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CAPSTONE PROJECT

A STUDY ON DESIGN AND CONTROL THE SUGARCANE COVER PEELING MACHINE

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ABSTRACT

This capstone project presents a study on designing a sugarcane peeler machine. Firstly, some current designs in the world are analyzed. Second, a design of sugarcane peeler machine is proposed. Finally, the peeling process is simulated with force simulation to check for peeling efficiency.

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CHAPTER 1: OVERVIEW

1.1 Introduction on sugarcane

1.1.1 Definitions

Sugarcane is the common name of a number of species in the sugarcane genus (Saccharum), besides vegetables and spleen. They are perennial grasses, belonging to the tribe Andropogoneae of the family Poaceae. They have a fat, segmented body, containing a lot of sugar, from 2-6m high. All forms of sugarcane grown today are complex hybrids. They are grown for harvesting mainly to produce sugar and sell sugarcane juice as a beverage. The residue from sugar production has many other usages such as biofuel, paper production,…

Sugarcane is an important industrial crop of the sugar industry. Sugar is a necessary food in the daily meal structure of countries around the world, as well as an important raw material for many light industries and consumer goods.

In Vietnam, sugarcane juice has become a familiar beverage, you can see the image of sugarcane juice shops everywhere. Selling sugarcane juice requires little investment, less labor and fast capital recovery, so it is a viable investment opportunity. With a fairly compact working vehicle, including a sugarcane juicing machine and a number of tables and chairs, it is not difficult to see that shops selling sugarcane juice appear on the sidewalks across the routes from the cities and tourist areas…

Sugarcane juice has a lot of sugar plus water to help drinkers cool off on summer days. Sugarcane juice can be used to cure diseases and improve health.

1.1.2 Physical properties of sugarcane

Sugarcane main body includes: root, stalk, leaf, node. Each has their own function.

Sugarcane roots are also divided into three groups according to their physiological functions: absorptive roots, supporting roots and deep roots (water-absorbing roots).

Sugarcane stalks are the combination many segments (also known as internodes) together, bringing a variety of colors and shapes. There are types of sugarcane with green stalk, while others have yellow, dark red or purple ones. Regarding the shape of internodes, there are types of sugarcane with cylindrical stalk, others with stem-shaped (dented), drum-shaped (bulged), or truncated cone-shaped facing up or down. In addition, there are also sugarcane varieties with curved stalk. The length of each internode is from 15-20 cm, the stalk diameter in mature sugarcane is about 20 - 50 mm, on each internode there are cane eyes (buds), growth belt, root belt,…

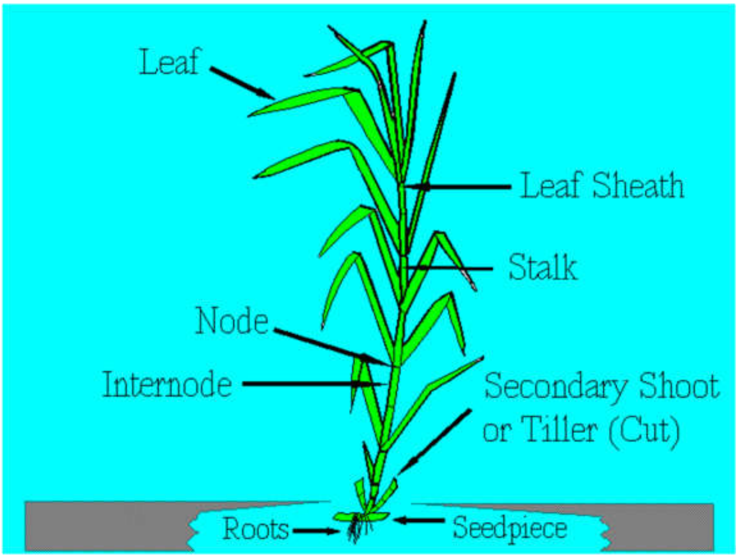

Fig 1.1 Sugarcane tree [1]

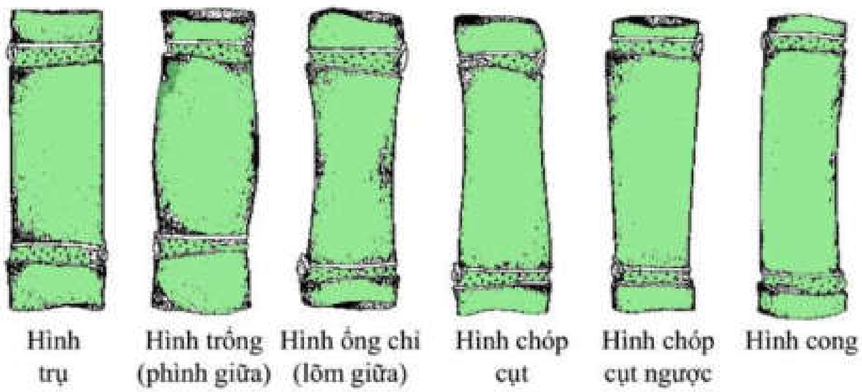

Fig 1.2 Sugarcane tree internode shape [1]

For production and processing industry, the stalk is the object of harvest, a place to store sugar, which is used as the main raw material for processing table sugar.

The leaf plays a crucial role in the growth of sugarcane. Respiration and photosynthesis are some of the leaves' functions.

The average height of the sugarcane plant is 2-3 m, some varieties can be 4-5 m tall. The diameter of sugarcane is about 20-50 mm.

1.1.3 Benefits of sugarcane to human

First, the benefits of sugarcane come from the nutrition content of sugarcane. There are fibers, sugar, water, minerals and important micronutrients such as calci, chromium, magnesium, manganese, phosphorus, potassium, zinc,…and also vitamin such as A, C, B1, B2, B3, B5 and B6 and protein, antioxidant,…All of which is good for the health of human in moderate amount.

1.1.4 Economic impact in Vietnam

Sugarcane is a backbone in sugar industry in Vietnam. It is ingredients for food and agriculture industries, which produces consumer goods for human daily necessities. The production of sugar has opened a lot of work opportunities for people.

The process of growing sugarcane is a meticulous one and require a lot of steps. From seeding, nuturing in the farm to the processing in factory. The factory will mill the sugarcane to extract the juice, then filtered, centrifugal screening to get the final sugar product. However, the technology and equipments in Vietnam is outdated, leading to high production cost.

Currently, sugar factories in Vietnam produce 1.5 million tons of sugar per year and ranked 4th in ASEAN in terms of output. But high production costs hindered Vietnam competition in pricing both inside and outside the country, because of cheaper foreign sugarcane. Many farmers have abandoned sugarcane and switched to other crops.

Reality has proven that in areas specializing in sugarcane cultivation where the government and businesses have methodically invested in seeds, fertilizers, cultivation and harvesting, sugarcane productivity has increased to 80-90 tons/ha, even reaching

100-120 tons/ha. At the same time, the application of automation in farming has helped 30-40% of costs and increased proactiveness in harvesting.

Thus, the development and application of advanced technology and technical processes in sugarcane growing has brought positive results, improving productivity and reducing costs for farmers, contributing to enhancing competitiveness of the sugar industry in Vietnam agriculture.

1.1.5 Sample sugarcane processing method and product

Processing fresh sugarcane into a commodity product is currently a new direction, still developing in the sugarcane industry in our country. To ensure quality, as well as strict requirements from foreign markets, research and development of sugarcane processing method is essential. For example, Truc Lam Phat Foods company sells frozen sugarcane that is ready for eating or pressing for juice. Their process is as follows:

1. Sorting: Sugarcane after being bought to the factory will be sorted to ensure that the quality of sugarcane is always okay for producing.
2. Peeling: The sugarcane is put into the peeler to completely remove the outer stalk layer.
3. Cutting: Sugarcane will be cut into pieces according to the requirements of customers. Normally, the size of sugarcane is cut from 40 to 50 cm/piece.
4. Washing: After being cut, sugarcane is put through a sink with ozone system to disinfect the surface of sugarcane. Washing time from 15-20 minutes
5. Drying: After being picked out, the cane will be put on a drying rack to dry, ensuring that they are completely dry before packing.
6. Packing: Cane after cutting is packed according to each required specification. They are packed in PE bags and 5-layer cartons.
7. Freezing: Sugarcane is frozen at a temperature of -21 degrees Celsius in 48 hours, with this temperature ensuring the center of the product reaches -18 degrees Celsius.

1.1.6 Challenge in sugarcane peeling

To get sugarcane juice, we have to go through the process of scraping sugarcane stalk. Although this stage is not difficult, it is laborous and time consuming. Normally now in the city, sugarcane juice business addresses will buy sugarcane that has been peeled from sugarcane granaries. The large sugarcane granaries operate at full capacity but still cannot meet the high demand of customers. The reason for this problem is that most of the sugarcane is peeled by traditional methods such as: pulling, hand scraping. These methods consume a large amount of time and labor, so the design and manufacture of a sugarcane peeler will meet the needs of supplying sugarcane consumption as well as solving labor costs.

1.2 Sugarcane Peeling/cutting process

1.2.1 Manual

Scraping method:

The worker uses a semicircular knife to shave, although it does not take much effort, but brings very low productivity, about 3 minutes to shave off a 2-3m long cane. This leads to the need for a large number of workers to meet the amount of sugarcane for the market.

Advantages:

- Most economical method

Disadvantages:

- Require constantly high manual effort
- Lowest productivity
- Inconsistent peel because the motion of cane and peel is managed by human.

Pulling method:

This method consists of using a metal tube with 4 peelers attached on it by bolted joints. Workers will move the cane forward and backward to peel it off. Lower manual efforts than peeling method but still consumes a lot of energy, at the same time, it is difficult to deal with curvy sugar cane. Although it requires less strength than hand peeler, it still only reaches 2-3 minutes for one tree. Requires workers to be in good health and in large numbers.

Fig 1.3 Manual hand peeler [2]

Fig 1.4 Peeling tube with 4 peelers [3]

Advantages:

- Simplify the motion of peeling
- Peel simultaneously on almost all sides of the cane so it's consistent

Disadvantages:

- Require constant human effort
- Low productivity

1.2.2 Semi-automatic

Wire brush peeler:

Machine consists of a rotating wire brush and a passive roller. The wire brush will peel either parallel or perpendicular to the sugarcane feeding inside. The movement of the sugarcane would still be handled manually.

With this method, the cane scraper consumes little effort, needs to move the cane back and forth to scrape off the peel. Productivity increased significantly compared to the above 2 methods (1 minute for 1 tree).

Wire brush is available in stainless steel or brass.

Parameters	**Value**
Weight	45 kg
Power	750 W
Compatible diameter of cane	2-6 cm
Dimension (Length-Width-Height)	51 x 60 x 105 cm
Productivity	1 cane/ minute

Table 1.1 Wire brush peeler specification

Fig 1.5 Wire brush peeler (Parallel peeling) [2]

Figure 1.6 Wire brush peeler (Perpendicular peeling) [2]

Figure 1.7 Wire brush [2]

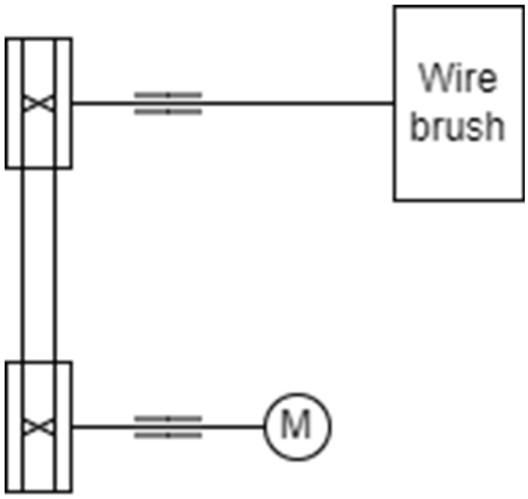

Figure 1.8 Kinematic diagram of wirebrush peeler

Advantages:

- Automated the motion of peeler, the workers only need to move and rotate the cane
- Reduce workers' effort

Disadvantages:

- Not fully automated, workers have to move the cane.
- Higher cost than the manual method

Reciprocal blade peeler

Manually feed the sugarcane into the machine. A worker will hold it and push it inside. There are 2 unidirectional mechanisms that help preventing the sugarcane from going backwards in the machine. Then it will go to peeling section with reciprocating blades frame. The motion will also pull the sugarcane inside. These groups of blades are mounted on the frame and at an angle to each other. The blades frame will periodically reciprocate, which makes the cane peeled almost wholy after going through the machine only once.

The blades in this machine usually arranged $45° - 60°$ to the feeding path of the cane, which can help prolong the blades life while peeling more easily even with hard and uneven rind.

The machine can peel variable diameter of sugarcane thanks to the spring installed for each blade.

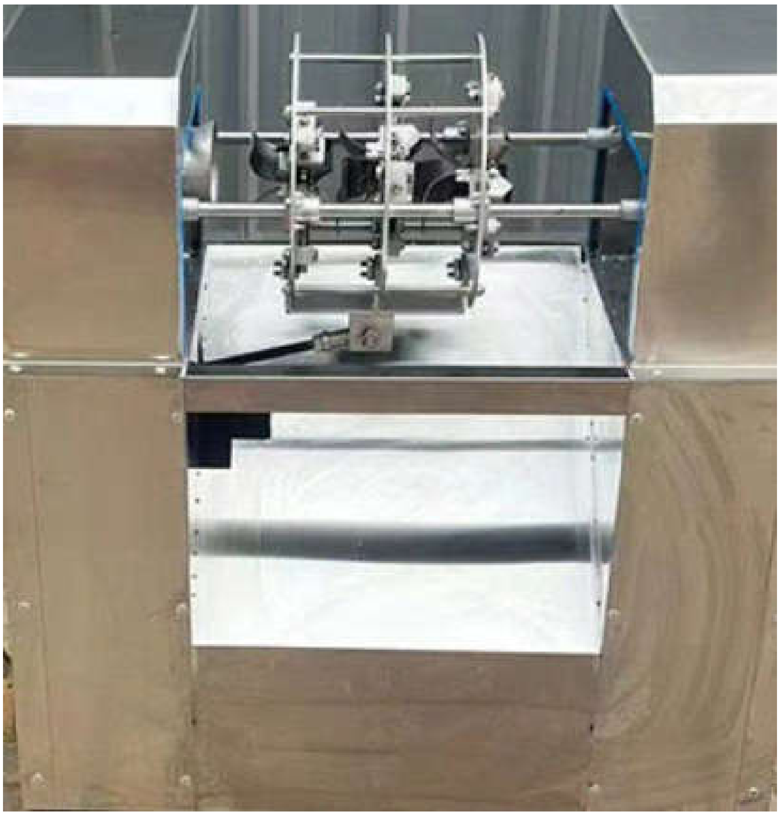

Fig 1.9 Xinjiate sugarcane peeler frame machine [4]

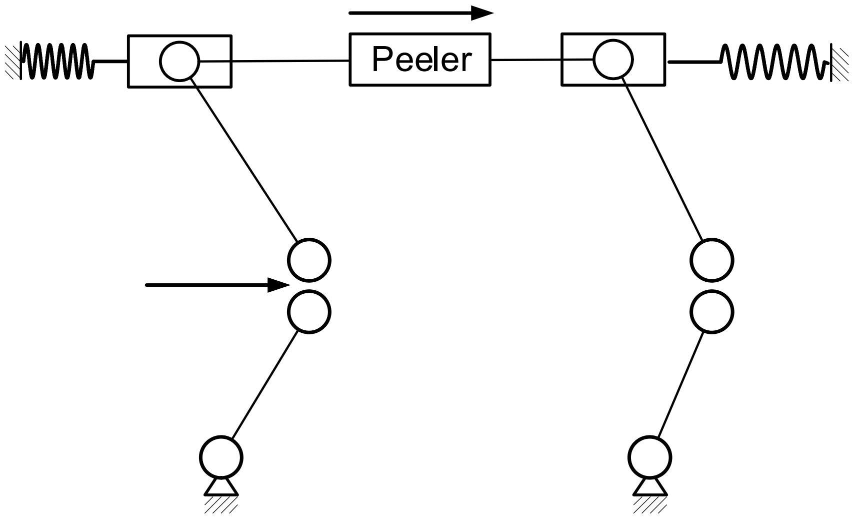

Fig 1.10 Xinjiate sugarcane peeler frame machine kinematic diagram

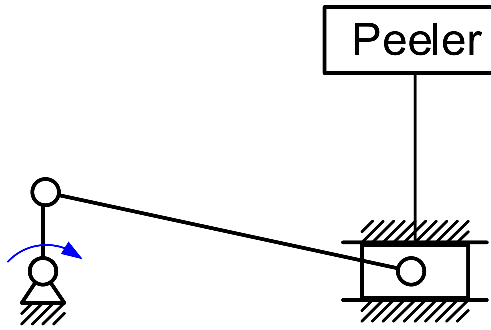

Fig 1.11 Xinjiate sugarcane peeler frame machine kinematic diagram front view

Parameters	Value
Weight	100 kg
Power	500W
Dimension (Length-Width-Height)	100 x 50 x 90 cm
Productivity	4 cane/ minute

Table 1.2 Xinjiate sugarcane peeler specifications

Advantages:

- Less human effort than wire brush peeler because the gripping mechanism prevents the cane from moving back
- Peel consistent in one run
- Only use one motor for both peeling and pulling

Disadvantages:

- High initial cost and maintenance cost
- Large area needed to operate because bagasse may spread long
- May not cover the whole circumference of the cane with few blades.

<u>Reciprocal turning blade peeler:</u>

Manually feed the sugarcane into the machine. A set of rollers will hold it and pull it inside. Then it will go to peeling section with reciprocating blades frame. These pairs of blades are mounted on the frame by a rubber band and at an angle to each other, when the cane is scraped, the blades frame will rotate evenly to help the blade contact all the outer shell, thereby helping to peel off the cane shell. The blades frame will periodically reciprocate, which makes the cane peeled wholely after going through the machine only once.

The blades in this machine usually arranged perpendicular to the feeding path of the cane, which can damage the blades early on and not peeling fully if the rind is too hard or uneven surface.

The machine can peel variable diameter of sugarcane thanks to the rubber bands that connect the couples of blades to the frame to hold it together.

Advantages:

- Less human effort than wire brush peeler because the roller feeds the cane inside
- Peel better, consistent in one run-through
- The rotation of the blade frame ensures bagasse does not stuck
- Able to peel curvy cane

Disadvantages:

- Highest initial cost
- Large area needed to operate
- Daily maintenance is needed

Figure 1.12 TanPhat’s Automatic sugarcane peeler [2]

Parameters	Value
Weight	230 kg
Power	1500W+100W+200W
Compatible diameter of cane	2-6 cm
Dimension (Length-Width-Height)	172 x 40 x 110 cm
Productivity	2 cane/ minute

Table 1.3 TanPhatManufacture’s machine specification

There are variations of this type of machine, which can supports at most 3 canes feeding in as once, tripling the productivity.

Example: Peelmax 2, Peelmax 3.

The 6 canes variation also exists. However the more sugarcane it can peel, the more human effort is needed at the feeding process.

Figure 1.13 Peelmax 2 and Peelmax 3 [5]

Type	Hand peel	Tube peel	Wire brush	Reciprocating blades (1 feeder)	Reciprocating turning blades (1 feeder)
Time to peel/ 3m cane	3 minutes	2 minutes	1 minutes	15 seconds	30 seconds
Productivity	20 trees/hour	30 trees/hour	60 trees/hour	240 trees/hour	120 trees/hour

Table 1.4 Productivity comparison of methods on 2m cane [6]

1.3 Literature study

For the reciprocating fixed frame peeling machine, Tian Shunxia [7] developed a gripping mechanism that works in harmony with the peeling and pulling process by using slidercrank mechanism to open and close the gripper.

For the peeling blade, Guan Yonggang [8] summarized that the optimal angle between the blade and sugarcane feeding direction is $45 - 60°$ for the productivity and longevity of blades service time. He also drafted a blade holder design for reciprocating peeling method.

1.4 Safety standards in processing equipments in the food industry

Equipment and tools that come into direct contact with food must be designed, manufactured and ensured to be safe, made from materials that do not contaminate food, meet production technology requirements, and are easy to clean, disinfect and maintain. Mobile production equipment and tools must be durable, easy to move, disassemble and clean.

Regarding food production equipment and tools must be sufficient and suitable for processing raw materials, food processing and packaging, are made of nontoxic materials, less prone to wear and corrosion, not contaminated with harmful substances in food, do not create strange odors or alter food, do not contaminate food with lubricating oil or metal fragments.

1.5 Problem statement

Through the process of studying machinery and technological processes for peeling sugarcane stalk around the world and in our country, even though the automation method has already been available, there are still many limitations in costs and the dimension of the machines is not suitable for the small-scale sugarcane granary. In Vietnam, many people choose to peel manually or go for an inexpensive tool to peel sugarcane.

For the above reasons, the objective of this topic is to propose a design of the sugarcane peeling machine for sugarcane preprocessed products which will be distributed to juice shops, capable of peeling it with higher productivity and high adaptability to various diameters, based on the available mechanisms and existing models on the market.

With reference to article [9], in Hue, a granary can sell up to 2000 trees on average a day, both peeled and unpeeled.

Assume that the opening hour occur in the morning when juice shop starts the preparation, and close in at night when there are no more customers. Let it be from 6AM-10PM, which means 16 hours of operating a day. On average, they have to peel 63 sugarcane trees per hour.

Therefore, we have the technical requirement for machine:

- Productivity: 240 trees/hour
- Able to peel straight tree
- Length of cane: $2 - 3m$
- Maximum diameter of cane: $50mm$
- Reduce the initial costs.
- The design ensures compliance with food safety and hygiene standards according to the Food safety regulations.
- The machine can be applied to an industrial production line to make sugarcane products.
- The machine's materials must withstand the cleaning chemicals daily because sugarcane juice and bagasse may stuck on machine.
- The machine size, electrical components, and controls should be simple, easy to use, and convenient for operators. It should have the ability to quickly stop the machine in case of any incidents or malfunctions.

CHAPTER 2: SELECTIONS OF METHOD

2.1 Sugarcane peeling principle selection

2.1.1 Technical requirements

- Productivity: 240 trees/hour
- Able to peel straight tree
- Reduce workers effort
- Reduce the initial costs

2.1.2 Selection

With the above requirements, we have two possible peeling methods: Peeling with reciprocating motion and peeling with rotational motion.

Reciprocating blades motion: The method of cutting an object with the knife moves straight is a simple cutting method in which the material is held stationary while the blade moves straight through the material to perform the cutting process. The operating principle of this method is based on the following basic components:

- Hold the material tight: The material needs to be placed in a device that holds the material firmly during the peeling period.
- Straight moving blade: The blade is usually attached to a peeling unit, controlled by mechanical or electronic systems. It moves straight on the direction of material.
- Peeling force and speed: Fore and speed need to be adjusted appropriately to ensure efficient and accurate removal of materials.

Rotational blades motion (wire brush): The method of peeling an object with the knife rotating following basic principles:

- Hold the material tight: The material needs to be placed in a device that holds the material firmly during the peeling period.
- Rotating blade: The wire brush is attached to a shaft, controlled by mechanical or electronic systems. It rotates perpendicular to the direction of the sugarcane.

- Peeling force and speed: Force and speed need to be adjusted appropriately to ensure efficient and accurate removal of materials.

	Wire brush machine	**Reciprocating blades machine**
Productivity	60 trees/hour	120-240 trees/hour
Workers' effort	• Continuous moving forward and backward the cane to peel it	• Only need to feed the cane into the machine
Advantages	• Lowest initial cost option	• Low human effort • Peeler blade reach almost all of the cane
Disadvantages	• Workers have to move the cane • Inconsistent peel if lacking human attention	• Bagasse form when peeling may stuck. • The blade frame have to be designed to remove the bagasse.
Initial cost	Low (6,000,000 VND)	Medium-High (10,000,000-27,000,000 VND)

Table 2.1 Peeling mechanism comparison

Conclusion: Select the reciprocating blades principle because it can meet the requirement producitivity and reduce human efforts.

2.2 Feeding method selection

Although the reciprocating turning blades principle machine currently requires 1 worker to feed it, another possible method is to use a belt conveyor to feed it in, so the workers don't always have to aim and put it in the machine hole.

	Manual feeding by worker	Conveyor feeding system
Advantages	Low space occupied Low initial cost	Conveyor will transport the cane into the machine.. The mechanism which feeds the conveyor have to be designed to be compatible and reduce manual work.
Disadvantages	Requires intermittent feeding. Requires more human efforts.	More maintenance is required More complex design Spaces is required

Table 2.2 Feeding method comparison

Conclusion: Select the manual feeding method for simplicity in design.

2.3 Selection of peeling

The blade frame consists of many peelers assembled on a single frame to enable itself to peel the whole area of the sugarcane. Currently there are two types, both utilised the slidercrank mechanism to move reciprocatingly forward and backward.

Curved frame

Fig 2.1 Curved frame peeler [6]

This is the current most popular type in Vietnam, it facilitates 20 peelers blade on the frame at all the angle of a circle to peel. 20 blades as 10 couples of blades, each couple will peel the cane perpendicularly.

The frame is connected to a hollow shaft through which the cane pass through. The frame is rotated by the shaft, and reciprocate by a platform on a guiding rail with slidercrank mechanism.

The sugarcane is peeled in returning stroke (opposite direction of sugarcane feeding in). The speed of feeding and reciprocating motion affect the peeling speed.

Advantages:

- Peel consistently and all around
- Does not leave bagasse on the peelers, lower the chance of being stuck

Drawbacks:

- Take up a long space to arrange the peelers.
- Usage of rubber bands to tighten the couples of blades.
- Blades susceptible to failure fast because perpendicular peeling
- Daily maintenance
- Highest cost in the market

<u>Fixed circular frame</u>

This is the current most popular type in China. The frame move only in translational direction and does not rotate, which will reduce the usage of 1 more motor or a transmission system. The blade in this frame is arranged at a certain angle between $45° - 60°$ to the sugarcane [8], which will help prolong the working hour of the blades compared to the perpendicular blade arrangement in the curved frame above.

The frame is connected to a guiding rail which supports the linear motion. The frame connect to a slider of a slidercrank mechanism.

The sugarcane is peeled in returning stroke (opposite direction of sugarcane feeding in). The speed of feeding and reciprocating motion affect the peeling speed.

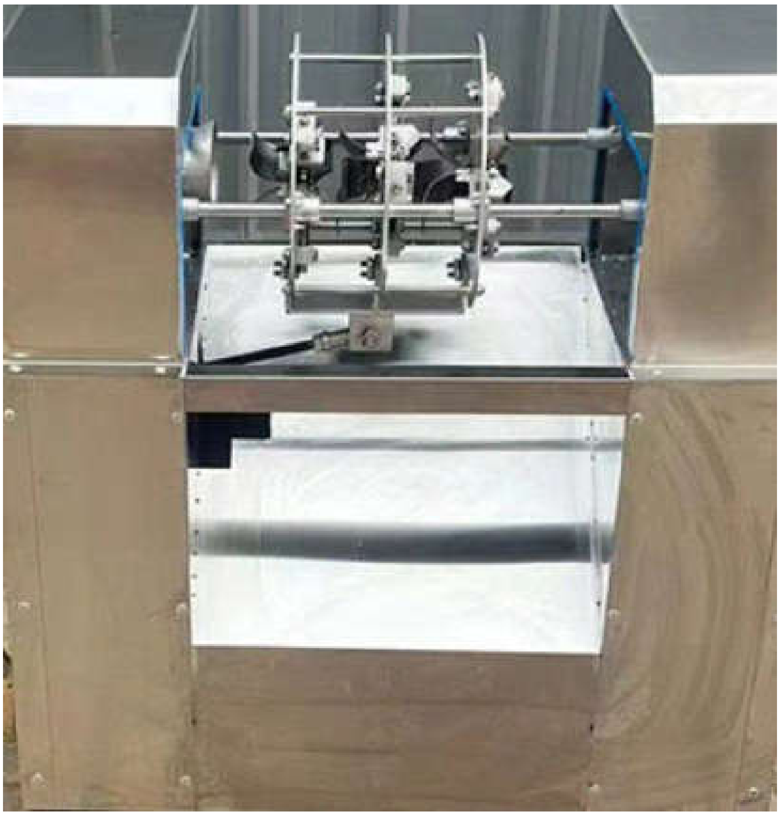

Fig 2.2 Xinjiate sugarcane peeler frame machine [4]

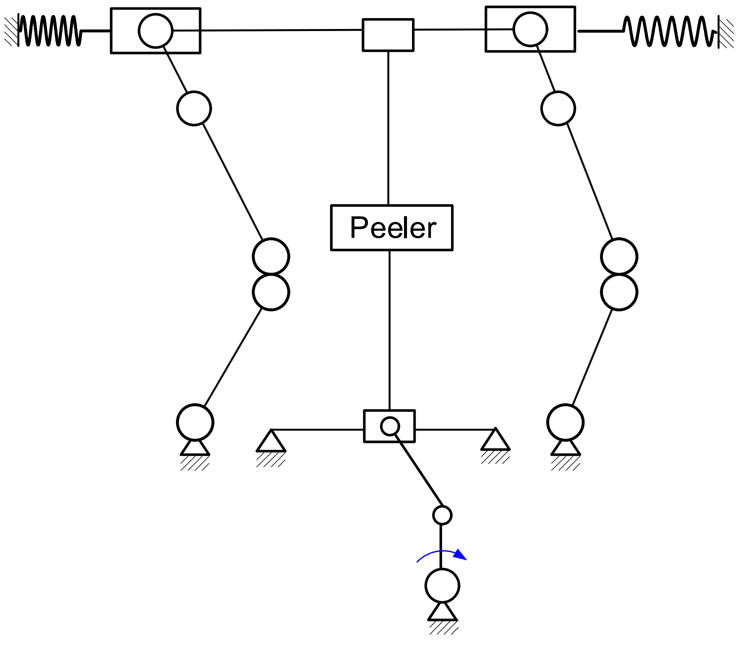

Fig 2.3 Xinjiate sugarcane peeler frame machine kinematic diagram [4]

Advantages:

- Peel consistently and all around
- Durable blades frame because it is fixed on a plate with spring
- Use one fewer motor to operate than rotating frame
- Fastest peeling method
- Assistance to pull the cane

Drawbacks:

- Bagasse stuck out when peeling
- Daily maintenance

Conclusion: Choose the fixed circular frame for because of higher productivity.

2.4 Selection of pulling feeding configuration

The pulling mechanism to help feeding sugarcane inside the machine must alleviate the human effort and stabilize the sugarcane to not let it crushed and to not let it pushed back because of the reciprocating peeling motion. There are 3 options: rollers with spring feeder, one-way mechanism using gripper, one-way rollers.

Fig 2.4 Xinjiate sugarcane peeler gripping mechanism [4]

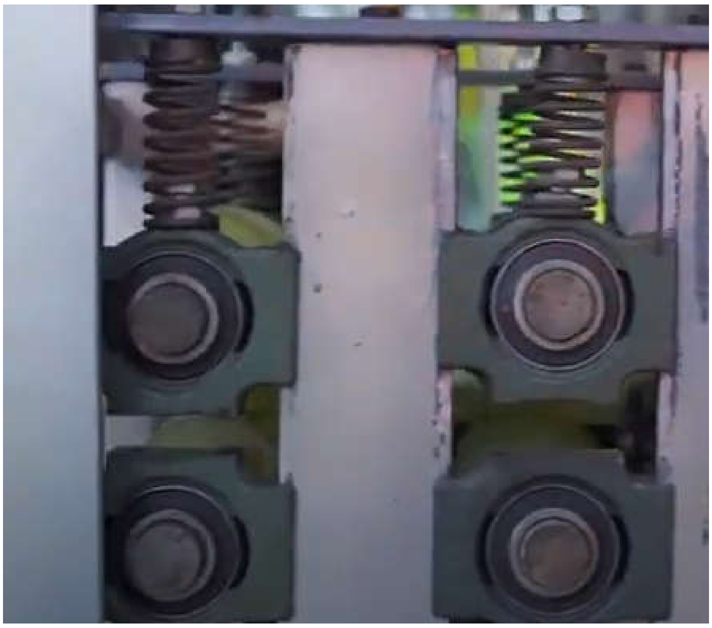

Fig 2.5 Rollers with spring feeding [2]

	Advantages	**Disadvantages**
Rollers with spring	• Automatically pull the sugarcane at contact • Reduce human effort	• Need a large transmission system • Low speed • Highest cost
One-way mechanism with gripper	• Reduce human effort • Simple mechanism • Use friction to hold the cane tight, so it can hold very tight.	• Require more force than rollers • Noisy because of surface contact. • Need conveyor at the ends to transport sugarcane outside
One-way rollers	• Reduce human effort • Simple mechanism • Use one-way bearing to prevent pull back • Use less force than gripper	• Need conveyor at the ends to transport sugarcane outside

Table 2.3 Pulling feeding mechanism comparison

Conclusion: We use one-way rollers configuration, because it will hold the cane tight during feeding, peeling process, while help ease up the manual feeding. This method will also require conveyor at the outfeed to transport the sugarcane outside.

2.5 Select actuators

From the peeling method choice above, the actuators should be able to move the peeling platform at desired speed and exert force sufficiently. Moreover, it should prove to safe to food, save space and require less maintenance. Possible actuators are listed below.

	Advantages	Disadvantages
Electric motor with slidercrank mechanism	• High force and high speed achievable • Clean, safe for food • Take up least space • Used in current machine	• Need to design the slidercrank mechanism
Pneumatic cylinder	• Simple design • High force and high speed achievable • Clean, safe for food	• Require a lot of equipments to work with, take up lots of space. • Noisy operation
Hydraulic cylinder	• Simple design • High force and high speed achievable	• Require a lot of equipments to work with, take up lots of space. • Leakage of oil during use • Noisy operation

Table 2.4 Comparsion of actuators

Conclusion: Choose electric motor with slidercrank mechanism as actuators because it will not spoil the food and take the least space.

2.6 Select transmission system

Requirements:

The blades frame, its chain drive and its motor will be placed on a linearly reciprocating platform. The motion will help peeling along the cane length. The platform will be heavy. The speed is not too fast. Observation from demo video [10] show that the speed is $0.6\ m/s$

We estimate that the weight of the blade frame will be less than a half of the whole machine. Using [4]'s machine weight, we deduce that the frame weights about 20kg.

To move such large weights, the transmission system must incorporate a gearbox. So there are four possible configurations: belt – gear – chain, belt – gear, gear- chain, gear only.

Transmission configuration	Advantages	Disadvantages
Belt – gear – chain	• Can transmit power between distant shafts • Transmit high torque • Work well with high speed input	• Short life span • Need a belt tensioner • Make system larger • Need lubricants, may not be safe for food
Belt – gear	• Can transmit power between distant shafts • Work well with high speed input • Transmit high torque	• Short life span • Make system larger • Need a belt tensioner
Gear only	• Transmit high torque • High efficiency	• Noisy operation • Limited distance of transmission

Table 2.5 Comparsion of various transmission configurations

Conclusion: We will use a motor with gear reducer only to move such a large weight because we are limited in space and do not want lubricants to contaminate the food.

2.7 Motor type selection

The selected motor type must be compact, suitable for limited installation space and is easy to replace.

	Advantages	**Disadvantages**
DC motor	• Simple speed control module • Diversity of the motor parameters	• Using DC voltage, which required switching supply • High maintenance cost
AC 1 phase motor	• Smooth, quiet operation • Simple installation, maintenance	• Low efficiency • High power consumption • Limited power range • High cost of speed control module
AC 3 phase motor	• Large power range • Simple installation	• Loud noise • High cost of speed control module

Table 2.6 Comparsion of motor types

Conclusion: Choose 3 phase AC motor, because it satisfies the requirement and is also generally available in industry environment.

2.8 Control algorithm and hardware

Requirements:

- Each module operates at its own speed
- Each module in the machine will normally run at a specific speed, so the motors would stay at that speed most of the time.
- Reduce cost

- Hardware is reliable in a continuous usage, hazard environment

We will use ON/OFF control algorithm.

For ON/OFF principle we have the following suitable hardware selections:

	Advantages	**Disadvantages**
PLC	• Stable operation • Ease of maintenance • Extendable with modules • Flexible control ability	• High cost
Relay – contactor	• Stable, simple operation for industrial use • Low cost	• Difficult wiring • Limited extensibility
Microcontroller	• Easy to repair and replace • Low cost • Flexible control ability	• Unstable long-term operation

Table 2.7 Control hardware selections

Conclusion: Choose relay-contactor because the machine operates continuously, simple and at a specific speed.

2.9 Summary of selection

Requirements	**Selection**
Peeling principle	Reciprocating fixed blade
Feeding method	Manual feed with one-way rollers
Control method	ON/OFF

Control hardware	Relay – contactor
Transmision system for reciprocating platform	Gearbox integrated in motor for slider crank mechanism

Table 2.8 Summary of selections

<u>Mechanical kinematic diagram</u>

The kinematic diagrams for the machine with peeling principle using reciprocating fixed blade, combined with feeding section utilizing one-way rollers. In the diagram below, blue arrow is the rotation direction of motor, green arrow is the allowed direction of one-way bearing, red arrow is the locked direction.

