of pulley is 60 mm and has smooth straight arms of elliptical straight arms of cross-section in which the major axis is twice the minor axis. Find the dimensions of the arm if the allowable bending stress is 15 MPa. Mention the plane in which the major axis of the arm should lie.

Let b_1= Minor axis, and

b_1= Major axis = $2b_1$

We know that the torque transmitted by the pulley,

$$T = \frac{PX60}{2X\pi XN} = \frac{5x10^3X60}{2X\pi X161.67} = \frac{30x10^4}{1815.67} = 295.34 \text{ N.M}$$

Maximum bending moment at the hub end,

$$M = \frac{2T}{n} = \frac{2x295.34\text{N.M}}{1} = 590.678 \text{ N.M}$$

And section modulus,

$$Z = \frac{\pi}{32} X b_1 X (a_1)^2 = \frac{\pi}{32} X b_1 X (2b_1)^2 = \frac{\pi (b_1)^3}{8} \quad (3.34)$$

We know that the bending stress σ_b

$$15 = \frac{M}{Z} = \frac{590.678X8X10^3}{\pi(b_1)^3} = \frac{47.25X10^5}{\pi(b_1)^3}$$

$$(b_1)^3 = \frac{47.25X10^5}{15X\pi} = 46.45 \text{ mm}$$

$$a_{1=}2b_1 = 2x46.45 \text{ mm} = 92.9 \text{ mm}$$

3.5.9 design and select Chain

The chains are made up of number of rigid links which are hinged together by pin joints in order to provide the necessary flexibility for wrapping round the driving and driven wheels. These wheels have projecting teeth of special profile and fit into the corresponding recesses in the links of the chain as shown in Fig. in below. The toothed wheels are known as *sprocket wheels or simply sprockets. The sprockets and the chain are thus constrained to move together without slipping and ensures perfect velocity ratio. The reason of using chain is because of In order to avoid slipping. [*D. N Sharma; S. Mukesh (2008 and 2010)*].

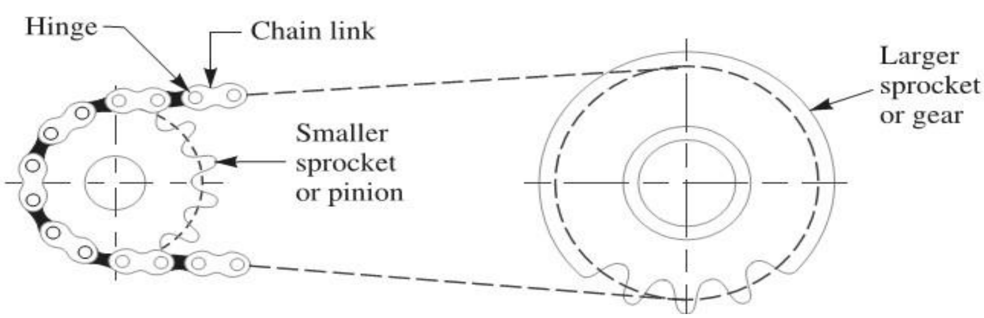


Figure 36 steel chains and sprocket wheels [D. N Sharma; S. Mukesh (2008 and 2010)].

The chains are mostly used to transmit motion and power from one shaft to another, when the center distance between their shafts is short such as in bicycles, motor cycles, agricultural machinery, conveyors, rolling mills, road rollers etc. The chains may also be used for long center distance of upto8 meters. The chains are used for velocities up to 25 m / s and for power up to 110 kW. In some cases, higher power transmission is also possible. [16]

3.5.9.1.1 Second stage chain

Harvesting machine operated by 5 kW diesel engines and the speed of engine shaft is 970 rpm and that of rotary is to be 1/3 x 970 rpm. The maximum distance between driver sprocket and driven sprocket gear 450 mm. calculate a suitable chain drive for the rotary. Since the speed of diesel engine in the present case is not very high, so we may take lower chain velocity and a roller chain is selected for the drive. Determine velocity ratio (VR): the velocity ratio of drive can be determined by following equation [*D. N Sharma; S. Mukesh (2008 and 2010)*].

$$V_{\mathrm{R}} = \frac{\mathrm{RPM_e}}{\mathrm{RPM_r}} = \frac{970}{\frac{1}{3}*970} = \frac{323.33}{277.14} = 3$$

Now, from the chain table which is available in the design handbook for a velocity ratio of 3. The number of teeth in the pinion is equal to 20.

Number = 20

Nr = Ne x V.R = 20 X 3 =60 teeth

<u>Calculate of chain pitch</u>

$$P = \frac{c}{30 \text{ to } 60} = \frac{450}{30 \text{ to } 60} = 15 \text{ from the to } 7.5 \text{ mm}$$

From the table in design data book we may select standard chain pitch = 12.7 mm

Chain velocity

$$V = \frac{P}{1000} X \frac{RPM}{60} XNe \frac{m}{s} \tag{3.35}$$

$$V = \frac{12.7}{1000} X \frac{323.33}{60} X20 \frac{m}{s} = 1.3 \text{ m/s}$$

3.5.9.1.2 Calculate of load on chain

$F_{chian} = F + F_C + F_f$ $\quad$ F_f = Frictional force chain

$F = \frac{P(KW)X1000}{V}$ F_C = centrifugal load on chain

F = 5 kW X $\frac{1000}{1.3}$ = 3652.9 KN F = load due to power transmission

$F_c = \frac{W \times V^2}{g}$, where; W = 13.6 N/m for duplex chain from design data book

$$F_C = \frac{13.6 \times 1.3^2}{9.81} = 2.34 \text{ N}$$

$F_f = K_f * W * C$, where; (3.36)

c = center to center distance (450 mm)

K_f = kinematics friction

For inclined drive (K_f) =2.0

For horizontal drive (K_f) = 4.0

For vertical drive (K_f) = 1.0

K_f= 2 X 13.6 X0.45 m = 12.24 N

F_C = 3652.9 N+ 2.34 N+ 12.24 N = 3667.48 N

New check the loading strength of chain from the chain table which is available in design hand book for duplex chain minimum breaking load = 431 KN

$$F_{os} = \frac{Fu}{c} = \frac{\text{breaking load}}{\text{chain load}} = \frac{431X10^3 \text{ N}}{3667.48 \text{ N}} = 117.52$$

When it is greater than 117.52 >11.00 the chain selection for rotary is safe.

Chain length = L_C = m x p, when p =chain pitch and m = number of chain link

$$M = \frac{2c}{p} + \frac{Nr - Ne}{p} + \frac{p(Nr - Ne)^2}{4\pi^2 c} \quad (3.37)$$

$$M = \frac{2 \text{ x } 450}{12.7} + \frac{20 + 60}{2} + \frac{12.7(60 - 20)^2}{4\pi^2(450)}$$

$$\text{M} = 70.866 + 40 + 1.133 = 135.63 \text{ pitches}$$

L_C = m x p = 135.63x 12.7 =1200.15mm

3.5.9.2 Center to center distance between sprockets

$$c = \frac{p}{4}(m - \frac{Ne+Nr}{2} + [\{m - \frac{(Ne+Nr)^2}{4\pi^2 c}\}^{1/2} - 8\frac{p(Nr-Ne)^2}{2\pi}]^{1/2} \quad (3.38)$$

$$c = \frac{12.7}{4}(135.63 - \frac{20+60}{2} + [\{135.63 - \frac{(20+60)^2}{4\pi^2 c}\}^{1/2} - 8\frac{12.7(60-20)^2}{2\pi}]^{1/2}$$

$$c = 303.53 + [11.63 - 160.85]$$

c = 452.75 mm

3.5.9.3 Power transmit ion chain in to wheel shaft

Harvesting machine operated by 5 kW diesel engines and the speed of engine shaft is 970 rpm and that of rotary is to be 1/3 x 1/3 of 970 rpm. The maximum distance between drive pulley and rotary shaft is to be 300 mm. calculate a suitable chain drive for the rotary. Since the speed of diesel engine in the present case is not very high, so we may take lower chain velocity and a roller chain is selected for the drive. Determine velocity ratio (VR): the velocity ratio of drive can be determined by following equation

$$\text{VR} = \frac{\text{RPM}_e}{\text{RPM}_r} = \frac{323.3}{107.78} = 3$$

Now, from the chain table which is available in the design handbook for a velocity ratio of 3 say 3. The number of teeth in the pinion is equal to 20.

Number = 20

Nr = Ne x V.R = 20 X 3 =60 teeth

Calculate of chain pitch

$$P = \frac{c}{30 \text{ to } 60} = \frac{400}{30 \text{ to } 60} = 13.33 \text{ from the to } 6.667 \text{ mm}$$

From the table in design data book we may select standard chain pitch = 12.7 mm

Chain velocity

$$V = \frac{P}{1000} X \frac{RPM}{60} XNe \frac{m}{s} \tag{3.39}$$

$$V = \frac{12.7}{1000} X \frac{107.78}{60} X20 \frac{m}{s} = 0.45 \text{ m/s}$$

Calculate of load on chain

$$F_{chian} = F + F_C + F_f$$

F_f = Frictional force chain

$$F = \frac{P(KW)X1000}{V}$$

F_C = centrifugal load on chain

$$F = 5 \text{ kW} X \frac{1000}{0.45} = 11111.11 \text{ N}$$

F = load due to power transmission

$F_c = \frac{W \text{ x } V^2}{g}$, where; W = 13.6 N/m for duplex chain from design data book

$$F_C = \frac{13.6 \text{ x } 0.45^2}{9.81} = 0.62 \text{ N}$$

$F_f = K_f * W * C$, where;

c = center to center distance (400 mm)

K_f = kinematics friction

For inclined drive (K_f) =2.0

For horizontal drive (K_f) = 4.0

For vertical drive (K_f) = 1.0

K_f= 2 X 13.6 X0.4 m = 10.88 N

F_C = 11111.11 N+ 0.62N+ 10.88 N = 11122.6 N

New check the loading strength of chain from the chain table which is available in design hand book for duplex chain minimum breaking load = 431 KN

$$F_{os} = \frac{Fu}{c} = \frac{\text{breaking load}}{\text{chain load}} = \frac{431X10^3 \text{ N}}{11122.6 \text{ N}} = 38.75 \text{ N}$$

When it is greater than 38.75 N >11.00 N the chain selection for rotary is safe.

Chain length = L_C = m x p, when p =chain pitch and m = number of chain link

$$M = \frac{2c}{p} + \frac{Nr - Ne}{p} + \frac{p(Nr - Ne)^2}{4\pi^2 c} \qquad (3.40)$$

$$M = \frac{2 \text{ x } 400}{12.7} + \frac{20 + 60}{2} + \frac{12.7(60 - 60)^2}{4\pi^2(400)}$$

$$M = 63 + 40 + 2 = 105 \text{ pitches}$$

L_C = m x p = 115 x 12.7 =1437.5mm

3.5.9.4 Center to center distance between sprockets

$$c = \frac{p}{4}(m - \frac{Ne + Nr}{2} + [\{m - \frac{(Ne + Nr)^2}{4\pi^2 c}\}^{1/2} - 8\frac{p(Nr - Ne)^2}{2\pi}]^{1/2} \qquad (3.41)$$

$$c = \frac{12.7}{4}(105 - \frac{20+60}{2} + [\{105 - \frac{(25+60)^2}{4\pi^2 c}\}^{1/2} - 8\frac{12.7(60-20)^2}{2\pi}]^{1/2}$$

c = 374 mm

3.5.9.5 Design chain tension

Design a chain to drive a dynamo generating 5 kW at1/3 0f 970 r.p.m. and fitted with a sprocket pinion 60 mm diameter.

Allowable stress for belt = 20 MPa

Density of rubber = 1000 kg / m3

Angle of contact for dynamo pulley = 165°

Coefficient of friction between belt and pulley = 0.3

Given: P= 5 kW = 5×10^3 W;

N= 323.33 r.p.m.; d= 60 mm = 0.06 m; η, d = 85% = 0.85; σ= 20 MPa = 20×10^6N/m2

ρ= 1000 kg/m^3; θ= 165° = 165 × π/180= 2.88 rad; μ= 0.3

T_1= Tension in the tight side of the chain, and

T_2= Tension in the slack side of the chain.

We know that velocity of the chain,

$$V = 1 + \frac{\pi.\mathrm{d.N}}{60} = 1 + \frac{\pi X0.06X323.33}{60} = 2.07\frac{M}{S}$$

And power transmitted (P)

$$5 \times 10^3 = (T_1 - T_2)\ v \qquad (3.42)$$

$$= (T_1 - T_2)X2.07$$

$$= 2.07\ (T_1 - T_2)$$

$$\therefore\ T_1 - T_2 = 5\ x10^3/2.07 = 2415.46\ N \qquad (3.43)$$

We know that

$$2.3 \log\frac{T_1}{T_2} = \mu.\theta = 0.3\ x\ 2.88 = 0.864$$

$$\log\frac{T_1}{T_2} = \frac{0.864}{2.3} = 0.3756$$

$$\frac{T_1}{T_2} = 2.375 \qquad (3.44)$$

... (Taking antilog of 0.3756)

From equations ((3.43)) and (3.44), we find that

T_1= 4172.16 N; and T_2= 1756.7 N

3.5.9.6 Design chain tension

Design a chain to drive a dynamo generating 5 kW at1/3 0f 970 r.p.m. and fitted with a sprocket pinion 120 mm diameter.

Allowable stress for belt = 20 MPa

Density of rubber = 1000 kg / m3

Angle of contact for dynamo pulley = 165°

Coefficient of friction between belt and pulley = 0.3

Given: P= 5 kW = 5 × 10^3 W;

N= 323.33 r.p.m.; d= 60 mm = 0.06 m; η, d = 85% = 0.85; σ= 20 MPa = 20 × 10^6N/m2

ρ= 1000 kg/m^3; θ= 165° = 165 × π/180= 2.88 rad; μ= 0.3

T_1= Tension in the tight side of the chain, and

T_2= Tension in the slack side of the chain.

We know that velocity of the chain,

$$V = 1 + \frac{\pi.\mathrm{d.N}}{60} = 1 + \frac{\pi X0.12X323.33}{60} = 3.03\,\frac{M}{S}$$

And power transmitted (P)

$$5 \times 10^3 = (T_1 - T_2)\,v$$

$$= (T_1 - T_2)X2.07$$

$$= 3.03\,(T_1 - T_2)$$

$$\therefore\ T_1 - T_2 = 5\,x10^3/2.07 = 1650.16\ N \qquad (3.45)$$

We know that

$$2.3\log\frac{T_1}{T_2} = \mu.\theta = 0.3\ x\ 2.88 = 0.864$$

$$\log\frac{T_1}{T_2} = \frac{0.864}{2.3} = 0.3756$$

$$\frac{T_1}{T_2} = 2.375 \qquad (3.46)$$

... (Taking antilog of 0.3756)

From equations (3.45) and (3.46), we find that

T_1= 2850.28N; and T_2= 1200.11 N

3.5.10 Shaft design

A shaft is a rotating machine element which is used to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within the shaft permits the power to be transferred to various machines linked up to the shaft. The shafts are usually cylindrical, but may be square or cross-shaped in section. They are solid in cross-section but sometimes hollow shafts are also used. But my shaft is cylindrical shape. The design of shaft material, the design of shaft material will be: -

1. It has high strength.
2. It has good machinability.
3. It has low notch sensitivity factor.
4. It has good heat treatment properties.
5. It has high wear resistant properties.

3.5.10.1 General layout of shaft

The general layout of the shafts, including axial location of pulleys, sprocket gear and bearings, must now is specified in order to perform a free-body force analysis and to obtain shear force and bending moment diagrams. If there is no existing design to use as a starter, then the determination of the shaft layout may have many solutions. A free-body force analysis can be performed without knowing shaft diameters, but cannot be performed without knowing axial distances between pulley (sprocket gears) and bearings. Strength of shaft, in designing shafts on the basis of strength.

(a) Shafts subjected to twisting moment or torque only,

When the shaft is subjected to a twisting moment (or torque) only, then the diameter of the shaft may be obtained by using the torsion equation.

T = Twisting moment (or torque) acting upon the shaft,

J = Polar moment of inertia of the shaft about the axis of rotation,

Ʈ= Torsional shear stress, and

r = d / 2; where d is the diameter of the shaft.

T/Ʈ=Ʈ/r (3.47)

T=(πX Ʈ X d^3)/(16) (3.48)

J=π/(16 X d^3) (3.49)

3.5.10.2 Frist stage pulley support shaft design

A shaft is supported by two bearings placed 0.7 m apart. A 200 mm diameter pulley is mounted at a distance of 150 mm to the right of left hand bearing and this drives a pulley directly below it with the help of belt having tension $T_{11} = 2132.435$ N and $T_{12} =$ 897.87 N [1ststage of belt]. Another Sprocket pinion 60 mm diameter is placed 78 mm to the left of right hand bearing and is driven with the help of diesel engine and belt having tension $T_{21} = 4172.16\,N$ AND $T_{22} = 1756.7$ N $\left[2^{nd}\right.$ stage of belt tension$\left.\right]$, which is placed horizontally to the right and third pulley 120 mm diameter is placed 323 mm to the right hand bearing a belt having tension $T_{31} = 2850.28$N AND $T_{32} = 1200.11$ N [2nd stage of belt 2 tension], The angle of contact for both the pulleys is 180° and μ =0.24. Determine the suitable diameter for a solid shaft, Ultimate tensile strength, 670 MPa. Factor of safety is 2. Assume that the torque on one pulley is equal to that on the other pulley. Vertical load acting on the shaft at A, T_1= 2132.435 N; and T_2= 897.87 N

$W_{C1} = T_{11} + T_{12} = 2132.435\text{ N} + 897.8\text{ N} = 3030.235\text{ N}$ And horizontal load on the shaft at A=0

The vertical load diagram is shown in Fig. 15 (c).

