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Palestine Polytechnic University



College of Engineering

Mechanical Engineering Department.

**A proper coupling of Engine-Generator: Design,
Analysis and Implementation.**

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Hebron-Palestine

2020

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**Submitted to the College of Engineering in partial fulfillment
of the requirements for the Bachelor degree in Mechanic
Engineering.**

Palestine Polytechnic University
College of Engineering
Department of Mechanical Engineering
Hebron - Palestine

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Analysis and Implementation**

By


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Mohammad Abu Dayeh

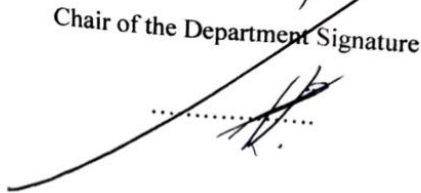
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Supervisor Signature


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Chair of the Department Signature


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Dedication

To our parents

To our university that we spent

Days in it...

To our supervisor who show us

The way to

Success...

To our college which

Provides us the

Knowledge

To E.Bader Al Mohtaseb Who Support us...

To our country Palestine which always in our hurts

Mohammad Abu Dayeh

Yousef Hamad

Khalil Mohammad

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Abstract

The internal composition engine laboratory has an essential device in Mechanical engineering study which is Dynamometer. This device let students consolidate the theoretical information in their study, but there is an essential need to run this device again to obtain this target, but the problem is: the mechanical coupling between engine and generator (two main parts of dynamometer) has been broken after few seconds of device's running when our colleagues tried to rehabilitate this device last year, since they chose improper mechanical coupling to their device So, the mission was: first step was to choose number of flexible couplings of this device not rigid type to avoid any serious damage if coupling break and to accommodate misalignment. Second step was to make a comparison between them to choose the most proper coupling to device condition. Third step was to make a design for our chosen coupling, all these steps had been done in the first semester and we implemented our coupling design in second semester , run the device again and make many experiments on it like brake power, fuel consumption and mass fuel rate.

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Chapter

one

Introduction

1.1 Introduction.

1.2 Project Overview.

1.3 Project Objectives.

1.4 Project Description.

1.5 Tasks Description Table.

1.6 Time Table.

1.7 Project Cost Estimation.

1.1 Introduction:

Dynamometer is one of the most important devices in internal combustion engine laboratory, since it let the automotive student perform their study's experiments, so our target was to rehabilitate this device by build a proper mechanical coupling for it.

1.2 Project Overview:

There is a device in internal combustion engine laboratory which called dynamometer consist of single cylinder internal combustion engine or motor, AC generator, electric and sensitive load unit .The motor converts the chemical energy from fuel to a mechanical energy through an output shaft and by a mechanical coupling converts this effective load to an AC generator to represent the motor's load electrically using a strain gauge, so our device is one of most important devices in the automotive engineering study, since it lets the automotive engineering students to perform what did they learn in internal combustion engine courses experimentally by using this device the laboratory.

In this device which has been rehabilitated, the coupling between motor and generator has been broken after run the device and the strain gauge sensor which test the rotating equipment broke by broken coupling, so after bought a new strain gauge sensor by Palestine Polytechnic University, our main goal and mission was to build a new mechanical coupling which converts the load from motor to AC generator without any problem or defect.

1.3 Project Objectives:

The objective of the project was to solve the current situation that the internal combustion engine laboratory faced in dynamometer device which was a selection of proper coupling for our device and steps of our project were as follow: -

1. Make a short research in universal websites and scientific papers to select the best coupling for our project.
2. Make a proper design to the coupling using one of design programs such as SOLIDWORKS.
3. Implement our work on our device and make an equilibrium to the device by put the motor-generator inline using special gauges.
4. Avoid vibration by put pieces of rubbers between device table which hold the system and dynamometer.

1.4 Project Description:

Simulated the effective load electrically to show the motor performance and its change graphically by using a P.C monitor and that was not happened if we not rehabilitated the dynamometer by design, analysis and implement a new mechanical coupling for dynamometer.

After faced many obstacles in the project, like there was a misalignment between motor and generator's shafts and vibration sources, so the task was to choose the most proper coupling to the device's condition to accommodate the misalignment and absorb the vibration to achieve the smoothest and regulate torque from motor to generator. After chose many types of coupling, compared them, chose the most proper one, chose coupling's material, designed the chosen coupling and fit it to the device to operate smoothly and correctly.

1.5 Tasks Description Table:

In this section the task description was as show in Table (1.1).

Table 1.1: Tasks Description

Task ID	Tasks Description
	1 semester
T1	Project selection.
T2	Collection data about our project.
T3	Put the major project's steps.
T4	Study the project's steps to proceeds it.
T5	Start to implement our project's steps.
T6	Maintained and repair.
T7	Make a project modification.
T8	Write the first semester project and presentation.
	Second semester
T9	Edit the discussed problem points in our project.
T10	Build our designed coupling.
T11	Install the mechanical coupling in our device.
T12	Test the device performance with a new coupling.
T13	Perform experiments on device and collect the data.
T14	Analysis the data that we collect.
T15	Writ the second semester project and presentation.

1.6 Time Table:

Our project's time description was as the following table (1.2).

Table 1.2: Time Table

Tasks	First semester
T1	Week 1,2
T2	Week 3,4
T3	Week 5,6
T4	Week 7,8
T5	Week 9,10
T6	Week 11,12
T7	Week 13,14
T8	Week 15,16
Second semester	
T9	Week 1,2
T10	Week 3,4
T11	Week 5,6
T12	Week 7,8
T13	Week 9,10
T14	Week 11,12,13
T15	Week 14,15,16

1.7 Project Cost Estimation:

This section includes all software and hardware costs of our project in the following table (1.3).

1 Table 1.3: Cost and budget

Component	Price (NIS)
Computer programs	500
Vibration dampers	200
Iron washer stability	200
Motor-generator coupling	800
Cut and welding	600
Total	2300

CHAPTER

TWO

Literature review

2.1 Coupling function.

2.2 General application or uses.

2.3 Requirements of a good coupling.

2.4 Types of couplings.

2.5 Difference between rigid and flexible couplings.

2.6 Rigid and flexible coupling comparison.

2.7 Why chosen flexible coupling type not rigid one?

2.8 Flexible element.

2.9 Advantages and Limitations of all coupling.

2.10 Conclusion.

2.1 Coupling function:

In simple words, couplings are mechanical devices used to transmit torque from one shaft to another shaft.

2.2 General application:

- To transmit power from drive shaft to driven one.
- To connect or couple two components which are manufactured separately e.g. output motor shaft and generator.
- To introduce an extra flexibility while transmitting power in case of space restrictions.
- To introduce protection against overloads.
- To reduce the transmission of shock loads from one shaft to another by using flexible couplings.

2.3 Requirements of a good coupling:

- It should be capable of transmitting torque from driving shaft to drive shaft.
- It should align the shafts properly.
- It should be able to dismantle for the purpose of maintenance.
- It should provide safety to human being and machine in case of coupling failure. [1]

2.4 Types of couplings:

There are two types of couplings:

1. Rigid coupling.
2. Flexible coupling.

Rigid couplings are a type of couplings that should only be utilized when shafts are precisely in line. A rigid coupling is only suitable for shafts in close alignment or held in alignment. If not aligned, torque will transfer to the shafts and bearings and likely cause premature failure.

Flexible couplings are used to transmit torque from one shaft to another when the two shafts are slightly misaligned. Flexible couplings can accommodate varying degrees of misalignment up to 3° and some parallel misalignment.

Examples of rigid coupling muff coupling, clamp coupling, rigid flange coupling etc. [2]

Example of flexible coupling is tire coupling jaw coupling etc.

2.5 Difference between rigid and flexible couplings:

- Flexible coupling can accommodate a slight misalignment between shafts but a rigid coupling cannot accommodate any misalignment.
- Flexible coupling can absorb shocks and vibrations but a rigid coupling cannot absorb shocks and vibrations.
- Rigid coupling is simple and inexpensive but flexible couplings are expensive. [2]

2.6 Rigid and flexible coupling comparison:

Table 2.1: Rigid and flexible coupling comparison.

	Rigid coupling	Flexible coupling
Purpose	Connect two shafts perfectly aligned	Connect two shafts having misalignment
Alignment	Cannot bear misalignment	Bear misalignment
Shocks	Cannot absorb shocks of vibration	Absorb shocks or vibration
Deflection	Less shaft's deflection	More shaft's deflection
Cost	Low	High

2.7 Why chosen flexible coupling type not rigid one?

Rigid coupling is a type of coupling that should only be vitalized when shafts are precisely in line which not achieved in our device between motor and generator. However flexible coupling can accommodate a slight misalignment between motor and generator shaft.

Flexible coupling can absorb shocks and vibration, but rigid coupling can't absorb these effects which occur in our device.

From previous contrast, it let us know that flexible coupling is more proper for our device not rigid coupling one.

2.8 Flexible element:

A) Oldham coupling:

The Oldham coupling consists of three discs. Two of the discs, typically made of aluminum or stainless steel, are connected to either side of the drive, while the third, made from one of several different plastics, is sandwiched in between with a tongue and groove design. The tongue and groove on one side is perpendicular to the tongue and groove on the other. Springs are often used to reduce the coupling's backlash. [3]

- 1) Handle high degrees of parallel misalignment to their sliding element design.
- 2) Some manufactures claim an ability to tolerate up to 5 degree angular.



Figure 2.1: Oldham coupling.

B) Jaw coupling:

A jaw coupling is a type of general purpose power transmission couplings that also can be used in motion control (servo) applications. It is designed to transmit torque (by connecting two shafts) while damping system vibrations and accommodating misalignment, which protects other components from damage. Jaw couplings are composed of three parts: two metallic hubs and an elastomer insert called an element, but commonly referred to as a "spider". The three parts press fit together with a jaw from each hub fitted alternately with the lobes of the spider. Jaw coupling torque is transmitted through the elastomer lobes in compression. [3]

Functions:

- 1) Used to transfer light and medium power level.
- 2) Make a vibration damping, since it's had a spider insert.
- 3) Parallel misalignment up to 0.25 (mm).
- 4) Angular misalignment up to 1 degree.
- 5) Torque below 110 (Nm).

Advantages:

- 1) Operate in a wide band of temperatures.
- 2) Can handle angular misalignment.
- 3) Can handle reactionary loads due to misalignment.
- 4) Have good torque to outside diameter capability.
- 5) Have good speed capability.
- 6) Good chemical resistance (spider dependent).
- 7) Decent dampening capability.

Limitations:

Traditional jaw couplings are limited in operating temperature to what can be handled by the elastomeric spider. These couplings do offer moderate misalignment handling capabilities (both in angular and parallel), but some applications may require more radial stiffness may introduce high reactionary loads in these couplings. [3]



Figure 2.2: Jaw coupling. 1

C) Pin bush coupling:

This is used for slightly imperfect alignment of the two shafts.

This is modified form of the protected type flange coupling. This type of coupling has pins and it works with coupling bolts. The rubber or leather bushes are used over the pins. The coupling has two halves dissimilar in construction. The pins are rigidly fastened by nuts to one of the flanges and kept loose on the other flange. This coupling is used to connect shafts which have a small parallel misalignment, angular misalignment or axial misalignment. In this coupling the rubber bushing absorbs

shocks and vibration during its operations. This type of coupling is mostly used to couple electric motors and machines. [2]

■ **Consist of:**

- 1) Two hubs.
- 2) Number of bolts, washers and nuts.
- 3) Flexible buffer with spherical shape.

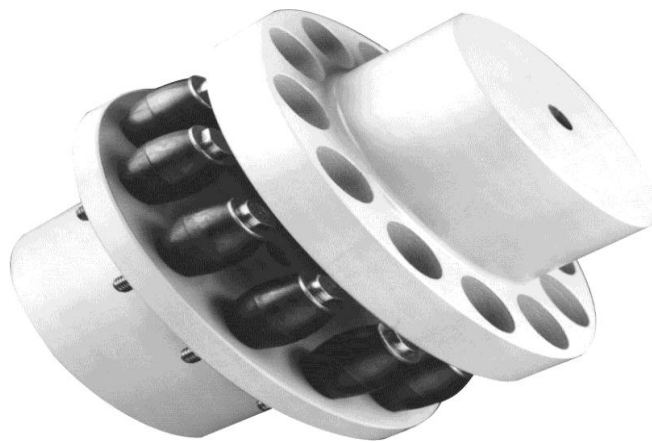


Figure 2.3: Pin bush coupling 1

Advantages:

- 1) Torsional stiff.
- 2) Free maintenance.
- 3) Good vibration damping and shock absorbing.

Limitations:

- 1) Sensitive to chemical and high temperature.
- 2) Not torsional stiff for positive displacement.
- 3) Difficult to balance as an assembly. [3]

D) Tire coupling:

These couplings have a rubber or polyurethane element connected to two hubs. The rubber element transmits torque in shear.

- **Consist of:**

- 1) Two hubs.
- 2) Rubber elements.
- 3) Bolts.



Figure 2.4: Tire coupling. 1

- **Advantages:**

- 1) No lubrication or maintenance are required.
- 2) Good vibration damping and shock absorbing.
- 3) Less expensive than metallic coupling.
- 4) Misalignment allowable than most metallic coupling.

- **Limitations:**

- 1) Sensitive to chemicals and high temperatures.
- 2) Usually not torsional stiff enough for positive displacement.
- 3) Large in outside diameter than metallic coupling.
- 4) Difficult to balance as an assembly. [4]

E) Diaphragm coupling:

Diaphragm couplings utilize a single or a series of plates or diaphragms for the flexible members. It transmits torque from the outside diameter of a flexible plate to the inside diameter, across the spool or spacer piece, and then from inside to outside diameter. The deflection of the outer diameter relative to the inner diameter is what occurs when the diaphragm is subject to misalignment. [4]

■ Function and Specification:

- 1) Used to transfer high power.
- 2) Used to transfer high torque.
- 3) High speed operation.
- 4) Lubrication isn't required.
- 5) Have flexible metal one or more.

■ Consist of:

- 1) Two hubs.
- 2) Plate pack membrane.
- 3) Bolts and nuts.
- 4) Collar catch ring.



1 Figure 2.5: Diaphragm coupling.

■ Failure cases:

- 1) Failure due to fatigue or misalignment.
- 2) Overload torque.
- 3) Angular misalignment.

2.9 Advantages and Limitations of all coupling:-

Each of the couplings mentioned above has advantages and limitations, which were mentioned in the following table.

Table 1.2: advantages and limitation of each couplings.

	Advantages	Limitations
Oldham coupling	<ol style="list-style-type: none"> 1) Handle high degrees of parallel misalignment to their sliding element design. 2) Some manufactures claim an ability to tolerate up to 5 degree angular. 	<ol style="list-style-type: none"> 1) It accommodates a relatively small angular misalignment. 2) It has lower peak torque and torsional stiffness. 3) There is an angular misalignment.
Jaw coupling	<ol style="list-style-type: none"> 1) Operate in a wide band of temperatures. 2) Can handle angular misalignment. 3) Can handle reactionary loads due to misalignment. 4) Have good torque to outside diameter capability. 5) Have good speed capability. 6) Good chemical resistance (spider dependent). 7) Decent dampening capability. 	Traditional jaw couplings are limited in operating temperature to what can be handled by the elastomeric spider. These couplings do offer moderate misalignment handling capabilities (both in angular and parallel), but some applications may require more radial stiffness may introduce high reactionary loads in these couplings.
Pin bush coupling	<ol style="list-style-type: none"> 1) Torsional stiff. 2) Free maintenance. 3) Good vibration damping and shock absorbing. 	<ol style="list-style-type: none"> 1) Sensitive to chemical and high temperature. 2) Not torsional stiff for positive displacement. 3) Difficult to balance as an assembly.

Tire coupling	<ol style="list-style-type: none"> 1) No lubrication or maintenance are required. 2) Good vibration damping and shock absorbing. 3) Less expensive than metallic coupling. 4) Misalignment allowable than most metallic coupling. 	<ol style="list-style-type: none"> 1) Sensitive to chemicals and high temperatures. 2) Usually not torsional stiff enough for positive displacement. 3) Large in outside diameter than metallic coupling. 4) Difficult to balance as an assembly.
Diaphragm coupling	<ol style="list-style-type: none"> 1) Less axial movement capability. 2) The outside diameter is larger for the same rated torque. 	<ol style="list-style-type: none"> 1) Failure due to fatigue or misalignment. 2) Overload torque. 3) Angular misalignment.

2.10 Conclusion:

In this project and after review those couplings, the jaw coupling was the most proper coupling for this project according to many reasons:

- 1) Easy to use, install and accommodate a misalignment.
- 2) Used to transfer light and medium power level which needed to transfer from motor to generator.
- 3) Make a vibration damping, since it has a spider insert.
- 4) Parallel misalignment up to 0.25 mm which would have in this device.
- 5) Angular misalignment up to 4 degree.
- 6) Torque below 110 (N.m).