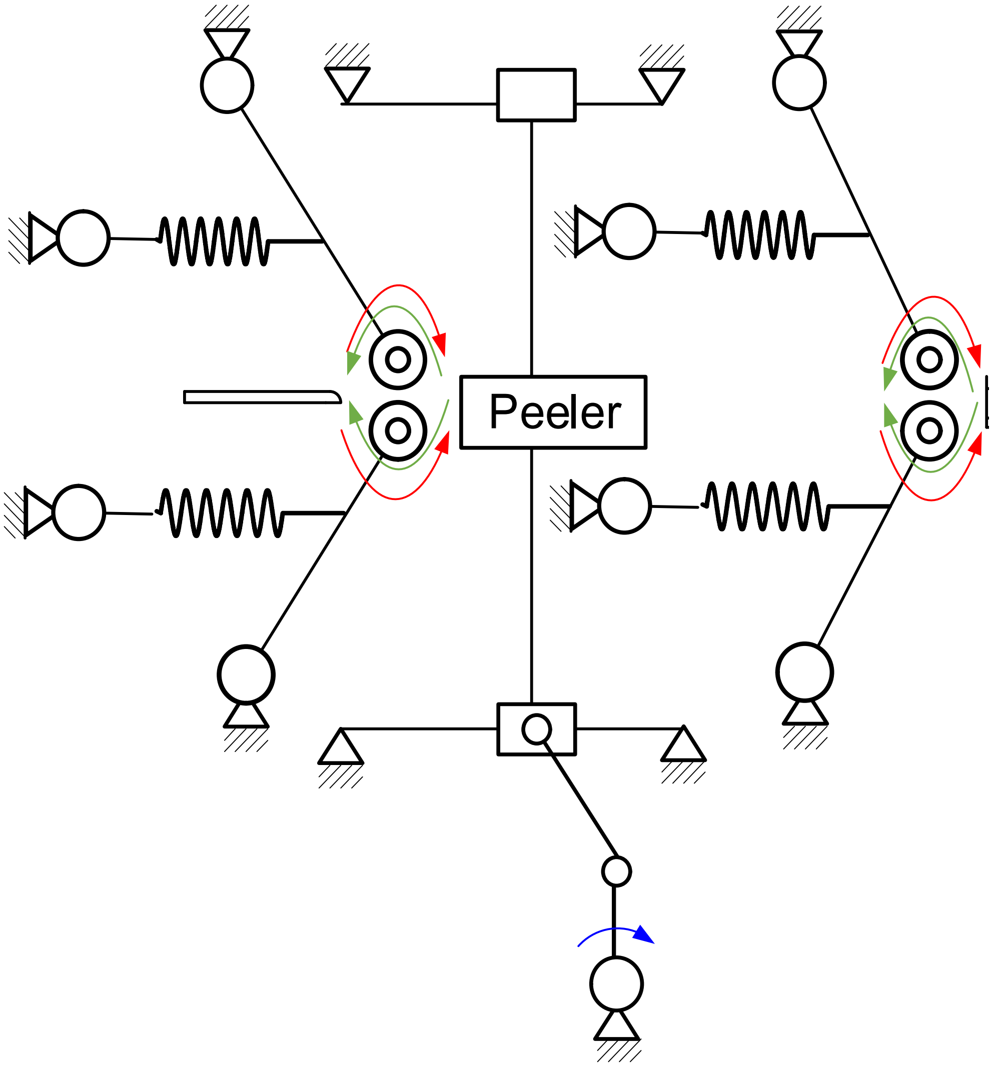

Fig 2.6 Kinematic diagram of the machine

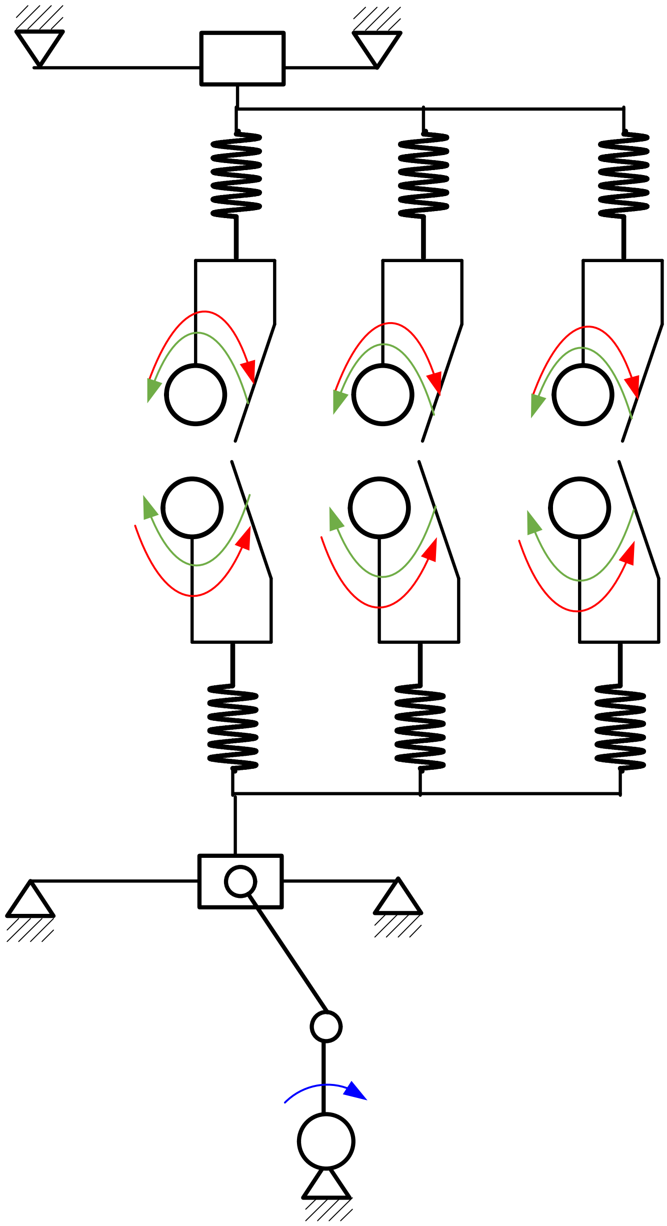

Fig 2.7 Kinematic diagram of the peeler module

<u>Electrical selection and algorithm flowchart</u>

Use 3 phases 380VAC sources to power to run motor. System has 3 states: ON, OFF, Emergency shutdown and 2 lights for indicating ON, OFF states. Use buttons to change between stages through relay – contactor control. The power circuit will include protective equipment such as circuit breaker, overload relay, fuse.

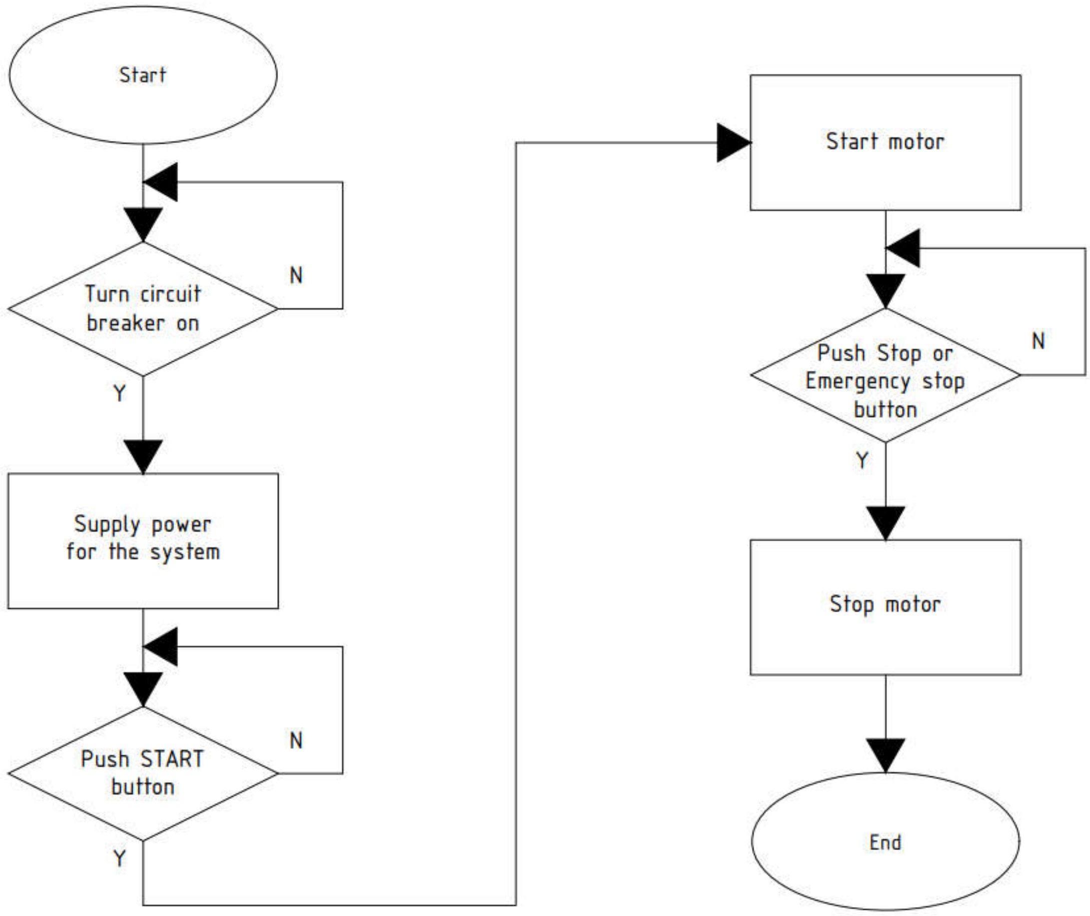

Fig 2.8 Algorithm preliminary flowchart

CHAPTER 3: MECHANICAL DESIGN

3.1 Reciprocating peeling section

3.1.1 Calculate peeling force of blades

We will experiment to get the peeling force F_p, the peeling area to get the stress needed to tear off the stalk. Set up the vice clamp to hold the sugarcane straight on the loadcell. Then proceed to press down the knife until the sugarcane stalk comes off.

The stress formula [11]:

$$\sigma_p = \frac{F_p}{A_{knife}} \tag{3.1}$$

Where σ_p is the stress needed to peel (MPa). F_p is peeling force that can be measured (N).

A_{knife} is the knife section that contact the sugarcane, calculated through formula: $A_{knife} = 0.4 \times 2\sqrt{2 \times R \times d - d^2}\ (mm^2)$, where d is depth of peel that can be measured and R is the sugarcane diameter, in this case it is $15mm$.

The measurement shall be taken 30 times to be elligible for center limit theorem, which stated that 30 is the lowest number of samples where this theorem could apply. And that means the sample distribution follows approximately normal distribution. This is to help us find the largest possible force needed to peel.

The experiment is setup as below figure 3.1, where $\overrightarrow{F_p}$ is peeling force, $\vec{F}$ is clamping force.

Set up the experiment with a sugarcane specimen.

Tools and devices: Vice, knife, loadcell, HX711 module, a microcontroller

Collect the data about the force need to peel off.

Then measure the depth of the stalk that was cut off using a vernier caliper. From that we deduce the area that was pressured.

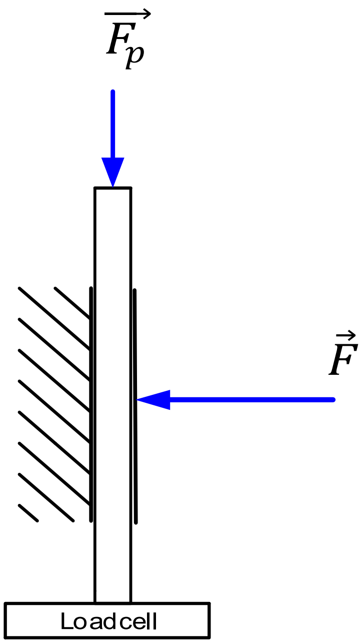

Fig 3.1 The experiment setup to find sugarcane peeling force

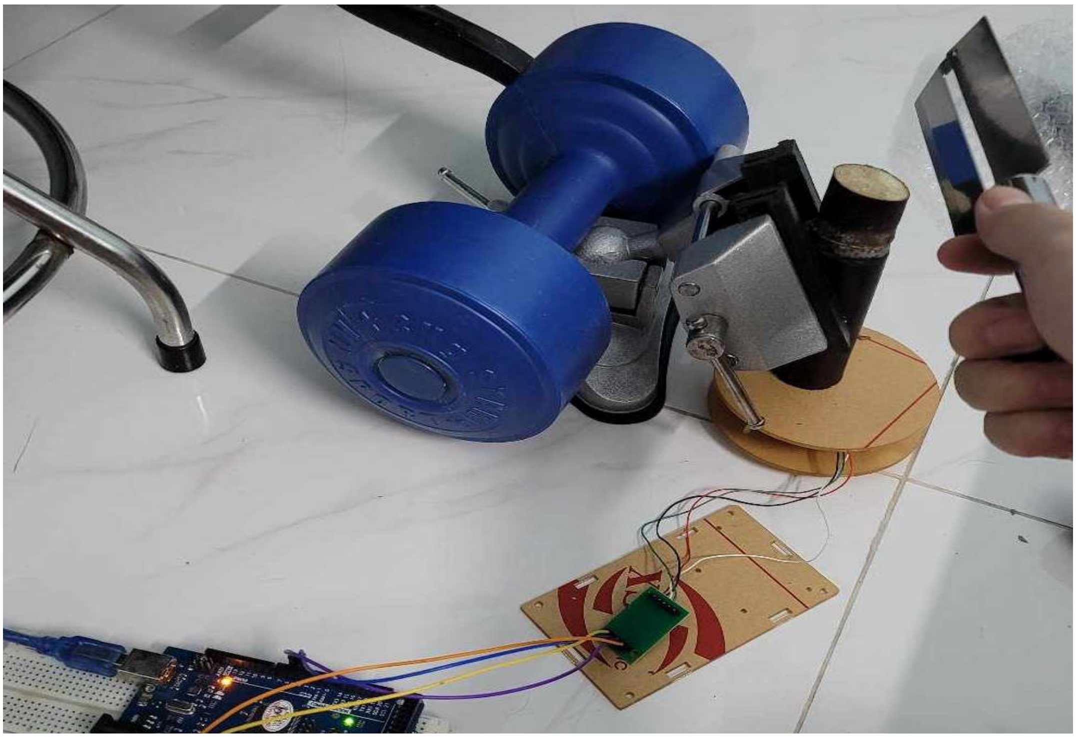

Fig 3.2 Experiment setup in reality

Fig 3.3 Measure the depth of the peeling.

Experiment number	Depth of peel (mm)	Weight ($gram$)	Calculated section area of knife (mm^2)	Calculated stress (MPa)
1	2.8	5813.98	3.490787877	8.327604262
2	1.92	4609.15	2.93703524	7.846603162
3	1.6	3730.78	2.696367927	6.918158242
4	3.1	6039.26	3.65272501	8.266787102
5	1.78	4403.63	2.834970194	7.766624865
6	1.56	3597.41	2.664324305	6.751073797
7	2.4	5542.26	3.255518392	8.512100582
8	1.8	4531.18	2.849842101	7.949879045
9	2.22	5464.65	3.141250706	8.698207356
10	1.7	3893.35	2.774454901	7.016423295
11	2.1	5036.46	3.061764197	8.224767937
12	2	4850.34	2.993325909	8.101924326

13	2.7	5625.54	3.434181125	8.190511501
14	1.3	2779.67	2.443276489	5.688406558
15	2.72	5737.56	3.445614024	8.32588903
16	1.76	4305.94	2.819997163	7.63465307
17	1.38	2916.07	2.513820996	5.800074875
18	1.7	4195.62	2.774454901	7.561160929
19	2.1	5094.37	3.061764197	8.319337597
20	2	4660.67	2.993325909	7.785102827
21	1.7	3971.94	2.774454901	7.158054719
22	2.22	5321.01	3.141250706	8.46957231
23	2.14	5120.27	3.088569896	8.289062856
24	2.24	5526.43	3.154232712	8.760339685
25	1.8	4502.23	2.849842101	7.899086757
26	1.74	4246.55	2.804921389	7.569819989
27	2	4717.91	2.993325909	7.880715536
28	3.14	6067.09	3.673481183	8.257957096
29	1.4	3389.53	2.531086723	6.69579981
30	2.2	5281.94	3.128194367	8.442474124

Table 3.1 Experimental result

We have mean stress is $\sigma_{p\ average} = 7.784\ (MPa)$ and standard deviation $SD = 0.773\ (MPa)$. We apply the normal distribution properties that is empirical rule: 99.7% of values lies within 3 standard deviations of the mean. Therefore, the stress that is needed for peeling sugarcane will have 99.7% chance lower than:

$$\sigma_{p\ 99.7} = \sigma_{p\ average} + 3SD = 7.784 + 3 \times 0.773 = 10.103\ (MPa) \quad (3.2)$$

Hypothetical section area that will be peel for sugarcane with diameter $50mm$

$$A_{50} = 0.4 \times 2 \times \sqrt{2 \times R \times d - d^2} = 0.4 \times 2 \times \sqrt{2 \times \frac{50}{2} \times 1 - 1^2} = 5.6\ (mm^2) \quad (3.3)$$

Then the highest force needed to peel sugarcane of diameter $50mm$ at $1mm$ depth is

$$F_p = \sigma_{p\ 99.7} \times A_{50} = 10.103 \times 5.6 = 56.5771\ (N) \quad (3.4)$$

Total force needed:

$$F_{p\ total} = 9 \times F_p = 9 \times 56.5771 = 509.1939\ (N) \quad (3.5)$$

3.1.2 Calculate the slider crank mechanism power

According to a demo video, a similar peeler module run in 1 second, peel 0.3 m and pull the cane 0.3 m. Which means in 1 second the peeler has run through 1 period, travelled 0.6 m. We need at least 4 trees per minutes, which converts to about 0.2 m peeled per seconds. The speed from this machine is faster than our requirement productivity, so we will calculate with the higher speed.

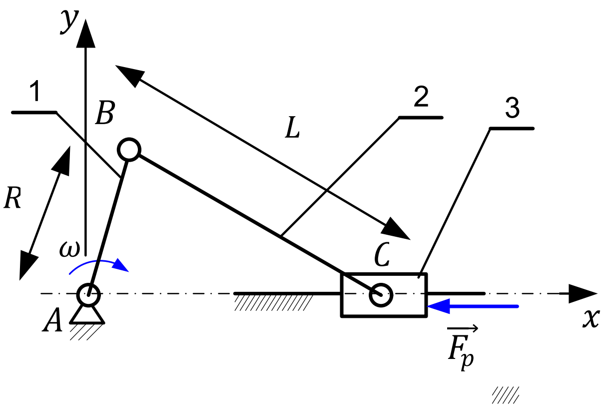

Fig 3.4 Slidercrank mechanism diagram

For the slidercrank, it means that the stroke of the slider will be $S = 0.3\ (m)$, and then the radius of the of the crank $R = \frac{0.3}{2} = 0.15\ (m)$

The period of the peeler

$$T = \frac{1}{1} = 1\ (s) \tag{3.6}$$

Angular velocity of peeler, which is also equal to angular velocity at the slidercrank shaft

$$\omega = \frac{2\pi}{T} = \frac{2\pi}{1} = 2\pi\ (rad/s) \tag{3.7}$$

<u>Position, velocity, acceleration analysis:</u>

Consider figure 3.5, let i be the unit vector for x-axis, while y be unit vector for y-axis. Point A is selected as the origin of the xy-frame. AB is link 1, BC is link 2, slider C is link 3.

We know $AB = 0.15\ m$. Let $BC = 0.4\ m$

The weight of link 1,2,3 have to be determined for simulation later. Using Solidworks mass properties tools, we find that $m_1 = 2\ kg, m_2 = 5\ kg, m_3 = 2\ kg$

Position vectors for point B, C:

$$r_B = x_B i + y_B j = ABcos(\alpha)i + ABsin(\alpha)j \tag{3.8}$$

$$r_c = x_c i + y_c j = \left(ABcos(\alpha) + \sqrt{BC^2 - AB^2 \sin^2 \alpha} \right) i \tag{3.9}$$

Velocity vectors for point B, C:

$$v_B = \omega AB(-sin\alpha)i + \omega ABcos(\alpha)j \tag{3.10}$$

$$v_C = -\omega ABsin(\alpha)i - \frac{\omega AB^2 \times sin\alpha cos\alpha}{\sqrt{BC^2 - AB^2 \sin^2 \alpha}} i \tag{3.11}$$

Acceleration vectors for point B, C

$$a_B = \omega^2 AB(-cos\alpha)i - \omega^2 ABsin(\alpha)j \tag{3.12}$$

$$a_C = \left(-\omega^2 ABcos(\alpha) - \frac{\omega^2 AB^2 cos2\alpha}{\sqrt{BC^2 - AB^2 \sin^2 \alpha}} - \frac{\omega^2 AB^4 \sin^2 x \cos^2 x}{\sqrt{BC^2 - AB^2 \sin^2 \alpha}^3} \right) i \tag{3.13}$$

Now we will solve the forces from link 3 to get the torque on the link 1 to calculate the motor power. The force on diagram may be negative, which means they go the opposite direction from the diagram.

<u>Dynamic force analysis on link 3:</u>

The center of gravity for link 3 is point C.

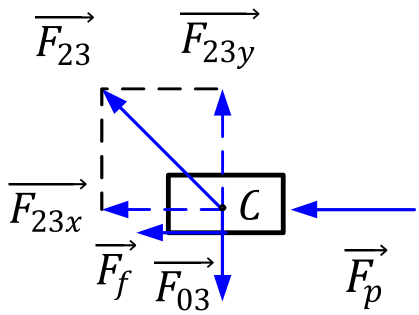

Fig 3.5 Link 3 force analysis

Net force according to Newton Second Law:

$$m_3 \vec{a}_{C3} = \vec{F}_{23} + \vec{F}_{03} + \vec{F}_f + \vec{F}_{ext} + \vec{G}_3 \tag{3.14}$$

Where F_f is friction force on the sliding rail of discs, G_3 is weight of link 3.

Projecting the equation on x and y axes gives

$$m_3 a_{C3x} = F_{23x} + F_{ext} - F_f \tag{3.15}$$

$$m_3 a_{C3y} = F_{23y} - F_{03} - G_3 \rightarrow F_{23y} = F_{03} + G_3 \left(because\ a_{C3y} = 0\right) \qquad (3.16)$$

While we also have:

$$F_f = P_{peeling\ platform} \times \mu_f = 20 \times 9.81 \times 0.006 = 1.1772\ (N)$$

$$\mu_f = 0.006$$

$$F_{ext} = 509.1939\ N$$

Dynamic force analysis on link 2:

The center of gravity for link 2 is

$$r_{C2} = \frac{r_B + r_C}{2} = \frac{2ABcos(\alpha) + \sqrt{BC^2 - AB^2 \sin^2 \alpha}}{2} i + \frac{ABsin(\alpha)}{2} j \qquad (3.17)$$

The acceleration vector for point C_2

$$a_{C2} = \frac{a_B + a_C}{2}$$

$$= -\omega^2 \left(ABcos(\alpha) + \frac{AB^2 cos2\alpha}{\sqrt{BC^2 - AB^2 \sin^2 \alpha}} + \frac{AB^4 \sin^2 x \cos^2 x}{\sqrt{BC^2 - AB^2 \sin^2 \alpha}^3} \right) i - \frac{\omega^2 ABsin(\alpha)}{2} j \qquad (3.18)$$

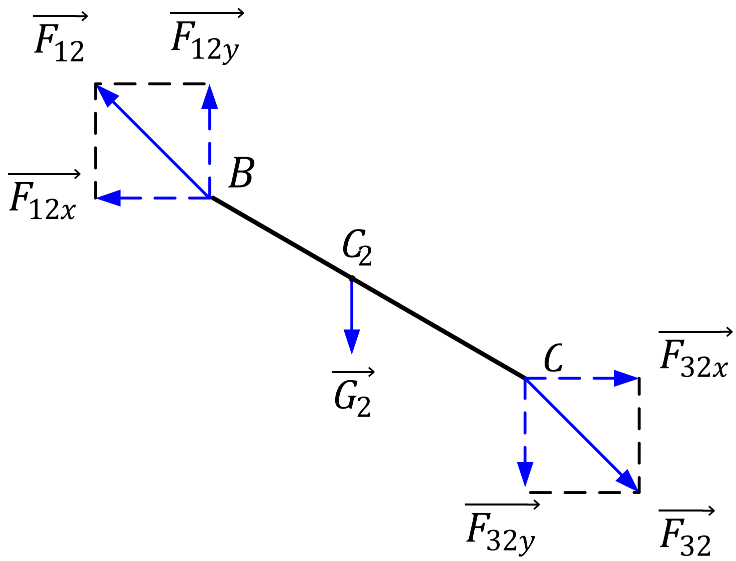

Fig 3.6 Link 2 force analysis

Force equilibrium equations:

$$m_2 \vec{a}_{C2} = \vec{F}_{12} + \vec{F}_{32} + \vec{G}_2 \qquad (3.19)$$

Projecting the equations onto x and y axes give:

$$m_2 a_{C2x} = F_{32x} + F_{12x}$$

$$\rightarrow -m_2\omega^2\left(ABcos(\alpha) + \frac{AB^2 cos2\alpha}{\sqrt{BC^2 - AB^2\sin^2\alpha}} + \frac{AB^4\sin^2 x\cos^2 x}{\sqrt{BC^2 - AB^2\sin^2\alpha}^3}\right) = F_{32x} + F_{12x} \quad (3.20)$$

$$m_2 a_{C2y} = F_{32y} + F_{12y} - G_2 \rightarrow -m_2\omega^2\frac{ABsin(\alpha)}{2} = F_{32y} + F_{12y} - G_2 \quad (3.21)$$

The moment equation concerning C_2

$$0 = (x_B - x_{C2})F_{12y} - (y_B - y_{C2})F_{12x} + (x_C - x_{C2})F_{32y} - (y_C - y_{C2})F_{32x} \quad (3.22)$$

<u>Dynamic force analysis on link 1:</u>

The center of gravity for link 1 is

$$r_{C1} = \frac{r_B}{2} = \frac{ABcos(\alpha)}{2}i + \frac{ABsin(\alpha)}{2}j \quad (3.23)$$

The acceleration vector for point C_1

$$a_{C1} = \frac{a_B}{2} = \frac{\omega^2 AB(-cos\alpha)}{2}i - \frac{\omega^2 ABsin(\alpha)}{2}j \quad (3.24)$$

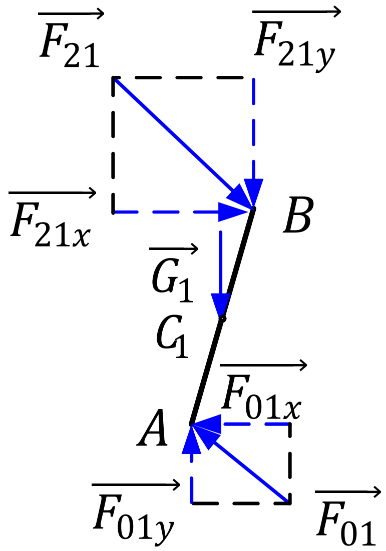

Fig 3.7 Link 1 force analysis

Force equilibrium equations:

$$m_1\vec{a}_{C1} = \vec{F}_{21} + \vec{F}_{01} + \vec{G}_1 \quad (3.25)$$

Projecting the equations onto x and y axes give:

$$m_1 a_{C1x} = F_{21x} + F_{01x} \rightarrow m_1\frac{\omega^2 AB(-cos\alpha)}{2} = F_{21x} + F_{01x} \quad (3.26)$$

$$m_1 a_{C1y} = F_{21y} - G_1 + F_{01y} \rightarrow m_1 - \frac{\omega^2 ABsin(\alpha)}{2} = F_{21y} - G_1 + F_{01y} \quad (3.28)$$

The moment equation concerning C_1

$$0 = (x_B - x_{C1})F_{21y} - (y_B - y_{C1})F_{21x} + (x_A - x_{C1})F_{01y} - (y_A - y_{C1})F_{01x} + M (3.29)$$

From reference [12], use MATLAB to solve the system of equations, thereby obtaining the value of moment M based on the angle alpha ranging from 0 to 360, the figure of moment M is shown below:

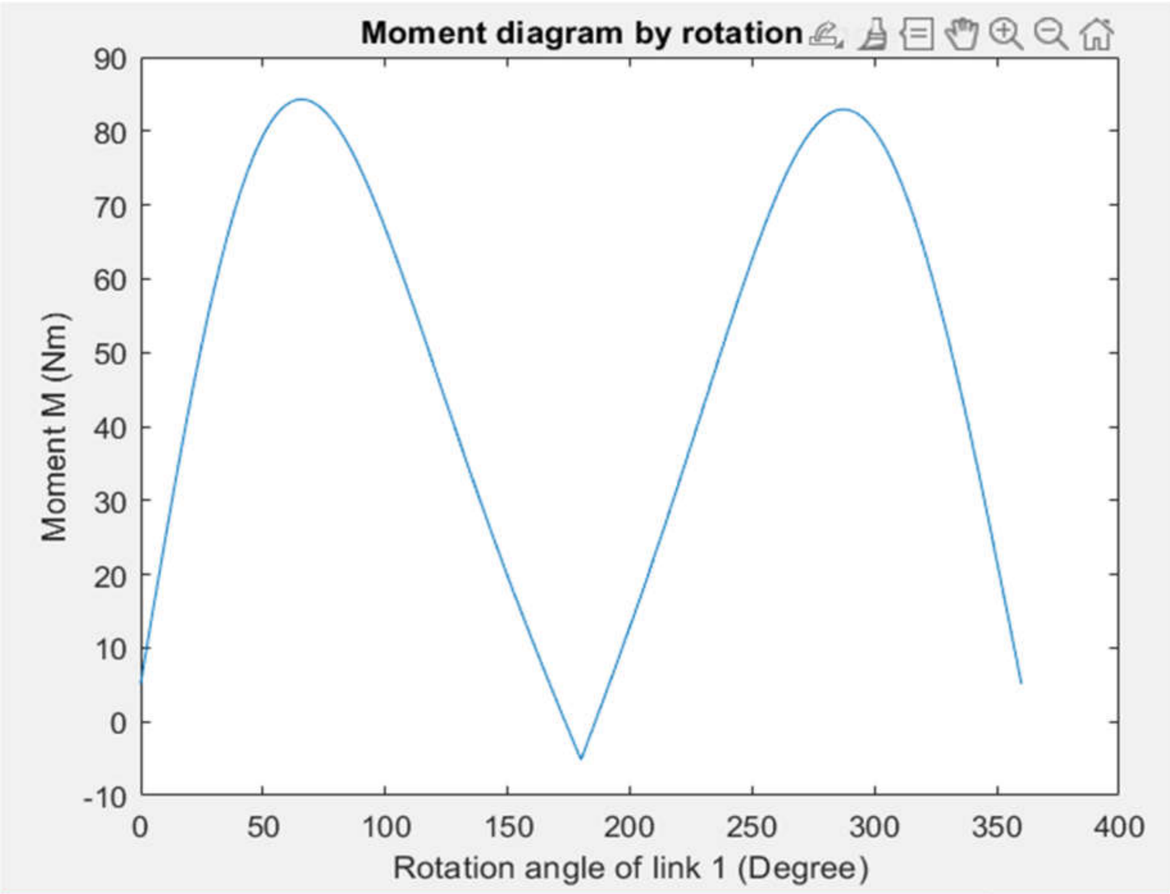

Fig 3.8 Moment diagram by rotation angle of link 1

From figure, the maximum torque is 84.2838 Nm.

Power at the actuating shaft, with safety factor $[s] = 1.3$ is:

$$P_{slidercrank} = [s]T\omega = 1.3 \times 84.2838 \times 2\pi = 688.442 \text{ (W)} \quad (3.30)$$

Choose H2L32M25-WB08TAVEN 3 phase AC geared motor with below specification

Parameter	Value
Power	$745\ W$
Rated current	$1.4\ A$
Gear ratio	25
Output shaft speed	$68\ rpm$
Output shaft diameter	$28.575\ mm$
Rated torque	$89.6\ Nm$

Table 3.2 Specification of 3 phase AC motor 745W with right gearbox

3.1.3 Calculate the peeling blade and frame geometric parameters

The overall design of the peeling blade and spring to comply the reciprocating mechanism. The blade should be place at an angle of 60 degrees to maximize the efficiency of peeling based on patent [8], the bladeholder will have some crucial parts to calculate such as the blade, spring.

The geometry of the peeling blade is similar to guillotine blade which is commonly used in food industry. Following [8], we use a peeler platform consisting of 3 blades per disc for a total of 3 discs. First, we decide the dimension of the blade. The arrangement of 3 blades on 1 disc forms an equilateral triangle. Then the sugarcane must be barely inside that triangle. Then the blade length has to be:

$$l \leq 2\sqrt{3}R_{sugarcane} = 2\sqrt{3} \times 15 = 51.9615\ (mm) \tag{3.31}$$

Choose $l = 50mm$. The other dimensions are selected arbitarily and will be checked for durability later.

Now we design a holder to keep the blade at correct angle to peel, and serve as connector to other elements such as spring and roller.

The dimension of these elements will follow the dimension of peeling blades.

The 3D design of these elements summarized below.

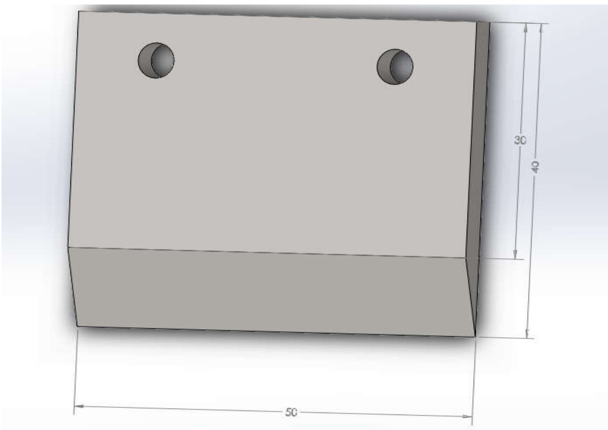

Fig 3.9 Peeling blade design

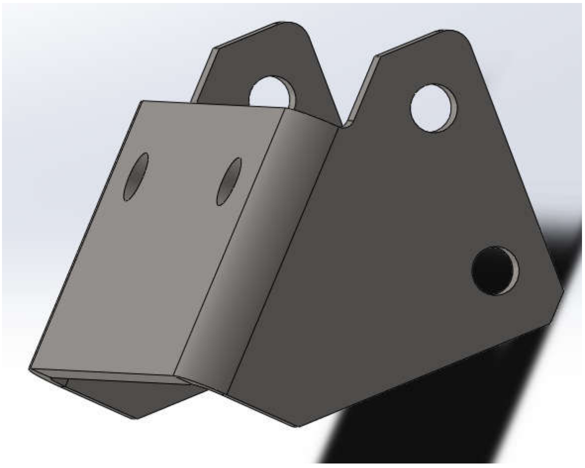

Fig 3.10 Blade holder design

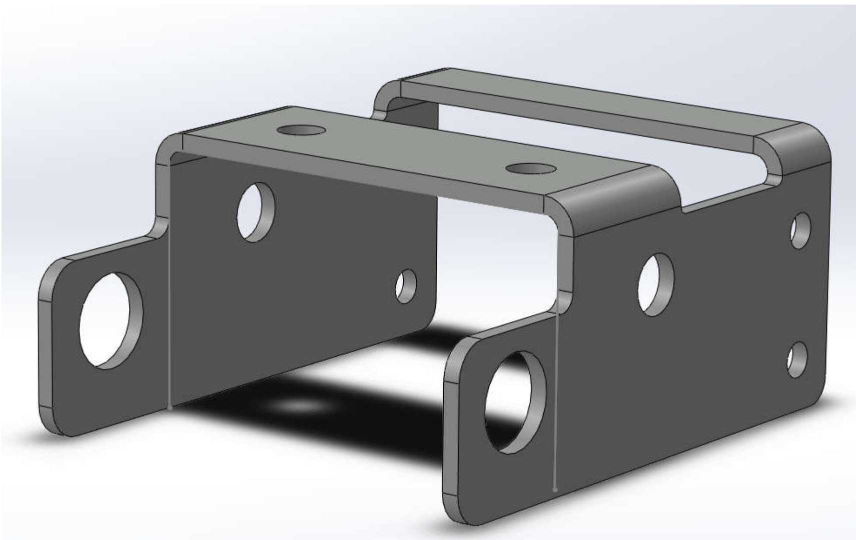

Fig 3.11 U-shaped holder

Checking the durability of peeling blade module

Since these parts' dimensional parameters are selected with limited calculations, we have to check through Solidworks. The blade is subjected to a force of $56.5771\ (N)$ acting on its surface at 60 °

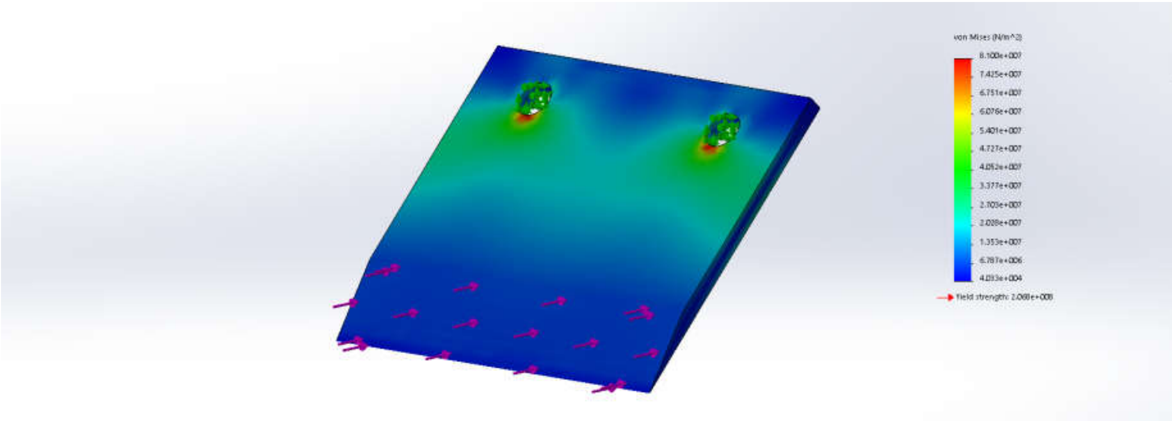

Fig 3.12 The stress of blade using Solidworks

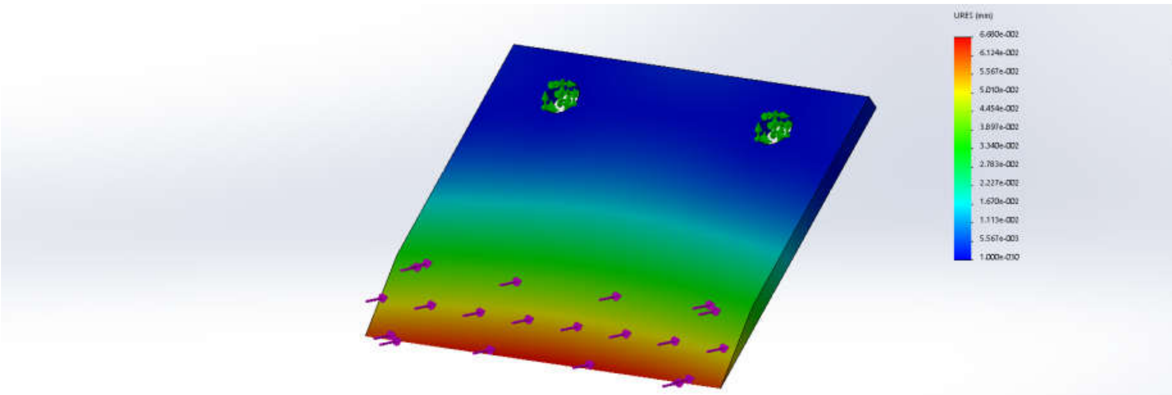

Fig 3.13 The displacement of blade using Solidworks

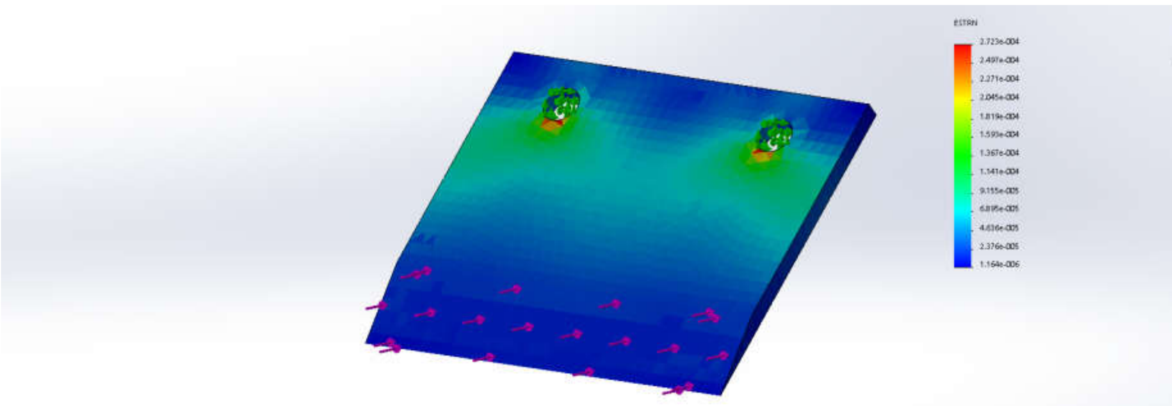

Fig 3.14 The strain of blade using Solidworks

The results show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition of blade is met.

