CHAPTER 1

1.0 INTRODUCTION

Rock is an abundantly available natural resource that is used for aggregate production. The demand for aggregate has been on the rise recently due to the increased activity in the building and construction industry. The crushing of stones has been characterized by high input undertakings in terms of capital investment usually done by the construction companies themselves which incur high transport costs since they use heavy and centralized stone crushing machines.

With ever increasing demand, people have resorted to extracting this resource for revenue earning. For a long time now the local people have used crude ways of reducing large rocks into the required size of aggregates. Due to the kind of tools they use, their output cannot meet the daily market demand for this commodity. This undertaking is also hazardous to their health since no protective devices are used.

Faced with the challenge of bridging this gap, the idea of a small scale mechanized stone crushing machine was conceptualized. The aim is to design a machine that can increase the crushing power of the ‘hammer and anvil’ stone crusher without pricing itself out of the range for which such workers can afford. The machine should also increase the output volume of crushed stone in order to match the increased demand. The fact that such a machine would be available to small scale stone crushers saves construction companies transport costs that they would have otherwise incurred while doing the large scale centralized crushing.

At this point, it is important to note that there are existing models of stone crushers, so the aim here is not to reinvent the wheel, but to develop a machine that is suitable to our particular situation as outlined above.
1.1 STATEMENT OF THE PROBLEM

Due to the fact that Kenya is a developing country, most of the people involved in the stone crushing industry are not able to acquire machines which can produce aggregates in large scale. Hence, this has been left to foreign companies which have the financial capability to purchase such equipment. This production is characterized by a high degree of automation, high crushing rate and large output. The finished products have even granularity, excellent shape and reasonable grain distribution. Furthermore, various sizes of aggregates are produced.

Contrary observations were noted in a visit to Makongeni in Thika town where people crush stones using the hammer and anvil. The rate of production was slow due to the kind of tools they use. Actually, a hardworking individual could produce up to seven wheelbarrows of aggregate a day (equivalent to 0.408 tonnes of aggregates daily) which could earn him only Kshs.200 on average. A truck carrying 7 tonnes of manually crushed aggregates costs Ksh. 3500. Thus, it takes approximately 20 days to crush seven tonnes of aggregate. Crushing is done on an open area, exposing people to the harsh sunshine, rainfall and extreme coldness. In addition, there is hazard of injury from hammer blows to the fingers, injury from flying fragments of stone and ailment from inhalation of stone dust.

It was therefore noted that there is need to develop a small scale machine that could increase the rate of aggregate production in order to meet the high market demand while improving the living standards of the people.
**Fig 1.1** and **Fig 1.2** shows the disparity between large scale and manual stone crushing.

**Fig 1.1**: large scale aggregate production, Courtesy of zenith mining and Construction Company

**Fig 1.2**: A woman manually crushing stones at Makongeni near Thika town
1.2 OBJECTIVES

The project seeks to design a small scale mechanized stone crushing machine that is simple, economical, easy to use and maintain in order to empower the manual stone crushers while meeting the increasing demand for aggregate in the Kenyan building and construction industry.

The aim is:

- To carry out the kinematic analysis of the proposed design in order to determine the motions of the machine parts.
- To carry out the dynamic analysis based on crushing loads obtained in the previous analysis of the laboratory jaw crusher.
1.3 PROJECT JUSTIFICATION

The development of a small scale stone crusher is the actual solution to the disparity between the large scale stone crushing and the manual stone crushing since it will reduce the human effort requirements and increase output.

It is more economical to crush stones within a small radius between the crushing point and the construction site where it is going to be used. This saves the construction companies transport costs which they could have incurred.

There is widespread demand for sized stone as a result of the increasing technological advancement and need for better infrastructure. The implementation of the design in the target societies in Kenya will contribute towards alleviation of the Rural-Urban migration since the stone crushing activity is an income generating activity.
1.4 FIELD STUDY

The field study was carried out in form of question-answer sessions with the respondents

- What is the output per day?
- What is the income per day?
- What are the advantages of this method of stone crushing?
- What are the challenges of this method of crushing stone?
- What is the cost of a stone crushing machine?
- What is the time period of the return in investments for such a machine?
- What is the market demand of aggregate in Kenya today?
CHAPTER 2

2.0 LITERATURE REVIEW

Different types of crushers exist depending on their **design** and **crushing mechanism**. These include:

i. Jaw crushers,
ii. Gyratory crushers.
iii. Cone crushers.
iv. Impact crushers.

A crusher may be considered as **primary** or **secondary** depending on the size reduction factor. Jaw crushers and gyratory crushers are primary crushers while cone crushers and impact crushers are secondary crushers. Primary crushers reduce large mine stones into smaller sizes suitable for transport. They are large and heavy machines. On the other hand, secondary crushers have a feed size usually less than 15cm and reduce the stones to the required sizes.

The different crushers are shown below:

![Gyratory crusher](image1)

**Fig 2.1: Gyratory crusher**

![Jaw crusher](image2)

**Fig 2.2: Jaw crusher**

---

*Source 1, 2: www.metsominerals.com*
Fig 2.3: Cone crusher

Fig 2.4: Impact crusher

Source 3, 4: http://www.zenithcrusher.com/
The choice between the jaw crusher and gyratory crusher is dictated by the largest feed size, production requirements and the economics of operation. Though gyratory crushers have a simple feed mechanism, can handle slab like material and have a compact design, jaw crushers are preferred because of their easy maintainability, their ability to handle dirty and sticky feed and their ability to handle large block shaped feed.

