NOVATEUR PUBLICATIONS
INTERNATIONAL JOURNAL OF INNOVATIONS IN ENGINEERING RESEARCH AND TECHNOLOGY [IJIERT]
ISSN: 2394-3696
VOLUME 2, ISSUE 12, DEC.-2015

LOW PROFILE TIRE IMPACT ON DOUBLE WISHBONE SUSPENSION

Prashant Bhat
CAE/MBD-PD,
Marsh Consultancy of Excellence,
Pune, India

Prof. Ravindra R. Navthar
Dept. of Mechanical Engineering,
PDVVP College of Engineering ,
Ahmednagar, India

ABSTRACT

High cornering and inclination stiffness characteristics of these lower profile tires have benefited the steering and yaw responses of the modern vehicle, the associated increase in radial stiffness has reduced the tire's effectiveness as an isolator for ride comfort and harshness. Consequently, the modern suspension system has to ameliorate this loss of tire compliance while retaining accurate wheel control if ride comfort is to be maintaining, for justifying the need, the new suspension System has to design and validate through multibody dynamics for minimum castor trail loss and elastic centre while breaking and isolation characteristics. This requirement for longitudinal compliance has an unfortunate side effect on hub control when braking forces are applied. Consequently, the hub rotates when the suspension is subject to braking forces. However, this has to study considering the vehicle dynamics using ADAMS software. Based on vehicle dynamics analysis it is confirm the lower the longitudinal stiffness of the suspension, greater the associated hub rotation under braking.

INTRODUCTION

The demands on vehicle suspension performance – in terms of both accurate wheel geometry control and isolation – have increased steadily over the past decade as the requirements of steering, handling and styling have driven carmakers toward ever-lower profile tire choices of a larger diameter.

Although the high cornering and inclination stiffness characteristics of these lower profile tires have benefited the steering and yaw responses of the modern vehicle, the associated increase in radial stiffness has reduced the tire's effectiveness as an isolator for ride comfort and harshness. Consequently, the modern suspension system has to ameliorate this loss of tire compliance while retaining accurate wheel control if ride comfort is to be maintain. In particular, soft longitudinal rates at the wheel centre for improved harshness and impact performance are increasingly desirable.

However, this requirement for longitudinal compliance has an unfortunate side effect on hub control when braking forces are applied to the suspension system. The longitudinal elastic centre of most suspension types generally lies somewhere in the region of the wheel centre. Consequently, the hub rotates when the suspension is subject to braking forces, yet it remains relatively stiff when subject to impact forces, which are resolved at the wheel centre. The lower the longitudinal stiffness of the suspension, the greater the associated hub rotation under braking. In the case of a front suspension, this gives potential for excessive castor trail loss and attendant steering instabilities. Traditionally, because the castor and longitudinal stiffness are coupled, the only way to prevent high castor loss while lowering the longitudinal stiffness has been to increase the effective knuckle length, as exhibited in 'high top arm' wishbone systems and the MacPherson strut type. However, these tall suspension systems require more package space and higher bonnet lines than a short knuckle design such as the narrow spaced double wishbone.

PROBLEM STATEMENT

The demands on vehicle suspension performance – in terms of both accurate wheel geometry control and isolation – have increased steadily over the past decade as the requirements of steering, handling and styling have driven carmakers toward ever-lower profile tire choices of a larger diameter.

OBJECTIVE

- To find the way for increase in longitudinal stiffness of the suspension
- Investigate the new design of double wishbone suspension system for effectively decoupling of castor and longitudinal stiffness
- Maintain the suspension geometry kinematics as same as traditional double wishbone suspension for front.
- Reduce the relative angle between king pin axis and steering axis during braking by controlling the hub rotations
- A better solution would be a suspension whose longitudinal elastic centre moved vertically down from the wheel centre to the ground plane region. Analyse the new design by multibody dynamic software like ADAMS/CAR and ADAMS/VIEW

SCOPE

It is very important to replicate the existing problem in simulation environment, so traditional double wishbone suspension system with low profile tire for front suspension system has to simulate in simulation software like ADAMS and find the castor trail loss and elastic centre during braking only, for this analysis we can use existing ADAMS template for double wishbone.
To design a new way of suspension mechanism, maintain existing interaction point of the suspension to the hub and chassis of lower control arm and upper control arm and suspension height. The new suspension design to limit only for sedan vehicle, there will not be any modification and update in the steering assembly, which is same as traditional double wishbone suspension. Steering analysis is limited to design only., handling and styling have driven carmakers toward ever-lower profile tire choices of a larger diameter.

