Design, Analysis and Optimization of Steering System

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***Abstract*- There are various suspension geometry that can be consider commercial vehicle. Macpherson strut suspension is one of those which are widely used. Since the Macpherson type of suspension with A-arm system consists of several connected parts, the dimensions of each member play a vital role in the variation of steering and suspension angles. In this study, importance has been given to the optimization of suspension geometry and obtaining the optimum locations of the mounting points i.e. hard points of the suspension geometry. A Macpherson strut suspension (front suspension) with A-arm was optimised in Lotus Shark software to obtain the most appropriate locations of the hard points. The main function of steering system in the vehicle is to maneuver both the wheels in desired direction and support the suspension links in riding/static condition. Steering system works along with suspension system. During optimization work of front geometry in vehicle, we have to match steering points and suspension points together to meet at the same Instantaneous Centre (ICR). It is necessary to meet ICR to avoid bump steer and roll steer during riding condition of vehicle. Bump steer is defined as unwanted steer when the vehicle goes in bump condition and roll steer is defined as unwanted steer when the vehicle goes in rolling condition. The other important factors pertaining to the wheel geometry like the caster, camber, toe in and out, scrub radius all depend on the steering geometry. Also, the tire wear is dependent on the steering geometry. To avoid all the problems occurring due to improper geometry and meet required performance parameters, we have to optimize front wheel steering geometry using LOTUS SHARK V4 software.**

***Index terms*- Ackerman steering geometry, Caster, Camber, CATIA V5R20, Instantaneous Centre (ICR), KPI, LOTUS SHARK V4, Wheel geometry, Scrub radius, Toe in/out**

Stages involved—

1. Define specification of vehicle model whose steering geometry has to be optimized.
2. Make kinematic model and meet ICRs with the help of CATIA.
3. Define all the points in the kinematic model.
4. Optimize the points of the model to meet the performance parameters.
5. Import the obtained satisfactory points in the LOTUS SHARK software.
6. Verify the performance results and obtain the graphs.

I. INTRODUCTION

The kinematics of suspension components describe how important characteristics change as the suspension moves, typically in wheel travel, roll and steering. The kinematic relationship between the linkages and joints in an automotive suspension is very complex, and it has a significant influence on tools, including dynamic analysis software and K&C rig testing, which are usually used to design, test and tune the vehicle suspension concepts. Kinematics has been defined in a general sense as the study of motion without reference to mass or force. Many design parameters relative to the static settings, such as the wheel alignment angles, roll centre height, caster trail, scrub radius and spring motion ratio, basically need to meet the overall vehicle targets. Furthermore, the kinematics of the suspension, such as roll centre height, track gain, camber gain, caster gain, Ackermann change with steering angle, roll steer and bump steer, also need to be verified.

The kinematics of an automotive suspension has been studied extensively. These studies have generally evaluated variations in the corresponding parameters in regard to individual wheel vertical motion, rolling motion or steering. In this paper, Macpherson strut suspension with rack and pinion steering system was modelled by CATIA V5 R19 software, and various suspension and steering parameters of vehicle are analysed by using LOTUS SHARK V4.

Arindam Pal, Sumit Sharma, Abhinav Jain [1] discuss optimised suspension design of an off-road vehicle. The development of this suspension and steering geometry design gives us appropriate camber and caster variations, toe angles, Ackermann geometry, proper flow of forces from chassis to ground and shock absorber characteristics when running on the challenges posed by a rugged off-road track. The design of geometry was achieved through the thorough study of its position, dimensions, of application and installation.

Daniel B. January, Hunter Engineering Company, Bridgeton, Missouri [2] discuss steering geometry and caster measurement. The purpose of this paper is to define caster angle of steerable wheel to be referenced to the thrust line of non-steerable wheel and further purpose is to characterize and derive an optimum method of measuring caster which is implemented in practical.

P. N. Belkhode, A. M. Mahalle, P. P. Holay [3] discuss comparison of steering geometry parameters of front suspension of an automobile. Steering performance parameters such as the kingpin inclination, caster, camber, scrub radius, toe in and toe out for front suspension of vehicle are determined by experimental observations, comparing data based model and artificial neural network.

R. P. Rajvardhan [4] discusses the effect of wheel geometry parameters on vehicle steering as wheel geometry is an important aspect which contributes a lot to the steer ability of a vehicle. Even a slight misalignment creates problems like steering pull and vehicle drift. The wheel geometry performance parameters that affect the steer ability of the vehicle are camber, caster, Steering Axis Inclination (SAI), scrub radius and toe.

