SULTAN QABOOS UNIVERSITY

College of Engineering

Department of Mechanical and Industrial Engineering

Final Year Project II – MEIE5194

Final Year Project II

Design and Fabrication of an Experimental Setup to Investigate Fatigue Failure in Drilling Pipes

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Abstract

This project is aiming to build up an experimental setup that imitates drilling for oil and gas in order to investigate fatigue failure of drill-string. The proposed setup is about subjecting the drill-string to torsional and bending loads by rotating it at some given number of revolutions per minute. The rotational mechanism of the tested drilling-string is, in principle, to mimic the rotary drilling operation. In the real case, the drill-string is attached to a drilling bit which cuts through a rock formation. Resistivity of the formation is simulated through designing and building braking mechanism. In order to maintain concentricity and provide adequate stabilization and guiding the drill-string without imposing excessive torque, support mechanism has been designed. The design methodology has been followed in two main stages. The concept generation and evaluation, and the embodiment design were established in the Fall semester. In spring semester, efforts were focused to fabricate the set-up. Important decisions were taken in terms of manufacturing the setup. Preliminary tests were conducted to validate the design integrity and robustness of the setup. Number of tests were conducted to study the motion and behavior of the drill-string and the fatigue problem due to lateral and torsional vibrations was generated successfully. A comprehensive finite element modeling and analysis also developed to better understanding the drilling pipe behavior and obtain the natural frequencies of the rotating test specimens, and consequently, their critical speeds.

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Chapter 1.0: Introduction

1.1. Introduction

In the recent years the operation of drilling has been developed significantly and the technology has become advanced and sophisticated. The drilling operation is a major aspect of the journey to the black gold. A lot of money is spent in this operation where the equipment used is so expensive. Figure 1 shows schematic diagram of an oil rig where the major components of the oil rig are indicated. While the operation is held, some of these components fail to perform their intended function and delay the process or suspend it. One of the major components that frequently fail is the drill-string.

Failure of a drill-string not only suspends the operation but also costs a lot of money in terms of fixing and replacement of the drill-string. The failure of the drill string is a common problem that oil companies suffer from and still no super solution exists to overcome this major issue. The drill-string could fail due to many reasons but the main source of failure is mechanical fatigue where the drill-string is exposed to cyclic stresses when it cuts through a very hard formations resulting in a very high friction forces that keep pressing on the drill-string. Most of the failures happen in directional drilling where the drill-string starts to drill horizontally. The fluctuating loads on the drill-string cause it to fail especially at the connection between strings as some recent studies have presented. There are many ways to delay or reduce the likelihood of this type of failure to occur such as enhancing mechanical behavior of the drill-string materials and strengthening the connection or predicating the failure before it occurs. Such solutions might sound easy but actually they are cumbersome. There are many factors that are involved in such problem where an extensive study of nature of the formation is needed first to decide which way is more suitable for the improvement.

During drilling, the drill-string rotates but because of its high length which might go to 6km, it will tend to whirl. Also vibration of different types is a main reason that induces the fatigue problem. A combination of rotation and vibration leads to a cyclic loading which is basically mechanical fatigue. Therefore, an experimental setup that simulates that situation is needed to observe cyclic loading and get, at least, partial understanding of the basic mechanism associated with fatigue failure of the drill-string. Moreover, the way to improve or solve the problem can be decided from the observation process.

Simulating the drilling operation in a whole scale is impossible so a small scale of that would give quiet good result. The project is about simulating the directional drilling operation and producing a fatigue problem on a drilling pipe.

1.2. Problem Statement

Long length and heavy weight of the drill-string cause lateral vibration and the high resistivity of the formation cause torsional vibration. These vibration modes result in fatigue failure in the drill-string.

1.3. Project Objective

The project objective is to design and fabricate an experimental setup to study fatigue failure in drilling pipes. The following tasks are planned to arrive at the project general objectives:

- Develop an experimental setup to simulate fatigue failure in drill-string.
- Facilitate accelerated life–fatigue testing using drilling strings with various geometric parameters and loading conditions.
- Create lateral and torsional vibration modes.
- Propose a comprehensive finite element analysis to study the mechanical response of the drilling string under some controlled loading conditions.
- Observe the general behavior of the drill-string when lateral and torsional vibrations are generated.

1.4. Motivation

Although there are many setups that can imitate the drill-string failure, they are limited to very specific failure types. Therefore, the reasons that motivate us to work on this project are:

- No setup that simulates the above mentioned problem exists.
- The topic is highly involved in oil and gas industry which is considered to be the main source of income of Oman.

- Fatigue failure is concern in oil and gas industry where most of the failures are a result of it.
- We can make money out of this project where some oil and gas companies may show some interest in it.

1.5. Management Plan

1.5.1. Gantt Chart

Gantt chart is usually used in the project management where it shows the project activities and tasks over a specified period of time. Implementing Gantt chart in this project has made the project well organized in terms of time. With this, there was less work pressure since every task and activity has been given a restricted period of time. Below here, the Gantt chart for the entire project has been followed very well. The planned project tasks were divided into four main phases as follows:

1. **Literature review.**
2. **Drilling facility design and development.**
   2.1. Setup design and selection of items.
   2.2. Structural support development.
   2.3.Rotation mechanism development.
   2.4. Braking system design and development.
   2.5. Instrumentation and measurement devices selections and implementations.
   2.6. Motor, bearings and linear actuators selections.
3. **Drill pipe fatigue failure testing.**
   3.1. Testing at normal condition (normal temperature and pressure; bending loading only; bending and torsion loadings).
   3.2. Testing at normal condition (normal temperature and pressure; bending and torsion loadings).
4. **Analysis, documentations and final report.**

Table 1: Gantt Chart

Phase	Task	Subtask		Month 1	Month 2	Month 3	Month 4	Month 5	Month 6	Month 7	Month 8	Month 9	Month 10
I	**1.1**		**Literature review**										
II	**2.1**		**Drilling facility design and development**										
		2.1.1	Setup design and selection of items										
III		2.1.2	Structural support development										
		2.1.3	Rotation mechanism development										
		2.1.4	Braking system design and development										
		2.1.5	Instrumentation and measurement devices selections and implementations										
		2.1.6	Motor, bearings and linear actuators and material purchase.										
	3.1		**Drill pipe fatigue failure testing**										
		3.1.1	Testing at normal condition (normal temperature and pressure; bending loading only)										
		3.1.2	Testing normal conditions (bending and torsion loadings)										
V	**4.1**		**Analysis, documentations and final report**										

Progress report

FYP I report

Progress report of FYPII

Final report of FYPII

Chapter 2.0: Literature Review

2.1. Oil Field

The income of Oman is mainly dependent on oil and gas fields. Therefore, a big attention and care are given to this source of income. High level of technology and sophisticated equipment are used in upstream oil industry. Thus, a huge amount of money is invested to explore and produce oil and then make a profit out of it. With all of this advanced technology used in the oil extraction process still they face difficulties due to the harsh nature of Oman's land and the geographic of the region. As result of the harsh rock formation of Oman's land a lot of failures occur in the drilling setup like the drill-string which costs a lot and causes major delays in the production process. The oil field itself is a big science and a lot of operations are carried there but since this project is mainly targeting the fatigue failure of the drill-string and therefore the drilling operation would be explained in some details.

2.1.1. Drilling Operation

In the oil drilling process, first the rig is set up and the drilling process gets started. First of all, the team drills a surface hole down to a pre-set depth which is above where the oil trap is located according to their expectations. There are five basic steps to drill the surface hole:

1- The drill bit, collar and the drill pipe are placed in hole.

2- The Kelly and the turntable are attached and then the drilling begins.

3- As drilling progresses, mud is circulated through the pipe and out of the bit to float the rock cuttings out of the hole.

4- As the hole gets deeper new sections (joints) are added to the drill pipes.

5- The drill pipe, collar and bit are removed (pulled out) when pre-set depth which is anywhere from a few hundred to a couple- thousand feet near the oil reservoir.

Once the set-depth is reached, they run and cement the casing and place the casing pipe section into the hole to prevent it from collapsing. There are spacers in the casing pip that keep it centered in the hole. The cement is pumped down to the casing pipe using a bottom plug, cement slurry, a top plug and drill mud. The drill mud pressurizes the cement slurry to move through the casing and fill the space between the casing and the well's walls. Then, the cement is allowed to harden. This procedure is repeated till the direction drilling takes place. In the direction drilling, the drill-string drills horizontally rather than vertically and this gives the advantage to reduce the number of well in the same region. Figure 1 shows a schematic diagram of an oil rig to illustrate its components.

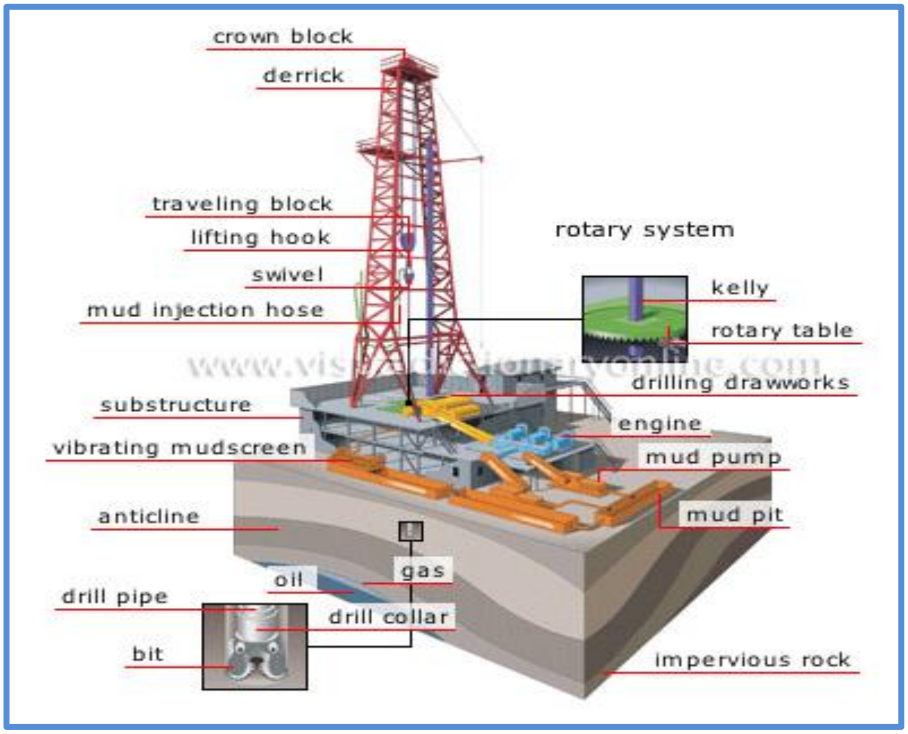


Figure 1: Schematic Diagram of Drilling Oil Rig (Baker, 2001)

2.2. Failure and Failure Modes

The word failure is a generalized word that can be implemented into different fields, the only concern here is about the Engineering Failure which is defined as the inability of component or material to perform the required function or to meet the performance criteria or to perform reliably and safely. Failure might have many different modes in the ductile and the brittle material and it might be caused by these factors:

- Unsuitable material used for the intended function.

- Improper design (unprofessional design).
- Misuse.
- Environmental impact.

The failure might cause catastrophic disasters and it might cost a lot of resources, unless all of the failure causes are considered which need comprehensive experimental, analytical and numerical techniques. Considering the failure causes will surly delayed the failure or it might prevent it. Engineering Failure can occurs on both brittle and ductile material, and it has deferent behaviors in each one of them, but in general the engineering failure can be categorized on any type of material as a fracture failure, fatigue failure or creep failure.

2.3. Fatigue Failure

Fatigue failure is one of the dynamic failures that are encountered in all aspects of industrial field such as the oil and gas field. There are many costly researches and studies about fatigue failure, since it is accounting for more than 50 % of metallic failure and causes heavy financial losses. Fatigue is a dynamic failure that happens when a structure is subjected to cyclic (fluctuated) stresses. Consequently, the failure could occur below the yield or tensile strengths of static loads. Therefore, fatigue failure occurs suddenly and without warning which leads to a catastrophic disaster. Since there is a little plastic deformation in the fatigue failure, the fatigue crack is similar to brittle crack even in ductile material.

Cyclic stresses in a structure take the form of sinusoidal shape as shown in figure 2 where *mean stress, stress amplitude and stress ratio* defined as:

$$\boldsymbol{\sigma_m = \frac{\sigma_{max}+\sigma_{min}}{2}} \quad \text{(1): Main Stress}$$

$$\boldsymbol{\sigma_a = \frac{\sigma_{max}-\sigma_{min}}{2}} \quad \text{(2): Amplitude Stress}$$

$$\boldsymbol{R = \frac{\sigma_{min}}{\sigma_{max}}} \quad \text{(3) Range Ratio}$$

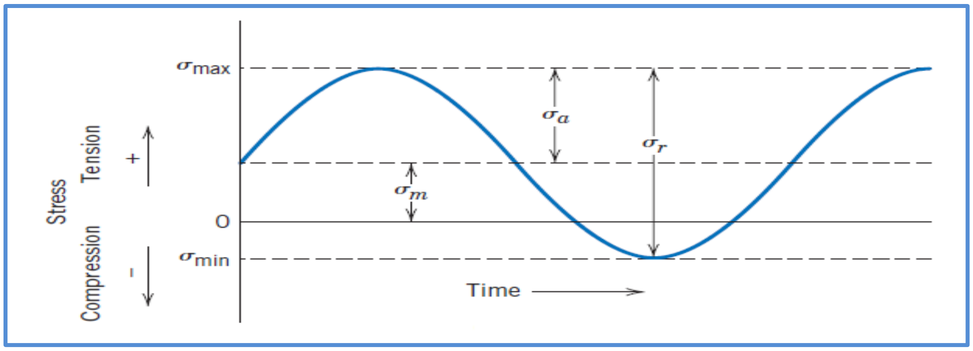


Figure 2: Repeated Stress Cycle (Callister, 2009)

Three types of loads that are countered in structure: axial (tension - compression), flexural (bending) and torsional (twisting). Therefore, three modes of different fluctuating stress could happen for the structure. First one is reversed stress cycle, figure 3, where the amplitude is symmetric about the zero mean stress level. Repeated stress cycle, figure 2 is another type where the mean stress is not at zero level. Third type called Random stress cycle, figure 4, varies in amplitude as well as frequency.

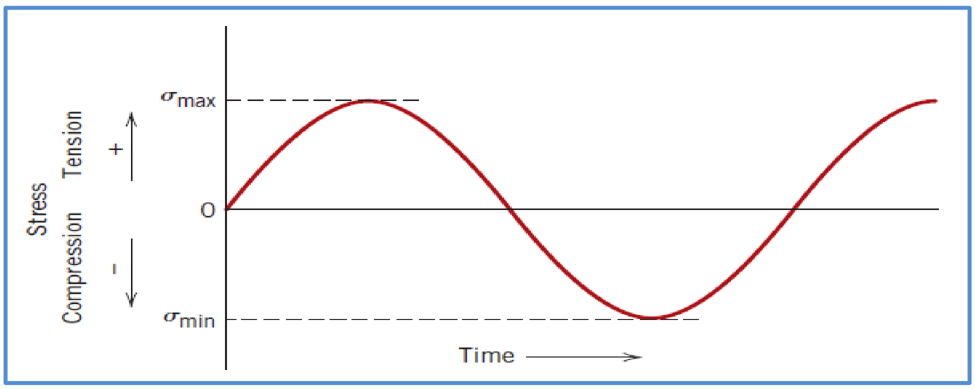


Figure 3: Reversed Stress Cycle (Callister, 2009)

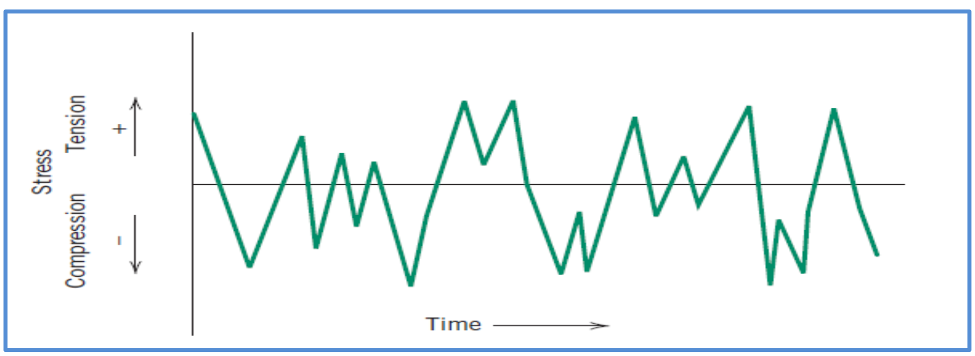


Figure 4: Random Stress Cycle (Callister, 2009)

Although, there is no sufficient explanation of how the fatigue failure occurs, engineers must implement safety in their design. Most common approach used in design to predict the fatigue failure is *fatigue life method* which predicts the number of cycles needed for structure to fail when it is subjected to a specific load. Fatigue could be classified further to low-cycle fatigue where fatigue life (N) $1 \leq N \leq 10^3$, and high-cycle fatigue where $N>10^3$.

2.3.1. The S-N Diagram

Figure 5 shows a rotating-bending test apparatus used for determining the fatigue properties. There are some stress conditions had to be implemented in fatigue testing includes stress level, time frequency and stress pattern. The specimen is imposed to a compression and tensile stresses simultaneously with bending and rotating the specimen. An alternating uniaxial tension-compression stress could also be used in fatigue test.

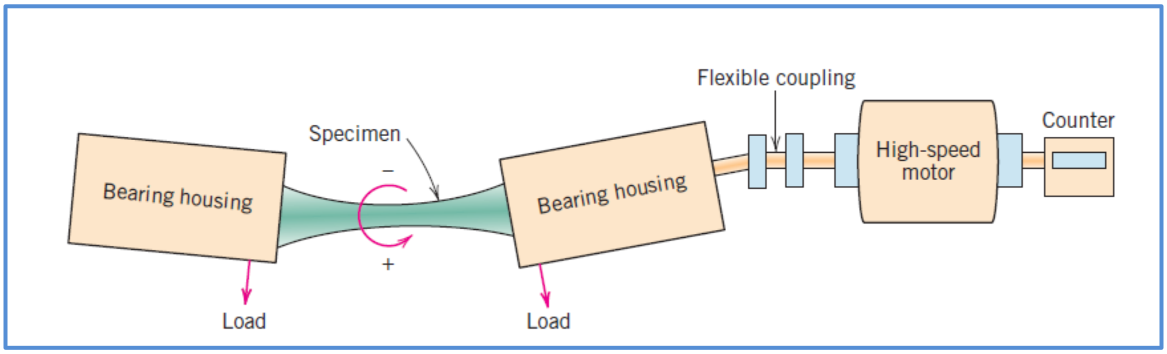


Figure 5: Fatigue-Testing Apparatus for Making Rotating-Bending Tests (Callister, 2009)

Many tests have to be conducted to come up with S-N diagram. In each test, the specimen is subjected to the cyclic stress at specific maximum stress amplitude. Then, the number of cycle to failure is counted. The process is repeated for many specimens with decreasing the stress amplitude. After that the data are plotted as Stress (S) versus number of cycles to failure (N). This plot indicates that as the strength of stress subjected to a specimen increases the number of cycle to failure decreases as shown in figures 6 and 7.

Two different type of S-N diagram are found after the tests are conducted for different materials. In first type, the S-N curve becomes horizontal at higher N values as shown in figure 7. The stress where the curve is constant is called fatigue limit or endurance limit. There is no failure due to fatigue happen below this limit. Endurance limit, however, exists for only some ferrous and titanium alloy. For nonferrous alloy, there is no endurance limit and the material behavior exhibits as shown in figure 6. Therefore, the fatigue will occur ultimately regardless of stress that is subjected to the structure.

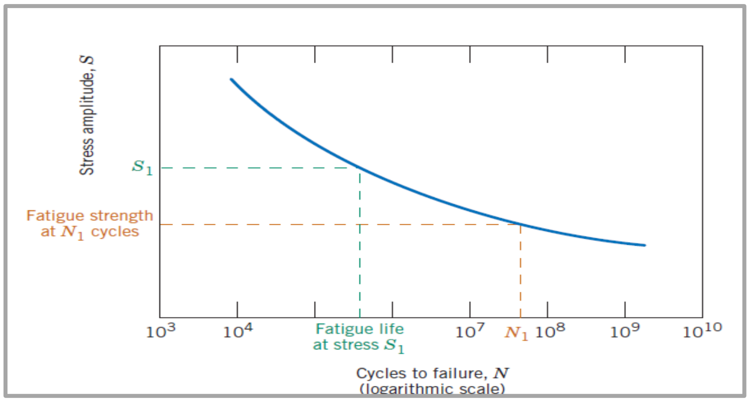


Figure 6: S-N Diagram of a Material that Doesn't Display Fatigue Limit (Callister, 2009)

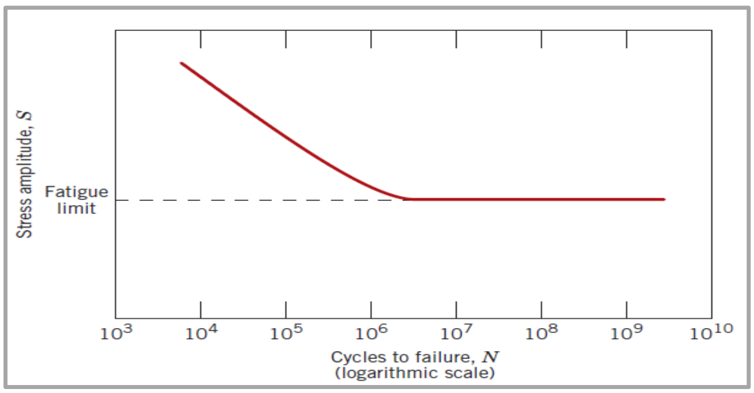


Figure 7: S-N Diagram of a Material that Displays Fatigue Limit (Callister, 2009)

2.4. Crack Growth

Every fatigue failure problem starts at the microstructure level on the bonds between the atoms. The material surface integrity defines the type of surface if that might has a good profile or not. Thus, the material surfaces that has any defects like pitting holes, voids or any type of discontinuity was mainly caused by insufficient manufacturing process or by the environment impact. The fatigue failure starts at the area where the stress concentration is really high, starting as a void nucleation, then the linkage between the voids occurs creating a series of connecting voids on form of a narrow crack, and because of the cyclic stress, those cracks starts to propagates until it reached to a critical size causing a sever failure. Figure 8 shows how the cracks on metals take a place on a metallic specimen under tensile stress leading to a fracture.

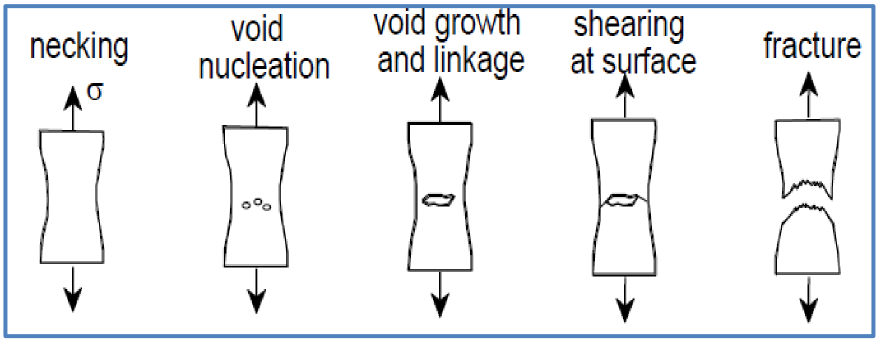


Figure 8: Crack Initiation and Growth in Ductile Material (Schmid, 2008)

Cracks have been categorized into two type, edge crack and center crack as shown in figure 9, whereas each type of cracks has different geometries and different effects on the fatigue life of any metallic parts. Understanding the fatigue crack propagation will predict fatigue life of the metallic part. Figure 9 shows the two types of the crack with their different geometries.

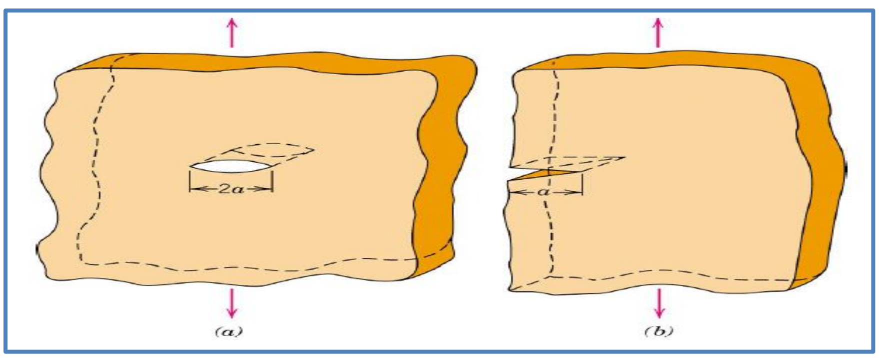


Figure 9: (a) Center Crack, (b) Edge Crack (Callister, 2009)

A closer look to the crack will surely provide fine details about the crack, which would help to predict the fatigue life. Once an sufficient stress is applied to any metallic part that has an initial cracks, The stress concentration would be higher at the crack tip, and the only thing that will increase the chance for the fatigue failure is that the maximum stress (σ_m) at the crack tip is about to reach to the critical stress (σ_c), so if the($\sigma_m > \sigma_c$) ;fatigue failure will definitely occurs leaving the chances for further operation of the metallic part. Below here, figure 10 shows a closer look at the crack tip.

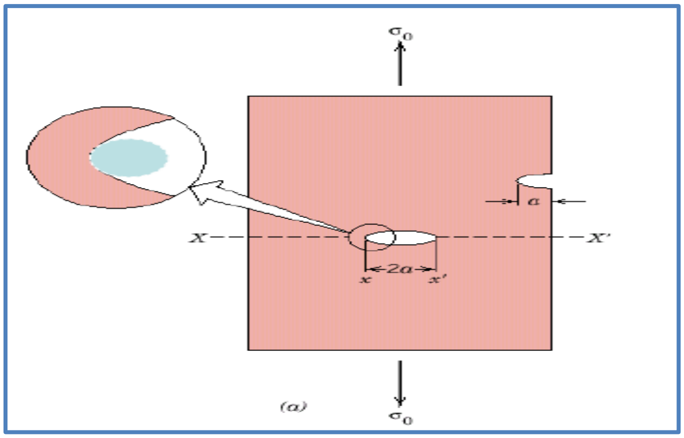


Figure 10: Crack Tip (Callister, 2009)

Another important parameter is needed to be justified is the stress intensity factor at the crack tip which depends on the amount of applied stress plus the crack size (a) and upon the different crack geometries. The stress intensity factor can be denoted as K_1, which is for mode 1 testing (a tensile stress perpendicular to crack surfaces), but there is K_{11} and K_{111} for the second and third mode. This factor can be found by using this equation:

$$K_I = Y\sigma\sqrt{\pi a}$$ **(4): Fracture Toughness**

Where σ is the applied nominal stress, a is the crack length and Y is a dimensionless constant that represents the crack geometry.

Apart from the stress intensity factor, the relationship between the applied stress and the crack length is expressed as the fatigue crack growth rate da/dN were N is denoted for the number of stress cycles. As a result, if the crack size a is too small, then da/dN will be also too small. A further note about the fatigue crack growth rate is that it is a function of the applied stress and the crack size and it is also a function of stress intensity factor K_1, and this can be shown as:

$$\frac{da}{dN} \propto f(\sigma, a)$$ **(5): *Crack Growth as* a *Function of Stress and Crack Size***

In addition, the crack growth rate can be calculated by using this equation:

$$\frac{da}{dN} = A(\Delta K)^{m} \quad \text{(6): Crack Growth Rate}$$

Where ΔK is the difference between the $K_{1,max}$ and $K_{1,min}$, and A & m are constant dependence on the material , frequency , environment , temperature and stress ratio. Below here, figure one shows the fatigue crack growth for a test specimen under two applied stress.