CHAPTER THREE

**Jaw coupling, features,
benefits, parts and design.**

- 3.1 Reasons and benefits of choose a jaw coupling.**
- 3.2 Construction of jaw coupling and its use advantages.**
- 3.3 Spider (Elastomer).**
 - 3.4.1 Types of spider.**
 - 3.4.2 Materials construct the previous types of spider.**
 - 3.4.3 The failure in jaw coupling happened in spider part, but in different cases.**
- 3.5 Why chosen (PP Copolymer) as a coupling spider's material?**
- 3.6 Why chosen Aluminum 1060 alloy as a coupling hub's material?**
- 3.7 Limitations.**
- 3.8 Sizing and selection of elastic jaw coupling.**
- 3.9 Shaft coupling's design equations and its calculation:**

3.1 Reasons and benefits of choose a jaw coupling:

Jaw coupling is the most popular coupling since it's:

1. Versatile and robust.
2. Used especially for light and medium power transmission.
3. Has a high temperature range.
4. Used for many general purposes.
5. Can transfer loads through a few degrees of misalignment.
6. Has a good torque and speed capability.
7. Has a good damping resistance, since it consists of a spider element.
8. Has a chemical resistance.
9. Easy to install.

3.2 Construction of jaw coupling and its use advantages:

Jaw coupling consist of three main parts: two hubs and one elastomeric spider.

If jaw coupling fail, it will not make any damage for the device, especially encoder part, down of jaw coupling.

After bought a new encoder, since fail of rigid coupling in previous coupling design.

However, chosen coupling which was jaw coupling have a spider insert, so if it comes to fail, the coupling will continue its work to transfer torque from motor to generator.

The only defect will be lost of damping capability; however, hubs will interlock and continue carrying loads.

Fail in spider alert the user to replace the coupling to avoid a serious damage. It should know that the spider and jaw coupling in general are maintenance free, for example it doesn't require any lubrication. [2]

3.3 Spider (Elastomer):

An essential component of jaw couplings, flexible elastomeric spiders transmit torque and accept misalignment. Design and material options for these flexible elements let you tailor them to the application.

3.4.1 Types of spider:

- Solid-center spider: this is the most commonly used design for applications where the distance between ends of driving and driven shafts is large enough to accommodate the spider thickness. [5]



Figure 3.1: Solid-center spider 1

- Open-center type (OCT) spider: this type is used in situations where shaft ends must be positioned closer together than the solid center design allows. A thin segment of elastomeric material connects the spider legs, and fits in a small space between the jaw inner edges and the hub bore. The spider ID is slightly smaller than the hub bore, so that it overlaps the bore. Because the spider's legs are joined only by a thin segment of material, they have limited support. Accordingly, speed is limited to 1,750 rpm for (PP copolymer) and 3,600 rpm for urethane and Hytrel. Cost of the open center type is about the same as the solid center type.



Figure 3.2: Open-center type spider 1

- Third variation is spider which wrap around the outside of coupling with ring to hold the spider on coupling which made its replacement easier than two previous of coupling, but this type has a higher cost and suitable only for low speeds.



Figure 3.3: Snap-wrap spider 1

3.4.2 The failure in jaw coupling happened in spider part, but in different cases:

- **Case one: normal wear similar on all contact point:**

The image below shows an example of normal even elastomeric jaw coupling spider wear for a coupling that is operated in one direction only. The wear is even and linear, indicating that misalignment does not appear to be an issue with this specific application. When a jaw coupling runs in one direction, only every other spider leg is put under compression. (If the application were to run in both directions, called a reversing load, you would expect to see similar wear on every leg of the spider).



Figure 3.4: Normal wear similar on all contact point.

- **Case two: overload failure similar on all contact points when it faced a very high torque.**

When sizing and selecting a coupling, it is always important to identify the appropriate service factor for the given application, and include it when calculating the coupling sizing torque. (For a great tutorial on how to calculate coupling sizing torque.

Overload or over-torque occurs when the coupling is subjected to peak torque load(s) in excess of the calculated coupling sizing torque.



Figure 3.5: Overload failure.

■ **Case three: wear due to misalignment.**

Flexible couplings are designed to accommodate for system misalignment. Couplings are designed and rated to handle a certain amount of angular, parallel, and axial misalignment, but users do not always stay within these published ratings. The tell tail sign that misalignment is causing premature jaw coupling is that the jaw coupling spider is wearing unevenly... either at an angle or not across the full face of the spider legs. Spider "dust" may also be present in coupling or coupling guard. [5]



Figure 3.6: Wear due to misalignment 1.

- **Case four: chemical failure.**

On chemical failures... if an elastomeric spider is in contact with chemicals that it is not compatible with, the spider will generally rapidly deteriorate and fail, sometimes breaking off in pieces. Coupling manufacturers will generally include compatibility guidelines in their catalogs for each elastomer material relative to traditional chemicals that a coupling may come in contact with. [5]



Figure 3.7: Chemical failure.

- **Case five: vibration failure.**

What causes this failure is a heat buildup in the spider caused by the spiders attempt to absorb damaging high natural frequencies in a system, generally caused by a reciprocating engine (diesel, gas, natural gas, liquid propane), pump, or compressor. [5]



Figure 3.8: Vibration failure.

3.4.3 Materials that construct the previous types of spider:

- 1) (PP Copolymer): This is the most economical and widely used coupling element material. It resembles natural rubber in resilience and elasticity, plus resistance to oil, hydraulic fluid, and most chemicals. Its operating temperature ranges from -40 to 100°C. (PP Copolymer) provides the best damping capability among elastomeric elements.
- 2) Hytrel: it has a high temperature range, high torque capability, with excellent resistance to oil and chemical but it recommended for low degree of misalignment and it doesn't suitable for start/stop application. Cost is 3 times of (PP Copolymer).
- 3) Bronze: Not really elastomeric, it's rigid suitable for low speed application (used 250 RPM), but its capability for misalignment is low. Cost is at least 3 times of (PP Copolymer).
- 4) Urethane: it has a high torque capability with temperature range of (-34 to 71°C), it has a good resistance to oil and chemical, but its damping capability is low and it doesn't suitable for start/stop application. Its cost is 1.5 to 2 times of (PP Copolymer). [5]

Table 3.1: Contrast between spider materials:

Spider material	Temperature range (°C)	Misalignment capability		Torque capability
		Angular (degree)	Parallel (mm)	
(PP Copolymer)	-40 to 100	1°	0.35	High
Urethane	-34 to 71	1°	0.35	High
Hytrel	-51 to 121	0.5°	0.35	High
Bronze	-40 to 232	0.25°	0.25	High

Spider material	Start/stop application	Damping capability	Oil and chemical resistance
(PP Copolymer)	recommended	High	Excellent
Urethane	Not recommended	Poor	Good
Hytrel	Not recommended	Good	Excellent
Bronze	recommended	Poor	Excellent

3.5 Why (PP Copolymer) has been chosen as a coupling spider's material?

The most proper material for coupling spider's material is (PP Copolymer), since it has best damping capability among elastomeric element's, good temperature ranges from (-40 to 100 °C), high torque capability, misalignment capability (1°) degree in

angular misalignment and 0.35 mm in parallel one and it has an excellent oil and chemical resistance.

3.6 Why [Aluminum 1060 Alloy] has been chosen as a coupling hubs material?

The most proper material for coupling hubs is Aluminum 1060 Alloy since, it has a high machinability and good wear resistance as well as good vibration damping. It can withstand greater load and has a good degree of resistance against corrosion.

3.7 Limitations:

Traditional jaw coupling has limitations in some field:

- a) Its temperature range according to its spider material.
- b) Its capability to hand misalignment is limited in both angular and parallel, since some applications may want more.
- c) Isn't well suited for motion control applications, since it hasn't a backlash.

3.8 Jaw coupling drawing assembly on the SOLIDWORKS program:

1- Jaw coupling: Jaw coupling drawing in SOLIDWORKS program in assembly view.

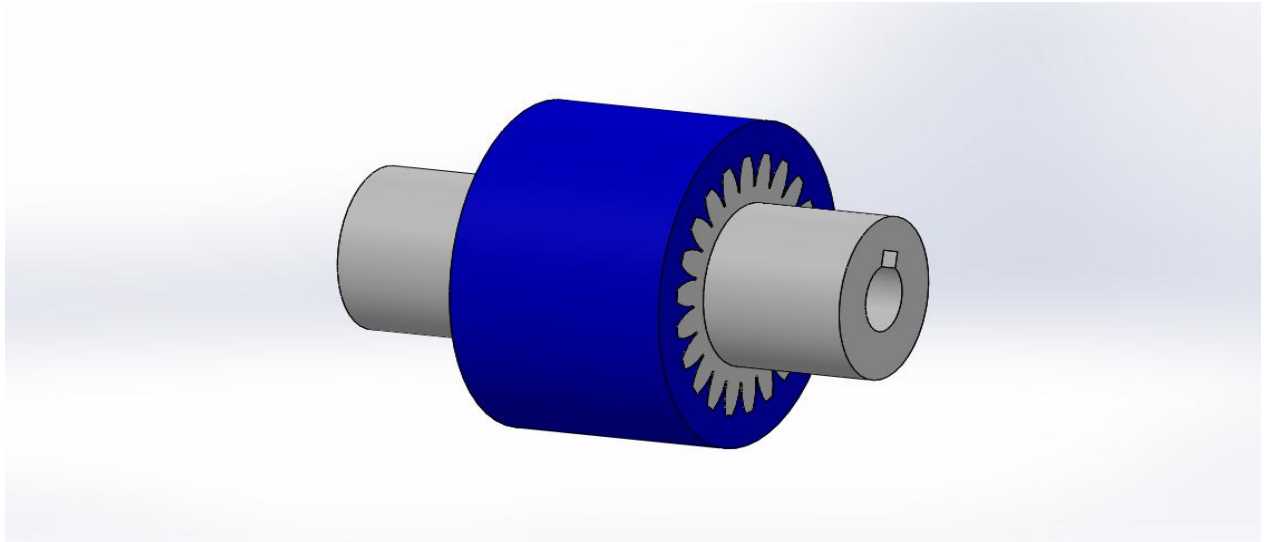


Figure (3.9): Jaw coupling drawing1

2-coupling spider: Coupling spider drawing in SOLIDWORKS program in part view.

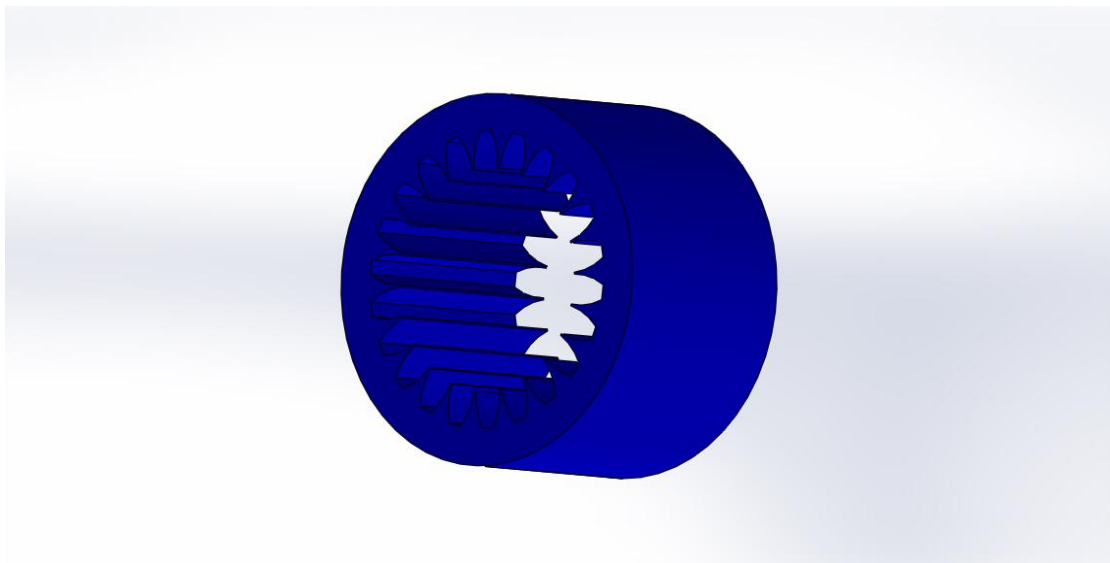


Figure (3.10): coupling spider drawing.2

3-Coupling hubs: Coupling hub drawing in SOLIDWORKS program in part view.

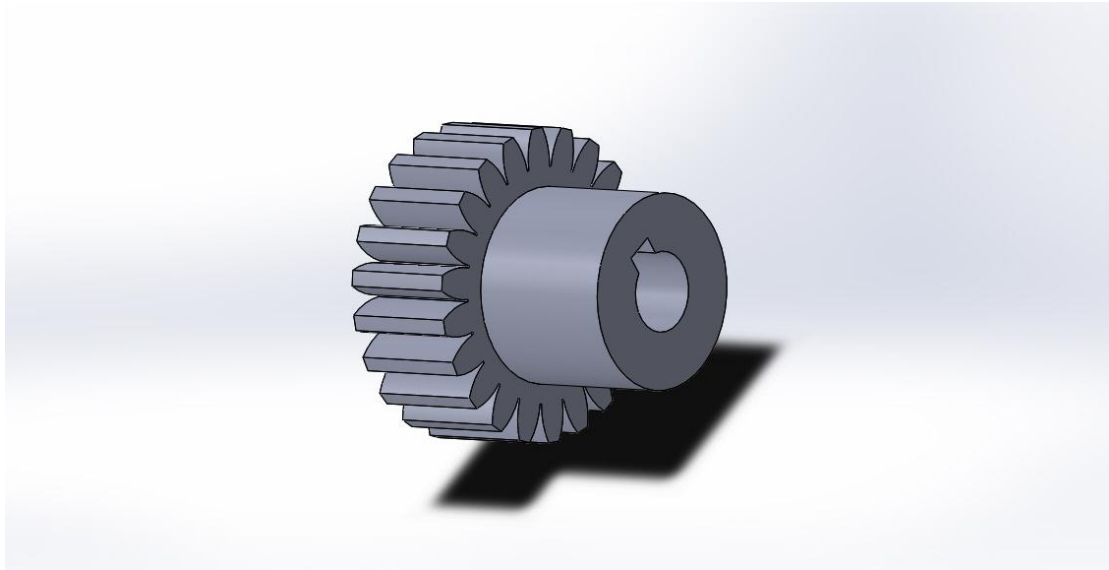


Figure (3.11): coupling hub drawing. 3

3.10 Shaft coupling's design equations and its calculation:

The main object of this section is to show how the system design works safely without structure failure. The maximum shear stress occurs at the outer radius of the coupling and at the interface of the two bores. The need was to derive and calculate the stresses because it shows the parallel and perpendicular forces applied to jaw coupling.

This stress derived from the torsion formula:

$$\tau_{max} = \frac{T \times D_o}{2J} \dots \dots \dots (5)$$

Where J, the polar second moment of inertia is:

$$J_{motor} = \frac{\pi}{32(D_o^4 - D_i^4)} = \frac{\pi}{32(50^4 - 20.5^4)} = 1.615 \text{ (mm}^3) \dots \dots \dots (6)$$

$$J_{gen} = \frac{\pi}{32(D_o^4 - D_i^4)} = \frac{\pi}{32(50^4 - 30^4)} = 1.803 \text{ (mm}^3) \dots \dots \dots (7)$$

$$\tau_{max(motor)} = \frac{T \times D_o}{2J} = \frac{12 \times 50 \times 10^{-3}}{2 \times 1.615 \times 10^{-9}} = 185.75 \text{ (MPa)} \dots \dots \dots (8)$$

$$\tau_{max(gen)} = \frac{T \times D_o}{2J} = \frac{12 \times 50 \times 10^{-3}}{2 \times 1.803 \times 10^{-9}} = 166.38 \text{ (MPa)} \dots \dots \dots (9)$$

The coupling must be sized so that, typically, the stress does not exceed 10 percent of the ultimate tensile strength of the material.

Other factor to consider are the length of engagement into the coupling. The shear stress over the keyway must not exceed the allowable shear stress as given above.

