A

Dissertation Report on

Design and Development of Dual Worm Self Locking Lifter

Submitted

in partial fulfilment of the requirements for the degree of

Master of Technology

in

Mechanical Design Engineering

by

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Sponsored by

Vedant Enterprises Pune, Maharashtra

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(An Autonomous Institute, Affiliated to Shivaji University, Kolhapur)

2020-2021

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CERTIFICATE

This is to certify that, Miss Shubhada Kalidas Gorave(1921001) has successfully completed the dissertation work and submitted dissertation report on (“Design and Development of Dual Worm Self Locking Lifter”) for the partial fulfillment of the requirement for the degree of Master of Technology in Mechanical Design Engineering from the Department of Mechanical Engineering , as per the rules and regulations of Rajarambapu Institute of Technology, Rajaramnagar, Dist: Sangli.

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TO WHOMESOEVER IT MAY CONCERN

This is to certify **MISS.GORAVE SHUBHADA KALIDAS** is the students of M.Tech MECHANICAL Engineering from RAJARAMBAPU INSTITUTE OF TECHNOLOGY, have completed the project work entitled **"Design And Development of Dual Worm Self locking lifter."** in our company from 1st Oct.2020 to 31st May2021 as part of their curriculum under my supervision, in the partial fulfilment of Masters of Engineering in Mechanical Engineering.

We have noticed that, during the period, she has shown keen interest in her assigned work and were also regular in attendance. The data provided for the project by her is satisfactory. This project is completed and validated by our engineer with the candidate successfully.

We wish her all the best for future endeavours.

Thanks & regards

Vedant Enterprises Pune

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DECLARATION

I declare that this report reflects my thoughts about the subject in my own words. I have sufficiently cited and referenced the original sources, referred or considered in this work. I have not misrepresented or fabricated or falsified any idea/data/fact/source in this my submission. I understand that any violation of the above will be cause for disciplinary action by the Institute.

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ABSTRACT

When an outside load applies a dynamic or static torque to the output worm gear shaft, and this torque does not result in any rotation of the input worm, the reducer is considered self-locking. Contingent on several design and load characteristics, worm gear speed reducers might be chosen which either self-lock or back-drive and in some restricted cases can do both depending on external loads and operational conditions. The worm gear drive when made self-locking with the view of safety becomes highly in efficient with drop in output power to 50 to 60% of the input power. Hence their use of lifting application becomes un-economical. But owingto the absence of any other effective self-locking devices the worm gear drives are still irreplaceable in applications where self-locking characteristics is desirable or inevitable

Keywords: Self-locking, Worm Gear, Lifting

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NOMENCLATURE

f	Coefficient of friction between the worm and the gear
ϑn	pressure angle of the gear train
Λ	Helix angle of the worm.
Cs	Service Factor
M	Module
W	Maximum Load
N	Neutron
d	Nominal Diameter
dc	Core diameter
dm	Mean diameter
Mt	Twisting Moment
α	Helix angle
φ	Frictional Angle
μ	Coefficient of friction
A	Constant
Le	Equivalent unsupported length of screw

K	Radius of gyration
Fc	Yield stress in compression
Wcr	Crippling Load
P	Equivalent dynamic load
X	Radial load constant
Fr	Radial load
Y	Axial load contact
$\delta 1$	Movement of ball clutch
$\delta 2$	Compression of ball clutch

ABBREVIATIONS

R.P.M.	Rotation per Minute
ASME	American society of mechanical Engineering
RH	Right hand
LH	Left hand

Chapter 1

Introduction

1.1 General

Lifting devices are used i industry to lift loads, in small machines even though the lifting capacity required is low, self-locking is very necessary. The term self-lockingas applied to gear systems denotes a drive which gives the input gear the freedomto rotate the output gear in both directions but the output gear locks with input when an outside torque attempts to rotate the output in either direction. This is desired in lifting machines meaning that if motor is lifting a load and it suddenlystops (due to power failure / breakage) load will stop moving and it will not come down due to gravity. thus, complete safety is ensured

1.2 Existing Methodlogy

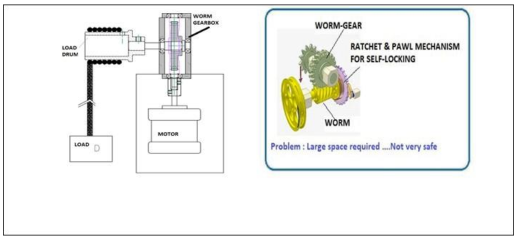

Figure 1.1: Existing methods of worm and worm wheel gear box with self-locking or worm gear box with Ratchet –pawl mechanism.

A worm gear is gear train where one of the meshing gears is worm (a screw-like gear) and another one is a spur gear. For power transmission, input torque is applied to the worm and it rotates and transmits the torque to the spur gear. As a result, the gear rotates. The above picture is showing a single start worm gear train. It is single start as you can see a single thread wraps around the worm. Similarly, two, three and multi start worm gear trains are also possible. A self-locking worm gear is a type of worm gear that does not allow the interchangeability of the input and output gears. As you know, in spur gear trains you can interchange the driving gear and the driven gear but the same is not possible for the self-locking type of worm gears. For this type of gear, the worm always acts as a driving gear and the spur gear as a driven gear- vice versa is not possible. If you try to run it otherwise, it will lock automatically. As a matter of fact, most of the worm gear trains used in industry are of the self-locking type. But you can of course design a non-self- locking type of worm gear. Approximately, if the tangent of the helix angle of the worm gear is less than the coefficient of friction between the worm and the gear, then the worm gear train should be a self-locking type. A more precise governing equation to ensure the self-locking feature of the worm gear train is:

$f < \cos\vartheta_n * \tan\wedge$. . . Eqn.1

Where,

f – Coefficient of friction between the worm and the gear.

ϑ n- is the pressure angle of the gear train

$\wedge$ - Helix angle of the worm.

Gears

You can achieve a large reduction ratio (as large as 200:1) from a self-locking worm gear without increasing the size of the gear box. How? A 360-degree rotation of a single start worm causes the meshing spur gear to rotate by one tooth. So, if a 10 teeth spur gear is meshed with a single start worm then you will get a reduction ratio of 10:1 straight away. Whereas, for achieving the same reduction ratio by using a spur gear train, you have to use another 100 teeth spur gear with that 10 teeth spur gear. So, imagine the comparative size reduction

A self-locking gear cannot be driven by the load in the opposite direction from its intended direction of rotation.

When worm gearing is self-locking or irreversible, this means that the worm gear cannot drive the worm

It is usually impractical to design irreversible worm gearing with any security.

The correct answer to all of these statements is true, under certain conditions. So how do you reconcile these seemingly contradictory statements: the load can- not backdrive in the opposite direction, yet it is impractical to design irreversible gearing? The explanation lies in understanding the difference between gear design theory and the practical aspects of applications engineering.
Applications teach hard lessons Within the last 3 years, our failure analysis engi- neers investigated two accidents involving worm gears where the user was injured.In each case, a designer selected a worm gear speed reducer from a catalog and incorporated it into a lift device intended to raise and lower people. In both cases, the designer was under the impression that the load (person plus platform) couldnot self-lower. They both relied on self locking worm gears to hold the cargo in a desired position and genuinely believed that no brake was necessary to supplement the worm gear in such a critical application

Self-locking worm gears: fact or fiction?
Friction angle experiment to understand the difference between conventional the- ory and application of worm gears, lets first conduct an experiment. Place a quarter on a book that is lying flat on a table. Slowly raise one end of the bookuntil the quarter begins to slide downward. Then lower the end slightly so the quarter is stationary. The angle between book and table, which just prevents slid- ing, is called the static friction angle. Its value depends on the amount of friction between the quarter and the book. Now with the book held at the friction angle, lightly tap the book and notice that the quarter slides down. The vibration (tap-ping) changes the friction from static to dynamic, which has a lower value, thus allowing the quarter to slide. A threaded fastener works the same way. The helix angle of its threads is small enough to prevent it from turning under static load.But add vibration, and the fastener loosens. The helix angle and the friction co- efficient are the key elements in determining the friction angle that makes “things stay put.” Worm gear theory A number of references will help you understand the theory of worm gear geometry. All theoretical analyses of self-locking worm gears deal with static conditions. In such an analysis, the load on the worm gear can’tdrive the worm if the coefficient of friction between worm gear and worm is larger

than the tangent of the worm's lead angle. In other words, the friction angle must be larger than the lead angle to prevent back driving.

1.3 Problem Statement

The efficiency of worm gear box depends upon coefficient of friction and lead angle , but lead angle must be between 1o to 3o to get self-locking , so efficiency of worm gear box is very low (below 50%) as shown in graph below The second method of Ratchet and pawl mechanism is not reliable or safe and it also takes a lot of space Hence a compact system with high efficiency is needed for lifting devices

1.4 Soution

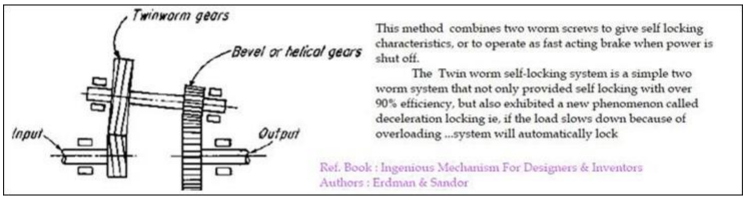

Figure 1.2: Twin worm arrangement

The twin worm gear drive is simple, two threaded rods, or worm screws (Input worm is Right hand & Output worm is Left hand and have a different pitch angle) are meshed together for proper mesh the worm axes are not parallel but inclined to each other, the drive shows self- locking or combination of self-locking and deceleration locking as required.

Chapter 2

Literature Review

2.1 Introduction

This Section describe Present and past studies done on worm gears. Different aspect of worm gear studied by different researcher. These reaserch paper are sumerise are as follow

YallamtiMurali et.al have studied the optimization of spur gear using genetic algorithm. The design variables for spur gear set are module, face width and number of teeth on the pinion, minimizing the centre distance, weight and tooth deflection of gears are taken as the objective function and subjected to constraints such as bending stress and contact stress. The proposed algorithm does not require gradient information of the objective function, which makes it very attractive. The results of proposed algorithm have been compared to those of the traditional techniques, such as, graphical technique, geometric programming, etc for solving the same problem and proposed traditional techniques [1]

Marimuthu, G. Muthuveerappan et al.analysed the effect of the module, gear teeth and addendum height on the load sharing and corresponding stress. In order to calculate stress due to applied load at highest pressure angle, they developed a multi-point contact model for finite element analysis. For the parametric study,they developed ANSYS parametric design language code. It was seen that in the application of load at the critical loading point, an increase in addendum height increases the bending stress. On the contrary, increase in module and number of teeth favourably decreases the bending stress [2]

Spur Gear In this paper Faisal S et al explained the parallel axis spur gear reduction unit which is the type, probably encountered most often in general practice. Optimized design of spur gear indicates that compact design of spur gears involves a complicated algebraic analysis. The author describes the development of such a design methodology and diagnostic tool for determining the modes of failure for spur gear and also the causes of the failures. The ray diagrams are incorporated to make the design more feasible with respect to the transmission ratio and number of teeth used in gearbox. The mode of failure curve in a design space shifts quite appreciably as torque increases. Further the author explains the mode of failures curves is not showing any change in the behavioural pattern when the pressure angle is changed and the large pressure angle gears have smaller value of pinion teeth. This clearly indicates that the lower number of teeth rather than the higher module which reduces the size of gear sets [3].

Christian Brechera et al showed the potential of optimization for face-milled bevel gears. The comparison of the measured and the simulated TE showed that the FE-based tooth contact analysis ZaKo3D is capable of giving a good agreement between simulations and testing. The requirement is that the scatter of the topography is known and, if necessary, considered in the simulation. Therefore, the method is applied for an exemplary gear set. Two variants are investigated, a ground target topography is calculated and as well a variant with a scatter in the topography deviations. The topography scatter is derived from a possible lapping process, but can be also applied on purpose in the grinding process. By applying the micro geometry scatter from tooth to tooth, the results of the TE calculation show that the amplitudes of the tooth mesh frequencies decrease. In parallel, the amplitudes of the frequencies in between the tooth mesh [4]

Werner Sigmund et.al evaluate load-carrying capacity as well as overall efficiency of large sized worm gears are carried out on the large-sized worm gear test rig developed by FZG. In order to apply a particular load on the test worm wheel, they have used hydrostatic torque motor which is connected to the reverse transfer gearbox. Via a summation gearbox, the bracing cycle is closed. Consequently, only occurring overall power losses have to be fed in by a direct current motor. There main aim for the project is to analyse large-sized worm gears with centre distance a=315mm with regard to their wear and pitting behaviour and project is to analyse large-sized worm gears with centre distance a=315mm with regard to their wear and pitting behaviour and overall efficiency at different operating as well as with various lubrication conditions. They have continuously measured and analysed the overall efficiency of worm gear test rig with centre distance 315mm. They find overall efficiencies of up to η=96% are measured at present operating conditions. Although an increasing pitting damage occurs up to η=45%, no significant influence on efficiency is recorded. With the use of lubricants with low viscosity (ISO VG 220) slightly lower efficiencies are reached when compared to lubrication with an oil of viscosity ISO VG 460. A mineral oil with comparable viscosity even leads to significantly lower values. [5]

Mr. Naeem et.al has designed and studied about dual worm self-locking sys- tem for improved transmission efficiency & deceleration locking property. In thishe compared the conventional system and mating worm gear. He said that the efficiency of worm gear depends on the coefficient of friction and the lead angle.In order to obtain a worm gear with high efficiency it is recommended to use thelead angle in the range between 1 and 3 . In his study he also concluded that mating worm pair system [6]

Yuvaraj Jadhav the efficiency of the worm gear pair was measured experimen- tally on the dynamometer. The efficiency of the newly designed worm gear pairfor plug valve application is high, and depends on the output torque.[7]

Pavel Melnikov et.al (recommended, the use of the hardest and wear-resistant material in the manufacture of the worm and the worm wheel will increase the life of the gearbox. An increase in the width of the worm wheel crown and advanced technologies for processing gearbox parts will lead to a higher efficiency value. [8]

2.2 Literature Gap

After careful study of literature, it is found that though various methods of self-locking are suggested a precise and compact method is needed that will give quick self-locking and higher efficiency of transmission hence there is need to produce such drive. More over 3-D printing is the latest technology available in market which gives very fast but extremely cheap method of manufacturing parts having complex shapes. This method will be tried and tested by means of the project to establish its utility in such applications.