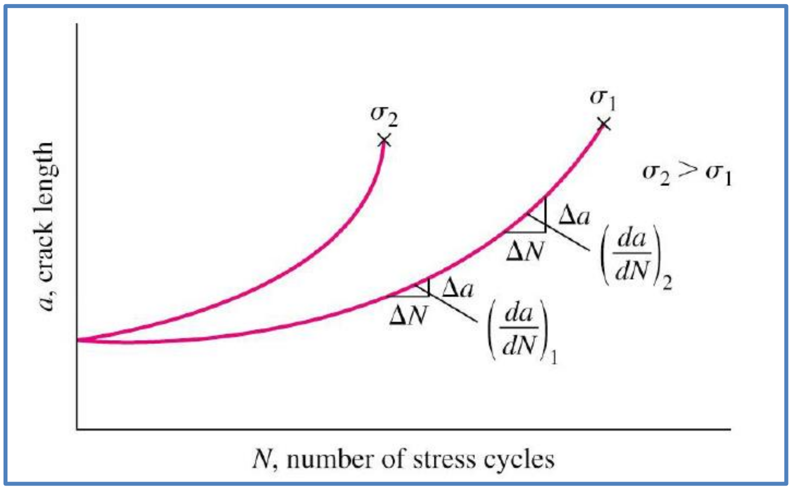


Figure 11: Fatigue Crack Growth Rate at Different Stress (Anderson, 2005)

Beside the fatigue crack growth, the fatigue life can be also predicted by using the below equation:

$$\mathbf{N_f = [\, a_f^{-(m/2)+1} - a_o^{-(m/2)+1} \,] / [\, AY^m \Delta\sigma^m \pi^{m/2} (-m/2+1)]} \quad \text{(7): Fatigue Life}$$

Where Y is independent of crack growth which can be justified as Y=f(a), and m≠2.

2.5. Fatigue Failure in Drill String

The drill-string in oil wells may fail due to several reasons. However, the most common failure criterion is fatigue. While the drill-sting is penetrating the formation, it goes under the effect of various cyclic stresses resulting from frictional and normal forces. These forces excite the drill-string and make it vibrate. Different types of drill-string vibration are discussed in the coming section.

2.5.1. Vibration

Vibrations can be categorized into four main types:

- Natural vibrations.
- Self-excited vibrations.
- Forced vibrations.
- Parameter-excited vibrations.

Rotary drilling systems that are used to drill deep boreholes for hydrocarbon exploration and production often exhibit complex self-excited and forced vibratory behaviors.

2.5.2. Drill String Vibration

The main content of this section on vibration mechanisms are taken from: Halliburton – Sperry Drilling, ADT Drilling Optimization Brochure, 2010: Drill string Integrity Service – Vibration Sensors and Vibration Mitigation Guidelines. For a drill string, there exist three modes of vibration. These modes are:

- Axial.
- Lateral.
- Torsional.

For each of the above modes, there exist popular vibrational mechanisms. An explanation for each mode along with its vibrational mechanisms is mentioned below.

2.5.3. Axial Vibration

In the axial mode, the vibration is longitudinal motion along the drill-string resulting in very intensive and occasionally compression and tension reversals. An example of axial vibration mechanism is bit bounce. Bit bounce is known as intermittently loose of contact between drill bit and the whole formation surface which results in damaging of the drill bit cutting structure, bearing and seals. Furthermore, it results flexing of the drill-string, causing more damage from lateral and axial shocks. When drilling with a three-cone bit, the initial cause may be some hard spot in the formation. Other reasons are pressure fluctuations in the drilling pipe, or when the drill-string is not rotating. For example, when using a steerable mud motor in oriented mode, longitudinal stick-slip friction can induce irregular vibrations. In some circumstances this can become so severe that it is impossible to hold a constant tool face. The axial motion damages the bit cutting structure, seals and bearings. Top drive and hoisting equipment may start to shake axially and if severe enough, lateral BHA vibrations may be introduced.

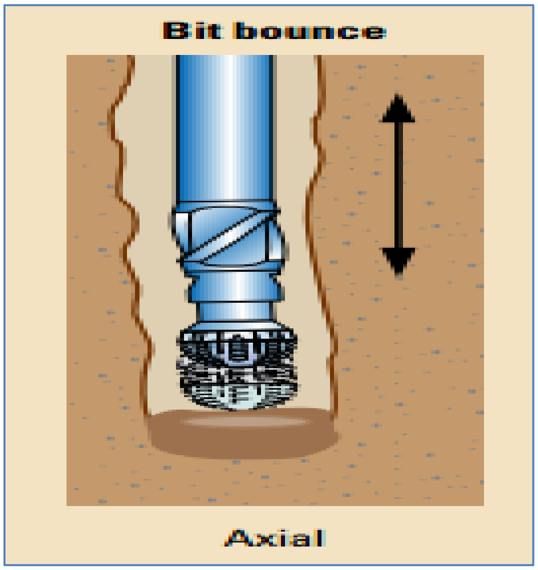


Figure 12: Axial Vibration (Dunayevsky, 1993)

2.5.4. Torsional Vibration

In torsional mode, the vibration is induced by rotation, resulting in twisting as torque is applied to overcome resistance causing irregular downhole rotation. The most popular torsional vibration mechanisms are:

1. **Stick-Slip Mechanism**

In stick-slip mechanism, the drill bit stops rotating at regular intervals, causing drill string to periodically torque up and spin free at 3-15 times the average surface rpm. Stick-slip is typically encountered in high angle and deep wells, when encountering hard formations or salt or when using aggressive PDC (Polycrystalline Diamond Compact) bits in combination with large WOB. Observable stick-slip effects are:

- More than 15% fluctuation in the average surface torque readings.
- Damaged PDC Bit's.
- Reduced ROP (Rate Of Penetration).
- Connection over-torque.
- Back-off and Drill-string spin-off.
- Mud Pulse telemetry detection problems.
- Wear on stabilizers and Bit gauge.

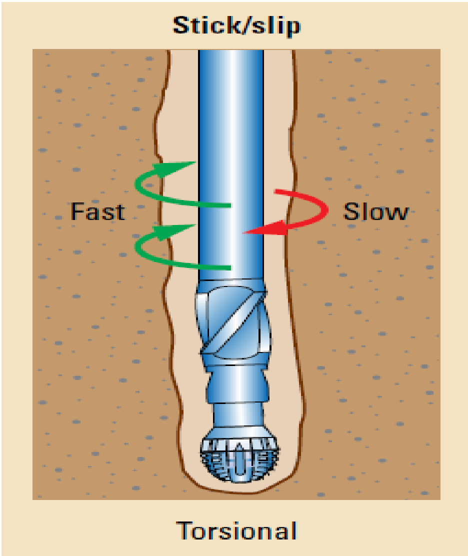


Figure 13: Torsional Vibration (Dunayevsky, 1993)

2. Torsional Resonance

More specifically this is drill collar torsional resonance, as this mode is related to the natural torsional frequency of the drill collars and is the consequence of the drill collars being excited. This very specific type of vibration is encountered when drilling in very hard rock's with PDC bits. This vibration mode is most damaging at higher rotational speeds. This is because at higher rotational speeds, higher amplitude resonance at the harmonics of the drill collar's natural frequency may occur. In some cases this high amplitude fluctuations may cause backward turning of the bit, damaged cutters as well as severe damage to downhole electronics.

3. Bit Chatter

Bit chatter is caused by the individual teeth of the bit impacting the rock. It is usually a low-level vibration with high frequency, 50 – 350 Hz, depending on the rotational speed and the number of bit teeth. Typical environment for this vibration mode is when using PDC bits in high comprehensive strength formations when the bit teeth have lost its shearing cutting action and the individual cutters are impacted on the formation. This results in cutter damage and high frequency vibrations as well as a bit dysfunction that may cause bit whirl.

2.5.5. Lateral Vibrations

Lateral vibrations are recognized as the leading cause of drill-string and bottom hole assembly (BHA) failures, and the most destructive type of vibration creating large shocks as the BHA impacts with the wellbore. There exist three mechanism of lateral vibration:

1. Bit Whirl

Lateral vibrations or walk of the bit is the eccentric rotation of the bit about a point that is not its geometric center. Two major factors are identified leading to drill bit whirl. It could be due to either: **(a)** a slight bend in the drill collars due to high lateral vibrations, or **(b)** an unbalanced drill bit. An in homogenous borehole could aggravate the conditions of whirling. As a result the bit cut's itself a hole larger than its own diameter and is there by allowed to walk around the hole, opposed to be rotating around its natural center. This vibration mode can't be seen on surface since the lateral vibrations are dampened throughout the string before it reaches the top. Bit whirl can be seen when excessive side-cutting bits have been used or when encountering soft and unconsolidated formations. The primary consequence of bit whirl is the damage it causes to the bit cutting structure. During the whirling motion the bit cutters are moving faster and are subjected to high impact loads. The high loads cause the cutters to chip thereby making the wear from abrasion and heat more prominent. The over gauge hole created by the bit whirling facilitate a down hole conditions that easily may cause the onset of BHA whirl.

2. Backward & Forward BHA Whirl

Similar to the bit Whirl, BHA whirl is the eccentric rotation of the BHA about a point other than its geometric center. The motion of the BHA is the same as described for bit whirl with both forward and backward whirling motions occurring.

Forward whirl: a drilling dysfunction during which the bit instantaneous center of rotation rotates forwards around the hole axis. Or in other words, it's the rotation of a deflected drill collar section around the borehole axis in the same direction as it rotates around its axis.

Backward whirl: a drilling dysfunction during which the bit instantaneous center of rotation rotates backwards around the hole axis. Or in other words, it's the rolling motion of the drill collar or the stabilizer over the borehole wall in opposite direction as it rotates around its axis.

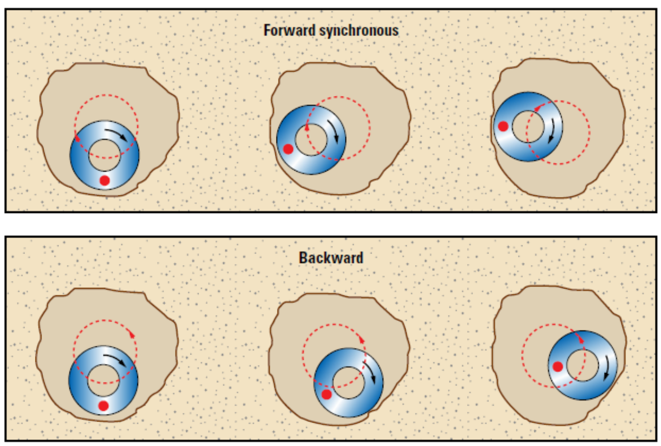


Figure 14 Forward and Backward Whirl (Dunayevsky, 1993)

BHA whirl is a complex motion of the BHA generating lateral displacements, shocks and increased friction against the well bore wall. BHA whirl is onset as a consequence of bit whirl, rotation of a drill-string in imbalance, or by the lateral movement induced from bit bounce. The consequences BHA whirl can be summarized into the following:

- MWD / directional equipment failure.
- Localized tool joint and stabilizer wear.
- Washouts.
- Twist-offs due to fatigue cracks of connection(s).

3. Parametric Resonance

Parametric resonance is severe lateral vibrations caused by the dynamic component of axial load. The dynamic component is primarily caused by bit-formation interactions and results in WOB fluctuations (Dunayevsky, 1993). These fluctuations generate a mechanical instability that is evident through rapidly growing lateral vibrations at a specific frequency. As an analog the mechanism behaves the same way as you would see if you induce a snakelike motion in the end of a hanging rope by moving the end up and down at a specific frequency. This may generate severe lateral vibrations than may accelerate drill-string failure or create the opportunity of hole enlargement which in turn may cause poor directional control and onset of whirl. When Parametric Resonance is encountered, it is typically in relation with interbedded formations and under gauge holes.

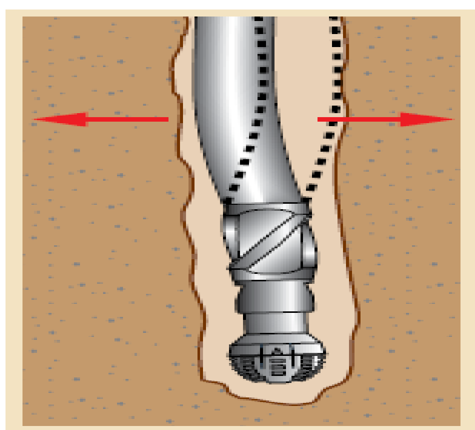


Figure 15: Parametric Resonance (Dunayevsky, 1993)

2.5.6. Modal Coupling

Modal coupling is when all the modes: axial, torsional and lateral are coupled together and there are vibrations in all three directions simultaneously. This is the most severe mode of vibration. It generates axial and torsional oscillations along the BHA and large lateral shocks. It is usually onset due to the failure of controlling one of the vibration modes thereby allowing it to initiate one or more other mechanisms simultaneously. Environments where stick-slip, whirl or bounce can be initiated are typical modal coupling environments and consequences are typically:

Measurements While Drilling (MWD) component failures such as motor/3D RSS, M/LWD tool, localized tool joint and/or stabilizer wear, washout or twist-offs due to connection fatigue cracks and increased average torque.

2.5.7. Stick-slip Whirl Interaction in Drill-string Dynamics

1. **Down-hole Measurement**

In the late 1980s the Institut Franc¸ais du Pe´trole designed the Trafor system. Trafor system is basically a research tool to measure downhole and surface data to improve knowledge about drillstring dynamics. It consists of:

- **Down-hole Measurement Device**

Called the Te´le´vigile, The Te´le´vigile is basically a tube much like a normal drill collar, but equipped with sensors that measure Weight On Bit, down-hole torque, down-hole accelerations in three orthogonal directions and down-hole bending moments in two directions. Three magnetic field sensors, known as magnetometers, measure a projection of the earth magnetic field in three orthogonal directions co-rotating with the Te´le´vigile.

- **Surface Measurement Device Known as the Survigile**

The signals of the Te´le´vigile and Survigile are gathered by a computer and synchronized. The advantage of the Trafor system is all about having the ability to measure both down-hole and surface data at real-time.

2. **Stick-Slip Whirl interaction**

The measurements reported of stick-slip whirl interaction were recorded at a full-scale research rig. The figure shows the downhole angular velocity, calculated from the magnetometer signals. The angular velocity at the surface, WOB and other parameters were kept constant during the experiment.

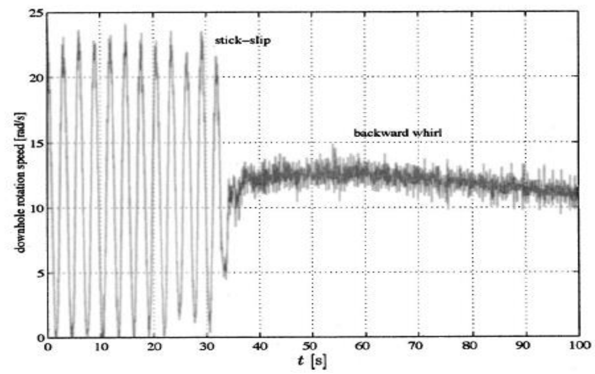


Figure 16: Measured Downhole Angular Velocity versus Time (CAMPEN, -)

Figure16 clearly shows that the drill-string undergoes stick-slip motion in the interval t<35, and this motion suddenly disappears and backward whirl is initiated.The friction curve of the part of the BHA beneath the Te´le´vigile, relating the torque to the down-hole angular velocity, could be reconstructed from the measurements as shown in figure (17).

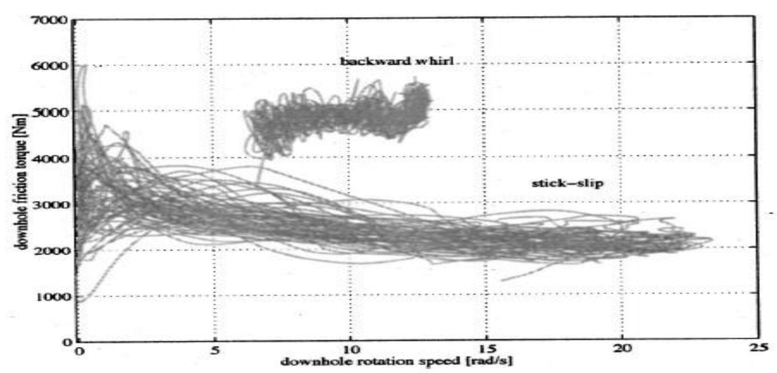


Figure 17:Measured Downhole Friction Curve (CAMPEN, -)

The torque on the Te´le´vigile consists of:

- The friction torque of the bit.
- Torque created by contact of the drill collar beneath the Te´le´vigile with the borehole wall.
- Viscous torque of the drilling mud.

During the stick-slip motion the part of the friction curve is traversed with the negative slope causing the steady rotation of the drill-string to be unstable, resulting in the stick-slip motion.

At transition to whirl, a switch to another part of the friction curve with a higher friction values with positive slope is made. Knowing that drill-string is not deflected in lateral direction during stick-slip motion. Consequently, the torque on the Te´le´vigile is during stick-slip motion is mainly due to the friction torque on the bit. The whirl motion has been identified as being backward whirl caused by rolling of a drill collar section over the borehole wall. The torque on the Te´le´vigile will be higher during whirl motion due to the additional torque created by the contact between drill collar and borehole wall. The slightly positive slope causes the constant rotation to be stable which prevents stick-slip motion as is shown in figure18. (Pavone, 1994) (Campen, -).

Figure18 shows the mean bending moment measured by the strain gauges in the Te´le´vigile versus the angular velocity at the surface.

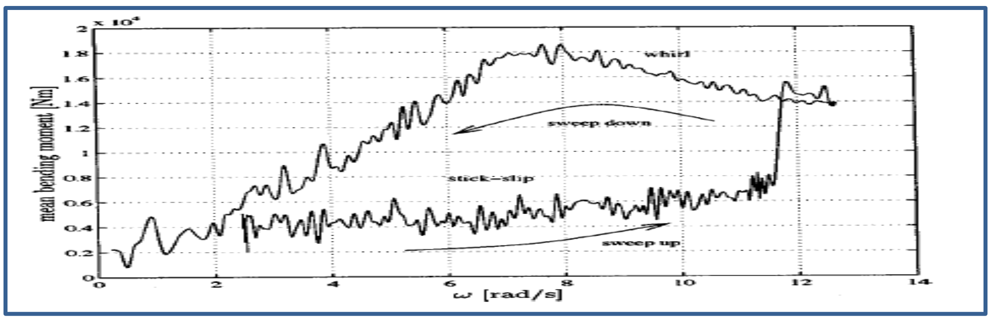


Figure 18: Measured Downhole Bending Moment versus Surface (CAMPEN, -)

The value of (w) was varied with a sweep-up followed by a sweep-down. The mean bending moment can be said to be a measure for the lateral displacement of the drill-string. The system was first in stick-slip motion with a low value of the mean bending moment. Then, as a consequence of increasing (w), motion switches from stick-slip to whirl and the mean bending moment jumps to a higher value, indicating a large radial deflection of the drill-string. Decreasing (w), the drill-string remains in whirl motion. It can be observed that the bending moment undergoes in three stages:

- Positive slope 2 < w < 7: when the drill-string is not in contact with the bore-wall.
- Decreasing slope w > 7: larger part of the drill-string will become in contact with the borehole wall which decreases the bending moment.

As a conclusion, combined stick-slip whirl motion is not observed.

2.6. Using Finite Element Method to Estimate the Fatigue Life

In micro-level of a structure, drill string in our case, two things have to be considered to estimate the fatigue life, crack initiation and crack propagation. However, the crack initiation is eliminated in studying the fatigue problem since cracks are always pre-existed in structure due to defects because of manufacturing method, corrosion, or environmental conditions.

Finite element analysis is an excellent tool in the modeling and analysis of cracks propagations to estimate the fatigue life. It uses a finite element over the structure intersected by the crack. The degree of freedom for each node in an element is determined to simulate the propagation of the crack. The defects are included in the numerical model without modifying the mesh discretization. However, where the stress concentration is expected to be high, the mesh has to be very fine to get more accurate results.

There are three fatigue life assessment methods that consider the numerical method such as FEM, nominal stress, hot spot stress and the effective notch spot methods. The first method is global while other methods are local. In local method which is based on the local failure approaches are highly sensitive to finite element modeling and mesh, since the stresses are often in an area of high stress concentration. Therefore, the global method is used for there is no stress concentration factor (ex. away from the welded region). The stress parameters used in the fatigue assessment methods are presented in Figure 19.

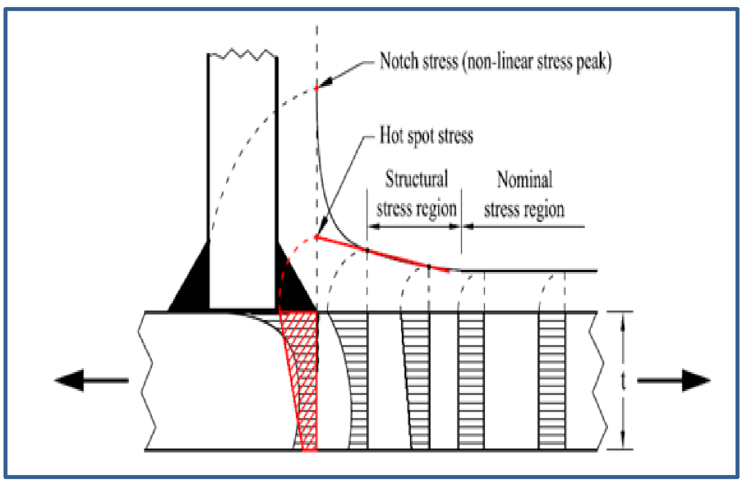


Figure 19: Stress Distribution through Plate Thickness and Along the Surface Close to the Weld (Mustafa, 2012)

2.6.1. Nominal Stress Assessment Method

The simplest way to estimate the fatigue life using FEM is nominal stress method because it is a global method that is not considering the stress concentration. Therefore, it only used in the regions away from stress concentration, figure 20. Thus, using nominal stress method in high stress concentration region will disregard the effects of the welds and critical regions and complex shapes.

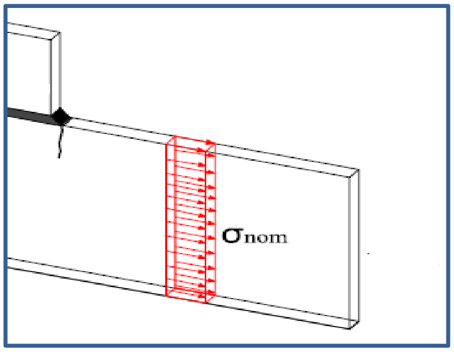


Figure 20: Nominal Stress Definition (Mustafa, 2012)

2.6.2 Hot Spot Stress Assessment Method

The hot spot stress method is used to evaluate the fatigue life of region where there is high stress concentration factor and more complexity, figure 21. The major advantage of the hot spot stress method is that effects of high stress concentration are taken into considerations in the fatigue stress calculations. It also applicable for this material which has reduced number of S–N curves. However, this method needs more effort in FEM modeling.

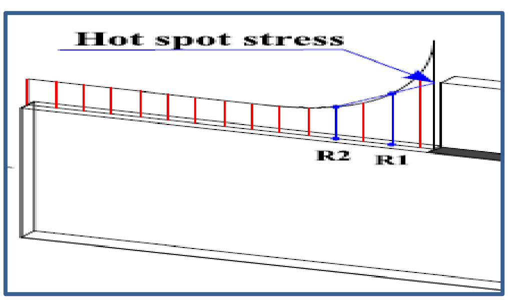


Figure 21: Hot Spot Stress Definition (Mustafa, 2012)

2.6.3 Finite Element Analysis of Drill String Lateral Vibration

When drill string reaches near the natural frequencies, severe vibrations will appear for the drill string. To solve the problem and avoid the resonance of the drill string, the analysis should be in the drill string to find its natural frequencies and modal shapes, and then by changing the drilling parameters. Therefore, advanced numerical methods like FEM are used for drill string static and dynamic of the great complexity of the problem. Also the connections in drill string have a high stress concentration that is needed to be considered carefully, the best way is to use a finite element method.

2.7. Existing Setup

There are many setups that introduce different types of failures in drill string. However, they are limited only to a specific type of failure. In our project different type of stresses are exposed to the drill- string including torsion and bending. Although, there is no existing setup that produces similar modes of failure, starting with similar setup would be a good start for designing our setup. Italian Pisa University's setup is taken as the reference for our design.

2.7.1. Pisa University's Setup

The principle in this setup is to use a concentric mass in order to generate a vibration in the specimen without rotating it as shown in figure 22. The vibration generates a fluctuating bending stress that leads to fatigue. They concluded that the specimen fatigue life is about 10^7 cycles for moderate stresses. The physical and functional decomposition of the setup is explained in section 3.3 in more detailed. Some of mechanism used in this setup will be used in our design.

Figure 22: Concentric (Rotating Mass) (Santus, 2012)

Chapter 3.0: Conceptual Design and Evaluation

This chapter presents the first stage of the procedure of designing of an experimental setup to test drilling pipes for investigation of the fatigue failure. The concepts are generated using morphological method and evaluated using Pugh chart. However, the physical and functional decompositions are done in advanced of the design process to get better understanding about the setup and, consequently, to get better design.

3.1. Design Methodology

The design methodology used here is taken from the Engineering Design book (E. Dieter). The main design stages are conceptual design, embodiment design and then detailed design as shown in figure 23.

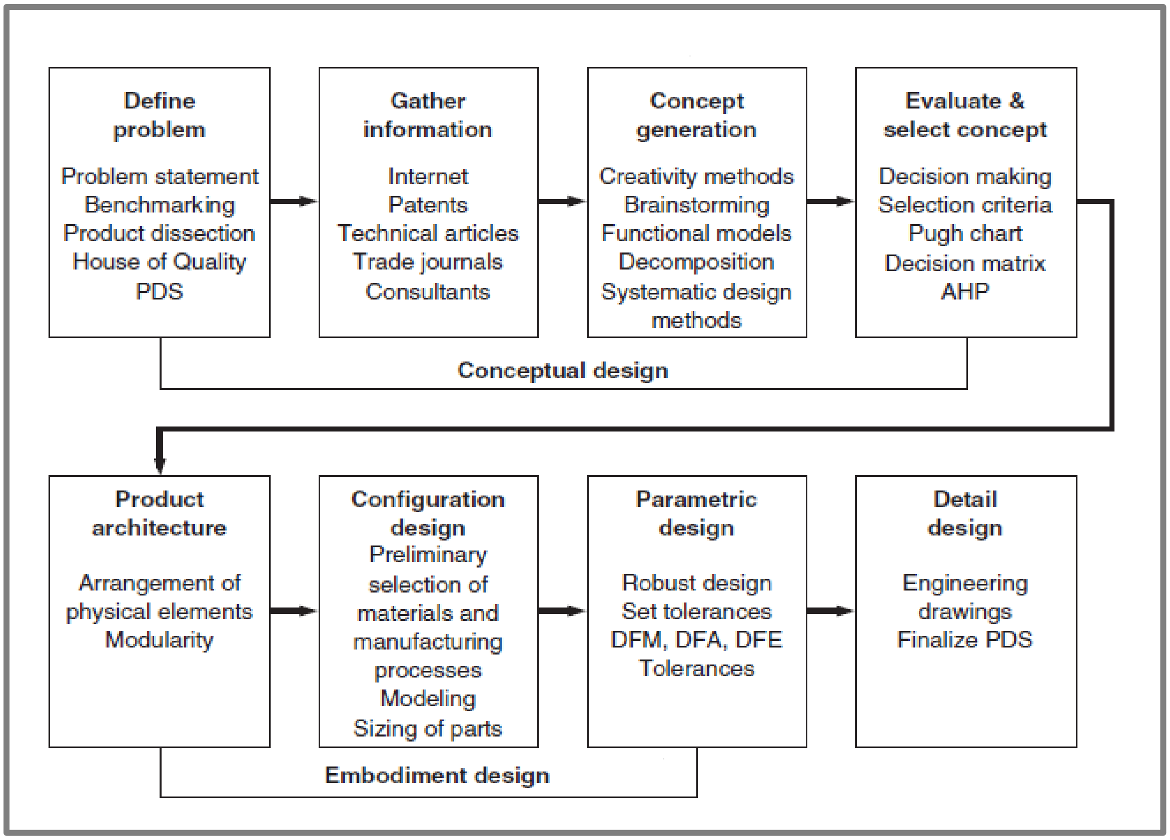


Figure 23: Design Process Stages (Dieter, 2009)

3.2. Realistic Constraints and Design Criteria

Determinacy of constraints and design criteria prior to the design procedure is very important in order to have a good design's making-decision. Since the setup has significant changes compared to other similar designs, realistic constraints and design criteria should be defined properly to avoid any bad decisions and wasting of time and resources.