Sneh Hetawal, Naik Shashank Giridhar, Baskar P. [5] discusses suspension geometry optimization and analysis of steering knuckle for weight reduction. Here, LOTUS SHARK software is used to optimize the results of suspension and steering performance parameter.

Yung Chang Chen, Po Yi Tsai and I An Lai [6] discuss Kinematic Analysis of Roll Motion for a Strut/SLA Suspension System. The roll centre is one of the key parameters for designing a suspension. Various driving characteristics of vehicle are affected significantly by the shifting of the roll centre during the suspension's motion.

II. METHODOLOGY

The LOTUS SHARK software allows us to conveniently vary the co-ordinates of the mounting points, which in turn alters the dimensions of wishbone arms and the knuckle mounting points. Thus, we are able to vary the dimensions of the suspension A-arms. In this study the Macpherson strut suspension system was analysed in the above mentioned software to obtain the optimum locations for the mounting points. We can see that steering and suspension angles can be controlled by setting the appropriate co-ordinates of the hard points. Formula SAE cars generally run on almost flat tracks, where the bumps and rebounds experienced are small. These cars also have to experience high speed turns that cause car body roll. The above mentioned track conditions cause variation of camber and toe. Variation of camber directly affects the contact patch of the tires with the road. The LOTUS SHARK allows us to simulate this variation of camber and toe change with set conditions of bump, rebound and roll, maintaining the integrity of the specifications.

In all the cases a linear variation is always preferred. The linear characteristics help us to maintain predictable handling characteristics of the vehicle under varying track conditions. The results for the optimized geometry in the software, to suit our requirements are discussed below:

Figure 1 shows position of different points whose coordinates need to be put in to LOTUS SHARK software to optimize the result of steering and suspension geometry.

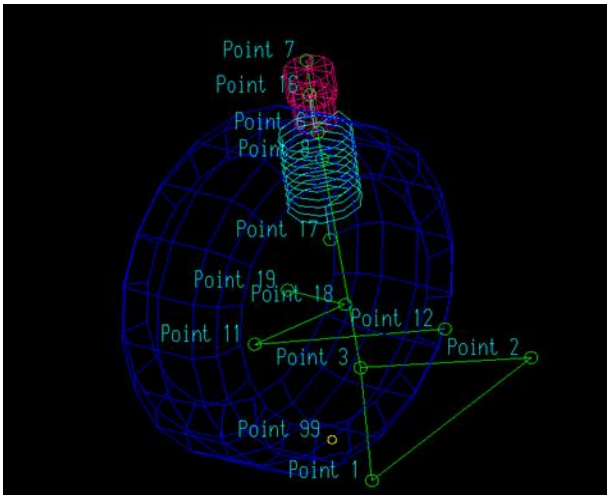

Fig.1. Macpherson strut suspension geometry modelled in Lotus Shark

III. KINEMATICS OF GEOMETRY

For the design and analysis of the vehicle steering system based on its kinematics and geometrical aspects, it is important to achieve following:

1.Optimize the % Ackerman in the vehicle so that it will be maneuver well in the sharp turns without facing under steer and over steer problems.