The holders consist of 2 sheet metals parts made to keep the blade. A bladeholder is under peeling force of 56.5771 (N), while a U-shaped toolholder is under additinoal load from roller shaft: 146.6667 (N) and 188.0342 (N)

The results show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition is met.

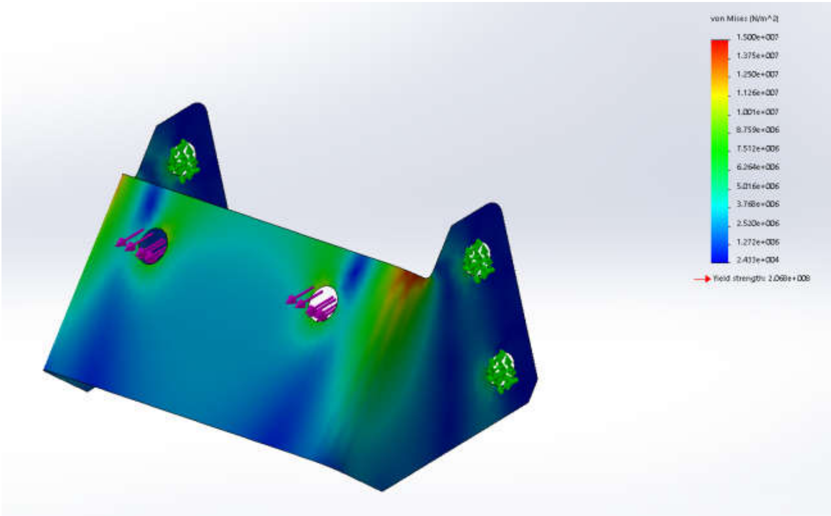

Fig 3.15 The stress of bladeholder using Solidworks

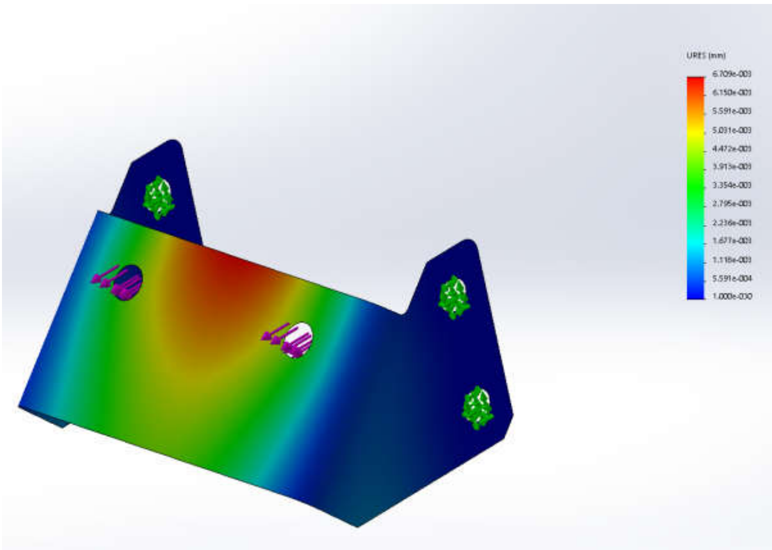

Fig 3.16 The displacement of bladeholder using Solidworks

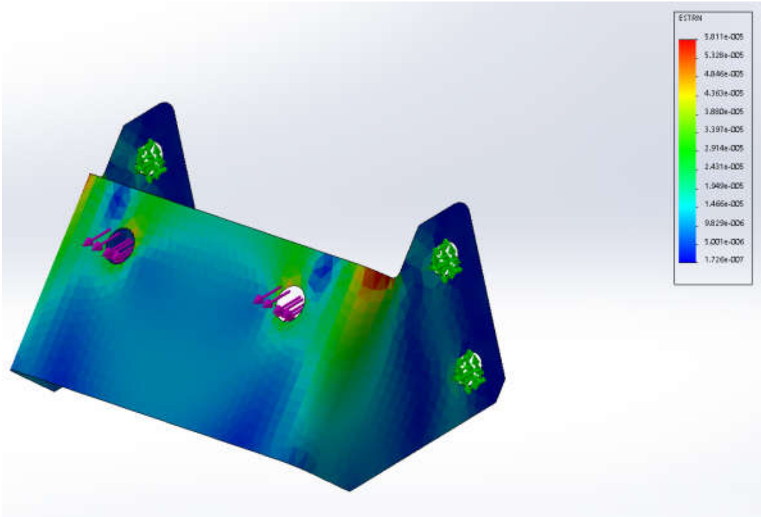

Fig 3.17 The strain of bladeholder using Solidworks

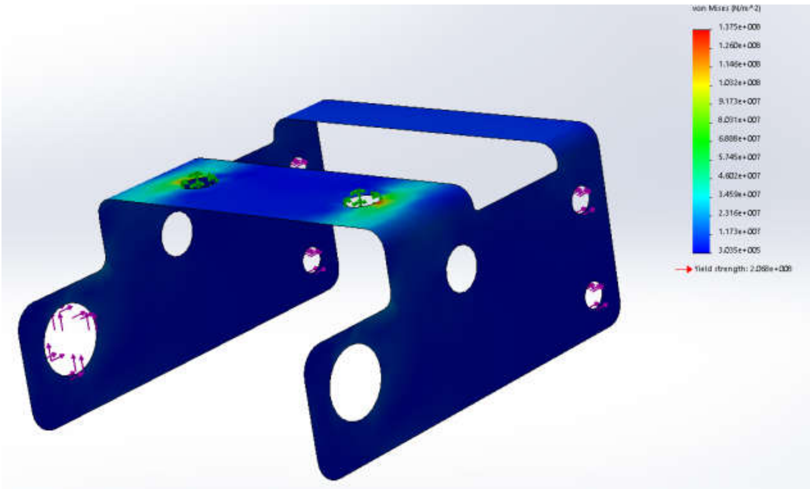

Fig 3.18 The stress of toolholder using Solidworks

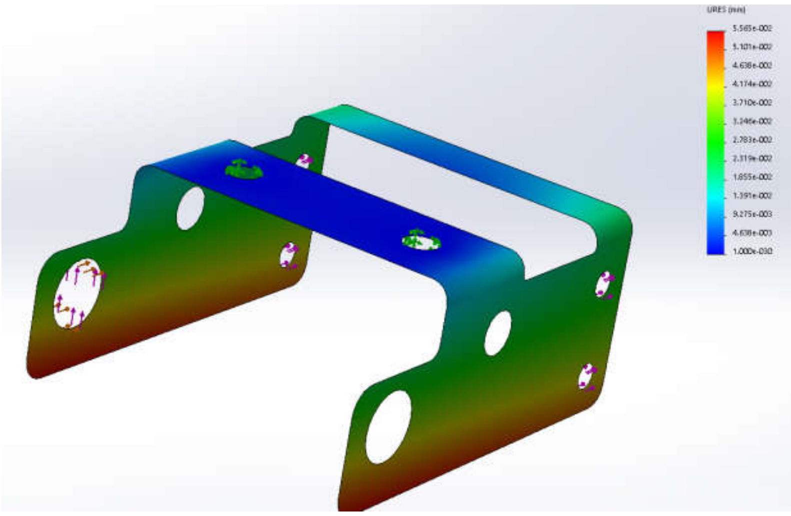

Fig 3.19 The displacement of toolholder using Solidworks

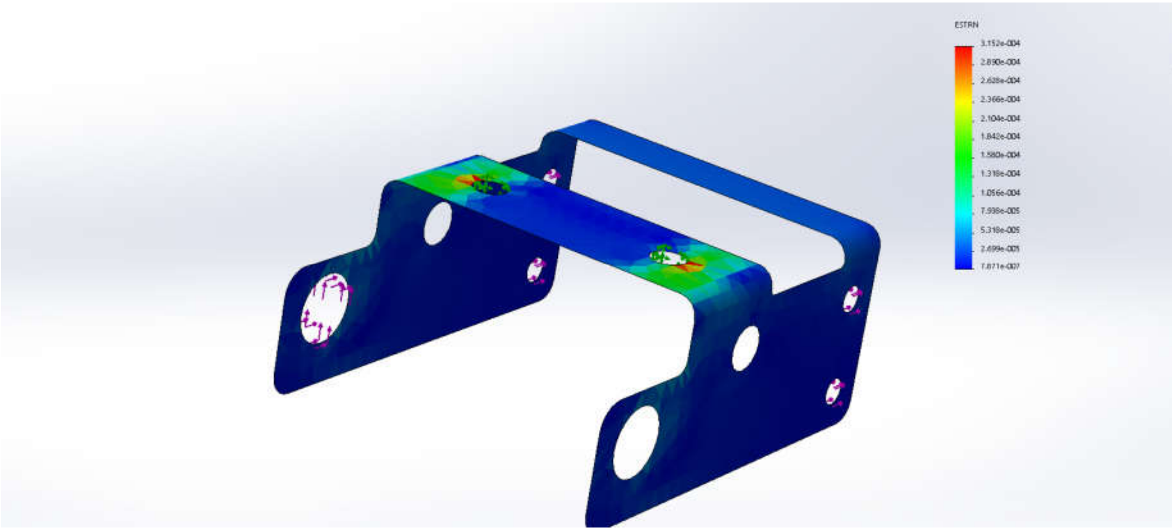

Fig 3.20 The strain of toolholder using Solidworks

3.1.4 Calculate the peeling frame spring

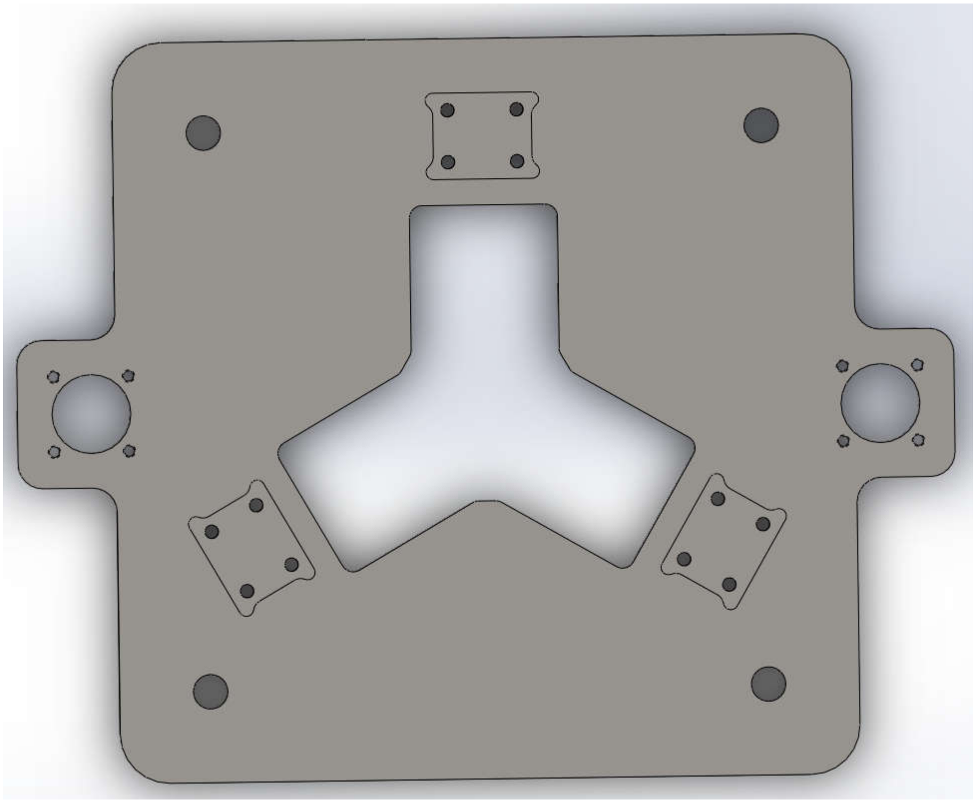

Fig 3.21 Plate 3D design

The spring will be attached to a shaft to allow the blade to adapt to the diameter of the sugarcane. And the shafts are supported by linear bearings, connected to a plate. There are 3 identical bladeholders on the same plate, each are separated by an angle of 120 degrees. This is to ensure the sugarcane will stay balanced in the center of the plate during machine operation. There are one-way roller and the blade inside the bladeholder.

The plates will be connected to shaft, and connected to a sliding rail with linear bearings to support the reciprocating motion.

<u>Calculate the spring to hold the depth of cut</u>

Consider 2 phases: peeling and pulling.

In peeling phase, the sugarcane must be held tight by the one-way roller at the feeding section, or else it will slide on the one-way roller because of the peeling force.

$$F_{f\ slide\ 1\ max} > F_{p\ total} \rightarrow \mu_{slide1} N_1 > F_{p\ total} \tag{3.32}$$

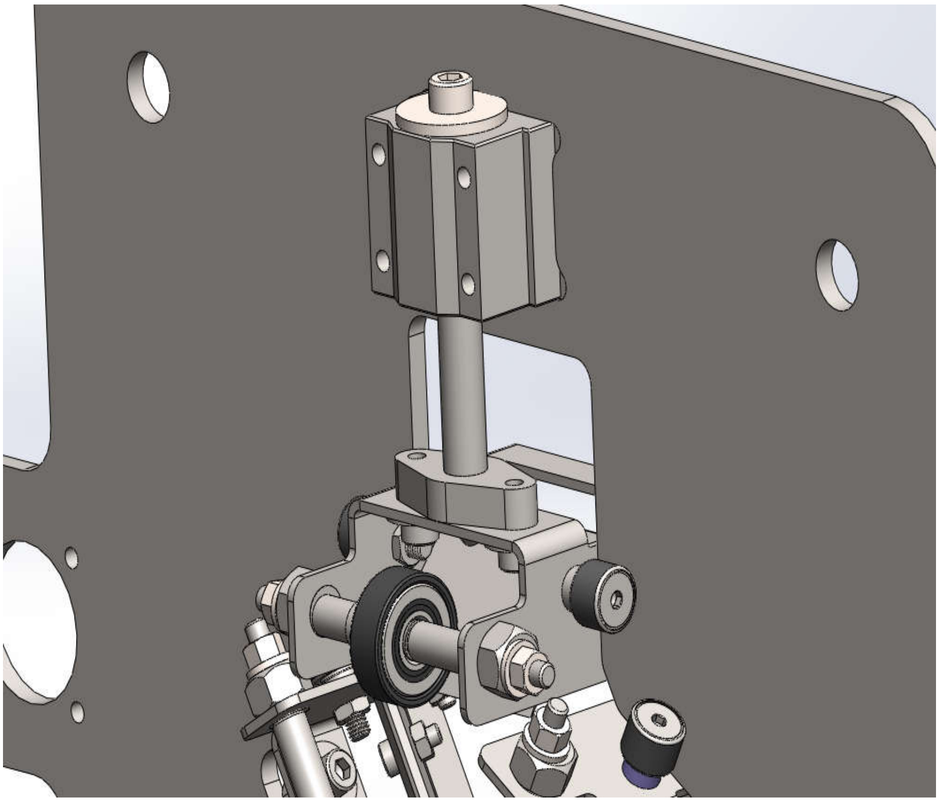

Fig 3.22 The bladeholder 3D design

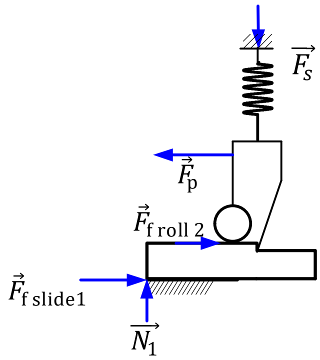

Fig 3.23 Forces diagram in peeling process

In pulling phase, the sugarcane must be held tight by the peeling platform to continue the operation, while also pulling it into the machine.

$$F_{p\ total} > F_{f\ roll\ 1} = \mu_{roll\ 1} N_1 \tag{3.33}$$

From references [13], we obtain the friction coefficient of sugarcane stalk and rubber:

$$\mu_{static\ 1} = 0.78, \qquad \mu_{slide\ 1} = 0.61, \qquad \mu_{roll\ 1} = 0.05$$

$$\rightarrow N_1 > \frac{F_{p\ total}}{\mu_{slide\ 1}} = \frac{509.1939}{0.78} = 652.8127\ (N) \tag{3.34}$$

This value also satisfies the pulling phase condition.

Value conditions of N_1 will be used to calculate the spring at the one-way roller feeder later.

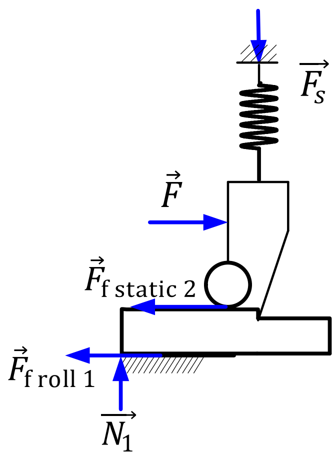

Fig 3.24 Forces diagram in pulling process

In pulling phase, when sugarcane and peeling platform moves together, the peeling platform act as an inertial frame. To pull the sugarcane:

$$\mu_{static\ 2} N_2 \geq m_{platform} a_{platform} \tag{3.35}$$

$$\mu_{static\ 2} = \mu_{static\ 1} = 0.78$$

From references [14], the acceleration of the peeling platform is

$$a = r\omega^2 \left(\cos\alpha + \frac{\cos 2\alpha}{r} l\right) = 0.15 \times (2\pi)^2 \times \left(\cos\alpha + \frac{\cos 2\alpha}{r} l\right) \tag{3.36}$$

$$\rightarrow a_{max} = 0.15 \times (2\pi)^2 \times \left(1 + \frac{l}{r}\right)$$

$$= 0.15 \times (2\pi)^2 \times \left(1 + \frac{0.4}{0.15}\right) = 22\ (m/s^2\ ) \tag{3.37}$$

Assume $m_{platform} = 10\ (kg)$

$$\rightarrow N_2 \geq \frac{m_{platform} a_{max}}{\mu_{static\ 2}} = \frac{10 \times 22}{0.78} = 282.0512\ (N) \tag{3.38}$$

Since the peeling frame need to pull the sugarcane in, at the initial time and end time, there may be only 3 blades in contact, so the reaction force magnitude for each, which equals the spring force's is:

$$N_{2s} = \frac{N_2}{3} \geq \frac{282.0512}{3} = 94.017\ (N) \tag{3.39}$$

The spring will be a compression spring to adapt to the diameter of sugarcane from $30 - 50\ mm$. The blade leaves a space enough for $30mm$ sugarcane to pass through.

Presume the space available for spring $37.44mm$.

Summary input parameters to calculate spring: Force range: $94.017\ \leq F_s$; Working displacement $x = 10\ (mm)$

Use **SPEC $\mathbf{2} \times \mathbf{18} \times \mathbf{68}$** compression spring with below specifications:

Parameters	**Value**
Outer diameter	$18\ mm$
Wire diameter	$2\ mm$
Length	$68\ mm$

Initial compressed length	$37.44\ mm$
Required compressed length	$27.44\ mm$
Maximum compressed length	$25.76\ mm$
Maximum load	$198.09\ N$
Spring constant K_b	$4.69\ N/mm$
Material	DIN 17223 - C

Table 3.3 Bladeholder first disc spring parameters

The normal force at initial compressed length for $30mm$-diameter sugarcane:

$$F_{s30b} = K_b \Delta x_{30b} = 4.69 \times (68 - 37.44) = 143.3264\ (N) \tag{3.40}$$

The normal force for $50mm$-diameter sugarcane:

$$F_{s50b} = K_b \Delta x_{50b} = 4.69 \times (68 - 37.44 + 10) = 190.2264\ (N) \tag{3.41}$$

The force satisfied the condition of normal force range.

<u>Calculate the roller to hold the depth of cut</u>

We analyze the force acting on the 1 roller the instance before it contacts the sugarcane:

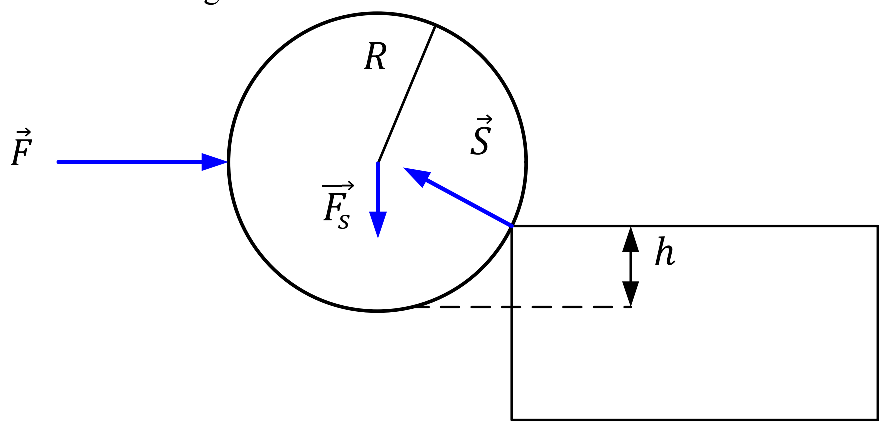

Fig 3.25 Force analysis of a roller in peeling frame

Where $\vec{F}$ is force pushing the roller; $\vec{F_s}$ is spring force; $\vec{S}$ is force of sugarcane acting on the roller; R is the radius of the roller; h is the height that the roller must go above to press down the sugarcane.

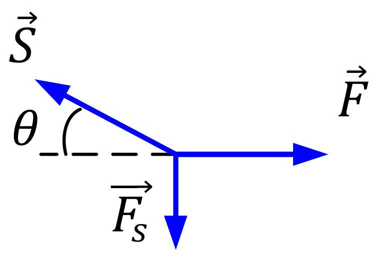

Fig 3.26 Force analysis simplified diagram

Applying Newton Second Law, we have:

$$\begin{cases} F = Scos(\theta) \\ F_s = Ssin(\theta) \end{cases} \quad (3.42)$$

Consider $F_s = F_{s30}$ before the roller contacts the sugarcane, and $h = 0.01\ (m)$ when roller is at initial position for $30mm$ sugarcane and expect to go over $50mm$ one, we take division:

$$\frac{F_{s20}}{F} = \tan(\theta) = \frac{R-h}{\sqrt{h(2R-h)}} \quad (3.43)$$

$$\rightarrow \frac{111.206}{509.1939} = \frac{R-0.01}{\sqrt{0.01 \times (2R-0.01)}} \rightarrow R = 0.01567(m) \quad (3.44)$$

Since the force is fixed, only rollers with $R \geq 0.01567\ m$ will be suitable. Choose **4752N11** shielded one way bearing with rubber rings.

Parameters	**Value**
Material	$Steel$
Roller diameter	$22\ mm$
Roller width	$9\ mm$
Shaft diameter	$8\ mm$
Radial static load	$860\ N$
Radial dynamic load	$3265\ N$

Table 3.4 Bladeholder's roller parameters

3.1.3 Calculation of shaft for bladeholder's rollers

We have $N_2 = 188.0342\ N$, $F_{static\ 2} = \mu_{static\ 2} N_2 = 0.78 \times 188.0342 = 146.6667\ (N)$

Assume the length of the shaft is $100\ mm$

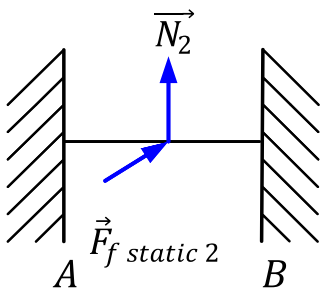

Fig 3.27 Force analysis on shaft for bladeholder's rollers

We calculate the load (in N) and bending moment (in Nm) using MDSolids.

$$M_{td-holder} = \sqrt{3.67^2 + 4.7^2} = 5.9631\ Nm \tag{3.45}$$

$$d \geq \sqrt[3]{\frac{32M_{td-holder} \times 10^3}{\pi[\sigma]_{304}}} = \sqrt[3]{\frac{32 \times 5.9631 \times 10^3}{\pi \times 80}} = 7.9859\ (mm) \tag{3.46}$$

Where $[\sigma]_{304} = 80\ MPa$ is the allowable bending stress of 304 steel. [11]

The selected feeder roller's shaft diameter satisfied the condition. Because the shaft does not rotate, we check the shaft by static durability.

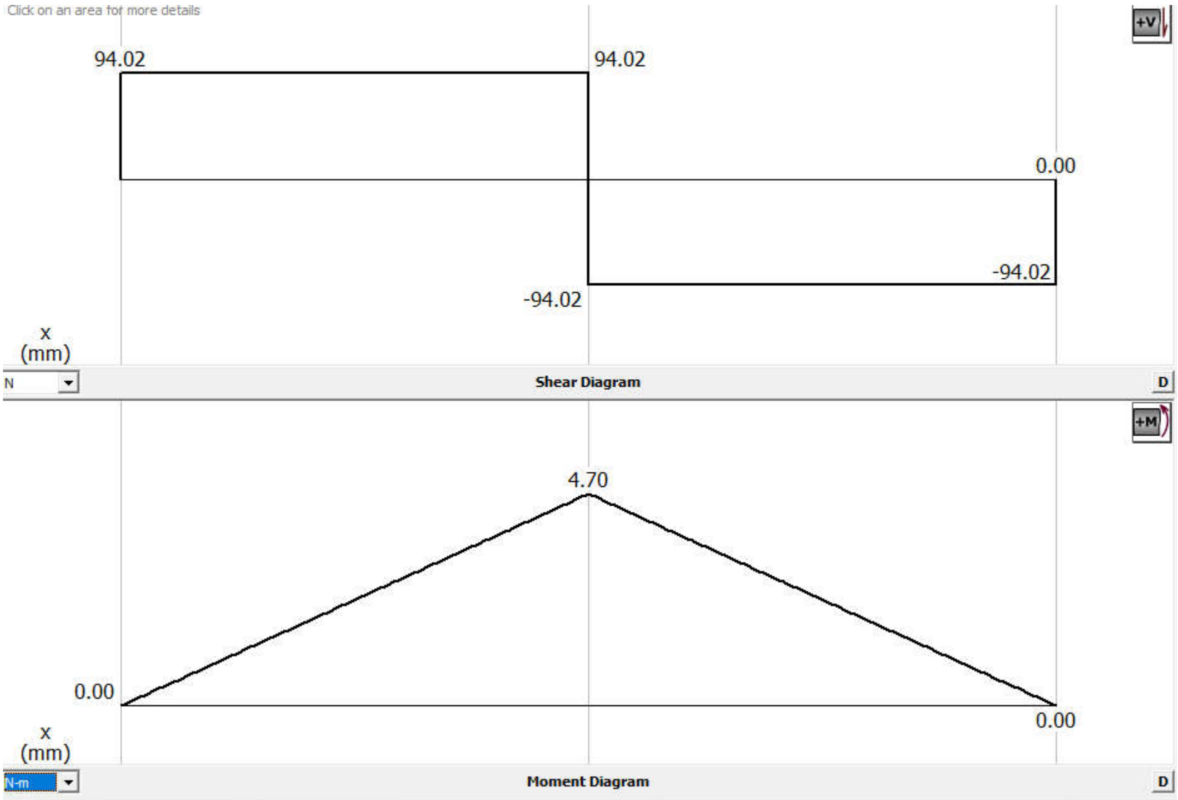

Fig 3.28 Shear force and moment diagram of axial load N_2

$$\sigma_{td-holder} = \frac{32M_{td-holder} \times 10^3}{\pi d^3} = \frac{32 \times 4 \times 10^3}{\pi \times 8^3} = 79.5775\ (MPa) \tag{3.47}$$

$$\rightarrow \sigma_{td-holder} < [\sigma]_{qt}$$

Where $[\sigma]_{qt} = 0.8 \times [\sigma]_{304\,yield} = 0.8 \times 205 = 164\ (MPa)$ is allowable stress at overload of 304 stainless steel.

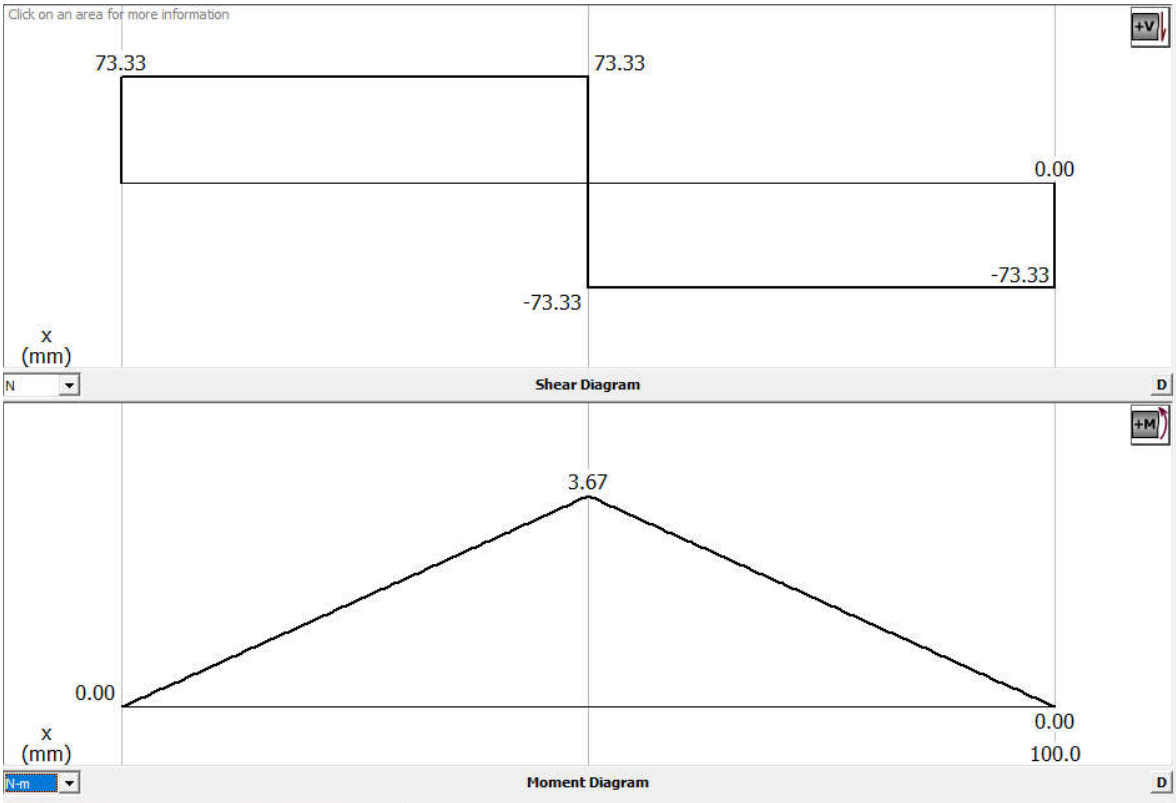

Fig 3.29 Shear force and moment diagram of tangential load $F_{static\ 2}$

3.2 Feeding section

3.2.1 Calculate the stiffness of the spring holding the roller

The minimum force on the sugarcane at the feeder roller for $50mm$ sugarcane.

$$N_1 \geq 652.8127\ (N)$$

We also need to make sure the force will not damage the sugarcane. According to [15], the crushing force of sugarcane is $1200\ (N)$

We use 2 tension springs hook at 2 ends of a screw, with roller in the middle of the shaft.

Peeling force needed for $30mm$ diameter sugarcane is

$$F_{p30} = 9 \times \sigma_{p\ 99.7} \times A_{20}$$

$$= 9 \times 10.103 \times 0.4 \times 2 \times \sqrt{2 \times \frac{30}{2} \times 1 - 1^2} = 391.7255\ (N) \qquad (3.48)$$

The normal force required to hold tight during peeling phase:

$$N_{120} \geq \frac{F_{p30}}{\mu_{static1}} = \frac{391.7255}{0.78} = 502.2122\ (N) \tag{3.49}$$

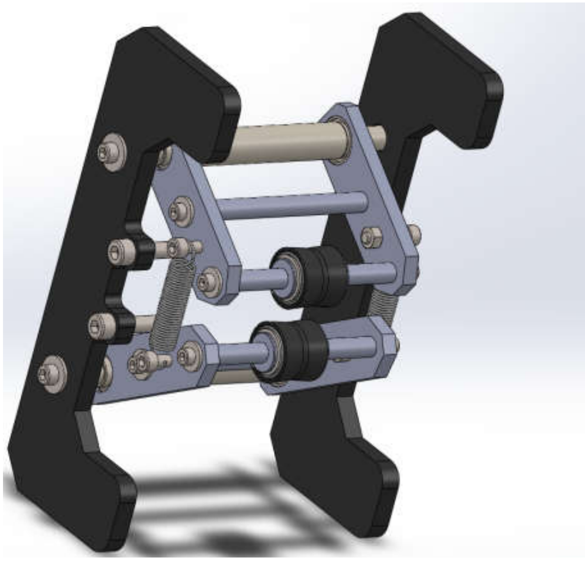

Fig 3.30 The feeder section design at one end

Summary input parameters to calculate spring: Force range:

$$502.2122 \leq N_2 \leq 652.8127 (N) \tag{3.50}$$

According to the plate design we have the lever moment equation

$$N_2 l_1 = F_s l_2 \rightarrow F_s = \frac{N_2 l_1}{l_2} = \frac{75 N_2}{55} = \frac{15}{11} N_2 \tag{3.51}$$

Where $l_1 = 75\ mm$ is the center distance from feeder roller shaft to revolute joint shaft of the lever. $l_2 = 55\ mm$ is the center distance from spring pin to revolute joint shaft of the lever.

$$\rightarrow 502.2122 \times \frac{15}{11} \leq F_s \leq 652.8127 \times \frac{15}{11} \tag{3.52}$$

$$\rightarrow 684.8348 \leq F_s \leq 890.1991 \tag{3.53}$$

Since we will use 2 springs, split the force in 2.

$$\rightarrow 342.4174 \leq F_s \leq 445.1 \tag{3.54}$$

Working displacement $x = 20\ (mm)$

Select tension spring Metrol $4 \times 28 \times 76$

The tension spring to durable enough for this application using Autocad Mechanical's shear stress test.

