UNIVERSITY OF NAIROBI

COLLEGE OF ARCHITECTURE AND ENGINEERING

SCHOOL OF ENGINEERING

DEPARTMENT OF MECHANICAL AND MANUFACTURING ENGINEERING

MECHANICAL DESIGNOF A SMALL SCALE MECHANIZED STONE CRUSHER

A final year project for the partial fulfillment for the award of bachelor's degree in Mechanical and Manufacturing Engineering of the University of Nairobi

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DECLARATION

We declare that this is our original work and has not been presented for a degree in any other university.

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ABSTRACT

Due to their simple design and easy maintainability jaw crushers are widely used as primary size reduction equipment in mechanical and mining industries. As jaw crushers break minerals & ores of high strength and the economy of many industries depends on its performance; it is essential to improve the efficiency of the present design.

The purpose of this project is to evaluate the kinematic and static force analysis of a single toggle jaw crusher that employs the simple technology of a four bar mechanism, and design a small scale mechanized jaw crusher for crushing the stones into aggregates. This mechanism will help to evaluate its effectiveness in comparison with other different types of mechanisms in use especially the double toggle and the modified single toggle mechanism. Based on this, a conclusion will be drawn based on the analysis found and recommendations given that will seek to further improve the designs of small scale stone crushing machines.

This will further evaluate the differences in having a small scale mechanized jaw crusher gap against hammer and anvil crushing mechanisms, from a point of view of economic viability. This is of great interest since construction is a key pillar towards the achievement of vision 2030.

Analysis of the design is done by modeling the machine as a four bar mechanism. Hence equations describing displacement, velocity as well as acceleration having been derived and analyzed by the previous group, the dynamic analysis is done by deriving the equations for calculating forces on each machine part and the reactions at the joints in addition to the torque equation applied at the crank, all in terms of the crushing force. Moreover, from the kinematic and dynamic analysis the maximum forces, and the mean torque applied on the crank were determined as varying from$39T_2 \leq T_3 \leq 60T_2\ i.e\ 10650\ N\ to\ 16\,380\ N$.

The kinematic analysis of single toggle jaw crusher shows that the forces on the moving jaw plate at different crank angle are different and hence power generated varies with crank angle. One way to increase the efficiency is to store the energy in a flywheel when the supply is more than the rate of consumption and to utilize the same when the supply falls down. Hence efforts are made to design a flywheel to minimize the wastage of power and to improvise the performance parameters of single toggle jaw crusher.

Jaw plate wear has considerable effect on the life of jaw Crusher which is caused by the slipping motion between the fed material and the jaws. This wear is predominantly serious in the fixed plate and hence the liners of the fixed jaw should be properly chosen. In addition to this the toggle bar which acts as a safety lever has to be precisely designed. The design aspects of flywheel, spring of tension bar and toggle bar are discussed in this paper.

The shaft speed of 384rpm, with a power requirement is 11kW at the shaft and hence it can be run by a 14kW engine. Two flywheels each 120 kg, one which acts as the sheave and the other with a design with radii of 40.98 cm, and a minimum shaft diameter of 6.5 cm with an eccentricity of 1.2 cm.

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LIST OF ABBREVIATIONS

T_2 is the torque driving the crank

T_3 = is the torque, acting about the swing jaw axis 0_3

F_2 *and* F_3 and are the forces in links 2 and 3 respectively, all assumed to be compressive

θ_3 is the angle of the swing jaw

θ_2 is the angle of the crank

L_T = length of throw

G = gape

W = width of jaw

R = Reduction ratio

V_C = Critical velocity

ρ_s = density of stones or rocks

$f(p_k)$ = packing factor

$f(\beta)$ = 1

S_c = surface characteristics of rocks

D_{P_b} = mean product size

D_{p_a} = mean feed size

w_i = work index, which is 16 for granite

Q = Crusher capacity

M = bending moment in the jaw

I_b = second moment of area of the jaw

t_b = thickness of the jaw

σ_y = yield strength of the jaw material

ΔE = maximum luctuation of energy

$$T_m\theta = \text{work done per cycle}$$

T_m = mean torque

θ = angle turned in one revolution of the crank.

ω_{mean} = mean angular velocity

ω_{max} = maximum angular velocity

ω_{min} = minimum angular velocity

$$K_e = \text{coef icient of luctuation of energy of the lywheel;}$$

E = mean kinetic energy of the lywheel;

ω_m = mean angular speed of the lywheel

I = moment of inertia of the lywheel;

m = mass of the lywheel;

k = radius of gyration of the lywheel

W_{rim} = Width of the rim

t_{rim} = thickness of the rim

$$A = \text{area of X} - \text{section of the rim} = W_{rim} * t_{rim}$$

D_f= mean diameter of the flywheel

R_f= mean radius of the flywheel

ρ= density of the flywheel

ω= angular speed of the flywheel

μ= linear velocity of the flywheel

σ_t = tensile or hoop stress

E_{rim} = energy stored in the rim of the lywheel

d_{Shaft} = diameter of the shaft

τ = maximum shear stress on the spring

K = spring factor

W_S = load on the spring

$D_1 = d_{sp} + D_{sp}$

d_{sp} = Diameter of steel wire, used for making the spring

D_{sp} = Mean diameter of spring

δ = de lection of the spring

G_m = shear modulus of elasticity of the material

n = no of active coils in the spring

$n' = n + 2$ = total no of coils in the spring

L_f = free length of the spring

P_{sp} = pitch of the spring

L_b = Rating life

C_b =Basic dynamic load rating

W_b = Equivalent dynamic load

N_d = speed in the small pulley in R. P. M

N_D = speed in the larger pulley, in R. P. M

L_{Key} = length of key

W_{Key} = width of key

τ_{Key} = shear strenth of key material

$I_g = 2^{nd}$ *moment of area of the toggle*

P_{cr} = *critical load*

CHAPTER ONE

1.0 INTRODUCTION

Rock is a natural occurring resource found in and on the earth surface. A rock is defined as an aggregate mineral petrified matter found on the earth crust. Rock is classified into various types depending on the mode of formation associated by it. These classes include sedimentary, metamorphic and igneous rocks. Rocks exist abundantly in almost all regions worldwide.

Crushing is the process of reducing the size of the lump of ore or over size rock into definite smaller sizes. The crusher crushes the feed by some moving units against a stationary unit or against another moving unit by the applied pressure, impact, and shearing or combine action on them. The strain in the feed material due to sufficiently applied pressure, impact forces, or shearing effect when exceeds the elastic limit of the feed material, the fracturing will occur on them. The crushers are very much rugged, massive and heavy in design and contact surfaces have replaceable high tensile manganese or other alloy steel sheet having either flat or corrugated surfaces. To guard against shock and over load the crushers are provided with shearing pins or nest in heavy coiled springs.

In man's quest for development and civilization, rocks have found various uses in different fields which include building and construction industry among others. Rock is used industrially in different forms, shapes and size, crushed stones being one of them. The exploitation of rock is majorly done in quarries. In Kenya, the demand for crushed stones has increased tremendously in the past few years due to the booming construction and real estate industry, the government strategy to improve infrastructure country wide and development agenda in realization of the vision 2030.

Stones required for the above purposes are of different sizes of aggregate hence the need for stone crushing. This has created a huge demand for ballast and it has become a major source of revenue for both large and small scale producers. Big stone crushing companies use highly sophisticated and expensive stone crushers. These stone crushers are centralized and located remote to the construction sites hence additional transportation expense to the place of use. A greater proportion of Kenyan citizens and the world over use crooked stone crushing methods such as the hammer and anvil, due to their lack of capital to acquire the highly mechanized, efficient and expensive stone crushers. These remote methods are labor intensive, low output, health hazardous, low quality and produce a large disparity in aggregate sizes.

1.1 STATEMENT OF THE PROBLEM

Stone crushing industry is currently on the upward trajectory in Kenya with tremendous government and private sector investment in construction, real estate industry and infrastructural developments. However, most of the people in this industry are unable to acquire machinery that can produce aggregates in large scale.

A study conducted during our visit to Kitengela in Kajiado County and Ruai in Nairobi County where people crush stones using hammer and anvil showed that a hardworking person could produce up an average of five wheelbarrows of aggregates a day which could earn him on averagely KSh. 300 only per day. This method is not very lucrative in its returns and cannot produce aggregates of desired shape and size, and also exposes the users to extreme cold, heat, rainfall, generally harsh weather conditions as well as being very prone to injuries. As opposed to the above method, mechanized crushers can produce marketable, high quality of precisely desired shapes and sizes of aggregates, with high crushing and production rate but higher production cost.

In view of this deviation between the manual hammer and anvil and large scale mechanized crushers, it was noted that there is need to develop a small scale mechanized stone crusher that could benefit the small scale stone crushing entrepreneurs and meet high market demand while at the same time speed up aggregate production of less deviation in required size and shape, with minimal cost while improving people's livelihoods.

Figure 1 1: A man manually crushing stones

1.2 OBJECTIVES

The objective of this project is to design a small scale stone crusher which is affordable, economical, and easy to use and maintain. Our aim will be to carry out:

- Kinematic analysis.
- Field workon the existing stone crushing methods; large scale and hammer and anvil.
- Stress analysis of the individual components of the machine.
- Inventor drawings of the mechanized stone crusher.

1.3 METHODOLOGY

- Literature review on the existing and previous designs.
- Carrying out field study on the labor intensive stone crushing method of hammer and anvil in places such as Kitengela in Kajiado County and Ruai in Nairobi County.
- Visiting various construction sites and firms to evaluate large scale stone crushing machines.

- Studying the laboratory stone crusher to understand its operational mechanisms and output.
- Analyze Data from the earlier single toggle kinematic and dynamic analysis.
- Undertaking a stress analysis of each machine component on the design.
- Coming up with the inventor drawings.

1.4 PROJECT JUSTIFICATION

The aim of the design and development of a mechanized small scale stone crusher is to bridge the wide gap currently existing between the large and small scale stone crushers. Large scale fabrication and finally usage of the machine will see an increase in production of high quality and regularly sized stone aggregates hence leverage in the overstretched already existing production processes.

The machine can also be used in areas where the hammer and anvil could not be used and hence increase the production capacity of the aggregates in the country. This will in turn increase the supply and lower the general cost of construction in the country, making the trade more profitable in the process as the current hammer and anvil method is very tedious, produces aggregates of lower quality and hence fetches a lower value despite its difficulty.

The usage of the machine will provide employment opportunities to citizens both locally and around the world, realizing a better return in investment in terms of time and money. Youth and women groups are poised to benefit from the production of the machine, improving the living standards of these groups. Creation of demand for the machine will invite ambitious investors to put their investments in mass industrial production of the machine providing further employment opportunities and a footstep towards the realization of Kenya becoming an industrialized nation.

1.5 FIELD STUDY

A field study on the relevance and impact of a small scale stone crusher was carried out. This was significant in understanding the challenges that the hammer and anvil stone crushers are experiencing and to find out firsthand merits that a mechanized stone crusher would mean for them and their business.

For this purpose we visited a mechanized quarry in Kitengela and an area where hammers and anvils where used in Ruai. The price of ballast per tonne in Kitengela ranged from KSh. 1250 to KSh 1400 inclusive of V.A.T and the 60 horse power crushers produced 60 tonnes per hour. The plant was highly mechanized with minimal workers and was hugely profitable.

In Ruai, a 7-tonne lorry of hardcore cost KSh. 5,000 while that of ballast was KSh.7, 500. A worker using a hammer and anvil could crush a 7-tonne lorry of stone into ballast from between one week to ten days. This means he could make between KSh. 250 to KSh. 350 per day depending on his working rate and working conditions, if he crushes the 7-tonne lorry in that time.

Therefore, in addition to the tough working conditions of the workers, the hammer and anvil method was more costly, far less productive and its output fluctuated widely depending on the weather conditions and worker outputs for each day. The hammer and anvil method is also a health hazard as the workers inhale a lot of dust working while the mechanized crusher could also crush harder rocks of better construction quality.

Figure 1 2: Site visit

CHAPTER TWO

2.0 LITERATURE REVIEW

Jaw crushers have been around for almost 175 years. All jaw crushers are distinguished by the presence of two plates, one of which is fixed and one that swings open and then closes, concurrently trapping and crushing material between the two surfaces. There are three types of jaw crushers:

- Blake Jaw Crusher
- Dodge Jaw Crusher
- Universal Jaw Crusher

These are classified according to the location of the pivot point on the swinging jaw.

a) Blake Type Jaw Crusher

In a Blake jaw crusher the swinging jaw is hinged at the top of the frame. They are used as primary crushers in the mineral industry; attains maximum amplitude at the bottom of the crushing jaws due to the position of the hinged. The size of the feed opening is referred to as the gape and the discharge end of the jaws is referred to as the set. These crushers are operated and controlled by a pitman and a toggle. The function of the toggle(s) is to move the pivoted jaw. The retrieving action of the jaw from its furthest end of travel is by springs for small crushers or by a pitman for larger crushers. Blake type jaw crusher may be divided into two types:

(i) Single toggle type

A single toggle bar is used in this type of crushers. It is comparatively lighter and cheap. It is normally preferred to crush larger material. The single toggle is taking over most new applications due to lower cost and higher capacity.

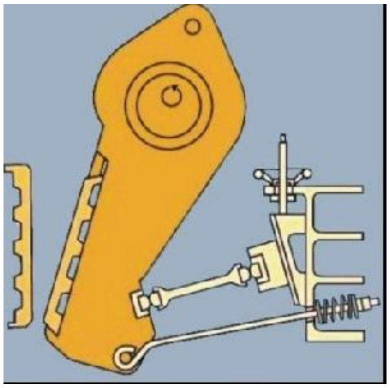


Figure 2 1: Single toggle jaw crusher

(ii) Double toggle type

One extra toggle bar is attached here. Commonly used in mines as their ability to crush materials is excellent, including tough and abrasive minerals.

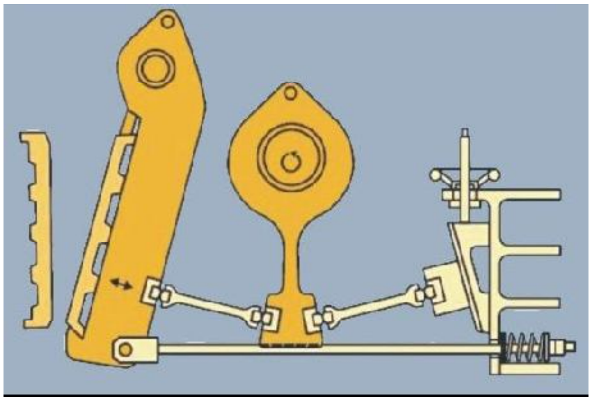


Figure 2 2: Double toggle

b) Dodge Type Jaw Crusher

The movable jaw is pivoted at the bottom and connected to an eccentric shaft. Maximum amplitude of motion is obtained at the top of the crushing plates. Dodge type crushers are not used for heavy duty operations and are commonly found in laboratories.

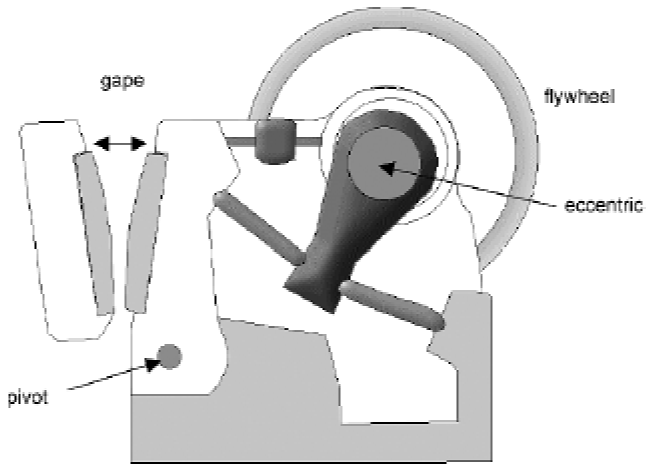


Figure 2 3: Dodge type crusher

c) The universal type jaw crusher

The crushers are pivoted in the middle so that the jaw can swing at the top and the bottom as well.

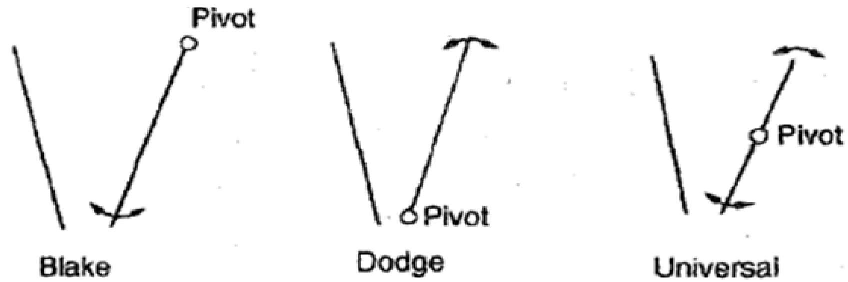


Figure 2 4: Different types of jaw crushers

Crushers are also classified according to the *stage of crushing* which they accomplish as:

- Primary
- Secondary
- Tertiary

a) Primary crusher

Receives the stones directly from a quarry after blasting and produces the first reduction size. The input of such crushers is relatively wide and the output products are coarse in size.

Example - Jaw crusher and Gyratory crusher.

b) Secondary crusher

The crushed rocks from primary crusher are sent to secondary crushers for further size reduction.

Example - Cone crusher, reduction gyratory crusher, spring rolls, disc crushers etc.

c) Tertiary/Fine crushers

Tertiary/Fine crushers have relatively small openings, and are used to crush the feed material into more uniform and finer product.

Example - Gravity stamp.

Another classification of crushers is based on the method of mechanically transmitted fracture energy to the rock.

Jaw, gyratory and roll crushers work by applying compressive force to the rock.

Single rotor and hammer mill apply high speed impact force to accomplish the fracturing.

2.1 WORKING PRINCIPLE

The mechanism of jaw crusher is to crush using impact on the upper parts of the jaw, with a little shear towards the bottom. Jawcrushers consist of two jaws. One fixed and the other reciprocating. The opening between themis largest at the top (gape) and decreases towards the bottom (set). The jaw moves on an eccentric shaftand the lower part is hinged on the toggles. The rock is thrown between two jaws and crushed bymechanical pressure.

A belt pulley; which is driven by a motor drives the eccentric shaft to rotate. This makes theattached jaw to approach and leave the other jaw repeatedly, to crush, rub and grind the feed. Hence the material moves gradually towards the bottom and finally discharges from thedischarge end. The fixed jaw mounted in a “V” alignment is the stationary breaking surface.

The swinging jaw exerts impact force on the material by forcing it against the stationaryplate. The space at the bottom of the “V” aligned jaw plates is the crusher product size gaper size of the crushed product from the jaw crusher. The rocks are crushed until they are small enough topass through the gap at the bottom of the jaws.

The ores are fed to the machine from the top where the jaws are atthe maximum distance apart. As the jaws come closer the ores are crushed into smaller sizes and slip down the cavity in the return stroke. In following cycle, further reduction of size is experienced and the ore moves down further. The process is continued till particles size is reduced to less than the bottom opening.

The toggle is used to guide the moving jaw. The retrieving motion of the jaw from its furthest end of travel is by springs for small crushers or by a pitman for larger crushers. For a smooth movement of the moving jaws, heavy flywheels are used.

2.2 CRUSHER SIZES AND POWER RATINGS

Table 2 1:Crusher sizes and power ratings

Model (Gape* Width) mm	Feed Opening (mm)	Max Feed Size (mm)	Set adjust ment range (mm)	Capacity (t/h)	Power (Kw)	Overall Dimensions (mm)	Weig ht (t)
PE150*250	150*250	125	10-40	1-3	5.5	896*745*935	1.5
PE250*400	250*400	210	20-50	5-20	15	1430*1310*1340	2.5
PE400*600	400*600	350	40-100	15-50	30	1700*1732*1650	6.8
PE500*750	500*750	425	50-100	30-85	55	2035*1921*2000	12.5
PE600*900	600*900	480	65-160	45-110	55-75	2290*2206*2370	18.5
PE750*1060	750*1060	630	80-140	105-195	90-110	2655*2302*3110	30.5
PE900*1200	900*1200	750	95-165	90-220	110-132	3800*3166*3045	52
PE150*750	150*750	120	18-48	5-16	15	1200*1500*1200	3.8
PE250*750	250*750	210	15-60	15-30	22-30	1667*1545*1020	5
PE250*1000	250*1000	210	15-60	15-50	30-37	1550*1964*1380	7
PE250*1200	250*1200	210	15-60	20-60	37-45	2192*1900*1950	9.8

2.3 JAW CRUSHER COMPONENTS

2.3.1 Crusher Frame

Crusher Frame is made of high welding. As a welding structure, it has been designed with every care so as to ensure that it is capable of resistant to bending stress even when crushing materials of extremely hard.

2.3.2 Jaw Stock

Jaw Stock is also completely welded and has renewable bushes, Particular importance has been given to jaw Stock of a design resistant to bending stresses. All jaw stocks are provided with a renewable steel Alloy or manganese steel toggle grooves.

2.3.3 Pitman

Pitman" means "connecting rod", but in a jaw crusher it doesn't connect two things. The Pitman refers to the moving jaw in a jaw crusher. It achieves the reciprocating movement through the eccentric motion of the flywheel shaft. This creates enormous force in each stroke. Pitman is fabricated from high quality steel plates and stresses are removed after welding. The Pitman is fitted with two replaceable high strength steel Alloy or manganese steel toggle bar. Grooves housings for the bearings are accurately bored and faced to gauge.

2.3.4 Manganese Liners

The jaw crusher pitman is covered on the inward facing side with dies made of manganese, an extremely hard metal. These dies often have scalloped faces. The dies are usually symmetrical top to bottom and can be flipped over that way. This is handy as most wearoccurs at the bottom (closed side) of the jaw and flipping them over provides another equalperiod of use before they must be replaced.

2.3.5 Jaw Crusher Fixed Jaw Face

The fixed jaw face is opposite the pitman face and is statically mounted. It is also covered with a manganese jaw die.Manganese liners which protect the frame from wear; these includethe main jaw plates covering the frame opposite the moving jaw, the moving jaw, and thecheek plates which line the sides of the main frame within the crushing chamber.

