ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

# Design and Assembly Analysis of a Worm-Assembly in a Gear Box

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## ABSTRACT:

Differential is a part of inner axle housing assembly, which includes the differential rear axles, wheels and bearings. The differential consists of a system of gears arranged in such a way that connects the propeller shaft with the rear axles. Worm gear speed reducers are comprised of the terms "gearbox" and "speed reducer" that are used interchangeably in the world of power transmission and motion control. Gearboxes are used for speed reduction and torque multiplication. A hybrid term of "gear reducer" is also commonly used when talking about gearboxes. This is simply a gearbox (or speed reducer, or gear reducer) with a motor directly mounted to the input. A gearbox designed using a worm and worm-wheel will be considerably smaller than one made from plain spur gears and has its drive axes at 90° to each other. With a single start worm, for each 360° turn of the worm, the worm-gear advances only one tooth of the gear. In the present work all the parts of differential are designed under Structural condition and modeled. The required data is taken from journal paper. Modeling and assembly is done in Solid Works. The detailed drawings of all parts are to be furnished. The main aim of the project is to focus on the mechanical design and contact analysis on assembly of gears in gear box when they transmit power at different speeds at 2500 rpm, 5000 rpm. Presently used materials are Cast iron and Cast steel. For validating design Structural Analysis is also conducted by varying the materials for gears, Cast Iron and of Aluminum Alloy. The analysis is conducted to verify the best material for the gears in the gear box at higher speeds by analyzing stress, displacement and also by considering weight reduction.

## 1. INTRODUCTION:

Originally, worm gearing was used to secure, by compact means, a large reduction of speed between driving and driven shafts with a proportionate increase (except for frictional loss) in the torque of the driven shaft. Worm gearing is still used for this purpose, and frequently the wheel is driven by a single-thread worm of such low helix angle that the drive cannot be reversed; that is the wheel cannot drive the worm as the gearing automatically locks itself against backward rotation. Although a multiple-threaded worm when applied under like conditions is much more efficient than a single-threaded worm, it does not follow that the multiple-threaded worm should always Be used. A single-threaded worm might be preferable when the most important requirement is to obtain a high ratio and especially if the worm must be self-locking.

When power is the primary factor, the multiple-threaded worms should be used. Lubrication is an important factor when using worm gearing. An increase in heat generated means a decrease in efficiency. The amount of power which can be transmitted at a given temperature increases as the efficiency of the gearing increases. Materials for worm and worm gears are generally confined to steel for worms and bronze or cast iron for gears. When steel worms are run with bronze gears at high speeds, the worm is usually hardened with ground threads. However, unlike a worm, a worm gear's diameter is usually much larger than the width of its face.

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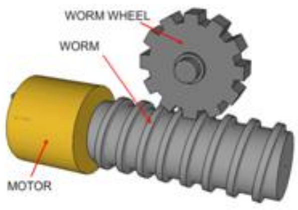


Figure 1.1: worm and worm wheel

Figure 1.2: worm gear and worm wheel

## 1.3 CIRCULAR (LINEAR) PITCH:

With a worm, circular (also referred to as linear) pitch is a distance measured along the pitch line of the gear. It can be determined by measuring – with an ordinary scale – the distance between any two corresponding points of adjacent threads parallel to the axis. (See Figure 1.4) With a worm gear, circular pitch is a distance measured along the pitch circle of the gear. It can be determined by measuring – with an ordinary scale – the distance between any two corresponding points of adjacent teeth. As noted above, this measurement should be taken on the pitch circle, which is approximately halfway down a tooth. (See Figure 1.6)

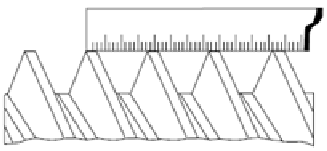

Figure 1.5: Worm

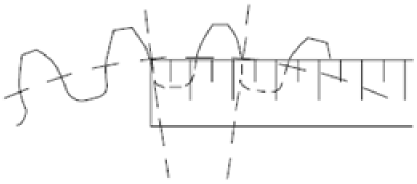

Figure 1.6: Worm Gear

## WORMS – THREAD DIMENSIONSNB:

The dimensions of a worm thread are important because they provide valuable information when determining a customer's needs. As noted earlier, a worm thread is the part of the worm that wraps (spirals) around the cylindrical base of the worm, similar to the way the threads of a screw are configured.

