

Optimization of Fork Tines Design on a 3-Ton Capacity Forklift Using Heel Radius and Thickness Geometry Variation via Finite Element Analysis

Ersyam Nur Wahid¹, Muhammad Prasha Risfi Silitonga^{1,*}

¹ Department of Mechanical Engineering, Politeknik Negeri Jakarta, Depok, 16242, Indonesia

* Correspondence: m.silitonga@mesin.pnj.ac.id

Abstract: Structural failure at the heel area of forklift fork tines due to high stress concentration is a serious safety concern in the logistics industry. This study aims to optimize the geometry of ISO Class 3 fork tines with a 3-ton capacity to achieve a design with maximum strength and efficient mass. The design methodology utilizes a 3² factorial design via the Taguchi approach (L9 orthogonal array) by varying the Heel Radius (65, 75, 85 mm) and Base Thickness (40, 50, 60 mm). Testing was performed numerically through static Finite Element Analysis (FEA) on AISI 4140 Alloy Steel, assuming a uniform load of 15,000 N per fork arm at a Load Center distance of 500 mm. Initial validation of the numerical model against analytical cantilever-beam calculations resulted in an error below 1%, confirming simulation accuracy. ANOVA results prove that base thickness is the primary and most dominant parameter significantly affecting stress (P-value 0.003 < 0.05). Through comparative analysis, Design 6 (D6), combining an 85 mm Heel Radius and 50 mm Thickness, was selected as the most optimal configuration, reducing maximum stress to 208.5 MPa and increasing the Factor of Safety (FoS) to 1.99, significantly safer than the factory-standard design (FoS 1.84) while maintaining a mass efficiency of 97 kg. The optimized AISI 4140 design is structurally safe for static operation and is recommended as a fabrication standard.

Keywords: Forklift; Fork Tines; Finite Element Analysis; Taguchi Method; AISI 4140.

Citation: Wahid, E. N., Silitonga, M. P. (2026). Optimization of Fork Tines Design on a 3-Ton Capacity Forklift Using Heel Radius and Thickness Geometry Variation via Finite Element Analysis. *Recent in Engineering Science and Technology*, 4(3), 176–184. Retrieved from <https://www.mbi-journals.com/index.php/riestech/article/view/179>.

Academic Editor: Noor Hidayati

Received: 12 May 2026

Accepted: 15 July 2026

Published: 31 July 2026

Publisher's Note: MBI stays neutral with regard to jurisdictional claims in published maps and institutional affiliations.



Copyright: © 2026 by the authors. Licensee MBI, Jakarta, Indonesia. This article is an open access article distributed under MBI license (<https://mbi-journals.com/licenses/by/4.0/>).

1. Introduction

Material handling efficiency is a critical requirement of modern logistics and warehousing systems, and the forklift remains the primary vehicle used to transport heavy loads that cannot be handled manually. The operational effectiveness of a forklift is strongly dependent on the reliability of its critical components, particularly the lifting system and the fork tines, which interact directly with the load and must withstand extreme, repetitive bending moments during service.

Structural failure of forklift fork tines is a serious safety issue in industrial operations. Field failure data indicate that damage most frequently occurs at the heel, the right-angle transition zone between the vertical shank and the horizontal blade. This failure mode is generally caused by fatigue resulting from high stress concentration at the heel radius; when the geometry of this critical area is not optimized, the risk of sudden fracture during operation increases sharply, endangering both equipment and operator safety [1].

Mechanically, the L-shaped geometry of the fork creates a substantial stress-concentration zone at the inner corner radius [2]. Consequently, the heel radius and the base thickness of the fork are the two dominant geometric parameters governing how bending loads are distributed and whether the resulting stresses remain below the

material's elastic limit. Evaluating and reducing the risk of structural failure therefore requires a design approach built on computer simulation. Finite Element Analysis (FEA) allows a wide range of fork-tine geometries to be tested virtually, enabling identification of the location and magnitude of the maximum von Mises stress and the resulting Factor of Safety in a comprehensive and repeatable manner [3].