2.3.6 Eccentric Shaft

The pitman is put in motion by the oscillation of an eccentric lobe on a shaft that goes through the pitman's entire length. This movement might total only 1 1/2" but produces substantial force to crush material. This force is also put on the shaft itself so they are constructed with large dimensions and of hardened steel. The main shaft that rotates and has a large flywheelmounted on each end. Its eccentric shape moves the moving jaw in and out. Eccentric Shaft ismachined out of Alloy Steel Fitted with anti-friction bearings and is housed in pitman anddust proof housing.Rotational energy is fed into the jaw crusher eccentric shaft by means of a sheave pulley which usually has multiple V-belt grooves. In addition to turning the pitman eccentric shaft it usually has substantial mass to help maintain rotational inertia as the jaw crushes material.

2.3.7 Toggle Plate Protecting the Jaw Crusher

The bottom of the pitman is supported by a reflex-curved piece of metal called the toggle plate. It serves the purpose of allowing the bottom of the pitman to move up and down with the motion of the eccentric shaft as well as serve as a safety mechanism for the entire jaw. Should a piece of non-crushable material such as a steel loader tooth (sometimes called "tramp iron") enter the jaw and be larger than the closed side setting it can't be crushed nor pass through the jaw. In this case, the toggle plate will crush and prevent further damage.

2.3.8 Tension Rod Retaining Toggle Plate

Without the tension rod & spring the bottom of the pitman would just flop around as it isn't connected to the toggle plate, rather just resting against it in the toggle seat. The tension rod system tensions the pitman to the toggle plate. The toggleplate provides a safety mechanism in case material goes into the crushing chamber thatcannot be crushed. It is designed to fail before thejaw frame or shaft is damaged. The seatsare the fixed points where the toggle plate contacts the moving jaw and the main frame.

2.3.9 Jaw Crusher Eccentric Shaft Bearings

There are typically four bearings on the eccentric shaft: two on each side of the jaw frame supporting the shaft and two at each end of the pitman. These bearings are typically roller in style and usually have labyrinth seals and some are lubricated with an oil bath system. Bearings that support the main shaft are normally spherical tapered roller bearings on anoverhead eccentric jaw crusher.

Anti-Friction Bearings are heavy duty double row self-aligned roller-bearings mounted in the frame and pitman is properly protected against the ingress of dust and any foreign matter by carefully machined labyrinth seals.

2.4 MATERIAL FOR COMPONENTS OF JAW CRUSHER

Table 2 2: Material for components of jaw crusher

1. Body	Made from high quality steel plates and ribbed heavily in welded steel construction
2. Swing jaw Plate	Manganese steel
3. Fixed jaw plate	Manganese steel
4. Pitman	Crushers have a light weight pitman having White-metal lining for bearing surface
5. Toggle	Double toggles, for even the smallest size crushers give even distribution of load
6. Flywheel	high grade cast iron
7. Tension Rod	Pullback rods helps easy movement, reduces pressure on toggles and machine vibration
8. Hinge plate	Strong hinge pin made from steel are used for crushing without rubbing
9. Shaft and bearings	Massive rigid eccentric shafts made from steel along with roller bearing ensures smooth running

CHAPTER THREE

3.0 STATIC FORCE AND KINEMATIC ANALYSIS

3.1 STATIC FORCE ANALYSIS

In performing the static force analysis it shall be assumed that the masses of the links, as well as friction forces are negligible.

T_2 is the torque driving the crank

T_3 = is the torque, acting about the swing jaw axis 0_3

F_2 *and* F_3 and are the forces in links 2 and 3 respectively, all assumed to be compressive

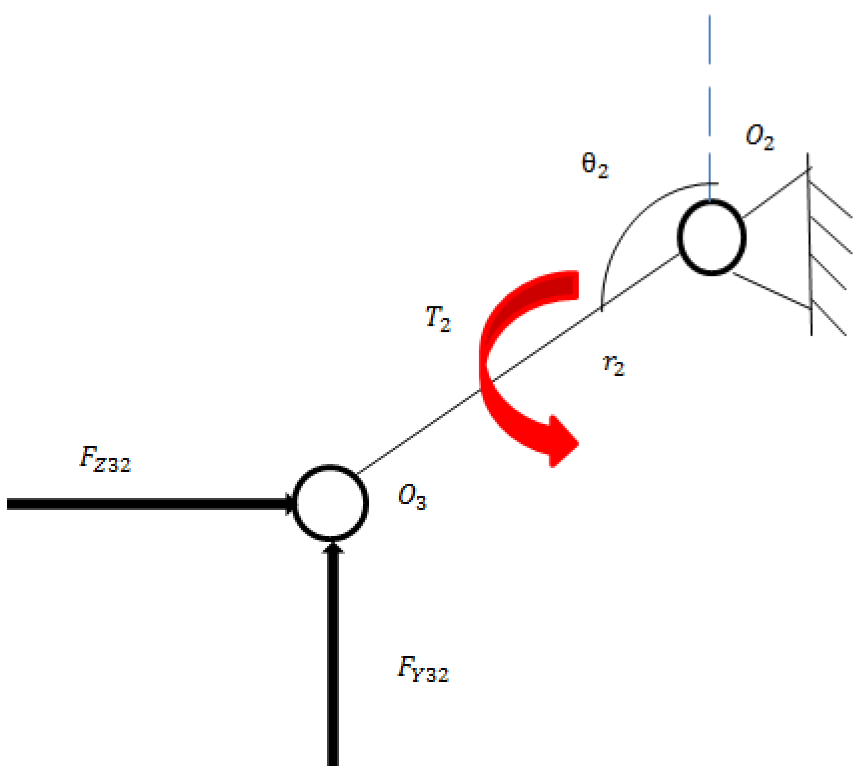


Figure 3 1: Free body diagram about joint O_2

And the angles,

θ_3 *is the angle of the swing jaw*

θ_2 *is the angle of the crank*

The equilibrium of moments on the crank, about the joint O_2, leads to the following result:

$$\left.\begin{aligned} 0 &= -F_{Y32} r_2 \sin\theta_2 + F_{Z32} r_2 \cos\theta_2 + T_2 \\ T_2 &= \left[F_{Y32} \sin\theta_2 - F_{Z32} \cos\theta_2\right] r_2 \end{aligned}\right\} \tag{3.1}$$

Now let us consider the rocker. The equilibrium of moments on the rocker, about the joint O_4, leads to the following result:

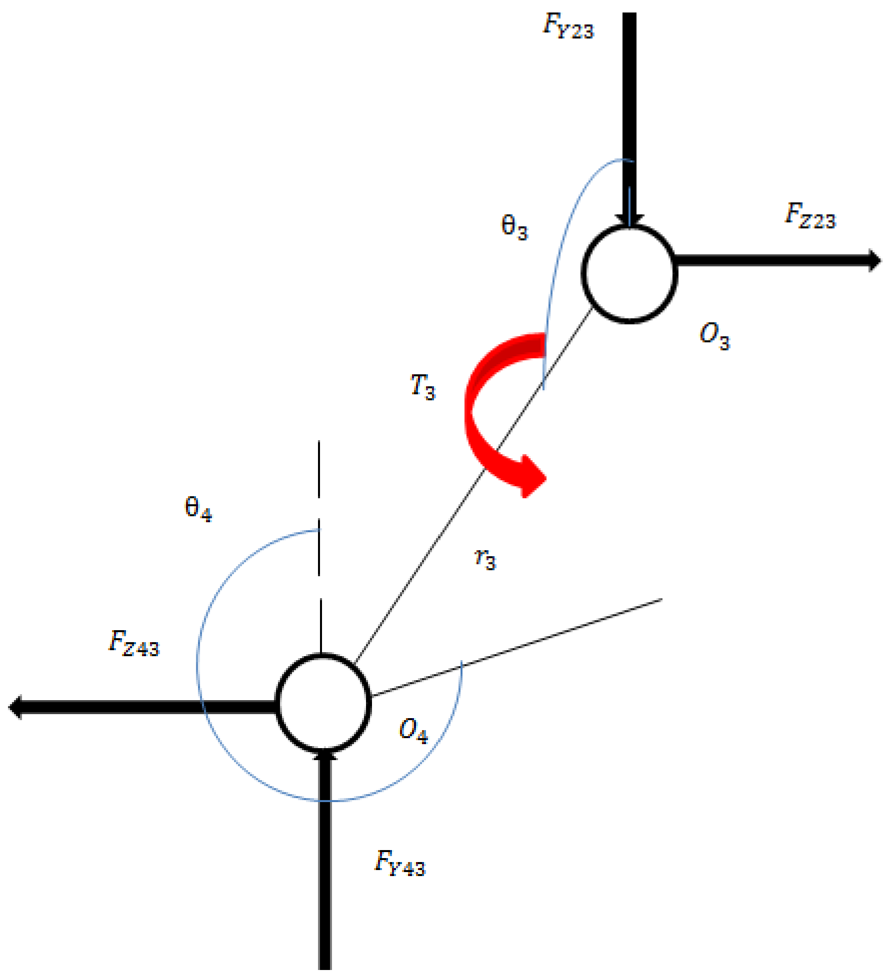


Figure 3 2: Free body diagram about O_4

$$\left.\begin{aligned} 0 &= -F_{Y43} r_3 \sin\theta_3 - F_{Z43} r_3 \cos\theta_3 + T_3 \\ T_3 &= \left[F_{Y43} r_3 \sin\theta_3 - F_{Z43} r_3 \cos\theta_3\right] r_3 \end{aligned}\right\} \tag{3.2}$$

Equilibrium of forces at joint O_4 leads to the following:

$$\left.\begin{aligned} F_{Z43} + F_3 \cos\left(\theta_3 + 90^0\right) &= 0 \\ F_{Z43} + F_3 \sin\theta_3 &= 0 \\ F_{Z43} &= -F_3 \sin\theta_3 = F_{Z32} \end{aligned}\right\} \tag{3.3}$$

$$F_{Z23} = F_{Z34} = F_3 \sin\theta_3$$

Similarly:

$$\left.\begin{aligned} F_{Y43} + F_3 \sin\left(\theta_3 - 90^0\right) &= 0 \\ F_{Y43} - F_3 \cos\theta_3 &= 0 \\ F_{Y43} &= F_3 \cos\theta_3 = F_{Y32} \end{aligned}\right\} \tag{3.4}$$

Substituting the values of forces in the equations of torque derived leads to the following results:

Equation (1) becomes:

$$\begin{aligned} T_2 &= \left[F_3 \cos\theta_3 \sin\theta_2 + F_3 \sin\theta_3 \cos\theta_2\right] r_2 \\ T_2 &= r_2 F_3 \left[\cos\theta_3 \sin\theta_2 + \sin\theta_3 \cos\theta_2\right] \\ T_2 &= r_2 F_3 \sin\left[\theta_2 + \theta_3\right] \end{aligned} \tag{3.5}$$

Equation (2) becomes:

$$\begin{aligned} T_3 &= \left[F_3 \cos\theta_3 \sin\theta_3 + F_3 \sin\theta_3 \cos\theta_3\right] r_3 \\ T_3 &= r_3 F_3 \left[\cos\theta_3 \sin\theta_3 + F_3 \sin\theta_3 \cos\theta_3\right] \\ T_3 &= r_3 F_3 \left[\sin 2\theta_3\right] \end{aligned} \tag{3.6}$$

A relationship between T_2 and T_3 can be obtained from equations (3.5) and (3.6), as follows:

$$\frac{T_3 r_2}{T_2 r_3} = \frac{\sin\left(2\theta_3\right)}{\sin\left(\theta_2 + \theta_3\right)} \tag{3.7}$$

Equation (3.7) is in dimensionless form. For a given crusher mechanism, values of θ_2 and θ_3 can be determined from purely kinematical considerations and then the value of the right-hand side of equation (3.7) can be determined.

3.2 KINEMATIC ANALYSIS

The mechanism that was proposed consists of a four bar eccentric shaft and rocker mechanism with the rocker being the swing jaw. A simple line diagram of this mechanism is shown below.

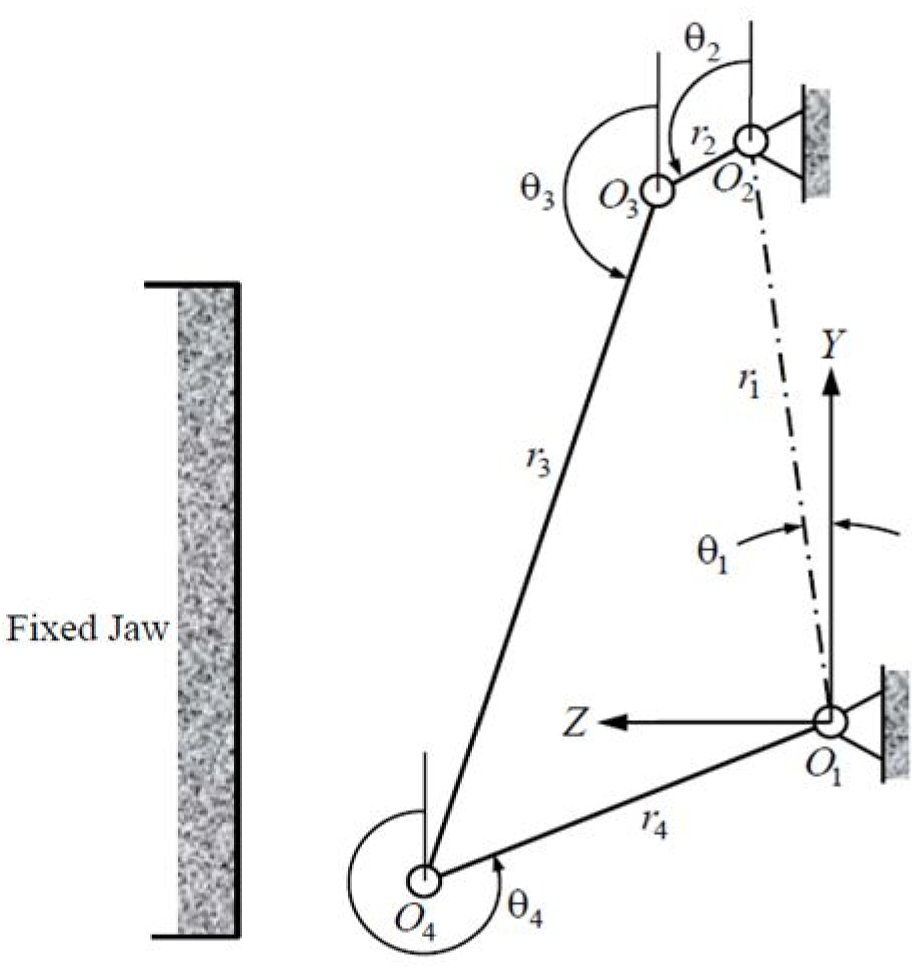


Figure 3 3Kinematic Model of a Single Toggle Jaw Crusher

In analysis of the kinematics of the above crusher, an understanding of the motion of the rocker, relative to the fixed jaw as the crank rotates through a complete cycle is mandatory. All angular displacements are taken counter clockwise, relative to the Y direction.

3.2.1 Position and Displacement analysis

The analysis of the position and displacement can be accomplished through use of the well-known vector loop closure method, which is illustrated in figure 3.4 below.

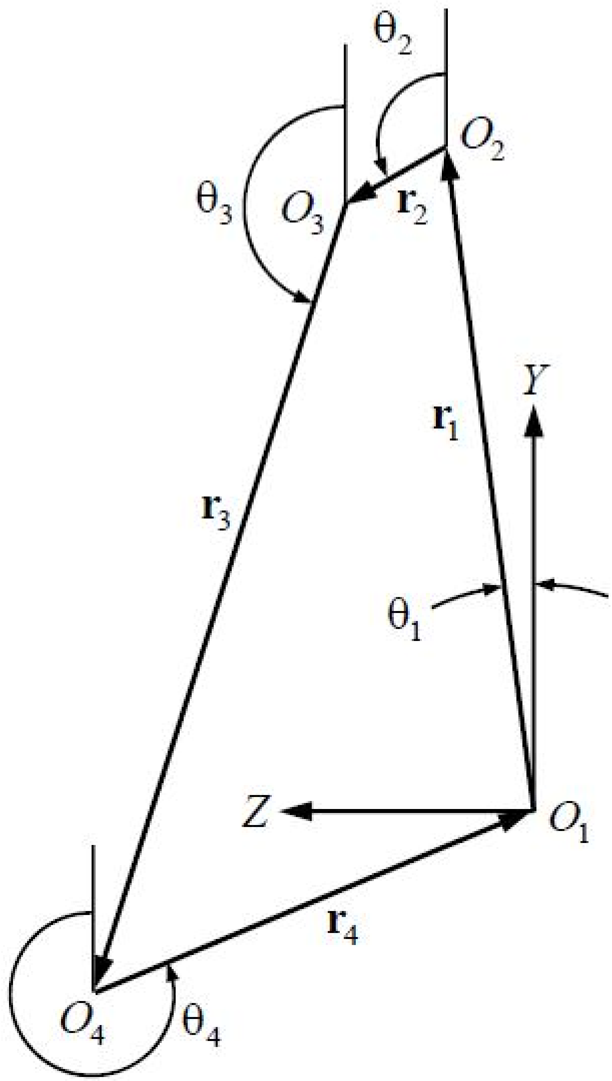


Figure 3 4Vector loop closure method

In figure 3.4, the vector loop equation can be written as follows:

$$\mathbf{r}_1 + \mathbf{r}_2 + \mathbf{r}_3 + \mathbf{r}_4 = 0 \tag{3.8}$$

Equation (3.8) above can be re-written in complex notation as follows:

$$r_1 e^{j\theta_1} + r_2 e^{j\theta_2} + r_3 e^{j\theta_3} + r_4 e^{j\theta_4} \tag{3.9}$$

Noting the Euler Identities:

$$\left.\begin{aligned} e^{j\theta} &= \cos\theta + j\sin\theta \\ e^{-j\theta} &= \cos\theta - j\sin\theta \end{aligned}\right\} \tag{3.10}$$

For conciseness the following notation can also be introduced:

$$\left.\begin{aligned} \cos\theta_i &= c_i \\ \sin\theta_i &= s_i \end{aligned}\right\} \tag{3.11}$$

Using equations (3.10) and (3.11), equation (3.9) can be re-written as:

$$r_1(c_1 + js_1) + r_2(c_2 + js_2) + r_3(c_3 + js_3) + r_4(c_4 + js_4) = 0 \tag{3.12}$$

The real and imaginary parts of the equation (3.12) could be separated to obtain:

$$\left.\begin{aligned} r_1c_1 + r_2c_2 &= -(r_3c_3 + r_4c_4) \\ r_1s_1 + r_2s_2 &= -(r_3s_3 + r_4s_4) \end{aligned}\right\} \tag{3.13}$$

Furthermore, both equations above could be squared to yield the following equations:

$$\left.\begin{aligned} r_1^2c_1^2 + 2r_1r_2c_1c_2 + r_2^2c_2^2 &= r_3^2c_3^2 + 2r_3r_4c_3c_4 + r_4^2c_4^2 \\ r_1^2s_1^2 + 2r_1r_2s_1s_2 + r_2^2s_2^2 &= r_3^2s_3^2 + 2r_3r_4s_3s_4 + r_4^2s_4^2 \end{aligned}\right\} \tag{3.14}$$

By adding the corresponding terms in equation (3.14) above and noting that $c_i^2 + s_i^2 = 1$, we obtain the following:

$$r_1^2 + 2r_1r_2(c_1c_2 + s_1s_2) + r_2^2 = r_3^2 + 2r_4r_3(c_4c_3 + s_4s_3) + r_4^2 \tag{3.15}$$

Rearranging equation (3.13) we have:

$$\left.\begin{aligned} r_3c_3 &= -(r_1c_1 + r_2c_2 + r_4c_4) \\ r_3s_3 &= -(r_1s_1 + r_2s_2 + r_4s_4) \end{aligned}\right\} \tag{3.16}$$

From trigonometry:

$$\cos\theta_i \cos\theta_k + \sin\theta_i \sin\theta_k = \cos(\theta_i - \theta_k) \tag{3.17}$$

By substituting equation (3.16) into (3.15) and using the identity in equation (3.17), we obtain:

$$\left.\begin{aligned}2r_1r_2\cos(\theta_2-\theta_1)+2r_1r_4\cos(\theta_4-\theta_1)+r_1^{\,2}+r_2^{\,2}-r_3^{\,2}+r_4^{\,2}\\=-2r_2r_4\cos(\theta_4-\theta_2)\end{aligned}\right\} \tag{3.18}$$

From figure (3.4) above, θ_1 is a fixed quantity and for given values of r_1, r_2, r_3 and r_4 the value of θ_1 will be known.

Also, motion of the crank O_2O_3 is the input motion. It may be considered to be a rotation at uniform angular velocity ω_2. At an instant time t, after commencement of motion, the value θ_2 in radians will be determined as follows:

$$\theta_2(t) = \omega_2 t \tag{3.19}$$

For given lengths of the four links in the mechanism, equation (3.18) can be used to determine the values of θ_4 that correspond to any given value of θ_2.

Equation (3.18) describes all possible spatial configurations of the mechanism, for given lengths of the four links. For the case where $\theta_1 = 0$, equation (3.18) becomes:

$$2r_1r_2\cos\theta_2 + 2r_1r_4\cos\theta_4 + r_1^{\,2} + r_2^{\,2} - r_3^{\,2} + r_4^{\,2} = -2r_2r_4\cos(\theta_4-\theta_2) \tag{3.20}$$

Dividing each term by $2r_2r_4$, we obtain the following equation:

$$\frac{r_1}{r_4}\cos\theta_2 + \frac{r_1}{r_2}\cos\theta_4 + \frac{r_1^{\,2}+r_2^{\,2}-r_3^{\,2}+r_4^{\,2}}{2r_2r_4} = \cos(\theta_4-\theta_2) \tag{3.21}$$

The above equation can also be written as follows:

$$\left.\begin{aligned}K_1\cos\theta_2 + K_2\cos\theta_4 + K_3 &= \cos(\theta_4-\theta_2)\\ K_1 &= \frac{r_1}{r_4}\\ K_2 &= \frac{r_1}{r_2}\\ K_3 &= \frac{r_1^{\,2}+r_2^{\,2}-r_3^{\,2}+r_4^{\,2}}{2r_2r_4}\end{aligned}\right\} \tag{3.22}$$

Equation (3.22) is called the Freudenstein equation. For given values of the lengths of the four links, the equation can be used to determine values of θ_4 that correspond to any given values of θ_2.

