International Journal of
Advances in Scientific Research and Engineering (ijasre)

E-ISSN : 2454-8006

DOI: http://doi.org/10.31695/IJASRE.2018.32975

Volume 4, Issue 11
November - 2018

A Novel Design for an all Terrain Custom-Built Vehicle Dynamics

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ABSTRACT

Rally vehicle dynamics design is a quintessential part of motorsports in today's do-it-yourself world. An aspiring hobbyist aims to design and implement a robust build which is simulated initially using software tools and furthermore, developed as a large-scale manufacturable product. This report aims to discuss the ABC's of a professional vehicle build which incorporates the use of all the aforementioned features of development. Consequently, the final build is an integration of many modern and futuristic features and hacks that are supposedly a niche requirement in today's automobile industry. The features highlight the modifications and improvements to the essential sub-systems of a typical commercial vehicle. Subsequently, the vehicle can then be used in rallying, off-roading and as a means of transportation in rugged and tough terrains.

This report elaborates a basic proof of concept of the features thereby implementable, a design summary of the wholistic build of such a vehicle, the adaptations performed and a conceptual generalized calculation that can be implemented on any generic vehicle. Some of the features elaborated are: On-demand drive change mechanism, Independent wheel braking system, An On-board diagnostics tool which is configurable to any required engine specification and a contrast of independent and partially dependent suspension.

Key Words: *Rally-car, Vehicle, Design, Motorsport, Retrofit, Mechanism.*

1 INTRODUCTION

Designed to conquer the challenges of any terrain, the vehicle consists of a powerful 800cm^3 engine which uses an automatic transmission. The engine manifests a dual cylinder, liquid cooled, 4 stroke Double overhead camshaft (DOHC) which uses a timed ECU controlled Ignition. The perceptible dimensions of the vehicle [1] include a length of 105in, breadth of 60in, a height of 70in and as estimated Kerb weight of 470kg. The suspensions [2] integrate Front Suspension: Independent dual A-arm with anti-sway bar which produces a 15in wheel travel, Rear Suspension: Semi-trailing arm with anti-sway bar which produces a 23in wheel travel. The dynamicity of the suspension height is brought about by the shock preload CAM adjustment mechanism which inherits an extended length of 16.32in and a compressed length of 11.01in. Wire Diameter: 0.48in with a spring weight of 1.919kg. The transmission used of an Inline F/N/R type. The tires used are thereby chosen with dimensions, Front: 25x8x12, Rear: 25x10x12. As opposed to groove-surfaced tires, button-surfaced tires are employed which exhibit maximum traction in marshy terrains. Braking incorporates an outboard hydraulic system mounted on a 220mm front rotor and a 180mm rear rotor that works on an H-split mechanism [3]. Any typical generic steering is employed that includes a Rack and Pinion [4] which has a steering ratio of 7:1 and provides a Turning Radius of 3.12m.

2 CAD (COMPUTER AIDED DESIGN) AND ANALYSIS

CATIA (Ver: V5R20), a professional automobile design tool was employed in the structural design for the tubular-chassis of the vehicle for accuracy and precision. A contrast of analysis was performed using CATIA which was commensurable with the same using ANSYS.

2.1 CHASSIS DESIGN

The frontal and side impact typically endures the maximum amount of stresses [1] [2] during a collision for which, an impact was simulated for the chassis design adhering to Newton's Second Law of Motion. The forces analyzed proved to be within the safety range for a practical implementation of the vehicle.

