Design and Development of Adaptive Headlight System (AHS)

Manish Bhargao, Student, MIT College of Engineering, Pune, India,mbhargao1997@gmail.com
Mihir Sonawane, Student, MIT College of Engineering, Pune, India, mihirsss10@gmail.com
Prof Abhijit.A.Kadam, Professor,MIT College of Engineering, Pune,India, aakadam31@gmail.com
Gopal Patil, Student, MIT College of Engineering, Pune, India, patilgopal500@gmail.com
Rahul Yadav, Student, MIT College of Engineering, Pune, India, rahul225599@gmail.com

Abstract **In today's fast moving vehicle scenario, road safety is of utmost importance. Many people have lost their lives while travelling, due to a road accident. So we should mitigate such accidents if we wish to travel safely. To cater this cause, we propose an adaptive steering controlled headlight setup. The system can be adopted in any type of four wheel vehicles/trucks or trailers etc. Without being an economic burden on the end user. The notion of steering controlled headlight is not new, but its adaptability according to the steering turning angle is its novel part. A lot of companies have developed technologies that incorporate turn able headlight to better illuminate the path, but these technologies are quite expensive and continue to be distant from the majority of car owners. So we felt the need of developing a mechanism that incorporates few simple components like gears, linkages etc. And can be readily fitted onto any steering column without much of a design variations. The setup contains a gear mounted on the steering column and it is mesh with the semicircular gear which is mounted on an axis parallel to the steering column. A linkage or wire mechanism is used to transmit the rotating motion to the headlight. The headlight is designed so that it has 1 degree of freedom i.e. it can rotate about its axis. This setup will be of huge aid to the driver, as it will permit him to see the incoming obstructions in hilly areas or in the regions with sharp turns as it provides a better illuminated path by the virtue of adaptable headlights.**

Keywords —Adaptive Headlight System (AHS), Bevel Gears, Spur Gears, Safety, Gear Ratio

I. Introduction

Safety is the biggest concern for automobile manufacturers and also for the driver of that automobile. The manufacturers who installs latest safety equipment in their vehicle wins the trust of its customers and hence makes it big in the market. Adaptive headlight is one such attempt. In fact, adaptive headlight technology first appeared on a 1960s Citroën DS. The French automaker developed a quad-headlight system where the inner pair of lights swiveled with the front wheels, making it the first car to "see" around corners.

Nowadays big companies such as Ford have patented their own adaptive headlight system. Based on stepper motors and sensors for safety purposes. But these systems are very costly. These system are precise and safety on road is guarantee.

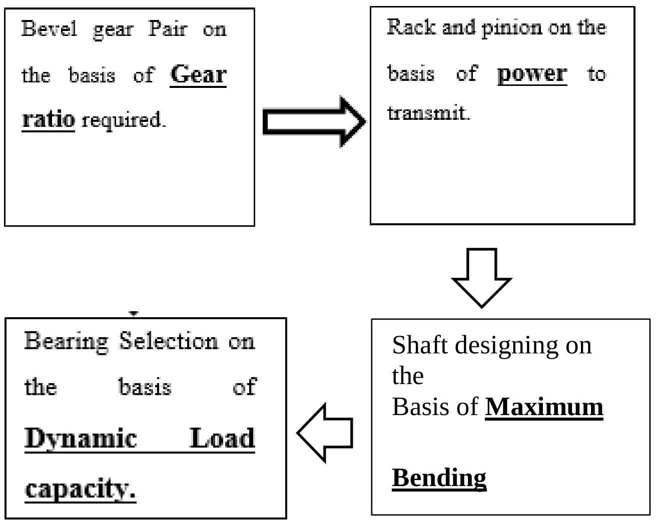
II. Design Methodology

Two Spur gears will be installed on the of **rack and pinion** mechanism of the steering of the car.

This gears will rotate along with the pinion of the Rack and pinion. By connecting a shaft to the spur gears a **bevel gears** assembly can be constructed having a suitable **gear ratio**.

This gear ratio will determine the actual relation between the angle of rotation of steering wheel and the angle of rotation of the headlight.

