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Suspension Geometry Optimization and Analysis of Steering Knuckle for Weight Reduction

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***Abstract*—there are various suspension geometries that can be considered in formula SAE cars. The most common type of Independent suspension geometry is a double wishbone type. It is the most extensively used suspension geometry right from FSAE cars to professional sized formula cars. Since the double wishbone type of suspension 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 optimization of suspension geometry and obtaining the optimum locations of the mounting points i.e. hard points of the suspension geometry. A double wishbone suspension (front suspension) with pushrod was analyzed in Lotus Shark software to obtain the most appropriate locations of the hard points. Steering knuckle being one of the important components of steering system is also analyzed in this study and emphasis has been given on alternate material for weight reduction. The knuckle was modeled in Solid Works and analyzed in ANSYS 15.0.**

Keywords—Suspension Geometry, Double wishbone, Steering Knuckle)

I. INTRODUCTION

The double wishbone suspension system is the most widely used type of front independent suspension system nowadays [1]. In its simplest form it consists of two arms namely upper and lower arms that stretch in transverse direction with respect to the longitudinal axis of the car. The spring damper arrangement can be provided in this type of suspension either in between the two arms or above the upper arm. The arms are conventionally made of tubular cross section using steel as material adding to the weight of the suspension. The dimensions of these arms are set by the location of the mounting points. Thus, for optimum dimensions, the relative positions of the mounting points should be obtained. Steering knuckle is an important member of the vehicle as it acts as housing for wheel hub or the spindle and attaches to the suspension components [3]. The knuckle also provides the mounting points for the brake pads. The upper and the lower wishbone arms are connected to the knuckle using ball joints. The working conditions of the knuckle demand it to be rigid as well as compact at the same time.

It is necessary to understand the concept of the sprung and the un-sprung masses in order to design and engineer an effective suspension system. The sprung mass is the part of the car that is supported by the car's suspension system. The un-sprung masses include the mass of the suspension itself along with the wheels, knuckle and other components connected to it. Lower is the un-sprung mass; lower is the inertia of the suspension system. Thus the suspension is able to respond quickly to the road conditions. Thus a large emphasis is being given on reducing the un-sprung mass of a vehicle without affecting the functionality of the suspension system.

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The suspension system should isolate vehicle body from road induced vibration and maintain contact between tire and road [4]. It should also be able to limit wheel movement, Keep the wheels in alignment and provide directional control during handling maneuvers. The steering and suspension angles are to a large extent dependent on the dimensions of the individual components constituting the double wishbone suspension. The steering and suspension angles generally include the camber, kingpin inclination, toe, and the scrub radius. Factors such as straight line stability, vehicle pull, tire wear and size of the contact patch are affected due to variation of the steering and suspension angles. The Variation of these angles must be allowed only within permissible limits.

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 arms. In this study the double wishbone suspension system was analyzed 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 do 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 straight characteristics (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.

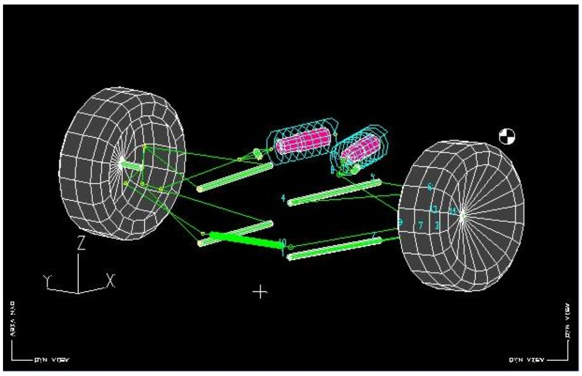

Fig.1. Double Wishbone suspension geometry modeled in Lotus Shark

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.

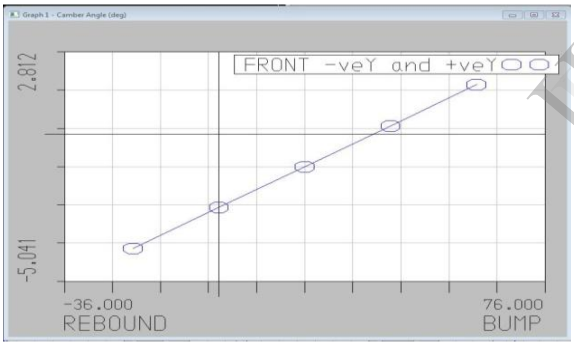

Fig2. Camber v/s bump and rebound

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 76mm and a rebound of 36mm (extreme values) were considered. The initial camber set in the geometry is 2.8^{o} (negative). The camber gain for bump is approximately 4.2^{o} (positive) and during rebound it is 1.4^{o}(negative). 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.