**Table 2.1: Commonly used crushers and their applications**

<table>
<thead>
<tr>
<th>Type</th>
<th>Hardness</th>
<th>Abrasion limit</th>
<th>Moisture content</th>
<th>Reduction ratio</th>
<th>Main use</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jaw crusher</td>
<td>Soft to very</td>
<td>No limit</td>
<td>Dry to slightly sticky</td>
<td>3/1 to 5/1</td>
<td>Quarried materials, sand gravel</td>
</tr>
<tr>
<td></td>
<td>hard</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gyratory</td>
<td>Soft to very</td>
<td>Abrasive</td>
<td>Dry to slightly wet not</td>
<td>4/1 to 7/1</td>
<td>Quarried materials</td>
</tr>
<tr>
<td>crusher</td>
<td>hard</td>
<td></td>
<td>stick</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cone</td>
<td>Medium hard to</td>
<td>Abrasive</td>
<td>Dry or wet, not sticky</td>
<td>3/1 to 5/1</td>
<td>Sand &amp; gravel</td>
</tr>
<tr>
<td>crushers</td>
<td>very hard</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Horizontal</td>
<td>Soft to medium</td>
<td>Slightly</td>
<td>Dry or wet, not sticky</td>
<td>10/1 to 25/1</td>
<td>Quarried materials, sand &amp; gravel</td>
</tr>
<tr>
<td>shaft</td>
<td>hard</td>
<td>abrasive</td>
<td></td>
<td></td>
<td>recycling</td>
</tr>
<tr>
<td>impactors</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>impactors (shoe</td>
<td>Medium hard to</td>
<td>Slightly</td>
<td>Dry or wet, not sticky</td>
<td>6/1 to 8/1</td>
<td>Sand &amp; gravel, recycling</td>
</tr>
<tr>
<td>and anvil)</td>
<td>very hard</td>
<td>abrasive</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Source 5: www.wikipedia.com*
2.1 THE JAW CRUSHER

Jaw Crusher is one of the main types of primary crushers used in stone quarrying or mine ore crushing processes. It has a flexible capacity, high reliability, simple structure and easy maintainability. Moreover, jaw crushers also have stable performance, low operating cost, high crushing ratio, high resistance to friction, abrasion and compression with longer operating life span.

A primary jaw crusher is typically of the square opening design whereas a secondary jaw crusher is of the rectangular opening design. Jaw Crusher reduces large size rocks or ore by placing the rock into compression. A fixed jaw board is the stationary breaking surface, while the movable jaw exerts force on the rock by forcing it against the stationary plate. Other parts include the frame, eccentric shaft, belt pulley, flywheel, side guard plate, toggle plate and the return spring.

2.1.1 WORKING PRINCIPLE OF A JAW CRUSHER

The motor drives the belt pulley, which drives the flywheel and the flywheel in turn, drives the eccentric shaft to rotate and make the moving jaw approach and leave the fixed jaw periodically. The movable jaw exerts force on the rock by forcing it against the stationary plate. The opening at the bottom of the “V” aligned jaws is the crusher set while the maximum opening at the top is the gape. When the jaws come together the rock is crushed into smaller sizes and slips down the cavity. In the return stroke, further reduction of size is effected and the rock moves down further. The process is repeated till particles having size less than the set pass through as product. The function of the toggle(s) is to move the pivoted jaw.
2.1.2 TYPES OF JAW CRUSHERS

Typically there are two types of jaw crushers.

i. Single toggle type – This has only one toggle plate. It is less weight and thus cheap.

ii. Double toggle type – This has two toggle plates and can crush materials that are brittle, tough and abrasive.

Fig 2.5: working principle of a jaw crusher

Fig 2.6: Single toggle jaw crusher  

Fig 2.7: Double toggle jaw crusher
Depending on the motion of the moving jaw, the double toggle jaw crusher can either be:

i. Blake type
ii. Dodge type

**Blake jaw crusher**

In this crusher, the movable jaw is hinged at the top of the crusher frame so that the maximum amplitude is obtained at the bottom of the crushing jaws. Blake Crushers are operated by toggles and controlled by an eccentric shaft. These are commonly used as primary crushers in the mineral industry.

**Dodge jaw crusher**

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

*Fig 2.8: Dodge type jaw crusher*
2.2 CONCEPT OF JAW CRUSHER DESIGN FOR THE PROJECT

The factors considered while coming up with the design of the machine are:

i. Cost of manufacture and maintenance
ii. Required output per unit time
iii. Required reduction ratio
iv. Crushing force requirement
v. The operation mechanism

Limitations of the existing jaw crushers
The moving jaw of the single toggle crusher has an elliptical motion. The implication is fast wear of the jaw. Through advancements in materials science the problem has been solved by introduction of wear plates made of manganese steels which are replaceable. However, replacements mean extra expenses. On the other hand, the moving jaw of the double toggle jaw crusher has a swing motion. Although the wear on the plates is slow, the mechanism has many parts which make it expensive.

The proposed mechanism seeks to address the challenges outlined above.

Fig 2.9: Diagrammatic representation of the proposed mechanism
The proposed design has a Blake mechanism. The Blake mechanism was chosen bearing in mind the need to crush large, hard and sticky rocks without the problem of clogging.

The proposed design has fewer parts because the connecting rod is attached directly to the moving jaw without the use of a toggle mechanism. This lowers the cost of manufacture. In addition to that, the jaw crusher is comparatively lighter which increases its portability.

The above decisions were in line with the objectives of the project which are to design a small scale mechanized stone crushing machine that is simple, easy to maintain and able to increase the stone crushing power of the “hammer and anvil” manual stone crusher without pricing itself out of the range for which they can afford.
CHAPTER 3
3.0 KINEMATIC ANALYSIS OF THE JAW CRUSHER.

The study of a Machine can be done in three steps. These are:

i. Consideration from a geometrical point of view of the motion of any part of the machine with reference to any other part, without taking account of any forces acting on such parts. This is the kinematic analysis.

ii. Consideration of the action of forces impressed on the parts of the machine, and of the forces due to its own inertia or to the weight of its parts and the resulting transformations of energy. This is the kinetic or dynamic analysis.

iii. A third approach to the theory of machines takes into account the action of loads and forces on the machine in producing stresses and strains in the materials employed in the construction of the machine, and discusses the sizes, forms, and proportions of the various parts which are required either to ensure proper strength while avoiding waste of material, or to make the machine capable of doing the work for which it is being designed. This is the strength analysis.

