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Design and Simulation of 4 Wheel Steering System

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Abstract*— In standard 2 Wheel Steering System, the rear set of wheels are always directed forward and do not play an active role in controlling the steering. While in 4 Wheel Steering System, the rear wheels do play an active role for steering, which can be guided at high as well as low speeds. Production cars are designed to under steer and rarely do they over steer. If a car could automatically compensate for an under steer/over steer problem, the driver would enjoy nearly neutral steering under varying operating conditions. Also in situations like low speed cornering, vehicle parking and driving in city conditions with heavy traffic in tight spaces, driving would be very difficult due to a sedan's larger wheelbase and track width. Hence there is a requirement of a mechanism which result in less turning radius. We have developed an innovative 4 wheel steering design to implement a mechanism that can serve the purpose of changing in-phase and counter-phase steering of rear wheels depending upon the conditions of turning and lane changing with respect to front wheels, thus enhancing the maneuverability of a sedan in accordance with its speed. Our 4 Wheel Steering System gives 64.4% reduction in turning circle radius of a sedan which is reduced from 5.394m to 1.92m, considering HONDA CIVIC as a standard car for our calculations, and steering ratio thereby obtained is 8.177:1 which gives much better maneuverability and control on the car even while driving at high speeds.

Abbreviations:

4W S -	**4 Wheel Steer**
2WS -	**2 Wheel Steer**
DRRC -	**Double Rear Rack Concept**
ECU -	**Electronic Control Unit**
AA -	**Ackerman Angle**
LD1PF -	**Linear Displacement of Rack for 1 rotation of front pinion**
LD1PR -	**Linear Displacement of Rack for 1 rotation of rear pinion**
PCD -	**Pitch Circle Diameter**
OD -	**Outer Diameter**
DFMEA -	**Design Failure Mode Effect Analysis**

***Index Terms*—Introduction, Literature Survey, The Concept, Calculations, Casing, Analysis of Components, DFMEA, Cost Analysis, Advantages, Disadvantages, Future Scope, Conclusion**

I. INTRODUCTION

4 Wheel Steering System is employed in vehicles to achieve better maneuverability at high speeds, reducing the turning circle radius of the car and to reduce the driver's steering effort. In most active 4 wheel steering system, the guiding computer or electronic equipment play a major role, in our project we have tried to keep the mechanism as much mechanical as possible which can be easy to manufacturing and maintenance.

This project focuses on a mechanically feasible & innovative design involving a double rack and pinion system for rear wheels enclosed within a casing, connected to the steering column by a combination of a bevel gear assembly & telescopic shaft. The movement of the rear wheels is done by the movement of the rear pinions which in turn move the newly designed spindle to achieve the required movement of the rear wheels.

II. LITERATURE REVIEW

New generation of active steering systems distinguishes a need of steering of rear wheels for the reason of directional stability from a need of steering of rear wheels for the reason of cornering at slow speed.

- **Condition for True Rolling**

  While tackling a turn, the condition of perfect rolling motion will be satisfied if all the four wheel axes when projected at one point called the instantaneous center, and when the following equation is satisfied:

$$\cot \phi - \cot \theta = c/b$$

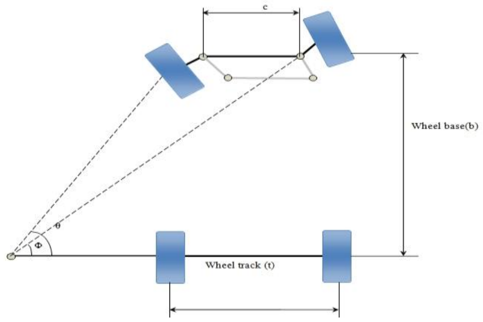

Fig 1: True Rolling Condition

- **Slow and High Speed Modes**

At Slow Speeds rear wheels turn in direction opposite to that of front wheels. This mode is used for navigating through hilly areas and in congested city where better cornering is required for U turn and tight streets with low turning circle which can be reduced as shown in Fig 2.

Fig 2: Slow Speed

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At High Speeds, turning the rear wheels through an angle opposite to front wheels might lead to vehicle instability and is thus unsuitable. Hence the rear wheels are turned in the same direction of front wheels in four-wheel steering systems. This is shown in Fig 3.

Fig 3: High Speed

- **In-Phase and Counter-Phase Steering**

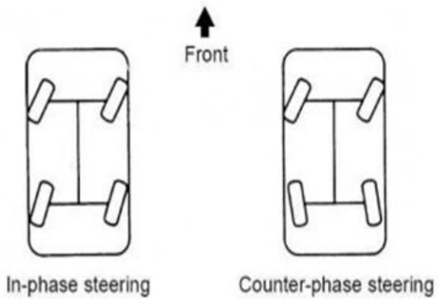

Fig 4: In-Phase and Counter-Phase Steering

The 4WS system performs two distinct operations: in- phase steering, whereby the rear wheels are turned in the same direction as the front wheels, and counter phase steering, whereby the rear wheels are turned in the opposite direction. The 4WS system is effective in the following situations:

- ✓ Lane Changes
- ✓ Gentle Curves
- ✓ Junctions
- ✓ Narrow Roads
- ✓ U-Turns
- ✓ Parallel Parking

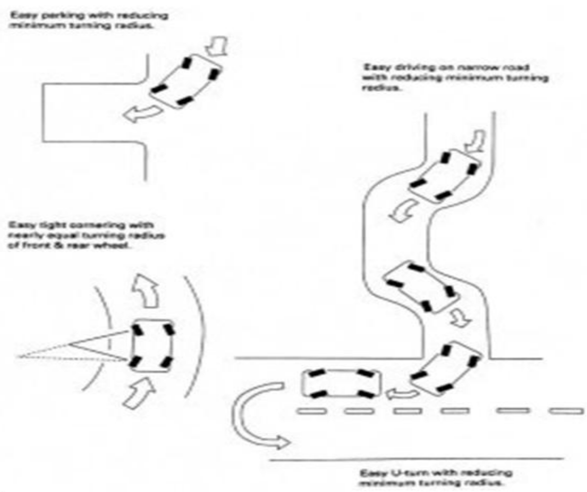

Fig 5: Car in Various Modes

- **U-Turns**

By minimizing the vehicle's turning radius, counter-phase steering of the rear wheels enables U-turns to be performed easily on narrow roads.

- **Parallel Parking**

Zero steer can significantly ease the parking process, due to its extremely short turning footprint. This is exemplified by the parallel parking scenario, which is common in foreign countries and is pretty relevant to our cities. Here, a car has to park it between two other cars parked on the service lane. This maneuver requires a three-way movement of the vehicle and consequently heavy steering inputs. Moreover, to successfully park the vehicle without incurring any damage, at least 1.75 times the length of the car must be available for parking for a two-wheel steered car.

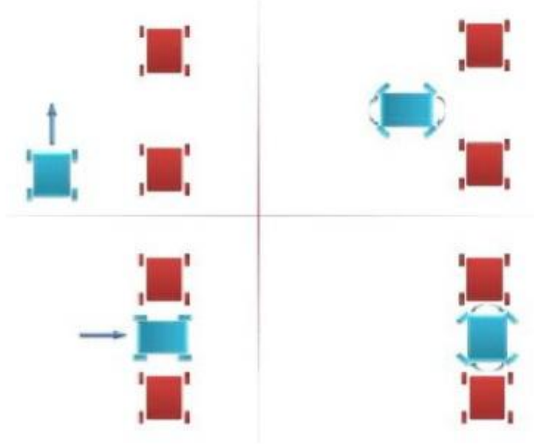

Fig 6: Parallel Parking

As can be seen clearly, the car requires just about the same length as itself to park in the spot. Also, since the 360 mode does not require steering inputs, the driver can virtually park the vehicle without even touching the steering wheel. All he has to do give throttle and brake inputs, and even they can be automated in modern cars. Hence, such a system can even lead to vehicles that can drive and park by themselves.

- **High Speed Lane Changing**

Another driving maneuver that frequently becomes cumbersome and even dangerous is changing lanes at fairly high speeds. Although this is less steering intensive, this does not require a lot concentration from the driver since he has to judge the space and vehicles behind him. Here is how crab mode can simplify this action as shown in Fig. 7

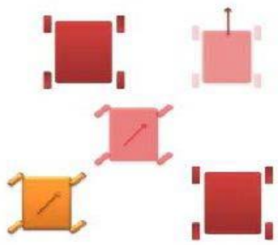

Fig 7: Lane Changing

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III. THE CONCEPT

This project consists of front rack and pinion mechanism assisted by three bevel gears of which one is connected to front pinion, one is connected to steering rod in which input is given by the driver and third one will be connected to rear pinion. Rear wheel system consists of two racks with two pinions. One of the racks will be in front of the rear wheel axis (primary rack) and the other will be behind the axis (secondary rack). Also at any point in the system, one of the rack & pinnion assembly will be engaged with the other being disengaged. Motion of pinion will be guided by an actuating pump connected to intermediate shaft which will receive input from speed sensors. The engaging & disengaging of the rack & pinion assembly will depend on the input received from the speed sensor. At lower speeds i.e. below 35kmph the pinion will be in contact with secondary rear rack so as to keep the wheel's motion out of phase while for speeds above 35kmph pinion will be in contact with front rack of rear steering system, giving in phase motion to wheels. This position of the rear pinion on the rack is controlled by a hydraulic circuit and an actuator mechanism. The angle turned by rear wheels will not be as high as that of front wheels because the function of rear steering system is to assist the motion of front wheels and not provide its own direction. This change of angle is obtained by changing gear ratio of rack and pinion.

