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FLOOR CRANE STRUCTURE DESIGN AS A TOOL FOR REPLACING BATTERY PACKS ON TMII TRAM MOVER

Achmad Z. Anwar[1], I Putu S. Asmara[2], Dhika A. Purnomo[3]*, Saiful Arif[4]
*E-mail Korespondensi: adityadhika@ppns.ac.id
[1,3]Engineering Design and Manufacture Study Program, Department of Mechanical Engineering
Politeknik Perkapalan Negeri Surabaya, Indonesia
[2]Ship Design and Construction Engineering Study Program, Shipbuilding Engineering Department
Politeknik Perkapalan Negeri Surabaya, Indonesia
[4]Mechanical Engineering Study Program, Department of Mechanical Engineering
Politeknik Negeri Malang, Indonesia

ABSTRAK

Tram mover adalah kereta buatan PT INKA (Persero) sebagai salah satu wahana di TMII (Taman Mini Indonesia Indah) yang menggunakan baterai sebagai sumber energi. Pada proses penggantian baterai masih menggunakan pengangkatan manual dengan membutuhkan 4 orang pekerja. Setiap penggerak trem memiliki 2 paket baterai, membutuhkan 4 lift di setiap proses penggantian. Tidak ada produk floor crane di pasaran yang dapat digunakan sebagai alat pengganti dalam kondisi stasiun TMII. Dalam penelitian ini, dilakukan perancangan alat untuk penggantian battery pack. Floor crane dengan kapasitas, kekuatan, dan dimensi disesuaikan dengan kondisi di TMII dan tram mover. Proses analisis akan menggunakan pendekatan FEM (fenite element methods) dengan menggunakan software berbasis FEM. Validitas hasil simulasi dilakukan terhadap nilai hasil faktor keamanan, von misses stress, dan defleksi. Beban yang digunakan dari berat battery pack adalah 275 Kg. Analisis tegangan statis lengan diperoleh nilai faktor keamanan minimum aktual 2,004, tegangan von mises maksimum 123,9 MPa, dan perpindahan total maksimum 5,206 mm. Pada kruk diperoleh nilai faktor keselamatan minimum aktual sebesar 2,518, tegangan von mises maksimum 98,59 MPa, dan perpindahan total maksimum 0,195 mm. pada sasis diperoleh nilai faktor keamanan minimum aktual sebesar 2,027, tegangan von mises maksimum 109,4 MPa, dan *displacement total maximum* sebesar 0,2813 mm.

Keywords: fusion 360, product redesign, optimization, FEA simulation, structural analysis

ABSTRACT

Tram mover is a train made by PT INKA (Persero) as one of the rides at TMII (Taman Mini Indonesia Indah) which uses batteries as an energy source. In the battery replacement process still uses manual lifting by requiring 4 workers. Each tram mover has 2 battery packs, requiring 4 lifts in each replacement process. There is no floor crane product on the market that can be used as a replacement tool in TMII station conditions. In this research, the design of tools for battery pack replacement is carried out. A floor crane with capacity, strength, and dimensions adapted to the conditions at TMII and the tram mover. The analysis process will use FEM (fenite element methods) approach using FEM based software. The validity of the simulation results is carried out on the value of the result of the safety factor, von misses stress, and deflection. The load used from weight of battery pack is 275 Kg. Arm static stress analysis obtained an actual minimum safety factor value of 2.004, maximum von mises stress of 123.9 MPa, and maximum total displacement of 5.206 mm. On the crutch obtained the actual minimum safety factor value of 2.518, stress von mises maximum of 98.59 MPa, and a maximum total displacement of 0.195 mm. on the chassis obtained the actual minimum safety factor value of 2.027, stress von mises maximum of 109.4 MPa, and a maximum total displacement of 0.2813 mm.

Keywords: fusion 360, product redesign, optimization, FEA simulation, structural analysis

INTRODUCTION

PT Industri Kereta Api (Persero) or PT INKA (Persero) is the first integrated train manufacturing State-Owned Enterprise (BUMN) in Southeast Asia which was founded on May 18 1981. PT INKA (Persero)'s commitment is to produce a variety of quality railway facilities products by continuing to innovate supported by high technology, modern production facilities, reliable human resources, as well as providing customer satisfaction and operational excellence so that the products produced can be well received by consumers. One of these commitments was realized by PT INKA (Persero) by participating in the success of the G20 Summit in Bali in November 2022, namely by producing tram movers for TMII.

