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Design and Analysis of Wishbone Suspension System

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ABSTRACT

An ATV is the vehicle which is designed to move through all terrains. We are using this vehicle for various range of purposes such as military purpose, rescue purpose during natural calamities and also for forest inspections. Suspension system of this vehicle should be strong enough so that it will give better ride quality and maximum comfort to the driver. Wishbone suspension system is selected and is designed in Unigraphics Software. A present work is focused in the field of the impact load, deformation of material, stress for ATV vehicle for improving the stability and handling of vehicle to minimizing the un-sprung mass, better durability. After designing the hard points are received and after that we selected the material to Analyze the various stresses acting upon it by ANSYS. This project aims at selecting a suspension system of ATV which will capable of handling at rough terrains.

Keywords: *ATV vehicle, Wishbone suspension system, impact load, deformation of material, stress*

INTRODUCTION

All-Terrain Vehicle (ATV) is a vehicle that travels on low pressure tires driven using handlebar or steering wheel for steering control. Although, ATVs were first designed only for a single operator but now-a-days many companies have developed ATVs with two or more seats. In most of the countries around the globe, these vehicles are banned on streets. All-Terrain Vehicles is a package of different systems that are designed to enrich the performance and to provide comfort to the driver. Different systems include chassis, steering system, suspension system, braking system and drive train. All these mentioned systems are inter-dependent. Failure of a single system or a part may lead to the death of the operator or driver. ATVs are also popular for their good aesthetics and their sporty look. Suspension system of as All-Terrain Vehicle is one of the most critical system that needs to be designed for better stability and comfort for the operator. Suspension system is generally designed in relationship with the steering system. [1]

Suspension System

Suspension system is referred to the springs, shock absorbers and linkages that connect the vehicle to the wheels and allows relative motion between the wheels and the vehicle body. Suspension system also keeps the driver or operator isolated from bumps, road vibrations, etc. Also, the most important role played by the suspension system is to keep the wheels in contact with the road all the time. Good suspension system and better handling is the characteristic of a good All-Terrain Vehicle (ATV). One of the functions of suspension system is to maintain the wheels in proper steer and camber attitudes to the road surface. It should react to the various forces that act in dynamic condition. These forces include longitudinal (acceleration and braking) forces, lateral forces (cornering forces) and braking and driving torques. It should resist roll of the chassis. It should keep the wheels follow any uneven road by isolating the chassis from the roughness of the road. All the dynamic parameters are to be considered while designing the suspension system, especially the behavior of the suspension for various loading cases. Besides the dynamic parameters, other factors considered in design process are cost, weight, package space, manufacturability, assembly, etc. [2]

DESIGN INPUT AND CALCULATIONS

Stiffness of spring

To design a spring, we need first of all we need to select the stiffness of the value of the spring. [3]

Stiffness of the spring= 10 (N/mm)

Material of the spring

After performing a brief study, our team has come to a conclusion that spring steel economically safe in all the aspects.

SPRING MATERIALS (ROUND WIRE)

Material	**Commercially Available Specification**	**Nominal Chemistry**	**Density (lb/in3)**	**Minimum Tensile Strength (psi x 106)**	**Modulus of Elasticity (E) (psi x 106)**	**Modulus in Torsion (G) (psi x 106)**	**Maximum Operating Temperature**	**Method of Manufacture / Primary Use**
Spring steel	ASTM A228	C 0.70 - 1.00% Mn 0.20 - 0.60%	0.284	230 – 399	30.0	12 (up to 0.100") 11.75 (over 0.100")	250 °F	Highest quality cold drawn, high carbon wire. Good surface finish.

Hard Drawn MB	ASTM A227	C 0.45 - 0.85% Mn 0.60 - 1.30%	0.284	Class I: 147 - 283 Class II: 171 – 324	30.0	11.5	250 °F	Cold drawn. Average stress applications. Can be readily electroplated.
Oil Tempered MB	ASTM A229	C 0.55 - 0.85% Mn 0.60 - 1.20%	0.284	Class I: 165 - 293 Class II: 191 – 324	30.0	11.5	250 °F	Cold drawn and heat treated before fabrication. Slight scale on surface. General purpose spring wire.

