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ANALYSIS AND OPTIMIZATION OF EPICYCLIC MECHANISMS WITH MUTUALLY MESHED SATELLITES FOR ENGINEERING AND INDUSTRIAL APPLICATIONS

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This study presents an in-depth kinematic and dynamic analysis of an epicyclic gear mechanism with mutually engaged satellites, aiming to optimize its characteristics and overall efficiency. The primary objective is to analyze its operational performance and propose optimization solutions to enhance its effectiveness. The research focuses on the complex interactions between the satellites, which ensure even load distribution, reduce localized stresses, and increase load capacity. Key parameters of the gears, including the shape of the involute profile, have been analyzed to minimize wear and extend the service life of the mechanism. Mathematical models based on the finite element method (FEM) were utilized to conduct simulations that illustrate the impact of various loads and operating conditions on the mechanism's durability and performance. Additionally, simulations conducted in the CAD environment of SolidWorks allowed for the optimization of the gear design and other system components, achieving significant efficiency under high-load operating conditions. The results indicate that this type of mechanism is particularly suitable for applications in industries such as automotive engineering, aerospace technology, and the energy sector, where reliability, durability, and precision in operational processes are of critical importance.

Key words: epicyclic gear mechanism, torque, satellites, involute profile, CAD systems.

1. Introduction

The kinematic behavior of interconnected satellites differs significantly from traditional planetary gear mechanisms. While standard planetary systems allow simple satellite motion around the carrier axis and rotation about their own axes, mechanisms with mutually engaged satellites introduce complex kinematic dependencies. This interaction, where the gears are directly engaged with each other, ensures even load distribution and higher system load capacity. These characteristics result in improved wear resistance, enhanced efficiency, and optimized operational parameters, making these mechanisms particularly suitable for applications requiring high reliability and durability.

In the context of modern industrial demands, such as globalization, accelerated technological progress, and increasing competition, engineering systems must be designed with an emphasis on efficiency and precision. This necessitates the extensive use of CAD-CAM-CAE systems, which provide powerful tools for three-dimensional modeling, analysis, and optimization of complex structures (He *et al.* [1]; Fuentes *et al.* [2]). These platforms, based on the finite element method (FEM), streamline the design process by reducing development time, minimizing errors, and increasing the efficiency of mechanical systems. Investments in such technologies and qualified professionals are essential for adapting to evolving industrial standards.

The process of three-dimensional modeling enables the visualization of technical objects from various perspectives, which supports the identification and correction of potential design errors before production. Visualization in 2D and 3D formats also enhances communication between engineering and production teams, facilitating a better understanding of the structure and functions of mechanical systems. These advanced technologies enable rapid and efficient design of new products that meet industry requirements.

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This study focuses on the analysis and optimization of various types of epicyclic gear mechanisms, including simple gear mechanisms (SGM), planetary gear mechanisms (PGM), and differential mechanisms (DGM1 and DGM2). The primary goal is to achieve higher output torque and improve the kinematic characteristics of the systems. The analysis results are based on previous studies (Souza *et al.*[3]; Shanmukhasundaram*et al.* [4]; Yang *et al.*, [5]), which explore the kinematic synthesis and structural optimizations of epicyclic mechanisms.

Furthermore, contemporary studies on epicyclic gears highlight the importance of innovations in structural synthesis and classification of mechanical systems. Pennestrí and Valentini [6] propose comprehensive methods for kinematic analysis and combining gear mechanisms, while Freudenstein and Yang [7] present key techniques for understanding the kinematics and dynamics of these systems. These studies provide essential guidance for understanding the kinematic dependencies and dynamic stability of the mechanisms.

Blocking satellite gear mechanisms find widespread applications across various industries. In automotive engineering, they ensure better torque distribution and enhanced resistance under high loads (Kahraman and Kharazi, [8]). In the aerospace sector, they are critical for flight control systems and robotic mechanisms in spacecraft, where precision and reliability are essential. In the energy sector, these mechanisms optimize the performance of wind turbines and other renewable energy systems. In industrial automation and medical technology, these mechanisms provide reliability, precision, and durability, which are crucial for high-tech applications (Yang and Ding, [9]).

In conclusion, the conducted studies on epicyclic mechanisms emphasize their significance for modern engineering applications, offering opportunities for innovation, optimization, and sustainable development of mechanical systems.

2. Parametric model of the gears

Designing the correct shape and geometry of the gear profile plays a key role in improving the efficiency of simulations and studies of the mechanism, as well as in analyzing loads. This applies regardless of the type of selected kinematic scheme being considered. According to studies (Wang *et al.*[10]), parametric modifications of helical gear transmissions can significantly enhance the contact characteristics and durability of mechanisms, which is essential for achieving optimal operational parameters.

The research in (Yang and Ding, [11]) presents a comprehensive set of planetary mechanisms with degrees of freedom that provides a foundation for the synthesis and optimization of gear transmissions. This approach enables structural analysis and refinement of the kinematic and dynamic characteristics of mechanisms.