Fig 8: Proposed Design in SOLIDWORKS

- **Bevel Gears**

Three bevel gears are used in this project to transmit the motion given to steering wheel by driver to front as well as rear wheels. Steering wheel is connected to vertical bevel gear by the means of connecting rod. This vertical bevel gear transmits motion to two horizontal bevel gears of which one will be connected to front pinion and other one to rear pinion. Depending on gear ratio front pinion will receive input from the gear and this will give the front wheels its respective motion. Also same in case for rear pinion it will be given input from gear assembly and the pinion will set its position on respective rack depending on speed of the vehicle controlled via the sensor, hydraulic pump and telescopic shaft.

Advantage: One is to one gear ratio helps in motion of rear pinions to act in accordance with the steering wheel and front pinion.

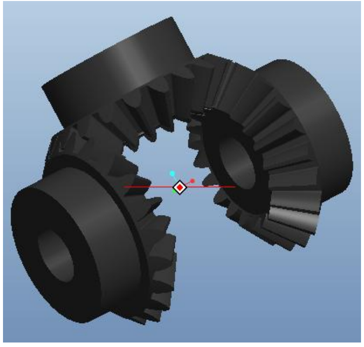

Fig 9: Bevel Gears

- **Telescopic Shaft**

This shaft plays a key role in setting the position of the rear pinions on rear racks. The movement of rear pinions is in synchronization with the common shaft attached with them which reciprocates with the force of hydraulic pressure in coordination with Hydraulic Circuit Directional (HCD) control system which is operated by Electronic Control Unit (ECU) as a reaction to changes in speed, the movement of shaft attached to the pinions is paradigm to telescopic actuator hence the name telescopic shaft. Advantage: The reciprocative movement of pinions become quick and synchronizes evenly with speed of the vehicle due to hydraulic pressure offered from Hydraulic Circuit Directional control system.

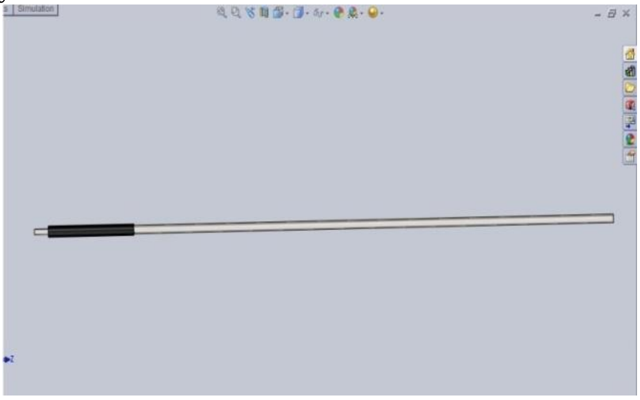

Fig 10(a): Telescopic Shaft Full View

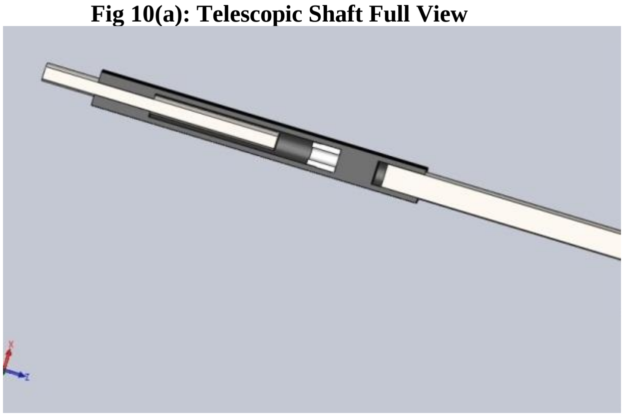

Fig 10(b): Telescopic Shaft Frone Sectional View

- **Pump and Sensors**

A sensor is connected to acceleration pedal which receives and transmits the signal to Electronic Control Unit (ECU),

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which operates the pump, initiating the control volve to apply hydraulic pressure on telescopic shaft, thus sets the pinion's motion from primary rack to secondary rack via the reciprocating movement of telescopic shaft. This system even improves control over traditional mechanical four-wheel steering systems. Advantage: The dynamic of these new systems is the full independent control of speed, direction and traction control through the high-speed sensor networks.

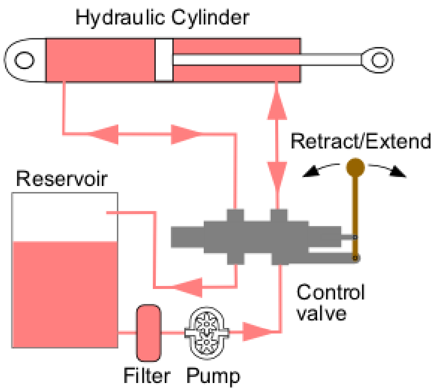

Fig 11: Hydraulic Circuit Directional Control

- **Double Rear Rack Concept (DRRC)**

Double rear rack concept is the first of its kind to be implemented in any car. These are connected by the means of two pinions. The pinion slides through the casing supported by bearing, which is connected to both racks. Thus in rear part, two rack and pinion have same module of teeth to avoid any type of interference between pinion and rack when it is not in its mean position. Casing along with bearing plays a major role for steering as it is the only method through which pinion can switch racks. When car is running is speed below 35kmph the hydraulic actuator will keep the pinion on the secondary rack to keep both wheels out of phase. As soon as the speed of car exceeds 35kmph, hydraulic actuator shifts the pinion to primary rack and this done easilty with the help of Telescopic Shaft. Thus it keeps all the wheels in phase assisting high speed motion.

Advantage: In- phase and out- phase turning of rear wheels become simpler and works with in accordance to speed of the vehicle balancing the maneuverability of vehicle.

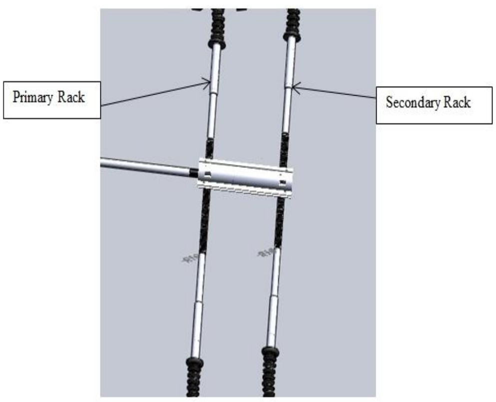

Fig 12: Double Rear Rack

- **Newly Designed Spindle**

A totally new type of spindle is designed for the rear wheels to assist motion of both racks. It consists of 4 arms of which two are for connecting the racks and remaining two for rear suspension.

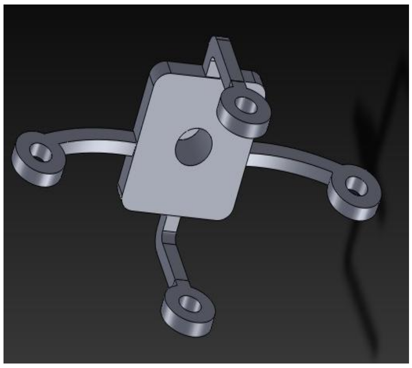

Fig 13: Newly Designed Spindle

IV. CALCULATIONS

(Note: The following calculations are done in SI units only, while for convenience supportive notes are been made)

1. Standard Specifications of Honda Civic

Sr. No	Standards	Dimensions
1	Wheelbase	105.1"
2	Steering Ratio	14.89
3	Lock to Lock Turns	2.87
4	Track Width Front and Rear	60"
5	King Pin Centre to Centre Distance	51"

Table I: Civics' Specifications for tie rod calculations

Assumed values for Rack and Pinion:
No of teeth on Front Pinion = 20
No of teeth on Rear Pinion = 16
Module of Front and Rear Pinion = 1.25mm
No of teeth on Front and Rear Rack = 26
Module of Rack = 1.25mm
These values on pinion are assumed so as to avoid interference with the rack and to obtain the required values of lock angles of all wheels.