DESIGN OVERVIEW

Double Wishbone Suspension

A double wishbone suspension Fig.1. This consists of two transverse links (control arms) either side of the vehicle, which are mounted to rotate on the frame, suspension subframe or body and, in the case of the front axle, are connected on the outside to the steering knuckle or swivel heads via ball joints. The greater the effective distance c between the transverse links, the smaller the forces in the suspension control arms and their mountings become, i.e. component deformation is smaller and wheel control more precise [1].

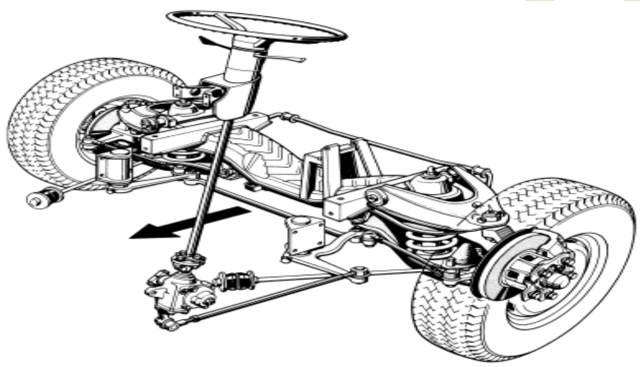

Figure 1: Double Wishbone Suspension System [1].

A cross-member serves as a subframe and is screwed to the frame from below. Springs, bump/rebound-travel stops, shock absorbers and both pairs of control arms are supported at this force centre. Only the anti-roll bar, steering gear, idler arm and the tie-rods of the lower control arms are fastened to the longitudinal members of the frame. The rods have longitudinally elastic rubber bushings at the front that absorb the dynamic rolling hardness of the radial tires and reduce lift on uneven road surfaces.
The main advantages of the double wishbone suspension are its kinematic possibilities. The positions of the suspension control arms relative to one another – in other words the size of the angles α and β – can determine both the height of the body roll centre and the pitch pole. Moreover, the different wishbone lengths can influence the angle movements [2].
Of the compressing and rebounding wheels, i.e. the change of camber and, irrespective of this, to a certain extent also the track width change. With shorter upper suspension control arms the compressing wheels go into negative camber and the rebounding wheels into positive. This counteracts the change of camber caused by the roll pitch of the body. A recent trend in the passenger car market is the adoption of low profile tire.

Low Profile Tire Impact on Suspension Geometry

The market trend in tires had previously been moving from bias tire (cross-ply tire) to radial tire, but a shift has now started towards low profile tire fig.2. As the lowering of the profile of a tire increases the cornering force, it can be regarded as improvement to the tire's handling performance. In addition, as the lowering of the tire profile decreases the section height and increases the wheel diameter of the tire even when the outside diameter is not changed as shown in fig 8, it can also be regarded as improvement in the visual appeal of the tire. bring

NOVATEUR PUBLICATIONS
INTERNATIONAL JOURNAL OF INNOVATIONS IN ENGINEERING RESEARCH AND TECHNOLOGY [IJIERT]
ISSN: 2394-3696
VOLUME 2, ISSUE 12, DEC.-2015

some advantages like greater power in steering performance Improved handling performance (Quick handling) and increased cornering force (CF) Improved tire performance and greater power in steering performance, however has some disadvantage Braking capabilities may vary, driving comfort and handling may change.

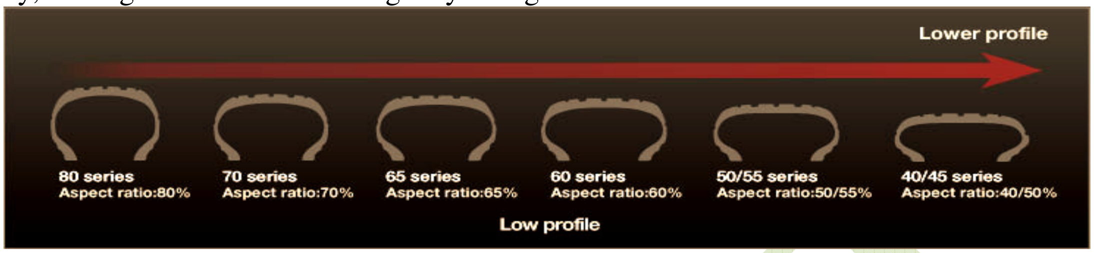

Figure 2: Low Profile Tier Series.