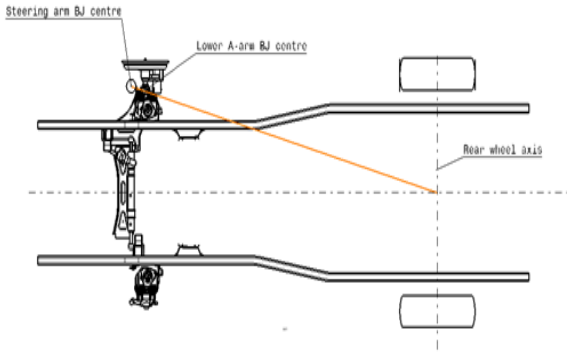

Fig.2: Kinematics and geometric aspects of vehicle

2. Minimize turning circle radius of vehicle.

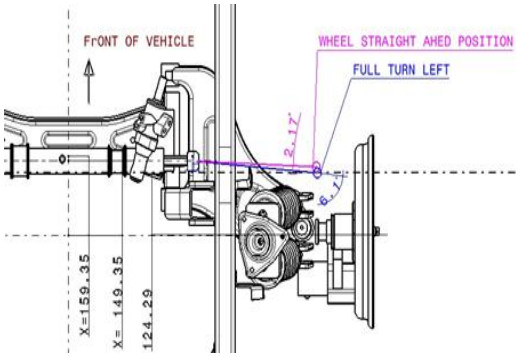

Fig.3: Tie rod criteria

3. Optimize the kinematic geometry.

4. Avoid bump steer problems in vehicle. This is interrelated to vehicle suspension geometry.

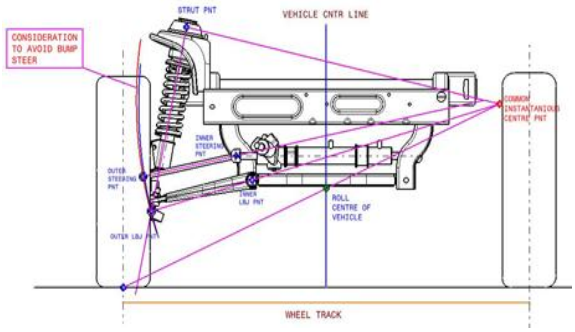

Fig.4: Kinematics and geometric aspects of vehicle

5. ICRs meeting to optimize performance parameters of vehicle like, Caster, Camber, KPI, Scrub radius, Toe.

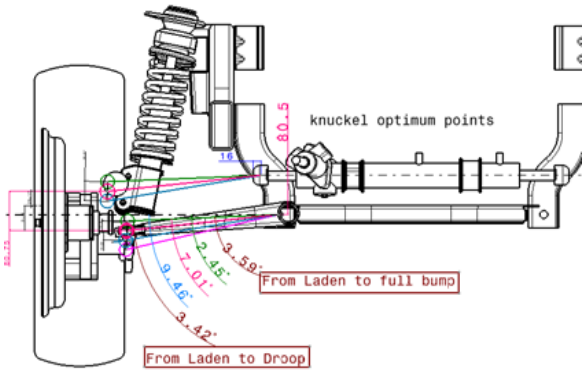

Fig.5: A-arm movement Droop to Full Bump

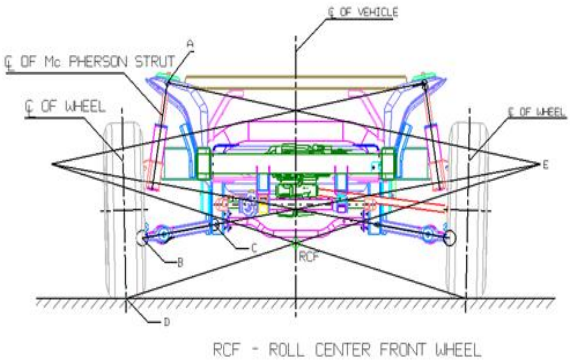

Fig.6: Instantaneous centre (virtual reaction point) meeting

Roll Centre optimum location

Roll center is the virtual pivot point in space that a car rotates around when subjected to cornering forces. The roll center is significant because its location determines how a car will handle and what factors must be considered when tuning its suspension.

The roll center is found by drawing lines from the center of the tire contact patch to the instant centers. When they intersect at the car's centerline, it is the roll center.

In the case of extreme angularity of the links, the roll center can be under ground. It is not ideal, although many race cars are like this due to a desire to have a lot of negative camber gain in the front suspension.

The biggest effects that roll center location has on a car's handling have to do with how the car responds to steering input in a corner, how much the car rolls over in a corner and how friendly the car's balance and mechanical grip at the limit are.

The distance between the roll center and the center of gravity is called the roll couple. The CG location for each end of the car can be located by jacking the car

up a known distance from side to side while it is on corner scales, and observing the change in corner weights. Then the data is fed into an equation to give you the coordinates of the CG. Since it is fairly safe to say that most people don't have a precisely flat surface and expensive corner scales, it is usually safe to estimate the CG for the front suspension around crankshaft height. In the rear, it is usually at the floor of the trunk. The roll couple is the lever arm that centrifugal force working on the CG uses to make a car lean over in a turnabout the roll center.

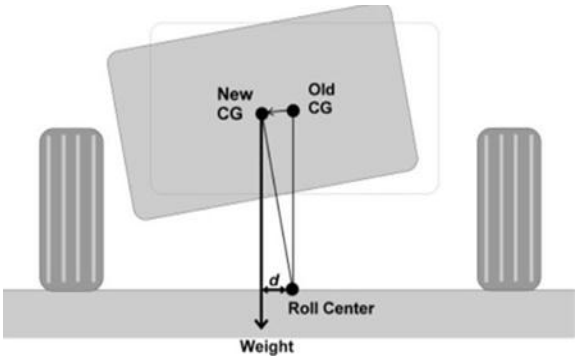

Fig.7: Position of CG, Roll centre i.e. rolls couple

The longer the roll couple, the more leverage centrifugal force has on the suspension through the center of gravity and the more the car will want to roll in a turn. A longer roll couple makes cars slower to respond to steering input. The resulting weight transfer from a long roll couple and high roll angles does not have that much effect on overall weight transfer but it will increase dynamic weight transfer which can make a car twitchy and harder to control at the limit of traction.