The headlight can be mounted of the Bevel gear which will be connected to the shaft



Design Flowchart

DESIGN OF BEVEL GEAR

For the design of bevel gears, certain assumption have to be made.

Initial Known data:

The material selected is C.I FG 220 (S_{ut}=220Mpa)

Pressure angle (ϕ) = 20°

Module of gears= 3

Gear ratio = 3:1

$F.O.S_{bending}$=2

We know that,

Minimum number of teeth on Bevel pinion is given by:

$$Z_{min} = \frac{2h_a cos\gamma_p}{msin^2\phi}$$ **……………………………..(eq 2.1)**

Putting h_a=1m,

γ_p $=\tan^{-1}\frac{1}{G}$

γ_p $=\tan^{-1}\frac{1}{3}$

Φ=20°

Substituting the values in eq 3.1.1

We have, Number of teeth on Bevel Pinion (Z_p) =18

We know that,

$$m = \frac{d}{z}$$

Hence, d_p=54

G=3 hence Z_g=54

d_g=162mm

Now, Pitch Cone diameter

AO=$\sqrt[2]{(d_p/2)^2 + (d_g/2)^2}$

AO=$\sqrt[2]{(54/2)^2 + (162/2)^2}$

AO=85.38mm

Face width b,

b= $\frac{AO}{3}$ or 10m } whichever is smaller

b=28.71 or 30mm

Selecting b=28.71mm

Consider Bending Failure

Bending force acting is given by,

$$F_b = \sigma_{bp} bmY\prime_p(1 - \frac{b}{AO})$$ **…………………………..Eq(2.2)**

$$\sigma_{bp} = \frac{S_{ut}}{3} = \frac{220}{3} = 73.33MPa$$

$b = 28.4mm$

m=3

$$Y\prime_p = 0.484 - \frac{2.87}{Z\prime_p}$$

Here,

$Z\prime_p$ is formative number of teeth.

$$Z\prime_p = \frac{Z_p}{cos\gamma_p} = \frac{18}{0.9487} = 18.97$$

Similarly,

$Z\prime_g$ =170.71

Substituting the obtained values in eq 3.1.2

We have, $\left(Fb = 73.33 * 28.44 * 3 * 0.3327 * \left(1 - \frac{28.4}{85.38}\right)\right)$

F_b=1385.7 N

Consider Wear Failure

Wear failure of gear is given by the following expression

$$F_w = \frac{0.75 d_p b Q\prime K}{cos\gamma_p}$$ **……………………………(Eq 2.3)**

Load stress factor

$$K = 0.21(\frac{BHN}{100})^2$$

$$K = 0.21(\frac{250}{100})^2$$

K=1.31

Ratio Factor for gear pair

$$Q' = \frac{2Z'_g}{Z'_g + Z'_p}$$

$$Q' = \frac{2170.71}{170.71 + 18.97}$$

$$Q' = 1.8$$

F_w=$\frac{0.75*54*28.4*1.8*1.31}{0.9488}$

Fw=2858.79 N

Comparing Fb and Fw we observe that Fb<Fw

Hence, Designing Gear for Failure

$F_{eff}=\frac{F_t}{K_v}$……………………………………..(Eq 2.4)

Here , Kv is called as the Barth Factor or velocity Factor.

Kv=$\frac{5.6}{5.6+\sqrt[2]{V}}$

V is Pitch line velocity and is given by;

$$V = \frac{\pi d_p n_p}{60000}$$

V=0.8482m/s

Putting this value is expression of Kv we have

Kv=0.8587

Putting the value in eq 3.1.4 we have

$F_{eff}=\frac{F_t}{0.8567}$

Fb=F.O.S *F_{eff}…………………………......(Eq 2.5)

1038.96=2*$\frac{F_t}{0.8567}$

Ft=593.56 N

Power that is transmitted is calculated as

$$P = F_t * v$$

P=503 Watts

Forces on Bevel gear are given as

$$F_{rp} = F_{aG} = F_t tan\phi cos\gamma_p$$

=593.56*tan20*cos18.43

=157.69 N

$$F_{ap} = F_{ap} = F_t tan\phi cos\gamma_g$$

=593.56*tan20*cos71.56

=68.33 N

Bevel gear Results

Z_p	18	Ft	593.56 N
Z_g	54	Frp=Fag	204.95 N
m	3	Fap=Frg	68.33
dp	54mm		
dg	162mm		

Table 1 Bevel gear results

SPUR GEAR CALCULATIONS

For the design of Rack and Pinion some data is to assumed

Initial Data

Material for Rack C.I FG 220 (S_{ut}=220 Mpa)