2.3 Project Objectives

I. To design of twin worm system for design load to achieve self-locking condition.
II. To use application of 3-D printing technology for fast and low-cost manufacturing of input and output worm.
III. To take test and trial to determine Torque-Speed-Power-efficiency of system

Chapter 3

DESIGN METHODOLOGY

3.1 Design

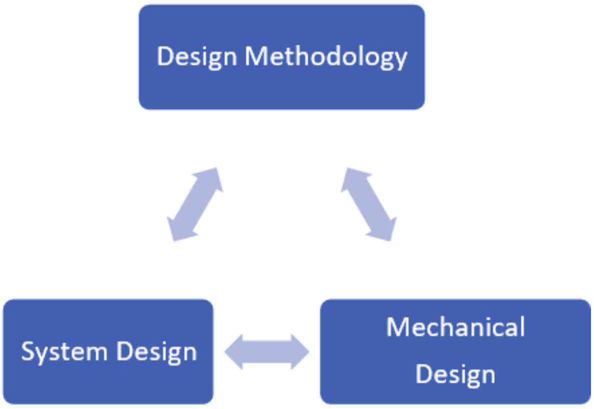

Figure 3.1: DESIGN METHODOLOGY

Hence Design consists of application of scientific principles, technical informa- tion and imagination for development of new or improvised machine or mechanism to perform a specific function with maximum economy & efficiency. a careful de- sign approach has to be adopted. The total design work, has been split up intotwo parts

1. System design
2. Mechanical Design.

System design mainly concerns the various physical constraints and ergonomics, space requirements, arrangement of various components on main frame at system, man + machine interactions, No. of controls, position of controls, working environment of machine, chances of failure, safety measures to be provided, servicing aids, ease of maintenance, scope of improvement, weight of machine from ground level, total weight of machine and a lot more. In mechanical design the compo- nents are listed down and stored on the basis of their procurement, design in two categories namely, designed parts and parts to be purchased for designed parts detached design is done & distinctions thus obtained are compared to next high-est dimensions which is readily available in market. This amplifies the assemblyas well as postproduction servicing work. The various tolerances on the works are specified. The process charts are prepared and passed on to the manufacturing stageThe parts which are to be purchased directly are selected from various cat- alogues & specified so that anybody can purchase the same from the retail shopwith given specifications.

3.1.1 System Design

In system design we mainly concentrated on the following parameters: -

1. System Selection Based on Physical Constraints

While selecting any machine it must be checked whether it is going to be used in a large-scale industry or a small-scale industry. In our case it is to be used bya small-scale industry. So space is a major constrain. The system is to be very compact so that it can be adjusted to corner of a room. The mechanical designhas direct norms with the system design. Hence the foremost job is to control the physical parameters, so that the distinctions obtained after mechanical design canbe well fitted into that.

2. Arrangement of Various Components

Keeping into view the space restrictions the components should be laid such that their easy removal or servicing is possible. More over every component should be easily seen none should be hidden. Every possible space is utilized in component arrangements.

3. Components of System

As already stated, the system should be compact enough so that it can be accommodated at a corner of a room. All the moving parts should be well closed & compact. A compact system design gives a high weighted structure which is desired

4. Man Machine Interaction

The friendliness of a machine with the operator that is operating is an important criteria of design. It is the application of anatomical & psychological principles to solve problems arising from Man – Machine relationship. Following are some of the topics included in this section.

Energy expenditure in foot & hand operation

Lighting condition of machine.

5. Chances of Failure

The losses incurred by owner in case of any failure is an important criterion of design. Factor safety while doing mechanical design is kept high so that there are less chances of failure. Moreover, periodic maintenance is required to keep unit healthy.

6. Servicing Facility

The layout of components should be such that easy servicing is possible. Especially those components which require frequents servicing can be easily disassembled

7. Scope of Future Improvement

Arrangement should be provided to expand the scope of work in future. Such as to convert the machine motor operated; the system can be easily configured to required one. The die & punch can be changed if required for other shapes of notches etc.

8. Height of Machine from Ground

For ease and comfort of operator the height of machine should be properly decided so that he may not get tired during operation.

3.1.2 Mechanical Design

In system design we mainly concentrated on the following parameters: -

1. Drive Motor

The drive motor is 12 VDC motor coupled to an planetary gear box. Specifications of motor are as follows:

A) Power 5 watt

B) Speed = 60 rpm

C) TORQUE = 0.833 N-m

selection of motor:

1. Drive Motor:

The drum is 100 mm in diameter hence the design torque at the end of the drive will be given as

T = Load x 9.81 x radius of Drum = 2 x9.81x $\frac{100}{2}$ = 0.98 n0m approx. 1.0 N-m

Thus, we are introducing a spur gear pair of following specifications drive train

Pinion = 12 teeth

Gear = 120 teeth

thus, transmission ratio = $\frac{120}{12}$=10

the final torque received by the system as input will be 0.833 x 10 = 8.3 N-m

Thus, the selected motor can easily lift the designed load of 2 kg

Therefore, the selection of motor is justified

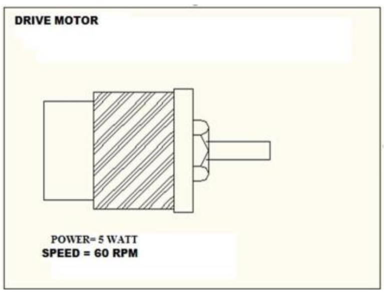

Figure 3.2: 12 V DC Motor

2. Design of spur gear pair for power transmission between motor and dual worm system

Drive has a pinion & gear arangement Maximum torque = 2 N-m

No of teeth on gear =120

No of teeth o pinion = 12

Module = 1.27 mm

Radius of gear by geometry =$\frac{120*1.27}{2}$=76.2mm

Maximum load =$\frac{t}{r}$ = $\frac{2*10^3}{76.2}$ =26.25 N

b = 10 m

Material of spur gear and pinion = Nylon 6 Sultpinion =Sult gear = 60N/mm2

Service factor (Cs) = 1.5

Peff=(W x Cs) = 39.3 N.

Peff = 39.3N (as Cv =1 due to low speed of operation)

Peff = 39.3 N (A)

Lewis Strength equation WT = Sbym

Where;

Y=0.484-$\frac{2.86}{Z}$

=0484-$\frac{2.86}{12}$

=0.245

Syp =14.74

As pinion is weaker

WT = (Syp) x b x m

=14.74 x 10m x m WT=147.4m2 (B)

Equation (A) & (B) 147.4 m2 = 39.3

m=0.51 mm

selecting standard module = 1.27 mm for ease of construction as we go for single stage gear box. . . making size compact . . . achieving maximum strength and proper mesh.

3. Input Shaft

ASME CODE FOR DESIGN OF SHAFT

MATERIAL SELECTION: -Ref: - PSG (1.10 & 1.12) + (1.17)

Table 3.1: Material Selection

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm2	YEILD STRENGTH N/mm2
En24	800	680

Since the loads on most shafts in connected machinery are not constant, it is necessary to make proper allowance for the harmful effects of load fluctuations

According to ASME code permissible values of shear stress may be calculated form various relation.

fs_{max} = 0.18 fult

= 0.18 x 800

= 144N/mm2

or

fs_{max} = 0.3 fyt

=0.3 x 680

= 204 N/mm2

considering minimum of the above values;

fsmax = 144 N/mm2

Shaft is provided with notch for locking; this will reduce its strength. Hence reducing above value of allowable stress by 25%

fs_{max} = 108 N/mm2

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

T= 2 x103 N-mm

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.

Assuming minimum section diameter on input shaft = 16 mm — GEAR BORE IS READY MADE OF 16 MM DIAMETER

d = 16 mm

$Td = \frac{\pi}{16} x\ fs_{act} \ x\ d^3$

$fs_{act}= \frac{16*T*d}{\pi*d^3}$

$=\frac{16*2*10^3*16}{\pi*d^3}$

=39.78 N/mm^2

$fs_{act} < fs_{all}$

I/P shaft is safe under torsional load

Analysis of input shaft

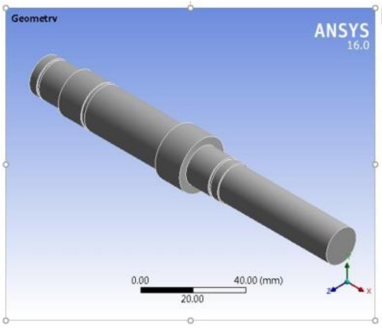

Figure 3.3: Input Shaft

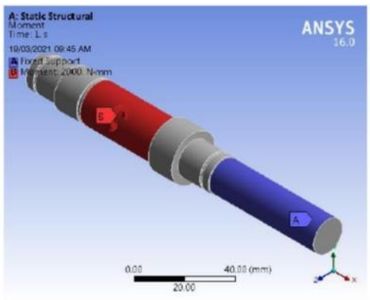

Figure 3.4: Input Shaft

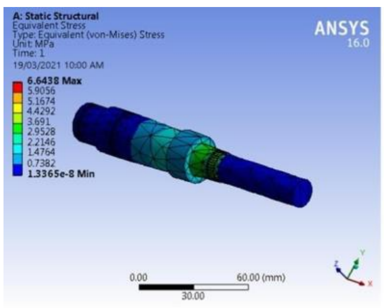

Figure 3.5: Equivalent Stress in Input Shaft

The maximum value of stress induced is 6.6438 Mpa which is well below the maximum, allowable stress of 39.78N/mm2 the input shaft is safe.

4. Design of Input worm

d = Nominal /outer diameter (mm) = 32.36mm

dc = core / inner diameter (mm) = 22.36mm

dm = mean diameter (mm) = 19mm

Mt = W x $\frac{am}{2}$ tan $(\vartheta + \mu)$

Where, W= Axial load

Ø = friction angle

μ = Heilx angle

tanμ = $\frac{L}{\pi dm}$

For the single start sq. thread lead is same as pitch=10

tanμ= $\frac{10}{\pi * 27.36}$

μ=6.63

Friction Angle ref: R.S.Khurmi (Table No 17.5)

Table 3.2: Coefficient of friction

Condition	Average coefficient of friction	
	Starting	Running
Average quality of material & workmanship & average running conditions	0.18	0.13

μ = tan ϑ

0.18 = tanϑ

ϑ= 10.2

Assuming that load of 800 g is carried by the drum of 120 mm diameter, then the resultant torque T= 8000X 60 =480000N-mm. ....(A)

M_t = WX27.32/2 x tan (10.2 + 6.63)

M_t = 4.13 X W N-mm. ..... (B)

Equating (A) & (B)

W =116.22K N

Material selection:

Ref: - (PSG – 1.12)

Table 3.3: Material selection

Designation	Tinsel Strength N/mm2	Yield Strength N/mm2
ABS POLYMER	60	42

Direct Tensile or Compressive stress due to an axial load: -

a) Torsional shear stress: -

$fc_{act}= \frac{w}{\pi/4*dc^2}$

$= \frac{116.42}{\pi/4*22.36^2}$

$=296.47 N/mm^2$

$T = M_t = \frac{\pi}{16} x\ f_{act}\ x\ dc^3$

$2x10^3 = \frac{\pi}{16}\ x\ f_{act}\ x\ (22.36)^3$

$fs_{act} = 0.88\ N/mm^2$

As fsact< fall ; the screw is safe in torsion.

Stresses due to buckling of screw: - According to Rankine formula,

$Wcr = \frac{FcA}{l+a(le/k)}$

Where,

Wcr = Crippling load on screw (N) A = Area of c/s at root (mm^2)

A= constant

Le= Equivalent unsupported length of screw (mm)

K= Radius of gyration = dc/4 (mm)

Fc= Yield stress in compression (N/mm^2)

le = 0.5L; as both end of screw is considered to be fixed)

le = 0.5x 45= 22.5mm

As the theoretical crippling load is well above the design load the worm is safe

Analysis of Input worm

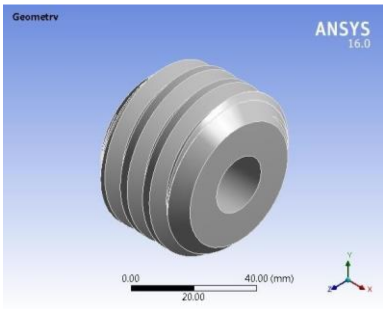

Figure 3.6: Input worm

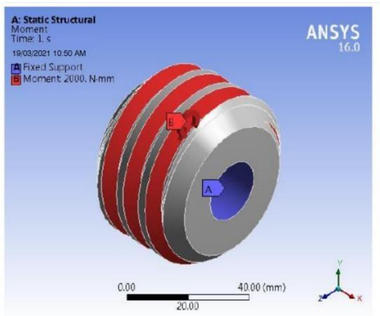

Figure 3.7: Input worm

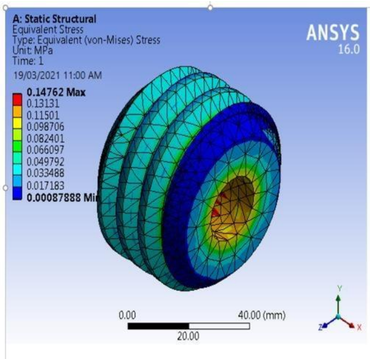

Figure 3.8: Equivalent Stress in Input worm

the maximum value of stress induced is 0.14 Mpa which is well below the maximum, allowable stress of 30 N/mm2 the input worm is safe.