The TMII tram mover is equipped with autonomous technology so that its operation does not require a driver. Apart from that, the tram mover also uses a propulsion system in the form of a motor and battery. The battery functions as an energy supplier to all electrical components in the tram mover, this makes the battery very vital as a power source for electrical components, including as a driving system for all existing elements such as AC, lights, sensors, etc.

TMII provides pre-charged replacement, namely replacing a discharged battery pack with a charged battery pack. This is intended to the tram mover can continue to operate without having to wait for the refilling process which takes 3 hours. Pre-charged replacement still uses conventional methods or uses human power and requires 4 workers. Each tram mover has 2 battery packs, so it requires 4 lifts for each replacement process. There is no floor crane product on the market that can be used as a replacement tool at TMII station conditions. The condition of the TMII station is that there is only one side of the platform and there is a gap between the tram mover and the platform, so the battery pack replacement tool requires you to get from the platform through the tram mover door.

Therefore, in this research, a floor crane structure was designed as a tool to help replace battery packs at TMII stations and it is hoped that this will reduce the number of workers involved in the battery pack replacement process.

Research Method

Identification and literature study are stages for determining the problem formulation and objectives of the research and studying previous research. Then proceed with a study of products on the market to plan the floor crane design, calculate floor crane components, and carry out FEM simulations, as well as selecting supporting components.

To be more concise, the research flow in this article can be seen in Figure 1.

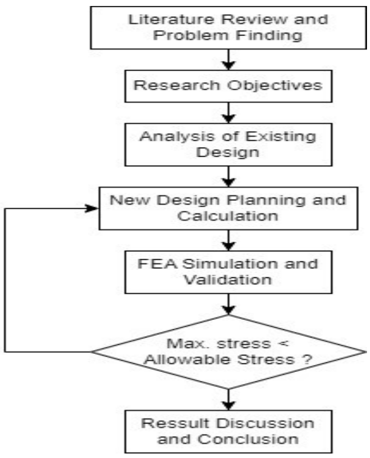


Figure 1. Research Method

RESULT AND DISCUSSION

Analysis of Existing Product Design

This study uses the Big Foot Floor Crane 280 product which can be seen in Figure 2.

Figure 2. Existing Product, Big Foot Floor Crane 280

Based on the specifications of the Big Foot Floor Crane 280, analysis is needed to determine the advantages and disadvantages when applied to lift the battery pack. The advantages and disadvantages of the Big Foot Floor Crane 280 can be seen in Table 1.

Table 1. Advantages and Disadvantages of Big Foot Floor Crane

No.	Advantages	Disadvantages
1.	Has 4 arm lifting/output distances	The longest output arm is 920 mm, unable to reach the battery pack from the platform
2.	Lift load capacity maximum 280 kg	The manufacturing process Lots
3.	Maximum lifting height 2700 mm	Counterweights cannot be moved all at once because there is no container
4.	Floor crane height is more lower than the entrance of the TMII Tram Mover which is 1820 mm	*Handle is not ergonomic because of the wrong height*
5.	Wheels can be used on uneven place	
6.	*Counterweight can be moved and adjusted*	
7.	Can be folded for easy storage	

Floor Crane Design

This design is drawn according to the conditions of the TMII station. The data inputted is the condition of the platform and dimensions of the Tram Mover. A modeling drawing of a floor crane can be seen in Figure 3.

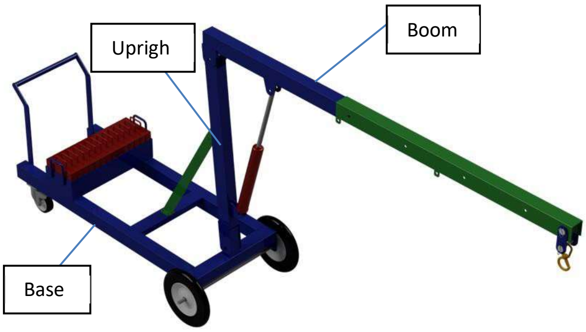


Figure 3. Floor Crane Design

Calculation of structural strength

Structural calculations are needed on the floor crane components to determine the profile, so that the floor crane is safe to use for lifting battery packs.

a. Boom

The following is a free body diagram on the arm according to predetermined dimensions which can be seen in Figure 4.

