INNOVATIVE MECHANICAL ENGINEERING **ISSN 2812-9229 (Online)**
University of Niš, Faculty of Mechanical Engineering
VOL. 4, NO 2, 2025, PP 32-43

Original scientific paper *

MASS MINIMIZATION-BASED OPTIMIZATION OF LOADER MANIPULATOR DRIVE MECHANISMS

Jovan Pavlović[1], Dragoslav Janošević[1], Predrag Milić[1], Vesna Jovanović[1]

[1]Faculty of Mechanical Engineering University of Niš, Serbia

Abstract. *This paper presents a structural optimization approach for the manipulator drive mechanisms of a wheel loader, with the primary objective function of minimizing the mass of kinematic chain members while ensuring allowable strength and stiffness. The study focuses on evaluating the influence of manipulator mechanism parameters on structural characteristics, such as cross-sectional dimensions and wall thicknesses, under static loading conditions derived from a typical operating cycle. Finite element analysis (FEA) is used to evaluate stress distribution and deformation, while optimization is performed by allowable stress constraints. Based on the analysis results, a minimum mass criterion was defined as part of a multi-criteria optimal synthesis procedure for the manipulator drive mechanisms. The analysis shows that reducing the mass of the manipulator not only enhances structural efficiency but also contributes to improved fuel economy, lower energy consumption, and enhanced overall operational performance of the loader.*

Key words: *Wheel loader, Drive mechanisms optimization, Mass reduction*

1. INTRODUCTION

In loaders and other mobile machines, the total mass and its distribution across the members of the kinematic chain play a crucial role in ensuring stable and reliable performance during manipulation tasks.

In loader design, particular attention is given to the position of the manipulator relative to the machine's potential rollover lines, with the goal of minimizing the mass of both the kinematic chain components and the manipulator's drive mechanisms. This optimization is constrained by various factors, including the machine's stability requirements, allowable load capacity, reliability and material availability.

There are several reasons why minimizing the weight of the loader manipulator is an important design objective. First, a lower manipulator mass reduces the influence of the gravitational moments of the kinematic chain members and mechanisms on the overall

*Received: June 05, 2025 / Accepted July 02, 2025.**
Corresponding author: Jovan Pavlović
University of Niš, Faculty of Mechanical Engineering, Serbia
E-mail: jovan.pavlovic@masfak.ni.ac.rs

overturning moment of the machine. This reduction allows for an increase in the effective payload capacity of the load handled by the manipulator tool. In addition, during load transfer and lifting operations – from the transport position to the unloading position – a lower manipulator mass decreases the influence of static and dynamic force moments on the total loading of the manipulator mechanisms. As a result, less energy is required from the drive system to move the manipulator. Furthermore, reducing the manipulator's mass leads to material savings in the production of the kinematic chain elements and mechanisms.

To achieve such reductions effectively, it is essential to first determine the external loads acting on the manipulator's working mechanism, as this forms the basis for structural stress analysis and optimization. These loads serve as boundary conditions for evaluating stress distribution within the structure. Models for determining loading resistance and the optimal bucket trajectory have been presented in papers [1–4], providing a foundation for more accurate prediction of operational loads and enhancing the efficiency of structural design and motion planning. Accurate load estimation ensures realistic simulation of operational scenarios, which is essential for identifying critical stress regions and guiding effective optimization.

Recent research has focused on material reduction, weight minimization, and structural optimization in mechanical design, aiming to improve performance while reducing costs and environmental impact. These investigations often employ advanced computational methods, such as topology optimization and finite element analysis, to develop lightweight and efficient components without compromising strength or safety. In [5], Zou et al. proposed an optimization methodology for the parametric design of a hydraulic excavator's manipulator mechanism, targeting weight reduction and stress minimization. A novel 3D force model, termed the limiting theoretical digging capability model, was introduced to accurately evaluate the maximum digging forces and moments. Based on this, joint forces and critical digging conditions were determined using a stress-based evaluation. In [6] the authors investigated the topology optimization of a boat crane with the objective of reducing weight while maintaining structural integrity. Using Autodesk Fusion 360, the initial geometry was created and optimized through iterative design and simulation. Static analyses, based on a one-ton load capacity, confirmed that the optimized design met strength and safety requirements, achieving a 42% reduction in mass while maintaining a safety factor of 3.7.