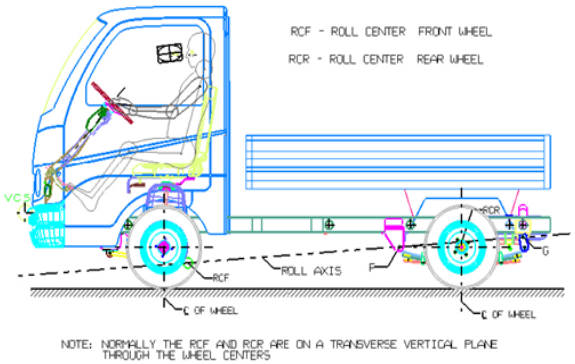

Fig.8: Roll axis about which vehicle rolls

A high roll center can cause the tires to jack and tuck under when cornering hard. This jacking is very dangerous and is the reason why old VW bugs and pre 1964 Conveyers with swing axles had a disturbing tendency to flip. On most cars the ideal location for the roll center is 2-4 inches above the ground for the front suspension and 4-10 inches above ground for the rear suspension, with the rear roll center higher than the front. This is so that the car will transfer more weight on to the front due to an increase in geometric anti-roll, giving a more predictable tendency to understeer at the absolute limit. Most purpose built racecars are like this.

IV. ANALYTICAL SIMULATION

Table 5.1 below shows coordinate values of different points and vehicle 3D parameters are shown in Table 5.2 and Table 5.3.

Table 1: Front Suspension Co-ordinates

Sr. No	Points	Description	Co-ordinates (mm)		
			X	Y	Z
1	[1]	Lower wishbone front pivot	657.68	254.33	-260.50
2	[2]	Lower wishbone rear pivot	952.68	254.33	-260.50
3	[3]	Lower wishbone outer ball joint	991.15	600.95	-354.72
4	[6]	Strut slider upper axis point	1008.11	516.36	38.20
5	[7]	Strut top point	1013.22	476.32	223.84
6	[8]	Strut slider lower axis point	1007.09	518.61	-22.46
7	[11]	Outer track rod ball joint	868.03	629.52	-250.06
8	[12]	Inner track rod ball joint	884.46	309.57	-186.73
9	[16]	Upper spring pivot point	1009.30	505.60	74.25
10	[17]	Lower spring pivot point	1004.02	545.02	-127.13
11	[18]	Wheel spindle point	999.36	604.85	-265.73
12	[19]	Wheel centre point	999.36	697.30	-265.73
13	[20]	Part 1 C of G	898.03	434.55	-293.54
14	[21]	Part 2 C of G	955.23	538.11	-72.61
15	[22]	Part 3 C of G	876.50	464.51	-217.38
16	[23]	Part 4 C of G	956.72	526.07	-26.38

Table 2: 3D Parameters

Sr.No.	DESCRIPTION	VALUE
1	Bump travel (mm)	61 mm
2	Rebound travel (mm)	-38 mm
3	Bump/rebound increment (mm)	20 mm

4	Roll angle (deg)	3 deg
5	Roll increment deg)	0.5 deg
6	Steer travel (mm)	72 mm
7	Steer increment (mm)	6 mm
8	Wheel base (mm)	1394 mm
9	C.G. Height (mm)	450 mm
10	Braking on front (%)	60 %
11	Drive on front (%)	0 %
12	Weight on front (%)	40 %
13	Front brake type (1/2 inboard/outboard)	1
14	Rear brake type (1/2 inboard/outboard)	1
15	Total sprung weight (kg)	-
16	Front suspension type (1/2 independent/rigid)	1
17	Rear suspension type (1/2 independent/rigid)	2

Table 3: Tire Specifications

Sr.No.	DESCRIPTION	VALUE
1	Rolling radius of wheel (mm)	300 mm
2	Tire width (mm)	185 mm

Putting all above parameters in to the LOTUS SHARK software we can obtained the graph as follows with respect to wheel travel.