**The following terms are used when describing the dimensions of a worm-thread:**

1. Addendum – the part of the thread from the pitch line of the worm to the outer edge of the thread. (See Figure 1.7A)
2. Dedendum – the part of the thread from the pitch line of the worm to the bottom of the thread. The dedendum is equal to one addendum plus the working clearance (defined below). (See Figure 1.7A)
3. Working Clearance – the distance from the working depth (defined below) to the bottom of the thread.
4. Working Depth – the space occupied by the mating worm gear tooth. It is equal to twice the addendum.
5. Whole Depth – the distance from the bottom of the thread to its outside diameter.

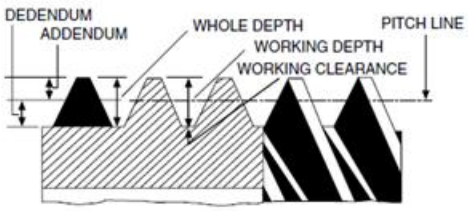


Figure 1.7A: Drawing of Worm showing cross section and full view of the thread.

## 1.4 WORMS–PITCH DIAMETER:

The pitch diameter of a worm is the diameter of the pitch circle (the "imaginary" circle on which the worm and worm gear mesh). There is no fixed method for determining the pitch diameter of a worm. (See Figure 1.7B)

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**Important:**

Pitch diameters can vary, but sound engineering practice dictates that they be as small as possible for the most efficient performance. Why? A small pitch diameter reduces the sliding velocity and, therefore, the efficiency of the worm.

### 1.8 WORMS–PRESSURE ANGLE:

The pressure angle is the angle at which a force is transmitted from the worm thread to the worm gear tooth. It determines the relative thickness of the base and top of the thread. (See Figure 1.10)

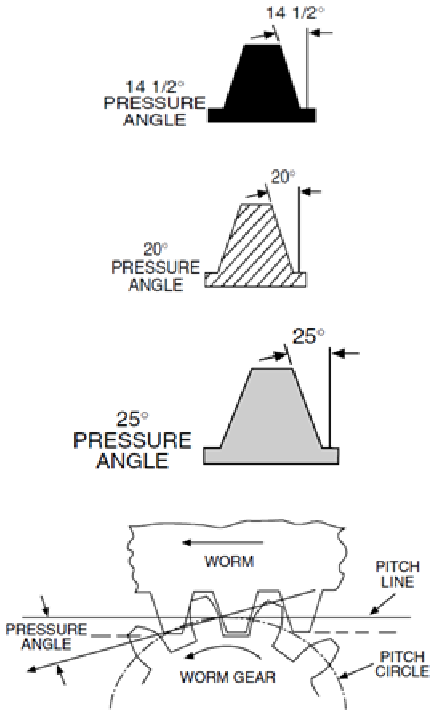


**Figure 1.10**

### 1.9 WORMS–PHYSICAL DIMENSIONS:

When ordering special (made-to-order) worms, the pitch, pitch diameter, pressure angle, number of threads and hand should always be specified, as should the physical dimensions illustrated in 1.11.

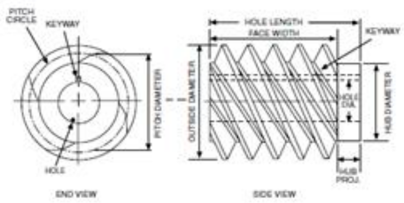


**Figure 1.11**

**Note:**

Sometimes a pinhole through the hub is required (rather than a keyway). If this is the case, be sure to specify the pin dimensions and location.

### WORMS GEARS–BASIC DIMENSIONS:

Here are definitions you need to know in order to determine the basic dimensions of worm gears. (See Figure 1.12)

1. Pitch Diameter – the diameter of the pitch circle (which, you will remember, is the "imaginary" circle on which the worm and worm gear mesh.
2. Working Depth – the maximum distance the worm thread extends into the tooth space of the gear.
3. Throat Diameter – the diameter of the throat circle at the center line of the worm gear face (the lowest point on the tooth face).
4. Outside Diameter – the largest diameter of the worm gear teeth. It is equal to the diameter of the outside circle.
5. Root Diameter – the smallest diameter of the worm gear. It is equal to the diameter of the root circle.