The centroidal radius is:

$$R_{c(motor)} = 0.5 \times \left[\frac{D_o}{(2 + D_i) \times (2 + h)} \right] = 0.5 \times \left[\frac{50}{(2 + 20.5) \times (2 + 6)} \right] = 0.138 \text{ (mm)} \dots \dots \dots (10)$$

$$R_{c(gen)} = 0.5 \times \left[\frac{D_o}{(2 + D_i) \times (2 + \square)} \right] = 0.5 \times \left[\frac{50}{(2 + 30) \times (2 + 6)} \right] = .097 \text{ (mm)} \dots \dots \dots (11)$$

The centroid of the bearing area is at radius $(D_i + h)/2$. If the transmitted torque is T, then the compressive force F is $2T / (D_i + h)$. The bearing stress σ_b is:

$$\sigma_b(motor) = \frac{F}{A} = \frac{4T}{(wL(D_i + h))} = \frac{4 \times 12}{(5 \times 10^{-3} \times 39 \times 10^{-3} \times (20.5 + 6) \times 10^{-3})} = 9.288 \text{ (MPa)} \dots \dots \dots (12)$$

$$\sigma_b(gen) = \frac{F}{A} = \frac{4T}{(wL(D_i + h))} = \frac{4 \times 12}{(5 \times 10^{-3} \times 45 \times 10^{-3} \times (30 + 6) \times 10^{-3})} = 5.925 \text{ (MPa)} \dots \dots \dots (13)$$

The allowable compressive stress from distortion energy theory of failure is

$$\sigma_{all} = \frac{\tau_{all}}{0.577} \dots \dots \dots (14)$$

Combining this equation σ_b with give :

$$\begin{aligned} \tau_{all (motor)} &= \left[\frac{0.577 \times 4 \times T}{w \times L(D_i + h)} \right] = \left[\frac{0.577 \times 4 \times 12}{5 \times 10^{-3} \times 39 \times 10^{-3} \times (20.5 + 6) \times 10^{-3}} \right] \\ &= 5.339 (MPa) \dots \dots \dots (15) \end{aligned}$$

$$\begin{aligned} \tau_{all (gen)} &= \left[\frac{0.577 \times 4 \times T}{w \times L(D_i + h)} \right] = \left[\frac{0.577 \times 4 \times 12}{5 \times 10^{-3} \times 45 \times 10^{-3} \times (30 + 6) \times 10^{-3}} \right] \\ &= 3.419(MPa) \dots \dots \dots (16) \end{aligned}$$

$$\sigma_{all (motor)} = \frac{\tau_{all}}{0.577} = \frac{5.339}{0.577} = 9.253 (MPa) \dots \dots \dots (17)$$

$$\sigma_{all (gen)} = \frac{\tau_{all}}{0.577} = \frac{3.419}{0.577} = 5.925 (MPa) \dots \dots \dots (18)$$

Next, the length of key stock, for keyed shafts, must be examined to keep its shear loading from exceeding the allowable shear stress. Referring to the equation for σ_b , we note that the shear force is:

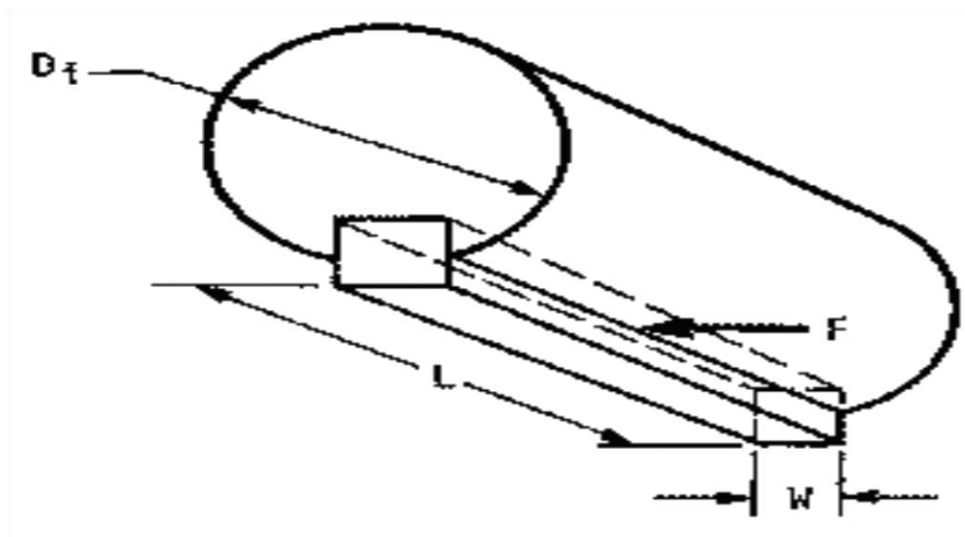
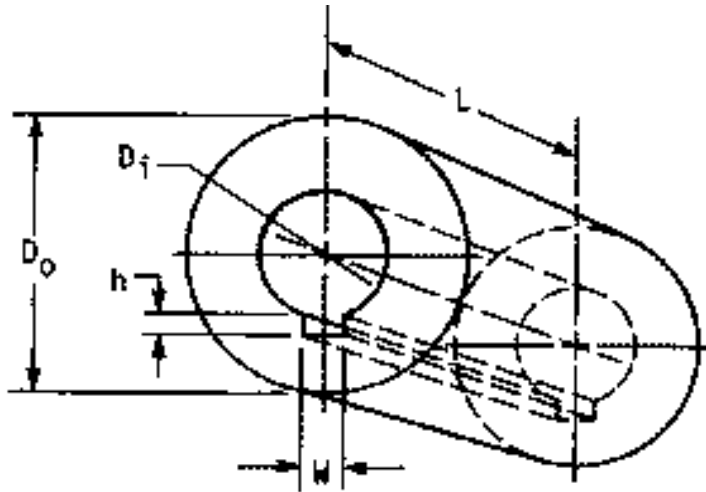
$$F = \frac{2T}{\tau} = \frac{2T}{D_i} \dots \dots \dots (19)$$

Therefore the average shear stress is

$$\tau = \frac{F}{A} = \frac{2T}{w \times L \times D_i} \dots \dots \dots (20)$$

$$\tau = \frac{F}{A} = \frac{2T}{w \times L \times D_i} = \frac{2 \times 12}{5 \times 10^{-3} \times 39 \times 10^{-3} \times 20.5 \times 10^{-3}} = 6.003 (MPa) \dots \dots \dots (21)$$

$$\tau = \frac{F}{A} = \frac{2T}{w \times L \times D_i} = \frac{2 \times 12}{5 \times 10^{-3} \times 45 \times 10^{-3} \times 30 \times 10^{-3}} = 3.555 (MPa) \dots (22)$$



Both keys must be checked, although experience has shown that small diameter shafts are more prone to failure of the key and keyway when these precautions are not followed because of their normally smaller key width and length of engagement.

Table 3.2: Maximum allowable shear stress 1

Maximum allowable shear stress	
material	Stress (psi)

Aluminum	4000
Grey Cast iron	4500
Ductile iron	6000
Brass	3500
Powered iron	4000
Steel	8000

Where:

T : Torque (Nm)

D_o : Diameter outer (mm).

D_i : Diameter inner (mm).

J : Polar second moment of inertia (mm^3).

R_c : Centroidal radius or distance (mm).

F : Compressive shear stress (Pa).

A : Cross sectional area (mm^2).

W : Width (mm).

L : Length (mm).

h : Keyway height (mm).

τ_{all} : Compressive shear stress (Pa).

τ : Shear stress (Pa).

CHAPTER

Four

Analysis using SOLIDWORKS simulation

4.1 Introduction.

4.2 Types of simulation in SOLIDWORKS program.

**4.3 Calculations were needed to the simulation to
be done:**

4.4 static simulation.

4.5 Dynamic simulation.

4.5.1 Frequency simulation mode.

4.5.2 Vibration simulation mode.

4.5.3 Nonlinear dynamic analysis.

4.1 Introduction:

SOLIDWORKS simulation is a virtual testing environment to analyze your design, evaluate its performance and make decisions to improve product quality. But how does it accomplish this? The software employs a numerical technique called Finite Element Analysis, or FEA.

So, the SOLIDWORKS enhance project's design and its analysis by improve the product quality and avoid the technical problem in the jaw coupling's design before implement it in real environment in the project by use the simulation in the SOLIDWORKS program.

4.2 Types of simulation in SOLIDWORKS program:

There is many types of simulation are available in this program and this project request three types of them which are: static, dynamic and natural frequency simulation.

By use this three types of simulation many results, calculations and analysis have been got to the jaw coupling part assembly.

In this simulation many factors were calculated like damping ratio, force and modules of elasticity.

4.3 Calculations were needed to the simulation to be done:

$$F = \frac{T}{R} = \frac{12}{25 * 10^{-3}} = 480 \text{ N} \dots\dots\dots (1)$$

$$K = \frac{\pi * G * (D^4 - d^4)}{32 * L} = \frac{\pi * 2.7 * 10^9 * (0.05^4 - 0.0205^4)}{32 * L}$$

$$= 11174 \text{ N.} \frac{M}{red} \dots\dots\dots (2)$$

$$\delta = \frac{F}{K} = \frac{480}{11174} = 0.043 \dots\dots\dots (3)$$

$$\varphi = \frac{\delta}{2 * \pi} = 0. \frac{043}{2 * \pi} = 0.0068 \text{ N. A} \dots\dots\dots (4)$$

$$a = \frac{F}{M} = \frac{480}{1.2} = 400 \frac{M}{S^2} \dots\dots\dots (5)$$

R: Radius of hub.

K: Modulus of elasticity.

G: Shear modulus.

F: Force.

T: Torque.

D: Outside diameter of hub.

d: Inside diameter of hub.

L: Length of coupling.

φ: damping ratio.

4.4 static simulation:

Static Simulation is a simulation model which has no internal history of both output and input values that were previously applied. It also represents a model in which time is not a factor. This type of simulation model usually has some function which is made of inputs. Each output in this type of simulation is dependent on the values of the function and inputs.

After made a static simulation on jaw coupling, the results were as the following:

Stress: After made the stress calculations on this part, the result was 3.606 Mpa and this is in the safe range because it's lower than the maximum range which is 9.25 Mpa in motor and 5.925 Mpa in generator.

Name	Type	Min	Max
Stress1	VON: von Mises Stress	2.888e+001N/m ² Node: 16043	3.606e+006N/m ² Node: 13308

Model name: 3As sem 3
Study name: Static 1 (Default)
Plot type: Static nodal stress Stress1
Deformation scale: 4901.78

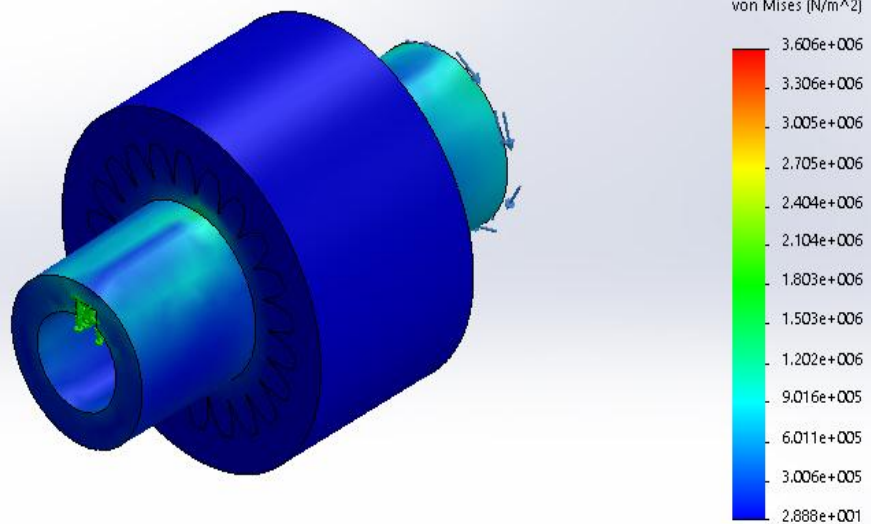


Figure 4.1: Static 1-Stress-Stress 4

Resultant Displacement: the displacement was in safe range after we get 2.965mm.

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+000mm Node: 1	2.965e-003mm Node: 15662

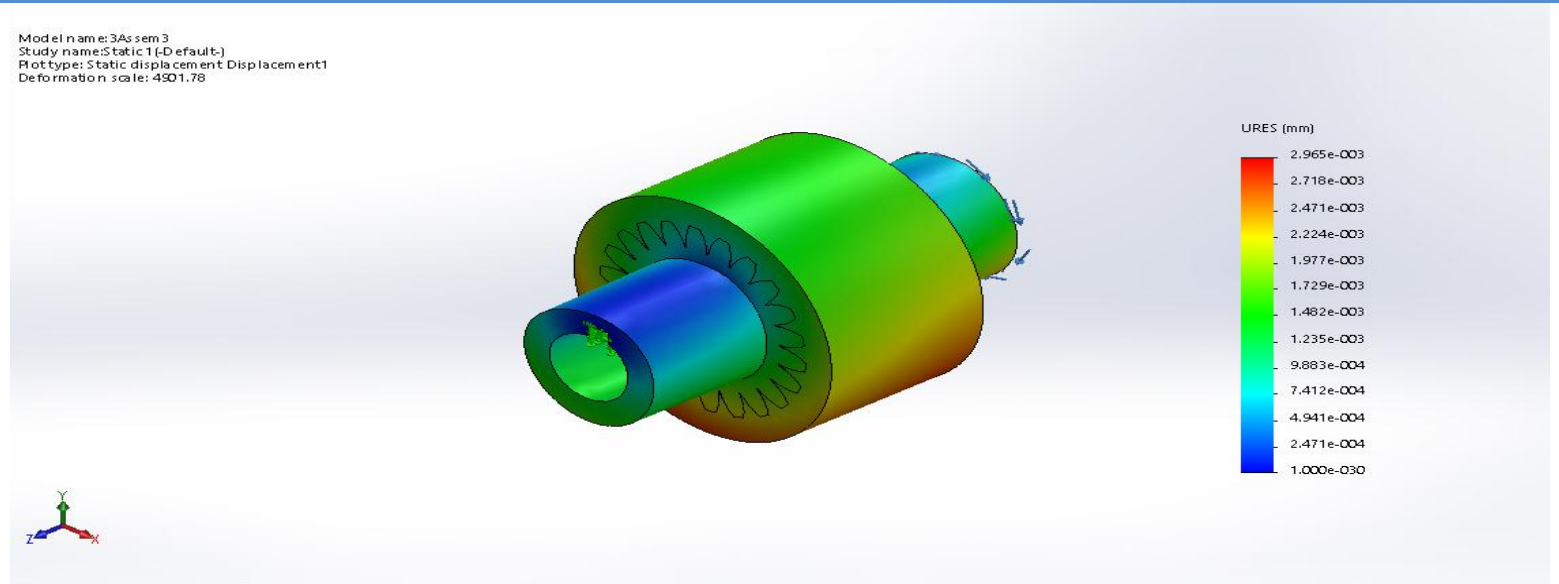


Figure 4.2: -Static 1-Displacement. 5

Resultant Strain: strain was in the safe range after get 3.453 in maximum range.

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	3.454e-008 Element: 1640	3.453e-005 Element: 7826

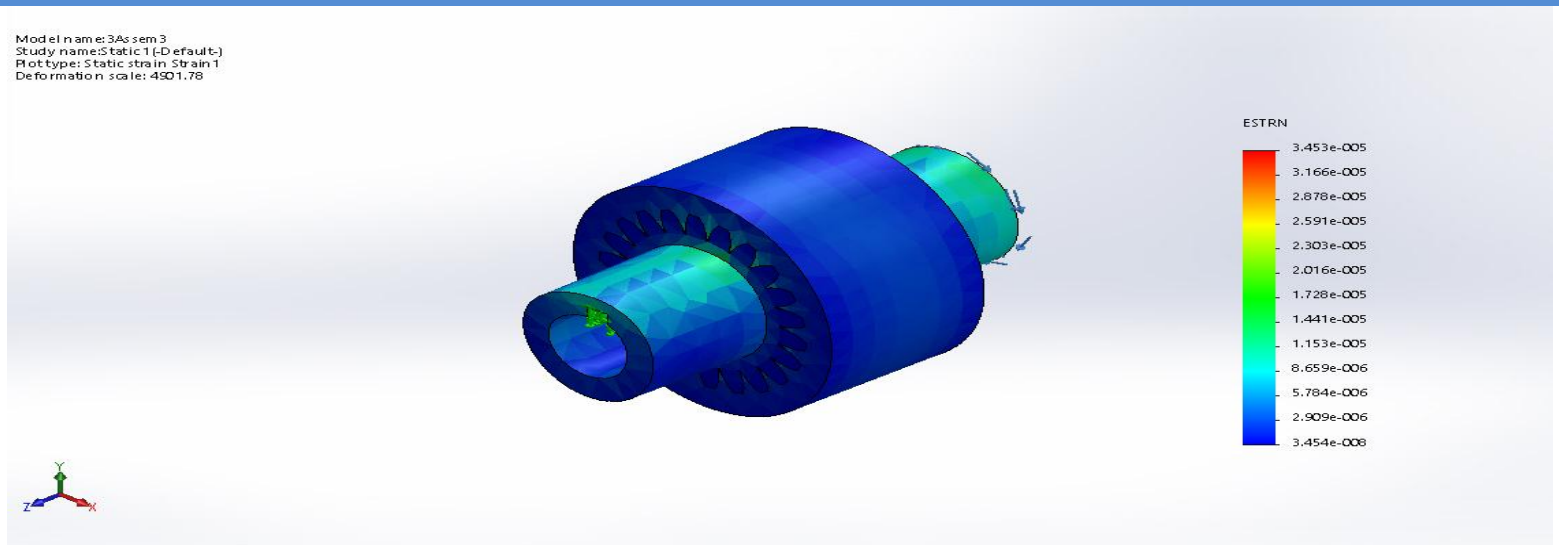


Figure 4.3: Static 1-Strain 6

4.5 Dynamic simulation:

SOLIDWORKS Simulation dynamic part analysis allows designers and engineers to validate product quality performance and safety throughout the design creation.

Greater Time and Cost Efficiency:-

SOLIDWORKS Simulation integrates seamlessly with the design process, allowing you to run linear stress analysis directly from your SOLIDWORKS model resulting in fewer costly prototypes, reduced rework and delays and greater time and cost efficiency.

Frequency, impact and drop tests are all covered within dynamic analysis. The primary unknown in a dynamic analysis is component displacement over time, but with this calculated, stresses, velocities, and accelerations can also be determined together with the natural modes of vibration.