Analysis of output worm

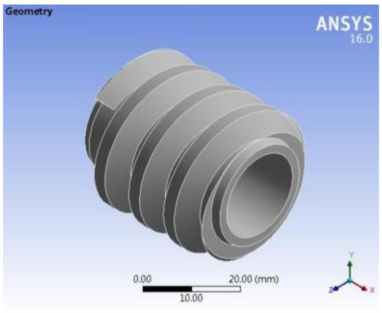

Figure 3.9: Output worm

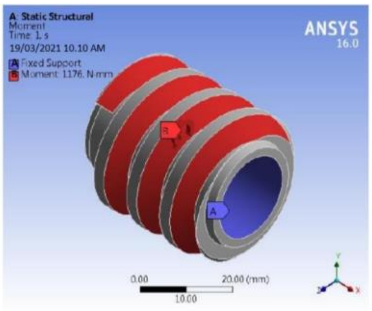

Figure 3.10: Output worm

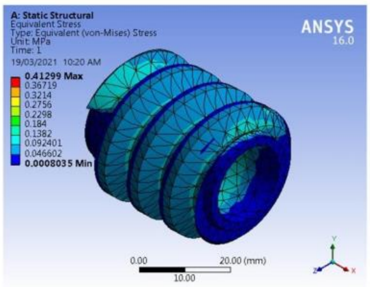

Figure 3.11: Equivalent Stress in Output worm

4. Design (selection of ball bearing) for shaft In selection of ball bearing the main governing factor is the system design of the drive i.e.; the size of the ball bearing is of major importance ; hence we shall first select an appropriate ballbearing first select an appropriate ball bearing first taking into consideration con- venience of mounting the planetary pins and then we shall check for the actuallife of ball bearing .

BALL BEARING SELECTION.

Series 60

Table 3.4: Ball Bearing Selection

ISI NO	Basic Design No (SKF)	d	D1	D	D2	B	Basic capacity	
							C kgf	Co Kgf
20B C003	6203	17	21	40	36	12	4400	750

P = XFr + Yfa. Where;

P=Equivalent dynamic load,(N) X=Radial load constant

Fr= Radial load(H)

Y = Axial load contact Fa = Axial load (N)

In our case;

Radial load FR= RA

= 69N

Axial load (Fa) = 0 P x Fr =69

L= $(C/P)^p$

here, p=3, for single row ball bearings

$$L = \frac{60nlh}{10^6} = \frac{60*1000*60}{10^6} \quad C = 106.6$$

AS; required dynamic of bearing is less than the rated dynamic capacity of bearing Bearing is safe

5. 2-D manufacturing Drawings

Base Frame

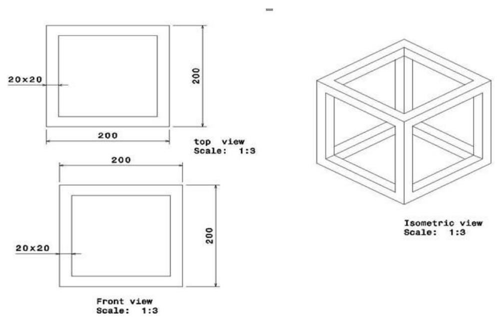

Figure 3.12: Base Frame

Base plate

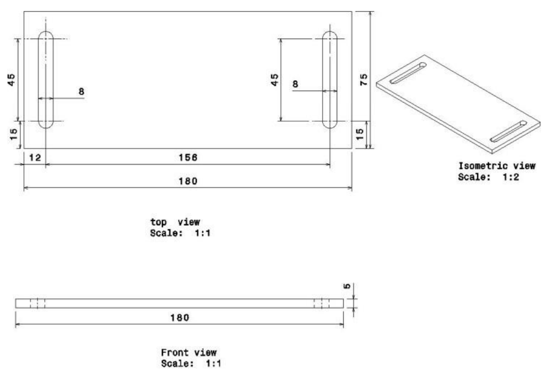

Figure 3.13: Base plate

LH Bearing Housing holder

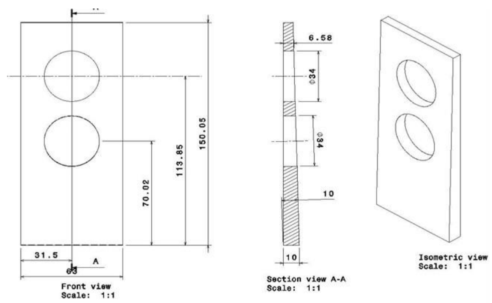

Figure 3.14: LH Bearing Housing holder

RH Bearing Housing holder

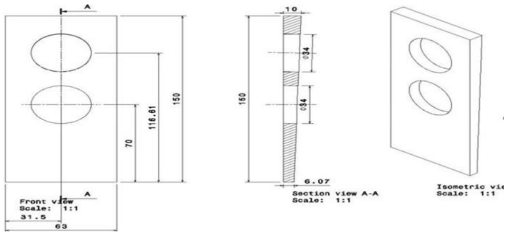

Figure 3.15: RH Bearing Housing holder

Input shaft

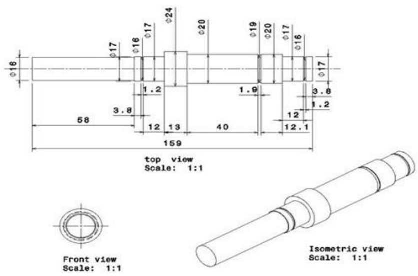

Figure 3.16: Input shaft

Output shaft

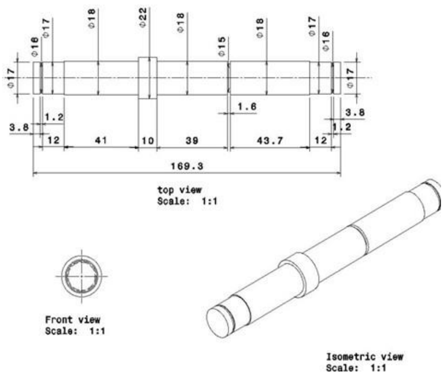

Figure 3.17: Output shaft

Load Drum

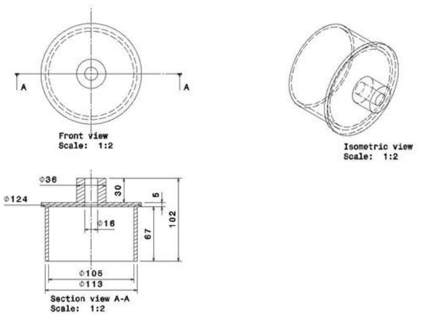

Figure 3.18: Load Drum

Input worm

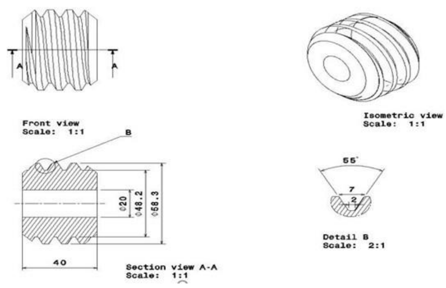

Figure 3.19: Input worm

Output worm

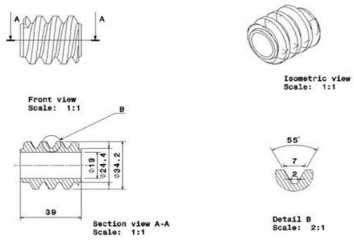

Figure 3.20: Output worm

Spur Pinion

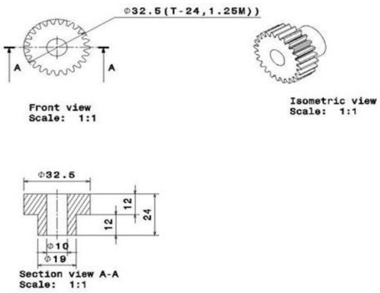

Figure 3.21: Spur Pinion

Spur Gear

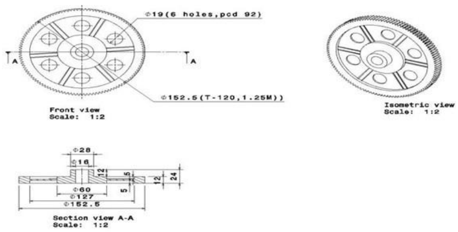

Figure 3.22: Spur Gear

Chapter 4

BALL CLUTCH

4.1 Introduction

Clutch is a mechanism which enables the rotary motion of one shaft to be transmitted, when desired to a second shaft the axis of which is coincident with that of the first

Positive clutches are used to transmit power between two coincident shafts. The positive engagement between the clutch elements ensures 100% torque transmission on but occasionally the output shaft may the subjected to a sudden overload which may make the driving motor or engine to stall; which will lead to burnout of the electric motor. In extreme cases this overload will lead to the breakage of drive elements or the clutch itself In order to avoid the damage of the transmission elements it is necessary that the input and output shafts be disconnected in case of sudden overloads .The isolation of the input driver member ie; motor from the output member is absolutely necessary to avoid damage and it is possible by called ball clutch.

4.2 Construction

The construction of test rig in general shall be as follows.

A] BALL CLUTCH

Ball clutch is the transmission element that connects the input shaft and output shaft . It is a self-contained and compact unit . Its members are keyed to the input and output shafts respectively.

B] BASE FLANGE

Base flange is an slightly hollowed out flange on its end face to leave an narrow annular band at the periphery. A series of radial Vee shaped serrations of identical size and shape and at equal pitch spacing a part, milled across the annular bandas shown. The sides of these serrations are inclined at 37 ^{0} (or 45^{0}) relative tothe axis of shaft. The included angle between the faces of the Vee serrations is90^{0}. This is important dimension, can of course, be varied within certain limitsin accordance with the load to be transmitted and The size of serrations is also determined by the diameter of the driving balls engaging in the The cylindricalsteel body of the ball clutch is keyed to the driven shaft but it is made slightlylonger than the shouldered end of that member so that the short smaller diame-ter concentric portion of its bore is a slip fit over the adjoining end of the shaft. Projecting beyond the base flange as shown. The purpose of this arrangement is to maintain a body of the clutch perfectly concentric with the base flange for ensuring the smooth and accurate engagement of the balls in the base flange ser- rations The left hand end of the body is recessed a small depth to admit the serrated portion of flange, the outside diameter of which has a tight clearance fitin recess. Six holes are accurately drilled and reamed passing axially through the body, The holes are spaced exactly 60^{0} a part around the same pitchcircle, thediameter of which is equal to pitch diameter of the serrations in the annular bandof baseflange. The right hand end of the body is reduced in diameter and threadsto receive the hardened steel casing.

C] CASING

Hardened steel casing is of the same outside diameter as the front end of the body. The sleeve is deeply bored at one side to be close fit over the reduced portion on the outside of the body. By fitting the casing over the body at that point its correct and accurate location relative to the body is not determined by the fit inthe threads . The three plungers bear simultaneously against the inner left hand face of the sleeve ; thus as that members advanced longitudinally, all springs willbe compressed or expanded by an equal amount.

D] LOCKNUT

A threaded lock nut is screwed on the body behind the sleeve for locking the sleeve in any desired setting.

4.3 Design of ball clutch

Ref: - PSG DESIGN DATA HAND BOOK (Fig.1)

Ball Clutch Nomenclature

d = Diameter of ball, mm

D = Pitch circle diameter of groove, mm

Ft = Total tangential force on balls, N

Fs = Total spring force, N

F = Spring force on each ball, N

α = Angle of inclination of groove k Spring stiffness, N/mm

Lf = Free length of spring, mm

Mt = Torque transmitted, N mm

n = Number of turns in the spring

p = Pitch of spring coil, mm

zb = Number of balls in the clutch

μ = Coefficient of friction

K1 = Stiffness per turn N /mm

$\delta 2$ = Movement of ball while Clutch is slipping ,mm

$\delta 1$ =Compression of spring to exert force F,mm

A) CALCULATION OF TANGENTIAL FORCE ON BALLS (Ft)

Ft= $\frac{2*mt}{D}$

Ft= $\frac{2*0.742*103}{90}$

(Assuming pitch circle diameter of the grooves(D) = 90 mm)

Ft = 16.489 N

B) CALCULATION OF TOTAL SPRING FORCE ON BALLS (Fs)

Fs= $\frac{Ft[cosa - \mu sina - \mu]}{sina + \mu cosa}$

where, α = Angle of inclination of groove = 45^0

μ = Coefficient of friction between the balls and body of the clutch= 0.08

Fs= $\frac{16.489[cos45 - 0.08sin45 - 0.08]}{sin45 + cos45}$

Fs = 12.727 N

C) CALCULATION OF FORCE ON EACH SPRING (F)

F = $\frac{Fs}{Zb}$

Zb = Number of balls in clutch

= 3 No's

F = $\frac{12.727}{3}$

=4.242 N

D) STIFFNESS OF SPRING (Ks)

Ks = $\frac{K1}{n}$

where K1 = Stiffness of spring per turn K1 (N/mm)

n = Number of turns of spring = 6

$Ks = \frac{7.98}{6}$

$=1.33$ N/mm

Ref. PSG DESIGN DATA HANDBOOK

Stiffness and permissible static and dynamic loads for helical compression springs

Table 4.1: Stiffness and permissible static and dynamic loads for helical compression springs

Wire Diameter mm	Outer Diameter mm	Stiffness Of spring Per turn K1 N/mm	Permissible load	
			Static Load N	Dynamic Load N
1.0	12.0	7.98	32.4	14.5

E) COMPRESSION OF SPRING TO EXERT A FORCE ‘ F’ (δ1)

$\delta 1 = \frac{F}{Ks}$

$= \frac{4.242}{1.33}$

$=3.18947$ mm

F) MOVEMENT OF BALL WHILE THE CLUTCH IS SLIPPING (δ2)

$\delta 2 = \frac{d(1-cosa)}{2}$

where,

d = Diameter of ball

=14 mm

$\delta 2 = \frac{14(1-cos45)}{2}$

$\delta 2 = 2.050$ mm

G) MAXIMUM DEFLECTION OF SPRING (δ max)

H)

δ max=δ 1 + δ 2

= 3.18947 + 2.050

δmax = 5.23947 mm

I) FREE LENGTH OF SPRING (Lf)

Lf = Solid length + Maximum deflection + Clearance between adjacent coils

= n'd + δmax + (n'-1) Where,

n' = n + 2

= 6 + 2

n' = 8

Lf = 8 x 1 + 5.23947 + (8-1)

Lf =20.23947mm

J) PITCH OF SPRING COIL (p)

$$p=\frac{Lf}{(n-1)}$$

$$=\frac{20.23947}{(6-1)}$$

=4.047

1.	Diameter of ball	(d)	14 mm
2.	PCD of grooves	(D)	90 mm
3.	Angle of inclination of grooves	(a)	45^0
4.	Rod diameter of springs	(d’)	1 mm
5.	Outside diameter of spring	(D’)	12 mm
6.	Pitch of coil	(p)	4 mm
7.	Free length of spring	(Lf)	20 mm

Table 4.2: Dimension of slipping clutch

4.4 Working

The overload slipping ball clutch is an safety device used in the transmission line to connect the driving and driven elements such that in case of occasional overload the clutch will slip there by disconnecting the input and output members. Thisprotects the transmission elements form any breakage or damage. For a particular loading conditions the clutch is preset to set the cylindrical body for slippingat a different overload , it is simply mounted on output member by means of akey .Casing is adjusted in the appropriate direction , during which the balls will remain pressed against the serrations ; thus setting operation is simple , rapid and reliable .The clutch is there connected to the output member or load. When the input shaft is in rotation through the reduction pulley and motor, the base flangeis rotated, along with it the balls pressed against Vee - serration also rotate. This motion is transmitted through springs; plunger to the cylindrical body which then rotates the outpushaft. When the load on the output shaft exceeds the preset design overload the resistance of the balls to more in direction of motion of base flange , there by balls start slipping in the Vee serrations . At one point the balls completely come out of the serrations into open space in base flange thereby disconnecting the base flange and the cylindrical body. Thus the input shaft keeps rotating whereas the output shaft comes to stand still The overload valueat which clutch slips can be designed and present the casing in either directionof the cylindrical body .To increase the overload value; move casing towards the base flange where as to reduce the overload ; move the casing away from the base flange The casing can be locked in position by means of the lock nut.