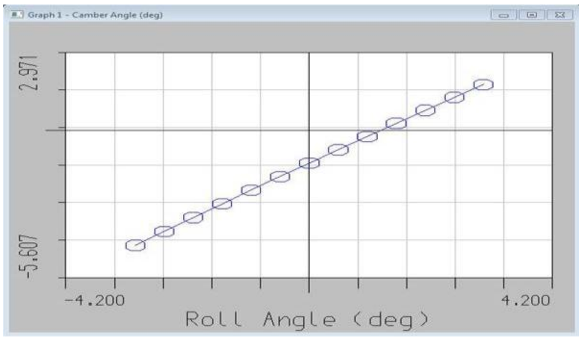

Fig3. Camber angle v/s Roll angle

The above graph indicates linear variation of camber angle with roll angle. The initial camber angle set to 2.8^{o} is clearly seen in the graph. Roll angle of 4.2^{o} (either side) was considered for this simulation. The camber gain on either side was found to be approximately 3^{o}. The linear characteristics obtained help us to maintain predictable handling characteristics.

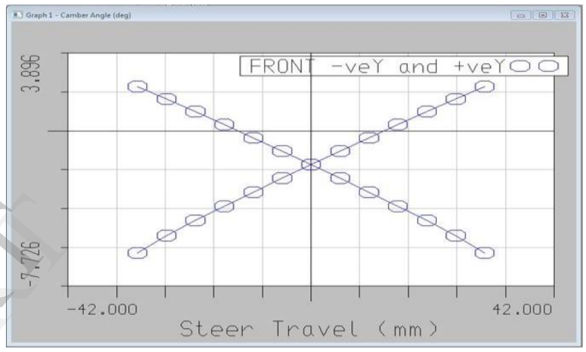

Fig4. Camber angle v/s Steer Travel

The above plot indicates camber angle variation for both the front wheels simultaneously. Steering travel refers to movement of the steering rack. The initial camber set is negative. When the wheels are turned in a particular direction, the outer wheel will have a negative camber gain whereas the inner wheel will have a positive camber gain. The self-aligning torque will try to reduce the camber gain in each case and maintain the contact patch. In case of a smaller camber gain, the self-aligning torque may cause loss of contact patch affecting the drivability of the vehicle. Thus a larger camber gain is preferable so as to have sufficiently large contact patch even under the action of self- correcting torque.

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; wheels aimed outward have toe-out. When the vehicle is moving, toe decreases (or disappears) because the wheels straighten out under acceleration and the steering linkage flexes slightly. Below are the plots of toe change with bump and roll.

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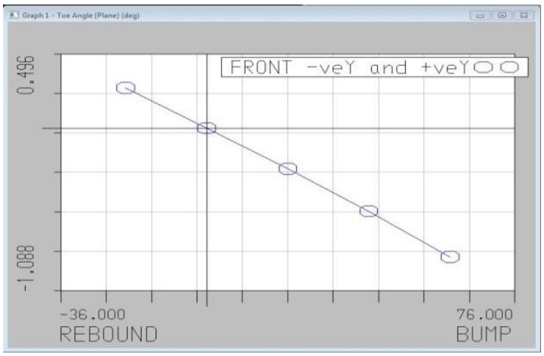

Fig5. Toe v/s Bump and Rebound

The above 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.

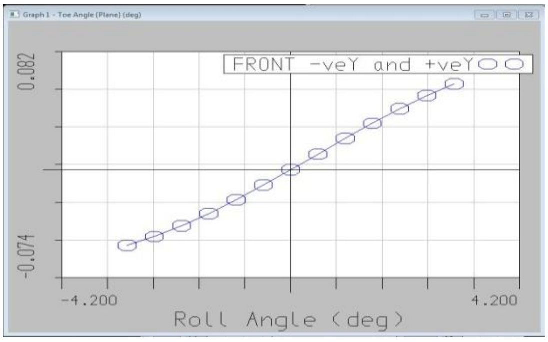

Fig6. Toe v/s Roll angle

The characteristic for variation of toe angle with roll is almost linear. The toe angle variation for a roll angle of 4.2^o (either side) is negligible. Thus, vehicle will maintain the straight ahead characteristics under roll.

