Design and Analysis of Dynamics and Kinematics of A Utility Terrain Vehicle

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Abstract- *Frame is the skeletal part of the automobile body which holds all the components which include the suspension geometry and engine mounts and hence it is essential for us to do a proper analysis on its structural integrity to predict possible outcomes or failures during impact so that passenger remains unharmed even in collision. Suspension and steering are designed to optimise comfort and provide maximum tyre traction always to increase life of the vehicle. Braking is selected such that all four wheels are locked when brakes are jammed. All the components are then analysed for safety and to check if they meet the required standards.*

Keywords- Brake analysis, Design and analysis,Dynamics and Kinematic study, Suspension, Steering.

I. INTRODUCTION

In the world of automotive, the functions of a utility terrainvehicle may seem odd and quirky. It isn't the fastest, nor the most efficient and yet is one of the most useful and the engineering efforts into their design and study can leave most perplexed. The areas of their use can range from military, sports, recreational and even to applications of small scale transport. A utility terrain vehicle (UTV) is aptly named from its characteristics of being able to take on almost any terrain with the same ease and comfort.

The dynamics and kinematics of UTV's are a refreshing turn from the monotonous staple design work of most other automobile sectors due to their very unpredictable ride conditions. The dynamics in the suspension design, braking force calculations and the adjustment of steering setups to achieve perfect harmony with the driver are all important aspects in the study.

The suspension setup starts as the backbone and lays the foundations for the rest of the vehicle to be built upon. The steering, frame design and the braking setups come next respectively. The entire design and analysis though is a purely iterative process whereby we go on rebuilding on the modifications made to any one system.

Being a sporting vehicle as it is popularly used for, safety also plays a role of utmost importance in the design and as it is usually thrown against hurdles of high shock and intensity. The UTV is a fighter and hence as such must be durable enough to bear the repeated loading and withstand failure due to fatigue.

II. METHODOLOGY

A. PROPOSED METHODLOGY

1. The rough definition of the goals and target within the set limitations. This includes a complete design procedure and selection of the parts needed to be used for optimal performance.
2. The design procedure starts from the selection of the tyres and rims based on the terrain conditions. Off-road ATV tires are preferred for their excellent grip in poor traction conditions.
3. With the selection of tyres done, the next step is the suspension geometry and suspension spring design. This is a critical stage as it concerns the ride, handling and comfort of the occupants inside.
4. The steering system is then analysed to fit in the design and provide proper steering control and feedback to the driver. The aim of the steering system is to allow the driver to accurately control the vehicle with minimal effort.
5. The frame is modelled step by step around the required suspension and spring design. The cockpit is designed to maximize space utilization and providing good ergonomics to the occupants in rough terrain. The cockpit must satisfy the safety concerns and be analysed for any faults.
6. The braking system is then selected for the vehicle to slow down and stop. Calculations are done to check the forces required in bring the vehicle to a stop without losing control. The effort required by the driver is again minimised.
7. The final suspension modifications are made to comply with the components added and the final revision and analysis is done to verify that the performance is at its peak.
8. Completion of the design phase progresses to the manufacture and fabrication of the vehicle. Here techniques with the aid of the software **SolidWorks**

2016 and **Fusion 360,** the fabrication of the space frame is made easy.

9. The tubes are then welded together to create the finalised assembly. The complete vehicle is then tested extensively to check for any faults.

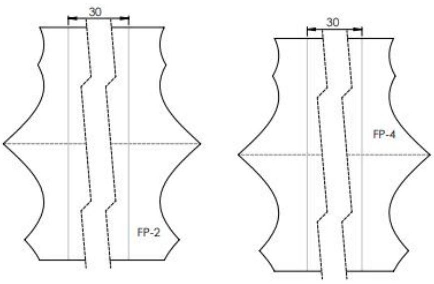

Fig 1 Pipe profiles for notching

B. SUSPENSION DESIGN

i. Front suspension

Mass of the vehicle with the driver and navigator = 587 kg
Wheelbase (L) = 2235 mm
Track width (front and rear) = 1372 mm
Height of center of gravity (CG) from the ground = 510 mm
Weight distribution ratio 34 (front): 66 (rear)
Maximum longitudinal acceleration = 0.7g
Maximum lateral acceleration = 0.3g
Distance of CG from rear axle (L_r) = 670.5 mm
Distance of CG from front axle (L_f) = 1564.5 mm
Static weight on front axle (SWF) = 0.343*5719= 1956.7 N
Static weight on rear axle (SWR) = 0.657*5719= 3762.3 N
Weight transfer (ΔW) = M*0.7g*CGH/L =919.8 N
Dynamic weight front axle (DWF) = SWF+ΔW = 1956.7 + 919.8 = 2876.5 N
Dynamic weight rear axle (DWR) = SWR -ΔW = 3762.3 – 919.8 = 2842.5 N

The spring stiffness was calculated based on the availability of the spring and the size in which it was available. Due to these limitations, by analyzing the size and stiffness of the spring, the mounting location to find the correct motion ratio was calculated by assuming a suitable ride frequency for travel.