2. Calculations for Front Rack-Pinion and Tie-Rod

Sr. No.	Particulars	20^0 full depth involute system	For T =20 (Front Pinion)	For T = 16 (Rear Pinion)
1.	Module	-	1.25mm	1.25mm
2.	Addendum	1m	1.25mm	1.25mm
3.	Dedendum	1.25m	1.5625mm	1.5625mm
4.	Working Depth	2m	2.5mm	2.5mm
5.	Minimum total Depth	2.25m	2.8125mm	2.8125mm
6.	Tooth	1.5708m	1.9635mm	1.9635mm

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	Thickness			
7.	Minimum Clearance	0.25m	0.3125mm	0.3126mm
8.	Fillet radius at root	0.4m	0.5mm	0.5mm
9.	PCD (D)	T x m	25mm	20mm
10.	Outside Diameter (OD)	PCD + (2 x Addendum)	27.5mm	22.5mm
11.	Inside Diameter (ID)	PCD – (2 x Dedendum)	21.875mm	16.875mm
12.	Circular Pitch	πD/T	3.927mm	3.927mm

Table II: Pinion Calculations

3. Calculations for Front Rack-Pinion and Tie-Rod

- Calculating Ackerman Arm Angle

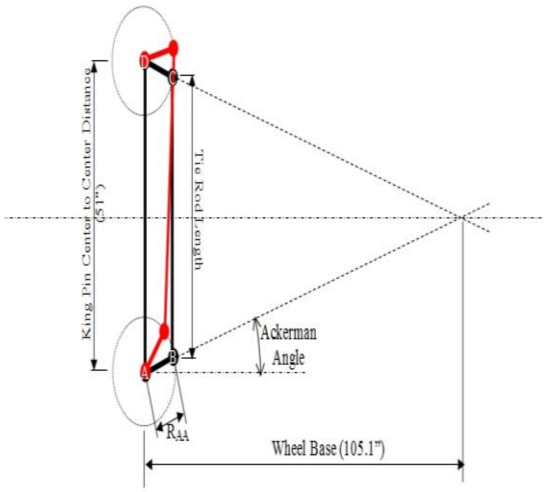

Fig 14: Ackerman Steering Mechanism

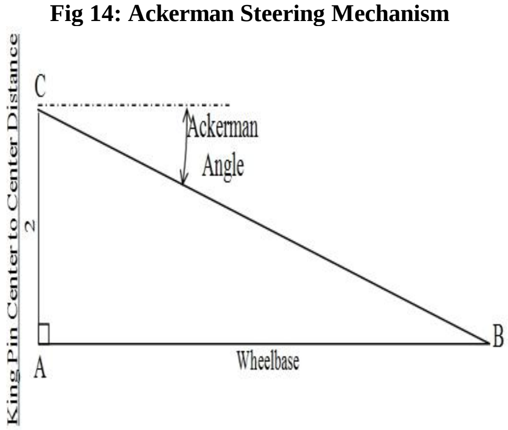

Fig 15: Ackerman Angle

$$\tan^{-1}\left[\frac{\text{king pin center to center distance}/2}{\text{Wheelbase}}\right] = \alpha$$

$$\tan^{-1}\left(\frac{51/2}{105.2}\right) = 13.64^{\circ}$$

Therefore,

Ackerman Arm Angle (α) = 13.64°

- Calculating Arm Base and Length of Tie Rod

To find the length of the tie rod, we can decompose the trapezoid ABCD into a rectangle and two triangles.

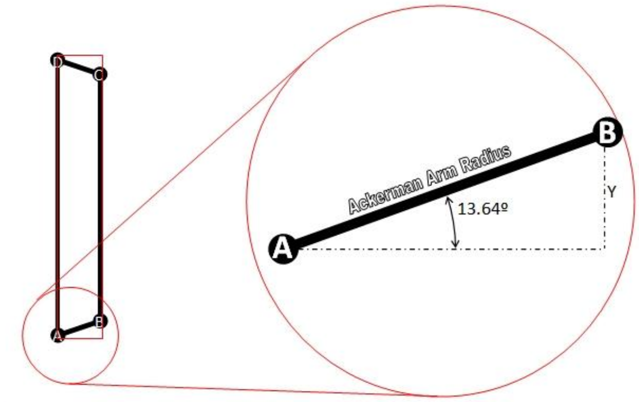

Fig 16: Ackerman Arm Radius

We will recall that the SIN of an angle is the ratio between the side opposite the angle and the hypotenuse. In shorthand it looks as follows:

$$\text{Sin } 13.64^{\circ} = \frac{Y}{R}$$

Where,

Ackerman Arm Radius R = 6" (Assumption)

As you know, the name of the game in Algebra is getting the variable by itself, so…

$$\textit{Arm Base } (Y) = 6 \times \sin 13.64^{\circ} = 1.4149$$

Arm Base (Y) = 1.4149"

- Verifying Arm Base Y

$$Y = \frac{(\text{King pin c} - \text{c distance}) - (\text{Length of tie rod})}{2}$$

$$Y = \frac{51 - 48.17}{2} = 1.415''$$

Arm Base (Y) = 1.415"

Therefore,

Arm Base (Y) is equal to the calculated value and is thus verified.

So, the tie rod is 1.414" inches shorter on the bottom and 1.414" inches shorter on the top than the kingpin center to center distance. Expressed mathematically:

$$L_T = D_{KC} - 2R_{AA}[\sin(\text{Ackerman Angle})]$$

Where:

L_T = length of the tie rod

D_{KC} = distance between kingpin's center to center

R_{AA} = radius of the Ackerman Arm (Assumed) = 6

$$L_T = 51 - 2 \times 6(\sin 13.64^{\circ}) = 48.17''$$

Length of Tie Rod = 48.17"

Another way to find Ackerman angle (α)

$$\sin\alpha = \frac{(\text{King pin c} - \text{c distance}) - (\text{Length of tie rod})}{2 \times 6}$$

$$\sin\alpha = \left[\frac{51 - 48.17}{2}\right]$$

Ackerman Arm Angle (α) = 13.64°

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- Calculating the transect, assuming that the Ackerman arm labeled AB steers 20 degrees to the left as shown:

(Note 1: This step is done only to understand the movement of Ackerman Arm of one side with respect to the movement of the other Ackerman Arm.)

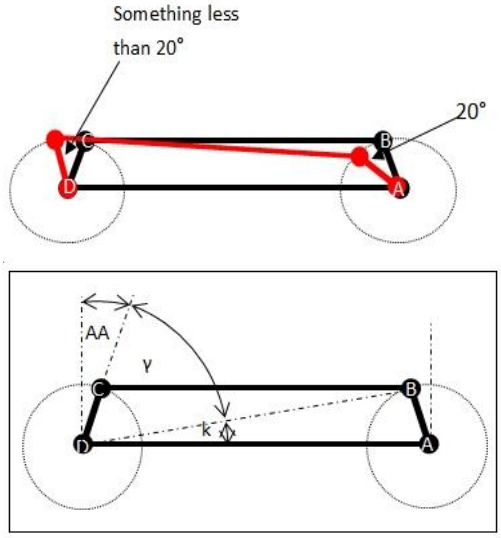

Fig 17: Ackerman Coordinates

(Note 2: Here we are considering that the car is taking right turn; hence the calculations are done accordingly)
Assigning point A = (0, 0)
And point D = (Kingpin C – C Distance, 0) = (51, 0)
Now, calculating the co-ordinates of point B,

$$\text{Point B's X coordinate} = R_{AA}\left[\cos(AA + SA_L)\right]$$

$$\text{Point B's Y coordinate} = R_{AA}\left[\sin(AA + SA_L)\right]$$

Where, R_{AA} is the Ackerman Arm Radius

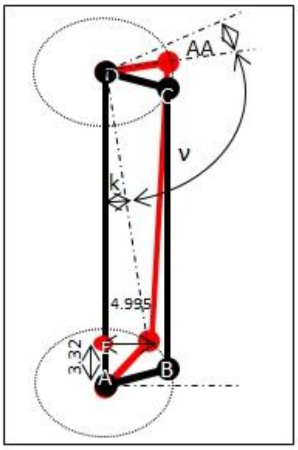

Fig 18: Other Angles

AA is the Ackerman Angle
SA_L is the steering angle of the left wheel. Zero degrees are straight ahead.
Positive values are a left turn; negative values are a right turn.
Plugging in our numbers for a 20° left turn

$$\text{Point B's X coordinate} = 6\left[\cos(13.64° + 20°)\right]$$

$$\text{Point B's Y coordinate} = 6\left[\sin(13.64° + 20°)\right]$$

$$\text{Point B's X coordinate} = 4.995$$

$$\text{Point B's Y coordinate} = 3.323$$

Therefore, Co-Ordinates of B = (4.995, 3.323)
We can project straight to the left of point B and straight up from point A to create a new point called point E.
Also, because point E falls on segment AD, we can calculate distance ED with the formula:

ED = AD – AE
ED = 51” – 3.323
ED = 47.677”

Now that we know EB and ED, we can find the length of BD because it is a hypotenuse of the triangle formed. Using Pythagorean Theorem:

$$BD = \sqrt{EB^2 + ED^2}$$

$$BD = \sqrt{(4.995^2 + 47.677^2}$$

BD = 47.93”

Furthermore, because we know the sides of the triangle we can determine angle k in the following manner:

$$\tan k = EB/ED$$

$$k = (\tan^{-1})\,(EB/ED)$$

k = 5.980°

- Calculating value of γ

We know that side DC is the length of the Ackerman arm, which we chose to be 6”. We know that side CB is the length of the tie rod, which we calculated earlier to be 48.17”. Finally, we know the distance BD, which we determined using Pythagorean Theorem to be 47.93”. From Law of Cosines,

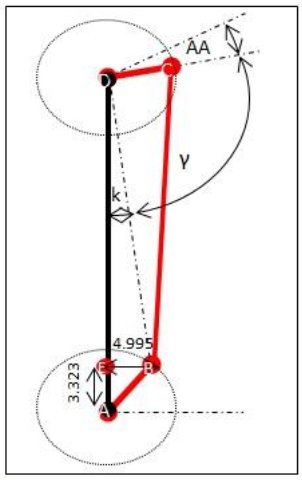

Fig 19: Steer Angle

$$\cos\gamma = \frac{A^2 + B^2 - C^2}{2AB}$$

Rearranging gives:

$$\cos^{-1}\left(\frac{A^2 + B^2 - C^2}{2AB}\right) = \gamma$$

$$\cos^{-1}\left(\frac{47.93^2 + 6^2 - 48.17^2}{2 \times 47.93 \times 6}\right) = \gamma$$

$$\gamma = 88.71°$$

Now if we add up angle k, γ and the Ackerman angle, we’ll have
the wheel’s steer angle from the line that connects the two kingpins.
To get the steer angle, we have to subtract 90°

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Steer Angle = k + γ + Ackerman Angle – 90°
Steer Angle = 5.980° + 88.71° + 13.64° - 90°

Steer Angle = 18.33°

This steer angle is related to Ackerman Angle, assuming that when left Ackerman Arm (AB) turns 20° the transect arm (CD) turns 18.33°. This gives the approximate relation between the angles of turning of the wheels.