Parameters	**Value**
Outer diameter	$28\ mm$
Number of rounds	$12\ rounds$
Wire diameter	$4\ mm$
Initial length	$76\ mm$
Initial tension	$71\ N$
Length at maximum load	$111\ mm$
Maximum load	$564\ N$
Stiffness	$18.8\ N/mm$
Material	Stainless steel wire 1.431

Table 3.5 Feeder spring parameters

Initial length for spring for $30mm$ sugarcane:

$$l_1 = \frac{342.4174 - 71}{18.8} = 14.4371\ (mm) \tag{3.55}$$

Length of spring for $50mm$ sugarcane:

$$l_2 = l_1 + 20 = 14.4371 + 20 = 34.4371\ (mm) \tag{3.56}$$

The length satisfies the spring boundary condition.

3.2.2 Calculate the feeding one-way roller

We analyze the force acting on the 1 roller the instance before it contacts the sugarcane

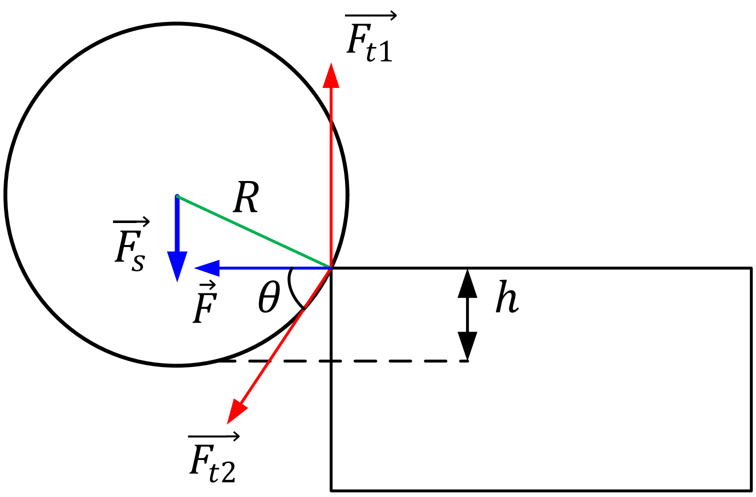

Fig 3.31 Force analysis of a roller in peeling frame

Where $\overrightarrow{F_s}$ is spring force; $\vec{F}$ is push force, equal to $\vec{F}_{t1} + \vec{F}_{t2}$; $\vec{F}_{t1}$ is the force pushing the roller vertically; $\vec{F}_{t2}$ is the force that make roller rotates. R is the radius of the roller; h is the height that the roller must go above to press down the sugarcane.

To push the roller up, $\vec{F}_{t1} \geq \overrightarrow{F_s}$

Assmume $F = 100\ (N)$ [16], $F_s = 652.8127\ (N)$ before the roller contacts the sugarcane, and $h = 0.015\ (m)$ when roller is at initial position for $20mm$ sugarcane and expect to go over $50mm$ one, we have:

$$F_{t1} = F\tan(\theta) = F\frac{\sqrt{h(2R-h)}}{R-h} \geq 652.8127 \tag{3.57}$$

$$\rightarrow \frac{652.8127}{100} \leq \frac{\sqrt{0.01 \times (2R - 0.01)}}{R - 0.01} \tag{3.58}$$

$$\rightarrow R \geq 0.01178(m) \tag{3.59}$$

Choose **1553N15** sealed one way bearing with rubber rings.

Parameters	Value
Material	$Steel$
Roller diameter	$28\ mm$
Roller width	$26\ mm$
Shaft diameter	$20\ mm$
Radial static load	$2449\ N$
Radial dynamic load	$5262\ N$

Table 3.6 Feeder's roller parameters

3.2.3 Calculation of shaft for feeder's rollers

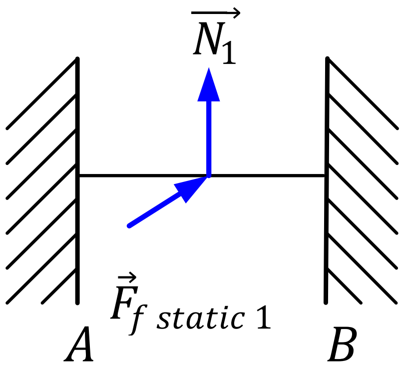

Fig 3.32 Load on shaft for feeder's rollers

Assume the length of feeder roller shaft is $100mm$.

Maximum radial load is $N_1 = 652.8127\ (N)$. Maximum tangential load is $F_{f\ static\ 1} = 509.1939\ (N)$

$$M_{td-feed} = \sqrt{16.32^2 + 12.73^2} = 20.6977\ (Nm) \quad (3.60)$$

$$d \geq \sqrt[3]{\frac{32 M_{td-feed} \times 10^3}{\pi[\sigma]_{304}}} = \sqrt[3]{\frac{32 \times 20.6977 \times 10^3}{\pi \times 80}} = 12.8102\ (mm) \quad (3.61)$$

The selected feeder roller's shaft diameter satisfied the condition. Because the shaft does not rotate, we check the shaft by static durability.

$$\sigma_{td-feed} = \frac{32 M_{td-feed} \times 10^3}{\pi d^3} = \frac{32 \times 16.51 \times 10^3}{\pi \times 35^3} = 4.9172\ (MPa) \quad (3.62)$$

$$\rightarrow \sigma_{td-feed} < [\sigma]_{qt}$$

Where $[\sigma]_{qt} = 0.8 \times [\sigma]_{304\ yield} = 0.8 \times 205 = 164\ (MPa)$ is allowable stress at overload of 304 stainless steel.

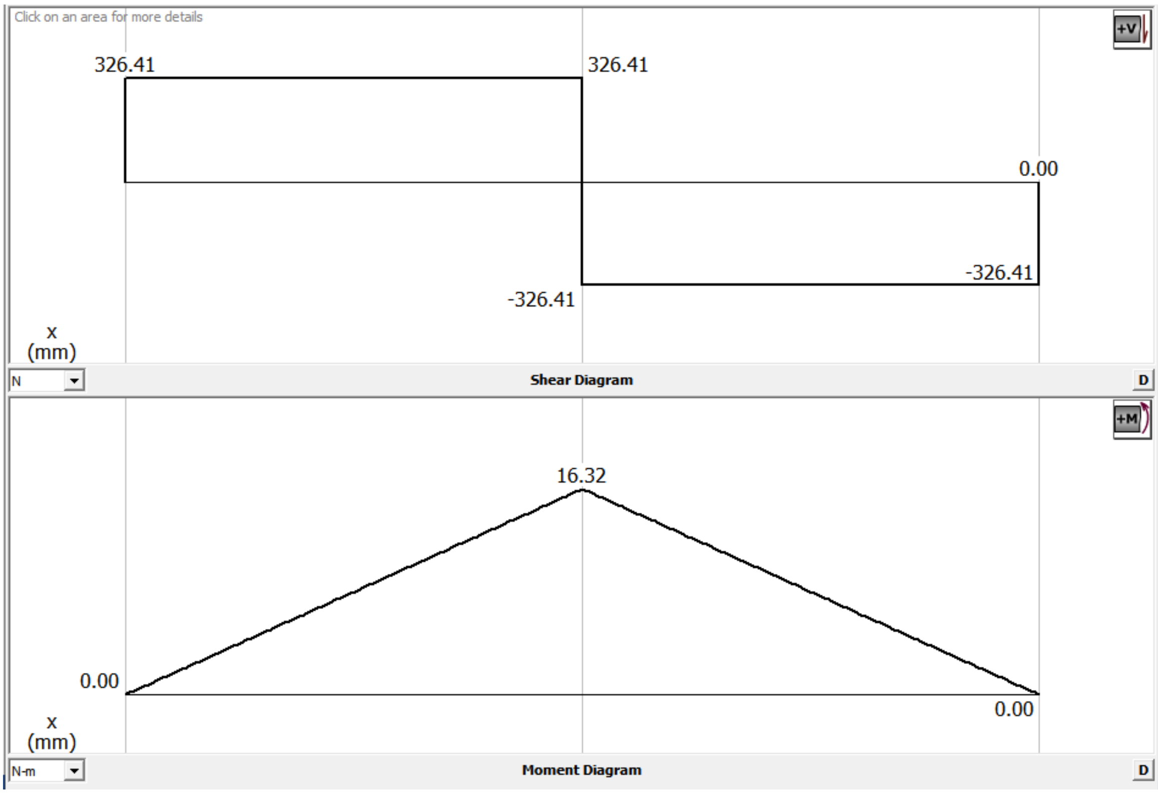

Fig 3.33 Radial load moment diagram of feeder roller shaft

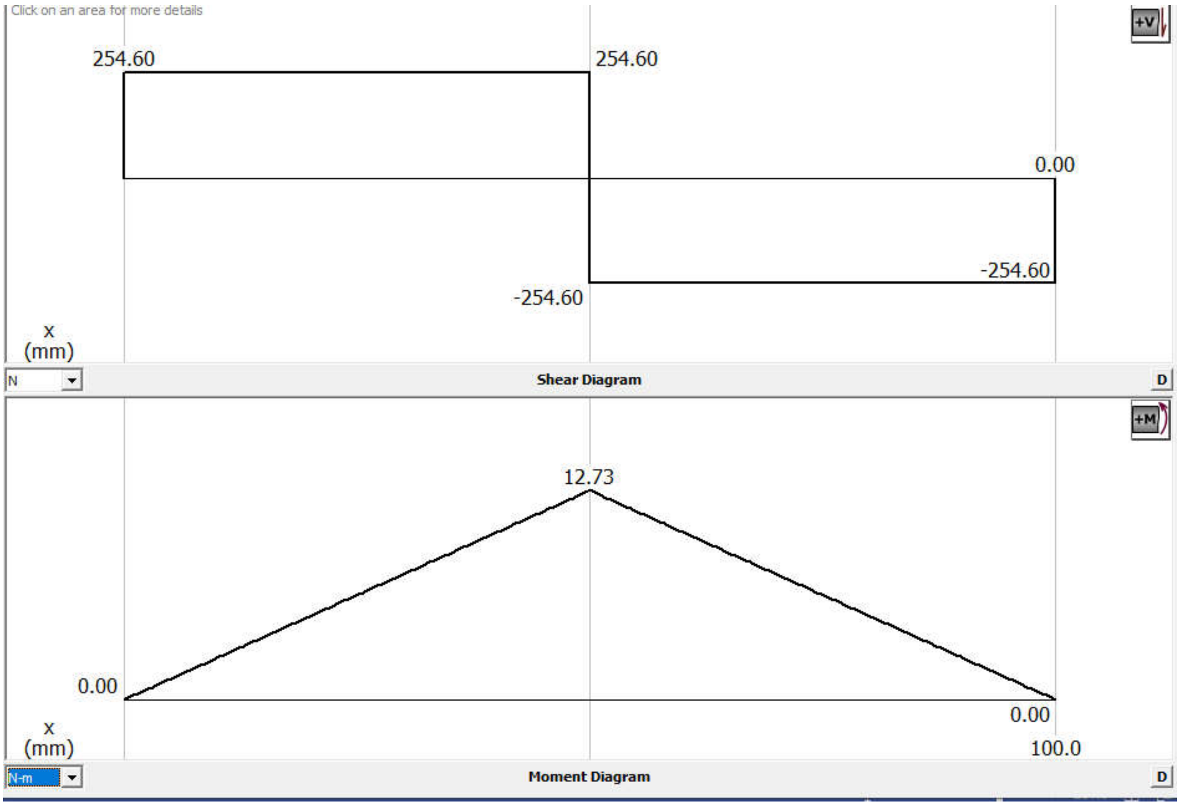

Fig 3.34 Tangential load moment diagram of feeder roller shaft

3.3 Machine frame

3.3.1 Calculation of sliding shaft strength

The sliding shaft is fixed at 2 ends to the machine frame by using a flange shaft mount.

Since the shaft does not rotate, we calculate the shaft strength by static load.

From current machine's frame dimension [4], let the length of shaft $l = 800\ (mm)$. Assume the peeling frame is applying a load equal to half its weight on each of two sliding shafts. Since the load move along the shaft so we can't exactly place a position of force application, instead we choose the most failible position, which is the middle of the shaft. Force on shaft:

$$P_1 = \frac{P_{peeling\ platform}}{2} = \frac{20 * 10}{2} = 100\ (N) \tag{3.63}$$

Use MDSolids to acquires the bending moment diagram, we see that the highest moment is $M_{bend} = 20\ Nm$. Then, according to [11] the diameter of the shaft at the dangerous section is:

$$d \geq \sqrt[3]{\frac{32M_{bend} \times 10^3}{\pi[\sigma]_{304}}} = \sqrt[3]{\frac{32 \times 20 \times 10^3}{\pi \times 80}} = 13.66\ (mm)$$

Where $[\sigma]_{304} = 80\ MPa$ is the allowable bending stress of 304 steel. [11]

Choose $d = 20\ mm$

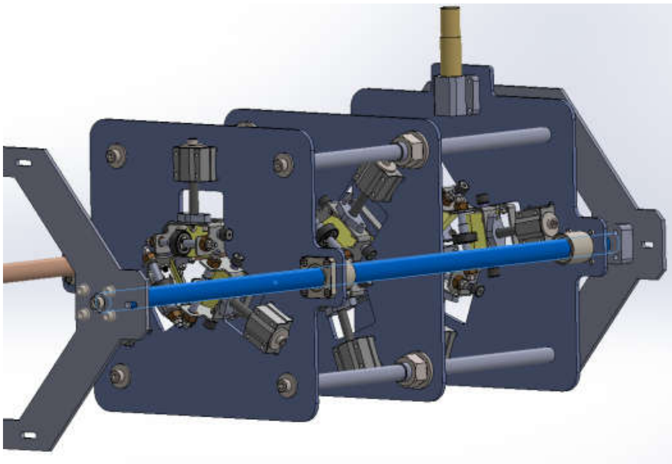

Fig 3.35 Sliding shaft with flange mount 3D view

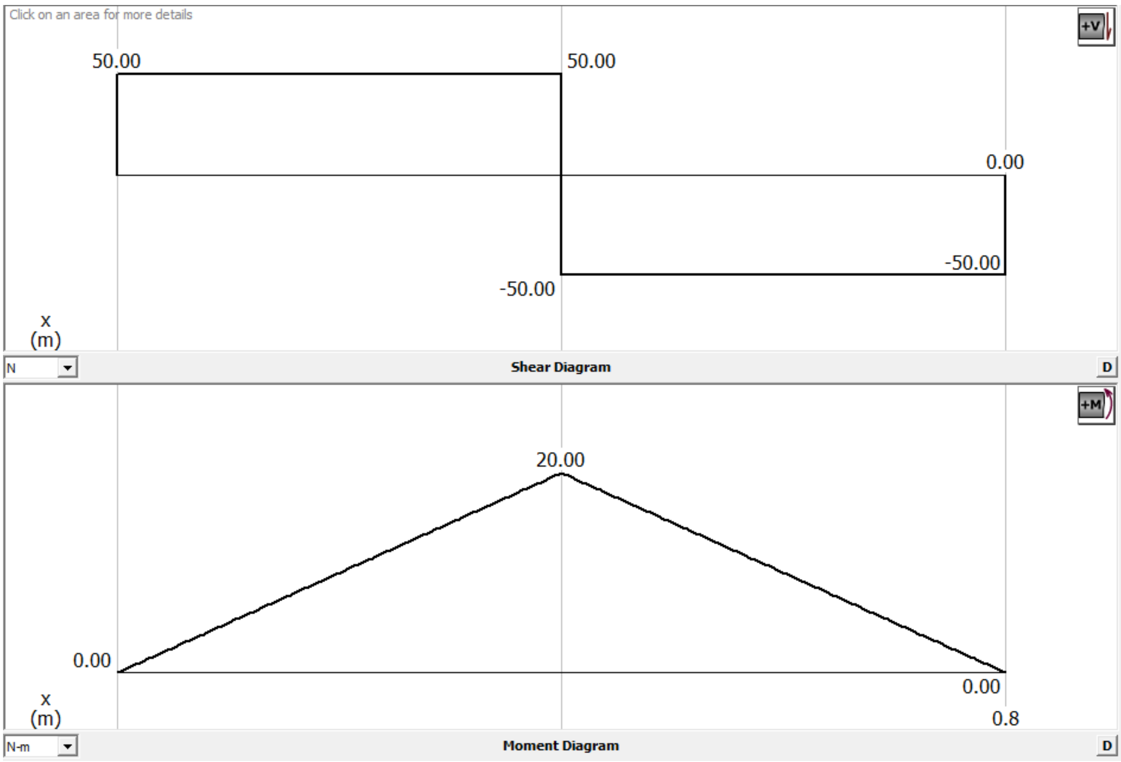

Fig 3.36 Shear force and bending moment diagram on MDSolids

3.3.2 Calculation of bolt forces of sliding shaft fixed support.

The sliding shaft will be fixed with bolt group joints at two ends. Assume the bolts is arranged in a rectangle with length $a = 50\ mm$, width $b = 40\ mm$ with center distance $c = 35\ mm$, $d = 25\ mm$.

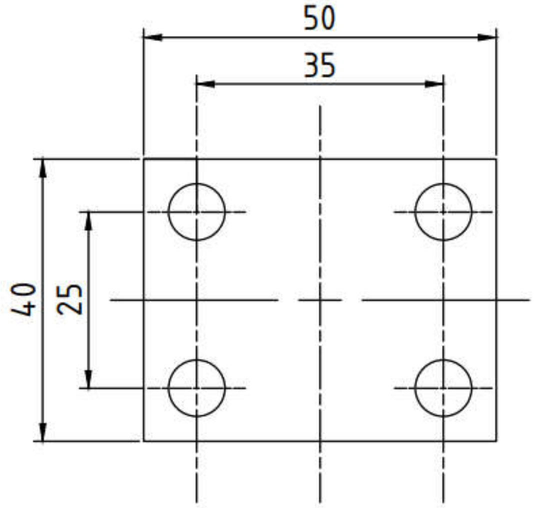

Fig 3.37 A shaft's bolt group dimension

Calculate the force to tighten the bolt by no sliding condition [11]:

$$V_1 = \frac{kF_H}{fz} = \frac{1.3 \times 50}{0.8 \times 4} = 20.3125\ (N) \tag{3.64}$$

Where:

k is safety factor

F_H is the force acting perpendicular to the bolt group plane, $F_H = \frac{P_1}{2} = \frac{100}{2} = 50(N)$

f is friction coefficient between bolted surfaces – steel and steel

z is the number of bolts in a group

Calculate the force to tighten the bolt by no detachment condition:

$$V_2 = \frac{k}{z}\left(\frac{F_h \times l \times A \times y_c}{J_X}\right)(1 - \chi)$$

$$= \frac{1.3}{4} \times \frac{50 \times 400 \times 50 \times 40 \times \frac{25}{2}}{\frac{50 \times 40^3}{12}} \times (1 - 0.25) = 457.0313\ (N) \tag{3.65}$$

Where:

l is the distance from center of bolt group plane to F_H force application point

A is the bolt group mount surface area

y_c is the distance from outer bolt to the rolling axis, $y_c = \frac{d}{2}$

J_X is the bolt group moment of inertia

χ is a force constant, $\chi = 0.25$ [11]

Since $V_2 > V_1$, we continue calculation with V_2. Corresponding force for computation:

$$F_{td} = 1.3V + \frac{\chi M Y_1}{\Sigma z_i Y_i^2}$$

$$= 1.3 \times 457.0313 + \frac{0.25 \times (50 \times 400) \times \frac{25}{2}}{4 \times \frac{25}{2}^2} = 694.1407\ (N) \tag{3.66}$$

Where:

Y_i is the distance from bolt i to the rolling axis

z_i is the number of bolts with distance Y_i

The diameter of bolt required:

$$d_1 = \sqrt{\frac{4F_{td}}{[\sigma_k]\pi}} = \sqrt{\frac{4 \times 694.1407}{32\pi}} = 5.2554\ (mm) \tag{3.67}$$

Where:

$$[\sigma_k] = \frac{[\sigma_{ch}]}{s} = \frac{240}{7.5} = 32\ (MPa) \tag{3.68}$$

Is the allowable tensile strength for 4.6 bolt type, alloy steel material and variable load safety factor $s = 7.5$ [11].

Choose bolt M8.

3.3.3 Calculation of bolt forces of connecting rod shaft fixed support on disc.

The shaft of connecting rod will transfer motion from slidercrank mechanism to the disc through a bolted housing. It transfers motion from motor to the sliding platform, hence the force acts on it will be inertial forces of platform and peeling forces.

Assume the bolts is arranged in a rectangle with length $a = 50\ mm$, width $b = 40\ mm$ with center distance $c = 40\ mm$, $d = 30\ mm$.

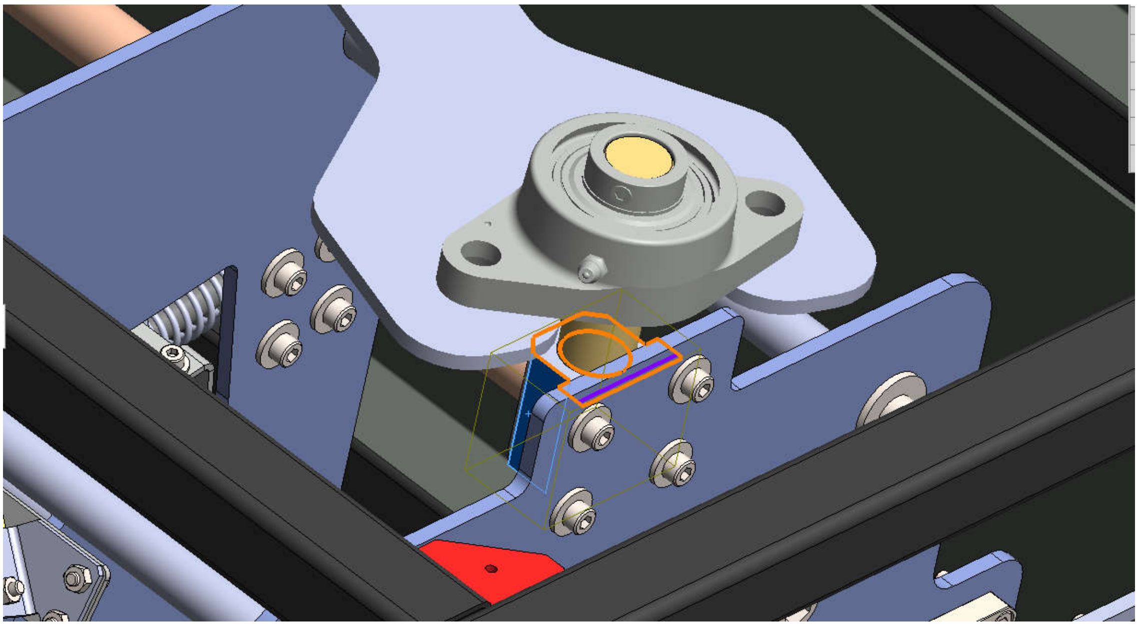

Fig 3.38 Position of bolt group mounting in 3D view

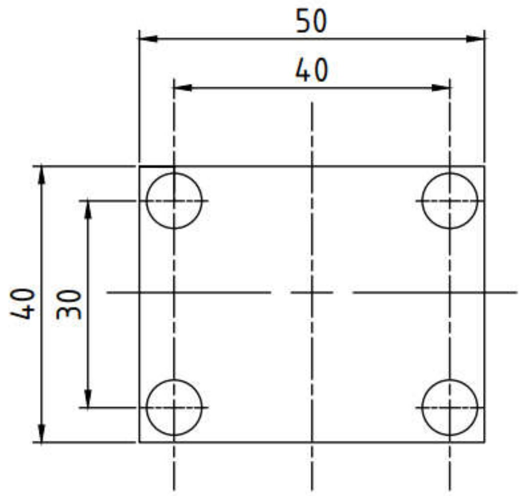

Fig 3.39 Mechanism's shaft's housing joint bolt group dimension

Calculate the force to tighten the bolt by no sliding condition:

$$V_1 = (1 - \chi) \times \frac{F_V}{z} = (1 - 0.25) \times \frac{1073.2965}{4} = 201.243\ (N) \tag{3.69}$$

Where:

$$F_V = F_{inertia} + F_p = 564.1026 + 509.1939 = 1073.2965\ (N) \tag{3.70}$$

Calculate the force to tighten the bolt by no detachment condition:

$$V_2 = \frac{k}{z}\left(\frac{F_v \times l_1 \times A_1 \times y_{c1}}{J_{X1}} - F_v\right)(1-\chi)$$

$$= \frac{1.3}{4} \times \frac{1073.2965 \times 50 \times 50 \times 40 \times \frac{30}{2}}{\frac{40 \times 50^3}{12}} \times (1-0.25) = 941.8177(N) \tag{3.71}$$

Where:

l_1 is the distance from center of bolt group plane to F_V force application point

A_1 is the bolt group mount surface area

y_{c1} is the distance from outer bolt to the rolling axis, $y_c = \frac{d}{2}$

J_{X1} is the bolt group moment of inertia

χ is a force constant, $\chi = 0.25$ [11]

Since $V_2 > V_1$, we continue calculation with V_2. Corresponding force for computation:

$$F_{td1} = 1.3V - \frac{\chi F_V}{z} + \frac{\chi M Y_1}{\Sigma z_i Y_i^2}$$

$$= 1.3 \times 941.8177 - \frac{0.25 \times 1073.2965}{4} + \frac{0.25 \times 50 \times 1073.2965 \times \frac{30}{2}}{4 \times \frac{30}{2}^2}$$

$$= 1380.8854\ (N) \tag{3.72}$$

The diameter of bolt required:

$$d_2 = \sqrt{\frac{4F_{td1}}{[\sigma_k]\pi}} = \sqrt{\frac{4 \times 1380.8854}{32\pi}} = 7.4124\ (mm) \tag{3.73}$$

Is the allowable tensile strength for 4.6 bolt type, alloy steel material and variable load safety factor $s = 7.5$.

Choose bolt M8.

3.3.4 Calculation of crank and connecting rod parameters.

There are 2 shafts on connecting rod, one connecting to the disc, one connecting to the crank. We will calculate the reaction force at the joint to determine the radial force acting on the shaft then design the shaft parameters. These forces act along the connecting rod or the crank. Using MATLAB simulation, we are able to determine the below value table.

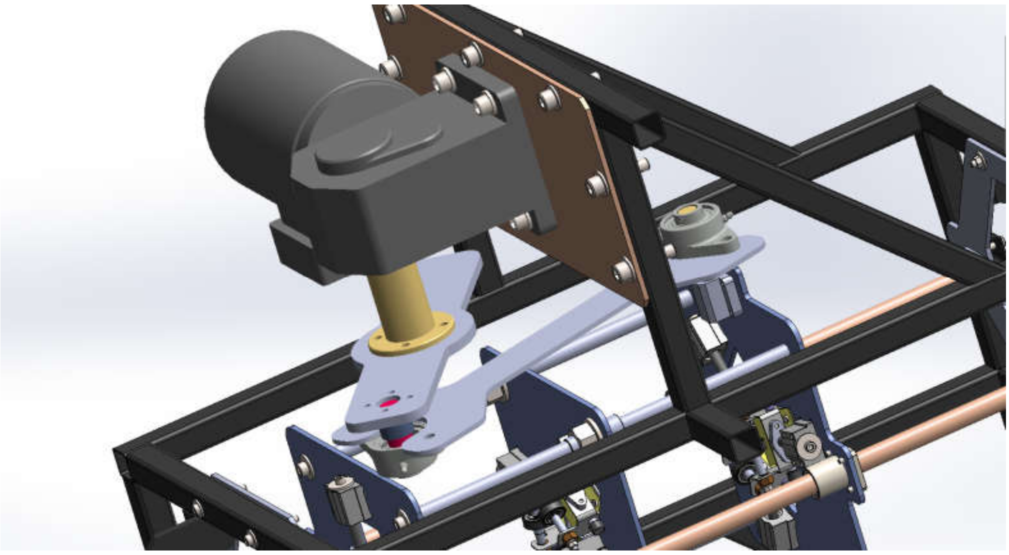

Fig 3.40 3D View of Slidercrank mechanism

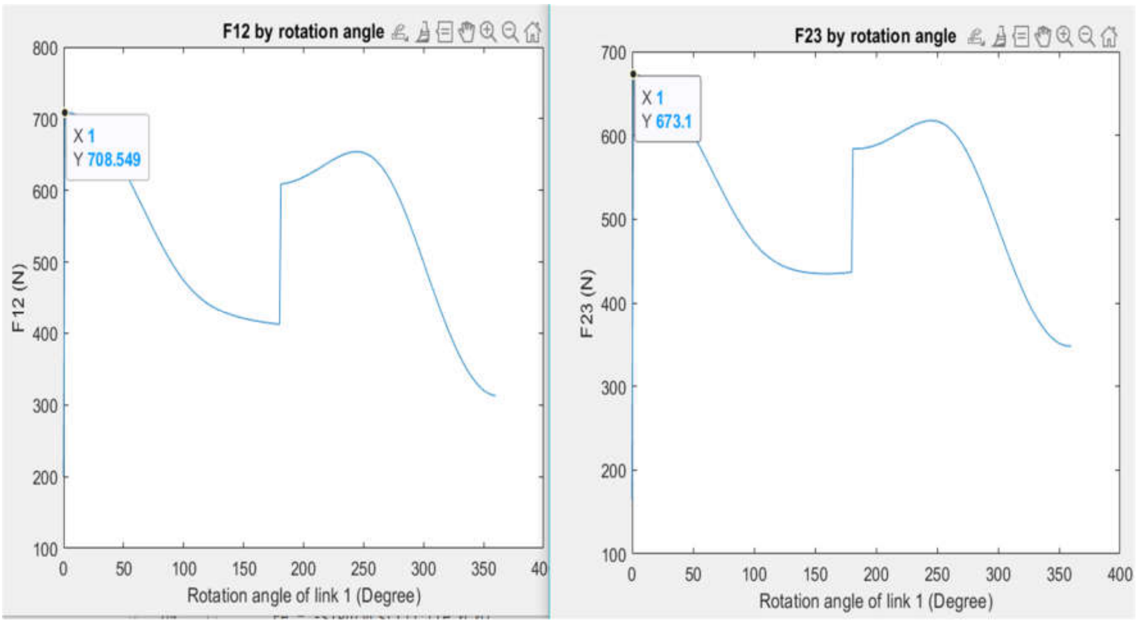

Fig 3.41 Reaction force at joints by rotation angle of link 1

Shaft	**Maximum reaction forces at joint**
Connecting rod – Disc	$F_{23} = 673.1\ N$
Crank – Connecting rod	$F_{12} = 708.549\ N$

Table 3.7 Summary of reaction forces of slidercrank

Select the preliminary value for $l_1 = 60\ mm$ and $l_2 = 110\ mm$ for shaft connecting the rod to disc and the crank to rod respectively. We calculate the diameter for each shaft. Using MDSolids to acquire the moment diagram.

For shaft connecting the crank to rod, we see that the highest moment is $M_{bend} = 31.88\ Nm$. Then, according to [11] the diameter of the shaft at the dangerous section is:

$$d \geq \sqrt[3]{\frac{32M_{bend} \times 10^3}{\pi[\sigma]_{304}}} = \sqrt[3]{\frac{32 \times 31.88 \times 10^3}{\pi \times 80}} = 15.9518\ (mm) \tag{3.74}$$

Where $[\sigma]_{304} = 80\ MPa$ is the allowable bending stress of 304 steel. [11]

Choose $d = 20\ mm$.

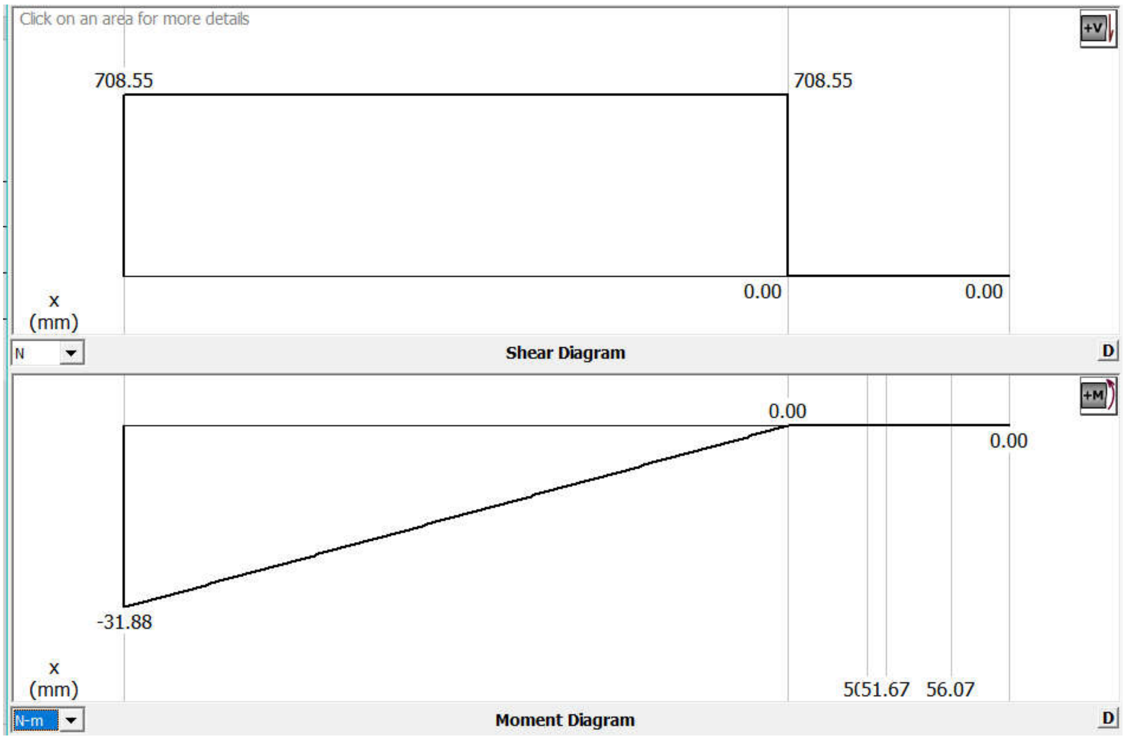

Fig 3.42 Force and moment diagram of shaft connecting the crank to rod

Use Solidworks to check the selected shaft diameter's durability show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition is met.