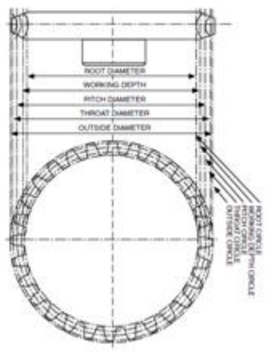


**Figure 1.12**

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### 1.19 Failure of Gears:

Failure of gears may be classified into four categories:

1. Surface fatigue (pitting)
2. Wear
3. Plastic flow
4. Breakage

The appearance of the various distress and failure modes can differ between gears that have through hardened teeth and those that have surface hardened teeth. These differences result from the different physical characteristics and properties and from the residual stress characteristics associated with the surface hardened gearing.

### Advantage of worms and worms gear:

1. Higher speed reduction could be secured; speed reduction could be secured up to 300: 1
2. Worm and worm gears operate silently
3. Worm and worm gears will have one characteristics i.e. self locking. Reverse movement will be restricted but this characteristic depends on lead angle and friction angle, we have discussed this concept in our discussion in previous post during study of worm and worm gear.
4. Worm and worm gear unit will be preferred to use if space is restricted as we have already discussed that worm and worm gear unit could be used for heavy speed reduction in compact space also.
5. Handsome output torque will be secured here with the application of worm and worm gear.

### Disadvantage of worms and worms gear:

1.Manufacturing cost is heavy as compared with manufacturing cost of bevel gear

2.Cost of raw material to manufacture the worm and worm gear set will be quite high

3.Worm and worm gear set will have heavy power losses.

4.Efficiency will be low

5.If speed reduction ratio is large, worm teeth sliding action will create lots of heat

6.Lubrication scheduled must be strictly maintained for healthiness of worm and worm gear as this unit requires much lubrication for smooth working of gearbox.

## 3. DESIGN CALCULATIONS OF WORM GEAR

### 3.1 ALUMINUM ALLOY7475-T761

### 3.1 2400 rpm

### 3.1.1 Worm Gear

Diameter of crown wheel = DG= 475mm

Number of teeth on gear = TG = 50

Number of helical teeth on shaft = TP = 6

Module = m=DG/TG=475/50=9.5=10(according to stds)

Diameter of shaft = DP = m x TP=10x6=60mm

Material used for both shaft and gear is aluminum alloy7475-T761

Brinell hardness number(BHN)=140

Pressure angle of teeth is 20° involute system Ø=20°

P=162BHP = 162x745.7w=120803.4w

We know that velocity ratio

V.R=TG/ TP= DG/DP= NP/NG

V.R=TG/ TP=50/6=6.25

V.R= NP/NG

6.25=2400/ NG

NG=384rpm

For satisfactory operation of gears the number of teeth in the shaft must not be

Less than $\frac{48}{\sqrt{1+(vr)^2}}$ where v.r=velocity ratio

$=\frac{48}{\sqrt{1+(6.25)^2}}=7.5$

Since the shafts are at right angles therefore pitch angle for the shaft

θp1=tan-1(1/v.r)

= tan-1(1/6.25)

=9.0

Pitch angle of gear θp2=90°-9=81

We know that formative number of teeth for shaft

TEP= TPsec θp1=8sec9 =6

And formative number of teeth for gear

TEG= TGsec θp2=50sec81 =319.622

Tooth form factor for the shaft

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y1P=0.154-0.912/ TEP, for 20° full depth involute system
=0.154-0.912/ 8
=0.04
And tooth form factor for gear
y1G=0.154-0.912/ TEG
=0.154-0.912/ 319.622
=0.151
Since the allowable static stresses(σO) for both shaft and gear is same (i.e σO=172.33 Mpa) and y1P is less than y1G, therefore the shaft is weaker. Thus the design should be based upon the shaft
Allowable static stress(σO) =σu/3=517/3=172.33 Mpa
σu=ultimate tensile strength=517 Mpa

