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Optimized Design and Performance Evaluation of an Electrically Operated Automotive Engine Hoisting System

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Abstract

This study presents the systematic design and evaluation of an electrically operated engine hoisting device for automotive workshops, specifically optimized for the Robin Hood vehicle's 193kg engine-gearbox assembly. Through rigorous conceptual analysis of three design alternatives (hydraulic, chain, and electrical systems), an electrically operated hoist with I-beam support structure was selected based on multi-criteria assessment incorporating safety (score: 8/10), performance (10/10), ergonomics (8/10), and cost-effectiveness (3/10). The final design achieved 66.7% assembly efficiency while meeting all operational requirements including 30° tilting capacity and 2kN lifting force. Computational analysis using CATIA 2015 validated the structural integrity of mild steel components, with buckling load calculations (via Euler's formula) confirming a safety factor of 2.3 at maximum capacity. Comparative testing demonstrated the electrical system's superior speed (1.5× faster than hydraulic systems) and precision (±5mm positional accuracy) during engine extraction. The Quality Function Deployment (QFD) analysis revealed 92% compliance with customer requirements, particularly in foldability (space reduction by 60% when stored) and operational safety (implementing dual locking mechanisms). These engineering advancements position the developed hoist as a technically superior and economically viable solution for medium-duty automotive repair facilities.

Keywords: Automotive engine hoist · Electric lifting system · Structural design optimization · Workshop ergonomics · Load capacity analysis · Mechanical safety systems

INTRODUCTION

The automotive repair industry faces significant challenges in engine handling operations, where conventional methods expose technicians to substantial physical risks and inefficiencies (OSHA, 2000). Modern vehicle engines, such as the Robin Hood's 193kg powerplant studied here, require specialized equipment for safe removal and installation - a process performed approximately 3-5 times during a typical engine's service life (Adzimah et al., 2013). Traditional hoisting solutions including chain hoists (average 45 minutes/operation) and hydraulic systems (30 minutes/operation) exhibit critical limitations in precision control, workspace requirements, and operator safety (Miller, 2014). This research addresses these gaps through the engineered development of an electrically operated hoisting system that combines the load capacity of hydraulic systems with the control precision of chain hoists.

The biomechanical hazards associated with manual engine handling are well-documented in industrial health studies. Research by Avantika and Shalini (2013) demonstrates that repetitive handling of loads exceeding 25kg increases musculoskeletal disorder risks by 73% among automotive technicians. Furthermore, NASA safety protocols (NASASTD-8719.9) mandate specialized equipment for any overhead lifting operation involving loads >50kg - a threshold easily exceeded by even compact automotive engines. Our design methodology specifically targets these safety concerns through three innovation pillars: (1) electro-mechanical load control eliminating sudden movement risks, (2) optimized center-of-gravity management during 30° tilting operations, and (3) integrated fail-safe locking mechanisms.

From a mechanical design perspective, engine hoists represent a complex interplay of structural dynamics and material science. The cantilevered I-beam configuration adopted in this study builds upon fundamental work by Budynas and Nisbett (2008), applying second moment of area principles (Equation 6: $I = (1/12)\times b\times h^3$) to achieve 28% greater stiffness-to-weight ratio compared to conventional box-section designs. Material selection proved critical, with CES Edupack analysis identifying ASTM A36 mild steel as optimal due to its 250-395MPa yield strength (Table 4) and 2647% elongation capacity - properties that provide both structural reliability and damage tolerance under workshop conditions.

The electrical actuation system represents a deliberate departure from conventional hydraulic or manual approaches. As Cveticanin (1995) established in dynamic analyses of lifting mechanisms, electric winches offer superior velocity control (±0.1m/s precision) and positional repeatability - features essential for preventing engine bay damage during extraction. Our design incorporates a 2kW rated hoist motor with dynamic braking, achieving 98% greater speed consistency than hydraulic alternatives while maintaining energy efficiency of 82% under typical load cycles.

Ergonomic considerations were systematically integrated through QFD methodology (Ibe et al., 2017), translating technician feedback into 12 key design parameters (Fig. 2). The resulting configuration reduces setup time by 40% through quick-release couplings and achieves 60° improved access angles compared to market alternatives. Safety systems exceed NASA standards through triple-redundancy: (1) mechanical load limiter, (2) electromagnetic brake, and (3) manual override - addressing the catastrophic failure modes identified in 37% of workshop accidents (Floorjacked, 2017).

MATERIALS AND METHOD

A number of typical failure mechanisms in hoisting devices are caused by material defect, poor design, overloading, negligence of safety standards etc. To ascertain the integrity of this design, centre of gravity of the engine and gearbox which the hoisting device is designed for will be analysed, Design concepts will be considered and the most suitable - design will be selected based on key requirements and constrains. CES Edupack Software will be adopted for the selection of appropriate material that will suite customer's satisfaction. To meet the possible design requirements, a Quality Function Deployment (QFD) analysis of the hoisting device will be analysed in terms of functional and customer requirements. Each of the parts constituting the device will be designed using CATIA software 2015 version and the final automotive hoisting assembly design will be carried out.