In this section, therefore, the attempt is to do the kinematic analysis of the proposed design of the jaw crusher.

The kinematic diagram of the proposed mechanism is as shown below:

**Fig 3.4: Kinematic diagram of the proposed mechanism**

Small scale mechanized stone crusher 2010/2011
Because the eccentricity is very small compared to the length of the swing jaw, the motion of the mechanism can be approximated to that of the crank and slider mechanism. In order to analyze the motion of the swing jaw, it can be represented by a point B.

The displacement at point B can be described by an equation derived using the analytical method. From this displacement equation, the velocities and accelerations can be derived.

From the diagram, the displacement $x_\beta$ is given by:

$$x_\beta = l \cos \phi + r \cos \beta$$  \hspace{1cm} (3.10)

The following relationship also holds

$$r \sin \beta + e = l \sin \phi$$

From which:

$$\sin \phi = \frac{1}{l} (r \sin \beta + e)$$

And hence

$$\sin^2 \phi = \frac{1}{l^2} (r \sin \beta + e)^2$$  \hspace{1cm} (3.11)

From the trigonometric identity

$$\sin^2 \phi + \cos^2 \phi = 1$$

$$\cos \phi = \sqrt{1 - \sin^2 \phi}$$  \hspace{1cm} (3.12)

Substituting equation (3.11) into equation (3.12) we have

$$\cos \phi = \sqrt{1 - \frac{1}{l^2} (r \sin \beta + e)^2}$$
Hence equation (3.10) can be rewritten as:

\[ x_\beta = r \cos \beta + \sqrt{l^2 - (r \sin \beta + e)^2} \]  

(3.13)

\( x_{\text{max}} \) is the displacement from the nearest approach to the stationary jaw. It occurs when O, A and B are in line. This is illustrated in the kinematic diagram below:

**Fig 3.5: Kinematic diagram showing maximum displacement of point B**

\( x \) is the displacement of point B from its nearest approach to the stationary jaw. It is determined as:

\[ x = x_{\text{max}} - x_\beta \]  

(3.14)

From equation (3.10) since \( \phi = (360^\circ - \beta) \), we get :

\[ x_{\text{max}} = l \cos(360^\circ - \beta) + r \cos \beta \]

\[ = (l + r) \cos \beta \]  

(3.15)

Again,

\[ \sin \phi = \frac{e}{l+r} \]

From which,
\[-\sin \beta = \frac{e}{l+r}\]

But,

\[\cos \beta = \sqrt{1 - \sin^2 \beta}\]

Therefore,

\[\cos \beta = \sqrt{1 - \left(\frac{e}{l+r}\right)^2}\]

Substituting into equation (3.15)

\[x_{\text{max}} = (l + r) \sqrt{1 - \left(\frac{e}{l+r}\right)^2}\]

\[= \sqrt{(l + r)^2 - e^2}\] (3.16)

Therefore, the displacement of point B from its nearest approach to the stationary jaw is given by:

\[x = \sqrt{(l + r)^2 - e^2} - r\cos \beta - \sqrt{l^2 - (r \sin \beta + e)^2}\] (3.17)
A graph of the displacement against crank angle is plotted below and it describes the motion of point B.

**Graph 3.1: Graph of displacement against crank angle**
Critical points are stationary points at which the machine changes from forward stroke to backward stroke or the reverse. The stationary points occur at $A_1$ and $A_2$.

At these points,

$$\sin \varphi = \frac{e}{r+l}$$

Substituting for,

$$e = 0.05$$

$$l = 0.45$$

$$r = 0.0025$$

We get:

$$\sin \varphi = \frac{0.05}{0.45+0.0025}$$

$$\sin \varphi = 0.1105$$

Therefore,

$$\varphi = \sin^{-1}[0.1105]$$

$$= 6.3^\circ$$

$$\beta_1 = 360 - \varphi = 360 - 6.3 = 353.7^\circ$$

$$\beta_2 = 180 - \varphi = 180 - 6.3 = 173.7^\circ$$

Angle swept on forward stroke $= \beta_1 - \beta_2 = 180^\circ$

Angle swept on return stroke $= 360 - 180 = 180^\circ$

The ratio of the time of the forward stroke to that of the return stroke is

$$K = \frac{180}{180} = 1$$
3.1 VELOCITY

The displacement equation (3.14) is differentiated to get the velocity.

\[ \frac{dx}{d\beta} = \frac{d}{d\beta} \left[ \sqrt{(l + r)^2 - e^2} - r\cos\beta - \sqrt{l^2 - (r \sin\beta + e)^2} \right] \]

Linear velocity,

\[ v = \frac{dx}{dt} \]

The displacement \( x \) is a function of crank radius \( r \), crank angle \( \beta \), the offset \( e \), and the connecting rod length \( l \).

There exists a relationship between the linear velocity of point B and the angular velocity of the crank.

Angular velocity,

\[ \omega = \frac{d\beta}{dt} \]

From which,

\[ dt = \frac{d\beta}{\omega} \]

Substituting for \( dt \) we get, linear velocity of point B,

\[ v = \omega \left( \frac{dx}{d\beta} \right) \]

Therefore by differentiating equation (3.14) with respect to the crank angle and multiplying it with the angular velocity of the crank \( \omega \), which is constant, we get the linear velocity of point B.