### TANGENTIAL TOOTH LOAD(WT)

WT = ( σO x Cv).b.Π.m. y1P((L-b)/L)
Cv=velocity factor =3/3+v, for teeth cut by form cutters
v=peripheral speed in m/s
b=face width
m=module=10
y1p=tooth form factor
L=slant height of pitch cone

$$= \sqrt{(\frac{D_G}{2})^2 + (\frac{D_P}{2})^2}$$

DG= pitch diameter of gear =475
DP= pitch diameter of gear =80

$V=\frac{\Pi\ Dp\ NP}{60\times1000}$

=10.048m/s
Cv==3/3+10.048=0.229

$$L=\sqrt{(\frac{475}{2})^2 + (\frac{80}{2})^2}$$

=240.844
The factor (L-b/L) may be called as bevel factor
For satisfactory operation of the bevel gears the face width should be from 6.3m to 9.5m
So b is taken as 9.5m
b= 9.5x10=95

WT =(172.33x0.229)x95xΠx10x0.04($\frac{240.844-95}{240.844}$) = 2922.51N

### STATIC TOOTH LOAD (WS)

The static tooth load or endurance strength of the tooth for worm gear is given by

WS= σe.b.Π.m. y1P($\frac{L-b}{L}$)

(Flexible endurance limit) σe = 1.75XB.H.N = 1.75X140=245

$$WS=245 \times 95 \times \pi \times 10 \times 0.041\ (\frac{240.844-95}{240.844})$$

WS=18145N
For safety against tooth breakage the WS ≥1.25 Wd=13165.2875

### WEAR LOAD (WW)

The maximum or limiting load for wear for worm gears is given by

$$Ww=\frac{D_{P\times b\times q\times k}}{\cos\theta_{p1}}$$

Dp,b,q,k have usual meanings as discussed in worm gears except that Q is based on formative or equivalent no.of teeth such that ratio factor $Q=\frac{2T_{EG}}{T_{EG}+T_{EP}}$

$$=\frac{2\times319.622}{319.622+8}=1.951$$

K= load stress factor (also known as material combination factor )in N/mm2 given by

$$K=\frac{\sigma_{es}{}^2\times\sin\emptyset}{1.4}(\frac{1}{E_P}+\frac{1}{E_G})$$

σes= surface endurance limit in mpa or N/mm2

$$\emptyset = pressure\ angle$$

σes=(2.8×B.H.N-70)N/mm2
= (2.8×517-70)=322 N/mm2

$$K=\frac{(322^2)\sin 20}{1.4}\left(\frac{1}{70300}+\frac{1}{70300}\right)=0.72$$

$\cos\theta_{p1}$=cos 9 =0.987

$$Ww=\frac{80\times95\times1.951\times0.72}{0.987}=10825.25N$$

### Forces acting

WT=WNCOS$^{\emptyset}$

WN=normal load=WT/ COS$^{\emptyset}$
WT=tangential force

$$WN=\frac{tangential\quad force}{COS\emptyset}=\frac{2922.51}{\cos 20}=3110.070N$$

Radial force WR=$W_T \tan\emptyset = 3110.070 \tan 20 =$ 1131.972

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Mean radius (Rm)=(L-b/2)$sin\,\theta_{p1}$
=(240.844-95/2) $sin$ 9
=32.111

Axial force acting on the shaft shaft
WRH=WT$tan\,\emptyset\,sin\,\theta$P1
Tangential force acting at the mean radius
WT=T/Rm
T= torque on the shaft
$T = \frac{p\times60}{2\pi\times N_P}$
$=\frac{120803.4\times60}{2\pi\times2400}$
T=480.661N-m=480661N-mm

WT=480661/32.111=14968.733N

Axial force
WRH=WT$tan\,\emptyset\,sin\,\theta$P1
=14968.733$tan$ 20 $sin$ 9
=850.010N
Radial force acting on the shaft shaft
WRH=WT$tan\,\emptyset\,cos\,\theta$P1
=14968.733$tan$ 20 $cos$ 9
=5366.752N