Lewis Form Factor Y_p=0.3644

Lewis form factor Y_r =0.484

F.O.S=1.5

Pressure angle(ϕ)=20˚

Machining Factor (K_a)=1

Application Factor(Km)=1.05

Barth Velocity factor(Kv)=0.9836

Tangential Force is transmitted by the rack to the spur pinion

Hence, Tangential Force is given as

$$F_t = \frac{2T_m}{Z_p m}$$ ……………………………………(Eq 3.1)

Here Tm is the torque which is given by,

$$T_m = \frac{P60000}{2\pi n_p}$$

$$T_m = \frac{503 * 60000}{2\pi * 300}$$

Tm=16010.9 KN

Substituting the value of Tm in Eq 3.2.1

$$F_t = \frac{1334.2}{m}$$ ………………………………………(Eq 3.2)

Now Applying Lewis Equation for Bending

$$F_b = \sigma_{bp} bmY_p$$

fF_b = $\frac{S_{ut}}{3}$ $10m\,m\,0.3644$

Fb=267.21 m^2…………………………………Eq(3.3)

Now we know F.O.S

$$F_b = F.O.S * F_{eff}$$

$$F_b = F.O.S * \frac{K_a K_m F_t}{K_v}$$

$$F_b = 1.5 * \frac{1.05 * 1}{0.9836} F_t$$ ……………………….(eq 3.4)

Fb=1.601 Ft…………………………………....(eq 3.5)

Putting equation 3.3 and 3.4 in

$$267.21m^2 = 1.601 * \frac{1334.2}{m}$$

$$\Longrightarrow m^3 = \frac{1.601 * 1334.2}{267.21} = 2.86$$

m= 1.99 ≈2

Now pitch line velocity is given by;

$$V = \frac{\pi d_p n_p}{60000}$$

$$V = \frac{\pi 48 * 300}{60000}$$

V=0.75 m/s

Tangential Force is given as

$$F_t = \frac{P}{V} = \frac{T_p}{\frac{d_p}{2}}$$

Ft = $\frac{16010.9}{24}$ =667.12 N

Now calculating the forces associated with the Gear

$$F_r = F_t * tan\phi$$

Fr=667.12*tan20

Fr=242.81 N

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Spur Gear Results

Material of Rack and pinion	C.I FG 220 (Sut=220 Mpa)
Number of teeth on pinion	$Z_p = 24$
Module	2
Tangential force (Ft)	667.12 N
Radial Force (Fr)	242.81 N

Table 2 Spur gear results

III. Shaft Calculation Procedure

Step 1: Calculate the torque on the shaft from power Calculate the torque on the shaft from power.
Step 2: Find the Find the torsional stress in the shaft stress in the shaft
Step 3: Calculate the loads coming from gears, belts Calculate the loads coming from gears, belts or chains or chains
Step 4: Calculate the bending moment due to the Calculate the bending moment due to the acting forces. If necessary combine the forces acting
Step 5: Calculate the bending stress in the shaft Calculate the bending stress in the shaft
Step 6: Combine the bending stress and the Combine the bending stress and the torsional stress using the theories text" should *not* be selected.