The data in Table 3.1 shall be used to demonstrate how the kinematic equations are applied.

Table 3 1: Dimensions of a Single Toggle Jaw Crusher

r_1 (mm)	r_2 (mm)	r_3 (mm)	r_4 (mm)
600	12	700	250

3.2.2 Angular Displacement of the Swing Jaw

Equation (3.13) can be re-arranged to obtain the following equation:

$$\left.\begin{aligned} r_1c_1 + r_2c_2 + r_3c_3 &= -r_4c_4 \\ r_1s_1 + r_2s_2 + r_3s_3 &= -r_4s_4 \end{aligned}\right\} \quad (3.23)$$

Equation (3.23) can be substituted into equation (3.15) and using equation (3.17), the following is obtained:

$$\left.\begin{aligned} 2r_1r_2\cos(\theta_2-\theta_1)+2r_2r_3\cos(\theta_3-\theta_2)+r_1^{\,2}+r_2^{\,2}+r_3^{\,2}-r_4^{\,2} \\ = -2r_3r_1\cos(\theta_3-\theta_1) \end{aligned}\right\} \quad (3.24)$$

For given lengths of the four links in the mechanism, along with the value of θ_1, equation (3.24) can be used to determine corresponding values of θ_3 for the given values of θ_2. When compared to equation (3.18), equation (3.24) is of greater utility in describing the motion of the swing jaw, relative to that of the crank.

For our case $\theta_1 = 0$ equation (3.24) reduces to the following:

$$\left. 2r_1r_2\cos\theta_2 + 2r_3r_1\cos\theta_3 + r_4^{\,2} - r_3^{\,2} - r_2^{\,2} - r_1^{\,2} = -2r_2r_3\cos(\theta_3-\theta_2) \right\} \quad (3.25)$$

Substituting the values of $\left. r_1 = 600mm, r_2 = 12mm, r_3 = 700mm, r_4 = 250mm \right\}$ into equation (3.25) above and for various values of θ_2, we obtain the following table:

Table 3 2: Crank angle, coupler angle and normalized torque relationships

θ_2	θ_3	T_3 / T_2
0	159.405	-109.204
15	159.704	-411.192
18	159.763	-970.001
21	159.821	2636.133
24	159.88	556.854
27	159.938	311.191
30	159.995	216.07
45	160.265	86.875
60	160.493	56.547
75	160.669	44.128
90	160.779	38.406
105	160.817	36.301
120	160.824	36.849
135	160.777	40.276
150	160.475	48.309
165	160.228	65.117
180	159.937	109.581

θ_2	θ_3	T_3 / T_2
195	159.622	406.292
198	159.559	896.376
201	159.496	-4421.31
204	159.432	-640.993
207	159.37	-346.713
210	159.308	-238.418
225	159.018	-95.825
240	158.772	-62.876
255	158.592	-49.258
270	158.488	-42.779
285	158.466	-40.092
300	158.528	-40.184
315	158.666	-43.163
330	158.86	-50.384
345	159.121	-66.283
360	159.405	-109.204

With the data given in Table 1, given the values of θ_2 , the corresponding values of θ_3 and θ_4 were computed and then used in equation (3.18) to determine the corresponding torque ratios. The results are plotted in Fig. below:

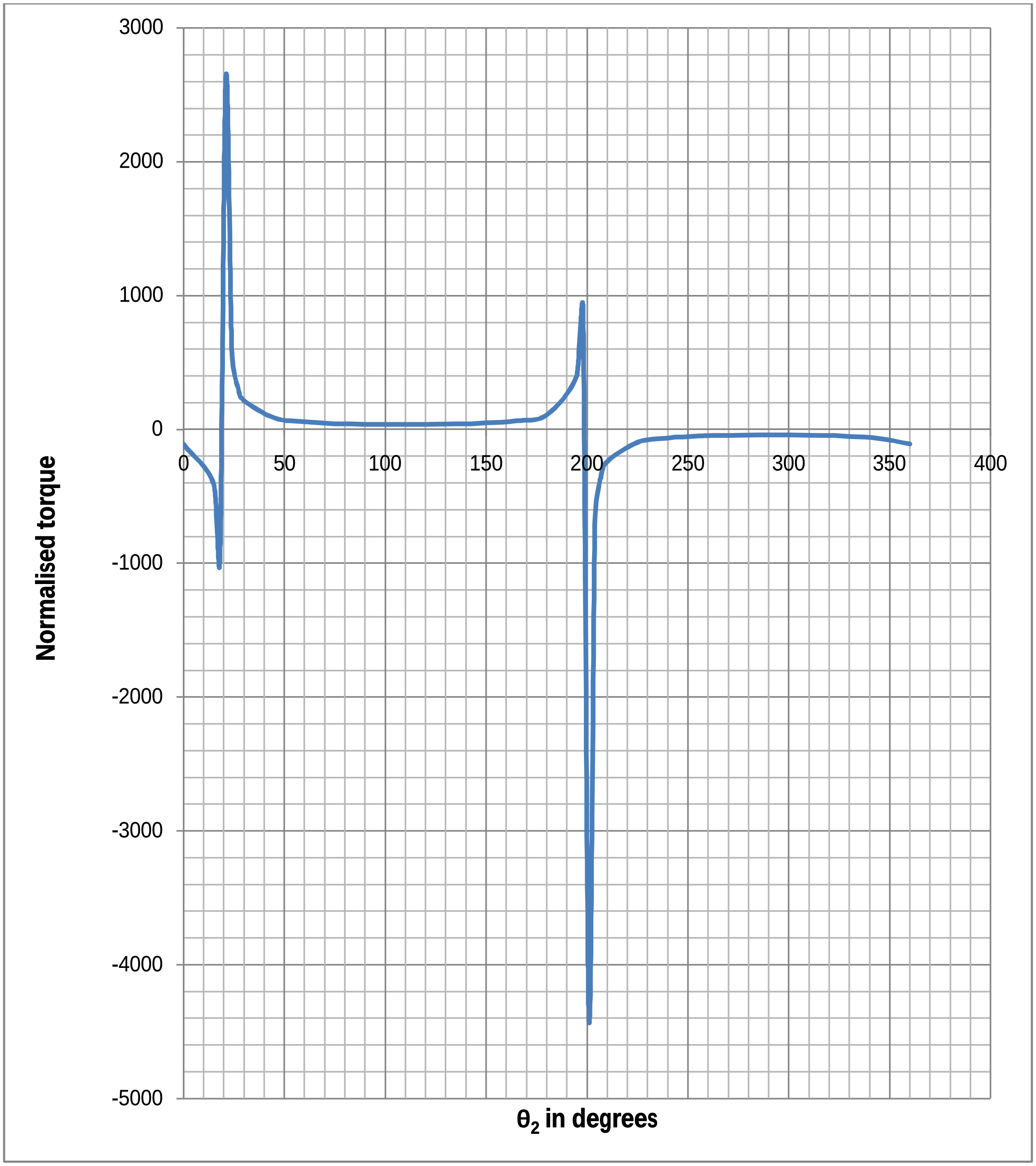


Figure 3 5: Graph for normalized torque versus crank angle θ_2

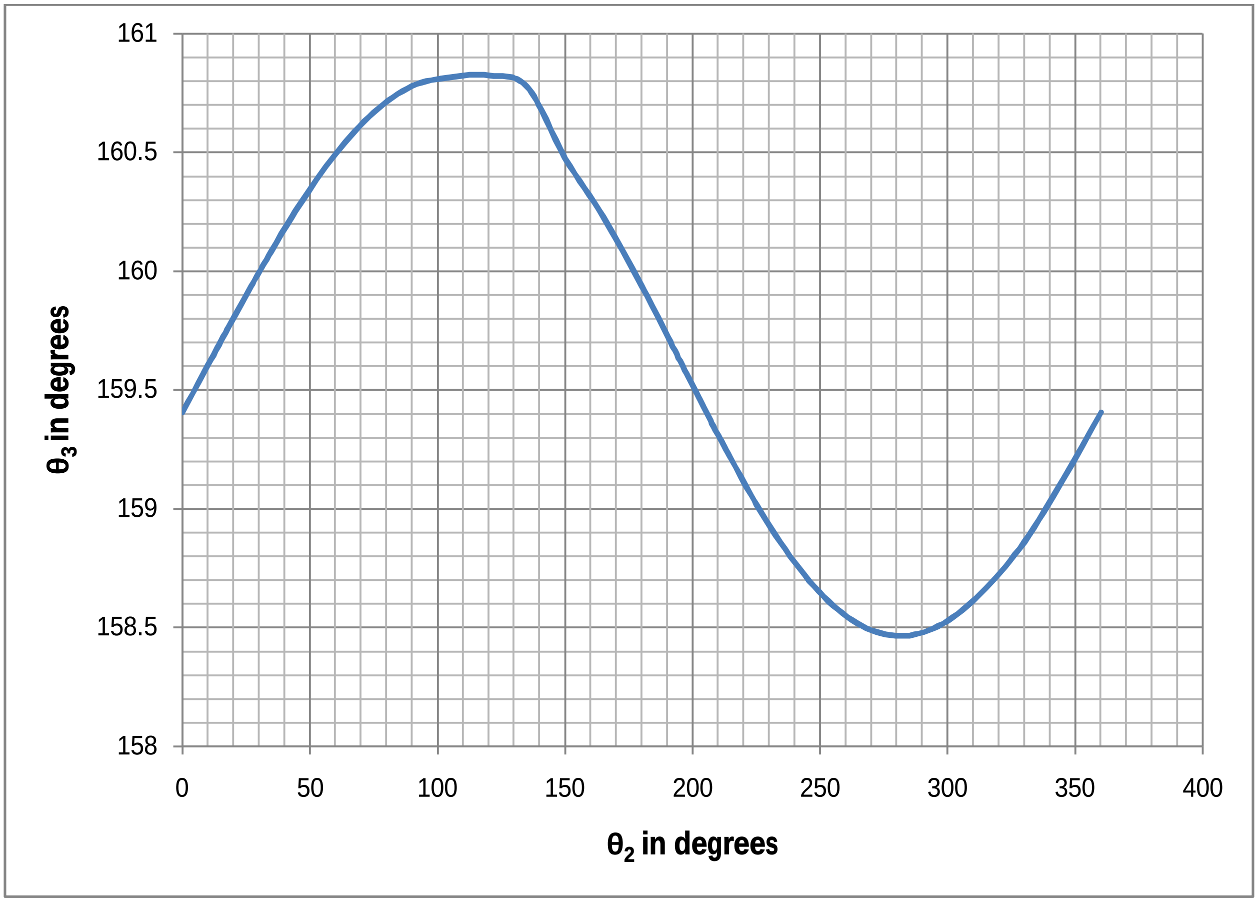


Figure 3 6: Graph for variation in coupler angle θ_3 against crank angle θ_2

CHAPTER FOUR

4.0 DESIGN OF PARTS

4.1 DETERMINATION OF CRUSHER DIMENSIONS

From table 3 1, we obtained the following design dimensions:

$L_T = 1.2\ cm$

Where $L_T = \ length\ of\ throw$

For reduction ratio,$R = 5{:}1$ we set the gape as, 0.3m and a set of 0.06m.

Therefore,

$$L_{min} = 54mm$$

$L_{max} = 66mm$

The rule of the thumb relating the gape and width of the jaw is;

$$1.3G < W < 3G$$

Where,

$G = gape$

$W = width\ of\ jaw$

We set the width of the crusher to be twice the gape. Hence,

$W = 2G, \quad W = 0.6m$

4.1.1 Determination of critical velocity

There exists a speed for a crusher, where the conditions are optimum. The output and power consumption are optimized at that speed. This speed is known as critical velocity.

For the above dimensions, the critical speed in R.P.M is given by the following formula (Rose and English),

$$V_C = 47\left(\frac{1}{{L_T}^{0.5}}\right) * (\frac{R-1}{R})^{0.5}$$

R = Reduction ratio

V_C = Critical velocity

$$V_C = 47\left(\frac{1}{0.012^{0.5}}\right)({}^{4}/_{5})^{0.5}$$

$$= 383.75$$

$$= 384\ R.P.M$$

4.1.2 Determination of Crusher Capacity, Q

a) From Rose and English,

$$Q = 2820L_T^{0.5}W(2L_{min} + L_T)(\frac{R}{R-1})^{0.5}\rho_s f(p_k)f(\beta)S_c$$

Where,

$\rho_s = density\ of\ stones\ or\ rocks$

$f(p_k) = packing\ factor$

$f(\beta) = 1$

$S_c = surface\ characteristics\ of\ rocks$

Substituting for values in the above equation,

$$Q = 2820 * (0.012)^{0.5} * 0.6 * (2 * 0.054 + 0.012) * ({}^{5}/_{4})^{0.5} * 2.65 * 0.4 * 0.5$$

$$Q = 13.17962\ tonnes/hour$$

b) From Michelson equation,

$$Q = 7.037 * 10^5 W K_X (\frac{L_{min} + L_T}{V_C})$$

Where,

$$K_X = 0.18\ to\ 0.3$$

$$Q = 7.037 * 10^5 * 0.6 * 0.2 * (\frac{0.054 + 0.012}{384})$$

$$Q = 14.5137\ tonnes/hour$$

From the two methods, it is clear that for the given dimensions of the crusher, the average crusher capacity, Q, is **14tonnes/hour**.

4.1.3 Power consumption

The jaw crusher is driven by a motor or a diesel engine. The power that the engine or motor should avail to the jaw crusher drive shaft is given by;

$$power, P = Q * 0.3162 w_i \left(\frac{1}{\sqrt{D_{P_b}}} - \frac{1}{\sqrt{D_{p_a}}} \right) * safety\ factor$$

Where,

$D_{P_b} = mean\ product\ size$

$D_{p_a} = mean\ feed\ size$

$w_i = work\ index, which\ is\ 16\ for\ granite$

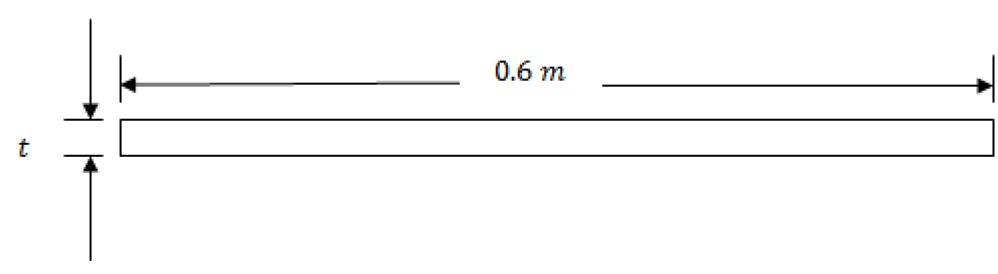
$Q = Crusher\ capacity$

$$P = 14 * 0.3162 * 16 * \left(\frac{1}{\sqrt{54}} - \frac{1}{\sqrt{270}} \right) * 2$$

$$P = 10.656 \cong 11\ kW$$

4.2 DESIGN OF THE JAW

Jaw is a critical part of the machine. Its motion provides the crushing mechanism. The jaw can be modeled as a simply supported wide beam carrying a moment equivalent toT_3;



The cross-section of the jaw plate

Figure 4 1: Jaw cross-section

From force analysis,

$$T_3 = 60T_2\,, and\ power = T_2 * 2\pi V_C/60$$

$T_2 = \dfrac{11000 * 60}{2\pi * 384} = 273\,Nm$

$T_3 = 60 * 273 = 16380\,Nm$

From beam theory,

$\sigma_y = \dfrac{M}{I}\dfrac{t_b}{2}$

Where,

$M = bending\ moment\ in\ the\ jaw$

$I_b = second\ moment\ of\ area\ of\ the\ jaw$

$$I_b = \frac{W t_b^3}{12}$$

$$\frac{\sigma_y}{2} = \frac{250 * 10^6}{2} = \frac{16380 * 12 * t_b}{0.6 * {t_b}^3 * 2}$$

$t_b = thickness\ of\ the\ jaw$

$\sigma_y = yield\ strength\ of\ the\ jaw\ material$

$t = 36.2mm \cong 36mm$

4.3 DESIGN OF FLYWHEEL

A flywheel is a device used in machines as a reservoir which stores energy during the period when the supply of energy is more than the requirement and releases it during the period when requirement of energy is more than the supply. In order to understand the design of a flywheel, we have a look at some terms concerning the operation of a flywheel.

4.3.1 Coefficient of fluctuation of energy, K_e

This is the ratio of maximum fluctuation of energy to the indicated work done by the engine during one revolution of the crank.

$$K_e = \frac{\Delta E}{T_m * \theta}$$

$$\Delta E = E_{max} - E_{min}$$

Where,

$\Delta E =$ maximum luctuation of energy

$T_m\theta =$ work done per cycle

$T_m =$ mean torque

$\theta =$ angle turned in one revolution of the crank

4.3.2 Coefficient of fluctuation of speed, K_s

This is the ratio of difference between maximum and minimum angular velocities of the crankshaft to its men angular velocity. This is the limiting factor in design of flywheel.

$$K_s = \frac{\omega_{max} - \omega_{min}}{\omega_{mean}}$$

$$\omega_{mean} = \frac{\omega_{max} + \omega_{min}}{2}$$

Where,

$\omega_{mean} =$ mean angular velocity

$\omega_{max} =$ maximum angular velocity

$\omega_{min} =$ minimum angular velocity

The following table gives specific values of K_s that have been recommended for different systems.

Table 4 1: Values of K_s for Different Systems[1] for most stone crushers the: $K_s = 0.2$

No	Type of machine or class of service	Coefficient of fluctuation of speed (K_S)
1.	Crushing machines	0.200
2.	Electrical machines	0.003
3.	Electrical machines (direct drive)	0.002
4.	Engines with belt transmission	0.030
5.	Gear wheel transmission	0.020
6.	Hammering machines	0.200
7.	Pumping machines	0.03 to 0.05
8.	Machine tools	0.030
9.	Paper making, textile and weaving machines	0.025
10.	Punching, shearing and power presses	0.10 to 0.15
11.	Spinning machinery	0.10 to 0.020
12.	Rolling mills and mining machines	0.025

4.3.3 Fluctuation of energy

Fluctuation of energyE_f is the excess energy developed by the engine between two crank positions. It is determined from the turning moment diagram for one complete cycle of operation.

$$E_f = K_e E$$

$$E = \frac{I\omega_m{}^2}{2}$$

$$I = mk^2$$

K_e = coef icient of luctuation of energy of the lywheel

E = mean kinetic energy of the lywheel

ω_m = mean angular speed of the lywheel

I = moment of inertia of the lywheel

m = mass of the lywheel

k = radius of gyration of the lywheel

As speed of flywheel changes from ω_{max} to ω_{min} the maximum fluctuation of energy is as in equation (5.6) and this can be expanded as:

$$\Delta E = E_{max} - E_{min} = \frac{I[\omega^2_{max} - \omega^2_{min}]}{2}$$

$$= \frac{1}{2} I(\omega_{max} + \omega_{min})(\omega_{max} - \omega_{min})$$

Combining the equations we obtain;

$$\Delta E = mk^2 K_s {\omega_{mean}}^2$$

k may be taken to be equal to the mean radius of the rim (R), since the thickness of the rim is very small as compared to the diameter of the rim, hence equation above becomes;

$$\Delta E = mR^2 K_s {\omega_{mean}}^2$$

Or

$$R = \sqrt{\left\{\frac{\Delta E}{mK_s {\omega_{mean}}^2}\right\}}$$

4.3.4 Stresses in a Flywheel

Assuming the rim is unstrained by the arms, the tensile stress in the rims due to centrifugal force is determined as a thin cylinder subjected to internal pressure.