4. Minimum Number of Teeth to avoid Interference

For Pinion,
The number of teeth on the pinion (T_p) in order to avoid interference may be obtained from the following relation:

$$T_p = \frac{2A_w}{G\left[\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1\right]}$$

Where,
A_w = Fraction by which the standard addendum for the wheel should be multiplied
G = Gear ratio or velocity ratio = T_R / T_P
T_R = Number of teeth on rack = 26
T_{PF} = Number of teeth on front pinion = 20
T_{PR} = Number of teeth on rear pinion = 16
ϕ = Pressure angle = 20°
Thus, substituting these values in the above equation,

$$T_{PF} = \frac{2\times 1}{1.3\left[\sqrt{1+\frac{1}{1.3}\left(\frac{1}{1.3}+2\right)\sin^2 20^0} - 1\right]}$$

$$T_{PF} = 13.071 \approx 14$$

$$T_{PR} = \frac{2\times 1}{1.625\left[\sqrt{1+\frac{1}{1.625}\left(\frac{1}{1.625}+2\right)\sin^2 20^0} - 1\right]}$$

$$T_{PR} = 13.66 \approx 14$$

Therefore,
Minimum number of teeth on front and rear pinion to avoid interference is = 14

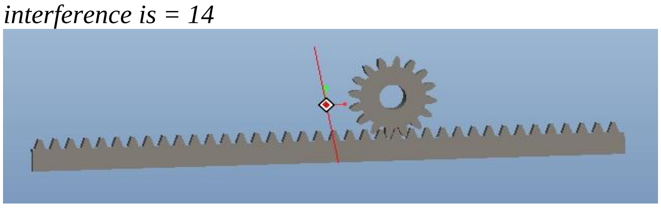

Fig 20: Rack and Pinion Assembly

5. Linear Displacement of Rack for one Rotation of Pinion

For Front Rack,
Linear Displacement of Rack for 1 rotation of front pinion (LD1PF)

$$LD1PF = \pi m z_p = \pi\,(1.25)(20)$$
$$LD1PF = 78.53mm = 3.1"$$

Therefore,
Linear Displacement of Front Rack for one rotation of pinion is = 3.1"

For Rear Rack
Linear Displacement of Rack for 1 rotation of rear pinion (LD1PR)

$$LD1PR = \pi m z_p = \pi\,(1.25)(16)$$
$$LD1PR = 62.832mm = 2.474"$$

Therefore,
Linear Displacement of Rear rack for one rotation of pinion is = 2.474"
(Note: The above calculated value of linear displacement of rear rack is applicable to both of the rear racks)

6. Bevel Gear Calculations

Sr. No	Particulars	Dimensions
1.	No of Teeth	20
2.	Pressure Angle	20°
3.	Tooth form	Straight
4.	Module	2.5mm
5.	Pitch Circle Diameter (PCD)	50mm
6.	Outer Diameter (OD)	55.22mm
7.	Face Width	15mm
8.	Overall Width	31.06mm
9.	Bore Diameter	12mm

Table III: Bevel Gear Specifications

Fig 21: Bevel Gear

7. Bevel Gear Calculations

In Force Analysis, it is assumed that the resultant tooth force between two meshing teeth of bevel gears is concentrated at the midpoint along the face width of the tooth.

(Note: These calculations are applicable to all the three Bevel Gears.)

- Beam Strength of Bevel Gears:

The size of the cross-section of the tooth of a bevel gear varies along the face width. In order to determine the beam strength of the bevel strength of the bevel gear, it is considered to be equivalent to a formative spur gear in a plane perpendicular to the tooth element.

$$\text{Cone Distance } (A_o) = \sqrt{(D_p/2)^2 + (D_g/2)^2}$$

$$A_o = \sqrt{(D_p/2)^2 + (D_p/2)^2}$$

D_p = D_g, considering same diameter for both driving and driven gear

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$$A_o = \sqrt{2D_p{}^2/4}$$

$$A_o = \frac{100}{\sqrt{2}}$$

Cone Distance (A_o) = 70.71mm

Therefore, Beam Strength of a Bevel Gear is,

$$S_b = mb\sigma_b Y\left(1 - \frac{b}{A_o}\right)$$

The above equation is known as Lewis Equation for Bevel Gears.

Where,

Material Type = Grey Cast Iron

D_p = Pitch Circle Diameter of driven gear = 50mm

D_g = Pitch Circle Diameter of driving gear = 50mm

S_b = Beam Strength of the tooth (N)

m = module at the large end of the tooth = 2.5mm

b = face width (mm) = $A_o/3$ = 15

σ_b = Permissible bending stress ($S_{ut}/3$) (N/mm^2) = 200MPa/3 = 66.67 N/mm^2

Y = Lewis Form Factor based on formative number of teeth = 0.320 (Ref. Design Data Book Pg. No. 8.53)

A_o = Cone Distance = 35.35

$$S_b = 2.5 \times 15 \times 66.67 \times 0.32\left[1 - \frac{15}{35.35}\right] = 460.56N$$

Therefore,

Beam Strength of Bevel Gear is 460.56N

- Wear Strength of Bevel Gears:

The contact between two meshing teeth of straight bevel gears is a line contact, which is similar to that in spur gears. In order to determine the wear strength, the bevel gear is considered to be equivalent to a formative spur gear in a plane which is perpendicular to the tooth at the large end. Applying Buckingham's equation to these formative gears,

$$S_w = \frac{bQD_pK}{\cos\gamma}$$

Where,

b = face width of gears =15mm

γ = Pitch angle = 45°

Q = Ratio factor

D_p = Pitch circle diameter of the pinion at the large end of the tooth (mm) = 50mm

K = material constant (N/mm^2)

z_g & z_p = 20

Ratio factor is calculated as:

$$Q = \frac{2z_g'}{z_g' + z_p'}$$

Where,

$$z_p' = \frac{z_p}{\cos\gamma}$$

$$z_g' = \frac{z_g}{\cos(90^0 - \gamma)}$$

Substituting these values,

$$Q = \frac{2z_g}{z_g + z_p \tan\gamma}$$

$$Q = \frac{2 \times 20}{20 + 20\tan 45^0}$$

Q = 1

Material Constant can be given as,

$$K = 0.16\left[\frac{BHN}{100}\right]^2$$

Where BHN = Brinell Hardness Number = 201 (Ref. Design Data Book Pg. 1.4)

Therefore,

$$K = 0.16\left[\frac{201}{100}\right]^2$$

Therefore,

Material Constant = 0.6464

Therefore,

Substituting the above calculated values in the equation shown below,

$$S_w = \frac{bQD_pK}{\cos\gamma}$$

$$S_w = \frac{15 \times 1 \times 50 \times 0.6464}{\cos 45^0}$$

$$S_w = 685.610N$$

Therefore,

Wear Strength of Bevel Gear = 685.610N

8. Bearing Calculations

These Bearings are placed between the two rear pinions, so as to support the easy movement of the pinion to switch the racks. According to the application in our project, We choose the 6003 type of Ball Bearing. (Ref. Table 15.5 Design of Machine Elements by V.B. Bhandari)

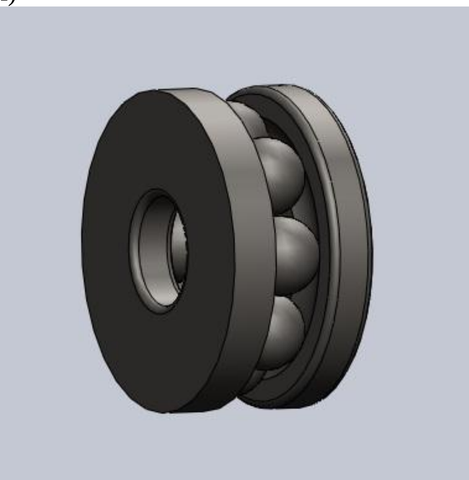

Fig 22: Ball Bearing

The expression for dynamic load is given by:

$$P = XF_r + YF_a$$

Where,
P = Equivalent dynamic load (N)
F_r = Radial load (N)
F_a = Axial or Thrust load (N)
D = 35mm
B = 10mm
C = 6050 N
C_o = 2800 N
F_r = 1000 N (Assumption on basis weight of casing, body weight, etc.)
F_a = 2000 N (Assumption on basis weight of casing, body weight, etc.)
Comparing the ratio of Radial and Axial load to get the value of X and Y

$$\frac{F_a}{F_r} = \frac{1000}{2000} = 0.5$$

For $(F_a/F_r) = 0.5$, e should be equal to 0.44
So, now

$$(F_a/F_r) > e, i.e., 0.5 > 0.44$$

Therefore,

$$X = 0.56, Y = 1$$

Thus,

$$P = 0.56(2000) + 1.4(1000)$$
$$P = 1120 + 1400$$
$$P = 2520 \text{ N}$$

Therefore,
Equivalent Dynamic Load Acting on Bearing is = 2520 N
Dynamic Load Capacity of a Bearing is given by:

$$C = P\,(L_{10})^{1/3}$$

Where,
L_{10} = Rated bearing life (in million revolutions)

$$6050 = 2800\,(L_{10})^{1/3}$$

L_{10} = 10.08 Million Revolutions
Therefore,
Rated Bearing life is 10.08 Million Revolutions

9. Honda Civics' Specifications

Particulars	Dimensions
Turning circle radius (R)	5.394m
Weight of car (W)	1250kg
Weight Distribution	60:40 (Front : Rear)
Wheelbase (L)	2.669m
Track width (t_w)	1.524m

Table IV: Civics' Specifications for Turning Circle Calculations

(Note: For the above rack & pinion and tie rod calculations we have to use the standard dimension as inches. While for the calculations of turning circle radius of car the standard dimension is in meter.)