IV. Analysis

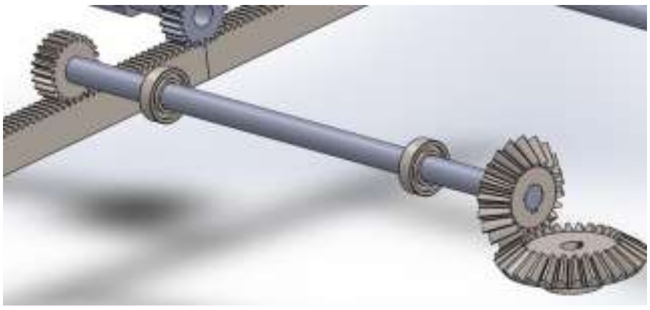

Fig1 Shaft CAD

Consider the shaft
The SFD and BMD in the horizontal and Vertical Plane are as follows

- In Horizontal Plane

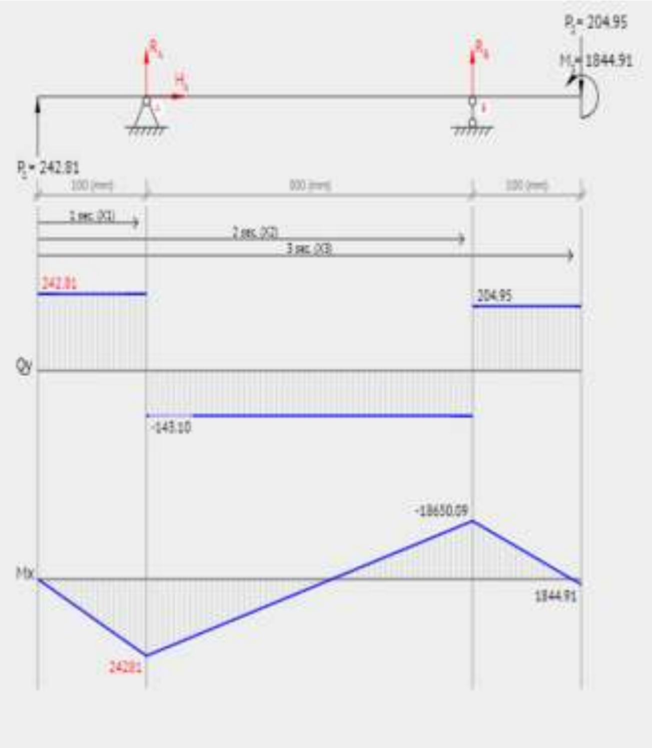

Fig 2 SFD BMD horizontal plane

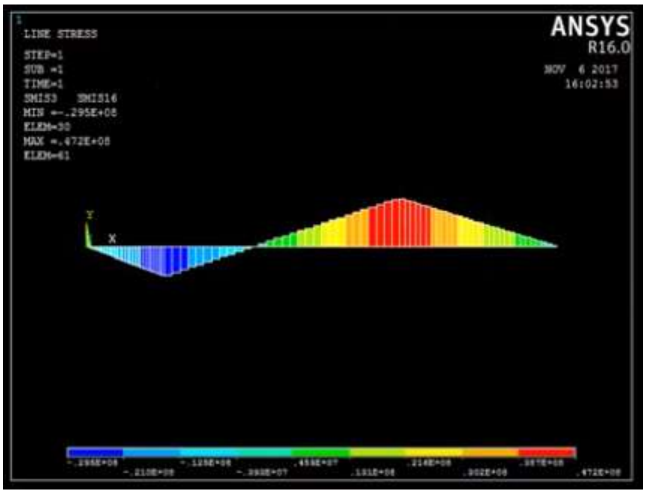

Fig 3 BMD on ANSYS APDL

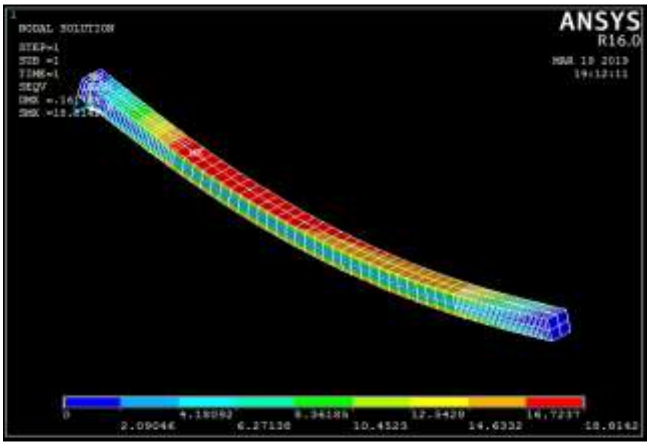

Fig 4 Deflection of shaft

- In Vertical Plane

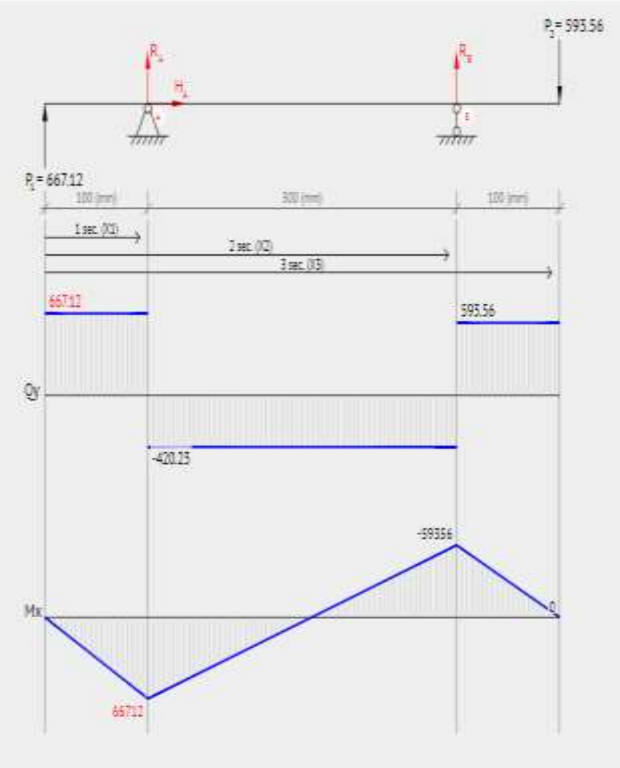

Fig 5 SFD BMD of Vertical shaft

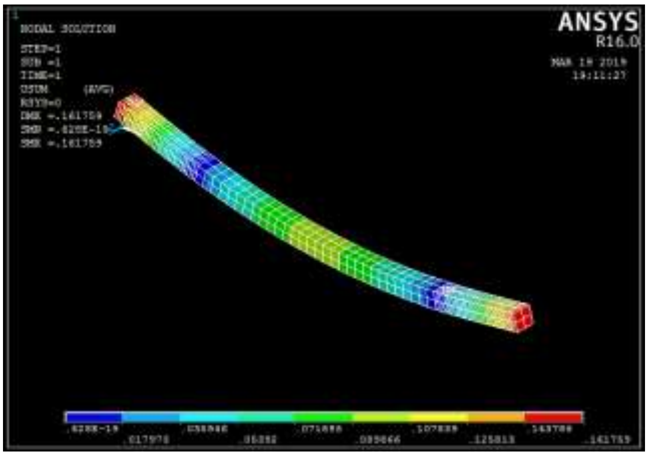

Fig 6 Von Mises stresses

V. Conclusion

After following the designing procedure for Rack and pinion, bevel gear pair,
Shaft and bearing the final assembly is done in CAD software Solid works.

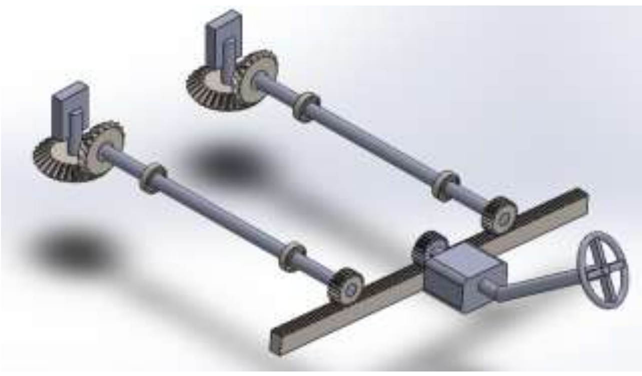

Fig 7 Final CAD assembly

VI. Advantages

- Enhanced existing functionality.
- Improved visibility.
- Simple Design.
- Decreased number of night time accidents.
- Increased safety for drivers and pedestrians.

VII. Disadvantages

- Less precision than electronic system.
- Increased steering effort.
- Increased weight.
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