A finite element model of tower crane arm was developed in [7] using Ansys Apdl to analyze stress, strain, strength, and stiffness under static loading. An optimization model was formulated with cross-sectional dimensions as variables and stress/strain under extreme conditions as constraints. Using fuzzy and genetic algorithms, the structure was optimized for minimal mass. The results showed a 309 kg weight reduction without compromising structural strength, meeting design and safety specifications.

The authors in [8] focused on multiobjective optimization of loader rims made from three different materials, aiming to balance safety, stiffness, and weight reduction. Using response surface methodology and genetic algorithms, various designs were evaluated through static, fatigue, and weight analyses. The optimal design was selected considering both performance and production costs. Field tests validated the reliability and safety of the optimized rims under real working conditions. In [9] the authors aimed to optimize the design of a loader arm by evaluating structural parameters such as dump height, digging depth, joint location, and interference through multiple design iterations. Finite Element

Analysis (FEA) using ANSYS was conducted to analyze load behavior, cylinder and ram pressure mapping, and machine stability with a standard bucket. A consistent factor of safety of 2 was maintained. The FEA results were validated through experimental testing using strain gauges, measuring stress and strain under real operating conditions, and comparing these with theoretical and simulated outcomes.

Previous research [10-14] on the optimization of loader manipulators has primarily focused on kinematic parameters, with the following objective functions: minimum change in the bucket angle, maximum mechanism ratio of the arm and bucket drive mechanisms, and minimum strokes of the boom and bucket hydraulic cylinders. However, fewer studies have addressed structural optimization, weight reduction, and stress analysis, which offer significant potential for further improvements in durability, energy efficiency, and overall machine performance.

In this paper, the criterion of minimum mass is introduced as the main objective in the optimization of the loader manipulator drive mechanisms. Initially, based on the defined mathematical model, the influence of mechanism parameters on the loads and resulting stresses in the components of the kinematic chain of the loader manipulator was analyzed. Then mathematical models for estimating the nominal masses of individual mechanism components were developed. These models formed the basis for defining the objective function where the goal is to minimize the total mass of the manipulator kinematic pair members while satisfying constraints related to stress limits, kinematic requirements, and geometric compatibility.

2. Mathematical Model of Wheel Loader

A mathematical model of the wheel loader was developed to analyze the influence of manipulator drive mechanism parameters on the loads within the kinematic chain members. The model represents the general configuration of the loader's kinematic system, which includes the following components: the rear structural-motion member L_1 (Fig. 1), the front structural-motion member L_2 and the manipulator featuring Z- kinematics. The manipulator consists of the arm L_3, bucket L_4, double-arm lever L_5, coupling rod L_6, hydraulic cylinders C_3 for arm actuation, and a hydraulic cylinder C_4 for bucket actuation.

The mathematical model of the wheel loader was developed based on the following assumptions: (1) the manipulator's kinematic chain is planar, meaning that the axes of all revolute joints are mutually parallel and the joint centers lie within the same plane; (2) the support surface and all components of the loader's kinematic chain are modeled as rigid bodies; (3) the loader's kinematic chain has an open configuration, with the final member – the bucket – subjected to material loading forces are applied at its center of gravity; (4) the kinematic chain is influenced by gravitational and inertial forces originating from both the kinematic chain members and the components of the drive mechanisms; (5) the coefficient of friction between the revolute joint elements in the manipulator's kinematic pairs is assumed constant, while friction within the hydraulic cylinders is neglected [15].