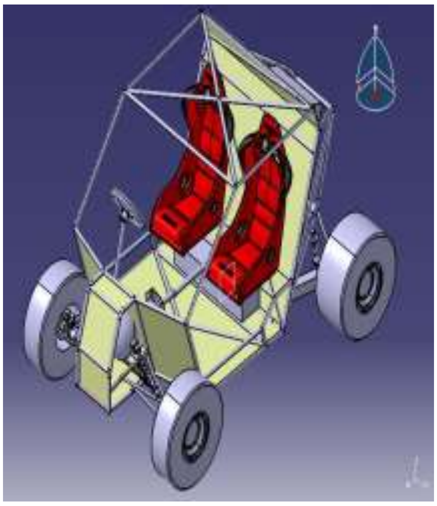
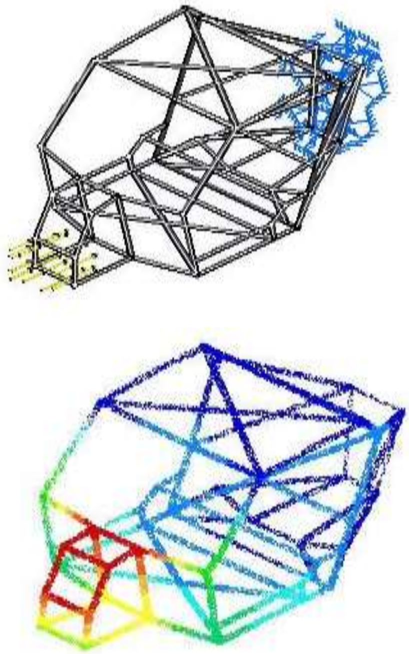
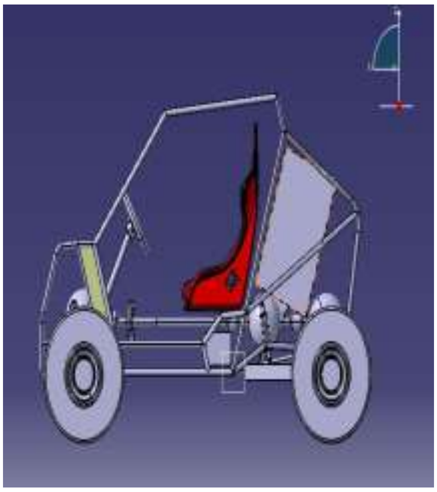
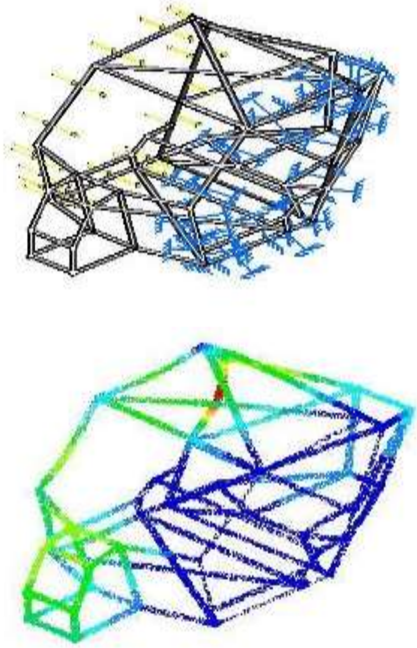
CAD MODEL	FEA (CATIA) [6]	RESULTS
1.Isometric View 		RANGE 0.202 - 0.182 0.182 - 0.162 0.162 - 0.141 0.141 - 0.121 0.121 - 0.101 0.101 - 0.0808 0.0808 - 0.0606 0.0606 - 0.0404 0.0404 - 0.0202 2 - 0
2.Side View 		RANGE 0.0374 - 0.0336 0.0336 - 0.0229 0.0229 - 0.0262 0.0262 - 0.0224 0.0224 - 0.0187 0.0187 – 0.015 0.015 - 0.0112 0.0112 – 0.00748 0.00748 – 0.00374 0.00374 - 0

Figure 1 DARK BLUE: MINIMUM DEFORMATION RED: MAXIMUM DEFORMATION

2.2 SUSPENSION DESIGN

In an off-road vehicle [1], the suspension design [2] [3] [1] is of paramount importance as they endure the majority of forces thereby, The A-arms, semi-trailing arms, and the knuckle have been independently designed to endure multiple loads individually.