The response of a traditional double wishbone type front suspension to braking forces is depicted in Figure 3; the hub and steering axis rotation, loss of castor trail and approximate elastic centre location are all apparent. The lower the longitudinal stiffness of the suspension, the greater the associated hub rotation under braking. In the case of a front suspension, this gives potential for excessive castor trail loss and attendant steering instabilities. Traditionally, because the castor and longitudinal stiffness are coupled.

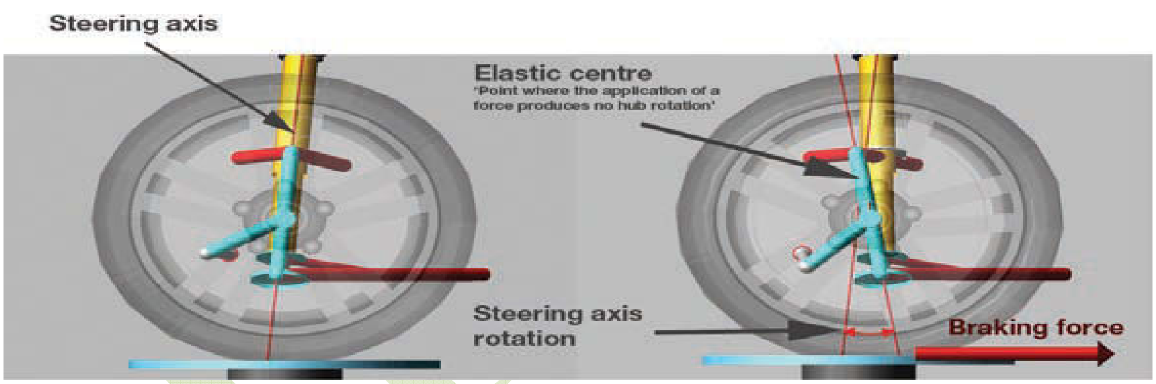

Figure 3: Low Profile Tier Series.

The continuing trend towards larger wheels and ever-lower profile tires has compounded the problems facing car manufacturers who now need to achieve more suspension isolation for comfort whilst retaining good wheel control for responsive handling and steering. Until now, there has generally been a need to compromise a point is reached at which either responsive handling or good isolation is determined to be the dominant concern and then begins the challenge of packaging the appropriate solution within the vehicle's architecture. by effectively decoupling the castor and longitudinal stiffness's of the traditional double wishbone suspension overcomes this compromise creating a solution that delivers the cornering, handling and steering performance of a double wishbone design but with the longitudinal isolation associated with a MacPherson strut arrangement (without its need for height and associated high wing/bonnet lines). Being both less complex and far more compact than the multi-link arrangements currently in widespread use.

Low Profile Tier Suspension Profile

Efficient cornering performance is best achieved when the suspension is capable of orienting the tire to an angle for which the tire provides maximum grip based on the forces required to achieve the desired maneuver and the anticipated roll of the vehicle. Caster, camber, and toe all significantly affect this performance, as is well documented conversely, maximum tire longitudinal force needed during braking or acceleration is usually produced with zero tire camber. Tire performance is also affected by parameters such as normal load, Fz, and various terrains.
The objective of establishing a side elevation rotation point for the suspension under longitudinal forces is also desirable for a rear axle, especially a driven rear axle, but for different reasons then the front. A rear driven axle is required to provide "toe in" under reverse traction loading to counter the ten-dency of the vehicle to over steer when the throttle is lifted mid corner. In particular, the outside rear wheel is required to "toe in" under these circumstances, whilst the inside wheel is required to either "toe in" to a lesser degree or even to "toe out". Moreover, longitudinal force changes at the rear axle generate low loads in comparison with braking loads FBrake, typically of the order of one quarter. A further issue to be dealt with is the requirement to generate adequate "toe" change under throttle lift off, without generating excessive "toe" change under braking.
With a conventional double wishbone suspension, which would have a hub rotation point in the side elevation somewhere near the wheel centre, an undesirable over steer component developed from the inside wheel generating more “toe in” than the outside wheel, will result. Fig.4 and fig.5 represent a conventional double wishbone front suspension viewed from the side. The front of the vehicle is towards the left hand side of the figures. An upper wishbone and a lower control arm are each attached at their inner ends to a vehicle body or sub frame, by means of compliant bushings. The upper wishbone and a lower control arm connected at their outer ends to the upper and lower ends respectively, of a swivel hub, by means of spherical joints. Track rods (not shown) connect a steering mechanism (not shown) to steering arms on the swivel hubs, by means of spherical joints, in conventional manner [4].

