Design Methodology of Parallel Steering System for Light Weight off Road Vehicles

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Abstract – "Design Methodology of Parallel Steering System for Light Weight off Road Vehicles" is to ensure the most efficient steering geometry selection for light weight off road vehicles. In this process various parameters are kept in mind for selecting most efficient steering geometry for the vehicle. The Steering system uses Rack and Pinion steering gearbox assembly along with Parallel geometry. In this steering system self-designed column assembly is used with rack and pinion gearbox. The steering wheel is designed with aim to keep driver efforts and weight of the system as minimum as possible. The rack and pinion gearbox is connected to steering arm with the help of tierods.Tierods are designed and analyzed for their load carrying capacity.

Key Words: Off Road Vehicles, Parallel geometry, Rack and pinion gearbox,tierods,steering arm

1.INTRODUCTION

Steering is one of the systems that connects driver to the vehicle. What if we have a vehicle but there is no mechanism to give direction to it, in such a case our vehicle will wander aimlessly and it is certainly going to hit something. Steering system helps driver to steer front wheels and provide directional control. However the actual steering angles are modified by steering geometry and the suspension geometry of the vehicle.

The rack and pinion gearbox system is used widely in passenger cars and light weight vehicles because it is simple, light and responsive system. It occupies less space and uses less number of linkages. The gearbox provides the numerical reduction between the rotational input from the steering wheel and rotational output of the wheels. The steering ratio i.e., the steering wheel angle to road wheel angle for commercial vehicles is around 20:1 and it varies upto 4:1 for racing cars. The rack and pinion system has fixed steering ratio and the output of it is only changed by changing the geometry of the system.

The lateral acceleration produced by the gearbox is passed through the tierods to steering arms on the left and right wheels and the vehicle turns according the steering geometry.

1.1 Selection of Parallel Steering Geometry

As we all know generally there are three types of steering geometry that we use namely:-

- Ackermann
- Anti Ackermann
- Parallel steering

Starting with the first one, when the vehicle is moving very slowly and 'free of lateral forces', it will only corner precisely when the verticals drawn in the middle of all four wheels meet at one point - the center of the bend. If the rear wheels are not steered, the verticals on the two front wheels must intersect with the extension of the rear axle center line at center of the bend whereby different steer angles δ_i and δ_o , occur on the front wheels on the inside and outside of the bend. The nominal value δ_o of the outer angle can be calculated from the larger inner angle δ_i :

$$\text{Cot}\ \delta_o = \cot \delta_i + b/l$$

where l is the wheelbase and b is the wheel track. The geometry which satisfies the above conditions is known as Ackermann geometry. Theoretically, this gives the perfect cornering condition but in practical situation things are different. Ackermann works on this assumption that during turning the lateral acceleration of the vehicle is zero, so there is no formation of slip angle and hence no lateral force generated. Keeping this assumption completely ignores the slip angle consideration of the tire. However the actual story is completely different from this one. This led to the generation of another steering geometry which is Anti Ackermann.

In Anti Ackermann front wheel on the outside of the bend makes more angle than on the inside of the bend. It takes into consideration the optimum slip angle for the tire at the reduced and higher vertical load. As seen in the graph of lateral force VS slip angle, more slip angle is required to reach the peak lateral force at higher vertical load and less slip angle is required to reach peak lateral force at lower vertical load. At high lateral acceleration, all the tires operate at significant slip angle and load on the inside wheel is much less than on the outside wheel. So if we have Ackermann the inner wheel has to have more steering angle equal to more slip angle but has much less vertical load. Thus the inside front tire is forced to a higher slip angle than required for maximum lateral force. This drags the inside tire along at high slip angle(above the peak lateral force)raises the temperature and slows the car down due to lateral force induced drag which is given by

$D_{induced}=F_{yF}\sin\alpha_F$

Where F_{yF} is lateral force on front tires and α_F is the slip angle formed on front tires

With Anti Ackermann the inner wheel has less steering angle than the outer wheel but has more grip as compared to Ackermann. Also as the car starts to transfer weight in the corner, the outside wheel gains effectiveness in turning a car. This is due to the fact that as the outside wheel builds lateral force the relative advantage of the camber gain increases outside tyre grip even further.