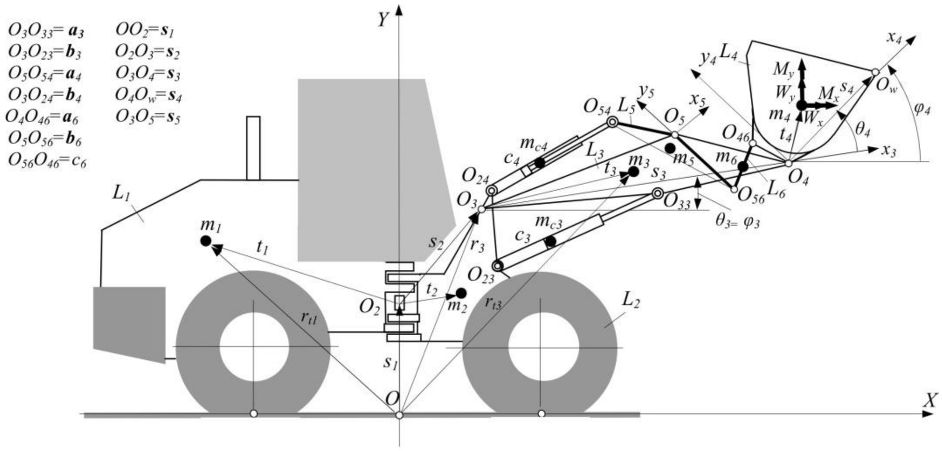

Fig. 1 Mathematical model of the loader drive mechanisms

2.1 The Nominal Mass of a Loader Kinematic Chain Member

The objective function for the minimal mass optimization criterion is defined based on the theory of lightweight structures, using the nominal masses m_{ni} of the kinematic chain members and the manipulator drive mechanisms, as expressed by the general formula:

$$m_{ni} = k_{m1} \cdot k_{m2} \cdot k_{m3} \tag{1}$$

where k_{m1} is the factor representing general conditions, which depend on the support method, loading, and deformation of the member; k_{m2} is the factor related to the profile shape, reflecting the geometric characteristics of the member's cross-section; k_{m3} is the material factor, accounting for the physical and mechanical properties of the member's material.

The expression used to determine nominal mass enables the separation and individual analysis of factors affecting the mass of mechanism components during the synthesis phase, with the goal of identifying strategies for mass reduction.

The nominal mass factors are defined according to the type of loading and the design criteria applied to the member. Relevant design considerations may include: (a) the stress state, strength, and load-bearing capacity; (b) deformation and stiffness requirements; and (c) the deformation energy induced by the applied load.

The parameters of the drive mechanisms in loader manipulators influence the general conditions factor k_{m1} of the nominal mass, since the members of the manipulator kinematic pairs differ in length, support types, loads, and transmission parameters – specifically, the coordinates of the joints in the kinematic pairs (Fig. 2a), as well as in transformation parameters, such as the sizes of the hydraulic cylinders (actuators). However, the parameters of the loader manipulator mechanisms do not affect the shape factor k_{m2} or the material factor k_{m3} of the nominal mass of the mechanism members.

Regarding the form factor of the nominal mass, the members of the loader manipulator's kinematic chain are primarily made from sheet metal, formed by welding, and most commonly have a rectangular cross-section (Fig. 2b). The sheet metal components are mainly made of structural steel, while the elements of the pivot joints in the kinematic pairs are made of non-ferrous metal alloys (joint sleeves) and hardened steel (pins).

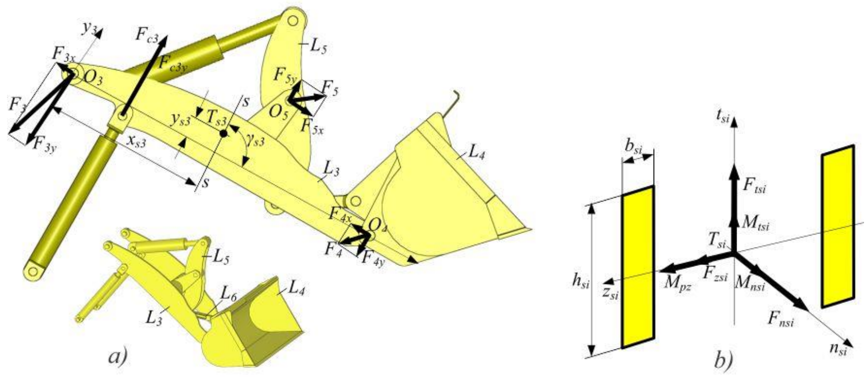

Fig. 2 Loads on manipulator members: a) forces in the joints of manipulator mechanisms, b) loads on the cross-section of the manipulator