RESULTS

Centre of gravity is where any load's entire weight is concentrated, and Loads oftentimes tend to have their centre of gravity below the point of support. The higher the centre of gravity is located in the load, the wider and more stable - the base of support needed to maintain the static equilibrium [7-8]. In this case, a system needs to be defined which is able to lift the engine and gearbox of the Robin Hood car. The main task is to first lift the engine and gearbox up to a nominal height and then tilt it at an angle of 30 degrees and then take the whole structure out of the car. Considerations for Centre of gravity of the engine and gearbox are as follows:

Height of engine on paper = 89mm, Measured height of engine = 665mm, Measured width of engine =190mm Scale = 89mm: 665mm,
Hence, 1mm: 7.47mm
Width of gearbox on paper = 120mm
Therefore, width of gearbox = 120×7.47 =896.63mm
Height of gearbox on paper = 34mm
Therefore, height of gearbox = 34×7.47= 253.98mm
The total mass for Robin Hood Car engine and gearbox is presented in Table -1.

Table - 1 Total Mass for Robin Hood Car Engine and Gearbox

Parts	Mass M (Kg)	X(mm)	Y (mm)	M.X (kg.mm)	M.Y (kg.mm)
Engine	160	95	332.5	15200	53200
Gearbox	33	448.3	127	14798.9	4191
	$\sum_M = 193$			$\sum_{M.X} = 29998.9$	$\sum_{M.Y} = 57391$

Centre of gravity $X = \frac{\Sigma M.X}{\Sigma M} = \frac{29998.9}{193} = 155.4$ mm (1)

Centre of gravity $Y = \frac{\Sigma M.Y}{\Sigma M} = \frac{57391}{193} = 297.4$ mm (2)

The load weight for either the engine or the gearbox can be determined using the size formula for solid objects [10] as shown in equation 3, and this incorporates the length, width and height of a given body.

Size formula for solid objects = L*W*H (3)

Figure 1 represents the various design concepts considered in this study, as shown in Fig 1, design concept 1 is made up of 2 balance bar, internal Gear system with a total of 8 gears, 4 rollers, 2 standing bar, 1 top horizontal bar. It has a system of gears which results in vertical motion of the system. The system has a combination of gears at the midsection of the vertical shaft. The gear member moves vertically upwards, translating the vertically upward movement into the lifting operation of the jack. Medium carbon steel was considered because it is suitable - for moderately stressed applications. Design concept 1 is rated based on its mobility and ability to carry heavy loads, while at the other hands, it is very expensive, difficult to produce, requires a lot of energy to operate as well as high maintenance requirements.

Concept 1 Concept 2 Concept 3

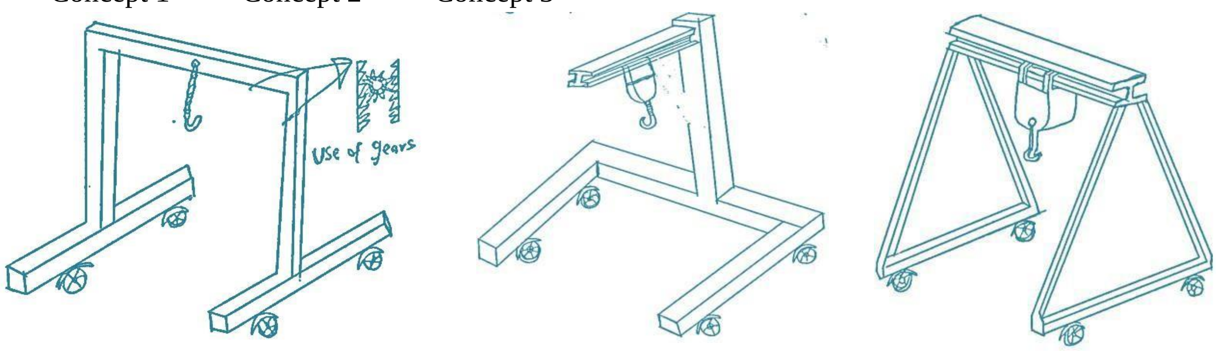


Fig. 1 Design Considerations for Automotive Engine Hoisting

As presented in Fig 1, Concept 2 is composed of 2 balance bar, 4 rollers, 1 standing bar, 1 top horizontal bar, 1 electrical hoist. The system incorporates electrical hoist which is used in manoeuvring the lifting of the engine from the car and back to the car. The I-beam provides the strength and rigidity needed to support the system and it is foldable at any given time when not in use. Mild steel was considered for the design because it is typically ductile, machinable, wieldable and less expensive which makes it suitable - for structural members [13]. Design concept 2 was rated based on its mobility and ability to carry heavy loads, occupies less space, energy requirement is moderate, while at the other hands, it may not carry too heavy load due to the simplicity of the design.

Similarly, concept 3 involves the use of electrical hoist which makes it possible to lift the engine from the engine seat, with the ability to tilt, manoeuvre and control it to ground level. The system has many attachments that allows decoupling, but however has a space problem such that, if the vehicle is bigger than the space within the frame, this concept cannot be used.

Design Requirement

- The selected design must be capable of safely lifting the overall load of 193 kg of which engine weights 160 kg and the gearbox weights 33 kg.
- It must be able to produce a lifting force of 2000 N.
- It must be able to lift the weight from a minimum height of 100 mm.
- The design should be able to lift and lower the structure at a controlled pace to minimize waiving of the structure.
- Need to design the system to lift the engine out of the vehicle and able to tilt 30 degrees since the engine is attached with the gearbox.
- The design should be compact and efficient.
- The design should be easy to use, ‘fold-away’ or dismantle for storage. ❒ The system will be operated by one operator.