10. Turning Circle Radius

To Calculate the Turning Circle Radius, we did the theoretical calculations then verified the Radius of all the Wheels and the Turning Circle Radius of the Car through our SOLIDWORKS draft.

- Calculation of Inside Lock Angle of Front Wheels (θ_{if})

By Ackerman Mechanism,

$$\sin(\alpha + \theta_{if}) = \frac{Y + X}{R}$$

Where,
α = Ackerman Angle = 13.64°
θ_{if} = Inside Lock Angle
Y = Arm Base = 1.415"
X = Linear Displacement of rack for one rotation of pinion
R = Ackerman Arm Radius = 6"

$$\sin(13.64^o + \theta_{if}) = \frac{1.415 + 3.1}{6}$$
$$\theta_{if} = 35.16^o$$

Therefore,
Inside Lock Angle of Front Wheel is = 35.16°

- Calculation of position of Centre of Gravity with respect to the rear axle
  From the benchmark vehicle (Honda Civic) we know that turning Radius is 5.394 m.
  We know that,
  $R^2 = a^2 + R_1^{\,2}$ .... (1)
  Where,
  R = Turning radius of the vehicle = 5.394m (Standard Specification of Civic)
  a_2 = Distance of CG from rear axle
  R_1 = Distance between instantaneous centre and the axis of the vehicle
  To find a_2
  $W_f = \frac{W \times a_2}{L}$ .... (2)
  Where,
  W_f = Load on front axle = 750kg (On basis weight distribution)
  W = Total weight of car = 1250kg
  L = Wheelbase = 2.669m
  Therefore,
  $a_2 = 1.60\text{m}$
  Substituting the value of a_2 in the above equation
  $R_1 = 5.15\text{m}$
- To find position of Instantaneous Centre from both the axles

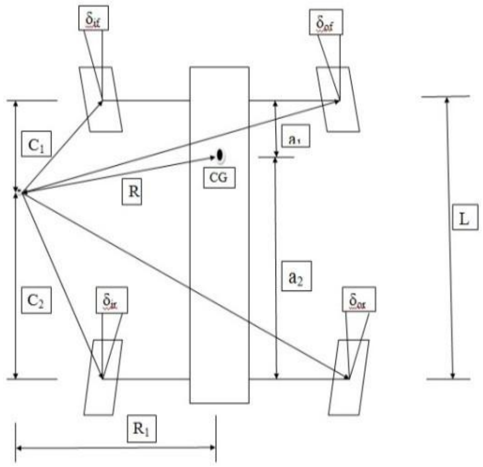

Fig 23: Turning Angles

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From our standard calculations of 2 Wheel Steering,

$\theta_{if} = 35.16^o$

$$\tan\theta_{if} = \frac{C_1}{R_1 - t_w/2} \quad \text{...........................} (3)$$

Where,

t_w = Front track width

θ_{if} = Inside Lock angle of front wheel

Therefore,

$$\tan 35.16^o = \frac{C_1}{5.15 - 0.762}$$

$C_1 = 3.09m$

$C_1 + C_2 = R$ .... (4)

Where,

C_1 = Distance of instantaneous centre from front axle axis

C_2 = Distance of instantaneous centre from rear axle axis

Therefore,

$C_2 = 5.394 – 3.09$

$C_2 = 2.304m$

Therefore, from equation (3) and (4)

$C_1 = 3.09m$

$C_2 = 2.304m$

- To find the remaining lock angles

To find $\emptyset_{of}$ = Outer Angle of Front Wheel

$\tan\emptyset_{of} = [C_1 / (R_1 + t_w/2)]$.................. (5)

$\tan\emptyset_{of} = 3.09/(5.15 + 0.762)$

$\emptyset_{of} = \tan^{-1}[3.09/(5.15 + 0.762)]$

$\emptyset_{of} = 27.59^o$

To find θ_{ir} = Inner Angle of Rear Wheel

$\tan\theta_{ir} = [C_2 / (R_1 - t_w/2)]$.................. (6)

$\tan\theta_{ir} = [2.304 / (5.15 - 0.762)]$

$\theta_{ir} = \tan^{-1}[2.304/(5.15 - 0.762)]$

$\theta_{ir} = 27.70^o$

To find $\emptyset_{or}$ = Outer Angle of Rear Wheel

$\tan\emptyset_{or} = [C_2 / (R_1 + t_w/2)]$.................. (7)

$\tan\emptyset_{or} = [2.304 / (5.15 + 0.762)]$

$\emptyset_{or} = \tan^{-1}[2.304 / (5.15 + 0.762)]$

$\emptyset_{or} = 21.29^o$

- Now considering the same steering angles for front and rear tires, we reduce in the turning radius of the vehicle but keeping the wheelbase and track width same as the benchmark vehicle.

- Calculations for turning radius for same steering angle

To find turning radius, R

$R^2 = a_2^{\,2} + L^2(\cot^2\delta)$ .... (8)

Where, δ = Total steering angle of the vehicle

To find δ

$$\cot\delta = \frac{\cot\theta + \cot\emptyset}{2} \quad \text{...........................} (9)$$

Where,

θ = total inner angle of the vehicle

ϕ = total outer angle of the vehicle

Therefore,

$$\cot\delta = \frac{\cot(35.16^o + 27.70^o) + \cot(27.59^o + 21.29^o)}{2}$$

Thus, $\cot\delta = 0.692$

Therefore, substituting the above values in equation (8)

R = 2.44 m

We put this above value of R in equation (1), to get the new value of R_1, i.e.

$R^2 = a^2 + R_1^{\,2}$

R_1 = 1.84m (For the new value of R)

Considering the turning radius as 2.44m,

Further calculation for C_1 and C_2 from equation (3) and (4)

$$\tan\theta_{if} = \frac{C_1}{R_1 - t_w/2}$$

$$C_1 + C_2 = R$$

$C_1 = 0.759m$

$C_2 = 1.681m$

Therefore, considering the new values of C_1 and C_2, we find that the inside and outside lock angle of front and rear wheels is as follows:

Thus, re-substituting the new values of C_1 and C_2 in equation (3), (5), (6), (7) to get the final values of Inside and Outside Angles, this is as follows:

$$\tan\theta_{if} = \frac{C_1}{R_1 - t_w/2}$$

$\tan\emptyset_{of} = [C_1 / (R_1 + t_w/2)]$

$\tan\theta_{ir} = [C_2 / (R_1 - t_w/2)]$

$\tan\emptyset_{or} = [C_2 / (R_1 + t_w/2)]$

$\theta_{if} = 35.16^o$ (Inside Lock Angle of Front Wheel)

$\emptyset_{of} = 16.98^o$ (Outside Lock Angle of Front Wheel)

$\theta_{ir} = 57.32^o$ (Inside Lock Angle of Rear Wheel)

$\emptyset_{or} = 32.86^o$ (Outside Lock Angle of Rear Wheel)

Therefore,

$\theta = \theta_{if} + \theta_{ir}$

$\theta = 35.16^o + 57.32^o = 92.48^o$

(Total Inner Angle of the Vehicle)

$\emptyset = \emptyset_{of} + \emptyset_{or}$

$\emptyset = 16.98^o + 32.86^o = 49.84^o$

(Total Outer Angle of the Vehicle)

From our SOLIDWORKS draft we find the following values of Radius of All Wheels:

Radius of inner front wheel (R_{if}) = 1.426m

Radius of outer front wheel (R_{of}) = 2.813m

Radius of inner rear wheel (R_{ir}) = 2.185m

Radius of outer rear wheel (R_{or}) = 3.264m

Considering the above values we drafted a SOLIDWORKS part modelling of Ackerman Steering Mechanism of our benchmark vehicle (Honda Civic) and we found that the Turning Circle Radius of our vehicle is reduced to 1.84m.