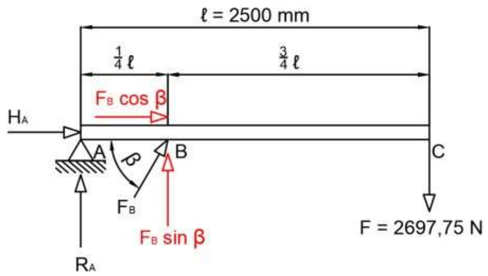


Figure 4. Boom free body diagram

$$
\begin{aligned}
{}^{+}\circlearrowleft \Sigma M_A &= 0\\
R_A \times 0 - F_B \sin\beta \times \tfrac{1}{4}\ell + F \times \ell &= 0\\
0 - F_B \sin\beta \times \tfrac{1}{4}\ell + F \times \ell &= 0\\
F_B \sin 60{,}927^\circ &= 10791 \text{ N}\\
F_B &= 12346{,}715 \text{ N}
\end{aligned}
$$

$$
\begin{aligned}
{}^{+}\uparrow \Sigma F_y &= 0\\
R_A + F_B \sin\beta - F &= 0\\
R_A &= -8093{,}250 \text{ N}
\end{aligned}
$$

$$
\begin{aligned}
{}^{+}\rightarrow \Sigma F_x &= 0\\
H_A + F_B \cos\beta &= 0\\
H_A &= -F_B \cos 60{,}927^\circ\\
&= -5999{,}641 \text{ N}
\end{aligned}
$$

From the results of the calculation of the equilibrium on the boom, the calculation of the shear stress and bending moment is carried out on the boom with the results in Figure 5. From Figure 4. of the moment diagram, it can be seen that Mmax = 5058281.25 Nmm.

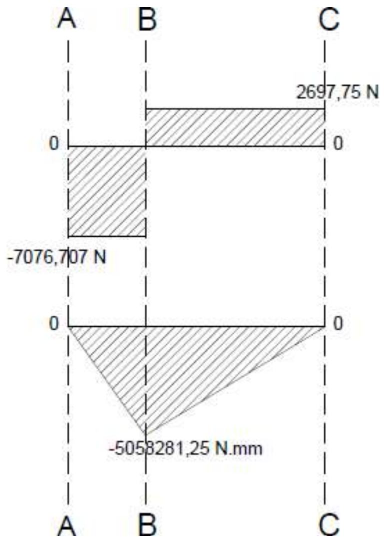


Figure 5. Boom SFD and BMD Diagram

b. Upright

The following is a free body diagram on a support according to predetermined dimensions which can be seen in Figure 6.

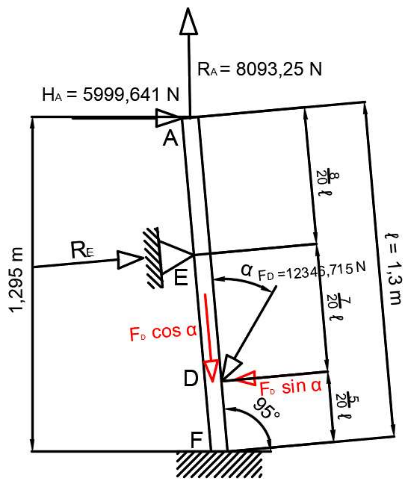


Figure 6. Upright free body diagram

Calculating the moment in the cantilever using the slope deflection method.

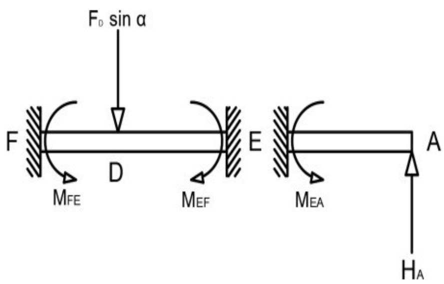


Figure 7. Upright Moment Diagram

$$M_{FFE} = -\frac{F_D \sin\alpha \times 0{,}324 \times 0{,}453^2}{1{,}295^2}$$
$$= -\frac{6917{,}231 \times 0{,}324 \times 0{,}453^2}{1{,}295^2}$$
$$= -275{,}891 \text{ Nm}$$
$$M_{FEF} = \frac{F_D \sin\alpha \times 0{,}324^2 \times 0{,}453}{1{,}295^2}$$
$$= \frac{6917{,}231 \times 0{,}324^2 \times 0{,}453}{1{,}295^2}$$
$$= 197{,}326 \text{ Nm}$$

