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Design of Suspension System with Driveshaft as a Camber Governing Link

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Abstract- **Suspension is the system that connects a vehicle (chassis) to its wheels and allows relative motion between the two with the help of suspension linkages, shock absorbers and wheel uprights. Suspension system is responsible for the handling, road holding and ride quality of the vehicle. It is important for the suspension system to keep the road and wheels in contact. In automobile suspension a suspension link or control link is a suspension member, that connects at two points, one point on the frame and other attached to the upright. A link provides pivot and can control only one degree of freedom by itself. Independent suspension system allows each wheel on the same axis to move vertically independently of the other wheel. Independent suspension in off-road typically offers better ride quality and handling characteristics, due to lower un-sprung weight and the ability of each wheel to address the road undisturbed by activities of the other wheel on the vehicle. The available options for independent suspension systems for an All-Terrain Vehicle are mainly: Macpherson Strut, Trailing Arm/Semi-Trailing Arm, Double A-arm Suspension, Multi-link Suspension, H-arm and Camber link. Two link independent suspension system having an H-arm and a camber link is chosen for the application for its simplicity in design and application. An innovative solution to reduce the weight of suspension system by eliminating the camber link and using driveshaft with double Hooke joint as a suspension link which governs the wheel camber throughout the suspension travel. The driveshaft is also responsible for transmitting torque and rotation from the differential to the wheel hub. Double Hooke joint is used to provide sufficient articulation angle to accommodate vertical wheel travel and also acts like a suspension link as it provides only one degree of freedom. The primary aim is to reduce weight as it improves the overall performance of the vehicle i.e. reduction in un-sprung mass, better acceleration, enhanced handling and reduce the overall production cost by eliminating an entire link.**

Index Terms- Camber link, Driveshaft, H-arm, Hooke joint, Parallelogram Geometry, Rear Upright, Suspension System.

I. INTRODUCTION

A two-link suspension system is used in the vehicle where one link is H-arm and drive shaft acts as a camber governing link. The system depicts a four-bar mechanism with simplest geometry and controlled wheel movement characteristics. A parallelogram type of geometry is used to further simplify camber gain and toe gain with wheel movement. Usage of Hooke joint eliminated the need to provide compliance to the half-shaft and does control one degree of freedom which gives driveshaft characteristics of a suspension link. The design is efficient as the torsionally safe shaft also satisfies the tensile strength requirements of the camber link.

Figure 1. Rear Suspension System

II. DESIGN CONSTRAINTS

A. Wheel-base and wheel-track

The wheel base and wheel track define the position of the wheel centre with respect to vehicle body and thus directly affect geometry design.

B. Rear Packaging

The vehicle has a rear mounted engine and is a rear wheel drive with CVT and a two-stage single setting gearbox coupled in line. The chassis design constraints packaging of power-train assembly and

subsequently constraints the chassis end of the driveshaft.
At the wheel the packaging of upright, hub and linkage assembly along with wheel centre position gives a tentative position for wheel-end of driveshaft.

C. Wheel travel
The bump and droop travel values affect the wheel alignment over the wheel travel. This restricts position of linkages and constraint suspension geometry.

D. Length & Articulation angle of driveshaft
The suspension geometry along with ride height, wheel track, wheel base and wheel travel values govern the articulation angle of drive shaft. UV joint used at the ends of drive shaft have a limited articulation angle and this in return constraints the former mentioned parameters and subsequently the design of system.

Figure 2. Driveshaft Angle

III. SUSPENSION GEOMETRY

A two-link suspension with Parallelogram geometry is selected for the rear suspension. The driveshaft acting as camber link consists of two Hooke's joints wherein, they connect gearbox spline to the hub spline through the shaft. For no loss of angular velocity from hub spline to upright spline they should necessarily be parallel to each. This gives rise to the need of parallelogram suspension. H-arm, Driveshaft, upright and the fixed link between H-arm axis and Inner Hooke joint forms a four-bar mechanism with parallel sides.

Figure 3. Parallelogram Suspension Geometry
The geometry is optimized to yield a roll centre of 16" and zero camber and toe change. It is in accordance with the wheel travel values. The total wheel travel is 10" divided as 8" bump and 2" droop.
High rolling stiffness in rear suspension gives over-steer tendency to the car which is desirable characteristic.

Parameter	Value
Centre of Gravity	20.5 in.
Roll centre Height	16 in
Static Camber	0^0
Camber gain	0^0
Static Toe	0^0
Toe gain	0^0
Anti-squat	15%
Roll Stiffness	4.5 deg/g