3.2.1. Realistic Constrains.

Table 2: Realistic Constraints of the Design

Category	Constraints
Performance	• The setup should be able to perform continuously for few days without any interruptions. • The setup should be able to generate self-excited lateral vibration.

Economic	• The budget should not exceed 2,400 OMR and the scale of the setup should be determined accordingly. • The cost of test specimens and maintenance cost of the main components of the setup should be taken into account. • If this project succeeds, it will leave an impact on the local economy as well it will provide solutions or will reduce drill-string problems.
Environmental	• No toxic materials are used. • Recyclable materials are used for testing (metallic materials specimen). • Production and operation of the setup is totally clean.
Conformity to Standards	• The components (bolts, pipes, rivets) of the setup should be designed according to the standards followed in the local or international market.
Safety	• The rotating specimen should be held perfectly at both ends to avoid any catastrophic potential failure due to self-excited vibration. • Supporting components are perfectly connected to the ground since the pipe could be highly vibrated. • Availability of labeling and safety warnings while test is running. The allowable distance from the setup will be labeled. • The setup is operated from outside. • SQU workshops and laboratory safety instructions should be followed. • The pipe specimen should be as close as possible to the ground.
Environmental	• Since that the device is to test vibrational modes that lead to material fatigue [not thermal or other types of fatigue], operating conditions are kept constant and atmospheric.
Time and Space	• The project should be done in two semesters • The size of the setup is limited to the size of the room allocated for the project.

3.2.2. Design Criteria:

Table 3: Design Criteria

Category	Criteria
Performance	• Operating RPM depends on the size and the critical speeds of the of pipes. RPM considered 50% less than the critical speed. • The input torque is ranging from 50-100 N.m. • The resistance torque should be 30 to 50 % of the input torque.
Geometry	• The specimen is not to be less than 4 m and more than 6 m. • The range of diameter of the pipe specimen is 0.5 to 4 inches.
Material	• Pipes material is mild steel as its properties similar to the real drilling pipes and they are available locally
Operating Environment	• The noise level should be as low as possible. • Operating at room temperature and atmospheric pressure.

3.3. Physical and Functional Decomposition

In order to generate concepts for fatigue failure test, it is better to have a good visualization about the mechanisms that operate the test. Although, there is no existing mechanism that could accomplish the ultimate objective which is investigating the fatigue failure in drill-string, it is better to start with similar mechanism that achieves similar results. Resonant rig test mechanism (section 2.7.1) was the base which was used to generate physical and functional decompositions for concept generation. Figure 24 shows the resonant rig test mechanism. Figures 25 and 26 show the Physical and functional decomposing of resonant rig test mechanism, respectively.

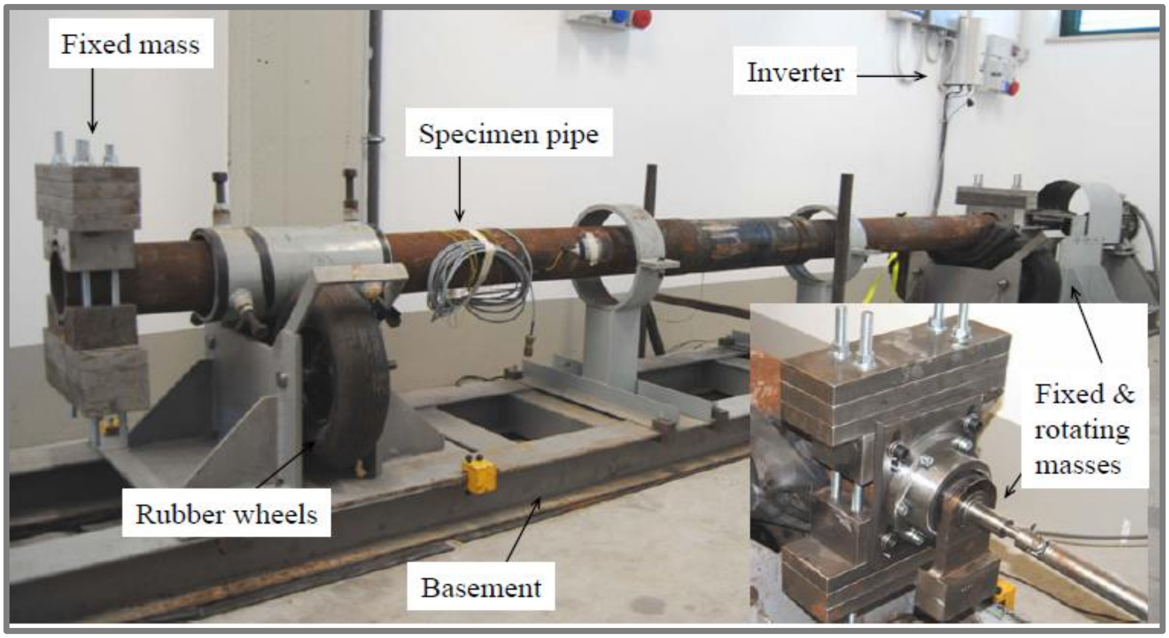


Figure 24: Resonant Rig Test Mechanism

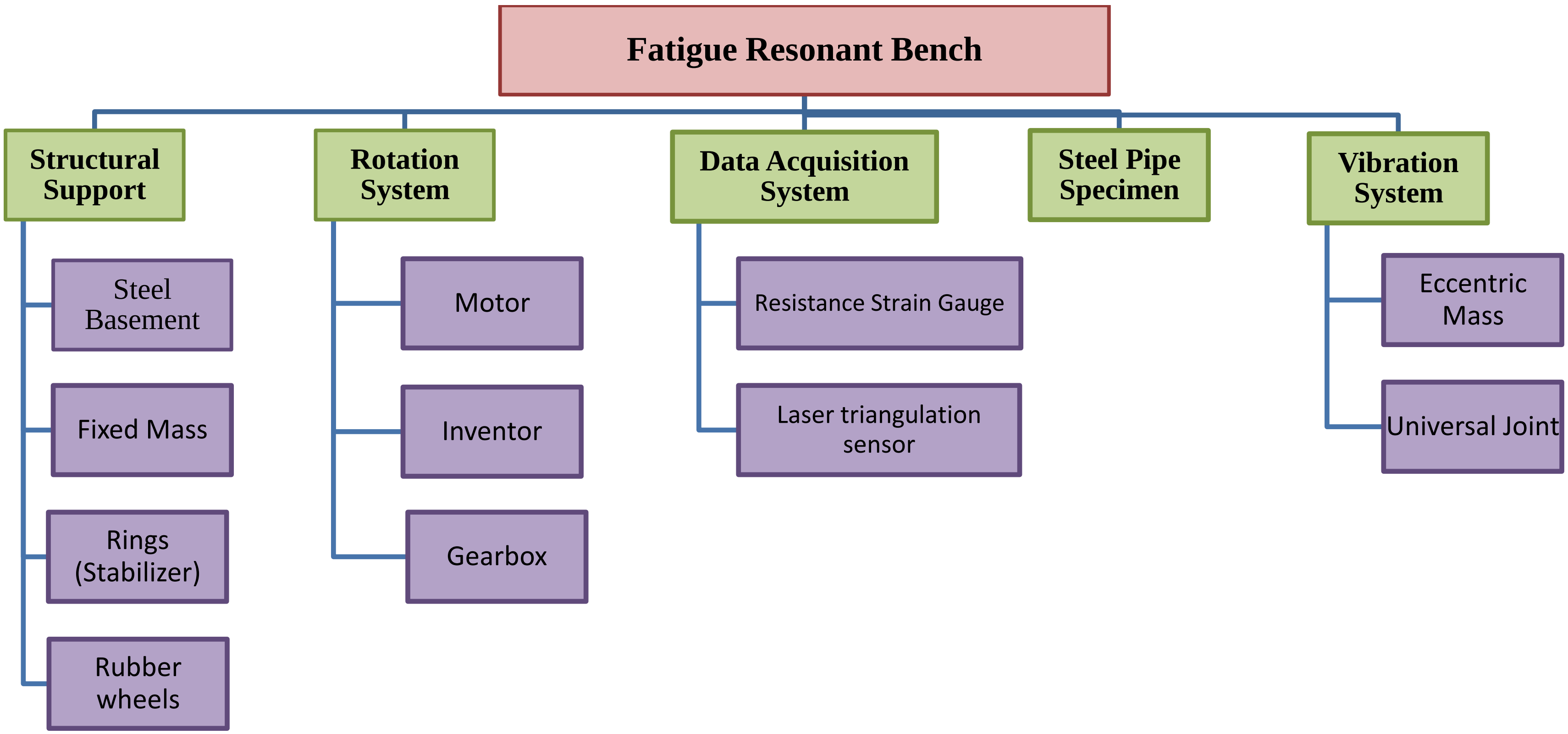


Figure 25: Physical Decomposition of Resonant Rig Test Mechanism

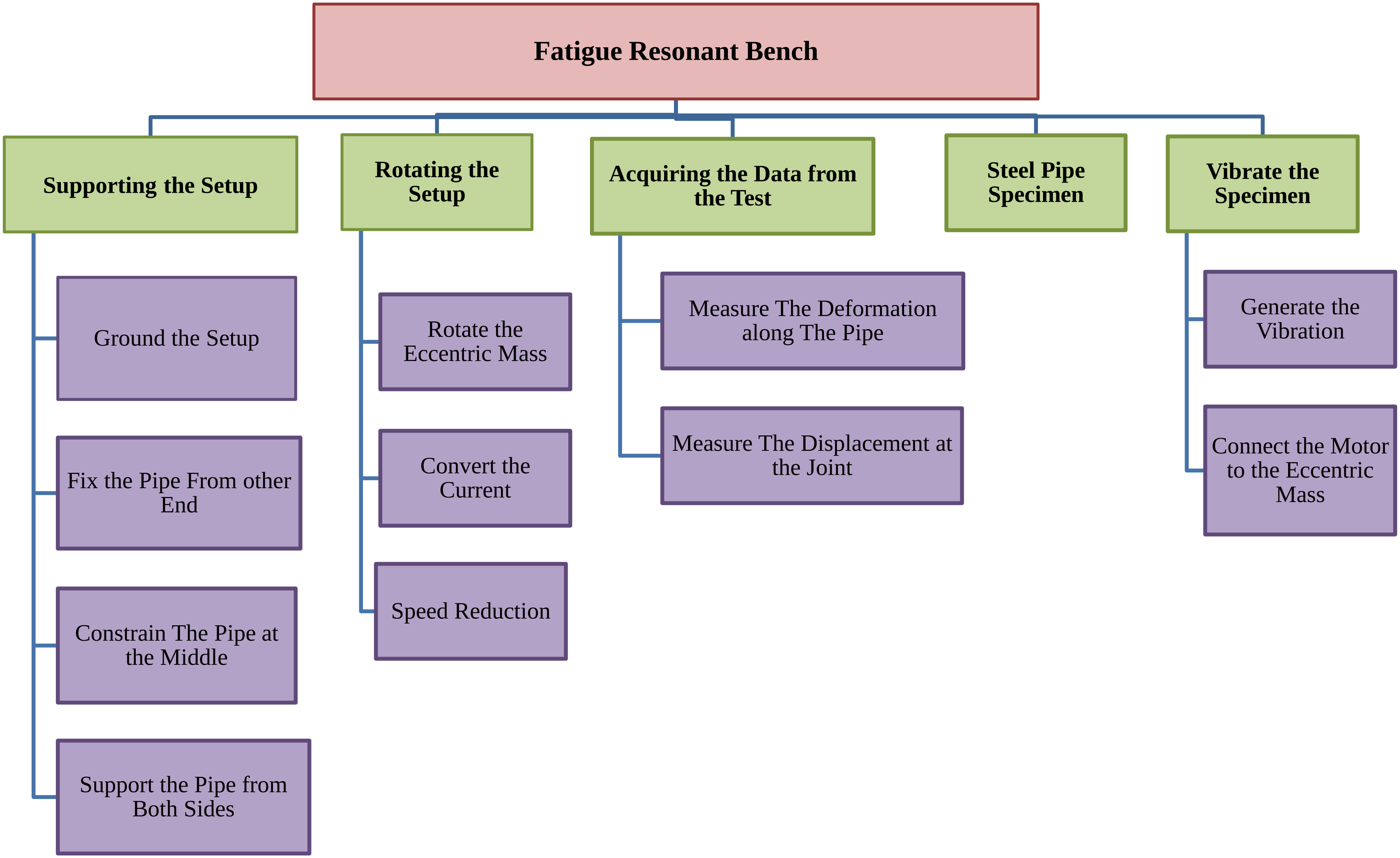


Figure 26: Functional Decomposition of Resonant Rig Test Mechanism

3.3.1. Difference Between Existing Setup and Required Setup

As mentioned before, the objective of resonant rig test mechanism does not match with the objective of required system which is testing the fatigue failure in drill-string. However, it has similar subsystems that could be considered as a base to start with. Therefore, the concepts will be generated based on the rig test mechanism but with significant changes. For example, the vibration system will be eliminated since the drill-string specimen in the new setup will have self-excited vibration. In addition, the resonant rig test uses fixed mass to support the specimen from the other end. However, in the required mechanism, chuck with bearings will be used for the same purpose. Since it is required to apply a resistance torque on the specimen to generate torsional fatigue, a braking system is added to the new setup, conversely to the existing setup which does not consider the torsional fatigue.

3.3.2. List of Future Features of the New Setup

The existing setup in the University of Pisa in Italy has some components that provide certain functions that are still needed in the new setup such as stability, Lateral vibration and data acquisition. However, new components must be added to provide the new requested functions such as rotation of the pipe and resistance to the rotation from the other end of the pipe specimen. Hence, a list of the improvements that are required to build up the new setup is shown below:

1. Safety against any sudden slip of the pipe from the ends (slipping from the chuck).
2. Rotation of the pipe with a high speed to accelerate the test.
3. Lateral vibration.
4. Low cost setup.
5. Robust setup (long life).
6. Easy to maintain.
7. Good simulation of the problem.
8. Resistance mechanism at the other end of the pipe specimen.
9. Tests are conducted at variety of rotational speeds and variety of resistance speeds.

3.4. Quality Function Deployment (QFD)

QFD is a powerful technique where it helps in relating the customer requirements and the engineering specifications and in ranking the higher priority of the customer requirements.

3.4.1. Identifying the Customer Needs

The customers first must be identified to gather the right information to design the fatigue failure setup. In this project the information is gathered from Prof. Jamil Abdo and Dr. Edris M. Hassan who are supervisors of the project and have the experience and knowledge about the drilling operation in oil and gas industries. The information that is related to manufacturing the setup is gathered from the ME workshop staff.

3.4.2. Customer Requirements

Based on the information that are given by Prof. Jamil Abdo and Dr. Edris M.Hassan, the customer requirements are listed as follow:

1. High rotational speed.
2. High input torque.
3. Effective resistance of the rotational motion. (to generate torsional fatigue)
4. Self-excited vibration.
5. Safety.
6. Easy to maintain and replace the specimen.
7. Easy to extract information from the setup.
8. Short time for testing one specimen. (due to limited time)
9. Low cost setup.
10. Not noisy.

3.4.3. Engineering Specifications

Table 2 shows how the customer requirements are converted into engineering specifications.

3.4.4. The House of Quality

In order to identify the most important engineering specifications that will be used in the next stage which is concept generation, QFD method is developed to determine the relative importance of the requirements that were established earlier. Four engineering specifications are selected based on their relative importance. According to QFD technique (see table 5) the most effective specifications are:

- Number of revolutions per minute.
- Power of motor.
- Reduction Percentage of input torque.
- Time to failure.

#	Customer Requirements	Engineering specifications	Units
1	High rotational speed	Number of revolutions per minute	Rpm
2	High Input torque	Power of motor	hp.
3	Effective resistance of the rotational motion.	Reduction Percentage of input torque	%
4	Self-excited vibration	Length of the pipe specimen	m
5	Safety	Elevation of the drilling pipe from the ground	m
		Thickness of supporting Structure	mm
6	Easy to maintain and replace the specimen	Less number of joints	#
7	Easy to extract information from the setup	Efficiency of data acquisition system	%
		Number of sensors	#
8	Short time for testing one specimen	Time to failure	hr
9	Low cost setup	Cost of manufacturing and operation	OMR
10	Not noisy	Noise level	dB

Table 4: Customer Requirements and Their Corresponding Engineering Specifications

3.5. Product Design Specifications (PDS)

PDS is the way where the designer gets more information about the product and starts to build in his/her mind how the product will ultimately appear and whether it will meet the intended functions or not. Basically, PDS is a list of the specifications and requirements that must exist in the designed product. However, in this project what is being designed is not a product but an experimental setup.

3.5.1. Project Title

Design and Fabrication of an experimental setup to investigate fatigue failure in drilling pipes.

Improvement Direction		↑	↑	↑	↑	↓	↑	↓	↑	↑	↓	↓	↓	Product meets requirement				
Units		rpm	Hp	rpm	m	m	mm	#	%	#	Hr	OM R.	dB	□ Resonant rig test mechanism. ★ Our Project. 1. Not at all, 2. Slightly, 3. Somewhat, 4. Mostly, 5.Completely				
Customer Requirements	Importance weight Factor	Number of revolutions per minute	Power of motor	Reduction in number of revolutions per minute	Length of the pipe specimen	Elevation of the drilling pipe from the ground	Thickness of supporting Structure	number of joints	High efficiency data acquisition system	Number of sensors	Time to failure	Cost of manufacturing and	Noise Level	1	2	3	4	5
High rotational speed	18	5	5	3		3					3		1	□				★
High Input torque	13	3	5	5							3			□			★	
Effective resistance of the rotational motion.	16	3	5	5	3			1			3	3		□			★	
Self-excited vibration	13	5	3	3	5	1		1			3		3		□		★	
Safety	5	3			3	5	5	5									□★	
Easy to maintain and replacing the specimen	9				5	5		5								□	★	
Easy to extract information from the setup	6								5	5		1					□★	
Short time for testing one specimen	7	5	3	5					3	1	5					□	★	
Low cost setup	9		3		3	1	5	1	5	3		5			□		★	
Not noisy	4	1	1	3			3						5		□	★		
Raw Sore		296	326	285	200	146	82	108	96	64	215	99	77	1994				
Relative Weight %		0.148	.163	.144	.10	.073	.041	.054	.048	.03 2	.108	.05	.039	1				
Rank Order		2	1	3	5	6	10	7	9	12	4	8	11					

Table 5: QFD Table

3.5.2. Setup Major Specifications

- Maximum rotational speed: 1400rpm.
- Maximum pipe length: 6m.
- Minimum pipe length: 4m.
- Motor power: 10hp (Might change according to the calculations of motor selection).
- The diameter of the pipe: 0.5 in, 1 in, 1.5 in, 2 in and 3 in.
- Resistance mechanism: 30%, 40% or 50 % of the input torque.
- Thickness of table metal sheets: 16 mm

3.5.3. Competition

The setup used here for testing the fatigue failure is considered to be an original and unique setup, since the other existing setups are meant to measure the effects of vibration, whirl or bending stress on the fatigue life of the specimen separately. However, the setup used here combines the effects of whirl and torsional stress on the fatigue life of drill-string. For example, the braking system which is used to generate torsional resistance is considered to be a new mechanism that is implemented to cause torsional fatigue.

3.5.4. Intended Market

The results of the experiments will be useful for oil and gas companies such as Petroleum Development Oman (PDO) and Oxidental.

3.5.5. Need for Product

Drilling is considered the most expensive operation in obtaining oil and gas as very advanced technologies used in addition to the high possibility of failure such as fatigue failure in drill-string. Therefore, this project is a great step in predicting the fatigue life in drill-string will result in minimizing the cost of the drilling operation.

3.5.6. Functional Performance

- Prediction of fatigue life of the drill-string under torsional and bending loads.
- Generate the fatigue problem due to lateral and torsional vibrations.
- Determination of the critical locations of fatigue crack.

3.5.7. Human Factors

- Stabilizers are used to restrict the whirl motion of drill-string and prevent any parts from flying.
- Chucks should be strong and big enough to hold the drill-string from both ends at same elevation.
- The height of drill-string from the ground should be as low as possible to be more safe.

3.6. Concept Generation

3.6.1. Gathering Information

The experimental setup that's being designed consists mainly of four mechanisms and systems. These mechanisms and systems are:

1. Rotation mechanism.
2. Rotation resistance mechanism.
3. Supporting mechanism (Mechanical and structural).
4. Instrumentation (speed measurement device).

Different suitable alternatives were generated for each function in a brain storming session and the most effective specifications mentioned in the QFD table were taken into account. Next, those alternatives were combined and final concepts sketches were drawn. In addition, a brief explanation along with the advantages and disadvantages for each concept was written to help in choosing the best concept. Finally, concepts were evaluated using the weighted decision matrix.

3.6.2. Morphological method

Morphological method is a method for generating different possible solutions by combining every function solution together for every possible outcome. This method is implemented in this project.

3.6.3. Basic Decision Matrix

The weighted decision matrix method is used to evaluate the available alternatives and choose the best of them among the others. This method was applied in this project and in order to use this method the following steps must be followed:

- Rank and assign weights to the criteria.
- Select a reference alternative (Datum).
- Evaluate each alternative using a scale of (1-5).
- Multiply the weights and the scales.
- Choose the highest score.

3.6.4. Concept Generation for the setup:

1. Rotation Mechanism

The main device used to basically provide rotation is a motor. However, the motor alone cannot provide the desired mechanism where it should come with different components like supports, gearbox, pulleys and belt or all of these components can be found in one device. Therefore, many alternatives were available to provide such a mechanism.

I. Lathe Machine

<table>
<tr>
<td>
Figure 27: Lathe Machine</td>
<td>Description:
Lathe machine can be a very good option to rotate the pipe although; it is mainly used in the industry to make threads on work pieces. The big advantage in this option is that all the components to make a rotation mechanism are in one device with a strong and heavy structural support and housing.</td>
</tr>
</table>

II. Milling Machine

<table>
<tr>
<td>
Figure 28: Milling Machine</td>
<td>Description:
Milling machine is used to remove material from a work piece surface. However, this machine has a very strong motor mounted and supported in a heavy duty structure, and can rotate a pipe horizontally by removing some of its parts.</td>
</tr>
</table>

III. Motor and Pulley Assembly

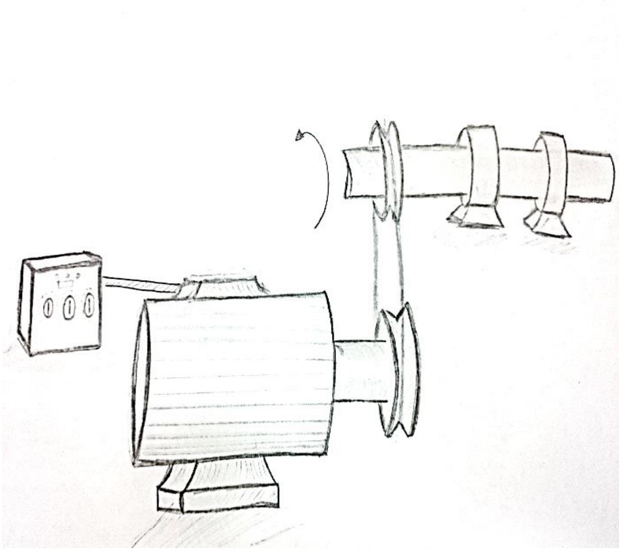


Figure 29: Motor and Pulley Assembly

Description:

This mechanism is not really a new one, where the previous options have it inside them but here it is alone without supports and housing.

IV. Motor and gearbox assembly

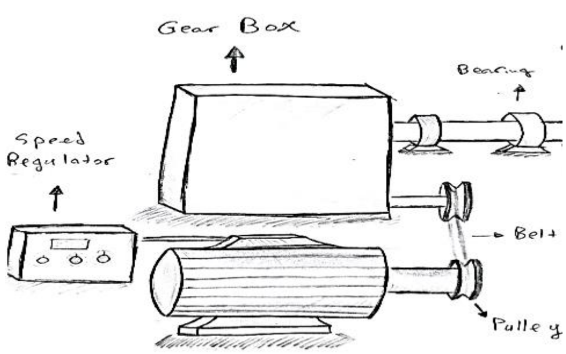


Figure 30: Motor and Gearbox Assembly

Description:

This mechanism is not really a new one, where the previous options (I, II) have it inside them but here it is alone without supports and housing. However, it has a gearbox.

2. Supporting Mechanisms

The supporting mechanisms are divided into two categories:

A. Mechanical support.

B. Structural support.

- **Mechanical Support**

I. Bearings

The bearings will be basically housed in Pillow block which is a special housing designed for the bearings as shown in figure 31. In application wise, it is usually used in a cleaner environment and for lesser load in general. Here, only two types of bearing are compared cylindrical roller bearing and ball bearing.

Cylindrical Bearing	Ball Bearing

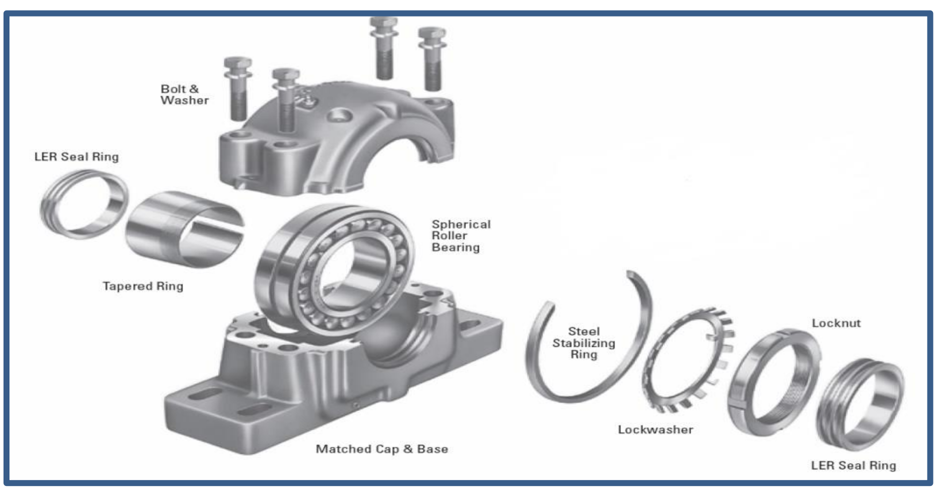


Figure 31: Cylindrical Bearing

It is one type of bearings that handles both, thrust and radial loads. This type of bearings consists of four main components, These components are:

1- Outer ring (cup).
2- Inner ring (cone).
3- Rollers.
4- Cage or the roller retainer.

This type of bearings can handle more load in radial direction.

Figure 32: Ball Bearing

This type of bearing can support both axial and radial loads. Where this type of bearing contains two races (inner and outer races). Usually one of the races is stationary and the other one is rotating with the bearing rotary part. Since the contact surface of the ball is small, the friction factor is too small and the load capacity of this type of bearing is also small.