calculation *final moment using slope deflection equation.*

$$M_{FE} = -275{,}891 + \frac{2EI}{1{,}3}(2\theta_A + \theta_B)$$
$$= -275{,}891 + 1{,}539\ EI\ \theta_B$$
$$M_{EF} = 197{,}326 + \frac{2EI}{1{,}3}(\theta_A + 2\theta_B)$$
$$= 197{,}326 + 3{,}078\ EI\ \theta_B$$
$$M_{EA} = H_A \times 0{,}52$$
$$= 3119{,}813 \text{ Nm}$$

Calculation on B.

$$M_{EF} + M_{EA} = 0$$
$$(197{,}326 + 3{,}078\ EI\ \theta_B) + 3119{,}813 = 0$$
$$3317{,}139 + 3{,}078\ EI\ \theta_B = 0$$
$$EI\ \theta_B = -1077{,}693$$

Moment on F.

$$M_{FE} = -275{,}392 + 1{,}539\ EI\ \theta_B$$
$$= -1933{,}962 \text{ Nm}$$

$$^{+}\circlearrowright\Sigma M_F = 0$$
$$M_F - F_D \sin\alpha \times 0{,}324 + R_E \times 0{,}777 + H_A \times 1{,}295 = 0$$
$$1933{,}962 - 2241{,}182 + R_E \times 0{,}777 + 7769{,}535 = 0$$
$$R_E = -9604{,}009 \text{ N}$$

$$^{+}\rightarrow F_x = 0$$
$$H_A - R_E - F_D \sin\alpha + F_{Fx} = 0$$
$$5999{,}641 - 9604{,}009 - 6917{,}231 = -F_{Fx}$$
$$F_{Fx} = 10521{,}599$$

$$^{+}\uparrow F_y = 0$$
$$R_A - F_D \cos\alpha + F_{Fy} = 0$$
$$8093{,}250 - 10227{,}086 = F_{Fy}$$
$$F_{Fy} = 2133{,}836 \text{ N}$$

From the results of the equilibrium calculation on the upright support, the shear stress and bending moment on the upright support were calculated with the results in Figure 8. From Figure 8. the moment diagram, it can be seen that Mmax = 3119.813 Nm = 3119813 Nmm

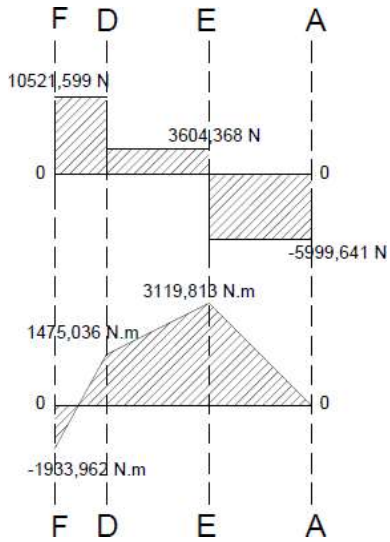


Figure 8. Upright SFD and BMD Diagram

c. Base

The following is a free body diagram of the arm according to the specified dimensions which can be seen in Figure 9.

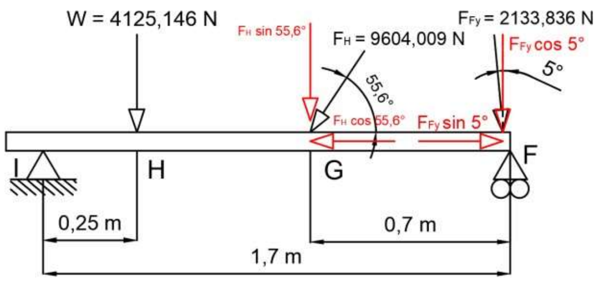


Figure 9. Base free body diagram

$$^{+}\circlearrowright\Sigma M_I = 0$$
$$F_{Fy} \cos 5° \times 1{,}7 + F_G \sin 55{,}6 \times 1 + W \times 0{,}25 - R_F \times 1{,}7 = 0$$
$$3613{,}717 + 7924{,}397 + 1031{,}287 - R_F \times 1{,}7 = 0$$
$$R_F = 7393{,}765 \text{ N}$$