The gear models were developed in the CAD environment of the SolidWorks system, which offers powerful tools for automated modeling. Despite the availability of libraries with standard gear designs, the software lacks built-in functionality for the automatic generation of involute gear profiles. To address this limitation, a specific methodology was developed for creating involute profiles based on a mathematical curve generated through a system of equations. This methodology ensures high accuracy and enables the creation of gear profiles with optimized shapes (Fig.1).

The designed gears, including the sun gear, satellites, and crown, were generated using this approach. A key advantage of the parametric models used is that when an input parameter is changed, the program automatically updates all dependent geometric characteristics of the gears. For instance, as shown in Tab.1, it is possible to apply a correction to the gear profile. In the current model, a correction was applied to the sun gear z_1 with a parameter $x = -0.417$, optimizing its contact characteristics. Similar parametric modeling approaches have been discussed in (Wu *et al.* [12]).

The development of parametric models not only increases the precision and flexibility of the design process but also facilitates the optimization of gears and their adaptation to specific operational requirements. This approach is particularly useful for studying contact stresses, wear, and the efficiency of the mechanism, making it an indispensable tool in modern engineering design.

Figure 2 presents a realistic three-dimensional model of an epicyclic gear mechanism with twelve mutually meshed satellite gears, created using a parametric modeling approach. The main components of the mechanism include the sun gear (Sun - Z_1), satellite gears (Satellites - Z_n), and the ring gear (Ring - Z_r).

This model demonstrates the complex kinematic interactions between the components, which ensure uniform load distribution and high torque transmission efficiency.

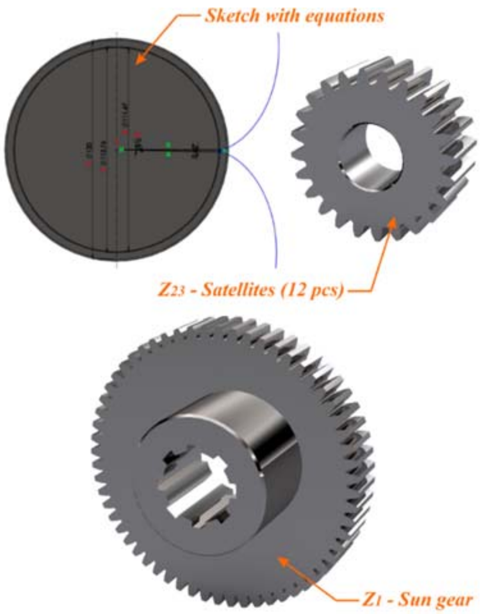


Fig.1. Parametrically modeled gears.

Figure 2 presents a realistic three-dimensional model of an epicyclic gear mechanism with twelve mutually meshed satellite gears, created using a parametric modeling approach. The main components of the mechanism include the sun gear (Sun - Z_1), satellite gears (Satellites - Z_n), and the ring gear (Ring - Z_r). This model demonstrates the complex kinematic interactions between the components, which ensure uniform load distribution and high torque transmission efficiency.

Fig.2. Geometric model of a mechanism with interlocking satellites.

The use of the parametric modeling approach allows for dynamic adjustment of the geometric characteristics of the gears, providing adaptability to different requirements and operating conditions. Each component is designed with optimal precision to ensure minimal contact stresses and extended durability of the mechanism. The mutually meshed satellite gears significantly enhance the system's wear resistance and increase its load-carrying capacity.

This parametric model is suitable for simulations and analyses related to mechanical stability, dynamic behavior, and the optimization of epicyclic gear mechanisms. It serves as a reliable foundation for improving the structural characteristics of the system and its integration into various industrial applications.

Tables 1 and 2 present the formulas and parameters introduced into the software environment for generating the involute profile of cylindrical gears with straight teeth. Table 1 contains the essential geometric characteristics and relationships required for accurate profile modeling, while Tab.2 provides specific equations and values used in various sketches of the model. This methodology ensures high precision and reliability in gear design and can be effectively applied to a wide range of engineering applications.

The methodology is specifically developed for cylindrical gears with straight teeth, which represent one of the most widely used types of gears in mechanical systems. The simplicity of their profile allows for efficient modeling and optimization, making them a preferred choice in practice.

The developed model can be further enhanced by including parameters such as the helix angle of the teeth, enabling the generation of helical cylindrical gears. This enhancement would expand the applicability of the methodology and adapt it to mechanisms requiring more complex kinematic interactions. Additionally, the methodology could be adapted for modeling other types of gears, such as worm gears, worms, and bevel gears. However, creating such models involves greater complexity due to the need for additional parameters and relationships stemming from the more intricate geometry of these mechanisms.

The applicability of the presented methodology, as well as its potential for expansion to more complex gear systems, underscores the significance of CAD tools in achieving high precision, optimization, and efficiency in modern engineering design.