C. Percentage Ackermann

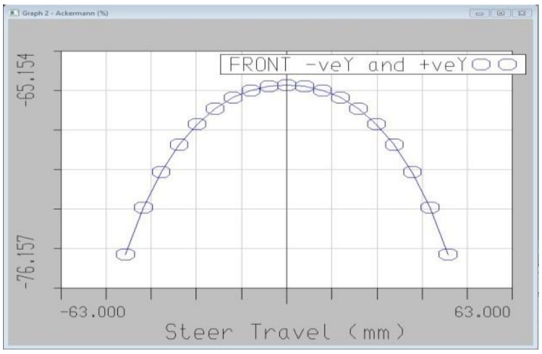

Fig7. Percentage Ackermann v/s steer travel

The Percentage Ackermann for the system varies within a narrow range (65% to 76%). The percentage Ackermann should not be 100% as it leads to loss of steering feel. The driver cannot sense the road if the percentage Ackermann is 100%. For the optimized geometry, a sufficient steering feel will be available to the driver.

D. Scrub Radius

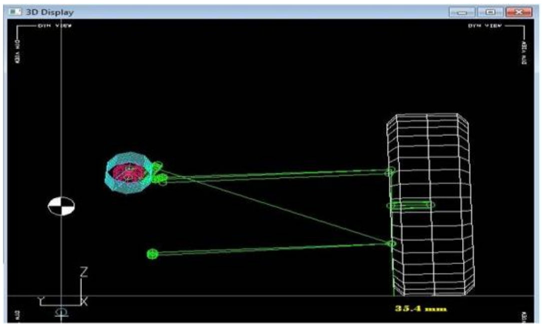

Fig8. Scrub Radius

Scrub radius is the distance between two imaginary points on the road surface - the point of center contact between the road surface and the tire, and the point where the steering-axis center-line contacts the road surface. If these two points intersect at the center of the tire, at the road surface, then the scrub radius is zero. If they intersect below the road surface, scrub radius is positive and vice versa. The scrub radius obtained is a positive with a magnitude of 35.4mm.

II. KNUCKLE ANALYSIS

The knuckle selected is from a formula SAE car. The knuckle is conventionally made of steel .The knuckle has arrangement for mounting the brake caliper. Hence, during braking there will be longitudinal load transfer. The knuckle provides connection for wishbone arms through ball joints at the top and the bottom, leading to lateral load transfer during cornering. The knuckle has spaces cut out in its structure for efficient material utilization. The knuckle can be optimized for weight reduction using an alternate material.

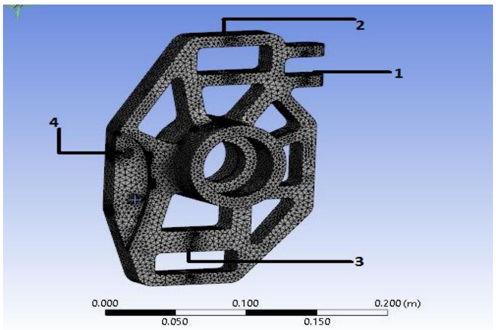

Fig9. Knuckle meshed in ANSYS 15.0

1-Tie Rod
2-Upper wishbone arm
3-lower wishbone arm
4-Brake caliper

The SAE car knuckle is made up of AISI 1020 steel. This knuckle is first analyzed under the conditions of cornering and braking. An alternate material is also considered for the knuckle and separately analyzed under same conditions as the actual knuckle. The alternate material chosen is Aluminium 2011 T3 alloy. The Aluminium alloy was chosen due to its

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light weight and high strength, high rigidity properties. It is expected that this alloy will sustain the working conditions while maintaining the functionality of the knuckle.

TABLE I. MATERIAL PROPERTIES

Material	AISI 1020 Steel	Aluminium 2011T3 alloy
Density	7900 Kg/m^3	2770 Kg/m^3
Poissons ratio	0.29	0.33
Ultimate strength	420 Mpa	310 Mpa
Yield strength	350 Mpa	280 Mpa

Given above is a table showing the properties of both the materials considered for this analysis. The results of the analysis for both the knuckled are discussed below.

A. *Lateral Load Transfer Under Cornering*

The knuckle modeled in Solid Works was analyzed by fixing the central hub and the loads were applied at the wishbone mounting points.