Wire diameter
From the above table wire diameter of the spring is taken as
Wire diameter d= 8mm
Spring diameter (D): We know that**,** From the equation

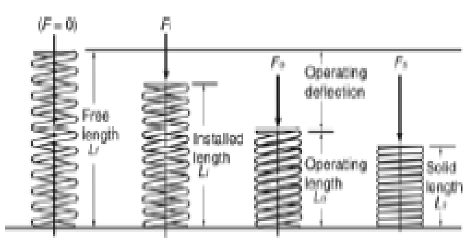
$$k=\frac{G*d^4}{8*D*n}\ \text{N/mm}$$

Where
G= modulus of rigidity
d=spring wire dia
D=spring dia
n= number of turns

k= stiffness of the spring

$$10=\frac{79289*8^4}{8*D^3*14}$$

D=68.35 mm≅68mm Solid length of spring



$$L_{solid}=(n_a+2)d$$

Where
n_a=number of turns
d=diameter of wire

L_{solid}=(14+2)8

L_{solid}=16*8

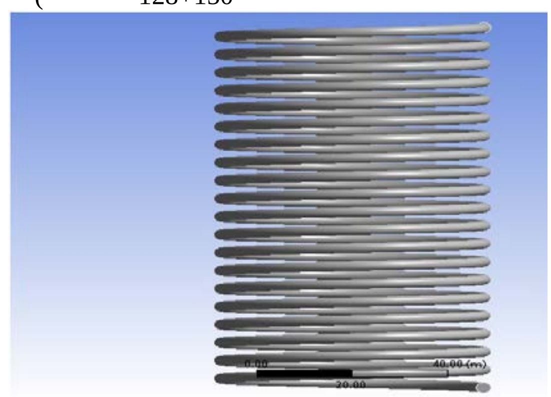
L_{solid}=128 mm

Fig Design of spring

Free length of spring (L_{free})

L_{free} =solid length +displacement for maximum force (i.e. 1500N)
We know that , F=k*x
Where
X= displacement for maximum force
L_{solid}=128 mm

1500=10*x
X=150
(L_{free})=128+150



Free length of spring (L_{free})=278mm
Coil pitch

Coil pitch= $\frac{\text{Free length of spring } (L_{free})}{n_a\text{=number of turns}}$

Coil pitch= $\frac{278}{14}$

Coil pitch=19.85 mm

Turn angle $\Theta=\tan^{-1}\frac{coil\ pitch}{\pi D}$

Coil pitch=19.85 mm
D=68 mm

$\Theta=\tan^{-1}\frac{19.85}{\pi*68}$

$\Theta=6^o$ degree

Length of wire

Length of wire=π*D($\frac{n_a}{\cos\Theta}+2$)
D=68 mm
$\Theta=6^o$ degree

Length of wire=[π*68($\frac{11}{\cos 6}+2$)]
Length of wire= 2642.17 mm

Maximum force (F_{max})

F_{max}=k(L_{free} - L_{solid})

F_{max}=10(278-128)

F_{max} ≅1500 N

Torisional force

$$T_{max} = \frac{8*W*D}{d^3*\pi}(F_{max})$$

W=size factor

$W = \frac{4c-1}{4c-4} + \frac{0.615}{c}$ ($c = \frac{D}{d}$)

($c = \frac{68}{8} = 8.5$)

$W = \frac{4*8.5-1}{48.5-4} + \frac{0.615}{8.5}$

W=1.167

$T_{max} = \frac{8*1.167*70}{8^3*\pi}$ (1500)

T_{max} =609.5 $\cong 610$ N/mm^2

Shock absorber angle

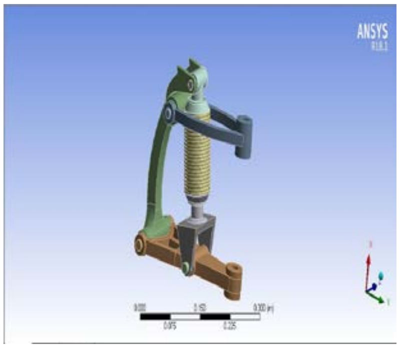
From the table

Shock Angle	Angle Correction Factor (ACF)
10°	.98
15°	.96
20°	.94
25°	.91
30°	.87
35°	.82
40°	.77
45°	.71