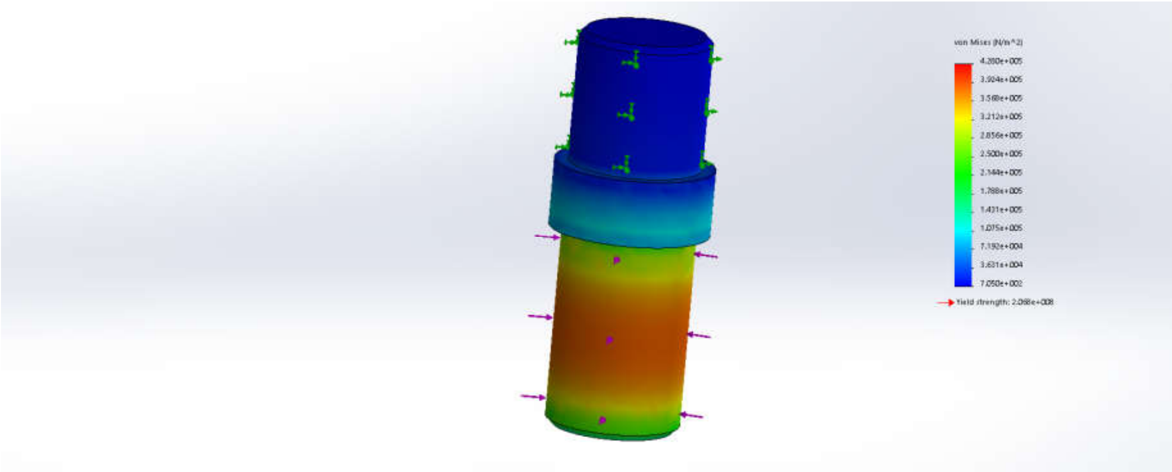

Fig 3.43 The stress of shaft connecting crank - rod using Solidworks

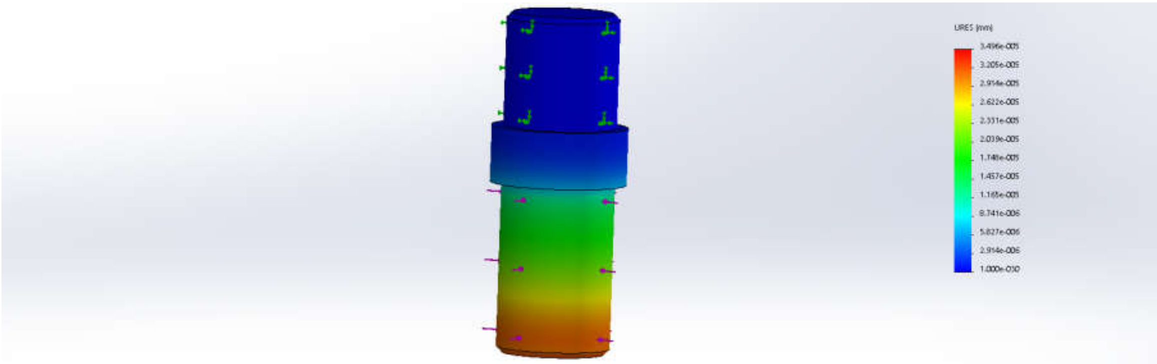

Fig 3.44 The displacement of shaft connecting crank - rod using Solidworks

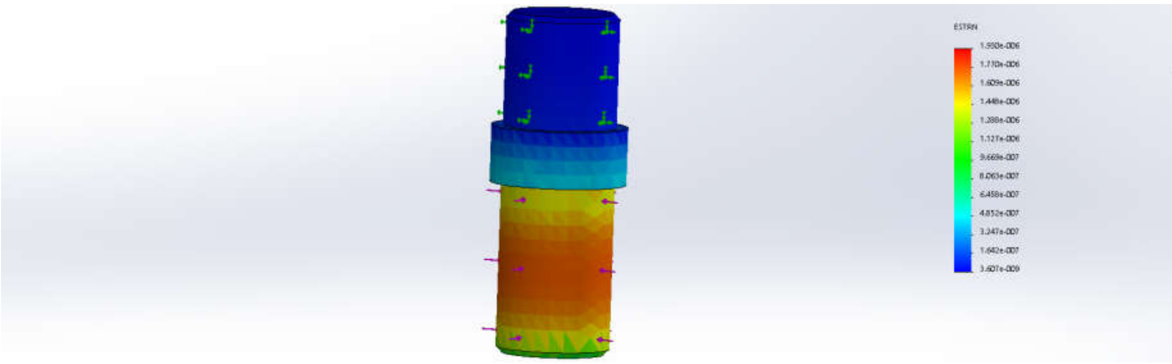

Fig 3.45 The strain of shaft connecting crank - rod using Solidworks

For shaft connecting the rod to disc, we see that the highest moment is $M_{bend} = 60.58\ Nm$.

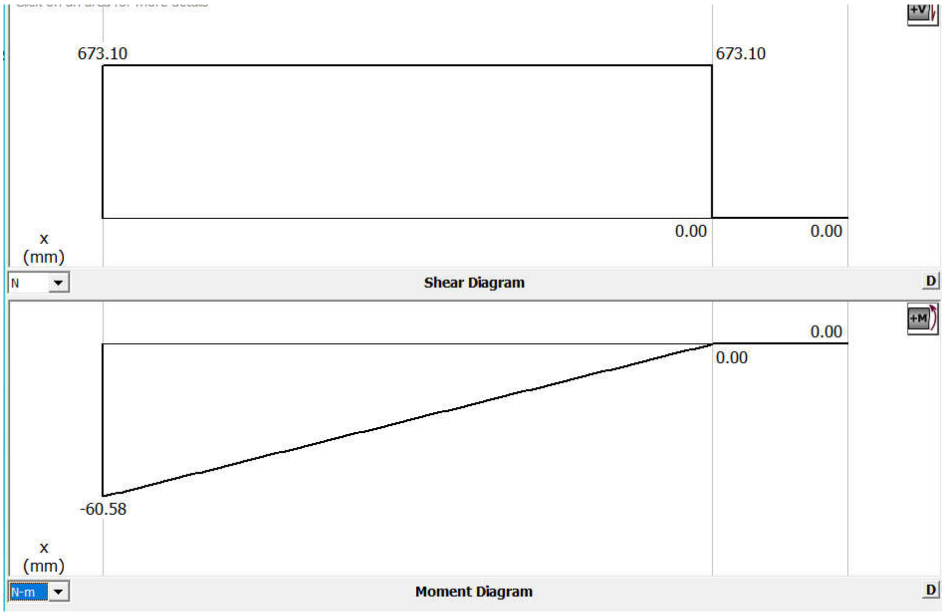

Fig 3.46 Force and moment diagram of shaft connecting the rod to disc

Then, according to [11] the diameter of the shaft at the dangerous section is:

$$d \geq \sqrt[3]{\frac{32M_{bend} \times 10^3}{\pi[\sigma]_{304}}} = \sqrt[3]{\frac{32 \times 60.58 \times 10^3}{\pi \times 80}} = 19.7581\ (mm) \qquad (3.75)$$

Where $[\sigma]_{304} = 80\ MPa$ is the allowable bending stress of 304 steel. [11]

Choose $d = 20\ mm$.

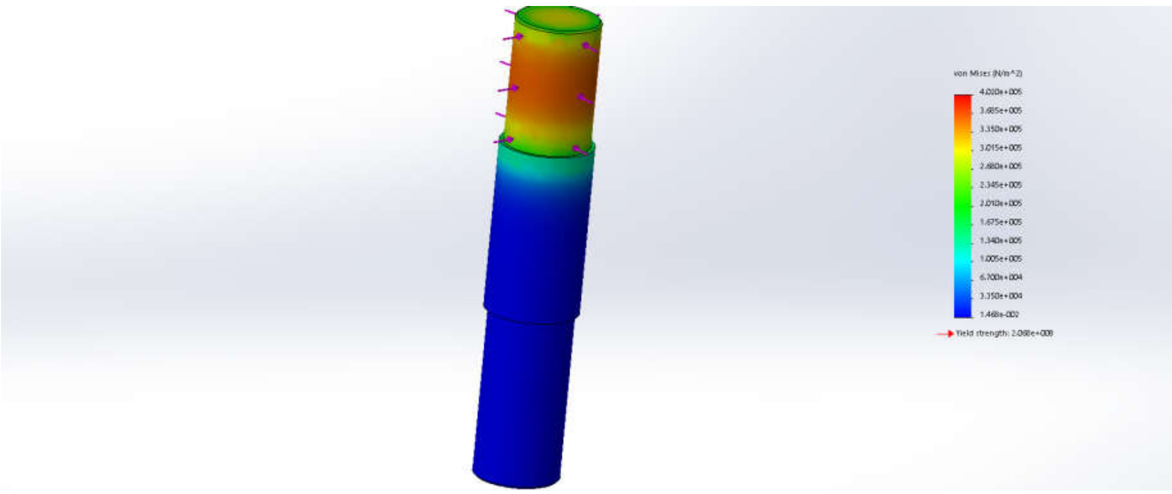

Fig 3.47 The stress of shaft connecting rod – disc using Solidworks

4

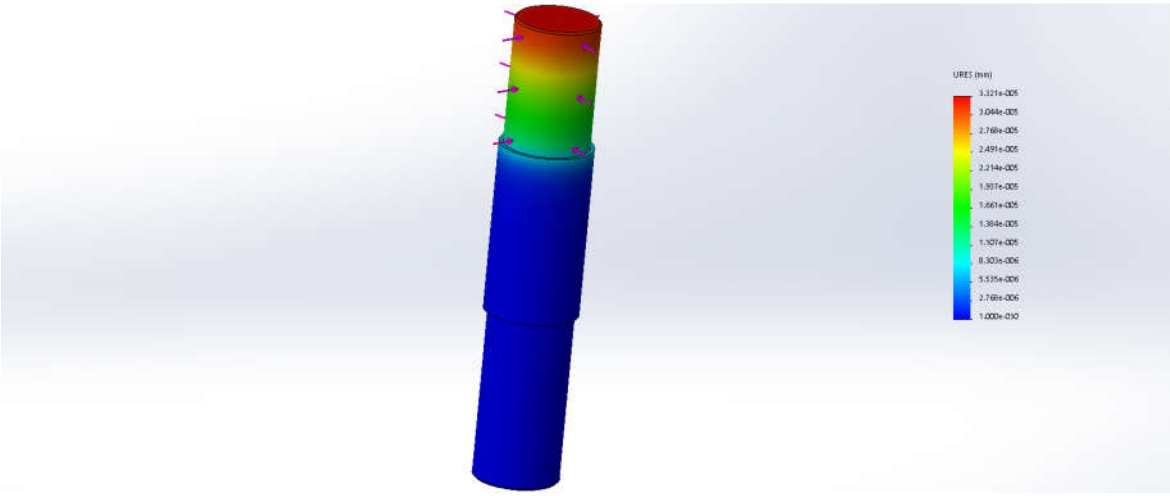

Fig 3.48 The displacement of shaft connecting rod – disc using Solidworks

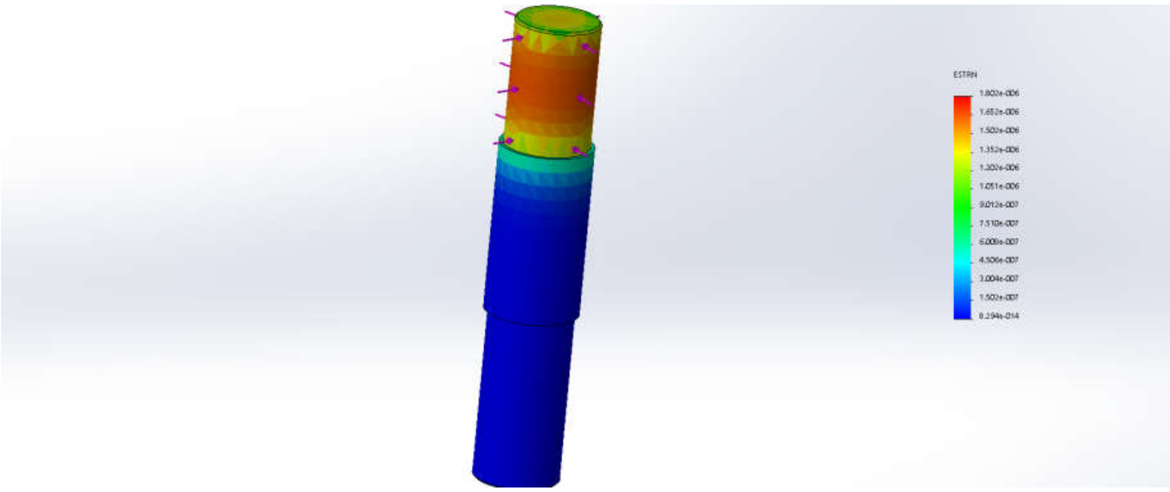

Fig 3.49 The strain of shaft connecting rod – disc using Solidworks

Use Solidworks to check the selected shaft diameter's durability show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition is met.

3.4 Check the static load durability of machine frame parts.

Use Solidworks simulation feature to check the durability of high load machine parts that is not accounted above.

3.4.1 Check durability of motor mount plate

The mount plate is a $5mm$ thick sheet metal to connect the motor to the machine frame. It uses bolt connections. The motor weights $20kg$ and assume the load is split evenly through 4 bolt joints.

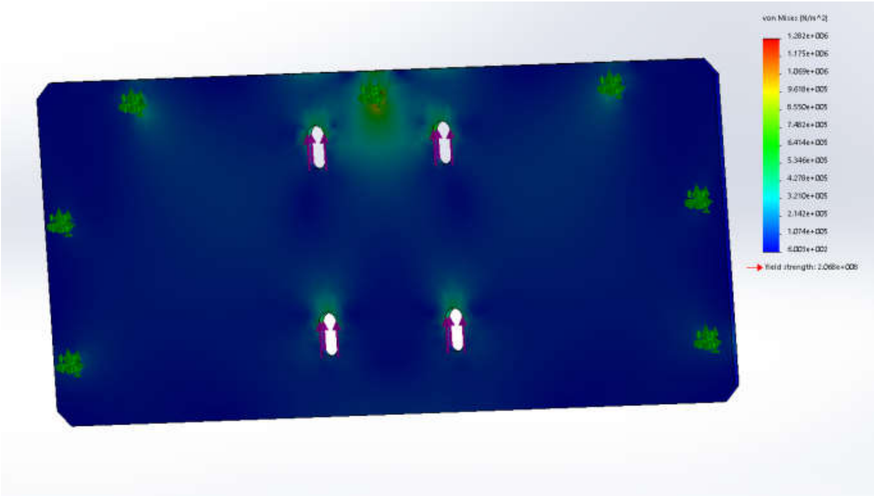

Fig 3.50 The stress of motor mounting plate using Solidworks

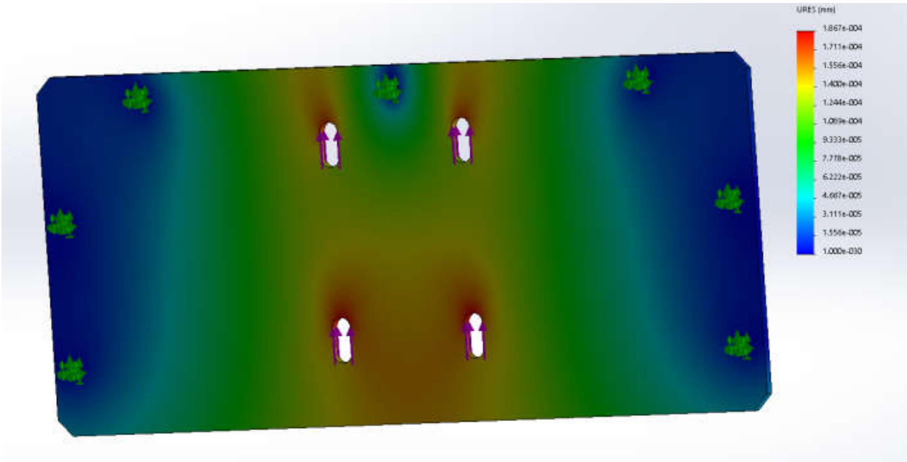

Fig 3.51 The displacement of motor mounting plate using Solidworks

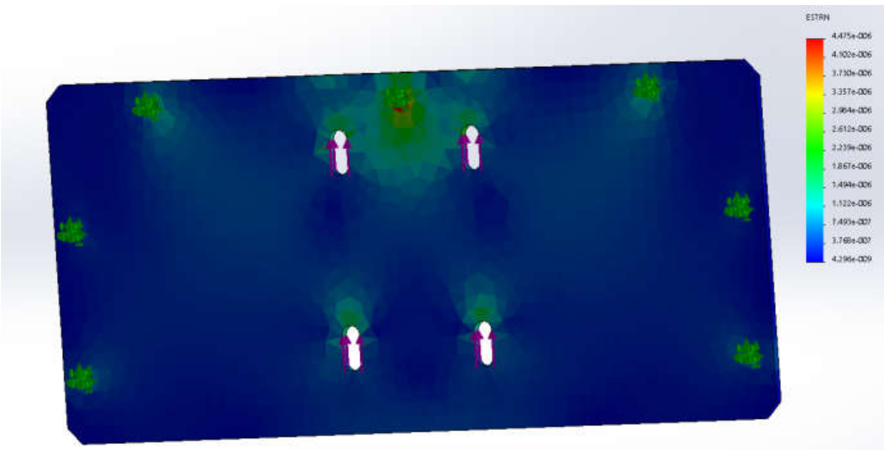

Fig 3.52 The strain of motor mounting plate using Solidworks

The results show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition is met.

3.4.2 Checking the durability of crank

The crank will transmit torque from motor to the rod. The reaction force at joint is $F_{12} = 708.549\ N$.

The results show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition is met.

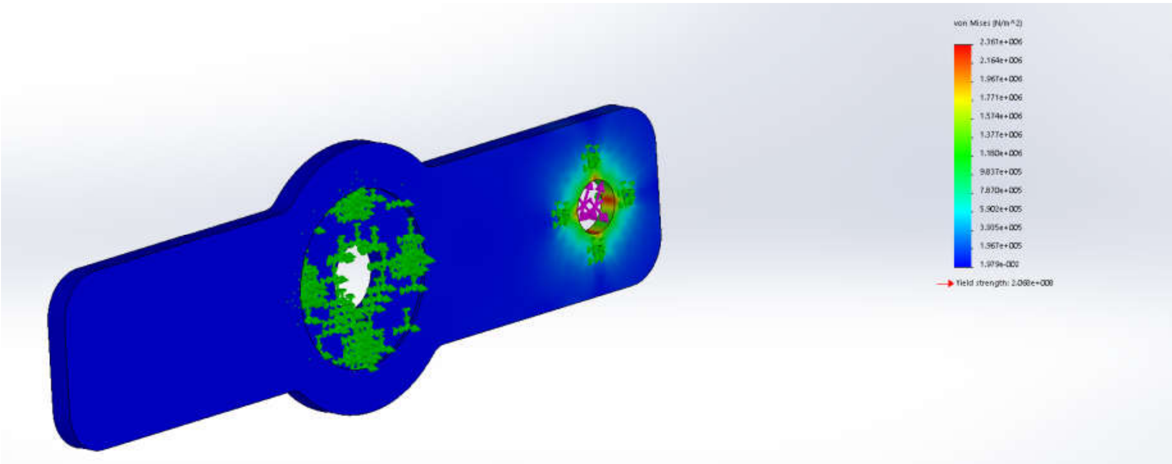

Fig 3.53 The stress of crank using Solidworks

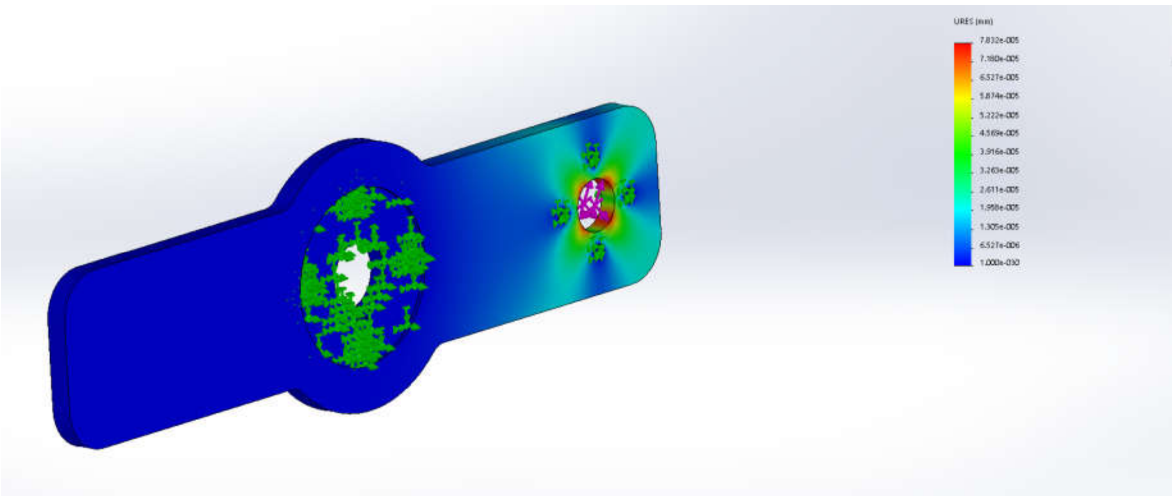

Fig 3.54 The displacement of crank using Solidworks

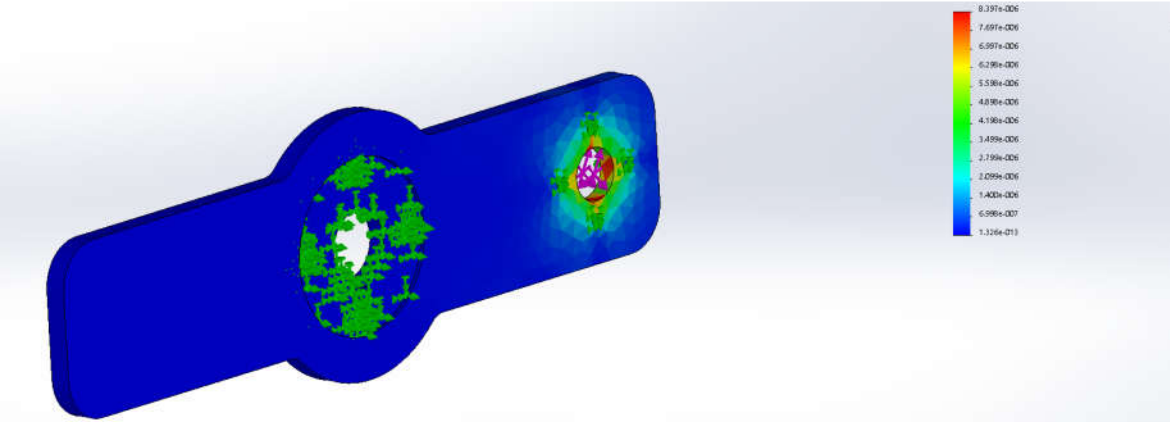

Fig 3.55 The strain of crank using Solidworks

3.4.3 Checking the durability of shaft connecting crank – rod

The rod will transmit torque from crank to the disc. The reaction force at joints is $F_{12} = 708.549\ N$ and $F_{23} = 673.1\ N$

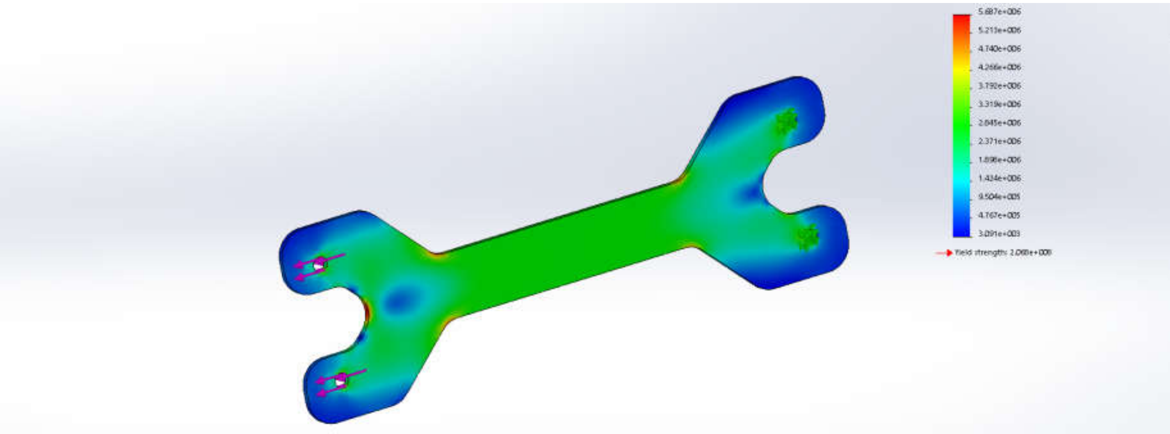

Fig 3.56 The stress of rod using Solidworks

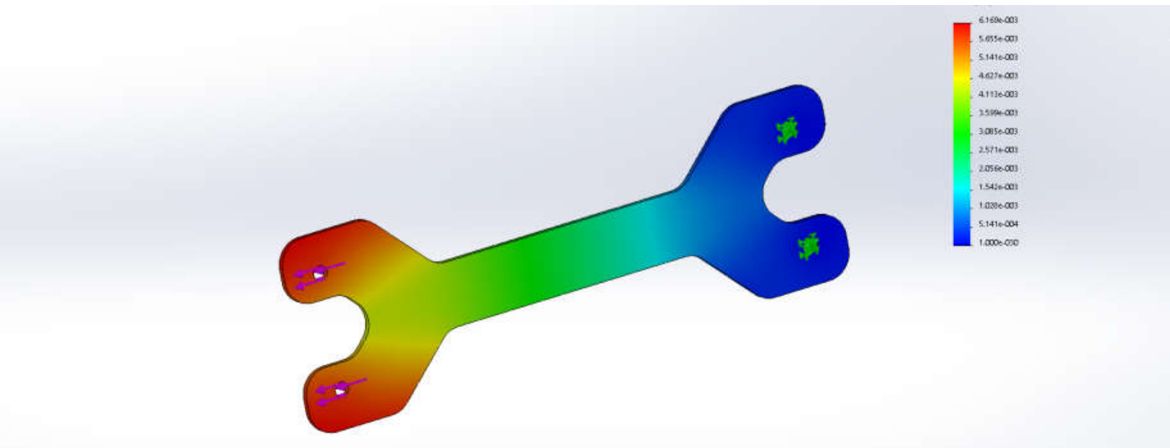

Fig 3.57 The displacement of rod using Solidworks

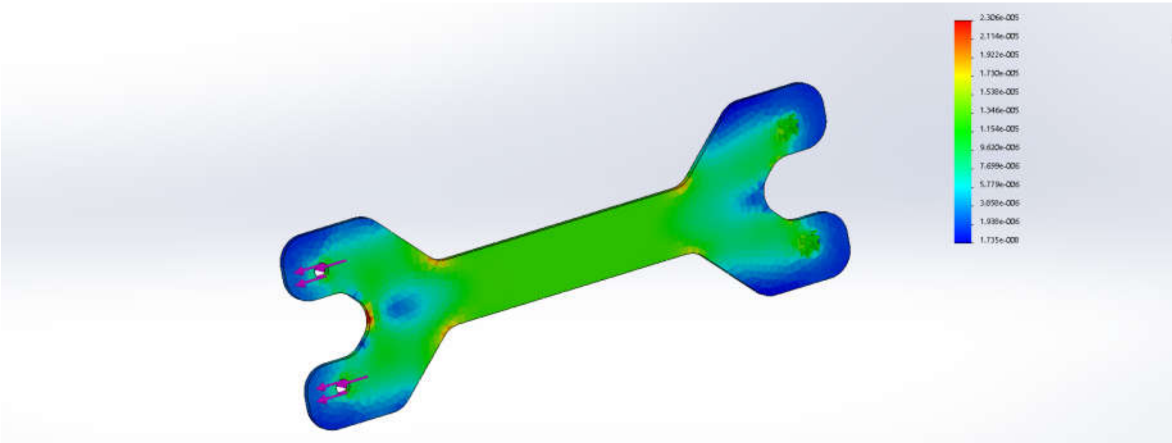

Fig 3.58 The strain of rod using Solidworks

The results show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition is met.

3.4.4 Check durability of peeler frame mounting bracket

The mount plate is a $5mm$ thick sheet metal to connect the peeler frame to the machine frame. It uses bolt group connections. Assume the weight of peeler frame is $20kg$ and assume the load is split evenly through 4 bolt groups.

The results show that all maximum value of simulation stress is under yield strength and displacement and strain are insignificant, so the durability condition is met.

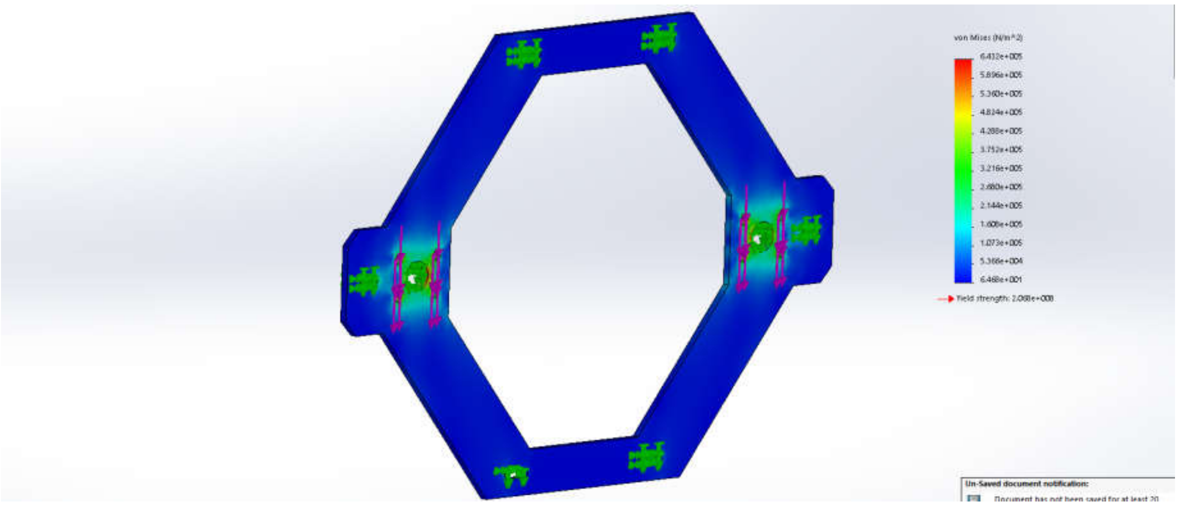

Fig 3.59 The stress of peeler mounting bracket using Solidworks

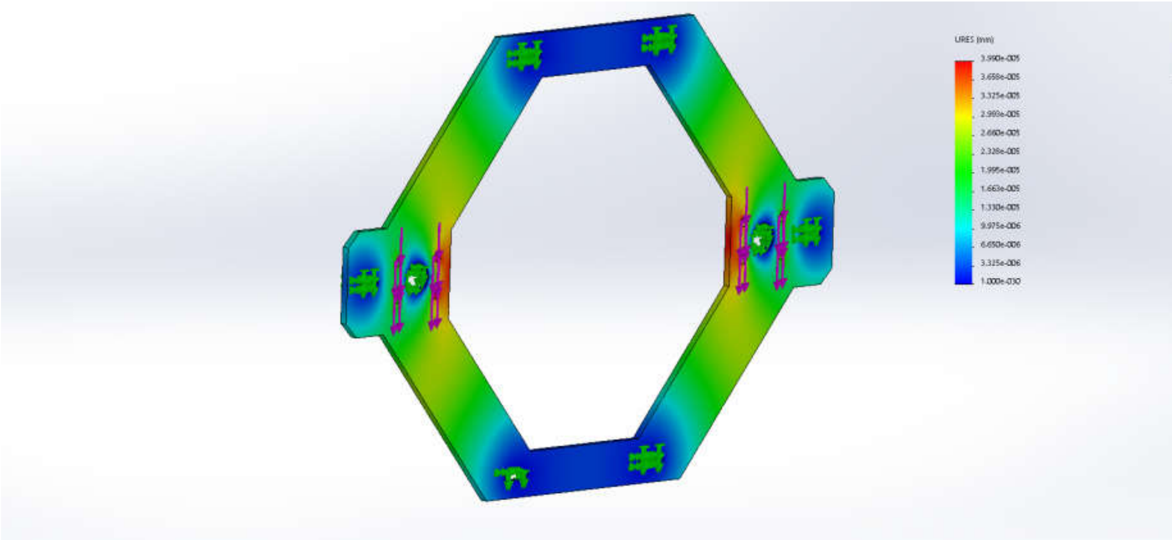

Fig 3.60 The displacement of peeler mounting bracket using Solidworks

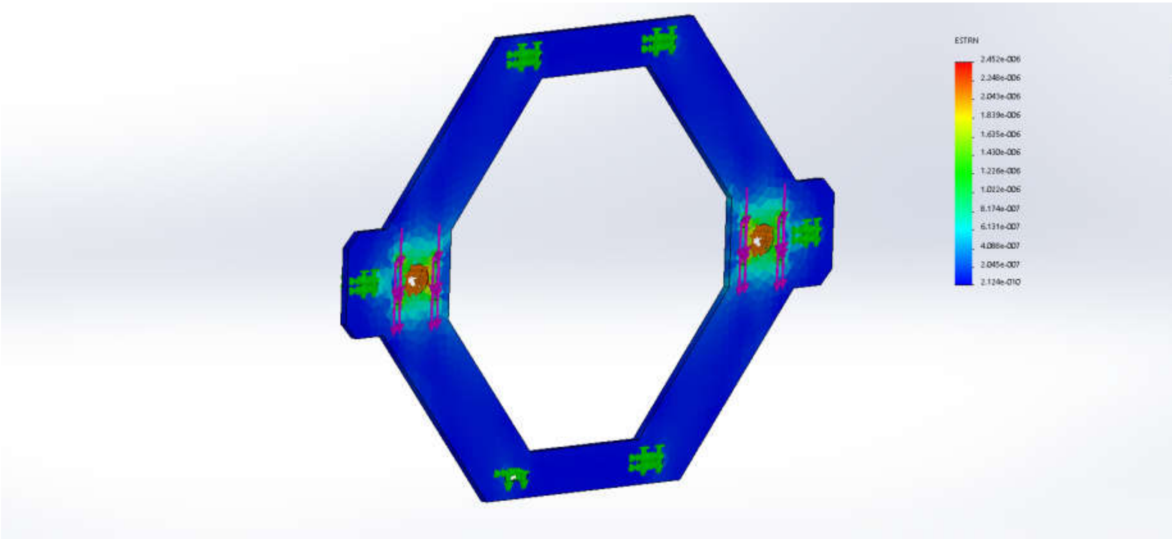

Fig 3.61 The strain of peeler mounting bracket using Solidworks

3.5 Summary of mechanical design

We summarize our calculations of mechanical design in a table below.

Then a 3D model of machine is shown.

Parameters	Value
All shafts material	Stainless steel 304
Peeling section	
Peeling forces	$56.5771\ N$
Motor power	$745\ W$
Motor torque	$86\ Nm$
Peeling blade length	$50\ mm$
Blade holder frame material	Stainless steel 304
Compression spring	SPEC $2 \times 18 \times 68$
Spring forces	$143.3264 - 190.2264\ N$
Spring working length	$10\ mm$
One-way bearing for roller	4752N11
Bearing dynamic load	$3265\ N$
Roller diameter	$22\ mm$
Roller shaft diameter	$8\ mm$
Feeding section	
Tension spring	Metrol $4 \times 28 \times 76$
Spring forces	$342.4174 - 445.1\ N$
Spring working length	$20\ mm$
One-way bearing for roller	1553N15
Bearing dynamic load	$5262\ N$
Roller diameter	$28\ mm$
Roller shaft diameter	$20\ mm$
Machine frame section	
Sliding shaft diameter	$20\ mm$
Bolt diameter for sliding shaft support	$8\ mm$
Bolt diameter for rod shaft support	$8\ mm$
Shaft connecting rod and disc diameter	$20\ mm$
Shaft connecting crank and rod diameter	$20\ mm$

Table 3.8 Summary of mechanical design

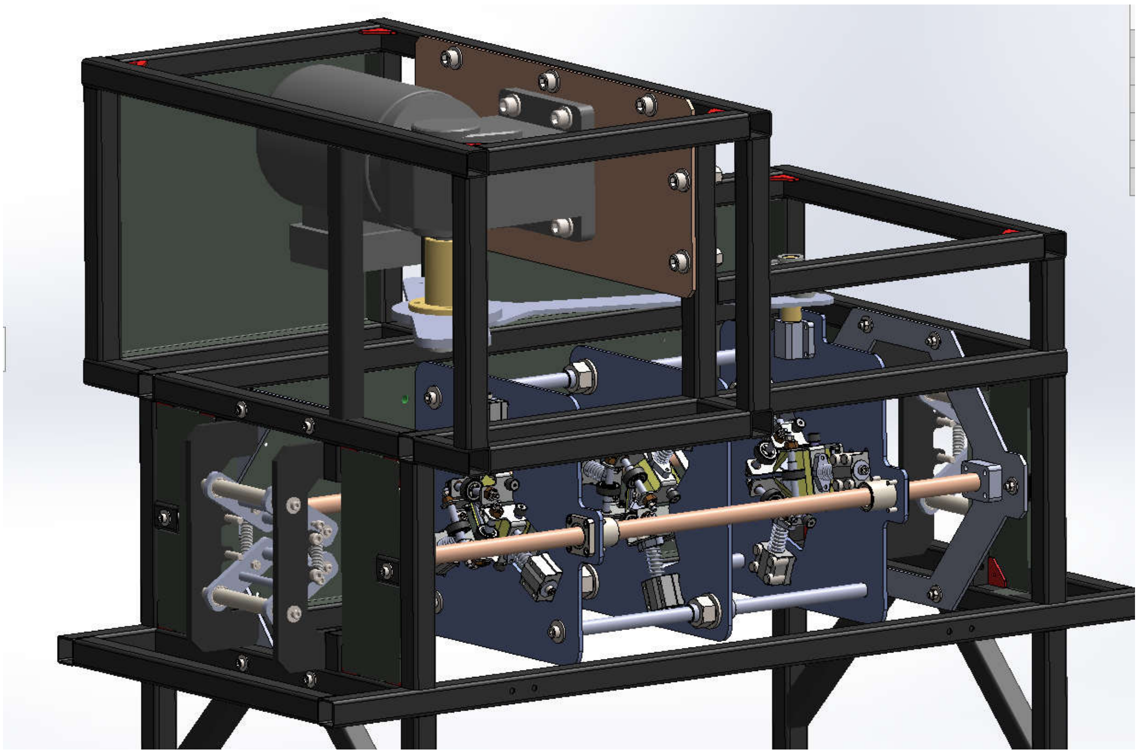

Fig 3.62 Machine design in 3D (covers partially omitted to show the peeler frame)

CHAPTER 4: CONTROLLER DESIGN

4.1 Criteria of controller

The main operating principle of the machine is the reciprocal motion of the peeler module made by slidercrank mechanism, driven by torque from the 3-phase AC motor. In other words, the system does not have any sensors or detection modules, so the control principle of this system is ON-OFF, meaning that when the motor is started, the system will operate without any additional conditions. However, it is necessary to determine the exact speed of the shafts because this parameter will greatly affect the quality of the output product desired by the operator. Therefore, a motor speed control module such as a variable frequency drive is required adjust the speed to match the theoretically calculated one in this thesis.