4.5 Design and analysis of RH gear base flange

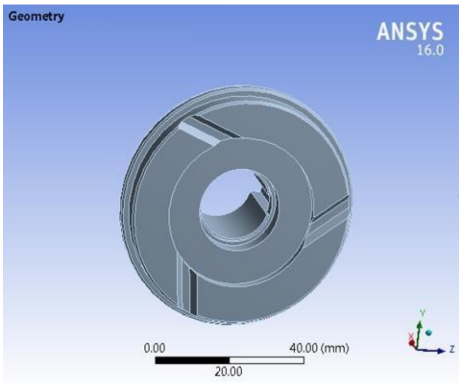

Figure 4.1: Base Flange

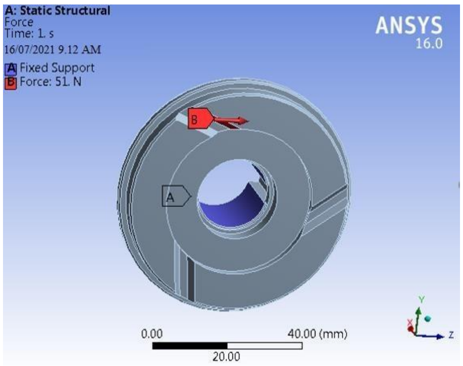

Figure 4.2: Base Flange

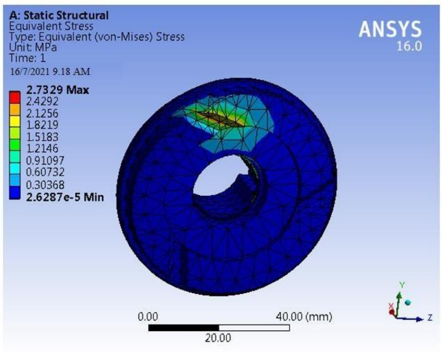

Figure 4.3: Equivalent Stress in base flange

4.6 Design and analysis of Clutch body

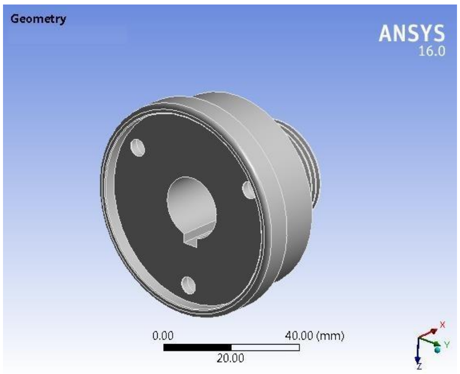

Figure 4.4: Clutch

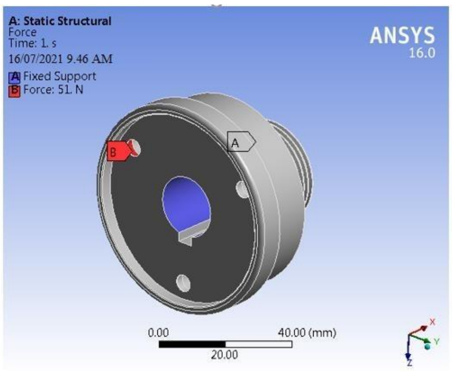

Figure 4.5: Clutch

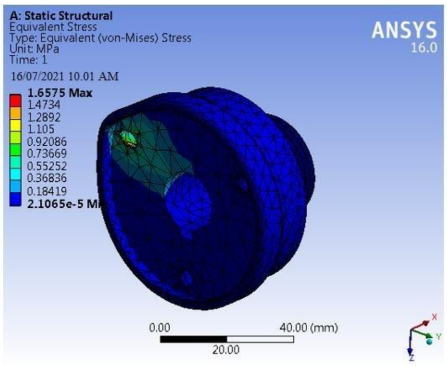

Figure 4.6: Equivalnt Stress in cluctch

Chapter 5

3D PRINTING TECHNILOGY

5.1 Introduction

Fused deposition modelling (FDM) is an additive manufacturing technology commonly used for modelling, prototyping, and production applications. It is one of the techniques used for 3D printing works on an ”additive” principle by laying down material in layers; a plastic filament or metal wire is unwound from a coil and supplies material to produce a part. The technology was developed by S. Scott Crump in the late 1980s and was commercialized in 1990. [1] The term fused deposition modelling and its abbreviation to FDM are trademarked by Stratasys Inc. The exactly equivalent term, fused filament fabrication (FFF), was coined by the members of the RepRap project to give a phrase that would be legally unconstrained in its use. It is also sometimes called Plastic Jet Printing (PJP).

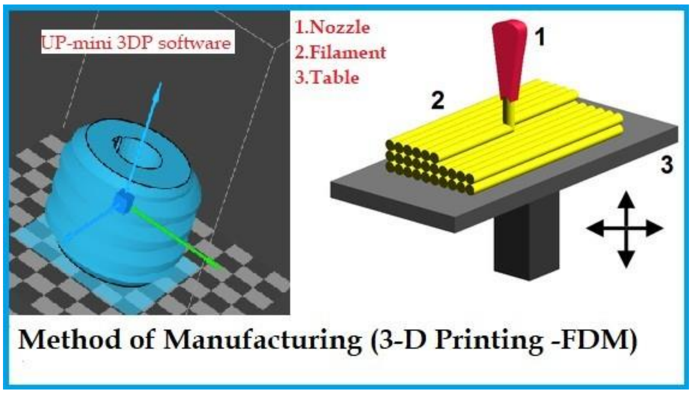

Figure 5.1: Method Of manufacturing

5.2 Working Pricipal of 3D Printing Technilogy

All 3D printing techniques are based on the same principle: a 3D printer takes a digital model (as input) and turns it into a physical three-dimensional object by adding material layer by layer.

It is way different than traditional manufacturing processes such as injection mold- ing and CNC machining that uses various cutting tools to construct the desired structure from a solid block. 3D Printing, however, requires no cutting tools: ob- jects are manufactured directly onto the built platform.

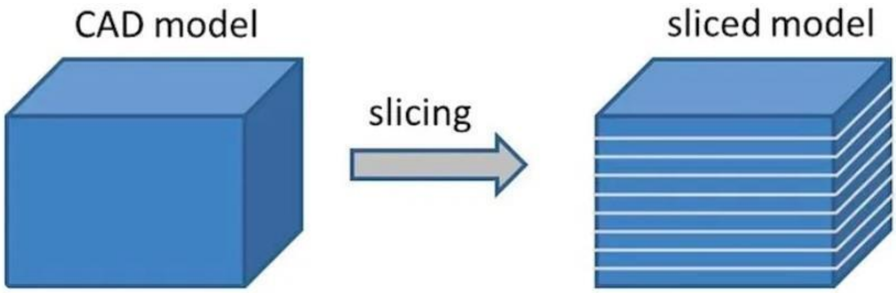

Figure 5.2: Workking Methodlogy For 3D Printing

The process starts with a digital 3D model (a blueprint of the object). The soft- ware (specific to the printer) slices the 3D model into thin, two-dimensional layers. It then converts them into a set of instructions in machine language for the printer to execute

Depending on the type of printer and size of the object, a print takes several hours to complete. The printed object often requires post-processing (like sand-ing, lacquer, paint, or other types of conventional finishing touches) to achieve the optimal surface finish, which takes additional time and manual effort.

5.3 Procedure of 3-dPrinting of Input worm

Interface of UP-mini 3-d printing software

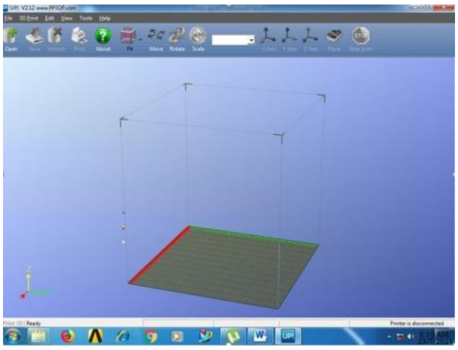

Figure 5.3: Interface of UP-mini 3-d printing software

The Solid model to be printed is exported as a .stl file and it is imported to the interface as shown below

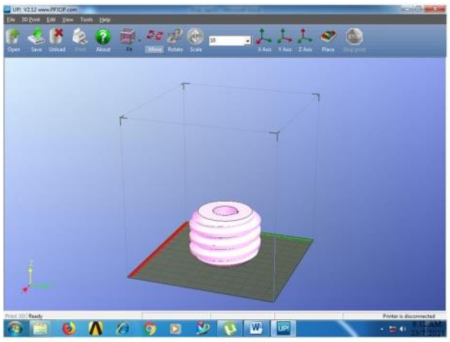

Figure 5.4: Interface of UP-mini 3-d printing software

Printing preferences are set:

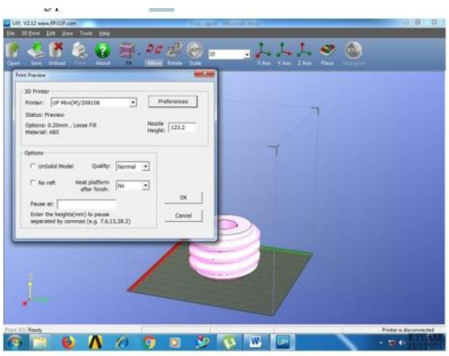

Figure 5.5: Interface of UP-mini 3-d printing software

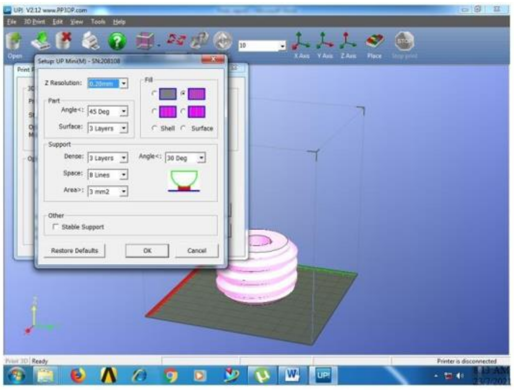

Figure 5.6: Interface of UP-mini 3-d printing software

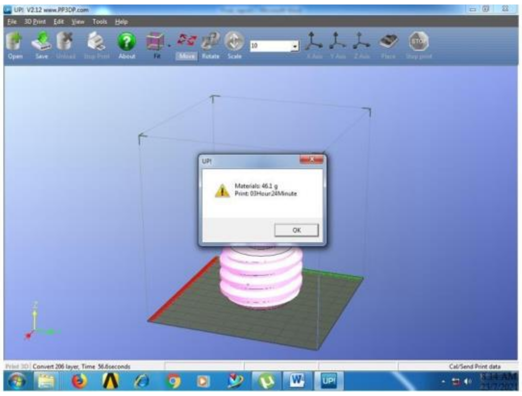

Figure 5.7: Interface of UP-mini 3-d printing software

46.1 Kg material and 3 hours 24 minute required to print input worm

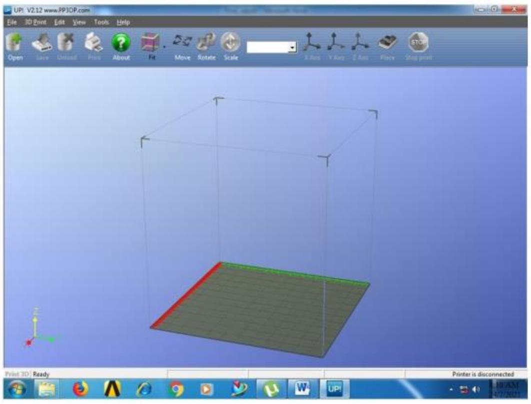

Figure 5.8: Interface of UP-mini 3-d printing software

The Solid model to be printed is exported as a .stl file and it is imported to the interface as shown below