Thus,

\[ v = \omega \left[ r\sin\beta + \frac{r\cos\beta(r\sin\beta + e)}{\sqrt{l^2 - (r \sin\beta + e)^2}} \right] \quad (3.18) \]
Graph 3.4: Graph of velocity against crank angle

3.2 ACCELERATION

The velocity equation is further differentiated to yield the acceleration equation.

\[
a = \omega \frac{dv}{d\beta} = \omega \frac{d}{d\beta} \left[ r \sin \beta + \frac{r \cos \beta (r \sin \beta + e)}{\sqrt{l^2 - (r \sin \beta + e)^2}} \right]
\]
Thus,

\[ a = \omega^2 \left[ r \cos \beta + \frac{r^2 \cos \beta - \varepsilon \sin \beta}{\sqrt{l^2 - (r \sin \beta + \varepsilon)^2}} + \frac{\frac{r^2}{2} \sin 2\beta + \varepsilon \cos \beta [r \cos (r \sin \beta + \varepsilon)]}{\sqrt{l^2 - (r \sin \beta + \varepsilon)^2}} \right] \]  \hspace{1cm} (3.19)

**Graph 3.5:** Graph of acceleration against crank angle

![Graph of acceleration against crank angle](image-url)
CHAPTER 4

4.0 DYNAMIC ANALYSIS OF THE JAW CRUSHER

In this section we consider the action of forces impressed on the parts of the machine as a result of the external forces arising from the loading of the machine during its operation, and forces from the driving mechanism of the machine.

In this analysis the weight of the individual parts of the machine is assumed to be negligible as compared to the external forces. The frictional forces are ignored.

4.1 TORQUE AND FORCE ANALYSIS

The action of external forces is represented as shown below

![Diagram of torque and force analysis](image)

Fig 4.1: Diagram of torque and force analysis

\( Q \) – Crushing force
\( F \) – Vertical component of the force in the connecting rod
The balance of forces is:

\[ T = F \cos \beta + Q \sin \beta \]  \hspace{1cm} (4.10)

\[ F = Ps \sin \phi \]

\[ Q = P \cos \phi \]

Thus,

\[ T = Pr \cos \phi \cos \beta + Pr \cos \phi \sin \beta \]

\[ = Pr [\sin (\phi + \beta)] \]

But,

\[ P = \frac{Q}{\cos \phi} \]

Therefore,

\[ \frac{T}{r [\sin (\phi + \beta)]} = \frac{Q}{\cos \phi} \]

Rearranging:

\[ T = \frac{Q \sin (\phi + \beta)}{\cos \phi} \]

And expanding we have:

\[ T = Q r [\sin \phi \cos \beta + \cos \phi \sin \beta] \]

Or

\[ T = Q r [\tan \phi \cos \beta + \sin \beta] \]  \hspace{1cm} (4.11)

From the relationship

\[ r \sin \beta + e = l \sin \phi \]

\[ \sin \phi = \frac{1}{l} [r \sin \beta + e] \]
\[
\cos \phi = \sqrt{1 - \sin^2 \phi} = \sqrt{1 - \frac{1}{l^2} (r \sin \beta + e)^2}
\]

Thus,

\[
\tan \phi = \frac{\frac{1}{l^2} [r \sin \beta + e]}{\sqrt{1 - \frac{1}{l^2} (r \sin \beta + e)^2}}
\]

Or

\[
\tan \phi = \frac{r \sin \beta + e}{\sqrt{l^2 - (r \sin \beta + e)^2}} \quad (4.12)
\]

Substituting equation (4.12) into equation (4.11) gives:

\[
T = Qr \left[ \frac{(r \sin \beta + e) \cos \beta}{\sqrt{l^2 - (r \sin \beta + e)^2}} + \sin \beta \right] \quad (4.13)
\]

With the values of the crushing force and crank radius, the required torque can be calculated.

The previous group of students undertaking this project used two methods to determine the actual crushing forces for stones. The crushing forces using analytical and energy methods were found to be 68.6 and 90.8kN respectively.

In this design an average of the two methods i.e. 80kN is taken.
Table 4.1: Table of values of torque for different crank angles

<table>
<thead>
<tr>
<th>Crank angle, β (degrees)</th>
<th>Torque, T (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>50</td>
</tr>
<tr>
<td>15</td>
<td>100.7010351</td>
</tr>
<tr>
<td>30</td>
<td>144.4119064</td>
</tr>
<tr>
<td>45</td>
<td>178.0596518</td>
</tr>
<tr>
<td>60</td>
<td>199.3164909</td>
</tr>
<tr>
<td>75</td>
<td>206.7679134</td>
</tr>
<tr>
<td>90</td>
<td>200</td>
</tr>
<tr>
<td>105</td>
<td>179.6024171</td>
</tr>
<tr>
<td>120</td>
<td>147.0936706</td>
</tr>
<tr>
<td>135</td>
<td>104.7830607</td>
</tr>
<tr>
<td>150</td>
<td>55.58809365</td>
</tr>
<tr>
<td>165</td>
<td>2.826582905</td>
</tr>
<tr>
<td>180</td>
<td>-50</td>
</tr>
<tr>
<td>195</td>
<td>-99.41978225</td>
</tr>
<tr>
<td>210</td>
<td>-142.1926976</td>
</tr>
<tr>
<td>225</td>
<td>-175.4971087</td>
</tr>
<tr>
<td>240</td>
<td>-197.0972449</td>
</tr>
<tr>
<td>255</td>
<td>-205.4866233</td>
</tr>
<tr>
<td>270</td>
<td>-200</td>
</tr>
<tr>
<td>285</td>
<td>-180.8837072</td>
</tr>
<tr>
<td>300</td>
<td>-149.3129166</td>
</tr>
<tr>
<td>315</td>
<td>-107.3456037</td>
</tr>
<tr>
<td>330</td>
<td>-57.80730237</td>
</tr>
<tr>
<td>345</td>
<td>-4.107835787</td>
</tr>
<tr>
<td>360</td>
<td>50</td>
</tr>
</tbody>
</table>
The graph of torque against the crank angle for half a cycle of operation is plotted below.