In the context of mechanism optimization, nominal mass represents a predefined or design-assumed value of a component's mass, used to facilitate consistent and simplified analysis. It serves as a reference in dynamic modeling, structural calculations, and optimization processes, especially when evaluating mass distribution, load effects, and inertia within the system.

Unlike actual mass, which may vary due to manufacturing tolerances or material inconsistencies, nominal mass ensures uniformity in comparative studies and numerical simulations. Accurate estimation and strategic reduction of nominal mass are often key objectives in mechanism design, particularly when aiming to enhance performance, reduce energy consumption, and minimize stress on actuators and structural elements. This chapter outlines the role of nominal mass in mechanism optimization and presents methods for its determination based on material properties and geometric parameters.

For a member L_i of a mechanism kinematic chain that is predominantly subjected to tension or compression, the design criterion based on allowable stress is expressed by the following inequality [16]:

$$\frac{F_{ins}}{A_{is}} \leq \sigma_{ide} \tag{2}$$

on the basis of which the nominal mass of the member m_{in} is determined:

$$m_{in} = \int_0^{l_{is}} \rho_i \cdot A_{is} \cdot dl_s = \frac{\rho_i}{\sigma_{ide}} \int_0^{l_{is}} F_{ins} \cdot dl_s \tag{3}$$

where: F_{ins} – the normal load force of the section, A_{is} – the cross-sectional area of the member L_i, l_{is} – the length of the neutral line of the member connecting the centers of the section along the kinematic length of the member, σ_{ide} – the allowable tensile stress of the member material, ρ_i – the density of the kinematic chain member material.

For a member L_i of a mechanism subjected to bending, the design criterion based on allowable stress is defined by the following:

$$\frac{M_{izs}}{W_{is}} = \frac{M_{izs}}{f_{is} \cdot A_{is}} \leq \sigma_{idf} \tag{4}$$

where the nominal mass is:

$$m_{in} = \int_0^{l_{is}} \rho_i \cdot A_{is} \cdot dl_s = \frac{\rho_i}{\sigma_{idf}} \int_0^{l_{is}} \frac{M_{ins}}{f_{is}} \cdot dl_{is} \tag{5}$$

where: M_{isz} – the bending moment at the section, $f_{is}=W_{is}/F_{is}$ – the factor representing the ratio of the section modulus to the cross-sectional area of the member, σ_{idf} – the allowable bending stress of the member material.

If the member L_i of the manipulator mechanism is simultaneously subjected to tension and bending, the design criterion based on allowable stress is expressed by the following equation:

$$\frac{F_{ins}}{A_{is}} + \frac{M_{izs}}{W_{is}} = \frac{F_{izs}}{A_{is}} + \frac{M_{izs}}{f_{is} \cdot A_{is}} \leq \sigma_{id} \tag{6}$$

where the nominal mass is:

$$m_{in} = \frac{\rho_i}{\sigma_{id}} \int_0^{l_{is}} (F_{ins} + \frac{M_{izs}}{f_{is}})\, dl_{is} \tag{7}$$

3. Load Analysis of the Different Drive Mechanism Variants

A procedure for generating variant solutions of the manipulator drive mechanism was developed for the purpose of analysis based on the defined mathematical model of the loader, taking into account two types of parameters**: transformational parameters**, which include the coordinates of the joints and the lengths of the levers within the mechanism, and **transmission parameters**, which refer to the diameters of the piston and piston rod in the hydraulic cylinders that actuate the arm and the bucket.