## TANGENTIAL TOOTH LOAD(WT)

WT =( σO x Cv).b.Π.m. y1P((L-b)/L)
Cv=velocity factor =3/3+v, for teeth cut by form cutters
v=peripheral speed in m/s
b=face width
m=module=7
y1p=tooth form factor
L=slant height of pitch cone
$= \sqrt{(\frac{D_G}{2})^2 + (\frac{D_P}{2})^2}$
DG= pitch diameter of gear =150mm
Dp= pitch diameter of gear =70mm

$V=\frac{\Pi\, Dp\ \ NP}{60\times1000}$

$=\frac{\Pi\times70\times2400}{60\times1000}$

=8.792m/s
Cv==3/3+8.792=0.254
$L=\sqrt{(\frac{150}{2})^2 + (\frac{70}{2})^2}$
=82.764

The factor (L-b/L) may be called as bevel factor
For satisfactory operation of the bevel gears the face width should be from 6.3m to 9.5m
So b is taken as 9.5m
b= 9.5x7=66.5
WT =(172.33x0.254)x66.5xΠx7x0.099$(\frac{82.764-66.5}{82.764})$
=1244.7N

## STATIC TOOTH LOAD (WS)

The static tooth load or endurance strength of the tooth for bevel gear is given by
WS= σe.b.Π.m. y1P$(\frac{L-b}{L})$
(Flexible endurance limit) σe = 1.75XB.H.N = 1.75X140 =245
$WS= 245 \times 66.5 \times \pi \times 7 \times 0.099(\frac{82.764-66.5}{82.764})$
WS=6966.47N
For safety against tooth breakage the WS ≥1.25 Wd=6892.2125
WS > Wd

## WEAR LOAD (WW)

The maximum or limiting load for wear for bevel gears is given by
$Ww=\frac{D_P\times b\times q\times k}{\cos\theta_{p1}}$
Dp,b,q,k have usual meanings as discussed in worm gears except that Q is based on formative or equivalent no.of teeth such that ratio factor $Q=\frac{2T_{EG}}{T_{EG}+T_{EP}}$
$=\frac{2\times42.55}{42.55+16.554}=1.439$
K= load stress factor (also known as material combination factor )in N/mm2 given by
$K=\frac{\sigma_{es}{}^2\times\sin\emptyset}{1.4}(\frac{1}{E_P}+\frac{1}{E_G})$
σes= surface endurance limit in mpa or N/mm2
$\emptyset$ = pressure angle
σes=(2.8×B.H.N-70)N/mm2

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$= (2.8\times140-70)=322\ N/mm2$