Table 1. Geometric parameters of the gear profiles

Name	Designation	Value/Formula	Result
Diametral pitch	dp	*12.7*	*12.7*
Module	m	*2*	*2*
Pi	Pi	*3.1416*	*3.1416*
Number of teeth	z	*24*	*24*
Profile angle	α	*20*	*20*
Tooth tip height	ha	$1*m$	*2*
Tooth root height	hf	$1.25*m$	*2.5*
Tooth height	he	$ha+hf$	*2.4*
Radial clearance	c	*0.25*	*0.25*
Pitch circle	d	$m*z$	*48*
Root circle	df	$d-2*m*(1+c-x)$	*43*
Tip circle	da	$d-ha*2$	*52*
Base circle	db	$d*\cos\alpha$	*45.11*
Phi	ϕ	$Sqr(d_2-db_2)db*180/\pi-\alpha)$	*0.854*
Fillet	r	$c*\cos\alpha*\tan((90+\alpha)/2)$	*0.22*
Adjustment	x	*0*	*0*

Table 2. Equations and parameters used in CAD sketches

Sketch - equation	Formula	Value
D1@Sketch 1	$d+2*m$	*52*
D1@Sketch 2	$z*m$	*48*
D2@Sketch 2	$d-2.5*m$	*43*
D3@Sketch 2	$d*\cos\alpha$	*45.11*
D4@Sketch 2	$360/(4*z)-\alpha$	*2.90*

3. Designing an epicyclical gear mechanism in a CAD environment

Based on the thorough investigation of the mechanisms presented in (Yan *et al.*[13]; Ye *et al.* [14]; Gao *et al.* [15]), it can be concluded that the EGM design is the most suitable for achieving a high output torque $M_{out}=M_{\max}$. This makes the model highly effective for applications requiring high reliability and load resistance.

The analysis results are summarized in Tab.3, which contains the key parameters used for the geometric modeling of the mechanism. The model was developed in the SolidWorks software environment, where a virtual simulation of the mechanism was implemented. The primary objective was to achieve full compliance between the virtual model and real operating conditions by using standard components for assembly and integration of the mechanism. This ensures precision and reliability in the design process.

The creation of a virtual test stand allows for the closest possible approximation to real working conditions, enabling an in-depth analysis of the load capacity and durability of the mechanism. This approach provides the ability to evaluate the loads occurring in individual components during operation. The analysis delivers valuable insights into the distribution of loads, which supports the optimization of the design and ensures high system efficiency (Hsu and Lam, [16]; Hsu and Fong, [17]).

Table 3. Key geometric and mechanical parameters of the EGM mechanism

z_1	z_2	z_3	z_4	$z_{1'}$	$z_{2'}$	$z_{3'}$	$z_{4'}$	$P_{dv}[kW]$	$M_{dv}[Nm]$	$M_{out}[Nm]$
60	*24*	*24*	*120*	*67*	*17*	*17*	*135*	*2.2*	*14.3*	*1924.64*

Table 3 presents the key characteristics of the mechanism, including parameters z_1, z_2, z_3, $z_{1'}$, $z_{2'}$, $z_{3'}$, rotor power (P_{motor}), and torques (M_{motor}, M_{out}). These were calculated to ensure a balance between output power and the mechanical stability of the system, with the high output torque $M_{out}=1924.64Nm$ highlighting the mechanism's efficiency under load.

The geometric model of the epicyclic gear mechanism was designed with a comprehensive structural framework, encompassing all key details and elements required to ensure its smooth and efficient operation. The primary working components-the gears of the differential gear mechanism (DGM2) and the simple gear mechanism integrated into the system-were developed using a parametric model. This approach ensures high accuracy and adaptability in modeling the components, which is essential for achieving optimal performance. All other components of the mechanism were designed in accordance with the principles of geometric design within a CAD environment, adhering to all requirements for creating assembled units. The model was constructed to simulate realistic operating conditions and facilitate the analysis of interactions between components. Figure 3 provides a visualization of the complete mechanism, mounted on an artificial base, with the main components positioned on standard profiles.

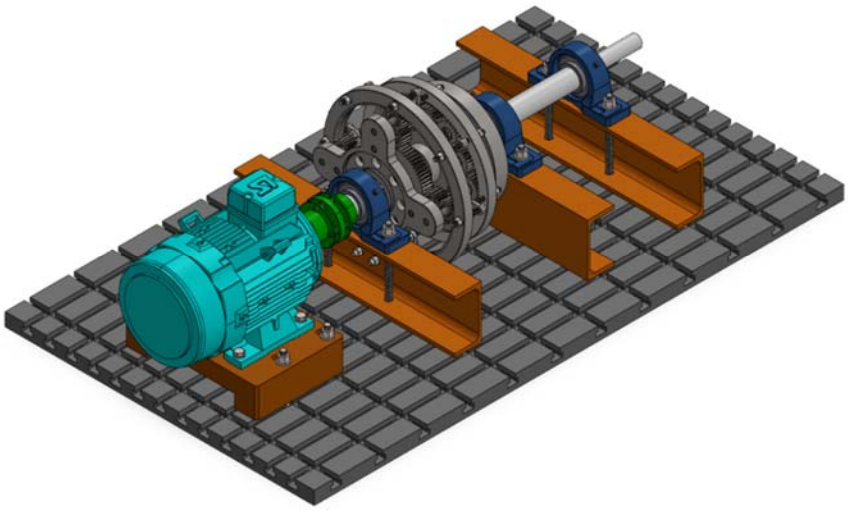


Fig.3. EGM – Complete construction built in a CAD environment.