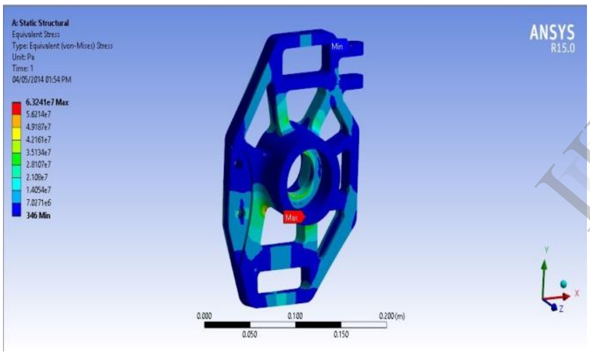

Fig10.Equivalent stress- AISI 2010 steel

The Equivalent (Von-Mises) stress obtained for the steel knuckle during lateral load transfer due to cornering is 63.24 Mpa. This is much below the yield strength of the material. Thus, the knuckle is acceptable.

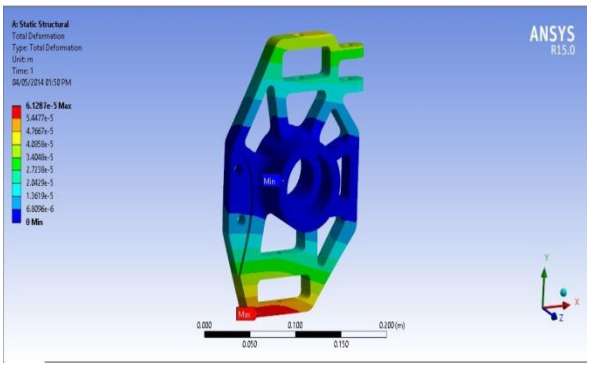

Fig11. Deformation- AISI 2010 steel

The maximum deformation in the steel knuckle is 0.0612mm. This deformation can be considered negligible.

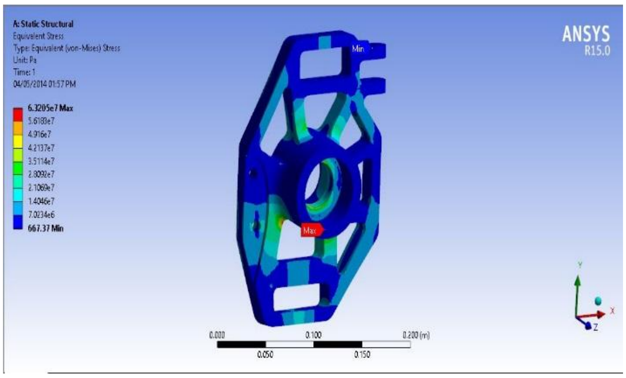

Fig12. Equivalent stress – Al 2011T3 alloy

The equivalent (von-mises) stress in case of the aluminium knuckle under the lateral load transfer condition is 63.20 Mpa. This value is sufficiently below the yield strength of the alloy.

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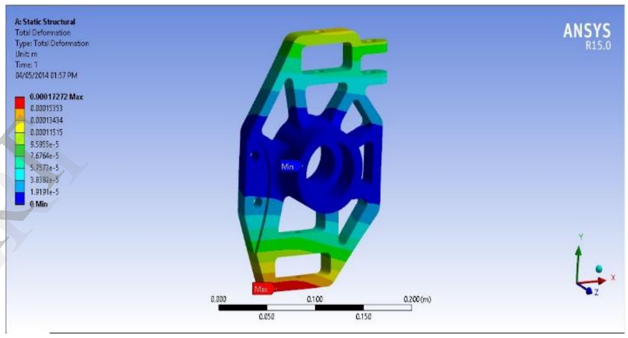

Fig13. Deformation -Al 2011T3 alloy

The maximum deformation in the Al2011T3 knuckle is 0.1mm. The deformation can be considered negligible.

B. *Longitudnal load transfer under braking*

The longitudinal load transfer to the knuckle due to braking is through the brake caliper mounting. The knuckle has holes drilled in to it for mounting the calipers. For this analysis, wishbone mounting points were fixed and the braking torque was applied at the caliper mounting points. This analysis was performed separately by considering each material separately. The results are discussed below.