Therefore,

$$\cot\delta = \frac{\cot 92.48^o + \cot 49.84^o}{2}$$

$$\cot\delta = 0.400$$

Therefore, substituting the above value in equation (8)

R = 1.92 m

Thus,

The Turning Circle Radius of whole car = 1.92m

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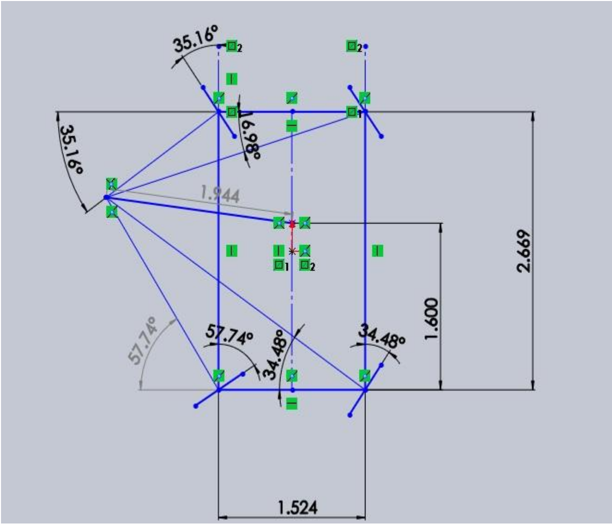

Fig 24: Turning Circle Radius drafted in SOLIDWORKS

Hence, our calculated value matches with the value obtained from the draft of Ackerman Steering Mechanism created in SOLIDWORKS. Hence verified!
Thus here we can see that the original Turning Circle Radius of 5.394m is reduced to 1.92m, i.e., the total reduction in Turning Circle Radius of the car is 64.4%.

- **Calculation of Steering Ratio**

Steering Ratio of car is calculated by the following formula:

$$R = \frac{s}{\sqrt{2 - 2\cos\left({}^{2a}/_{n}\right)}}$$

Where,
R = radius of curvature (same as units of wheelbase) = 1.92m = 75.59”
s = wheelbase = 105.1”
a = steering wheel angle = 360^{o} (assumed for one rotation of steering wheel)
n = steering ratio (Eg. for 16:1 its 16)

$$75.59 = \frac{105.1}{\sqrt{2 - 2\cos {}^{720}/_{n}}}$$

$$1.39 = \sqrt{2 - 2\cos {}^{720}/_{n}}$$

$$1.9321 = 2 - 2\cos {}^{720}/_{n}$$

$$0.069 = 2\cos {}^{720}/_{n}$$

$$\cos {}^{720}/_{n} = 0.034$$

$$\frac{720}{n} = 88.051$$

$$n = 8.177$$

Thus, the steering ratio of our car is 8.177:1, i.e. for 8.177^{o} of rotation of steering wheel the tire is turned by an angle of 1^{o}. Thus from the above obtained value of Steering Ratio, we can conclude that driver has to apply less effort to turn the car, giving much better maneuverability and control on the car.

V. CASING

Casing is the outer covering of components mainly used to protect the components from the damage by the external forces acting on them. It also holds the assembly together and maintains the proper engagement between the parts of the assembly. Casings are usually made from steel, and produced by the process of casting, in order to obtain the desired shape of casing. In this project we have used two casings:-

- **Bevel Gears Casing**

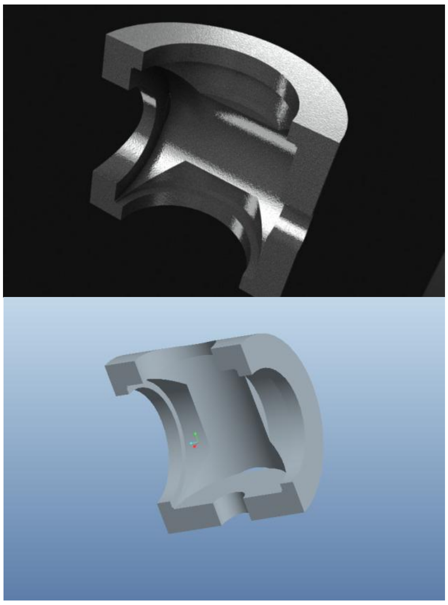
Fig 25: Bevel Gear Casing Half Section

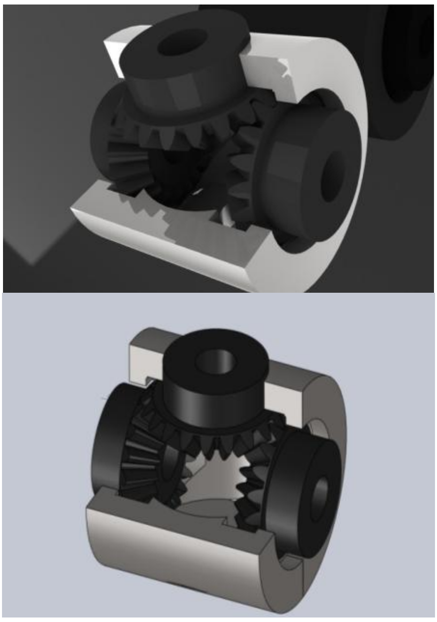
Fig 26: Bevel Gear Casing Assembly

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Bevel gear casing is used in order to protect the bevel gear assembly from forces. It also holds the three bevel gears in its proper position and maintains the engagement between the teeth of bevel gears. It is cylindrical in shape and it is made of steel. It is produced by the process of casting.

- **Double Rack and Pinion Casing**

Double rack and pinion casing is made in order to:-

1) Support the pinion over the rack and constraint is motion sideways.
2) It also helps in engagement and disengagement of pinion over the rack.
3) It protects the whole DRRC system from the external forces which are acting on them.
4) It allows rotational motion of the telescopic shaft properly.

It is made of steel. And it is produced by the process of casting.

VI. ANALYSIS OF COMPONENTS

Analysis is process of analyzing the components by applying external factors such as loads, temperature, pressure etc. and obtaining the values such as stresses (bending, tangential and normal), deformations etc. in order to determine the safety of the components when implemented in practical use. It gives optimum result of the safety of components and very easy to understand various factors applicable in the process.

These Analyses gives optimum result of safety of components and minimize the chances of failure. There are various packages in market to carry out these simulations on computer such as ANSYS, HYPERWORKS, and FLOTRAN etc.

In this project we have used ANSYS 14.0 as the software to analyze the safety of our components under various loading conditions.

Two major analyses carried out in this project are:

1) Deformation analysis
2) Stress analysis

Various components analyzed in this project are:

1) Bevel gear (top surface)
2) Bevel gear (side)
3) Roller bearing
4) Telescopic shaft
5) Rack and pinion system
6) Spindle
7) Bevel gear casing
8) Double rack /pinion casing

Process for performing the analysis:

1) Making or importing the geometry to software interface (GUI).
2) Defining the field.
3) Applying the material properties.
4) Meshing the components with appropriate element size.
5) Applying the actions such as load, pressure etc. on the body.
6) Applying the boundary conditions such as fixed supports (constraints).
7) Solving using the solver.
8) Obtaining required reactions such as stresses deformations etc.

- **Bevel Gear Analysis**

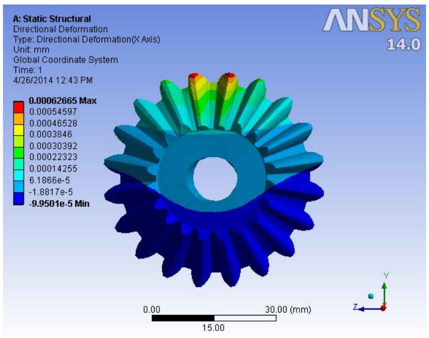

Fig 28: Bevel Gear on top deformation

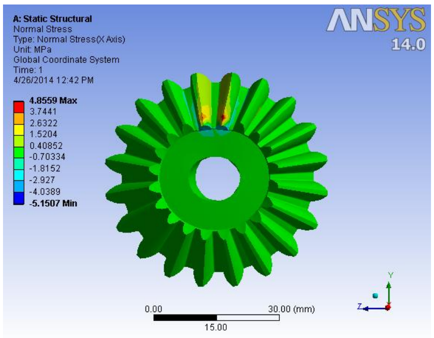

Fig 29: Bevel gear on Top Normal Stress

Material Type: Grey Cast Iron
Ultimate Tensile Strength = 297 MPa
Maximum Stress Obtained = 4.855 MPa
Factor of Safety = 297/4.855 = 61.174
Design Completely Safe
Maximum Deflection = $6.2 * 10^{-4}$mm

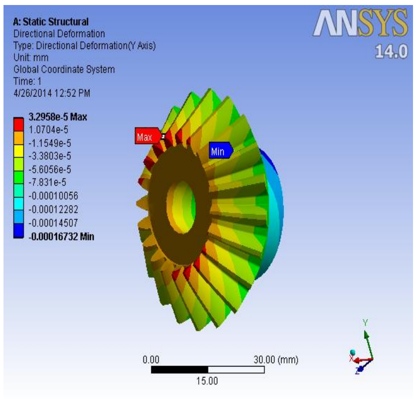

Fig 30: Bevel Gear Side Deformation

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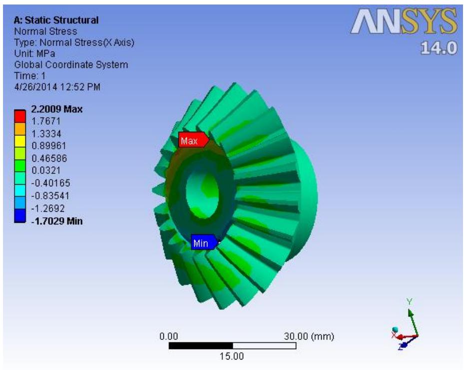

Fig 31: Bevel Side Normal Stress (Bending)

Material Type: Grey Cast Iron
Ultimate Tensile Strength = 297 MPa
Maximum Stress Obtained = 2.2009 MPa
Factor of Safety = 297/2.2009 = 134.94
Design Completely Safe
Maximum Deflection = 3.295 * 10^{-5}mm