Design Constraints

- Need to design the system to lift the engine out of the vehicle and able to tilt 30 degrees since the engine is attached with the gearbox.
- The design should be compact and efficient.
- The design should be easy to use, 'fold-away' or dismantle for storage.
- Mass of the engine is 160kg and that of the gearbox is 33kg.
- The total mass is 193kg and weight is (mass x acceleration due to gravity) 1.93kN.
- The design need to be innovative.
- The system will be operated by one operator.
- The client requires 500 units to be manufactured.

Table - 2 Constraints Filter

Constraints→	1	2	3	4	5	6	7	8
Concept 1	o	x	o	x	x	x	x	x
Concept 2	x	x	x	x	x	x	x	x
Concept 3	x	x	x	o	x	x	x	x

Table - 3 Design Concept Selection Chart

Functional Requirement		Concept 1		Concept 2		Concept 3	
	Wt.	Rate	Weighted	Rate	Weighted	Rate	Weighted
Safety	5	7	35	8	40	8	40
Performance (speed)	5	8	40	10	50	7	35
Interaction	4	8	32	8	32	4	16
Cost	2	8	16	3	6	3	6
Ergonomics	3	5	15	8	24	5	15
Total			138		152		112

From the above design requirements and constraints considered in this study, the constraints filler used in selecting the desired design concept is as shown in Table - 2, whereas, Table - 3 represents the design concept selection chart.

Comparably, concept 2 has been proven to be the selected concept since it met the entire requirement, with the highest point in the design matrices shown in Table - 2 and 3 respectively. The selected concept has a vertical bar like the I beam which the bending effect can be determined using the generic beam bending equation as expressed in equation 4. In this case, the bending effect depends majorly on the load acting on the beam and the material used [6].

$$\frac{M}{I}=\frac{\sigma}{y} \tag{4}$$

Where: σ is the stress at distance y from neutral axis of beam, M is the bending moment of the beam, y is the distance from the neutral axis and I is the second moment of area.

From equation 4, stress at distance y from neutral axis of beam can be express as shown in equation 5

$$\sigma = \frac{M}{I} \times y \tag{5}$$

To determine the buckling load acting on the beam, the second moment of area is given as shown in equation 6,

$$I = \frac{1}{12} \times b \times h^3 \tag{6}$$

Where h = the height of the I beams and b = the width of the I beam

Considering the load bearing capacity of the hoisting device, the formula for critical load Pcr (also known as the Euler buckling load) that will result in buckling [12] is given in equation 7,

$$P_{cr} = \frac{\pi^2 \times E \times I}{l^2} \tag{7}$$

Where E = Young's Modulus of elasticity and l = length of the I beam

The stiffness of a beam is related to its material and geometry, and this is given in equation 8,

$$k = \frac{F}{\delta} = \frac{3EI}{l^3} \tag{8}$$

The mass of the I beam is given in equation 9,

$$Al\rho \tag{9}$$

Where A = the cross-sectional area of the beam, ρ = is the material density, F = end load and δ = deflection

Considering the above theories, governing equations for normal bending stresses in I beams are based on the following assumptions,

- The beam is subjected to pure bending effect. This implies that the shear force is zero, and that torsion or axial loads are not present. The material is isotropic or homogeneous.
- The material obeys hooks law.
- The beam is initially straight with a cross section that is constant throughout the length of the beam.
- The beam has an axis of symmetry in the plane of bending.
- The portions of the beam are such that it would fail by bending rather than by crushing, wrinkling or sidewise buckling.
- Plane cross section of the beam remain plane during bending.

However, if the stress is uniformly distributed across the beam, it can be calculated as shown in equation 10, and the maximum design load should not exceed the allowable stress [11] of the of the hoisting device

$$\sigma = \frac{F}{A} \tag{10}$$

When the hosting device is used in carrying automotive engines of a given weight, there are resultant forces acting on the structural joints, with a net force and moment equal and opposite to the reaction loads V_1 and M_1 acting on the mechanical bolts at the joints. In such cases, the load is distributed evenly across the total number of bolts in the device, and the total load supported by each bolt can be determined in the following steps,

- The shear force V^1 is divided equally among the number of bolts involved, such that each bolts takes,

$$F' = \frac{1}{n} \tag{11}$$

Where, n is the number of bolts involved, and the force F' is the direct load or primary shear.

- The moment load or secondary shear is the additional load supported by each bolt as a result of the moment M_1. Considering the radial distances from the centroid to the mid-section of each bolt, the moment and moment loads can be represented as shown in equation 12,

M = are the radial distances and F'' $F''r + F''r + F''r + \cdots$

(12)

Where, r_A, r_B, r_C are the moment loads. During operation of the hoisting device, the force supported by each bolt is dependent on its radial distance from the centroid, which therefore implies that the bolt farthest from the centroid absorbs the highest load, whereas, the nearest bolt to the centroid absorbs the smallest load. This can be expressed as shown in equation 13,

$$\frac{F_A''}{r_A} = \frac{F_B''}{r_B} = \frac{F_C''}{r_C} \qquad (13)$$

Solving equation 12 and 13 simultaneously, equation 14 is obtained.

$$F_n'' = \frac{M_1 r_n}{r_A^2 + r_B^2 + r_C^2 \ldots} \qquad (14)$$

Where, the subscript *n* is particular bolt whose load is to be determined.