The size of the spring available in the market was of an ATV spring of 32 N/mm stiffness and a free coil length of 215.6 mm. The spring parameters are given in the table below.

Parameters	Value
Eye to Eye length	317.5 mm
Spring Material	Chromium Vanadium Alloy Steel
Free Spring Coil Length	215.6 mm
Spring Stiffness	32 N/mm
Outer Diameter of Coil	64 mm
Wire Diameter	8 mm
Spring Index	7
Solid Spring Length	85 mm
Maximum Load on Spring	4176 N
Maximum Allowable Travel	146.8 mm

With these values, the required motion ratio was calculated with the assumption of a few parameters.

K_{Spring} = 32 N/mm
$f_{Natural}$ = 0.95 Hz (Commercial street cars have a value of the range 0.5-1 Hz)
m_{Sprung} = 80 kg (Sprung weight at that corner neglecting the tyres, rim, brakes and upright)
MR – Motion Ratio = (Spring Travel)/ (Wheel Travel)
From these, we can arrive at the equation to find the motion ratio.

$f_{Natural} = (1/2\pi) *(MR)*\sqrt{(K_{Spring}/ m_{Sprung})}$
$MR \quad = 2\pi*f_{Natural}*\sqrt{(m_{Sprung}/ K_{Spring})}$
$\quad = 2\pi*0.95*\sqrt{(80/32*1000)}$
$\quad = 0.298\sim 0.3$

Hence by the definition of Motion Ratio,
MR = (Spring Travel)/ (Wheel Travel)
Spring Travel = 0.3* 152.4
= 45.72 mm

To achieve this Motion Ratio is a process of trial and error which can be achieved in most CAD software. **Fusion 360** was used in this case to draw the line diagram and the line compression was tried repeatedly in various positions to achieve this motion ratio.

To make the process easier, the spring were mounted upright. This also reduces the force experienced by the spring in supporting the body.

Hence by repeated iterative trials, the spring was mounted on the lower control arm at 224 mm from the ball joint center of the arm.

This is shown in Fig 6.3. Due to this type of mounting, the geometry is analogous with a Type-II Lever system which acts as a force multiplier.

Hence to get the static sag or initial compression of the spring due to the sprung weight at that corner, we use this simple equation of levers.

F_{Spring} = m_{Sprung}* Lower control arm length/ Spring mount point
= 80* 9.81* 325/ 224
= 1138.66 N

Therefore, the static compression of the spring would be,

x_{Spring} = F_{Spring}/ K_{Spring}
= 1138.66/ 32
= 35.5 mm

Therefore, total spring compression would be,

X_{Spring} = Dynamic compression + Static compression
= 45.72 + 35.5
= 81.22 mm

A preload force must be set such that the spring doesn't go limp or lose its compressive force. This is calculated by the static compression and the spring stiffness.

Therefore, the preload force required is 1152 N.

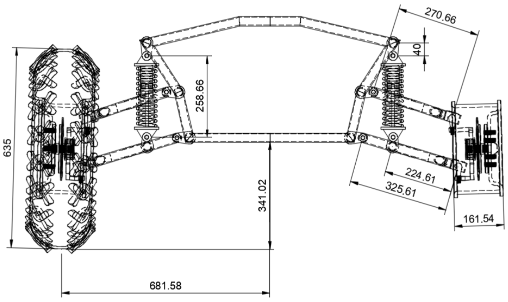

Fig 2 Schematic of the front suspension

ii. Rear suspension

For the rear, the values vastly remain the same excluding the natural frequency and the corner weight. So, repeating the procedure as for the front,

$f_{Natural}$ = 0.65 Hz (Softer suspension in the rear is preferred for improved comfort)
m_{Sprung} = 173.5 kg
MR = $2\pi * f_{Natural} * \sqrt{}$ (m_{Sprung}/ K_{Spring})
= 2π*0.65*$\sqrt{}$ (173.5/32*1000)
= 0.3
Spring Travel = 0.3* 152.4
= 45.72 mm

Thereby, we achieve similar spring travel rates in the rear as well. The location of the spring mount here was simplified by using the same ratio and mounting angle as the front to obtain similar values. This helps in reducing time spent on repetitive calculation and factors in as a bonus during the manufacturing stage where all the spring are similarly oriented. This gives a visual aid and ease in maintenance during periods of servicing and repair.