A. Camber

Camber is the tilt of the wheel from true vertical as viewed from the front of the vehicle. It is measured in degrees. If the top of the tire appears to tilt outwards, it is positive camber. High positive camber causes the outer tread of the tire to wear more than the inner tread; negative camber has the opposite effect. Below are the plots of camber angle variation with bump; roll and steer travel.

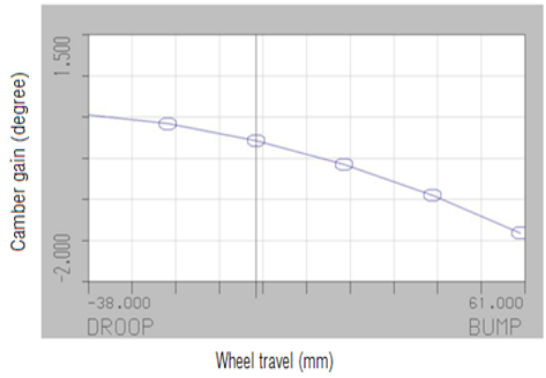

Fig.9: Camber v/s bump and rebound

The above graph shows that the camber angle for the optimized geometry varies linearly with the bump and the rebound. For this analysis, a bump of 61mm and a rebound of 38mm (extreme values) were considered. The initial camber set in the geometry is 0 degree. The camber gain for bump is approximately 1.28 degree (negative) and during rebound it is 1.05(positive). The above camber angle variations are taken care by the tire width. Wide tires will prevent reduction of the contact patch and subsequent loss of control. The camber gain comes into picture only for the duration of the bump/rebound.

Total camber gain is 2.2 deg for wheel travel 99 mm.

B. Toe angle

Toe is how the wheels are aimed, as viewed from top. Pair of front or rear wheels aimed inward at the forward edges has toe-in while wheels aimed outward have toe-out.

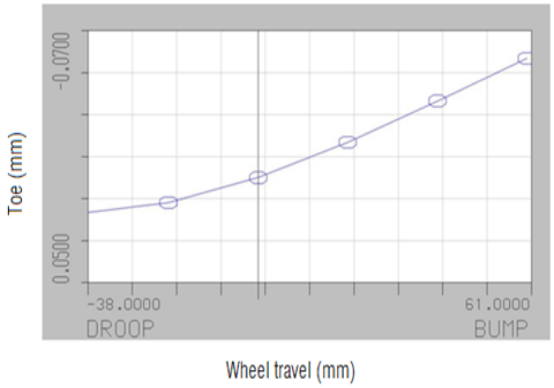

Fig.10: Toe v/s Bump and Rebound

When the vehicle is moving, toe decreases (or disappears) because the wheels straighten out under acceleration and the steering linkage flexes slightly. Above are the plots of toe change with bump and roll. The plot indicates variation of toe with bump and rebound. The toe gain for the rebound (positive) and bump (negative) are negligible. Hence, the vehicle will be able to maintain straight ahead characteristics under the conditions of bump and rebound.

Toe in or Toe out is another important suspension setting; it affects the handling of the car in terms of tire/ tire wear, straight line and cornering characteristics.

In terms of the best braking and acceleration capacity for the tires/ tires, it is best to have neutral toe settings.

a. Too much toe in causes the outsides of the tires/ tires to wear out.

b. Too much toe out causes the insides of the tires/ tires to wear out.

Total toe is 0.02 mm for 99mm wheel travel.

C. Kingpin inclination

On most modern designs, the kingpin is set at an angle relative to the true vertical line, as viewed from the front or back of the vehicle. This is the kingpin inclination or KPI (also called steering axis inclination, or SAI). SAI is non-adjustable, since it would change only if the wheel spindle or steering knuckles are bent.

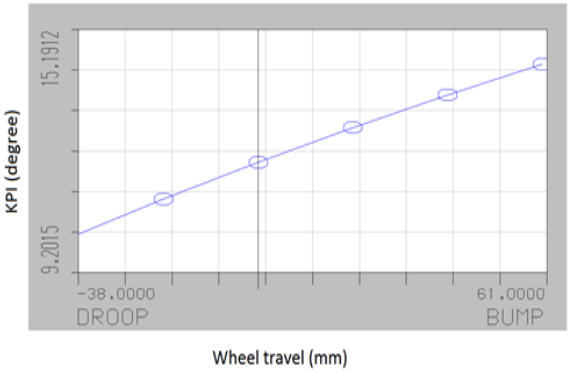

Fig.11: Kingpin inclination v/s Wheel travel

This has an important effect on the steering, making it tend to return to the straight ahead or centre position. This is because the straight ahead position is where the wheel is at its highest point relative to the suspended body of the vehicle - the weight of the vehicle tends to return the kingpin to this position. A second effect of the kingpin inclination is to set the scrub radius of the steered wheel. This is the offset between the tire's contact point with the road surface and the projected axis of the steering down through the kingpin.