$$^{+}\uparrow F_y = 0$$
$$-F_{Fy} \cos 5° - F_G \sin\delta - W + R_F + R_I = 0$$
$$-2125{,}716 - 7924{,}397 - 4125{,}146 + 7393{,}765 + R_I = 0$$
$$R_I = 6781{,}494 \text{ N}$$

$$^{+}\rightarrow F_x = 0$$
$$F_{Fy} \sin 5° - F_G \cos 55{,}6 + H_I = 0$$
$$185{,}976 - 5425{,}948 + H_I = 0$$
$$H_I = 5239{,}972 \text{ N}$$

From the results of the equilibrium calculation on the base, the shear stress and bending moment on the base were calculated with the results in Figure 10. From Figure 10. of the moment diagram below, it can be seen that Mmax = 3687.634 Nm = 3687634 Nmm

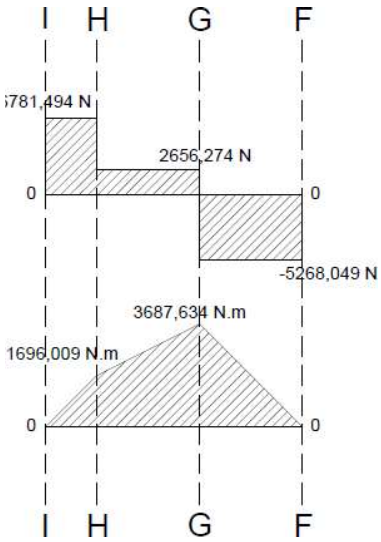


Figure 10. Base SFD and BMD Diagram

Calculation of Profile Components

The frame profile is planned to use a rectangular hollow profile with ASTM A36 material with a tensile strength value of 250 N/mm2.

Allowable Stress

$$\sigma_i = \frac{\sigma_b}{Sf} = \frac{250}{2} = 125\ \text{N/mm}^2$$

Determine the minimum cross-sectional modulus of the boom profile

$$Z = \frac{M}{\sigma_i} = \frac{5058281{,}25}{125} = 40466{,}25\ \text{mm}^3 = 40{,}466\ \text{cm}^3$$

Obtained profile dimensions from the catalog for rectangular hollow with dimensions 100×80×5 mm for main arm and U profile with dimensions 100×100×5 mm for extend boom.

Determine the minimum sectional modulus of the upright support:

$$Z = \frac{M}{\sigma_i} = \frac{3119813}{125} = 25572{,}238\ \text{mm}^3 = 25{,}572\ \text{cm}^3$$

The profile dimensions are obtained from the catalog for a rectangular hollow with a size of 100×80×5 mm for boom supports.

Determine the minimum sectional modulus of the Base:

$$Z = \frac{M}{\sigma_i} = \frac{3687634}{125} = 29501{,}072\ \text{mm}^3 = 29{,}501\ \text{cm3}$$

Obtained profile dimensions from the catalog for rectangular hollow with dimensions of 100×80×5 mm for the Base.

Deflection of the Boom

The amount of deflection shows how much shape change occurs in a component. The normal deflection is said to be adequate if the value is smaller than the allowable deflection. The allowable deflection and normal deflection can be known using the equation.