- **Ball Bearing Analysis**

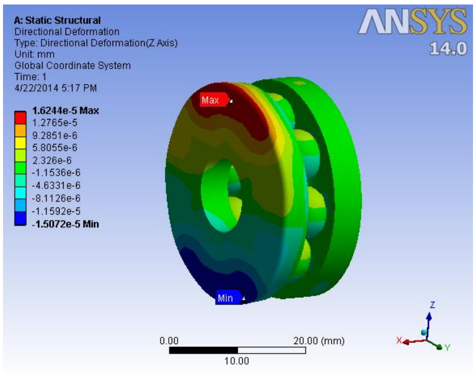

Fig 32: Ball Bearing Deformation

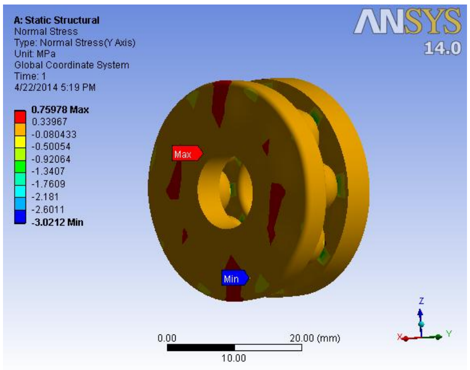

Fig 33: Ball Bearing Normal Stress

Material: Mild Steel (AISI 1020)
Ultimate Tensile Strength = 394MPa
Maximum Stress Obtained = 0.75 MPa
Factor of safety = 394/0.75 = 525.33
Design Completely Safe
Maximum Deflection = 1.6 * 10^{-5}mm

- **Telescopic Shaft Analysis**

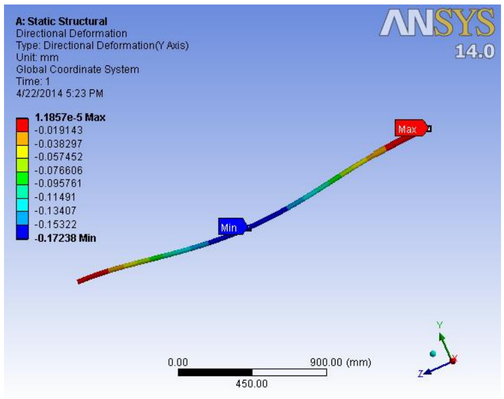

Fig 34: Deformation of Telescopic Shaft

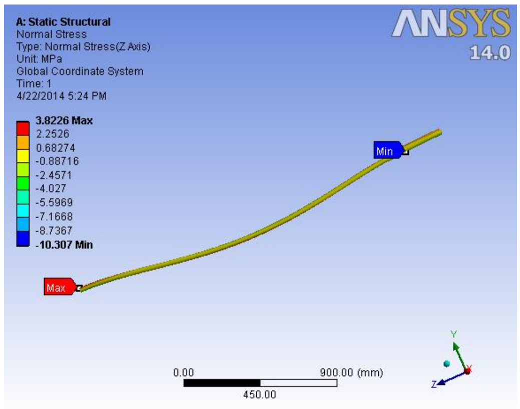

Fig 35: Normal Stress in Telescopic Shaft

Material: Mild Steel (AISI 1020)
Ultimate Tensile Strength = 394 MPa
Maximum Stress Obtained = 3.82 MPa
Factor of safety = 394/3.82 = 103.41
Design Completely Safe
Maximum Deflection = 1.18 * 10^{-5}mm

- **Pinion Analysis**

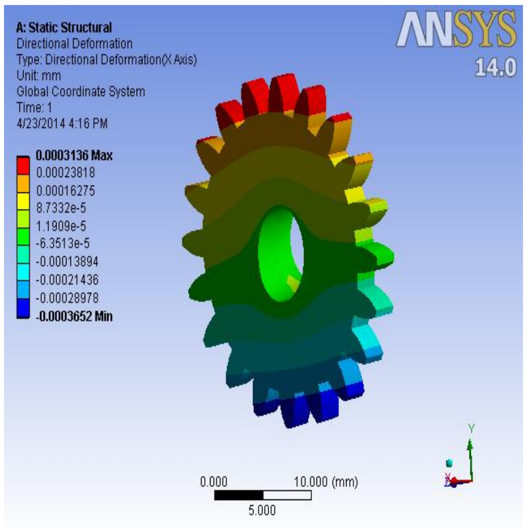

Fig 36: Pinion Deformation

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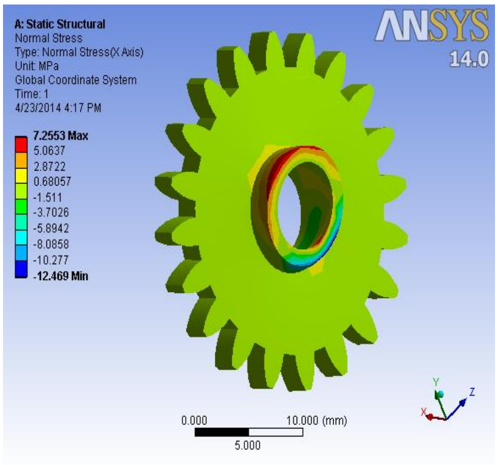

Fig 37: Pinion Stress

Material: Grey Cast Iron

Ultimate Tensile Strength = 297 MPa
Maximum Stress Obtained = 7.25 MPa
Factor of safety = 297/7.25 = 40.96
Design Completely Safe
Maximum Deflection = 3.136 * 10^{-4}mm

- **Rack Analysis**

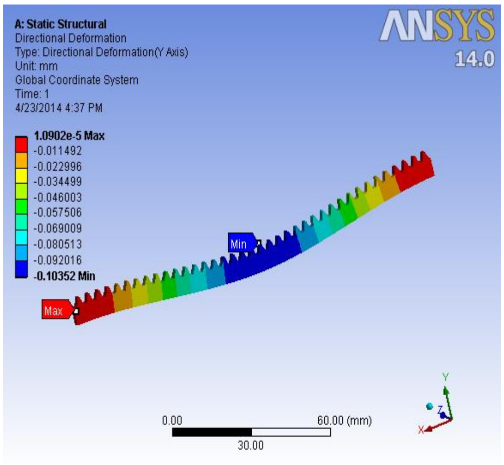

Fig 38: Rack Deformation

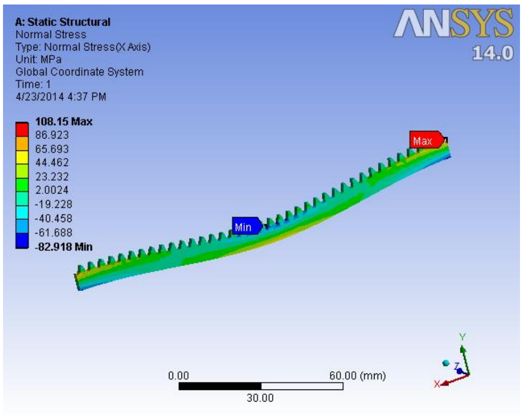

Fig 39: Rack Stress

Material: Grey Cast Iron
Ultimate Tensile Strength = 297 MPa
Maximum Stress Obtained = 108.15 MPa
Factor of safety = 297/108.15 = 2.746
Design is Safe
Maximum Deflection = 1.0902mm

- **Spindle Analysis**

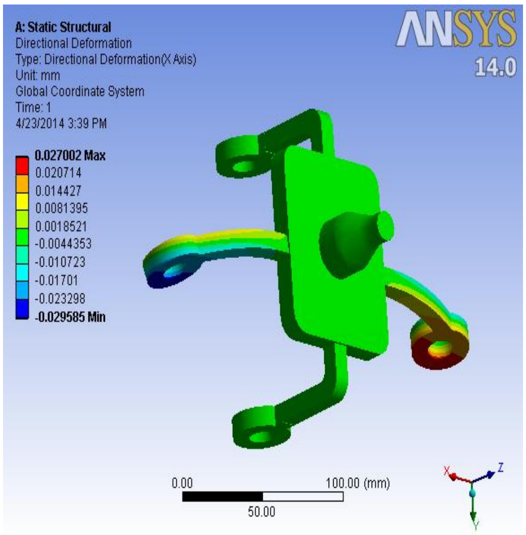

Fig 40: Spindle Directional Deformation

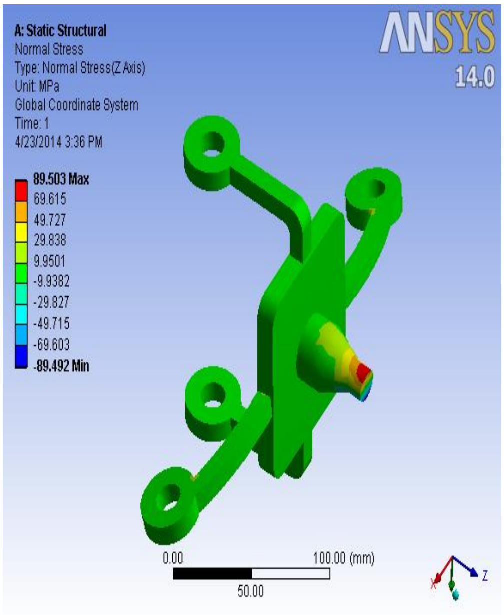

Fig 41: Spindle Normal Stress

Material: Mild Steel (AISI 1020)
Ultimate Tensile Strength = 394 MPa
Maximum Stress Obtained = 89.5 MPa
Factor of safety = 394/89.5 = 4.4
Design is Safe
Maximum Deflection = 0.027mm