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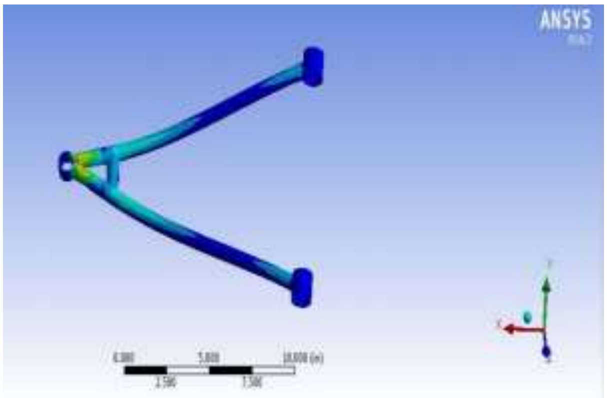
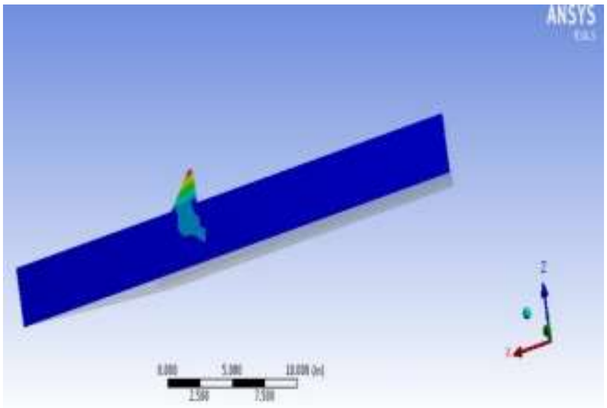
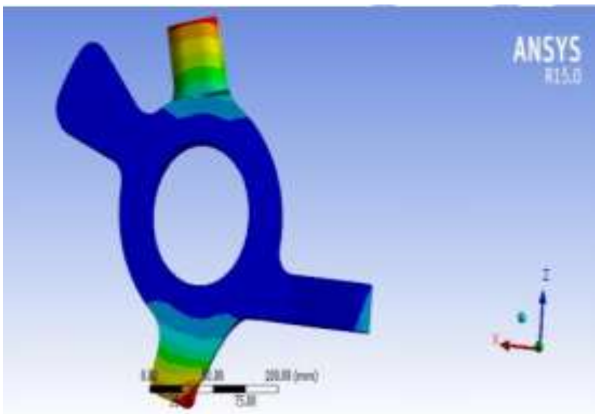
FEA MODEL	RESULTS
A-ARM 	RANGE 2053.4 -1825.3 1825.3-1597.1 1597.1-1369 1369-1140.8 1140.8-912.64 912.64-684.48 684.48-456.32 456.32-228.16 228.16-1.167e-6
TRAILING ARM 	RANGE 4.1146e-8 - 3.6574e-8 3.6574e-8 - 3.2002e-8 3.2002e-8 - 2.7431e-8 2.7431e-8 - 2.2589e-8 2.2589e-8 - 1.8287e-8 1.8287e-8 - 1.3715e-8 1.3715e-8 - 9.1436e-9 9.1436e-9 - 4.5718e-9 4.5718e-9 - 0
KNUCKLE 	RANGE 0.0017204-0.0015292 0.0015292-0.0013381 0.0013381-0.0011469 0.0011469 – 0.00095577 0.00095577 – 0.00076462 0.00076462 – 0.00057346 0.00057346 – 0.00038231 0.00038231 – 0.00019115 0.00019115 - 0

Figure 2 DARK BLUE: MINIMUM DEFORMATION
RED: MAXIMUM DEFORMATION

3 THEORETICAL SUBSYSTEM DESIGN

SUB-SYSTEM	CALCULATIONS	SPECIFICATIONS [1]
TRANSMISSION	• Minimum gear [1]ratio=7750/552=14.04=a*b*c=4*4.86*.75 where a=ratio of open differential ,b=ratio of Torsen Differential ,c=min ratio of PVT • Maximum gear ratio =a*b*3.86=72.25 • Maximum torque available=max gear ratio *21= 1642.39Nm @2500rpm • Actual torque available=1642.39*.85=1396.43Nm@25000rpm • Minimum speed	Tire radius = 305mm Circumference=1.963m Max Torque=34Nm @ 7000rpm PVT will lock at 2500rpm Torque at 2500rpm = 21Nm PVT max ratio=3.86:1 min ratio=1:0.75 Mechanical efficiency =85%