Let

$$W_{rim} = \text{Width of the rim}$$

t_{rim} = thickness of the rim

$$A = \text{area of X} - \text{section of the rim} = W_{rim} * t_{rim}$$

D_f = mean diameter of the lywheel

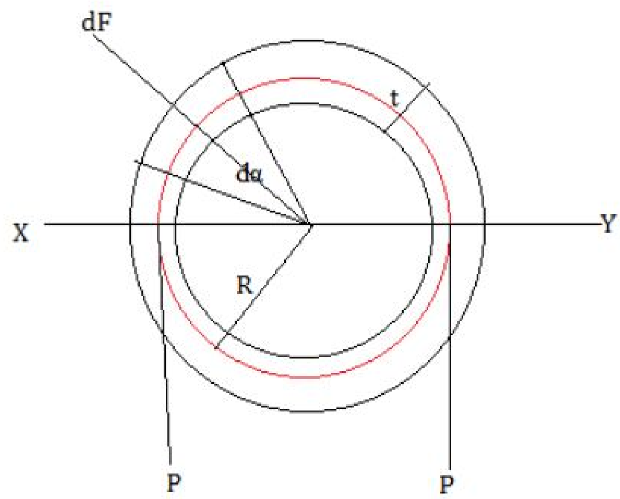
R_f = mean radius of the lywheel

ρ= density of the flywheel

ω= angular speed of the flywheel

μ= linear velocity of the flywheel

σ_t = tensile or hoop stress



The volume of this small element= $A.R_f.d\alpha$

Mass of the element= volume x density = $A.R_f.d\alpha\,\rho$

Centrifugal force on this element = $dF = dm.\omega^2.R_f = \rho A.R_f{}^2.\omega^2.d\alpha$

Vertical component= $dF \operatorname{Sin}\alpha = \rho A.R_f{}^2.\omega^2.d\alpha \operatorname{Sin}\alpha$

Total vertical force across the rim diameter,$X - Y = \rho A.R_f{}^2.\omega^2.\int \operatorname{Sin}\alpha\, d\alpha = 2\rho A.R_f{}^2.\omega^2$

This vertical force is restricted by a force 2P such that

$$2P = 2\sigma_t * A = 2\rho A.R_f{}^2.\omega^2$$

$$\sigma_t = \rho R_f{}^2.\omega^2 = \rho R_f{}^2.\omega^2$$

For our crusher: $\omega = 40.212$ rad/s$\rho = 7250$kg/m^3Factor of safety = 4

$$\sigma_{yp} = 6 * 10^6 N/m^2$$

$$T_{max} = \frac{1.4(273 * 4\pi)}{1.571} = 3057.2035\, Nm$$

$$Power\ stroke = 273 * 1.4 * 4\pi = 4802.8668\, Watts$$

$$\frac{Work}{cycle} = 273 * 4\pi = 3430\, joules/cycle$$

$$\Delta E = Power\ stroke * (\frac{T_{max} - T_{mean}}{T_{max}})^2 = 3983.3989\, Joules$$

For stone crushers, $K_s = 0.2$

$$I = \left(\frac{\Delta E}{\omega^2}\right) * K_s = 12.3169\, Kgm^2$$

For flywheels, $W_{rim} = 2t_{rim}$

$$\sigma_t = \rho R^2 \omega^2$$

$$1.5 * 10^6 = 7250 * R^2 * 40.212^2$$

$$R = 0.3576\, m$$

$$E = \frac{\Delta E}{2K_s} = 9958.49\, joules$$

$$E_{rim} = 0.92E = 9161.81756 joules = \frac{1}{2} m R^2 \omega^2$$

$$m = \frac{2E_{rim}}{R^2 \omega^2} = 88.612\, kgs$$

$$m = W_{rim} t_{rim} * 2\pi R * \rho$$

Therefore, $t_{rim} = 52.15\, cm\ and\ W_{rim} = 104.30\, cm$

Where, W_{rim} = Width of the rim

t_{rim} = thickness of the rim

$E_{rim} = energy\ stored\ in\ the\ rim\ of\ the\ flywheel$

4.4 DESIGN OF THE SHAFT

Given the weight of the rim is 88.612Kg, we estimated the weight of the flywheel to be 120Kg each i.e. 1177.2N.

$$Mass\ of\ jaw\ and\ wear = density * volume = (8400 * 0.08 * 0.6 * 0.7) = 282.24\, Kg$$

$$Mass\ of\ rocks = \left(\frac{1}{2} * 0.3 * 0.76 * 0.6\right) * 2650 = 181.6\, Kg$$

But the rocks normally occupy just about 60% of the capacity hence the mass of the rocks is just about 100 Kg. The mass of the jaw and plate approximated as 300 Kg. But since the shaft is a very critical component and works at very high speed, we designed it to carry twice that load. Hence the weight distributed on the point of contact with the shaft was 3924N on each of the points of contact at the bearings.

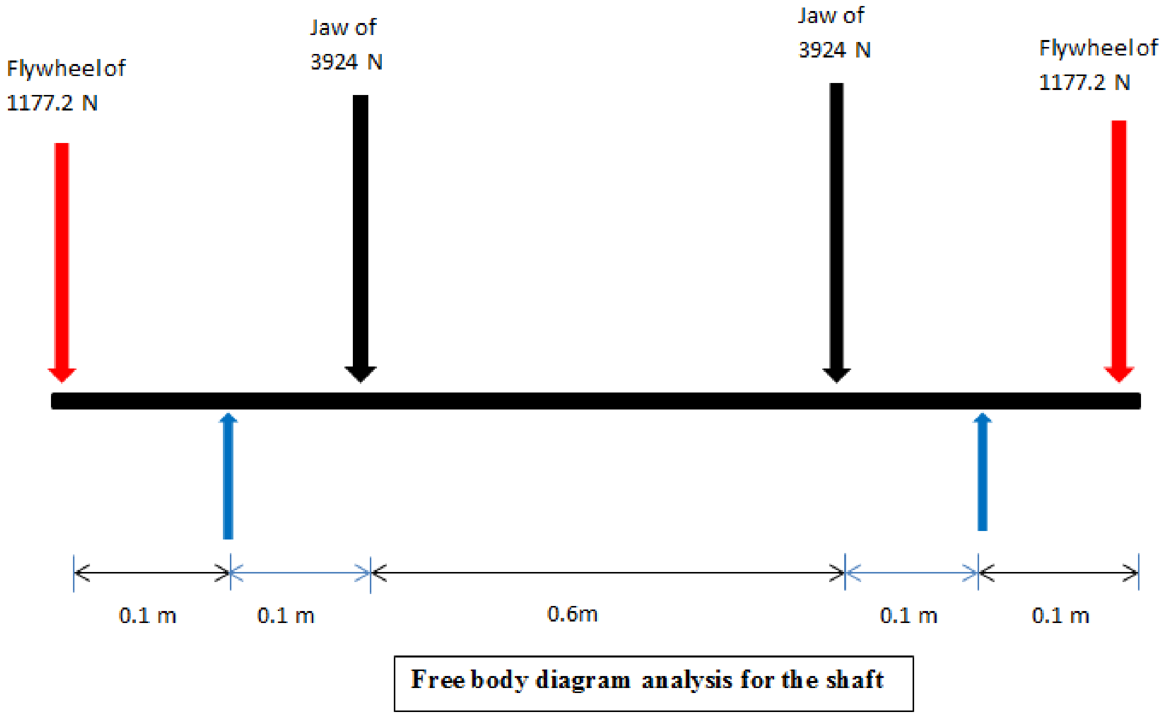


Figure 4 2: Free body diagram of the shaft

$$\sum F_y = 0$$

Reactions at the supports from equality of action and reaction forces,

$$R_S = 1177.2 + 3924 = 5101.2\ N$$

Using singularity method;

$$M_x = -1177.2(x-0) + R_s(x-0.1) - 3924(x-0.2) - 3924(x-0.8) + R_s(x-0.9) - 1177.2(x-1)$$

$$At\ X = 0.1, M_{X=0.1\ m} = -117.72\ Nm$$

$At\ X = 0.2, M_{X=0.2\ m} = 274.68\ Nm$

$At\ X = 0.3, M_{X=0.3\ m} = 274.68\ Nm$

$At\ X = 0.4, M_{X=0.4\ m} = 274.68\ Nm$

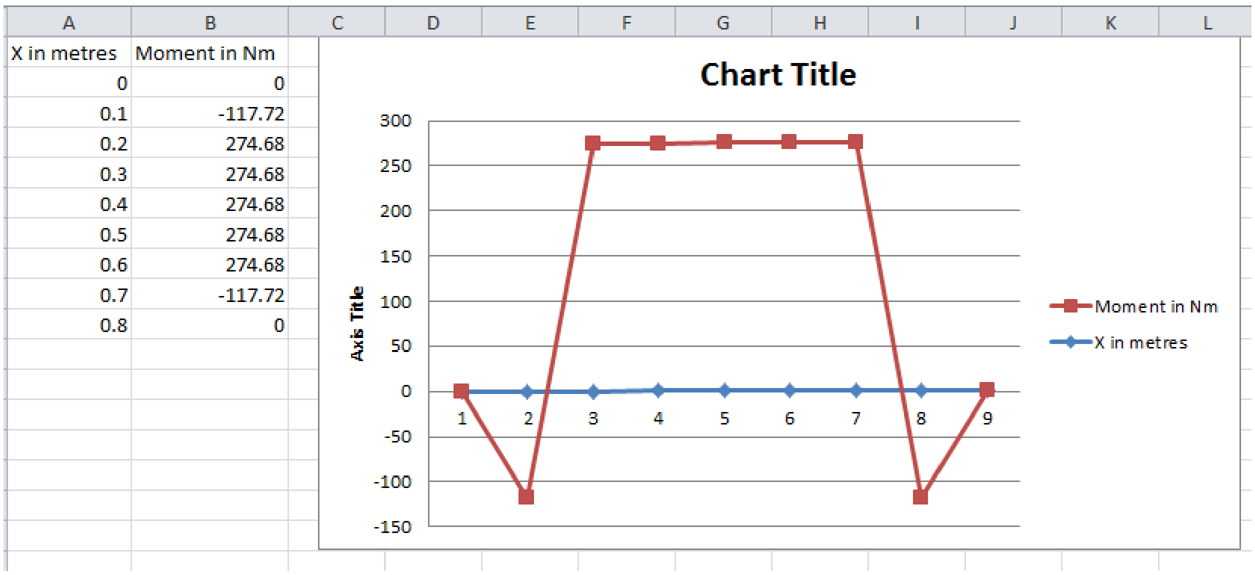


Figure 4 3: Moment distribution

Given M_{max} = 274.68 Nm andT_{max} = 3057.2035;

$$d_{Shaft}{}^{3} = 16/\pi\sqrt{K_b T^2 + K_t m^2} * 2\ s.f/\sigma_{yp}$$

For σ_{yp} = $300 * 10^6 N/m^2$

d_{Shaft} = 6.3465 cm

But from inventor application, we designed a shaft of 6.5 cm due to the standardization of bearings.

The loading on the shaft involves shock and fatigue. To account for these, the equation has to be modified using factors that depend on the type of loading.

Table 4 2: Shock and fatigue factor table

Type of loading	**k_b**	**k_τ**
Load gradually applied	1.5	1
Load suddenly applied with minor shock	1.5 – 2.0	1.0 – 1.5
Load suddenly applied with heavy shock	2.0 – 3.0	1.5 – 2.0

In jaw crushers, the load is applied suddenly with a moderate shock. Therefore, from the table of shock and fatigue factors above, the appropriate values chosen are:

$$K_b = 2.0$$

$$K_\tau = 1.5$$

Where

K_b = shock and fatigue factor for bending moment

K_τ = shock and fatigue factor for torsion

4.5 DESIGN OF HUB

Diameter of the hub = $twice\ the\ diameter\ of\ the\ shaft$ = 13.0 cm

Length of the hub = $width\ of\ the\ rim$ = 10.4305 $cm \simeq$ 10.43 cm

4.6 DESIGN OF SPRING

For a spring, the value of C should lie between 4 and 16. We opted for the minimal ratio of 4 to minimize the spring size.

$$K = \frac{4C - 1}{4C - 4} * \left(\frac{0.615}{C}\right) = 0.1921875$$

From the normalized torque ratios, the values for the torque on the jaw varied from

$$39T_2 \leq T_3 \leq 60T_2$$

Hence it varied from 10647 Nm to 16380 Nm, and these were the variations during the cycle that where absorbed by the spring, a variation in the torque of 5733 Nm. For a length of 0.7 m of the jaw, this implied a force of $\frac{5733}{0.7} = 8190\ N$

Resolving this force in the z-direction $F_{Z43=}8190 \cos 20 = 7696\ N$

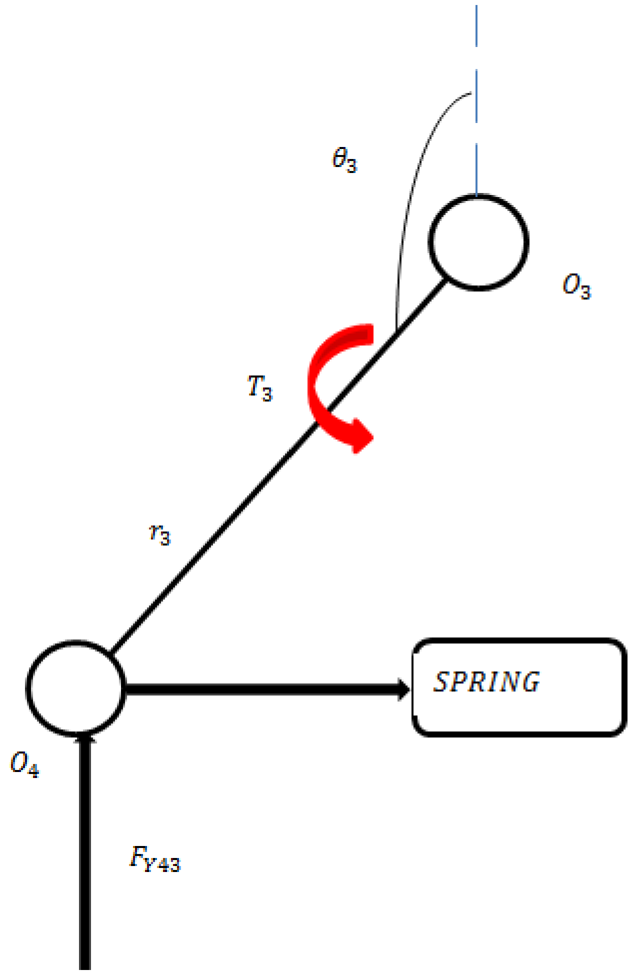


Figure 4 4: Free body illustration of force on the spring

For a load W_S of 7696 N $\simeq$ 7700 N , d_{sp} = 13mm and D_{sp} = 52 mm

$$\tau = \frac{K * 8 * W_S * D_1}{\pi * {d_{sp}}^2} = 89.19269 \text{ M Pa}$$

τ = $maximum\ shear\ stress\ on\ the\ spring$

K = $spring\ factor$

W_S = $load\ on\ the\ spring$

$D_1 = d_{sp} + D_{sp}$

d_{sp} = $Diameter\ of\ steel\ wire, used\ for\ making\ the\ spring$

$D_{sp} = \ Mean\ diameter\ of\ spring$

Using a carbon steel of 2nd grade with 8 active coils,

$$\delta = \frac{8 * W_S * C^3 * n}{G_m * d_{sp}} = 30.13\ mm$$

$\delta = deflection\ of\ the\ spring$

$G_m = shear\ modulus\ of\ elasticity\ of\ the\ material$

$n = no\ of\ active\ coils\ in\ the\ spring$

$n' = n + 2 = total\ no\ of\ coils\ in\ the\ spring$

$L_f = free\ length\ of\ the\ spring$

$\delta_{max} \simeq 1.5\delta$

$$L_f = n' * d_{sp} + \delta_{max} + 0.15\delta_{max} = 175.884\ mm$$

$P_{sp} = pitch\ of\ the\ spring$

$$P_{sp} = \frac{L_f}{n' - 1} = 19.542\ mm$$

4.7 DESIGN OF BEARINGS

4.7.1 Dynamic Load Rating Variable Loads

The approximate rating (or service) life of ball or roller bearings is based on the fundamental equation;

$$L_b = \{\frac{C_b}{W_b}\}^k \times 10^6\ rev$$

Where k=3 for small bearings

L_b = Rating life

C_b = Basic dynamic load rating

W_b = Equivalent dynamic load

The reliability (R) is defined as the ratio of the number of bearings which have successfully completed $L_b$million revolutions to the total number of bearings under test. According to Wiebull, the relation between the bearing life and the reliability is given as;

$$log_e\left\{\frac{1}{R}\right\} = \{L_b / a\}^b$$

$$where \;\; a = 6.84$$
$$b = 1.17$$

The total force on one bearing is approximately 3924N.
The bearings carry maximum load of 3924N during working hours of about 33% of the total time and a lesser load of1.962KN for about 67% of the time.
We fix our bearings to have an expected life of 20×10^6 revolutions at 95% reliability. Since most catalogues are tabulated at 90% reliability, the following formula converts to 90% reliability:

$$\{L_{95} / L_{90}\} = \{ln(1/R_{95}) / \ln(1/R_{90})\}^{1/b}$$

Substituting the values in the above equation,

$$\frac{L_{95}}{L_{90}} = \{\frac{\ln\frac{1}{0.95}}{\ln\frac{1}{0.9}}\}^{1/1.7}$$

=0.54.

Considering life adjustment factors for the operating conditions and material to be 0.9 and 0.85 respectively,

$$\frac{L_{95}}{L_{95}} = .90 \times 0.85 \times 0.54 = 0.4131$$

Therefore,

$$L_{90} = (20 * 10^6) / 0.4131 = 48.4 \times 10^6 \; revolutions$$

Equivalent radial load

$$W_b = \{(n_1 w_1^3 + n_2 w_2^3) / (n_1 + n_2)\}\text{^}(\frac{1}{3})$$

$$W_b = \{\frac{0.33 * 3924^3 + 0.67 * 1962^3}{0.33 + 0.67}\}^{\frac{1}{3}}$$

Therefore, $W_b = 2.92398 \; KN$

Dynamic load rating C_b is given the following expression,

$$C_b = \left(\frac{L_{90}}{10^6}\right)^{\frac{1}{K}} = 2.92398\{(48.4 * 10^6)/10\text{^}6\}^{1/3}$$
$$= 10.656\ KN$$

From the catalogue of bearings, we choose for a bearing type with

$$C_0 = 10.656\ KN$$
$$L_{90} = 48.4 \times 10^6\ revolutions$$

Based on the above values, the available bearings for the shaft is JIS 1523 SKF, self-aligning ball bearings double row with cylindrical bore SKF, mild steel. For the 2 sets of bearings, we choose those with the following specifications:

Table 4 3: Bearing specifications

FILE NAME	**JIS 1523 SKA(A) SKF 2218**
INSIDE DIAMETER	90mm
OUTSIDE DIAMETER	160mm
FILLET RADIUS	2 mm
STATIC LOAD RATE	28500N
LIMITING SPEED	3600 RPM
FRICTION FACTOR	0.001

FILE NAME	**JIS 1523 SKA(A) SKF 2213E**
INSIDE DIAMETER	65 mm
OUTSIDE DIAMETER	120mm
FILLET RADIUS	1.5 mm
STATIC LOAD RATE	20000N
LIMITING SPEED	5000 RPM
FRICTION FACTOR	0.001

4.8 DESIGN OF V-BELTS

The *V*-belts are mostly used in situations where great amount of power is to be transmitted from one pulley to another and when the two pulleys are very near to each other. They are made of fabric and cords moulded in rubber and covered with fabric and rubber.

The wedging action of the *V*-belt in the groove of the pulley results in higher forces of friction. Analysis shows that the wedging action and the transmitted torque is more if the groove angle of the pulley is small. But a small groove angle require more force to pull the belt out of the groove which result in loss of power and excessive belt wear due to friction and heat. Hence the selected groove angle is a compromise between the two. Usually the groove angles of 30° to 40° are used.

A clearance must be provided at the bottom of the groove in order to prevent touching of the bottom as it becomes narrower from wear. The *V*-belt drive may be inclined at any angle with tight side either at top or bottom. In order to increase the power output, several *V*-belts may be operated side by side. It may be noted that in multiple *V*-belt drive, all the belts should stretch at the same rate so that the load is equally divided between them. When one of the set of belts breaks, the entire set should be replaced at the same time. The figure below shows the cross-section of belt and pulley.

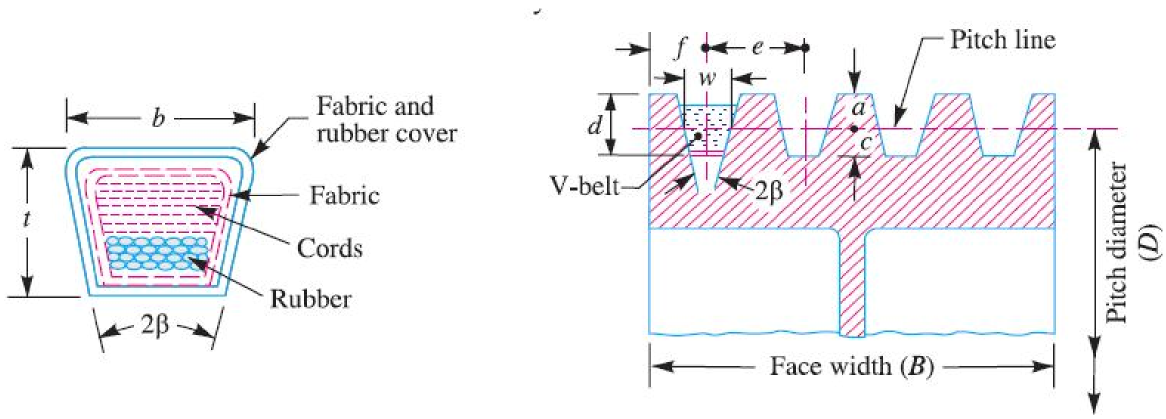


Figure 4 5: Belt Cross section

V-belts are advantageous over other types in the following ways:

- The *V*-belt drive gives compactness due to the small distance between centers of pulleys.
- The drive is positive, because the slip between the belt and the pulley groove is negligible.
- Since the *V*-belts are made endless and there is no joint trouble, therefore the drive is smooth.
- It provides longer life, 3 to 5 years.
- It can be easily installed and removed.
- Less expensive than gear or chain drives.
- Have flexible shaft center distances where gear drives are restricted.
- Operate smoothly and with less noise at high speeds.
- They can be designed to slip when an overload occurs in the machine.
- They require no lubrication, as do chains and gears.

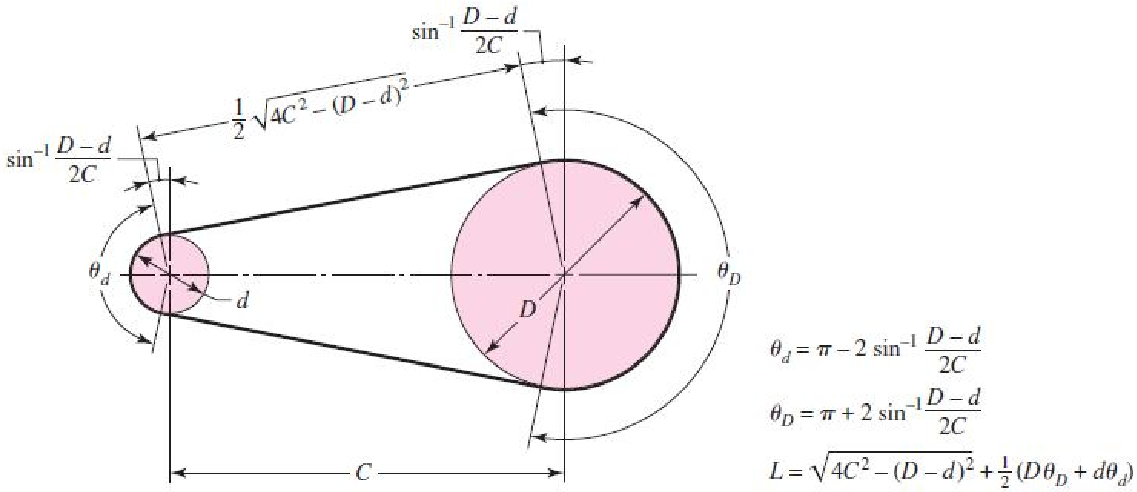


Figure 4 6: Belts and Pulley

The crusher is designed to run at 384 R.P.M.