We know that torque acting on the pulley A,

$$T = (T_{11} - T_{12})R_A = (2132.435\text{ N} - 897.8\text{ N})0.1 = 123.46\text{ N.M}$$

The torque diagram is shown in Fig. 15 (b).

Vertical load acting on the shaft at B,

$$W_{C2} = (T_{31} + T_{32}) = (4172.16\text{ N} + 1756.7\text{ N}) = 5928.86\text{ N}$$

We know that torque acting on the pulley b,

$T = (T_{21} - T_{22})R_B = (2850.28N - 1200.11\ N)0.3 = 495.051\ N.M$

We know that torque acting on the pulley C,

$W_{C3} = (T_{21} + T_{22}) = (2850.28N + 1200.11\ N) = 4050.39N$

We know that torque acting on the pulley C,

$T = (T_{31} - T_{32})R_B = (4172.16\ N - 1756.7\ N)0.3 = 724.638\ N.M$

The maximum torque at B or C pulley is 289.85 N. M

The torque diagram is shown in Fig. 15 (b).

First of all, considering the vertical loading at C. Let RAV and RBV be the reactions at the bearings A and B respectively. We know that

$R_{XV} + R_{YV} = 3030.235\ N + 5928.86\ N.M - 4050.39N = 4908.705\ N$

Taking moments about X,

$R_{YV} \times 0.4 = 4908.705 \text{ N} \times 0.15$

$R_{YV} = 1840.76 \text{ N}$

and $R_{XV} = 3030.235 \text{ N} - 1840.76 \text{ N} = 1189.47 \text{ N}$

We know that B. M. at Aand B,

$M_{XV} = M_{YV} = 0$

B. M. at A, MCV

$$M_{AV} = R_{XV} \times 0.1 = 1189.47 \text{ N} \times 0.1 = 118.9 \text{ N.M}$$

B. M. at B, MDV

$$M_{BV} = R_{XV} \times 0.12 = 1189.47 \text{ N} \times 0.12 = 142.7 \text{ N.M}$$

B. M. at B, MDV

$$M_{CV} = R_{YV} \times 0.323 = 1840.76 \text{ N} \times 0.323 = 594.56 \text{ N.M}$$

The bending moment diagram for horizontal loading is shown in Fig. 16 (f).

horizontal loading is on there fore $M_{Xh} = 0$ and $M_{Yh} = 0$ shown in Fig. 15 (e).

Resultant B.M. at A, $\sqrt{(M_{AV}{}^2 + M_{Ah}{}^2} = \sqrt{1189.47^2 + 0^2} = 1189.47 \text{ N.M}$

Resultant B.M. at B, $\sqrt{(M_{BV}{}^2 + M_{Bh}{}^2} = \sqrt{142.7^2 + 0^2} = 142.7\ N.M$

Resultant B.M. at B, $\sqrt{(M_{CV}{}^2 + M_{Ch}{}^2} = \sqrt{594.56^2 + 0^2} = 594.56\ N.M$

Maximum bending moment

The resultant B.M. diagram is shown in Fig. 16 (g).

We see that the bending moment is maximum at B, therefore Maximum B.M,

$M = M_A = 1189.47 \ N.M \ Ans.$

$Let\ d = \ Diameter\ of\ the\ shaft.$

$We\ know\ that\ equivalent\ twisting\ moment,$

$$T_e = \sqrt{M^2 + T^2} = \sqrt{1189.47^{\,2} + 123.46^{\,2}} = 1195.86\ N.M = 1195.86\ X10^3 N.mm$$

$We\ also\ know\ that\ equivalent\ twisting\ moment\ (T_e),$

Table 4The mechanical properties of these grades of carbon steel Richard G. Budynas and J. Keith Nisbett (2011)

standard designation	Ultimate tensile strength, MPa	Yield strength, MPa
40 C	560 - 670	320
45 C	610 - 700	350
50 C	640 - 760	370
50 C	700 Min	390

$\tau = \ 0.3\ \sigma_{el}\ or\ 0.18\ \sigma_u$

$$\tau = \ 0.3\ X\ 1195.86\ N.M = \frac{478.344}{f} = \frac{478.344}{1.5} = 239.172\ mpa$$

$$T_e = \frac{\pi * \tau * \ d^3}{16}$$

$$1195.86\ X10^3 N.mm = \frac{\pi * 60.3\ \ MPa * \ d^3}{16}$$

$$16\ x\ X\ 1195.86\ \ X10^3 N.mm = \pi * 239.172\ mpa * \ d^3$$

$$d^3 = 25464.8$$

$$d = 29.42\ mm\ say\ 30\ mm$$

$Again\ we\ know\ that\ equivalent\ bending\ moment,$

$M_e = 1/2(M + \sqrt{M^2 + T^2} = 1/2(M + T_e)$

$M_e = 1/2(\ 1189.47\ N.M + 1195.86\ N.M)$

$M_e = 1192.665\ N.M$

$We\ also\ know\ that\ equivalent\ bending\ moment\ (Me$

$\sigma_b =\ 0.6\ \sigma_{el} or\ 0.36\ \sigma_u$

$\sigma_b = 0.36\ X\ 1192.665\ = \frac{429.3594}{f} = \frac{429.3594}{2} = 214.67\ mpa$

$M_e = \frac{\pi * \sigma_b *\ d^3}{16}$

$1192.665\ \ N.M\ X10^3 N.mm = \frac{\pi * 214.67\ MPa *\ d^3}{16}$

$16\ X\ 1192.665\ X10^3 N.mm = \pi * 214.67 MPa *\ d^3$

$d^3 = 28295.5$

$d = 30\ mm$

Bending, torque and load diagram of Frist stage pulley support shaft design

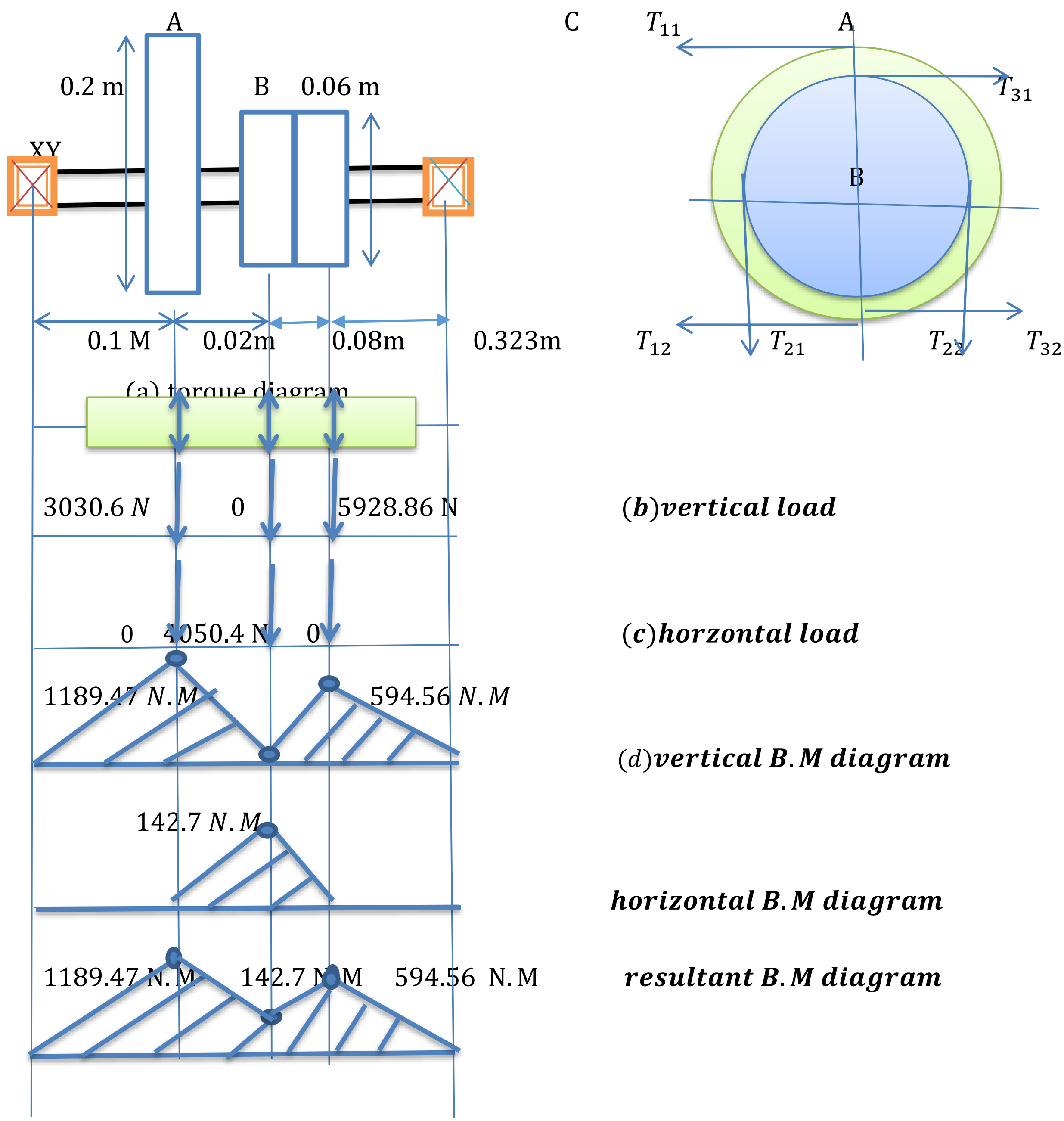


Figure 37 Bending, torque and load diagram of shaft

3.5.10.3 Second stage Pulley support shaft design

Shafts subjected to twisting moment or torque only,

A shaft supported at the ends in ball bearings carries a straight tooth spur gear at its mid span and is to transmit 5 kW at 161.67 r.p.m. The pitch circle diameter of the pulley is 150 mm. The distances between the center line of bearings and pulley are 100 mm each. If the shaft is made of steel and the τ_u = 610MPa = 610 N/mm², determine the diameter of the shaft. Show in a sketch how the gear will be mounted on the shaft; also indicate the ends where the bearings will be mounted? Factor of safety is 4

L= 40 mm = 0.04 m; τ_u = 610MPa = 610 N/mm²

$$\tau = \frac{\tau_\mu}{f} = \frac{610}{4} = 152.5N/mm2$$

Let d = Diameter of the shaft.

$$T_1 = \frac{P * 60}{2\pi N} = \frac{5000w * 60}{2\pi N} = \frac{300000}{2\pi * 161.67} = 295.33\ N.m = 295333\ N.mm$$

$$T = 295.33N.mm$$

We also know that torque transmitted by the shaft (T),

$$295333\ N.mm = \frac{\pi X\ \ X\ d^3}{16} = \frac{\pi X\ 152.5\ X\ d^3}{16}$$

d ^{3}=295333 $N.mm^4$/29.94 N.mm

d^3 =9863mm^3

d =21.44 mm say 25 mm shaft diameter

3.5.10.3.1Shafts Subjected to Bending Moment Only

$$T_1 = \frac{P * 60}{2\pi N} = \frac{5000w * 60}{2\pi N} = \frac{300000}{2\pi * 161.67} = 295.33\ N.m = 295333\ N.mm$$

$$T_1 = 295333\ N.mm$$

Since the sprocket is mounted at the middle of the shaft, therefore maximum bending moment at the center of the sprocket,

$$M = \frac{WL}{4} = \frac{4050.39\ N * 800\ mm}{4}$$

M = 810078 N.mm

Let d = Diameter of the shaft.

We know that equivalent twisting moment.

$T_e = \sqrt{M^2 + T^2} = \sqrt{(810078\ N.mm)^2 + (295333\ N.mm)^2}$

T_e = 862234.3 N.mm

862234.3N.mm = π /16 x Ʈ x do $^3(1-k^4)$ = π /16 x 40 Mpa x d ^{3} x $(1-0.8^4)$

$862234.3\ \mathrm{N.mm} = \frac{\pi X\ Ʈ\ X\ d^3}{16} = \frac{\pi X\ 152.5\ X\ d^3}{16}$

d ^{3}=862234.3N.mm ^{4}/29.943 N.mm

d^3 =28795.8 mm^3

d =30 mm shaft diameter

3.5.10.4 Wheel shaft design subjected to twisting moment or torque

The design of cylindrical key way Shaft rotating at 107.78 r.p.m. is to transmit 5 KW. The shaft may be assumed to be made of mild steel with an Ultimate tensile stress of 610 MPa. A shaft is supported by two bearings placed 1 m apart. A 300 mm diameter sprocket is mounted at a distance of 160 mm to the right of left hand bearing and this drives a sprocket directly below it with the help of chain having maximum tension of 11.92 KN. The angle of contact for both the pulleys is 180° and μ =0.24. Determine the diameter of the shaft, neglecting the bending moment on the shaft. Assume a factor safety as 4

N = 107.78 r.p.m.;

P = 5,000 W;

Ʈ u = 610MPa = 610 N/mm2

$$\tau = \frac{\tau_\mu}{f} = \frac{610}{4} = 152.5N/mm2$$

Let d = Diameter of the shaft.

$$T_1 = \frac{P * 60}{2\pi N_G} = \frac{5000w * 60}{2\pi N_G} = \frac{300000}{2\pi * 107.78} = 443\ N.m = 443008\ N.mm$$

$$T\ = 443008\ N.mm$$

We also know that torque transmitted by the shaft (T),

$$443008\ N.mm = \frac{\pi X\ Ʈ\ X\ d^3}{16} = \frac{\pi X\ 152.5\ X\ d^3}{16}$$

d ^{3}=443008$N.mm$ ^{4}/29.943 N.mm

d^3 =14795.06 mm^3

d =24.5mm say 25 mm shaft diameter

(b) Shafts subjected to bending moment only,

(c) Shafts subjected to combined twisting and bending moments, and

(d) Shafts subjected to axial loads in addition to combined torsional and bending loads.

We shall now discuss the above cases, in detail, in the following pages.

3.5.10.5 Shafts Subjected to Bending Moment Only

When the shaft is subjected to a bending moment only, then the maximum stress (tensile or compressive) is given by the bending equation.

M = Bending moment,

I = Moment of inertia of cross-sectional area of the shaft about the axis of rotation,
δ_b = Bending stress, and

y = Distance from neutral axis to the outer-most fiber.

We know that for a round solid shaft, moment of inertia y=d/2

$$T = \frac{\pi \; X \; d^4}{64} \qquad y = \frac{d}{2}$$

$$M = \frac{\pi * \delta_b * d^3}{16}$$

3.5.10.6 The design of Shafts Subjected to Bending Moment

The design of cylindrical key way Shaft rotating at 107.78 r.p.m. is to transmit 5 KW. The shaft may be assumed to be made of mild steel with an Ultimate tensile stress of 610 MPa. A shaft is supported by two bearings placed 1 m apart. A 120 mm diameter sprocket is mounted at a distance of 160 mm to the right of left hand bearing and this drives a sprocket directly below it with the help of chain having maximum tension of 11.92 KN. If the shaft is made of steel and the allowable shear stress is 152.5 MPa, determine the diameter of the shaft. Show in a sketch how the gear will be mounted on the shaft; also indicate the ends where the bearings will be mounted? The pressure angle of the gear may be taken as 20°.length of shaft is 1000 mm.

N = 107.78 r.p.m.;

P = 5,000 W;

τ_u = 610MPa = 610 N/mm^2

$$\tau = \frac{\tau_\mu}{f} = \frac{610}{4} = 152.5 N/mm2$$

Let d = Diameter of the shaft.

$$T_1 = \frac{P * 60}{2\pi N_G} = \frac{5000w * 60}{2\pi N_G} = \frac{300000}{2\pi * 107.78} = 443\ N.m = 443008\ N.mm$$

$$T = 443008\ N.mm$$

$$F = \frac{2T}{D} = \frac{443008\ \ N.mm}{120} \qquad Tangential\ force\ on\ the\ Sprocket\ gear$$

F_t = 3691.73 N

W = $F_{t/COS}$ α =3691.73 N/COS 20^0

$$W = \frac{F_t}{COS\ \alpha} = \frac{3691.73\ \text{N}\ N}{COS\ 20^0}$$

W = 3928.6 N

Since the sprocket is mounted at the middle of the shaft, therefore maximum bending moment at the center of the sprocket,

$$M = \frac{WL}{4} = \frac{=\ 3928.6\ N * 1000\ mm}{4}$$

M = 982165.14 N.mm

Let d = Diameter of the shaft.

We know that equivalent twisting moment.

