Design and Dynamic Characteristics of Suspension system for All-terrain vehicle

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Abstract:

Suspension performance plays an important role in All-Terrain vehicles (ATV). The intention of the work is to fabricate the suspension system for the ATV using the multi body dynamics approach using LOTUS SHARK software. The components of the suspension are modeled in solid works and analysis is carried out in Ansys workbench. The front and rear suspensions are double wishbone type due to its stability and good distribution of loads. Using the multi body dynamics we will study the camber and toe angles with respect to wheel travel. The cambers are of different type's i.e. Positive camber, negative camber and zero camber. The camber and toe angles were studied in LOTUS software and based on the performance, particular angles of toe and camber were selected. The important dynamic characteristics like ride rate, natural frequencies, roll rate were taken to design the spring stiffness and factor of safety for crucial components that include in suspension. The materials like Al-6082-T6, Structural steel and mild steel were chosen to carry out the analysis in suspension components. The suitable hard points were chosen to reduce the bump steer geometry, by simulating the whole front and rear wheel alignment in multi body dynamics software, we will be able to know the camber and toe angles with respect to wheel travel. Finally the components are manufactured and assembled to vehicle chassis.

Keywords : Spring stiffness, Ride rate, Roll rate, Natural frequency, Camber angle, Toe in and out angles, Factor of safety.

Introduction:

ATV'S are the vehicles that travel on low pressure tyres. ATV'S were first designed only for a single operator, but nowadays many companies have developed ATV'S with two or more seats. ATV'S are the combination of different systems that are designed to enhance endurance ability of the vehicle. These units include steering, suspension system, power train, brakes, and chassis. The suspension system is inter linked with steering system, where bump steer geometry should be solved to reduce the stresses on tie rods and steering arms due to high wheel travel. The wishbones lengths and ground clearances play a major part in vehicles dynamic characteristics like cornering and stability during vehicle drives in irregular cross ditches, for better comfort of the driver the ground clearance and shock travel have important role. Twin wishbone of unequal length arms were chosen to meet the requirements and for easy adjusting of the cambers as per the drive. Though this model need more space and material to manufacture, it gives a tough stability during high speed cornering and cross bumps. The complete vehicle suspension was simulated using multi body dynamics (Lotus shark) to ensure the design was secure and stable.

Assumptions and Modeling:

Before the modeling of the suspension some basic inputs are necessary like, ground clearance, track width, wheel base, chassis dimensions, tyre dimension, steer travel, wheel travel, and knuckle mounting points. This should be given as input to lotus software to simulate the geometry for the results. The standard dimensions were chosen from the SAE (Society of Automotive Engineers) rule book and choosing ground clearance the CAD model is drawn in solid works and lengths of the control arms were obtained and were designed and modeled in solid works and analysis was carried out in Ansys workbench. While modeling the A-arm the installation ratio is taken as 0.75 for mounting the shock on lower A-arm. The kingpin inclination is taken as 7 degrees while for the knuckle due to its positive steering advantage. While clearing GO NO GO test, the track width of the vehicle should not exceed more than 64 inches. So we have designed our vehicle to track width of 62 inches out to out and therefore, we have optimized the front cabin width for obtaining good wishbone lengths. Based up on the vehicle dynamics principles by Milliken and Milliken author, we have designed our wishbones based up on the types like parallel wishbones and unequal wishbones and three dimensional view layout of the front cabin, wishbones hard point positions those we have got it from stark suspension simulation software by doing multiple simulations giving different hard points to avoid camber angles, toe-in and toe-out angles adjustment and offset of wheel while simulating the suspension. However, taking all these constraints in to criteria and after conforming the 3D model view of suspension and lengths of wishbones, then we had designed the wishbones upper and lower arm that has suspension mountings were modeled in solid works and analysis is done in ansys workbench and following these loading conditions, material and factor of safety.

