Design, Development And Performance Evaluation of Self Locking Dual Worm Gears Manufactured By FDM Process

Swapnil Ramdas Jagdale[1], Prof. Ravindra Lahane[2]
[1, 2] Dept of Mechanical Engineering
[1, 2] D.Y.P.C.O.E.,Akurdi, Pune, Maharashtra, India.

Abstract- *In many applications of special purpose machinery, a self-locking drive is desired. The self-locking worm gear does not allow interchangeability between the input and output shaft and thus find many industrial applications. A large gear reduction ratio can be achieved by using the self-locking worm gear without increasing the size of the gear box. conventionally 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, self-locking devices using the worm and worm gear drives are still needed to be replaced by another self-locking arrangement which will be more efficient and compact as compared to the earlier conventional system.*

Manufacturing of the worms will be done using FDM process using an UP mini 3-D printer using ABS polymer. Purpose of using FDM process is to facilitate quick and cost effective manufacturing for low torque like packaging machines. FDM manufactured dual worm system apply two worm with opposite hand of drive i.e., one is left hand and other is right hand. Both worms are positioned at a skew angle to each other. Thus, the input worm can drive the output shaft i.e. the load drum whereas the load drum cannot reverse direction of rotation of the input worm thus giving the desirable self-locking effect. 3-D modeling of set-up will be done using Unigraphix Nx-8.0 and CAE of critical component and meshing using ANSYS Work-bench-16. The experimental validation part of the lifting force developed by the dual worm system will be validated using test-rig. Performance characteristics of torque, Power and efficiency vs Speed will be plotted.

Keywords- Worm, Self-Locking, Efficiency.

I. INTRODUCTION

The term self-locking as applied to gear system denotes a drive which gives the input gear the freedom to rotate the output gear in either directions but the output gear locks with input when an outside torque attempts to rotate the output in either direction. This characteristic is often sought after by designers who want to be sure that the loads on the output side of the system cannot affect the position of the gears. Worm gears are one of the few gear systems that can be made self-locking, but at the expense of efficiency, they seldom exceed 45% efficiency, when made self-locking. Thus, we can say that if worm gear drives when used for lifting applications with self-locking as the primary objective for safety considerations the drives are extremely in-efficient. Hence there is a need of special purpose drive that will provide better transmission efficiency in self-locking condition to reduce power consumed by the device.

Plastic gears are serious alternatives to traditional metal gears in a wide variety of applications. The use of plastic gears has expanded from low-power, precision motion transmission into more demanding power transmission applications. As designers push the limits of acceptable plastic gear applications, more is learned about the behavior of plastics in gearing and how to take advantage of their unique characteristics. While metal gears handle loads better than comparably sized plastic gears and have better dimensional properties as temperature and humidity changes, plastic offers many cost, design, processing, and performance advantages over metal.

3D printing is a method of Additive Manufacturing that adds material to an object layer by layer to create the final product. It is also known as rapid prototyping, layered manufacturing, or additive manufacturing. 3D Printing refers to a relatively new class of manufacturing methods which quickly produce physical prototypes from 3D CAD data.

Fused deposition modeling, (FDM) is one of the techniques used for 3DPrinting. It was developed by Scott Crump in the late 1980s and was commercialized in 1990 by Stratasys. FDM is commonly used for modeling, prototyping, and production applications. FDM 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. FDM uses the Thermoplastics ABS, Polyphenylsulfone (PPSF), Polycarbonate (PC), Ultem 9085. It is one of the typical Rapid Prototyping processes that provide functional prototypes of ABS plastic. FDM produces the highest-quality parts in Acrylonitrile Butadiene Styrene (ABS) which is a common end-use engineering material that allows you to perform functional tests on sample parts. FDM process is a filament based system which feeds the material into the heated extrusion head and extruding molten plastic that hardens layer by layer to form a solid part. ABS parts are sufficiently resistant to heat, chemicals, and moisture that allows them to be used for limited to extensive functional testing, depending upon the application.