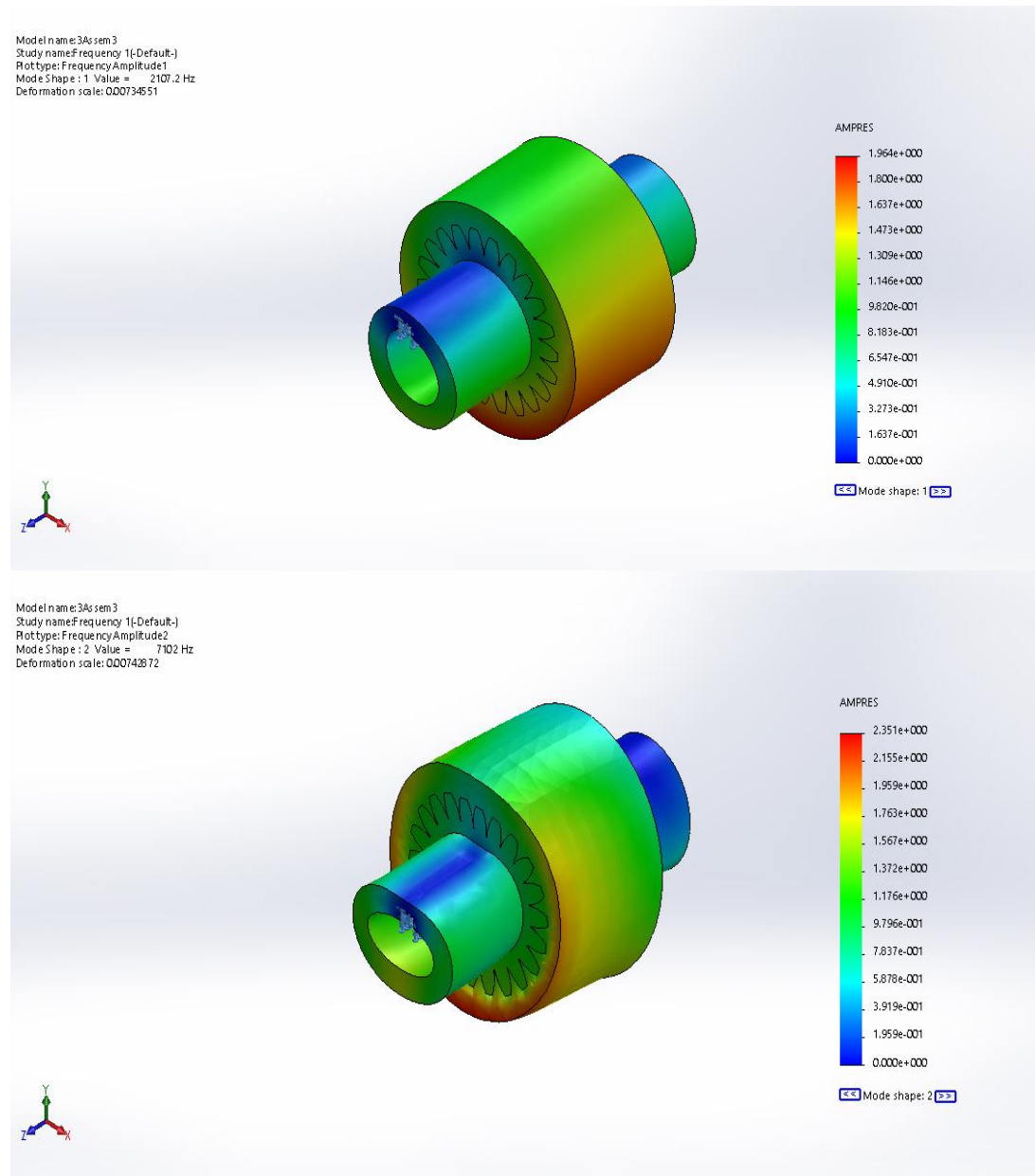
SOLIDWORKS Simulation makes use of one two methods when conduction dynamic analysis:

- Linear modal analysis which is able to calculate natural modes of vibration and then the displacements, stresses, strains, velocities and accelerations.
- Linear dynamic consist also of vibration and frequency simulation mode, vibration mode have more than three types of motion which are harmonic, random and modal vibration.
- Nonlinear dynamic analysis calculates the displacement field at each time step taking into account the applied loads and initial component velocities, following this it is able to calculate the nonlinear stresses, strains, velocities and accelerations.

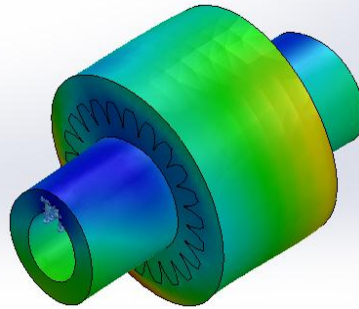
4.5.1 Frequency simulation mode:

SOLIDWORKS have the frequency simulation mode that let their users to know the natural frequency so it's easy to avoid the resonance phenomena by make a frequency study to their projects to avoid any serious damage to their parts.

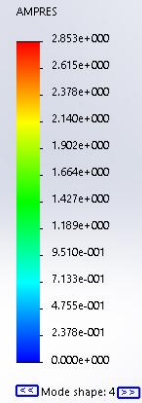
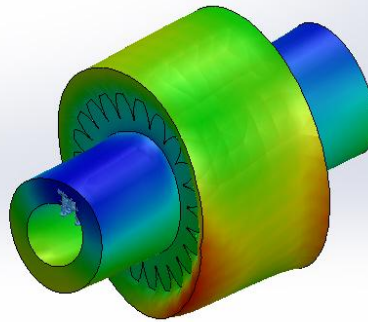
After made a frequency study the results and drawing were as below:-



Model name: 3Ac sem 3
Study name: Frequency 1(-Default-)
Plot type: Frequency Amplitude 3
Mode Shape : 3 Value = 7885.1 Hz
Deformation scale: 0.00560987



Model name: 3Ac sem 3
Study name: Frequency 1(-Default-)
Plot type: Frequency Amplitude 4
Mode Shape : 4 Value = 8005.9 Hz
Deformation scale: 0.0060952



Model name: 3Ac sem 3
Study name: Frequency 1(-Default-)
Plot type: Frequency Amplitude 5
Mode Shape : 5 Value = 8694.4 Hz
Deformation scale: 0.00460551

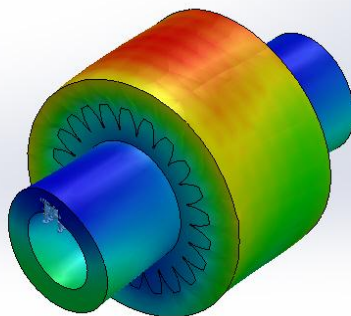


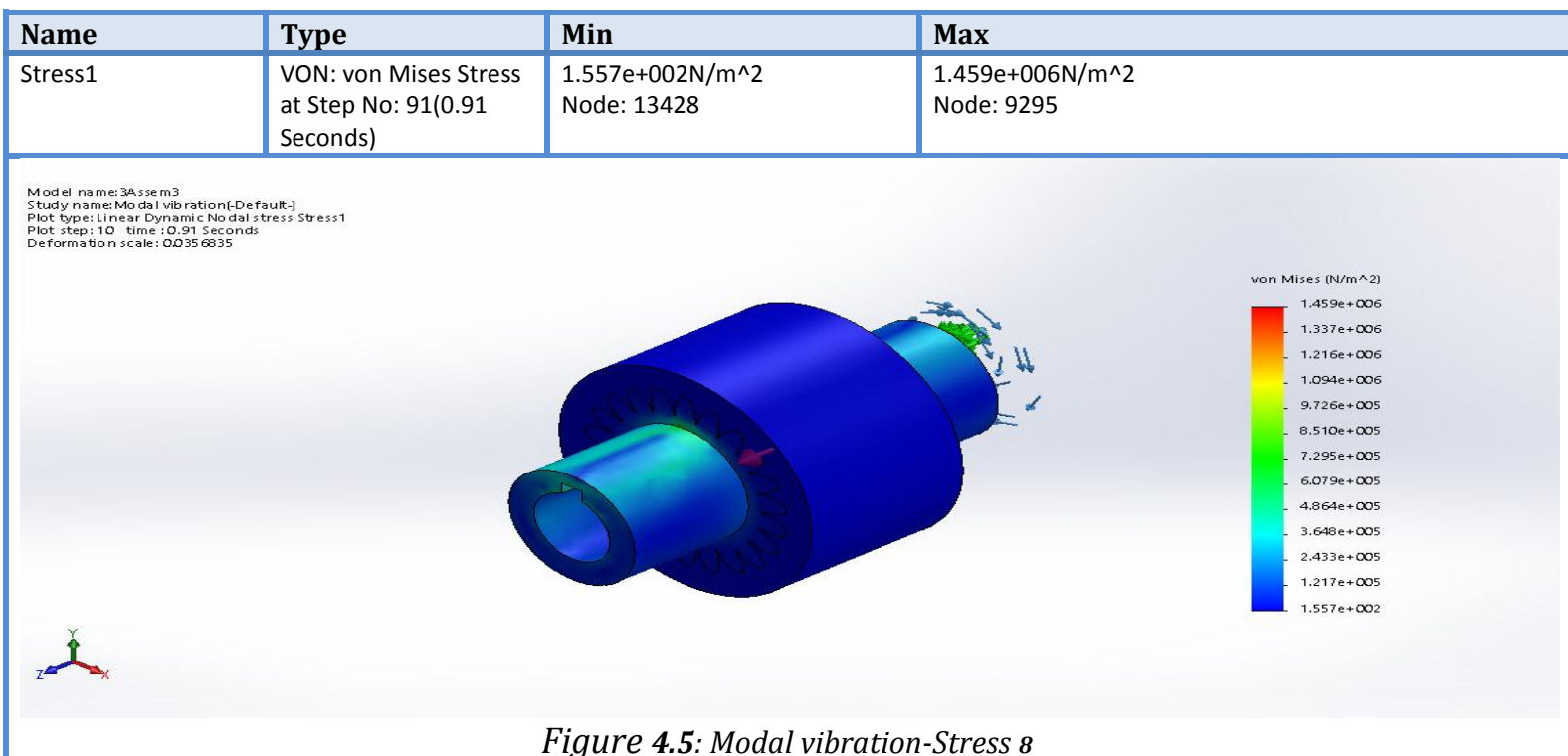
Figure 7(4.4): frequency simulation.

4.5.2 Vibration simulation mode:

SOLIDWORKS Vibration simulation tools provide easy-to-understand results for three different kinds of stresses:

- **Modal analysis** – which determines the natural frequencies and mode shapes to avoid resonance.
- **Harmonic analysis** – which avoids catastrophic failures that can occur when rotating components match product resonance.
- **Random Vibration analysis** – which is used to determine the response of product designs and components to vibration loads that are random such as bumpy roads, rocket launches, earthquakes, etc.

1) Modal vibration analysis:



Name	Type	Min	Max
Displacement1	URES: Resultant Displacement at Step No: 91(0.91 Seconds)	4.035e+002mm Node: 13338	4.035e+002mm Node: 13563

Model name: 3Assem3
 Study name: Modal vibration-(Default-)
 Plot type: Linear Dynamic Displacement Displacement1
 Plot step: 10 time :0.91 Seconds
 Deformation scale: 0.0356835

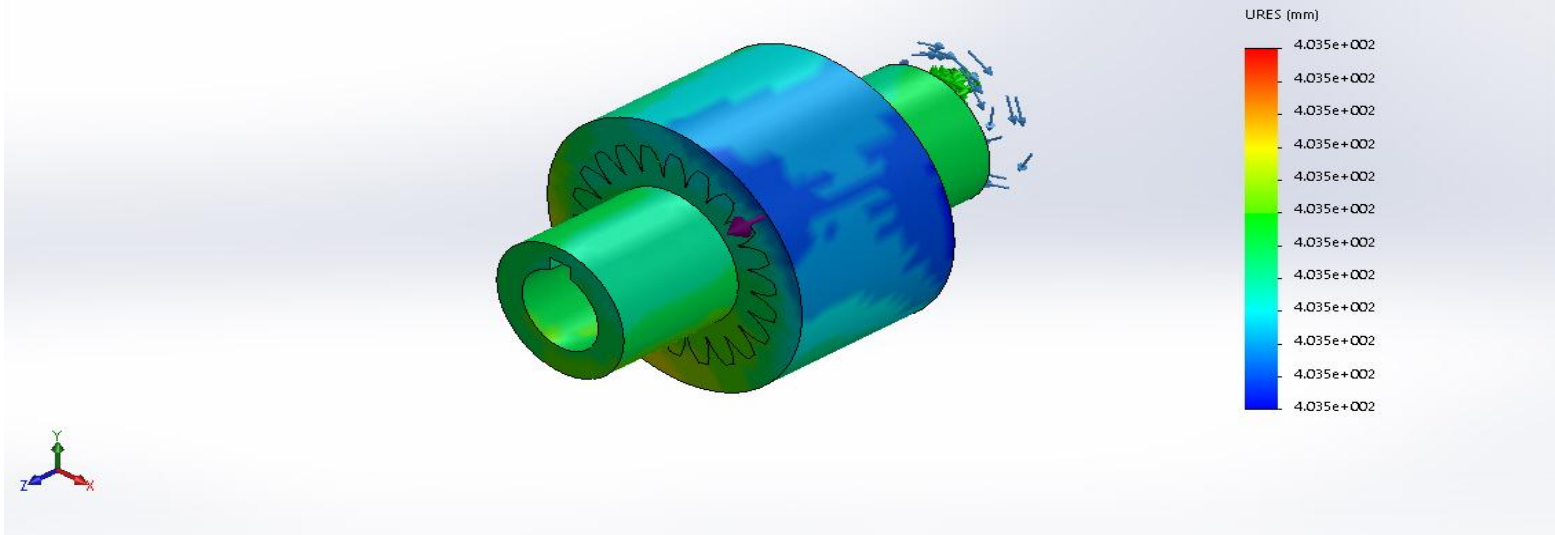


Figure 4.6: Modal vibration-Displacement. 9

2) Harmonic analysis:

Name	Type	Min	Max
Stress1	VON: von Mises Stress at Step No: 166(Frequency = 14000 Hz)	1.465e+003N/m ² Node: 15781	9.254e+005N/m ² Node: 7449

Model name: 3Assem3
 Study name: Harmonic analysis-(Default-)
 Plot type: Linear Dynamic Modal stress Stress1
 Plot step: 166 Frequency : 14000 Hertz
 Deformation scale: 10341.3

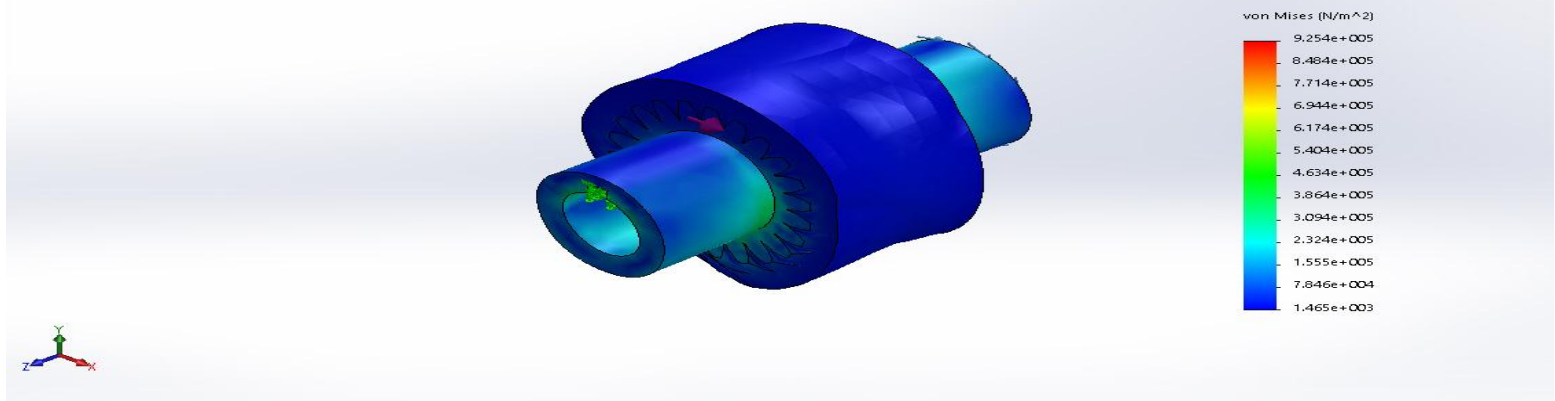


Figure 4.7: Harmonic analysis Stress. 10

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement at Step No: 166(Frequency = 14000 Hz)	0.000e+000mm Node: 1	1.416e-003mm Node: 13533