Previous studies have examined forklift design and structural performance from several perspectives. Sun et al. investigated the structural design and load-carrying performance of a transportation fork used in a hot-stamping production line [4], while Wu et al. successfully integrated finite element analysis with Taguchi statistics to reduce the mass of a precision machine-tool bed without compromising its stiffness, a methodology directly relevant to fork-tine optimization [5]. More specifically, Mahdiah et al. redesigned the lifting system of a large-capacity (10-ton) forklift using FEM simulation [6], although that work focused on validating a design for an extreme load case rather than exploring the combined effect of heel radius and base thickness at standard capacities.

Despite this body of work, the combined influence of heel radius and base thickness on the Factor of Safety of a 3-ton class fork tine manufactured from AISI 4140, a chromium-molybdenum alloy steel known for its favorable fatigue resistance, has not been thoroughly explored [7], [8], [9]. This research gap motivates the present study, which applies Finite Element Analysis together with a Taguchi 3^2 factorial design to systematically vary the heel radius and base thickness of a standard 3-ton forklift fork tine, with the objective of identifying an optimal configuration that satisfies industrial safety standards while remaining mass-efficient [10], [11].

Under the international classification ISO 2328, forklifts are grouped by lifting capacity, and a unit rated at 3000 kg falls under Class 3, a category widely used in manufacturing, logistics, and warehousing for pallet mobilization [9]. The overall performance of such a unit is governed not only by its ability to lift a rated load but also by its safety margin, commonly expressed as the Factor of Safety (FoS), the ratio between the material's yield strength and the maximum working stress. A component is considered structurally optimal when it can sustain the maximum operating load without plastic deformation or fracture while avoiding unnecessary mass, since excessive dead weight reduces the efficiency of the vehicle's hydraulic lifting system and its overall load-to-weight ratio [8], [14]. Balancing these two competing requirements, strength and efficiency, is therefore the central engineering problem addressed by this research, and it is approached here through a combined FEA-Taguchi optimization framework.

The remainder of this paper is organized as follows. Section 2 describes the fork-tine geometry, material properties, Taguchi experimental design, and the finite element simulation procedure, including the analytical validation method. Section 3 presents the validation results, the FEA outcomes for all nine design variations, the Signal-to-Noise ratio analysis, and the ANOVA-based statistical evaluation of factor significance, culminating in the selection of the optimal design. Section 4 concludes the paper and outlines directions for future work, including experimental fatigue validation and dynamic loading analysis.

2. Materials and Experiment Methods

The object of this study is an ISO Class 3 forklift fork tine with a nominal lifting capacity of 3 tons (3000 kg), which operates on the mechanical principle of a cantilever beam fixed at the carriage and free at the loaded end [9], [12]. The material selected for the fork tine is AISI 4140 alloy steel (Chromium-Molybdenum Steel), a low-alloy steel widely used in heavy-equipment components such as axles, gears, and forklift forks because of its combination of high toughness, good tensile strength, and favorable fatigue resistance [7]. The mechanical properties of AISI 4140 used as simulation input were: Yield Strength 415 MPa, Tensile Strength 655 MPa, Elastic Modulus 210 GPa, and Poisson's Ratio 0.29. A

3^2 factorial design based on the Taguchi L9 orthogonal array was applied to obtain nine design variations while minimizing the number of required simulations. Two control factors were varied: Heel Radius (Factor A) and Base Thickness (Factor B), each at three levels, as summarized in Table 1. Other dimensions, including the total blade length and the 140 mm cross-sectional width, were held constant across all variations.

Table 1. Factors and levels of the 3^2 factorial design for geometry variation.

Factor	Level 1	Level 2	Level 3
Heel Radius (A)	65 mm	75 mm	85 mm
Base Thickness (B)	40 mm	50 mm	60 mm

Nine 3D CAD models (D1–D9), each corresponding to one row of the Taguchi L9 matrix, were constructed in SolidWorks. Each model was analyzed using static structural Finite Element Analysis. A fixed-geometry (fixed support) boundary condition was applied to the vertical shank, representing attachment to the carriage, while a uniform load of 15,000 N was distributed over the horizontal blade at a Load Center distance of 500 mm from the base, in accordance with standard forklift load-rating conventions [10], [13]. Each model was then meshed and solved, as described below, to extract three response variables for every design: maximum von Mises stress, total displacement, and the resulting Factor of Safety (FoS), computed as the ratio of the material's yield strength to the maximum von Mises stress.

