Calculation of Dynamic Forces and Analysis of Front Upright for ATV

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Abstract – *Paper presents the calculation of various dynamic forces that acts on the front upright (also called Knuckle) of an ATV so that we can precisely design and analyze the uprights. Precise calculation of forces leads to the optimization of weight. Uprights play an important role in carrying the complete wheel assembly with itself. Loads of the tires are directly transferred to the Uprights, making it more prone to failure if not properly designed with the right forces. The Front Upright has a steering arm, brake caliper mounting, upper and lower suspension arm attached to it. Lower the weight of the Front Upright, higher is the dynamic stability of the vehicle as it contributes to unsprung mass.*

Key Words: **Front Upright, ATV, A arm, Dynamic forces, Knuckle, Vehicle Dynamics, Analysis**

1. INTRODUCTION

The Front Uprights are the most vital element in the front suspension assembly of an ATV (All-Terrain Vehicle). It connects the suspension arms from the frames to the wheel forming the multi-link structure. The Upright also connects the rack and pinion assembly to the wheels with the help of tie rods for the proper steering of the wheels. It carries a free single-axis rotating hub coupled with the wheel having the disc rotor over the spindle in the center. The brake caliper is mounted on the Upright so that the brake rotor is in-between the brake pads of the caliper. The different parts are shown in the fig-1:

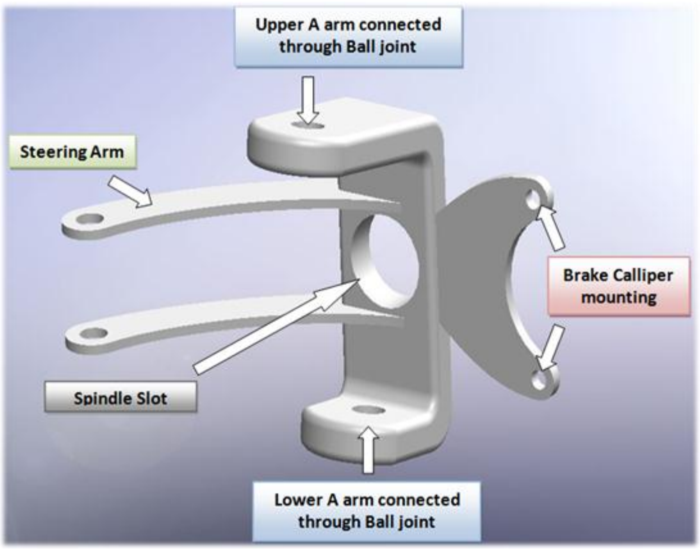

Fig -1: Front Upright

1.1 Vehicle Specifications

The ATV was designed for the BAJA SAEINDIA event. Thus it's designing is done according to the rules specified in the rulebook [1].

Table -1: Vehicle Specifications

Dimension	Front	Rear
Overall length, width and height	1.97 m, 1.54 m and 1.51 m	
Wheelbase (L)	1.34 m	
Track width (B)	1.32 m	1.27 m
Height of center of gravity (H)	0.56 m	
Kerb Mass of vehicle	200 kg	
Mass with 60 kg driver seated	110 kg	150 kg
Sprung mass*	80 kg	120 kg
Unsprung mass	30 kg	30 kg
Turning radius	2.5 m	

* Static weight distribution is assumed to be 40:60 (Front : Rear)

1.2 Knuckle and Tire Specifications

Knuckle is designed using suspension hard points in such a way that it fits in the rim of wheel with caliper mounted on it.

Table – 2: Knuckle and Tire specifications

Scrub radius	20.66 mm
Length of steering arm	80 mm
Distance of hub centre to lower A arm point (a)	0.045 m
Distance of hub centre to upper A arm point (b)	0.050 m
Pitch circle radius of caliper bolt (r)	0.067 m