Suspension Calculations:

W1 = 230.35 lbs W2 = 230.35 lbs
W_F (55%) = 460.7 lbs

W3 = 188.49 lbs W4 = 188.49 lbs
W_R (45%) = 376.99 lbs

W= weight on 4 wheels
W_T = 837.7 lbs

T_F = 4.58 ft (Track width- front) T_R = 4.58 ft (Track width- Rear) l = 9 ft (Wheel Base)

h(height of CG from ground) = 1.98 ft; H(CG to Roll-Axis height) = 1.47 ft; α = -10 deg (banking inclination); R = -600 feet (radius to vehicle central axis, measured horizontally); V = 127.6 kmph = 116.28 ft/sec

Roll center heights from ground for Front and Rear:
measured using a line diagram layout.

Z_{RF} = 0.509 feet, Z_{RR} = 0.65 feet.

We assume the Roll- Rates as:

Front: $K_{\emptyset F}$ = 27217.6 lb-ft/rad Rear: $K_{\emptyset R}$ = 19441.176 lb-ft/rad

Calculation of CG position from Front end and Rear end of an ATV:

b = W_F*l/W_T = 460.7*9/837.7 = 4.94 ft a = l-b = 9-4.94 = 4.05 ft

$A_\alpha = V^2/R*g = (116.28)^2/(-600*32.2) = -0.699$ g's

$A_Y = A\alpha \cos(\alpha) - \sin(\alpha)$ = -0.699*Cos(-10) – sin(-10) = -0.514 g's

Where A_α = Horizontal Lateral acceleration; A_Y =Lateral acceleration in the car axis system.

The operative weight of ATV caused by banking inclination:

$W' = W (A_\alpha \sin(\alpha) + \cos(\alpha))$

= 837.7* (-0.699 sin (-10) +cos(-10)) = 1021.44 lbs

The Effective front and rear axle weights:

W_F' = W' *b / l = (1021.44 *4.94)/9 = 560.65 lbs

W_R' = W'*a / l = (1021.44*4.05) / 9 = 459.64 lbs

The Roll gradient is:

$\emptyset/A_Y = -W*H / (K_{\emptyset F}+K_{\emptyset R})$ = -837.7 * 1.47 /(30000+20000) = -0.024

$\emptyset$ = -0.048 * -0.514 = 0.024 rad/g Where $\emptyset$ = vehicle spin angle.

The fore-end and back lateral load shift ascribed to lateral acceleration:

$W_F = A_Y * W/t_F[\ [H* K_{\emptyset F}/(K_{\emptyset F}+K_{\emptyset R})] + [(b/l)*Z_{RF}]]$

= -0.514 *837.7/4.58 [[1.47*30000/(30000+20000)] + [(4.94/9)*0.509]] = -108.99 lbs

$W_R = A_Y * W/t_R[\ [H* K_{\emptyset R}/(K_{\emptyset F}+K_{\emptyset R})] + [(b/l)*Z_{RR}]]$

= -0.514 *837.7/4.58 [[1.47*20000/(30000+20000)] + [(4.05/9)*0.65]] = -82.77 lbs

Individual Wheel (tire) Loads:(in lbs

Front outside W_{Fo} = 560.65/2 + 108.99 = 389.31

Front inside W_{FI} = 560.65/2 -108.99 = 171.

Rear Outside W_{RO} = 459.64/2 +82.77 = 312.

Rear Inside W_{RI} = 459.64/2 – 82.77 = 147.05

Static loads measured on level ground:

W_{Fo} = 389.31-230.35 = 158.96

W_{FI} = 171.33 -230.35 = -59.02

W_{RO} = 312.59 – 188.49=124.1

W_{RI} =147.05 – 188.49 = -41.44

Ride rates:	**Natural Frequencies:**
Front Ride Rate: K_{RF}= W_{FO}/6 K_{RF} = 158.96 / 6 = 26.49 lb/in Rear Ride Rate: K_{RR}= W_{RO}/6 = 124.6/6 = 20.68 lb/in	W_F = 1/2π * Sqrt [K_{RF} * 12*32.2 / W2] = 1/2π * Sqrt [26.49 * 12*32.2 / 230.35] = 1.060 Hz = 63.65 cpm W_R = 1/2π * Sqrt [K_{RR} * 12*32.2 / W4] = 1/2π * Sqrt [20.68 * 12*32.2 / 188.49] = 1.036 Hz = 62.17 cpm