$K=\frac{(322^2)\sin 20}{1.4}\left(\frac{1}{70300}+\frac{1}{70300}\right)=0.72$

$\cos\theta_{p1}=\cos 25.025=0.906$

$Ww=\frac{70\times66.5\times1.439\times0.72}{0.906}=5322.62N$

### Forces acting

$WT=WNCOS^{\emptyset}$

$WN=\text{normal load}=WT/\ COS^{\emptyset}$

WT=tangential force

$WN=\frac{tangential\ \ force}{COS\ \emptyset}=\frac{1244.7}{0.939}=1324.582N$

Radial force $WR=WT\ \ \tan\emptyset=1324.582\tan20=482.108N$

Mean radius $(Rm)=(L-b/2)\sin\theta_{p1}$

$=(82.764-66.5/2)\sin 25.025$

$=20.944N$

Axial force acting on the shaft shaft

$WRH=WT\tan\emptyset\sin\theta P1$

Tangential force acting at the mean radius

WT=T/Rm

T= torque on the shaft

$T=\frac{p\times60}{2\pi\times N_P}$

$=\frac{120803.4\times60}{2\pi\times2400}$

T=480.661N-m=480661N-mm

WT=480661/20.944=22949.818N

Axial force

$WRH=WT\tan\emptyset\sin\theta P1$

$=22949.818\tan 20\sin 25.0259$

$=3533.340N$

Radial force acting on the shaft shaft

$WRH=WT\tan\emptyset\cos\theta P1$

$=22949.818\tan 20\cos 25.025$

$=7567.863N$

## 4. RESULTS TABLE

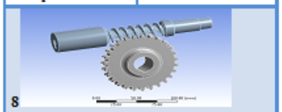
### 4.1 2400 RPM

| TANGENTIAL | Aluminum Alloy | Cast Iron |
|---|---|---|
| LOAD (N) | 2922.51 | 3243.08 |
| DISPLACEMENT (mm) | 0.0241696 | 0.0100566 |
| STRESS (N/mm2) | 3.19018 | 3.57544 |
| STRAIN | 4.1593e-5 | 1.69558 e-5 |
| STATIC | | |
| LOAD (N) | 18143.3 | 37933.7 |
| DISPLACEMENT (mm) | 0.150063 | 0.11763 |
| STRESS (N/mm2) | 19.8068 | 41.8212 |
| STRAIN | 0.000258239 | 0.000198329 |

## 5. INTRODUCTION TO CAD:

Computer-aided design (CAD), also known as computer-aided design and drafting (CADD), is the use of computer technology for the process of design and design-documentation. Computer Aided Drafting describes the process of drafting with a computer. CADD software, or environments, provide the user with input-tools for the purpose of streamlining design processes; drafting, documentation, and manufacturing processes. CADD output is often in the form of electronic files for print or machining operations. The development of CADD-based software is in direct correlation with the processes it seeks to economize; industry-based software (construction, manufacturing, etc.) typically uses vector-based (linear) environments whereas graphic-based software utilizes raster-based (pixelated) environments.

| | |
|---|---|
| Total Nodes | 39721 |
| Total Elements | 21842 |
| Maximum Aspect Ratio | 250.8 |
| % of elements with Aspect Ratio < 3 | 63.3 |
| % of elements with Aspect Ratio > 10 | 1.11 |
| % of distorted elements(Jacobian) | 0 |
| Time to complete mesh(hh;mm;ss): | 00:01:25 |
| Computer name: | WIN |



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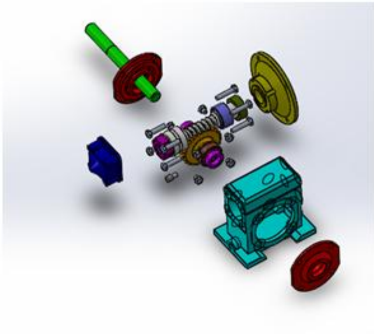
### 5.2.22 ASSEMBLE OF WORM GEAR BOX IN EXPLODE VIEW



### 6.1 INTRODUCTION TO ANSYS

ANSYS is a useful software for design analysis in mechanical engineering. That's an introduction for you who would like to learn more about ANSYS WORKBENCH. ANSYS is a design analysis automation application fully integrated with Solid Works. This software uses the Finite Element Method (FEM) to simulate the working conditions of your designs and predict their behavior. FEM requires the solution of large systems of equations. Powered by fast solvers, ANSYS WORKBENCH makes it possible for designers to quickly check the integrity of their designs and search for the optimum solution.

### 6.2 STRUCTURAL ANALYSIS OF WORM GEAR

### 6.2.1 2400 rpm

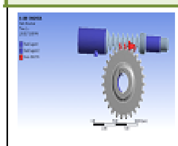
### 6.2.1.2. ALUMINUM ALLOY

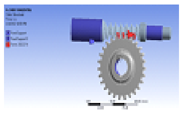
### 6.2.1.2.1 TANGENTIAL LOAD

### 6.2.1.2.2 Material Properties

### 6.2.1.1.7 Mesh Information – Details

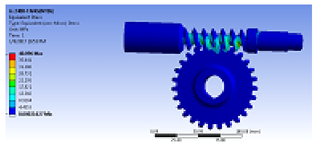
### 6.2.1.2.3 Loads and Fixtures

| Fixture Image | Fixture Details |
|---|---|
|  | Entities: 2 face(s)<br>Type: Fixed Geometry |

| Load name | Load Image | Load Details |
|---|---|---|
| Force-1 |  | Entities: 9 face(s)<br>Type: Apply normal force<br>Value: 2922.51 N |