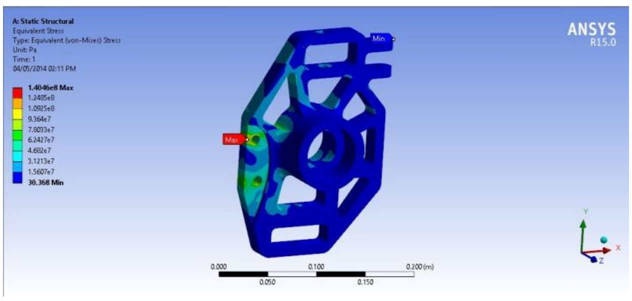

Fig14. Equivalent stress- AISI 2010 steel

The Equivalent (Von-Mises) stress obtained for the steel knuckle during longitudinal load transfer due to braking is 140.46 Mpa. This is much below the yield strength of the material. Thus, the knuckle is acceptable.

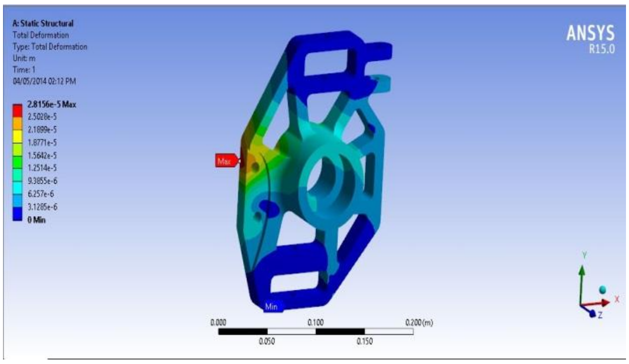

Fig15. Deformation-AISI 2010 steel

The maximum deformation in the steel knuckle is 0.028mm. This deformation can be considered negligible.

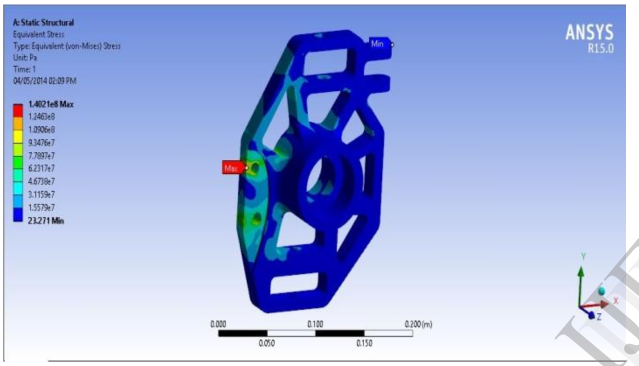

Fig16. Equivalent stress- Al 2011 T3 alloy

The Equivalent (Von-Mises) stress obtained for the Al 2011T3 alloy during longitudinal load transfer due to braking is 140.21 Mpa. This is about 50% of the yield strength of the material (factor of Safety=2). Thus, the knuckle is acceptable.

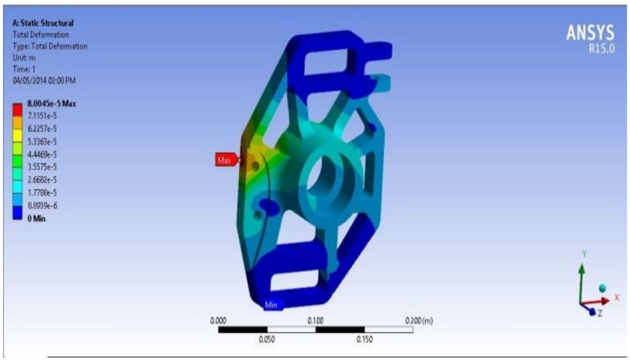

Fig17. Deformation- Al 2011T3 alloy

The maximum deformation in the steel knuckle is 0.08mm. This deformation can be considered negligible.

C. Mass Consideration

The steel knuckle made of the AISI 2010 has a mass of 3.64 Kg, whereas the Al 2011T3 alloy knuckle has 1.28 Kg. As can be seen from the analysis performed for various conditions,the stresses encountered for either materials are similar in magnitude but the mass of the Al 2011T3 alloy knuckle is almost 65% lesser than the steel knuckle. Thus, we can reduce the un-sprung mass of the suspension system making it respond more accurately and rapidly to the road conditions.

III. Conclusions

1. The Double wishbone 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. The knuckle was optimized for material considerations. Thus, the suspension un-sprung masses can be reduced, improving the response of the suspension.
4. The optimized suspension geometry is expected to give comfortable ride.
5. The knuckle can be optimized for lighter and stronger materials like carbon fiber and a composite knuckle is also possible.

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