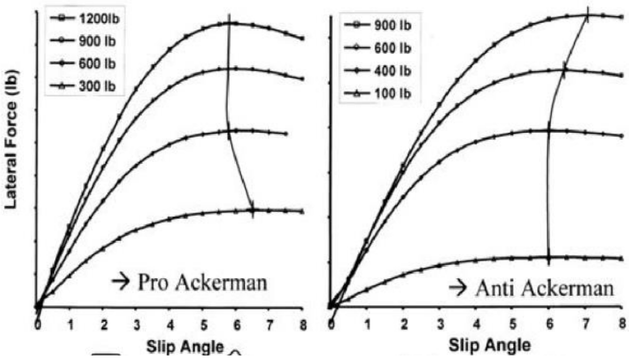

Figure-1: Lateral force vs slip angle

If we look at tire curve for anti-Ackermann, it is clear that for outer wheel(at higher load) when maximum cornering force is reached the curve flattens out. So it seems we do have a large window of slip angle where the outside tyre grip will be OK(as compared to Ackermann). This means the outside tyre particularly, can handle more slip angle variation, and still offer near maximum grip. Thus at mid corner, even though the toe angles might be pretty wild, we can have near maximum outside grip. Also the shape of the curve determines how the car will behave in high acceleration and grip limit situation. For example, If the curve has a sharp top, it indicates an abrupt change from transition into a skid, with little warning to the driver. Curves with almost flat top indicate a smoother transition, giving much more warning about the grip situation on the tyres. Thus in anti Ackermann curve it is clear it will give more skid warning to the driver as compared to Ackermann.

Inspite of the above positive factor, this can be too risky if proper data of tire lateral force and slip angle is not known. This is because of the fact that Anti Ackermann that does not satisfies proper cornering condition. The verticals drawn in the middle of all four wheels do not meet at one point - the center of the bend. Instead the verticals drawn from the center of both front wheel meet each other outside the bend. So if perfect load is not transmitted on the front wheels at any given lateral acceleration then maximum lateral force will not be achieved at the wheels and the vehicle will lose grip moving outside the bend.

Now coming back to the Ackermann condition, according to Ackermann geometry turning circle of a vehicle is the circle which the outer front wheel traces with the largest steering angle. The turning circle of a vehicle should be as small as possible to make it easy to turn and park. The formula for turning circle is,

$D_s=2(l/\sin\delta_o+r_\sigma)$

Where, l =wheelbase

δ_o = maximum outer wheel steered angle

r_σ = scrub radius

by using this formula it is clear that to get shortest turning circle we need less wheelbase and maximum outer wheel steered angle. This in turn requires an even greater steering angle applied to the wheel at the inside of the turning circle, though this is limited by the fact that the tyre should not come into contact with the wheel arch or any of the front-axle components. The inner angle δ_i is therefore limited, whereas the wheel angle on the outside (for functional reasons a smaller angle) is not. This may be the same size as the inner one. The disadvantage is that it impairs the cornering behavior of the vehicle , but with the advantage that the track circle becomes smaller and the lateral tire force capacity on the outside of the bend increases. For this reason, the outer steering angle should be larger, i.e. the actual value δ_o should be greater than the nominal angle δ_{oA} calculated according to Ackermann. A series of test measurements has shown that a reduction by Ds ≈ 0.1 m per 1° steering deviation can be achieved. This lead to generation of parallel steering where both the inner and outer wheel makes the same steer angle. But like anti ackermann this also doesn't satisfies the perfect cornering condition and leads to a lot of tire wear with time. So to get better results it would be more preferable if we keep a little less steer angle of outer wheel than inner wheel. In other words approx. parallel steer geometry would be the ultimate answer. For this reason and keeping in mind all the above factors we decided to have approx. parallel steer geometry. This can be achieved if the rack is moved forward from the line joining the outer point of tie rod such that the angle made between tie rod and steering arm (when the front wheels are not steered) is 90 deg

1.2 Methodology

The process followed for design of parallel steering geometry involves following steps.

1. Benchmarking the geometries of light weight vehicles
2. Defining the objective of new geometry of the vehicle.
3. Steering Geometry iterations.
4. Design Validation
5. Steering system parts modelling.
6. Steering system assembly.