The finite element mesh was generated in SolidWorks Simulation using second-order (10-node) tetrahedral solid elements and a curvature-based mesher. A global element size of 10 mm was applied to the fork body, while a local mesh control with an element size of 2 mm was imposed on the heel fillet surfaces, where the peak stress occurs, so that at least 8 elements span the fillet arc. For the baseline design D5 this produced 145,230 elements and 215,480 nodes, and the element count across the nine designs ranged from 138,500 to 152,400. Mesh independence was verified on the baseline design D5 by progressively reducing the local element size at the heel fillet and monitoring the maximum von Mises stress. The mesh was considered converged when the change in maximum stress between two successive refinements was below 2%; the converged local mesh settings were then applied unchanged to all nine designs. As shown in Figure 1, reducing the local element size from 10 mm to 5 mm changed the maximum von Mises stress by 17.6%, whereas further refinement from 3 mm to 2 mm and from 2 mm to 1.5 mm changed it by only 0.45% and 0.13%, respectively. A local element size of 2 mm was therefore adopted, giving a converged maximum stress of 225.5 MPa for D5, the value used in the validation and parametric study.

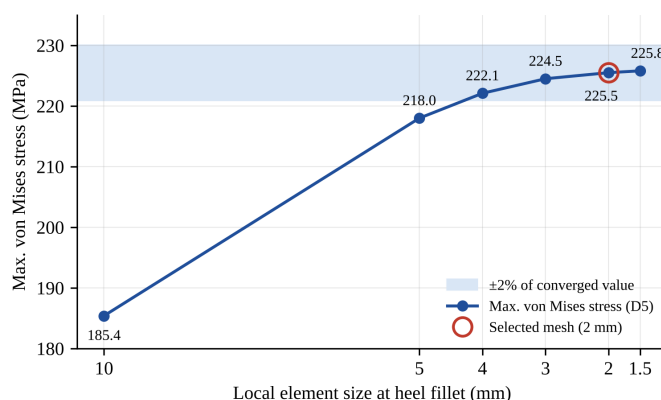


Figure 1. Mesh-convergence study for the baseline design (D5): maximum von Mises stress versus local element size at the heel fillet.

Prior to the parametric study, the numerical model was validated against analytical cantilever-beam calculations. The bending stress was computed from $\sigma = M \cdot y / I$, where M is the bending moment ($F \cdot L$), y is the distance from the neutral axis to the outer fiber, and I is the second moment of area of the rectangular cross-section, then corrected using a stress-concentration factor (K_t) derived from Peterson's charts for a filleted cantilever corner. Validation was accepted when the percentage error between the analytical and simulated stress values remained below the 10% tolerance limit [3], [6].

The nine sets of simulation responses were subsequently processed using the Taguchi Signal-to-Noise (S/N) ratio, with the Smaller-is-Better criterion selected because the optimization objective was to minimize stress and deformation. Analysis of Variance (ANOVA) was then performed at a 95% confidence level ($\alpha = 0.05$) to statistically determine which of the two geometric factors, heel radius or base thickness, exerts the more significant influence on the stress and deformation responses, following an approach analogous to Taguchi-FEA optimization frameworks applied in other precision-manufacturing contexts [5], [14]. Because each FEA run is deterministic and the L9 (3^2) design contains a single run per factor-level combination (no replication), only the two main effects were included in the ANOVA model. The Heel Radius $\times$ Thickness interaction, which has $(3 - 1)(3 - 1) = 4$ degrees of freedom, cannot be estimated separately in an unreplicated 3×3 design, since fitting it would leave no degrees of freedom for error. Consequently, the "Error" term (DF = 4) reported in Tables 4 and 5 represents the pooled interaction and higher-order effects rather than random measurement error, and the resulting F-tests for the main effects are conservative, because any genuine interaction inflates the error mean square.

The bending stress produced by the cantilever loading condition was computed analytically as $\sigma = M \cdot y / I$, where the bending moment M equals the applied force F multiplied by the load-center distance L , y is half of the base thickness (distance from the neutral axis to the outer fiber), and I is the second moment of area of the rectangular cross-section, $I = (1/12) \cdot b \cdot h^3$, with b the 140 mm section width and h the local thickness. The nominal stress was then multiplied by an empirical stress-concentration factor K_t , obtained from Peterson's charts for a filleted right-angle cantilever, to yield the theoretical maximum stress used for validation.