The geometry is also analogous to a Type II Lever system where force is multiplied but for a shorter travel range.

Hence to get the static sag or initial compression of the spring due to the sprung weight at that corner, we use this simple equation of levers.

F_{Spring} = m_{Sprung}* Upper control arm length/ Spring mount point
= 173.5* 9.81* 255/ 175
= 2480.12 N

Therefore, the static compression of the spring would be,

x_{Spring} = F_{Spring}/ K_{Spring}
= 2480.12/ 32
= 77.5 mm

Therefore, total spring compression would be,

X_{Spring} = Dynamic compression + Static compression
= 45.72 + 77.5
= 123.22 mm

A preload force must be set such that the spring doesn't go limp or lose its compressive force. This is calculated by the static compression and the spring stiffness.
Therefore, the preload force required is 2481 N.

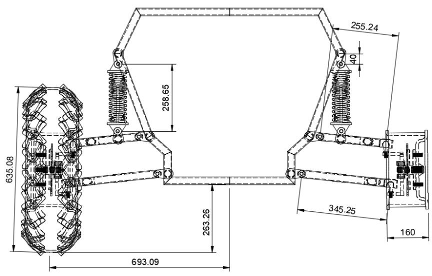

Fig 3 Schematic of rear suspension

C. STEERING DESIGN

The steering system is designed to follow Ackerman's steering condition to get the wheels to get maximum traction and reduce tyre wear during turning.The following parameters are considered in the design of the steering geometry.

Track width (T) = 1372 mm
Wheelbase (L) = 2235 mm

Inner wheel angle α = 25° (Maximum slip angle before tyre starts skidding)

Outer wheel angle β - In degrees

$\tan\alpha = L/(R - T/2)$
$R = L/\tan\alpha + T/2$
$R = 2235/\tan(25) + 1372/2$
$R = 5479$ mm

Where R is the turning radius of the vehicle.

Using R, we can find the outer wheel angle, β in degrees

$\tan\beta = L/(R + T/2)$
$\beta = \tan^{-1}(L/(R + T/2))$
$\beta = \tan^{-1}(2235/(5479 + 1372/2))$
$\beta = 19.92°$

The Ackerman Steering can be found theoretically by the equation

Ackerman angle $= \tan((T/2)/L)$
Ackerman angle $= \tan(1372/2 * 2235)$
Ackerman angle $= 17.06°$
$\cong 17.1°$

Therefore, the steering arm was integrated into the upright design with the corresponding Ackerman angle. This helps in reducing weight as well as better space utilization inside the rim to a more compact packaging.The angle of 17.1° was included in the lower arm of the upright where the lower control arm would also be mounted. This solves a major issue of most off-road vehicles which have large suspension travel, that is, bump steer and roll steer.

Having the steering tie-rod mounted on the lower arm and having the steering rack mounted at the base of the frame aligns the **Instantaneous Centers** of both the suspension and the steering systems. This prevents the rods from turning the steering wheel during wheel travel.

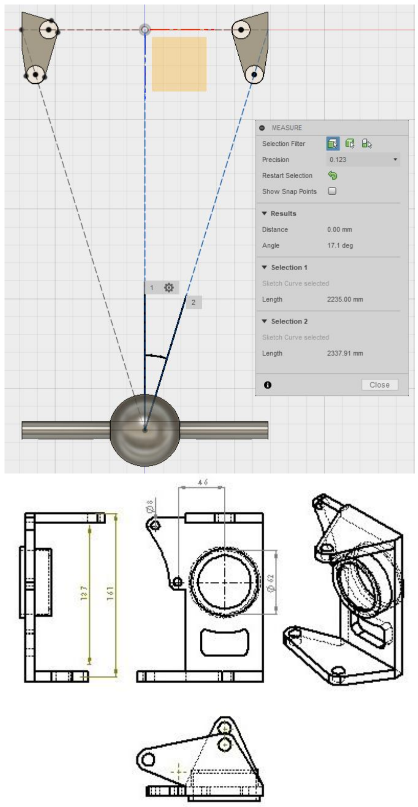

Fig 4 Ackerman angle representation along with the inclusion of steering geometry on the upright

D. BRAKING SYSTEM

We have used 50:50 biased disc brake system using single Tandem Master Cylinder as our car is rear heavy with a weight distribution of 34.3: 65.7 (Front: Rear) and the dynamic weight distribution is almost the same in the front and rear (during braking).