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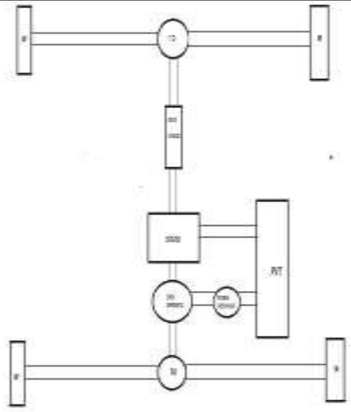
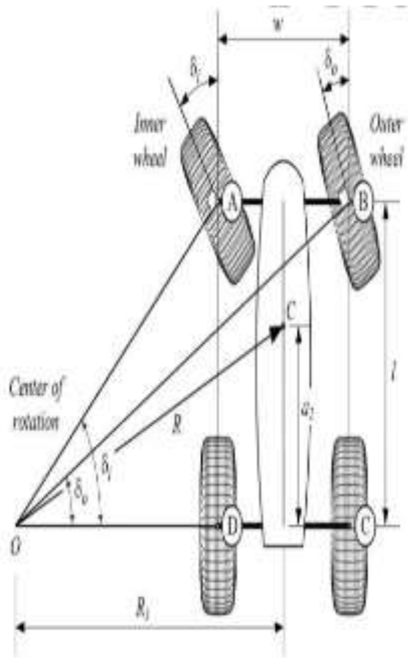
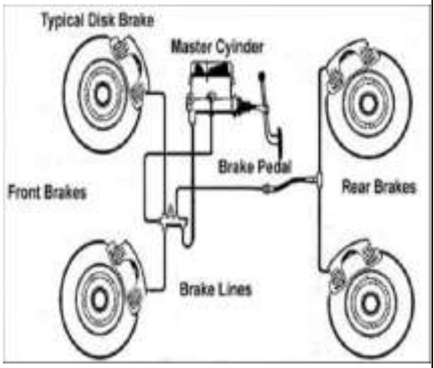
	=1.963*(2500/72.25)*60/1000=3.48KmpH @2500rpm • actual min speed=3.48*.85=2.958KmpH	Max speed =65KmpH
STEERING  	**ACKERMANN CONDITION** [2] General equation: Cot (θo)-cot (θi) =B/L Where: θo= posterior turn angle Θi= interior turn angle W= track width L= wheelbase b= distance from rear axle to center of mass From the general equation we can calculate the interior turn angle of the wheel outside and inside: W=1397 mm L=2032mm Θi=35˚ θo=25.69˚ The minimum radius of turn R can be determined from the geometry: R1= (W tanθi) + (L 2) = (1379 tan35) + (2032 2)=3011.12mm R=√$R1$ 2 + W2=√3011.122 + 838.2 2=3125.6mm =3.12m Therefore, the minimum turning radius of the vehicle around the center of gravity for a maximum inside wheel turn of 35 degrees is approximately 3.12 meters.	Wheel Base 80 in Track Width 52 in Tie rod length 15.8 in Turning radius 3.12 m Steering arm length 3 in Maximum turn angle of inner wheel 35 deg Maximum turn angle of outer wheel 25.7 deg Steering ratio 7:1 Rack travel [10] 4.25 in Caster 12 deg SAI(KPI) 7 deg camber 3 deg Scrub radius 2.097 in Toe out 3 deg C-factor 131.6 Ackerman % 26 Steering wheel diameter 12 in Torque due to the caster and KPI 30.68Nm Torque by driver 24.38Nm
BRAKING  	**Assumptions made**: [3] • Total weight of the vehicle=550kgs • Co-efficient of rolling resistance=0.33 • Effective radius of the tire=0.31m • Co-efficient of friction of brake pad=0.7 **Brake pedal:** Fbp=Fd x (L1/L2) The force applied by the driver=160N The leverage or pedal ratio =6.25:1 The Fbp=160x6.25=1000 N the master cylinder According to Pascal's law; Pcaliper=Pmc Fcaliper Acaliper = FMC Amc The diameter of the master cylinder=19.05mm (say the area be X) The diameter of the caliper piston =26.5mm (say the area be 3X) Therefore, 1120 x = Fcaliper 3x	Pedal Force Considered = 160N Pedal Ratio = 6.25: 1 Front Rotor Size = 240 mm Rear Rotor Size = 180 mm Total Force on All Wheels = 6166.62 N Co-efficient of friction between rotor and calliper pad = 0.7 Tandem Master Cylinder bore x stroke = 19.05mm x 17.05 mm Stopping Distance = 10.71 m Weight Transfer = 37.695 %