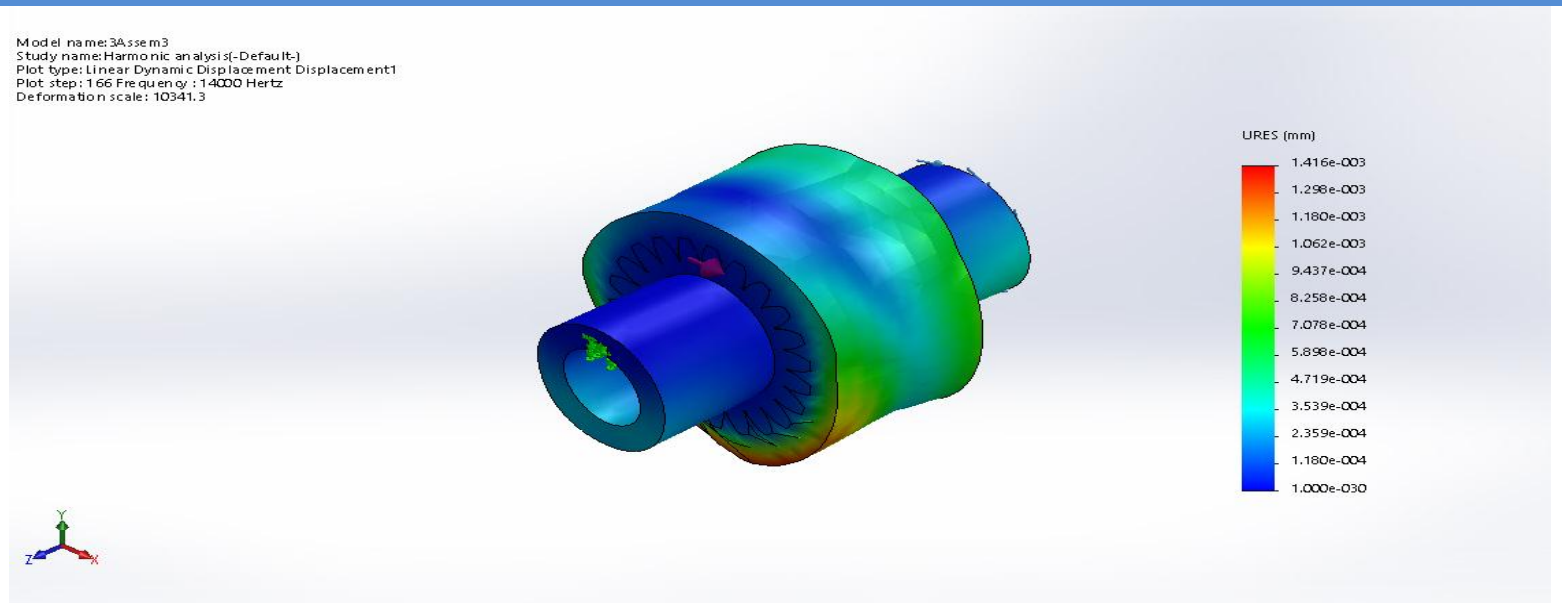


Figure 4.8: Harmonic analysis-Displacement.11

3) Random Vibration analysis:

Name	Type	Min	Max
Stress1	VON: von Mises Stress	9.511e+001N/m ² Node: 15644	1.308e+005N/m ² Node: 6752

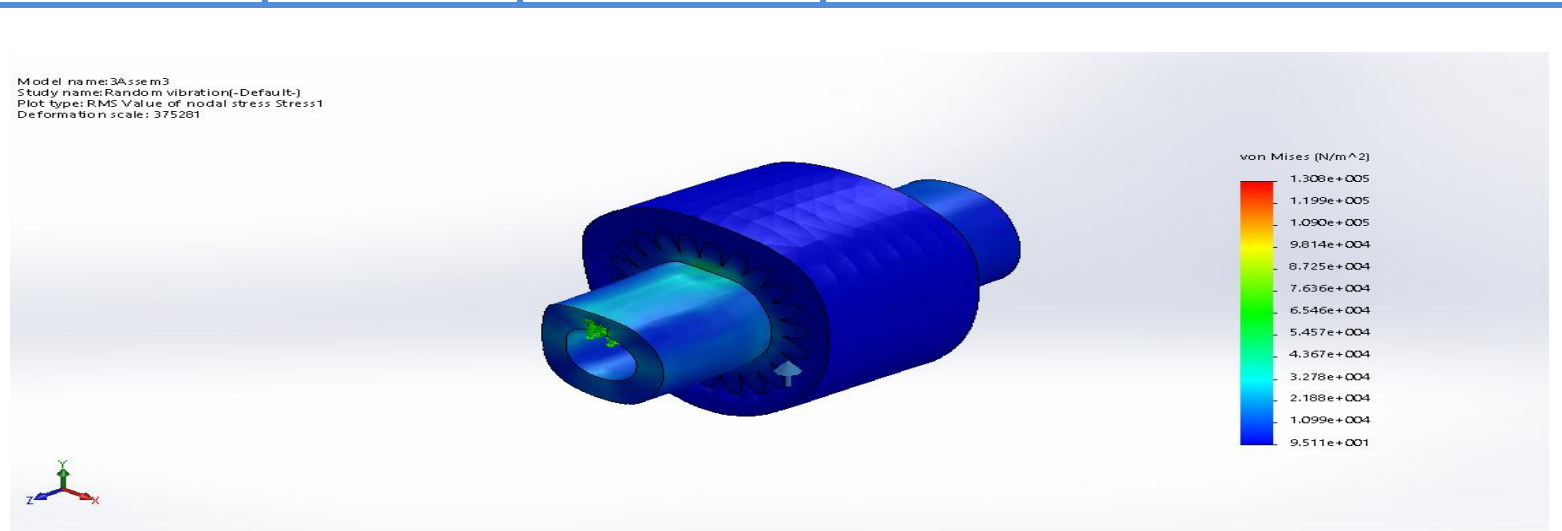


Figure 4.9: Random vibration-Stress12

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+000mm Node: 1	4.028e-005mm Node: 13523

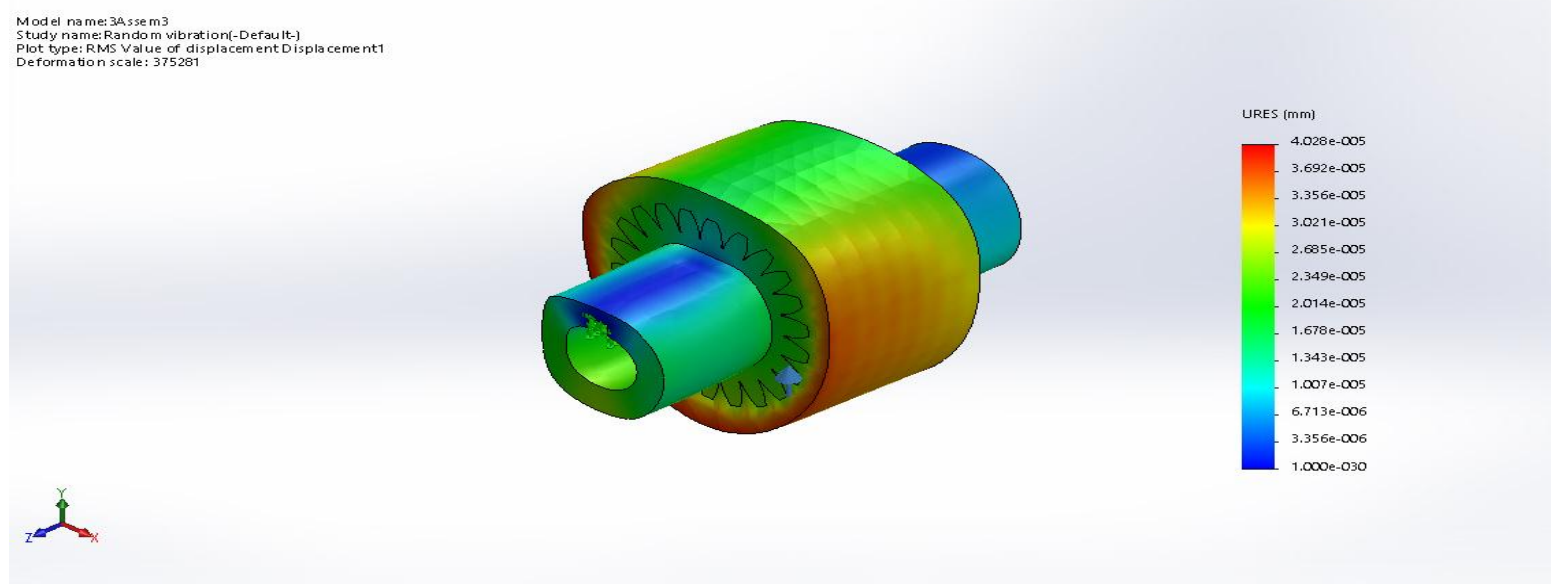


Figure 4.10: Random vibration-Displacement13

Name	Type	Min	Max
Acceleration1	ARES: Resultant Acceleration	5.083e+002mm/s^2 Node: 5861	8.177e+004mm/s^2 Node: 13522

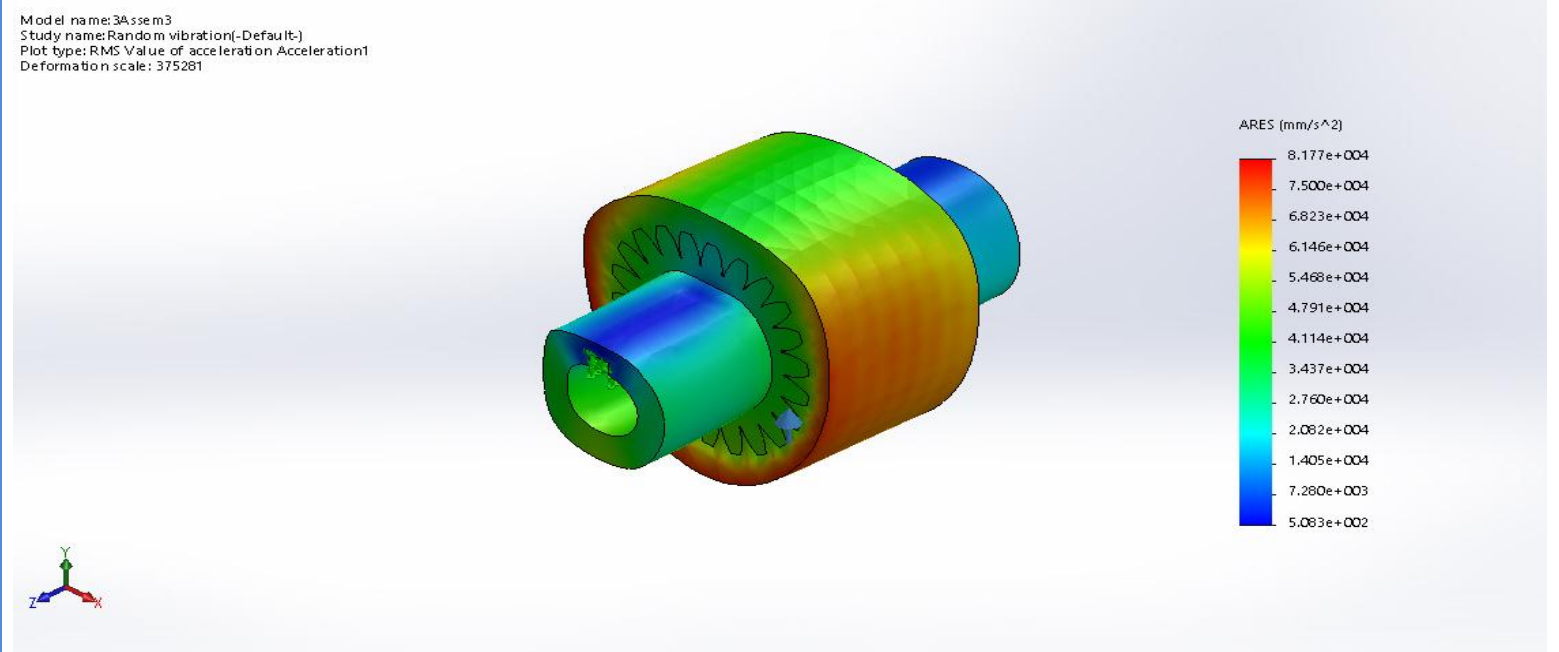


Figure 4.11: Random vibration-Acceleration.14

Name	Type	Min	Max
Velocity1	VRES: Resultant Velocity	0.000e+000mm/sec Node: 1	1.738e+000mm/sec Node: 13522

Model name: 3Aassem3
 Study name: Random vibration(-Default-)
 Plot type: RMS Value of velocity Velocity1
 Deformation scale: 375281

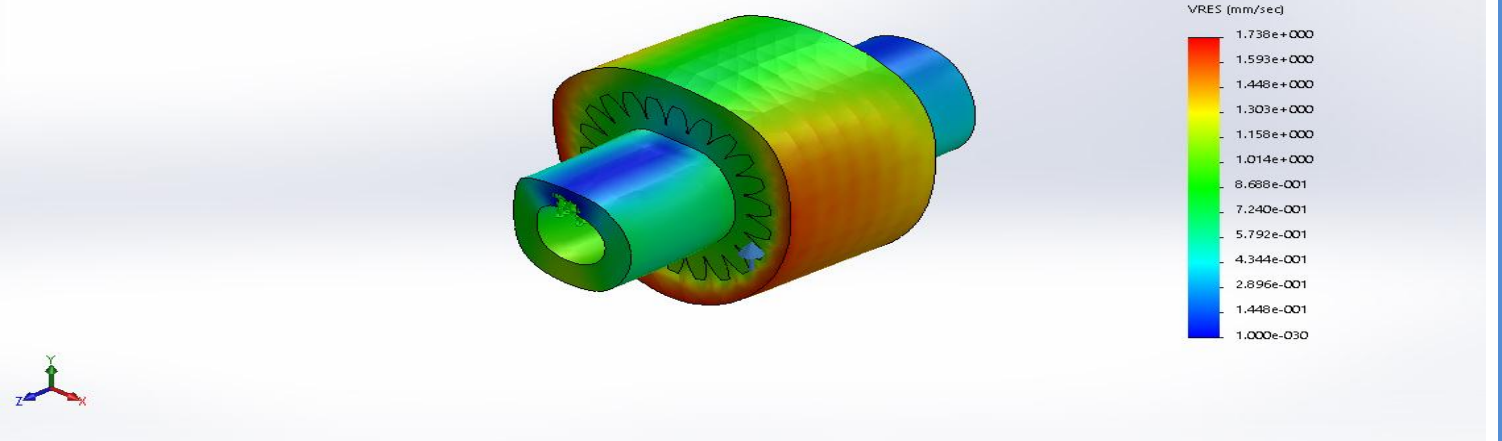


Figure 4.12: -Random vibration-Velocity 15

Name	Type	Min	Max
Strain1	SEDENS: Strain Energy Density	6.926e-006N.m/m^3 Element: 11195	1.168e+000N.m/m^3 Element: 10862

Model name: 3Aassem3
 Study name: Random vibration(-Default-)
 Plot type: RMS Value of strain Strain1
 Deformation scale: 375281