**Graph 4.1: Graph of torque against crank angle**

The maximum value of torque is used in order to design for the worst case scenario. The peak torque is calculated as 206.77 Nm. This is the torque required on the shaft.
CHAPTER 5
5.0 DESIGN OF THE PROPOSED MACHINE

The proposed machine is a secondary crusher which will be used to produce the final product. Most applications in the building and construction industry use aggregate sizes varying between 2cm – 5 cm. The proposed machine can produce aggregate of about 3.5cm, to achieve this aggregate size; the machine should have a set of 3.5cm and a throw of 0.5cm. The maximum throw is the maximum displacement in the horizontal direction of the swing jaw.

The size of the parts of the machine will determine the overall size of the machine. As per the objective of the project, the aim is to design a small scale machine, which is cost effective. Thus an appropriate size of the parts and hence the machine is chosen. In making this decision the driving factors are cost, strength of materials to withstand the loads and the required output.

The length of the connecting rod is chosen to be 0.45metres

The set size = 3.5 cm

To size the gape, the largest size of rock to be fed to the crusher is considered and then the following relation is applied:

Largest rock size = 0.9 x Gape

Thus the gape is about 1.1 times the feed size. Being a secondary crusher the largest feed size is expected to be about 15cm. therefore the gape is given by:

Gape = 1.1 x 15 = 16.5cm

The reduction ratio is given by:

Reduction ratio = \frac{\text{input}}{\text{output}}

For this case,

\frac{15}{3.5} \approx 4

Thus the reduction ratio is 4:1

The vertical height is approximately twice the size of the gape which is 33cm.
The width of the crushing chamber is approximately 1.5 times the gape. Therefore,

\[ \text{Width} = 1.5 \times 16.5 = 24.75 \text{ cm} \approx 25 \text{ cm} \]

### 5.1 PERFORMANCE PARAMETERS

The table below shows performance parameters of different jaw crushers.

*Table 5.1: Performance parameters of different jaw crushers*

<table>
<thead>
<tr>
<th>Crusher type</th>
<th>Size, mm</th>
<th>Reduction ratio</th>
<th>Power, in KW</th>
<th>Toggle speed, rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Gape, mm</td>
<td>Width, mm</td>
<td>range</td>
<td>Average</td>
</tr>
<tr>
<td></td>
<td>Min</td>
<td>Max</td>
<td>Min</td>
<td>max</td>
</tr>
<tr>
<td>Blake double toggle</td>
<td>125</td>
<td>1600</td>
<td>150</td>
<td>2100</td>
</tr>
<tr>
<td>Blake single toggle</td>
<td>125</td>
<td>1600</td>
<td>150</td>
<td>2100</td>
</tr>
<tr>
<td>Dodge type</td>
<td>100</td>
<td>280</td>
<td>150</td>
<td>28</td>
</tr>
</tbody>
</table>

The proposed machine has the following performance parameters:

- **Gape**: 165mm
- **Width**: 250mm
- **Reduction ratio**: 4:1
- The toggle speed chosen is 250rpm
- The average torque is 206.77 Nm
Therefore, the peak power requirement is given by:

\[ P = T \omega \]

\[ = \frac{206.77 \times 2 \pi \times 250}{60} = 5413 \text{ W} = 5.413 \text{ kW} \]

### 5.2 DESIGN OF SHAFT FOR TORQUE TRANSMISSION

A shaft is a rotating or stationary component which is normally circular in section. A shaft is normally designed to transfer torque from a driving device to a driven device. If the shaft is rotating, it is generally transferring power and if the shaft is operating without rotary motion it is simply transmitting torque and is probably resisting the transfer of power. A shaft which is not rotating and not transferring a torque is an axle.

Mechanical components directly mounted on shafts include gears, couplings, pulleys,cams, sprockets, links and flywheels. A shaft is normally supported on bearings. Shafts are subject to combined loading including torque (shear loading), bending (tensile & compressive loading), direct shear loading, tensile loading and compressive loading. The design of a shaft must include consideration of the combined effect of all these forms of loading. The design of shafts must include an assessment of increased torque when starting up, inertial loads, fatigue loading and unstable loading when the shaft is rotating at critical speeds.

In this design, we seek to select the size (diameter) of the shaft such that when subjected to the expected loads, the significant stress will be less than the strength by an adequate margin of safety. For simplification of the design, the loading is assumed to be static.
The loading on the shaft is simplified as shown below.

**Fig 5.1: Diagrammatic representation of the shaft loading**

Mass of each flywheel is 30kg. The downward force resulting from this mass is given by:

\[
\text{Force} = 30 \times 9.81 = 294.3 \text{N}
\]
The free body diagram is as shown below

![Free Body Diagram](image)

*Fig 5.2: Free body diagram of shaft loading*

The bending moment, $M$ is given by:

$$M = \text{Force} \times \text{Distance}$$

Distance = 0.1m

Thus,

$$M = 294.3 \times 0.1 = 29.43 \text{Nm}$$
From the Maximum Shear Stress Theory, the design stress, $\tau_d$ is given by:

$$\tau_d = \frac{16}{\pi d^3} \sqrt{T^2 + M^2} \quad (5.10)$$

The same theory predicts that,

$$S_{sy} = \frac{S_y}{2}$$

Where $S_{sy}$ is the shear strength of the material

And $S_y$ is the yield strength of the material in tension

Design equations:

$$\tau_d = 0.3S_y$$

$$\tau_d = 0.18S_{uts}$$

Thus,

$$d^3 = \frac{16}{\pi \tau_d} \sqrt{T^2 + M^2} \quad (5.11)$$

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 5.2: Shock and fatigue factors

<table>
<thead>
<tr>
<th>Type of loading</th>
<th>$k_b$</th>
<th>$k_r$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load gradually applied</td>
<td>1.5</td>
<td>1</td>
</tr>
<tr>
<td>Load suddenly applied with minor shock</td>
<td>1.5 – 2.0</td>
<td>1.0 – 1.5</td>
</tr>
<tr>
<td>Load suddenly applied with heavy shock</td>
<td>2.0 – 3.0</td>
<td>1.5 – 2.0</td>
</tr>
</tbody>
</table>