Roll – Rates:	**Wheel Rate :**
$K_{\phi F}$ = 12* K_{RF} * t_F^2 /2 = 12* 26.49 * 4.5 ^{2}/2 = 3218.53 lb-ft/rad $K_{\phi R}$ = 12* K_{RR} * t_R^2 /2 = 12* 20.68 * 4.58^2/2 = 2512.62 lb-ft/rad	K_T (Vertical Tire Rate Stiffness) = -42.44 lb/in K_{WF} = K_{RF} * K_T/(K_T – K_{RF}) = 26.49 * -42.44 /(-42.44-26.49) = 16.31 lb/in K_{WR} = K_{RR}* K_T/(K_T – K_R) = 20.68* -42.44 /(-42.44-20.68) = 14.06 lb/in

Spring Rate/Stiffness: Installation Ratio = 0.75

K_{SF} = K_{WF}/IR^2 = 16.31 / (0.75)^2 = 29N/mm K_{SR} = K_{WR}/IR^2 = 14.06 / (0.75)^2 = 25N/mm

Spring calculations for an ATV vehicle: To determine spring rate, d = coil Diameter, D outer = total width of spring, D = Mean Diameter, E = Young's Modulus, G = Shear Modulus, L free = allowed extent of spring, k = stiffness, N_A= Active Coils, Spring Index C = D/d; C for most ATV vehicles would be 9-12

1st ITERATION :	
<u>FRONT</u>	**<u>REAR</u>**
Assume C = 9, d = 10mm	Assume C = 9, d = 10mm
D = 90 mm	D = 90 mm
K_W= {[4C – 1] / [4C – 4]} + (0.615/C)	K_W= {[4C – 1] / [4C – 4]} + (0.615/C)
= {[4*9 -1]/ [4*9 – 4]} + (0.615 / 9) = 1.165	= {[4*9 -1]/ [4*9 – 4]} + (0.615 / 9) = 1.165
τ_{Max} = 8* K_W* D*F_{MAX}/(π*d^3)	τ_{Max} = 8* K_W* D*F_{MAX}/(π*d^3)
τ_{Max} = 8 * 1.165 *90 * 2048.2 / (π * 10^3)	τ_{Max} = 8 * 1.165 *90 * 1675.8 / (π * 10^3)
= 546.86 Mpa	= 447.43 Mpa
K = G * d^4/(8*D^3*N_A)	K = G * d^4/(8*D^3*N_A)
29 = 79300 * 10^4/(8*90^3 *N_A)	25 = 79300 * 10^4/(8*90^3 *2*N_A)
N_A = 5	N_A = 3
Inactice coils = 4, Total coils = 9	Inactice coils = 4, Total coils = 7
Solid length = n*d = 9*10 = 90 mm	Solid length = n*d = 7*10 = 70 mm
Free Length = 90 + 152.4 = 242.4 mm	Free Length =solid length +spring travel= 70 + 152.4 = 222.4 mm
Pitch = 242.4 /5 = 48.48 mm	Pitch = coil free length /Active coils = 242.4 /3 = 80.8 mm
Stress Concetration Factor,K_S =1 + (1/C)	Stress Concetration Factor,K_S =1 + (1/C)
= 1 + (1/9)	= 1 + (1/9) = 1.11

= 1.11

Wahl Correction factor, K_W = 1.165

Fm = (Minimum Load +Maximum Load)/2

Fa = (Maximum Load – Minimun Load)/2

Mean Load F_m = 2048.2+ 1024.1 /2 = 1536.15 N

Stress Amplitude = Fa = 2048.2 – 1024.1/2 = 512.05 N

τ_m = 1.11 *8 * 2324.42 *90 /(π * 10^3)

= 591.31

τ_a = 1.165 *8*303.17 * 90 /(π * 10^3)

= 80.94

W_T = 380 kgs

W_F = 380 * 55% = 209 Kg's at Front

For One Wheel = 209 /2 = 104.5 kg's * 9.8 = 1024.1 N (Static Load)

2 g value,Maximum Applied Load = 2 * 1024.1 = 2048.2 N

Yield Point, $\tau_Y = \sigma_{ult}$*0.51 = 675.2 Mpa.