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The caliper:
F clamp=2 X Fcaliper =2 X 3360=6720N

The brake pads:
Ffriction=F clampX μ bp disc 06 Co-efficient of friction of brake 0.7
Based on the materials in contact 07
Tandem master cylinder bore x stroke 19.05mm x17.05mm - 08 Stopping distance
10.71m - 09
Weight transfer 0.37695 or 37.695% - μ bp=0.7
F clamp = 0.7 X 6720 = 4704N

The rotor:
The torque on the wheel is equal to the torque on the rotor
Therefore, Tr = Tw Ffriction x Reff = Ftyre X Rtyre Ftyre (rear) = 0.5 x mg x (a2 /l) Ftyre (rear) =1134.375 N R t= Effective rolling radius of the tyre = 0.316 m 4200 X Reff= 1134.375 x 0.3165 Reff Reff = 1134.375 x 0.3165 4200
So the size of the rotor = 180mm Ftyre (front) = 0.5 x mg x (a1 /l) Ftyre (front) =1615.56 N Rt = Effective rolling radius of the tyre = 0.316 m 4200 X Reff
=1615 x 0.3165 Reff =115
So the size of the rotor = 240 mm Ftotal = (2 X 1134.75) + (2 X 1615.56) = 6166.62 N

Deceleration of a vehicle in motion [3]
Av = Ftotal m = 6166.62 550 =11.23 m/s2 Stopping distance of a vehicle: SDv = v2 a x 2 V =56 kmph =56 X (5/18)m/s2 SDv = 15.55x 15.55/(11.23 x 2)= 10.17 m

Vehicle static weight distribution Percent front weight = Vf Vt X 100 = 226.872x100 550 = 41.25 Percent rear weight = Vr Vt X100=323.125X100 550 =58.75 Dynamic impacts of vehicle experiencing deceleration: WT = (Av/g) x (hcg/WB) x V t = 0.37695
The dynamic weight of the vehicle is = 37.695 %

ELECTRICAL LAYOUT

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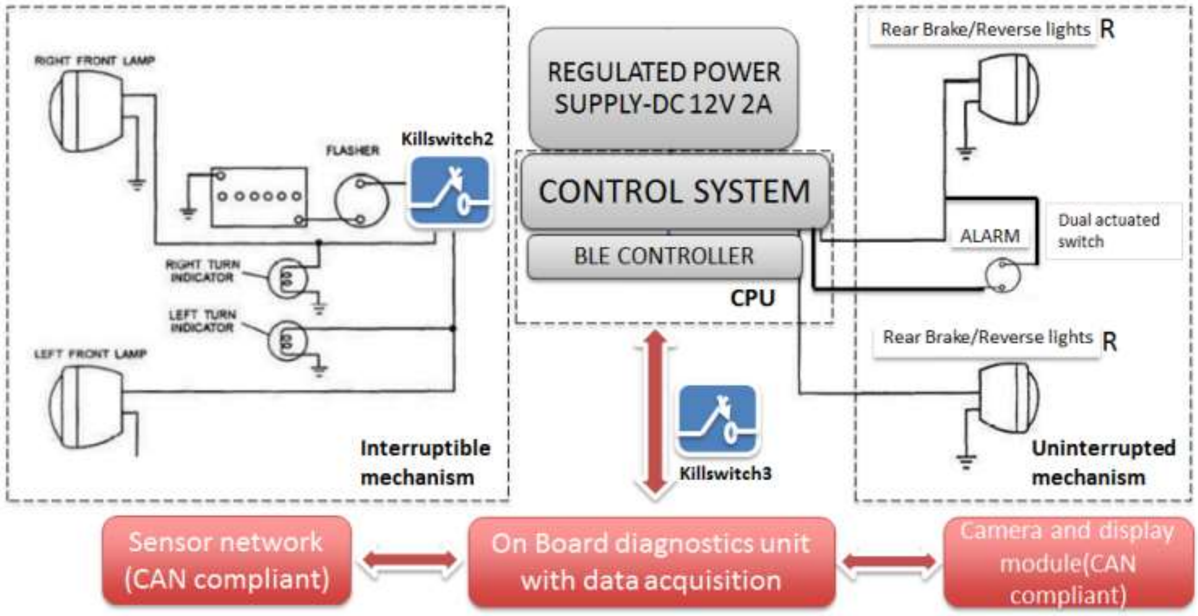