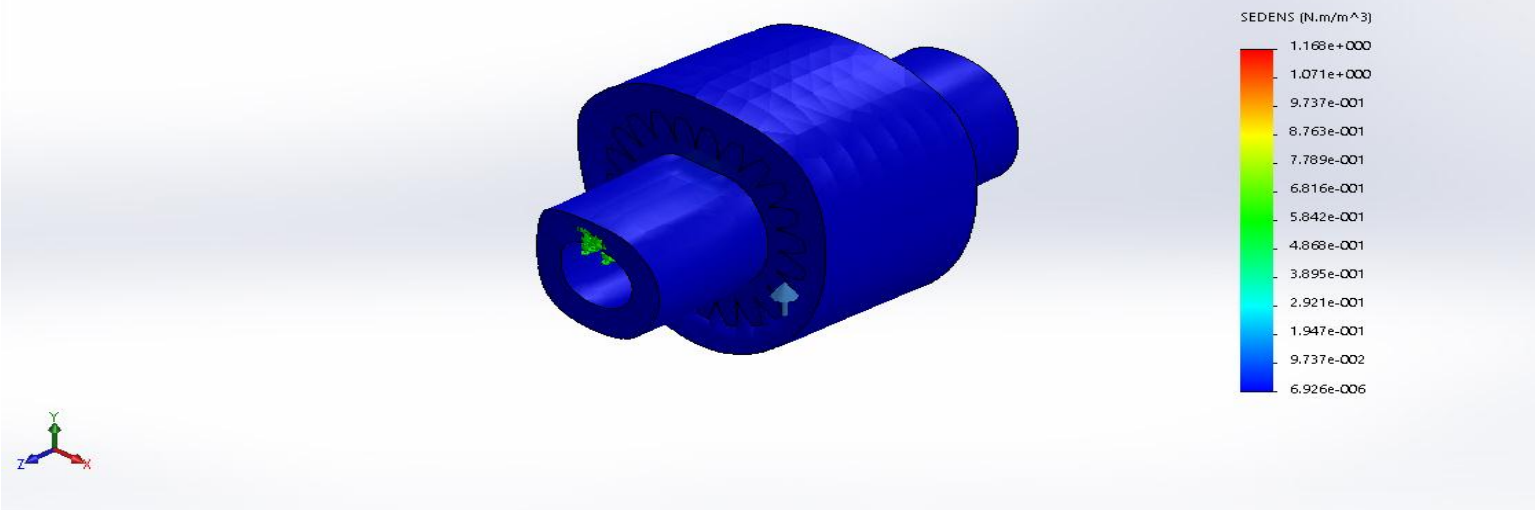
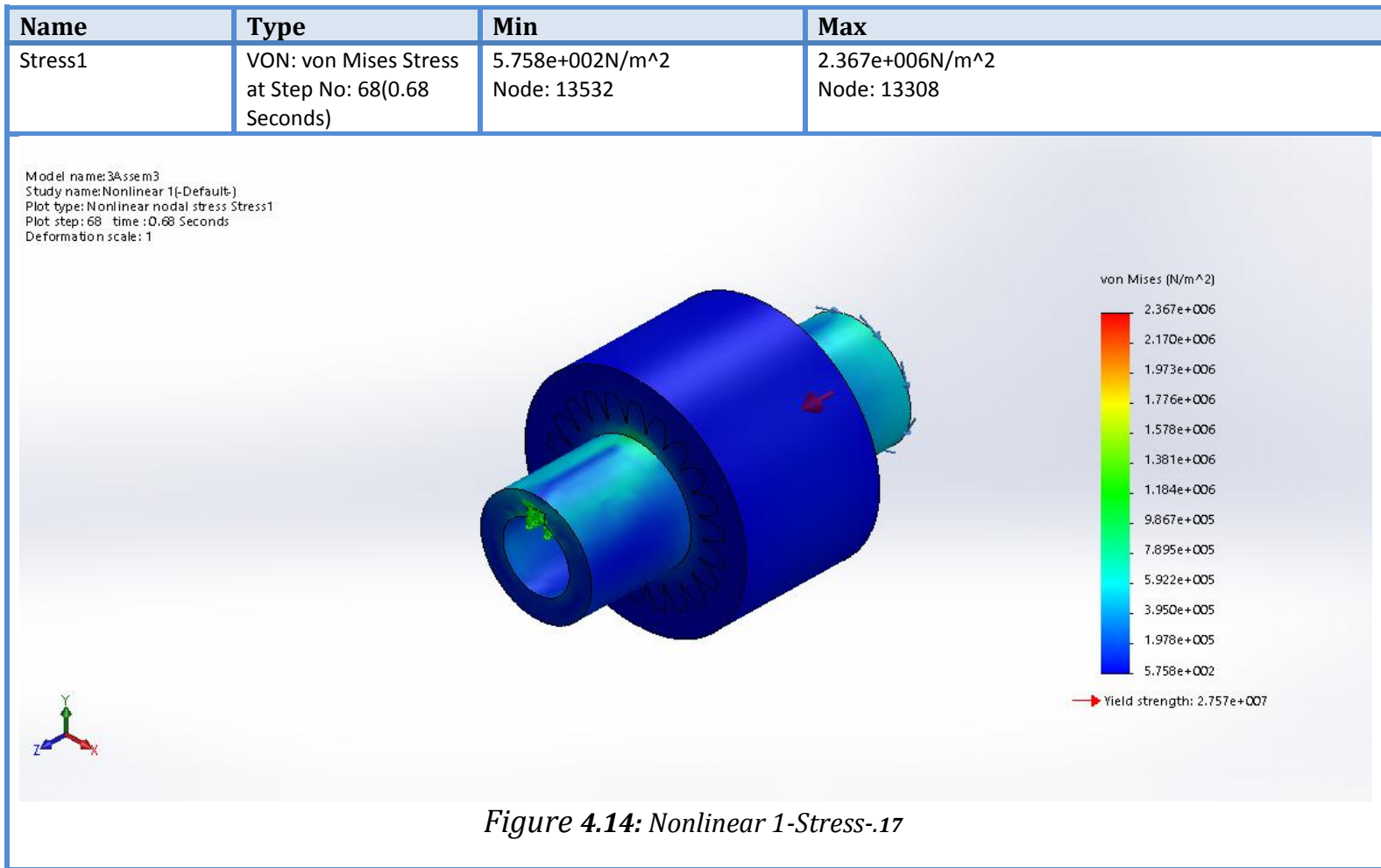


Figure 4.13: Random vibration-Strain.16

4.5.3 Nonlinear dynamic analysis:

In nonlinear dynamic analysis, the non-linearity of the structure is taken into consideration. It is a very complex model to analyze the effect of dynamic loading and it can be used for any no. of degrees of freedom. It combines various methods like SRSS (square root of sum of squares) and it is used for irregular structure.

It is an accurate estimation technique but involves various types of analysis.



Name	Type	Min	Max
Displacement1	URES: Resultant Displacement at Step No: 68(0.68 Seconds)	0.000e+000mm Node: 1	1.935e-003mm Node: 13557

Model name: 3Asse.m3
 Study name: Nonlinear 1(-Default-)
 Plot type: Nonlinear Displacement Displacement1
 Plot step: 68 time : 0.68 Seconds
 Deformation scale: 1

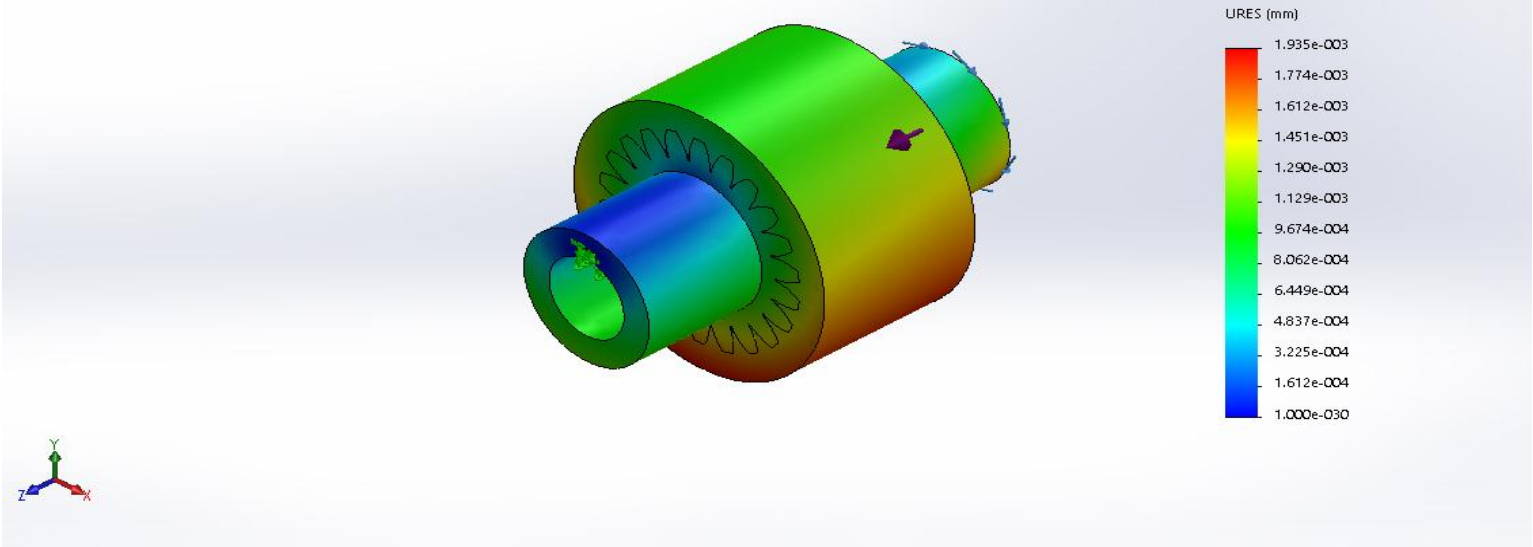


Figure 4.15: Nonlinear 1-Displacement.18

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain at Step No: 68(0.68 Seconds)	3.090e-008 Element: 11690	2.272e-005 Element: 7826

Model name: 3Asse.m3
 Study name: Nonlinear 1(-Default-)
 Plot type: Total Strain Strain1
 Plot step: 68 time : 0.68 Seconds
 Deformation scale: 1

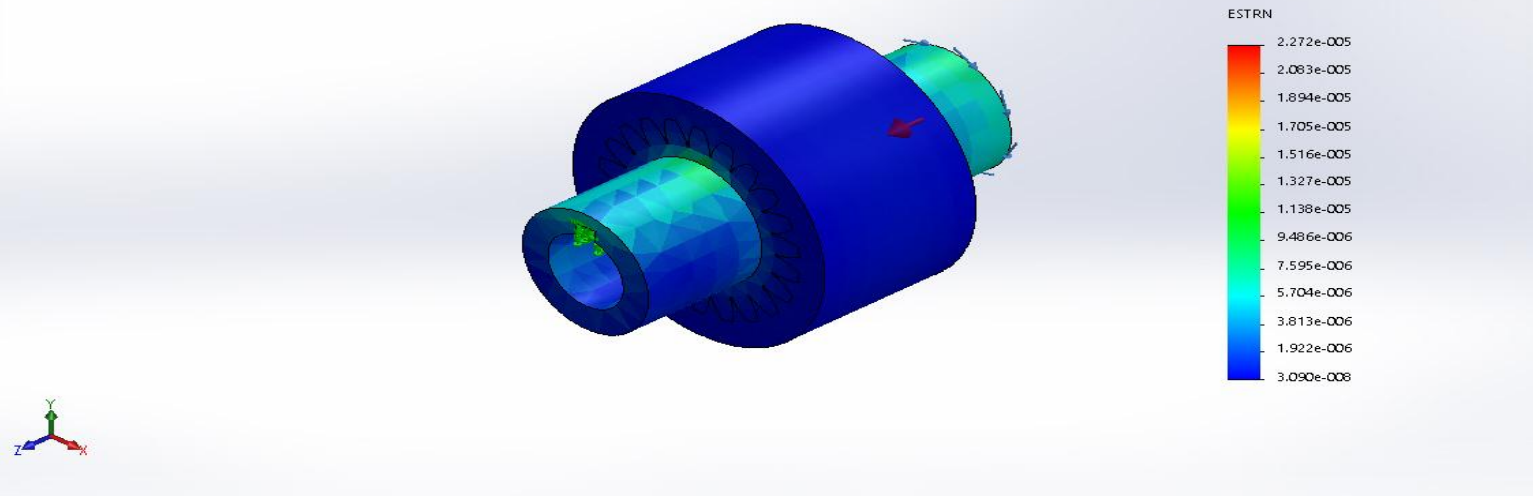


Figure 4.16: Nonlinear 1-Strain.19

Conclusion:

After the project is finished successfully, and the objectives achieved, all the devices and the dynamometer have been assembled and worked correctly without any vibration, misalignment or any problem else which was faced before the project get done.

After spent days and nights to find the best coupling and design for dynamometer's project and after ask many experts in this field and using SOLIDWORKS design and simulation program, the best coupling was the jaw coupling that could convert the torque from single cylinder internal combustion engine to generator without any problem like vibration or misalignment... etc. Usage of SOLIDWORKS program saved a lot of time, money and effort that would spent if it didn't used, SOLIDWORKS gave a good simulation environment that let the project's parts designed ,simulated and analyzed.

By finished this project, the automotive engineering students could perform several experiments on dynamometer that enhance their knowledge in practical way to convert the theoretical data from just a piece of papers to a real thing that could calculated and perform near to what happen in real life.

CHAPTER

FIVE

Engine Performance Parameters

And

Mathematical Relationships

5.1 Introduction.

5.2 Engine Swept Volume.

5.3 Compression Ratio.

5.4 Engine power.

5.4.1 Engine Brake Power (bp).

5.4.2 Engine Indicated Power (ip).

5.4.3 Engine Friction Power.

5.4.4 Indicated Mean Effective Pressure (imep).

5.5 Efficiencies (ζ).

5.5.1 Engine Volumetric Efficiency.

5.5.2 Engine Mechanical Efficiency.

5.5.3 Engine Brake Thermal Efficiency.

5.6 Fuel Consumption SFC.

5.7 The Torque.

5.1 Introduction:

To check engine performance well, we must check engine efficiencies by loading the engine with different loads and analyzing emissions, these efficiencies include (engine swept volume, compression ratio, engine indicated power, engine brake power, engine friction power, engine mechanical efficiency, specific fuel consumption SFC, engine brake thermal efficiency, and engine volumetric efficiency). These all parameters which used to check an engine performance and describe an engine efficiencies completely, we have to learn about energy flow in an engine which desirable to have a general knowledge of the energy flow or energy balance of the engine so that we become familiar with the various performance parameters to determine the performance by different loads.

Internal combustion engine generally operates within a useful range of speed. Some engines are made to run at fixed speed by means of a speed governor, which is its rated speed. At each speed within the useful range the power output varies and it has a maximum usable power value. The ratio of power developed to the maximum usable power at the same speed is called the load. The specific fuel consumption varies with load and speed. The performance of the engine depends on inter-relationship between power developed, speed and the specific fuel consumption at each operating condition within the useful range of speed and load.

The following factors are to be considered in evaluating the performance of an engine:

- Maximum power and torque available at each speed within the useful range of speed.

- The range of power output at constant speed for stable operation of the engine. The different speeds should be selected at equal interval within the useful speed range.
- Brake specific fuel consumption at each operation condition within the useful range of operation. [7]

Engine performance characteristics can be determined by the following two methods:

- By using experimental results obtained from engine tests.
- By analytical calculation based on theoretical data.

5.2 Engine Swept Volume:

Table 5.1 Engine Swept Volume Equation. [7]

$V_e = \left(\frac{\pi}{4} B^2 \times L\right) \times n = A_{cyl} \times L \times n = V_{cyl} \times n \quad [m^3]$		
V_e	Engine swept volume	m^3
B	Cylinder bore (diameter)	M
L	Stroke length	M
N	Number of cylinders	
A_{cyl}	Cylinder (piston) area	m^2
V_s	Cylinder volume	m^3

5.3 Compression Ratio:

Table 5.2 Compression Ratio Equation. [7]

$r = \frac{V_c + V_s}{V_c} = 1 + \frac{V_s}{V_c}$		
R	Compression ratio	
Vc	Clearance volume	m³
Vs	Cylinder volume	m³

5.4 Engine Power:

In general, the energy flow through the engine is expressed in three distinct terms. They are indicated power (ip), friction power (fb), and brake power (bp). indicated power can be computed from the measurement of forces in the cylinder and brake power may be computed from the measurement of force at the crank shaft of the engine but in this project the force will be measured by a load cell .the friction power can be estimated as the difference between the (ip) and (bp) if these two are known, then: [7]

ip= bp + fb (KW)	fb = ip – bp (KW)
-------------------------	--------------------------

5.4.1 Engine Brake Power (bp):

Brake power used to indicate the power actually delivered by the engine, so the brake power usually measured by attaching a power absorbing device to the drive shaft of the engine. Such a device sets up measurable forces counteracting the forces delivered by the

engine, and determined value of these measured forces is indicative of forces being delivered.

Measurement of brake power is one of the most important measurements, it involves the determination of the torque and angular speed of the engine output shaft. The torque measuring device is called a dynamometer. In the dynamometer, the rotor of the generator driven by the engine under test. By using geometry, the shaft that mounted on the body of the generator known dimension (length) work as the torque arm, this arm with the strain gauge scale makes it easier to load the cell.

The work done by the engine (Torque) has been defined as the product of a force and the distance (shaft length from the center of the generator) through which the point of application of force movement. [7]

Table 5.3 Engine Brake Power (bp) Equation.

$P_b = T_e \times \omega = T_e \frac{2\pi N}{60} \text{ [kW]}$		$T = R * F$
P_b	Engine brake power	kW
T_e	Engine torque produced by the engine	kN.m
ω	Engine angular speed of the engine	rad/sec
N	Engine speed (RPM)	rev/min
F	The force by the arm	N
R	The length arm	m

5.4.2 Engine Indicated Power (ip):

Power is defined as the rate of doing work. In the analysis of cycles, the network is expressed in “KJ/Kg” of air. This may be converted to power by multiplying by the mass flow rate of air through the engine in kg per unit time. So indicated power of an

engine tells about the health of the engine and also gives an indication regarding the conversion of chemical energy in the fuel into heat energy. Indicated power is an important variable because it is the potential output of the cycle. In working with actual engines, it is often desirable to compute (ip) from a given (pim) and given engine operating condition. The necessary formula may be developed from the equation of network based on the mean effective pressure and piston displacement. [7]

Table 5.4 Engine Indicated Power Equation.

$P_i = \text{imep} \times A \times L \times n \times \frac{N}{2 \times 60 \times 1000} = \text{imep} \times V_e \times \frac{N}{2 \times 60 \times 1000} \quad [\text{kW}] \text{ (Four stroke engine)}$		
$P_i = \text{imep} \times A \times L \times n \times \frac{N}{60 \times 1000} = \text{imep} \times V_e \times \frac{N}{60 \times 1000} \quad [\text{kW}] \text{ (Two stroke engine)}$		
Pi	Engine indicated power	kW
Imep	Indicated mean effective pressure	N/m²
A	Cylinder (piston) Are	m²
L	Stroke	m
N	Number of cylinders	
N	Engine revolution per minute	rpm
Ve	Engine volume	m³

5.4.3 Engine Friction Power:

Table 5.5 Engine Friction Power Equation.

$P_f = P_i - P_b \quad [\text{kW}]$		$P_i = P_b + P_f \quad [\text{kW}]$
Pf	Engine friction power	kW
Pi	Engine indicated power	kW
Pb	Engine brake power	kW

5.4.4 Indicated Mean Effective Pressure (imep):

It has been stated in section 4.1 that “ip” can be computed from the measurement of forces in the cylinder, with the pressure of the expanding gases.