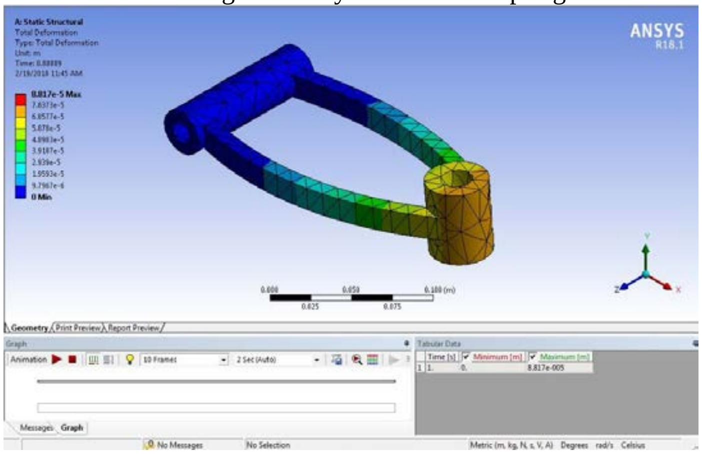
From above table and the calculations, we have done by taking formula we obtain the shock angle 15 degrees which has correction factor as 0.96 which results for good suspension system

RESULT AND DISCUSSIONS

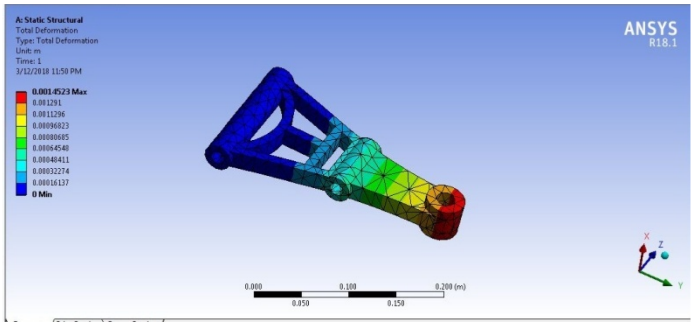
In previous chapter we studied about designing suspension system where stiffness of the spring is calculated and analysis is done for variant loads and obtained result is shown as follows. In this chapter, we also see the analysis of Upper Arm, Lower Arm and spring of von-misses stress. Results are shown as follows



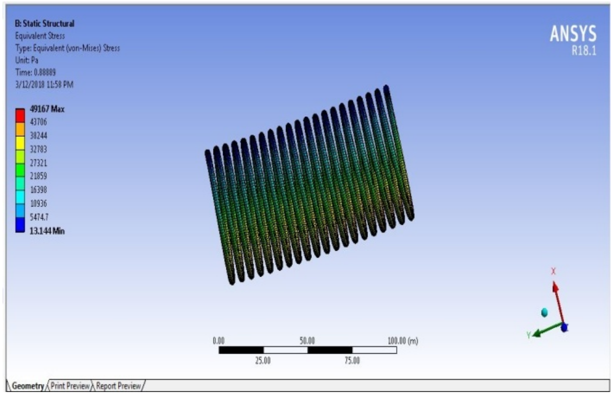
Here is the Fully Designed and Assembled part of Wishbone suspension system. Compare to older type of wishbone systems this model is said to be more appropriate model. In this Project we have done some analysis methods on Upper arm and Lower Arm and also a Design & Analysis of Helical Spring.



In the above picture it shows the Upper Arm analysis equivalent stress has ma of 8.8175 max and minimum of 0.1254 min. Here for given load there is total safety with blue colour and only red colour is scene which carries maximum loads when bump is obtained. [4]



In the above picture it shows the Lowe Arm analysis equivalent stress has ma of 8.0014 max and minimum of 0.2354 min. Here for given load there is total safety with blue colour and only red colour is scene which carries maximum loads when bump is obtained.