The analysis considered different manipulator drive mechanism configurations: the arm drive mechanism (Table 1) and the bucket drive mechanism (Table 2).

Table 1 Generated variant solutions of the arm drive mechanism.

Variant	D_3 *[mm]*	d_3 *[mm]*	b_{3x} *[mm]*	b_{3y} *[mm]*	a_{3x} *[mm]*	a_{3y} *[mm]*	c_{3p} *[mm]*	c_{3k} *[mm]*
V.001	*125*	*90*	*0*	*-524*	*1655*	*-129*	*1340*	*2058*
V.108	*150*	*100*	*-16*	*-362*	*1360*	*-86*	*1179*	*1618*
V.135	*125*	*90*	*137*	*-506*	*1764*	*-115*	*1340*	*2073*

Table 2 Generated variant solutions of the bucket drive mechanism

Variant	D_4 [mm]	d_4 [mm]	b_{4x}/b_{4y} [mm]	s_{5x}/s_{5y} [mm]	a_{6x}/a_{6y} [mm]	a_{4x}/a_{4y} [mm]	b_{6x}/b_{6y} [mm]	c_6 [mm]	c_{4p} [mm]	c_{4k} [mm]
V.001	*150*	*100*	*206/ -209*	*1703/ 577*	*13/ 369*	*-182/ 676*	*0/ -751*	*756*	*1340*	*1897*
V.108	*180*	*125*	*236/ -83*	*1703/ 577*	*-13/ 299*	*- 75/835*	*0/ -840*	*637*	*1397*	*1727*
V.135	*150*	*100*	*206/ -207*	*1703/ 577*	*12/ 354*	*-196/ 725*	*0/ -751*	*756*	*1393*	*1862*

Generated variant solution mechanisms V.001 and V.108 feature the same transfo-rmational parameters but differ in transmission parameters, whereas mechanisms V.001 and V.036 differ in transformational parameters while exhibiting similar transmission characteristics.

When defining the minimal mass criterion for the manipulator system, the influence of the drive mechanism's parameters on the nominal mass of the bucket is not considered. This is because the bucket is regarded as a standardized, pre-designed module with a fixed volume and mass, independent of the manipulator's configuration, as a result, its properties are treated as constants in the analysis.

As an example, by numerical dynamic simulation, using the MSC Adams program, the force components (Fig. 3) and moment components (Fig. 4) of the load on the s-s section of the arm L_3 were determined, with the coordinates of the center position ($x_{s3}=0.5\ m$, $y_{s3}=0.1\ m$, $z_{s3}=0\ m$) and the angle $\gamma_{s3}=90^o$ of the section plane, for a loader with selected variants of the manipulator mechanisms V.001, V.108 and V.036.

The section *s-s* (Fig. 2a) of the member L_i of the kinematic chain of the manipulator is determined, in the local coordinate system $O_ix_iy_iz_i$ of the member, by the coordinates x_{si}, y_{si}, z_{si} of the center of the section T_{si}, the angle γ_{si} of the section plane and the coordinate system of the section $T_{si}n_{si}t_{si}z_{si}$ (Fig. 2b). When the kinematic chain of the manipulator is virtually cut at section *s–s* of a member, the force components F_{nsi}, F_{tsi}, and F_{zsi}, as well as the moment components M_{nsi}, M_{tsi}, and M_{zsi}, act on the cross-section of the member.

The loading resistance was simulated using the Discrete Element Method (DEM), which provides representation of particle-scale interactions. This numerical technique models granular materials as assemblies of discrete particles, each with defined properties such as mass, shape, stiffness, and friction. Through time-stepped calculations, DEM tracks the motion and interaction of individual particles.

This method allows for a detailed assessment of contact forces and moments that arise during the loading process, including normal and tangential forces at contact points, rolling resistance, and local displacements. As a result, DEM offers realistic insights into the mechanical behavior of granular materials under manipulation.

