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# Retrofitting of Internal Combustion Scooter Using Electric Power Source

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***Abstract* - In the era of Technological Advancement, Innovations and Development, all the industries were focusing on the future trend i.e., Electric Mobility. Electric vehicles are the future of the automobile industries. All the renowned industries were focusing on electrification rather to go with the conventional mode of power used for propulsion. In the upcoming future, IC vehicles were completely outclassed by the electrical one. So, this project delas with the problem of public transportation by providing them cost efficient as well as eco-friendly alternative with the vehicles they have. The proposed design consists of changing the whole drive train i.e., Belt-drive with pure electric power Source drive by doing the necessary changes in the OEM scooter design. The proposed swing arm is design is most crucial part of our vehicle because it holds all the necessary components which directly affects the driver's comfort, performance and life of our vehicle. This project is very cost efficient, light weight, interchangeable and most of all it is completely eco-friendly because it drastically reduces the use of hazardous Substances which may cause the accident or cause death.**

***Index Terms* - Technological Advancement, Innovation, Development, Electrification, Electric mobility, Eco-friendly.**

## I.INTRODUCTION

It is required to convert Internal combustion Operated scooter into Electric operated Scooter. Retrofitting of IC operated vehicle with electric power Source deals with reusing of those vehicle were about to scrap i.e., vehicles whose manufactured date are overdue. The electric vehicles are currently experiencing a growing demand due to lack of fossil fuel and due to carbon dioxide emission from exhaust in conventional internal combustion vehicle. Ev's purely utilize electric drive motor which runs on electric energy stored in battery to power the vehicle.[7] Raising Global fuel price have improved business prospects for manufacturing of fuel saving system.[8] A motorized retrofit kit for light weight collapsible mini scooter is a complete conversion for light weight push scooter to a motor driven unit. Owners of an existing scooters can install a mini electric motor battery system, wiring and on/off button with limited skills and equipment to drive the rear wheel of the scooter.[9] The customized swing arm for the retrofit vehicle is going to be light in weight and has sufficient strength to withstand all dynamic forces coming on the vehicle. The current vehicle available in market are very much expensive as compared to in IC engine vehicles. so, retrofitting is the best alternative available to use the scrap item i.e., scooter or bike and convert it into useful product.

## II.OBJECTIVES OF DESIGN

1. To make effective use of all the spares available on vehicle with desired design upgradation without changing original design.
2. To convert IC vehicle into Electric powered vehicle.
3. The vehicle should be electrically powered and driven with the help of powerful dc motor.
4. To provide onboard battery facility with one extra detachable battery.
5. To design a special swing arm with suitable strength and light weight.
6. To design a suspension system.
7. To design cost efficient, light weight and eco-friendly vehicle.[4]

## III. METHODOLOGY

Design of retrofit electrically powered vehicle is carried out in systematic way. The detailed analysis are as follows-

1. Input Study-

This is the first step, and it involves gathering and analyzing the equivalent information. The main source of information is current market and future trends. Correct, complete and accurate data allows designer to identify and finalized the specific requirement of end product.

2. Deciding the geometry and material-

The swing arm is designed in such way that it forms compact structure with sufficient strength and withstand dynamic load in running condition.

3. Selection of motor-

An electric motor which suffices the needs of power, speed and compactness is selected, from overall available motors based on space constraints required acceleration and torque.

4. Selection of controller-

A battery composition of Li-ion is selected as it has best performance and minimal drain overtime. For an assumed range, the battery capacity is calculated, and a controller is selected to protect, provide right amount of power to motor.

5. Making cad model of vehicle-

Using the above collected and calculated data, a CAD model is prepared of the proposed vehicle.

6. Structural analysis of cad model-
7. The final CAD model of the vehicle is structurally analyzed using ANSYS for failure of parts.
8. Thermal analysis of battery pack-

On the battery pack consisting of cells in series and parallel, thermal analysis is done. The corresponding values are noted, and the design process is iterated until a safe design is concluded.

9. Manufacturing phase-

The finalized design is used for manufacturing of vehicle.