Endurance Limit, $\tau_e = \sigma_{ult}$*0.2 = 264.8 Mpa.

From Soderberg's Equation

1/FS = ($\tau_{m/}$ τ_Y) + $\tau_{a/}$ τ_y[(2* τ_Y/ $\tau_{e)}$ – 1] = 591.31/675.2 + 80.94/675.2[(2*675.2/264.8) – 1]

FS = 0.73

2nd ITERATION :

FRONT :

C = 8 , d = 11mm ,

D = 88 mm

K_W= {[4C – 1] / [4C – 4]} + (0.615/C)

= {[4*8 – 1]/ [4*8 – 4]} + (0.615 / 8) = 1.18

τ_{Max} = 8* K_W* D*F_{MAX}/(π*d^3)

τ_{Max} = 8 * 1.18 *88 * 2048 / (π * 11^3)

= 406.87 Mpa

Wahl Correction factor, K_W = 1.165

Fm = (Minimum Load +Maximum Load)/2

Fa = (Maximum Load – Minimun Load)/2

Mean Load F_m = 1675.8 + 837.9 /2 = 1256.85 N

Stress Amplitude = Fa = (1675.8 - 837.9) /2 = 418.95 N

τ_m = K_S * 8 * F_m *D /(π * d^3)

τ_m = 1.11 *8 * 1256.85 *90 /(π * 10^3) = 319.73

τ_a = K_W * 8 * F_a *D /(π * d^3)

τ_a = 1.165 *8*418.95 * 90 /(π * 10^3) =111.85

W_T = 380 kgs

W_F = 380 * 45% = 171 Kg's at Front

For One Wheel = 171 /2 = 85.5 kg's * 9.8 = 837.9 N (Static Load)

2 g value,Maximum Applied Load = 2 * 837.9 = 1675.8 N

Yield Point, $\tau_Y = \sigma_{ult}$*0.51 = 675.2 Mpa.

Endurance Limit, $\tau_e = \sigma_{ult}$*0.2 = 264.8 Mpa.

From Soderberg's Equation

1/FS = ($\tau_{m/}$ τ_Y) + $\tau_{a/}$ τ_y[(2* τ_Y/ $\tau_{e)}$ – 1] = 319.73/675.2 + 111.85/675.2[(2*675.2/264.8) – 1]

FS = 0.87

REAR:

C = 8 , d = 11mm ,

D = 88 mm

K_W= {[4C – 1] / [4C – 4]} + (0.615/C)

= {[4*8 - 1]/ [4*8 – 4]} + (0.615 / 8) = 1.18

τ_{Max} = 8* K_W* D*F_{MAX}/(π*d^3)

τ_{Max} = 8 * 1.18 *88 * 1675.8 / (π * 11^3)

= 332.92 Mpa

K = G * d^4/(8*D^3*N_A)

$K = G * d^4/(8*D^3*N_A)$

$29 = 79300 * 11^4/(8*88^3*N_A)$

$N_A = 7$

Inactive coils = 4, Total coils = 11

Solid length = n*d = 11*11 = 121 mm

Free Length = solid length + Spring travel = 121 + 152.4 = 273.4 mm

Pitch = coil free length / Active coils = 242.4 /7 = 34.62 mm

Stress Concentration factor, $K_S = 1 + (1/C)$

$= 1 + (1/8) = 1.125$

Wahl Correction factor, $K_W = 1.18$

Fm = (Minimum Load +Maximum Load)/2

Fa = (Maximum Load – Minimun Load)/2

Mean Load F_m = (2048.2 + 1024.1) /2 =1536.15 N

Stress Amplitude = Fa = (2048.2 – 1024.1) /2 = 512.05 N

$\tau_m = K_S * 8 * F_m * D /(\pi * d^3)$

$\tau_a = K_W * 8 * F_a * D /(\pi * d^3)$

$\tau_m = 1.125 *8 * 1536.1 *88 /(\pi * 11^3) = 290.94$

$\tau_a = 1.18 *8*512 * 88 /(\pi * 11^3) = 101.71$

W_T = 380 kgs

W_F = 380 * 55% = 209 Kg's at Front

For One Wheel = 209 /2 = 104.5 kg's * 9.8

= 1024.1 N (Static Load)