Having the following data and considering the worst-case condition

Mass of car, m= 583 Kg.
Velocity = 60 km/h = 16.67 m/s
Coefficient of Friction (worst case scenario), μ= 0.35
Stopping Distance is given by
$D = (V^2)/(2\mu g)$
$= (16.67^2)/(2*0.35*9.81)$
$= 40.46$ m
Stopping Time = D/V
$= 40.46/16.67$
$= 2.43$ s
Deceleration, $a = 16.67/2.43$
$= 6.86\ m/s^2 = 0.7\ g$

The braking force is magnified by a factor of 5.3 as shown in the diagram. This acts as a Type-II Lever system which reduces TMC piston travel but magnifies the drivers pedal force. It also reduces the effort by the driver in stopping the car.

Magnification factor = 390/ 75
= 5.3

Fig 6.9 Schematic of the brake pedal in force multiplication
Assuming the driver applies 75 N of force on the pedal

Force on TMC F_{TMC} = 5.3 * $Force_{Driver}$
= 5.3 * 75N
= 397.5N

Force by Callipers on each Disc

$F_{Caliper}$ = ($Force_{TMC}$ *Number of pistons* Area of Calipers)/ (Area of TMC bore)
= $(397.5 * 2 * 25.4^2)/ (19.08^2)$
= 1408.89 N

Brake Torque B_T= $F_{Caliper}$ * Effective Radius * μ_f
= 1408.89 * .09 * 0.4
= 50.72 Nm

KE_{Car} experienced by each wheel = $(0.5 * M * v^2)/4$
= $(0.5 * 583 * 16.67^2)/4$
= 20251.15 J

Force required = KE_{Car}/D
= 20251.15/40.46
= 500.52 N

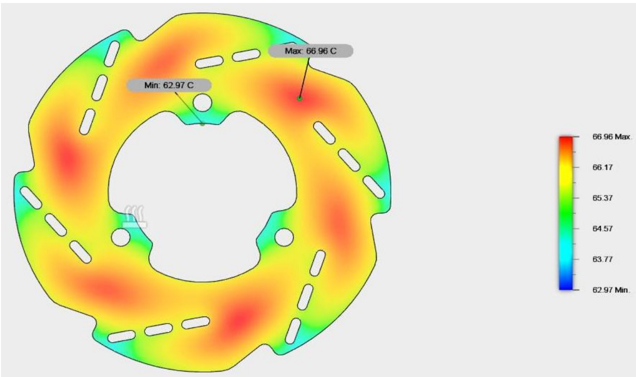

Fig 5 Thermal analysis of brake disc

Master Cylinder	Tandem Master Cylinder with bore diameter of 19.08mm
Disc	Disc of TVS Apache with 200mm OD and 100mm ID 4mm thickness and 3 bolts

V. CONCLUSION

In this paper, comparative studies of different controllers are studied and performance is evaluated according to time domain functions. It is observed that all controllers able to maintain the set point at the desired value but ZN-PID ,Fuzzy based controllers has slight overshoot, Model Reference Adaptive controller has no overshoot and settles quickly. So it conclude that Model Reference Adaptive Controller is the best controller then other controllers

III. CONCLUSION

The dynamic and kinematic study of any vehicle is a highly complicated task and in turn the calculations and assumptions are simplified for better understanding of the basic concepts and principles. From the study conducted, we got to know to approximate spring rates for the suspensions of a UTV, the basic grasp of Ackerman steering and why it is necessary, and the braking forces involved and how leverage is taken advantage of for magnifying the pedal force applied.

The general design considerations during the manufacture of any vehicle and some software hacks to reduce the manufacture time were also explored. This allows for rapid prototyping in large scale companies which allows testing and improving products at a rapid rate.

From the study, the necessary stiffness values of the springs, steering angles and braking force were calculated and recorded.

IV. ACKNOWLEDGEMENT

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