The mechanism is powered by a preselected electric motor, with the connection between the motor and the mechanism implemented via a rigid coupling. This configuration ensures stability and reliability in transmitting torque to the gears. The shafts of the mechanism are supported by standard bearings, which ensure proper movement of the components while minimizing friction and wear. The arrangement and mounting of the components were designed to provide durability and reliability under various operating conditions.

This structural approach, implemented in a CAD environment, enables detailed analysis and optimization of the mechanism while ensuring its functionality and adaptability for various applications.

The epicyclic gear mechanism EGM occupies a central position in the presented model (Fig.4) and serves as a key component of the construction. The mechanism is powered by a preselected electric motor, which transmits torque to the system's input. At the output stage of the EGM, an additional mechanism or device can be mounted, tailored to the specific requirements and purpose of the application. The flexibility of the design allows it to be adapted to various industrial applications, such as drive systems, robotic mechanisms, or power transmissions.

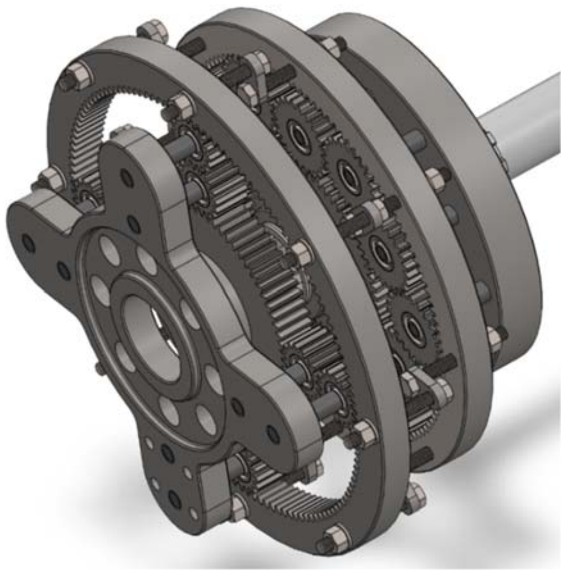


Fig.4. EGM – Construction of the mechanism itself.

This mechanism is designed to ensure high load capacity, efficiency, and durability under various operating conditions. The construction includes mutually meshing gears, which contribute to the even distribution of loads and the minimization of wear on the contact surfaces. The stability and reliability of the system are guaranteed through precisely manufactured components and optimal distribution of mechanical loads.

The functionality of the EGM makes it a crucial component for modern mechanical systems, offering the potential for integration with various subsystems and mechanisms in accordance with the requirements of engineering design and production needs.

In the presented gear configuration and the additional mechanism (Fig.5), it is established that there are no significant differences in the dimensions and overall size of the system. This is a key advantage, as it allows the mechanism to retain its compactness, which is an important factor in the design and optimization of gear transmissions. The achieved result aligns with the primary objective–an increase in the output torque (M_{out}) without requiring significant changes to the system's structural parameters.

The structural comparison between different mechanisms, as illustrated in the figure, shows that the integration of additional elements does not adversely affect the system's overall dimensions. This result is particularly important for applications where spatial constraints are critical. Moreover, the adaptation of the mechanisms to achieve higher efficiency and load capacity has been realized without compromising compactness or structural integrity.

This structural configuration demonstrates efficiency in the development process and enables the achievement of significantly higher output torque values, which is crucial for a wide range of industrial applications.