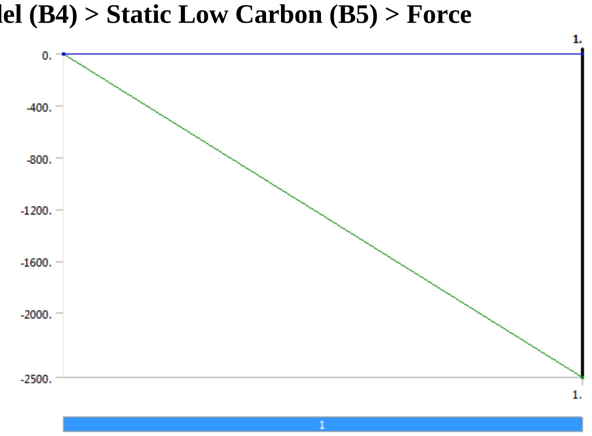


In this Helical Spring analysis equivalent stress has ma of 49167 max and minimum of 13.144 min. Where it can be seen in above figure. Here for given load there is total safety with blue colour and only red colour is scene which carries maximum loads when bump is obtained.

Model (B4) > Static Low Carbon (B5) > Loads

Object Name	*Force*	*Fixed Support*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	1 Face	
Definition		
Type	Force	Fixed Support
Define By	Components	
Coordinate System	Global Coordinate System	
X Component	0. N (ramped)	
Y Component	-2500. N (ramped)	
Z Component	0. N (ramped)	
Suppressed	No	

FIGURE 1
Model (B4) > Static Low Carbon (B5) > Force



Suppressed	No		
Results			
Minimum	0. m	26.289 Pa	3.842e-010 m/m
Maximum	2.8285e-002 m	49167 Pa	2.4597e-007 m/m
Minimum Occurs On	Solid		
Maximum Occurs On	Solid		
Minimum Value Over Time			
Minimum	0. m	13.144 Pa	1.921e-010 m/m
Maximum	0. m	26.289 Pa	3.842e-010 m/m
Maximum Value Over Time			
Minimum	1.4143e-002 m	24584 Pa	1.2299e-007 m/m
Maximum	2.8285e-002 m	49167 Pa	2.4597e-007 m/m
Information			
Time	1. s		
Load Step	1		
Substep	2		
Iteration Number	2		
Integration Point Results			
Display Option		Averaged	
Average Across Bodies		No	

FIGURE 2
Model (B4) > Static Low Carbon (B5) > Solution (B6) > Total Deformation

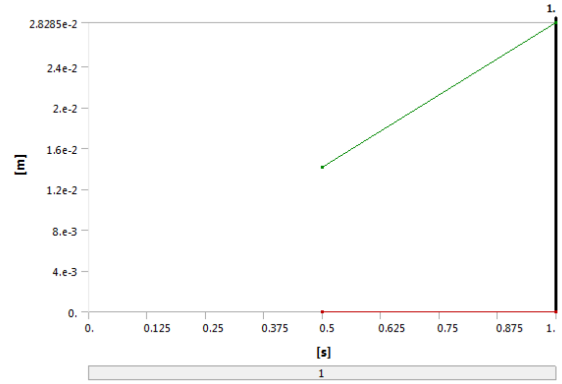


TABLE 1
Model (B4) > Static Low Carbon (B5) > Solution (B6) > Total Deformation

Time [s]	Minimum [m]	Maximum [m]
0.5	0.	1.4143e-002
1.		2.8285e-002

FIGURE 3
Model (B4) > Static Low Carbon (B5) > Solution (B6) > Equivalent Stress

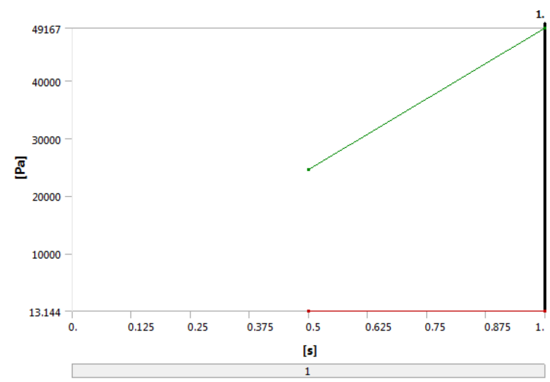


TABLE 2
Model (B4) > Static Low Carbon (B5) > Solution (B6) > Equivalent Stress

Time [s]	Minimum [Pa]	Maximum [Pa]
0.5	13.144	24584
1.	26.289	49167

FIGURE 4
Model (B4) > Static Low Carbon (B5) > Solution (B6) > Equivalent Elastic Strain