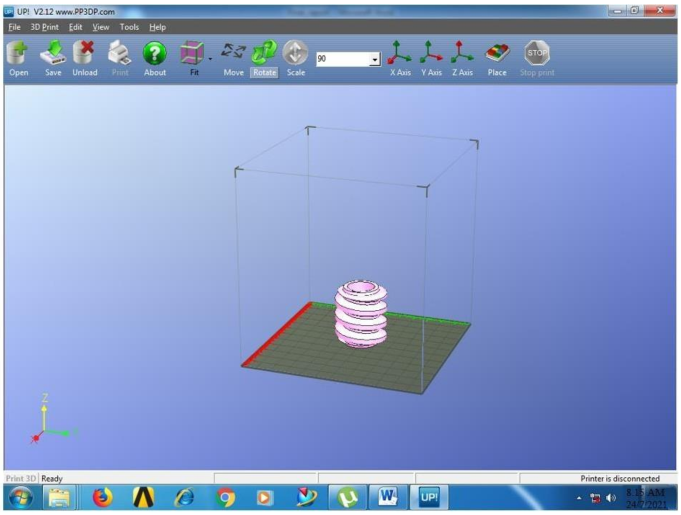

Figure 5.9: Interface of UP-mini 3-d printing software

Printing preferences are set

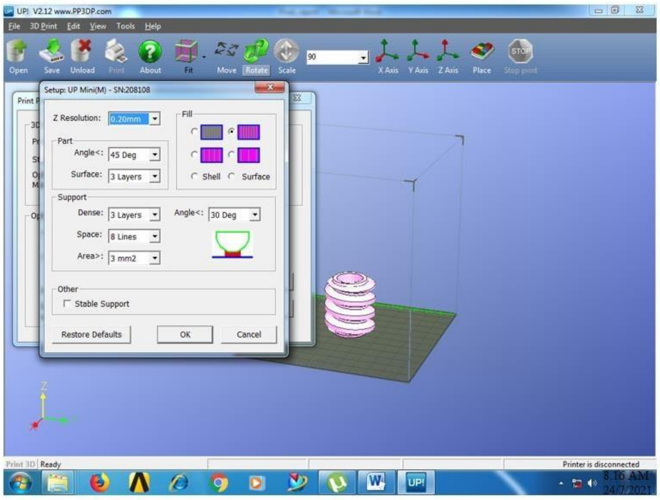

Figure 5.10: Interface of UP-mini 3-d printing software

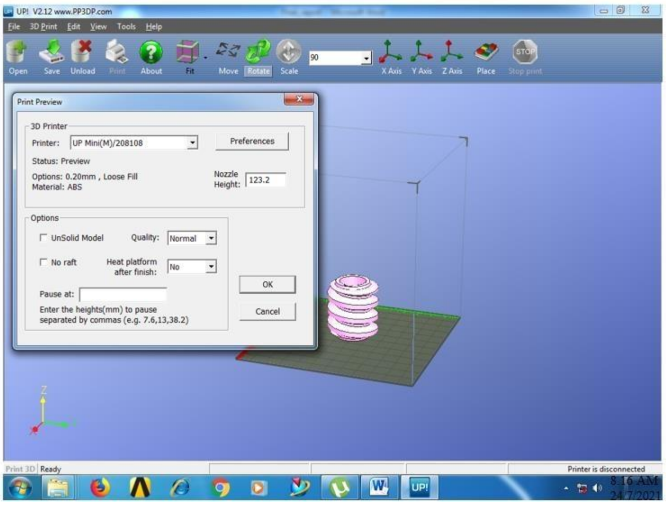

Figure 5.11: Interface of UP-mini 3-d printing software

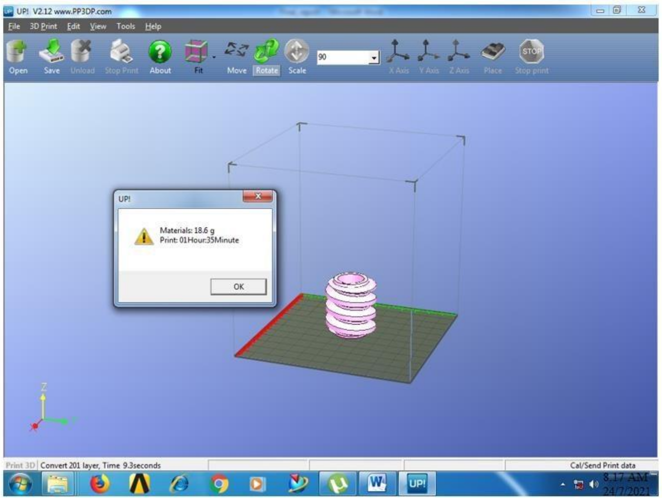

Figure 5.12: Interface of UP-mini 3-d printing software

18.6 Kg material and 1 hours 35minute required to print output worm

Chapter 6

CONSTRUCTION AND WORKING

6.1 Assembly Front View

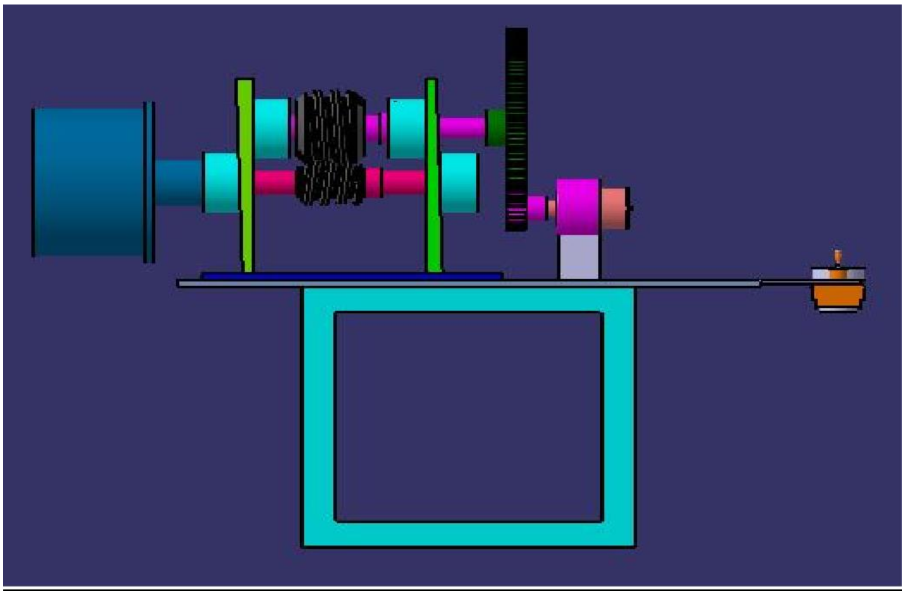

Figure 6.1: Assembly Front View

6.2 Assembly Top View

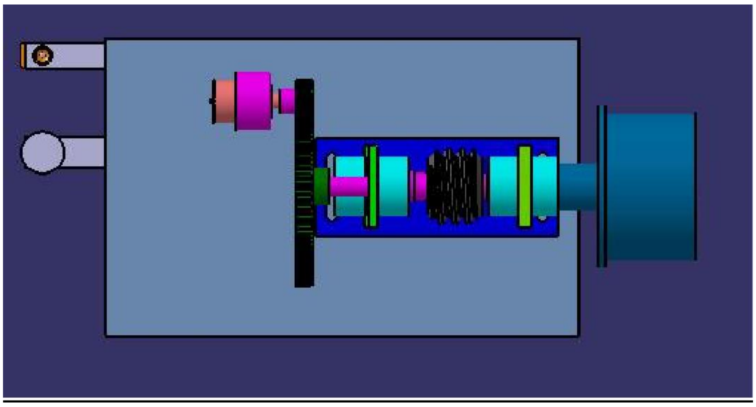

Figure 6.2: Assembly Top View

6.3 Construction

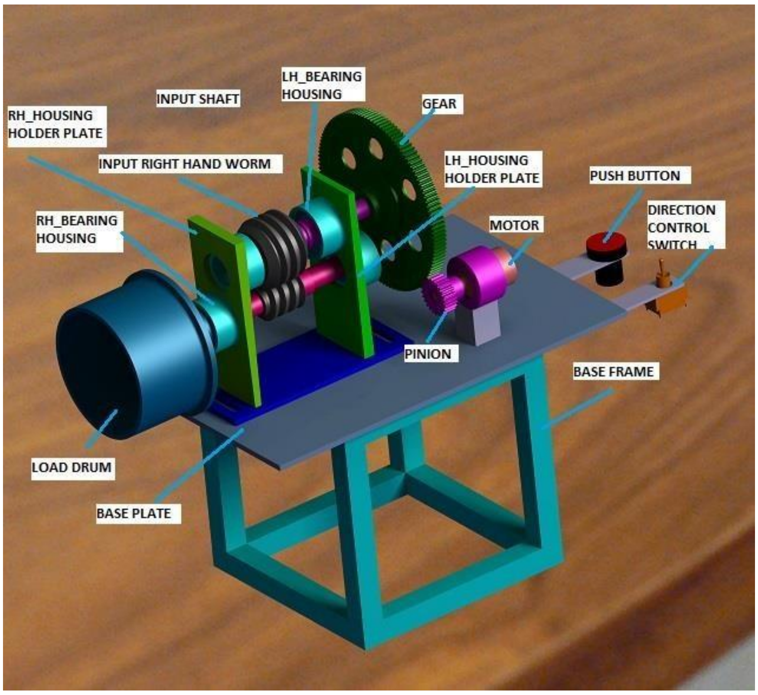

Figure 6.3: Design of Lifter

1. Right hand worm gear:

The right hand worm gear is the input gear of the system. The manufacturing process used is 3-D printing method.

2. Left hand worm gear:

The left-hand worm gear is the output gear of the system. The manufacturing process used is 3-D printing method.

3. Input shaft:

Input shaft is made of high grade alloy steel EN24, it is held in ball bearings at both ends, at one end a spur gear is mounted to receive power from the motor through a pinion.

4. Output shaft:

Output shaft is made of high-grade alloy steel EN24, it is held in ball bearings at both ends, at one end carries the load drum to lift or lower the load

5. Bearings:

Four ball bearings are used in the project. Single row deep groove ball bearing 6004zz is used at each end of shaft. Bearings are held in the bearing housings which are fitted on to the bearing housing holders.

6. LH/RH bearing housings:

These are round members of the system that support the above two shafts inball bearings, the bearings on the output shaft are appropriately locked, they are mounted on housing plates.

7. Housing mounting plates

Housing mounting plates are vertical members that hold the bearing housings, they are machined at angle of 2.5 degrees to provide proper angle of inclination between the input and output shat so that the desired self-locking characteristic is achieved.

8. Baseplate:

The base plate is the base member that houses the entire assembly of system. The housings are bolted to the base plate

9. Load drum

The load drum is fastened to the output shaft of the system. The rope is wound on the drum with its one end fixed to the drum whereas the other free end carries the load

10. Motor

The drive motor is 12 VDC motor coupled to a planetary gear box.

11. BaseFrame:

Base frame comprises of the base plate and base frame structure both made from mild steel.

12. Direction Control Switch:

2-pole 2-way (DP /DT) switch is used to control the direction of motor to rotate motor clockwise or anti-clockwise which will be used to lift or lower the load.

13. Push button

It is used for point to point control i.e.; it is push-to 'On' switch

6.4 Working

The twin worm gear self-locking system input shaft is connected to the drive motor, which provides the input power. The input right hand worm gear drives the output worm gear in the direction such that load connected to the load drum(not shown) on the output shaft is raised, now if the motor power is shut off, the input seizes to rotate , and the load will have a tendency to move down due to gravity , thereby the output the input shaft and thus the motion of the load in the down warddirection is stopped. Thus, the self-locking is effectively attained

Chapter 7

EXPERIMENTAL SETUP

7.1 Experimental setup

Testing Equipment

- Load pan
- Digital Tachometer
- Weights

Expected outcome

- Load capacity of device
- Torque
- Speed
- Power
- Efficiency

Graphs

- Load Vs Speed
- Torque Vs Speed
- Power Vs speed

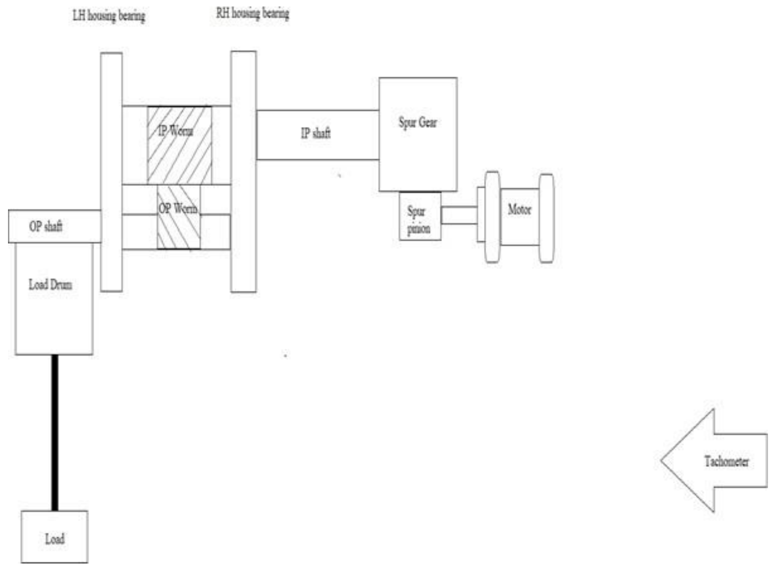

Figure 7.1: Schematic of experimental setup

Chapter 8

TEST AND TRIAL

8.1 Test and Trial on dual

To conduct trial

a) TORQUE Vs SPEED CHARACTERISTICS

b) POWER Vs SPEED CHARACTERISTICS

Input Data

Drive Motor

12 V dc

Power = 5watt

Diameter of Dynobrake drum = 100 mm.

Procedure: - Start motor

1. Let mechanism run & stabilize at certain speed (say 45 rpm)

2. Place the pulley cord on dyno brake pulley and add 0.5 KG weight into, the pan, note down the output speed for this load by means of tachometer.