2 g value,Maximum Applied Load = 2 * 1024.1 = 2048.2 N

Yield Point, $\tau_Y = \sigma_{ult} *0.51 = 675.2$ Mpa.

Endurance Limit, $\tau_e = \sigma_{ult} *0.2 = 264.8$ Mpa.

From Soderberg's Equation

$1/FS = (\tau_m / \tau_Y) + \tau_a / \tau_y [(2* \tau_Y / \tau_e) – 1] = 290.94/675.2 + 101.71/675.2[(2*675.2/264.8) – 1]$

FS = 1

$25 = 79300 * 11^4/(8*88^3*N_A)$

$N_A = 8$

Inactive coils = 4, Total coils = 12

Solid length = n*d = 12*11 = 132 mm

Free Length = Solid Length + Spring travel = 132 + 152.4 = 284.4 mm

Pitch = Coil Free Length /Active Coils = 284.4 /8 = 35.55 mm

Stress Concentration factor, $K_S = 1 + (1/C)$

$= 1 + (1/8) = 1.125$

Wahl Correction factor, $K_W = 1.18$

Fm = (Minimum Load +Maximum Load)/2

Fa = (Maximum Load – Minimun Load)/2

Mean Load F_m = (1675.8 + 837.9) /2 = 1256.85 N

Stress Amplitude = Fa = (1675.8 - 837.9) /2 = 418.95 N

$\tau_m = K_S * 8 * F_m * D /(\pi * d^3)$

$\tau_a = K_W * 8 * F_a * D /(\pi * d^3)$

$\tau_m = 1.125 *8 * 1256.85 *88 /(\pi * 11^3) = 238.05$

$\tau_a = 1.18 *8*418.95 * 88 /(\pi * 11^3) = 83.23$

W_T = 380 kgs

W_F = 380 * 45% = 171 Kg's at Front

For One Wheel = 171 /2 = 85.5 kg's * 9.8

= 837.9 N (Static Load)

2 g value,Maximum Applied Load = 2 * 837.9 =1675.8 N

Yield Point, $\tau_Y = \sigma_{ult} *0.51 = 675.2$ Mpa.

Endurance Limit, $\tau_e = \sigma_{ult} *0.2 = 264.8$ Mpa.

From Soderberg's Equation

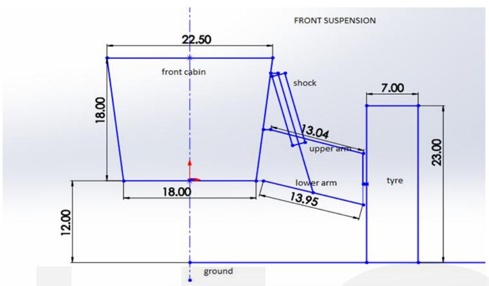
$1/FS = (\tau_m / \tau_Y) + \tau_a / \tau_y [(2* \tau_Y / \tau_e) – 1] = 238.05/675.2 + 83.23/675.2[(2*675.2/264.8) – 1]$

FS = 1.16

After calculating the above two iterations using the same stiffness value. The standard values were chosen for varying spring constants and coil diameter of the spring, as we got more active coils with good factor of safety compared to the first iteration, hence the second iteration values were chosen for spring design.

Suspension front view and A- arm lengths:

The front view of the suspension system with the original lengths of the components were drawn in solid works as line diagram using the standard input values as track width, ground clearance, rim and tyre dimensions chosen from the transmission calculations.