Figure 33: Bearing Housing Assembly

- **Structural Support**

I. Table/ Box

Heavy duty table or box is required to support the mechanism which will support chuck, bearings and motor. Vibrations will be highly induced as the length and the weight of the pipe are increased proportionally with rotational speed. Therefore, it is very important to have such a table and a box to provide high safety measures.

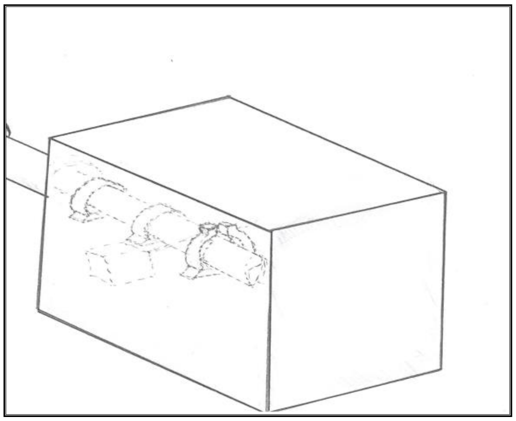
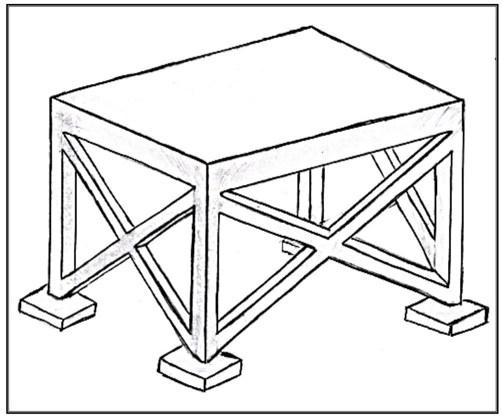
 

Figure 34: Box Support and Table Support

I. Stabilizer

Stabilizers are required to constrain the rotary motion and prevent any scattered pieces from flying away. As the rotational speed is increased the pipe will whirl more which can cause damage of the setup or danger to humans.

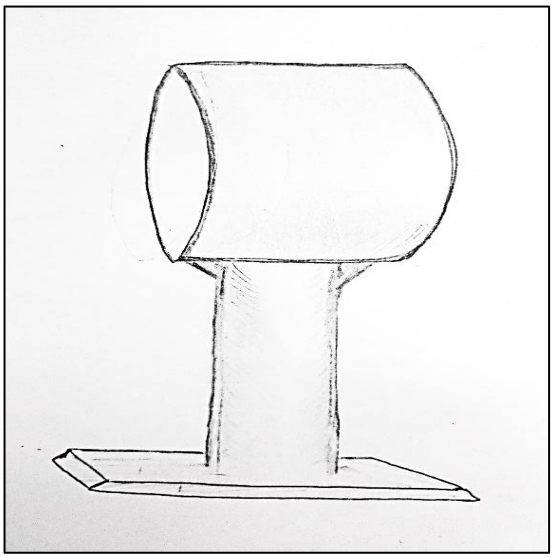


Figure 35: Stabilizer

II. Chucks

Three Jaws Chuck	Four Jaw Chuck

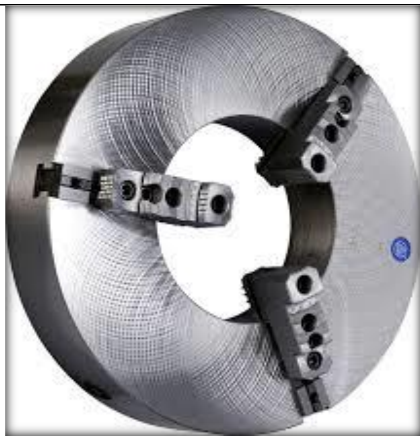


Figure 36: 3-jaw Chuck

It has a low centering accuracy because of the scroll wear and the tolerance buildup reduced by the moving part. In general, this one can be used for average machining accuracy with capability for high repeatable work.

Figure 37: 4-jaw Chuck

It has a higher centering accuracy than the 3 jaws chuck. Where it is slower and difficult for a repeatable work but in general it has a good machining accuracy.

3. Braking System

The system is intended to be used to resist the rotational movement the testing pipe. This basically simulates the resistivity of the formation to the drilling pipes. There is a various mechanism that can be implemented as a braking system and each one of them is used for a special purpose. Here only two braking systems are compared; car braking system and clutch clamps braking system.

Clutch Clamps	Car Braking System

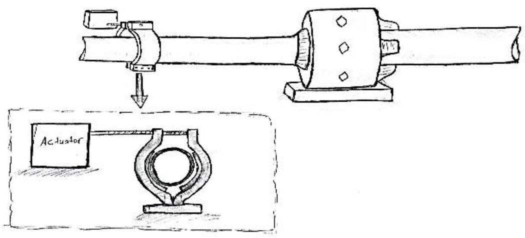


Figure 38: Clutch Clamps

Two clutches are surrounding the pipe specimen closed by linear actuators. The speed and power of actuator are set required amount to implement the required reduction in torsion force.

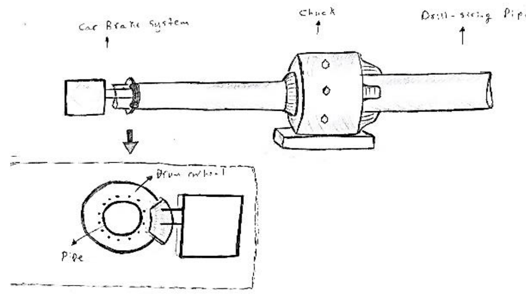


Figure 39: Car Braking System

Same as car brake mechanism where braking pads are used to reduce the speed of a flange that is attached to the pipe test specimen.

4. Instrumentations:

Instrumentations are required in this project to achieve the intended functions. The required instrumentations are speed measurement device and a variable speed controller. The speed measurement device is used to measure the rotational speed of the testing pipe and already existed which Tac-meter as shown on figure. Variable speed controller is required to control the speed of the motor and gives more options to carry out many tests with different speeds.

Figure 40: Tac-Meter

3.6.5. Final Concepts

Many concepts were generated from combining the different minor alternatives that were generated for each mechanism discussed before. However, only three concepts were chosen for evaluation after studying all alternatives and making go/no go decisions. These three concepts are:

Concept 1

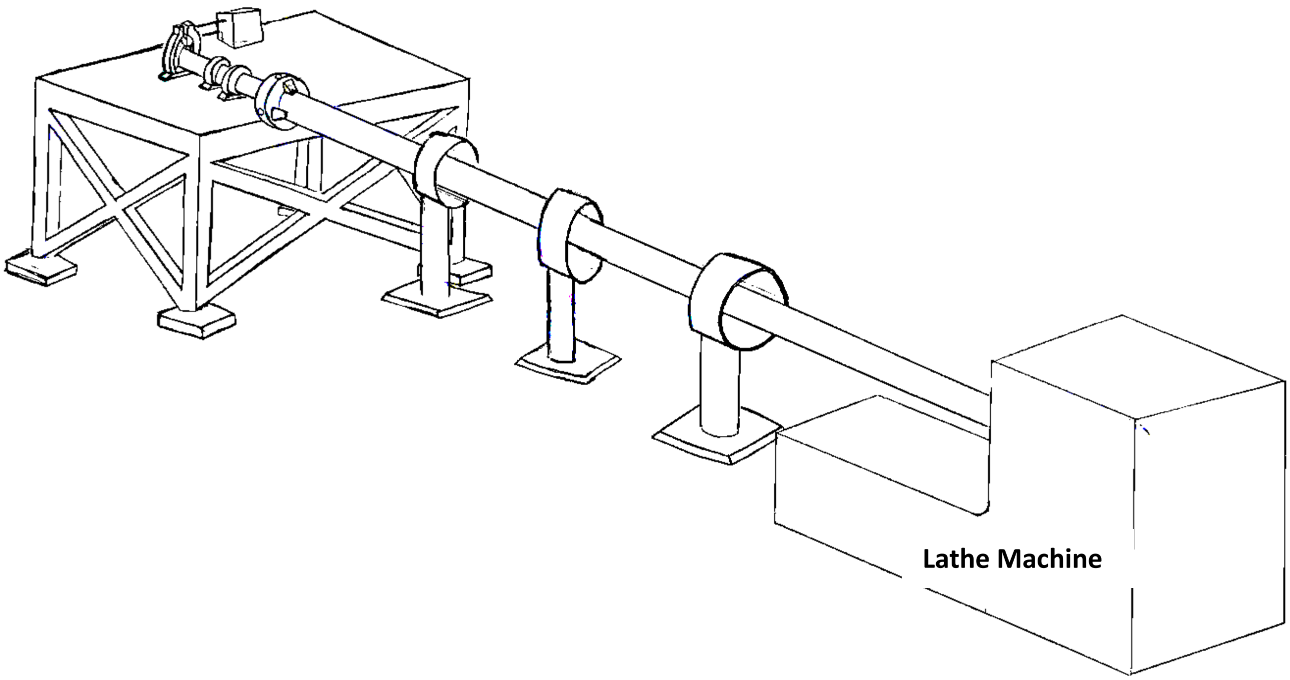
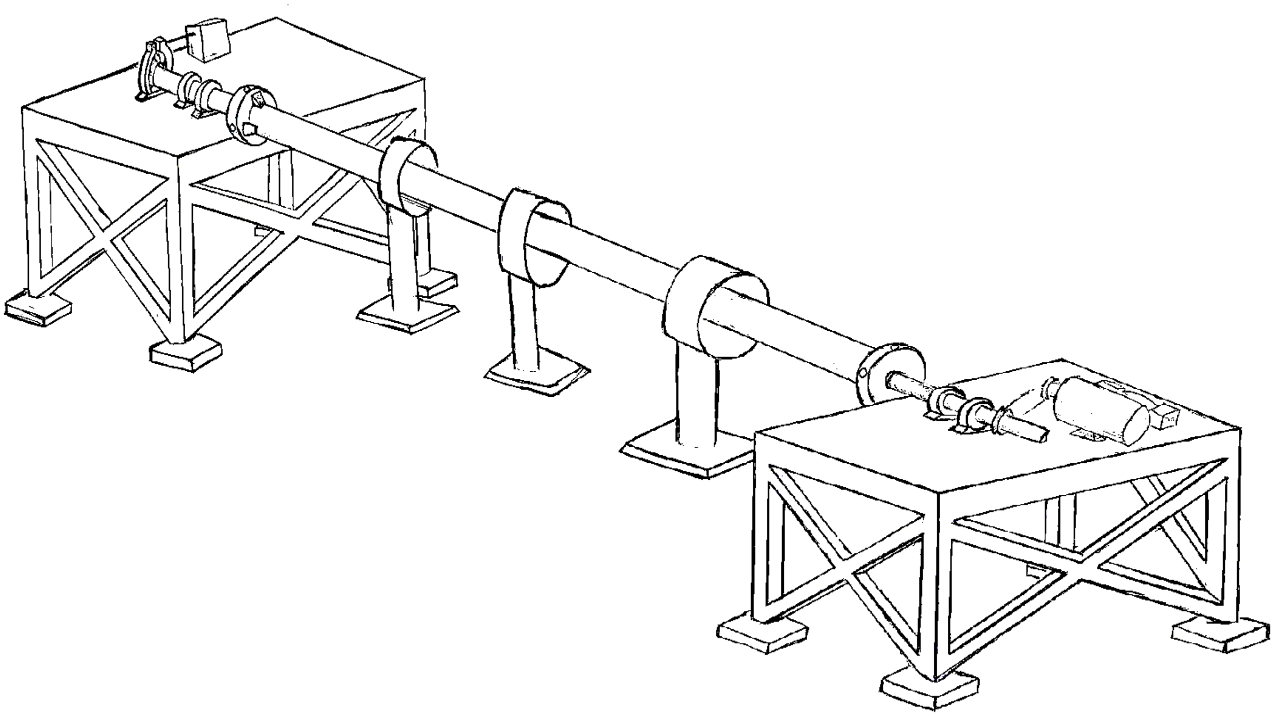
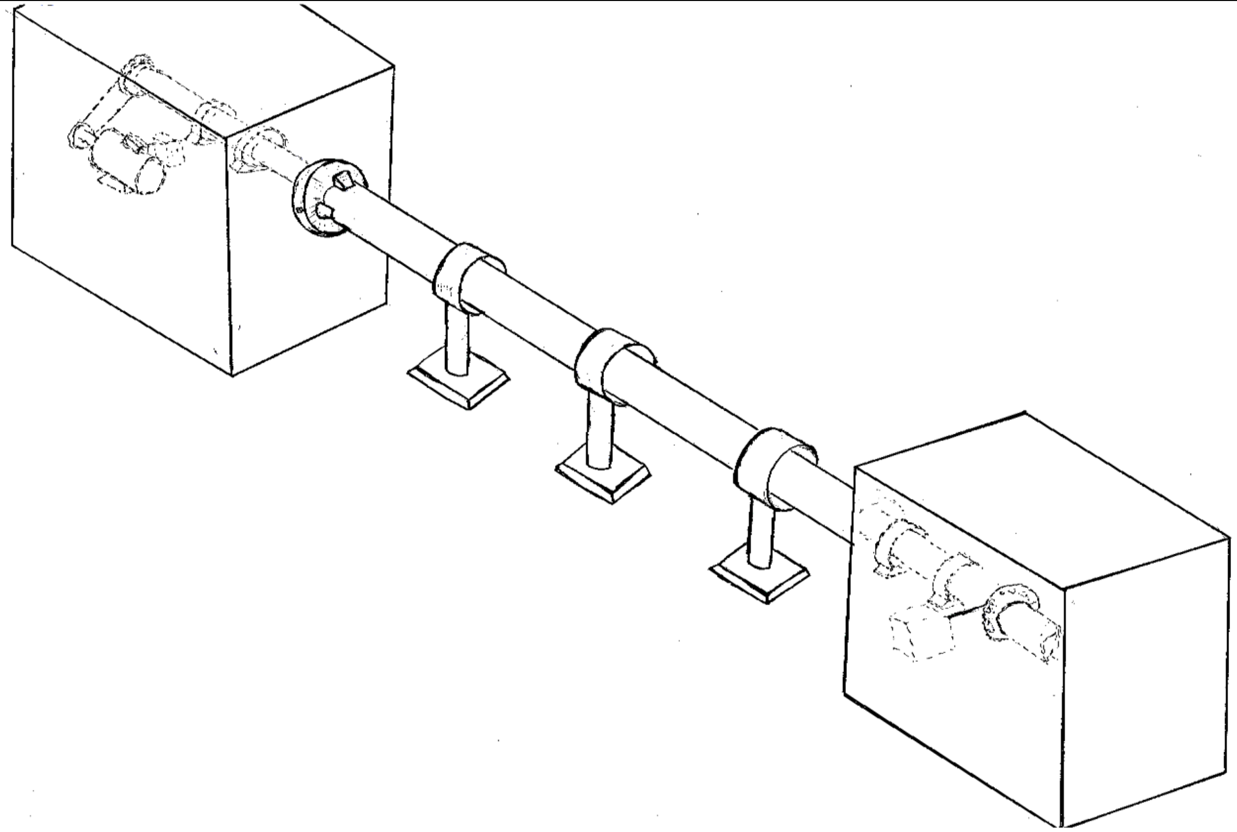


Figure 41: First Modification

Description:

- The rotating mechanism used here is lathing machine which is already has an excellent support mechanism.
- Support (heavy duty table, three stabilizers, 4-jaw Chuck, Ball Bearing).
- Braking mechanism is clutch Clamps mechanism with linear actuators.

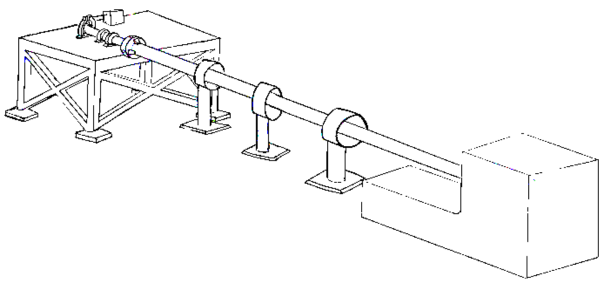
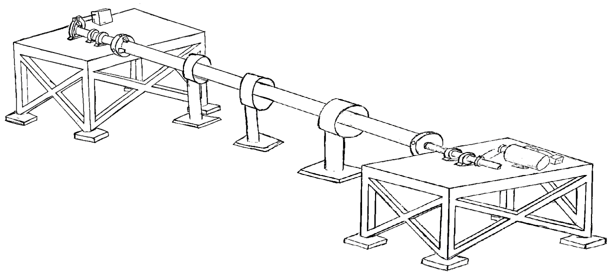
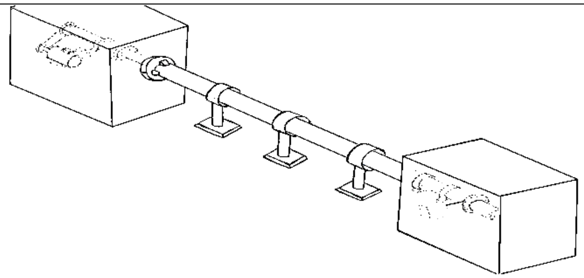
Concept 2
 **Figure 42: Second Modification**
Description: • This rotation mechanism uses a motor with a pulley mechanism and the motor is fixed on the table. • Supports used are: 2 heavy tables, 3 stabilizers, cylindrical Bearing and 3-jaw chuck. • Braking mechanism is clutch Clamps mechanism with linear actuators.

Concept 3
 **Figure 43: Third Modification**
Description: • This rotation mechanism uses a motor with a gearbox mechanism. The motor is fixed inside a steel box structure. • Supports used are: 2 box structures, 3 stabilizers, cylindrical Bearing, and 3-jaw chuck. • Braking mechanism is a car breaking mechanism.

3.7. Concepts Evaluation

3.7.1. Advantages and disadvantages of final concepts

Table 6: Advantages and Disadvantages of the Final Concepts

Alt.	Sketch	Advantages	Disadvantages
1		• Heavy support at the rotating side. • Safer relatively. • Less labor work required • Variable torque and speed. • Simple braking system. • High vibration induced.	• Lathe machine is expensive. • Other parts of lathe machine are not needed. • Elevation of the testing pipe is higher than 1m which's dangerous.
2		• Easy to assembly and manufacture supports. • Cheap relatively. • Simple braking system. • Low height. • High vibration induced.	• Many labors are required. • Heavy support is required.
3		• Low height. • High vibration induced. • inexpensive relatively.	• difficult to assembly and manufacture supports. • Complex connection required inside the box.

3.7.2. Evaluation Criteria

In this stage, the detailed evaluation phase is applied to all the three designs generated in the concepts generation stage. The basic evaluation techniques such as "go\no go" screen decision and feasibility technique have been applied by the group members in concept generation stage. For detailed evaluation Pugh chart is used. But before that, weights are needed to be assigned for each criterion (i.e. engineering specification) in QFD Table.

3.7.3. Weighted Criteria

As shown in table 7 the weights of each criterion have been determined after assessing relative importance of each criterion. The Reduction Percentage of input torque is found to have the highest weight which is 0.182.

Table 7: Weighted criteria

criteria	1	2	3	4	5	6	7	8	9	10	11	Row Total	Weighting Factor, w_i
Number of revolutions per minute	-	0	0	1	1	1	1	1	1	1	1	8	0.145
Power of motor	1	-	0	1	1	1	1	1	1	1	1	9	0.164
Reduction Percentage of input torque	1	1	-	1	1	1	1	1	1	1	1	10	0.182
Length of the pipe specimen	0	0	0	-	1	1	1	1	1	0	1	6	0.110
Elevation of the drilling pipe from the ground	0	0	0	0	-	0	1	0	0	0	0	1	0.018
Thickness of supporting Structure	0	0	0	0	1	-	1	1	1	0	1	5	0.091
Less number of joints	0	0	0	0	0	0	-	0	0	0	0	0	0
High efficiency data acquisition system	0	0	0	0	1	0	1	-	0	0	0	2	0.036
Number of sensors	0	0	0	0	1	0	1	1	-	0	0	3	0.054
Time to failure	0	0	0	1	1	1	1	1	1	-	1	7	.127
Cost of manufacturing and operation	0	0	0	0	1	0	1	1	1	0	0	4	.073
											Total	55	1

3.7.1. Selection of the Final Concept

After all criteria are determined and selected, the final concept is selected using Pugh Chart as shown in table 8. The rating is calculated using the following equation:

$$\boldsymbol{Sum} = \sum \boldsymbol{\beta_i}\, \boldsymbol{\omega_i} \quad \text{(8): Weighted Criteria Rating}$$

The second design has the highest rate of 9.782 score.

Table 8: Pugh Chart

Design criterion	Wi	Units	Alternatives								
			Concept 1			Concept 2			Concept 3		
			Magnitude	Score	Rating	Magnitude	Score	Rating	Magnitude	Score	Rating
Number of revolutions per minute	.145	rpm	1000	8	1.16	1400	10	1.45	1000	8	1.45
Power of motor	.164	hp	8	7	1.148	10	10	1.64	10	10	1.64
Reduction Percentage of input torque	.182	%	50	9	1.638	50	9	1.638	30	8	1.456
Length of the pipe specimen	.110	m	12	10	1.1	12	10	1.1	12	10	1.1
Elevation of the drilling pipe from the ground	.018	m	1.5	7	0.126	.5	10	0.18	.5	10	0.18
Thickness of supporting Structure	.091	mm	25	8	.728	50	10	.91	25	8	.728
High efficiency data acquisition system	.036	%	90	8	0.288	95	9	0.324	95	9	0.324
Number of sensors	.054	#	5	8	.432	8	10	.54	8	10	.54
Time to failure	.127	Hr.	72	8	1.016	48	10	1.27	72	8	1.016
Cost of manufacturing and operation	.073	OR	2600	7	.511	2300	10	.73	3400	5	.365
				Sum	8.147		sum	9.782	sum		8.799

Chapter 4.0: Embodiment and Detailed Design

Embodiment and detail design phases are important stages of the design process. Embodiment design includes main layouts and configurations. The actual sizes of the parts will be defined as well as the selection of the standard parts that will be used in the selected design. The next stage is the detail design which represents the types of materials, the cost analysis for the entire facility, the parts will be modeled and sizes will be decided. It requires a lot of skills. For example, tolerance, dimensioning, heat treating, welding, surface finish and many others.

4.1. Configuration Design

In this design phase all the parts that have been modeled are showed in the way that they are configured and fit to each other. A list with name of the parts is given below.

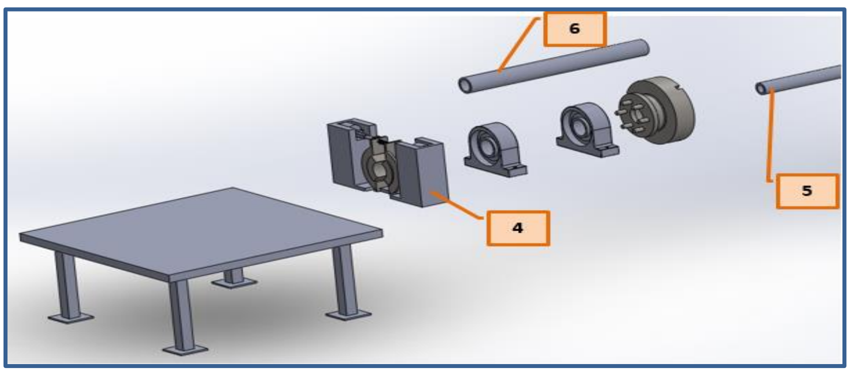


Figure 44: Design at Braking Mechanism Side

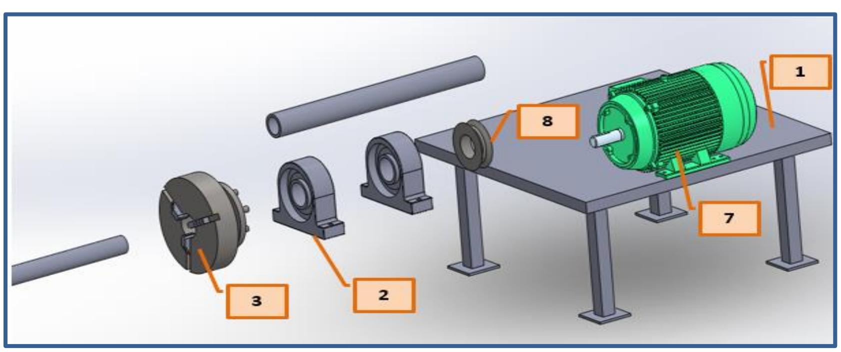


Figure 45: Design from Rotating Mechanism Side

1. Supporting table.
2. Cylindrical bearing.
3. Chuck and flange assembly.
4. Braking mechanism.
5. Drill-string specimen.
6. Housing pipe.
7. Motor.
8. Pulley.

4.2. Selection and Parametric Design

This section is considered to the most important section where all the decisions regarding to the design will be taken based on the calculations and assumption that are set in this section. Parametric design is about calculating or estimating the factors that will affect the performance of the setup or product being design. For example, the forces and loads acting upon a certain structure should be calculated to know how much load that structure can carry without failing.

In this project a lot of calculations were done to make decisions regarding to bearings and motor selections. Moreover, some calculations on the stability of the structural support of the setup were carried out to ensure that no failure will occur due to a failure in the structural support. Some of the selections were done based on experience or trial and error.

4.2.1. Bearing Selection

As mentioned previously in the conceptual design section, bearings are mechanical supporting components which are needed to support rotating shafts. Bearings which are mechanical components fail due to different reasons. An example is excessive load or working for a long period of time. Therefore, applying the parametric design to these components is crucial.

Assumptions:

- Housing pipe outer diameter is 4 in (101.6 mm) and the inner diameter is 3 inches (76.2 mm).
- The power of motor is 12 hp (8.9484 KW). (based on some experiments conducted in ME workshops)
- The maximum rotational speed is 1400 rpm.
- All components are perfect hallowing cylinders and have density of 7800 kg/m^3.