Tire model	24*8*12
Tire Radius (R)	0.3048 m

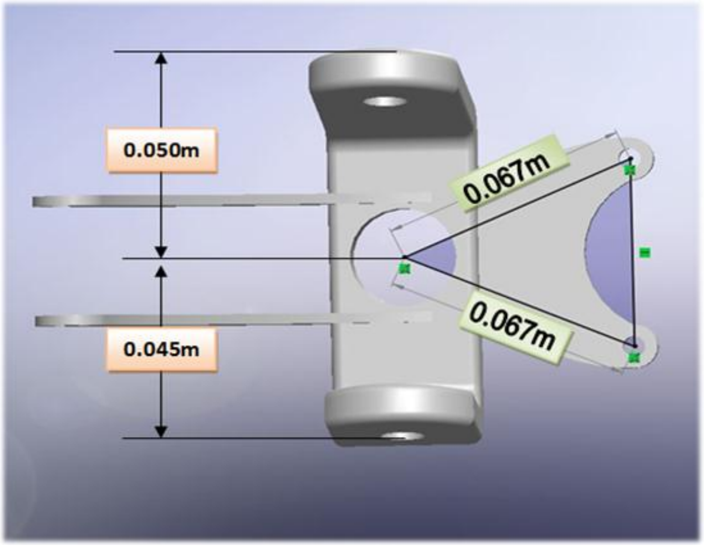

Fig -2: Knuckle dimensions

2. DYNAMIC FORCE CALCULATIONS

At present, the automobile industry is focusing on increasing the efficiency of the vehicle to reduce fuel consumption and encourage sustainability. One of the major factors by which efficiency can be increased is weight reduction. This can be achieved by the introduction of composite materials and optimizing the design of various components of the vehicle. We aim to design the front upright considering the stiffness, strength, and durability of the component. At the same time minimizing its weight and manufacturing cost. For achieving this goal precise calculation of dynamic forces is required. In this section we will calculate the following forces acting on front upright:

1. Braking torque acting on the caliper mounting.
2. The lateral force acting due to cornering
3. Bump forces due to suspension geometry
4. Force due to push and pull of tie rod

2.1 Braking torque

When the brake is applied on the vehicle, longitudinal mass transfer occurs in the front wheels. This mass transfer increases the vertical load and hence applies longitudinal reaction forces on the upper and lowers suspension arm ball joint points on the Upright. The system is considered as a beam with braking torque and reactions on the hard points of Knuckle. For calculating dynamic mass transfer we are considering only sprung mass.

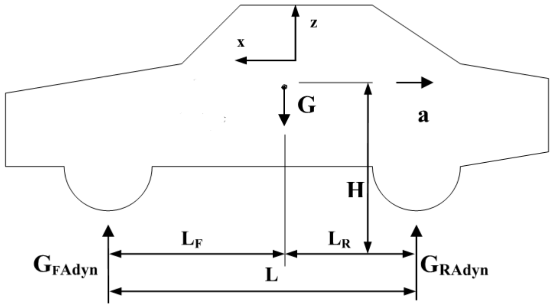

Fig -3: FBD of the vehicle during braking

For calculating dynamic mass transfer on front wheels, moment about rear axle is taken as shown in Fig-3.

Total mass on front axle (sprung + unsprung) = 104kg
Braking distance (S) = 4.57 m
$L_R = 2/5$
$L_F = 3/5$
Maximum velocity of vehicle (u) = 45 km/h
= 12.5 m/s

$v^2 = u^2 + 2aS$ (1)
$0 = (12.5)^2 + 2 * a * 4.57$
$a = 17.07\ m/s^2$ (deceleration)
$= 17.07/9.81$
$= 1.74\ g$

$G_{FA\ dyn} = (G\ L_R + m\ a\ H)/\ L$ (2)
$=(200*9.8*(2/5)*1.34 + 200*1.74*9.8*0.56)/1.34$
= 2209.24 N
$G_{FA\ dyn}$ = 1104.6 N (on one wheel)

Vertical load due to Unsprung mass = Unsprung mass on one wheel * g (3)
= 15* 9.81 N
= 147.15 N

Total vertical load acting on each wheel will be equal to sum of load transferred due to dynamic mass transfer and vertical load due to Unsprung mass on each wheel.
Total vertical load on one wheel = $G_{FA\ dyn}$ + Load due to Unsprung mass (4)
= 1104.6 +147.15 N
= 1251.75 N

Frictional force = Coefficient of friction * Total vertical load on one wheel (5)
= 0.6 * 1251.75 N
= 751.05 N

Braking torque on the wheel is the effect of frictional force acting in the contact patch of tire.
Braking torque (Tb) = Frictional force * Radius of tire(R) (6)