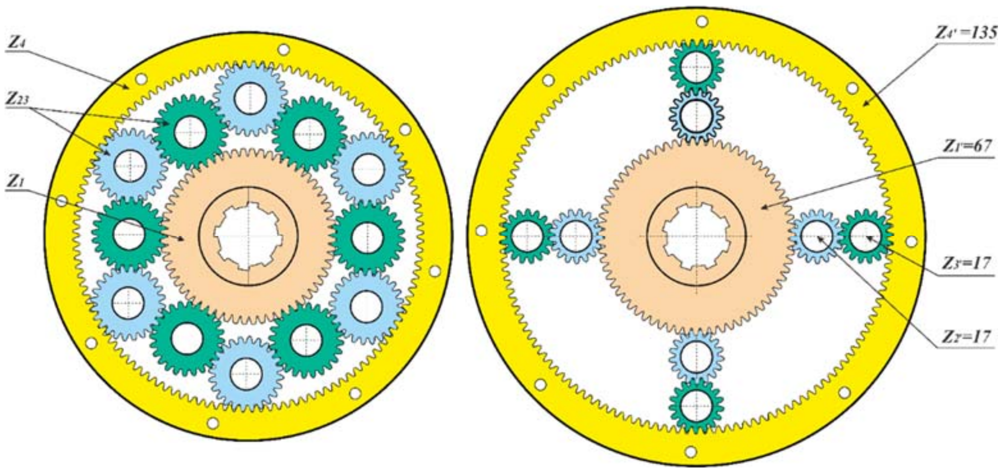


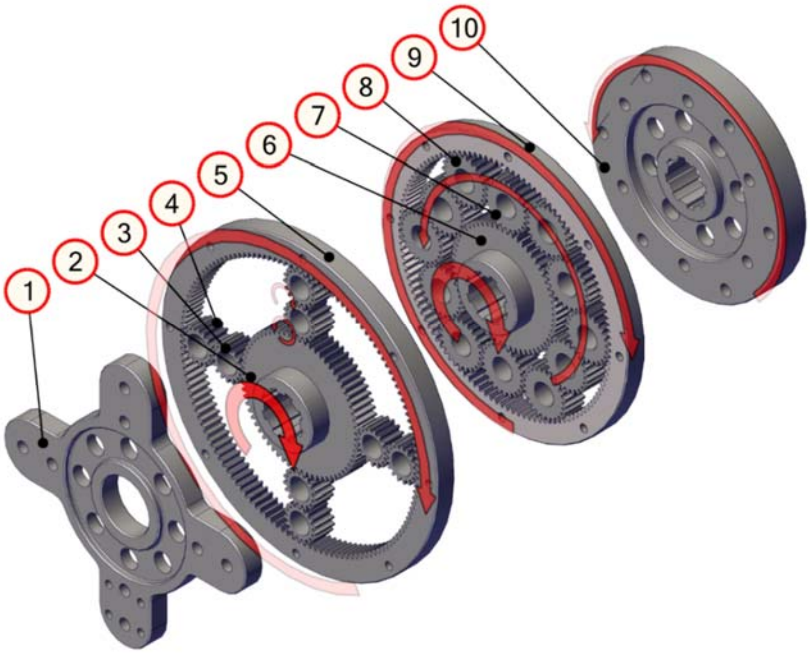
Fig.5. Structural comparison between the mechanisms.

Figure 6 presents the main components of the epicyclic gear mechanism EGM in an exploded view. The construction includes a front and rear mounting plate, a sun gear, planetary gears, and a ring gear. The central shaft ensures the transmission of torque, while the bearings and mounting components provide stability and smooth operation. The components are designed to ensure efficient load distribution and durability of the mechanism.

Table 4 provides a detailed analysis of the kinematic behavior of the mechanism's components, including their number of teeth, direction of rotation about their own axis, and planar motion. The central stationary element H' serves as the primary reference point for analyzing the other components. The gears z_2 and z_3 perform complex combined motions, which include rotation about their own axes as well as additional planar movements around the axis of z_1.

This complexity is characteristic of intermeshed mechanisms and ensures even load distribution. The components $z_{1'}$, $z_{2'}$, $z_{3'}$ and $z_{4'}$ are designed with varying numbers of teeth, contributing to the system's smooth motion and efficiency. Particular attention has been given to the directions of rotation, which play a critical role in synchronizing the elements and achieving optimal kinematic stability.

The table clearly illustrates the distinctions between stationary elements and those in active motion, demonstrating how each position contributes to the overall functionality of the mechanism. This information is essential for understanding dynamic interactions and optimizing the gears under different operational conditions.



1. front mounting plate; 2. sun gear; 3. planetary gears; 4. ring gear; 5. bearings and supports; 6. mounting screws and bolts; 7. central shaft; 8. protective cover; 9. intermeshing gear elements; 10. rear mounting plate.
Fig.6. Main elements of EGM.

Table 4. Kinematic characteristics of the EGM components

Position	Designation	Number of teeth	Direction of rotation around its axis	Movement plane
1	H'	-	Fixed	Movable
2	$z_{1'}$	*67*	-	not
3	$z_{2'}$	*17*	+	not
4	$z_{3'}$	*17*	-	not
5	$z_{4'}$	*135*	+	not
6	z_1	*60*	-	not
7	z_2	*24*	+	yes
8	z_3	*24*	-	yes
9	z_4	*120*	-	not
10	H	-	+	not

4. Selection of standard elements – driving and assisting the movement process

The selection of standard components for the mechanism's construction plays a critical role in ensuring its proper operation and enabling subsequent simulations and experimental studies. In this research, three key standard components were chosen: an electric motor with predefined parameters (1), a UCP 208 bearing housing (2), and a double disc coupling (3) (Fig.7). These components were selected to ensure optimal load distribution and compact dimensions for the mechanism while facilitating its integration into the overall design of the EGM.

The appropriate selection of standard components is essential for avoiding structural errors and discrepancies that could arise during simulations or under load. Incorrect component selection could result in significant operational difficulties, including inefficiency or premature wear. Therefore, it is crucial to perform a preliminary analysis of the mechanical requirements and align them with the characteristics of the chosen components. In this case, compatibility with the motor's power output has been ensured by analyzing the mechanical demands and the specifications of the selected components. These elements guarantee optimal performance and durability of the mechanism.