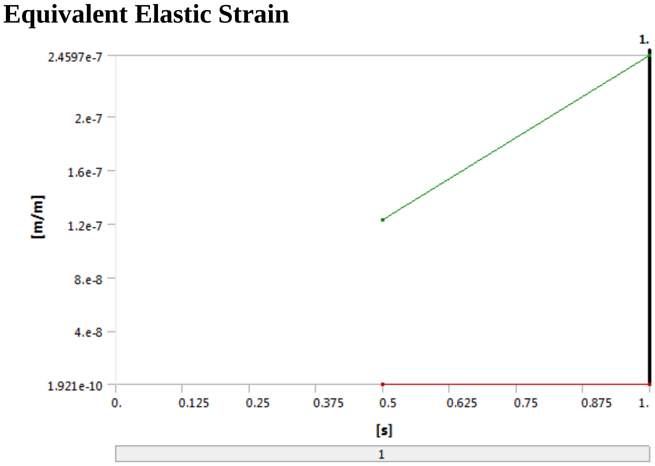


TABLE 3
Model (B4) > Static Low Carbon (B5) > Solution (B6) > Equivalent Elastic Strain

Time [s]	Minimum [m/m]	Maximum [m/m]
0.5	1.921e-010	1.2299e-007
1.	3.842e-010	2.4597e-007

TABLE 4
Low Carbon Steel > Constants

Density	7850 kg m^-3
Isotropic Secant Coefficient of Thermal Expansion	1.2e-005 C^-1
Specific Heat	434 J kg^-1 C^-1
Isotropic Thermal Conductivity	60.5 W m^-1 C^-1
Isotropic Resistivity	1.7e-007 ohm m

TABLE 5
Low Carbon Steel > Compressive Yield Strength

Compressive Yield Strength Pa
2.5e+008

TABLE 6
Low Carbon Steel > Tensile Yield Strength

Tensile Yield Strength Pa
2.5e+008

TABLE 7
Low Carbon Steel > Tensile Ultimate Strength

Tensile Ultimate Strength Pa
4.6e+008

TABLE 8
Low Carbon Steel > Isotropic Secant Coefficient of Thermal Expansion

Zero-Thermal-Strain Reference Temperature C
22

TABLE 9
Low Carbon Steel > Alternating Stress Mean Stress

Alternating Stress Pa	Cycles	Mean Stress Pa
3.999e+009	10	0
2.827e+009	20	0
1.896e+009	50	0
1.413e+009	100	0
1.069e+009	200	0
4.41e+008	2000	0
2.62e+008	10000	0
2.14e+008	20000	0
1.38e+008	1.e+005	0
1.14e+008	2.e+005	0
8.62e+007	1.e+006	0

TABLE 10
Low Carbon Steel > Strain-Life Parameters

Strength Coefficient Pa	Strength Exponent	Ductility Coefficient	Ductility Exponent	Cyclic Strength Coefficient Pa	Cyclic Strain Hardening Exponent
9.2e+008	-0.106	0.213	-0.47	1.e+009	0.2

TABLE 11
Low Carbon Steel > Isotropic Elasticity

Temperature C	Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa
	2.e+011	0.3	1.6667e+011	7.6923e+010

CONCLUSION

The present work is optimum design and analysis of a suspension spring for motor vehicle subjected to static analysis of helical spring the work shows the strain and stress response of spring behavior will be observed under prescribed or expected loads. We have known that camber, caster plays vital role for any suspension such that positioning of spring and damper also made 15 degrees for better suspension system. We have designed the suspension spring and analyzed it on Ansys Software where we found the safe result to desired condition of ride.

The suspension system can be further modified for decreasing the weight and cost. Transverse leaf spring can be used to decrease the weight of the suspension assembly. Pneumatic suspensions can be incorporated in the future for better performance.

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