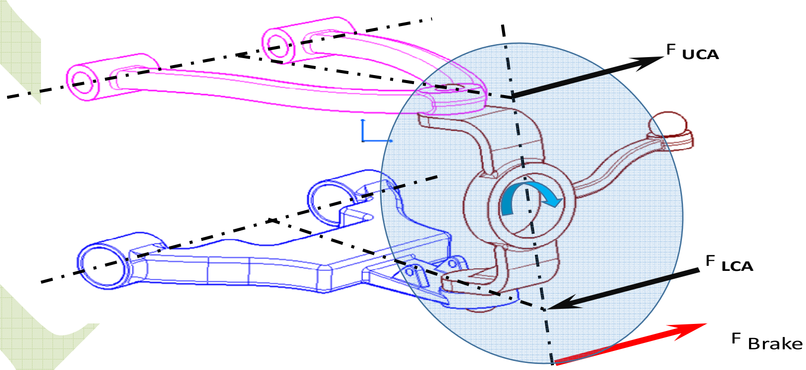

Figure 4: Double Wishbone Braking Force at Contact Patch (CP).

WC represents the wheel centre and CP represents the contact patch of the wheel upon the ground. For the purposes of simplicity, the moments about the steering axis have been ignored i.e. the forces have been resolved as if the outer spherical joints of the upper and lower control arms lie in the wheel plane.

Referring to Fig.4, if a braking force FB is applied at the contact patch CP, the reaction forces F_{UCA} and F_{LCA} are experienced by the swivel hub at the connections with the upper and lower control arm then (1);

$$F_{LCA} = FB\ B/A \text{ and } F_{UCA} = FB\ (B/A-1). \quad (1)$$

Actions 1 refer fig.5 due to braking force the axial deflection of the lower control arm to chassis or sub frame bushings resulting mostly from longitudinal forces at the lower ball joint. Upon axial deflection, which should be the overriding direction if conventional bushings are used. Then, longitudinal defection at the bushings (lower control arm to body) = the deflection at the lower spherical joint (lower control arm to hub lower joint) = 2X. the corresponding deflection at the outer spherical joint of upper control arm and hub upper joint = X, If the outer spherical joint of upper control arm moves 2X and the lower spherical joint moves 2X then the rotation point of the swivel hub is away the contact patch CP.

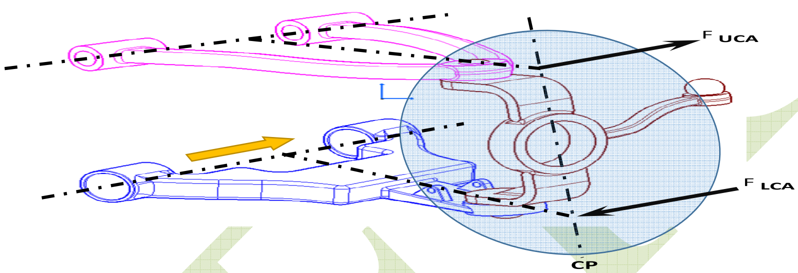

Figure 5: Double Wishbone Axial Force.

Generated radial deflection of the lower control arm to chassis or sub frame bushings at right angle to the plane of the lower control arm, resulting mostly from loads transferred through the upper control arm and hub. Upon radial out of plane deflection, lower control arm pitches about an axis through the inner spherical bushing of upper control arm and chassis, passing between the compliant bushings of LCA and body, then, deflection at the lower spherical bushing of LCA and hub =2X. If the outer spherical joint UCA and hub moves x and the lower spherical joint does not move, then the swivel hub will rotate about the WC, Thus, it is seen that in the simplistic case described, the stiffness at the wheel centre WC is 10 times the stiffness at the contact patch CP. The stiffness at the wheel centre WC relates to the level of isolation of the sprung mass from road induced shock inputs. The stiffness at the contact patch CP is indicative of the change in castor experienced because of braking. Thus, it is seen that the conventional double wishbone concept is not ideal if both good levels of isolation and good castor control are to be realized.

CONCLUDING REMARK

Conventional double wishbone front suspensions, in which both lower and upper control arm are located within the diameter of the wheel, exhibit behaviour that is well suited to front suspension application in terms of roll centre control, camber control, and toe control. They offer advantages over other concepts by virtue of their compactness and the close proximity of their structural attachments to the body or sub-frame, which allows for an efficient lightweight under structure. Impending pedestrian impact legislation will also tend to favour such systems against the widely used MacPherson strut suspensions which typically have mounting points high in the structure, close to the surface of the bonnet, which is undesirable from the point of view of protecting a pedestrian's head which might impact in that region. If double wishbones are spaced widely apart, so that the upper control arm is located above the tire, the resulting disparate body/sub frames mountings.