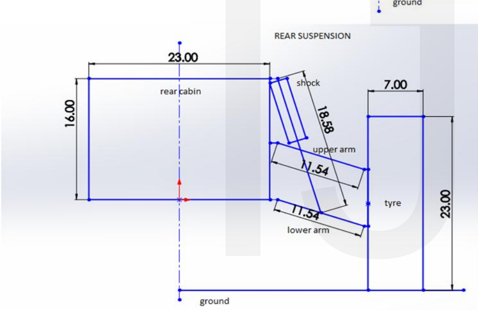


Fig.1 Line diagram of front view of suspension Fig.2 Line diagram of rear view.

Simulation Analysis in LOTUS software:

Suspension simulation is done in lotus software, the camber and toe angles were adjusted to avoid the bump steer and camber gains during bumps. Here the below figure shows the values while simulating of the front suspension, it shows the the changes taking place in camber and toe angles of the wheel geometry with respect to wheel travel.it also shows the change in kingpin inclination angle, castor angle with the change in wheel travel and the fig.3 and fig.4 shows the simulated values of the front and rear suspension. The fig.5 shows the graph between the change in toe and camber angles with respect to wheel travel and fig.6 shows the simulated image of whole vehicles suspension. The wire frame model of the suspension mounting points called as hard points, spherical joints, knuckle and hub with shock mounting were simulated in the lotus software. The points that take the peak amount of forces a saperate analysis is done on those parts like knuckle and wishbones and all the joints were analyzed with the degree of freedom that is necessary for the joint between the upper and lower wishbones which were assembled to the knuckle and the hard points of the wishbones. The original equipment manufacturer (OEM) components like spherical joints and steering ball joints were used. The suspension geometry is simulated in lotus software and results were plotted with respect to camber angles, toe angles, anti-wobbling rod positions to avoid toe angles and scrub radius of the wheel.

FRONT SUSPENSION - BUMP TRAVEL

RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

Bump Travel (mm)	Camber Angle (deg)	Toe Angle {SAE} (deg)	Castor Angle (deg)	Kingpin Angle (deg)	Damper1 Ratio (-)	Spring1 Ratio (-)
160.00	0.1077	-0.2953	-4.7650	5.3221	1.824	1.802
140.00	0.1568	-0.2745	-4.8899	5.2729	1.863	1.841
120.00	0.1901	-0.2502	-5.0180	5.2391	1.902	1.878
100.00	0.2072	-0.2223	-5.1496	5.2211	1.939	1.914
80.00	0.2074	-0.1902	-5.2846	5.2197	1.975	1.949
60.00	0.1892	-0.1534	-5.4233	5.2360	2.010	1.982
40.00	0.1507	-0.1106	-5.5657	5.2721	2.043	2.014
20.00	0.0890	-0.0603	-5.7121	5.3303	2.075	2.044
0.00	0.0000	0.0000	-5.8629	5.4147	2.105	2.072
-20.00	-0.1221	0.0740	-6.0185	5.5305	2.132	2.097
-40.00	-0.2857	0.1675	-6.1795	5.6854	2.156	2.119
-60.00	-0.5040	0.2898	-6.3468	5.8914	2.176	2.136
-80.00	-0.7974	0.4569	-6.5217	6.1667	2.190	2.147

REAR SUSPENSION - BUMP TRAVEL

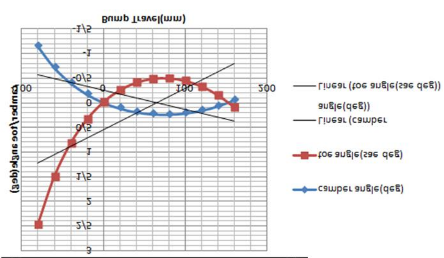
RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