2. Objective of the Geometry

- Turning radius: 1.65m
- Reduced steering ratio
- Less lock to lock rotation
- Lighter steering system
- Parallel steering geometry

3. Steering Geometry

The Parallel steering geometry is selected for the vehicle because this geometry enables the vehicle to take tight turns without under steer and less slippage of the vehicle while turning. According our geometry consideration the graphical representation of the anti-Ackermann geometry is as shown in the fig below:

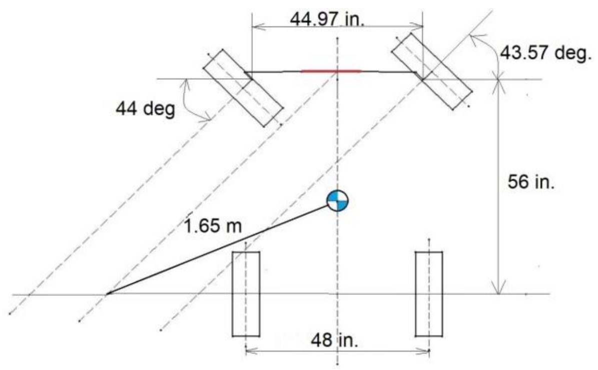

Figure-2: Parallel Steering geometry

After selecting all parameters like pivot to pivot distance, steering arm angle, steering arm length we draw geometry in CATIA software. From this we get inner and outer angle also turning radius. We have to give outer steered angle as 44° because as we are exceeding angle tyre shows skidding. Then with the help of suspension geometry we are able to decide tie rod length and position. After this we decided the pinion size and rack size.

Geometry in CATIA and simulation in LOTUS software

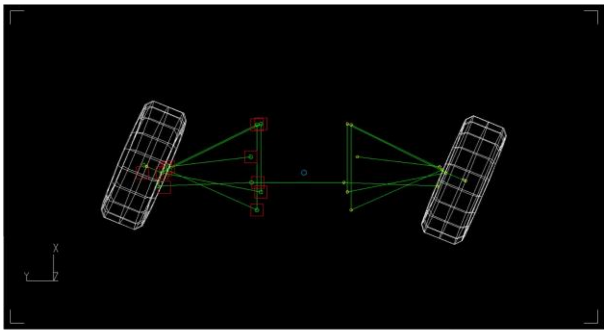

Figure-3: Top view of steering geometry in LOTUS

We are considering suspension geometry for designing the steering system. The various parameters like static camber, camber change rate, static toe and toe change rate and roll steer during dynamic conditions are to be considered. After doing so many iterations we decided to keep following values which give effective suspension system.

i) Steering arm length
ii) Steering arm angle

It was observed that for above listed parameters the dynamic toe change and camber change rate was negligible.

After doing so many iterations the geometry was finalised with following parameters:

Parameter	Value
Steering arm angle	90^0 (from horizontal)
Steering arm length	2.75 in
Inside locked angle	44°
Outside locked angle	43.57°
Turning radius	1.65m
Rack travel	46.5mm

Table-1: Steering geometry parameters

Design of the rack and pinion:

While designing the pinion we are considering following factors:

Parameter	Value
Module	1.5 mm
No. of teeth	25
Pressure angle (φ)	20^0
Addendum	1.5 mm
Dedendum	1.875 mm
Pitch circle diameter	37.5 mm
Tooth thickness	2.3562 mm
Tooth space	2.3562 mm
Addendum diameter	40.5 mm
Deddendum diameter	33.75 mm

Table-2: Rack and pinion parameters

4. Gear Calculations

Rack Travel : 46.5 mm , α (angle of rotation of steering wheel for one side) = 138.6° = 2.41 rad

p.c.r.(pitch circle radius) = 46.5/2.41

p.c.d.(pinion) = 38 mm

β (bending strength) = 440 / 3

= 146.67 MPa

Beam strength (b_m) = m*b*β*y(lewis factor)

=$(38/Z_p)$*10*$(38/Z_p)$*146.67*((0.484–$(2.865/Z_p)$) ………….(b = facewidth= 10*m(module))

= $(1025070.76/Z_p^2)$ - $(6067825.902/Z_p^3)$ ………………..(1)

(Z_p = no. of teeth's on pinion)

Service Factor = 1.5 = for Medium Loads

Velocity factor = C_v

V = $(\pi*d_p*n_p)$ / (60*1000) ……………………..(d_p = 38 mm(p.c.d) , n_p = no. of revolutions of pinion /min.=20 (assumption))

= 0.039

As velocity factor is less than 10,

C_v = 3 / (3+V)

= 3/(3+0.039)

= 0.98

P_t = ($2M_t$ / d_p) ………………(P_t = force transmitted along tangent to pitch circle while meshing)

M_t = F * r …………………(M_t = steering wheel torque

r = steering wheel radius = 130 mm)

= 80 * 130

= 10,400 N – mm

P_t = (2*10,400)/38

= 547.36 N

$P_{eff.}$ = (C_s / C_v)*P_t

= (1.5/0.98)*547.36

= 837.79 N

S_b = P_{eff} * f.o.s.