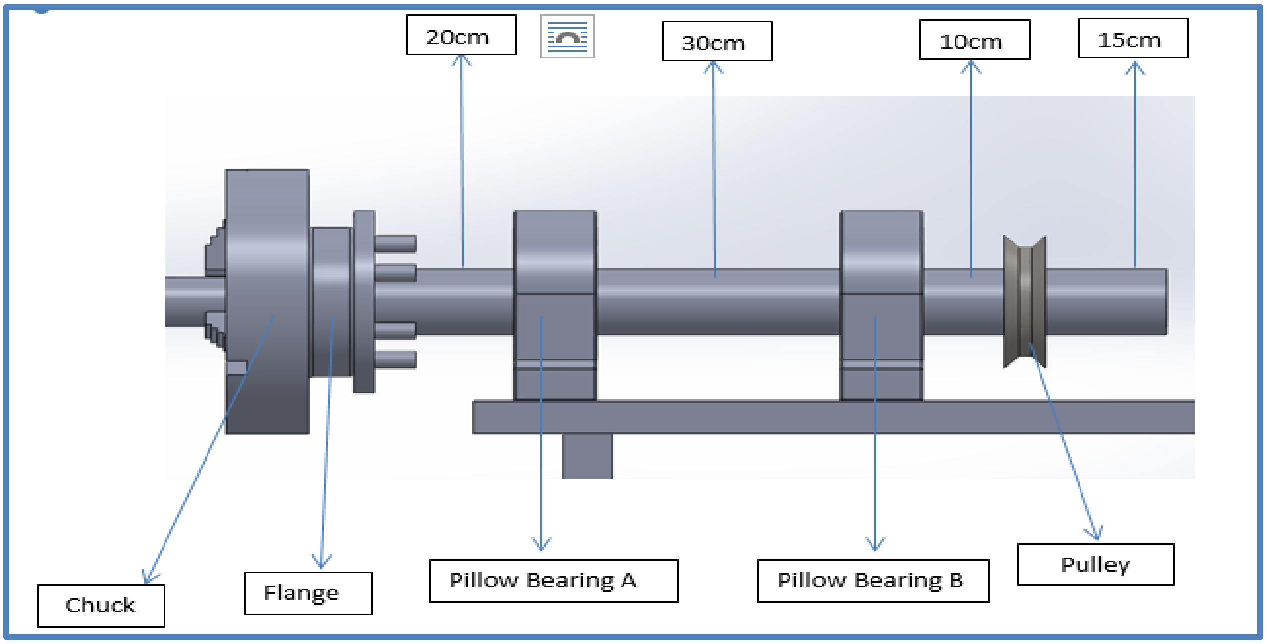


Figure 46: Side View of the Housing Pipe with Bearing, Chuck, Falnges and Pulley

First, it is required to calculate the mass of pulley, chuck, and coupling.

$$m_{pulley} = \rho V_{pulley} = 7800 * \left(\frac{\pi}{4}\right) * (0.1524^2 - 0.1016^2) * 0.0508 = 4.01\ kg$$

$$m_{chuck} = \rho V_{chuck} = 7800 * \left(\frac{\pi}{4}\right) * (0.4064^2 - 0.1016^2) * 0.1016 = 96.37\ kg$$

$$m_{coupling} = \rho V_{coupling} = 7800 * \left(\frac{\pi}{4}\right) * (0.2286^2 - 0.1016^2) * 0.0254 = 10.54\ kg$$

For further accuracy, the weights of these components which exist in the SQU mechanical workshop were measured to be:

$$m_{pulley} = 4\,kg \qquad W_{pulley} = m_{pulley} * g = 39.24\,N$$

$$m_{chuck} = 105kg \qquad W_{chuck} = m_{chuck} * g = 1030N$$

$$m_{coupling} = 10 \qquad W_{coupling} = m_{coupling} * g = 98.1N$$

It can be seen that the measured values are close to the calculated ones. The measured values will be taken into account in the coming calculations.

Given from the constraints, the maximum mild steel testing pipe has an outer diameter of 3in with a thickness of 1in and a total length of 12m. Thus the total testing pipe mass can be found as:

$$m_{testing,pipe} = \rho V_{testing,pipe} = 7800 * \left(\frac{\pi}{4}\right) * (0.0762^2 - 0.0508^2) * 12 = 237.14kg$$

$$W_{testing,pipe} = m_{testing,pipe} * g = 2326.34N$$

Considering that only $\frac{W}{2} = 1163.17N$ will effect on one chuck.

Given a housing pipe of a 1m length, 4in outer diameter and a thickness of 1in, its mass can be found as:

$$m_{housing,pipe} = \rho V_{housing,pipe} = 7800 * \left(\frac{\pi}{4}\right) * (0.1016^2 - 0.0762^2) * 1 = 27.66kg$$

$$W_{housing,pipe} = m_{housing,pipe} * g = 271.4$$

Dynamic Load Analysis:

Based on *NTN Paper* of bearing load calculation, the dynamic load on the bearing is calculated as follows:

For belt drives, an initial tension is applied to give sufficient constant operating tension on the belt and pulley. Therefore, the radial force equation is:

$$\boldsymbol{K_r = f_b * K_t} \quad \text{(9): radial force from motor}$$

where K_r is pulley radial load (N) and f_b is belt factor. K_t can be calculated as follows:

$$\mathbf{K_t = \frac{19.1e6*H}{D_p*n}} \quad \text{(10): Tangential load on Pulley}$$

where K_t is pulley tangential load (N), H is transmitted force (kW) , D_p is pulley pitch diameter (mm) and n is the rotational speed (min^{-1}).Therefore, according to the assumption the tangential load is:

$$K_t = \frac{19.1e6 * 8.9484}{101.6 * 8796.46} = 191.23\ N$$

and the radial force is:

$$K_r = 1.75 * 191.23 = 334.668\ N$$

Now, applying moment equation about bearing A and B to find the radial forces applied by each bearing:

$\sum M_A$ =(1163.17+1030)*0.25 – (271.4)*0.2 - (39.24+334.668)*0.575 - 0.4F_B + (98.1)*0.2625=0

F_B=762 N.

$\sum M_B$ =(1163.17+1030)*0.65 + (98.1)*0.6627 - 0.4F_A +(271.4)*0.2 – (39.24+334.668)*0.175=0

F_A=3698.5 N.

Since $F_A > F_B$. Therefore, calculations to find the bearings diameters are done considering F_A.

Now, calculating the bearing's load rating is required.

Since $F_A > F_B$. Therefore, calculations to find the bearings diameters are done considering F_A.

Now, calculating the bearing's load rating required.

$$\boldsymbol{C_{10} = F_A * \left[\frac{X_D}{X_o+(\theta-X_o)(1-R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}} \quad \textbf{(11):} \textit{ Load Rate}$$

$$\mathbf{X_D = \frac{n_D * L_D * 60}{10^6}}$$

For the objective of this setup, cylindrical roller bearing would be required to resist high radial load. Thus, the constants are:

$$X_o=0.02$$

$$(\theta - X_o) = 4.439$$

$$a = \frac{3}{10}$$

Given the constraints, n_D =1400 rpm, L_D=70,000 hr. and substituting the above values, it is found that $C_{10} = 82\ kN$.

The corresponding minimum bearing's bore can be found from Shigley's Mechanical Engineering Design to be= 75 mm=2.95276 in.

Therefore, it can be concluded that a 4in bearing will be safe enough to resist the applied load, rotational speed and the number of working hours.

	02-Series				03-Series			
Bore, mm	OD, mm	Width, mm	Load Rating, kN C_{10}	C_0	OD, mm	Width, mm	Load Rating, kN C_{10}	C_0
25	52	15	16.8	8.8	62	17	28.6	15.0
30	62	16	22.4	12.0	72	19	36.9	20.0
35	72	17	31.9	17.6	80	21	44.6	27.1
40	80	18	41.8	24.0	90	23	56.1	32.5
45	85	19	44.0	25.5	100	25	72.1	45.4
50	90	20	45.7	27.5	110	27	88.0	52.0
55	100	21	56.1	34.0	120	29	102	67.2
60	110	22	64.4	43.1	130	31	123	76.5
65	120	23	76.5	51.2	140	33	138	85.0
70	125	24	79.2	51.2	150	35	151	102
75	130	25	93.1	63.2	160	37	183	125
80	140	26	106	69.4	170	39	190	125
85	150	28	119	78.3	180	41	212	149
90	160	30	142	100	190	43	242	160
95	170	32	165	112	200	45	264	189
100	180	34	183	125	215	47	303	220
110	200	38	229	167	240	50	391	304
120	215	40	260	183	260	55	457	340
130	230	40	270	193	280	58	539	408
140	250	42	319	240	300	62	682	454
150	270	45	446	260	320	65	781	502

Figure 47: Dimensions and Basic Load Ratings for Cylindrical Roller Bearings

4.2.2. Motor selection

The motor is one of the major components in the setup where it provides the rotational motion of the drill-string. Therefore, the selection process of the motor should be done in a very proper way to get the most suited motor to the job. Some information is needed to carry out the calculations to get the right motor.

Important Parameters and Assumptions:

1- Pipe length is 12m.
2- The supporting pipe and the testing are made of mild steel with a density of 8000 kg/m^3.
3- The tested pipe diameter is 3 in and the supporting pipe diameter is 4 in.
4- Maximum rotational speed that the test might be carried at is 1400 rpm.

$$\text{Power of motor} = \text{Torque } x \text{ Rotational speed}$$

where it is clear from the above equation that the power and torque are still unknown. Therefore, the required torque to rotate the pipe, chucks and flanges.

Torque Calculations:

The torque in rotational mode is given by the following equation:

$$T = \mathbf{I\alpha} \quad (\text{N.m})$$ **(12): Torque**

Where $\boldsymbol{I}$ is the mass moment of inertia and α is the angular acceleration. Both of these parameters are unknown yet, but there is enough information to calculate them. For all the components being rotated by the shaft, their moment of inertia must be taken into account to find the total moment of inertia as shown below:

$$\mathbf{I_{Total}} = (2)\,\mathbf{I_{Chuck}} + (\mathbf{2})\mathbf{I_{housing\ pipe}} + \mathbf{I_{test\ pipe}} + (2)\mathbf{I_{flang}}$$ **(13): Total Moment of Inertia**

1- Chuck Mass Moment of Inertia:

The chuck is made of cast iron which its density is ranging from 6800 kg/m^3 to 7800 kg/m^3. Considering the worst case that density of the cast iron is 7800 kg/m^3.

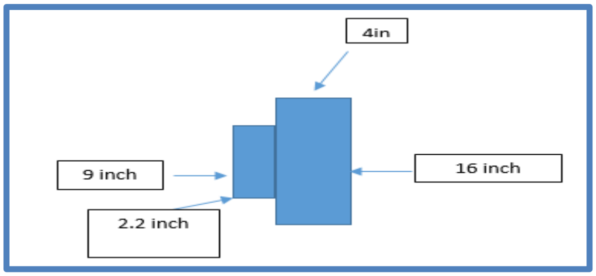


Figure 48: Side View of the Chuck

The above diagram shows a side view of the chuck where the outer diameter of the bigger portion is 16 inch and the outer diameter of the smaller portion is 9 inch. So the moment of the inertia is found in two stage as indicated in the equation below:

$$\mathbf{I_{chuck}} = \mathbf{I_{big}} + \mathbf{I_{small}} \qquad (\mathrm{Kg.m^2})$$

$$\mathbf{I_{big}} = \frac{\boldsymbol{\pi\rho L}}{\mathbf{32}}[\mathbf{D_{Out}^4} - \mathbf{D_{In}^4}]$$

$$\mathrm{I_{big}} = \frac{\pi 7800 * 0.1016}{32}[0.4064^4 - 0.1016^4] = 2.113\ \mathrm{Kg}.m^2$$

$$\mathrm{I_{Small}} = \frac{\pi 7800 * 0.05588}{32}[0.2286^4 - 0.1016^4] = 0.11224\ \mathrm{Kg}.m^2$$

$$\mathrm{I_{chuck}} = 2.225159\ \mathrm{Kg}.m^2$$

2. Flange Moment of Inertia:

Flanges are also known as coupling whenever they are connected to rotating parts. They are always made of steels and depending upon the service condition the steel type is specified. In this project, the used couplings are assumed made out of steel in general with a density of 7600 $\mathrm{Kg/m^3}$. The figure below shows the used coupling dimensions to connect the housing pipe to the chuck.

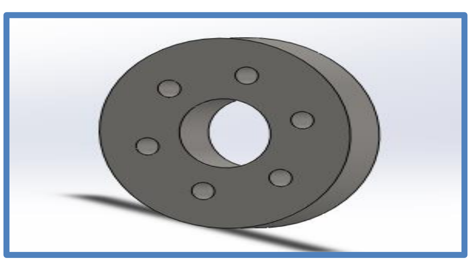


Figure 49: 3D- Drawing of Coupling

$$I_{Flange} = \frac{\pi\rho L}{32}[D_{Out}^4 - -D_{In}^4]$$

$$I_{Flange} = \frac{\pi * 7600 * 0.0254}{32}[0.2286^4 - -0.1016^4] = 0.0497 \text{ Kg.m}^2$$

3. Housing Pipes:

Bearings, chucks and pulley are all connected to the housing pipe which serves as the holder of the tested pipe and also it delivers the rotation motion to the tested pipe.

$$I_{Housing\ pipe} = \frac{\pi\rho L}{32}[D_{Out}^4 - D_{In}^4]$$

$$I_{Housing\ pipe} = \frac{\pi * 8000 * 1}{32}[0.116^4 - 0.0762^4] = 0.05717 \text{ Kg.m}^2$$

Therefore, the total moment of inertia is equal to as what is indicated in equation:

$$I_{Total} = 0.255 + 2 * (.005717) + 2 * (2.225159) + 2 * (0.05717) = 4.9339 \text{ Kg.m}^2$$

Hence, the total moment of inertia has been found to be approximately 5.0 kg.m^2. Only the angular acceleration is still needed to be found so that the torque can be obtained.

Angular Acceleration:

The angular acceleration is defined to be the rate change of the rotational speed with time. Assuming that the maximum speed will be reached in 8 seconds and the minimum speed will be reached in 5 seconds.

Note: The timings to the reach the maximum and minimum speeds were found by experiment which was held in the mechanical engineering department workshop using the lathe machine to spin the pipe.

$$\boldsymbol{\alpha} = \frac{\mathbf{dw}}{\mathbf{dt}} = \frac{\boldsymbol{\Delta}\mathbf{w}}{\boldsymbol{\Delta}\mathbf{t}} \quad \text{(14): Angular Acceleration}$$

$$\alpha_{max} = \frac{\left[\frac{1400}{60}\right] * 2\pi}{8 \text{ sec}} = 18.316 \text{ rad/s}$$

$$\alpha_{\min} = \frac{\left[\frac{600}{60}\right] * 2\pi}{5 \text{ sec}} = 12.56 \text{ rad/s2}$$

By having the angular acceleration and the total mass moment of inertia the torque is easily found as follows:

$$T_{max} = \mathbf{I} * \boldsymbol{\alpha}_{\mathbf{max}}$$

$$T = 5.0(18.316) = 91.6 \text{ N.m}$$

$$T_{min} = \mathbf{I} * \boldsymbol{\alpha}_{\mathbf{min}}$$

$$T_{min} = 5(12.56) = 62.8 \text{ N.m}$$

Since the torque and the angular acceleration have been calculated, the motor desired maximum and minimum power can obtained straight away as follows:

$$P_{max} = T_{max}\ \boldsymbol{\omega} \qquad \text{(15): Power of the motor}$$

$$P_{max} = 91.6 \left(1400 \frac{2\pi}{60}\right) = 13.429 \text{ Kw} = 18.0013 \text{ hp.}$$

$$P_{min} = T_{min}\ \boldsymbol{\omega}$$

$$P_{min} = 62.8 \left(600 \frac{2\pi}{60}\right) = 3.945 \text{ Kw} = 5.2882 \text{ hp.}$$

Taking the average power which's 12hp which the power of the selected motor.

4.2.3. Selecting the Motor Speed Regulator:

The speed controller will be selected according to the power of the selected motor where it depends on the current required to control the motor. Consultation of an expert is required to select the right motor and from that it was found that 25 A controller is required to control 12 hp motor.

4.2.4. Braking System

In this system two main things must be selected which are the linear actuator and the spring.

Spring Selection:

The spring is required in the braking system to connect the brake pads and give them some freedom to move relative to each other. Figure 52 shows a simple free body diagram of the left brake pad and the forces acting upon it. The main goal is to calculate the force that the spring induces, since the spring force is needed in the selection process.

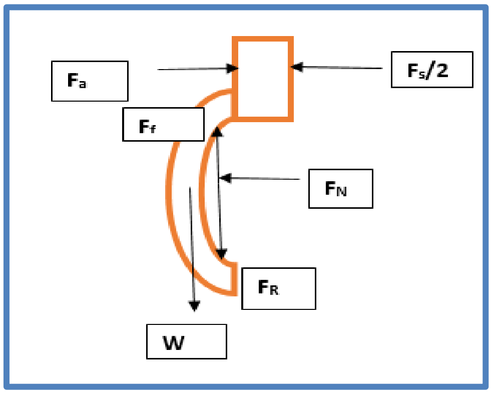


Figure 50: Free Body Diagram of Braking Clutch

Where:

F_a: is the linear actuator pushing force.

F_s: Spring force where it is divided by two just to account for the other end.

F_f: Friction force resulting due to the contact between the rotating pipe and the brake pad.

F_R: Rotational force coming the rotating pipe.

F_N: Normal force.

W: the weight of the left part of the braking system.

Pushing Force:

The pushing force of the actuator is specified in its catalog as 50kg maximum dynamic load. It can carry force of (9.81*50) = 450.9N.

Rotational Force:

The rotational force can be obtained from the following equation:

$$F_r = \boldsymbol{M\omega^2 r} \qquad \text{(16): Rotational Force}$$

Where M is the mass of the housing pipe. So:

$$F_r = 28 * 146.5^2 * 0.0508$$

$$F_r = 208.43N$$

The mass rings of the braking system is 22kg assuming that they are made of cast carbon steel so the mass of one ring is 11kg. Therefore, the weight of one ring is found to be 108N.

From the free body diagram shown in figure 52:

$$\sum F_Y = 0$$

$$F_f - W - F_r = 0$$

$$F_f = W + F_r = 108 + 208.43 = 316.43N$$

So the frictional force works out to be 316.43 N. The reduction that the braking system will reduce can be calculated as follows:

$$T = \boldsymbol{F_f * r} \qquad \textbf{(17): Reduced Torque}$$

Where r is the radius of the housing pipe.

$$T = 316.43 * 0.0508 = 16.07 \text{ N.m}$$

This reduction from one side only of the brake where it will be 32.12N in total with the help of using two actuators of the same specifications from maximum torque which calculated above to be 92N.m in case the rotational speed is 1400rpm. This reduction is still very good in case of the using rotational speed of 600rpm which needs torque of 62N.m so almost reducing it to half of it.

For selecting a spring the reaction force it provides when compressed is needed to be calculated. From figure 52:

$$\sum F_X = 0$$

$$F_a - \frac{F_s}{2} - F_N = 0$$

$$F_s = 2\,(-F_a + F_N)$$

Where $F_N = \frac{F_f}{\mu}$, and μ in this case is found from an engineering hand book, shown in figure 53, and the value of it is 0.6.

Materials and Material Combinations		Static Frictional Coefficient - μ_s -	
		Clean and Dry Surfaces	Lubricated and Greasy Surfaces
Aluminum	Aluminum	1.05 - 1.35	0.3
Aluminum-bronze	Steel	0.45	
Aluminum	Mild Steel	0.61	
Brake material[2)]	Cast iron	0.4	
Brake material[2)]	Cast iron (wet)	0.2	
Brass	Steel	0.35	0.19
Brass	Cast Iron	0.3[1)]	
Brick	Wood	0.6	
Bronze	Steel		0.16
Bronze	Cast Iron	0.22[1)]	
Bronze - sintered	Steel		0.13
Cadmium	Cadmium	0.5	0.05
Cadmium	Chromium	0.41	0.34
Cadmium	Mild Steel	0.46[1)]	
Cast Iron	Cast Iron	1.1, 0.15[1)]	0.07[1)]
Cast Iron	Oak	0.49[1)]	0.075[1]
Cast iron	Mild Steel	0.4, 0.23[1)]	0.21, 0.133[1)]
Car tire	Asphalt	0.72	
Car tire	Grass	0.35	
Carbon (hard)	Carbon	0.16	0.12 - 0.14
Carbon	Steel	0.14	0.11 - 0.14
Chromium	Chromium	0.41	0.34
Copper-Lead alloy	Steel	0.22	
Copper	Copper	1	0.08
Copper	Cast Iron	1.05, 0.29[1)]	
Copper	Mild Steel	0.53, 0.36[1)]	0.18[1)]
Diamond	Diamond	0.1	0.05 - 0.1
Diamond	Metal	0.1 - 0.15	0.1
Glass	Glass	0.9 - 1.0, 0.4[1)]	0.1 - 0.6, 0.09-0.12[1)]
Glass	Metal	0.5 - 0.7	0.2 - 0.3
Glass	Nickel	0.78	0.56
Graphite	Steel	0.1	0.1
Graphite	Graphite (in vacuum)	0.5 - 0.8	
Graphite	Graphite	0.1	0.1
Hemp rope	Timber	0.5	
Horseshoe	Rubber	0.68	

Figure 51: Shows a Table Contains Friction Coefficients of Common Material Adopted from the Engineering Toolbox Website

According to the previous equation F_N is found to be 316/0.6 = 526.7

$$F_s = (-F_a + F_N) = (-490.5 + 526.7)2 = 72.4 \text{ N}$$

The spring reaction is known and hence, the stiffness of the required spring can be calculated as follows:

$$Fs = K{*}X \qquad \text{(18): Spring Restoring Force}$$

Where X is the displacement of the spring when compressed. X is known since it is one of the design constrains. The actuator will push the spring from both ends of the spring and distance it can push for is only 0.15 in from each end. Thus, 0.3in in total.

$$K = \frac{F_s}{x} = \frac{72.4}{.000762} = 9501.3 \ {}^{N}/_{m} = 9.5 \, {}^{KN}/_{m}$$

Note: The above calculations of the spring are done based figure 52 which indicates that the spring is being compressed. However, the concern when selecting the spring in this project is when in tension to hold the brake system setup together as shown in figure54 below.

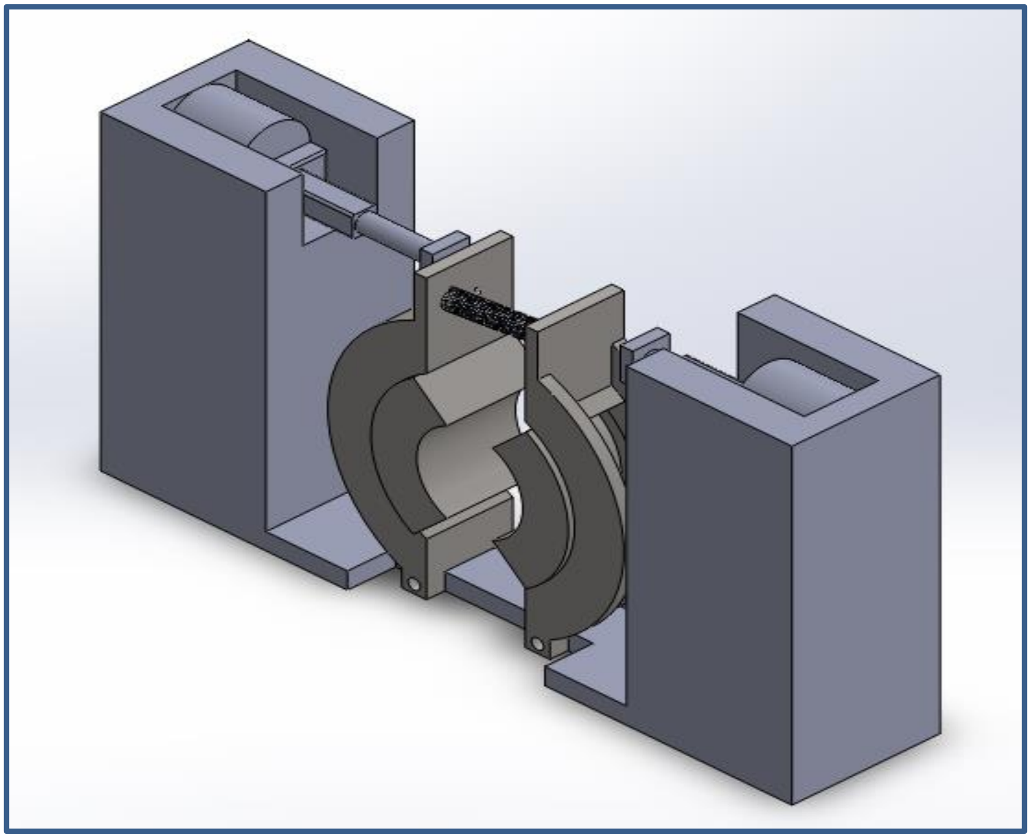


Figure 52: 3D Model of Braking System with the Spring

The figure below shows the free body diagram when under tension.

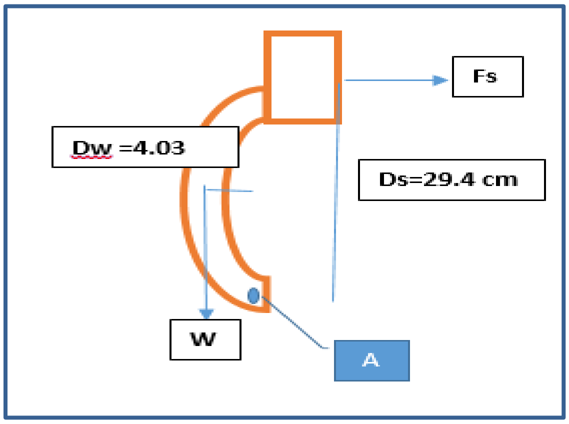


Figure 53: Free Body Diagram When Spring is in Tension

The summation of the moment around point *A* is taken to calculate the restoring force of the spring. The weight of the brake pad is 8Kg and assumed to be at the center.

$$\sum M_A = 0$$

$$-\frac{F_S}{2} = * (0.2946) + 78.5 * 0.0403 = 0$$

$$F_s = 21.40\text{N}$$

The main goal is calculate the stiffness of the spring. The spring will be set in a tension from both ends with value of 4cm. Hence:

$$K = \frac{F_s}{x} = \frac{21.40}{.04} = 535\ {}^{N}/{}_{m}$$

The spring is selected based on the stiffness. More calculation regarding the selection of the spring will be done next semester.

4.2.5. Linear Actuator Selection

The linear actuator is required to give a Push to the brake pads to be in contact with the rotating pipe in a regular basis of time. According to the dimensions as shown in the figure below, the linear actuator must have a stroke length of 2 in.

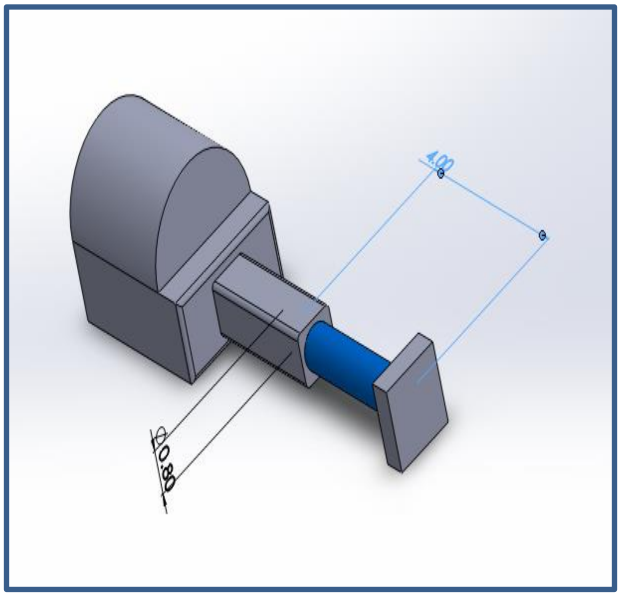


Figure 54: Linear Actuator

The required speed to achieve the stroke distance is 0.5 in/s which means the linear actuator will achieve the stroke length in 4s. This implies that the brake pads will be in contact with the rotating pipe every 12s. The actuator will in contact with pipe for 2s since the distance between the pad and the pipe is 1in and speed of the actuator is 0.5 in/s. The linear actuator is selected using the above specifications and by using POLOLU Robotics and Electronics website, the selected linear actuator is shown below.

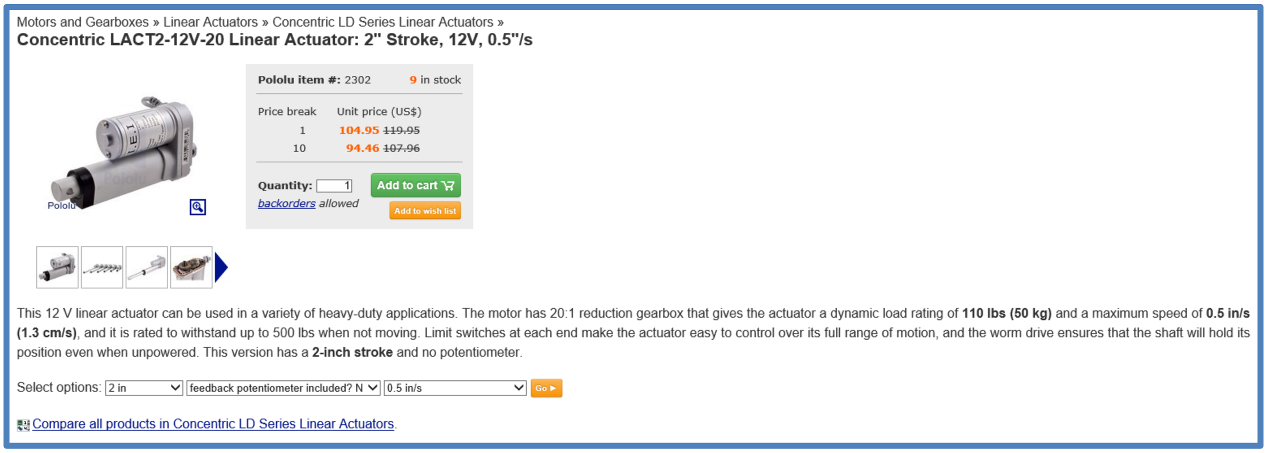


Figure 55: Desired Linear Actuator in Market

The figure above shows the picture of the actuator and states some of its specifications which are as follows:

1- 12V and can be used in variety of heavy duty applications.