The Factor of Safety for each design was computed as $FoS = \sigma_{yield} / \sigma_{von-Mises}$, where σ_{yield} is the 415 MPa yield strength of AISI 4140 and $\sigma_{von-Mises}$ is the maximum von Mises stress obtained from the FEA solution for that design. Model accuracy was quantified using the percentage error, $\%Error = |(\sigma_{theoretical} - \sigma_{FEA}) / \sigma_{theoretical}| \times 100\%$, and a design was accepted as valid when this error remained below the 10% tolerance threshold commonly applied in FEA validation studies [6]. Finally, the Taguchi Smaller-is-Better S/N ratio for each factor level was computed as $S/N = -10 \cdot \log_{10}[(1/n) \cdot \sum(1/y^2)]$, where y represents the response value (stress or displacement) and n is the number of observations at that level; higher S/N values indicate a more favorable (lower-stress or lower-deflection) combination of factor levels.

3. Results and Discussion

Validation was performed by comparing the analytical bending stress of the baseline design (D5: 75 mm heel radius, 50 mm thickness) with the corresponding FEA result. The analytical nominal stress was calculated as 128.57 MPa, and after applying an estimated stress-concentration factor of $K_t \approx 1.75$, the theoretical maximum stress was 225.0 MPa, compared with an FEA result of 225.5 MPa, an error of only 0.22%.

The same procedure applied to all nine design variations produced errors consistently below 1%, well within the 10% tolerance limit, confirming that the mesh density and boundary conditions used in the simulation accurately represent the underlying mechanics of the fork tine.

Table 2. Recapitulation of FEA results for the nine fork-tine design variations.

Design	Heel Radius (mm)	Thickness (mm)	Max Stress (MPa)	Displacement (mm)	FoS	Mass (kg)
D1	65	40	261.4	24.61	1.59	76
D2	75	40	307	29.57	1.35	77
D3	85	40	315.5	31.42	1.32	78
D4	65	50	224.5	15.09	1.85	95
D5 (Standard)	75	50	225.5	15.82	1.84	96
D6 (Optimal)	85	50	208.5	16.54	1.99	97
D7	65	60	164.8	9.11	2.52	114
D8	75	60	169.9	9.53	2.44	115
D9	85	60	159	9.96	2.61	116

Two clear trends emerge from Table 2. First, increasing base thickness consistently and substantially reduces both stress and displacement: the 40 mm group (D1–D3) reached stresses as high as 315.5 MPa and deflections up to 31.42 mm, while the 60 mm group (D7–D9) remained below 170 MPa with deflections under 10 mm. Second, the effect of heel radius depends strongly on thickness. In the thin-plate group (40 mm), enlarging the radius from 65 to 85 mm unexpectedly increased stress from 261.4 to 315.5 MPa, a throat-thinning effect in which a larger inner radius, without a corresponding increase in outer wall thickness, reduces the net cross-sectional area at the heel. In the thicker groups (50 mm and 60 mm), the same increase in radius instead reduced stress by smoothing the stress-flow path around the corner, indicating that radius and thickness interact rather than act independently.

With respect to mass, Table 2 shows that the fork mass increases by approximately 19 kg for each 10 mm increase in base thickness and by only about 1 kg for each 10 mm increase in heel radius at constant thickness. This pattern is consistent across all three thickness groups and reflects the fact that thickness changes the cross-section along the entire blade and shank, whereas the heel radius only modifies the material volume locally at the corner.

The Taguchi S/N ratio analysis (Smaller-is-Better) shows that thickness has a far greater effect on the response than heel radius. For the stress response, the Delta value for thickness (130.1) is roughly 7.5 times larger than that for heel radius (17.2); a similar pattern is found for the displacement response, where thickness produced a Delta of 19.0 compared with only 3.0 for heel radius. Both response tables rank thickness first and heel radius second, indicating that thickness is the dominant control factor for both structural strength and stiffness. Table 3 presents the mean stress response for each factor level, from which the Delta and rank values were derived.