Total KPI is 11.9 deg for 99mm wheel travel.

D. Scrub radius

The scrub radius is the distance in front view between the king pin axis and the center of the contact patch of the wheel, where both would theoretically touch the road.

The kingpin axis is the line between the upper and lower ball joints of the hub. On a Macpherson strut, the top pivot point is the strut bearing, and the bottom point is the lower ball joint. The inclination of the steering axis is measured as the angle between the steering axis and the centerline of the wheel. This means that if the camber angle is adjustable within the pivot points, the scrub radius can be changed. This alters the width and offset of the tires on a vehicle.

If the scrub radius is small then the contact patch is spun in place when parking, which takes a lot more effort.

Total scrub radius is 35.4 mm for 99mm wheel travel.

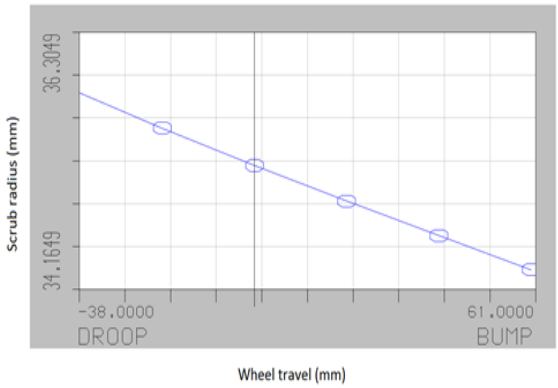

Fig.12: Scrub Radius v/s Wheel travel

E. Caster angle

The pivot points of the steering are angled such that a line drawn through them intersects the road surface slightly ahead of the contact patch of the tire on the pavement. The purpose of this is to provide a degree of self-centering for the steering — the wheel casters around so as to trail behind the axis of steering. This makes a car easier to drive and improves its directional stability (reducing its tendency to wander). Excessive caster angle will make the steering heavier and less responsive, although, in racing, large caster angles are used to improve camber gain in cornering. Caster angles over 7 degrees with radial tires are common. Power steering is usually necessary to overcome the jacking effect from the high caster angle.

Caster angle is 3.5 deg for 99mm wheel travel.

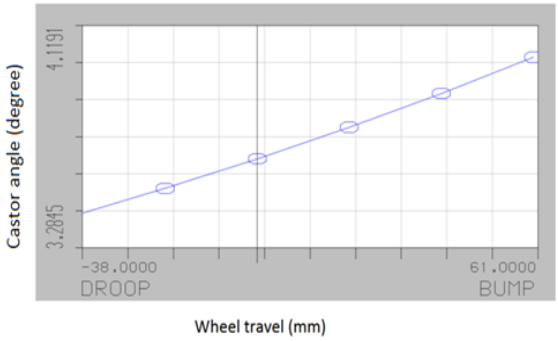

Fig.13: Caster angle v/s Wheel travel

F. Caster trail

The steering axis (the dotted line in the diagram above) does not have to pass through the center of the wheel, so the caster can be set independently of the mechanical trail, which is the distance between where the steering axis hits the ground, in side view, and the point directly below the axle.

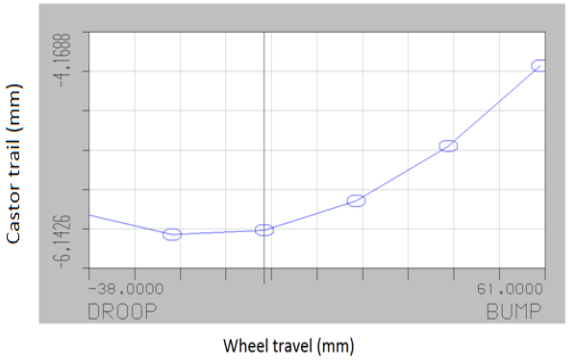

Fig.14: Caster trail v/s Wheel travel

The interaction between caster angle and trail is complex, but roughly speaking they both aid steering. Caster tends to add damping, while trail adds 'feel', and returnability. In the extreme case of the shopping trolley wheel, the system is undammed but stable, as the wheel oscillates around the 'correct' path. The shopping trolley/cart setup has a great deal of trail, but no caster. Complicating this still further is that the lateral forces at the tire do not act at the center of the contact patch, but at a distance behind the nominal contact patch. This distance is called the pneumatic trail and varies with speed, load, steer angle, surface, tire type, tire pressure and time. A good starting point for this is 30 mm behind the nominal contact patch.