2- The motor has a gearbox reduction ratio of 20:1.

3- The actuator has a dynamic load of 50Kg.

4- Maximum speed of the actuator is 0.5in/s(1.3cm/s).

5- 2in stroke length and no potentiometer.

The specific name of the actuator in the market is Concentric LACT2-12V-20Linear Actuators.

4.2.6. Material Selection for the Braking System

Problem Statement:

The rotating shaft is to be resisted by amount of 50% of input torque using a braking mechanism. The material of the braking mechanism that is in contact with the rotating shaft has to be selected for long life and robust design and the strength is to be maximized or the rotational speed is to be minimized.

Boundaries of the Problem:

The input torque is about 75 N.m. and the braking system is reducing the 50% of the input torque that is about 35 N.m. The diameter of the rotating shaft is 100 mm.

Available Information:

Ashby Charts and data of material properties from internet are used.

Assumption:

- Basic material relationship will be used.
- The contact between braking blades and rotating shaft is all over the perimeter of the shaft.
- Toughness $\boldsymbol{k_{Ic}}$ is the parameter of the design and it is considered to be at maximum of 50 $MPa\sqrt{m}$ as conventional design.
- The 10 ${}^{Mg}/_{m^3}$ density is considered to be maximum.

Model of the Problem:

$$\boldsymbol{F = ma = m\omega^2 R = \frac{T}{R}}$$

$$\boldsymbol{\sigma = \frac{T}{RA} = \rho\omega^2 R^2}$$

$$\boldsymbol{\sigma = \frac{K_{Ic}}{\sqrt{\pi a_c}} = \rho\omega^2 R^2}$$

$$\omega \leq \left(\frac{1}{\sqrt{\pi a_c}}\right)^{.5} \frac{1}{R} \left(\frac{K_{Ic}}{\rho}\right)^{.5}$$

$$\boldsymbol{M = \left(\frac{K_{Ic}}{\rho}\right)^{.5}}$$

Selection of Material:

Using Ashby chart and material property found in the previous section, the materials are limited to few specific materials that give the required strength according to Ashby chart in figure 58, the selected material are:

1. Glass-ceramic fiber.
2. Nodular cast iron.
3. Clay-copper ceramic.
4. Carbon fiber.

Then, one material is selected using approximated material index of $\frac{K_{Ic}}{\rho}$ as shown in table 8. The selected material is clay-copper ceramic material which is composed of clay and porcelain bonded to copper flakes and filaments.

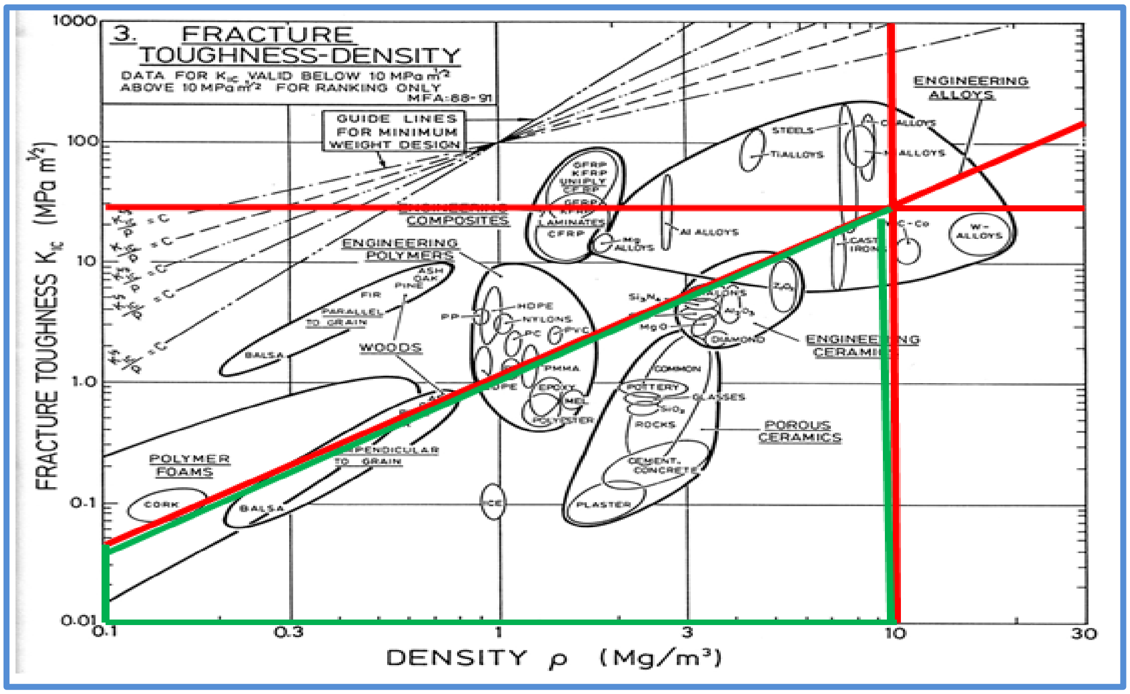


Figure 56: Ashby Chart (Fracture Toughness Verses Density)

Table 9: Material Selection's Calculation

Material	$K_{Ic}\ MPa\sqrt{m}$	Density $^{kg}/_{m^3}$	$\left(\frac{K_{Ic}}{\rho}\right)$
Glass-ceramic fiber	.83	2500	.000332
Nodular cast iron	7.1	7200	.000986
Clay-copper ceramic	1.2	100	.012
Carbon fiber	3.8	1550	.00245

4.2.7. Structural Supports Analysis

Two tables are required in the setup to carry the weight of the components like chucks, housing pipes, bearings, flanges and the motor. The total weight of these components must be calculated. Hence, the total force acting upon the table is known and safety measures are taken into account to avoid any sudden failure. The table below shows the mass and the weight of the setup components. Some of the components weights have been calculated in the earlier stages of the parametric design.

Table 10: Weight Distribution on the Supporting Table

Setup Components	Mass (Kg)	Weight (N)
Motor	79	990.81
Bearing with the housing	15	147.15
Flange	10.5	103
Housing pipe	27.66	271.35
Chuck	105	1030.05
Pulley	4	39.24

Some of the masses have been measured by weighing existing parts in the mechanical engineering department workshop and were estimated by calculating the volume and multiplying it by density.

$$W_{Total} = 990.81 + 147.15 + 103 + 271.35 + 1030.05 + 39.24 = 2581.55N = 2.6KN$$

Hence, the total force that acts on the table from the motor side is 2.6KN which is a huge force in addition to an expected vibrational force resulting due to the rotation of the testing pipe. Safety is an important aspect and therefore the following dimensions of the table that are shown in the figure below are assumed to be enough to carry and stand this much of force. The table is desired to be made out of mild steel to match the load requirements.

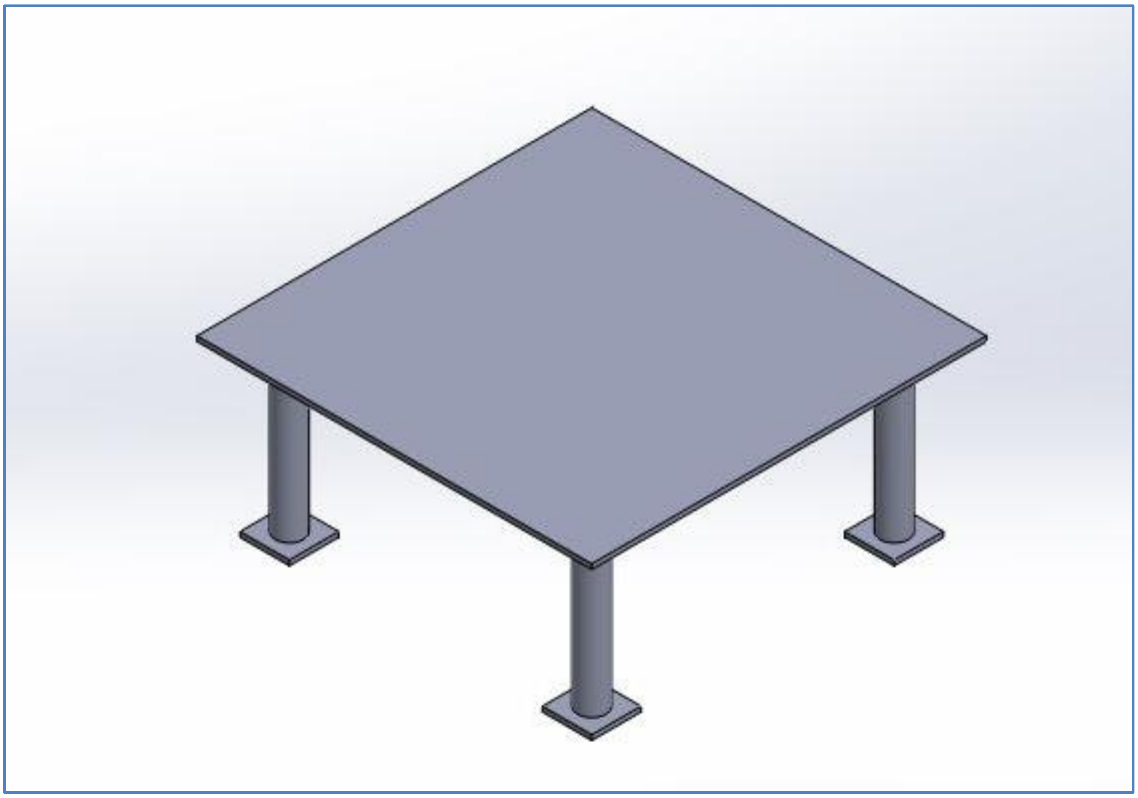


Figure 57: Table Support

4.3. Detailed Design

Detail design phase document the final design by producing engineering drawings of the final design to show the dimensions and views of each part along with the bill of material and cost estimation.

4.3.1. Engineering Drawing

The final detailed drawings are attached in Appendix A. Figure 57 shows the sample of detailed drawing of supporting table.

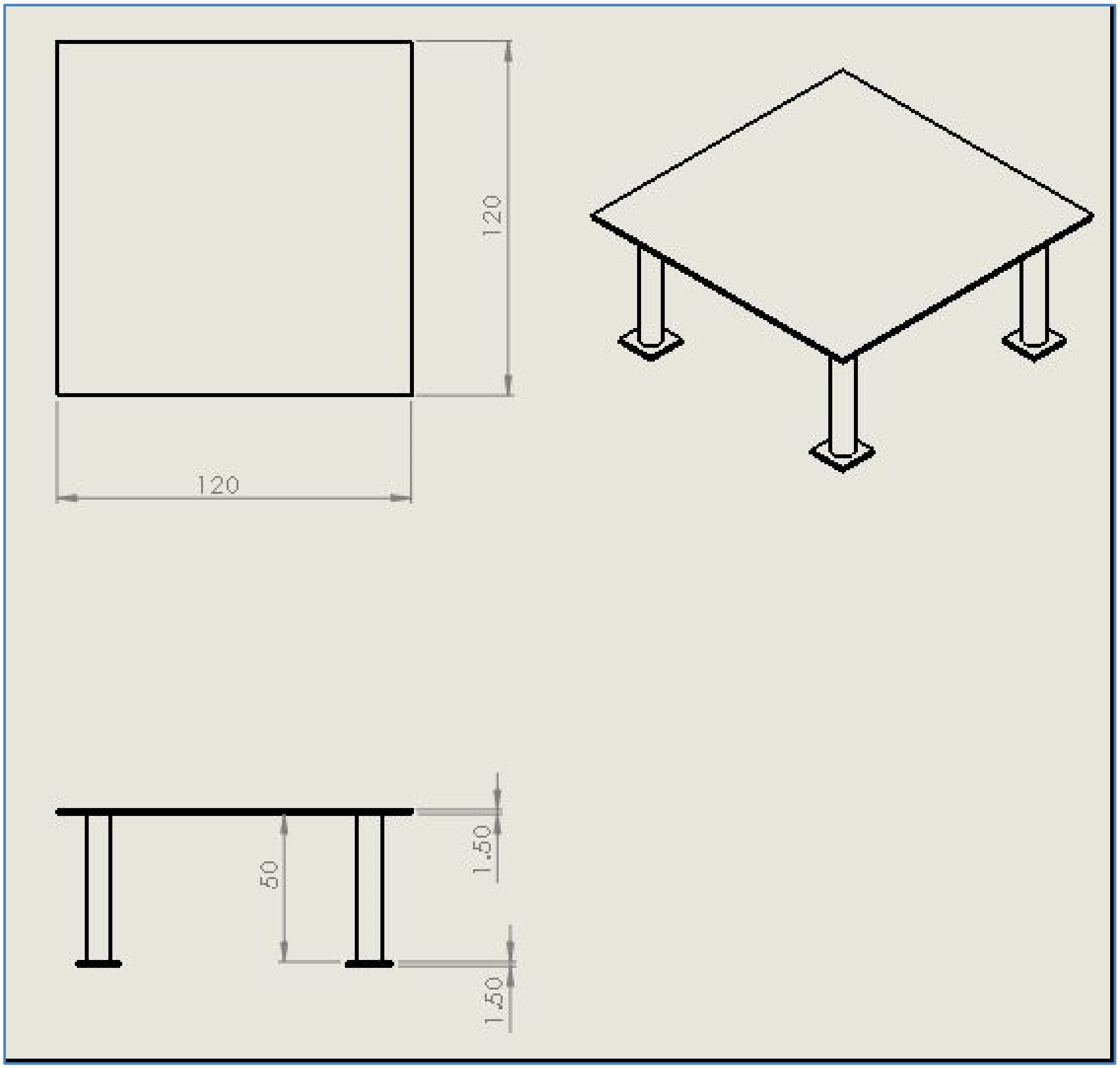


Figure 58: Detailed Drawing of Table Support

4.3.2. Assemble Drawing

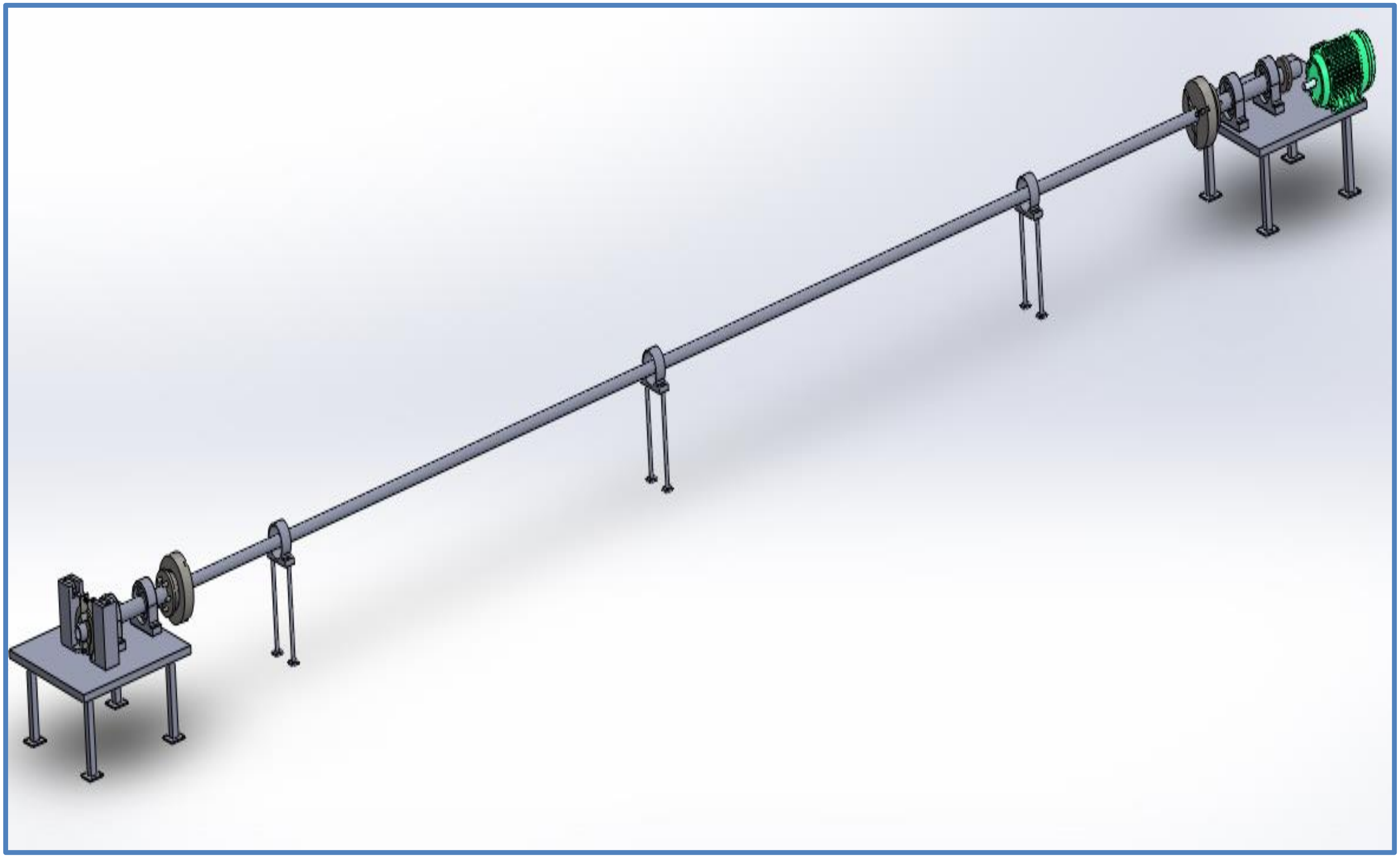


Figure 59: Assembly View

4.3.3. Exploded View

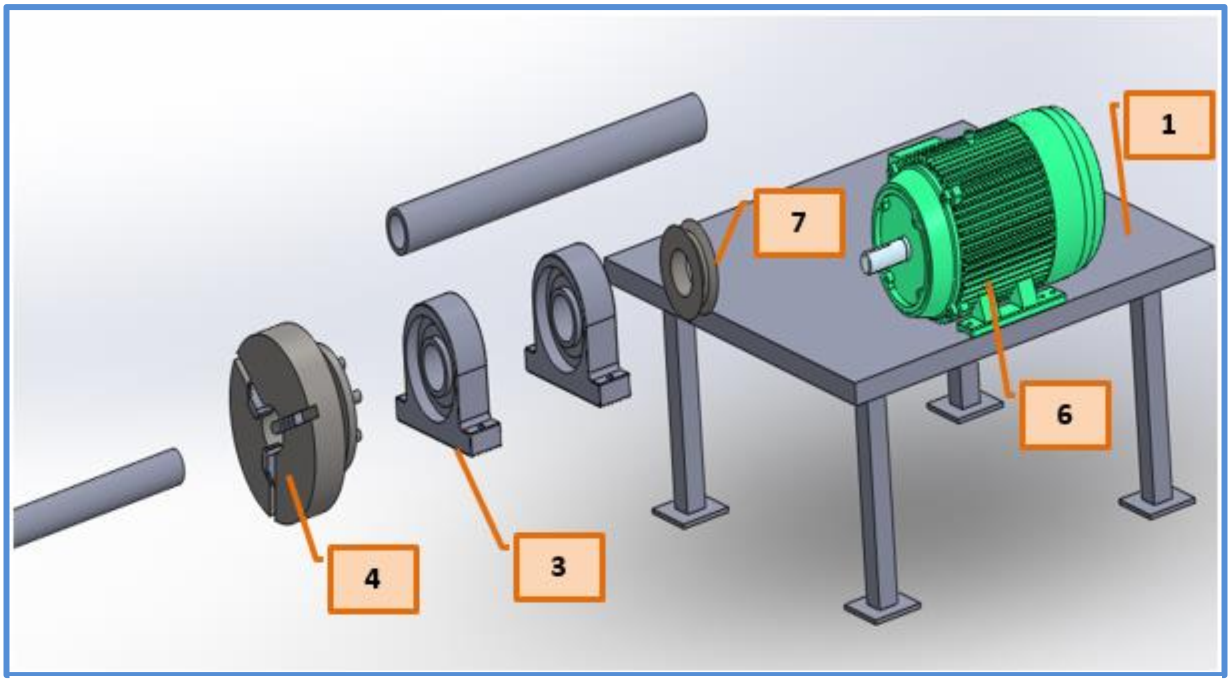


Figure 60: Disassembly View at Rotation Mechanism Side

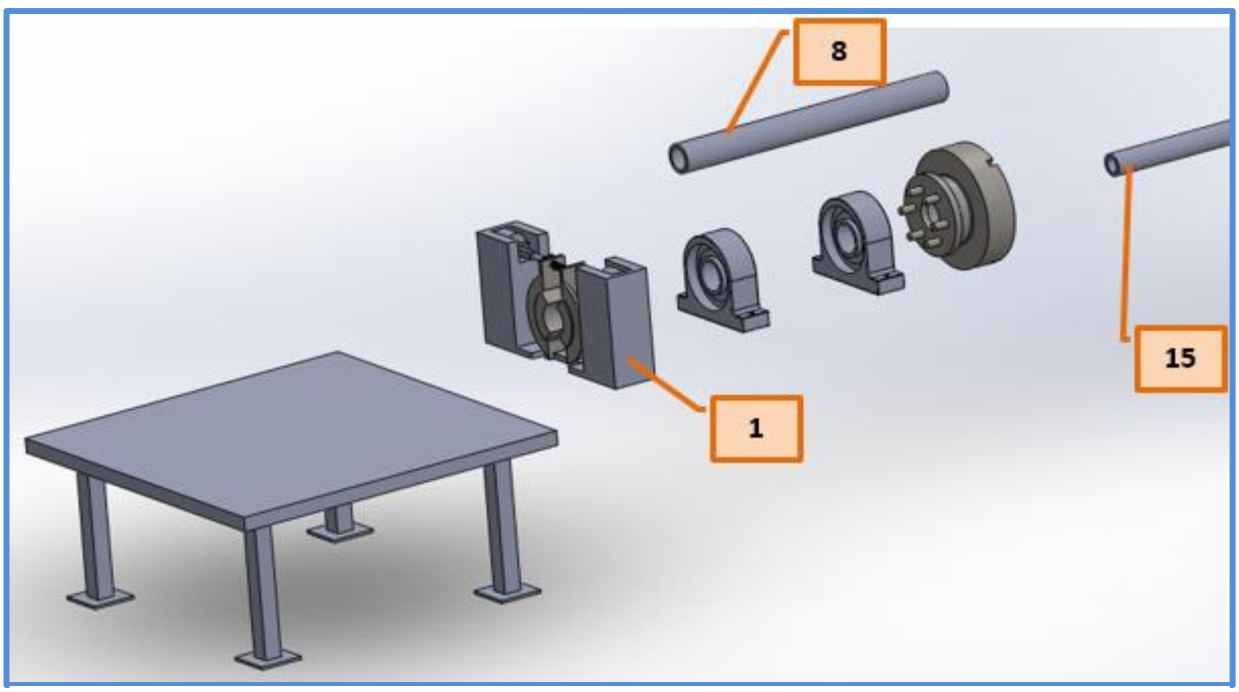


Figure 61: Disassembly View at Braking Mechanism Side

Table 11: Bill of Materials

#Part	name	material	Quantity
1	Table	Mild Steel	2
2	Stabilizer	Mild Steel	3
3	Cylindrical bearing	-	4
4	Chuck	Cast Iron	2
5	Bolts and nuts	Mild Steel	24
6	12 hp Motor	-	1
7	Three stages Pulley	-	2
8	belts		3
9	Housing shaft	Carbon Steel	2
10	Variable Speed Controller	-	1
11	Linear actuator	-	2
12	Clutches + brake pads	Clay-copper ceramic	1
13	6m long, 0.5in Dimeter-Pipe specimen	Mild Steel	4
14	6m long, 1 in Diameter-Pipe specimen	Mild Steel	4
15	6m long, 1.5 in Diameter-Pipe specimen	Mild Steel	4
16	6m long, 2 in Diameter-Pipe specimen	Mild Steel	4
17	6m long, 3 in Diameter-Pipe specimen	Mild Steel	1

Chapter 5.0: Finite Element Analysis

In practice, lateral vibrations are induced by combination of loadings applied on the drill-string due to contact with formation. These loadings add up to result in an equivalent alternating forces with a frequency equal to the drill-strings' natural frequency. Thus, the drill-string goes into resonance mode accompanied with significant lateral deflections.

With the source of lateral vibrations clearly defined, and according to the fact that the experimental setup is mimicking to some extent real drilling practice. Consequently, the test rig developed in this project may not completely introduce load carrying possibilities to the testing pipes. On the other hand, knowing that rotating a pipe with an angular velocity equal to its critical speed will lead to resonance and thus lateral vibrations.

5.1. Modal Analysis:

It was essential to find the frequency modes of the testing pipe using Finite Element Analysis.

The consequence of steps was as follows:

1. Pipe geometry:

Pipe geometry was created using SolidWorks software with the following dimensions:

D_{in}= 2.1 inch D_{out}=2.4 inch Length= 4.2 m

``

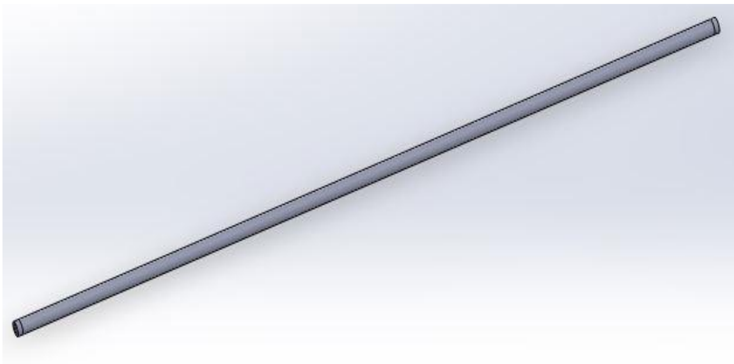


Figure 62 Pipe Geometry

1. Boundary conditions:

Since that the testing pipe is fixed from its both ends by a means of chucks. Chucks allow pipe's rotation about its axis and restricts its displacement in X, Y and Z coordinates. Thus, chucks can be modeled as bearing supports by switching allowing self-alignment off.

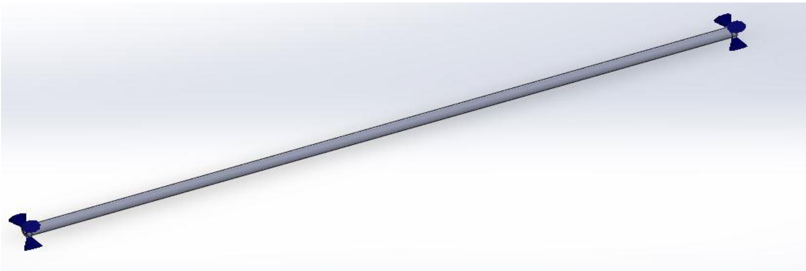


Figure 63: Application of Boundary Conditions

2. Meshing:

After applying boundary conditions, the part was meshed using fine finite elements for higher results accuracy.