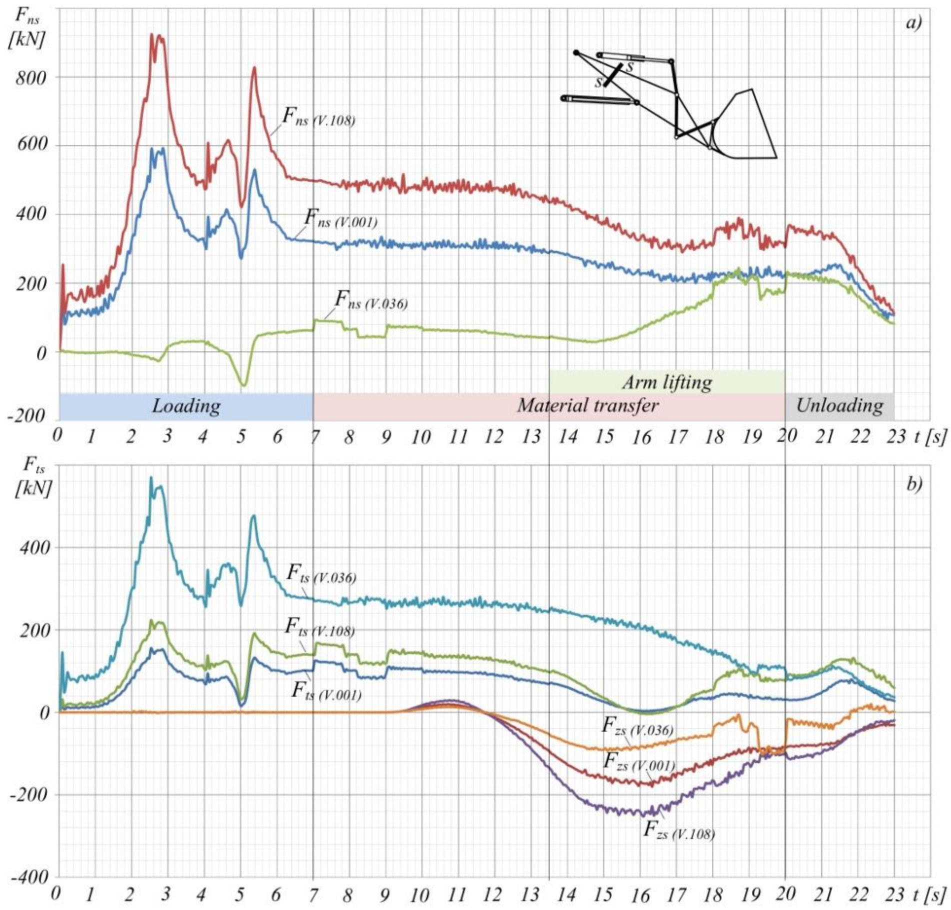

Fig. 3 Load forces of the *s-s* section of the arm: a) normal, b) tangential forces in the directions of the coordinate system $T_{s3}n_{s3}t_{s3}z_{s3}$ of the section for different variants of the manipulator mechanisms

The analysis results (Fig. 3 and Fig. 4) indicate that the loads on the arm section are highest during the loading operation and that they significantly depend on the parameters of the manipulator drive mechanisms. A comparison between variants V.001 and V.036 both having the same transmission parameters (D_3/d_3*=125/90 mm*, D_4/d_4*= 150/100 mm*) but differing in transformational parameters shows that variant V.001 exhibits higher normal forces (F_{ns}), i.e., greater section extension forces, while variant V.036 experiences a significantly higher section bending moment (M_{zs}). In variant V.108, which features larger transmission parameters (D_3/d_3*=150/100 mm*, D_4/d_4*=180/125 mm*) compared to variant V.001, the normal forces (F_{ns}) responsible for section extension are substantially higher, whereas the section bending moment (M_{zs}) remains nearly the same.