Criteria:

- Ensure safety when the machine is in operation
- Ease of adjustment and maintenance
- Turn an electrical device on or off using push buttons with relay – contactor
- Have LEDs to show the operating state of the circuit
- Motor do not need to reverse during operation

4.2 Algorithm flowchart

An algorithm flowchart for the machine is presented below. The calculation and selection of necessary electrical equipments is shown in Chapter 5.

First the miniature circuit breaker will be closed to provide power to the system, led H2 will turn ON to indicate that control circuit has electricity and but the inverter does not. If any errors or fault occur, press Emergency stop button to stop electricity flowing into system.

Press START1 button to power the inverter. From here, led H1 is ON, H2 is OFF and we can config the inverter parameters as stated above. After doing this once, the future run does not need initial adjustment unless we want to change it. Click the switch Run/Stop to start the motor. To adjust the speed, rotate the potentiometer.

To stop the motor, click the switch Run/Stop again. Motor will slow down gradually according to inverter's setting. To run motor again, click the switch Run/Stop again.

To turn off the inverter, press STOP1 and led H2 will be ON, H1 will be OFF. To run inverter again, press START1, then led H1 is ON, H2 is OFF.

To disconnect all electricity, press Emergency stop button. To power the system again, rotate the Emergency stop button. Led H2 will turn ON to indicate that control circuit has electricity

In case of overcurrent in power circuit, the overload relay inside the inverter will trip and disconnect electricity from the whole circuit.

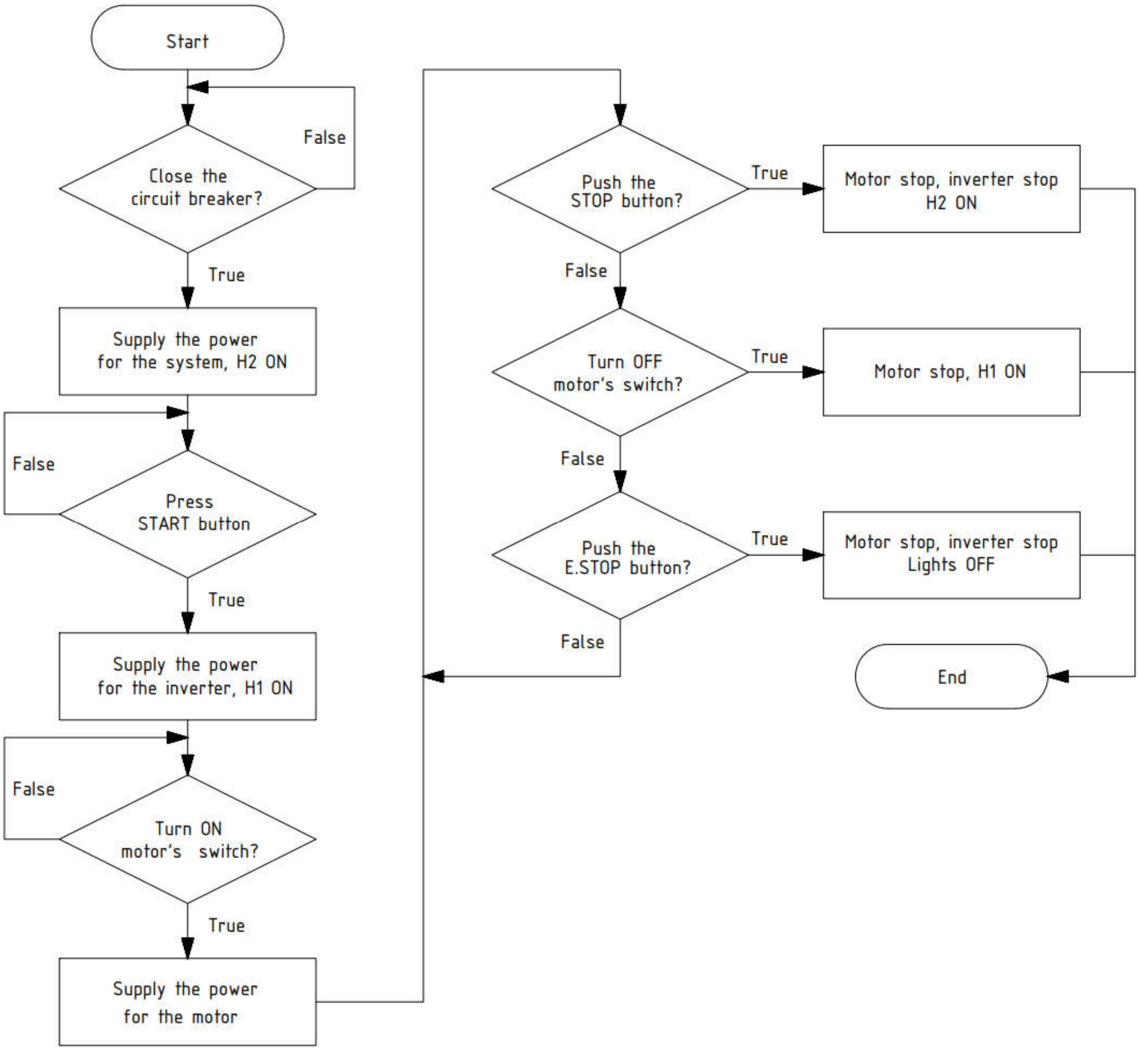

Fig 4.1 Machine operation flowchart

CHAPTER 5: ELECTRICAL DESIGN

Before starting the design, calculation and selection of electrical equipment, we summarize the power of each module to calculate the total power required for the system.

Using one 3-phase AC electric motor with the required power $P_{req1} = 745\ W$ for the peeler module.

Requirements:

- Protective devices for power circuit
- Protective devices for control circuit
- Use variable frequency drive to adjust difference of speed in theory and reality

5.1 Peeler module

5.1.1 Calculation of the electrical equipment for powering motor

The current of overload relay with starting current safety coefficient equal 3 [17]:

$$I_{relay} = 3I_{motor} = 3 \times 1.4 = 4.2\ (A)$$

The rated current of the system [18]:

$$I_{MCB} = 3I_{motor} = 3 \times 1.4 = 4.2\ (A)$$

With calculated current:

Select Circuit breaker Schneider GZ1E10, 4-6.3A tripping current range, 3 phase 380VAC.

Select Schneider LC1D09M7 contactor, 9A rated current, 3 phase 380VAC power circuit, 220VAC control circuit, with 3 NO contacts and 1 NO, NC auxiliary contacts.

Select fuse DF2CA05, 5A rated current for control circuit.

5.1.2 Calculation of the wire

The conducting wire cross-sectional area is:

$$S = \frac{I_{MCB}}{J} = \frac{4}{6} = 0.6667\ (mm^2)$$

Where J is the density of current allowed to flow through a copper wire [19]

The diameter of the wire:

$$S = \frac{\pi D^2}{4} \rightarrow D = \sqrt{\frac{4S}{\pi}} = 0.9213\ (mm)$$

Choose wire diameter $D = 1.13\ (mm)$ corresponding $S = 1\ mm^2$ according to IEC 60228 standard for wire cross-sectional area.

5.2 Variable frequency drive selection

Since the calculated speed and motor speed have a little difference, we can use variable frequency drive (VFD), also called inverter to control the speed of 3 phase motor. Choose inverter Mitsubishi FR-E740-0.75K corresponding with the motor power $P = 745W$.

Parameter	**Value**
Applicable motor capacity	$750W$
Rated capacity	$2.0\ kVA$
Rated output current	$2.6\ A$
Overload current rating	150% in 60s and 200% in 3s
Rated output voltage	Three phase 380 to 480V
Rated input voltage	Three phase 380 to 480V 50Hz/60Hz
Permissible AC voltage fluctuation	$325 - 528V$, 50/60Hz
Permissible frequency flucation	Fluctuation $\pm 5\%$
Output frequency range	From $0.2 - 400Hz$
Frequency setting resolution	$0.01Hz$
Frequency accuracy	$\pm 0.5\%$

Table 5.1 FR-E740-0.75K inverter specifications.

5.3 Variable frequency drive parameters setup

The adjustment of parameters for the Mitsubishi VFD FR – E740 – 0.75K inverter parameters for the motor 3 phases 380 $VAC - 0.75\ kW$ are shown as below.

Inverter parameter	Value	Function description
P-79	2	Run inverter in external operation mode, to make use of external Run/stop switch and potentiometer to change output frequency.
P-80	0.75	Motor power
P-81	4	Number of motor poles
P-82	0.7	Motor no load current
P-9	1.6	Rated current of motor
P-83	380	Rated voltage of the motor
P-1	50	Maximum output frequency of the inverter
P-2	0	Minimum output frequency of the inverter
P-72	1	Default carrier frequency
P-150	150	Motor overcurrent protection threshold
P-22	120	Motor overcurrent protection during operation
P-251	1	Activate the protection for the loss of inverter output phase
P-872	1	Activate the protection for the loss of inverter input phase
P-7	5	Acceleration time of motor from 0Hz in seconds
P-8	5	Deceleration time of motor from max frequency in seconds
P-244	1	Activate fan when the inverter is hot

Table 5.2 Mitsubishi FR – E740 – 0.75K initial setup value

5.4 Electrical diagram and principle

The AC motor's speed will be adjusted through the usage of potentiometer on inverters. From the 3 phase 380VAC power source will directly distribute power to the variable frequency drive FR-E740-0.75 K. When the circuit breaker is closed and the START button is clicked, it will activate the coil of contactor K1 and the green LED H1 are turned on. The variable frequency drive has power but the motor is not yet started. We use toggle switch S to select the start – stop mode to activate the motor.

An industrial potentiometer of 1000Ω with three pins 1-2-3 is connected correspondingly to terminal 10-2-5 of the variable frequency drive to adjust the frequency value and thereby control the motor speed. The system operates in one direction, so the motor only needs to rotate in one direction.

When the STOP button is clicked or relay of the variable frequency drive trips, the coils of the contactor K1 will be de-energized and the red LED H2 will turn on. The EMC (Emergency stop button) button is used to interrupt the system circuit in case of emergencies.

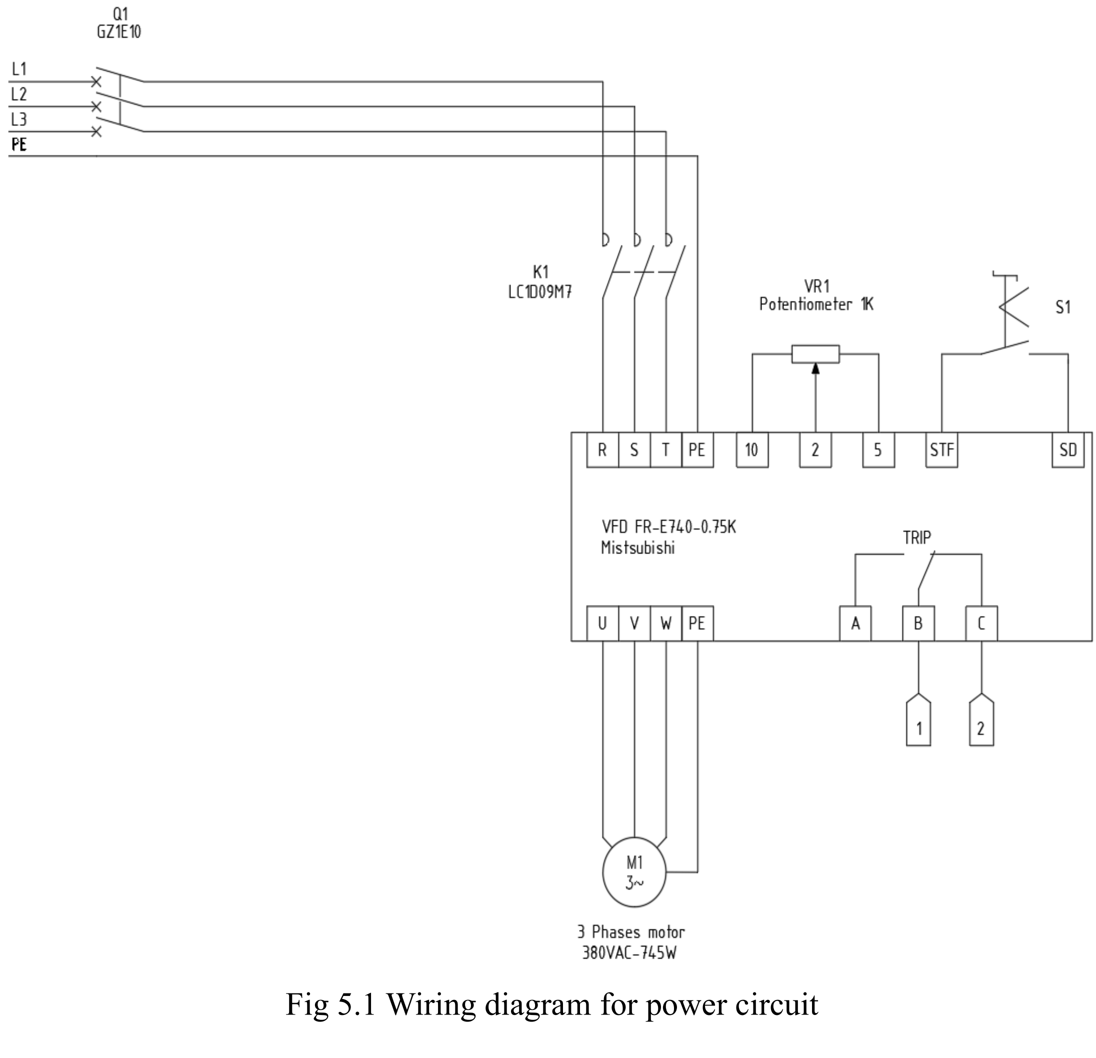

Fig 5.1 Wiring diagram for power circuit

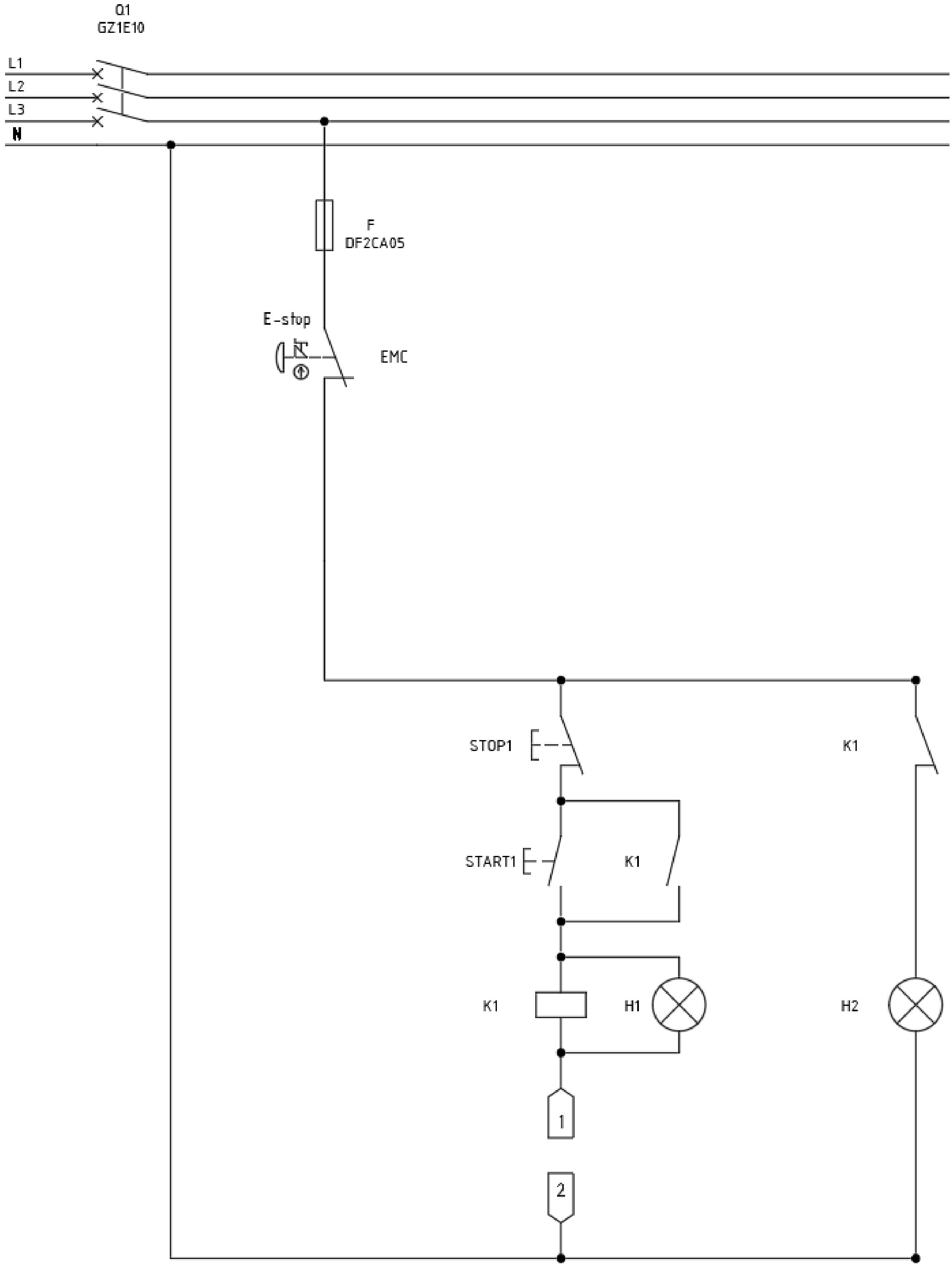

Fig 5.2 Wiring diagram for control circuit

CHAPTER 6: SIMULATION

Use Solidworks motion study feature to show the movements of the machine.

In peeling phase, when the peeling frame move toward the feeding section, the feeder's roller hold tight the sugarcane through spring force and doesn't rotate. When the peeler's roller contacts the sugarcane, it rolls on the sugarcane surface through spring force and help hold it to ensure the blade can reach enough depth of cut.

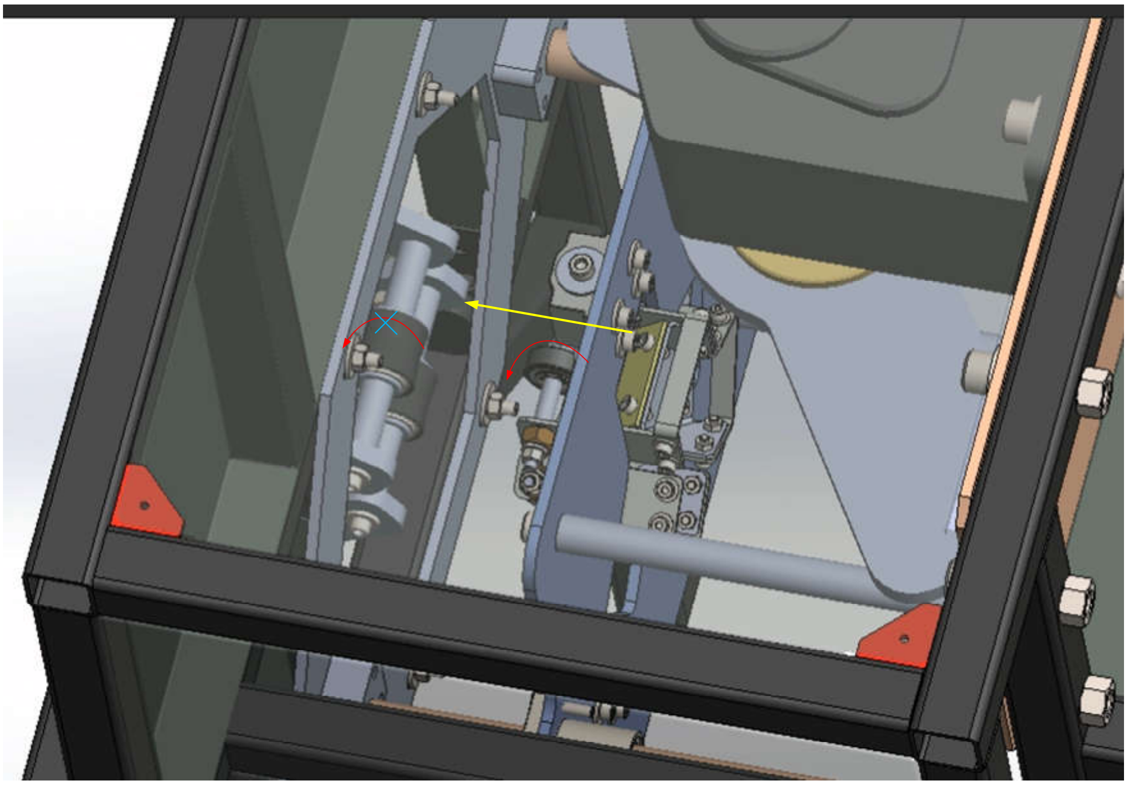

Fig 6.1 Motion of blade rollers and feeder rollers in peeling phase.

In pulling phase, when the peeling frame move away from the feeding section, the feeder's roller rotates and let go of the sugarcane. When the peeler's roller contacts the sugarcane, it rolls on the sugarcane surface and help hold it to ensure the blade can reach enough depth of cut.

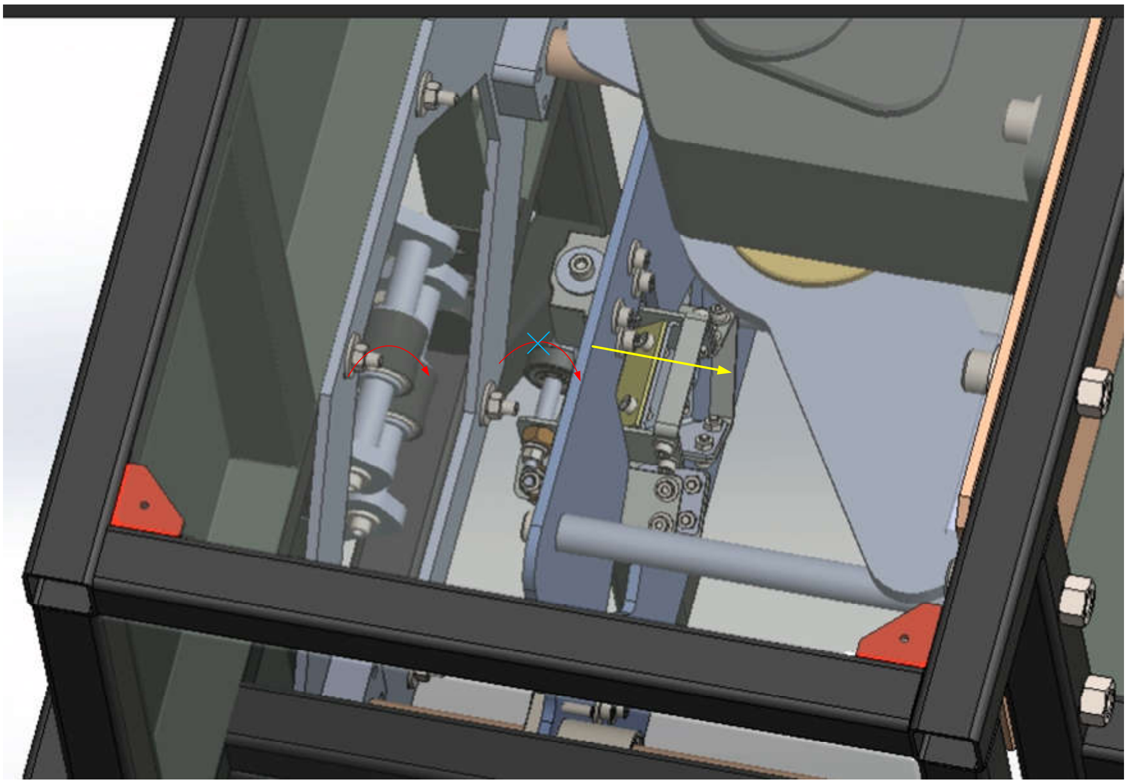

Fig 6.2 Motion of blade rollers and feeder rollers in pulling phase.

CHAPTER 7: CONCLUSIONS

7.1 Accomplishments

The thesis has provided an overview of the necessity of sugarcane, current situation of sugarcane industry in Vietnam, and the operation that is involved in the sugarcane processing.

In the process of designing the mechanical part for the sugarcane peeling machine, the main objective is to ensure productivity and working safety regulations for the subsequent production machines such as cutting and packaging. With a target output of 240 canes/hour, the peeling machine needs to be designed to peel the required amount of flour within the allowed time.

Regarding the design of the electrical circuit and control system, the requirements are to be simple yet ensure safety and convenience for the users. The electrical circuit and control system have the ability to manually change the parameters to adapt to minor difference of speed in theory and reality. It is also able to pause the machine immediately in case of any incidents, ensuring the safety of the operator.

7.2 Limitations

In the mechanical aspect, the machine has not accounted for vibration. Additionally, the configuration of the peeling blade is lacking bagasse cutter which might stuck on the machine and reduce efficiency.

Limited in simulation of machine motion due to the complexity in modelling the machine, and there is no actual prototype yet, so we cannot verify the productivity goal.

The preparation of input materials for the system and the outfeed are handled manually by workers.

Machine needs daily cleaning to prevent hazardous or toxic elements to appear and keep the machine last longer.

In terms of electrical aspect, the system does not have an immediate brake for the motor. It takes some time to stop, which might endanger the workers in case of emergency.

7.3 Development direction

Development direction involves mostly mechanical design: The peeler to need account for vibration that can occur due to large load by a damping mechanism. Additionally, the peeler module should include a bagasse cutter to prevent it from stucking in the blade. A gutter should also be connected to machine frame underneath the peeler to catch all the bagasse.

Moreover, a fast-braking mechanism should be developed either mechanically or electrically to ensure safety when the machine is in use.

Research on automation for the feeding process and the transfer process at the outfeed.

Implement actual model to test and callibration of mechanism such as gripping and cutting.

In electrical aspect: research and design to use an HMI to input the desired parameters for the machine to run instead of adjusting speed through the use of potentiometers to develop a more user – friendly and precise interface.

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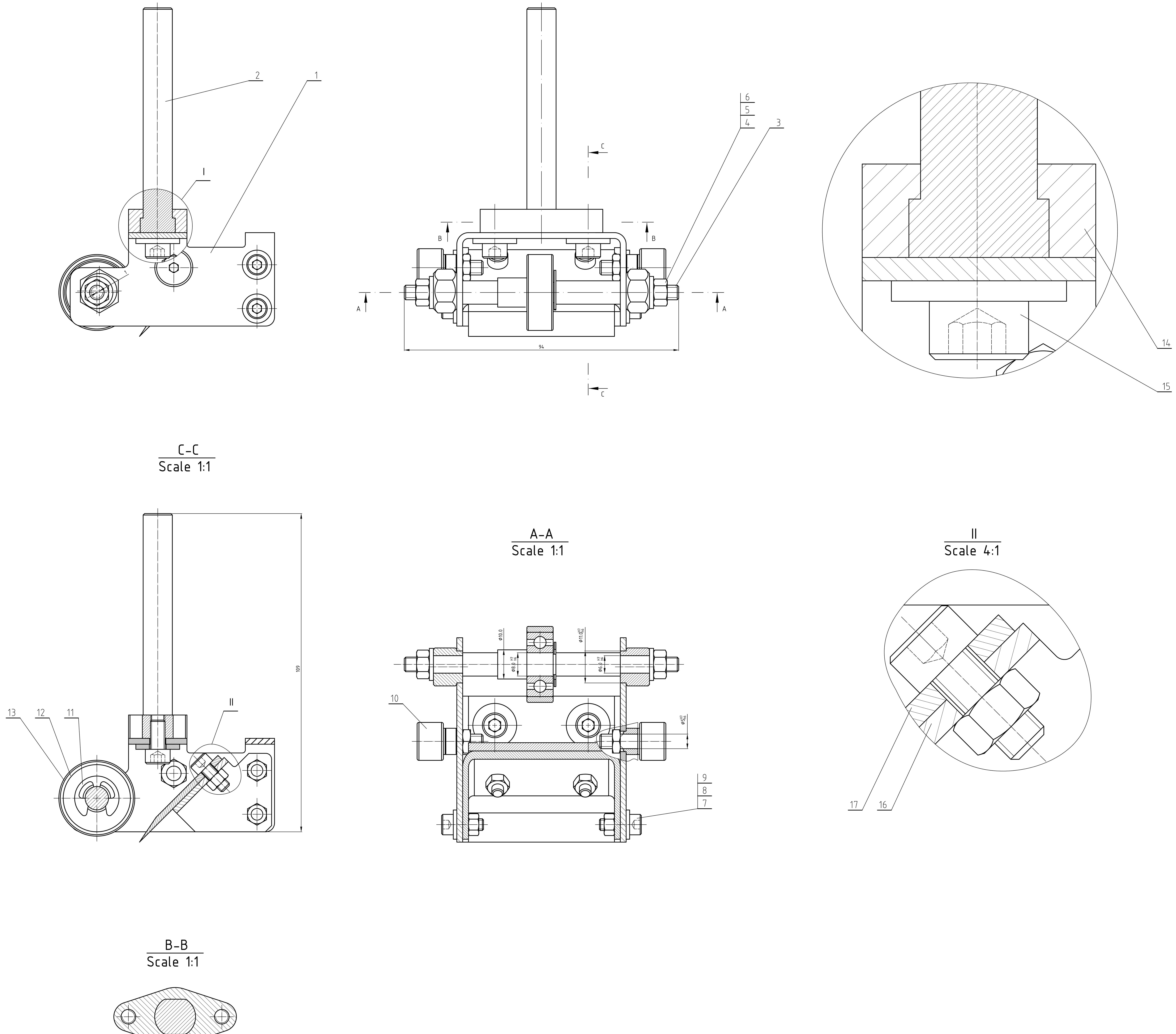
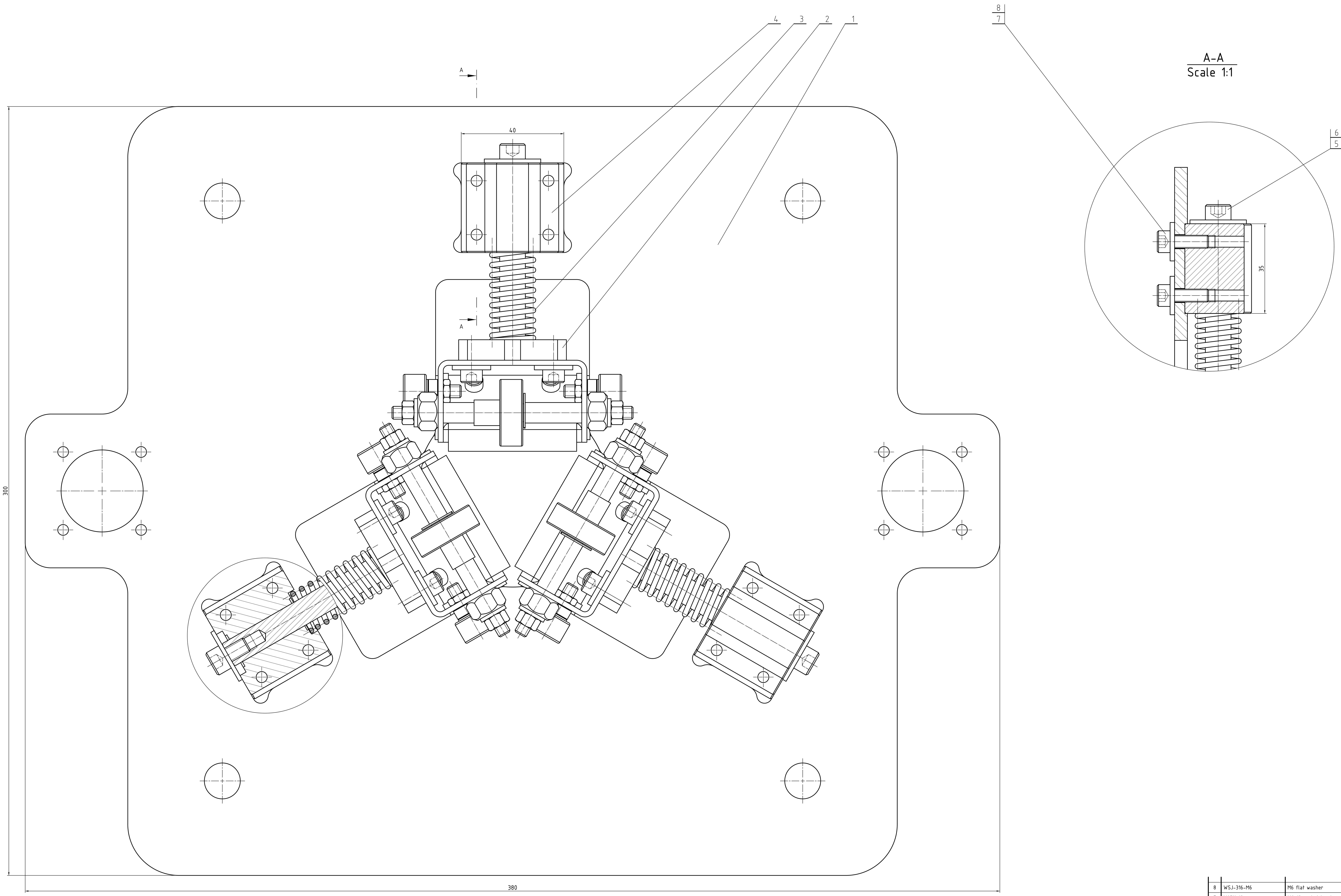


17	ME1-007	Peeling blade	1	SUS304	Workshop
16	ME1-006	Blade holder	1	SUS304	Workshop
15	M5SH-10	M5 socket head screw	2	SUS304	Misumi
14	ME1-005	Shaft and bracket connector	1	SUS304	Workshop
13	ME1-004	Rubber ring	1	Polyurethane	Workshop
12	4752N11	One-way bearing	1	SUS304	Mcmaster
11	90768A103	Retaining ring E type	1	SUS304	Mcmaster
10	1470T643	Threaded track roller	2		Mcmaster
9	HNT-SUS-M4	M4 hexagon nut	6	SUS304	Misumi
8	WSJ-316-M4	M4 flat washer	6	SUS316	Misumi
7	M4SH-10	M4 socket head screw	4	SUS304	Misumi
6	B09CYK9P43	M5 Eccentric nut	2	SUS304	Amazon
5	WSJ-316-M5	M5 flat washer	4	SUS316	Misumi
4	HNT-SUS-M5	M5 hexagon nut	4	SUS304	Misumi
3	ME1-003	Roller shaft	1	SUS304	Workshop
2	ME1-002	Spring shaft	1	SUS304	Workshop
1	ME1-001	U bracket	1	SUS304	Workshop
No.	Symbol	Part's Name	Qty	Material	Note

A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE					Capstone Project		
Func.	Full name	Sign	Date	PEELER	Quantity	Weight	Scale
Design	Huynh Ba Loc		25/12				2:1
Instruct.	Vo Tuong Quan				Sheet: 1	Total sheet: 6	
Approve.	T. Viet Hong			C1	HO CHI MINH CITY UNIVERSITY OF TECHNOLOGY FACULTY OF MECHANICAL ENGINEERING CLASS CC19CDT1		



No.	Symbol	Part's Name	Qty	Material	Note
8	WSJ-316-M6	M6 flat washer	3	SUS316	Misumi
7	M6SH-10	M6x10 socket head screw	3	SUS304	Misumi
6	WSJ-316-M5	M5 flat washer	12	SUS316	Misumi
5	M5SH-12	M5x12 socket head screw	12	SUS304	Misumi
4	9338T52	Linear bearings	3		Mcmaster
3	D250	Compression spring	3	DIN 17223 - C	Mcmaster
2	C1	Peeler	3		Workshop
1	ME1-001	Peeler disc	2	SUS304	Workshop

A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE

Capstone Project

Func.	Full name	Sign	Date
Design	Huynh Ba Loc		25/12
Instruct.	Vo Tuong Quan		
Approve.	T. Viet Hong		