$T_e = \sqrt{M^2 + T^2} = \sqrt{443008\ N.mm^2 + 982165.14\ \text{N.mm}^2}$

T_e = 1077452.76 N.mm

1077452.76 N.mm = π /16 x Ʈ x do ^{3}(1-k^4) = π /16 x 40 Mpa x d ^{3} x (1-0.8^4)

$1077452.76\ N.mm\ = \frac{\pi X\ Ʈ\ X\ d^3}{16} = \frac{\pi X\ 152.5\ X\ d^3}{16}$

d ^{3}=1077452.76 N.mm ^{4}/29.943 N.mm

d^3 =35983.46 mm^3

d =33 mm say 35 mm shaft diameter

Step 1 calculation of reactions

The two reaction (R_A and R_B) of the beam are obtained from a transverse EE and a rotational EE written at E, which is the midpoint of the distributed load P. distributed load P_1 which is found on the middle part of the machine and distributed load P_2 which is found on the front part of the machine

$P_{load} = m * g$

mass on the middle part of the machine 90kg and front part of the machine 60 kg

$P_{d\ load\ 1} = m_{middle} * g$

$P_{load\ 1} = 90\ kg * 9.81\ m/s^2$

$P_{load\ 1} = 882.9\ N$

$P_{load\ 2} = m_{front} * g$

$P_{load\ 2} = 60\ kg * 9.81\ m/s^2$

$P_{d\ load\ 2} = 588.6\ N$

$P_{total\ load} = 882.9\ N + 588.6\ N$

$P_{total\ load} = 1471.5\ N$

$$\sum_{y} M: \quad R_A + R_B - P_{y_{total}} = 0$$

$R_A + R_B - P_{load\ 1} - P_{load\ 2} = 0$

$R_A + R_B - 1471.5\ N = 0$

$$\sum_{a} M: \quad 0 * MR_A + 1 * MR_B - 0.1m * P_{d\ load\ 1} - 0.99m * P_{dis\ load\ 2} = 0$$

$0 + 1MR_B - 0.1m * 882.9\ N - 0.99m * 588.6\ N = 0$

$1.1MR_B - 88.29Nm - 582.714\ NM = 0$

$1.1MR_B = 671\ Nm$

$R_B = 671\ Nm/1.1\ m$

$R_B = 610\ N$

$the\ result\ of\ R_A\ \ is$

$R_A + R_B - 1471.5\ \ N = 0$

$610\ N + R_B - 1471.5\ \ \ N = 0$

$R_B = 861.5\ N$

Bending, torque and load diagram of wheel shaft design

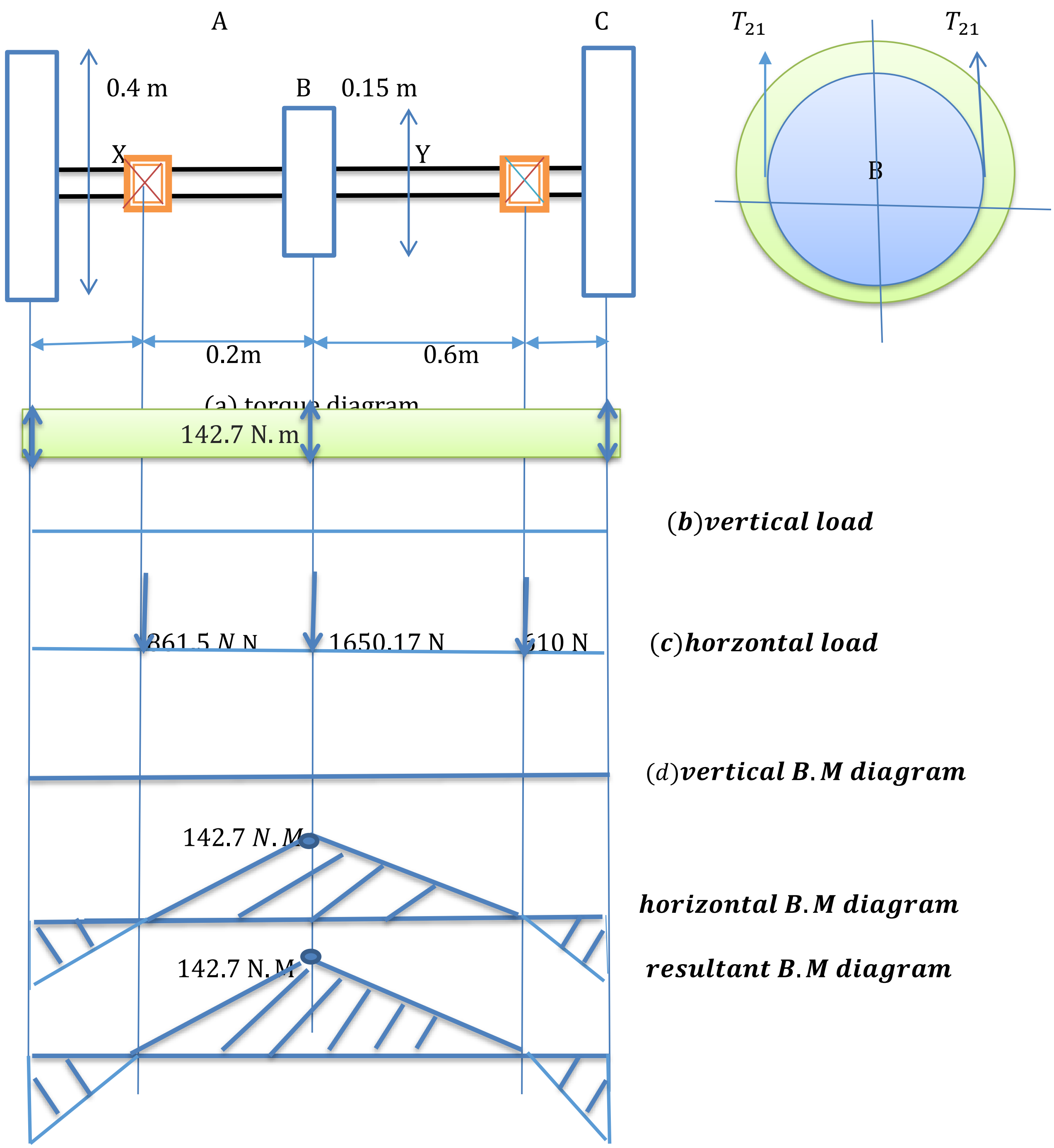


Figure 38 Bending, torque and load diagram of shaft

3.5.11 Bearing Selection

Bearing is used to support a load while permitting relative motion between two elements of a machine. The term rolling contact bearings refers to the wide variety of bearings that use spherical balls or some other type of roller between the stationary and the moving elements. The most common type of bearing supports a rotating shaft, resisting purely radial loads or a combination of radial and axial (thrust) loads. Some bearings are designed to carry only thrust loads. Most bearings are used in applications involving rotation, but some are used in linear motion applications. According to the design Ball select bearing [Richard G. Budynas and J. Keith Nisbett (2011)].

The main advantages of ball bearings are

1. Low starting and running friction except at very high speeds.

2. Ability to withstand momentary shock loads.

3. Accuracy of shaft alignment.

4. Low cost of maintenance, as no lubrication is required while in service.

5. Small overall dimensions.

6. Reliability of service.

7. Easy to mount and erect.

8. Cleanliness.

Figure 39 bearing[Richard G. Budynas and J. Keith Nisbett (2011) select six different ball bearings according to the shaft diameter

- Four 30mm bearings (20mm is internal diameter of bearing or the shaft diameter)
- One 35mm bearing

3.5.11.1 Bearing selection for input shaft

By selecting ball bearings for the intermediate shaft for above reason with a reliability of 99 percent. The problem specifies a design life of 12 000 h. The intermediate shaft speed is1250 rev/min. The estimated bore size is 35mm and the estimated bearing width is 30mm. From the free-body diagram of above first shaft;

R_{AV} = 3030.6 N R_{AH} = 1189.47N R_{AR} = 3255.67N

R_{BV} = 4050.39 N R_{BH} = 594.56 N R_{BR} = 4093.8 N

At the shaft speed of 250rev/min, the design life of 12 000 h correlates to a bearing life of

L_D = (12 000 h) (60 min/h) (1250 rev/min) = 9×10^8 rev.

Since the average life of the bearing is 5 years at 10 hours per day, therefore life of the bearing in hours,

$L_H = 5 \times 300 \times 10 = 15\,000$ hours ... (Assuming 300 working days per year) and life of the bearing in revolutions,

$L = 60\,N \times L_H = 60 \times 1250 \times 15\,000 = 1.125 \times 10^7$ rev

$$F = C_0 = R_f \left[\frac{C}{X_O + (\theta - X_O)(1 - Re)}\right]^{1/b} \qquad (3.50)$$

Where F = Basic static load rating or, the design load

R_F =reaction force at the bearing

C = Basic dynamic load rating,

R_e = reliability

x = life measure dimensionless vitiate, L/L10

X0 = guaranteed, or “minimum,” value of the vitiate $x_0 = 0.02$

Characteristic parameter corresponding to the 63.2% value of the vitiate, which is

(θ – x0) =4.439 from Richard G. Budynas and J. Keith Nisbett (2011)

b = shape parameter that controls the skew mess for ball bearing b = 1.483.

$$C = \frac{LD}{L} = \frac{9 \times 10^8}{= 1.125 \times 10^7} = 80$$

Since the maximum reaction force is at point A we must do the design load this reaction force

R_A =6264.83N

$$F=C_0 =Rf \left[\frac{C}{X_O+(\theta-X_O)(1-Re)}\right]^{1/b} \tag{3.51}$$

$$F=C_0 =6264.83N\left[\frac{80}{0.02+(4.439)(1-0.99)}\right]^{\mathbf{1/1.483}} =764498.45N =764.5KN$$

From Table VI, the bearing number 200 having C = 43Kn with bore diameter=10mm, outside diameter=35mm, and width of 13mm may be selected, and the bearing number 202 having C = 160kN, which bore diameter=15mm, outside diameter=35mm, and width of 11mm may be selected.

3.5.11.2 Bearing selection for intermediate shaft

By selecting ball bearings for the intermediate shaft, for above reason with a reliability of 99 percent The problem specifies a design life of 12 000 h. The intermediate shaft speed is 250 rev/min. The estimated bore size is 60mm and the estimated bearing width is 30mm.

From the free-body diagram of above wheel shaft;

R_{AH} = 90 N R_{BH} = 60 N R_{BR} = 108.16 N

From the free-body diagram of above;

R_{AV} = 775N R_{AH} = 316N R_{AR} = 946.65N

R_{BV} = 1200N R_{BH} = 446N R_{BR} = 1466.35

At the shaft speed of 250rev/min, the design life of 12 000 h correlates to a bearing life of

LD = (12 000 h) (60 min/h) (250 rev/min) = 1.8×10^8 rev.

Since the average life of the bearing is 5 years at 10 hours per day, therefore life of the bearing in hours,

LH = 5 × 300 × 10 = 15 000 hours ... (Assuming 300 working days per year) and life of the bearing in revolutions,

L = 60 N × LH = 60 × 250 × 15 000 = 22.5×10^6 rev

$$F=C_0=R_f\left[\frac{C}{X_O+(\theta-X_O)(1-Re)}\right]^{1/b} \quad (3.52)$$

Where F = Basic static load rating or, the design load

R_F =reaction force at the bearing

C = Basic dynamic load rating,

R_e = reliability

x = life measure dimensionless variate, L/L10

X0 = guaranteed, or "minimum,'' value of the variate $x_0 = 0.02$

Characteristic parameter corresponding to the 63.2121 percentile value of the variate, which is $(\Theta - x0) = 4.439$

b = shape parameter that controls the skew ness for ball bearing b = 1.483.

$$C = \frac{LD}{L} = \frac{1.8 \times 10^8}{= 22.5 \times 10^6} = 80$$

Since the maximum reaction force is at point B we must do the design load by this reaction force

$R_B = 1466.35$

$$F = C_O = R_f = \left[\frac{C}{X_O + (\theta - X_O)(1 - Re)}\right]^{1/b}$$

$$F = C_O = 1466.35\left[\frac{80}{0.02 + (4.439)(1 - 0.99)}\right]^{\mathbf{1/1.483}} = 178939\text{N} = 179\text{KN}$$

From Table above, the bearing number 204 having C = 160kN, width bore diameter=20mm, outside diameter=47mm, and width of 14mm may be selected, and the bearing number 206 having C = 160kN, with bore diameter=30mm, outside diameter=62mm, and width of 16mm may be selected.

3.5.12 design and select cutter bar and cutter guard lips.

The cutter bar consists of a single action cutter bar, which has 28 knifes triangular in shape located above fixed one has the same number of guards. The cutter bar is 140 cm in length takes its reciprocating motion from the machine engine through transmission system.

Figure 40cutter bar

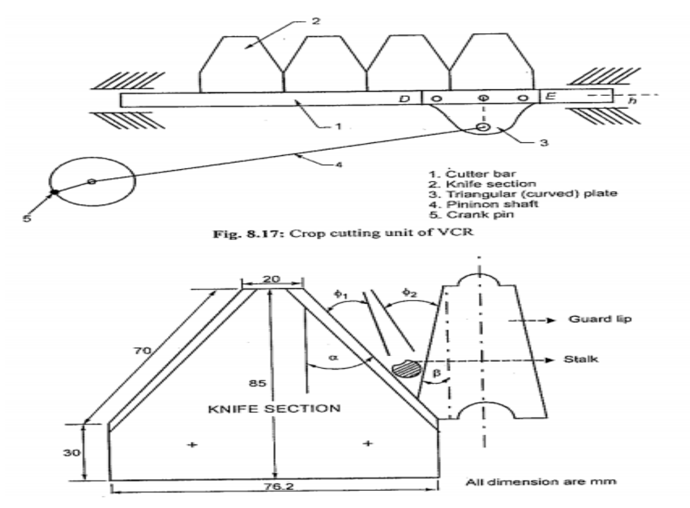


Figure 41 cutter bar and cutter guard lips [D. N sharma; S.mukesh (2008 and 2010).]

3.5.12.1 Size and shape of knife section:

It is selected based on following parameters: [*D. N sharma; S. mukesh (2008 and 2010).*]

(i) Gripping of stalk by cutting pair: The crop is cut by impact and shear action between the knife section and guard lip. Owing to the deflection of the crop, the height of stubble is somewhat greater than the height of the cutter bar above ground surface. For normal cutting action, the stalk should be pinched between cutting edge of knife and guard lip as given in Fig. 41 This condition is satisfied when

$$< \alpha + < \beta \leq < \varphi_1 + < \varphi_2 \tag{3.53}$$

where, $< \alpha$ = angle between cutting edge and axis of knife section

$< \beta$ = angle between cutting edge and axis of twin guard

$< \varphi_1 \& < \varphi_2$= angles of friction between the crop and cutting edge of knife and guard lip respectively

(ii) Rake angle, sharpness and thickness of cutting edge: According to Klenin (1985) the knives of rake angle of 22 ^{0} are sharp for longer time but force requirement for cutting is higher as compared to rake angle of 19 ^{0}, But, with the existing tool grinders the rake angle obtained varies from 22-25 ^{0}. So, rake angle of 22 ^{0} with sharp cutting edges is selected for the knife section. Thickness of knife section should not exceed 120-130 um for most of the cereals.

(iii) Pitch of serrated knives: In order to avoid slipping off the stalks, the knife section should be serrated. The pitch of serrated knives is selected two or three times smaller than the diameter of paddy and wheat stalk i.e. the pitch should be 1-1.2 mm.

(iv) Clearance between knife section and twin guard: Best results are obtained when clearance between knives and ledger plates is maintained at 0.3 mm. So, a clearance of 0.5-1.0 mm is selected.

(v) Velocity of knife section: The cutting of stalk is greatly affected by knife speed. The velocity of knife section is expressed as:

$$V_k = R * V_m \tag{3.54}$$

where, V_k = average knife velocity, m/s

V_m= forward speed of machine, m/s

R = velocity ratio

Type of cutter bar: A reciprocating type cutter bar having standard knife section of 76.2 mm stroke length and two cuts per stroke is selected. Number of knife sections required for the cutter bar of Reaper machine would be:

On the basis of above discussed design parameters, the dimensions of different components of crop cutting unit selected.

$$N_C = L_c/L_k \tag{3.55}$$

$N_C = number\ of\ cutter\ bar$

$L_K = length\ of\ knife(7.62\text{ cm})$

$L_C = length\ of\ cutter\ bar(120\ cm)$

$$N_C = \frac{120}{7.62} = 15.77 \sim 15$$

$$N_C = 15$$

3.5.12.2 Calculate knife velocity

For best results, according to Klenin (1985) for cc of 3 1 0 the knife velocity should be .5 m/s. The value of R ranges from 1.3 to 1.4 with available cutter knives (Bansal, 1989). Taking 'R' as 1 .3 and V_m-of 5.0 km/hr (l .39 m/s) and putting values in eqn. 4.3, we get

V_k = 1.3 x 1.17 m/s = 1.52 m/s Also,

we know that

$$V_k = X * \frac{N_k}{30} \tag{3.56}$$

$where,$

$X =\ stroke\ length\ \ 7.62\ mm$

$N_k =\ RPM\ of\ knife\ section$

Therefor, $N_k = V_k * \frac{30}{X} = 1.52\ m/s\ (100\ cm/m * \frac{30}{7.62})$

$N_k = 598.42\ rpm$

$N_k = 598.42\ rpm\ or\ say\ 600\ rpm.$

For a standard 76.2 mm knife section, the above velocity translates into 1280 strokes/min or 640 rpm of knife section. Now, we know that engine rpm = 3200 rpm. Assume a speed reduction of 1: 2 from engine to intermediate shaft and I: 3 from intermediate shaft to cutter bar shaft. Therefore, Knife rpm (N_k) is given by

$$N_k = 970\ rpm * \frac{7.4}{12} * \frac{1}{3} * 3 = 598.16\ rpm\ or\ say\ 600\ rpm.$$

Assume 10% slippage in belt drives

Therefore,

$$N_k = 485\ rpm * \frac{90}{100} = 540\ rpm$$

The actual average knife speed of cutter bar of self-propelled reaper would be

$$N_k = 540\ rpm\ \left(\frac{7.62}{30*100}\right) = 1.372\ m/s$$

For this knife speed rake angle of 19^0-22^0 is selected.