3. Add another 0.5 Kg weight & take reading.

4. Tabulate the readings in the observation table

5. Plot Speed Vs Load characteristic

SR NO	LOAD	SPEED (rpm)	TORQUE (N.M)	POWER (watt)	EFFICINCY
1	0.5	40	0.286943	1.2021	24.042
2	1	39	0.573885	2.344091	46.88181
3	1.5	37	0.860828	3.335823	66.71647
4	5	34	1.14777	4.087132	81.74265
5	2.5	30	1.434713	4.507868	90.15736
6	3	24	1.721655	4.327552	86.55104
7	3.5	19	2.008598	3.996976	79.93952

Table 8.1: Experimental Values

Chapter 9

RESULT AND DISCUSSION

9.1 Sample Calculations- (At2.5 kg Load)

1) Output speed = 30 rpm
2) Output Torque: -

T = Weight in pan x Radius of Dyno brake Pulley = (2.5x 9.81) x 58.5=1.434713

3) Input Power: - (Pi/p) = 5 WATT
4) Output Power:

$$P=\frac{2\pi nT}{(60)}$$

$$P=\frac{2*\pi*30*1434713}{(60)}$$

=4.507868

5) Efficiency

$$\eta= \frac{OutputPower}{(InputPower)}$$

$$=\frac{4.507868}{(5)}$$

=90.15%

9.2 Result and Discussion

From the above experimentation it is found that the use of twin worm system found to be useful in lifting load with less effort.

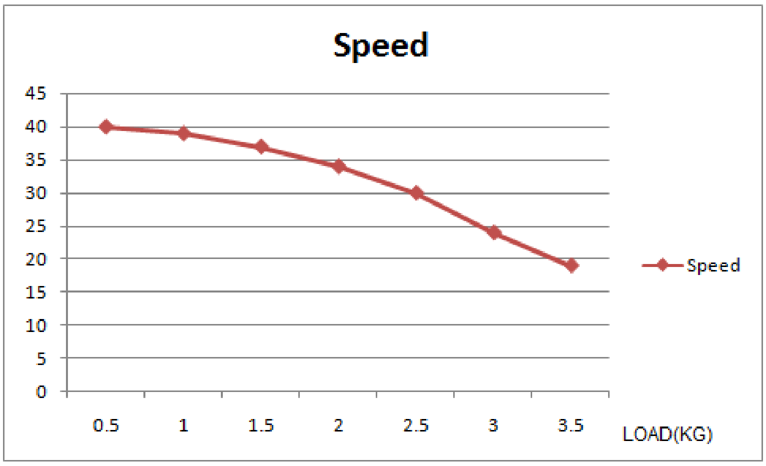

Figure 9.1: Speed Vs Load

In above Speed Vs Load graph, it is found that as load increase speed decreases

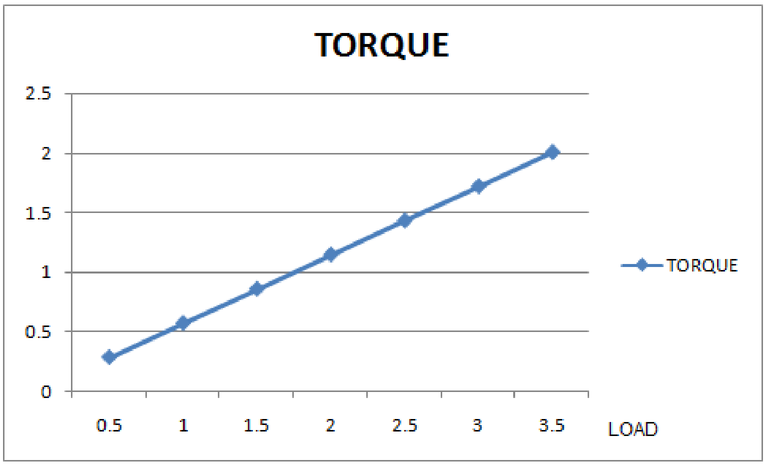

Figure 9.2: Torque Vs Load

In above graph Torque Vs Load, it is found that as the load increases the torque is also increases because of motor is slow down.

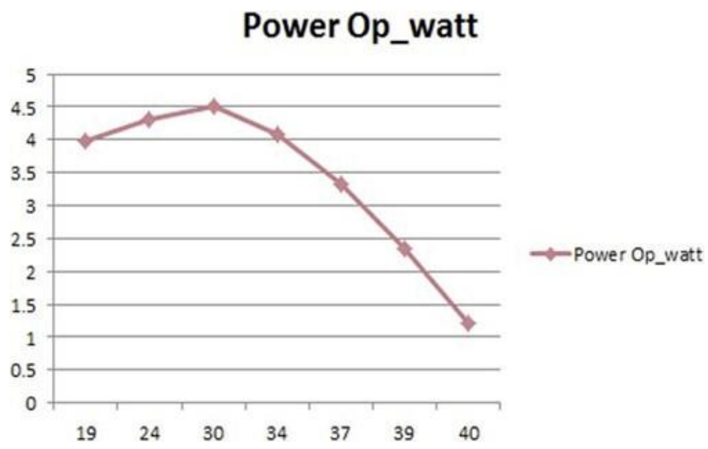

Figure 9.3: Power Output Vs Speed

In the above graph of Power output Vs Speed, it is found that power required is maximum for increasing load.

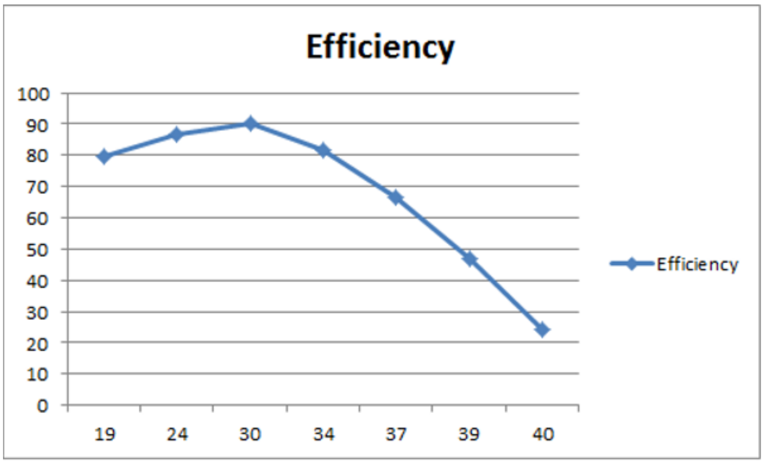

Figure 9.4: Speed Vs Efficiency

In the above graph Speed Vs Efficiency, Maximum efficiency is at speed of 30 RPM and at load 2.5 kg. The efficiency is maximum for increasing load

Features

Input worm and output worm have opposite hand of operation.

The axis of the input and output worm shafts are inclined to each other

Device is designed to produce instantaneous self –locking

Backlash or back driving is almost absent

Load can be raised as well as lowered with equal efficiency

Advantages

Twin worm system offers 90 % efficiency

Lower power consumption

Compact in size

Low weight

Low production cost

Deceleration locking possible.

Very low maintenance cost

No lubrication required for light loads

Applications

Mechanical Cranes

Hoists and lifts

Machine tool feed drives

Propulsion lifts

Power winches

Chapter 10

CONCLUSION

From the above experimentation and result and discussion it is found that the use of twin worm system found to be useful for lifting of load. The conclusion is drawn which are as follow.

In the above work we have modelled a RH worm and LH worm from theo- retical calculation and the 3D drafting is done through Unigraphics Nx 08. The both gears are analysed through ansys. It is found that maximum stress by the- oretical and analyticalmethods are well below the allowable limit. is negligible. Hence the RH Worm gear and LHWorm gears are safe under the rated torque

The torque required is more for as the load increases because of motor slow down and speed is also reduced with increasing load.

The Power Required is maximum for increasing load and the efficiency is maxi- mum for increasing load. We got maximum efficiency is at speed of 30RPM and at load of 2.5 kg is 90.17%

A simple, compact, high efficiency, low cost device is developed, also a new technology of 3-d printing which give complex shapes at low cost learnt through the project. The project gave the industry with a new device to solve backlash problems in many machines in many applications

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LIST OF PUBLICATIONS ON PRESENT WORK

[1] S.K. Gorave 1, Dr. S.S. Gawade 2, “Design and Analysis of Dual Worm Self Locking Lifter” , Recent Advances in Engineering and Technology (RIT-RAIET), pp. 1-4, Aug 2021.

[2] S.K. Gorave 1, Dr. S.S. Gawade 2, “Design and Analysis of Dual Worm Self Locking Lifter” , International Journal of Applied Sciences and Technology, 2021 Vol.6, No.3, pp. 1-4.

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Certificate

This Certificate is awarded to Mr./Ms. S.K. Gorave

of Rajarambapu Institute of Technology, Rajaramnagar presenting paper entitled Design and Analysis of Dual Worm Self Locking Lifter in the Track Mechanical - Design at National Conference on "Recent Advances in Engineering and Technology" (RIT - RAIET) held on 6th August 2021.

Dr. M. S. Kumbhar	Dr. S. S. Gawade	Dr. L. M. Jugulkar	Dr. S. K. Patil	Dr. Mrs. S. S. Kulkarni
Organizing Secretary	*Co-Convener*	*Convener*	*Dean, Academics*	*Director*

Design and Analysis of High Efficiency Self Locking Lifter

Authors:

S.K.Gorave, Dr. S.S. Gawade

Abstract

When an external load applies a dynamic or static torque to the output worm gear external load applies a dynamic or static torque to the output worm gear shaft, and this torque does not result in any rotation of the input worm external load applies a dynamic or static torque to the output worm gear shaft, and this torque does not result in any rotation of the input worm ***Keywords:*** Self-locking, Worm Gear, Lifting

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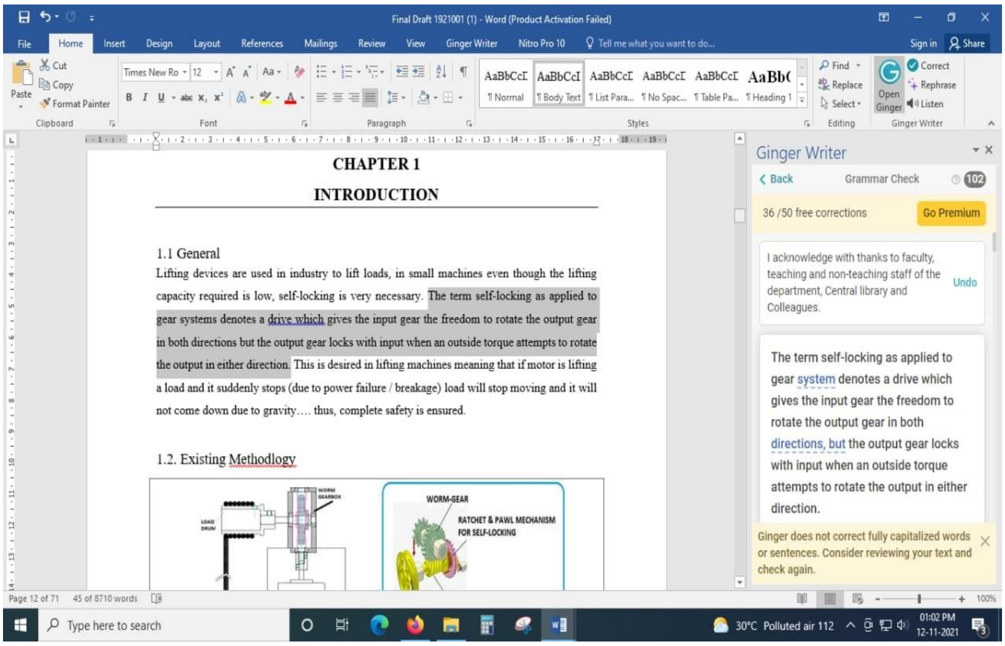