PEELER DISC

C2

Quantity	Weight	Scale
		2:1

Sheet: 2 | Total sheet: 6

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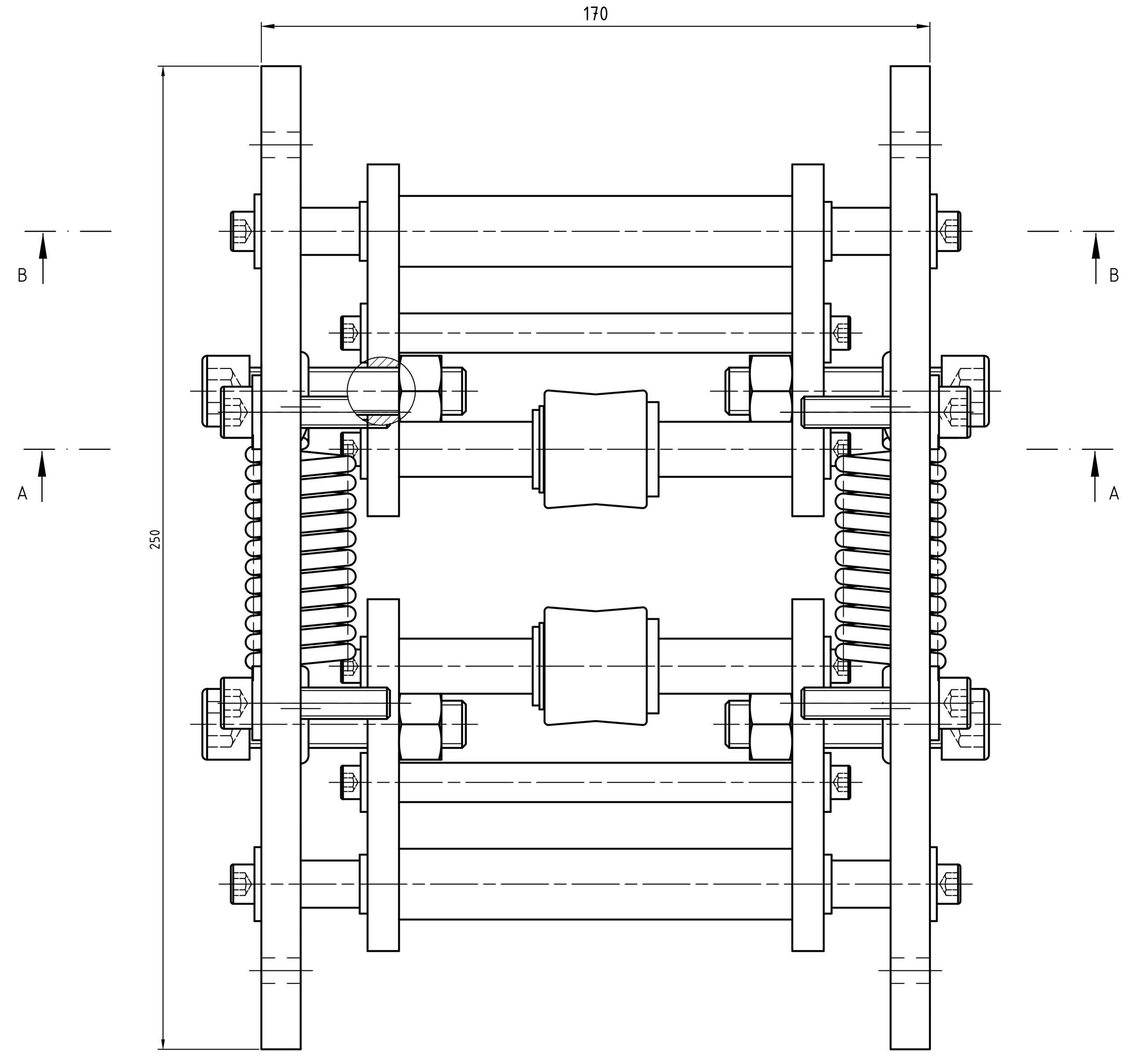
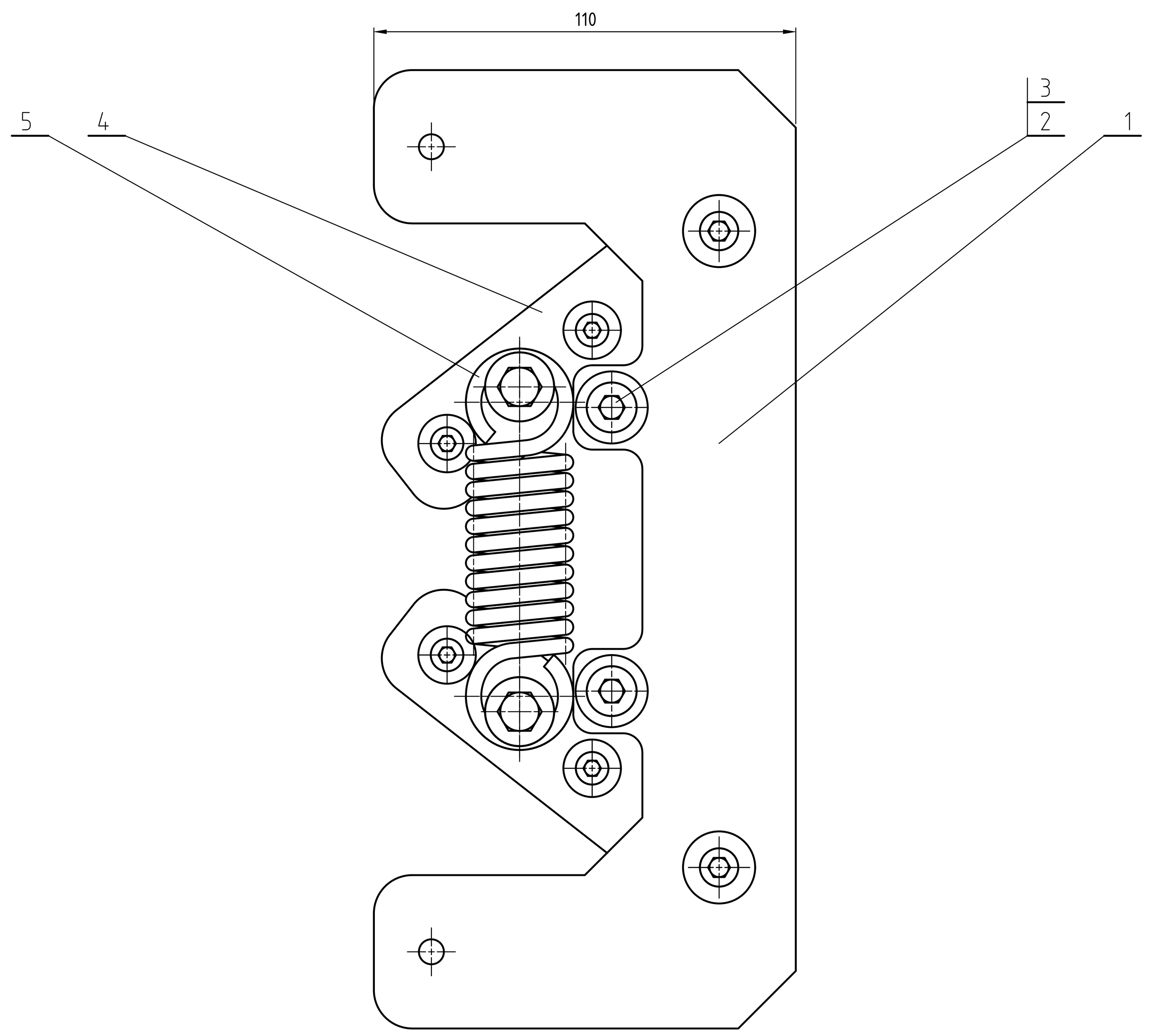
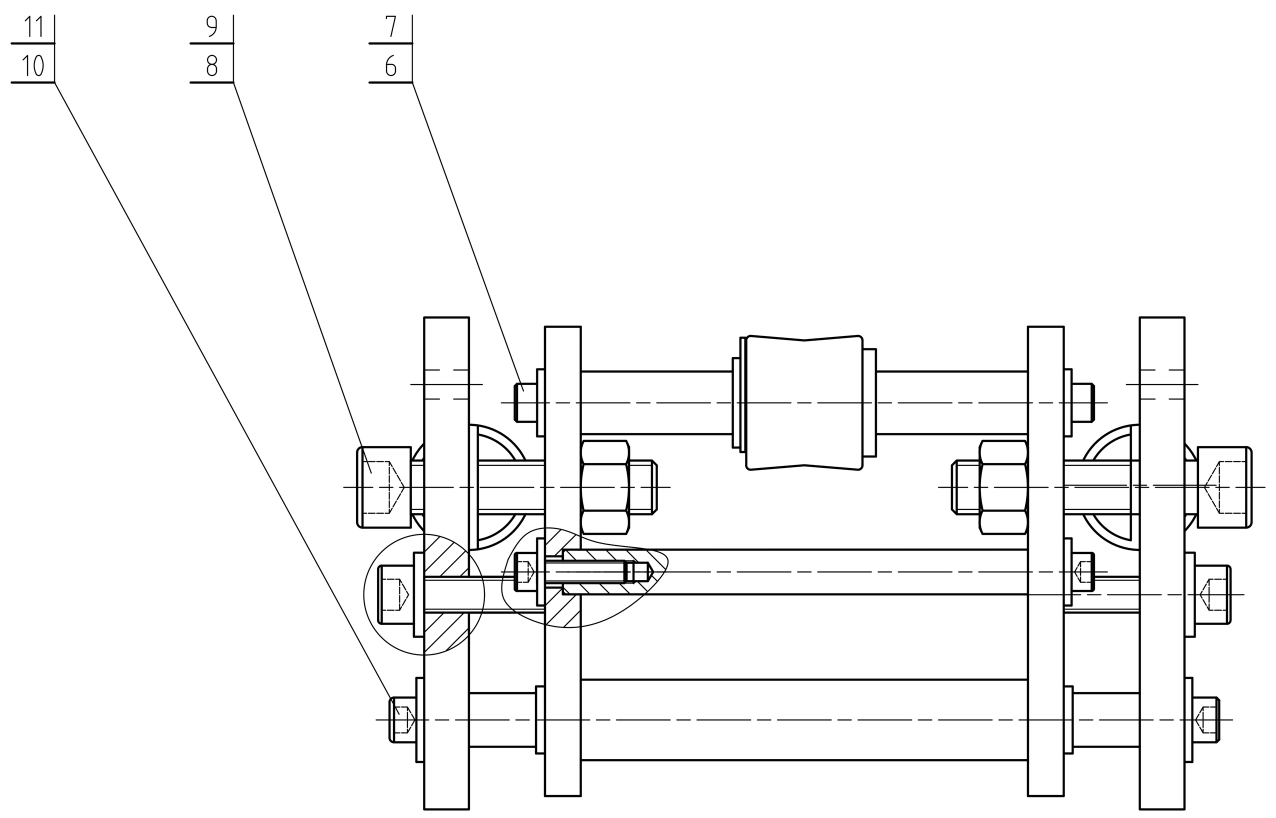
CLASS CC19CDT1







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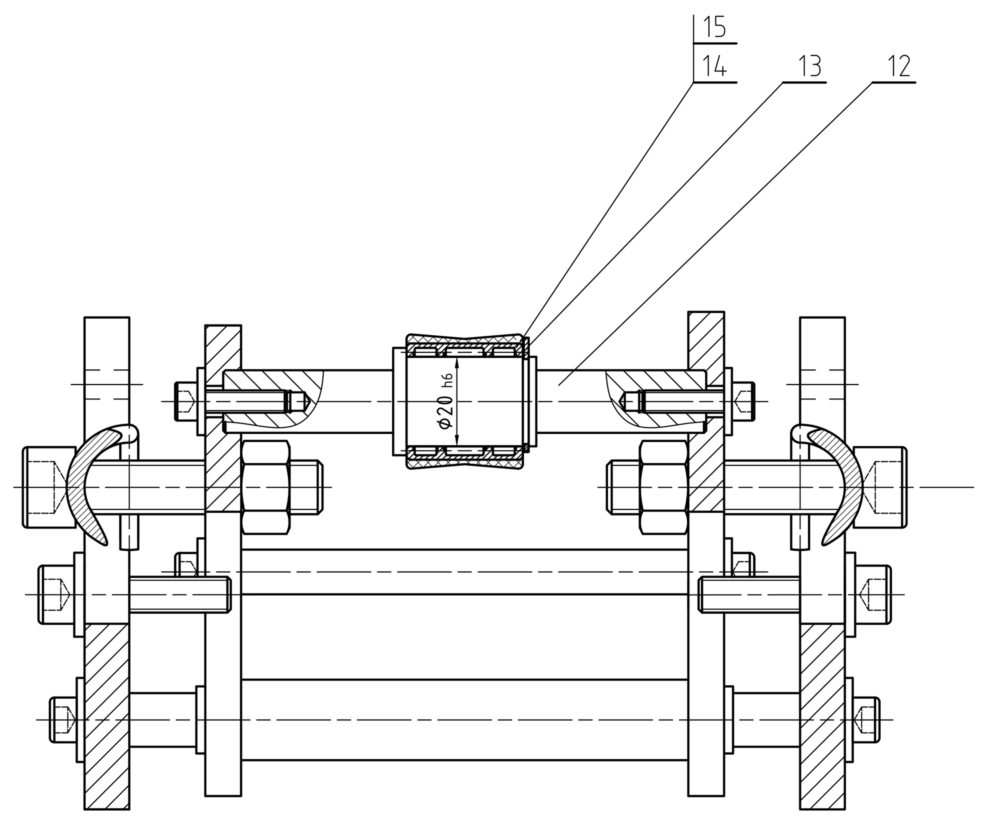

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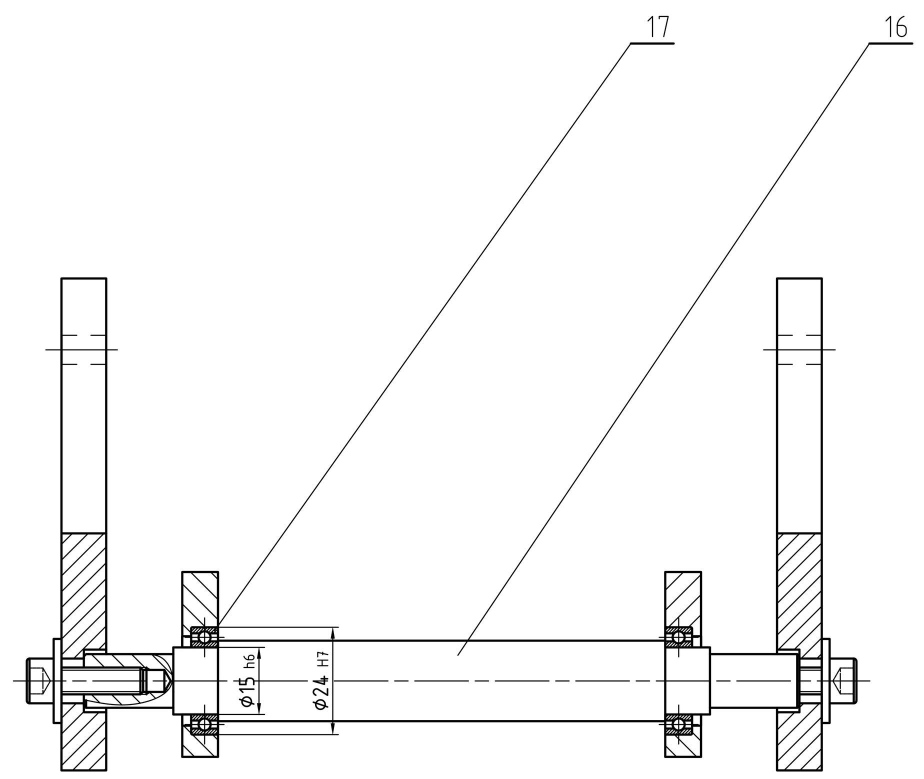

No.	Symbol	Part's Name	Qty	Material	Note
17	AFBMA 20.1 - 28 -15	Radial ball bearing	1		Mcmaster
16	ME1-004	Passive rotating shaft	2	SUS304	Workshop
15	ME1-003	Rubber rings	2	Polyurethane	Workshop
14	1553N15	One-way needle roller bearing	2		Mcmaster
13	STWS-20	Retaining ring C type	2	SUS304	Misumi
12	ME1-002	Feeder roller shaft	2	SUS304	Workshop
11	WSJ-316-M6	M6 flat washer	4	SUS316	Misumi
10	M6SH-20	M6 socket head screw	4	SUS304	Misumi
9	HNT-SUS-M12	M12 hexagon nut	4	SUS304	Misumi
8	M12SH-55	M12 socket head screw	4	SUS304	Misumi
7	WSJ-316-M5	M5 flat washer	8	SUS316	Misumi
6	M5SH-20	M5 socket head screw	8	SUS304	Misumi
5	SES42876	Tension spring	2	SUS301	Metrol
4	SES42876	Mounting plate	4	SUS304	Workshop
3	WSJ-316-M8	M8 flat washer	4	SUS316	Misumi
2	M8SH-35	M8 socket head screw	4	SUS304	Misumi
1	ME1-001	U bracket	2	SUS304	Workshop

A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE

Capstone Project

Func.	Full name	Sign	Date
Design	Huynh Ba Loc		25/12
Instruct.	Vo Tuong Quan		
Approve.	T. Viet Hong		

FEEDER

C3

Quantity	Weight	Scale
		1:1

Sheet: 3 | Total sheet: 6

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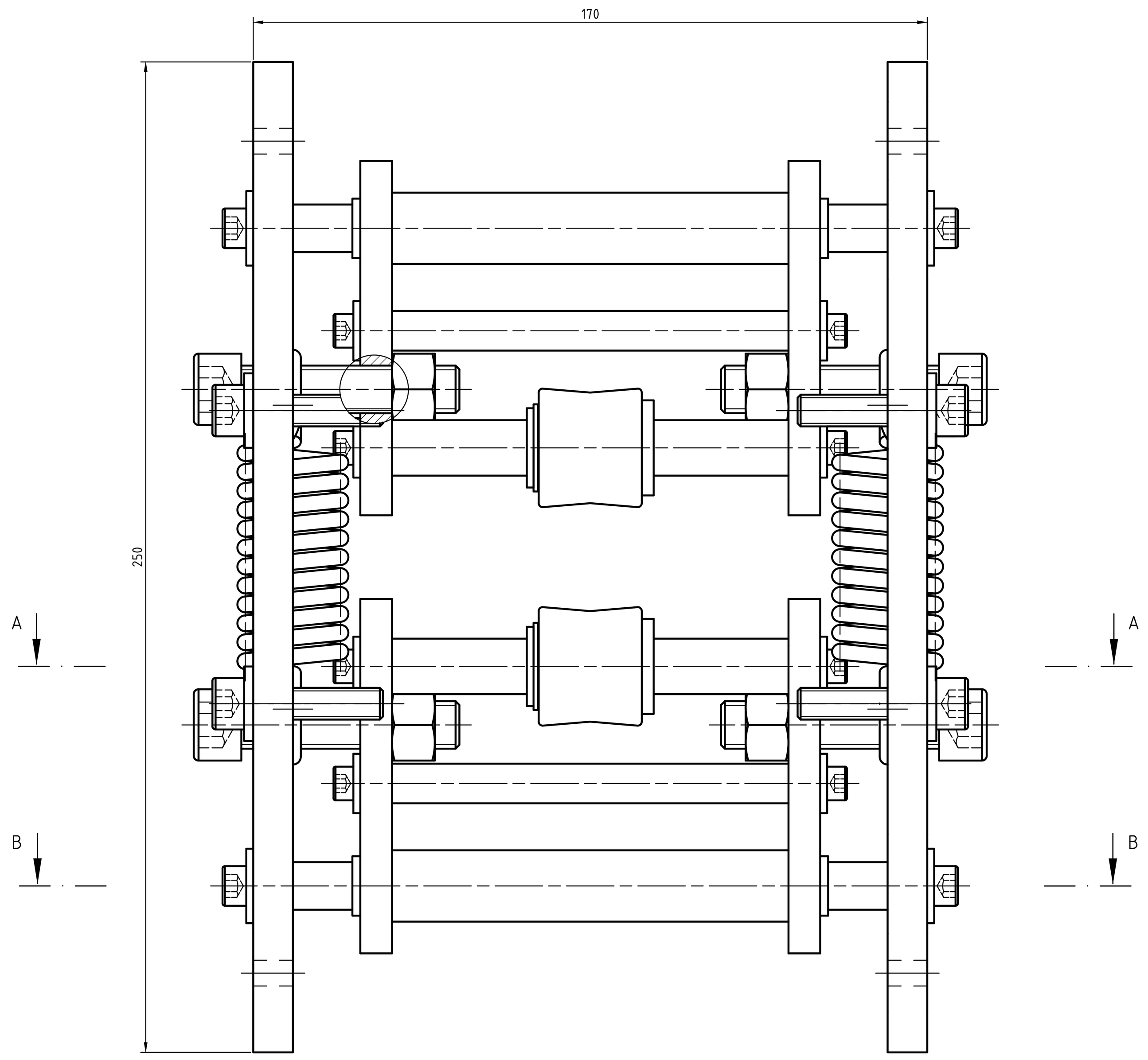
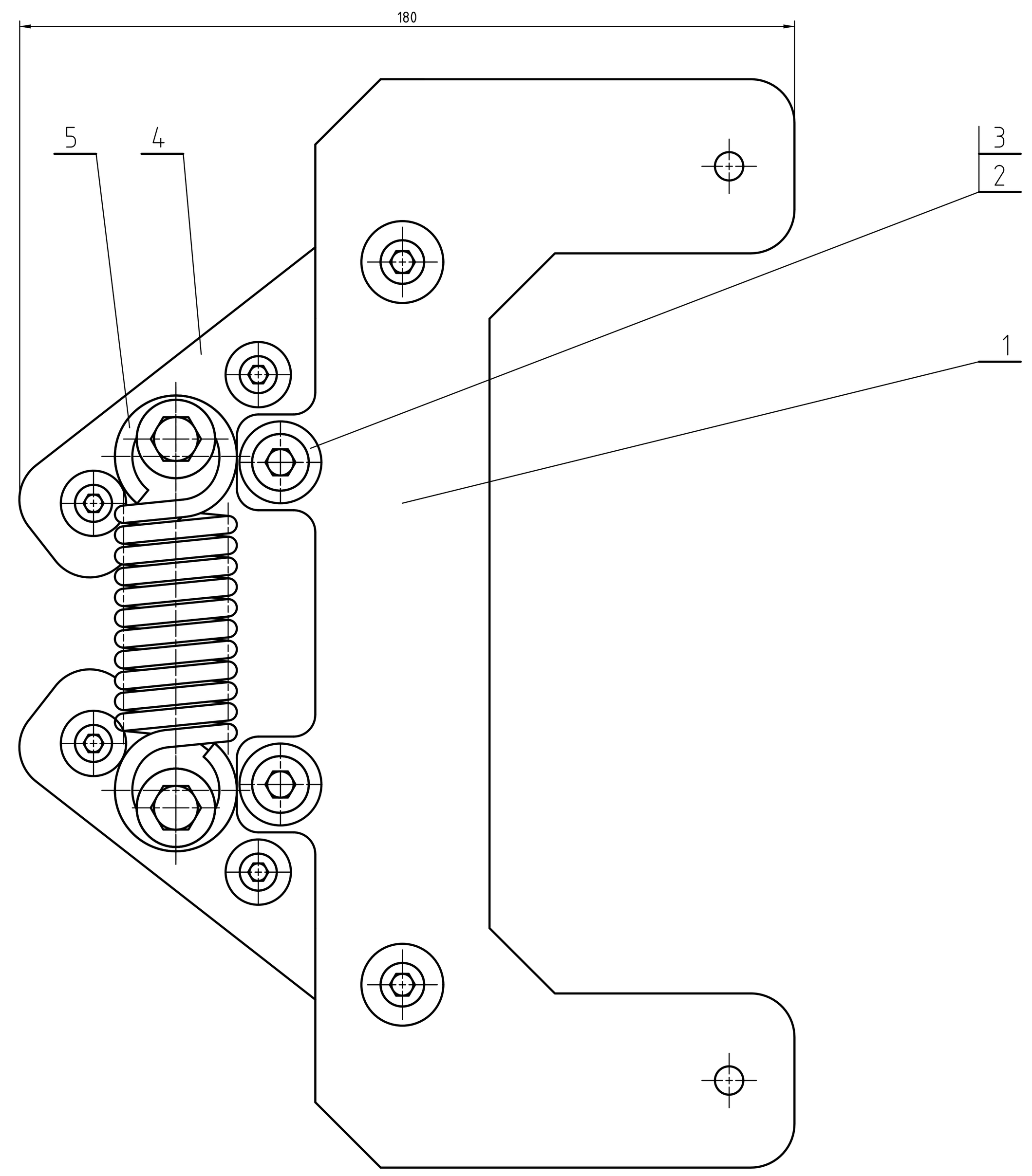
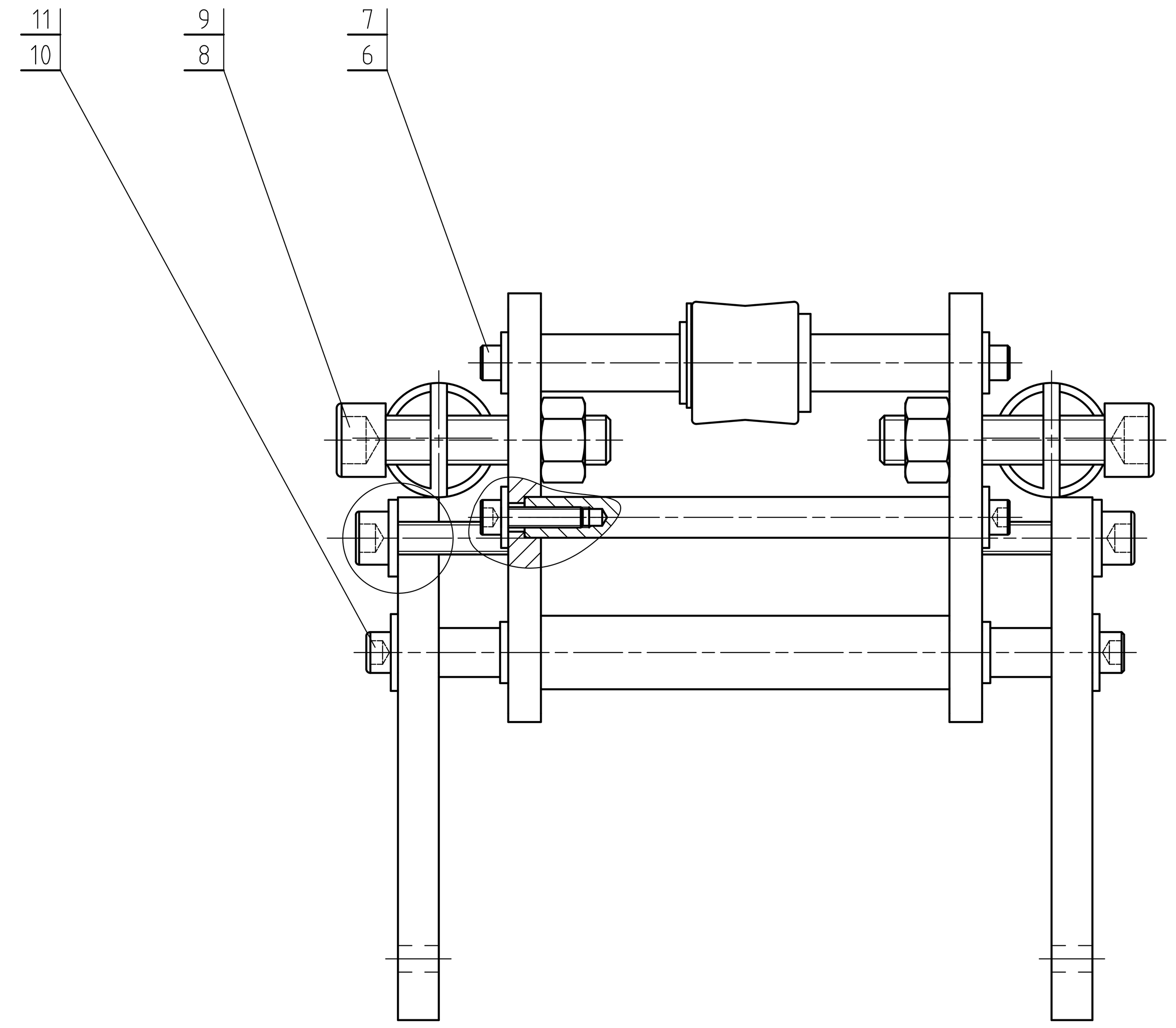
FACULTY OF MECHANICAL ENGINEERING
CLASS CC19CDT1







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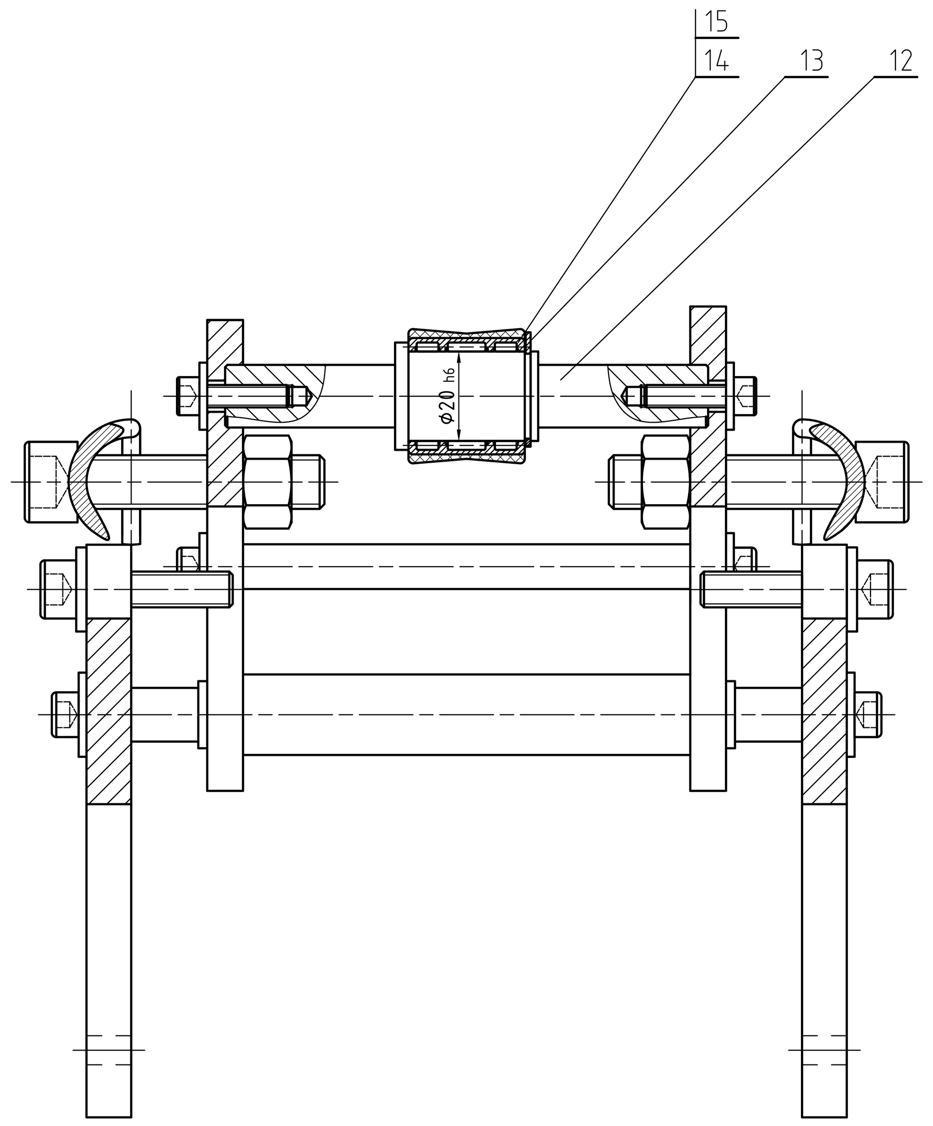

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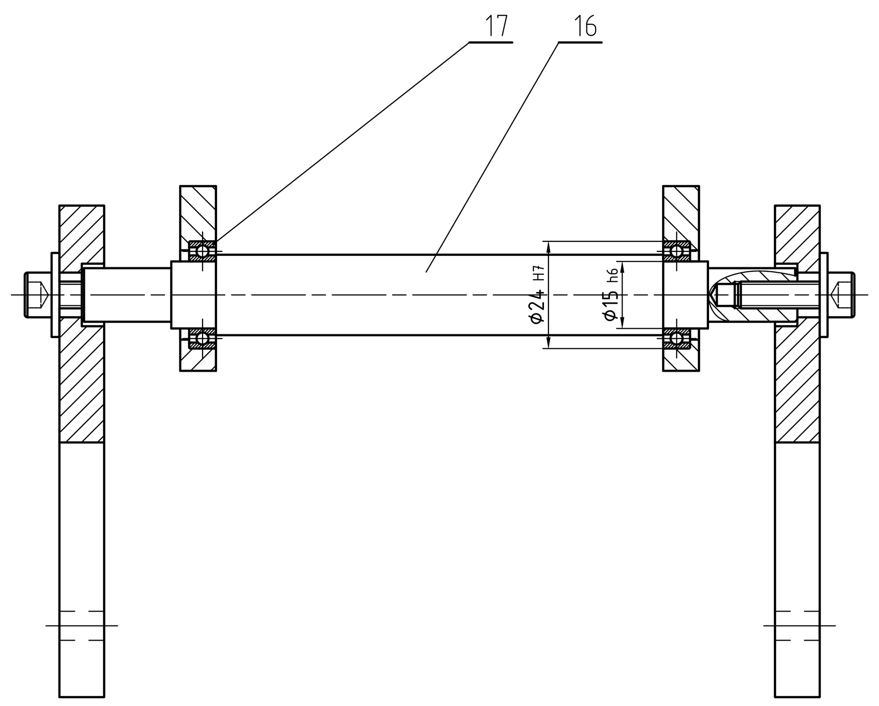

No.	Symbol	Part's Name	Qty	Material	Note
17	AFBMA 20.1 - 28 -15	Radial ball bearing	1		Mcmaster
16	ME1-004	Passive rotating shaft	2	SUS304	Workshop
15	ME1-003	Rubber rings	2	Polyurethane	Workshop
14	1553N15	One-way needle roller bearing	2		Mcmaster
13	STWS-20	Retaining ring C type	2	SUS304	Misumi
12	ME1-002	Feeder roller shaft	2	SUS304	Workshop
11	WSJ-316-M6	M6 flat washer	4	SUS316	Misumi
10	M6SH-20	M6 socket head screw	4	SUS304	Misumi
9	HNT-SUS-M12	M12 hexagon nut	4	SUS304	Misumi
8	M12SH-55	M12 socket head screw	4	SUS304	Misumi
7	WSJ-316-M5	M5 flat washer	8	SUS316	Misumi
6	M5SH-20	M5 socket head screw	8	SUS304	Misumi
5	SES42876	Tension spring	2	SUS301	Metrol
4	SES42876	Mounting plate	4	SUS304	Workshop
3	WSJ-316-M8	M8 flat washer	4	SUS316	Misumi
2	M8SH-35	M8 socket head screw	4	SUS304	Misumi
1	ME1-001	U bracket	2	SUS304	Workshop

A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE

Capstone Project

Func.	Full name	Sign	Date
Design	Huynh Ba Loc		25/12
Instruct.	Vo Tuong Quan		
Approve.	T. Viet Hong		