3.5.12.3 Forward speed of self-prospered riding reaper machine

Check for self-propelled riding reaper forward speed: The power tiller is powered by a 5 hp, 970 rpm diesel engine. It is fitted with iron wheels with outside diameter of 40 cm. The engine is fitted with 6 cm diameter pulley and there is speed reduction at two stages and overall transmission speed ratio of 18: l.

Therefore, theoretical forward speed of power tiller is

$V_m = D_w x\ 970\ /\ (60\ x\ 6)\ = \pi\ x\ 40\ x\ 970/\ (60x100x6)\ = 3\ .386\ m/s$ Assuming 15-25%-wheel slippage during motion of reaper.

Therefore, actual speed of VCR$V_{m1} = 3\ .386 \frac{m}{s} * \frac{75}{100} = 2.56\ m/s$ or 9.14 km/hr.

3.5.12.4 Forces acting on cutter bar of a reaper

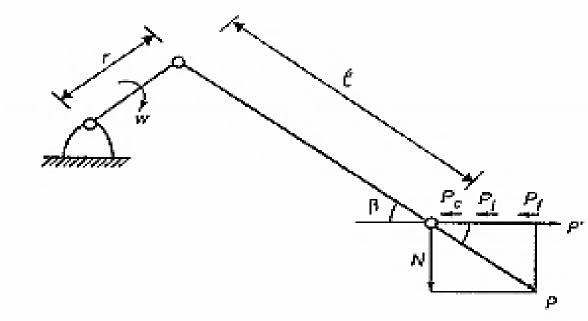


Figure 42 reciprocating knife type cutter bar [N sharma; S. mukesh (2008 and 2010)]

$$p = p_c + p_f + p_i \qquad (3.57)$$

$p - total\ resisting\ force\ in\ cutting\ bar,\ N\ or\ Kg$

$p_c - average\ resistance\ to\ cutting,\ N\ or\ Kg$

$p_f - frictional\ \ force\ N\ or\ Kg$

$p_i - inertia\ force\ of\ knife\ section\ N\ or\ Kg$

Where,

Resistance to cutting (Pc): To estimate resistance to cutting for cereal crops, it is assumed that the plants are uniformly distributed in the rows and two rows will be cut by manually operated machine.

$$P_C = E.F_1\,{}^{N}/_{X_C} \qquad (3.58)$$

$E = 1.25N - cm/cm2\ for\ wheat\ and\ 2.0N - cm/cm2\ for\ rice$

$F_l = knife\ load\ area;\ 41.9\ cm^2 for\ a\ single\ stroke\ cutter\ with\ 5.5\ cm\ knife\ sweep$

Where, length of knife (x)=7.62 cm

$knife\ load\ area = 7.62\text{ cm} * 5.5\text{ cm} = 41.9\ cm^2$

$N = number\ of\ knife\ sections\ in\ cutter\ bar, 15\ for\ 120\ cm\ cutter\ bar$

$Xc = displacement\ of\ knife\ from\ start\ to\ end\ of\ cutting$ (2.811 $cm\ for\ standard$

$knife\ section$)

$P_C = E.F_1 {}^{N}/_{X_C}$ $P_C = 1.25 * 41.9 * {}^{15}/_{2.81}$

PC = 279.58 N in wheat crop

$P_C = E.F_1 {}^{N}/_{X_C}$ $P_C = 2 * 41.9 * {}^{15}/_{2.81}$

PC =447.33 N in rice crop

3.5.12.4.1Inertia force (F_l): the inertia is governed by the mass (M_k) of reciprocating

cutter bar which is given by [(3.58)]

$$P_l = r.\omega^2.M_k(1 - {}^{x}/_{r}) \tag{3.59}$$

Where,

$M_k = mass\ of\ knife\ section, kg$

$r = radius\ of\ crrank, cm$

$\omega = angular\ velocity, \frac{rad}{s}, 0.84$

$x = length\ of\ stroke, cm$

$$P_l = r.\omega^2.M_k \tag{3.60}$$

For reaper weight of cutter bar per meter length is 20-22 Weight (M_k) of 120 cm long 28 N. at V m/sec and r= 3.81 cm of a 120 cm long cutter bar.

$P_{lmax} = 3.81\ cm * (0.84)^2 * 21\ N$

$P_{lmax} = 56.45\ N$

Frictional force(p_f): it acts on the knife as it slides over the finger bar and is given by

$$p_f = p_{f1} + p_{f2} \tag{3.61}$$

p_{f1}=force due to weight of cutter bar, N

$p_{f2} = force\ due\ to\ action\ of\ connecting\ rod, N$

$$p_{f1} = M_k * f \tag{3.62}$$

$f = 0.2\ to\ 0.3\ take\ as\ 0.25$

$p_{f1} = 28 * 0.25$

$p_{f1} = 7\ N$

$p_{f2}\ is\ the\ force\ caused\ by\ normal\ component\ of\ its\ force\ exerted\ by\ connecting$

$rod\ on\ knife\ as\ under:$

For wheat crops , r =3.81 cm and l 10 cm,$tan\,\beta = 0.22\ \ and\ p_c = 335.5\ N$)

$$p_{f2} = \left\{(p_c + p_{lmax} + p_{f1}) - \frac{tan\,\alpha}{l - f\ tan\,\beta}\right\} * f \tag{3.63}$$

$$p_{f2} = \left\{(\ 279.58\text{ N}\ + 56.45\ N + 7\ N) - \left(\frac{tan\ 29^0}{l - 0.25\ tan\ 22^0}\right)\right\} * 0.25$$

$p_{f2} = \{343.03 - (0.616)\} * 0.25$

$p_{f2} = 85.6035\ N$

For rice crops , r =3.81 cm and l 10 cm,$tan\,\beta = 0.22\ \ and\ p_c = 536.8\ N$)

$$p_{f2} = \left\{(447.33\text{ N}\ + 56.45\ N + 7\ N) - \left(\frac{tan\ 29^0}{l - 0.25\ tan\ 22^0}\right)\right\} * 0.25$$

$p_{f2} = \{510.78 - (0.63)\} * 0.25$

$p_{f2} = 127.5\ N$

Therefore, the total friction force (p_f) for wheat crops is

$p_f = p_{f1} + p_{f2}$

$p_{f1} = 7\ N$

$p_{f2} = 85.6035\ N$

$p_f = 7\ N + 85.6035\ N = 92.6\ N$

Therefore, the total friction force (p_f) for rice crops is $p_f = p_{f1} + p_{f2}$

$p_{f1} = 7\ N$

$p_{f2} = 127.5\ N$

$p_f = 7\ N + 127.5\ \ N = 134.54$ N

Therefore, the total resisting the motion cuter bar for wheat crop is

$$p = p_c + p_f + p_i \qquad (3.64)$$

$p = 279.58\ N + 92.6\ N + 56.45\ N = 298.1\ N$

Therefore, the total resisting the motion cuter bar for rice crop is

$p = p_c * p_f\ \ * p_i p$

$p = 447.33$ N $* 134.54$ N $* 56.45\ N = 638.32$ N

Power requirement of reaper machine: Total power required for operation of self-propelled harvesting machine (W_m) is given by the equation [(3.63)

where, $$W_m = W_r + W_n + W_w \qquad (3.65)$$

W_r =power for rolling machine,

W_n= power for no-load working of parts of machine,

W_w = power required for full load working of machine, w

Power for rolling of machine (W_r) is given by

$$W_r = p * v/(\eta_{tr} * \eta_s) \qquad (3.66)$$

where, P = resistance to rolling of machine,

η_{tr}= transmission efficiency, percentage

η_s = coefficient of skidding of machine

According to Devnani (1985), the power required for cutting crop at a knife speed of 1.372 m/s of cutter bar can be taken as 0.513 hp/m length of cutter bar and power required for conveying cut crop as 50% of the cutting' power.

Therefore, power required for full load working of machine $W_w = W_c + W_{con}$

$W_w = 0.513\ hp\ +\ 50\%\ of\ 0.513\ hp$

$W_w =\ 0.513\ kW\ +\ 0.2565\ hp$

$W_w =\ \ 0.7695\ hp$

Power for rolling of machine (W_r) is estimated as: W F P, v/75.

Let the coefficient of rolling resistance be 0.2 and weight of machine as 250 kg operating at a forward speed of 2.54 m/s.

Therefore, $W_r = 250\ kg * \ 0.2 * 2.54\ /75\ =\ 1.7\ hp$

Total power W = $W_w + W_r$ = 0.7695 hp + 1.7 hp = 2.47 hp.

Assuming 90% power transmission efficiency at each stage and there would be three reductions from engine to cutting and conveying unit and two reductions from engine to drive wheels of power tiller. Let us assume, the average efficiency as 70%.

Therefore, power requirement of self-propelled riding reaper = 2.47 /0.70 = 3.53 hp.

Therefore, 5 hp. diesel engine will be adequate to operate the machine in the actual field conditions for harvesting of crop.

3.5.13 Finger Conveyor design and developed

In a pin jointed four bar mechanism, as shown in Fig. 4.23, AB = 325 mm, BC = 300, CD = 100mm, and AD = 320 mm. The angle BAD = 60°. The crank AB rotates uniformly at 138.57 r.p.m. Locate all the instantaneous centers and find the angular velocity of the link BC. [R.S. KHURMI and J.K. GUPTA (2005)].

$N_{AB} = 138.57\ \mathrm{r.p.m}$ or $\omega_{BA} = 2\pi * \dfrac{138.57}{60} = 14.51\ \mathrm{rad/s}$

Since the length of crank AB = 325 mm = 0.325 m, therefore velocity of point B on link AB,

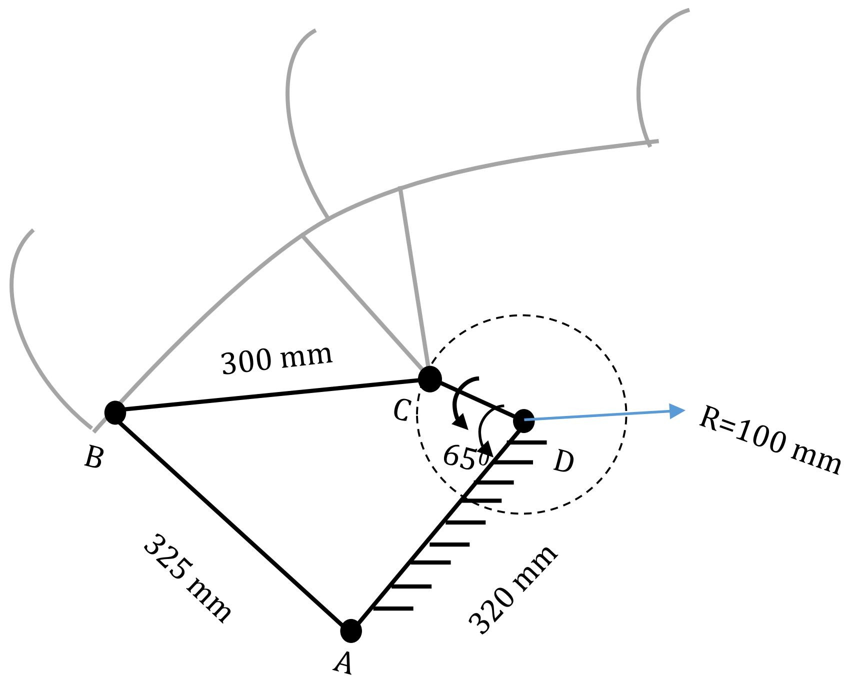


Figure 43 Finger Conveyor

$V_B = \omega_{BA} * \mathrm{AB} = 14.51\ \mathrm{rad/s} * 0.325 = 4.716\ \mathrm{m/s}$

Location of instantaneous centers

The instantaneous centers are located as discussed below: [Richard G. Budynas and J. Keith Nisbett (2011)].

1. Since the mechanism consists of four links (i.e. n = 4), therefore number of instantaneous centers,

$$N = \frac{n(n-1)}{2} = \frac{4(4-1)}{2} = 6$$

2. For a four bar mechanism, the book keeping table may be drawn as discussed in Art. 6.10.
3. Locate the fixed and permanent instantaneous centers by inspection. These centers are I^{12}, I^{23}, I^{34} and I^{14}, as shown in Fig. 35.

4. Locate the remaining neither fixed nor permanent instantaneous centers by Aron hold Kennedy's theorem. This is done by circle diagram as shown in Fig. 36. Mark four points (equal to the number of links in a mechanism) 1, 2, 3, and 4 on the circle

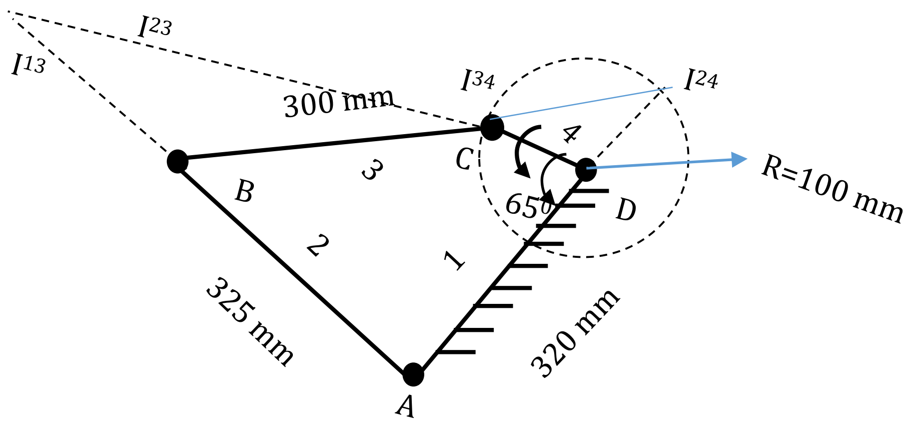


Figure 44 Finger Conveyor angular velocity

5. Join points 1 to 2, 2 to 3, 3 to 4 and 4 to 1 to indicate the instantaneous centers already located i.e. I12, I23, I34 and I14.
6. Join 1 to 3 to form two triangles 1 2 3 and 3 4 1. The side 13, common to both triangles, is responsible for completing the two triangles. Therefore, the instantaneous center I^{13} lies on the intersection of the lines joining the points $I^{12}I^{23}$ and I^{34} I^{14} as shown in Fig. 36. Thus center I^{13} is located. Mark number 5 (because four instantaneous centers have already been located) on the dotted line 1, 3.
7. Now join 2 to 4 to complete two triangles 2 3 4 and 1 2 4 The side 2 4, common to both triangles, is responsible for completing the two triangles. Therefore, center I24 lies on the intersection of the lines joining the points I^{23} I^{34} and I^{12} I^{14} as shown in Fig. 37. Thus center I^{24} is located. Mark number 6 on the dotted line 2 4. Thus all the six instantaneous centers are located.

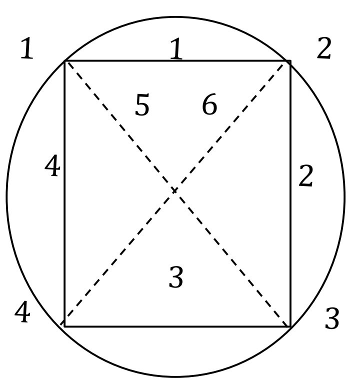


Figure 45 angular velocity

Angular velocity of the link BC

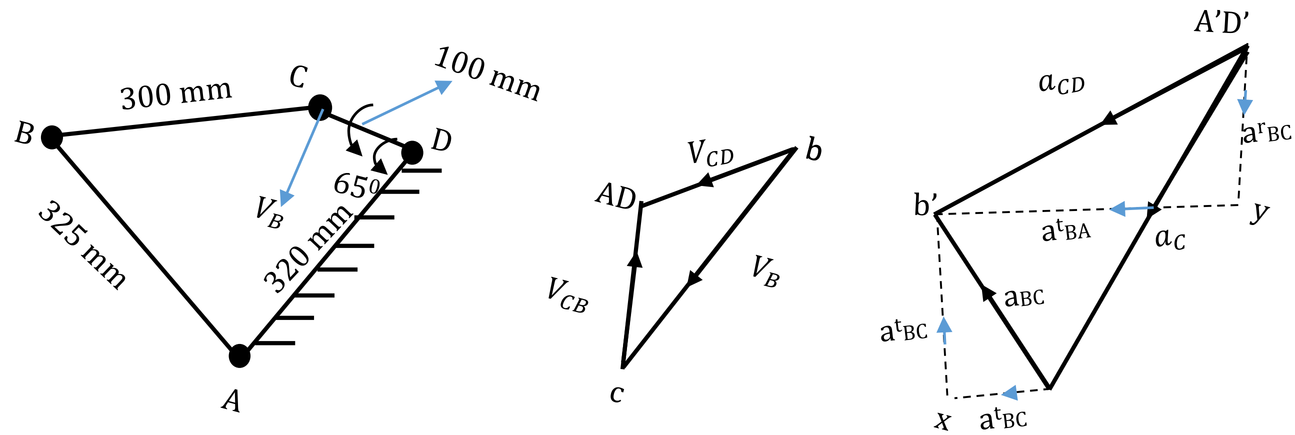
Let ω_{BC} = Angular velocity of the link BC.