= 751.05 * 0.3048
= 228.92 N-m

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Force exerted on caliper mounting (Fc) =
Braking torque / Distance from centre of spindle(r) (7)
= 228.92 / 0.067 N
= 3416.71 N

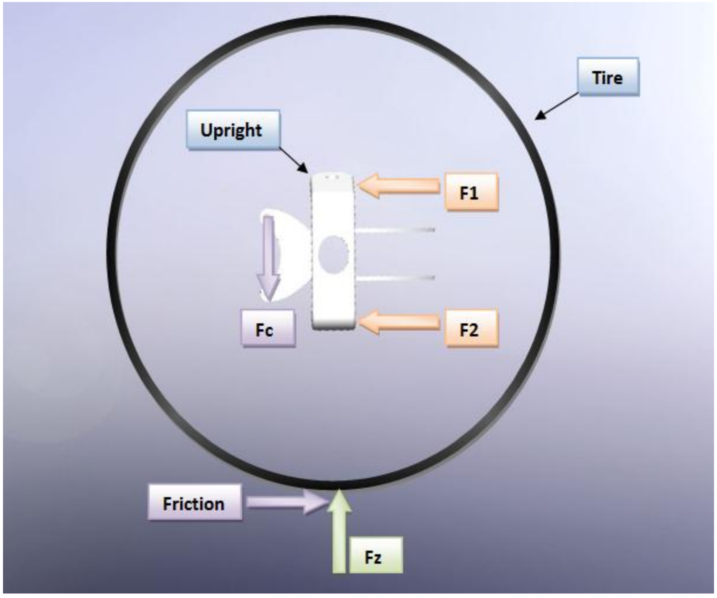

Fig -4: FBD from side view

Moment of reaction forces due to braking and force exerted on the caliper is taken about the centre of spindle as shown in Fig- 4.
F1 * b – F2* a+ Fc * r =0 (8)
0.05 F1- 0.045 F2 = -228.91
Net horizontal force is zero.
F2+F1= 751.05 (9)
F1= 2053.82 N (towards right)
F2= 2804.87 N (towards left)

2.2 Lateral force

Lateral force is exerted on the upright due to centrifugal force and gyroscopic effect which acts due to cornering. As effect of couple due to centrifugal force, weight is transferred on the outer wheels. Due to gyroscopic effect of wheels during cornering results in mass transfer on outer wheel. Thus total load on outer wheel is calculated by adding load due to Unsprung mass, Gyroscopic effect and centrifugal force.

Vertical load acting on each wheel = {(Sprung mass + Unsprung mass)/2}* g (10)
= 55 *9.81 N
=539.55 N

Vertical force due to centrifugal couple = {Total mass with driver * (velocity)^2 * Height of COG(H)}/{2*Cornering radius * Track width (B)} (11)
= 260 * 12.5^2 *0.56/ (5* 2* 1.32)
=1723.48 N

Vertical force due to gyroscopic effect = I ω ω_p (12)
= 4 * {Mass of each wheel * (Radius of tire(R))^2/2}* {(Velocity of vehicle)^2/ (Radius of tire *Cornering radius)}
= 4 * (8 * (0.3048)^2 / 2) *{(12.5)^2/(0.3048 * 5)}
= 152.4 N

Net vertical load acting on wheel = 539.55 +1723.48 +152.4 N
=2415.43 N

Lateral force is frictional force acting on the wheel due to load transfer during cornering.
Lateral force = Coefficient of friction * Net vertical load
= 0.6 * 2415.43 N
= 1449.26 N

Fig- 5 shows the location of vertical load, lateral force and reaction forces on the Upright and tire. For calculating reaction forces in knuckle moment about point of action of force F4 is taken. And other equation is taken by balancing horizontal forces.