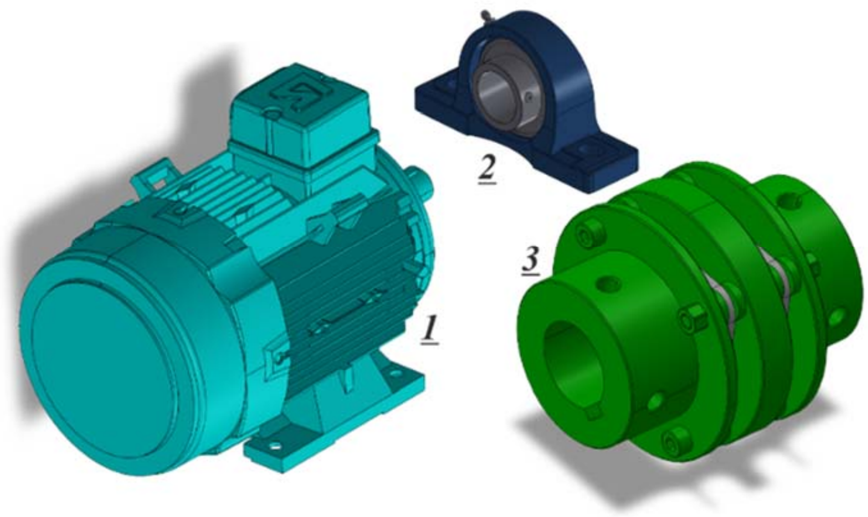


Fig.7. Selected standard components: 1 – electric motor; 2 – UCP bearing; 3 – disc coupler.

The selected configuration can be modified as needed by utilizing motors with different dimensions and power ratings. This provides significant flexibility for the mechanism, which is particularly important when higher input power is required. For instance, increasing the input power will result in a substantial increase in the output torque ($M_{out} = M_{in}$), which is essential for mechanisms designed for high-load applications.

Additionally, the ability to adjust the height and position of the shaft axis offers considerable advantages when adapting the mechanism to various operating conditions. This highlights the importance of the chosen standard components, which are easy to replace and modify in accordance with the principles of modular construction design.

Figure 8 presents a central view of the mechanism and illustrates the positioning of the UCP bearings, which play a critical role in maintaining the stability and proper functioning of the system. The electric motor, positioned on the left, serves as the primary driving element, transmitting torque to the shaft via a coupling. This coupling is designed to compensate for minor alignment deviations and ensure smooth torque transmission while reducing vibrations.

The UCP bearings are positioned along the shaft and function as stable supports to prevent displacements and minimize bending. This ensures an even distribution of loads during the operation of the mechanism. The shaft is held in a stable position thanks to the precise arrangement of the bearings, which significantly extends its service life.

All components are mounted on a robust and durable frame, which serves to reduce vibrations and provide structural stability. This configuration enables the mechanism to operate efficiently and reliably, ensuring resistance to dynamic loads and maintaining accuracy during operation.

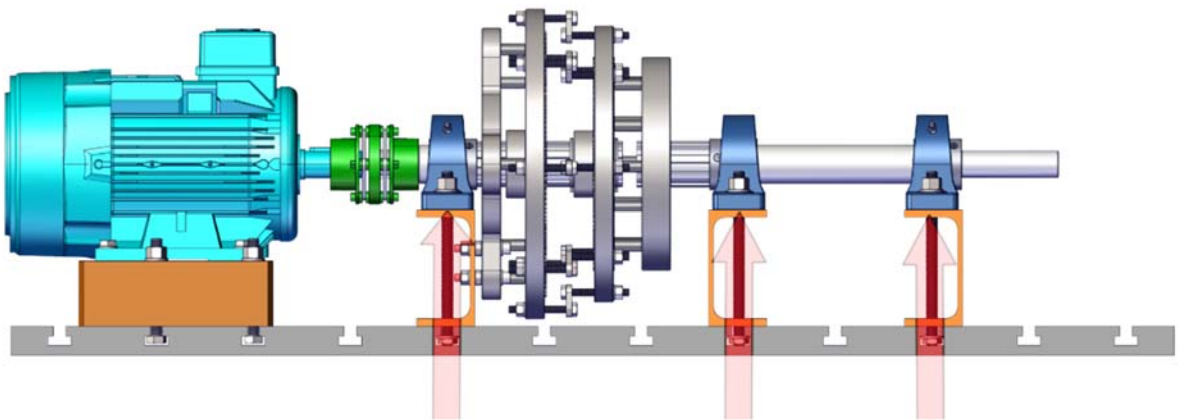


Fig.8. Selected central view and arrangement of UCP bearings.

5. Simulation analysis and study of an epicyclic gear mechanism

To analyze the motion and efficiency of the gear mechanism with mutually meshing satellites, several fundamental mathematical models and simulations are utilized. The kinematic models are based on equations of motion that describe angular velocities, gear ratios, and kinematic relationships between the various components of the gear mechanism. Willis' method is employed to transform the epicyclic mechanism into a simple gear mechanism and determine the gear ratios. The finite element method (FEM) is used to analyze stresses and deformations in the gears and other components of the mechanism. This method enables a detailed examination of the mechanical characteristics of the elements, considering boundary conditions and material properties.