Table 3. Response table for mean stress by factor level (Taguchi analysis).

Level	Heel Radius (mean stress, MPa)	Thickness (mean stress, MPa)
1	216.9	294.6
2	234.1	219.5
3	227.7	164.6
Delta (Max–Min)	17.2	130.1
Rank	2	1

Table 4. ANOVA results for the maximum von Mises stress response

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Heel Radius	2	454.7	227.4	0.61	0.585
Thickness	2	25580	12790	34.58	0.003
Error (pooled A×B interaction)	4	1479.6	369.9	-	-
Total	8	27514.3	-	-	-

Table 5. ANOVA results for the total displacement response.

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Heel Radius	2	14.37	7.185	2.43	0.204
Thickness	2	562.2	281.097	94.94	< 0.001
Error (pooled A×B interaction)	4	11.84	2.961	-	-
Total	8	588.4	-	-	-

At a 95% confidence level, the ANOVA in Table 4 confirms that thickness is a statistically significant factor for the stress response ($P = 0.003 < 0.05$), whereas heel radius is not significant on its own ($P = 0.585 > 0.05$). H_0 is therefore rejected for thickness and accepted for heel radius with respect to the stress response. It should be noted that, because the Error term in Tables 4 and 5 pools the Heel Radius × Thickness interaction (see Section 2), the interaction between the two factors discussed above, most evident in the opposite stress trend of the 40 mm-thick designs, is identified from the response pattern in Table 2 rather than from a separate significance test.

An analogous ANOVA performed on the total displacement response, shown in Table 5, yielded $P < 0.001$ for thickness and $P = 0.204$ for heel radius, again showing that thickness is the sole statistically dominant factor governing both stress and stiffness, while heel radius does not produce a statistically detectable change in deflection on its own. This finding is consistent with elementary beam-bending theory, since the second moment of area of a rectangular cross-section scales with the cube of thickness ($I \propto h^3$), giving thickness a much stronger leverage on both bending stress and deflection than the fillet radius alone. Similar dominance of a single control factor identified through combined Taguchi–ANOVA analysis has also been reported in other precision-manufacturing optimization studies, for example in the selection of spindle speed, feed rate, and depth of cut for CNC machining of metallic components, reinforcing the robustness of this statistical approach across different manufacturing domains [5], [16].

From a design standpoint, these statistical results imply that engineers seeking to improve the safety margin of a fork tine should prioritize increasing base thickness over enlarging the heel radius, since thickness delivers a far greater and more predictable reduction in stress per unit of design change. Heel radius should nonetheless not be disregarded: although not statistically significant in isolation, its interaction with thickness, evident in the throat-thinning behavior observed for the 40 mm-thick designs, means that radius selection must always be evaluated jointly with the local wall thickness rather than as an independent design choice.

Design 9 (D9) achieved the lowest stress (159.0 MPa) and the highest Factor of Safety (2.61) among all variations, but this came at the cost of the heaviest fork mass (116 kg), an overdesign that penalizes hydraulic lifting efficiency and material cost without a proportional safety benefit. Conversely, the 40 mm-thick designs (D1–D3) failed to reach the industry-recommended Factor of Safety of 2.0. Balancing structural safety against mass efficiency, Design 6 (D6), combining an 85 mm heel radius with a 50 mm thickness, was selected as the optimal configuration. D6 limits the maximum stress to 208.5 MPa and raises the Factor of Safety to 1.99, close to the 2.0 industry benchmark and clearly safer than the factory-standard design D5 (FoS 1.84), while adding only 1 kg of mass relative to D5 (97 kg versus 96 kg). This result demonstrates that a modest, well-placed increase in heel radius, when combined with an adequately thick base section, can deliver a substantial safety improvement without a significant weight penalty.