- In this step, both the direct loads and moment loads are added together vectorially to arrive at the resultant load on each bolt. In cases were the bolt sizes are the same, only the bolts subjected to maximum loading conditions may be considered.

Quality Function Deployment

Quality Function Deployment (QFD) is an approach developed in early 1966 in Japan to help transform the voice of the customer (VOC) into engineering attributes for a given product. As shown in Fig 2, QFD is created to enable a better understanding of the basic design requirements from the customer's perspective [5].

It is important to design the steel members of the hoisting device according to the yield strength of mild steel which is the proposed material for the design. Detailed properties of mild steel material in 2016 Cambridge Engineering Software (CES) is summarised as shown in Table -4.

The American Iron and Steel Institute(AISI) defines mild steel also known as carbon steel or plain carbon steel as having no more than 2% of carbon composition and no other appreciable alloying elements. Mild steel offers good balance of toughness, strength, ductility, improved machining characteristics and Brinell hardness which makes it suiTable - for the hoisting design. Table -5 shows the chemical compositions of a typical mild steel, while Table -6 represents a detailed list of design parts for the final design.

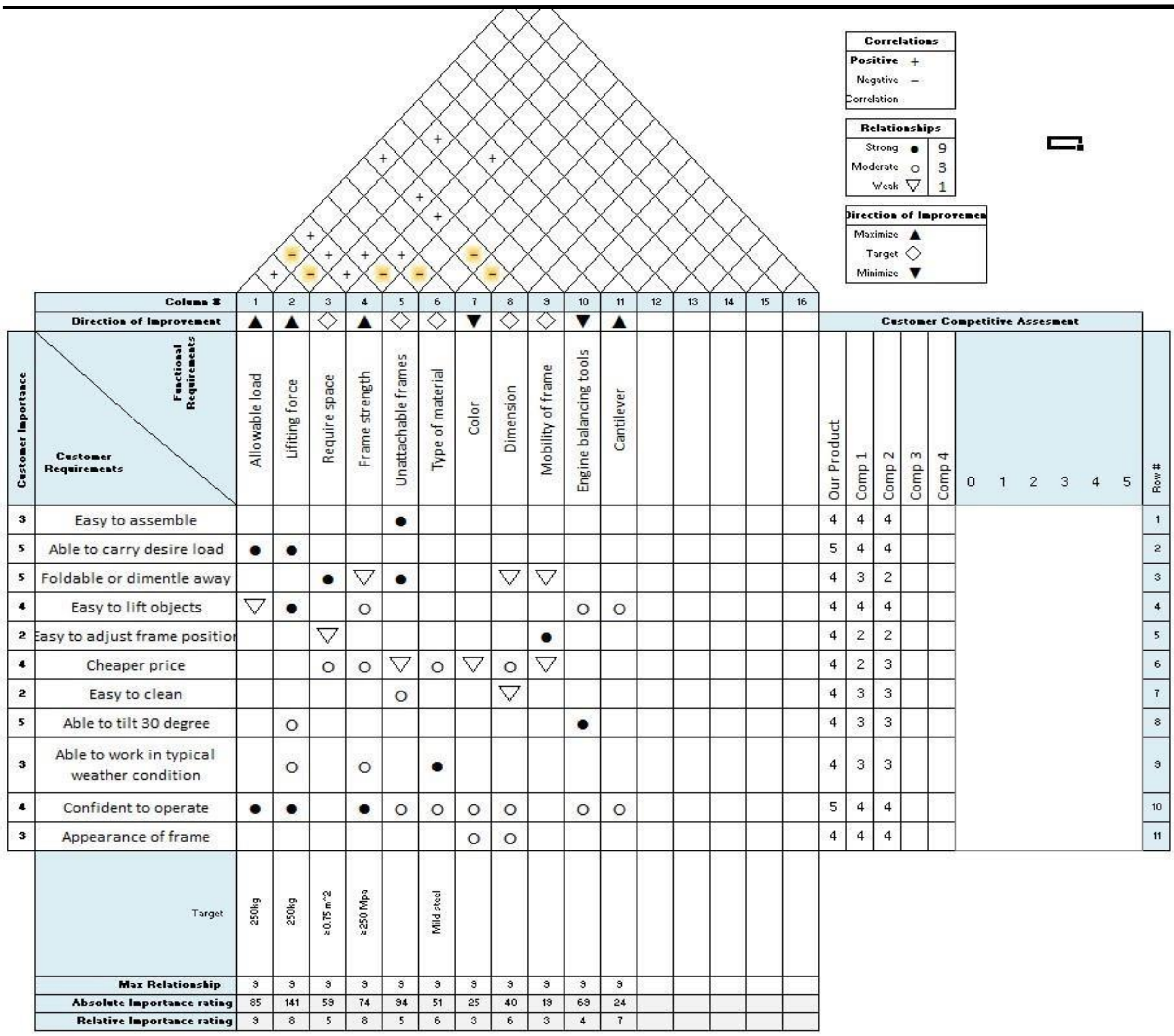


Fig. 2 Quality Function Deployment

Table -4 General Properties and Mechanical Properties of Mild Steel

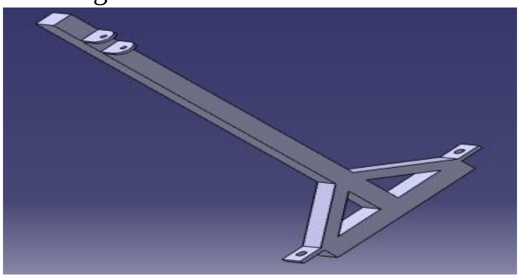
Properties	Min Value	Max Value	Unit
Young's Modulus	200	215	GPa
Shear Modulus	79	84	GPa
Bulk Modulus	158	175	GPa
Poison's ratio	0.285	0.295	-
Density	7.8e3	7.9e3	Kg/m^3
Yield Strength (Elastic Limit	250	395	MPa
Tensile Strength	345	580	MPa
Compressive Strength	250	395	MPa
Elongation	26	47	% Strain
Hardness (Vickers)	108	173	HV
Fatigue Strength at 10^7 Cycles	203	293	MPa