Using specialized CAD and CAE software, such as SolidWorks and its Simulation module, simulations of the real dynamic behavior of the mechanism are performed. These simulations provide a visualization of the gear motion and determine the mechanism's efficiency under various loads and torques. Additionally, equations are applied to calculate the load-carrying capacity and durability of the gears based on their geometric and material characteristics, which are crucial for optimizing the mechanism. All these methods and simulations contribute to improving the design and efficiency of the gear mechanism by optimizing its parameters and predicting its behavior under real-world conditions.

The sizing of complex-shaped elements involves solving intricate systems of differential equations that lack analytical solutions. In engineering practice, the finite element method (FEM) has proven particularly applicable. FEM can be described as dividing the body or structure into a finite number of parts, called finite elements, characterizing the behavior of each individual element using simplified means, and then reconnecting the elements at points known as nodes. As a result of this process, a system of algebraic equations is obtained, which, in the case of stress analysis, represent equations for the static equilibrium of the nodes.

A significant advantage of FEM over analytical methods is its ability to solve problems without restrictions on body shape, arbitrary static boundary conditions (loading), and arbitrary geometric conditions (supports). Constructing the geometric model, setting adequate static and geometric boundary conditions, and understanding the properties of the finite elements comprising the FEM model can only result from a deep study of the fundamentals and practical application of FEM. Extensive experience and knowledge are also required in the field of solid deformable body mechanics to assess the results obtained through FEM.

The theory of finite elements, particularly in the context of modeling the strength of physical bodies, is based on the fourth strength theory (von Mises), which has been further developed in numerous modern theoretical studies. According to the finite element theory, in the context of strength analysis, the magnitude of the equivalent stress is defined as follows:

$$\sigma_{von} = \sqrt{\frac{(\sigma_1 + \sigma_2)^2 + (\sigma_2 + \sigma_3)^2 + (\sigma_3 + \sigma_1)^2}{2}} \,. \tag{5.1}$$

Formula (5.1) provides a reliable tool for evaluating the mechanical properties of the material under complex stress conditions. The results of the simulations confirm that the analyzed mechanism is robust and reliable, maintaining its performance characteristics across a wide range of operational loads.

Figure 9 illustrates the finite element model used to simulate the stress distribution within the mechanism. The geometry of the model has been designed to accurately reflect the real interactions between the gear pairs. The stress concentration is evenly distributed due to the optimal tooth profile and the material properties of AISI 1045. This simulation enables the identification of zones with potentially high stress, such as the contact points between the gear teeth. Such information is essential for enhancing the durability and reliability of the mechanism.

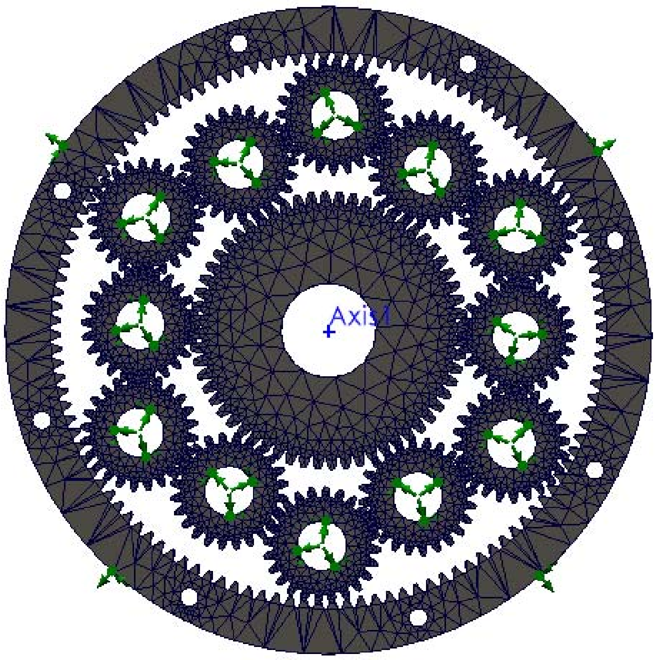


Fig.9. Finite element model of the gear mechanism.

The materials and their physical-mechanical properties were selected from the SolidWorks-Simulation database. In selecting the material, it is assumed that deformation is linearly dependent on stress, which is a fundamental assumption for isotropic materials. AISI 1045 steel (cold-drawn) was chosen as a suitable material due to its high strength, good ductility, and predictable behavior under load. One of the key advantages of this material is that even under significant deformations, the stress changes minimally, ensuring high stability and reliability of the mechanism during operation.

The specified material properties, including density, Young's modulus, and Poisson's ratio, are presented in Tab.5. These characteristics are critical for the accuracy of the simulations and for assessing the mechanical stresses and deformations in the structural elements.