Total caster trail is 6 mm for 99mm wheel travel.

G. Spring rate

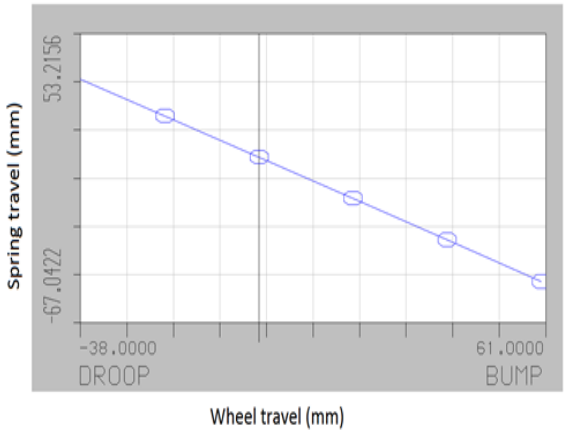

Fig 15.1: Spring travel vs. Wheel travel

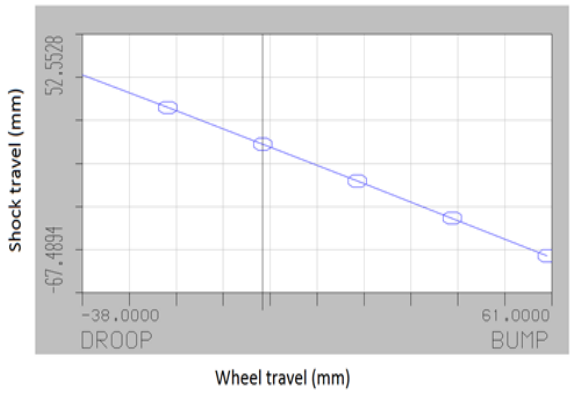

Fig 15.2: Shock travel vs. Wheel travel

It is a ratio indicating the resistance of a spring during bump or rebound (compression or expansion). Also known as suspension rate, it is critical for setting the correct ride height and is proportionate to the movement of the length of component travel in its stoke phases. As we know, the whole job of suspension is to keep the wheels and tires in contact with the ground at all times for total performance.

By having the ability to change the spring rate, heavier vehicles can have a higher setting to stop the suspension bottoming out under extreme loads, or if the car has big down force generation abilities. Softer spring rates could be an advantage in rougher terrains or raised curbs at apexes. If you have seen a car jump off a curb at speed, then lower spring rates are needed.

Sometimes people complain when driving sports or track focused cars with competitive suspension on normal roads, this is because of higher spring rates.

Dampers work in conjunction with springs to form the basis for car suspension, they are sometimes incorrectly referred to as shock absorbers. If a car was only fitted with spring and not dampers, then any movement in the cars suspension in the vertical plane would effectively keep bouncing up and down until the kinetic energy is displaced. This would make the car very difficult to drive from a performance point of view, as the geometry would be constantly changing.

Dampers effectively help to dissipate any vertical movement in the suspension and keep the springs movements controlled also they help the wheel keep full motion under different loads, while keeping the wheel in contact with the ground.

a. 1-Way adjustable Suspension: Rebound adjustable only.

b. 2-Way adjustable Suspension: Bump and rebound adjustable.
c. 4-Way adjustable Suspension: High and low speed bump setting, high and low speed rebound settings (high speed for bumps, low speed for corners and braking).

It is possible to adjust the dampers to give different hydraulic resistance to fine tune the handling of the suspension for given conditions, the movement is broken down into bump (compression) and rebound (extension) motions. Dampers don't control load levels, but affect the speed of how fast the suspensions reacts to load changes and how fast the tire/tires contact patch receives these loads.

a. Stiffer front damping (bump and rebound), increases grip at the rear of the car, increasing under steer characteristics.
b. Stiffer rear damping (bump and rebound), increases grip at the front of the car, increasing over steer characteristics.

H. % Ackermann

Ackermann steering geometry relates the steer angle of an inside tire to that of the outside tire. When turning, the inside tire travels a shorter radius than the outside tire. Hence, steering assembly must have a greater steer angle to avoid tire scrub.

Classic Ackermann minimizes scrub by positioning both tires perpendicular to the turn center. It can have a significant impact on tire wear. Ackermann analysis can also be used as a tuning tool in cases where classic Ackermann may not be the objective.