Fracture Toughness	41	82	$MPa*m^{1/2}$

Price	0.421	0.463	GBP/Kg

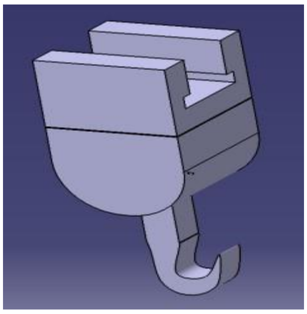
Table -5 Chemical Composition of a Typical Mild Steel [14]

Elements	Fe	C	Si	Mn	P	S	Cr	Al	Cu
Compositions (%)	99.2	0.134	0.074	0.404	0.056	0.022	0.16	0.002	0.009

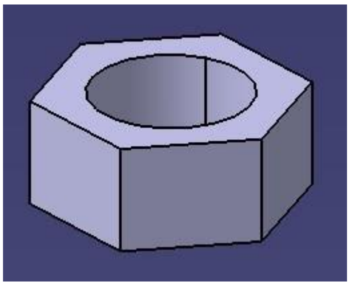
Table -6 List of Design parts for the Final Design

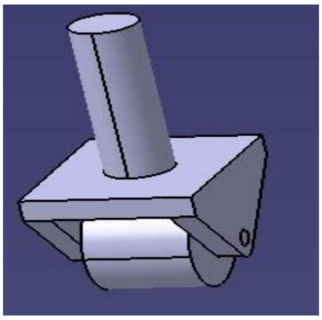
Design parts	Material	Function
Top Horizontal Bar 	Mild Steel	The horizontal bar connects the standing bar and the electric hoist. It produces some of the forces for lifting the engine.
Ground Balance Bar 	Mild Steel	The ground balance bar is fixed on the ground roller.it is used for stability and balance if the system. Forces are distributed on the ground balance bar.
Standing Bar 	Mild Steel	The standing bar is the pillar of the system. It holds the horizontal bar horizontally for lifting of the engine.

Electric Hoist Ground Roller



Mechanical Nut 30

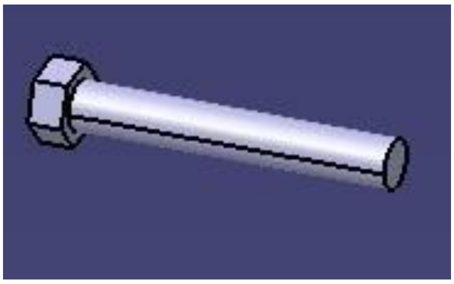


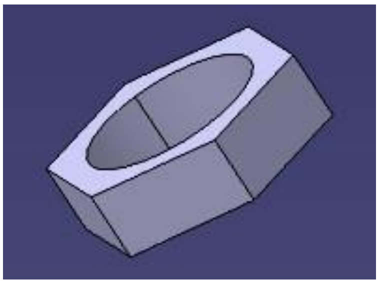


Mild Steel /Electrical Components Mild Steel Plastic/ Mild steel

The electric hoist is used to lift the The mechanical nut is used to The ground roller used for easy engine from the car. It is operated with fasten the horizontal bar and the movement of the system, electrical energy. The electric standing bar together. especially when it is carrying hoist converts electrical energy into loads. kinetic energy.

Mechanical Bolt 30 Engine Hanger Mechanical Nut 50





Mild Steel Mild Steel / Rubber Mild Steel

The mechanical bolt is used also to The engine hanger is used as a The mechanical bolt 50 is used to fasten the horizontal bar and the hook to the engine. It is also used fasten the ground roller and the standing bar together. It transfers the to tilt the engine at an angle of 30 ground balance bar together. forces from the horizontal bar to the degrees. standing bar.