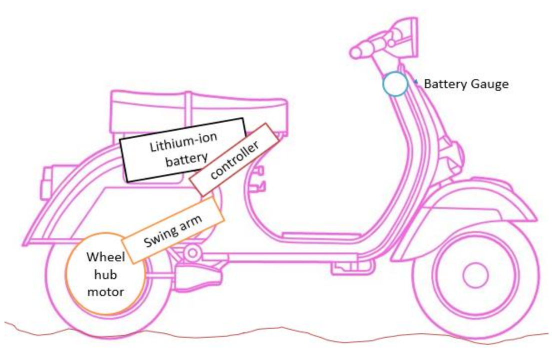



Figure 1 Virtual Image of Retrofit Vehicle

## IV.LITERATURE REVIEW

Wolfgang Gruber B *et al.* [3] have described the optimization, design, build-up and measurements of a wheel hub motor for an electric scooter. This type of drive system increases the power and torque output significantly. The drive was designed to replace the standard hub motor of the E-max scooter. Summarizes the characteristic data of the novel and conventional drive system. It can be seen that especially the power and torque density have been improved significantly. It is concluded that the dynamic performance and low noise level is maintained in the entire speed range of the running vehicle.

João Diogo da Cal Ramos *et al.* [4] have described FEA procedures for swing arm study, as According to his works, the swing arm pivot must be locked, while the rear end of the swing arm is loaded. Standard structural solutions as frames and swing arms, after decades of little alteration in concept are now facing a thorough reassessment. FEA was used through the iterative process of simulating different swing arm models under vertical, torsional and lateral loads.

Wojciech Pawlak, Kacper Leszczyński *et al.* [5] have described FEM analysis was conducted on couple of different driving conditions, such as: 1G, 1.9G, longitudinal stiffness, torsional stiffness, turning conditions and fatigue. Electric vehicles are both a solution and a new design issue. What was already designed and used throughout decades, now needs to be changed. Following, in turning conditions, motorcycle is not only exposed on acting of centrifugal forces, but also standard gravitational ones. Torsional stiffness is calculated based on tires type, coefficient of friction and CoG of motorcycle with a driver.

Nur Hazima Faezaa Ismail *et. al.* [2] have discussed that Heat is produced in batteries from two sources; electrochemical operation and Joule heating. Electric and hybrid vehicle may present the most promising solution for pollution problems caused by the emission of conventional Internal combustion engine. The role of battery as the power source in electric drive vehicle is important. Due to the high energy density, Lithium-ion batteries have gained much attention as a viable candidate to increase vehicle range and performance of electric drive vehicle application. However, there is a variety of thermal limitations for Lithium-ion battery. The heat transfer inside the battery can be divided into three parts, i.e., the heat generated by the cell's internal resistance, the change of entropy that occur in the cell components during discharge and heat transferred to ambient conditions by convection.

Yinjiao Xing, Eden W. M. Ma, Kwok L. Tsui, and Michael Pecht, *et.al.* [6] Have studied that purpose of the BMS is to guarantee safe and reliable battery operation. BMS indicators should show the state of the safety, usage, performance, and longevity of the battery. BMS needs to monitor and control the battery based on the safety circuitry incorporated within the battery packs. Whenever any abnormal conditions, such as over-voltage or overheating, are detected, the BMS should notify the user and execute the pre-set correction procedure.

Leonard Allan Dodd, *et. al*. [3] a motorized retrofit kit for light weight collapsible mini scooter is a complete conversion for a lightweight push scooter to a motor driven unit. Owners of existing scooters can install a mini electric motor battery system, writing and on/off button with limited skills and equipment to drive the rear wheel of the scooter. From the above papers and patents, it is concluded that,

- BLDC hub motor is most suitable one with minimum system losses.
- Lithium-Ferro phosphate cell is used in battery pack because of its characteristics.
- Circular cross section has high bending strength as compared to square cross section for swing arm.