= 837.79 * 1.5 ……………………….(fos = 1.5 (assumption))

= 1256.69 N

From (1)…….,

$1256.69 = (1025070.76 / Z_p^2) - (6067825.902 / Z_p^3)$

$Z_p = -31.1 , 24.9 ,6.21$

Now, 24.9 is approx. = 25

M = 38 / 25

= 1.52 approx. = 1.5

$S_b = (1025070.76/25^2) - (6067825.902/25^3)$

= 1251.77 N

$S_b > P_{eff.}$

Hence , pinion is safe

Now , for Wear strength ,

$S_w = b*Q*d_p*k$

$= 10*1.5*38*0.16*(BHN/100)^2$ ………………[$Q = 2Z_g/(Z_p + Z_g)$. Assume = $Z_g = Z_p$]

$= 9.12*10^{-3}*(BHN)^2$

$P_{eff.} = C_sP_t + P_d$

$P_d = \{21V\ (C*e*b + P_t\ )\} / \{21V + (C*e*b + P_t\ )^{1/2}\}$ ………………………..(2)

$e= e_p + e_g$

$e_p = 8 + 0.63*\varphi_p$

$\varphi_p = 1.0 + 0.25*(38)^{1/2}$

= 2.56

$e_p= 8 + 0.63*2.54$

= 9.6

$e_p=e_g$

e = 19.2

$C = k / (1/E_p + 1/E_g)$ ………………………..(k = form factor for tooth)

K = 0.111 for 20° full depth. $E_p = E_g$ = modulus of elasticity

As pinion and rack having same material, $E_p = E_g$ = 70 GPa (Al 7075 – T6) = $70 * 10^3$ MPa)

$= 0.111/\{1/(70 *10^3) + 1/(70 * 10^3)\}$

= 3885

From (2)…….,

$P_d = \{21 * 0.039\ (3885 * 19.2 * 10 + 547.36)\} / \{21*0.039 + (3885 *19.2 *10 +547.36)^{1/2}\}$

= 706.93 N

$P_{eff.} = C_s*P_t + P_d$

= 1.5 * 547.36 +706.93

= 1527.97 N

$S_w = P_{eff.} * f.o.s$

$9.12*10^{-3} * (BHN)^2 = 1527.97 * 1.5$

BHN = 501.3

$S_w = 9.12 * 10^{-3} *(501.3)^2$

= 2291.87

$S_w > P_{eff.}$

Hence, Design is safe.

5. Wheel Alignment Parameters

Table-3: Wheel alignments

Parameter	Value
Camber	0deg
Caster	6deg
King pin inclination	7deg
Scrub radius	44 mm
Toe	0deg

6. UNDERSTEER GRADIENT CALCULATIONS

Considerations

b = 0.015 m (belt thickness)
E = 27*10^6 MPa (compression modulus of tire)
S= 0.15 mm (sidewall vertical deflection when loaded)
a = tire aspect ratio = 6/7 = 0.8571
r = wheel radius = 11 in. = 0.2794 m
C= cornering stiffness
C_f = cornering stiffness at front
C_r = cornering stiffness ar rear
W_f = weight on front = 85.5 kg
W_r = weight on rear = 104.5 kg
W = belt width = 8 in. = 0.2032 m

Cornering Stiffness =

$$C = \frac{2Ebw^3}{(r+wa)^2 \sin\left\{acos\left[\frac{1-swa}{r+wa}\right]\right\}\left\{\pi - \sin\left(acos\left[\frac{1-swa}{r+wa}\right]\right\}}$$

Cornering stiffness after putting values is,

C = 821.9 N/deg

6.1 Understeer Gradient of tires:

Understeer gradient of tires = $K_{us} = \left(\frac{W_f}{C_f}\right) - \left(\frac{W_r}{C_r}\right)$