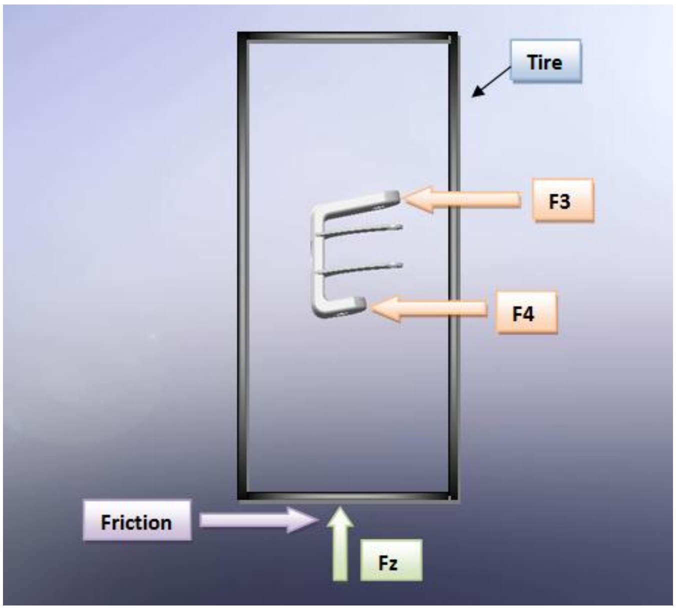

Fig -5: FBD from front view

Lateral force * (R-a) = F3 * (a+b) (13)
1449.26 * (0.3048-0.045) = F3 * 0.095
F3 = 3963.34 N (towards right)
F4 = Lateral force + F2 (14)
= 1448.97 + 3963.34 N
= **5412.31 N (towards left)**

2.3 Bump force

Bump force on Upright is due to force exerted by Coil Spring on suspension arm. Force due to Coil Spring is not applied directly to the Upright. Bump force is a result of bending moment on suspension arm due to force exerted by Coil Spring.

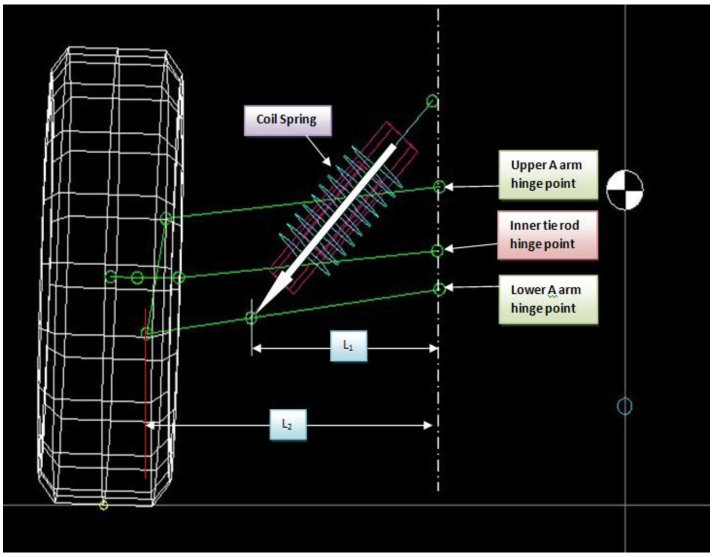

Fig -6: Suspension Geometry

For calculation of this force moment of vertical component of Coil Spring force and bump force on Upright is balanced about the axis passing through hinge point A of suspension arms as shown in Fig- 6.

Stiffness of Coil Spring (k) = 50 N/mm
Maximum compression of spring (x) = 100 mm
Coil spring makes an angle of 47˚ from horizontal

Force applied by Coil Spring = Stiffness (k) * Maximum compression (x) (15)
= 50 * 100 N
= 5000 N
Vertical component of above force = Force applied * sin (47˚)
= 3656.77 N

Torque applied about hinge A = Force applied by Coil Spring * Perpendicular distance (L_1) (16)
= 3656.77* 0.24513 N-mm
= 896.38 N-m

Bump force on Upright = Torque applied about hinge A / Length of suspension arm (L_2) (17)
= 896.38 /0.40855 N
= **2194.05 N**

2.4 Force on steering arm

Force on steering arm is applied due push and pull of tie rod due to linear movement of rack, according to the input given by steering wheel. The force acts perpendicular to the steering arm.