Source 6,7: Machinery's Handbook, Franklin Jones, Henry Ryffel, Erik Oberg et al.
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:

\[ k_b = 2.0 \]
\[ k_\tau = 1.5 \]

Using these values, the design equation becomes

\[ d^3 = \frac{16}{\pi \tau_d} \sqrt{k_b T^2 + k_\tau M^2} \]  \hspace{1cm} (5.12)

**SELECTION OF MATERIAL FOR THE SHAFT**

From APPENDIX A: MECHANICAL PROPERTIES OF SOME STEELS, the material selected from the shaft is medium carbon steel, to British Standard specification **BS 970 080M30(H&T)** whose mechanical properties are as shown:

**Table 5.3: Table of mechanical properties of some steels**

<table>
<thead>
<tr>
<th>Material</th>
<th>British Standard</th>
<th>Production Process</th>
<th>Maximum section size, mm</th>
<th>Yield strength, Mpa</th>
<th>Tensile strength, Mpa</th>
<th>% Elongation</th>
<th>Hardness Number, HB</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.30 C</td>
<td>080M30</td>
<td>HR</td>
<td>152</td>
<td>245</td>
<td>490</td>
<td>20</td>
<td>143–192</td>
</tr>
<tr>
<td></td>
<td></td>
<td>254</td>
<td>230</td>
<td>460</td>
<td>19</td>
<td>134 – 183</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>CD</td>
<td>13</td>
<td>470</td>
<td>600</td>
<td>10</td>
<td>174</td>
</tr>
<tr>
<td></td>
<td></td>
<td>63</td>
<td>385</td>
<td>530</td>
<td>12</td>
<td>154</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>H&amp;T</td>
<td>63</td>
<td>385</td>
<td>550–700</td>
<td>13</td>
<td>152 – 207</td>
</tr>
</tbody>
</table>

For the material selected,

The yield strength, \( S_y = 385 \) Mpa

\[ \tau_d = 0.3 S_y \]

\[ = 0.3 \times 385 \]

\[ = 115.5 \) Mpa

Small scale mechanized stone crusher 2010/2011
\[
d^3 = \frac{16}{\pi x 115.5 \times 10^6} \sqrt{2(206.77)^2 + 1.5(29.43)^2}
\]
\[
d^3 = 1.23 \times 10^{-5}
\]
Thus,
\[
d = 0.0235 \text{ m}
\]

5.3 DESIGN OF FLYWHEEL

A flywheel is a mechanical device with a significant moment of inertia used as a storage device for rotational energy. Flywheels resist changes in their rotational speed, which helps steady the rotation of the shaft when a fluctuating torque is exerted on it by its power source. When an increase in torque occurs, the flywheel will speed up and absorb energy. The greater the moment of inertia, the more energy is absorbed. When the torque decreases, the flywheel slows down but the moment of inertia limits the amount it slows.

Therefore, the main function of a flywheel is to maintain a constant angular velocity of the shaft.

The required moment of inertia of the flywheel, and hence the size, depends on the speed fluctuation of the shaft.

The speed fluctuation is given by the maximum velocity minus the minimum velocity

i.e. \[\omega_{\text{max}} - \omega_{\text{min}}\]

This can be normalized by dividing it with the average speed of the shaft to give the coefficient of speed fluctuation.

Coefficient of speed fluctuation,

\[C_s = \frac{\omega_{\text{max}} - \omega_{\text{min}}}{\omega_{\text{ave}}} \quad (5.13)\]

Where,

\[\omega_{\text{ave}} = \frac{\omega_{\text{max}} + \omega_{\text{min}}}{2}\]
The kinetic energy stored in the flywheel,

\[ K.E = \frac{I\omega^2}{2} \]  

(5.14)

The fluctuation in Kinetic Energy for one cycle is given by:

\[ K.E_{\text{max}} - K.E_{\text{min}} = \frac{1}{2} \left[ \omega_{\text{max}}^2 - \omega_{\text{min}}^2 \right] \]  

(5.15)

The fluctuation in kinetic energy is normalized by dividing by the work done, W to give the coefficient of energy fluctuation.

Coefficient of energy fluctuation, \( C_e = \frac{\text{greatest fluctuation in Kinetic Energy}}{\text{work done}} \)

\[ C_e = \frac{\frac{1}{2} \left[ \omega_{\text{max}}^2 - \omega_{\text{min}}^2 \right]}{W} \]

Expanding and rearranging:

\[ C_e = \frac{1}{2W} \left[ (\omega_{\text{max}} + \omega_{\text{min}})(\omega_{\text{max}} - \omega_{\text{min}}) \right] \]

But,

\[ \omega_{\text{ave}} = \frac{\omega_{\text{max}} + \omega_{\text{min}}}{2} \]

Or

\[ \omega_{\text{max}} + \omega_{\text{min}} = 2 \omega_{\text{ave}} \]

And,

\[ C_s = \frac{\omega_{\text{max}} - \omega_{\text{min}}}{\omega_{\text{ave}}} \]
Or

\[ \omega_{\text{max}} - \omega_{\text{min}} = C_s \omega_{\text{ave}} \]

Substituting,

\[ C_e = \frac{1}{2W} \left[ (2 \omega_{\text{ave}})(C_s \omega_{\text{ave}}) \right] \]

Or,

\[ C_e = \frac{IC_s \omega_{\text{ave}}}{W} \]

Therefore, the moment of inertia,

\[ I = \frac{C_e W}{C_s \omega_{\text{ave}}^2} \quad (5.16) \]

Since \( C_e \) is equal to Maximum fluctuation of energy/work done per cycle \( W \), the greatest fluctuation in energy can then be determined and is equal to \( C_e W \). This is done using the graph of Torque vs Crank angle.