PROPOSED DESIGN

Design Calculation- Electric Powertrain

a. Motor Power-

For calculating the motor power some assumptions were made. Those assumptions are as follows,

Tyre size= 10”

Assumption-

Coefficient of Rolling = 0.015

Coefficient of drag = 0.5

Air Density = 1.27 kg/m$^3$

Frontal Area = 0.6m$^2$

- For Maximum Speed

Maximum Speed = 70 Kmph = 17.44 m/sec

Weight of the Vehicle with two people,

(Gross weight) Gwgt = 220kg

Range= 80Km/charge @ speed of 70kmph

Peak power= (Gwgt $\times$ 9.81$\times$V $\times$ C$_r$) + ($\frac{1}{2}\rho$ $\times$A$\times$C$_d$$\times$V$^3$)

= (220$\times$ 9.81 $\times$ 19.44 $\times$ 0.015)+ ($\frac{1}{2}$1.27 $\times$0.6$\times$0.5$\times$19.44$^3$)

Peak power=2.028 KW………. (at top speed)

- For Nominal Speed

At speed of 50kmph……. (i.e., V= 13.88 m/sec)

Peak power at 50kmph = 1.165 kW

Peak power at 50kmph $\cong$ 1.2 kW

1. Rolling Resistance ($R_{r)}$ = 0.015$\times$ Gwgt$\times$ 9.81

 = 0.015$\times$220$\times$ 9.81

 =32.373 N

2. Air Drag ($R_a$) =$\frac{1}{2}\times\rho\ \times A\times V^2\times C_d$

$=\frac{1}{2}\times 1.27\ \times 0.6\times 19.44^2\times 0.05$

= 71.99 N

3. Total Resistance = $R_r$+$R_a$

= 72+32.373

= 104.36 N

4. Torque,

T= 13.25 N.m

N= $\frac{13.25*60*10\text{^}3}{\pi*254}$

N= 996.6 RPM

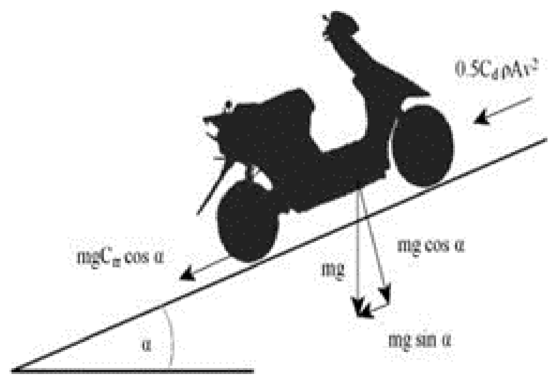



Figure 2 Road Gradient

At Gradient Speed of the Vehicle= 25kmph

$R_r$ = 32.373 N

$R_a=\frac{1}{2}\times\rho\ \times A\times V^2\times C_d$

$=\frac{1}{2}\times 1.27\ \times 0.6\times 6.94^2\times 0.5$

$R_a$ = 9.186 N

Gradient Force ($R_g$) = w*9.81*Sin $\propto$

Gradient Angle ($\propto$) =10°

$R_g$= 374.76 N

Total Resistance = $R_r$+ $R_a$+$R_g$

=374.76+9.186+32.373

= 416.32N

Torque Required = 416.32*0.127

 = 52.87N.m

b. Motor Selection-

1. Normally, Brushless DC (BLDC) motor is used in all electric portable vehicles due to its speed torque characteristics and also it gives maximum starting torque.

2. Torque decreases with increasing speed so no external reduction gearbox is required.
3. Also available in wide power ranges while not being expensive. [1]

Table 1 Motor Specification

| Description | Rated Value |
|---|---|
| Motor Peak Power | 2KW |
| Rated Power | 1KW |
| Rated current | 23A |
| Rated Voltage | 48V |
| Min. Torque | 37N.m |
| Max. Torque | 72N.m |
| Max. RPM | 1500rpm |

c. Battery Pack-

Selection of battery and manufacturing of battery pack is most important aspect of construction of battery pack. Is directly and indirectly affects the performance of the motor. Battery pack with high discharge current reduces the economy of the vehicle and vice a versa. So, the battery is selected by considering all the necessary factors. Battery selection is as follows,

Selection of Battery-

Power Output Required: 1000w

Required range in one charge: 70 km

Speed of Vehicle: 45kmph

Time: 1Hr.