These results are consistent with the broader literature on structural optimization of L-shaped and cantilevered steel components. Bozkurt et al. showed that localized geometry refinement at high-stress structural parts of a diesel forklift truck can meaningfully lower peak stress without a wholesale redesign of the component [1], while Li et al. demonstrated that the fillet geometry of cold-formed steel L-shaped built-up columns strongly governs their impact and load-bearing performance, echoing the sensitivity of the heel region observed in the present study [12]. The throat-thinning effect identified for the 40 mm-thick designs also aligns with fatigue and stress-concentration principles reported for threaded and filleted components, where increasing a local radius without sufficient surrounding material can paradoxically raise, rather than lower, the peak stress [2]. Compared with the factory-standard design D5, the proposed D6 configuration offers a more favorable balance between the stress-reduction benefits of a larger heel radius and the stiffness benefits of adequate wall thickness, and it does so using the same AISI 4140 material and the same overall envelope dimensions, meaning that D6 could be adopted as a drop-in replacement without requiring new tooling, raw-material specifications, or assembly procedures.

From a manufacturing standpoint, it is worth noting that the same Taguchi-based optimization logic used here, minimizing a critical response through a small, statistically efficient set of experimental trials, has also proven effective outside structural design, for instance in optimizing spindle speed, feed rate, and depth of cut to reduce machining time in CNC milling of metallic bipolar plates for fuel-cell applications [16]. This cross-domain

consistency supports the use of the L9 orthogonal array as a general-purpose, resource-efficient tool for engineering optimization problems in which two or three control factors must be balanced against a critical performance response, whether that response is structural stress, as in this study, or manufacturing cycle time, as in CNC machining process optimization.

4. Conclusions

This study applied a Taguchi 3^2 factorial design combined with static Finite Element Analysis to optimize the heel radius and base thickness of a 3-ton capacity forklift fork tine made of AISI 4140 alloy steel. The numerical model was validated against analytical cantilever-beam calculations with an error below 1% and a mesh-independence study performed on the baseline design, confirming the reliability of the meshing and boundary-condition setup used for the parametric study. Heel radius alone did not significantly affect stress ($P = 0.585$) or displacement ($P = 0.204$) in the ANOVA, and on thin cross-sections a larger radius could even raise stress through a throat-thinning effect caused by the reduction of the net cross-sectional area at the heel. Base thickness, in contrast, was confirmed as the dominant, statistically significant parameter ($P = 0.003$ for stress and $P < 0.001$ for displacement) governing both the strength and stiffness of the fork tine, a result consistent with the cubic dependence of the section's second moment of area on thickness.

Among the nine design variations tested, Design 6 (85 mm heel radius, 50 mm thickness) was identified as the optimal configuration, reducing the maximum stress to 208.5 MPa and raising the Factor of Safety to 1.99, an improvement over the factory-standard design (FoS 1.84) achieved with a mass increase of only about 1 kg. These findings confirm that AISI 4140 fork tines with the D6 geometry are structurally safe for static operation and can be recommended as a fabrication standard for 3-ton class forklifts. For future work, it is recommended that (1) the D6 design be validated experimentally through physical fatigue and tensile testing of a fabricated prototype; (2) the analysis be extended to dynamic loading scenarios, such as sudden braking or travel over uneven surfaces, to better represent real operating conditions [15]; and (3) alternative alloy systems be evaluated against AISI 4140 to explore whether a lower-density material can achieve an equivalent or higher yield strength, further improving the mass efficiency of the optimized fork tine.

Acknowledgments: The authors thank the Department of Mechanical Engineering, Jakarta State Polytechnic, and the double-degree program partner, St. John's University, Taiwan, for administrative and technical support during this research.

References

1. Ö. Y. Bozkurt, İ. C. Dai, and Ö. Özbek, "The Finite Element Analysis and Geometry Improvements of Some Structural Parts of a Diesel Forklift Truck," *Periodicals of Engineering and Natural Sciences*, vol. 5, no. 2, pp. 202–209, 2017, doi: 10.21533/pen.v5i2.118.
2. J. R. M. dos Santos, M. F. Fernandes, V. M. O. Velloso, and H. J. C. Voorwald, "Fatigue Analysis of Threaded Components with Cd and Zn-Ni Anticorrosive Coatings," *Metals*, vol. 11, no. 9, p. 1455, 2021, doi: 10.3390/met11091455.
3. J. Wu, "Finite element analysis and simulation of cargo fork dynamics," *Journal of Physics: Conference Series*, vol. 2419, no. 1, p. 012072, 2023, doi: 10.1088/1742-6596/2419/1/012072.
4. Y. Sun, L. Wang, B. Zhu, Y. Wang, and Y. Zhang, "The structure design and performance analysis of the transportation fork in hot stamping production line," pp. 1953–1956, 2015.
5. H. Wu, H. Li, X. Wang, J. Yang, R. Jiang, X. Deng, and J. Wang, "Research on multi-objective optimization design for high-precision turning-milling machine tool bed based on Taguchi method-FEA," *Scientific Reports*, vol. 15, p. 27307, 2025, doi: 10.1038/s41598-025-13263-1.
6. M. S. Mahdieh, F. Nazari, and F. A. Ghlaime, "Design of forks and lifting system of a 10-ton forklift truck by FEM simulation," *Journal of Simulation and Analysis of Novel Technologies in Mechanical Engineering*, vol. 16, no. 2, pp. 59–67, 2024, doi: 10.71939/jsme.2024.1092071.