$$\delta_{izin} = \frac{L}{150} = \frac{2500}{150} = 16.67\ \text{mm}$$

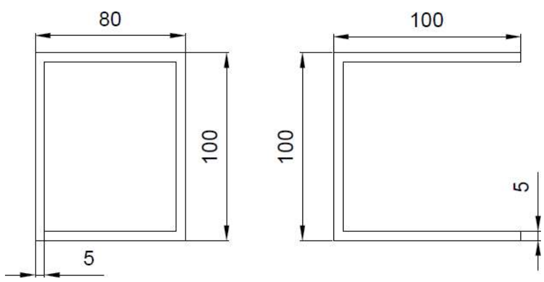


Figure 11. Main boom and extended boom Profile Dimension

Boom Momen Inersia

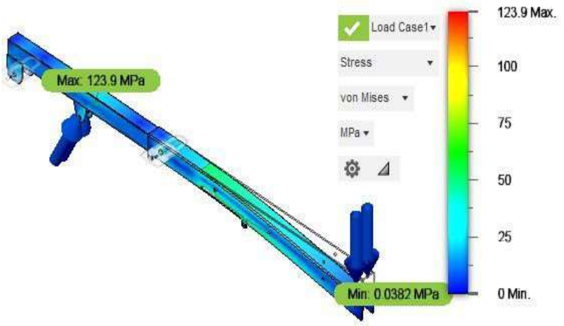
$$I_1 = \frac{1}{12}\left[(b \times h^3) - (b_1 \times h_1^3)\right]$$
$$= \frac{1}{12}\left[(80 \times 100^3) - (70 \times 90^3)\right]$$
$$= 2489166{,}667 \text{ mm}^4$$
$$I_2 = \frac{1}{12}\left[(b \times h^3) - (b_1 \times h_1^3)\right]$$
$$= \frac{1}{12}\left[(100 \times 100^3) - (95 \times 90^3)\right]$$
$$= 2562083{,}333 \text{ mm}^4$$
$$I_{total} = I_1 + I_2$$
$$= 5051250 \text{ mm}^4$$
$$\delta = \frac{PL^3}{3EI} - \frac{Pa^2}{6EI}(3L - a)$$
$$= \frac{2697{,}75 \times 2500^3}{3 \times 200000 \times 5051250} - \frac{2697{,}75 \times 325^2}{6 \times 200000 \times 5051250}(3 \times 2500 - 325)$$
$$= 13{,}908 - 0{,}337$$
$$= 13{,}571 \text{ mm}^2$$

13,571 mm < 16.67 mm

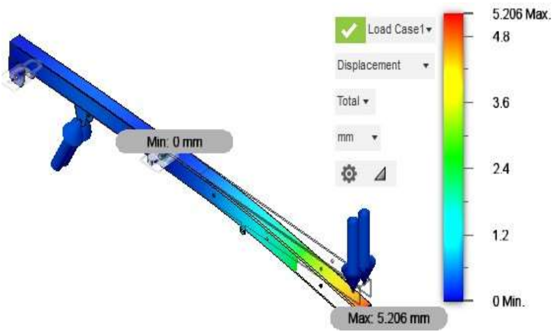
FEA Simulation

a. Boom

From the simulation results on the boom, the maximum von Mises stress is 123.9 MPa, and the maximum total displacement is 5,206 mm.



(a)



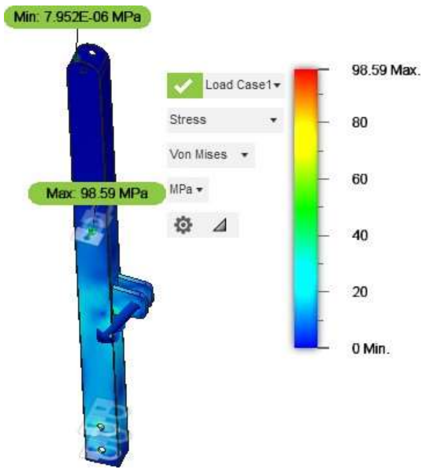
(b)

Figure 12. Simulation Ressult, a) Von misses stress, b) Displacement

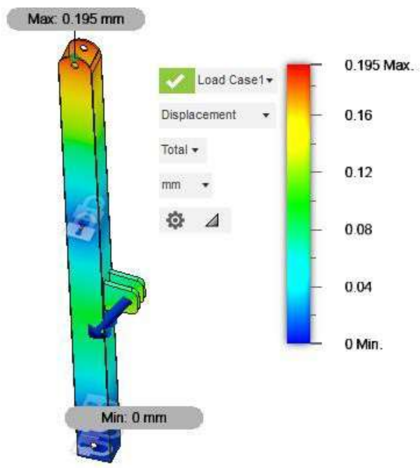
From these values, it can be concluded that the material is considered strong and safe because the simulated max stress is smaller than the allowable stress (123.9 N/mm2 < 125 N/mm2) and the displacement value is smaller than the allowable deflection value (5.206 mm < 16.667 mm).

b. Upright Support

From the simulation results on the upright support, the maximum von Mises stress is 98.59 MPa, and the maximum total displacement is 0.195 mm.