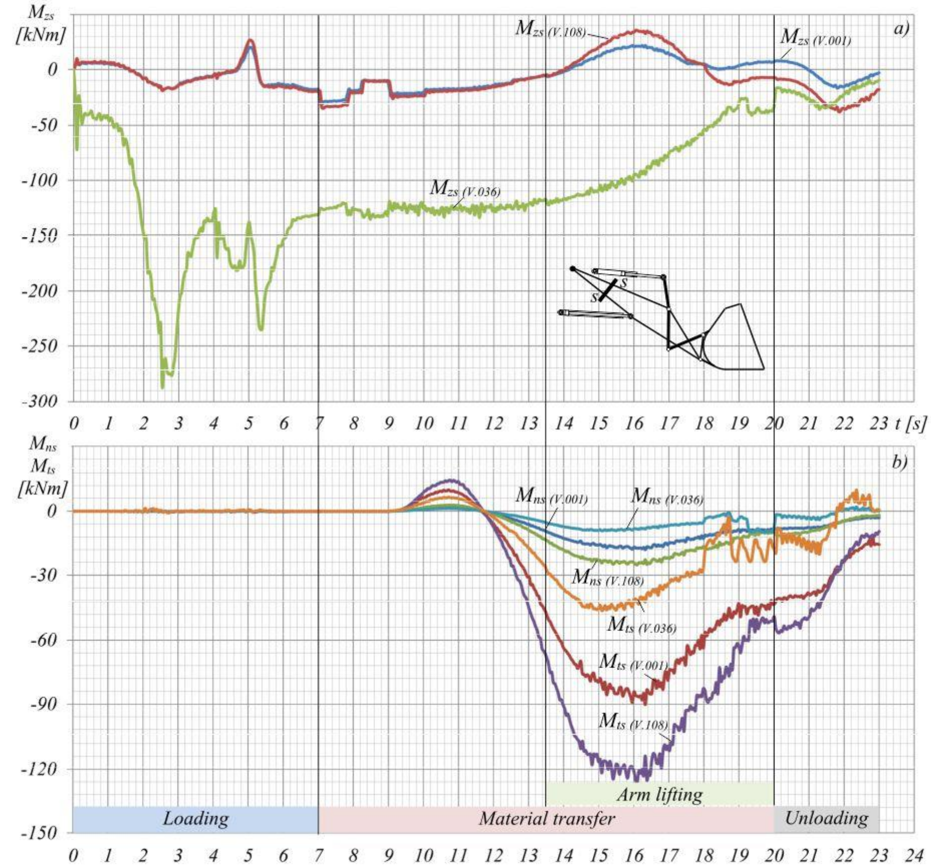

Fig. 4 Load moments of the s-s section of the arm about the axes: a) z_{s3}, b) t_{s3} and n_{s3} of the section coordinate system $T_{s3}n_{s3}t_{s3}z_{s3}$ for different variants of manipulator mechanisms

3.1 FEA Analysis of selected mechanism variants

The influence of the drive mechanism parameters on the stresses in the kinematic chain members of the manipulator was evaluated through structural analysis of the generated manipulator variants V.001, V.108, and V.036, using the Femap Siemens PLM software package. The Von Mises equivalent stresses in the arm L_3 (Fig. 5) for each variant were determined by linear static analysis, employing three-dimensional solid finite elements (752,320 elements and 160,243 nodes).

The model was constrained at joints O_3 and O_{33}, while the loads were applied at joints O_4 and O_5 under identical manipulation task conditions – specifically, the same components of the force vector $\boldsymbol{W}$ and the moment $\boldsymbol{M_w}$ representing the loading resistance of the material obtained from discrete element simulation.