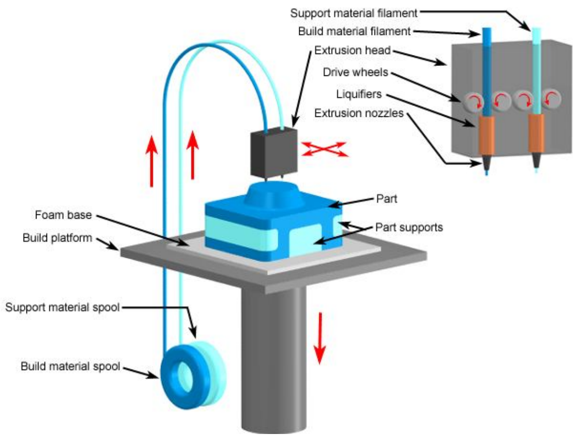

Fig.1: Fused Deposition Modelling Process

FDM Process: FDM begins with a software process which processes an STL file (Sterio Lithiography File), mathematically slicing and orienting the model for the build process. The part is produced by extruding small flattened strings of molten material to form layers as the material hardens immediately after extrusion from the nozzle. A plastic filament or metal wire is unwound from a coil and supplies material to an extrusion nozzle which can turn the flow on and off. The nozzle is heated to melt the material. The thermoplastics are heated past their glass transition temperature and are then deposited by an extrusion head. The nozzle can be moved in both horizontal and vertical directions by a numerically controlled mechanism. The nozzle follows a tool-path controlled by a Computer Aided Manufacturing (CAM) software package, and the part is built from the bottom up, one layer at a time.

II. METHODOLOGY

1. System design of opposite hand self-locking gear as for the component selection, geometry and profile selection, charge system selection, mounting & orientation.
2. Mechanical design of components under given system of forces to determine functional dimensions of the components to be used using various formulae and empirical relations.
3. 3-D modeling of set-up using Unigraphix.
4. CAE of critical component and meshing using ANSYS Workbench-16.
5. Manufacturing, assembly of the device and test-rig for experimental analysis and validation.
6. Testing and trial to derive performance characteristic of equipment under various load conditions.

III. DESIGN PROCEDURE

1. Drive Motor

The drive motor is 12V DC motor coupled to a Spur gear pair.

Specifications of motor are as follows:

1. Power = 5 watt
2. Speed = 60 rpm
3. Torque = 0.833 N-m

2. Design of Spur Gear Pair

Spur gear pair is designed for primary power transmission between motor and dual worm system.

Maximum torque = 2 N-m (considering 100 % overload)
No of teeth on gear =120
No of teeth o pinion = 24
Module = 1.25 mm
Radius of gear by geometry = 120 *1.25/2 = 75mm
Maximum load = T/r = 2 x 10^3/75 =26.67 N
b = 10 m
Material of spur gear and pinion = Nylon 6
S_{ult} pinion = S_{ult} gear = 60N/mm^2
Service factor (Cs) = 1.5
Pt = (W x Cs) = 40 N
(as Cv =1 due to low speed of operation)

Peff = 40N …(1)

Lewis Strength equation is
Sb = m x b x fs xY
Where;
Y= 0.484 – (2.86/Z)
Yp = 0.484 – 2.86/24 = 0.365
$(S_{ult})p$ = 60 …(As pinion is weaker)
$S_b = (S_{ult})_p$. y. b. m
= (60) (0.365) x10m^2
= 219 m^2 …(2)
Equation (1) & (2)
147.4 m^2 = 40
$\Rightarrow$m=0.521 mm

Selecting standard module = 1.25 mm,
For ease of construction, as we go for single stage gear box, making size compact and achieving maximum strength and proper mesh.