W_f =85.5 kg
W_r = 104.5 kg

K_{us} = -0.02311 deg/g

Negative value indicates oversteering behaviour of vehicle

7. LATERAL WEIGHT SHIFT :

- Weight distribution = 57 : 43
- Total weight of vehicle = 190 kg
- Height of C.G.(h) = 18 in.
- Wheel track (t) = 52 in. (1.32 m)

Lateral Weight Transfer (ΔW) = (W * A_Y * h) / (g * t)
(this is difference in weight of front left and right)
= (81.7 * 1g * 0.457) / (g * 1.32)
= 28.28 kg ....(1)

8. MOMENTS ON STEERING AXIS

8.1 Moment due to vertical forces

M_v= - (F_{ZL} + F_{ZR}) *d*sinλ*sin δ + (F_{ZL} – F_{ZR})*d*sinν*cosδ

- M_v = moment due to vertical force
- λ = KPI
- d = scrub radius
- δ = steer angle
- ν = caster angle

= -(81.7 * 9.81)*0.044* sin7°*sin(44)° + (28.28*9.81) ....{
(F_{ZL} – F_{ZR})=28.28 kg
*0.044*sin6°*cos(44°)
from (1)}
M_v = -2.06 N-m

8.2 Moment due to Lateral force :

M_L = (F_{yL} + F_{yR})*r*tanν
r= radius of tyre (23 in = 0.2921 m)
= 81.7 * 9.81 * 0.2921 *tan6°
M_L = 25.75 N-m
Moment due to Tractive Force :
M_T = (F_{XL} – F_{XR}) * d
= F_{XL} = F_{XR}
F_{XL} = μ*m_o*g*cosφ
= 0.65 * (81.7/2 + 28.28) * 9.81 * cos44°
= 317.09 N
F_{XR} = μ*m_i *g*cos42°
= 0.65 * (81.7/2 – 28.28)*9.81*cos43.57
= 58.07 N
M_T = (317.09 – 58.07) * 0.04
= 11.39 Nm

Net Aligning Torque :
= (11.39 + 25.75 – 2.06) * cos(7^2 + 6^2)$^{(1/2)}$
= 34.6 Nm

Force on Steering Arm :
F_{arm} = Aligning Torque/ steering arm length
= 30 / 0.06985
F_{arm} = 429.49 N
Force on Tie Rod *:*
F_{tie} = F_{arm}*cos(steering arm angle with vertical in top view geometry)*cosα
where α= angle between tie rod and steering arm in front view geometry

= 429.49*cos0°*cos8°

F_{tie} = 425.31 N

Force on Rack :

$F_{rack} = F_{tie} * \cos\alpha$

= 425.31 * cos8°

F_{rack} = 421.17 N

Torque on Pinion:

T_P = (Force on rack)*(p.c.r. of pinion)

= 421.17 * 0.019

T_P = 8 N-m

Steering Effort:

Driver Efforts = Torque on pinion / radius of steering wheel

= 8 / 0.13

Driver Efforts = 61.53 N

9. Material selection

Precise operation and light weight components was the target behind selecting the material for the steering system. Although precision and weight are the top priorities, **manufacturability, cost and reliability were also taken into consideration.**

Deflection in any component leads to steer compliance, hence resulting in an unresponsive steering system.

Hence we selected material which is light in weight and can perform operation very precisely and smoothly.

Table:4 Steering components material

Sr no.	Component	Material
1	Rack and Pinion	Aluminium 7 series
2	Steering casing	Aluminium 7 series
3	Tie Rods	Aluminium 7 series
4	Steering wheel and Column	Aluminium 6 series

10. Conclusion

The objective of designing parallel steering system for light weight off road vehicle is accomplished with the use of design software like CATIA and simulation software LOTUS and also use of engineering principles. The parallel steering system was designed to achieve minimum turning radius of the vehicle without any understeer and also to have minimum weight for the system. The parallel steering system is also designed to have minimum bump steer in the vehicle. The achieved bump steer in the vehicle is almost zero in this parallel geometry.

11. REFERENCES

[1] Thomas D. Gillespie- Fundamentals of Vehicle Dynamics.

[2] Caroll Smith- Tune to Win.

[3] William F. Milliken And Douglas L. Milliken- Race Car Vehicle Dynamics; ISBN 1- 56091-526-9.

[4] Kirpal Singh- Automobile Engineering Vol. 1