Required Power =

Power/km= $\frac{1000}{45}$ = 22.22 W.hr /Km

For Range of 70 Km =22.22× 70 = 1555.55 W.hr

Capacity of Battery Pack = $\frac{1555.55}{48}$ = 32.40 A ≅ 40A

Battery Pack capacity = 40A, 48V

Battery pack

- Lithium ion 18650,

Voltage = 3.7V, Current= 2.6A

Series cell = 13

Parallel cell =16

Total No. of cell = 208

Table 2 Specification of Battery Pack

| Battery pack | Lithium ion 18650 |
|---|---|
| Voltage | 3.7V |
| Current | 2.6A |
| Series cell | 13 |
| Parallel cell | 16 |
| Total No. of cell | 208 |

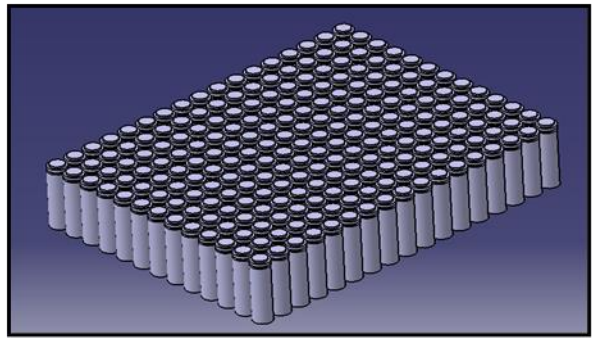


Figure 3 Battery pack Model

Swing Arm-

Swing arm is the part of the transmission which carries most of the weight which will be arise due to uneven surface of the load while driving. Swing arm should sustain in all dynamic conditions. It should have high bending strength and toughness to withstand the shocks from ground.

Below table shows the dimensions of the swing arm

Table 3 Dimensions of Swing Arm

| | |
|---|---|
| Diameter of a front wheel | 10” |
| Diameter of a rear wheel | 10” |
| Wheelbase | 1238 mm |
| Length of a swing arm; axle to axle | 488 |
| Width of the BLDC motor mounting | 144 mm |
| Maximum carrying mass | 109Kg |
| BLDC motor peak power | 2 kW |
| Maximal tilt in turning conditions | 30° |
| Approximate location of CG from rear axle | $h$ =579<br>b=355 |
| Dynamic longitudinal coefficient of Friction | 0.7 |
| Static longitudinal coefficient of friction | 1 |
| Motor Torque | 72N-m |
| Front wheel radius | 127mm |
| Rear wheel radius | 127mm |

e. Analytical Calculations of Swing Arm-

Below table shows the analytical calculations of Swing arm.[4]

Table 4 Analytical Calculations of swing arm

| | |
|---|---|
| Load on a front axle | $N_{sf}$ = F% × m × g = 0.4*109*9.81 = 427N |
| Load on a rear axle | $N_{sr}$ = R% × m× g= 0.6*109*9.81=641.57N |
| Driving force | $P_r= \frac{T}{R}$= 614.17N |
| Vertical force of inertia | $N_{tr} =\frac{579*355}{614.17}$ =334.671 |
| Dynamic load on the front wheel during acceleration | $N_{XCF}$ = $N_{sf}$- $N_{tr}$ =427-334.671 =92.32N |
| Dynamic load on the rear wheel during acceleration | $N_{ACR}$ = 641.57+334.671=976.245N |

| | |
|---|---|
| Static braking force on a front wheel | $F_F = N_{SF} \times \mu_S = 427$ |
| Static braking force on a rear wheel | $F_r = N_{Sr} \times \mu_S = 449.10$ |
| Total static braking force | $F_T = 427 + 449.10 = 876.1$ |
| Dynamic braking force on a front wheel | $F_{BF} = 427 + 876.10\frac{579}{355}$ =1855.90N |
| Dynamic braking force on a rear wheel | $F_{BR} = N_{SR} - F_B\frac{h}{p} =$ $=641 - 857\frac{579}{355} = 116.1249$ |
| Maximum speed | 70Km/hr. |
| Minimal turning radius with maximum speed | $\frac{377.91}{Tan\ 30^o * 9.81} = 66.75$m |
| Centrifugal force during turning conditions | $\frac{200*377.91}{66.75} = 1132.32$N |
| Torsional stiffness | 679.39N |

d. Bearing Selection-

Single deep groove ball bearing 6000 is selected.[10]

Ball bearing has been selected because of its load carrying capacity. It carries both axial and radial load.