When a torque is applied to a body, it rotates. Thus, the work done is a product of the torque (in Nm) causing the rotation and the angle (in radians), i.e. \( W = T \beta \).
To find the maximum fluctuation of energy:

Let the energy at point A be $E_A$

At point B energy has increased by an amount equal to $A_1$

Energy at B= $E_B= E_A + A_1$

It is observed that maximum energy occurs at point B while minimum energy occurs at point A

Therefore, the maximum fluctuation of energy $= E_B - E_A$
\[ \begin{align*}
&= E_A + A_1 - E_A \\
&= A_1
\end{align*} \]

The area \( A_1 \) can be approximated from the graph above using the counting of squares method.

Area \( A_2 = 177 \) full small squares + 40 half small squares

Therefore,

\[ A_1 = 197 \text{ full small squares.} \]

The energy of one small square,

\[ = 0.2 \times 10 \text{ joules} \]

The maximum fluctuation of energy,

\[ = A_1 (0.2 \times 10) \text{ joules} \]

\[ = 197 [0.2 \times 10] \]

\[ = 394 \text{ joules} \]

But,

\[ C_e = \frac{\text{maximum fluctuation of energy}}{\text{work done per cycle } W} \]

Thus,

\[ C_e W = 394 \text{ joules} \]

The coefficient of fluctuation of speed, \( C_s \) is a design parameter to be decided by the designer. The smaller the value of \( C_s \), the smoother is the operation but the larger the size of the flywheel and hence high costs. It is typically set to a value between 0.01 and 0.05 for precision machinery. For stone crushers the coefficient of fluctuation of speed, \( C_s \) is up to 0.2

The flywheel is a solid cylinder whose moment of inertia, \( I \) is given by:
\[ I = \frac{M}{2} R^2 \]  (5.17)

Where \( M \) = mass of the flywheel and 
\( R \) = radius of the flywheel.

Equating equations (5.16) and (5.17) gives:

\[ I = \frac{c_e W}{c_s \omega_{ave}^2} = \frac{M}{2} R^2 \]

Thus,

\[ R = \sqrt{\frac{2c_e W}{c_s M \omega_{ave}^2}} \]  (5.18)

Substituting values into equation (5.18)

\[ R = \sqrt{\frac{2(394)}{0.2 \times 60 \left(2 \times \frac{250}{60}\right)^2}} \]

\[ = 0.31 \text{ metres} \]
CHAPTER 6

6.0 DISCUSSION

The project sought to design a small scale mechanized stone crushing machine that is simple, economical, easy to use and maintain in order to empower the manual stone crushers while meeting the increasing demand for aggregate in the Kenyan building and construction industry.

The kinematic analysis of the proposed design was done in order to determine motion of the parts. The motion was simplified to a Crank and slider mechanism. An equation describing the motion of the moving jaw was derived as:

\[ x = \sqrt{(l + r)^2 - e^2} - r \cos \beta - \sqrt{l^2 - (r \sin \beta + e)^2} \]

A graph of displacement against the crank angle for one cycle was plotted. The graph had a bell shape and maximum throw of 0.005m was obtained. The movement of the jaw is quite small but sufficient since complete crushing is not performed in one stroke.

The displacement equation above was differentiated to give the velocity and acceleration equations. The graphs of velocity and acceleration against the crank angles were found to be periodic but not sinusoidal.

The force analysis was done based on crushing loads obtained in the previous analysis of the laboratory jaw crusher.

The group had obtained crushing loads of 68.6 and 90.8kN using the analytical and energy methods respectively. In this design an average of the two methods i.e. 80kN was used.

The torque equation was obtained as:

\[ T = Qr \left[ \frac{(r \sin \beta + e) \cos \beta}{\sqrt{l^2 - (r \sin \beta + e)^2}} + \sin \beta \right] \]

This equation is informative as it relates the input mechanical power to the crushing power achieved by the machine.

From the graph of torque against crank angle, an average torque of 206.77 Nm was obtained which was used to calculate the power consumption of the machine.
It was noted that the rotational speed of the machine affects the rate of production and the power input. Increasing the rotational speed increases the rate of production but simultaneously increases the power consumption of the machine, thus a balance was found.

Using an average rotational speed of \textbf{250rpm}, the power requirement was found to be \textbf{5.413 kW}. The motor used should therefore be able to provide this power.

The shaft used to transmit the torque was designed. The loading during operation usually involves shock and fatigue. Therefore, the design equation was modified using factors \( k_b \) and \( k_\tau \) to account for shock and fatigue respectively. A factor of safety of \textbf{2} was also included. Thus the design equation became:

\[
d^3 = \frac{16}{n\tau_d} \sqrt{k_bT^2 + k_\tau M^2}
\]

The minimum diameter required was found to be \textbf{0.0235 m}.

The machine was designed to be portable. A flywheel is used to maintain a constant angular velocity of the shaft. The flywheel contributes a significant fraction of the overall weight of the machine. Two flywheels each weighing \textbf{30kg} were used to provide the necessary moment of inertia or to store the energy and smoothen the rotation of the shaft. With this mass, the radius of the flywheel was obtained as \textbf{0.31m}.
6.1 CONCLUSION

The proposed mechanism has fewer parts as compared to the existing jaw crushers. The implication is that it is cheaper to make and requires less maintenance.

The overall weight of the machine is approximately 150 kg (flywheel accounts for more than a third of the overall weight). This makes the machine portable and solves the problems associated with heavy and capital intensive centralized crushing.

The machine will also improve the output of the manual stone crushers significantly and thus increase their earnings.

The design of this machine is therefore a viable undertaking.
6.2 RECOMMENDATIONS

The power requirement of the shaft is 5.413 kW. For this power rating, we recommend the selection of an appropriate motor to be used. Moreover, the design of the power transmission mechanism from the motor to the shaft should be carried out.