Since B is also a point on link BC, therefore velocity of point B on link BC,

$$V_B = \omega_{BA} * \mathrm{I}_{13} \qquad (3.67)$$

By measurement, we find that I^{13} B = 480 mm = 0.48 m

$$\omega_{BC} = \frac{V_A}{\mathrm{B} * \mathrm{I}_{13}} = \frac{4.716\ \mathrm{m/s}}{0.48} = 9.825\ rad/s$$



a Space diagram (All dim. in mm). b, Velocity diagram c, Acceleration diagram.

Figure 46 angular acceleration

Since the length of crank CD = 100 mm = 0.1 m, therefore velocity of B with respect to A or velocity of B, (because A is a fixed point),

$$\text{vectors ab } = V_{BA} = V_B = \omega_{BA} * CD = 4.716\frac{\mathrm{m}}{\mathrm{s}} * 0.1\mathrm{mm}$$

$$V_B = 0.4716\ m/s$$

First of all, draw the space diagram to some suitable scale, as shown in Fig. 46 (a). Now the velocity diagram, as shown in Fig. 46 (b), is drawn as discussed below: [R.S. KHURMI and J.K. GUPTA 2005]

1. Since the link AD is fixed, therefore points a and d are taken as one point in the velocity diagram. Draw vector ab perpendicular to BA, to some suitable scale, to represent the velocity of B with respect to A or simply velocity of B (i.e. V_{BA} or V_B) such that

   vector ab = V_{DC} = V_C = 0.4716 m/s

2. Now from point b, draw vector bc perpendicular to CB to represent the velocity of C with respect to B (i.e. V_{CB}) and from point d, draw vector dc perpendicular to CD to represent the velocity of C with respect to D or simply velocity of C (i.e. V_{CD} or V_C). The vectors bc and dc intersect at b. By measurement, we find that

$V_{AB} = V_B = \text{vectors ab} = 0.361\ m/s$

We know that BA = 325 mm = 0.325 m

∴Angular velocity of link BA,

$$\omega_{BA} = \frac{V_{AB}}{\text{BA}} = \frac{0.361\ m/s}{0.325\ \text{m}} = 1.11\ rad/s(s\text{clockwise about } A)$$

We know that angular velocity of link BC=300mm=0.3mm, the vectors bc and ba intersect at c. By measurement, we find that $V_{BC} = \text{vectors bc} = 0.251\frac{m}{s}\ and\ V_{BC} = 0.321\frac{m}{s}$

$$\omega_{BC} = \frac{V_{BC}}{\text{BC}} = \frac{0.251\ m/s}{0.3\ \text{m}} = 0.83735\ rad/s(\text{Anticlockwise})$$

and angular velocity of link AB,

$$\omega_{AB} = \frac{V_{BA}}{\text{BA}} = \frac{0.251\ m/s}{0.325\ \text{m}} = 0.772\ rad/s(\text{Clockwise})$$

Angular acceleration of links CB and BA

Since the angular acceleration of the crank DC is not given, therefore there will be no tangential component of the acceleration of C with respect to D.

We know that radial component of the acceleration of C with respect to D (or the acceleration of C),

$$\omega^{r}{}_{CD} = \omega_{CD} = \omega_{C} = \frac{V^{2}{}_{CD}}{\mathrm{CD}} = \frac{0.4716^{2}\ \mathrm{m/s}}{0.1\ \mathrm{m}} = 2.224\ rad/s$$

Radial component of the acceleration of B with respect to C,

$$\omega^{r}{}_{CB} = \frac{V^{2}{}_{CB}}{\mathrm{CB}} = \frac{0.251\ m/s}{0.3\ \mathrm{m}} = 0.837\ rad/s$$

and radial component of the acceleration of B with respect to A (or the acceleration of B),

$$\omega^{r}{}_{CB} = \frac{V^{2}{}_{BA}}{\mathrm{BA}} = \frac{0.321\ m/s}{0.325\ \mathrm{m}} = 0.99\ rad/s$$

The acceleration diagram, as shown in Fig. 46 (c) is drawn as follows:

1. Since d and a are fixed points, therefore these points lie at one place in the acceleration diagram. Draw vector d'c' parallel to DC, to some suitable scale, to represent the radial component of acceleration of C with respect to D or acceleration of C i.e a_{Cr} d or a_Q such that

   vectors $\mathrm{D'C'} = a^{r}{}_{CD} = 0.4716\ \mathrm{m/s}$

2. From point c', draw vector c' x parallel to CB to represent the radial component of acceleration of B with respect to C i.e. $a^{r}{}_{BC}$ such that

   vectors $\mathrm{C'}x = a^{r}{}_{CB} = 0.251\ m/s$

3. From point x, draw vector xb' perpendicular to QR to represent the tangential component of acceleration of B with respect to c i.e BC at whose magnitude is not yet known.
4. Now from point A', draw vector A'y parallel to AB to represent the radial component of the acceleration of B with respect to A i.e. BA ab such that

   vectors $\mathrm{A'}x = a^{r}{}_{BA} = 0.321\ \ m/s$

5. From point y, draw vector yb' perpendicular to AB to represent the tangential component

of acceleration of B with respect to A i.e. $a^r{}_{BA}$ at.

6. The vectors xb' and yb' intersect at b'. Join d 'b and d' b'. By measurement, we find that

$a^t{}_{BC}$ = vectors $B'x = 0.64\ \mathrm{rad}/s^2$, $a^t{}_{BA}$ = vectors $b'y$

We know that angular acceleration of link CB,

$$a^t{}_{BC} = \frac{a^t{}_{BC}}{\mathrm{CB}} = \frac{0.251\ m/s}{0.3\ \mathrm{m}} = 0.772\ rad/s$$

and angular acceleration of link BA,

$$a^t{}_{BC} = \frac{a^t{}_{BA}}{\mathrm{B\ A}} = \frac{0.251\ m/s}{0.325\ \mathrm{m}} = 0.772\ rad/s$$

3.5.14 design spring on hinder

In a spring loaded reaper binding as shown in Fig. 38, the cutting crops binding put on the vertical arms of the binder lever, the horizontal arms of which lift the sleeve against the pressure exerted by a spring. The mass of binding crops is 10 kg and the lengths of the vertical and horizontal arms of the binding crank lever are 400 mm and 150 mm respectively. Design a suitable close coiled round section spring for the governor. From Table IV Assume permissible stress in spring steel as 420 MPa, modulus of rigidity 84 kN/mm2 and spring index 8. Allowance must be made for stress concentration, factor of which is given by [R.S. KHURMI and J.K. GUPTA 2005].

$$\frac{4C-1}{4C-4} + \frac{0.615}{C} \quad where\ C\ is\ the\ spring\ index$$

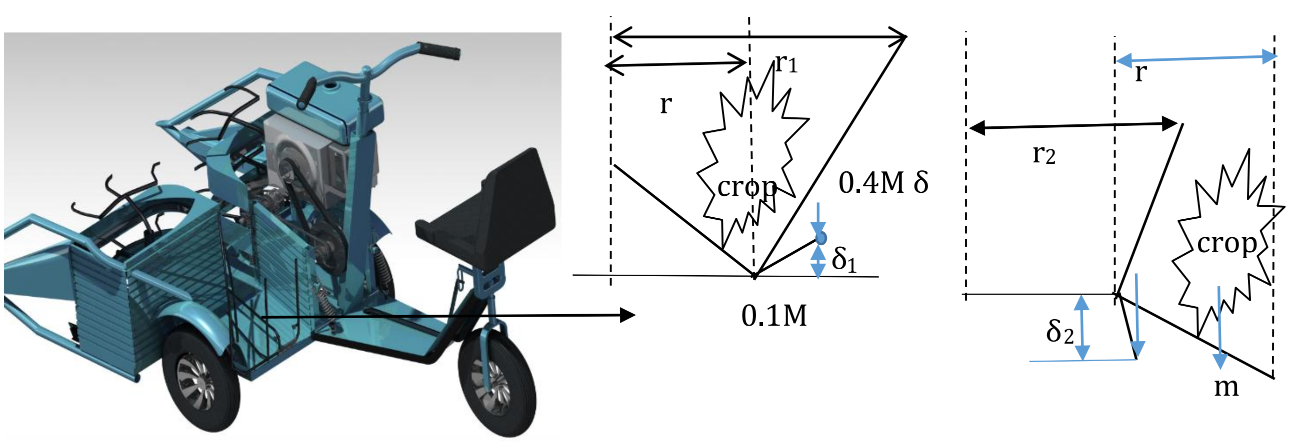


Figure 47 crop binder spring design

We know that the spring force.

$$S_1 = Mxg\ x\ r_1\ x\ \frac{x}{y} = 10\ kg\ x\ 9.81\frac{m}{s^2x}0.9m\ x\ \frac{0.5m}{0.1m}$$

$$S = 441.45\ N$$

$$S_2 = Mxg\ x\ r_2\ x\ \frac{x}{y} = 10\ kg\ x\ 9.81\frac{m}{s^2x}0.5m\ x\ \frac{0.5m}{0.1m}$$

$$S_1 = 245.25N$$

compression of the spring,

$$\delta = (r_1 - r)x\ \frac{y}{x} + (r_2 - r)x\ \frac{y}{x} \tag{3.68}$$

$$\delta = (0.9 - 0.5)x\ \frac{0.1}{0.5} + (0.5 - 0.5)\ x\ \frac{0.1}{0.5}0.08m = 80mm$$

Diameter of the spring wire

Let d = Diameter of the spring wire in mm.

We know that Wahl's stress factor,

$$k = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4x8-1}{4x8-4} + \frac{0.615}{4} = 1.184$$

We also know that maximum shear stress (τ)

$$\tau = kx\frac{8xwxc}{\pi d^2} = 1.184\ x\frac{8x441.45\ Nx8}{\pi d^2} \quad (Substituting\ W\ =\ S, the\ maximum\ spring\ force)$$

$$d^2 = 10647.9/420 = (25.35)^2 = 5.035mm$$

From Table V, we shall take the standard wire of size SWG 1 having diameter

(d) = 5.38 mm Ans.

Mean diameter of the spring coil

Let D = Mean diameter of the spring coil. We know that the spring index,

C = D/d or D = C.d = 8 × 5.38 = 46.64 mm Ans.

Number of turns of the coil

Let n = Number of active turns of the coil.

We know that compression of the spring (δ),

$$s = \frac{8xwxc^3xn}{Gxd^2} \tag{3.69}$$

$$80mm = \frac{8x196.2x8^3xn}{84x10^3x5.38^2} \qquad \ldots(Substituting\ W\ =\ S_1 - S_2)$$

$n = 44.4{\sim}45$

and total number of turns using squared and ground ends,

n' = n + 2 =45 + 2 = 47

Free length of the coil

Since the compression produced under a force of $245.25N$ is 80 mm, therefore maximum compression produced under the maximum load of 441.45 N is,

$$\delta_{max} = \frac{80\ mm}{245.25}x441.45 = 144mm$$

We know that free length of the coil,

$$L_F = n' x d x \delta_{max} + 0.15\delta_{max} = 144mm$$

$$L_F = 47' x5.3x144 + 0.15x144 = 418.46\ mm$$

Pitch of the coil

We know that pitch of the coil

$$\text{pitch of the coil} = \frac{\text{free length}}{n' - 1} = \frac{418.46}{47 - 1} = 9.1$$

3.5.15 Steering

When the vehicle is moving straight on a flat surface, the direction of the center of gravity (CG) keeps the same with the orientation of the vehicle. When the reaper turns, the yaw rate causes the change of the orientation. The reaper demonstrates a velocity component perpendicular to the orientation, known as the lateral velocity. Then, the orientation of the vehicle and the direction of the travel are no longer the same. The vehicle is moving under the influence of different forces. If a lateral force is acting on the tire, an angle is formed between the direction of movement of the tire and the tire straight line. This angle is called the sideslip angle α (shown in Fig. 48). Reason for this sideslip angle or the tire slip is the elastic lateral deflection of the rolling tire in the tire contact area under the effect of the lateral force between tire and road. For analyzing the motion behaviors of a single track model, a linearization of the tire lateral force and the tire slip angle is assumed via a tire stiffness C_a [Lorenzo Morello, Lorenzo Rosti Rossini, Giuseppe Pia and Andrea Tonoli (2011)].

$$C_a = \frac{F_a}{\alpha} \tag{3.69}$$

When the vehicle moves at low speed, the wheels roll without a tire slip angle since the lateral cornering force, F_a , is small and can be ignored. The vehicle model can be seen as the assumption of Rudolf Ackermann with the elongations of all wheel center lines intersecting at one point, the center of the turning curve (Fig. 48).

The steering angle, δ , can be simply calculated as:

$$\delta = arctan\left(\frac{l}{r}\right)$$

r: radius of the curve (r=10)

$$\delta = arctan\left(\frac{l}{10\ m}\right) = 0.11\ rad = 6.3^0$$

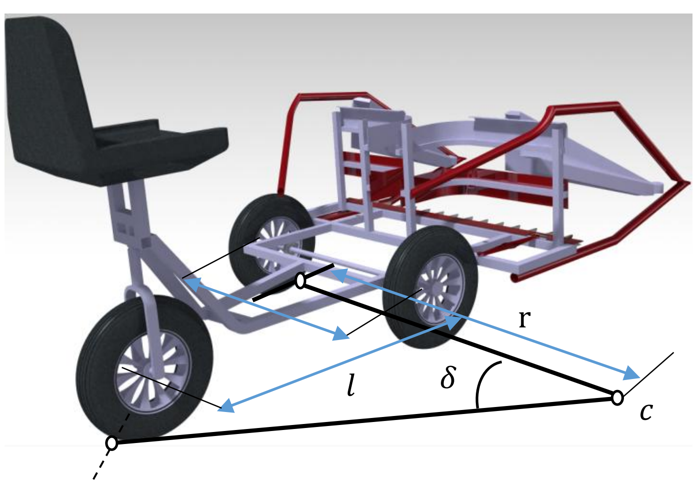


Figure 48 The steering angle

However, when the lateral force appears, the vehicle front wheel orientation and the vehicle movement direction is no longer the same. A simplified description of the vehicle lateral dynamics is demonstrated in a reaper wheel model (Fig.49). The tire contact points are in the center of tires. Longitudinal forces in the tire contact points as well as wheel load fluctuations are not considered. The height of the center of gravity is zero. The Newton's law equation of the motion for the vehicle lateral direction is: [Lorenzo Morello, Lorenzo Rosti Rossini, Giuseppe Pia and Andrea Tonoli [2011].

$$m\alpha_y = F_{If} + F_{Ir} \qquad (3.70)$$

The force of inertia acting on the vehicle center of gravity, $m\alpha_y$, corresponds to the centrifugal force:

$$m\alpha_y = m\frac{v^2}{r} + F_{Ir} + m\frac{v}{r}\dot{v}r = mv(\dot{\Psi} - \dot{\beta}) \qquad (3.71)$$

$$F_{cent} = m\frac{v^2}{r} \qquad (3.72)$$

Where;

m = mass of the machine (250 kg)

v = velocity (2.54 m/s)

r: radius of the curve (r=10)

$$F_{cent} = 250\ kg\,\frac{2.54\ \mathrm{m/s^2}}{10\ m}$$

$$F_{cent} = 63.5\ N$$

Moment balance in front point,

$$F_{cent}l_f = F_{Ir}l \qquad (3.73)$$

$$F_{Ir} = F_{cent}\frac{l_f}{l} = 63.5\ N * \frac{0.6}{1.1}$$

$$F_{Ir} = 34.63\ N$$

$$F_{If} = F_{cent}\frac{l_r}{l} = 63.5\ N * \frac{0.5}{1.1}$$

$$F_{Ir} = 28.86\ N$$

Where;

$l_f = 0.6\ m$

$l_r = 0.5\ m$

$l = 1.1\ m$

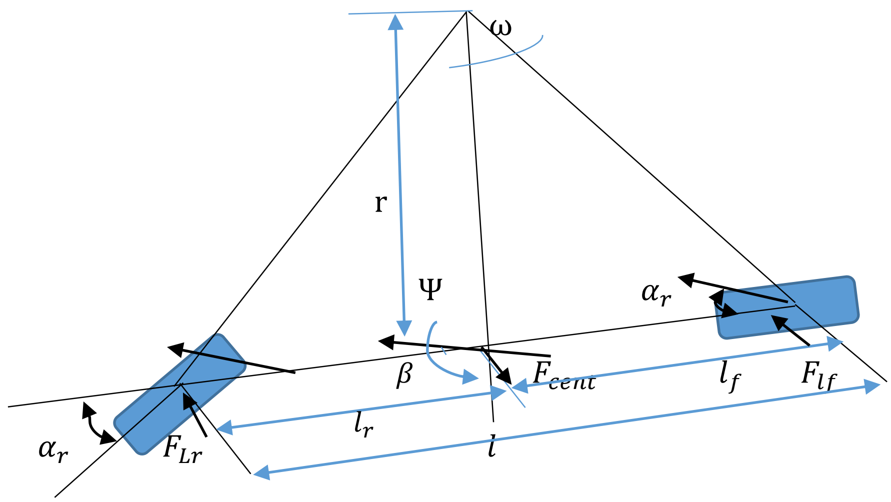


Figure 49 Deflection of the rolling tire by a lateral cornering force

where, v : Vehicle velocity, ω : Vehicle angular velocity, r : radius of curve, Ψ : Yaw angle, β : Side slip angle, δ : Steering angle, a : Tire slip angle, l : Wheel base. And the gyroscopic effect on the z-axis at the vehicle center of gravity:

$$J\ddot{\Psi} = F_{If}l_f + F_{Ir}l_f \tag{3.74}$$

where J is the vehicle moment of inertia on z-axis. The tire side forces can be calculated from the given tire slip rigidity, c_a , in equation ((3.74)) for the front wheel:

$$F_{If} = c_{af}a_f \tag{3.75}$$

And for the rear wheel

$$F_{Ir} = c_{ar}a_r \tag{3.76}$$

and from the rear tire slip, a_r:

$$a_r = \frac{F_{Ir}}{c_{ar}} \tag{3.77}$$

Assume;

$c_{ar} = c_{af} = for\ small\ tire$ 30,000N/rad

$$a_r = \frac{34.63\ N}{30{,}000\text{N/rad}}$$

$$a_r = 0.00115\ rad = \frac{180^o * 0.00115\ rad}{\pi} = 0.066\ ^o$$

$$a_f = \frac{F_{If}}{c_{af}} \tag{3.78}$$

$$a_f = \frac{28.86\ N\ N}{30{,}000\text{N/rad}}$$

$$a_f = 0.000962\ rad = \frac{180^o * 0.000962\ rad}{\pi} = 0.055\ ^o$$

The side slip angle for the vehicle at the center of gravity, β , can be formulated from the front tire slip, a_f:

$$a_f = \delta + \beta - \frac{l_f \dot{\Psi}}{v} \tag{3.79}$$

$$\alpha_r = \beta + \frac{l_f \dot{\Psi}}{v} \tag{3.80}$$

It is noted that the tire slip rigidity or the sideslip stiffness, c_α , is an elastic property for each rubber tires, normally in the range of 30,000-50,000 N/rad.