(a)



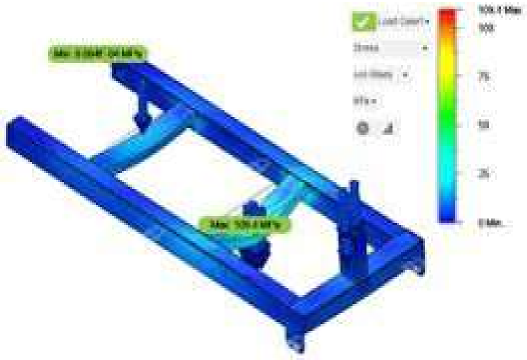
(b)

Figure 13. Simulation Ressult, a) Von misses stress, b) Displacement

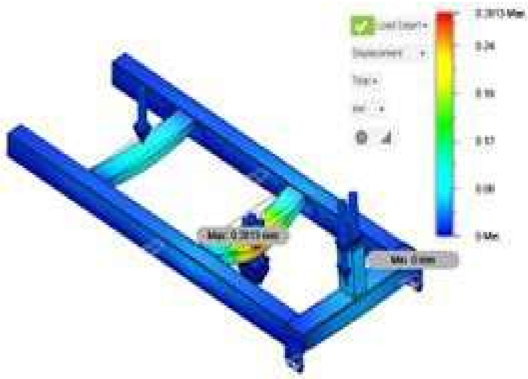
From these values it is concluded that the material is considered strong and safe because simulated max. stress is smaller than the allowable stress (98.59 N/mm2 < 125 N/mm2).

c. Base

From the simulation results on the base, the maximum von Mises stress is 109.4 MPa, and the maximum total displacement is 0.2813 mm.



(a)



(b)

Figure 14. Simulation Ressult, a) Von misses stress, b) Displacement

From these values it can be concluded that the material is considered strong and safe because the simulated max. stress is smaller than the allowable stress (109.4 N/mm2 < 125 N/mm2).

Component Selection

a. Hook

Based on the planning, the maximum lifting load is 275 kg. Hook selection is by choosing a hook that has specifications where the lifting load is greater than the specified lifting load. The hook chosen is a swivel hook from the Green Pin product catalog, namely Sling Hook S/S-GR5 with specifications, including:

Material	=	AISI 316l grade 5, *polished*
Max. lifting load	=	0,7 Ton = 700 Kg

b. Hydraulic

Based on the maximum hydraulic load capacity and stroke length, hydraulics with specifications that meet the FB lifting load are selected =12346.715 N = 1258.584 Kg. The hydraulics chosen were hydraulics with long ram jacks type D-51010-C from the US Jack catalogue. Selected hydraulic specification data:

Capasity	=	3 Ton = 3000 Kg
Stroke	=	24 1/8" – 41 1/8"
	=	572,775 – 1044,575 mm
Ram diameter	=	1 3/16" = 30,1625 mm
Base dimension	=	4 5/16" – 5 7/8"
	=	109,5375 – 149,225 mm
Weight	=	25 lbs = 11,34 Kg

c. Casters

The load received by each front wheel is:

$$R_F = 7393{,}765 \text{ N}$$

$$P = \frac{7393{,}765}{2}$$

$$= 3696{,}882 \text{ N}$$

$$= 376{,}848 \text{ Kg}$$

The load received by each rear wheel is:

$$R_I = 6781{,}494 \text{ N}$$

$$P = \frac{6781{,}494}{2}$$

$$= 3390{,}747 \text{ N}$$

$$= 345{,}641 \text{ Kg}$$

After obtaining the load received by each front and rear wheels, the casters are selected from the TENTE catalog. We get rigid caster wheels with a wheel height of 240 mm and a load capacity of 400 Kg for the front wheels, while the rear wheels are swivel caster wheels with a wheel height of 240 mm and a load capacity of 400 Kg.

CONCLUSION

The results of the design of the floor crane structure as a tool to replace the battery pack were carried out using 3D modeling which was adapted to the conditions of the TMII station using Autodesk Fusion 360 software. Analysis of the overall structural strength of the floor crane showed that the condition was safe because the actual minimum safety factor value exceeded the specified safety factor value, namely 2, the maximum von Mises stress obtained is smaller than the allowable stress, namely 125 N/mm2, and the maximum displacement on the arm shows a safe condition because it is less than the allowable deflection, namely 16.667 mm.

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