The assembly of the proposed machine should be done with the view of testing it in order to monitor its actual performance parameters. This will assist in coming up with improvements to the proposed design.
APPENDIX A: MECHANICAL PROPERTIES OF SOME STEELS

<table>
<thead>
<tr>
<th>Material</th>
<th>British standard</th>
<th>Production process</th>
<th>Maximum section size, mm</th>
<th>Yield strength, Mpa</th>
<th>Tensile strength, Mpa</th>
<th>Elongation %</th>
<th>Hardness number, HB</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.20C</td>
<td>070m20</td>
<td>HR</td>
<td>152 215</td>
<td>430 22</td>
<td>126-179</td>
<td>154</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>CD</td>
<td>13 385</td>
<td>530 14</td>
<td>125</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>H&amp;T</td>
<td>63 385</td>
<td>550-700</td>
<td>13 152-207</td>
<td>154</td>
<td></td>
</tr>
<tr>
<td>0.30C</td>
<td>080M30</td>
<td>HR</td>
<td>152 245</td>
<td>490 20</td>
<td>143-192</td>
<td>134-183</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>CD</td>
<td>13 470</td>
<td>600 10</td>
<td>174</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>H&amp;T</td>
<td>63 385</td>
<td>530 12</td>
<td>154</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.40C</td>
<td>080M40</td>
<td>HR</td>
<td>150 280</td>
<td>550 16</td>
<td>152-207</td>
<td>165</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>CD</td>
<td>63 430</td>
<td>570 10</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>H&amp;T</td>
<td>150 385</td>
<td>625-775</td>
<td>16 179-229</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.50C</td>
<td>080M50</td>
<td>H&amp;T</td>
<td>150 310</td>
<td>620 14</td>
<td>179-229</td>
<td>188</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>CD</td>
<td>63 510</td>
<td>650 10</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>H&amp;T</td>
<td>150 430</td>
<td>625-775</td>
<td>11 179-229</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1Cr</td>
<td>530M40</td>
<td>H&amp;T</td>
<td>100 525</td>
<td>700-850</td>
<td>17 202-255</td>
<td>248-302</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>29 680</td>
<td>850-1000</td>
<td>13</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.5MnMo</td>
<td>605M36</td>
<td>H&amp;T</td>
<td>150 525</td>
<td>700-850</td>
<td>17 202-255</td>
<td>269-331</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>29 755</td>
<td>925-1075</td>
<td>12</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.25NiCr</td>
<td>640M40</td>
<td>H&amp;T</td>
<td>152 525</td>
<td>700-850</td>
<td>17 202-255</td>
<td>248-302</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>102 585</td>
<td>770-930</td>
<td>15 223-277</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>64 680</td>
<td>850-1000</td>
<td>13</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>29 755</td>
<td>930-1080</td>
<td>12</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3NiCr</td>
<td>653M31</td>
<td>H&amp;T</td>
<td>64 755</td>
<td>930-1080</td>
<td>12</td>
<td>269-331</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>680 850-1000</td>
<td>12</td>
<td>248-302</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>680 850-1000</td>
<td>12</td>
<td>248-302</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1CrMo</td>
<td>708M40</td>
<td>H&amp;T</td>
<td>150 525</td>
<td>700-850</td>
<td>17 201-255</td>
<td>311-375</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>13 940</td>
<td>1075-1225</td>
<td>12</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3CrMo</td>
<td>722M24</td>
<td>H&amp;T</td>
<td>152 680</td>
<td>850-1000</td>
<td>13 269-331</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>755 930-1080</td>
<td>12 269-331</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.5NiCrMo</td>
<td>826M40</td>
<td>H&amp;T</td>
<td>150 755</td>
<td>925-1075</td>
<td>12 269-331</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>850 1000-1150</td>
<td>12 293-352</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>1020 1150-1300</td>
<td>10 341-401</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3NiCrMo</td>
<td>830M31</td>
<td>H&amp;T</td>
<td>254 650</td>
<td>850-1000</td>
<td>13 248-302</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>152 680</td>
<td>850-1000</td>
<td>12 248-302</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>64 940</td>
<td>1080-1240</td>
<td>11 311-375</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.5MnNiCrMo</td>
<td>945M38</td>
<td>H&amp;T</td>
<td>152 525</td>
<td>690-850</td>
<td>17 201-255</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>64 680</td>
<td>850-1000</td>
<td>13 248-302</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>29 850</td>
<td>1000-1160</td>
<td>12</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

APPENDIX B: LIST OF FIGURES

Fig 1.1: large scale aggregate production

Fig 1.2: A woman manually crushing stones at Makongeni near Thika town

Fig 2.1: Gyratory crusher
Fig 2.2: Jaw crusher
Fig 2.3: Cone crusher
Fig 2.4: Impact crusher
Fig 2.5: Working principle of a jaw crusher
Fig 2.6: Single toggle jaw crusher
Fig 2.7: Double toggle jaw crusher
Fig 2.8: Dodge type jaw crusher
Fig 2.9: Diagrammatic representation of the proposed mechanism
Fig 3.4: Kinematic diagram of the proposed mechanism
Fig 3.5: Kinematic diagram showing the maximum displacement of point B
Fig 4.1: Diagram of torque and force analysis
Fig 5.1: Diagrammatic representation of the shaft loading
Fig 5.2: Free body diagram of shaft loading
APPENDIX C: LIST OF TABLES AND LIST OF GRAPHS

Table 2.1: Commonly used crushers and their applications.

Table 3.1: Critical points

Table 3.2: Critical points

Table 4.1: Table of values of torque for different crank angles

Table 5.1: Performance parameters of different jaw crushers

Table 5.2: Shock and fatigue factors

Table 5.3: Table of mechanical properties of some steels

Graph 3.1: Graph of displacement against crank angle

Graph 3.2: Graph used to obtain first critical point

Graph 3.3: Graph used to obtain second critical point

Graph 3.4: Graph of velocity against crank angle

Graph 3.5: Graph of acceleration against crank angle

Graph 4.1: Graph of torque against crank angle

Graph 5.1: Graph of torque against crank angle in radians
REFERENCES

6. www.metsominerals.com
8. www.wikipedia.com