3.5.15.1 STEERING IN STEADY STATE

In steady state condition, the vehicle speed, v, is a constant, then, the yaw velocity $\dot{\Psi}$, and the sideslip, β, are also constant, i.e, $\ddot{\Psi} = 0$ and $\dot{\beta} = 0$. The torque balance equations can be formulated at the rear contact point:

$$F_{Lfl} = m_{\alpha y} l_r \tag{3.81}$$

Replaced with the tire slip rigidity in equation (3.81):

$$c_{af}\left[\delta + \beta - \frac{l_f \dot{\Psi}}{v}\right] = \frac{l_r}{l} m_{ay} \tag{3.82}$$

And in equation (3.81):

$$c_{\alpha r}\left[\beta + \frac{l_r \dot{\Psi}}{v}\right] = \frac{l_f}{l} m_{\alpha y} \tag{3.83}$$

Because in the steady state, $\beta = 0$, then from equation (3.80),$\Psi = \frac{v}{r}$. The transformation from equation ((3.86)) and ((3.87)) leads to:

$$\delta = \frac{l}{r} + \frac{m}{l}\left[\frac{l_r}{c_{\alpha f}} - \frac{l_f}{c_{\alpha r}}\right] a_y \tag{3.84}$$

From the above equation, the necessary steering angle, δ , during the steady state driving along a curve composes of two parts. The first part, $\frac{l}{r}$, or Ackermann angle, depends on the vehicle geometrical parameters. And the second part, $\frac{m}{l}\left[\frac{l_r}{c_{\alpha f}} - \frac{l_f}{c_{\alpha r}}\right] a_y$, is characterized by the influences of the lateral acceleration and the tire rigidities, which can increase, if $\left[\frac{l_r}{c_{\alpha f}} > \frac{l_f}{c_{\alpha r}}\right]$, or reduce, if $\left[\frac{l_r}{c_{\alpha f}} < \frac{l_f}{c_{\alpha r}}\right]$, the steering angle.

From equation ((3.83)) and ((3.84)), the sideslip angle difference between the front and the rear wheel is:

$$\Delta\alpha = \alpha_f - \alpha_r = \delta - \frac{l_f \dot{\Psi}}{v} \tag{3.85}$$

With $v = \dot{\Psi} r$, then,. $\Delta\alpha = \delta - \frac{l}{r}$ Replace with δ in equation ((3.85)):

$$\Delta\alpha = \frac{m}{l}\left[\frac{l_r}{c_{\alpha f}} - \frac{l_f}{c_{\alpha r}}\right] a_y \tag{3.86}$$

Calculate β

$$\beta = \frac{l_r \dot{\Psi}}{v} - \alpha_r = \alpha_r - \frac{l_r}{r} \quad (3.89)$$

$$\beta = 0.0136\, rad - \frac{0.6}{10}$$

$$\beta = -2.66^0$$

From equation ((3.84)) Calculate δ

$$\delta = a_f - \beta + \frac{l_f}{r} \quad (3.90)$$

$$\delta = 00.055\,^o + 2.66^0 + 2.866^0$$

$$\delta = 5.57^0$$

Assess the stationary steering behavior of this vehicle.

$$\Delta\alpha = \alpha_f - \alpha_r$$

$$\Delta\alpha = 0.055\,^o - 0.066\,^o$$

$$\Delta\alpha = -0.011\,^o$$

Then, the difference of the sideslip angles depends on the vehicle and the tire parameters. The driver has to compensate the sideslip angle difference, $\Delta\alpha$, with the steering angle, δ . This forms a basic knowledge of oversteer and under-steer definition: Over-steer is if $\Delta\alpha = \alpha_f - \alpha_r < 0$, neutral is if $\Delta\alpha = \alpha_f - \alpha_r = 0$ and under-steer is if $\Delta\alpha = \alpha_f - \alpha_r > 0$ fig ()

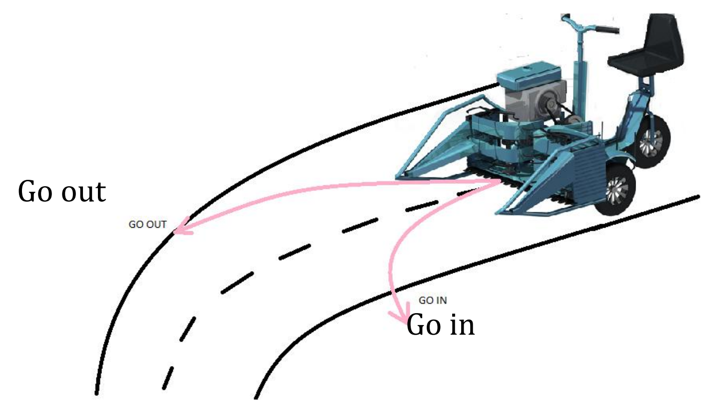


Figure 50 Over-steer (left) and Under-steer (right)

Under-steer and over-steer are vehicle dynamic characteristics used to demonstrate the sensitivity of a vehicle steering system. The under-steer happens if the vehicle turns less than the steering control of the driver. Conversely, over-steer happens if the vehicle turns more than the steering control of the driver.

3.5.15.2 Steering in Transient Response

Transformation of equations ((3.80)-(3.86)) can lead to the following expressions:

$$mv(\dot{\Psi} - \dot{\beta}) = c_{af}\left[\delta + \beta - \frac{l_f \dot{\Psi}}{v}\right] + c_{ar}\left[\beta + \frac{l_r \dot{\Psi}}{v}\right] \tag{3.91}$$

And

$$J_z \ddot{\Psi} = c_{af}\left[\delta + \beta - \frac{l_f \dot{\Psi}}{v}\right] l_f + c_{ar}\left[\beta + \frac{l_r \dot{\Psi}}{v}\right] l_r \tag{3.92}$$

Equation ((3.92)) can be represented by yaw velocity,

$$\dot{\Psi} = \frac{mv\dot{\beta} + c_{af}(\delta + \beta) + c_{ar}\beta}{mv - c_{ar}\frac{l_f}{v} + c_{af}\frac{l_r}{v}} \tag{3.93}$$

For the steady state, v = const., then, $\ddot{\Psi}$:

$$\ddot{\Psi} = \frac{mv\ddot{\beta} + c_{af}(\dot{\delta} + \dot{\beta}) + c_{ar}\dot{\beta}}{mv - c_{ar}\frac{l_f}{v} + c_{af}\frac{l_r}{v}} \tag{3.94}$$

Replace $\dot{\Psi}$ and $\ddot{\Psi}$ in equation (3.91):

$$\ddot{\beta} + \left[\frac{c_{af} + c_{ar}}{mv} + \frac{c_{af} l_f^2 + c_{ar} l_r^2}{v J_z}\right]\dot{\beta} + \left[\frac{c_{af} l_r + c_{ar} l_f}{J_z} + \frac{c_{af} l_f^2 + c_{ar} l_r^2}{J_z m v^2}\right]\beta \tag{3.94}$$

$$= \left[\frac{c_{af} l_f}{J_z} + \frac{c_{af} c_{ar}(l_f l_r + l_r^2)}{J_z m v^2}\right]\delta - \frac{c_{af}}{mv}\dot{\delta} \tag{3.95}$$

The characteristic polynomial of the dynamic equation in ((3.95)) can be represented in an inhomogeneous linear differential equation of 2nd order for the vehicle slip angle β . The homogeneous part of this differential equation has the form of a simple oscillating motion with damping:

$$\ddot{\beta} + A\dot{\beta} + B\beta = 0$$

or in the vibrated frequency form:

$$\omega_o^2 + 2D\omega_o s + s^2 = 0$$

Thus, the differential equation for the slip angle β can be viewed with a yaw undamped natural frequency, ω_o :

$$\omega_o = \sqrt{\frac{c_{ar}l_r - c_{af}l_f}{J_z} + \frac{c_{af}l^2}{J_z m v^2}}$$

Where:

$J_z = 1960\ kgm^2$

$$\omega_o = \sqrt{\frac{30{,}000 * (0.6) - 30{,}000(0.5)}{1960} + \frac{30{,}000(1.1)^2}{1960 * 250(2.54)^2}}$$

$$\omega_o = \sqrt{1.53}$$

$$\omega_o = 1.24$$

$$D = \frac{\frac{c_{af}l_r + c_{ar}l_f}{J_z} + \frac{c_{af}l_f^2 + c_{ar}l_r^2}{J_z m v^2}}{2\omega_o}$$

$$D = \frac{\frac{30{,}000 * (0.6) + 30{,}000(0.5)}{1960} + \frac{30{,}000(0.6)^2 + 30{,}000(0.5)^2}{1960 * 250(2.54)^2}}{2(1.24)}$$

$$D = \frac{165.115}{2(1.24)}$$

$$D = \frac{165.115}{2(1.24)}$$

$$D = 66.58$$

Then, the dynamic yaw frequency, ω_{omD}, is:

$\omega_{omD} = \sqrt{1 - D^2}\omega_o$

$\omega_{omD} = \sqrt{1 - 66.58^2} * 1.24$

$\omega_{omD} = 82.56$

The yaw natural frequency and damping rate can be represented for the movement of the vehicle around the vertical axis (z).

3.6 Production Ramp-Up

3.6.1 Prototype development

Before a new machine is built, it must be researched, designed, and developed into a workable product. Researchers analyze market trends, consumer surveys, and buying patterns to determine what consumers want, and then suggest what kinds of cars to make. Designers work to shape these new ideas into tangible parts or products. Engineers adapt existing parts for the new model and draw up new plans for the prototype. A prototype is a custom-built working example of a new design. Manufacturers begin by building a few prototypes before they set up a factory to build the new car. Product planners monitor the process along the way and make sure that an approved new car program finishes on time and within budget.

3.6.2 Modeling of Reaper machine

Before going to making prototype, the modeling of the Self-Prospered Reaper Harvesting Machine should be made. The figure given below shows the model of the Reaper machine made in CATIA V5-R19 software. For the designing of the reaper prototype, factors such as working width, minimum ground clearance were taken into consideration, finger conveyor and binding mechanism along with the harvesting methods. The designed prototype consisted of a cutter, a driving and braking system, a steering system, and a drive train, such that it was able to cut two rows of soybean stalks with a single cut and combine the cut portions into a single row. The prototype drove with three wheels (a two front wheel and one rear wheels). The rear wheels provided the driving function and the front wheel provided the steering function.

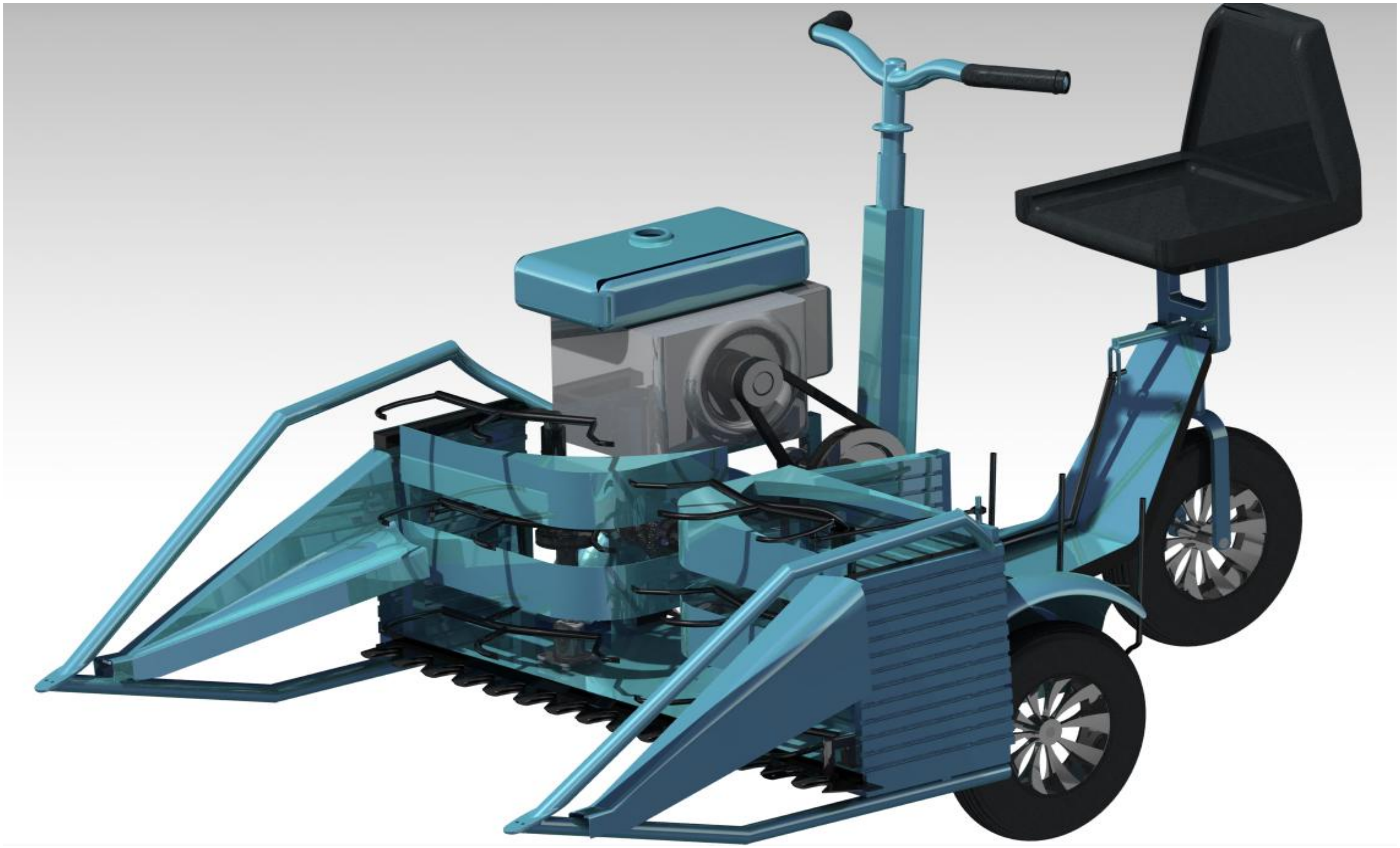


Figure 51 Isometric front view of reaper machine by CATIA V5

3.6.3 Manufacturing process of machine components

manufacturing process are the steps through which raw materials are transformed into a final product. The manufacturing process begins with the product design, and materials specification from which the product is made. These materials are then modified through manufacturing processes to become the required part. For manufactured parts process procedure, better to use process sheet method.

3.6.3.1 Uses of process sheet

A process sheet is a very important document which forms a basis for all Methodology, scheduling and dispatching function; also it helps in advance planning and for purchase of raw materials, design and manufacture of special tools, jigs, fixtures and inspection devices. Its helps in estimating the cost of the product before is an actually manufactured. It also helps in planning for man power required for doing the job.