Analytical Calculations of Bearing –

ID=d=10mm

OD=D=19mm

Width = w = 8mm

Dynamic Load Capacity=C=4620 N

Static Load Carrying Capacity=C0=41960 N

Assume, Bearing Life $L_{10}$ =35 million revolution

Inner Race rotating outer race stationary

V=1,

- By interpolation,

e=0.3916

$$\frac{F_a}{F_r} = \frac{641.547}{1132.32} = 0.56$$

$$\frac{F_a}{C_o} = \frac{641.57}{1960} = 0.3273$$

$$\frac{F_a}{F_r} > e$$

By interpolation,

Radial Factor= 0.56

Axial Factor= 1.13816

- Equivalent Dynamic Load,

$P_e = XVF_r + YF_a$

$= 0.56 \times 1 \times 1132.32 + 1.13816 \times 641.57$

= 1364.30 N

$C_r = P_e\,(L_{10})^{1/3}$

$=1364.30\,(35)^{1/3}$

= 4462.71 N

$C_r <$ C

Hence selected bearing is suitable.

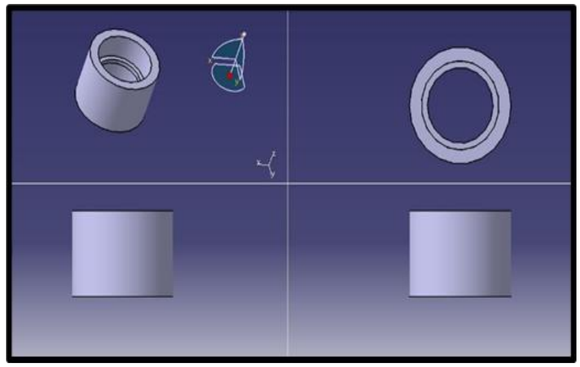


Figure 4 Bearing housing for Selected Bearing

g. Suspension System-

Suspension system is the most important system which affects swing arm design and comfort of the vehicle. It carries all the load of vehicle along with the driver and the other components of the vehicle. For effective suspension travel depends upon the location of the lower pivot from the swing arm pivot.

Suspension geometry-

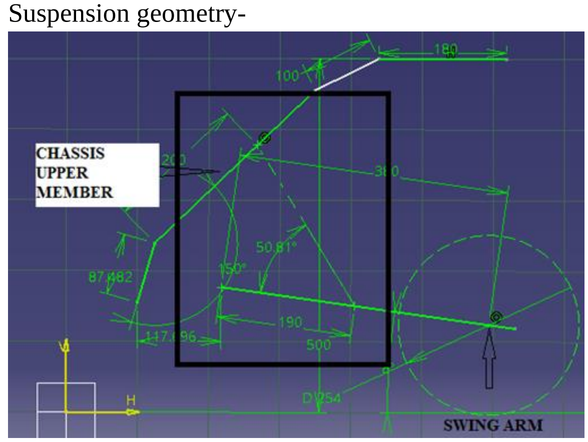



Figure 5 Suspension Geometry

Achieved motion ratio- 0.5

e. Calculation for Helical Compression Spring-[10]

Material: Spring Steel (modulus of rigidity)

G = 78600N/mm$^2$

Mean diameter of a coil, D=27.375 mm

Diameter of wire, d = 9.375 mm

Total no of coils, $n_1$= 17

Height, h = 210mm

Outer diameter of spring coil, D0 = D +d = 36.75 mm

Hence Square and ground end=number of active turns

= ($n_1$ -1)

= (17-2)

No of active turns, n= 15

Weight of bike = 118 kg
Let weight of 1person = 100Kg
Weight of 2 persons = 100×2=200Kg
Weight of bike + persons = 338 Kg
Considering dynamic loads, it will be double
W = 338 Kg = 3315.78N
We know that compression of spring,

$$\delta = \frac{8WD^3n}{Gd^4}$$

$$\delta = \frac{8 \times 3315.78 \times 27.375^3 \times 15}{78600 \times 9.375^4}$$

$$\therefore \delta = 23.44\ mm$$

C = spring index

$\therefore\ \frac{D}{d} = \frac{36.75}{9.375} = 3.92$

Solid length, Ls =n1×d=17×9.375= 159.375 mm
Free length of spring,
Lf = solid length + maximum compression + clearance between adjustable coils
= 159.375 + 13.44 + (13.44 ×0.15)
Lf= 174.831 mm