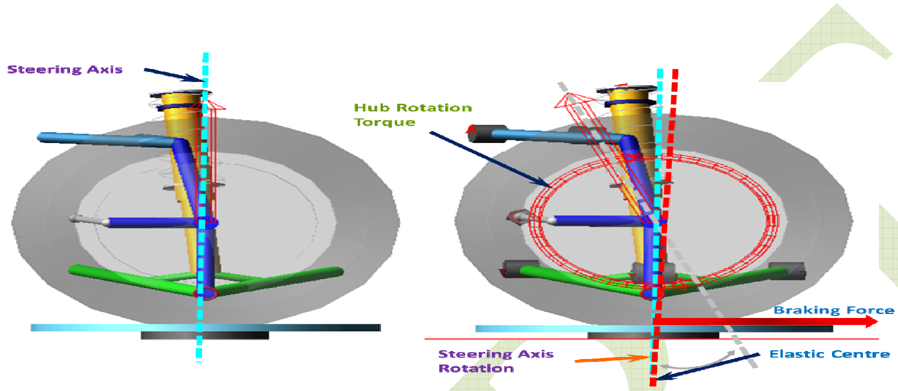

Figure 6: Elastic centre moved near to contact patch hence loss of castor trail minimize [3].

The longitudinal stiffness and castor stiffness need to be closer as shown in fig.6 and the elastic centre point should be near to contact patch of tier and road or below the road [2]. By isolating the longitudinal stiffness and castor stiffness, the better control on hub rotation will be there, by virtue of its elastic centre placement, has now effectively decoupled the longitudinal and castor stiffness, allowing the suspension system to be tuned independently.

REFERENCES

[1] David A Croll, *Automotive Engineering Powertrain, Chassis System and Vehicle Body, First edition, 2009, Butterworth-Heinemann Press.*

[2] Thomas D. Gillespie, *Fundamentals of Vehicle Dynamics,2nd ed, SAE inc. pp. 25-76.*

[3] MSC Software Corporation Ltd, *MSC Adams/Solver 2014 User Manual, pp 80-320.*

[4] MSC Software *Corporation Ltd, MSC Adams/Car 2014 User Manual, pp. 1-206.*

[5] T.J. Drach and D.C. Holloway, Mechanical Engineering, University of Maryland, *Development of a Mobile Tire Test Fixture and Lateral Force Variations Due to Large Camber Angles, SAE Technical Papers, 1991. pp. 3-12.*

[6] Wang, D. and Rui, Y, *The Effects of Front Suspension Parameters on Road Wheel Toe Dynamics, SAE 2001 World Congress Detroit, Michigan, 2001. pp. 3-8.*

[7] MB Gerrard, Kinematic Suspension Linkages - *A Model for Their Behavior & a Procedure for Their Design, SAE Technical Paper 2002-01-0281, World Congress Detroit, Michigan. March 4-7, 2002. pp. 14-20.*

[8] Tamiyoshi Kasahara, Naoki Morioka, Masaharu Satou, Toru Yagi and Kenichiro Muraok, *Development of a New Multi-Link Suspension, SAE 2000 World Congress Detroit, Michigan March 6-9, 2000. pp. 3-10.*

[9] Pascali, L., Gabrielli, P., and Caviasso, G., *Improving Vehicle Handling and Comfort Performance Using 4WS, SAE 2003 World Congress Detroit, Michigan March 3-6, 2003. pp. 4-9.*

[10] Steven P. Fuja, Henry *A. Schmid, and Joseph P. Ryan, Synthesis of Chassis Parameters for Ride and Handling on the 1997 Chevrolet Corvette, International Congress & Exposition Detroit, Michigan February 24-27, 1997. pp. 4-11.*

[11] Duygu GÜLER, İzmir *Institute of Technology, Dynamic analysis of Double wishbone suspension, July 2006.*

[12] Masaaki Minakawa, Jun Nakahara and Jiro Ninomiya, *A Vibration Transfer Reduction Technique, Making Use of the Directivity of the Force Transmitted from Road Surface to Tire, SAE Technical Papers, 2000. pp. 1-10.*

[13] Luc Pelkmans (Vito)Staffan Hultén, Robin Cowan (Merit) Gotzon Azkarate (Opti-Inasmet) Panayotis Christidis (Jrc-Ipts), *Trends in vehicle and fuel technologies review of past trends, Seville, Spain, 2003 , ESTO,May 2003. pp. 174-185.*

[14] SAE, *Surface Vehicle Recommended Practice-J670e, REV JUL76, pp. 11-24.*