Table 5 uses of process sheet

No of jobs	Description of operation	Machine tool	Positioning and clamping	Measuring tool
1.	Inspection of row blank Size and material type	Measuring tool	Reference point	Venire caliper, and spectrometer
2.	Profile cutting and hole making	Lathe machine	Reference Fixture	profile gauge and angle gauge
3.	Burrs removing	Bench work	Vise	Visual inspection
4.	Profile cutting	Milling lathe	Fixture	Visual inspection
5.	tampering	Electric furnace	Packing container	Surface hardens inspection
6.	welding	Arc welding	-	Angle gauge
7.	Surface treatment (cleaning and coloring)	Bath	Fixture	Visual inspection and coloring thickness tester

3.6.3.2 Frame and body Manufacturing process

- First prepare a biller material from a sheet metal to the specified sheet material by primary process
- Second: - measure and cots frame material to the require dimensions
- Bending and welding
- Joint support material with bending by using rivet or bolt with nut

3.6.3.3 Finger conveyor Manufacturing process

- First prepare a biller material from a cast material according to the specified finger conveyor material by primary process.
- Second: - Bending and welding billet material to the require.
- Pin joint movable material for conveyor mechanism.

3.6.3.4 Crop Binder Manufacturing process

- First prepare a biller material from a cast material according to the specified finger conveyor material by primary process.
- Second: - Bending and welding billet material to the require.
- Pin joint movable material for conveyor mechanism.
- Connect with spring.

3.6.3.5 Shafts Manufacturing process

- First prepare a biller material from a cast material according to the specified shaft material by primary process
- Second: - roll the billet material to the require dimensions either by hot rolling and subsequent turning to the correct size or by cold rolling to the final size
- Finally provide generous filet radius in stepped point and surface finish by machining in order to reduce stress concentrations.

3.6.3.6 Pulley Manufacturing process

The pulley can be manufactured by

- Permanent mold casting

- pulley cutters

But it is proffered that the pulley must me manufactured with CNC gear hobbling machine with the specified pulley materials.

3.6.3.7 Keys Manufacturing process

- Keys can be manufacture in a milling machine to the required dimensions

3.6.3.8 Tapestry driver sit Manufacturing process

- First prepare a biller material from a foam and synthetic lather to the specified dimensions.
- Cover driver sit by foam and synthetic lather

3.6.3.9 Standard parts

Standard parts of the reaper machine like, sprocket gear, chain, bolts, nuts and bearings can be manufactured by the standard manufacturing procedures.

3.6.4 Assembly and disassembly procedure

3.6.4.1 Assembly

When assembling the reaper harvesting machine
All necessary bolts, screws, rivets, nuts, washers and locking devices are to be furnished as assembly material for the equipment. All bolts and screwed rods shall be galvanized including the threaded portions. All nuts shall be galvanized with the exception of the threads, which shall be oiled.

Where for any type of reaper machine parts tensile steel bolts are employed then bolts of this type shall be used for all connections for every type of reaper machine parts on that line in order to avoid the use of mild steel bolts in error where high tensile type should be employed. High tensile steel bolts shall bear a mark on the heads to allow identification of grades.

All type of washers shall be included under this Contract, including locking devices and anti-vibration arrangements, which shall be subject to the approval of the Engineer. Taper washers shall be fitted where necessary. Nuts shall be finger tight on the bolt and will be rejected if they are, in the opinion of the Engineer, considered to have an excessively loose or tight fit. Nuts and bolts of the same size shall be interchangeable and supplied from one manufacturer. The Manufacturer shall supply the net quantities plus 5 percent of all permanent bolts, screws, nuts and other similar items and materials required for installation of the works at the site.

3.6.4.2 Disassembly

When disassembling the reaper harvesting machine

1. Remove the assembly wheel using proper tools for removing the bolts
2. Remove the belt in the engine pulley and first stage pulley
3. Remove the engine on the engine supporter
4. Remove the chain and sprocket gear using proper tools for removing the bolts
5. Remove the shaft where service is needed by smoothly rotating it to disengage the meshing gear on it. Then Remove the bearing.
6. Remove the gears by sliding over the shaft to a direction where a lower diameter is found. and finally remove the keys to finalize the overall disassembly.

Figure 52 Manufacturing process

Figure 53 Manufacturing process

CHAPTER FOUR

4 Results and Discussion

In this study, the discussions will cover the effect of harvesting systems as function of machines forward speeds and crop collecting on total grain losses, cutting efficiency, field capacity and efficiency, energy consumed and total cost. requirements for harvesting wheat crop. The mechanization of self-prospered riding reapers machine operation is essential to increase performances of field capacity, field efficiency, cutting efficiency by reduce manual effort, the hardness of collecting crops or making bundles. During harvesting of the crop the cutting unit collects the harvest crop from on both left and right side and pushes towards the middle position of machine for dropping or released to the ground in the form of bundles the bundles where the machine powered by 5 kW, single cylinder, water cooled diesel engine having rated engine speed of 970rpm. the Reaper machine frame structure designed and developed according to crop binding mechanism.

4.1 Working principle of self-prospered reapers harvesting machine

Working width of the reaper machine 120 cm and length of the reaper machine are 200 cm and100 cm Height of the machine. the crop is conveyed vertically by finger conveyor to the binding mechanism and released to the ground in the form of bundles. During harvesting of the crop the cutting unit collects the harvest crop from on both left and right side and pushes towards the middle position of machine for dropping the bundles where the machine powered by 5 kW, 4 cylinder, water cooled diesel engine having rated engine speed of 970rpm. It has reciprocating cutter bar of cutting width about 120 cm which get rotary motion powered also from the diesel engine and this engine provided two driving wheels in the front pneumatic wheels. Other systems include brakes, steering, mechanism, and power transmission and an operator seat is available to make the machine riding type.

Table 6 Specification of this reaper harvesting machines

NO.	**Descriptions**	**Details**
1.	Name of the harvesting machine	reaper
2.	Working width	120cm
3.	Cutting bar width	110 cm
4.	Color of the machine	Light blue
5.	Height of the machine	125 cm
6.	Length of the machine	200cm
7.	Weight of the machine	250 kg
8.	Source of power	4 cylinder gas oil engine
9.	engine power	5 hp
10.	Source of manufacture	Local(Ethiopia, Addis Ababa university 5 kilo)
11.	Cost of equipment, Birr	25,000 birr
12.	Types of vehicle	Three wheel vehicle
13.	steer control	handle bar
14.	Position of steer	Back wheel steer
15.	Number of machine Operator	One man
16.	Speed of operation	2.54 m/s

4.2 Harvesting process in filed operation

The cutting unit of reaper harvesting machine is reciprocating cutter bar type. After cutting, the crop is conveyed vertically to the binding mechanism and released to the ground in the form of bundles.

4.3 Test procedure:

The testing of experimental self-propelled riding type vertical conveyor reaper was done.

4.3.1 Laboratory tests:

The main objectives of laboratory tests were to confirm the dimensions and to study the performance of the components designed. Such a study assisted the modifications and improvement in the design of machine.

4.3.2 Specifications:

Self-propelled riding type Reaper was kept on firm level and horizontal surface of the workshop floor and various dimensions were measured.

4.3.2.1 Visual observations and checking provision for adjustment:

The reaper was inspected thoroughly paying attention in particular to the power transmission components, moving parts, correctness and ease of operations along with various adjustments, tightness of bolts and nuts, etc.

4.3.2.2 Field test:

The observations on header loss and conveyance loss were recorded for constant speed of vertical conveyor reaper. Pre-harvesting loss from header and also conveyance loss was measured using 1 m x 1 m size area. The grains were collected from 1m^2 area. Time and diesel required for the operations were calculated with the help of stopwatch and measuring cylinder, respectively (Bukhari and Nawaz, 1991).

4.4 Machine performance

The machine performances for all different types under study were evaluated measuring machine capacity and the labor requirement.

4.5 Workability

The second parameter is workability, which is calculated measuring the two mean factors, mainly, the length of harvesting period and potential harvesting hours. The economic evolution is dun calculating fixed costs, operating costs and optimal harvesting capacity

4.6 Field capacity

Field capacity of a machine is the actual rate of land preparation/harvested or crop processing in a given time, based on total field time. In other words, effective field capacity of a machine is a function of the rated width of the machine. The percentage of rated width actually utilized the speed of travel and the amount of field time lost during the operation. In order to determine effective field capacity, the rated width of the implement (cutting width), speed of travel and field efficiency were measured. Field capacity and field efficiency were calculated using the following formula (Hunt, 1973):

During harvesting of the crop the cutting unit collects the harvest crop from on both left and right side and pushes towards the middle position of machine for dropping or released to the ground between six meter in the form of bundles it shows in figure 54-55. This Crop gathering mechanism is use to increased Field capacity and field efficiency.

Figure 54 binding mechanism

4.6.1 Theoretical field capacity:

The theoretical field capacity was also calculated as per- equation

$$T.F.C\left(\frac{ha}{hr}\right) = \frac{\mathrm{e} * \mathrm{v}}{10}$$

v is the working speed (km/h), 9.144km/h = 2.54 m/s

e is the effective working width (m), 1.2m

$$T.F.C\left(\frac{ha}{hr}\right) = \frac{1.2\ \mathrm{m} * 9.144 km/h}{10} = 1.1 ha/hr$$

4.6.2 Actual field capacity

The actual field capacity was also calculated as per equation

$$A.F.C\left(\frac{ha}{hr}\right) = \frac{\mathrm{b} * \mathrm{L}}{T_{C0} * 10{,}000}$$

Where; T_{C0} time to covering total area ,

$$T_{C0} = \big((p * n) + k + m + s + c\big)(1 + \mathrm{q})$$

$$T_{C0} = \big((0.656 * 83.33) + 20 + 30 + 30 + 30\big)(1 + 5\%)$$

$$T_{C0} = 3\ hr$$

b is the field width (m), 100 m

L is the field length (m), 100 m

p is the time for turning (min), $p = \frac{100\ m}{2.54 m\ /s} = 39.37\ s = 0.656\ min$

n is the number of turnings, $n = \frac{100\ m}{1.2 m} = 83.33$

Assume;

k is the turnings on treatment of headland (min), k= 20 min

S the time of adjustments, control, tending of machine, etc. (min), s = 30 min

m is the preparation for unloading (min), m = 30 min

c is the net unit of unloading time (min), and 30 min

$$A.F.C\left(\frac{ha}{hr}\right) = \frac{100 * 100}{3\ hr * 10{,}000}$$

$$A.F.C\left(\frac{ha}{hr}\right) = 0.33\ ha/hr$$

4.6.3 Field efficiency

The field efficiency is the ratio of actual field capacity (ha/h) to the theoretical field capacity (ha/h).

$$Field\ efficiency\ = \frac{A.F.C}{T.F.C}$$

$$Field\ efficiency\ = \frac{0.33}{1.09}$$

$$Field\ efficiency\ = 0.306$$

4.7 Energy requirements

The power consumption (Er, kW) for different systems were calculated according to the following equation (Hunt- 1995)

$$Er = Fc\left(\frac{1}{60 * 60}\right) P.F * L.V.C * 427 * \eta_{th} * \eta_m * \frac{1}{75} * \frac{1}{1.36}$$

$$Er = 18\left(\frac{1}{60 * 60}\right) 0.85 * 11000 * 427 * 35\% * 80\% * \frac{1}{75} * \frac{1}{1.36}$$

$$Er = 54.8\ kw$$

Fuel flow calculation:

engine: 5 kW diesel engine at full power

Assume fuel 40.6MJ/kg, thermal efficiency 0.42:

$$Fuel\ input\ power\ = \frac{5\ kw}{0.42} = \ 12kW$$

$$specific\ fuel\ consumption\ = \ 12(kJ/s)/40600(kJ/kg) \times \frac{3600\left(\frac{s}{h}\right)}{5(kW)} = 0.21\ kg/kWh$$

$$\text{fuel consumption } = \frac{engine\ power *\ specific\ fuel\ consumption}{\text{the fuel density}\frac{\text{kg}}{\text{L}}}$$

$$\text{fuel consumption } = \frac{5\ Kw *\ 0.21\ kg/kWh}{0.9\frac{\text{kg}}{\text{L}}}$$

$$\text{fuel consumption } = 1.17\ l/h$$

Fc = the fuel consumption L/h 18 km/h

P.F = the fuel density, kg/L (for solar = 0.9)

L.C.V. = the lowest calorific value of fuel (kcal/kg) average L.C.V of diesel is 11000 kcal/kg)

η_{th}= the thermal efficiency of the engine, (considered to be about 35% for diesel engine)

427 = thermo-mechanical equivalent, kg.m/kcal

η_m= the mechanical efficiency of the engine, (considered to be 80% for diesel engine)

So, the energy can be calculated as following:

$$productivity(kg/h) = Fe * u$$

$$productivity(kg/h) = 0.306 fed/h * 10{,}000 \text{ kg/fed}$$

$$productivity(kg/h) = 3058.1 kg/h$$

$$Energy\ requirement = \frac{engine\ power(kw)}{productivity(kg/h)}$$

$$Energy\ requirement = \frac{5\ kw}{110 kg/h} = 0.0454\ kwkg/h = 45.45\ wkg/h$$

Manual energy: the manual energy (EH) per feddan was calculated by considering the power of one labor about 0.1 hp, Chancellor (1981).

$$E_H = \frac{0.1 * 0.746 * N_L}{pa} = \frac{N_L}{pa} * 0.0746$$

Where:

0.1 = hp of agricultural labor (hp/man)

N_L = number of labor $N_L = 4$

Pa = actual productivity (fed/h) Pa = 0.418 ha/hr

0.746 = co-efficiently per conversion from hp to kW.

$$E_H = \frac{4}{0.418\ ha/hr} * 0.0746 = 0.71$$

4.8 Cutting efficiency

Pre-harvest losses were measured by placing a square frame (1 m x 1m) in the un-harvest area. All the grains lying on the ground were collected within the frame which used to remain below the cutter bar in operation. Usually, the pre-harvest loss is occurred where the grain is on the ground due to a result of wind shatter, lodging, down crop or weather conditions. A few gains were found within the square from in the field. Three random sampling were done and made average the numbers of the threshed grains found.

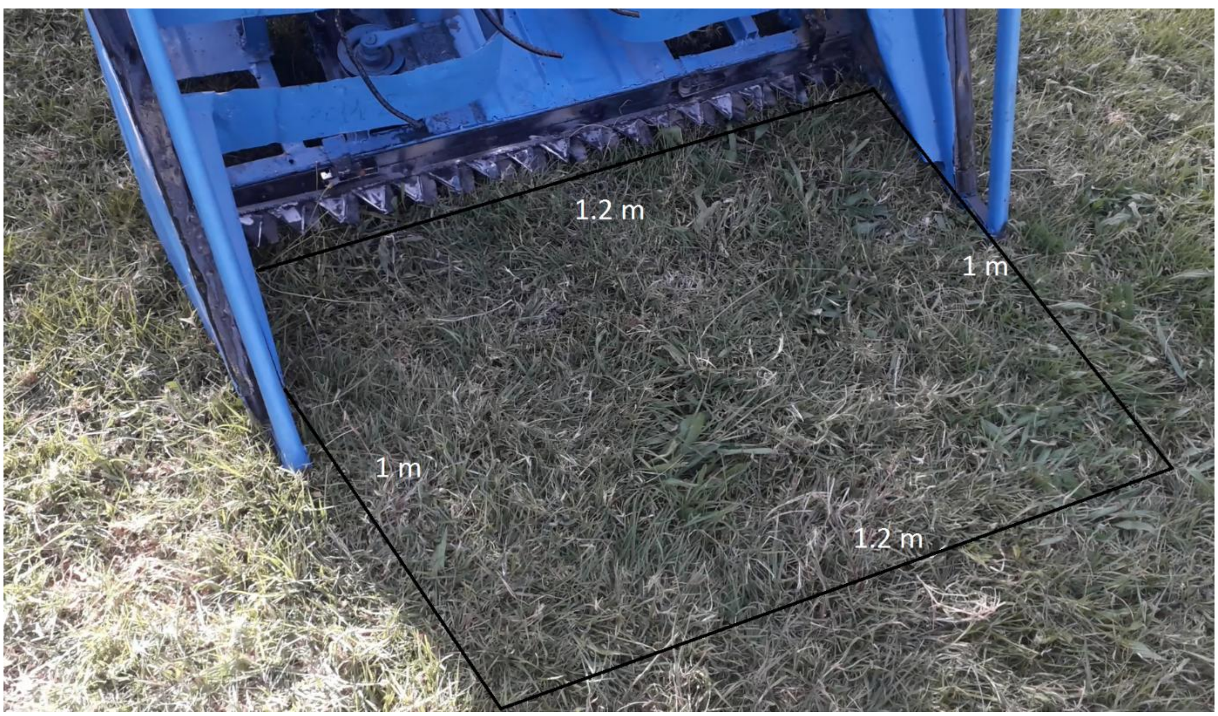


Figure 55 harvest area

The average grains loss per hectare was calculated. By using new modified cutter bar or knife

Figure 56 new modified cutter bar

CE_{wheat} = Cutting Efficiency, %

W_1 = Number of plants before cutting = 171

W_2= Number of plants left after cutting = 10

$$CE_{wheat} = \frac{171 - 10}{171} * 100 = 94.1\%$$

Cost Economics

The cost of mechanical harvesting can be calculated on basis of fuel required for the operation, labor charges, etc and was compared with the manual operations.

$$\text{Less Cost } = \frac{C_m - C_r}{C_M}$$

Where:
C_m = Harvesting cost by manual, K/day
C_r = Harvesting cost by reaper, K/day

Percentage of Grain Losses

Percentage of grain losses was obtained by the following equation.