The basic Electrical layout consists of a main control system that performs tasks which a re interruptible and non-interruptible. The interruptible mechanism is used to switch flashers, indicators and lamps. The uninterrupted mechanism has reverse lights, alarm systems and safety features. The Onboard diagnostics unit and the Camera and display module were also utilized in the same controller task schedule. Kill Switches were used for safe turn-on and turn-off.

4 PRACTICAL IMPLEMENTATION

4.1 Advancement In Conventional Designs

a) **On-demand drive change mechanism:** Conventionally, the driver had to stop the vehicle to shift from an RWD to a 4WD. Similarly, an AWD vehicle was an all-time AWD vehicle. As opposed to this, this rally car mechanism changes from RWD to AWD dynamically on-demand providing momentum while shifting and better torque. The mechanism so used is an arrangement of a brake system and a centrifugal clutch in a strategic manner so as to restrict power to the front wheels and supply the same when required. The RWD provides better acceleration whilst the AWD provides better torque vectoring for steering.

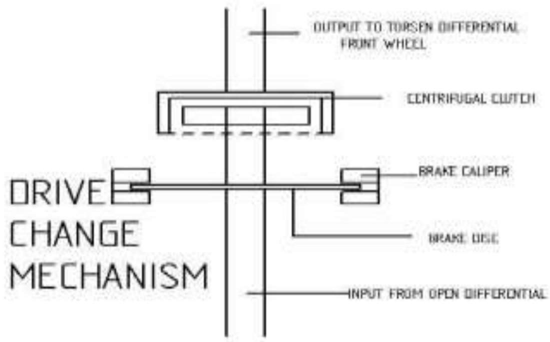

Figure 3 DRIVE CHANGE MECHANISM

The brake system restricts the power by holding the shaft, consequently, the clutch disengages and the power is supplied only to the rear wheels. When required, the brake system disengages and the centrifugal clutch is engaged whilst providing a smooth flow of undeterred power.

b) **Dual Suspension mode:** Conventionally, a manufactured vehicle used either a dependent or an independent suspension which contrastingly provided advantages and limitations. This mechanism used in the rally car utilizes the above modes on terrain demand. Based on speed requirements, for steady cruising over a rugged terrain, an independent suspension is adapted. However, a dependent suspension is used for articulation. This mechanism uses a simplified dog-clutch which is used over the anti-sway bar for a vehicle with independent suspension.

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c) **Independent Braking:** In the rally car design, the advantages of an AWD and a 4WD vehicle are integrated using independent wheel braking which serves the purpose of torque-vectoring in a high-speed rally and an optimum power distribution to the wheels when required in a slow trail. Individual brakes were employed so as to perform the limiting of excessive power in case of a wheel slip utilizing cutting brake technology.

d) **Onboard diagnostics platform:** The rally car has an onboard data acquisition unit which procures sensory data at about 2000 samples per minute. The sensor data is generated from IMU and ambient temperature and humidity sensors. The GY-521 is used for acquiring speed, tilt, and acceleration whereas the DHT11 sensor is used to acquire Temperature and Humidity data. The sensors are mounted in strategic locations such as the DHT11 sensors are mounted near the Engine, Exhaust, Fuel Tank and to measure ambient temperature and humidity. This data obtained in processed via, an onboard processor which runs a Linux based operating system. This data is then sent to the cloud and can also be remotely checked for real-time applications customizable in the future.