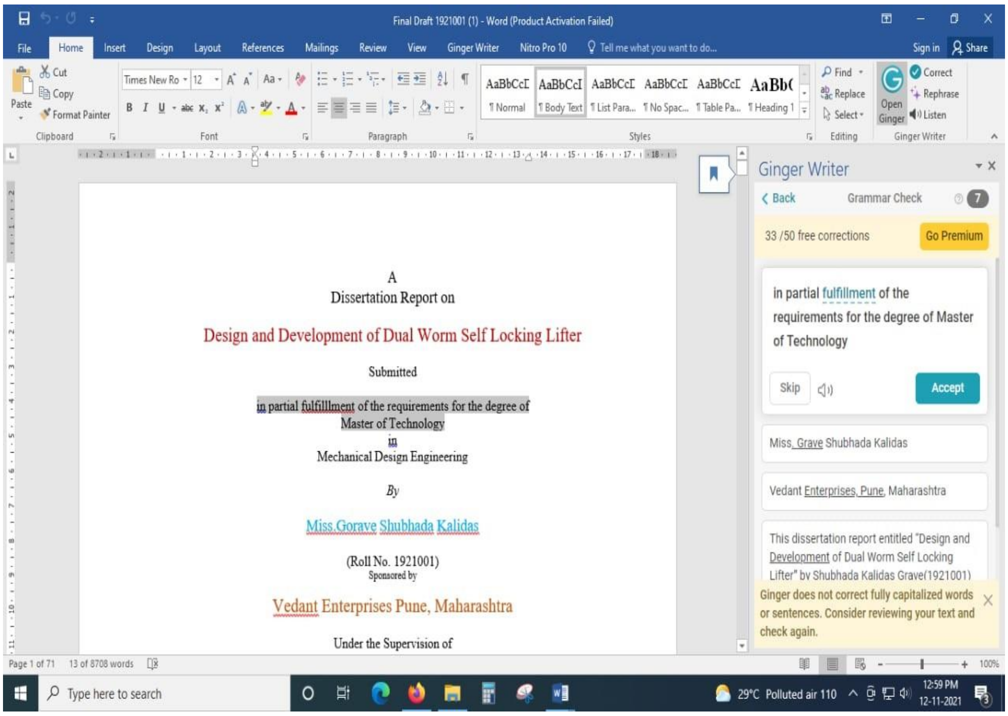

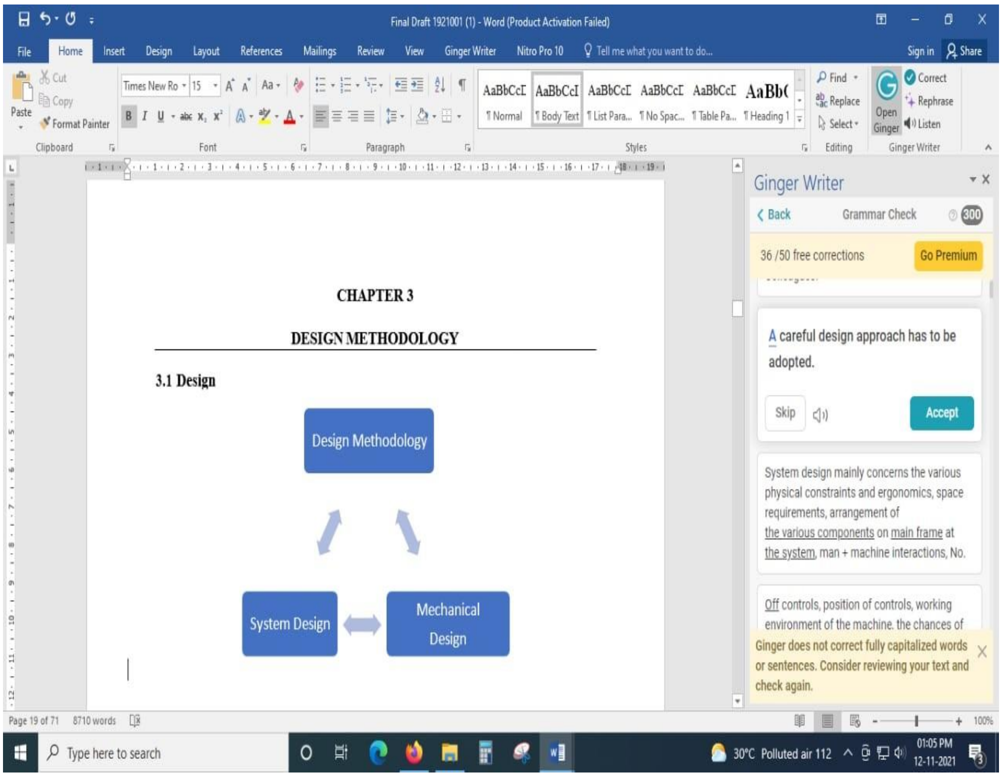
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DESIGN METHODOLOGY
3.1 Design
Design Methodology
System Design
Mechanical Design
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A careful design approach has to be adopted.
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Off controls, position of controls, working environment of the machine. the chances of
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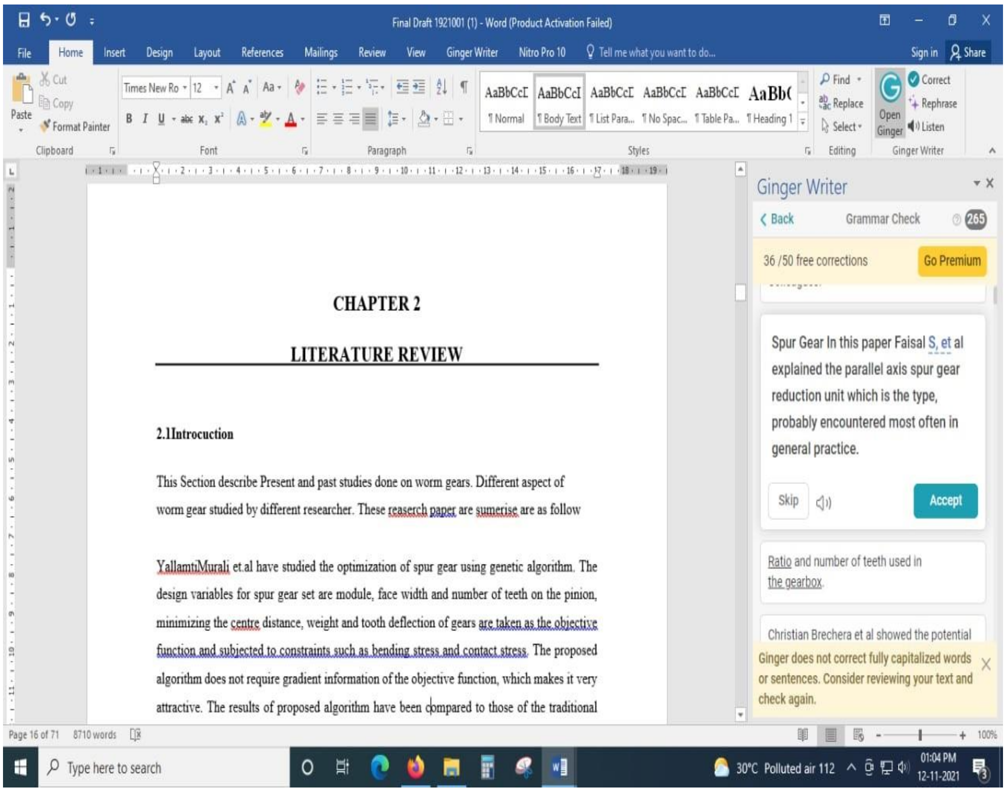
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CHAPTER 2
LITERATURE REVIEW
2.1Introcuction
This Section describe Present and past studies done on worm gears. Different aspect of worm gear studied by different researcher. These reaserch paper are sumerise are as follow
YallamtiMurali et.al have studied the optimization of spur gear using genetic algorithm. The design variables for spur gear set are module, face width and number of teeth on the pinion, minimizing the centre distance, weight and tooth deflection of gears are taken as the objective function and subjected to constraints such as bending stress and contact stress. The proposed algorithm does not require gradient information of the objective function, which makes it very attractive. The results of proposed algorithm have been compared to those of the traditional
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Spur Gear In this paper Faisal S, et al explained the parallel axis spur gear reduction unit which is the type, probably encountered most often in general practice.
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Ratio and number of teeth used in the gearbox.
Christian Brechera et al showed the potential
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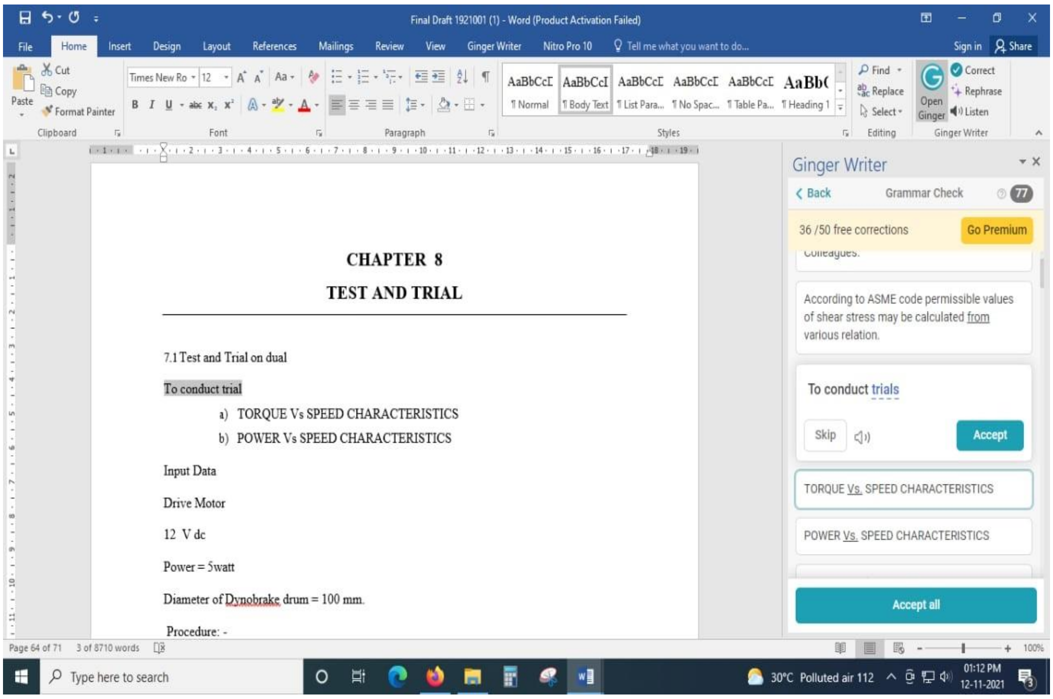
CHAPTER 8
TEST AND TRIAL
7.1 Test and Trial on dual
To conduct trial
a) TORQUE Vs SPEED CHARACTERISTICS
b) POWER Vs SPEED CHARACTERISTICS
Input Data
Drive Motor
12 V dc
Power = 5watt
Diameter of Dynobrake drum = 100 mm.
Procedure: -
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According to ASME code permissible values of shear stress may be calculated from various relation.
To conduct trials
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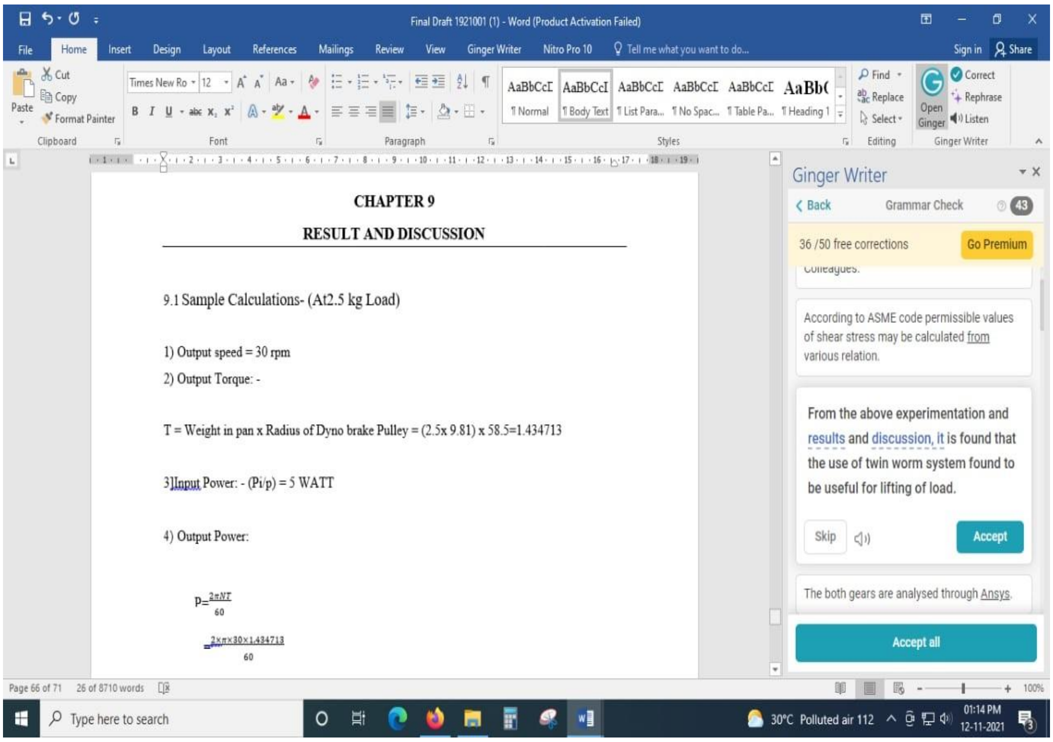
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2) Output Torque: -
T = Weight in pan x Radius of Dyno brake Pulley = (2.5x 9.81) x 58.5=1.434713
3]Input Power: - (Pi/p) = 5 WATT
4) Output Power:
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From the above experimentation and results and discussion, it is found that the use of twin worm system found to be useful for lifting of load.
The both gears are analysed through Ansys.
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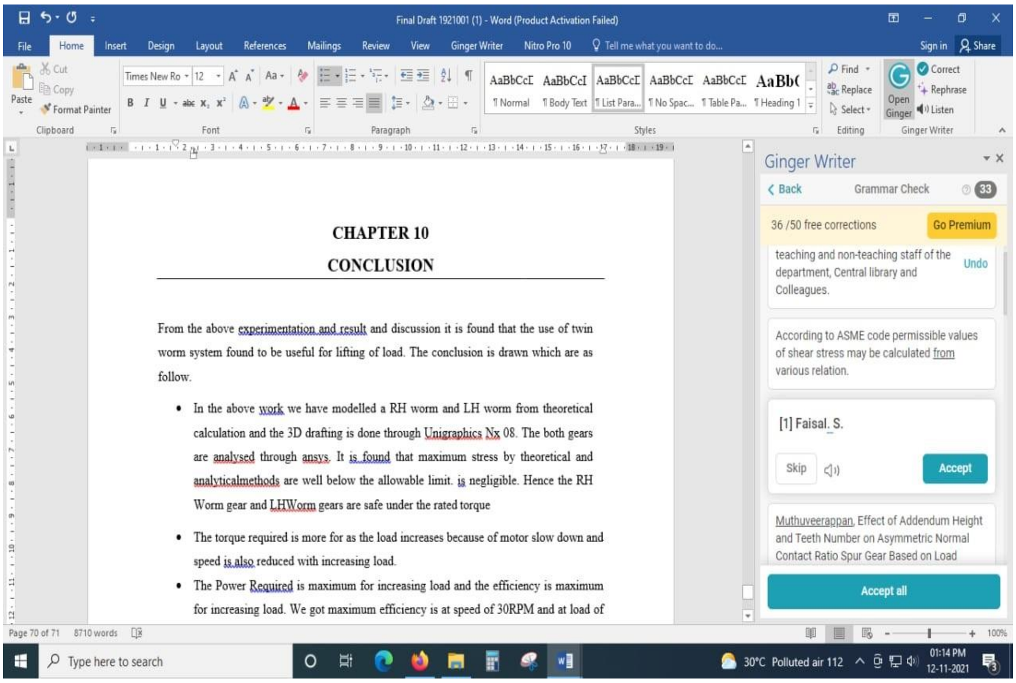
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The Power Required is maximum for increasing load and the efficiency is maximum for increasing load. We got maximum efficiency is at speed of 30RPM and at load of
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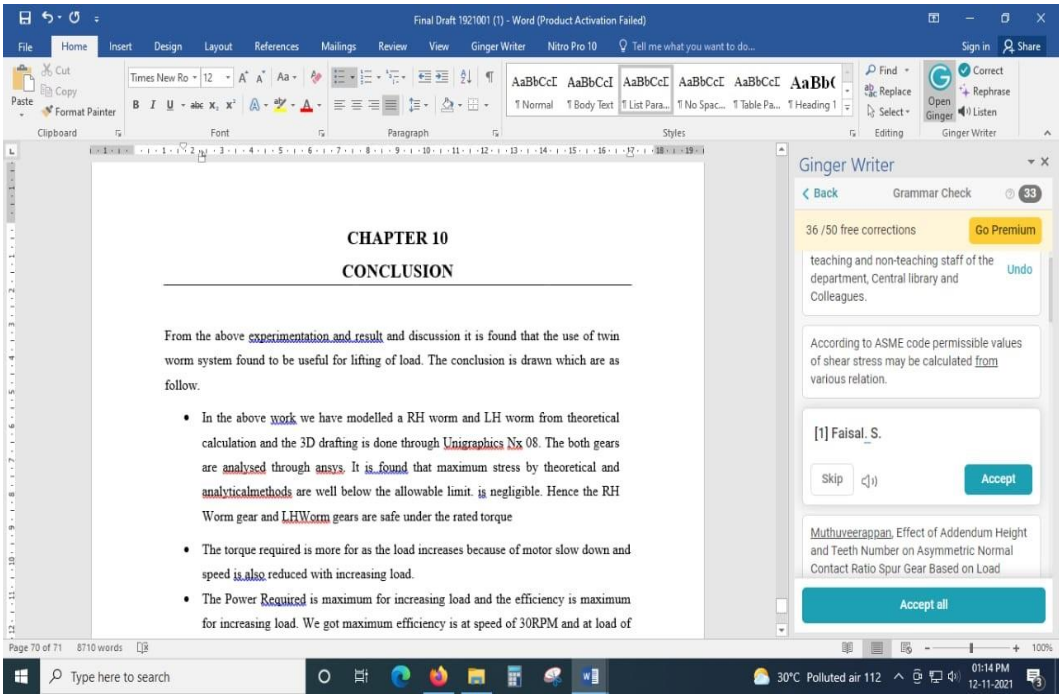
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SYNOPSIS OF M. TECH DISSERTATION