Ackermann has been around longer than the motor vehicle - over a century - but there is a little rigorous analysis in the literature. There are two common measurements of Ackermann which give very different results. Both are used in texts and computer programs. Yet a literature search revealed only a couple sentences discussing the relationship between the two.

Calculation of %Ackerman

%Ackerman = [(αi-αo) / (βi-βo)] X 100
= (40.-32.5) / (40.8-30.1) X 100
= (8.31) / (10.7) X 100
= 77.6 %

αi, αo = Obtained inner and outer angle (degree) from wheel geometry.

βi, βo = Required Inner and outer angle (degree) for 100% Ackerman

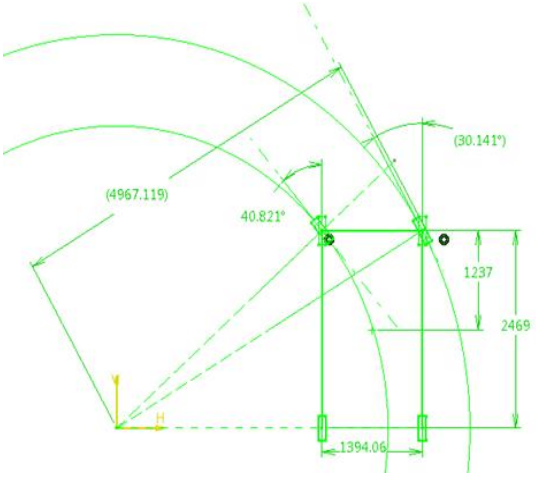

Fig.16: Ideal I/P and O/P angle for 100% Ackerman

V. OPTIMIZED RESULTS

Table 4: Optimized results of steering and suspension geometry parameters.

Sr. No.	LOADING CONDITION	STRUT HEIGHT IN (mm)	STRUT REMAINING DIST	WHEEL TRACK IN (mm)	CAMBER IN (DEGREES)	TOTAL TOE IN (mm)	CASTER IN (DEGREES)	KPI	Scrub Radius (mm)	Turning Radius (Outer)	Inner wheel angle	Outer Wheel Angle	Difference between 2 angles	Difference required for 100% Ackerman	% Ackerman
1	UNLADEN	370	130	1395	1.05	-0.0114	3.5	11.9	35.4	4.96	40.82°	32.51°	8.31	10.7	78
2	FULLY LADEN WITH 2 PERSON SITTING	350	110		0.533°	-0.006									
3	DESIGN CONDITION	340	100		0°	0									
4	FULLY LADEN WITH 2 PERSON SITTING ON BUMP, WITHOUT REBOUND STOP IN ACTION	320	80		-1.01	0.008									
5	FULLY LADEN WITH 2 PERSON SITTING, ON BUMP, WITH REBOUND STOP IN ACTION	310	70		-1.28	0.12									

VI. CONCLUSIONS

1. The Macpherson strut suspension geometry was optimized to obtain the suspension mounting points that gave linear variation of the suspension angles with the varying driving conditions.
2. It now possible to predict the handling characteristics of the car under different driving conditions.
3. Geometry optimized is expected to give comfort ride of vehicle.

4. Table above shows the steering performance parameters computed by experimental observations, experimental data-based model and LOTUS SHARK SIMULATION.
5. Based on the optimization results of commercial vehicle, for different values of wheel geometry parameters, the following conclusions are drawn:
   i. Caster helps for aligning wheel at the relatively small Steering angles used during normal and high-speed driving. Positive caster increases steering effort whereas negative caster reduces steering effort due to presence and absence of sufficient torque.
   ii. KPI has an important effect on the steering, which makes it to return to the straight ahead position.
   iii. Camber has an overall effect to promote stable, straight ahead running.
   iv. Too much toe in causes the outsides of the tires to wear out and too much toe out causes the insides of the tires to wear out.
   v. If the camber angle is adjustable within the pivot points the scrub radius can be changed. This changes the offset and width of the tires on a vehicle. If the scrub radius is small then the contact patch is spun in place at the time of parking, which takes lot of effort. It affects the torque required to turn steering wheel in the driver cabin.
   vi. If the scrub radius is zero, driver cannot experience steering feel.
6. With this analysis we are successful in avoiding bump steer and roll steer by meeting ICs at same point.
7. Minimum possible turning radius achieved is 4.9 metres.
8. Meeting all values of suspension and steering performance parameters, we have achieved 77% Ackerman.

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