Speeds in the two pulleys are related by the equation

$$\frac{N_d}{N_D} = \frac{D}{d}$$

N_d = *speed in the small pulley in R.P.M*

N_D = *speed in the larger pulley, in R.P.M*

Fixing the smaller diameter $d = 0.1m$,

Then, $D = \frac{2500}{384} \times 0.1 = 0.651 \simeq 0.65m$

From the above formula for

$$\theta_d = \pi - 2sin^{-1}\frac{D-d}{2C}$$

$$\theta_d = \pi - 2sin^{-1}\frac{0.65 - 0.1}{2 \times 1.2}$$

$$= 153.5^{\circ} = 2.679\ radians$$

$$\theta_D = \pi + 2sin^{-1}\frac{0.65 - 0.1}{2 \times 1.2}$$

$$= 206.5^{\circ} = 3.604\ radians$$

The length of the belt is given by the expression,

$$L = \sqrt{4C^2 - (D-d)^2} + {}^1\!/_2\,(D\theta_D + d\theta_d)$$

$$= \sqrt{4 \times 1.2^2 - (0.65 - 0.1)^2} + {}^1\!/_2\,(0.65 \times 3.604 + 0.1 \times 2.679)$$

$$L = 3.641\,m$$

Mass of one belt is given by, $m = A * L * \rho$

Taking density, $\rho = 1200\,{}^{kg}\!/_{m^3}$ and $2\beta = 40^\circ$ as from Shigleys Table 17–9

$$mass, m = 1200 * 3.641 * A = 4369.2A$$

Centrifugal tension in the belt is given by, $T_C = mv^2$

$$T_C = m * 13.069^2 = 170.8 * 4369.2A = 746\,259.36A$$

Maximum tension in the belt is given by, $T = \sigma * \mathrm{A}$

Given the allowable stress in the belt is, $\sigma_{all} = 7.0\,MPA$,

Therefore,$T = 7.0 * 10^6 * A\,newtons$

Tension on the belt in the tight side,$T_1 = T - T_C = (7.0 * 10^6A - 746\,259.36A)$

$$T_1 = 6.2537 * 10^5A$$

$$T_2 = Tension\ in\ the\ slack\ side$$

The above parameters are related to the groove angle and coefficient of friction by the following expression,

$$2.3log\left(\frac{T_1}{T_2}\right) = \mu\theta_d cosec\,\beta$$

Substituting for the values in the above equation,

$$2.3log\left(\frac{6.2537 * 10^5\,A}{T_2}\right) = 0.12 * 2.679cosec\,20$$

$$T_2 = 2.44 * 10^5A$$

Power transmitted by the belts is given by, $P = (T_1 - T_2)v * no\ of\ belts$

But from earlier calculations, the power transmitted is 11 KW

Taking the number of belts to be 3,

Then, $11 * 10^3 = (6.2537 * 10^5 A - 2.44 * 10^5 A) \times 13.069 * 3$

Therefore, the cross sectional area of each v-belt is $A = 735\ mm^2$

From Shigleys, the standard dimensions for v-belts, we choose the dimensions that give the nearest area to the one calculated above.

Belt Section	Width a, in	Thickness b, in
A	$\frac{1}{2}$	$\frac{11}{32}$
B	$\frac{21}{32}$	$\frac{7}{16}$
C	$\frac{7}{8}$	$\frac{17}{32}$
D	$1\frac{1}{4}$	$\frac{3}{4}$
E	$1\frac{1}{2}$	1

Figure 4 7: V-Belts dimensions

(Table 17–9Standard V-Belt Sections, page 899)

From the above table, belt section D gives a cross-section area of $734 mm^2$

Therefore, our selection for the v-belt will be,

Table 4 4: V-Belt specifications

Width	**38.1 mm or 1.5 in**
Thickness	**25.4mm or 1 in**
Groove angle 2β	$\mathbf{40^{\circ}}$
Length	$\mathbf{3.641\ m}$
Coefficient of friction	**0.12**
Allowable tensile stress	**7 M Pa**

4.9 DESIGN OF KEY

For standard keys, according to IS: 2292 and 2293-1974 (Reaffirmed 1992).

Proportions of standard Parallel Tapered and Gib head keys.

Shaft diameter(mm) for values up to (mm)	Key cross-section Width (mm)	Key cross-section Thickness (mm)	Shaft diameter(mm) for values up to (mm)	Key cross-section Width (mm)	Key cross-section Thickness (mm)
6	2	2	85	25	14
8	3	3	95	28	16
10	4	4	110	32	18
12	5	5	130	36	20
17	6	6	150	40	22
22	8	7	170	45	25
30	10	8	200	50	28
38	12	8	230	56	32
44	14	9	260	63	32
50	16	10	290	70	36
58	18	11	330	80	40
65	20	12	380	90	45
75	22	14	140	100	50

$$T = L_{Key} * W_{Key} * \tau_{Key} * (d_{Shaft})/2$$

For a shaft of diameter 65 mm, the standard parallel, tapered and gib key, it should have a key of width 20 mm.

$$273 = L_{Key} * 0.02 * 20 * 10^6 * 0.065/2$$

$$\text{Therefore, } L_{Key} = 21\ mm$$

$L_{Key} = length\ of\ key$

$W_{Key} = width\ of\ key$

$\tau_{Key} = shear\ strenth\ of\ key\ material$

4.10 DESIGN OF TOGGLE

From table 3 2, at the point of maximum T_3/T_2 and using equation 3.6, we can obtain the maximum value of F_3 as;

$$F_3 = (T_2)/(r_3 sin2(\theta_3))$$

$$F_3 = \frac{-4421.31 * 273}{0.7 \sin(2 * 159.496)} = 2{,}627{,}864\ N$$

$$P_{cr} = (\pi^2 * E * I_g)/{L_g}^2$$

$$I_g = \frac{P_{cr} * {L_g}^2}{\pi^2 * E} = \frac{2{,}627{,}864 * 0.25^2}{\pi^2 * 200 * 10^9} = 8.3205 * 10^{-8}$$

$$I_g = b * t^3/12$$

$$t = \sqrt[3]{12 * I_g/b} = 0.0149\ m = 1.49\ cm$$

$I_g = 2^{nd} moment\ of\ area\ of\ the\ toggle$

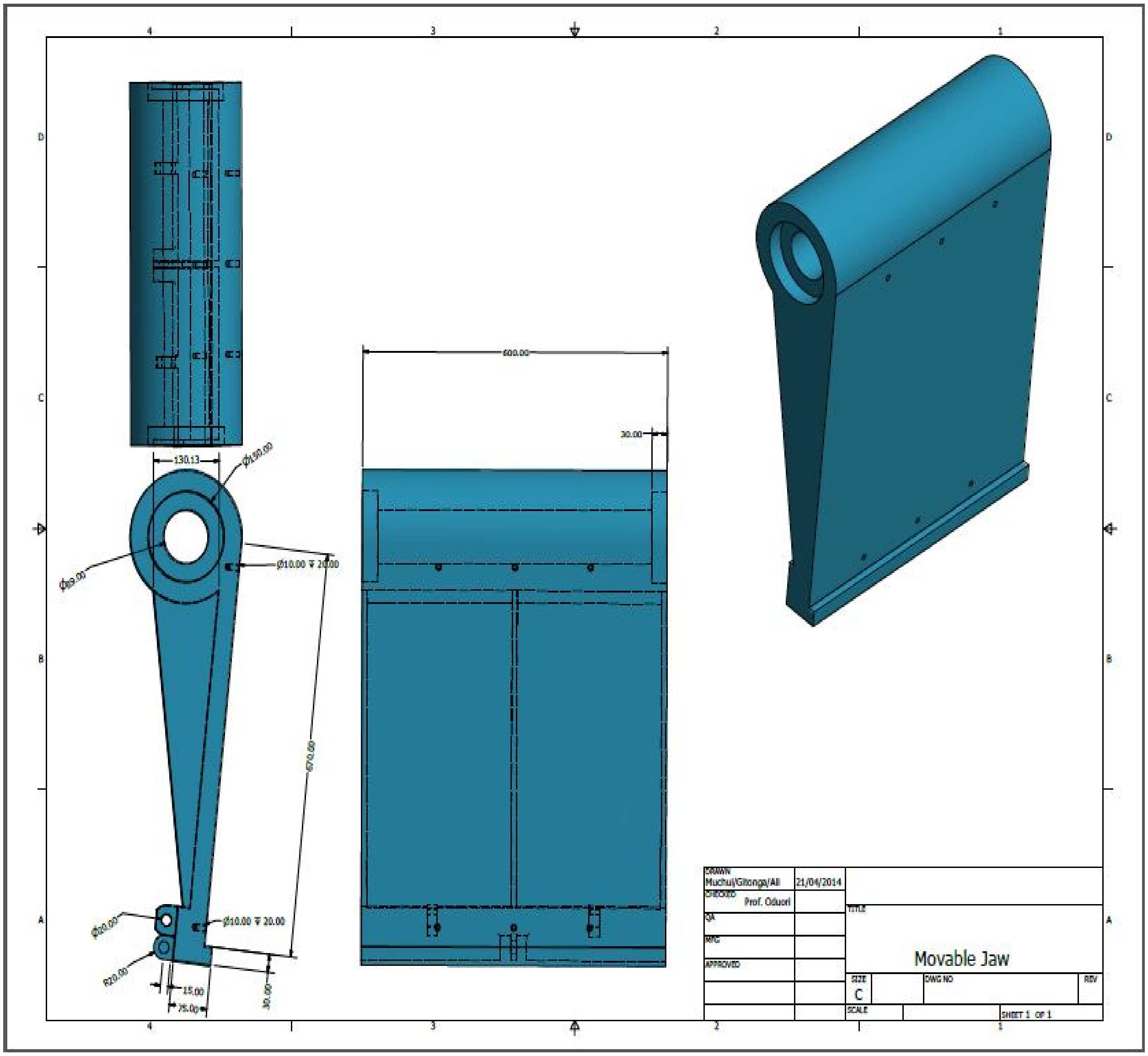
$P_{cr} = critical\ load$

Based on this, we designed a toggle with a thickness of 2 cm.