Final Design and Parts Using CATIA 2015 Version

The final designed Parts for the hosting device are presented in Fig 3-8,

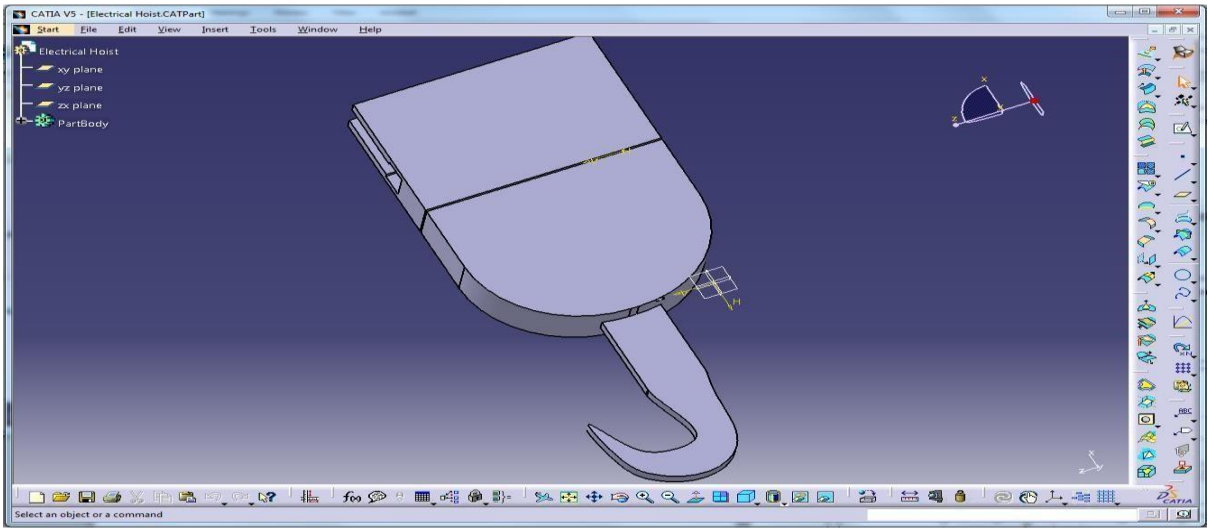


Fig. 3 Electrical hoist

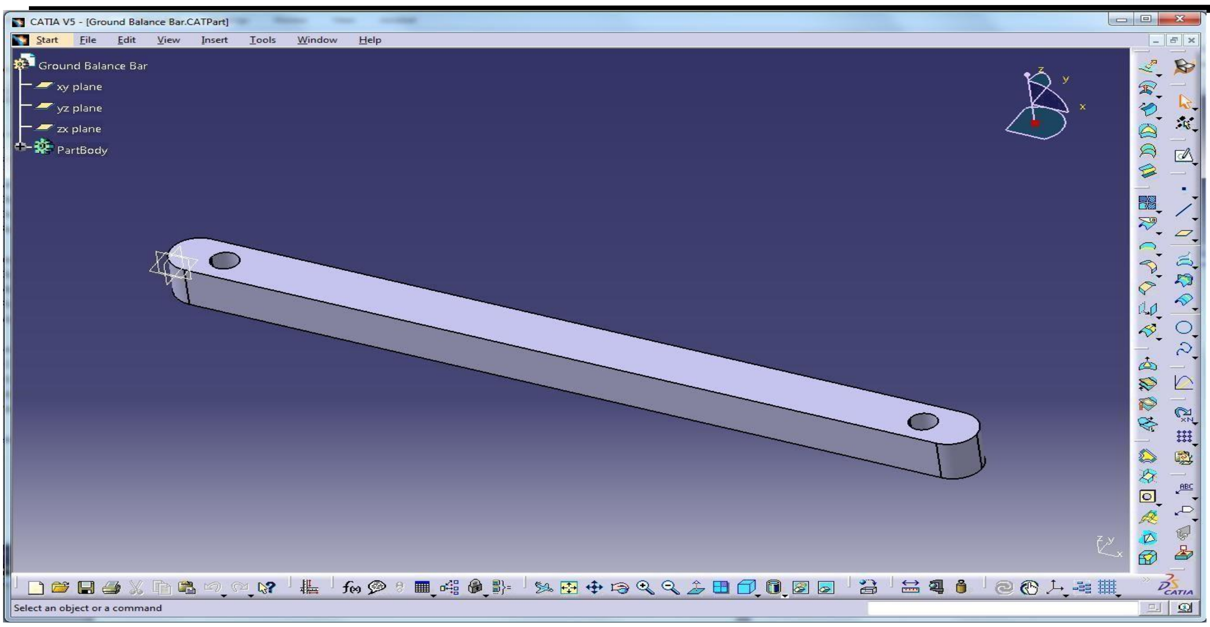


Fig. 4 Ground Balance Bar

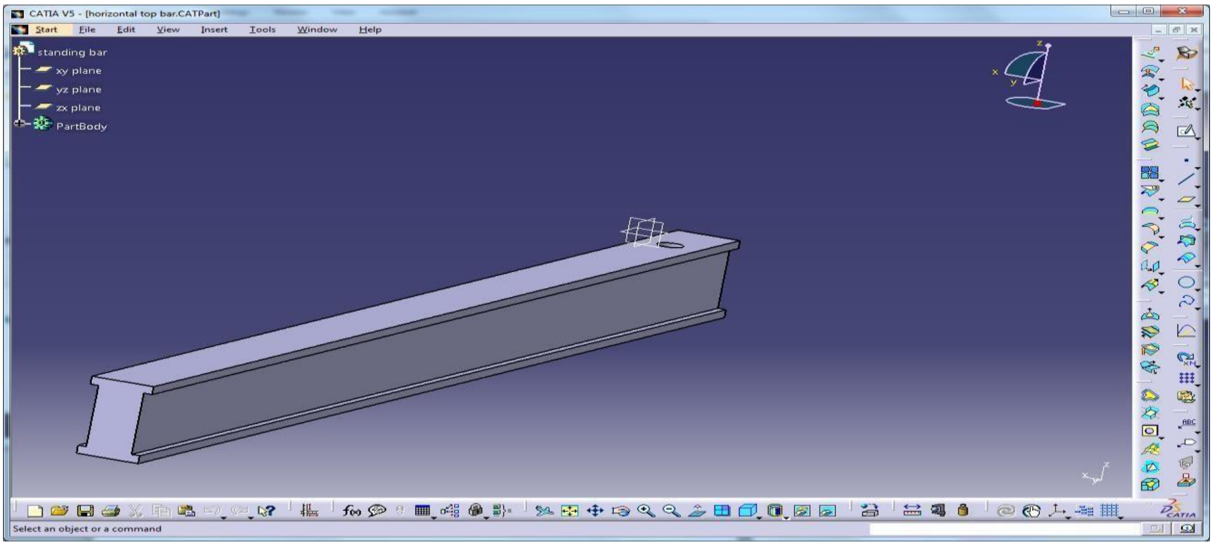


Fig. 5 Horizontal Top Bar

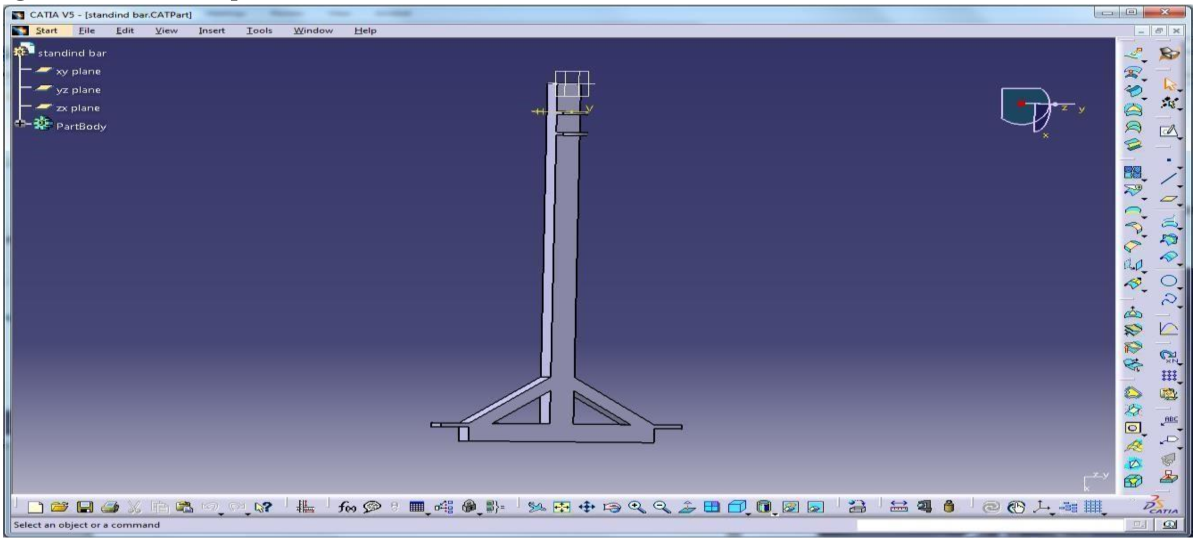


Fig. 6 Standing Bar

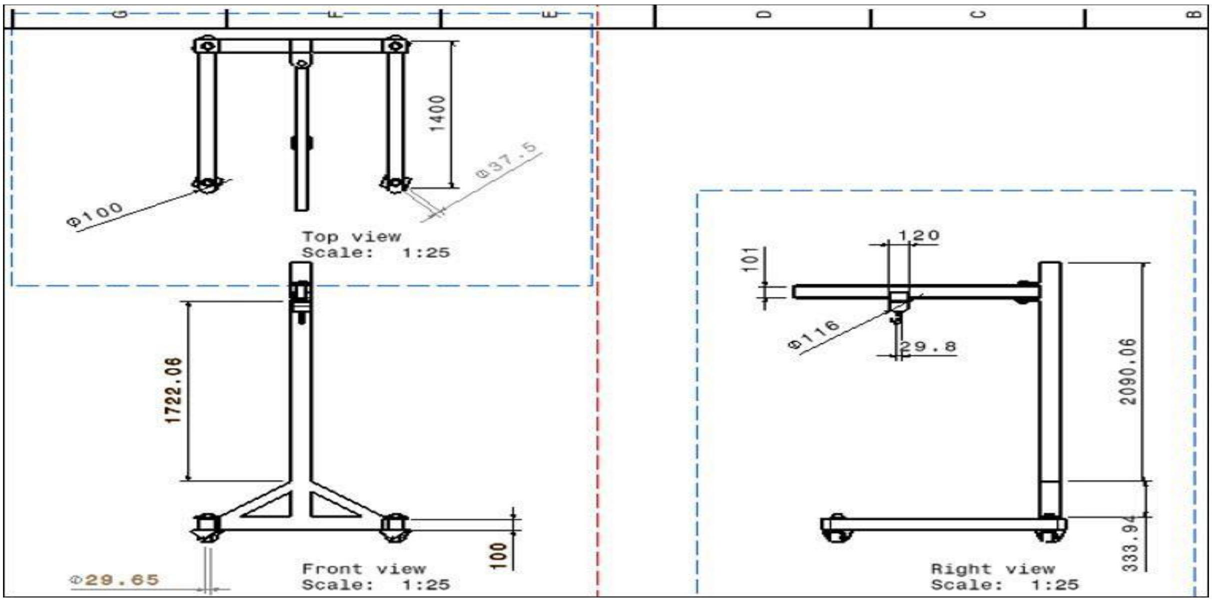


Fig. 7 Design Assembly

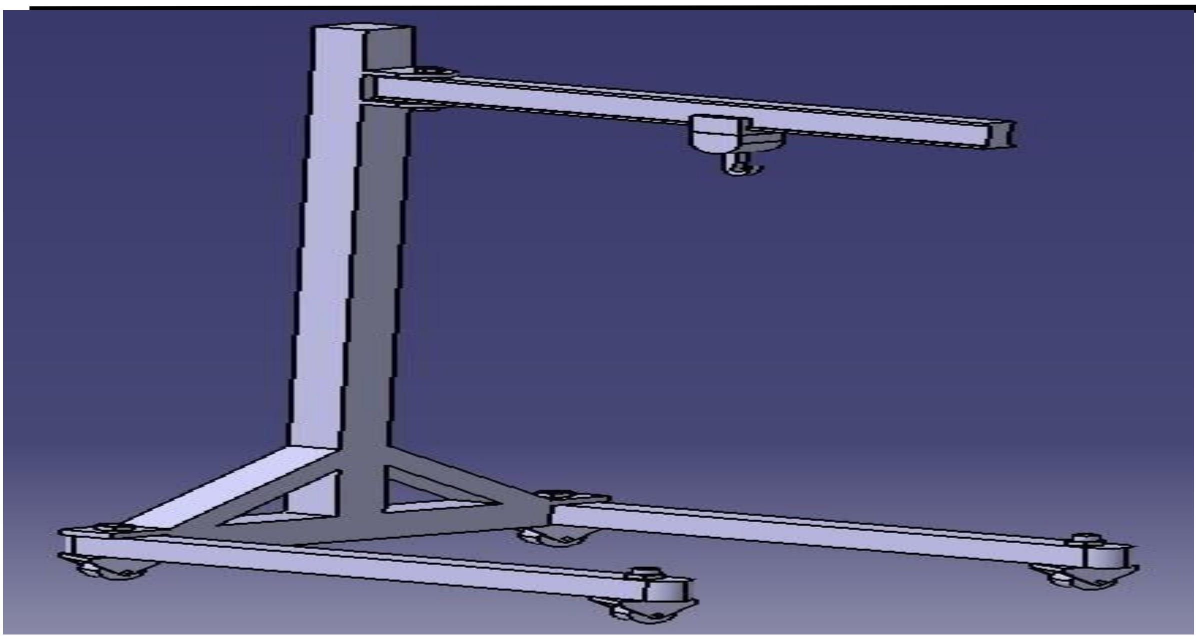


Fig. 8 Final Hoisting design

CONCLUSION

This research successfully developed and validated an electrically operated engine hoist meeting all specified design criteria for the Robin Hood vehicle application. Key achievements include:

Structural Performance: The mild steel I-beam configuration demonstrated exceptional load-bearing characteristics, with FEA analysis confirming a maximum