Table 1. Parameters obtained from suspension geometry

IV. DRIVESHAFT CALCULATIONS

A. Important Parameters:
Lateral Force (F_L): 8000 N
Bump Force (F_B): 6000 N
Maximum Torque (T): 600 N-mm

B. Shaft Diameter Calculations:
On the basis of Tensile failure of shaft due to lateral force:

$$\sigma_t = \frac{F_L}{\frac{\pi}{4} * D^2(1 - C^2)}$$

Where,

$$C = \frac{D}{D_i}$$

On the basis of Shear failure of shaft due to bump force:

$$\tau = \frac{F_B}{\frac{\pi}{4} * D^2(1 - C^2)}$$

On the basis of Torsional shear failure of shaft:

$$\tau = \frac{16\ T}{\pi * D^3(1 - C^4)}$$

C. Yoke dimension calculations:
From the empirical relationship:
Thickness of Yoke:

$$t = 0.75 * d$$

Inner to Inner distance of Yoke:

$$a = 1.5 * d$$

Width of Yoke:

$$b = 1.2 * d$$

D. Yoke shaft diameter calculations:

On the basis of Tensile Failure of fork due to lateral force:

$$\sigma_t = \frac{F_L}{2 * t * (b - d)}$$

On the basis of Crushing Failure of fork due to lateral force:

$$\sigma_c = \frac{F_L}{2 * t * d}$$

On the basis of Shear Failure of fork due to bump force:

$$\tau = \frac{F_B}{2 * t * (b - d)}$$

E. Spline calculations:

Shear stress at pitch diameter of spline resulting from applied torque T:

$$\tau = \frac{2 * T * K_s}{D * z * L * S}$$

Compressive stress on teeth resulting from applied torque T:

$$\tau = \frac{4 * T * K_s}{D * z * L * h}$$

Where,

K_s =Service factor

D = Pitch diameter of spline (mm)

S =Tooth thickness (mm)

z = Number of teeth

IV. LINKAGE DESIGN

The system is simulated on 'Lotus Shark' and geometric parameters are optimized. Linkages are designed accordingly.

A. H-arm

H-arm and camber link (drive-shaft) are designed as such to provide structural integrity as well as functional conformance. The structure of H-arm fits with the packaging of vehicle avoiding interference with other components. Ribs are provided in the structure for strength and triangulations are given to reduce weight.

Figure 4. H-arm

B. Upright

Rear Upright habitats common assembly point for H-arm and Suspension strut. It houses Wheel hub and drive shaft assembly too. It is made up of Aluminium 6061-T6 and optimized for weight and strength.

Figure 5. Rear upright

C. Driveshaft

Driveshaft is designed to transmit power to the wheel as well as to act as a camber governing link. Double Hooke Joint is used to improve articulation movement and restrict axial movement of the shaft. EN-36 Steel is used for Driveshaft.

Figure 6. Double Hooke Joint Driveshaft

VI. FINITE ELEMENT ANALYSIS

The model was prepared using 3-D technique in SOLIDWORKS. The model is then transferred to iges format and exported into ANSYS. Finite element analysis is applied. It is a computational tool for performing engineering analysis. It includes the use of mesh generation techniques for dividing a complex problem into small elements. The meshing is adequately done to obtain the accurate results while computation. Model is meshed with 3-D element type of tetrahedron shape with element size of 1mm where number of elements are 148249 and number of nodes are 599795.

A. Loading conditions:

Torque of 600 N-mm is transmitted to driveshaft via splines at its end hence it is subjected to torsional shear stresses. Lateral force of 8000 N is applied axially. Bump force of 6000 N is applied in transverse direction at its ends applying tensile, bending and shear stresses.

B. Results:

As EN -36 material is ductile, Von Mises yield criterion (also known as maximum distortion energy criterion) is used. Results from ANSYS shows that maximum stress operated in component is contoured portion of Driveshaft part as shown in Figure 8. The maximum stress is 709.5 MPa which is less than yield strength of EN-36 (900MPa). Hence the values are within safety limits.

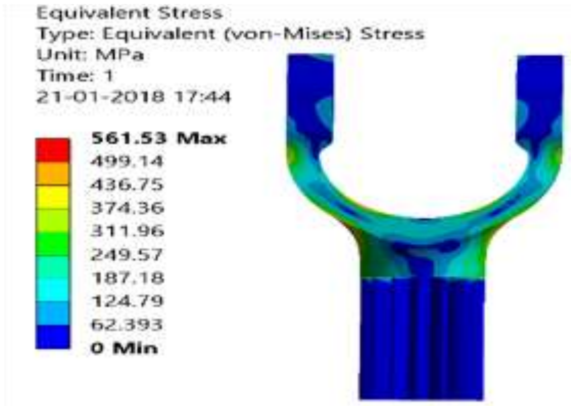

Figure 7. Gearbox side Universal joint

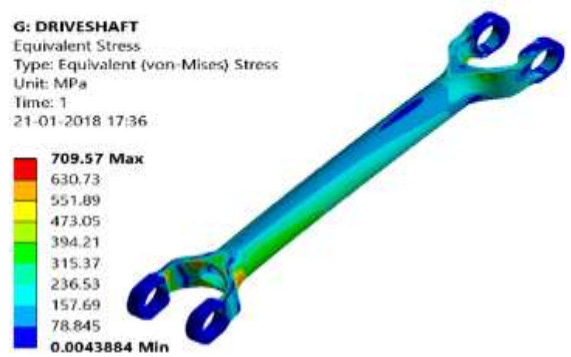

Figure 8. Driveshaft

Figure 9. Wheel side Universal joint

V. CONCLUSION

The proposed design is a lighter alternative to a conventional suspension design. There is reduction in unsprung mass by 2.2kg (14%) from previous design. Reduction in number of component and better serviceability with lower maintenance are the highlights. The suspension geometry achieved is at par with conventional designs with suspension parameters within acceptable range. On the basis of study on modelling and analysis technology using ANSYS, the FEA of the system was realized. The stress and deformation of the model were under control. The feasibility of the system was hence verified.

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