Static load on one wheel = 539.55 N (from equation 10)
Frictional force = Coefficient of friction * Static load
= 0.6* 539.55 N
= 323.73 N

Friction force rotates the wheel assembly about the Steering axis.
Torque acting about Steering axis = Frictional force * Scrub radius (18)
= 323.73 * 20.66 N-mm
= 6688.27 N-mm

Force acting on tie rod = Torque about Steering axis / length of Steering arm (19)
= 6688.27 / 80 N
= **83.60 N**

3. MATERIAL SELECTION AND ANALYSIS

Forged steel EN 47 is used for front upright due less weight to strength ratio and availability. It has high tensile strength and toughness. Physical and chemical properties of EN 47 are stated below:

Table - 3: Mechanical properties of EN 47

Quantity	**Value**
Density	7.7 g/cm3
Ultimate Tensile Strength	700 MPa
Yield Tensile Strength	420 MPa
Young’s Modulus	200 GPa
Poisson’s Ratio	0.3

Table - 4: Chemical Properties of EN47

Element	**Weight percentage**
Carbon ,C	0.45-0.55
Manganese, Mn	0.50-0.80
Silicon, Si	0.05 Max
Chromium, Cr	0.80-1.20
Sulphur, S	0.05 Max
Phosphorus, P	0.05 Max

The analysis of the Front Upright is done in Ansys Workbench 19.2. The analysis is a very crucial part as it gives us the simulation of the real-time forces to validate our design. We performed the static structural analysis by fixing the spindle and applying the different forces in the hard points of the Upright. Equivalent von-mises stress is calculated and according to the place of maximum stress, the design of upright is changed for the next iteration. The front Upright is prescribed to have a factor of safety of 1.5. The maximum area has equivalent stress up to 181.77 MPa, So this design is favorable.

The directions of the different forces are shown in the fig-6.

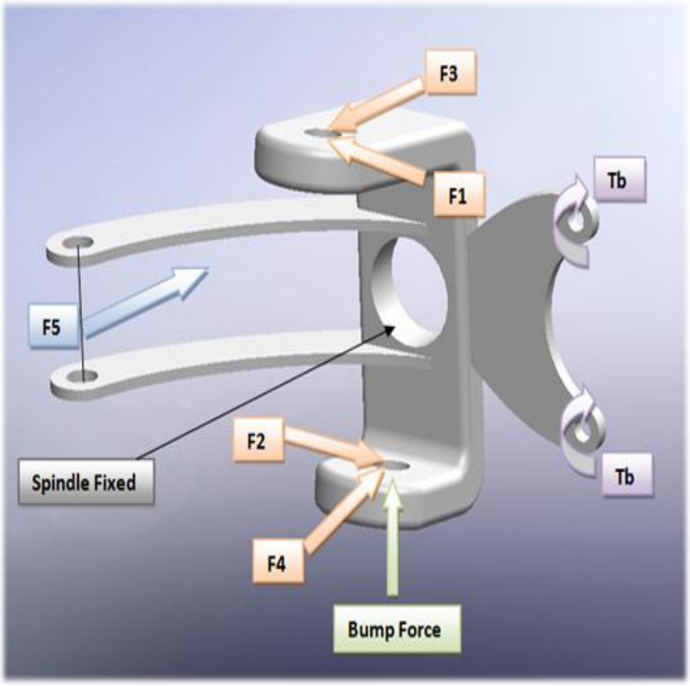

Fig -7: Direction of Forces

Table - 5: Magnitude of Forces and torque applied

Force/ Torque	Value
F1	2053.82 N
F2	2804.87 N
F3	3963.34 N
F4	5412.31 N
F5	83.60 N
Bump Force	2194.05 N
Braking Torque (Tb)	228.92 N-m