3. Design of Input Shaft

Material selection:

Table1: Material Selection For Shaft

Designation	Ultimate Tensile Strength, N/mm^2	Yield Strength, N/mm^2
EN24	800	680

: Ref-PSG (1.10 & 1.12) + (1.17)
According to ASME code permissible values of shear stress may be calculated form various relation,

$fs_{max} = 0.18\ f_{ult}$
$= 0.18 \times 800$
$= 144 N/mm^2$

OR
$fs_{max} = 0.3\ f_{yt}$
$= 0.3 \times 680$
$= 204$ N/mm

Considering minimum of the above values;
$\Rightarrow fs_{max} = 144 N/mm^2$

Shaft is provided with notch for locking; this will reduce its strength. Hence reducing above value of allowable stress by 25%
$\Rightarrow fs_{max} = 108 N/mm^2$

This is the allowable value of shear stress that can be induce in the shaft material for safe operation.
$T = 2 \times 10^3$ N-mm
Assuming 25% overload.

Torsional Shear Failure of Shaft:

Assuming Minimum Section Diameter on Shaft = 16mm (Gear Bore Is Ready Made Of 16 Mm Diameter)
$\Rightarrow d = 16$ mm
$Td = \Pi/16 \times fs_{act} \times d^3$
$\Rightarrow fs_{act} = \frac{16 \times Td}{\Pi \times d^3} = \frac{16 \times 2 \times 10^3}{\Pi \times 16^3}$
$\Rightarrow fs_{act} = 2.5/mm^2$
As, $fs_{act} < fs_{all}$
I/P shaft is safe under torsional load.

Ball Bearing selection

Table-2: Ball Bearing Specification for Series 60

C	Brg Basic Design No(SKF)	d	D1	D	D2	B	Basic capacity	
							C kgf	Co Kgf
20B C003	6203	17	21	40	36	12	440	750

$P = X.Fr + Y.fa$
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 (F_a)
$F_a = 0$
$$L = \frac{60\ n\ Lh}{10^6} = \frac{60 \times 1000 \times 60}{10^6}$$
= 3.6 mrev
P = Fr = 69
For single row ball bearings, p=3
$\therefore L = (C/P)^p$
$\therefore C = 106.6$
As, it's less than the rated dynamic capacity of bearing
Thus, Bearing is safe.

4. Design of Output Shaft

Material selection:

Table-3: Material Selection for Shaft

Designation	Ultimate Tensile Strength, N/mm^2	Yield Strength, N/mm^2
EN24	800	680

: Ref-SG (1.10 & 1.12) + (1.17)
According to ASME code permissible values of shear stress may be calculated form various relation,
$fs_{max} = 0.18\ f_{ult}$
$= 0.18 \times 800$
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OR
$fs_{max} = 0.3\ f_{yt}$
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$= 204$ N/mm
Considering minimum of the above values;
$\Rightarrow fs_{max} = 144 N/mm^2$

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

$\Rightarrow \quad fs_{max} = 108\ N/mm^2$

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

$T= 2 \times 10^3$ N-mm

Assuming 25% overload.

Torsional Shear Failure of Shaft:

Assuming Minimum Section Diameter on Shaft = 16mm

$\Rightarrow d = 16$ mm

$Td = \Pi/16 \times fs_{act} \times d^3$

$$fs_{act} = \frac{16 \times Td}{\Pi \times d^3} = \frac{16 \times 1.12 \times 10^3}{\Pi \times 16^3}$$

$\Rightarrow fs_{act} = 1.39\ N/mm^2$

As, $fs_{act} < fs_{all}$

O/P shaft is safe under torsional load.