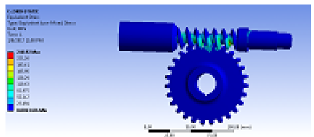
### 6.2.1.2.4 Study Results

| Name | Type | Min | Max |
|---|---|---|---|
| Stress1 | VON: von Mises Stress | 2.16e-04 N/mm^2 (MPa) | 40.096 N/mm^2 (MPa) |



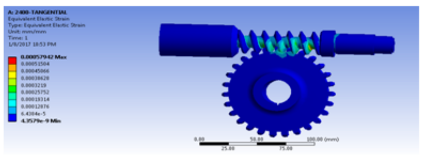
2part_assm-2400_aluminiumally_tangential_load-Stress-Stress1

| Name | Type | Min | Max |
|---|---|---|---|
| Stress1 | VON: von Mises Stress | 1.34E-4 N/mm^2 (MPa) | 548.92 N/mm^2 (MPa) |



2part_assm-2400_aluminiumally_static_load-Stress-Stress1

| Name | Type | Min | Max |
|---|---|---|---|
| Strain1 | ESTRN:Equivalent Strain | 4.357E-9 | 5.79E-4 |



2part_assm-2400_aluminiumally_tangential_load-Strain-Strain1

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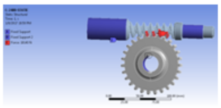
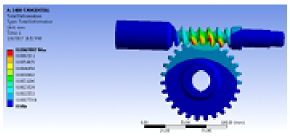
### 6.2.1.2.5 STATIC LOAD

### 6.2.1.2.6 Loads and Fixtures

| Fixture name | Fixture Image | Fixture Details |
|---|---|---|
| Fixed-1 |  | Entities: 2 face(s)<br>Type: Fixed Geometry |

| Load name | Load Image | Load Details |
|---|---|---|
| Force-1 |  | Entities: 9 face(s)<br>Type: Apply normal force<br>Value: 18145 N |

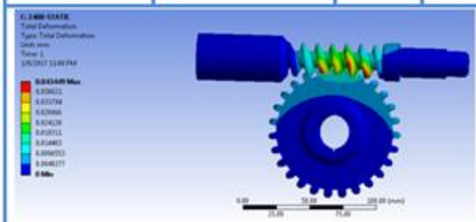
| Name | Type | Min | Max |
|---|---|---|---|
| Displacement1 | URES: Resultant Displacement | 0 mm | 0.00699 mm Node: 29507 |



2part_assm-2400_aluminiumally_tangential_load-Displacement-Displacement1

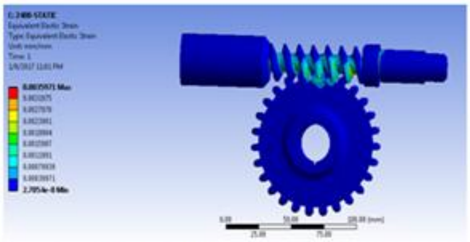
### 6.2.1.2.7 Study Results

| Name | Type | Min | Max |
|---|---|---|---|
| Displacement1 | URES: Resultant Displacement | 0 mm | 0.0434 mm |



2part_assm-2400_aluminiumally_static_load-Displacement-Displacement1

| Name | Type | Min | Max |
|---|---|---|---|
| Strain1 | ESTRN: Equivalent Strain | 2.705E-8 | 0.00359 |