Where,

$$p_{gl} = \frac{G_y - W_g}{G_y}$$

p_{gl} = Percentage of grain losses, %

N_g = Number of output grain, kg/ha

N_y= Number grain yield, kg/ha

No. of wheat Grains

N_g = No. of Grains/ear head * Plant population on 1 m * width of the machine (1.2 m)

$N_g = 39 * 171 = 6669$

N_y = No. of Grains/ear head * Plant population on 1 m * width of the machine (1.2 m)

$N_y = 39 * 161 = 6279$

Percentage of wheat grain losses

$$p_{gl} = \frac{6669 - 6279}{6669} * 100 = 0.585\%$$

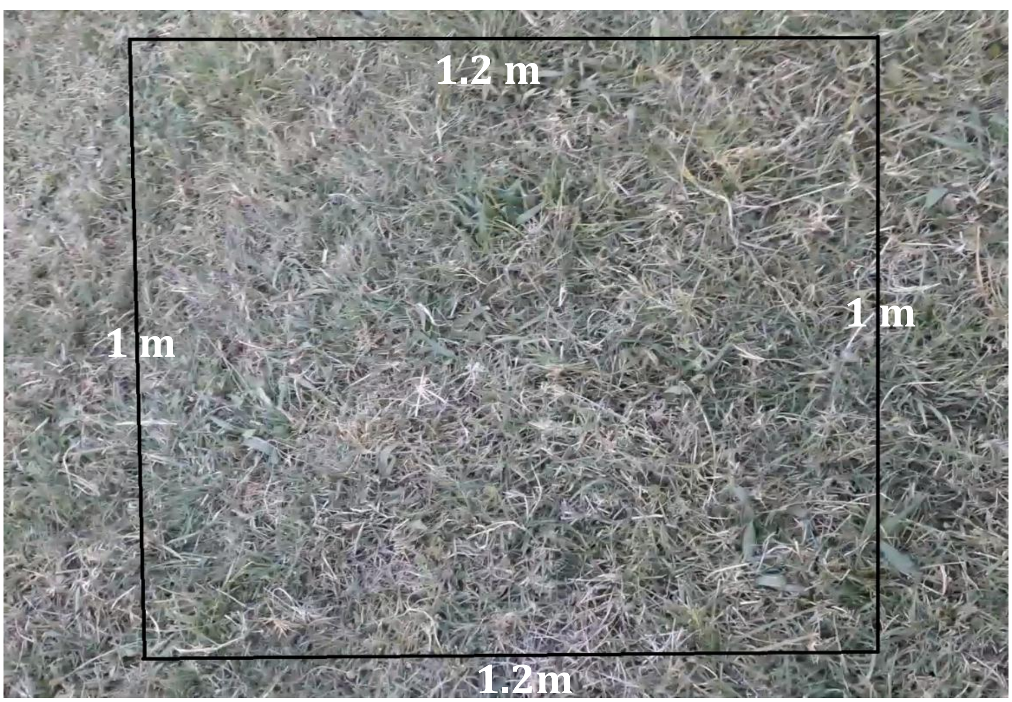


Figure 57 After cut the corps

4.9 self- prospered riding reapers machine efficiency with respect to other similar production

This new Reaper changed the way the world worked at the time; It created a leap in the Industrial Revolution as farming suddenly provided enough food to be exported worldwide as food was suddenly efficiently harvested. In the field operation of this new self- prospered riding reapers machine have: -

- The required labor for harvesting one hectare of wheat and rice field in present reaper machine was manually was 4 man-hr/ha compared to 2 man-hr/ha for this new reapers machine. The Mechanical Reaper enabled more production of food in a smaller amount of time. This means that manual labor was decreased, enabling former farmers and their families to pursue other careers and an education.
- easy to gathering cutting crops. During harvesting of the crop the cutting unit collects the harvest crop from on both left and right side and pushes towards the middle position of machine for dropping or released to the ground between in the form of bundles.
- Before operation it never needs to prepare the field by using traditional harvesting on the four sides of field and on four corners of field, this new reaper machine can be simply turn in field.
- During harvesting, the reaper machine easily changes its direction without stopping harvesting process.
- The farmers harvesting by riding the reaper machine without tiredness.
- Low cost machine compere to other reaper harvesting machine.

The average value of some of the parameters that include total operating time, total cutting area, forward speed, effective field capacity and field efficiency are shown in Table 5. The field capacity 0.24 ha/hr, respectively.

Table 7 Comparative performance characteristics of self-prospered reaper with this new reaper machine

No.	**Parameters**	**Harvesting Methods**	
		Previse machine	**New machine**
1.	Power source	5 HP diesel engine	5 HP diesel engine
2.	Date of harvesting	7-5-2014	9-15-2018
3.	Study area(m^2)	1	1
4.	Speed of operation(km/h)	2.18	9.144
5.	Width of operation(cm)	120	120
6.	Time required(h/ha)	4 h 10 min	1 h 30 min
7.	Theoretical field capacity(ha/h)	0.262	1.1
8.	Field efficiency (%)	0.24	0.306
9.	Labor requirement /day (including collecting and building)	4	2
10.	Fuel consumption(L/h)	1.89	1.17 l
11.	Crop gathering	one end and windrows the crops on the ground uniformly.	binding the crops and released to the ground between 6 m in the form of bundles.

CHAPTER FIVE

5 Conclusion and Recommendation

5.1 Conclusion

Among agriculture operations, harvesting is one the most tedious jobs and requires large numbers of laborers. self-propelled riding reapers machine can be use successful for quick and economical harvesting of wheat and rice crops, and to reduce the peaks labor shortage during the harvesting. The machine can be operated by single labor. Frame and other parts of the reaper machine are made angle iron, square bar, flat bar, pipe. The materials include standard parts and manufactured parts used in the above material the machine.

Working width of the reaper machine 120 cm and length of the reaper machine are 200 cm and125 cm length of the machine. The cut crop material was tied into bundles by its gathering mechanism and left on the field. During harvesting of the crop the cutting unit collects the harvest crop from on both left and right side and pushes towards the middle position of machine for dropping or released to the ground in the form of bundles the bundles where the machine powered by 5 kW, single cylinder, water cooled diesel engine having rated engine speed of 970rpm. This self-propelled reaper is operated at forward speed of 2.54 m/s. The field capacity 0.306 ha/hr, respectively. The required labor for harvesting one hectare of wheat and rice reaper harvesting field operation are needs 2 man-hr/ha. The quantity of fuel required to fill the tank fully after harvesting the plot was measured to determine the quantity of fuel consumed for reaping the test plot and fuel consumption 1.17 l/h. Before starting the harvesting operation in the test plot, the fuel tank of the reaper was filled up to its full.

5.2 Recommendation

By designing this machine that can have standard component and other small unknown issues that could only be found and fixed if a machine were built in manufacture company and further time was spent on design and testing. This was not accomplished due to restrictions in time, shortage of resources includes standard machine component and lack of manufacturing workshop. Ideally, other designers would look at this model and prototype determine a plan for the constructing of my design in manufacturing company level.

Budget Breakdown/details of expenditure:

Table 8 Material cost

NO.	COMPONENT/CATALOG NO. /	DESCRIPTION LENGTH (WITH IN M)	UNIT PRICE	QUANTITY	TOTAL PRICE
1	*5 HP diesel engine or Petrol engine*	*Engine*	*8,00birr*	*1 pic*	*7,000birr*
2	*1 thk. M.S. sheet metal*	*1 x 2 m*	*350birr*	*2 pic*	*350birr*
3	*42thk hard rubber*	*1 x 2.5 m*	*200 birr*	*1 pic*	*200 birr*
4	*10 thk. Hard rubber*	*0.15 x 1m*	*100birr*	*1 pic*	*100birr*
5	*30x30 square tubing*	*6 m*	*200 birr*	*2 pic*	*400birr*
	Angle bar				
6	*3.0 thk. X 30.0 x 30.0 angle bar*	*6m*	*350*	*2pic*	*350birr*
7	*4.0 thk. X 40.0 x 40.0 angle bar*	*6m*	*450 birr*	*2 pic*	*1000 birr*
	Round bar				
8	*5 Ø O.D. round bar*	*6m*	*150birr*	*1pic*	*150birr*
9	*20 Ø O.D. round bar*	*6m*	*450birr*	*1pic*	*450birr*
10	*30 Ø O.D. round bar*	*2.5m*	*500birr*	*1pic*	*500birr*
	Pipe	*Length*			
11	*19.05 Ø O.D. sch. 40 B.I. pipe is steel*	*6 m*	*200birr*	*1 pic*	*200birr*
12	*27 Ø O.D. sch. 40 B.I. pipe is steel*	*6 m*	*250birr*	*1 pic*	*250birr*
13	*33.4 Ø O.D. sch. 40 B.I. pipe is steel*	*6 m*	*350birr*	*1 pic*	*350birr*
	Timing chain	*Length*			
14	*Chain*	*1.3 m*	*400birr*	*2 pic*	*400birr*
15	*sprocket gear*	*180mm Ø*	*400birr*	*2 pic*	*400birr*
16	*sprocket gear*	*60mm Ø*	*400birr*	*2*	*400birr*
	V-belt and flat belt	*Length*			
17	*"B" section*	*1.2 m*	*120birr*	*2 pic*	*240birr*
18	*"B" section*	*1.5 m*	*130birr*	*2 pic*	*260birr*
	Bearing	*In mm*			

19	*Deep groove ball brg,#6203*	*(17x40x12)*	*50birr*	*6 pic*	*300birr*
20	*Taper roller bearing,#m88048-N*	*30x62x22*	*80birr*	*8 pic*	*640birr*
21	*Deep groove ball brg,#6206 RS*	*(30x62x16)*	*80birr*	*8 pic*	*640birr*
	BOLTS AND NUTS				
22	*Hex. Head bolt, with nut*	*M10Øx1.25x20.0 long*	*3birr*	*80 pic*	*240birr*
23	*Hex. Head bolt with nut and with lock washer*	*M22ØX3.25 x60long*	*12birr*	*6 pic*	*72birr*
24	*Flat(escrow) head bolt with nut*	*M8Øx1.25x25.0 long*	*2.5birr*	*70 pic*	*175birr*
25	*Flat washer*	*1.5 thk.x8.0I.D x16.0 O.D*	*0.5birr*	*100 pic*	*50birr*
26	*Hex. Head bolt with lock washer*	*M8Øx1.25x19.0 long*	*3bir*	*12 pic*	*36birr*
27	*Hex. Head bolt*	*M14Øx2.0x150long*	*5birr*	*6 pic*	*30bir*
28	*Hex. Head bolt*	*M12Øx1.75x50.0 long*	*10birr*	*8 pic*	*80birr*
29	*Flat washer,*	*1.5 thk.x10.0 I.D x20.0 O.D*	*0.5birr*	*12 pic*	*6birr*
30	*Key way c-45*	*10.0 mmx8.0 mm x1.2 m*	*120birr*	*1 pic*	*120birr*
	Pulley	*In mm*			
31	*Pulley cast iron*	*(75.0 ø x 45)*	*200birr*	*2 pic*	*400birr*
32	*V-pulley cast iron*	*(180.0x45)*	*350birr*	*2 pic*	*700birr*
33	*V- pully cast iron*	*(60.0øx45)*	*180birr*	*2 pic*	*360bir*
34	*Pneumatic tire 4x12 with curcke and kemendary*	*Motor cycle tire*	*600birr*	*3 pic*	*1,800birr*
35	*Painting*	*Painting spray white, dark red and black*	*100 birr*	*6 pic*	*600 birr*
36	*Breaking spare*	*motorcycle breaking parts*		*All part*	*1,000 birr*
37	*Additional material*				*1,000 birr*
	Sub total				*20,669birr*

Table 9 Equipment

N.O	Item	Quantity	Unit price	Total price(birr)
1.	Software	*Catia and ansys*	*200 birr*	200birr
	Sub Total			*300 birr*

Table 10 Consumables

N.O	Item	Quantity	Unit price	Total price(birr)
1.	*Printing papers (Ream)*	*1*	*250*1*	*250 birr*
3.	*Mobile card*	*2*	*100 *2 birr*	*200 birr*
	Sub Total			*450 birr*

Table 11Travel & Subsistence (Destination, No. of persons and days) Equipment

N.O	Item	Quantity	days	Unit price	Total price(birr)
1.	*Transportation To/adama metec (tractor)*	<<	5	*80 birr*	*400birr*
2.	*Transportation To/awash melkasa*	<<	5	*120 birr*	*600 birr*
	Sub Total				*1,000 birr*

Table 12 Personnel and other costs

N.O	Item	Quantity	days	Unit price	Total price(birr)
1.	labor	1 researchers	5	*100 birr*	500 birr
2.	welder	<<	4	*265 birr*	1060 birr
3.	*machine cost*	<<	2	*500birr*	*1000 birr*
	Sub Total				*2,560 birr*

Grand Total: Total = 24,979 birr

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Appendix

I. *Coefficient of friction between belt and pulley [R.S. KHURMI and J.K. GUPTA (2005)]*

Belt material	Pulley material						
	Cast iron, steel			Wood	Compressed paper	Leather face	Rubber face
	Dry	Wet	Greasy				
1. Leather oak tanned	0.25	0.2	0.15	0.3	0.33	0.38	0.40
2. Leather chrome tanned	0.35	0.32	0.22	0.4	0.45	0.48	0.50
3. Convass-stitched	0.20	0.15	0.12	0.23	0.25	0.27	0.30
4. Cotton woven	0.22	0.15	0.12	0.25	0.28	0.27	0.30
5. Rubber	0.30	0.18	—	0.32	0.35	0.40	0.42
6. Balata	0.32	0.20	—	0.35	0.38	0.40	0.42

II. *Physical properties of metals. [R.S. KHURMI and J.K. GUPTA (2005)]*

Metal	Density (kg/m^3)	Melting point (°C)	Thermal conductivity (W/m°C)	Coefficient of linear expansion at 20°C (μm/m/°C)
Aluminium	2700	660	220	23.0
Brass	8450	950	130	16.7
Bronze	8730	1040	67	17.3
Cast iron	7250	1300	54.5	9.0
Copper	8900	1083	393.5	16.7
Lead	11 400	327	33.5	29.1
Monel metal	8600	1350	25.2	14.0
Nickel	8900	1453	63.2	12.8
Silver	10 500	960	420	18.9
Steel	7850	1510	50.2	11.1
Tin	7400	232	67	21.4
Tungsten	19 300	3410	201	4.5
Zinc	7200	419	113	33.0
Cobalt	8850	1490	69.2	12.4
Molybdenum	10 200	2650	13	4.8
Vanadium	6000	1750	—	7.75

III. *Standard width of pulley [R.S. KHURMI and J.K. GUPTA (2005)]*

Belt width in mm	Width of pulley to be greater than belt width by (mm)
upto 125	13
125-250	25
250-375	38
475-500	50

IV. *Values of allowable shear stress, Modulus of elasticity and Modulus of rigidity for various spring materials. [R.S. KHURMI and J.K. GUPTA (2005)]*

Material	Allowable shear stress (τ) MPa			Modulus of rigidity (G) kN/m^2	Modulus of elasticity (E) kN/mm^2
	Severe service	Average service	Light service		
1. Carbon steel					
(a) Upto to 2.125 mm dia.	420	525	651	80	210
(b) 2.125 to 4.625 mm	385	483	595		
(c) 4.625 to 8.00 mm	336	420	525		
(d) 8.00 to 13.25 mm	294	364	455		
(e) 13.25 to 24.25 mm	252	315	392		
(f) 24.25 to 38.00 mm	224	280	350		
2. Music wire	392	490	612		
3. Oil tempered wire	336	420	525		
4. Hard-drawn spring wire	280	350	437.5		
5. Stainless-steel wire	280	350	437.5	70	196
6. Monel metal	196	245	306	44	105
7. Phosphor bronze	196	245	306	44	105
8. Brass	140	175	219	35	100

V. *Standard wire gauge (SWG) number and corresponding diameter of spring wire [R.S. KHURMI and J.K. GUPTA (2005)]*

SWG	Diameter (mm)	SWG	Diameter (mm)	SWG	Diameter (mm)	SWG	Diameter (mm)
7/0	12.70	7	4.470	20	0.914	33	0.2540
6/0	11.785	8	4.064	21	0.813	34	0.2337
5/0	10.973	9	3.658	22	0.711	35	0.2134
4/0	10.160	10	3.251	23	0.610	36	0.1930
3/0	9.490	11	2.946	24	0.559	37	0.1727
2/0	8.839	12	2.642	25	0.508	38	0.1524
0	8.229	13	2.337	26	0.457	39	0.1321
1	7.620	14	2.032	27	0.4166	40	0.1219
2	7.010	15	1.829	28	0.3759	41	0.1118
3	6.401	16	1.626	29	0.3454	42	0.1016
4	5.893	17	1.422	30	0.3150	43	0.0914
5	5.385	18	1.219	31	0.2946	44	0.0813
6	4.877	19	1.016	32	0.2743	45	0.0711