CHAPTER FIVE

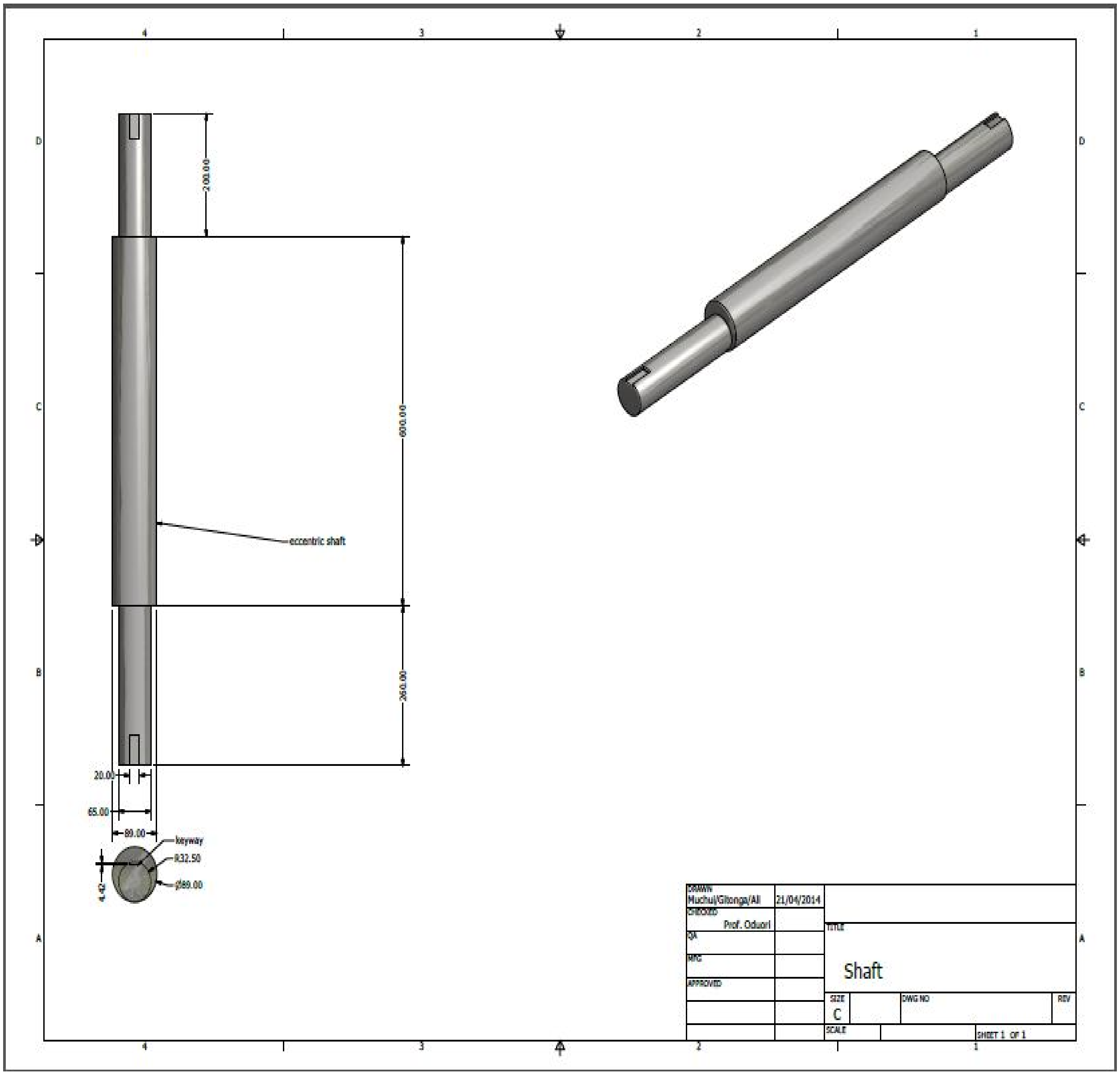
5.0 INVENTOR DRAWINGS

5.1 JAW PLATE



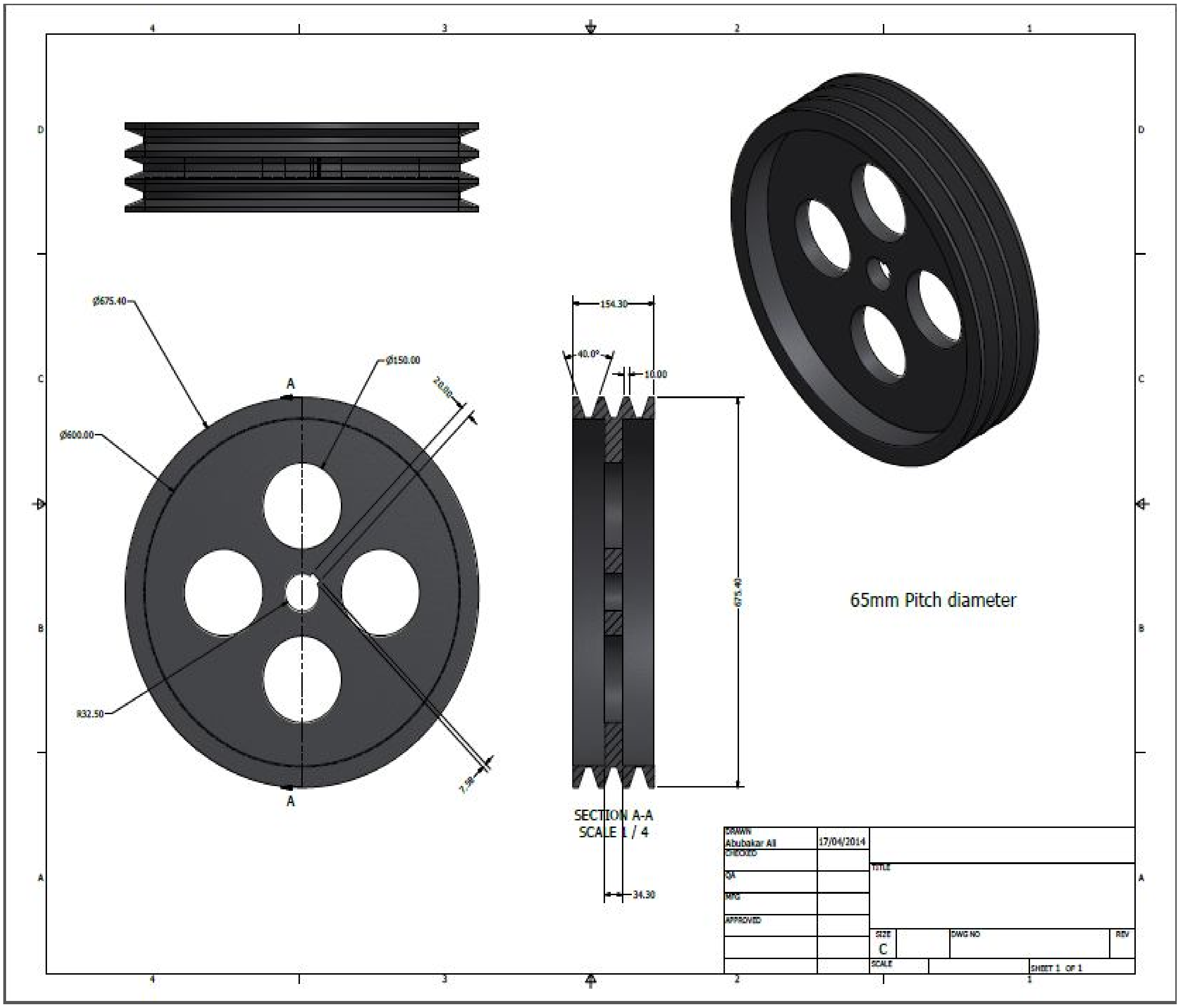
5.1 Inventor Drawing 1JAW PLATE

5.2 SHAFT



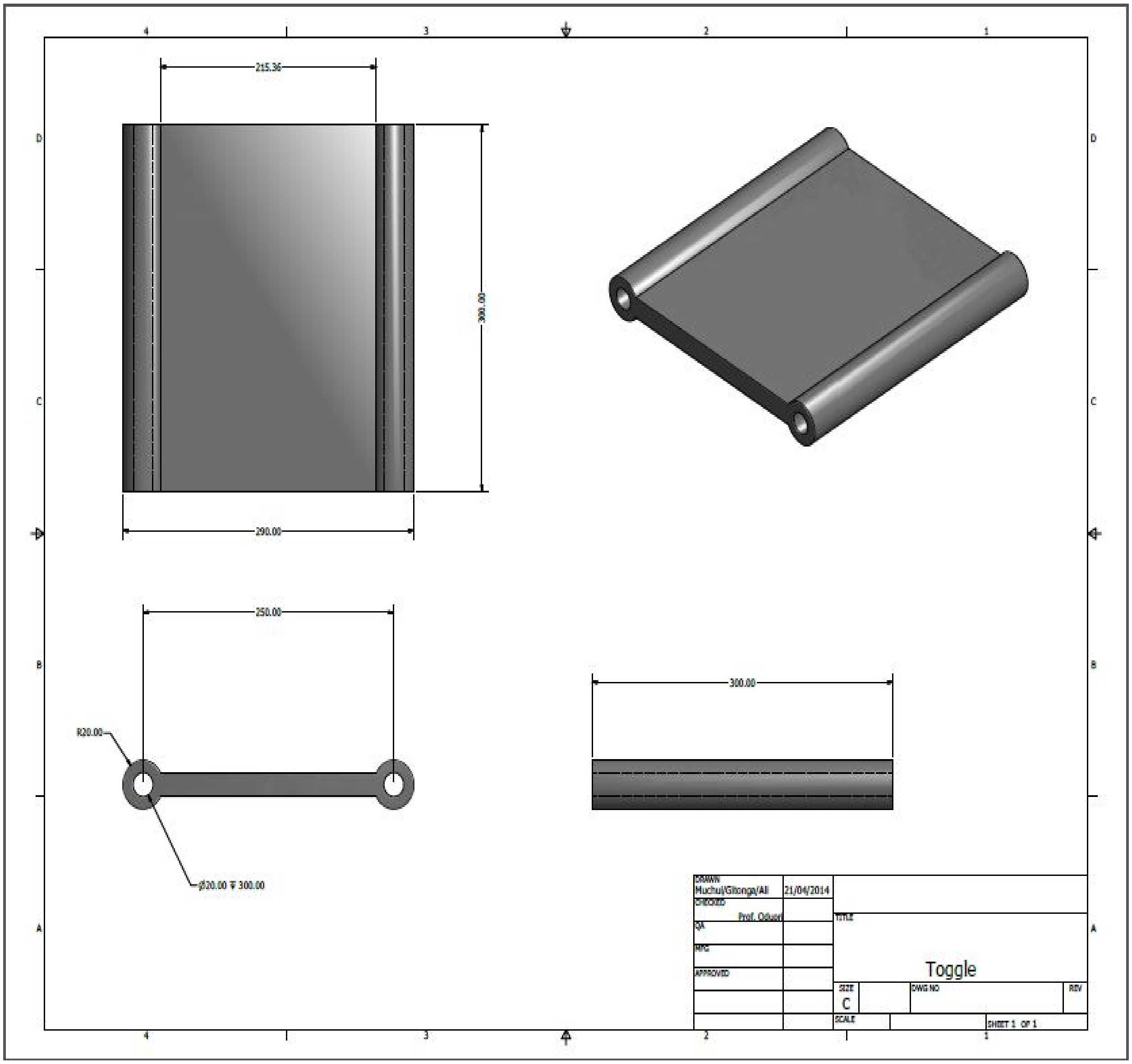
5.2 Inventor Drawing 2SHAFT

5.3 SHEAVE



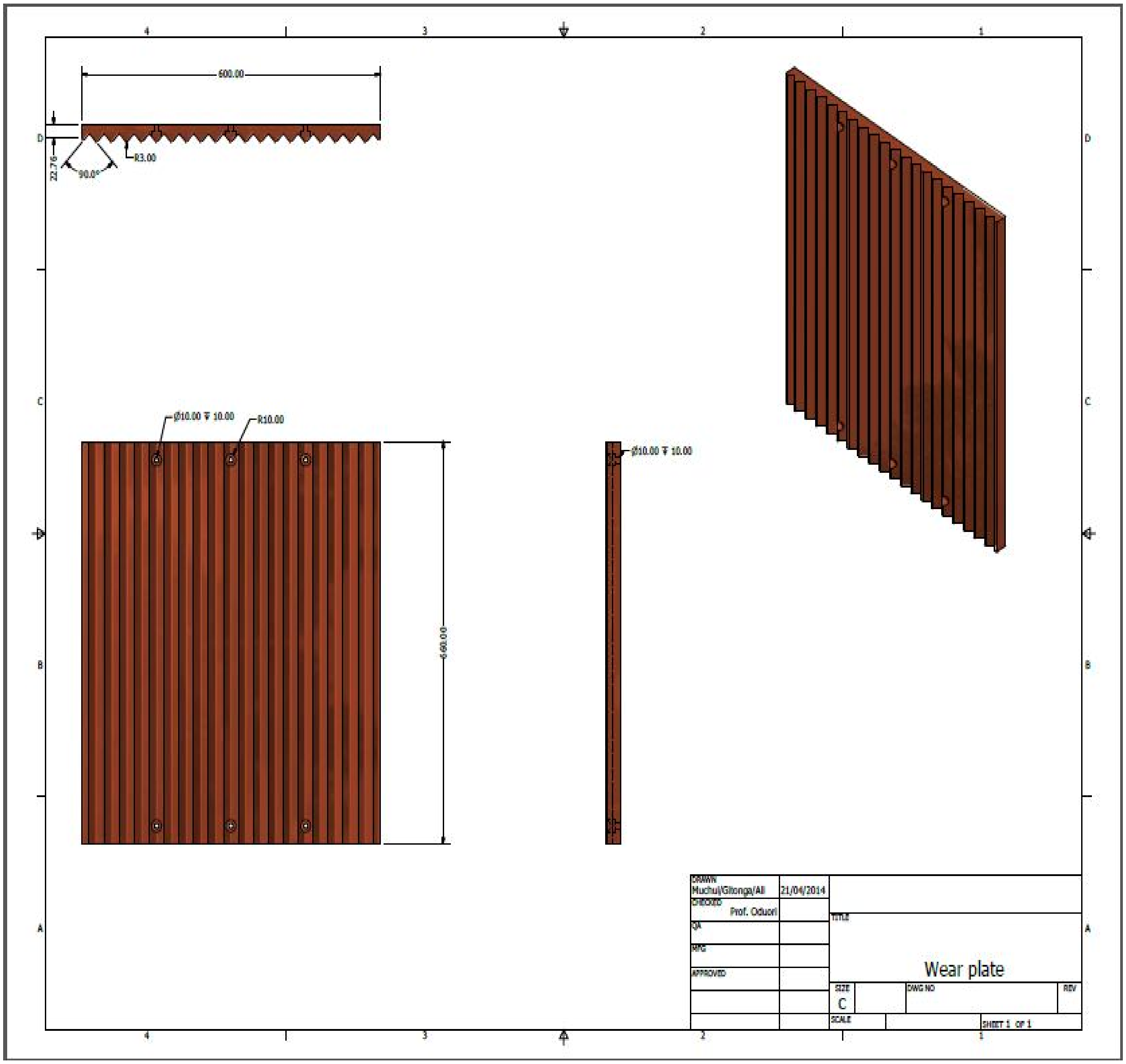
5.3 Inventor Drawing 3SHEAVE

5.4 TOGGLE



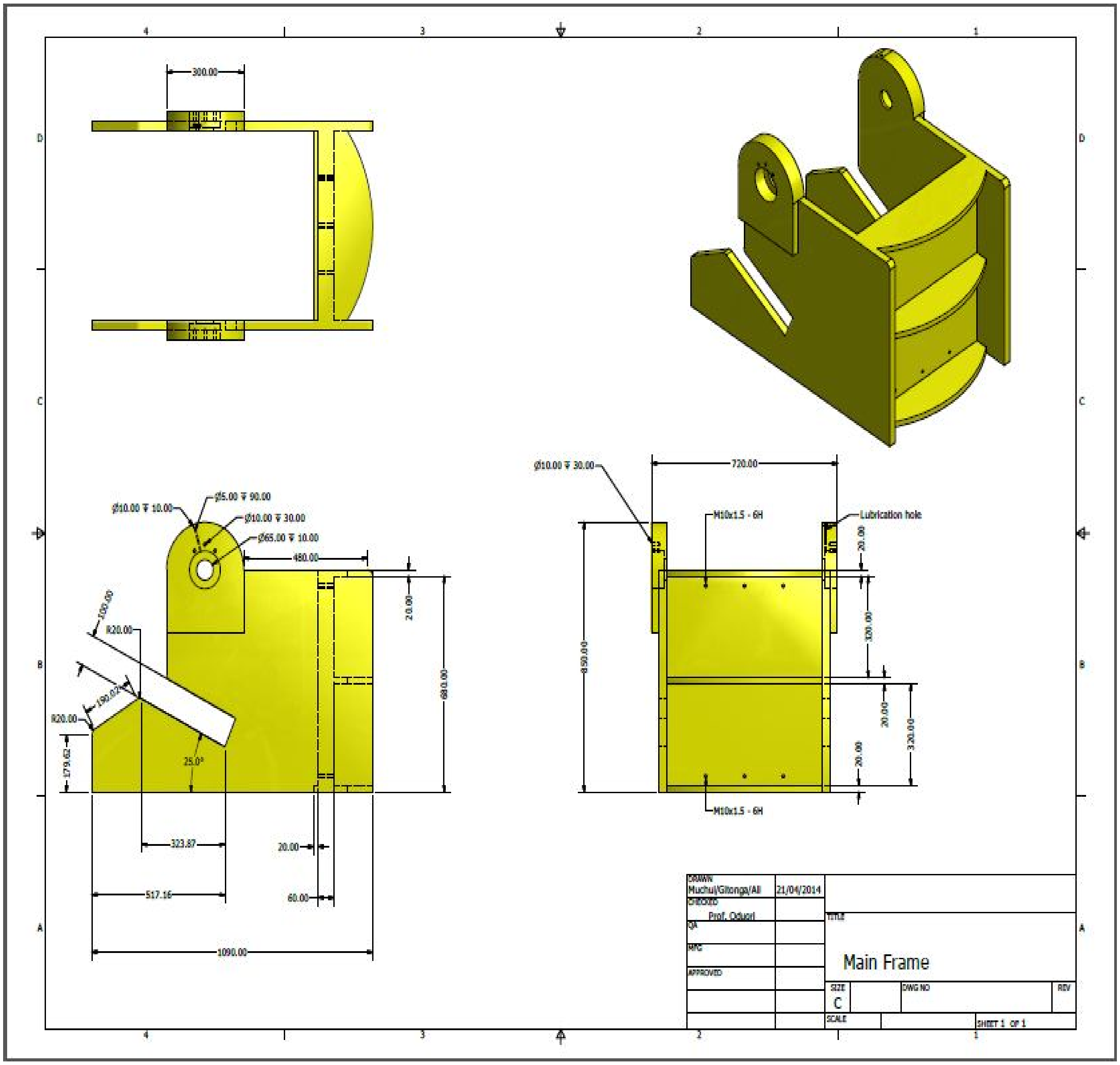
5.4 Inventor Drawing 4TOGGLE

5.5 WEAR PLATE



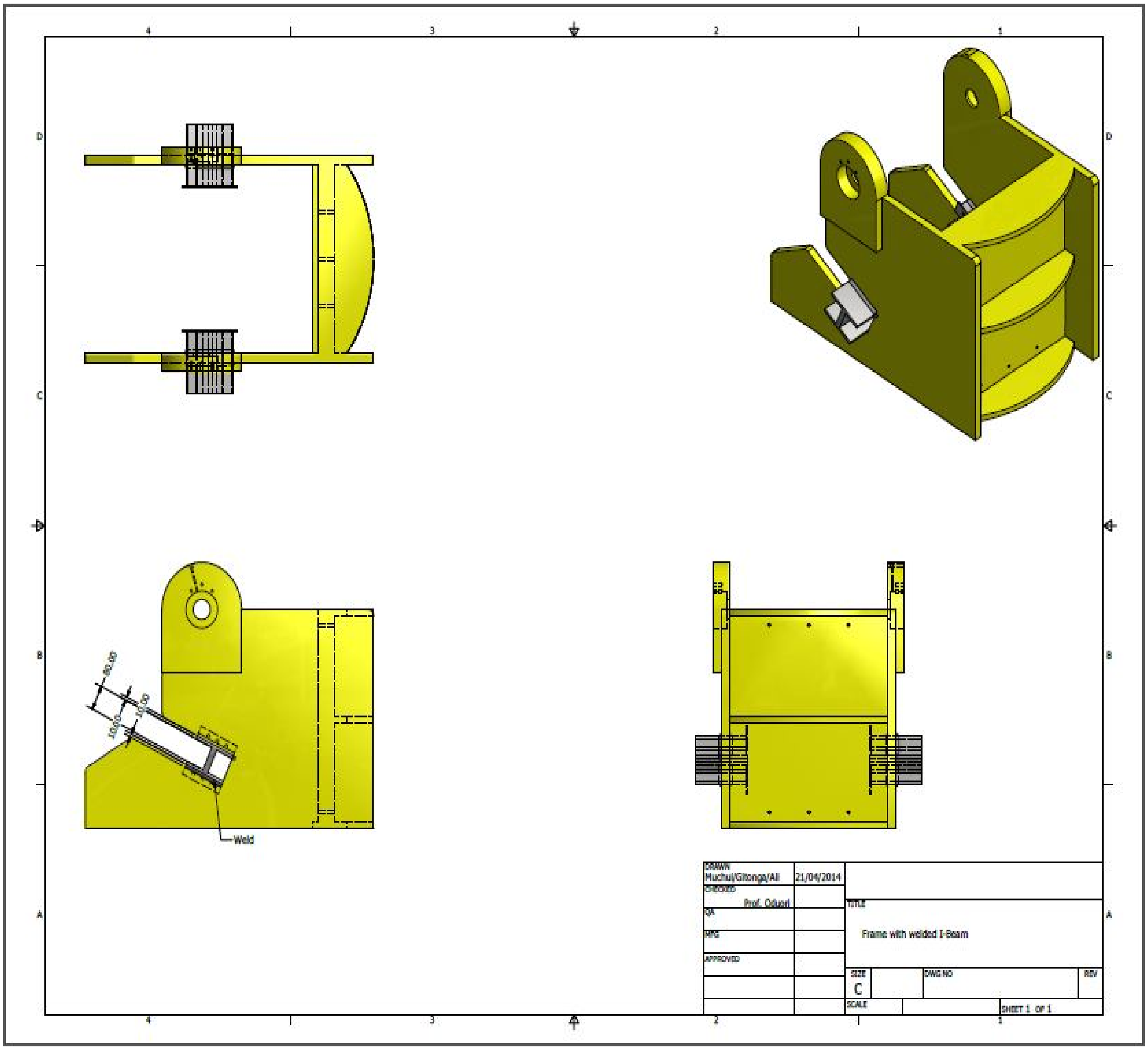
5.5 Inventor Drawing 5WEAR PLATE

5.6 MAIN FRAME



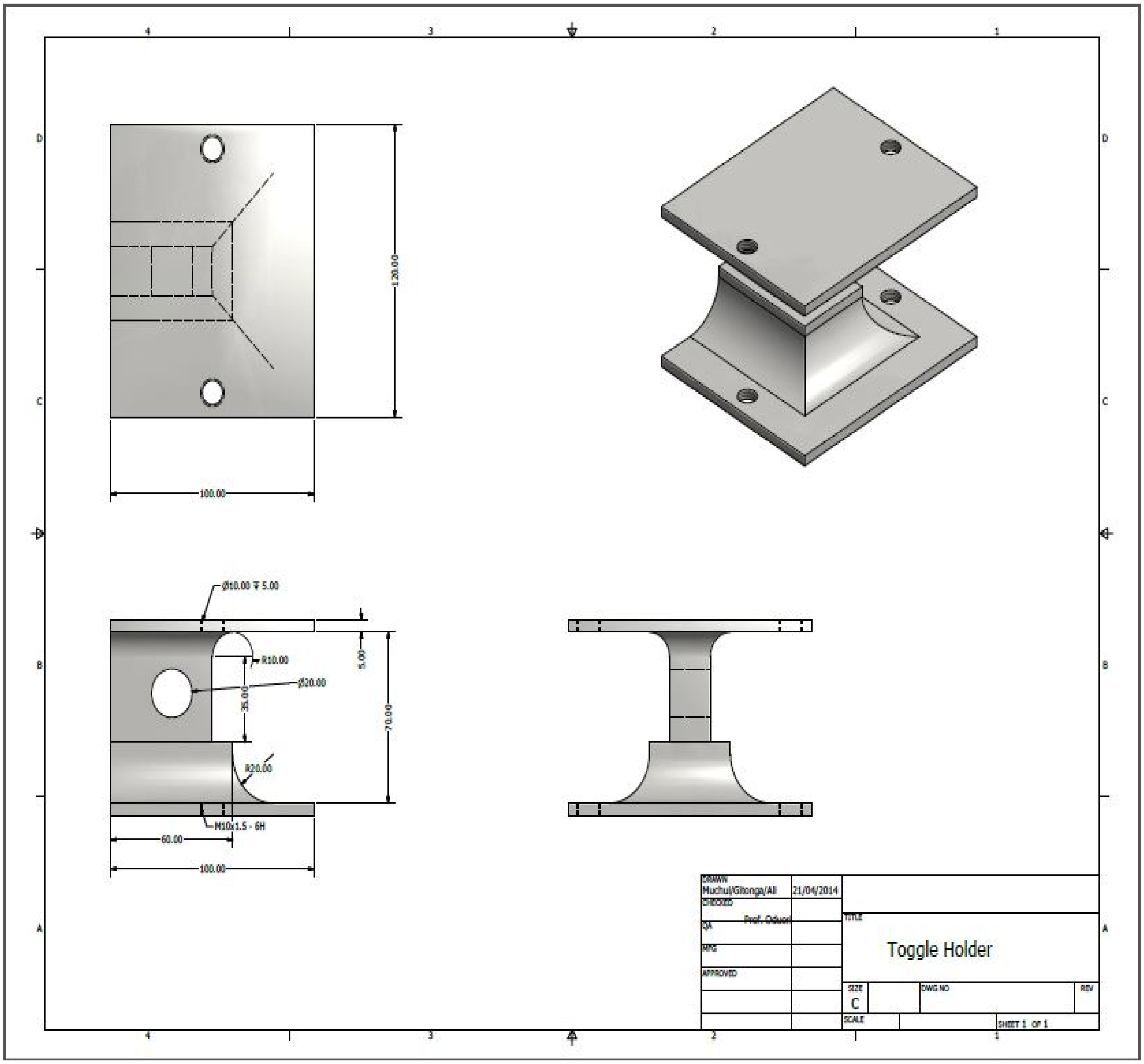
5.6 Inventor Drawing 6MAIN FRAME

5.7 MAIN FRAME SUB ASSEMBLY



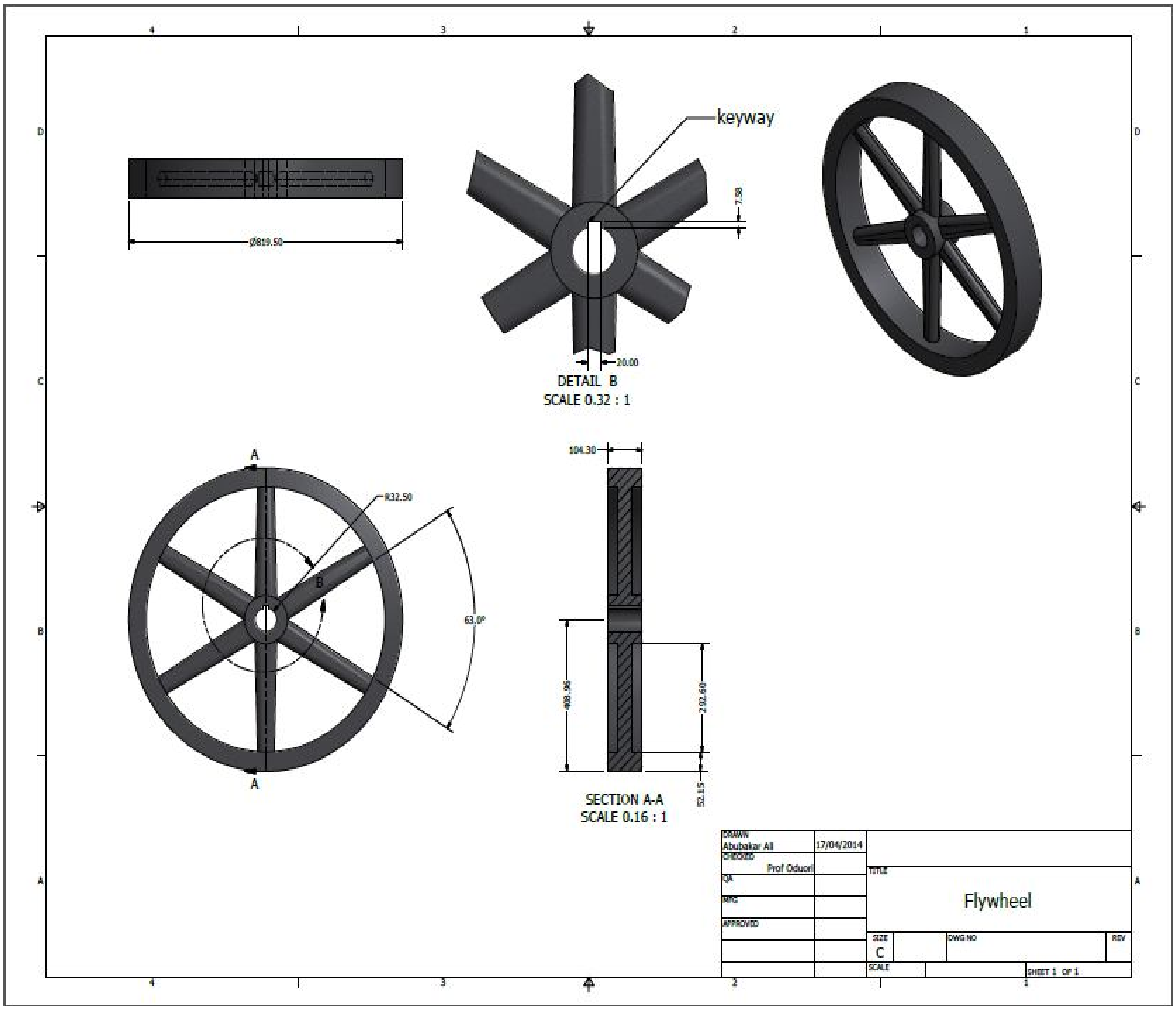
5.7 Inventor Drawing 7MAIN FRAME SUB ASSEMBLY

5.8 TOGGLE ADJUSTMENT HOLDER



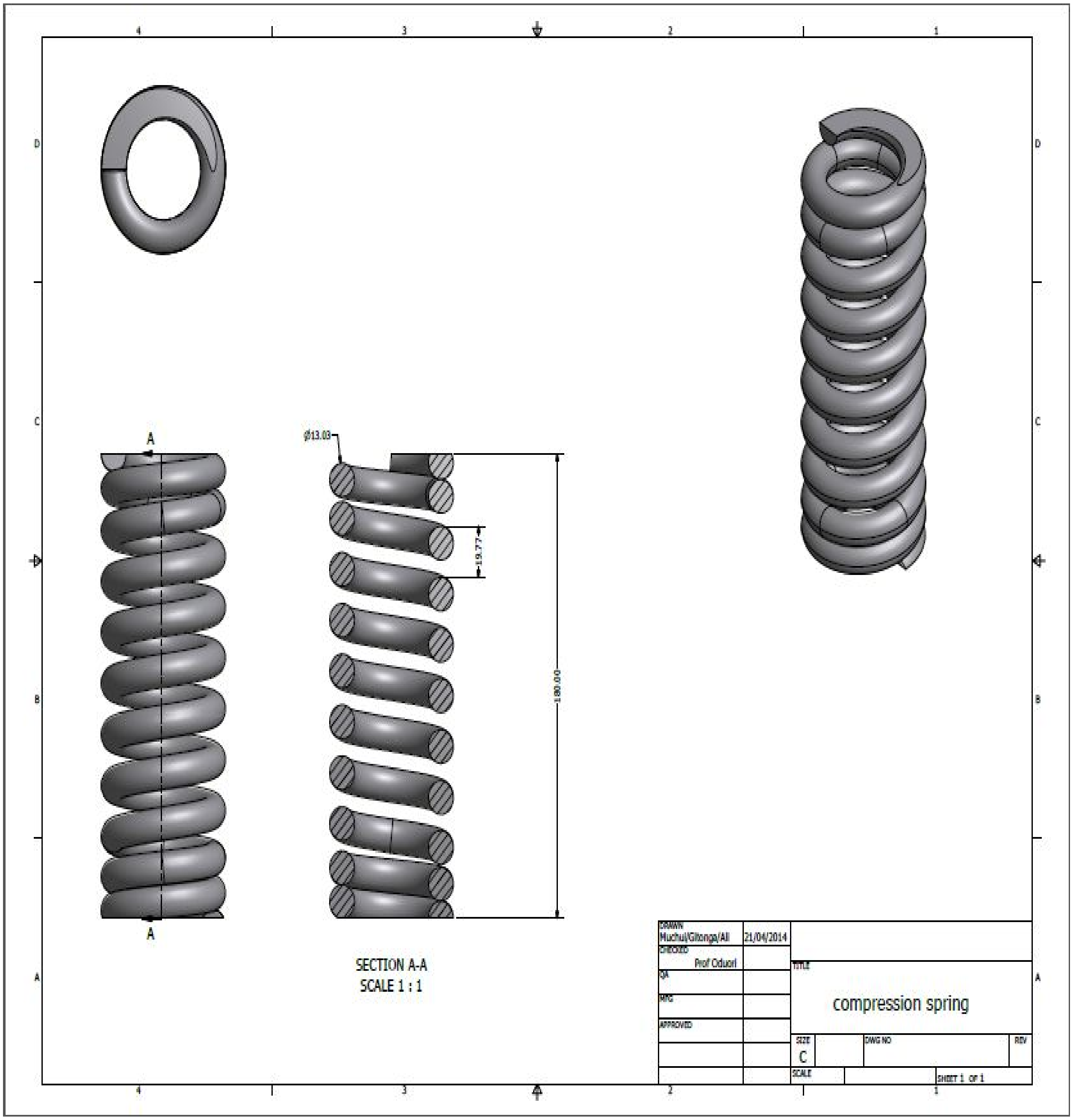
5.8 Inventor Drawing 8TOGGLE ADJUSTMENT HOLDER