5. Design of Worm

Design Check

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

a) dc = core / inner diameter (mm) = 22.36mm
b) dm = mean diameter (mm) = 27.32mm
c) $M_t = W \times (dm/2) \tan (\downarrow + \propto)$
d) Where, W= Axial load
e) Ø = friction angle

$\propto$ = Helix angle

Helix angle,

$$\tan \propto = \frac{L}{\pi \times Dm} = \frac{10}{\pi \times 27.36}$$

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

$\Rightarrow \propto = 6.63$

a. *Friction angle:*

Table-4: Coefficient of Friction

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

: Ref-R. S. Khurmi (Table17.5)

$\mu = \tan Ø$

$0.18 = \tan Ø$

$\Rightarrow Ø = 10.2$

If max load of 10 kg is carried by the drum of 100 mm diameter, then the resultant torque will be,

$T= (10 \times 9.81) \times 50= 4905$N-mm ___(3)

We know, the effort required to raise the load,

$P = W \tan (\phi + \alpha)$

Now, it is necessary to consider the torque required to raise the load. Torque requirement is given as,

Mt= Force x Perpendicular Distance

$Mt= W \times \tan (\phi + \alpha) \times (d_m/2)$ ___(4)

Where, dm = Mean diameter of the screw jack

$M_t = W \times (27.32/2) \times \tan (10.2 + 6.63)$

$M_t = 4.132 \times W$ N-mm ___(5)

Equating (3) & (5)

W = 1187.076 N

This is the load carrying capacity of the worm.

b. Material selection:

Table-5: Material Specification

Designation	Tensile Strength, N/mm^2	Yield Strength, N/mm^2
ABS Polymer	60	42

: Ref-(PSG-1.12)

Direct Tensile or Compressive stress due to an axial load,

$$fc_{act} = \frac{W}{\frac{\pi}{4} X\, d_c^2} = \frac{1187.076}{\frac{\pi}{4} X\, 22.36^2}$$

$\Rightarrow fc_{act} = 3.0245\ N/mm^2$

As $fc_{act} < fc_{all}$, Screw is safe in compression.

c. Torsional shear stress:

$T = M_t = \pi/16 \times fs_{act} \times dc^3$

$2 \times 10^3 = \pi / 16\ fs_{act} \times (22.36)^3$

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

As $fs_{act} < fs_{all}$; the screw is safe in torsion.

d. Stresses due to buckling of screw: -

According to Rankine formula,

$$W_{cr} = \frac{f_c X A}{1+a(\frac{Le}{K})^2}$$

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) decided by end conditions.

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

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

Le = 0.5L (as both ends of screw considered to be fixed)

…(Ref. PSG Design Data Pg. No. 6.8)

$\Rightarrow$ Le = 0.5x 45= 22.5mm

$$W_{cr} = \frac{30 \times (\frac{\pi}{4} \times 22.36^2)}{1+(1/7500)(\frac{22.5}{22.36/4})^2}$$