1.	**Name of college**	:	Rajarambapu Institute of Technology, Rajaramnagar
2.	**Name of course**	:	M. Tech (Mechanical)-Design Engineering.
3.	**Name of student**	:	Shubhada Kalidas Gorave (PRN-1921001)
4.	**Date of registration**	:	October 2020
5.	**Name of the Guide**	:	Prof.Dr.S.S.Gawade
6.	**Proposed Title**	:	Design and Development of Dual Worm Self-locking Lifter
7.	**Sponsored by**	:	Vedant Enterprises,Pune,Maharashtra.
8	**Company supervisor**		Mr. Santosh Gaikwad
9.	**Synopsis of proposed work**		

9.1) Relevance:

A self-locking worm gear is a type of worm gear that does not allow the interchangeability of the input and output gears. The term self-locking as applied to gear systems denotes a drive which gives the input gear the freedom to rotate the output gear in both directions. Lifting devices are used in industry to lift loads in small machines even though the lifting capacity required is low self-locking is very necessary. A simple, compact, highly efficient, low cost device will be developed, so also new technology used of 3-D printing will be learnt through the project. The project will provide the industry with a new device to solve backlash problem in many machines in many industrial applications.

9.2) Literature Review:

Faisal et al. [1] have explained the parallel axis spur gear reduction unit which is the type, probably encountered most often in general practice. Optimized design of spur gear indicates that compact design of spur gears involves a complicated algebraic analysis. The author describes the development of such a design methodology and diagnostic tool for determining the modes of failure for spur gear and also the causes of the failures. The ray diagrams are incorporated to make the design more feasible with respect to the transmission ratio and number of teeth used in gearbox. The mode of failure curve in a design space shifts quite appreciably as torque increases. Further the author explains the mode of failures curves is not showing any change in the behavioural pattern when the pressure angle is changed and the large pressure angle gears have smaller value of pinion teeth. This clearly indicates that the lower number of teeth rather than the higher module which reduces the size of gear sets.

Jian Chen et al. [2] have discussed about the feature of involute gear. According to the design principle of involute gear cutter, the index able gear insert with three cutting edges is designed. The milling FEM of index able gear insert is built in Deform 3D software, the FEA milling is analysed with different relief angle and the best relief angle is 6°. Considering cutting force and processing efficiency, the optimal cutting speed is 186.83mm/min and cutting depth is 2.5mm, which the relief angle of index able gear insert is 6°

R. Thirumurugan et al.[3] have calculated maximum contact and fillet stress for normal and high contact ratio gear. The research is based on load contact ratio implementing finite element method and performed for single point load model and multipoint contact model. The effect of various gear parameters such as pressure angle, teeth number, gear ratio, tooth size and addendum on the load sharing ratio and corresponding stress was investigated. Calculation of maximum fillet and contact stress in the case of normal contact ratio gear and high contact ratio gear using the load sharing ratio was performed.

R.P.Sekar et al. [4] have explained the maximum stress on the gear and the pinion at the fillet section is unequal. By removing the maximum fillet stress the load carrying capacity of the gear can be increased. The performance of the gear can be improved by designing the gear with uniform fillet stress which is the replacement of unbalanced fillet stress. Changing the tooth thickness of basic racks from non-standard tooth thickness to standard one,uniform fillet strength can be achieved.

Marimuthu et al. [5] have analysed the effect of the module, gear teeth and addendum height on the load sharing and corresponding stress. In order to calculate stress due to applied load at highest pressure angle, they developed a multi-point contact model for finite element analysis. For the parametric study, they developed ANSYS parametric design language code. It was seen that in the application of load at the critical loading point, an increase in addendum height increases the bending stress. On the contrary, increase in module and number of teeth favourably decreases the bending stress

Marunic[6] has explains the deformation in the middle web of thin rimmed involute spur gear in mesh with solid spur gear is expressed in the form of displacement as nondimensional form is analysed. It is concluded that the comparison of maximum rim and web displacements shows that rim deforms considerably more than the web. This result additionally spurred to the necessity of approach that fully respects the actual gear structure and the contribution of every part that the gear teeth are supported

Rushil H. Sevak et al.[7] have suggested that before making gearbox, verification of its work, performance, efficiency, which effects gearbox performance, is necessary. DOE techniques are used to achieve desired design of gearbox for control of temperature and noise levels. The following parameters such as input speed, back lash, axial play of pinion, output shaft and oil viscosity are very crucial for the gearbox noise and the oil temperature. By optimizing input parameter, the life of the gearbox can be increase

Wenqingming et al. [8] have discovered the shortage of the traditional "modification" theory, and then a research method of new modification principle is brought forward. Accordingly, the principle of curvature modification is established and the effect of the curvature modification theory is also analysed. From research and analysis, the curvature modification principle solves the long-time unsolved problem in the modification of toroidal worm-gearing. The new toroidal worm gearing modified on the principle of curvature modification have higher carrying capacity and transmission efficiency than the traditional toroidal worm-gearing. From the experiment higher bearing ability and transmission efficiency can be achieved .

Research Gap:

After careful study of literature, it is found that though various methods of self-locking are suggested a precise and compact method is needed that will give quick self-locking and higher efficiency of transmission hence there is need to produce such drive. More over 3-D printing is the latest technology available in market which gives very fast but extremely cheap method of manufacturing parts having complex shapes. This method will be tried and tested by means of the project to establish its utility in such applications.

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9.3) Problem statement:

The efficiency of worm gear box depends upon coefficient of friction and lead angle , but lead angle must be between 10 to 30 to get self-locking , so efficiency of worm gear box is very low (below 50%)The second method of Ratchet and pawl mechanism is not reliable or safe and it also takes a lot of space .Hence a compact system with high efficiency is needed for lifting devices.

9.4) Objectives

- To design of twin worm self-locking lifter for design load to achieve self-locking condition.
- To use of application of 3-D printing technology for fast and low-cost manufacturing of input and output worm
- To test and trial to determine Torque-Speed-Power-efficiency of system

9.5) Proposed work

Phase 1: Literature Survey

This phase consists carrying out literature survey by referring reputed journals from reputed publishers like research gate, Science Direct, IEEE. Also, the work related to different measurement principle and methodology will be studied. This will help to gain theoretical background regarding project.

Phase 2: Mechanical Design

The parts mentioned above in the part list will be designed for stress and strain under the given system of forces, and appropriate dimensions will be derived. The standard parts will be selected from the PSG design data handbook

Phase 3: Formulation of Model / Analysis of Model

Production drawings of the parts are prepared using Auto Cad, Unigraphics Nx 8 with appropriate dimensional and geometric tolerances. Raw material sizes for parts are also determined.

Phase 4: Material Procurement & Process Planning

Material is procured as per raw material specification and part quantity. Part process planning is done to decide the process of manufacture and appropriate machine for the same.

Phase 5: Manufacturing

Parts are manufactured as per the part drawings by using 3D printing technology method.

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Phase 6 : Assembly -Test & Trial

Assembly of device is done as per assembly drawing, and test and trial is conducted on device for evaluating performance characteristics like torque, speed, power, efficiency.

9.6 Expected outcome

The Dual Worm Self-Locking Lifter manufactured will increase the efficiency, it also exhibits the phenomenon deceleration locking i.e. if the load slows down because of the overloading system will automatically lock.

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9.7) Plan of proposed work:

Sr No	Activity/month	October	November	December	January	February	March	April	May
1	Literature Survey								
2	Mechanical Design								
3	Formulation of Model / Analysis of Model								
4	Material Procurement & Process Planning								
5	Manufacturing								
6	Assembly -Test & Trial								
7	Report Writing								

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9.8) Expected date of completion of work: -May 2021.

Date: 13/01/2021

Place: RIT, Islampur.

Gorave K
Student

(Shubhada Kalidas Gorave)

Dept. of Mechanical Engg.
RIT
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M. Tech
Mechanical Design

Guide	HOP	HOD
(Dr. S.S.Gawade)	**(Dr. M. B. Mandale)**	**(Dr.S. K. Patil)**

RIT, ISLAMPUR

CURRICULAR VITAE

Name of Student:

Shubhada Kalidas Gorave.

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Date of Birth: 27/03/1997

Ms. Shubhada Kalidas Gorave had completed BE in Mechanical Engineering from Sinhgad Institute of Technology and Science, Pune in 2019. Her bachelor's thesis was related to the Design and Analysis of Wave Spring for replacing compression spring. She has completed M. Tech in Mechanical Design Engineering from Rajarambapu Institute of Technology, Islampur in 2021. Her dissertation topic was "Design and Development of Dual Worm Self Locking Lifter", sponsored by Vedant Enterprises, Pune. Her research interest is in the field of Machine Design, Heat and Mass Transfer and Simulation. She has submitted one research paper in IJEAST Journal and attended one national level conference of Recent Advances in Engineering and Technology.

SHUBHADA KALIDAS GORAVE

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CAREER OBJECTIVE

To secure a challenging position in a progressive organization to expand my knowledge and skills, while making a significant contribution to the success of organization.

EDUCATIONAL QUALIFICATIONS

Qualification	College/ Institute	Board	% / CGPA
MTech (Design Engineering) 2021	Rajarambapu Institute of Technology, Sangali	Shivaji University	7.59
B.E. (Mechanical Engineering) 2019	Sinhgad Institute of Technology & Science, Narhe, Pune	Savitribai Phule Pune University	68.13
H.S.C 2014	Chate Junior College, Pune	Maharashtra State Board (HSC)	59.23
S.S.C 2012	S.N.R. Highschool, Indapur	Maharashtra State Board (SSC)	88

TECHNICAL SKILLS

- Designing software's like CREO Parametric, CATIA.
- Simulation Software like ANSYS software
- Effective use of MS-Office

ACADEMIC PROJECTS

MTECH PROJECT:

DESIGN AND DEVELOPMENT OF DUAL WORM SELF LOCKING LIFTER

PROJECT DESCRIPTION:

- To design of twin worm system for design load to achieve self-locking condition.
- Ansys is used as a Simulation software.
- It can give maximum efficiency 89.19% at 30r.p.m
- A simple, compact, high efficiency, low cost device is developed, so also a new technology of 3-d printing is learnt through the project.

BE PROJECT:

DESIGN AND ANALYSIS OF WAVE SPRING

- To be work more precisely than that of helical coil spring.
- Ansys is used as simulation software.
- These springs can easily replace various types of Damping equipment's
- Wave spring can reduce 50% height using the same force and deflections.

CO CURRICULAR & EXTRA CURRICULAR ACTIVITIES

- SOFT TO HARD WORKSHOP | SINHGAD KARANDAK TECHTONIC 2K15 TECHNICAL EVENT OF MAKING A COMPONENT ON LATHE MACHINE.
- RESEARCH PAPER PUBLISHED: DESIGN AND ANALYSIS OF WAVE SPRING AT (INTERNATIONAL JOURNAL OF EMERGING TECHNOLOGIES AND INNOVATIVE RESEARCH)
- COMPLETED CERTIFIED COURSE OF CATIA AT CAD CENTER.
- CONFERENCE PAPER PUBLISHED: DESIGN AND ANALYSIS OF DUAL WORM SELF LOCKING LIFTER (NATIONAL CONFERENCE RIT RAIET 2021)
- CETIFIED COURSE AT NPTEL PLATFORM; FUNCTIONAL AND CONCEPUTIONAL DESIGN

STRENGTHS

- Quick learner
- Hard working.
- Achieve the targets and superb team work.
- Event management ability

PERSONAL PROFILE

Nationality : Indian
Gender : Female
DOB : 27/03/1997
Languages : Marathi, Hindi and English
Hobbies : Reading books, Writing a poem.

DECLARATION

I hereby declare that all the information and facts given above are true to the best of my knowledge and belief.

Date: Shubhada kaliads Gorave