The pressure in the cylinder varied through the cycle and the variation can be expressed with respect to volume or crank angle rotation to obtain “P-V” or “P- α ” diagrams respectively. However, such a continuous variation does not readily lend itself to simple mathematical analysis in the computation of (ip). If an average pressure for one cycle can be used, then the computation becomes far less difficult.

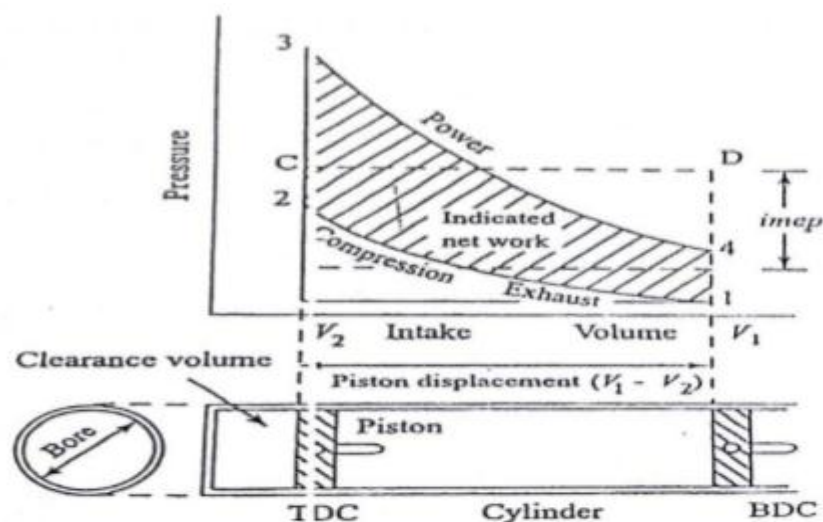


Figure 5.1 P-V Diagram for an Ideal Four-Stroke Cycle Engine

As the piston moves back and forth between TDC and BDC Figure.4.1, the process lines on the P-V diagram indicate the successive states of the working fluid through the cycle. The indicated network of the cycle is represented by the area 1234 enclosed by the process lines for that cycle. If the area CD rectangular equals area 1234, the vertical distance between the horizontal lines CD represents the indicated mean effective pressure “imep”, it is a mean

Value expressed in N/m^2 , which, when multiplied by the displacement volume, V_s , gives the same indicated network as is produced with the varying pressures.

Table 5.6 Indicated Mean Effective Pressure (imep) Equation.

$\text{imep} = \frac{\text{Net work of cycle}}{V_s}$	$\text{imep} = \frac{\text{Area of the indicator diagram}}{\text{Length of the indicator diagram}}$
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On an actual engine, the P-V diagram (called the indicator diagram) is obtained by a mechanical or electrical instrument attached to the cylinder. The area enclosed by the actual cycle on the indicator card divided by the piston displacement, results in the mean effective pressure.

5.5 Efficiencies (ζ):

5.5.1 Volumetric Efficiency:

The volumetric efficiency (ζ_v) is a measure of the success with which the air supply, and thus the charge, is inducted into the engine. It is a very important parameter, since it indicates the breathing capacity of the engine. The volumetric efficiency is defined as the ratio of the actual mass of air to the theoretical mass which should have been in during that same period of time. [7]

Table 5.7 Engine Volumetric Efficiency Equation.

$\eta_v = \frac{V_{\text{air}}}{V_s}$		$\zeta_v = m^{\bullet}_{\text{act}} / m^{\bullet}_{\text{th}}$
η_v	Volumetric efficiency	%
V_{air}	Actual air volume intake	m^3
V_s	Cylinder volume	m^3

5.5.2 Mechanical Efficiency:

The mechanical efficiency (ζ_m) it takes into account the losses in an engine. Mechanical losses of an engine may be further subdivide into following groups:

- Friction losses in case of piston, bearings, gears, valve mechanisms.
- Power is absorbed by engine auxiliaries.
- Ventilating action of the flywheel.
- Work of charging the cylinder with fresh and discharging the exhaust gases during the exhaust stroke.

The mechanical efficiency is defined as the ratio of the brake power to the actual net indicated power. [7]

Table 5.8 Engine Mechanical Efficiency Equation.

$\zeta_m = \frac{bp}{ip} * 100\%.$		
η_m	Mechanical efficiency	%
Pb	Engine brake power	kW
Pi	Engine indicated power	kW

5.5.3 Engine Brake Thermal Efficiency:

Brake Thermal Efficiency is defined as break power of a heat engine as a function of the thermal input from the fuel. It is used to evaluate how well an engine converts the heat from a fuel to mechanical energy. [7]

Table 5.9 Engine Brake Thermal Efficiency Equation.

$\eta_{th} = \frac{P_b \times 3600}{m \cdot CV} = \frac{P_b \times 3600}{V \cdot \rho \times CV}$		
η_{th}	Thermal efficiency	%
Pb	Engine brake power	kW
m.	Mass of fuel consumed per hour	kg/h
V	Volume of fuel consumed per hour	m³/h
ρ	Fuel density	Kg/m³
CV	Fuel calorific (heat) value	kJ/kg

5.6 Fuel Consumption SFC:

Table 5.10 Specific Fuel Consumption SFC Equation. [7]

$SFC = \frac{m_f}{P_b} \quad \left[\frac{\text{kg}}{\text{kW h}} \right]$		
SFC	Specific fuel consumption	kg/(kW.h)
m.f	Mass of fuel consumed per hour	kg/h
Pb	Engine brake power	kW

5.7 Torque:

We can know the torque by the force acting or by known Amp, voltage and speed in N.m. [7]

The Torque Equation

$$T = F * R \quad [N.m]$$

OR

$$T = \frac{I * V * 1.73 * 60}{2 * 3.14 * N} \quad [N.m]$$

CHAPTER

SIX

Practical Test and Discussion

6.1 Introduction.

6.2 Engine Operation Modes.

6.2.1 Warming up.

6.2.2 Constant Speed Wide Open Throttle (WOT) With Variable Loads.

6.3 Experiment No 1: Brake Power Calculation.

6.4 Experiment NO 2: Fuel Consumption.

6.5 Experiment No 3: (P- α) and (P-V) Diagrams.

6.6 Maintenance and Rehabilitation.

6.1 Introduction:

In this chapter, we will record and calculate experimental engine data (torque, power, and fuel consumption and the rated efficiency) with its graphs and create P-V & P- α Diagrams.

6.2 Engine Operation Modes:

6.2.1 Warming up:

In this mode, the engine requires high amount of fuel to allow it reach normal operation temperature quickly, so the intake choke valve must be closed. To allow fast warming, ignition timing must be retarded to appropriate value which achieved by automatic ignition control device.

- Do not increase engine speed until the oil reaches all engine parts normally. Shut the engine immediately down if any Suspicious notices.
- Explosion hazard warning: Do not use volatile starting aids such as ether, propane, or gasoline in the air intake system. The glow plugs/AIH will ignite the vapors and cause severe engine damage or personal injury. [7]

6.2.2 Constant Speed Wide Open Throttle (WOT) With Variable Loads:

In the operating mode the throttle valve must be opened to the maximum limit, the engine speed increased to specific speed then hold it constant until the end of experiment, then start applying loads sequentially until maximum load applied on the engine. [7]

6.3 Experiment No 1: (Brake Power Calculation)

Brake power measured firstly by turning the engine on then apply the load by the dynamometer; we can see the force on the load cell and engine rotation speed by the analyzer. [7]

Objectives:

To measure the brake power of the engine.

Procedure:

1. Check Engine Oil
2. Fill your tank with correct fuel for your test engine.
3. Open both valves on the fuel gauges and make sure that the fuel has passed down in the fuel feed pipes toward the test engine.
4. Make sure that your computer is operating correctly and has started TQ software (VDAS software, Engine Cycle Analyzer Software).
5. Check load control panel connections, and make sure that all loads switched off.
6. Turn ignition switch to on position, rotate the output shaft to the TDC mark, and start the engine.
7. Allow the engine to reach normal operating temperature

8. Calculate the brake power using its equation and graph it proportional to engine speed.

9. Graph the torque proportional to engine speed. [7]

Shut Down:

1. Use engine throttle lever to reduce engine speed to minimum.
2. Allow engine to run for two minutes at minimum speed.
3. Turn off all loads switches in the load control panel.
4. Turn ignition switch to off position.
5. Turn off fuel supply to the engine.

Calculation:

Table: 6.1 Variable Speed, Variable Load.

Speed [rpm]	Load/Power [KW]	Torque [N.m]	Variable current (min-max)	Average Current [A]	Voltage [V]	Brake Power [Kw]
2000	0.26	2.14	2 - 3	2.5	104	0.447
2000	0.42	3.47	3 - 4	3.5	120	0.726
2000	0.63	5.22	3 - 5	4	158	1.093
2300	0.47	3.38	1 - 5	2.5	157	0.815
2300	0.68	4.88	3 - 5	4	170	1.764
2300	0.84	6.036	4 - 5	4.45	187	1.45
2600	0.38	2.42	1 - 5	3	127	0.66
2600	0.71	4.516	2 - 5	3.5	203	1.23
2600	1.08	6.86	2- 6	4	270	1.86

3300	0.88	4.43	1 - 6	3.5	251	1.52
3300	0.96	4.81	2 - 6	4	240	1.662
3300	1.03	5.13	3 - 6	4.5	228	1.782
3600	0.686	3.15	1 - 4	2.5	274.5	1.187
3600	1.15	5.27	2 - 6	4	287.5	1.99
3600	1.39	6.38	4 - 7	5.5	253	2.41

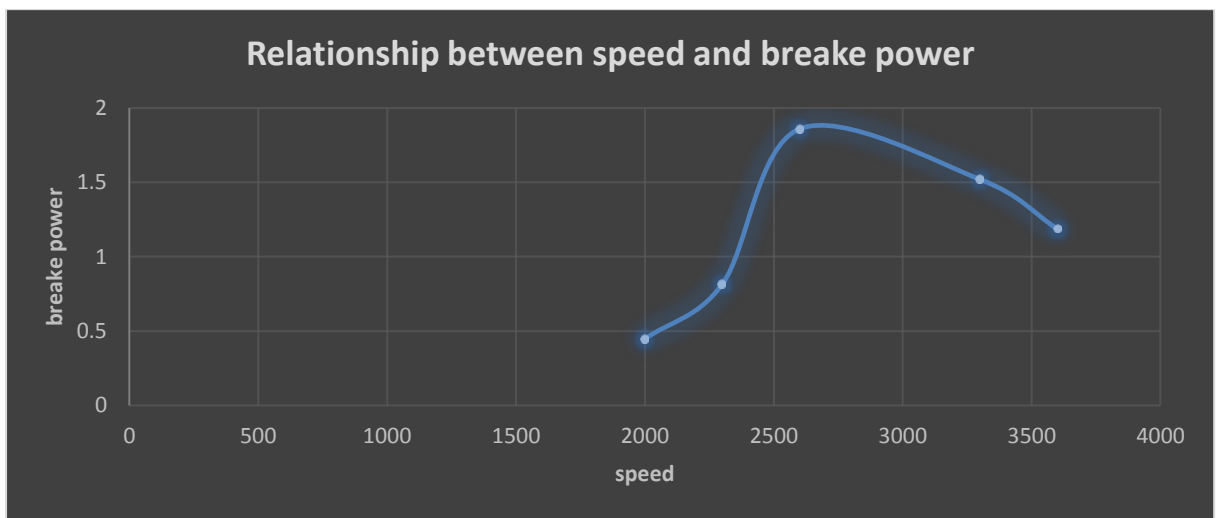


Figure 6.1: Brake Power and Speed.

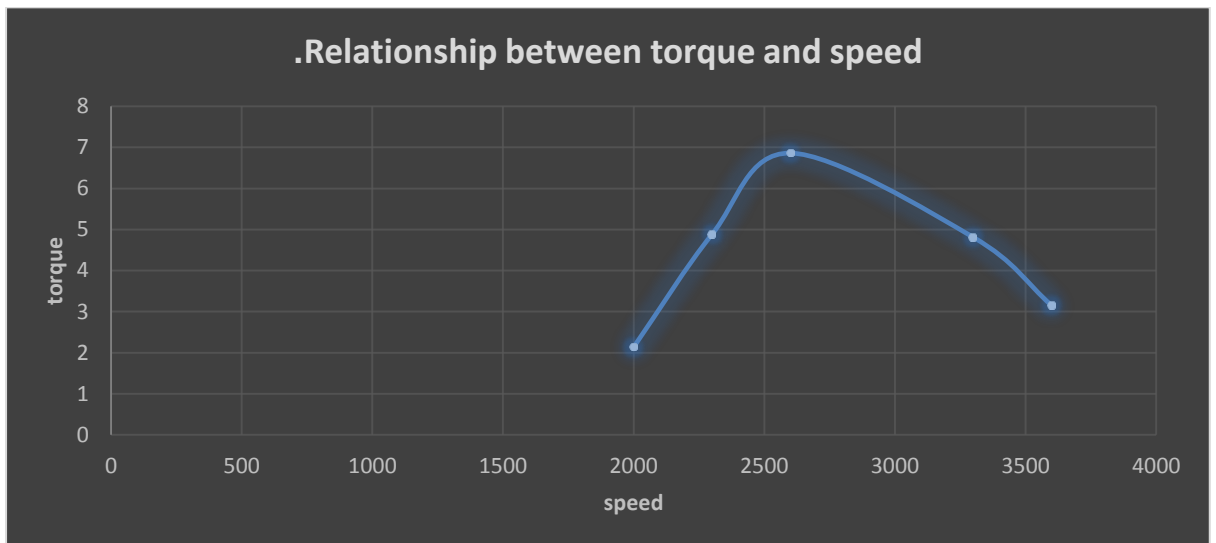


Figure 6.2: Torque and Speed.

Result discussion:

Figure 6.1 show the engine brake power with engine speed , with various load at constant speed operation starting in the last speed in the table and decreasing by applying load sequentially, every load applied we record the data.

Figure 6.2 show the engine torque with engine speed, also in this calculation using the mode of constant speed with varied load, using the load cell.

The load applied in all the experiments as the power consumed in the thermal resistance, from the ammeter device collecting the current.

The rated efficiency, the efficiency from the out power consumed by the thermal resistance on the input power as the brake power.

6.4 Experiment NO 2: (Fuel consumption)

Objective:

To measure the fuel consumption and specific fuel consumption.

Theory:

Specific Fuel consumption calculation.

The fuel mass of the engine calculated by the relation as shown in table 6.4.

Table 6.4: the Fuel Mass of Engine Calculation Equation.

$m_{\text{fuel}} = V * \rho$		
M	Mass of fuel	Kg
V	Volume of fuel, which can used	$8 \text{ ml} * 10^{-6} \text{ m}^3$
ρ	Fuel density	740 Kg/m^3

Another definition is a fuel mass flow per second and calculated as in the relation that shown in table 6.5.

Table 6.5: The Fuel Mass of Engine Calculation Equation.

$\dot{M}_{\text{fuel}} = \frac{m_{\text{fuel}} * 3600}{t} \text{ [Kg/ hr.]}$		
T	The time which can be calculated	S

In engine test, the fuel consumption measured as a flow rate-mass flow per unit time. A more useful parameter is Specific Fuel Consumption (SFC), the fuel flow rate per unit power output. It measures efficiently an engine is using the fuel supply to produce work as shown in table 6.6:

Equation the Specific Fuel Consumption:

$SFC = \frac{\dot{M}}{bp}$
--

Procedure:

To measure the fuel consumption of the engine.

1. Fill your tank with correct fuel for your test engine.
2. Open both valves on the fuel gauges (turn the valves so that they are in-line) and make sure that the fuel has passed down in the fuel feed pipes toward the test engine.
3. Make sure that your computer is operating correctly and has started TQ software (VDAS software, Engine Cycle Analyzer Software).
4. Check load control panel connections, and make sure that all loads switched off.
5. Turn ignition switch to on position, rotate the output shaft to the TDC mark, and start the engine.
6. Allow the engine to reach normal operating temperature
7. For the manual fuel gauge shut the fuel inlet valve and use stop watch to measure the time taken to drain 8 ml at varies engine speed
8. Increase Engine speed until the generator start supplying output.
9. Start applying loads sequentially one by one, record all engine test results and fill the table.
10. Calculate the fuel consumption and specific fuel consumption of the engine, and then graph the mass flow rate proportional to engine speed.

Shut Down:

1. Use engine throttle lever to reduce engine speed to minimum.
2. Allow engine to run for two minutes at minimum speed.
3. Turn off all loads switches in the load control panel.
4. Turn ignition switch to off position.
5. Turn off fuel supply to the engine.

Results:

Table 6.6: Fuel Flow Rate and Specific Fuel Consumption.

Speed (rpm)	Time (sec)	Mass Flow rate (Kg/hr)	SFC (Kg/Kw.hr)
2000	33.27	0.641	0.586
2300	29.21	0.729	0.414
2600	26.1	0.817	0.439
3300	24.76	0.861	0.566
3600	22.38	0.952	0.802

Table 6.7: Fuel Flow Rate [Without Load].