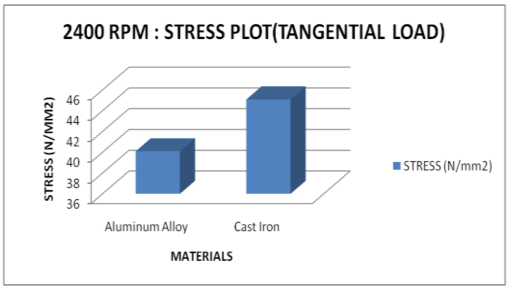


2part_assm-2400_aluminiumally_static_load-Strain-Strain1

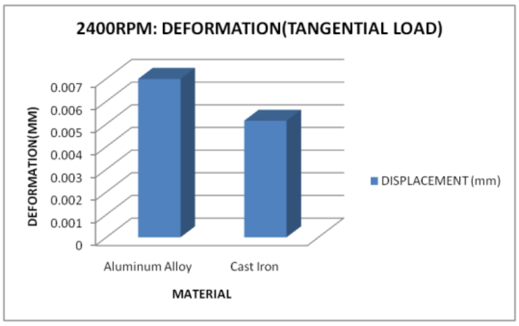
## 7. RESULTS TABLE

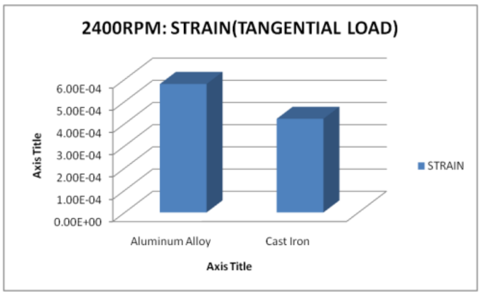
### 7.1 2400 RPM

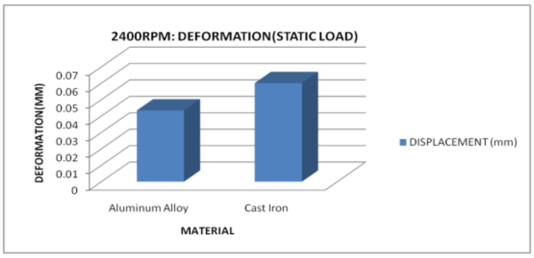
| S.NO. | TANGENTIAL | Aluminum Alloy | Cast Iron |
|---|---|---|---|
| 1 | LOAD (N) | 2922.51 | 3243.08 |
| 2 | DISPLACEMENT (mm) | 0.00699 | 0.005145 |
| 3 | STRESS (N/mm2) | 40.096 | 45.084 |
| 4 | STRAIN | 5.79E-04 | 4.22E-04 |
| 5 | STATIC | | |
| 6 | LOAD (N) | 18143.3 | 37933.7 |
| 7 | DISPLACEMENT (mm) | 0.0434 | 0.0601 |
| 8 | STRESS (N/mm2) | 248.92 | 527.34 |
| 9 | STRAIN | 0.0035 | 0.0049 |

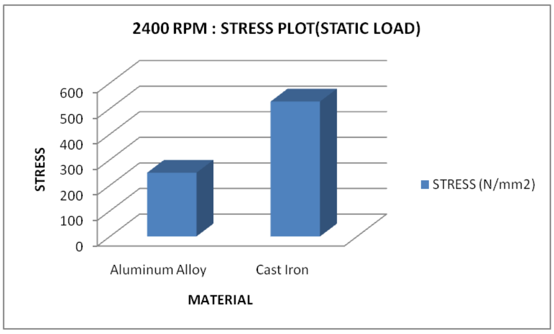


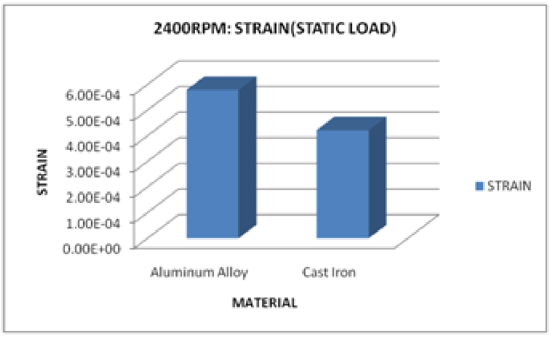
ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal




## 8. CONCLUSION:

In our project we have designed a Worm gear box. Loads are calculated when the gears are transmitting different speeds 2400rpm and 5000rpm and different materials Aluminum Alloy and Cast Iron. By observing the structural analysis results using Aluminum alloy the stress values are within the permissible stress value. So using Aluminum Alloy is safe for differential gear. When comparing the stress values of the three materials for all speeds 2400rpm and 5000rpm, the values are less for Aluminum alloy than Cast Iron. And also weight of the Aluminum alloy reduces almost 3 times when compared with d Cast Iron since its density is very less. Thereby mechanical efficiency will be increased. By observing analysis results, Aluminum Alloy is best material for Differential.

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