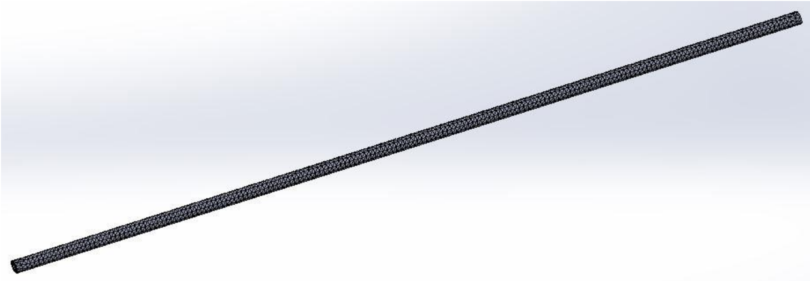


Figure 64: Meshed Pipe

3. Solve (Run):

The outputs of this study are the pipe's resonant frequencies as well as the maximum nodal

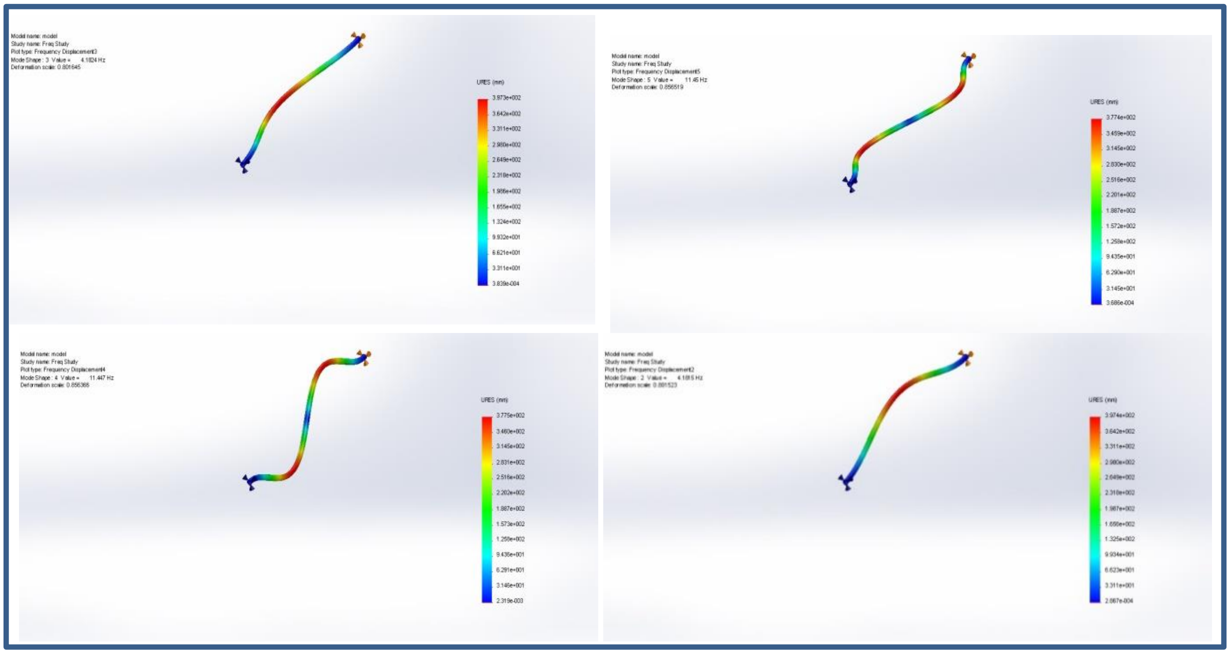


Figure 65:Resonant Frequencies

displacement.

Table 12: List of Resonant Frequencies

Mode Number	Frequency(Rad/Sec)	Frequency(Hertz)	Frequency(RPM)
1	26.273	4.1815	250.89
2	26.279	4.1824	250.944
3	71.922	11.447	686.82
4	71.943	11.45	687

SolidWorks simulation shows that the pipe's first natural frequency is reached at 4.1815Hz. Based on proposed FEA, it can be seen that rotating the pipe with an angular velocity of 250.9 RPM will put the pipe into resonant mode and consequently lateral vibrations will occur.

Chapter 6.0: Fabrication and Setup Performance

The setup is fabricated according to the design and specifications developed in the embodiment design. Some modifications are also considered and implemented to improve the setup like using C-channel to support the setup structure and utilizing pneumatic actuators instead of electromechanical linear actuators. The setup after fabrication is shown in figure 67. These changes incorporated due to the difficulties in fixing the setup to the ground by bolts.

The setup consists of three major mechanisms which are rotation mechanism, Structural support and braking mechanism. This section indicates the stages of the fabrication and assembly processes of these mechanisms. Also, it illustrates the performance of the setup and tests which were carried out to checkup it performance.

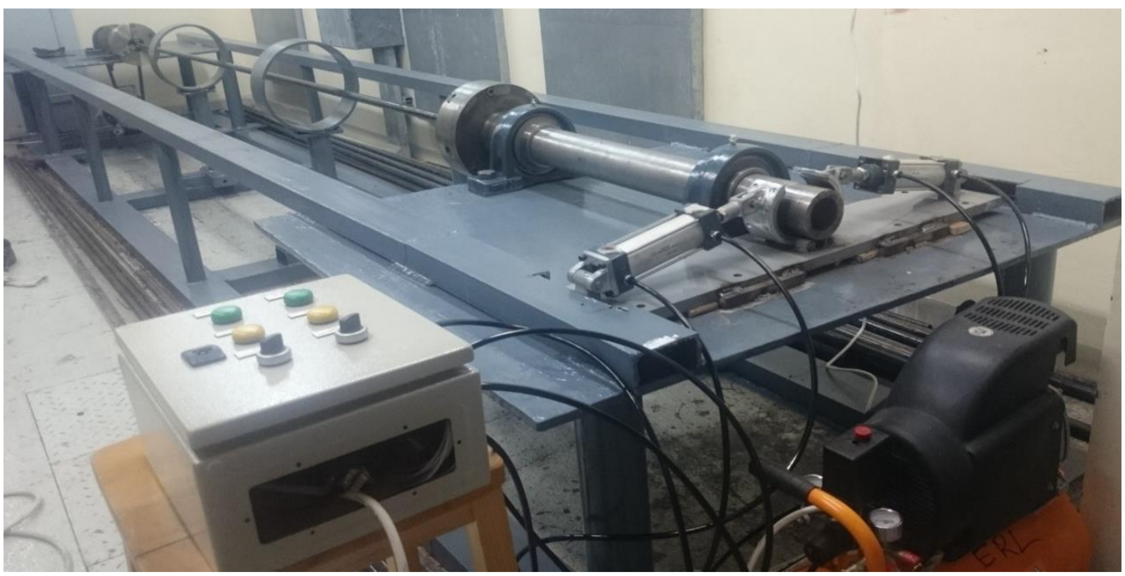


Figure 66 Fabricated Setup

6.1. Fabrication

The fabrication process of the three major systems.

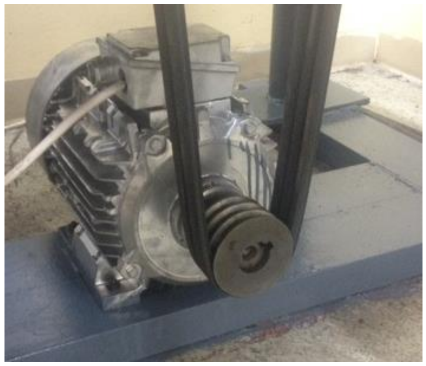
6.1.1. Rotation mechanism:

Figure 67 Rotation mechanism

Rotation mechanism is a mechanism that is responsible of creating lateral vibration in the specimen pipe beside rotating the pipe. It consists of motor, controller, pulleys with belts, bearings, housing pipes and chucks with flanges as shown in figure 67. The following table indicates the specifications of each of these components.

Table 13 : Pictures of the Rotation Mechanism Components and their specifications

Part	Picture	Specifications

Motor		• 12 hp • 1200 rpm maximum • 3-phase.
Controller		• 3-phase • Motor: 11 KW • 50 Hz maximum • 25 A
Pulleys with Belts		• The motor's pulley is 3.5 inch in diameter (3 grooves) • The housing shaft's diameter is 5.5 inch in diameter (3 groves) • The belt perimeter is 57 inches.

Bearings		• The internal diameter of the bearing is 85 mm. • Number of pulleys is 4 (2 for each housing pipe)
Housing Pipe		• Outside diameter is 85 mm • Thickness is 16 mm • Length is 1 meter.
Chucks with Flanges	 	• 3-jaws chuck • 12 inches in diameter • Number of chucks is 2

Figure 68 structure support

6.1.2. Structural Support:

The structural support (shown above in figure68) consists of two tables, ring shape stabilizers and four C-channel connecting the tables to the stabilizers and providing stability to the setup. The structural support is responsible of providing safe working environment and holding the other mechanism like the rotation mechanism and braking system. The following table shows the components of the structural support long with their specifications.

Table 14 Pictures of the Structural Support Components and their specifications

Part	Picture	Specifications
Metal sheet		• Two metal sheets • 16mm thick • One is 1.2X1.1m and other one is 1.2X1.2 • Carbon steel
Stabilizer		• Two ring shape stabilizers. • 23cm in diameter. •

C- Channel		• Four C-channel. • 8m long. • Two of them are 4X3in and the other two are 5X2in.
Table support		• 8 supporting pipes for the tables • 0.5m in length • 85mm in diameter • Carbon steel

6.1.3. Fabrication procedure of the structural support:

Based on the available size in the project room and the load calculations mentioned in the parametric design, the sizes and standards of the components of the structural supports were decided. Metal sheets of the tables, Rings of the stabilizers and supporting pipes of the tables were bought from the scrape market.

- **Tables Fabrication:**

First of all, the metal sheets were brought from the scrape market and unloaded to the workshop. The sheets were not in the required size so some measurements were carried out and the required dimensions were marked on the sheet. Next, the undesired parts of the sheets were cut out by gas cutting machine. The legs of the table were bought in length of 1m and therefore were cut to the required size by saw cutting machine. The removed parts of the table were cut to eight square pieces 20X20mm to be used as the bases of the tables. Grinding process was done on the metal sheets to remove the sharp edges and to clean the surface of the sheets. The pipes were welded to the metal sheets and the square pieces were welded to the supporting pipes using arc welding machines.

- **Stabilizers Fabrication:**

The bases that carry the rings of the stabilizers already existed in the workshop. The rings were bought from the scrape market as 50cm long pipe and 23cm in diameter. Then, the pipe was cut to two pipes each one is 15cm length to the rings of the stabilizers. These rings were welded to the bases using arc welding machines.

- **C-Channel Connection:**

The design had no c-channel before but a modification to it just happened while the manufacturing process was taking place. It was difficult to fix the tables and the stabilizers to the ground as planned before. Therefore, the idea of connecting the tables by C-channel came up to have the setup stable and fixed. The standard C-channel exists in the market is 6m long but 8m long C-channel was required. Therefore, Three C-channel of size 4X2in were bought and another three with a size of 5X2in to compensate the missing length. The smaller size C-channel were used to connect the table from above and the bigger size were used to connect the table from bottom. The connections were all arc welding connections. In the bottom side two small pieces of the C-channel were extended horizontally to connect the two C-channel on the ground where the stabilizers were welded on them.

Table 15: Pictures of the Manufacturing Process of the setup with Explanation

Picture	Process
	Gas cutting: To cut the unwanted parts of the sheet and cut the square bases of the tables

	Saw cutting: To cut the pipes to the required size.
	Grinding: To remove the sharp edges of the sheet and make the edge profile smoother.
	Arc welding: welding the pipes to the tables and the square pipes to the pipes.

7.1.3. Braking Mechanism:

This mechanism is used to provide time varying resistance to simulate the formation resistance and cause torsional fatigue to the testing pipe. The design of the brake was modified then the one which was illustrated in figure53 where the design is shown in figure5. It consists of steel hosting structure, two linear pneumatic actuators, steel ring and ceramic pads in core of them to present the braking pads.

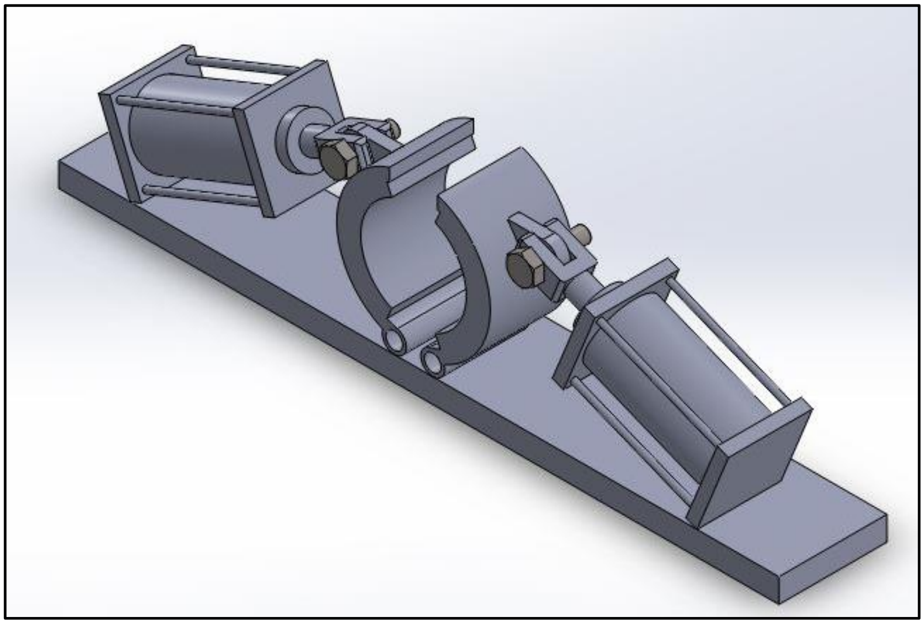


Figure 69 Modified Design of the Braking System

The braking system after fabrication is shown below with its controlling unit allows to control the braking time and the time of engagment.

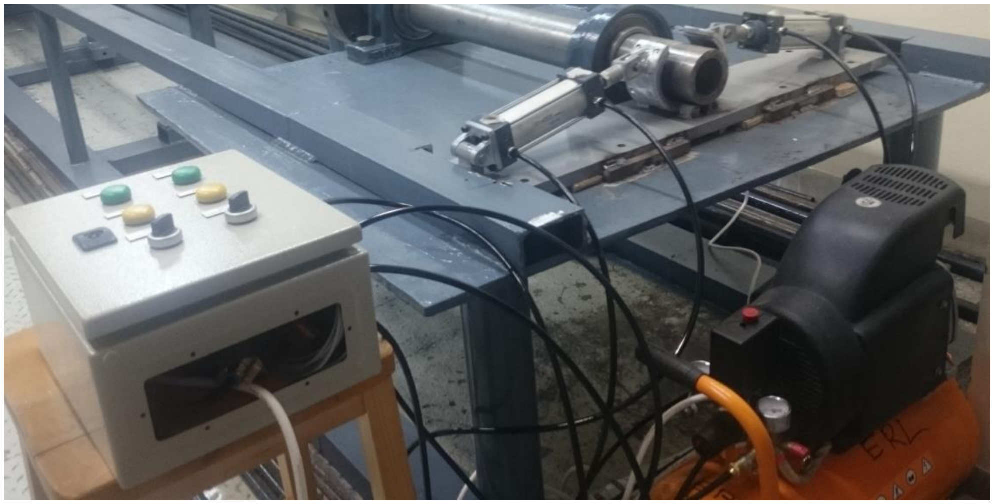
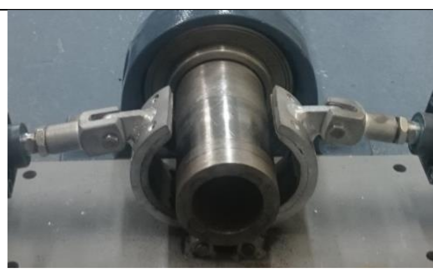


Figure 70 Braking Mechanism Assembly

The electric-mechanical actuators that were supposed to push the rings to stop the rotating pipe and to release it and repeat the same thing over and over were not found in the market. Pneumatic double acting actuators were bought with their controlling valves to do the function that electrical actuators were supposed to do. The following table shows the components of the structural support long with their specifications.

Table 16 Pictures of the Braking Mechanism Components and their specifications

Part	Picture	Specifications
pneumatic cylinders	 	• 10 cm strock length • Maximum pressure is 8 bar • Linear motion • One inlet and one outlet
Braking pads		• Consist of two curved ceramic composites, same as the one used in cars • Holds 4in pipe
Control Unit	 	• Each actuator has a control switch • Can be controlled automatic or manual • For automatic control, the delay tame vary from 1second up to 1 hour

Compressor		• The maximum provided pressure is 8 bar. • It has no control switch, it has only a valve where the flow rate can be changed.

6.2. Performance of the Setup:

After the setup was successfully built, its performance was observed by conducting one simple test on 0.5 inch-diameter pipe. The setup was able to laterally vibrate the pipe by rotating it at high rotational speed. The braking system was also effective and able to vibrate the pipe. There was a backward whirl on the pipe because of braking system when it is operating in high speed. This situation is considered as the most severe problem in the real field operation.

Figure 71: Bending Failure on 0.5 inch-diameter testing pipe.

In high speed operation, however, the 0.5 inch-diameter pipe exposed to bending failure since the stress due to high amplitude vibration reaches the yield stress as shown in figure71. The table, that has the braking system installed on, was also vibrating because of small instability in the structure. Therefore, it is suggested to remove the instability before conducting the tests in next stages to avoid any noise.

In general, all components in the three main mechanisms (i.e. rotation, braking and supporting mechanism) have excellent performance in different conditions.

Chapter 7.0. Revised Realistic Constraints:

Table 17: Revised Realistic Constraints

Category	Constraints
Performance	• The setup should be able to perform continuously for few days without any interruptions. • The setup should be able to generate self-excited lateral vibration.
Economic	• The budget should not exceed 2,400 OMR and the scale of the setup should be determined accordingly. • The cost of test specimens and maintenance cost of the main components of the setup should be taken into account. • If this project succeeds, it will leave an impact on the local economy as well it will provide solutions or will reduce drill-string problems.
Environmental	• No toxic materials are used. • Recyclable materials are used for testing (metallic materials specimen). • Production and operation of the setup is totally clean. • Since that the device is to test vibrational modes that lead to material fatigue [not thermal or other types of fatigue], operating conditions are kept constant and atmospheric.
Conformity to Standards	• The components (bolts, pipes, rivets) of the setup should be selected based on the standards followed in the local or international market.
Safety	• The rotating specimen should be held perfectly at both ends to avoid any catastrophic potential failure due to self-excited vibration. • Supporting components should be perfectly connected to the ground since the pipe could be highly vibrated. • Availability of labeling and safety warnings while test is running. The allowable distance from the setup will be labeled. • SQU workshops and laboratory safety instructions should be followed.

	• The pipe specimen should be as close as possible to the ground.
Time and Space	• The project should be done in two semesters • The size of the setup is limited to the workshops space in SQU.

The revised realistic constraints of the project are shown in the table below. These constraints have been implemented so far in the project where table 6 shows the way how each constraint is implemented.

Table 18 Implementation of Realistic Constraints

Category	Implementation of Realistic Constraint Mentioned in Table2
Performance	• The technician of the lab was told not turn off the power of the workshop while test is running. • The setup was constrained to tackle long pipe in length and the motor was bought with high speed.
Economic	• All the setup components were bought so that they do not exceed 2400 OMR in total..
Environmental	• No toxic materials were used in constructing the setup . • Recyclable materials were used for testing (metallic materials specimen). • Operating conditions were kept constant.
Conformity to Standards	• The components (bolts, pipes, rivets) of the setup were selected based on the standards followed in the local or international market.
Safety	• Two 12” chucks with 3 jaws are used to hold the testing pipe tightly • Four C-channels were used to Connect all the components of the setup. • labels and safety warnings were placed around while the test is running • SQU workshops and laboratory safety instructions were followed. • The tables were fabricated to have height of 0.5m from the ground.
Time and Space	• The dimensions of the setup were set according to the available space in the project room.

Chapter 8.0: Cost Analysis

The economic constraint is one of the most important constraints in this project. The budget is very limited (2400 RO from TRC and 200RO from the department) and it is important to make sure that the project cost will not exceed the limited amount. Therefore, conducting a cost analysis for this project is very critical in terms of determining whether it is possible to proceed in this project or not. Table 11 shows the cost of every component in the setup and whether it will be purchased or manufactured. The total cost as it appears in the table below is roughly estimated to be 1928 RO.

Table 19: Cost Analysis of the Setup

#Part	Name	Source	Price OR	Quantity	Total Price OR
Support Structure					
1	Table	In–house manufactured	45	2	90
2	Stabilizer	In–house manufactured	6	2	12
3	Cylindrical bearing	Local Market	38	4	76
4	Chuck	Local Market	90	2	180
5	Bolts and nuts	Local Market	-	24	-
Rotating Mechanism					
6	12 hp Motor	Local Market	120	1	120
7	Three stages Pulleys	Local Market	27.5	2	55
8	Housing shaft	Local Market	97.5	2	195
9	Belts	Local Market	-	3	-
Braking Mechanism					
10	Pneumatic actuator	ebay	15	2	30
11	Structure	In–house manufactured	200	1	200
12	Braking pads	Local Market	25	2	50
Instrumentation					
13	Variable speed Controller	UAE	470	1	470
14	Control panel of Braking system	Local Market	115	1	115

Others					
15	6m long, 0.5in Diameter-Pipes specimen	Local Market	12	6	72
16	6m long, 0.75in Diameter-Pipes specimen	Local Market	13	6	78
17	6m long, 1 in Diameter-Pipes specimen	Local Market	14	5	70
18	6m long, 1.5 in Diameter-Pipes specimen	Local Market	15	3	45
19	6m long, 2 in Diameter-Pipes specimen	Local Market	16	2	32
20	6m long, 3 in Diameter-Pipes specimen	Local Market	23	1	23
21	C-channels	Local Market	28.5	6	170
22	Welding job	-	-	-	125
23	cables	Local Market	-	-	20
24	miscellaneous	Local Market	-	-	100
				Total	2328

Chapter 9.0: Difficulties and Recommendations

9.1. Difficulties

During all stages of the project starting from studying the problem until testing the performance of the setup, there were many problems had to be solved and critical decisions had to be made.

9.1.1. Literature Review:

The fatigue problem is widely encountered in many industrial fields not just the oil and gas field. However, the problem is very difficult to simulate and investigate. There are only few setups around the world that were built to study the problem and they are limited to just one type of fatigue. Therefore, finding the proper references to base our study on was very difficult to accomplish. The team had to go through many papers and studies that explain the fatigue problem and the basic background of drilling operation to start confidentially the next stage.

9.1.2. Concepts Generation and Embodiment Design:

To build a setup that is able to create two types of fatigue, lateral and torsional, we had to make critical decisions about the design. There were many constraints that have to be considered especially budget and performance constraints. We had to generate concepts from scratch to come up with a proper design. In the embodiment design, especially in parametric phase, there were many unknown variables in selecting the motor, determining the proper size of the housing pipes and calculating the proper input torque to run the setup. these problems are considered in a way that the difficulties in the fabrication stage are reduced.

9.1.3. Fabrication Process:

In this stage, many problems were encountered because of lacking of the technical skills and inexperience about the local market. The materials that were selected in earlier stages to build the supporting mechanism were very expensive and we had to buy the materials from the scrap. Also, during the manufacturing process the workshop technicians were not available and there were many delays in the schedule.

There were many decisions had to be made during the fabrication process about the size of the components and there were some changes in the original design. For example, we used pneumatic actuators instead of the electrical one since the latest one is not available in the local market and the time was very limited. Therefore, we had difficulties in controlling the braking system since the pneumatic actuators are electrically difficult to control since they need to have precise pressure provided by a compressor. Also, the tables in the original design should be fixed to the ground. However, the tables were not stable during manufacturing, so we had to support them by C-channels and we used welding instead of bolts and nuts since it is easier.

9.2. Recommendation:

- Find a compressor that has a proper pressure control, otherwise an electrical actuator could be used for the braking mechanism instead of the pneumatic actuator.
- Replace belts and bullies with gear box or a chain to avoid any slippage that could happen between the belt and bully, and to enhance the applied torque.
- The setup structure should be fixed to the ground by any means to avoid the vibration that might buildup while running the test.
- The motor speed controlling unit should be monitored wirelessly in sake of safety.
- The chuck jaws should be well tightened to avoid any slippage that might occur between the tested specimen and the chuck jaws.
- A data acquisition system must be added to the setup to measure the pipe different mechanical parameters that could be affected while running the test.
- A surveillance system has to be used to record the test, especially when there is no one to observe the test 24 hours and to shot down automatically the setup power once the fatigue failure occurs.
- Speed sensors should be fixed along the setup so that the relative speeds of the tested specimen at different points can be measured directly; especially when the brake is applied.
- A proper instrumentation should be used to measure the angle of twist of tested pipe and measure the reduction in the torque applied by the motor.
- Add stress concentration like joints or initiate cracks with different orientations to study the effect of pre-existing to the time to failure.

This project needs a further development and a proper fund that could take the project into highly advanced stages to be much closer to real case of fatigue failure in drilling pipes.

Chapter 10.0: Conclusion

In conclusion, a study about the fatigue failure theory, the oil drilling operation, the types of failures occur in the drill-string and the common problem faced in the oil drilling operation has been conducted. A review of the related projects considering the fatigue failure resulted in the drill-string by reading the published journals about that has been done. It was found out that the common type of failure faced in the drill-string is the fatigue failure which causes the drill-string to fracture or induces cracks on it which will propagate and ultimately results in failure. In many related projects it was found that cracks occur at the pipes connection or nearby them. However, there was no project simulating the fatigue failure due to the rotation and vibration of the drill-string together only they were considering one type of motion. Therefore, it is needed to investigate more about the fatigue failure resulted due to the combination of rotation and vibration of the drill-string in the directional drilling for the sake of simplicity and the space constrain where it is difficult to simulate the vertical portion the drill-string. An experimental setup to simulate the fatigue failure due to lateral and torsional vibration in drilling pipes was designed and fabricated. The setup was able to generate the fatigue problem successfully. Many observation tests were carried out to check the general behavior of the pipe where the lateral and torsional vibration were generated. Hence, cyclic loading is there which is fatigue. Future work is to install proper instrumentations to be able get results and conduct a full test to find out the time to failure, number of cyclic to failure, maximum lateral shock that leads to failure and reeducation in torque that the brake provides.