Spring Rate=K$\frac{3315.78}{13.44}$

K=246.70 N-mm

E=$\frac{1}{2} \times W \times \delta$

E = strain energy stored in spring (N-mm)
E=22,282.04 N-mm
Stresses in helical spring: maximum shear stress induced in the wire

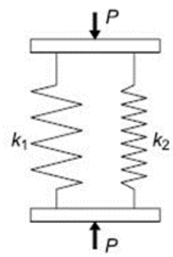
$$\tau = Ks \times \frac{8WD}{\pi . d^3}$$

$$k_s = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

$K_S$ = 1.41373

$\tau$ = 1032.39 $N/mm^2$

For Series combination,




$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2}$$

$k_1=k_2$ = 246.70
k= 123.35 N-mm
Pitch=p

$$\frac{l_f - l_s}{n_1} + d$$

$$= \frac{174.831 - 159.39}{17} + 9.375$$

p = 10.28 mm
Crippling load under which a spring may buckle KL = 0.1 (for hinged end spring)

The buckling factor for the hinged end and built -in end spring

$$Wcr = k \times k_l \times l_f = 123.35 \times 0.1 \times 174.83$$
$$= 2156.65\ N$$

## VI.FINITE ELEMENT ANALYSIS-

Topological optimization of Swing Arm-
Rough CAD Model of Swing Arm

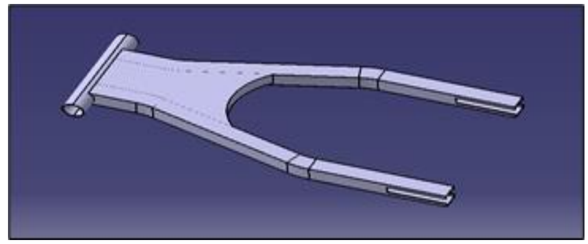


Figure 6 Rough Model

Applied Load-

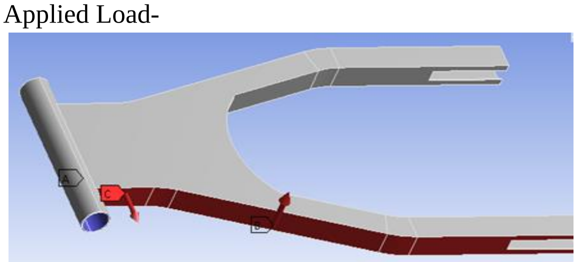


Figure 7 Applied Loads

Processed Model in Inspire

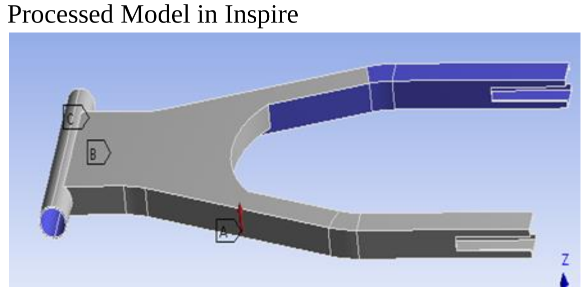
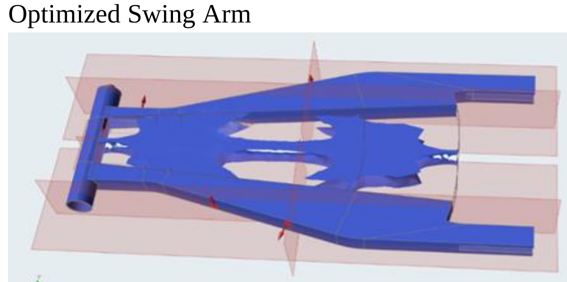