5.9 FLYWHEEL



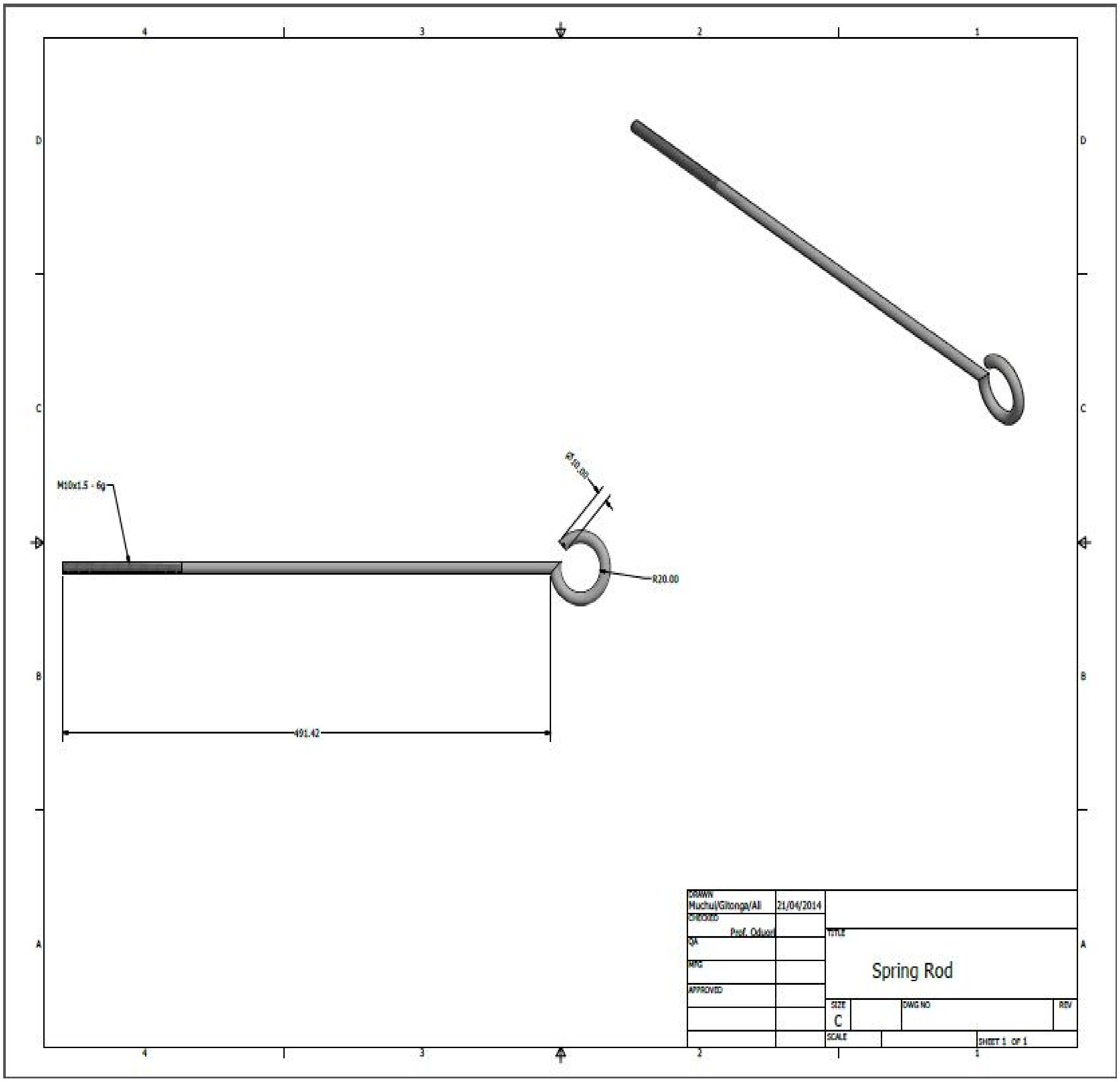
5.9 Inventor Drawing 9FLYWHEEL

5.10 SPRING



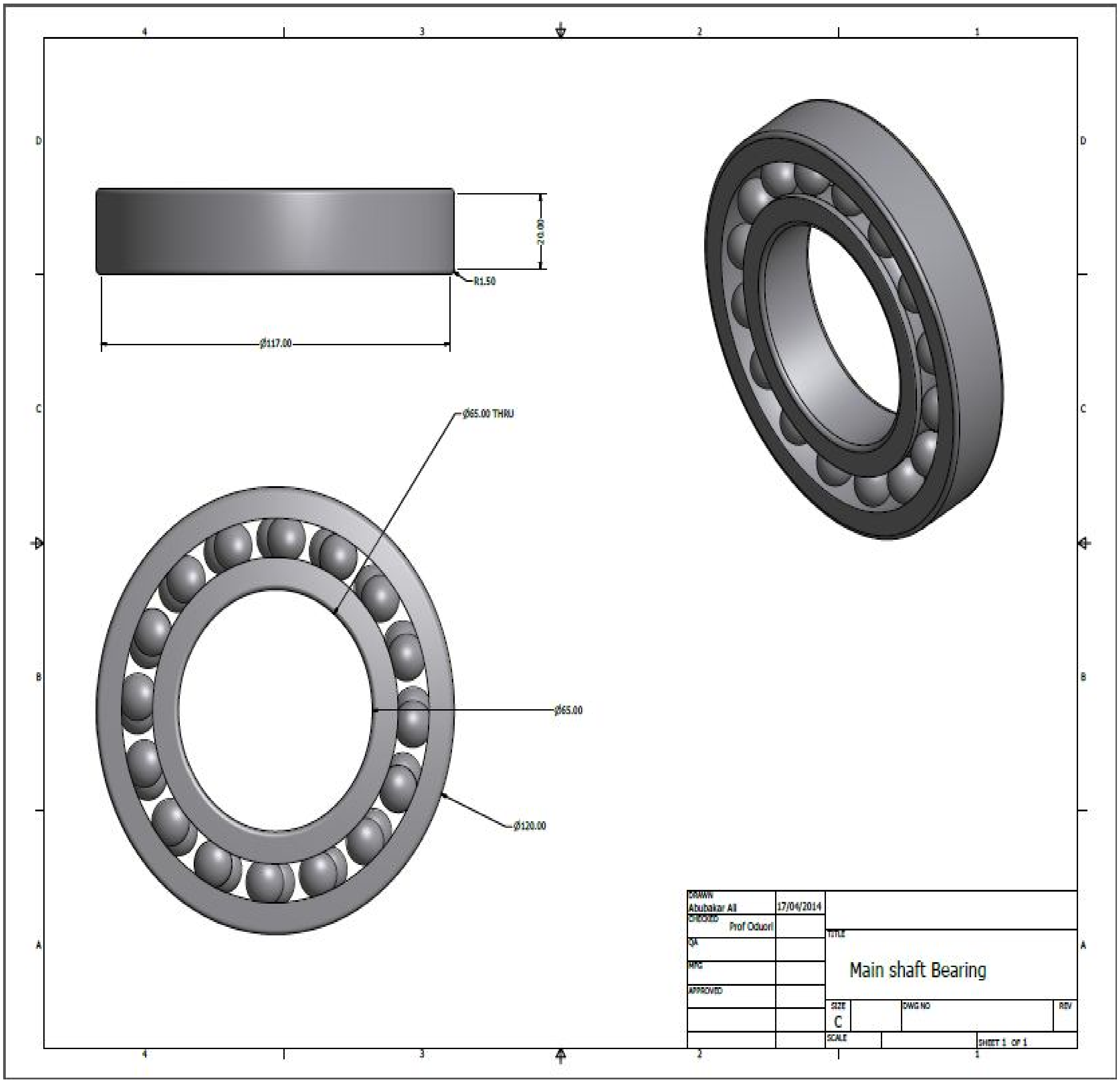
5.10Inventor Drawing 10SPRING

5.11 SPRING ROD



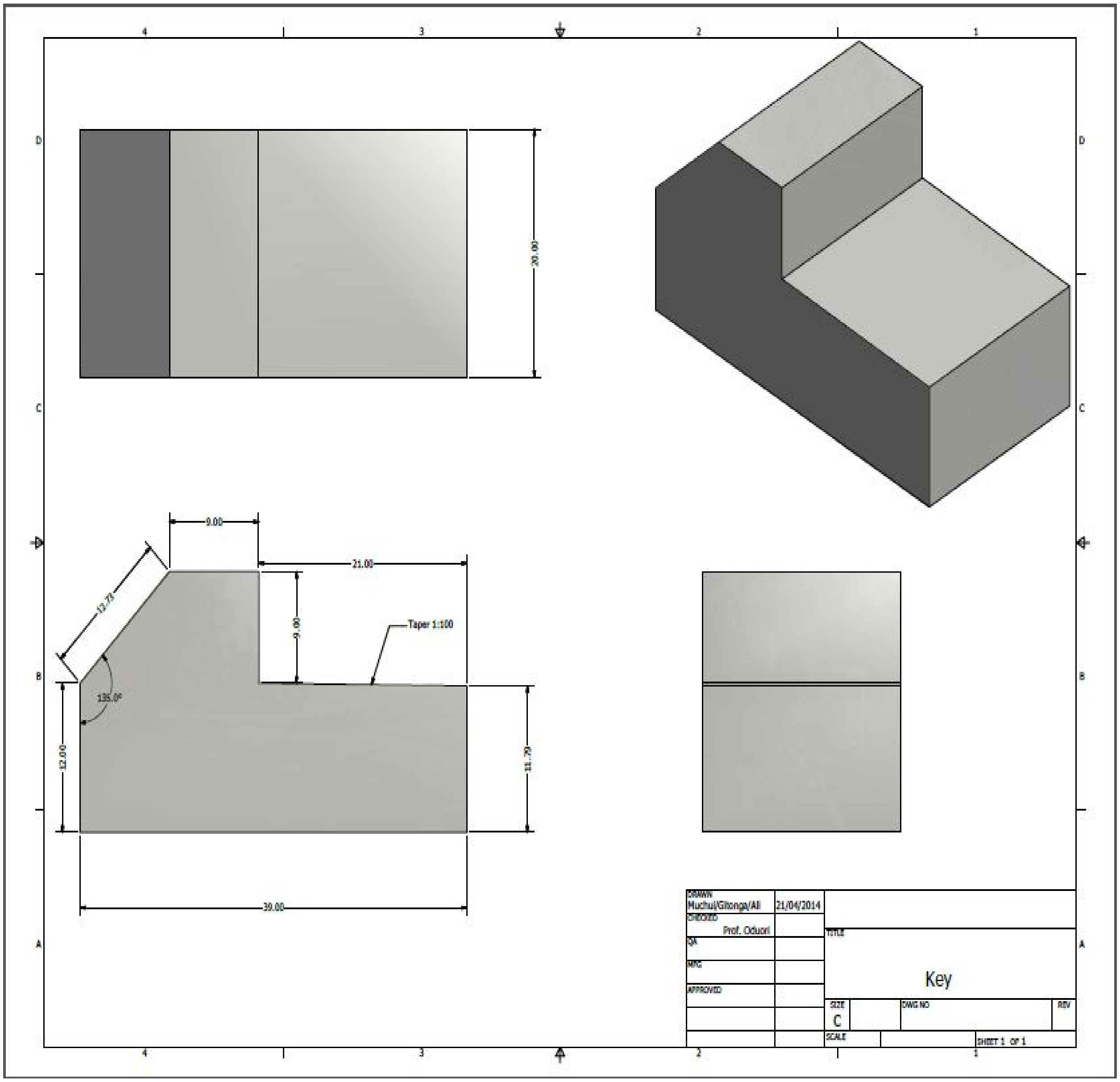
5.11 Inventor Drawing 11SPRING ROD

5.12 MAIN SHAFT BEARING



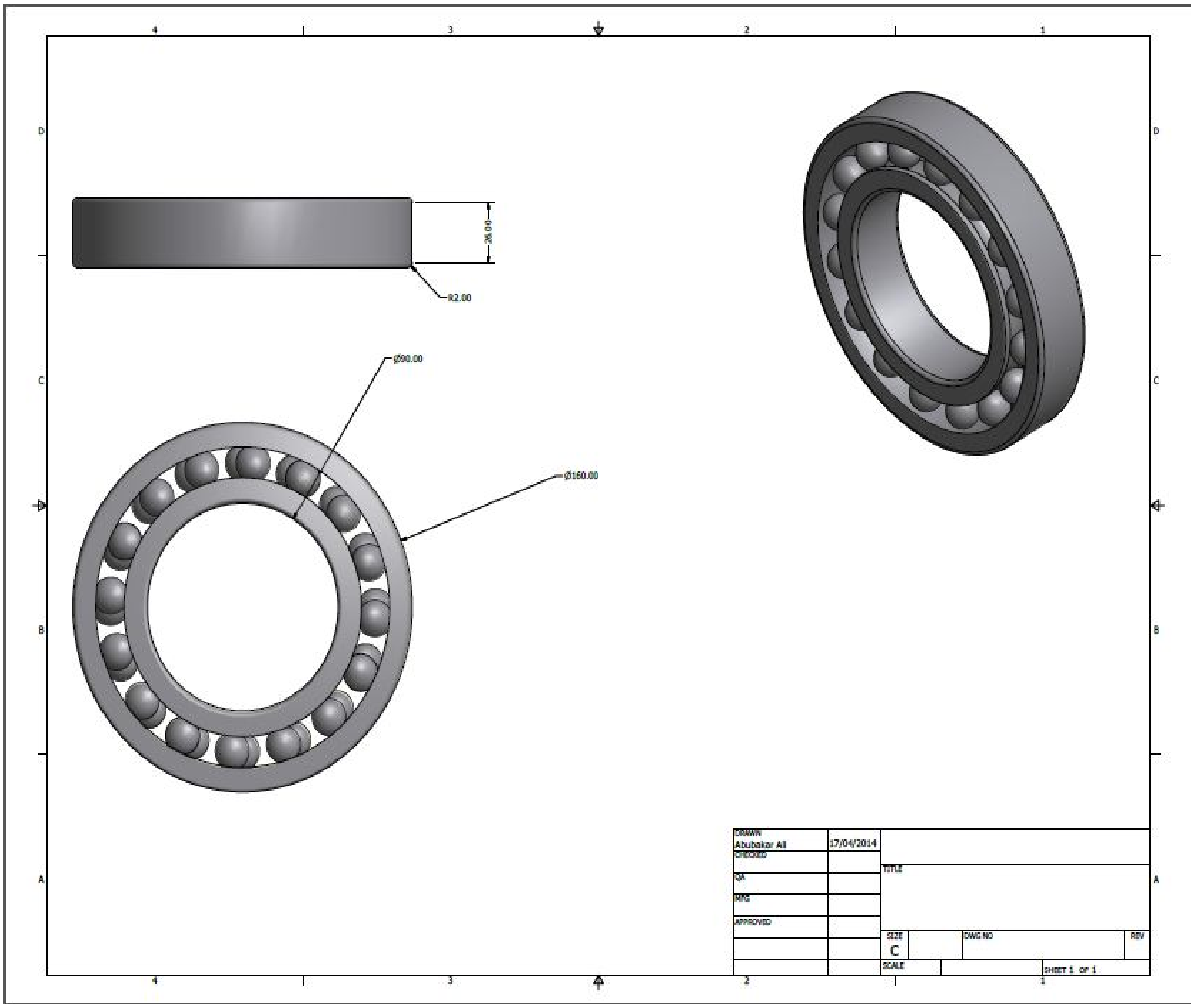
5.12 Inventor Drawing 12 MAIN SHAFT BEARING

5.13 KEY



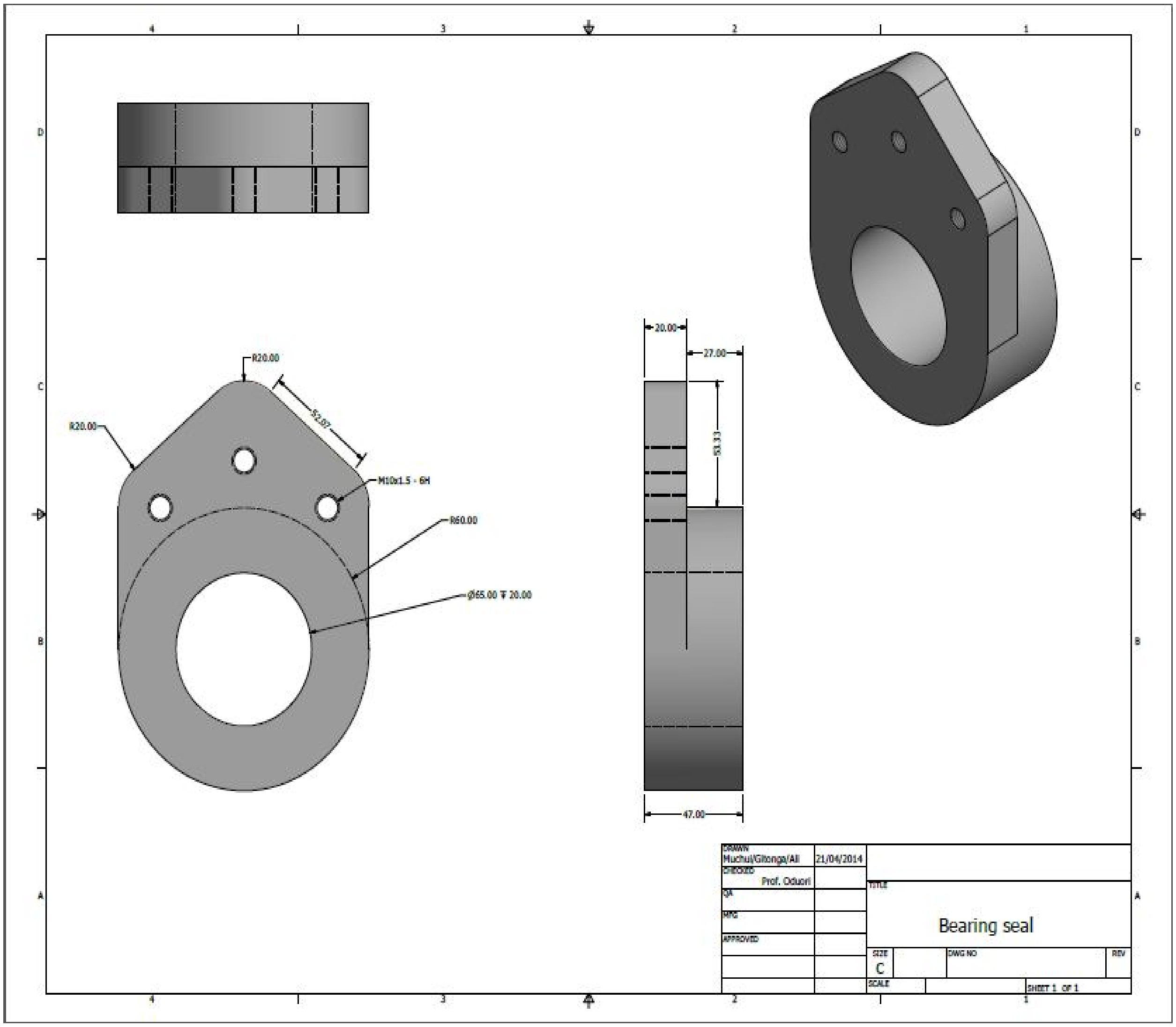
5.13 Inventor Drawing 13KEY

5.14 ECCENTRIC SHAFT BEARING



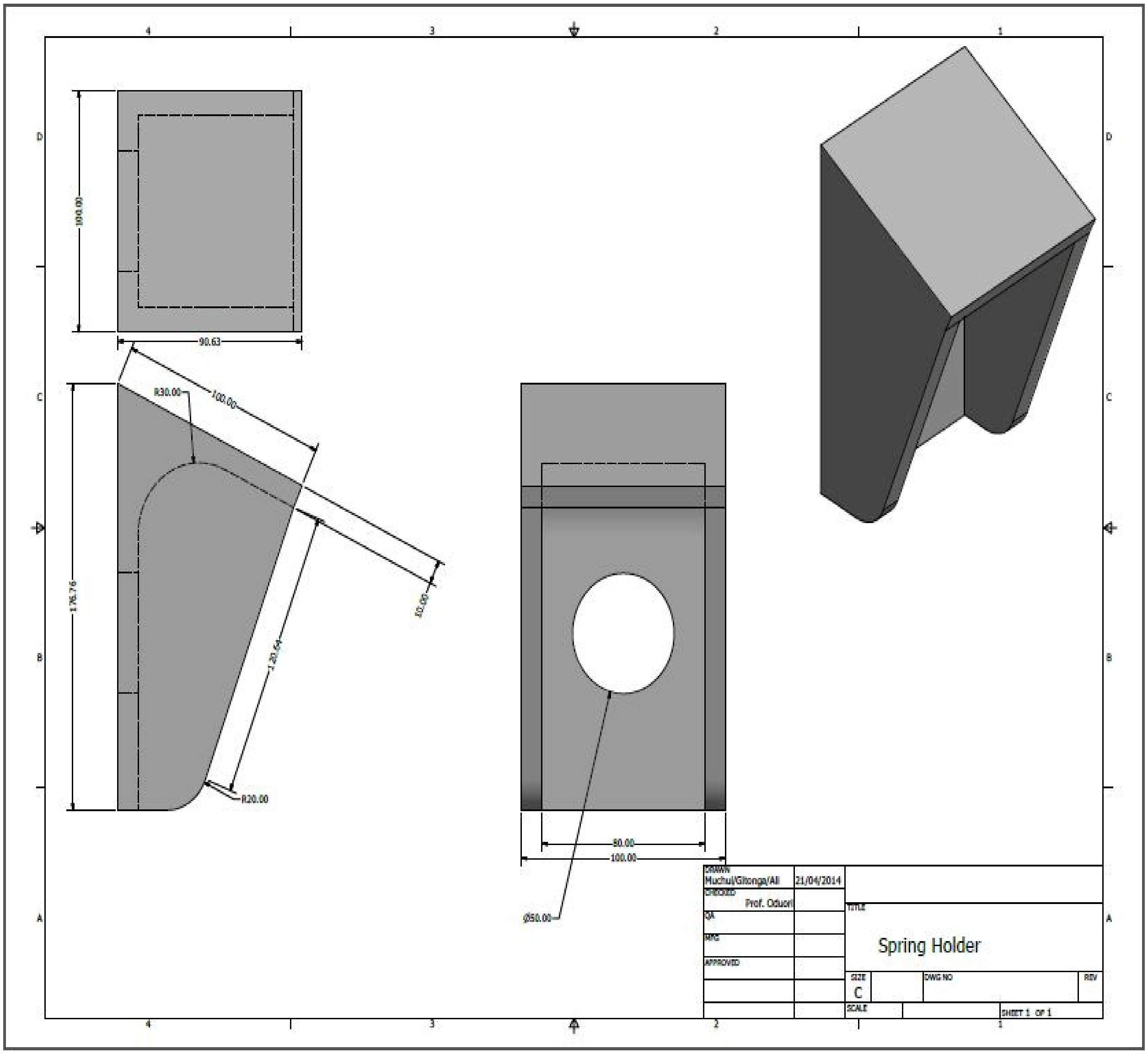
5.14 Inventor Drawing 14ECCENTRIC SHAFT BEARING