TAILER

C4

Quantity	Weight	Scale
		1:1

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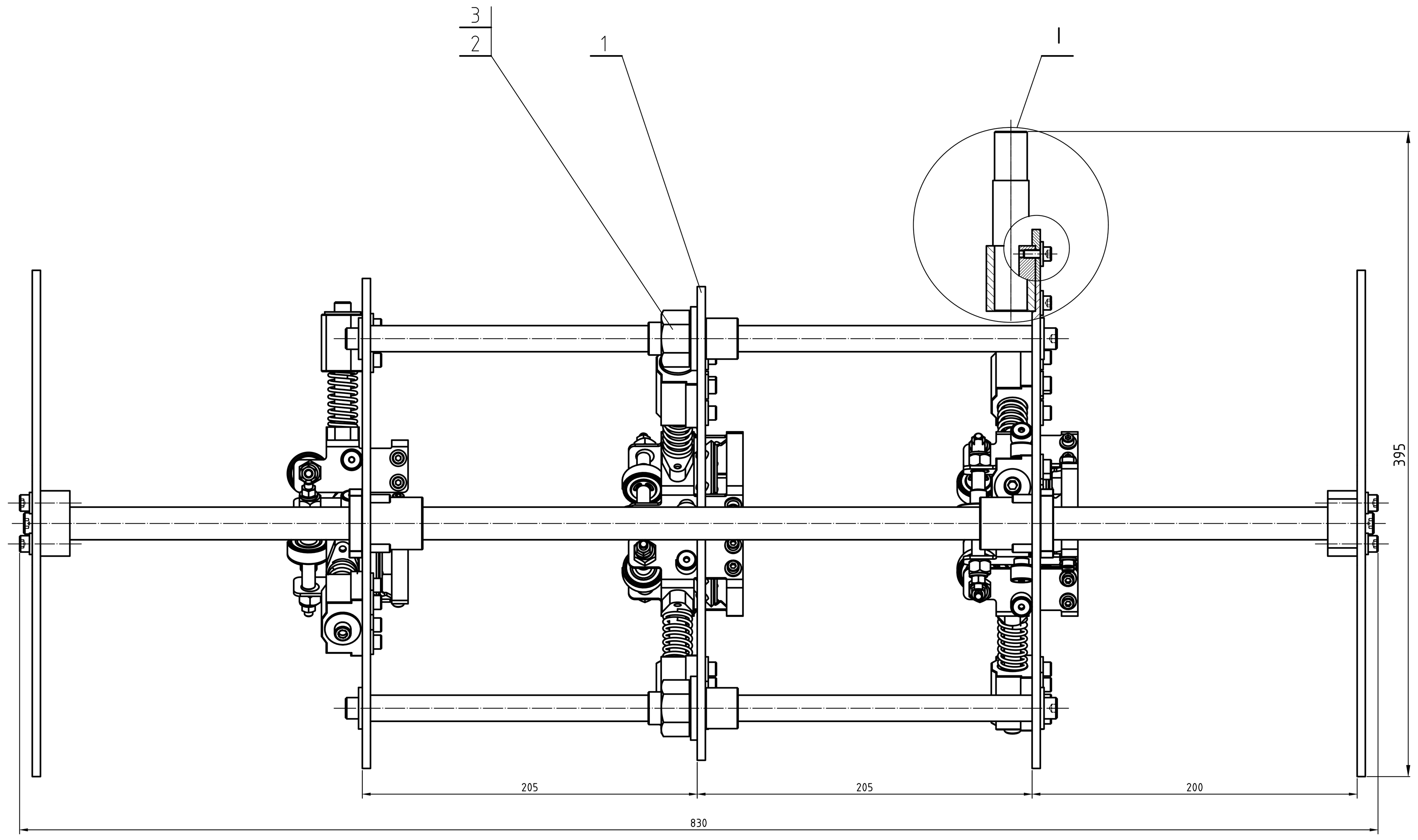
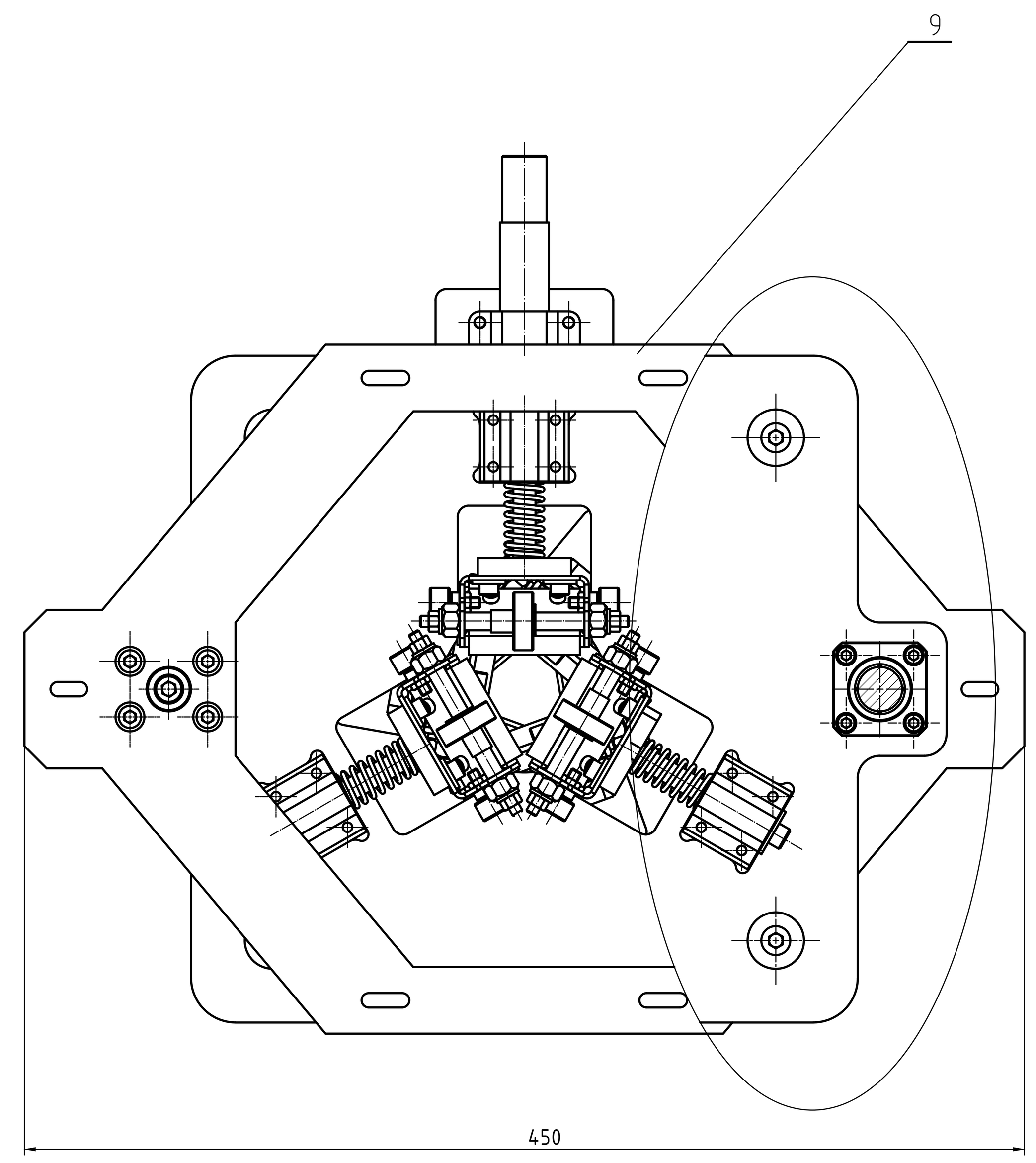
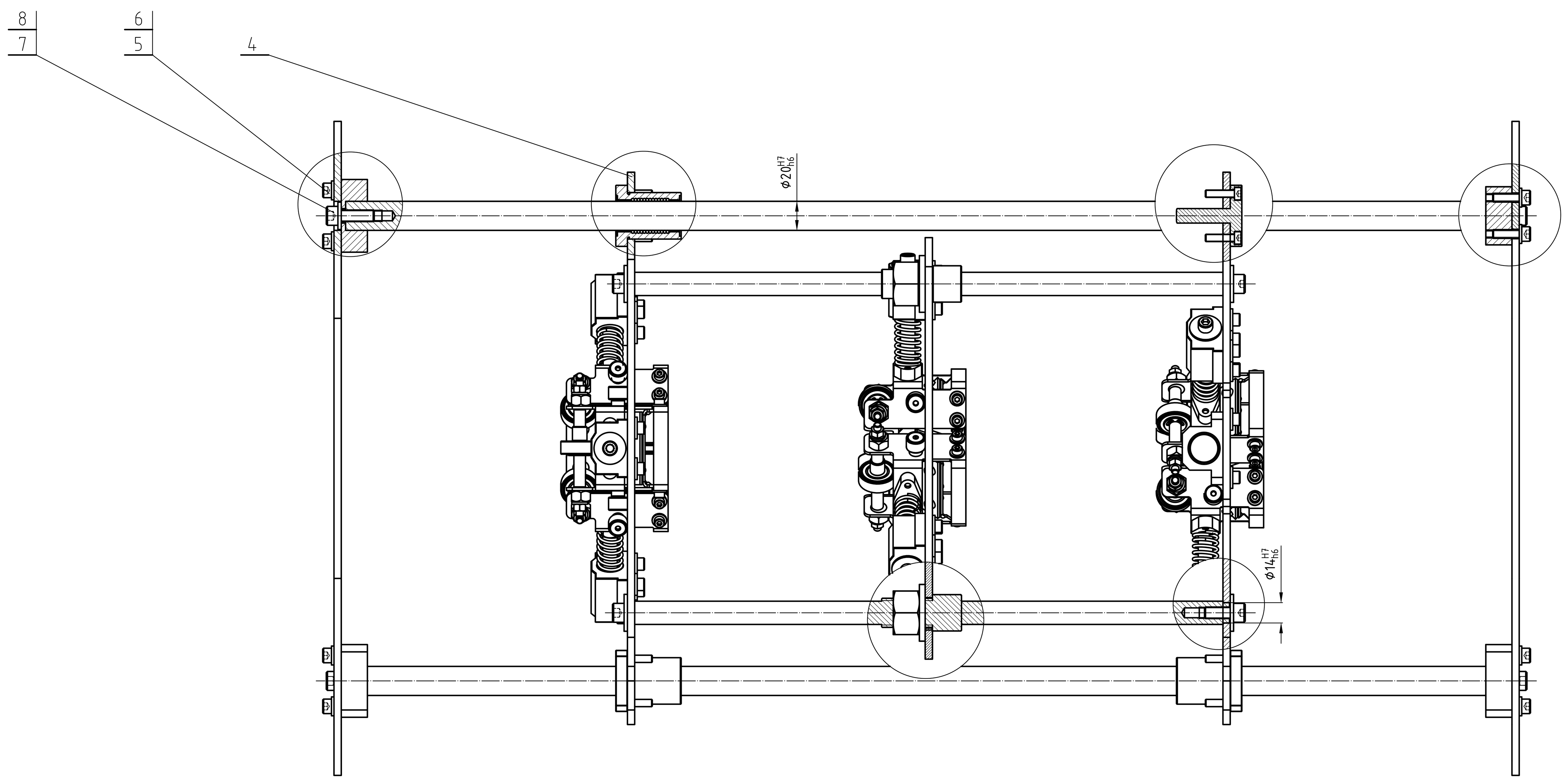
FACULTY OF MECHANICAL ENGINEERING
CLASS CC19CDT1







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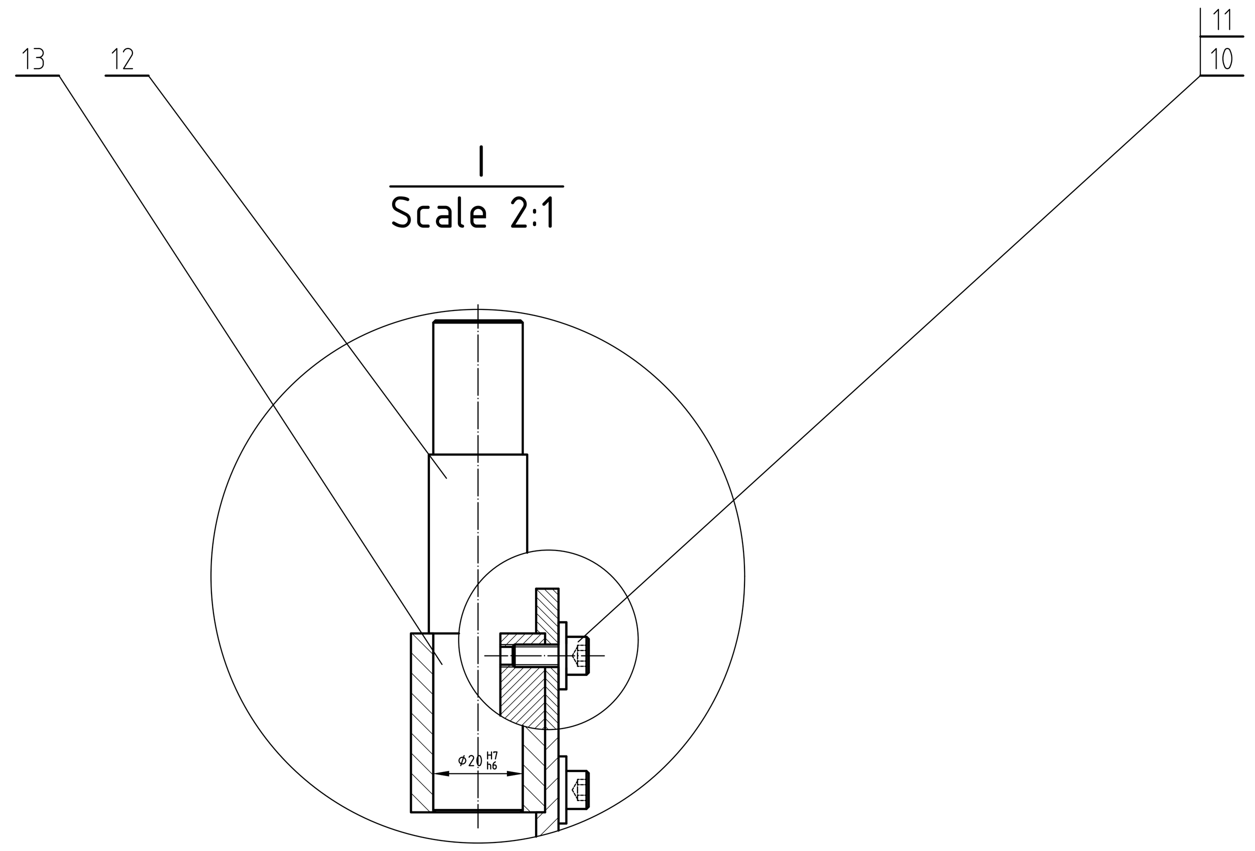

No.	Symbol	Part's Name	Qty	Material	Note
13	ME1-003	Shaft housing	1	SUS304	Workshop
12	ME1-002	Shaft connecting to rod	1	SUS304	Workshop
11	WSJ-316-M5	M5 flat washer	4	SUS316	Misumi
10	M5SH-12	M5X12 socket head screw	4	SUS304	Misumi
9	ME1-001	Mounting frame	2	SUS304	Workshop
8	WSJ-316-M8	M8 flat washer	8	SUS304	Misumi
7	M8SH-25	M8 socket head screw	8	SUS304	Misumi
6	WSJ-316-M6	M6 flat washer	16	SUS316	Misumi
5	M6SH-20	M6 socket head screw	16	SUS304	Misumi
4	6483K139	Linear ball bearing	4		Mcmaster
3	WSJ-316-M20	M20 flat washer	4	SUS304	Misumi
2	HNT-SUS-20	M20 hexagon nut	4	SUS304	Misumi
1	C2	Peeler disc	3		

A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE

Capstone Project

Func.	Full name	Sign	Date
Design	Huynh Ba Loc		25/12
Instruct.	Vo Tuong Quan		
Approve.	T. Viet Hong		

PEELER FRAME

C5

Quantity	Weight	Scale
		1:2

Sheet: 5 | Total sheet: 6

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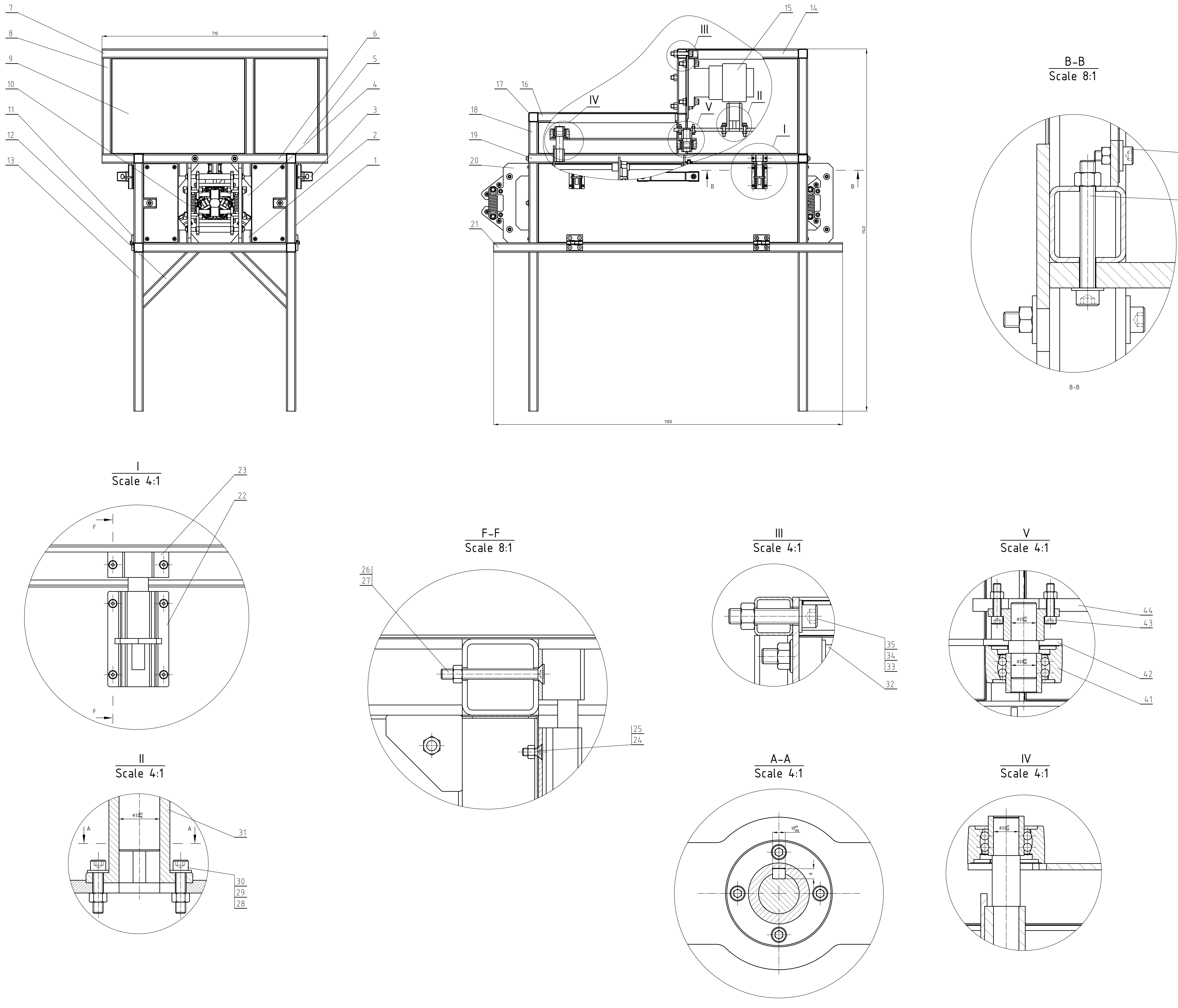
FACULTY OF MECHANICAL ENGINEERING
CLASS CC19CDT1



44	ME1-018	Crank	1	SUS304	Workshop
43	M6SH-25	M6 socket head screw	4	SUS304	Misumi
42	ME1-017	Connecting rod	1	SUS304	Workshop
41	SKF-FYTB-20-TF	Ball bearing	2		SKF
40	WSJ-316-M4	M4 flat washer	16	SUS316	Misumi
39	M4SH-10	M4 socket head screw	16	SUS304	Misumi
38	HNT-SUS-M6	M6 hexagon nut	24	SUS304	Misumi
37	WSJ-316-M6	M6 flat washer	20	SUS316	Misumi
36	M6SH-50	M6 socket head screw	20	SUS304	Misumi
35	HNT-SUS-M12	M12 hexagon nut	11	SUS304	Misumi
34	WSJ-316-M12	M12 flat washer	11	SUS316	Misumi
33	M12SH-55	M12 socket head screw	7	SUS304	Misumi
32	M12SH-50	M12 socket head screw	4	SUS304	Misumi
31	ME1-016	Shaft housing	1	SUS304	Workshop
30	HNT-SUS-M8	M8 hexagon nut	4	SUS304	Misumi
29	WSJ-316-M8	M8 flat washer	4	SUS316	Misumi
28	M8SH-30	M8 socket head screw	4	SUS304	Misumi
27	HNT-SUS-M4	M4 hexagon nut	24	SUS304	Misumi
26	93395A605	M4 countersunk hex screw	8	SUS316	Mcmaster
25	HNT-SUS-M3	M3 hexagon nut	16	SUS304	Misumi
24	93395A354	M3 countersunk hex screw	16	SUS316	Mcmaster
23	1913A35	Slide bolt latch B	4	Nylon plastic	Mcmaster
22	1913A35	Slide bolt latch A	4	Nylon plastic	Mcmaster
21	ME1-015	Square tube 30x30x1098	2	Galvanized steel	Workshop
20	C4	Tailer	1		
19	ME1-014	Square tube 30x30x878	2	Galvanized steel	Workshop
18	ME1-013	Square tube 30x30x100	6	Galvanized steel	Workshop
17	ME1-012	Square tube 30x30x510	1	Galvanized steel	Workshop
16	ME1-011	Square tube 30x30x438	2	Galvanized steel	Workshop
15	H2L32M25-WB08TAVEN	Geared 3 phase motor	1		Brother
14	ME1-010	Square tube 30x30x350	4	Galvanized steel	Workshop
13	ME1-009	Square tube 30x30x500	4	Galvanized steel	Workshop
12	ME1-008	Square tube angled 30x30x250	4	Galvanized steel	Workshop
11	3582N12	Metal hinge with holes	4	SUS303	
10	C3	Feeder	1		
9	ME1-007	Machine cover	12	SUS304	Workshop
8	ME1-006	Square tube 30x30x300	8	Galvanized steel	Workshop
7	ME1-005	Square tube 30x30x710	8	Galvanized steel	Workshop
6	ME1-004	Square tube 30x30x450	8	Galvanized steel	Workshop
5	ME1-003	Machine electric box cover	1	SUS304	Workshop
4	C5	Peeler frame	1		
3	2471N4	Pull handle	2	Nylon plastic	Mcmaster
2	ME1-002	Square tube 30x30x40	12	Galvanized steel	Workshop
1	ME1-001	Square tube 30x30x250	4	Galvanized steel	Workshop
No.	Symbol	Part's Name	Qty	Material	Note

A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE					Capstone Project		
Func.	Full name	Sign	Date	SUGARCANE COVER PEELING MACHINE	Quantity	Weight	Scale
Design	Huynh Ba Loc		25/12				1:4
Instruct.	Vo Tuong Quan				Sheet: 6	Total sheet: 6	
Approve.	T. Viet Hong			C6	HO CHI MINH CITY UNIVERSITY OF TECHNOLOGY FACULTY OF MECHANICAL ENGINEERING CLASS CC19CDT1		

A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE

Requirement of the sugarcane peeling machine:
- The productivity: 240 cane per hour
- The adaptability to cane diameter from 30 - 50 (mm)
- The length of sugarcane: 2-3 (m)
- Compliant of safe, hygienic standard for food
- Have resistance to daily cleaning chemicals

MECHANICAL SELECTIONS

Peeling principle

Wire brush peeling

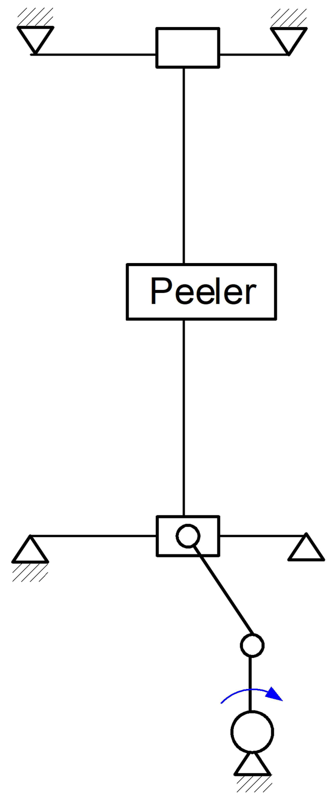
Advantages:
- Lowest cost
- Simple design

Disadvantages:
- Require constant human effort
- Inconsistent peel if lacking human attention

Reciprocate peeling

Advantages:
- Low human effort
- High productivity

Disadvantages:
- Bagasse may stuck inside
- Needs daily maintenance



Conclusion

Choose reciprocating peeling method because of higher productivity

Feeding method

Manual feeding

Advantages:
- Low cost
- Small space occupied

Disadvantages:
- Require intermittent feeding.
- Require human effort

Conveyor feeding

Advantages:
- Able to transport sugarcane in automatically

Disadvantages:
- Complex design

Conclusion

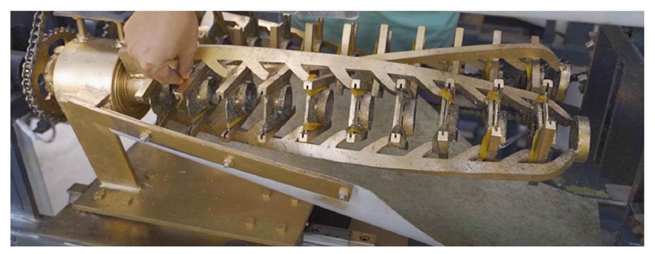
Choose manual feeding method to simplify the machine design

Peeling frame shape

Curved rotating frame

Advantages:
- Low waste material.
- Does not leave bagasse stuck.

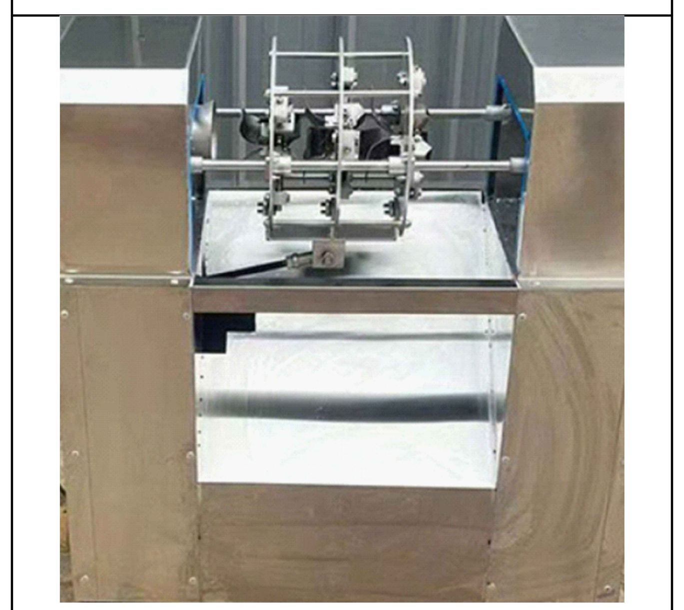
Disadvantages:
- Blade susceptible to failure fast because perpendicular peeling.
- Peel may not be clean



Fixed concentric frame

Advantages:
- Peel consistently.
- Fast peeling
- Simple design

Disadvantages:
- Bagasse stuck out
- Daily maintenance



Conclusion

Choose fixed concentric frame for simplicity and high productivity

Tightening method

Active rollers

Advantages:
- Automatically pull in the sugarcane.
- Reduce human effort.

Disadvantages:
- Need a long distance transmission system.
- The system can be harder to troubleshoot and maintain.
- Low speed
- May be pushed back when the peeling force is high

Gripping mechanism

Advantages:
- Hold the sugarcane very tight during peeling.
- Reduce human effort.

Disadvantages:
- Noisy due to surface contact.
- Have to operate the gripper opening-closing with timing of peeling process
- Complex design

One-way passive rollers

Advantages:
- Hold the sugarcane very tight during peeling.
- Prevent push back through one-way bearing

Disadvantages:
- Feeding force may be large
- Not automatic, require human effort

Conclusion

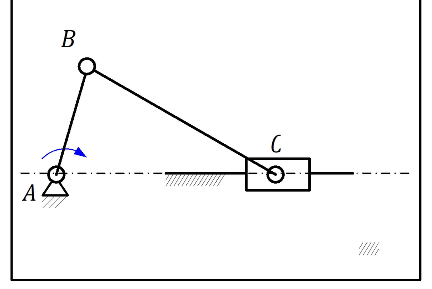
Choosing one-way rollers as a tightening option as it does not require connecting to the main peeling operation, simplify design

Actuators

Slidercrank mechanism with electric motor

Advantages:
- High force and high speed achievable.
- Clean, safe for food.
- Take up small space.

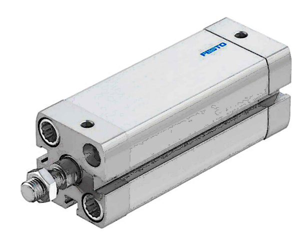
Disadvantages:
- Need to design a slidercrank mechanism to transmit motion.



Pneumatic actuator

Advantages:
- Simple design.
- High speed and force achievable.
- Clean, safe for food.

Disadvantages:
- Take up large space with pneumatic equipment.



Conclusion

Choosing electric motor with slidercrank mechanism because it saves space

Transmission system

Belt and gear

Advantages:
- Can transmit motion between distant shaft.
- Work well with high speed input.

Disadvantages:
- Short life span.
- Need a belt tensioner

Gear only

Advantages:
- Take up least space.
- High efficiency.

Disadvantages:
- Noisy operation.
- Limited distance of transmission.

Conclusion

Chose to use gear transmission only to move large load, reduce maintenance effort

ELECTRICAL METHOD

Electric motor type

DC motor

Advantages:
- Simple speed control
- Diversity of motor parameters.

Disadvantages:
- Require switching supply.
- High maintenance cost.

AC 1 phase motor

Advantages:
- Smooth, quiet operation.
- Simple installation.

Disadvantages:
- Low efficiency.
- Limited power range.

AC 3 phase motor

Advantages:
- Simple installation
- Wide power range.

Disadvantages:
- Loud noise.

Conclusion

Choosing 3 phase AC motor because it is suitable in industrial environment.

CONTROL METHOD

Control hardware

Relay - contactor

Advantages:
- Stable, simple operation.
- Suitable for industrial use.
- Low cost.

Disadvantages:
- Difficult wiring.
- Limited extensibility.

PLC

Advantages:
- Stable operation.
- Ease of maintenance.
- Flexible extensibility.

Disadvantages:
- High cost.

Microcontroller

Advantages:
- Flexible control ability.
- Ease of maintenance

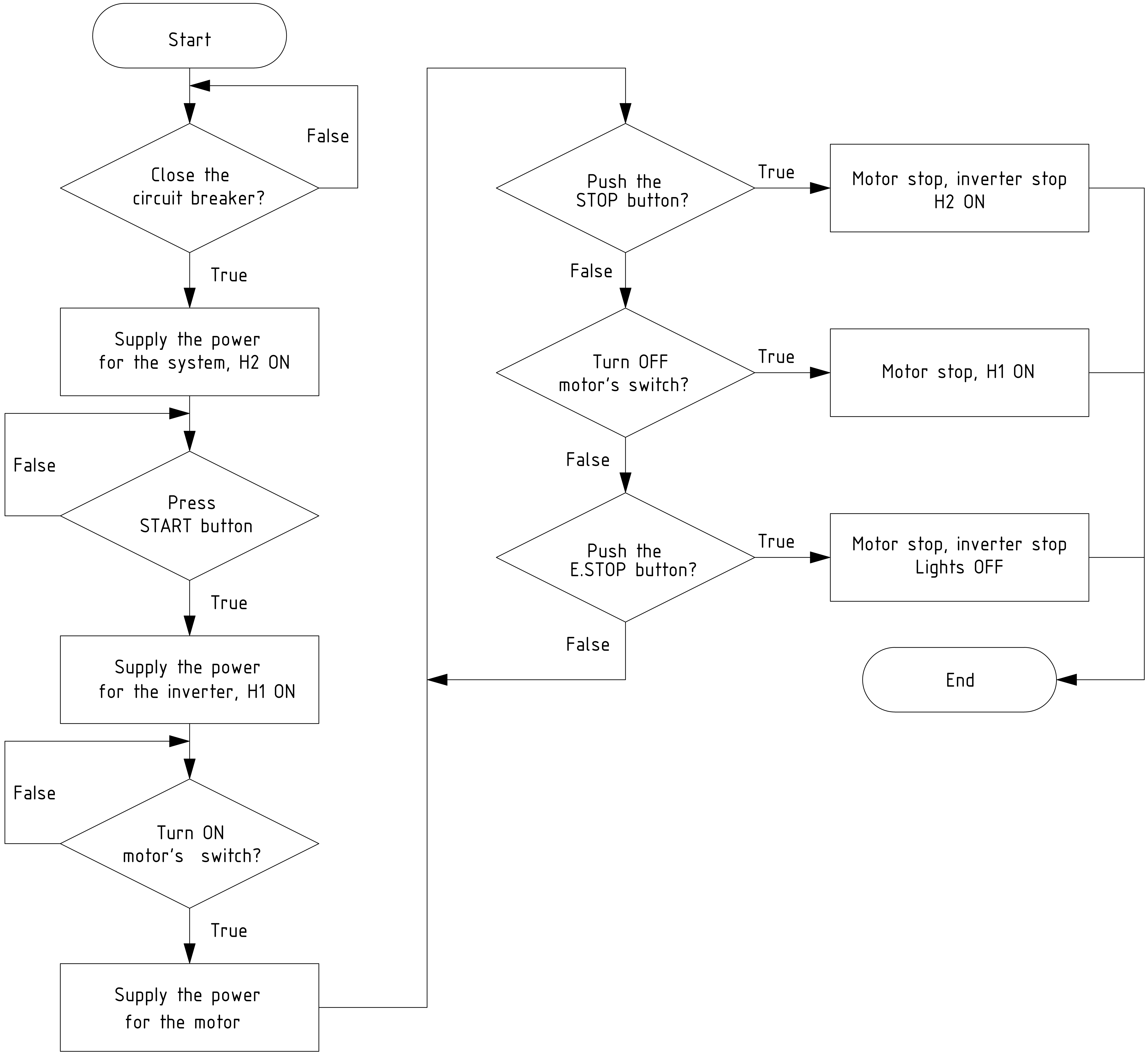
Disadvantages:
- Unstable long term operation.

Conclusion

Choose Relay-Contactor because the machine operation is simply at one specific speed

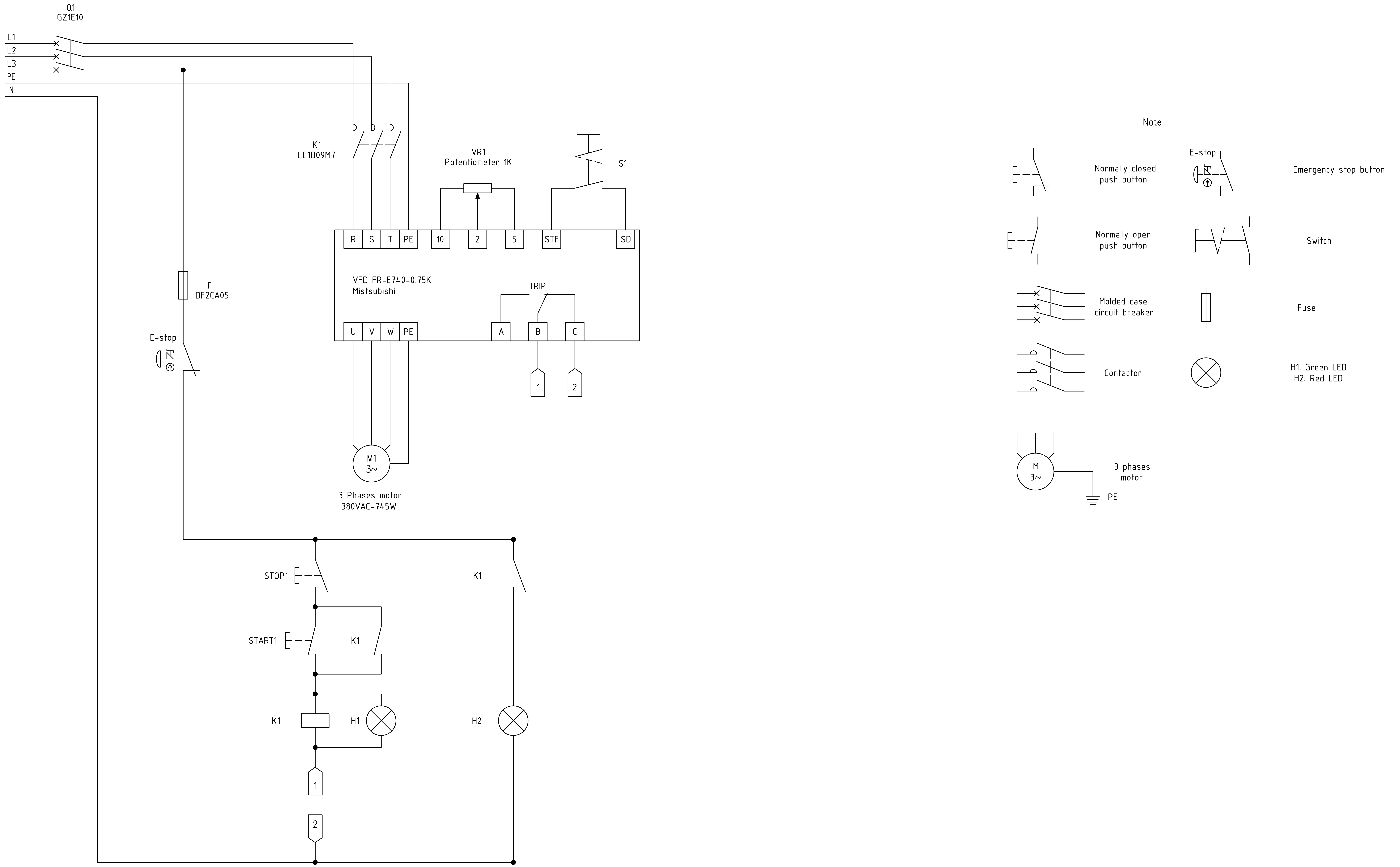
A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE					Capstone Project		
Func.	Full name	Sign	Date	METHODS SELECTION DRAWING	Quantity	Weight	Scale
Design	Huynh Ba Loc		25/12				
Instruct.	Vo Tuong Quan				Sheet: 7	Total sheet: 9	
Approve.	T. Viet Hong				HO CHI MINH CITY UNIVERSITY OF TECHNOLOGY FACULTY OF MECHANICAL ENGINEERING CLASS CC19CDT1		

MAIN ALGORITHM



A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE					Capstone Project		
Func.	Full name	Sign	Date	ALGORITHM DRAWING	Quantity	Weight	Scale
Design	Huynh Ba Loc		25/12				
Instruct.	Vo Tuong Quan				Sheet: 8	Total sheet: 9	
Approve.	T. Viet Hong				HO CHI MINH CITY UNIVERSITY OF TECHNOLOGY FACULTY OF MECHANICAL ENGINEERING CLASS CC19CDT1		

WIRING DIAGRAM



A STUDY ON DESIGN AND CONTROL A SUGARCANE COVER PEELING MACHINE					Capstone Project		
Func.	Full name	Sign	Date	ELECTRICAL DRAWING	Quantity	Weight	Scale
Design	Huynh Ba Loc		25/12				
Instruct.	Vo Tuong Quan				Sheet 9	Total sheet 9	
Approve.	T. Viet Hong				HO CHI MINH CITY UNIVERSITY OF TECHNOLOGY FACULTY OF MECHANICAL ENGINEERING CLASS CC19COT1		