7. R. Kurniawan and A. P. Budijono, "Analisis Gaya dan Mekanisme Angkut Forklift Toyota 8FBMT50 Berdaya Angkat 5 Ton dengan Sistem Hidrolik," *Jurnal Teknik Mesin (JTM)*, vol. 6, no. 1, pp. 51–59, 2018.
8. A. Ishak, K. Siregar, G. Valkov, and V. Nikolov, "Failure Mode and Effect Analysis (Fuzzy FMEA) Implementation for Forklift Risk Management in Manufacturing Company PT. XYZ," *IOP Conference Series: Materials Science and Engineering*, vol. 528, no. 1, p. 012027, 2019, doi: 10.1088/1757-899X/528/1/012027.
9. K. N. Khebude, "Design and Analysis of Mechanical Forklift," *International Journal of Scientific Research and Engineering Development*, vol. 3, no. 2, pp. 685–690, 2020.
10. M. Kumaran, N. Muthuram, and S. Hari Chealvan, "Redefining forklift longevity: a study on fork design optimization and lifecycle enhancement," *International Journal on Interactive Design and Manufacturing (IJIDeM)*, vol. 19, no. 9, pp. 6171–6187, 2025, doi: 10.1007/s12008-024-02195-1.
11. A. Jalal, M. Farooq, I. Anwer, N. Hayat, A. Munir, I. Zahid, N. S. Akbar, M. Hamza, M. Nouman, and F. Riaz, "Design and development of a low-cost, eco-friendly forklift for sustainable logistics management," *Frontiers in Built Environment*, vol. 10, p. 1506495, 2025, doi: 10.3389/fbuil.2024.1506495.
12. M. Li, J. Sun, Y. Hu, L. Jiang, S. Zhou, G. Dai, and N. Wu, "Lateral Impact Performance of Cold-Formed Steel L-Shaped Built-Up Columns," *Materials*, vol. 18, no. 19, p. 4548, 2025, doi: 10.3390/ma18194548.
13. S. N. Pokharkar, A. Dumbre, S. Bhagat, R. Pawar, and P. Borude, "Design and Analysis of New Flexible and Safe Forklifts," *International Journal for Science and Advance Research in Technology (IJSART)*, vol. 8, no. 5, pp. 404–410, 2022.
14. S. Fu, H. Chen, H. Ren, T. Lin, C. Miao, and Q. Chen, "Potential Energy Recovery System for Electric Heavy Forklift Based on Double Hydraulic Motor-Generators," *Applied Sciences*, vol. 10, no. 11, p. 3996, 2020, doi: 10.3390/app10113996.
15. Y. Y. Huang and L. Horvath, "Measurement and Analysis of Industrial Forklifts Vibration Levels for Unit Load Testing Purposes," *Applied Sciences*, vol. 11, no. 7, p. 2901, 2021, doi: 10.3390/app11072901.
16. F. Mulyana, R. Maladzi, B. Matsaany, N. Yudisha, M. P. Rifi, F. I. Rahman, B. Ramadhan, and Z. M. C. Alam, "Optimization of Bipolar Plate Machining Process Using the Taguchi Method on CNC 3-Axis," *Recent in Engineering Science and Technology*, vol. 4, no. 01, pp. 44–50, 2026, doi: 10.59511/riestech.v4i01.140.