Bump Travel (mm)	Camber Angle (deg)	Toe Angle {SAE} (deg)	Castor Angle (deg)	Kingpin Angle (deg)	Damper1 Ratio (-)	Spring1 Ratio (-)
160.00	0.1388	-0.4158	0.1689	2.2660	2.211	2.300
140.00	0.1508	-0.3743	0.1711	2.2541	2.254	2.341
120.00	0.1550	-0.3305	0.1719	2.2500	2.297	2.380
100.00	0.1515	-0.2844	0.1716	2.2537	2.338	2.418
80.00	0.1399	-0.2354	0.1700	2.2654	2.378	2.455
60.00	0.1200	-0.1832	0.1671	2.2854	2.417	2.491
40.00	0.0909	-0.1271	0.1629	2.3147	2.454	2.524
20.00	0.0515	-0.0664	0.1574	2.3543	2.490	2.556
0.00	0.0000	0.0000	0.1505	2.4060	2.524	2.586
-20.00	-0.0662	0.0736	0.1420	2.4723	2.555	2.612
-40.00	-0.1507	0.1566	0.1318	2.5570	2.584	2.636
-60.00	-0.2592	0.2522	0.1197	2.6658	2.609	2.655
-80.00	-0.4009	0.3654	0.1054	2.8076	2.630	2.669

Fig.3-Simulation values of front suspension
Fig.4- Simulation values of rear suspension



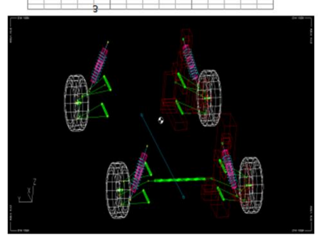


Fig.5- Wheel travel (vs) camber and toe angles

Fig.6- Simulation picture

Result: The camber angle ranges between 0.10 to -0.79 degrees for front suspension and 0.13 to -0.40 for the rear suspension with respect to wheel travel. The toe angle ranges between -0.29 to 0.45 for front suspension and -0.41 to 0.36 for rear suspension with respect to wheel travel. The second iteration spring values were chosen for manufacturing due to its good factor of safety and active coils. While the other important values that are obtained from the above calculations are mentioned in below table.

SPECIFICATIONS	Front / Rear
Spring rate [N/mm]	29 / 25
Roll rate*10^4 [lb-ft/rad]	3218.53 / 2512.62
Ride rate [lb/in]	26.49 / 20.68
Wheel rate [lb/in]	16.31/ 14.06
Natural frequency [Hz]	1.1 / 1.04
Suspension travel [mm]	152.4 / 152.4
Max Damper stroke[mm]	177.8 / 177.8
CG height [mm]	660.4
Ground clearance [mm]	304.8
Castor [deg]	7
KPI [deg]	7
Scrub radius [mm]	+38.1
Sprung mass [N]	2940
Unsprung mass [N]	784.8
Motion ratio	0.75:1 / 0.75:1
Roll center height [mm]	173 / 181
Roll gradient [rad/g]	0.024

Conclusion: The distance between upper A-arm and lower A-arm is 7 inches and the distance between the knuckle upper A-arm mounting point and lower A-arm mounting point is 7 inches. From this we can conclude that upper and lower wishbones are parallel. For the parallel double wishbone suspension system the camber and the toe angles can be controlled so that the stress on the tie rod and steering arm will be less and less chances of failure. The objective of designing suspension for a single-passenger off-road vehicle with high safety and low production costs seems to be accomplished. The design is first conceptualized based on personal experiences during the previous projects under SAE competitions. Engineering principles and design processes are then used to verify and create a vehicle with optimal performance, safety, manufacturability and ergonomics. The main important things about this suspension is the high ground clearance as 15 inches and the shock travel is up to the maximum of 6 inches, whereas the wishbone hard points were mounted to the nodes of the triangulated chassis where point can bear the peak amount of stresses, all the 8 hard points of the front suspension were mounted to the nodes of the chassis and rear suspension follows the same. To achieve good weight transfer stability that is 55% front and rear side as 45%, this weight transfer can achieve good stability during the maneuverability and hill climbing events, more importantly during the hill climbing weight transfer can achieve good stability during sudden accelerations technically called as anti-squat. The traction and camber adjustment of wheels and wishbone lengths plays an important role in achieving good stability during the high speed cornering's avoiding roll out.

Fig.7-Fabricated All-Terrain vehicle.

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