Figure 8 Processed Model

Optimized Swing Arm



Finite Element Analysis Result-
Swing Arm-

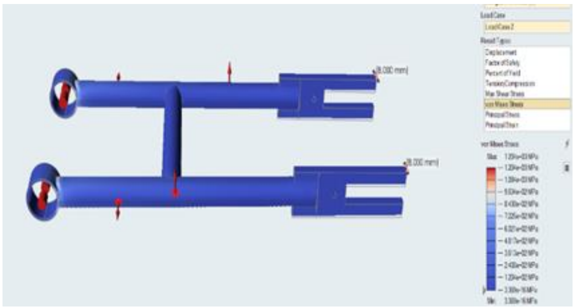


Figure 9  Max Developed Stress

Table 5  Tabulated date of FEA Result

| Material | Maximum Deformation | Maximum Stress | Maximum Yield % |
|---|---|---|---|
| AISI 4130 | 12.7mm | 120.4Mpa | 334.12% |

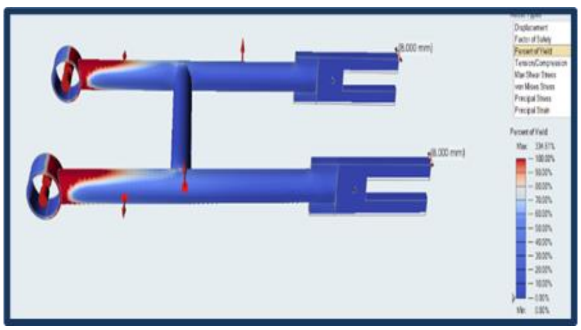


Figure 10  Max. Deformation

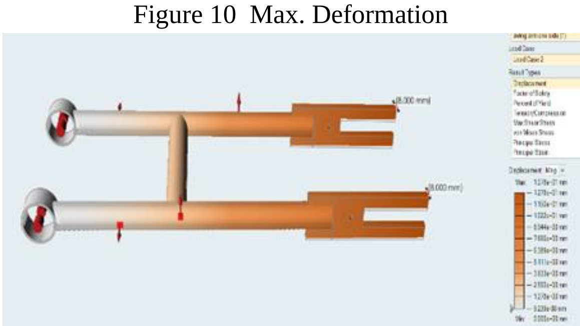


Figure 11 Yield Stress

Helical compression Spring-

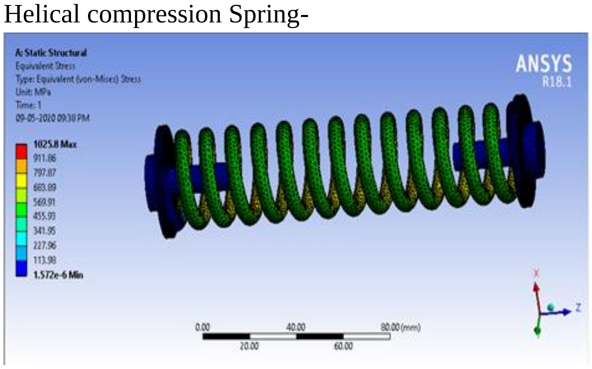



Figure 12 Von Mises Stress

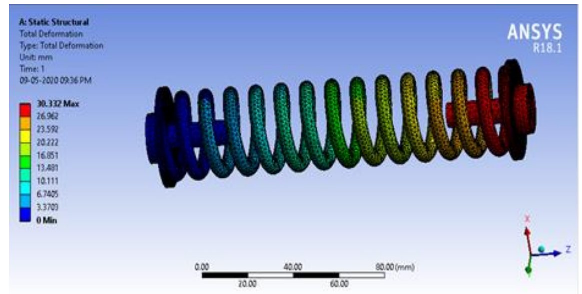



Figure 13 Deformation

Table 6  Result of Spring

| Maximum Deformation | Maximum Stress |
|---|---|
| 30.22 mm | 1025.8Mpa |

## VII.CONCLUSION

It is then concluded from the research paper and patents that; Retrofitting is best alternative available in market for those peoples who could not afford to buy the new Electrical vehicle because of the cost factor. Form the technical point of view there is very slight modification need to be done in OEM vehicle to convert it into electrical vehicle. The components those were selected are sufficient enough to propel the vehicle under the extreme conditions which were not easily applied on the vehicle.

Li-on cell has the maximum specific energy and capacity as compared to the other batteries present in market. using the composite cell of lifepo4 gives the best performance as compared to other cells. Material chosen for the swing arm is composite tube with circular cross section with high bending strength and withstand all dynamic loads comping from the ground. By using the scrap vehicle to convert into electric one is economical and eco-friendly and cost efficient as compared to other vehicles.

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