5.15 BEARING SEAL



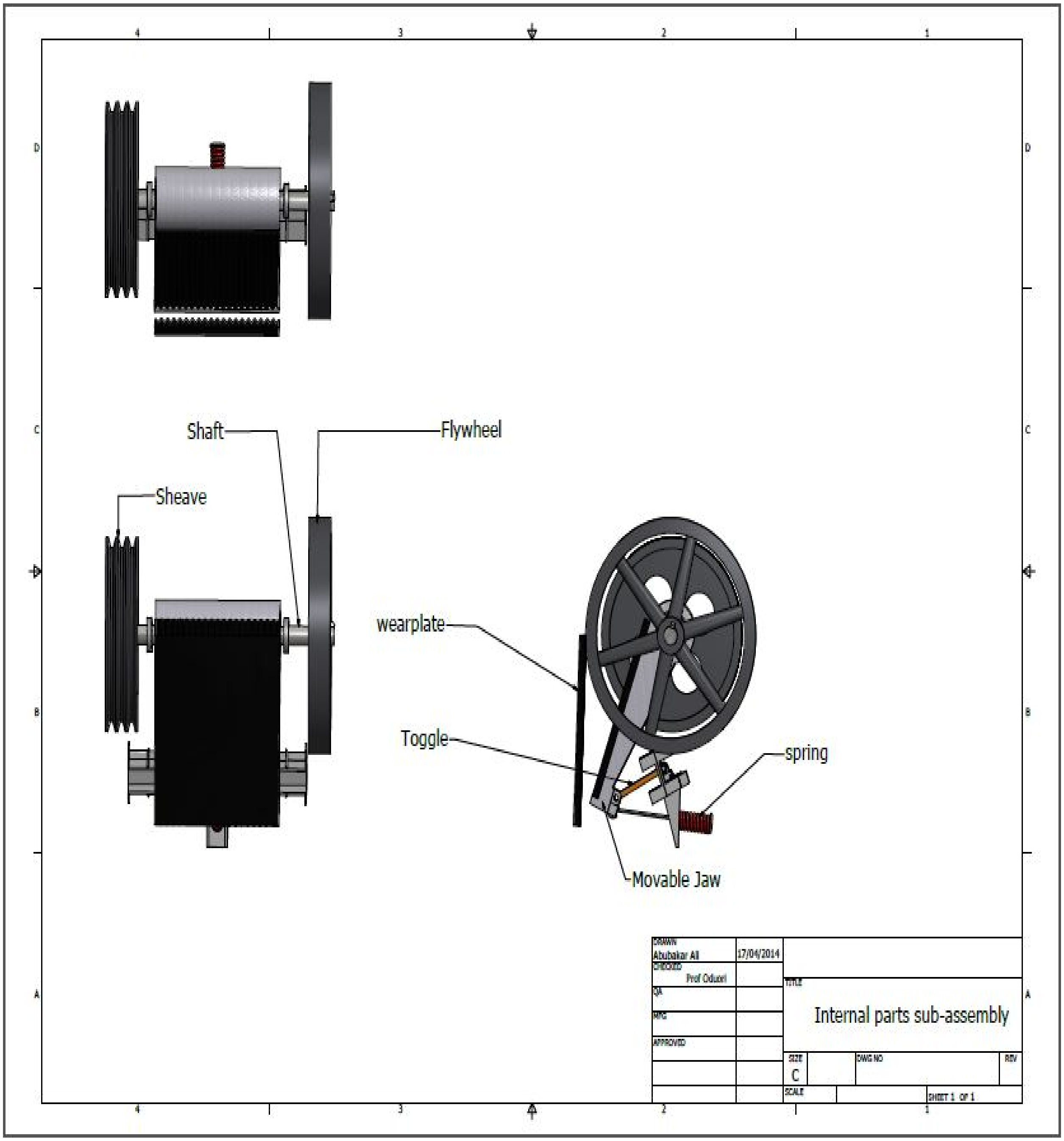
5.15 Inventor Drawing 15BEARING SEAL

5.16 SPRING HOLDER



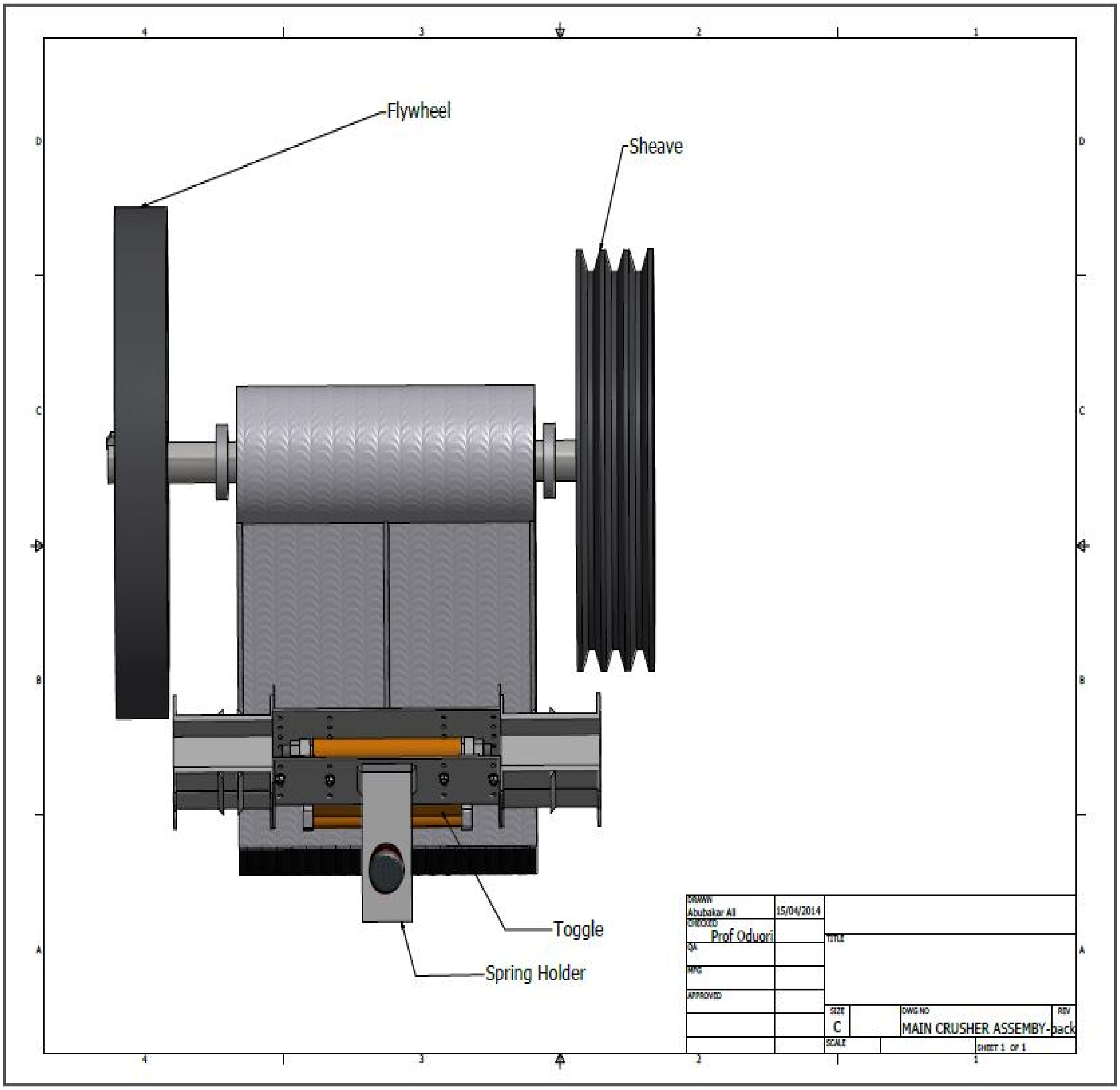
5.16 Inventor Drawing 16SPRING HOLDER

5.17 INTERNAL PARTS SUB ASSEMBLY



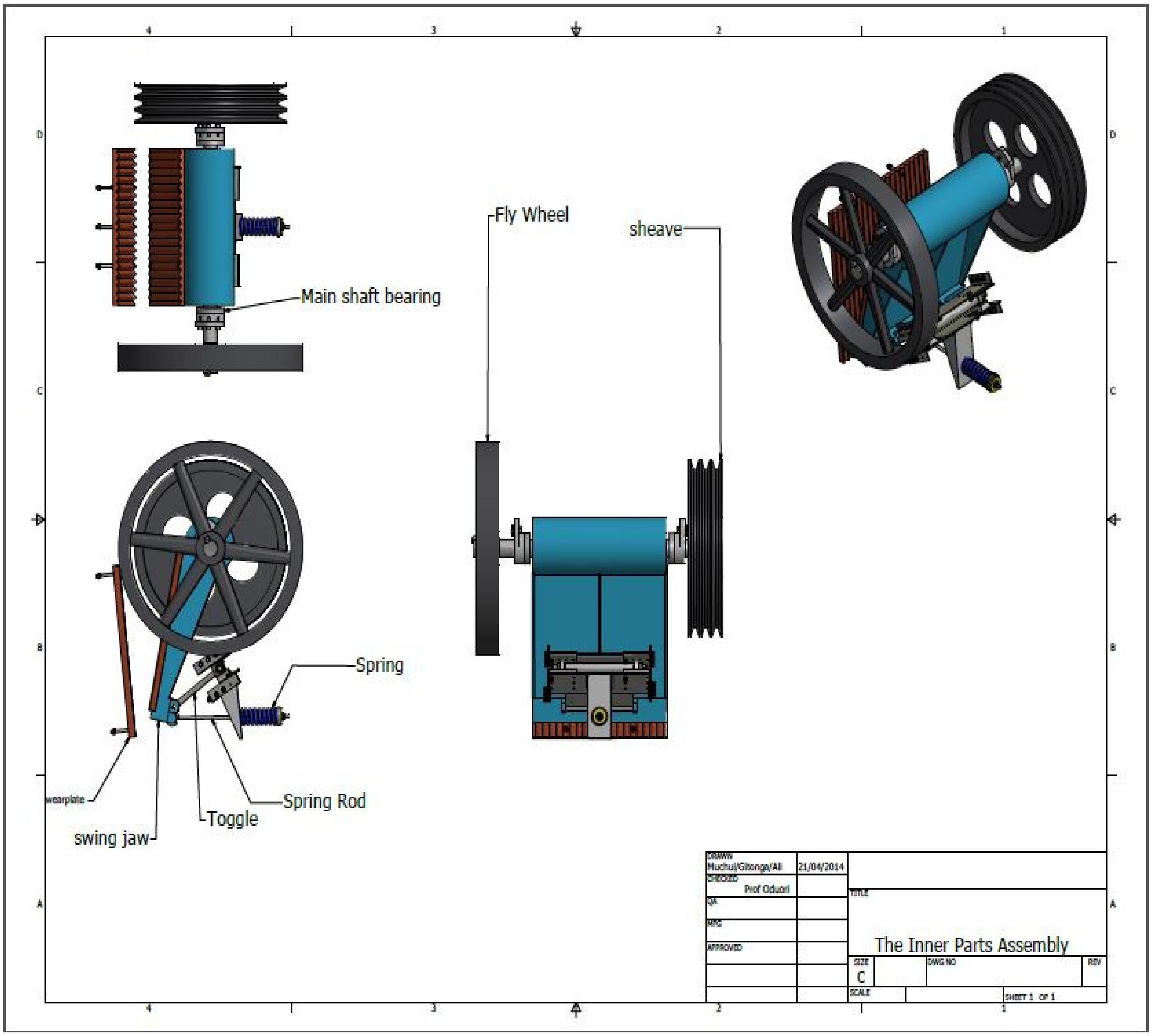
5.17 Inventor Drawing 17INTERNAL PARTS SUB ASSEMBLY

5.18 FINAL ASSEMBLY- REAR VIEW



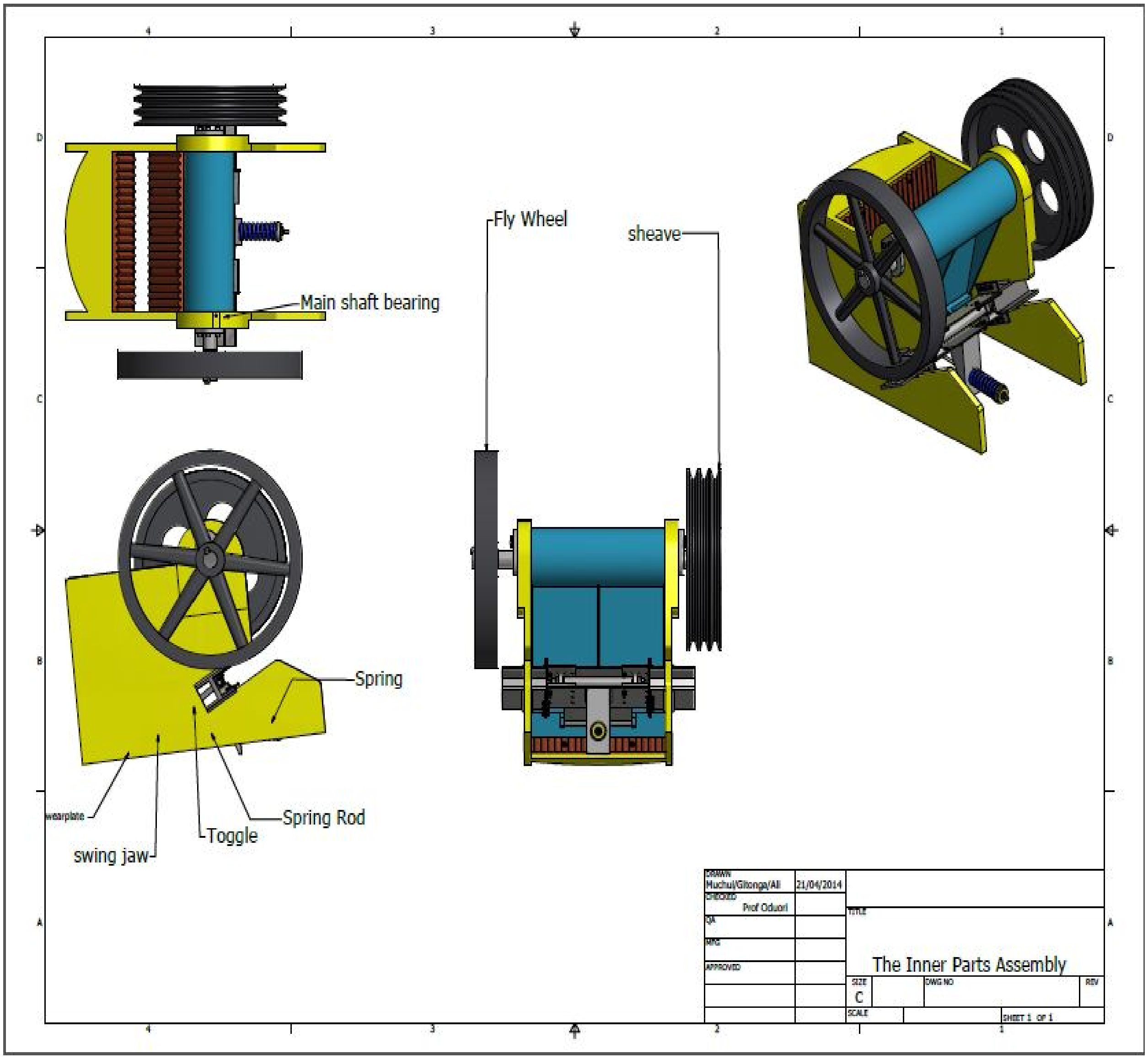
5.18 Inventor Drawing 18FINAL ASSEMBLY- REAR VIEW

5.19 INNER ASSEMBLY



5.19 Inventor Drawing 19INNER ASSEMBLY

5.20 FINAL ASSEMBLY



5.20 Inventor Drawing 20FINAL ASSEMBLY

CHAPTER SIX

6.0 DISCUSSION

The objective of this project was to design a small scale stone crusher which is affordable and economical. The analysis of the design was done by simplifying the machine as a planar crank and rocker mechanism, with the eccentric shaft being modeled as a short crank, the swing jaw of the crusher being modeled as the coupler link and the toggle link being modeled as the rocker. The kinematic analysis, which was essential in force analysis, was based on this.

An economic size model of 14tonnes /hour was chosen. For this size, the power required to crush 14 tonnes of stones every hour was determined to 11kW. This is the amount of power that should be delivered at the jaw shaft. To cater for losses due to mechanical inefficiencies in the belt drives and power loss in the bearing due to friction, the motor or diesel engine should have a slightly higher power rating, like 14 kW due to power losses, assuming an efficiency of 80%.

Most crushing sites in Kenya are not near power lines, or areas connected to the national grid. For this reason, most jaw crushers of this size operate on diesel engines. Diesel engines are cheaper to buy and operate. They have better torque characteristics and diesel is cheaper in Kenya. For this reason, a diesel engine is recommended for use with this crusher.

Power output from the diesel engine is uneven, rising to peak during the expansion stroke and falling during the expulsion and sucking stroke. To take care of these variations, the jaw crusher is designed with a powerful flywheel to absorb the excess power and give it out when the engine output is low.

The flywheel is a heavy duty casting, with a rim weight of about 88 Kg, distributed over a radius of 40.98 cm, this weight and distance was enough to absorb the power variations. The hub and the arms absorb about 8% while the rest is absorbed by the rim. Lesser power variations are, however, experienced when working with motors.

From kinematic analysis, it is evident that most crushing occurs on the upper parts of the jaw. The compressive forces gradually change to shear forces on the lower end of the jaw. For this reason, the upper part of the jaw was designed strengthen with wide web to make it stronger.

Also, the plate surface is designed in a manner to allow the replaceable manganese wear plates to be reversed in orientation when the lower side wears out. This extends the life of the wear plates and consequently reduces operation cost. The plate behaves like a wide beam, and the simple bending theory of beams was used to determine thickness of the plate. The plate carries a bending moment of 16380Nm. Using the simple bending theory, the thickness of the plate was determined to be 36.2mm, for a steel of 250 MPa yield strength, with a safety factor of 2. This thickness is enough to carry the above bending moment, while the direct stresses of 320MPa required for breaking the hardest granite rocks is borne by the extremely hard manganese wear plates.

It was established that a jaw crusher operates best at a certain optimum speed. This speed is known as the critical velocity. This critical velocity optimizes power and output. This jaw crusher was designed with critical speed of 384 R.P.M. This speed was within the operating speed range for small jaw crushers.

A small diesel engine with power rating of 14kW, running at a 2500 R.P.M. was selected in determining the radius of the belt drive pulleys. The pulley was found to have a pitch diameter 0.65 m on the larger pulley and 0.1m on the driver pulley. The pulley system is designed to run on 3 v-belts each with a cross-sectional area of$736\ mm^2$. Standards from shigleys were used in determining the dimensions of v-belts. The v-belts are 3.641 m long, each and the sheaves have a center distance of 1.2 m. The v-belts have a coefficient0.12μ, and a grove angle of 40°.

The shaft was designed to carry a load of up to 8731N this was including the weights of the flywheel, pulley and the weight of stones during operation. Impact loads were also taken into consideration when determining the maximum load on the shaft. The shaft was determined to be 65mm in diameter, though the nearest size dimension can be taken safely.

Self-aligning ball bearings were chosen for this shaft. The size and material was generated from the inventor material library, and tested for these loading. A simulation for the operation of the bearings showed that it would be able to withstand the required loading and speed. The bearing was designed for $48.4 * 10^6$revolutions in its entire lifetime.

The above key parameters for the jaw were arrived at after though consideration and a computer analysis report generated for each. Inventor drawings for each component were drawn.

6.1 CONCLUSION

Kinematic and force analysis was applied in determining the dimensions of various elements of the jaw crusher. The machine was designed with few components and hence, easy to maintain and cheaper in fabrication.

The machine was designed to give an output of 14tonnes/hr. hence met its design objective.

6.2 RECOMMENDATIONS

- Thorough tests should be undertaken to determine how forces are distributed over the plate length. This will enable more accurate design of the jaw, which should be a constant strength jaw designed to have a thickness varying from where the forces are highest near the gape, to where they are least, near the set.
- Also material classification should be more precise, by studying the material of each component to ensure the strongest and most reliable material is chosen for each part.
- Research should carry out on the cost of producing each element and its material availability.
- Further analysis should be carried out for the support frame.

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