Speed [rpm]	Time (sec)	Mass Flow rate (Kg/hr)
2000	99.58	0.214
2300	64.62	0.329
2600	54.09	0.394

3300	47.19	0.451
3600	37.92	0.562

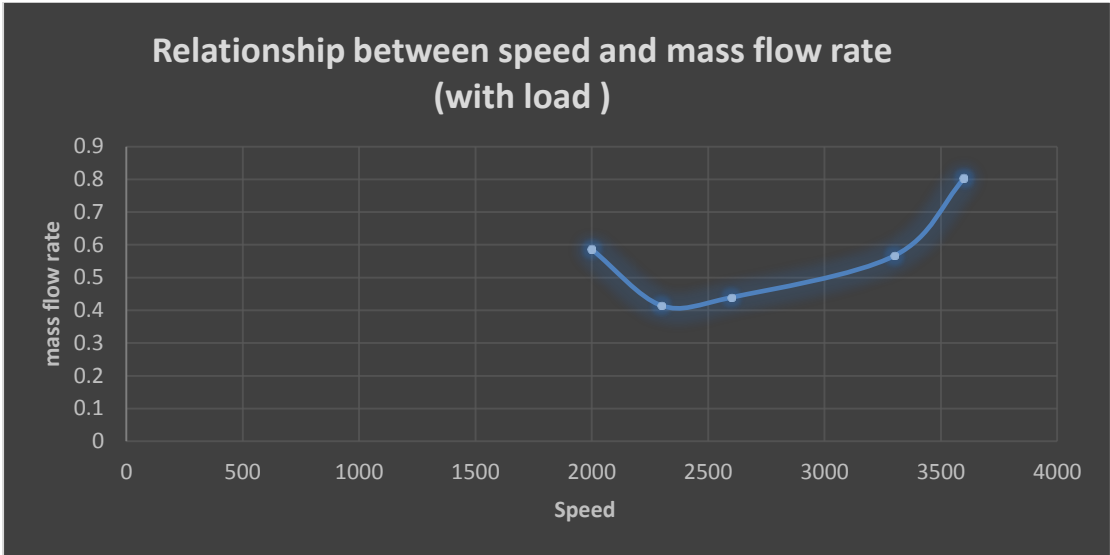


Figure 6.3: Specific Fuel Consumption Vs Speed (with load).

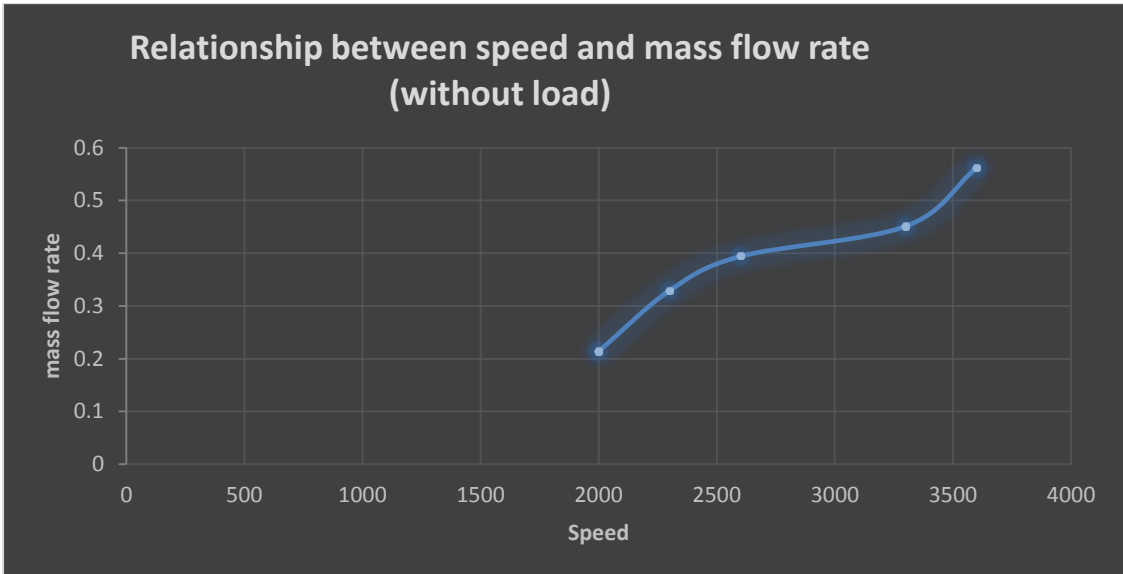


Figure 6.4: Mass Flow Rate Vs Speed [Without Applying Loads]

Result Discussion:

Figure 6.3 show the specific fuel consumption of the engine, this parameter depends on the brake power and the mass flow rate. The specific fuel consumption decreases firstly until it reaches the economic fuel consumption at approximately 2400 rpm, then starts increasing.

Figure 6.4 show the mass flow rate for the first table, and it increased with speed increasing, without any load applied.

6.5 Experiment No 3: (P- α) and (P-V) Diagrams

Objective: [7]

- 1- To facilitate understanding the thermodynamic processes occurring inside the engine cylinder such as variation in pressure.
- 2- To help the student to notice the actual pressure cycle (p- α) and (p-V) diagrams compared with the theoretical knowledge.

Procedure:

- 1- Fill your fuel tank with the correct fuel for your test engine.
- 2- Connect the exhaust pipe of the engine to the laboratory exhaust system.
- 3- Open both valves on the Fuel Gage-(turn the valves so that they are in-line with the fuel pipe).
- 4- Make sure that the fuel has passed down the fuel feed pipe to the test engine.
- 5- Check load control panel connections, and make sure that all loads switched off.
- 6- Turn on the engine cycle analyzer and make reset.
- 7- Make sure that your computer is operating and has started the equipment software
- 8- Turn the ignition switch to the on position and run the engine.

- 9- Allow the engine to reach normal operating temperature.
- 10- Make connection between engine cycle analyzer and the pc computer.
- 11- After engine reaches the normal operating temperature, make the reset to the engine cycle analyzer.
- 12- Start applying loads sequentially one by one, record all engine test results and fill the table.
- 13- Record all test results.
- 14- Turn off the engine.

Results:

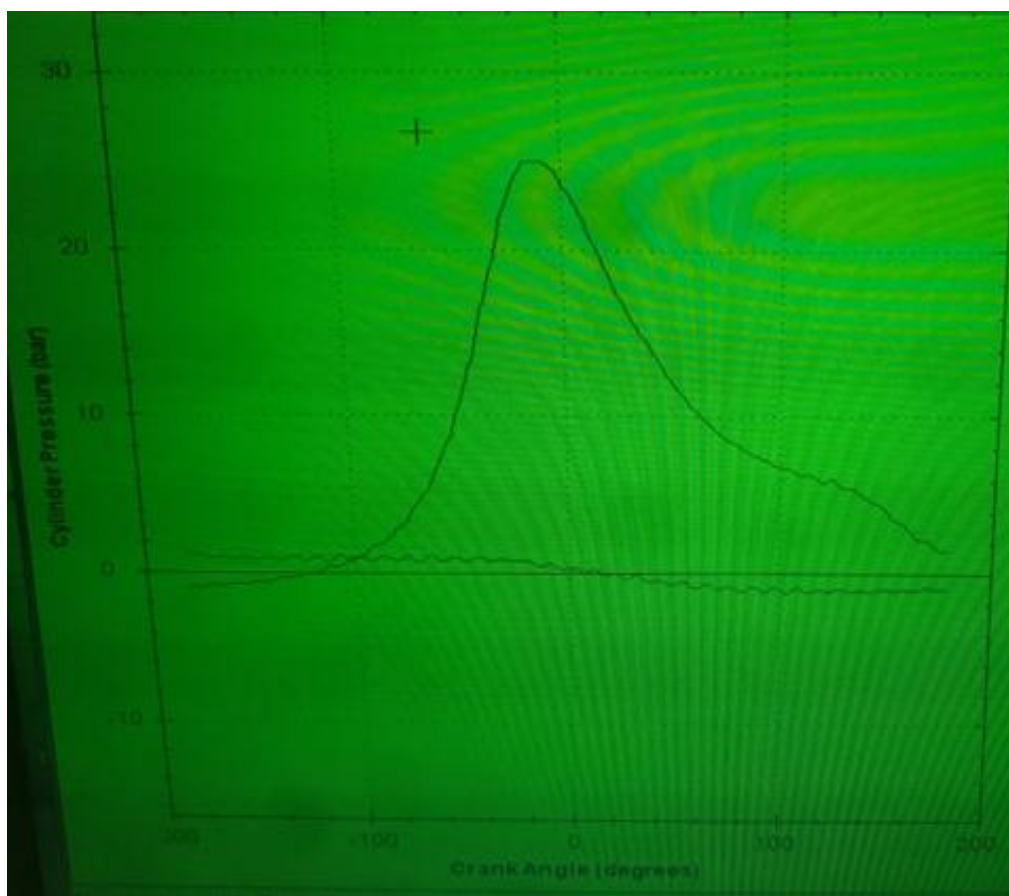


Figure6.5.1: Pressure-Crank Angle Diagram (P-α) at T=6.036[Nm], 2600[RPM].

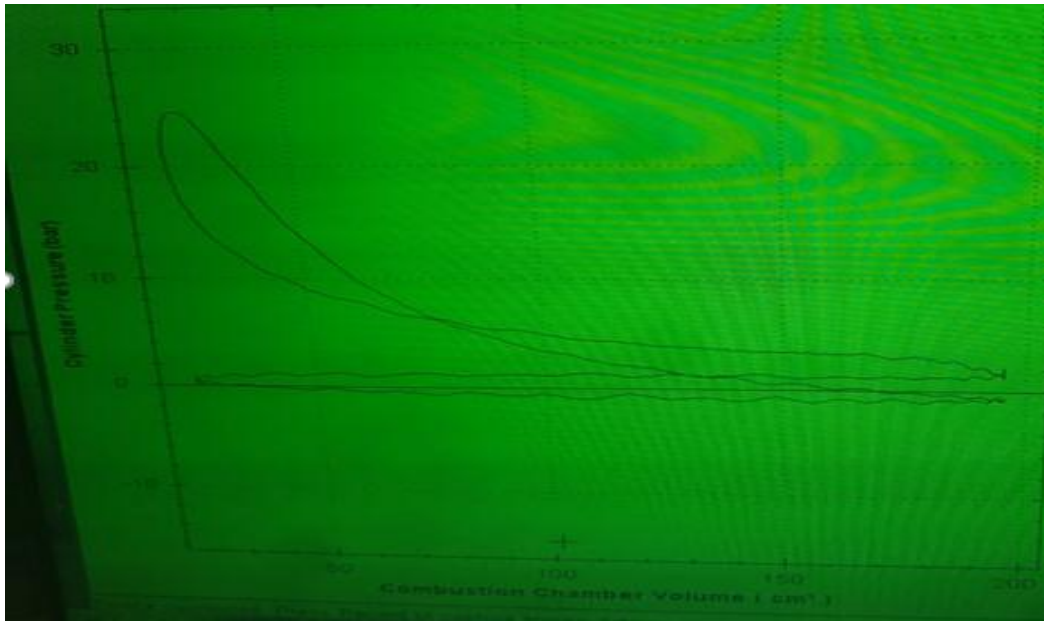


Figure 6.5.2: Pressure-Volume (P-V) at $T=6.036$ [Nm], 2600[RPM]

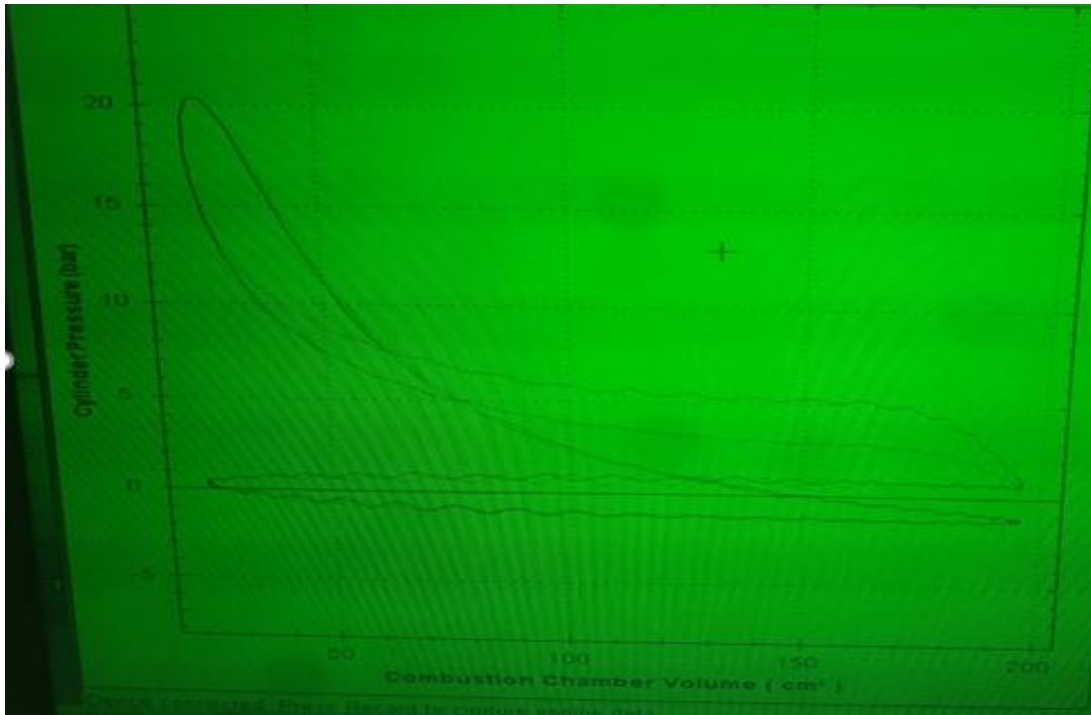


Figure 6.6.1: Pressure-Volume (P-V) at $T=3.47$ [Nm], 2000 RPM.

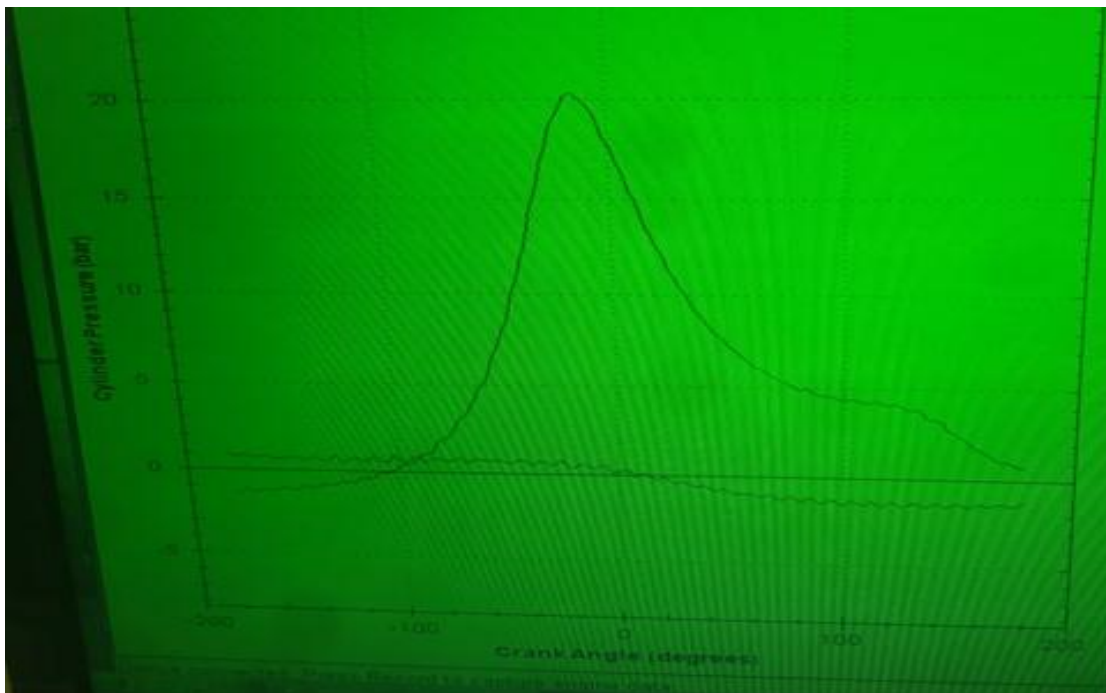


Figure 6.6.2: Pressure-Crank Angle Diagram (P- α) at $T=3.47$ [Nm], 2000 RPM.

Result discussion:

When the piston is at bottom dead center, the cylinder will have its largest volume. As the piston moves up the cylinder, the volume is reduced. At top dead center, the cylinder is at its minimum volume.

Once the piston reaches top dead center on the exhaust stroke, it starts back down the cylinder for the intake stroke. If the intake manifold is restrictive (or you have a small carb), the pressure in the cylinder will drop (more vacuum) as the piston moves down the cylinder during the intake stroke. This is represented by the line being at an even lower level as it moves back to the right during the intake stroke.

The piston now moves back up the cylinder on the compression stroke. This is represented on the diagram by the line swinging up the graph as the line moves back to the left. Near the end of the compression stroke, the ignition starts the combustion, at this point the line turns sharply and heads up to the peak pressure again.

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