W_{cr} = 11.74x10^3 N

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

IV. ANALYSIS OF INPUT RH WORM

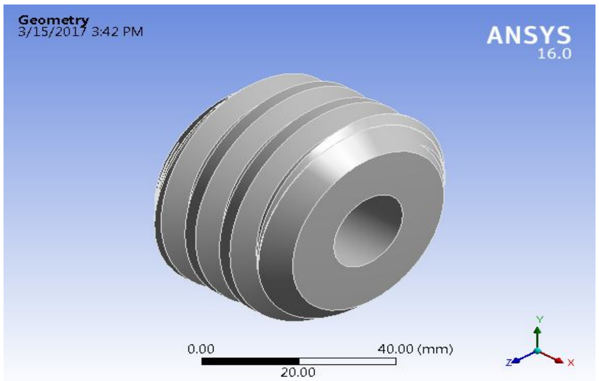

Fig.2: Modeling of RH Worm

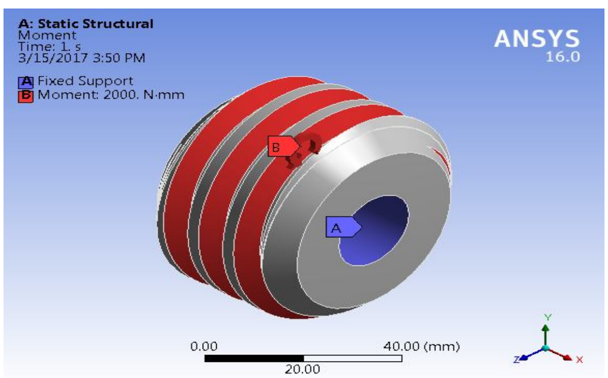

Fig.3: Boundary condition and Application of Torque

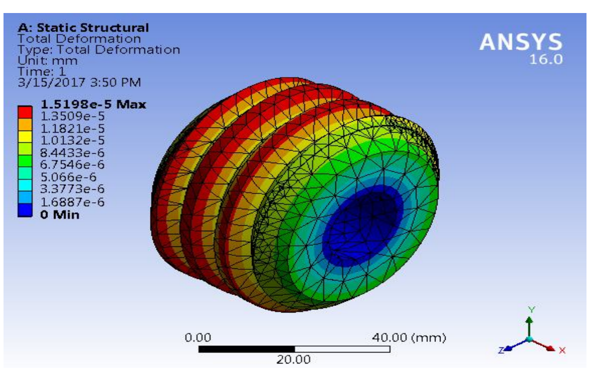

Fig.4: Total Deformation of RH Worm

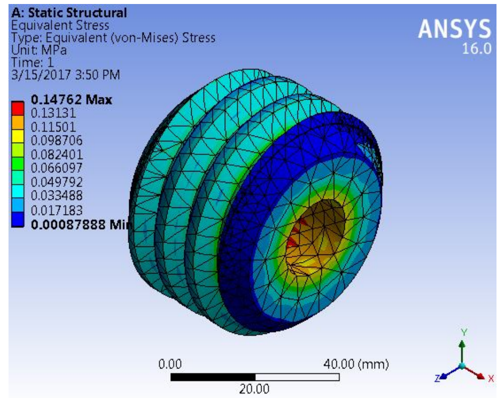

Fig.5: Von-Mises Stress

As the maximum value of stress induced is 0.14MPa which is well below the allowable stress of 30 MPa, the output worm is safe