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VII. DESIGN FAILURE MODE EFFECT ANALYSIS

	COMPONENTS	FUNCTION	POTENTIAL FAILURE MODE	POTENTIAL EFFECTS OF FAILURE	SEV	POTENTIAL CAUSES	OCC	CURRENT DESIGN CONTROLS	DET	RPN	RECOMMENDED ACTIONS	SEV	OCC	DET	RPN
2	RACK AND PINION	To convert rotational motion from steering wheel into linear displacement	Interference of teeth	Wear and Tear of parts resulting in damage of system	9	Improper meshing of gears	1	Accurate design following the standards	2	18	Number of teeth on pinion should be more than the calculated theoretical value	2	1	1	2
3	BEVEL GEARS	To transfer rotational motion from steering wheel to pinion in front & rear	Interference of teeth	Wear and Tear of parts resulting in damage of system	9	Improper meshing of gears	1	Accurate design following the standards	2	18	Design, manufacture and assemble the bevel gears to the best precision	2	1	1	2
4	CASING OF BEVEL GEAR	To protect & support the bevel gears	1) Breaking of casing 2)Corrosion	1) Damage of bevel gear assembly 2) Improper meshing	6	1)Improper selection of material 2) Improper assembly of casing	1	Quality control of casing	8	48	Proper selection of material , periodical checking, proper lubricating	4	1	4	16
5	TELESCOPIC SHAFT	To displace the rear pinions upto rear racks	Malfunctioning of 2-way control check valve	1) Improper displacement of inner shaft 2)Rear racks stops functioning	9	1) Improper selection of valve 2) Improper fitting of valve and parts of shaft	1	Hydraulic circuit directional control system	5	45	1)Designing layout of hydraulic circuit directional control system 2)checking of valve in quality control	3	1	3	9
6	DOUBLE REAR RACK SYSTEM	To turn the rear wheels in counter phase & inphase with front wheel	1)Malfunctioning of rack displacement 2)Malfunctioning of both pinions	1) Undesirable turning of rear wheels 2) accidents due to high speed turning	9	1) Improper calculations of design of DRRC	3	Design of DRRC	2	54	1) Proper design of DRRC	2	1	1	2
7	DOUBLE PINION CASING	To protect & support both pinions and bearings on rear side	1) Breaking of casing 2)Corrosion	1) Damage of pinion and bearing assembly 2) Improper meshing of pinions with respective racks	6	1)Improper selection of material 2) Improper assembly of casing	1	Quality control of casing	8	48	Proper selection of material , periodical checking, proper lubricating	4	1	4	16
8	SPINDLE	To Move the wheels as per lock angles & displacement of spindle	1) Excessive stress on spindle	1) Improper analysis of design	5	Nasty jerks	1	Quality ontrol of analysis	8	40	Proper analysis of design	2	1	3	6
9	BEARING	To assist linear and rotational movement of both pinions at rear side	Improper movement of pinion on DRRC	Wear and Tear of pinion or other parts of casing	5	Improper fitting and assembly	1	Design layout and assembly of casing alongwith bearing	6	30	replace the bearing	1	1	3	3

Table V: DFMEA

VIII. COST ANALYSIS

	COMPONENTS		PINION (FRONT)	RACK (FRONT &REAR)	BEVEL GEARS	MIDDLE ROD	OUTER COVER OF TELESCOPIC SHAFT	INNER ROD OF TELESCOPIC SHAFT	PINION (REAR)	SPINDLE FITTING PART	SPINDLE MAIN PART	KNUCLE ARMS	
2	MATERIAL		cast iron	cast iron	cast iron	mild steel	stainless steel (ss304)	stainless steel (ss304)	cast iron	mild steel	mild steel	mild steel	
3	RAW MATERIAL DIMENSIONS	OD (mm)	120	180	150	30	30	20	22.5	20	0	0	
4		WIDTH (mm)	20	20	70	250	350	250	20	60	20	150	
5		FACE WIDTH (mm)	20	20	30	-	-	-	20	-			
6		RADIUS m)	0.06	0.09	0.075	0.015	0.015	0.01	0.01125	0.01	0	0	
7		AREA	0.011304	0.025434	0.0176625	0.0007065	0.0007065	0.000314	0.000397406	0.000314	0.007872	0.007872	
8		mass	0.0887364	0.1996569	0.138650625	0.005546025	0.005546025	0.0024649	0.003119639	0.0024649	0.0617952	0.0617952	
9		WEIGHT (KGS)	1.774728	3.993138	9.70554375	1.38650625	1.94110875	0.616225	0.062392781	0.147894	1.235904	9.26928	
10	rate of material		75	75	75	85	250	250	75	85	85	85	
11	cost of raw material		133.1046	299.48535	727.9157813	117.8530313	485.2771875	154.05625	4.679458594	12.57099	105.05184	787.8888	
12	MACHING COST		35.49456	79.86276	194.110875	27.730125	38.822175	12.3245	1.247855625	5.91576	24.71808	185.3856	
13	HARDENING COST		79.86276	179.69121	436.7494688	62.39278125	87.34989375	27.730125	2.807675156	8.87364	74.15424	556.1568	
14	GEAR CUTTING	MODULE	1.25	2	5	0	0	0	1.25	0	0	0	
15		NO.OF TEETH	20	26	20	0	0	0	16	0	0	0	
16		FACE WIDTH (mm)	20	20	30	0	0	0	20	0	0	0	
17		FACE WIDTH (inch)	0.787401575	0.787401575	1.181102362	0	0	0	0.787401575	0	0	0	
18		rate	0.75	1.25	2	0	0	0	0.75	0	0	0	
19	total cost of gear cutting		14.76377953	51.18110236	236.2204724	0	0	0	11.81102362	0	0	0	
20	TOTAL PER COMPONENT		263.2256995	610.2204224	1594.996597	207.9759375	611.4492563	194.110875	20.546013	27.36039	203.92416	1529.4312	
21	NO.OF COMPONENTS		1	3	3	1	1	1	2	2	2	4	TOTAL COST OF ALL EXTRA PARTS
22	TOTAL		263.2256995	1830.661267	4784.989792	207.9759375	611.4492563	194.110875	41.09202599	54.72078	407.84832	6117.7248	14513.79875

Table VI: Cost Analysis

The cost analysis was carried out by exploring rates of raw materials required, machining required and manufacturing processes required to build the entire assembly which serves our concept of 4 wheel steering.

Entire design model was classified into three parts:

1. Bevel gear assembly
2. Middle portion including telescopic shaft
3. Double rear rack

Each of this parts had individual components and cost regarding each components was quoted and analysis was carried out by comparative calculative of each components based on common criteria.

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The rates are considered as highest available price as per the local market keeping in mind the factor of safety on higher side. The final resultant cost is the maximum possible cost of each component, since higher limit of rates are considered.

IX. ADVANTAGES

- The vehicle's cornering behavior becomes more stable and controllable at high speeds as well as on wet or slippery road surfaces.
- The vehicle's response to steering input becomes quicker and more precise throughout the vehicle's entire speed range.
- The vehicle's straight-line stability at high speeds is improved.
- Negative effects of road irregularities and crosswinds on the vehicle's stability are minimized.
- The vehicle is less likely to go into a spin even in situations in which the driver must make a sudden and relatively large change of direction.
- By steering the rear wheels in the direction opposite the front wheels at low speeds, the vehicle's turning circle radius is greatly reduced. Therefore, vehicle maneuvering on narrow roads and during parking becomes easier.

X. DISADVANTAGES

Following are the places and positions where there are chances of failure of rear wheel steering system:

1) Car while turning at speed of 50kmph suddenly reduces its speed to 30kmph there is transition from in-phase to out-phase steering. Since the car is turning there is also possibility of pinion stuck between two racks inside casing. Then for that instance car will become two wheel steering but this will not have any effect on front wheels and thus will not cause any damage or accident.

2) Pump and Sensors should be checked regularly to avoid its failure.

XI. CONCLUSION

As per the focus of the project we have created an innovative 4 wheel active steering mechanism which is feasible to manufacture, easy to install and highly efficient in achieving in-phase and counter-phase rear steering with respect to the front wheels using DRRC.

This system assists in high speed lane changing and better cornering. It combats the problems faced in sharp turning. It reduces the turning circle radius of the car and gives better maneuverability and control while driving at high speeds, thus attaining neutral steering.

Moreover components used in this system are easy to manufacture, material used is feasible, reliable and easily available in market. The system assembly is easy to install and light in weight and can be implemented in all sections of cars efficiently.

XII. FUTURE SCOPE

Having studied how 4WS has an effect on the vehicle's stability and driver maneuverability, we now look at what the future will present us with. The successful implementation of 4 Wheel Steering using mechanical linkages & single actuator will result in the development of a vehicle with maximum driver maneuverability, uncompressed static stability, front and rear tracking, vehicular stability at high speed lane changing, smaller turning radius and improved parking assistance. Furthermore, the following system does not limit itself to the benchmark used in this project, but can be implemented over a wide range of automobiles, typically from hatchbacks to trucks. This coupled with an overhead cost just shy of Rs. 15,000 provides one of the most economical steering systems for improved maneuverability and drivers' ease of access. With concepts such as "ZERO TURN" drive as used in 'Tata Pixel' and "360^{o} Turning" used in 'Jeep Hurricane', when added to this system, it will further improve maneuverability and driver's ease of access.

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