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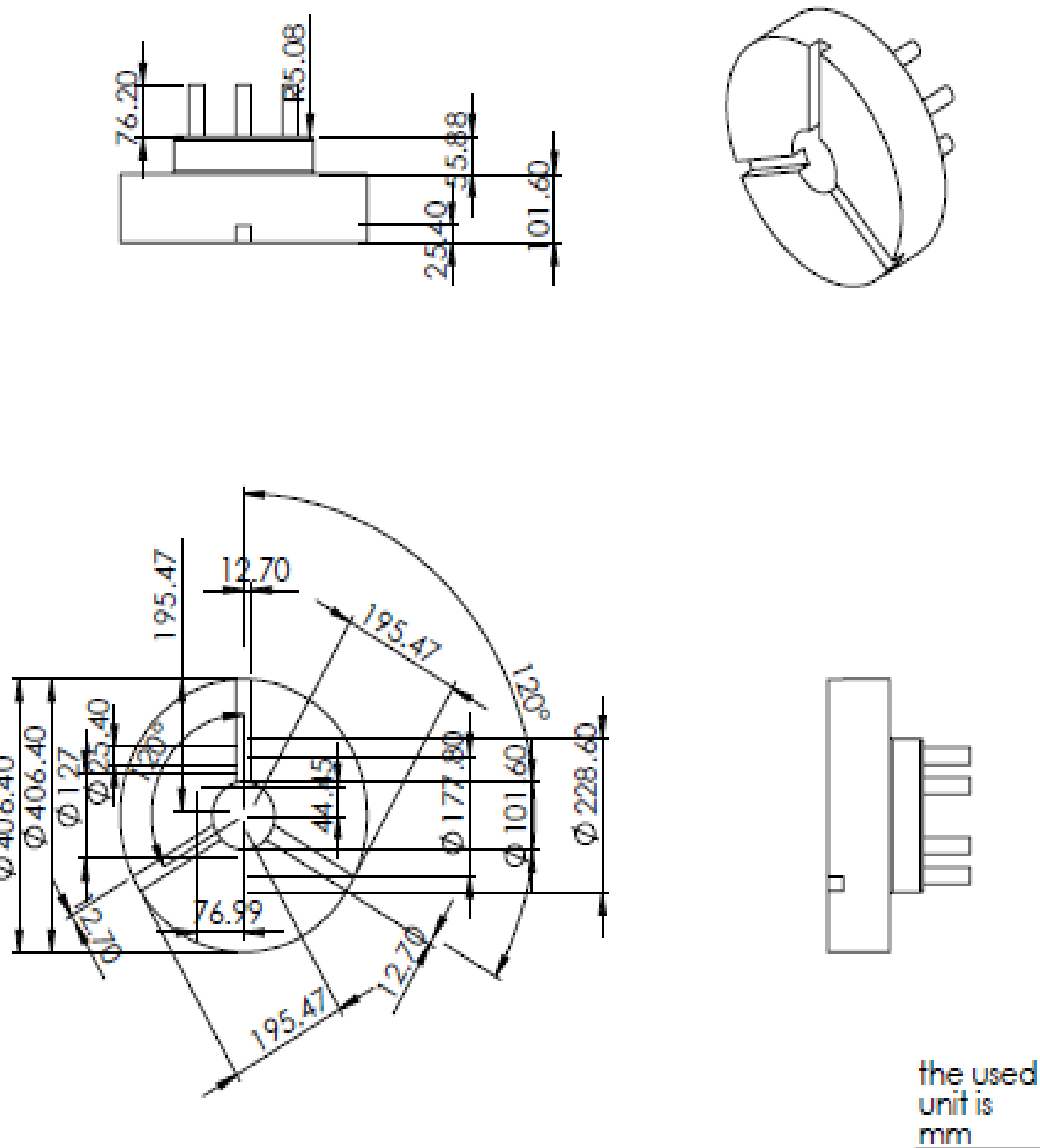
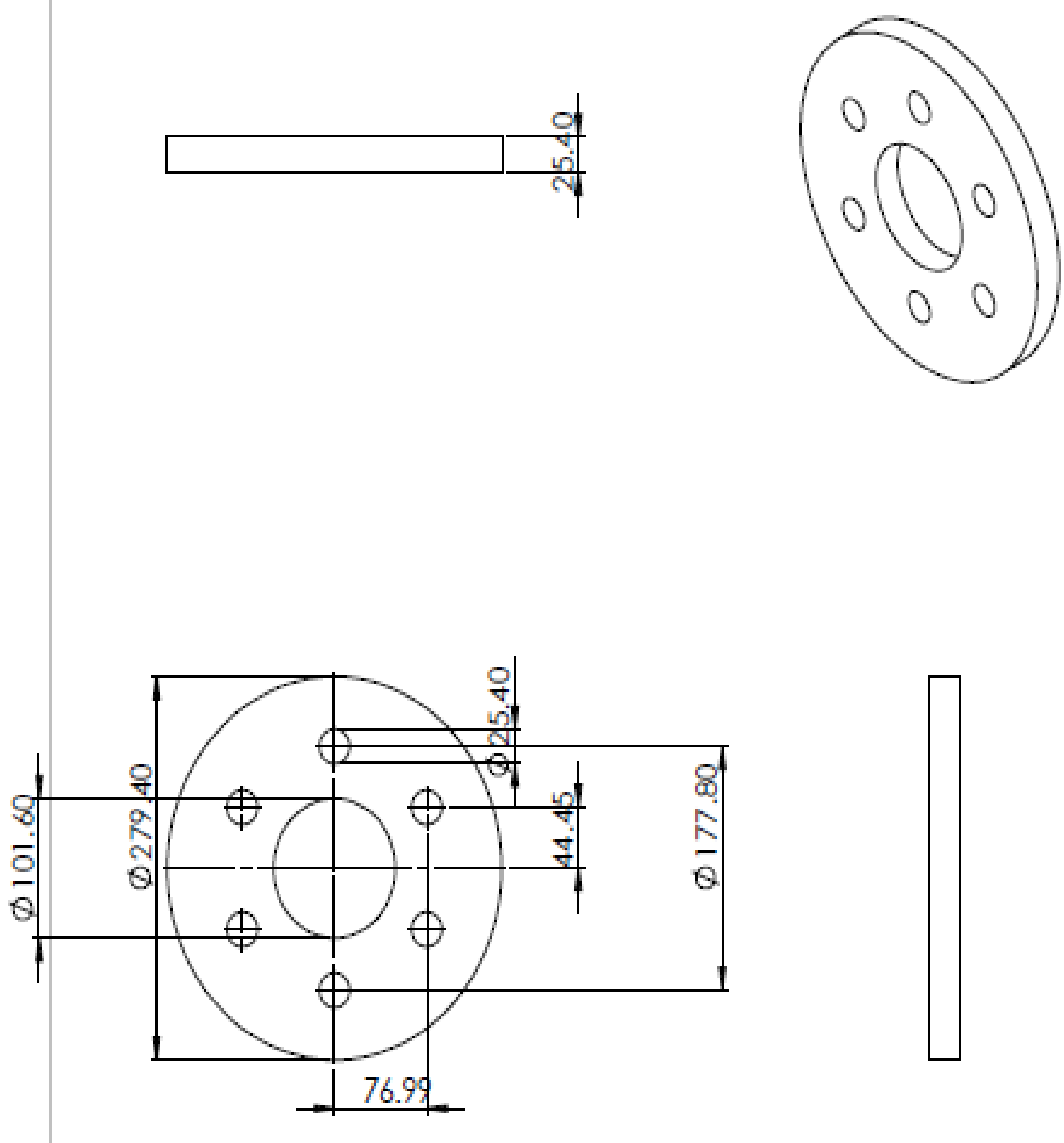
Appendix A

(Detailed Drawings and Codes and Standards)

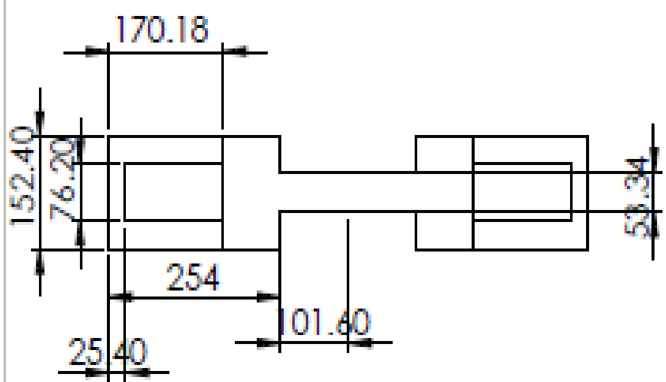
Chuck

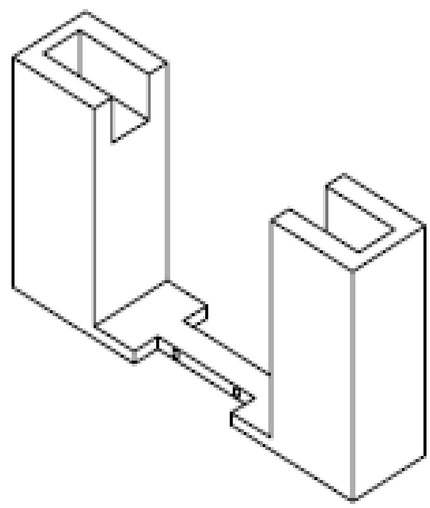

76.20
R5.08
55.88
101.60
25.40
195.47
12.70
195.47
120°
Ø 406.40
Ø 406.40
Ø 127
Ø 25.40
120°
44.45
Ø 177.80
Ø 101.60
Ø 228.60
76.99
12.70
12.70
195.47
the used
unit is
mm

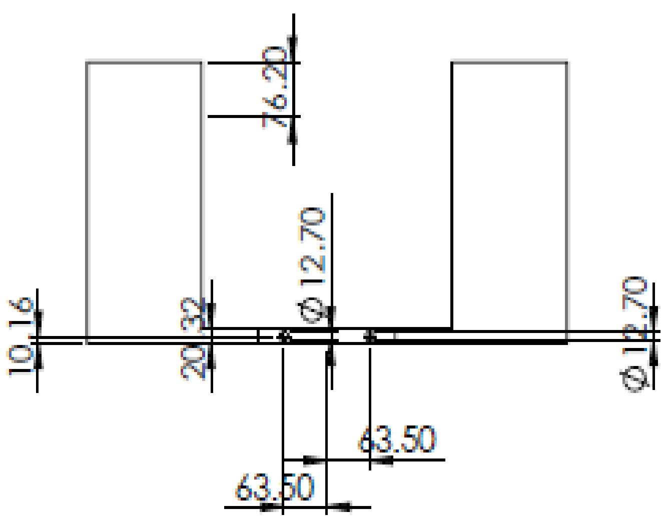
Flange (Coupling)

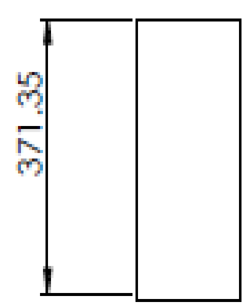


Supporting Structure of the Braking System

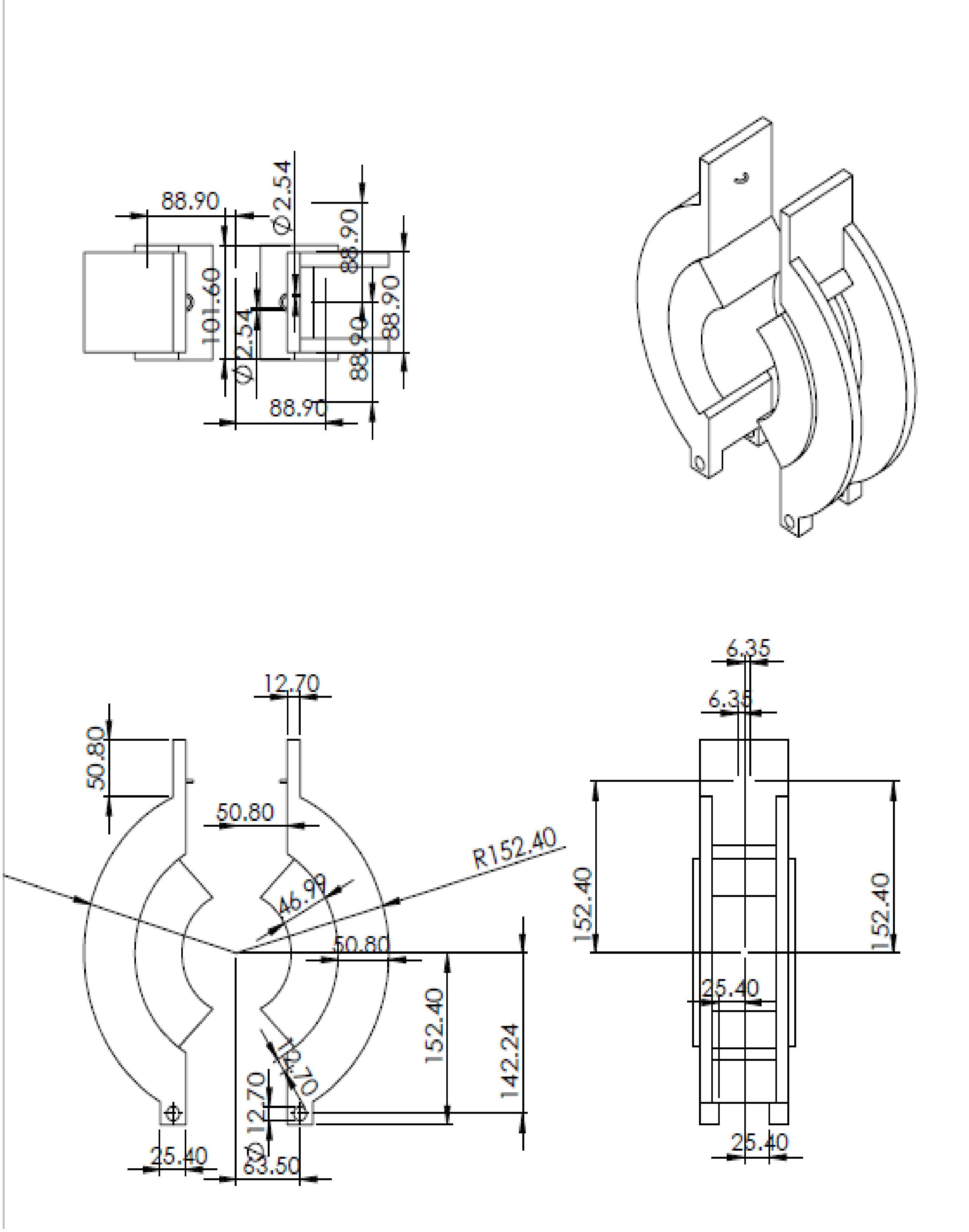




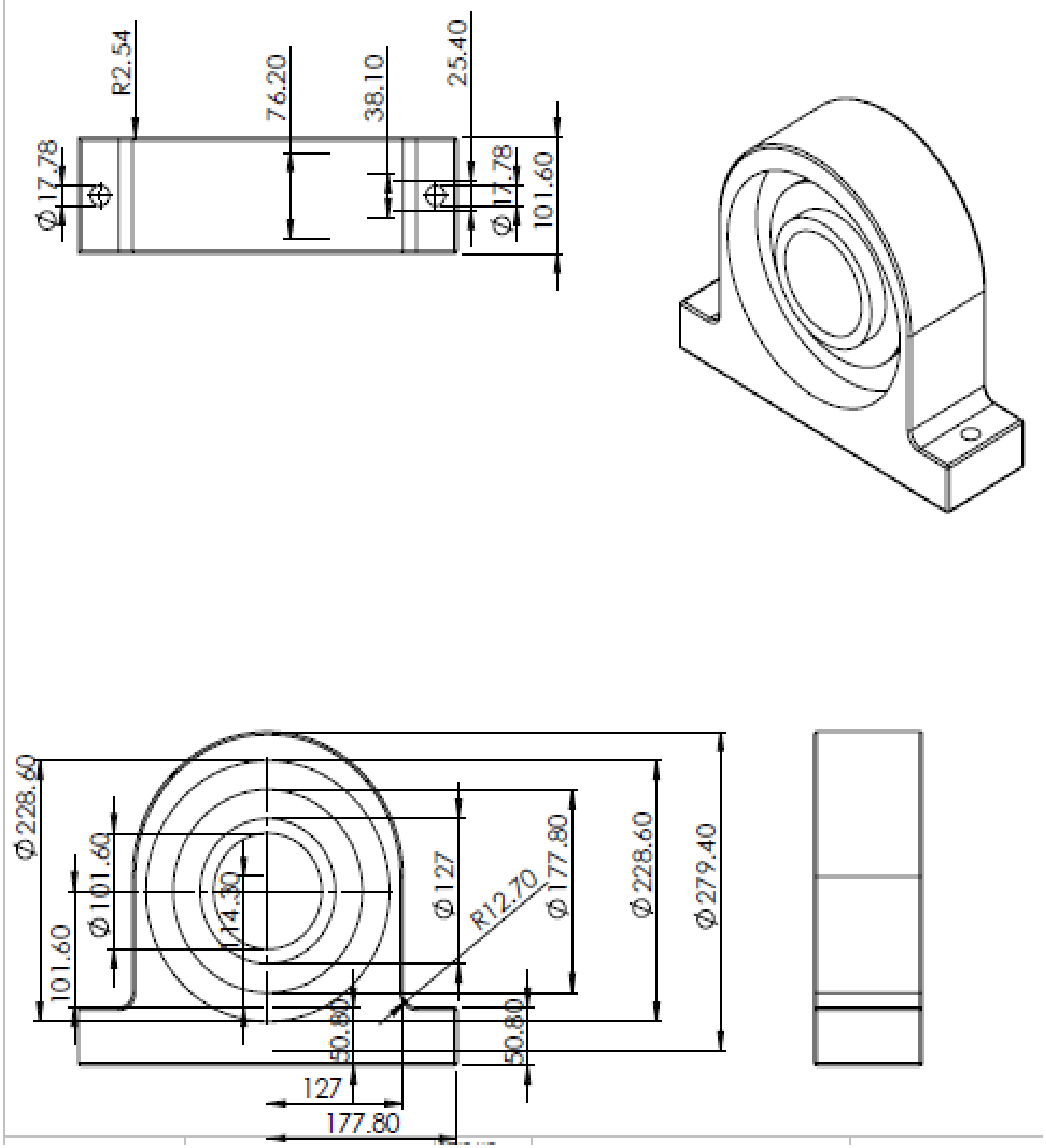




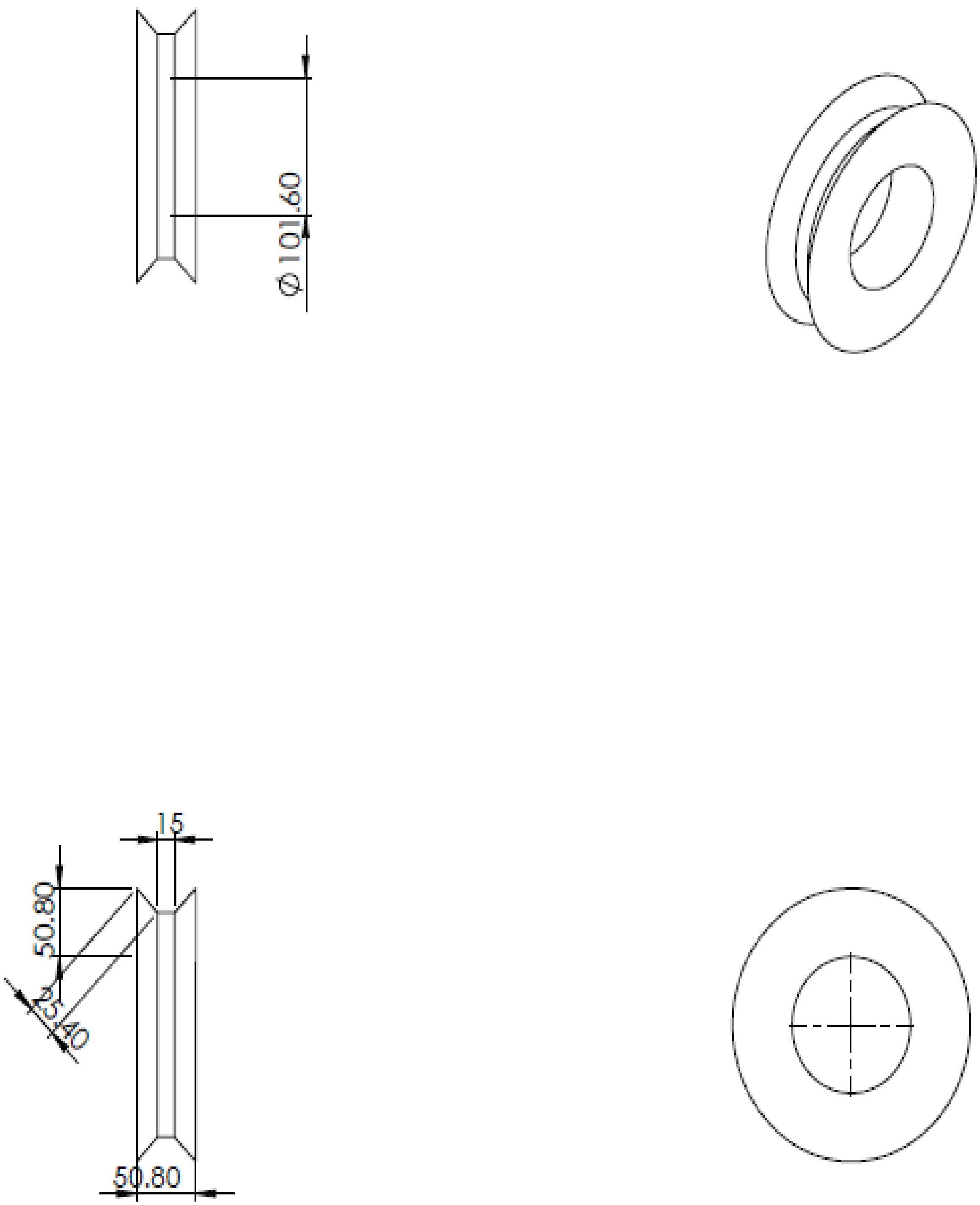
Clutch clamp



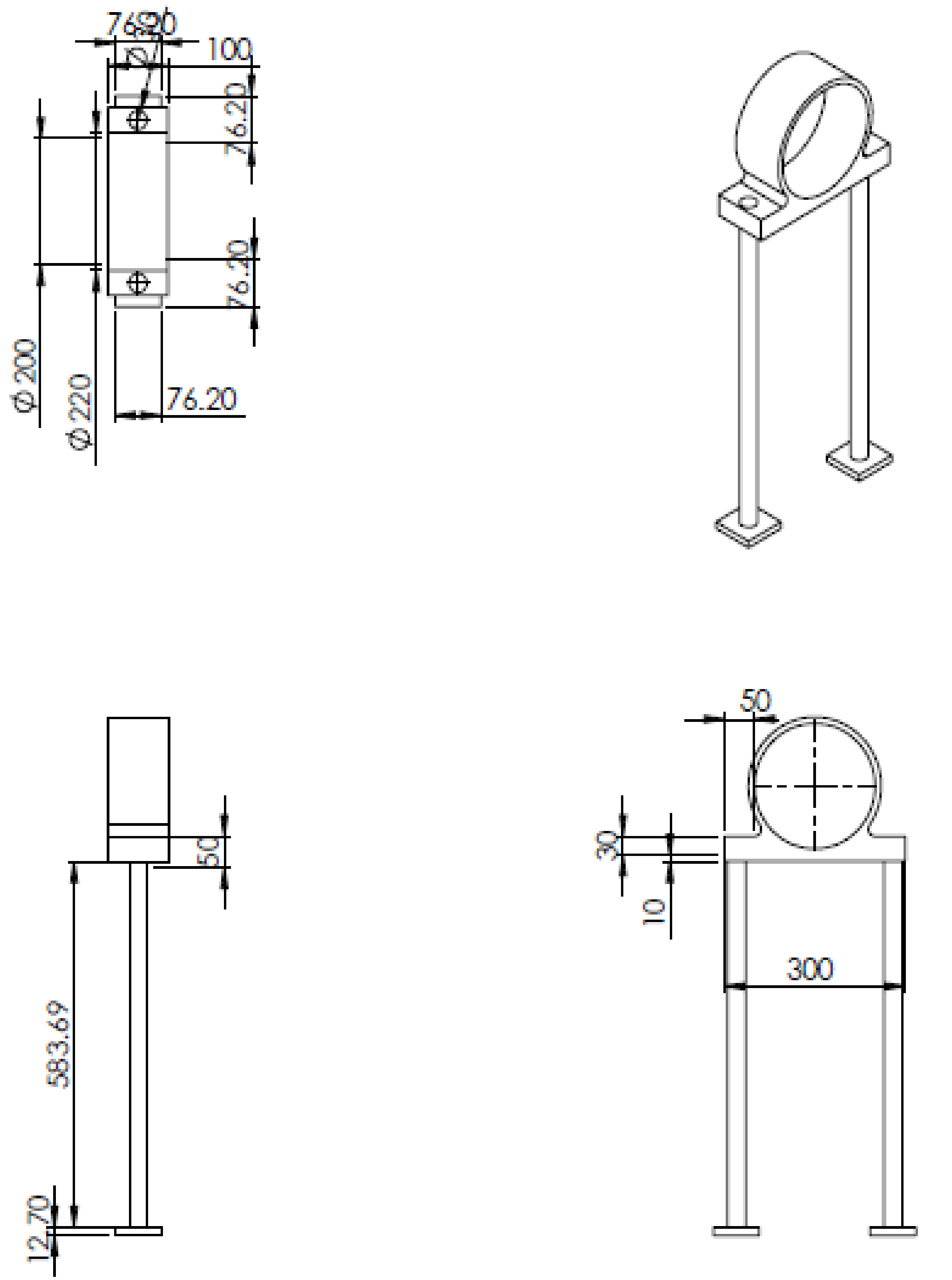
Pillow Bearing



Pulley



Stabilizer



➢ Codes and standards:

- Bearings selection:

In selecting the bearing, the available standards in Shigley's Mechanical Engineering Design textbook were used as shown below.

	02-Series				03-Series			
Bore, mm	OD, mm	Width, mm	Load Rating, kN C_{10}	Load Rating, kN C_0	OD, mm	Width, mm	Load Rating, kN C_{10}	Load Rating, kN C_0
25	52	15	16.8	8.8	62	17	28.6	15.0
30	62	16	22.4	12.0	72	19	36.9	20.0
35	72	17	31.9	17.6	80	21	44.6	27.1
40	80	18	41.8	24.0	90	23	56.1	32.5
45	85	19	44.0	25.5	100	25	72.1	45.4
50	90	20	45.7	27.5	110	27	88.0	52.0
55	100	21	56.1	34.0	120	29	102	67.2
60	110	22	64.4	43.1	130	31	123	76.5
65	120	23	76.5	51.2	140	33	138	85.0
70	125	24	79.2	51.2	150	35	151	102
75	130	25	93.1	63.2	160	37	183	125
80	140	26	106	69.4	170	39	190	125
85	150	28	119	78.3	180	41	212	149
90	160	30	142	100	190	43	242	160
95	170	32	165	112	200	45	264	189
100	180	34	183	125	215	47	303	220
110	200	38	229	167	240	50	391	304
120	215	40	260	183	260	55	457	340
130	230	40	270	193	280	58	539	408
140	250	42	319	240	300	62	682	454
150	270	45	446	260	320	65	781	502

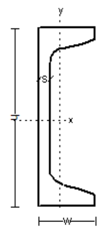
Figure 72 Dimensions and Basic Load Ratings for Cylindrical Roller Bearings

- C-channel:

The C-channels selected according the standards that exist in the local and international market. Figure8 shown screen capture of the standards

American Standard Steel Channels

Dimensions and static parameters of American Standard Steel Channels



Properties in imperial units of American Steel Channels are indicated below.

Designation	Dimensions					Static Parameters			
						Moment of Inertia		Elastic Section Modulus	
Imperial (in x lb/ft)	Depth h (in)	Width w (in)	Web Thickness s (in)	Sectional Area (in^2)	Weight (lb/ft)	I_x (in^4)	I_y (in^4)	W_x (in^3)	W_y (in^3)
C 15 x 50	15	3.716	0.716	14.7	50	404	11.0	53.8	3.78
C 15 x 40	15	3.520	0.520	11.8	40	349	9.23	46.5	3.37
C 15 x 33.9	15	3.400	0.400	9.96	33.9	315	8.13	42.0	3.11
C 12 x 30	12	3.170	0.510	8.82	30	162	5.14	27.0	2.06
C 12 x 25	12	3.047	0.387	7.35	25	144	4.47	24.1	1.88
C 12 x 20.7	12	2.942	0.282	6.09	20.7	129	3.88	21.5	1.73
C 10 x 30	10	3.033	0.673	8.82	30	103	3.94	20.7	1.65
C 10 x 25	10	2.886	0.526	7.35	25	91.2	3.36	18.2	1.48
C 10 x 20	10	2.739	0.379	5.88	20	78.9	2.81	15.8	1.32

Figure 73 Screen Capture of the Standards of C-Channel

- Nuts and Bolts:

Nuts and bolts and other fitting requirement were selected according to their availability in SQU's tools store.