V. ANALYSIS OF OUTPUT LH WORM

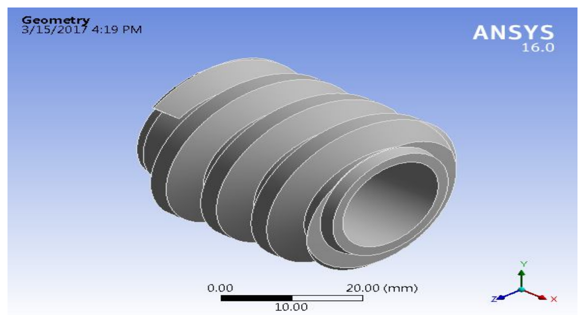

Fig.6: Modeling of LH Worm

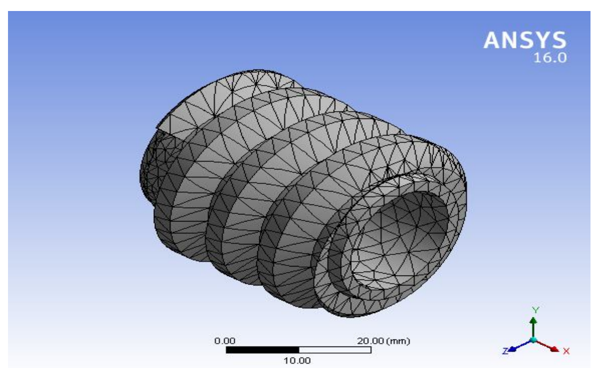

Fig.7: Meshing of LH Worm

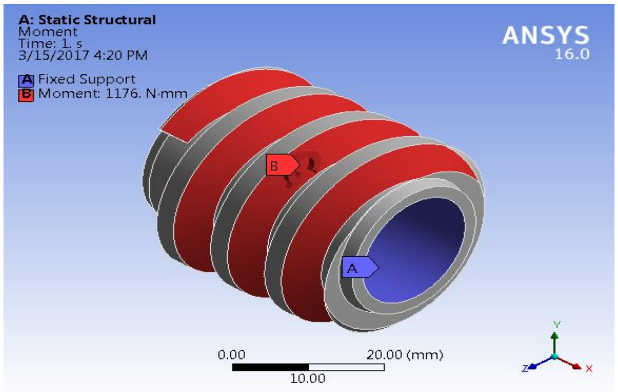

Fig.8: Boundary condition and Application of Torque

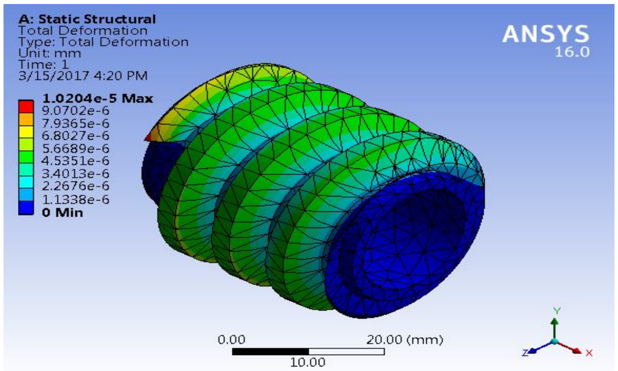

Fig.9: Total Deformation of LH Worm

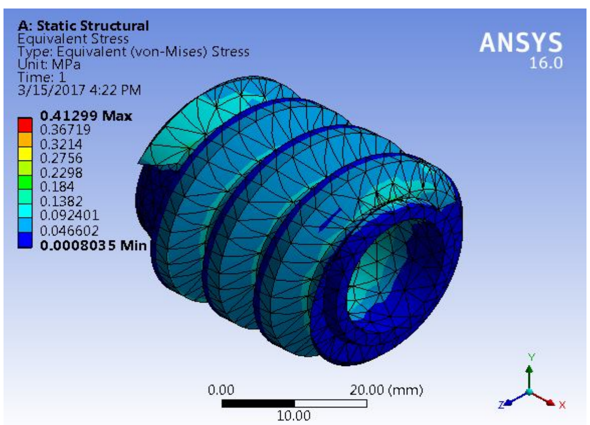

Fig.10: Von-Mises Stress

VI. RESULT OF EXPERIMENTAL SETUP

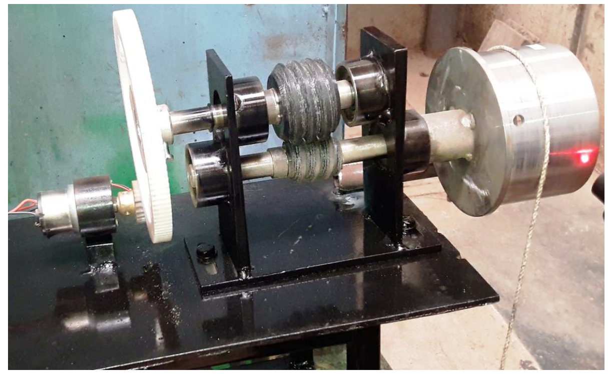

Fig.11: Experimental Setup

Input Data:

Motor power = 5 watt, 12V DC

Speed = 60 rpm

Reduction ratio of Spur gear pair = 5

Objectives of Trial:

1. To determine torque transmitted by system at various output loads
2. To determine efficiency of system

Table-6: Experimental Readings

Sr. No.	Load (Kg)	Speed (rpm)	Torque (N-m)	Power (watt)	Efficiency
01	1	12	0.981	1.232921	30.82302
02	2	11	1.962	2.260355	56.50887
03	3	10.6	2.943	3.26724	81.681
04	4	9.2	3.924	3.780957	94.52393
05	5	7.3	4.905	3.750134	93.75335
06	6	6	5.886	3.698762	92.46906
07	7	4.9	6.867	3.524099	88.10247
08	8	4.1	7.848	3.369984	84.24959

a. Graph of Load Vs speed:

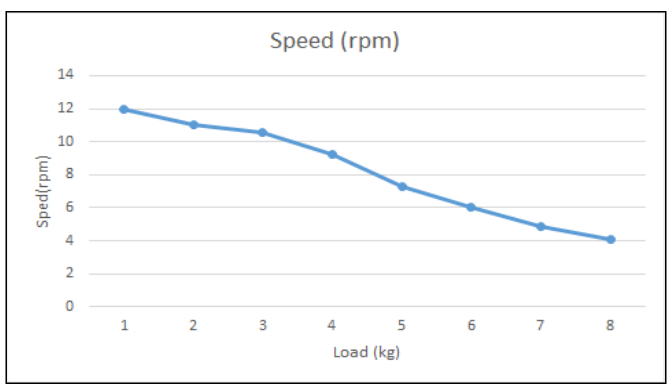

Fig.12: Graph of Efficiency vs Speed

Output speed drops with increase in load on output pulley

b. Graph of Torque Vs speed

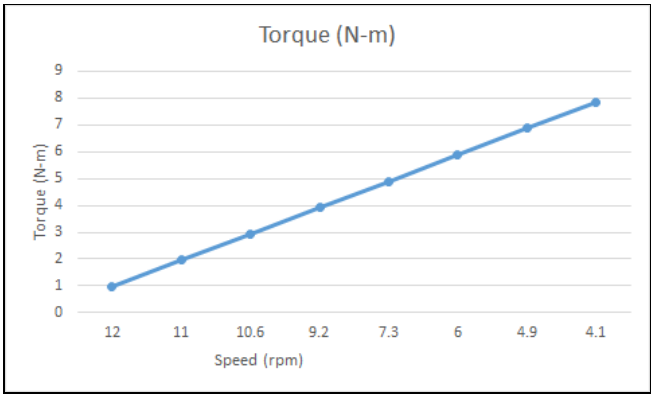

Fig.13: Graph of Torque vs Speed

Output torque increases with drop in output speed.

c. Graph of Power Vs speed

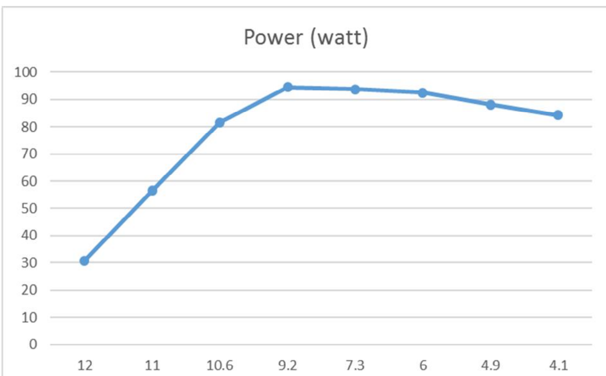

Fig.14: Graph of O/P Power vs Speed

Output power is maximum at output speed in range of 8 to 10 rpm.

VII. CONCLUSION

In this work, the opposite hand worm is designed and it is manufactured by using 'Fused Deposition Modeling' method from ABS Polymer material. Plastic made gears offers cost, design, processing, and performance advantages over metal.

The experimental validation part of the lifting force developed by the dual worm system is validated using test-rig. Following characteristics are plotted,

a) Torque Vs Speed
b) Power Vs speed
c) Power consumption of motor under rated load.

Simultaneously, the analysis of both the gears is done through ANSYS. It is found that maximum stress by theoretical and analytical methods are well below the allowable limit. Also, deformation is negligible. Hence the RH and LH Worm gears are safe under the rated torque.

From above results and observation, we can conclude that the mating worm pair system also exhibits a self-locking ability as that of conventional worm gear system. Also, the efficiency of the mating worm pair system is also greater than that of conventional worm gear system. So, with having some further modifications related to dimensions of worms such as its helix angle, lead angle and other parameters we can replace this conventional worm gear system with new worm pair self-locking system.

VIII. ACKNOWLEDGMENT

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