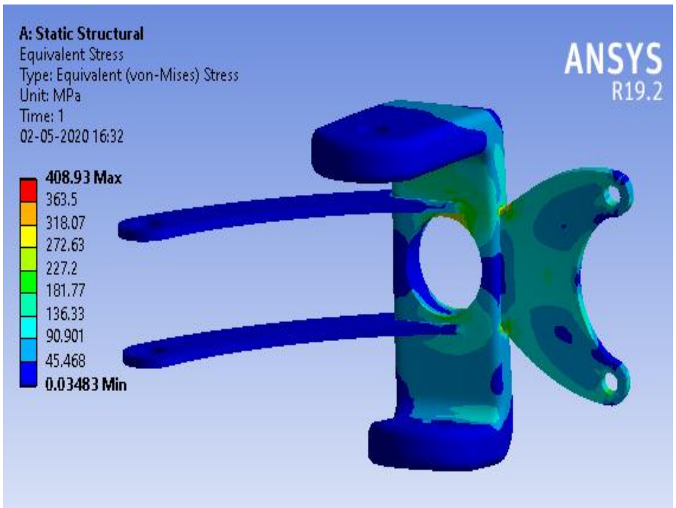

Fig -8: Equivalent Stress

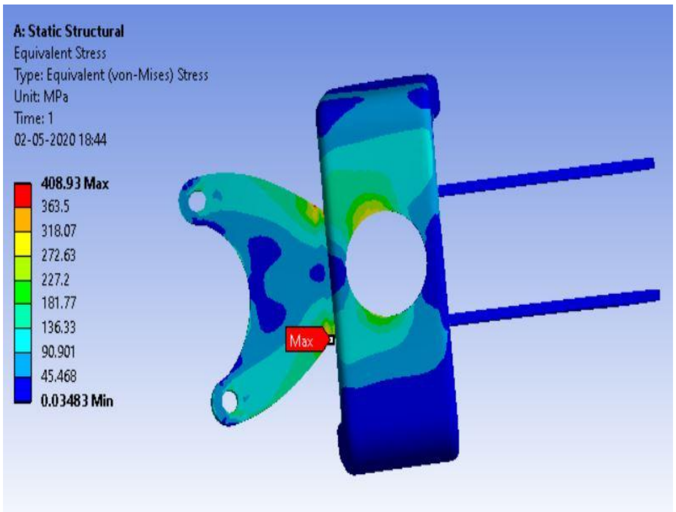

Fig -9: Equivalent Stress

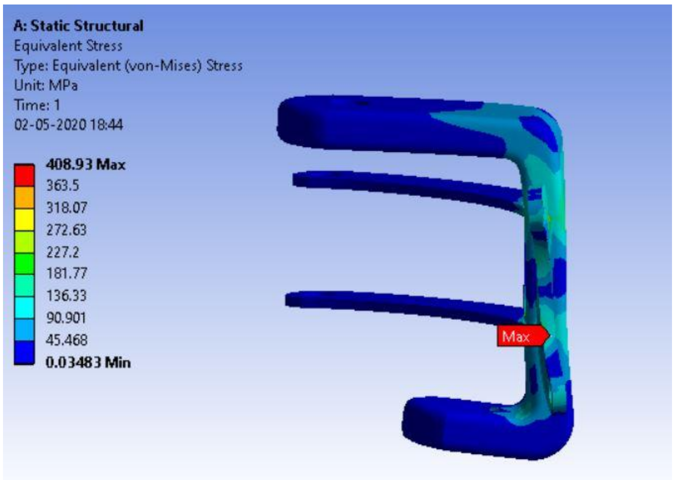

Fig -10: Equivalent Stress

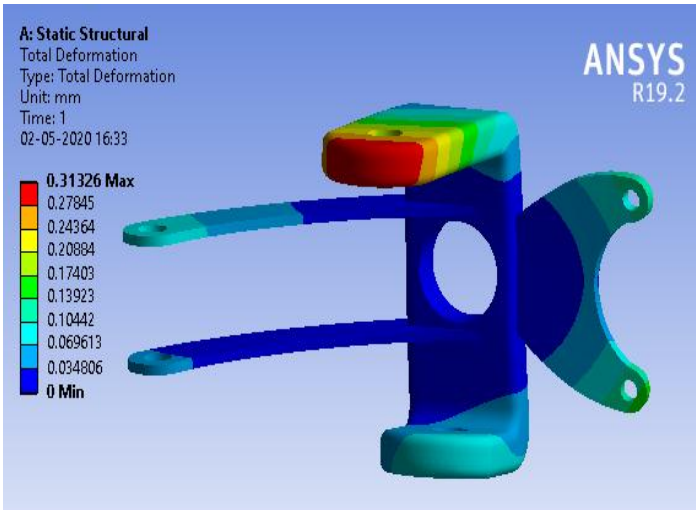

Fig -11: Deformation

4. CONCLUSIONS

The purpose of this paper was to design the Front Upright by calculating the dynamic forces. The force is calculated using basic concepts. This paper gives a clear idea of how forces act on the Upright. Material is selected based on calculated forces. Upright is designed using suspension points and dynamic force applied considering the factor of safety. Design is validated by using Ansys software. This design is fabricated and tested in the ATV in harsh conditions. No failure occurred at the time of testing, it can be concluded that forces calculations and design are up to the mark.

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