4.2 VEHICLE PROTOTYPE

Figure 4. VEHICLE PROTOTYPE
1. TRANSMISSION
2. SUSPENSION
3. STEERING
4. BRAKING
5. ELECTRICAL LAYOUT

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4.3 DFMEA (DESIGN FAILURE MODE EFFECT ANALYSIS)

SL.NO	FAILUREMODE	FAILURE CAUSE	FAILURE EFFECT	S	O	D	RPN	PREVENTIVE ACTION
1)	Bending and braking of frame	Axial stress exceeds yield stress of material due to excess load and impact loading	Overall damage to the roll cage. Frame breaks or bends. Driver's safety is endangered	08	02	02	32	Material with an appropriate factor of safety and effective design, analysis and constant testing.
2)	Structural failure of pedals due to fatigue, bending, and braking	insufficient strength in the brake pedals	Damage to vehicle in undesired circumstances and it'll lead to fatal accidents to driver	08	03	03	72	Choose a material with suitable FOS and constant testing
3)	Breakage of tie rods	Due to uneven terrains in desert track	Steering failure; Safety of driver and others compromised	07	02	01	14	Usage of material with high resistance to compression and tensile loads
4)	Brake failure due to high torque	All-wheel to rear wheel can't be converted when desired	Acceleration cannot be attained to the required level in AWD	05	02	02	20	To ensure the leakage of brake failure doesn't occur
5)	CVT belt failure	Accelerating while braking	No power transmission	09	02	02	36	Choose better belt and avoid accelerating while braking

4.4 DVP (DESIGN VALIDATION PLAN)

SL NO	Factors	Calculated value	Probable failure	validation
1	Top speed	55.25KmpH	The vehicle doesn't reach 55.25Kmph	Due to low power train efficiency
2	Stopping distance	10.17m	The vehicle doesn't stop within 10.17m	Due to insufficient pedal force
3	Turning radius	3.12m	Less steer angle	Decreasing the toe out angle
4	Camber change rate	1.38	Handling quality will decrease if CCR increases	Damper adjustment
5	Wheel travel	15in	The jounce and rebound will be more	Adjustment of springs

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5 CONCLUSION

A plethora of case studies was collected before realizing the advancements and adopted features. On evaluating existing designs for various commercially available off-road vehicles, it became clear that an indigenous design could be employed in a typical vehicle build that also possesses features and technology that is commensurable in a commercial vehicle. The hypothesis is thereby justified by a practical implementation of the same. The results so obtained were deterministic and also futuristic in nature.

These results establish the facts such as an onboard drive change mechanism from an AWD to RWD, Dual suspension mode (Interchangeability from dependent to independent mode), Independent Braking system (Individual Wheel braking achieved using cutting-brake technology) and an Onboard Diagnostics Unit in the control system which is used for data acquisition and for remote monitoring of a vehicle using GPRS and GSM technologies.

6 NOMENCLATURE AND ABBREVIATIONS

CAD(Computer aided design), CAE(Computer aided engineering), FEA(Finite Element analysis), ANSYS, CATIA, Ackermann Condition, Turning Angle, Turning Radius, Kill Switch, Regulated Power Supply, Control System, Starter, Ignition Switch, BLE(Bluetooth Low energy) Controller, CAN(Controller area network) Network, Indicator, CPU, DFMEA(Design Failure mode effect analysis), DVP(Design validation plan), RPN(Risk priority number), Tubular-Chassis, Kerb Weight, IMU(Inertial momentum unit), RWD(Rear wheel drive), 4WD(Four wheel drive), AWD(All wheel drive), Brake disk, Centrifugal clutch, Torque Vectoring, Articulation, Dog Clutch, Cutting Brakes. [8] [3] [1] [7] [2]

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