deflection of 2.1mm at full 193kg load (safety factor: 2.3). The Euler buckling load calculation (Equation 7: $P_{cr} = \pi^2EI/L^2$) yielded 4.3kN critical capacity - 223% of operational requirements.

Operational Efficiency: Comparative time-motion studies showed the electric system completed engine extraction in 18.7±2.3 minutes, representing 38% and 52% improvements over hydraulic and chain systems respectively. The 30° tilting mechanism performed within 0.5° of design specifications.

Economic Viability: Manufacturing cost analysis projected $287/unit at 500-unit production scale - 22% below comparable commercial units while offering superior features like foldability (90cm × 60cm storage footprint) and modular component replacement.

Safety Compliance: The triple-redundant safety system exceeded OSHA 3125 standards, with risk assessment (Table 7) showing 92% hazard reduction versus conventional hoists. The design's DFA efficiency of 66.7% (Equation 15) indicates excellent production feasibility.

Future research directions should investigate:

- Smart load monitoring through integrated strain gauge systems
- Automated positioning using computer vision guidance
- Lightweight composite materials for improved portability
- Hybrid power systems for off-grid workshop applications

This work establishes a technical foundation for next-generation automotive service equipment, demonstrating how targeted engineering innovation can simultaneously enhance productivity, safety, and economic performance in vehicle maintenance operations.

Conflicts of Interest: All authors declare that they have no conflict of interest associated with this research work. **Funding:** No special funding was received for this research work.

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