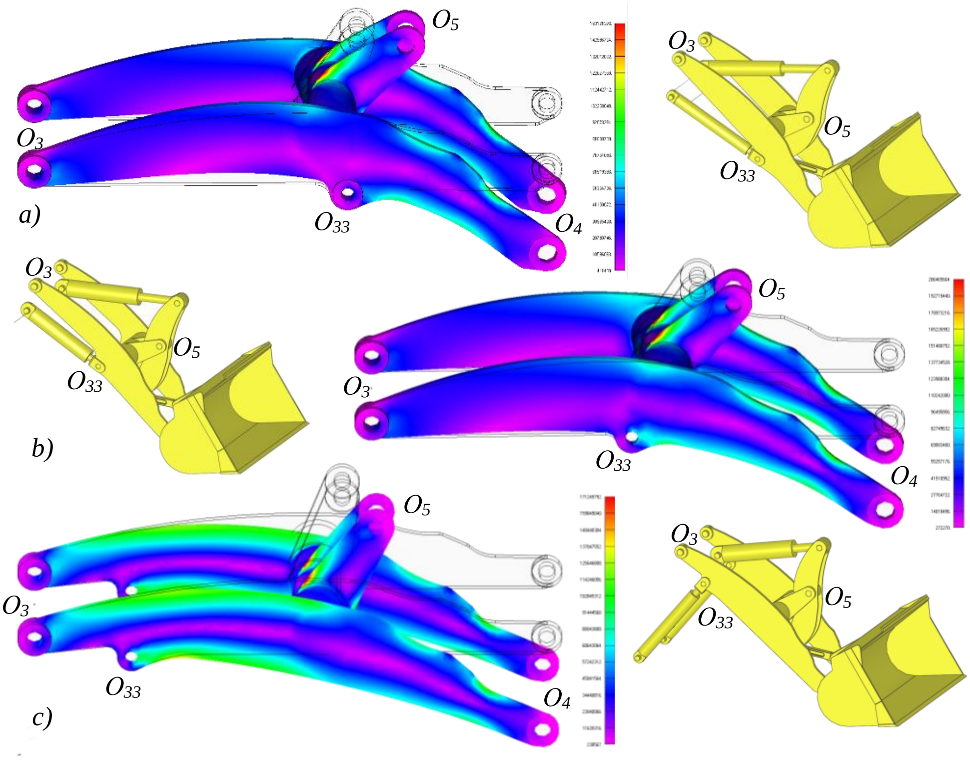

Fig. 5 Arm stresses of generated variant solutions of manipulator mechanisms: a) V.001, b) V.108, c) V.036 for the same manipulator loads that occur during the manipulation task

The structural analysis results (Fig. 5) show that, despite identical manipulator loads, the stresses in the kinematic chain members vary depending on the mechanism parameters. This is evident from the differences in stress distribution in the arm. Variant V.001 (Fig. 5a) exhibits a Von Mises stress of $p_{\max}=153.18\ MPa$, while variant V.036 (Fig. 5c) reaches $p_{\max}=171.25\ MPa$. Both variants share the same transformation parameters ($D_3/d_3=125/90\ mm$, $D_4/d_4=150/100\ mm$), but differ in their transmission parameters. The highest Von Mises stress is observed in variant V.108, with $p_{max}=206.46\ MPa$, which features larger transmission parameters ($D_3/d_3=150/100\ mm$, $D_4/d_4=180/125\ mm$).

4. Conclusion

The analysis results indicate that variations in transformation parameters, such as joint coordinates, linkage lengths, along with changes in transmission parameters, particularly the diameters of the piston and piston rod in the hydraulic cylinders, have an influence on the loading conditions in the selected cross-section of the kinematic chain member.

A comparison between design variants that share identical transmission parameters but differ in transformational parameters reveals distinct differences in internal loading. These findings highlight the sensitivity of internal force distributions to both geometric configuration and actuator sizing, underscoring the importance of parameter selection in structural optimization.

The analysis confirms that it is feasible to introduce a minimum mass criterion within the design and evaluation process. By integrating this criterion, the overall efficiency and structural optimization of the system can be significantly improved. A lighter structure not only contributes to better fuel efficiency and reduced environmental impact but also enhances the dynamic performance and maneuverability of the machine.

Moreover, implementing the minimum mass criterion enables a more effective use of materials, potentially lowering production costs without compromising structural integrity or functional requirements. This approach can serve as a valuable guideline in the early stages of design.

Acknowledgement: *This research was financially supported by the Ministry of Science, Technological Development and Innovation of the Republic of Serbia (Contract No. 451-03-136/2025-03/200109).*

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