Table 5. Mechanical properties of AISI 1045 steel

Material	Young modulus *MPa*	Poisson ratio	Yield strength *MPa*	Tensili strength *MPa*
AISI 1045, Steel cold drawn	2.05×10^5	*0.29*	*530*	*625*

When the physical properties of AISI 1045 steel, presented in Tab.5, play a key role in ensuring the mechanical stability and reliability of the gear mechanism. The yield strength (*530 MPa*) guarantees resistance to plastic deformation, even under high loads, while the ultimate tensile strength (*625 MPa*) provides additional safety and reliability under intensive operating conditions. The high Young's modulus (2.05×10^5 *MPa*) and Poisson's ratio (*0.29*) emphasize the material's ability to maintain its elasticity and resistance to stress, which is critical for minimizing wear in the contact zones of the teeth.

During deformation, the planetary gears do not penetrate the gear wheel (crown) but can only separate from it. To ensure proper simulation functionality, a fixed support "On Cylindrical Faces" was applied to the inner surfaces of the planetary gears (Fig.9). This support is specifically designed for cylindrical surfaces and restricts three degrees of freedom, blocking axial and radial displacement with a maximum value of *0 mm*. This ensures accurate simulation of mechanical loads, preventing motion distortion and maintaining realistic boundary conditions for the system.

The fixed supports "On Cylindrical Faces" play a critical role in ensuring mechanical stability during the simulation. By restricting axial and radial displacements, they accurately model real-world constraints, preventing structural instability. This ensures that the movements of the components align with theoretical calculations, providing a solid basis for the validity of the simulation results.

In the meshing process, the driving elements include the crown gear (element 2) and the sun gear (element 1), to which torque is applied. This torque drives the planetary gears (element 3), with the load being transmitted through the contact teeth between the intermeshed satellites. The interaction between these elements ensures that the planetary gears remain in equilibrium under the action of the torque and the resistance moment.

The applied torque on the crown and sun gears generates complex stresses that are transmitted through the intermeshed satellites. This stress distribution is exceptionally uniform, thanks to the optimal design of the mechanism. The even load distribution minimizes localized stress concentrations, which not only improves the efficiency of the mechanism but also significantly extends its operational lifespan. Furthermore, minimizing wear in the contact zones is critical for applications requiring high precision and reliability, such as the automotive and energy industries.

The load was implemented by applying torque to the designated surfaces, as shown in Fig.10. Three different torque values were specified for the study: *50*, *100*, and *150 Nm*, to simulate real operating conditions and evaluate the mechanism's behavior under varying load levels.

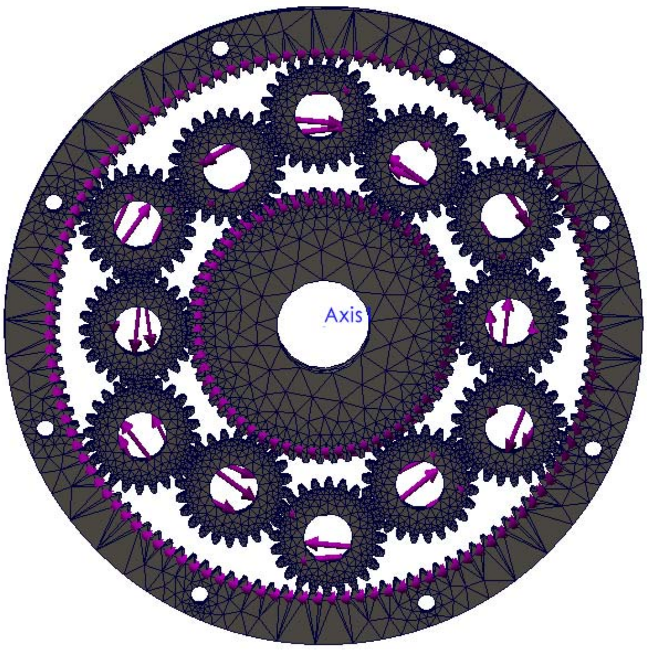


Fig.10. Setting the load on the gear mechanism.

Figure 10 illustrates the torsional loading applied to the gear mechanism. Through precisely defined boundary conditions and torques, realistic modeling of mechanical loads was achieved, enabling a detailed analysis of stresses and structural resilience. The analysis shows that for all tested torque values, the maximum stresses remain below the allowable limits for the material used, AISI 1045 steel. This confirms that the mechanism operates in a safe mode, with no risk of structural damage or compromise to its durability.

The simulation results highlight the mechanism's efficiency under high loads, demonstrating uniform stress distribution and minimal localized concentrations. This resilience is critically important for industrial applications requiring reliability, precision, and long operational life, such as automotive manufacturing, the energy sector, and the aerospace industry.

5.1 Kinematic modeling of the gear ratio

5.1.1 Gear ratio of a simple gear mechanism (SGM)

The gear ratio of a simple gear mechanism (SGM) can be determined using the following equation:

$$i_{14} = \frac{\omega_1}{\omega_4} = -(1)^b \frac{z_2}{z_1} . \frac{z_3}{z_2} . \frac{z_4}{z_3} = \frac{z_4}{z_1} = i_{ob} \tag{5.2}$$

where b represents the number of externally meshing gear pairs, which in this case is - $b = 2$.
z_1, z_2, z_3, z_4 and z_1', z_2', z_3', z_4' denote the number of teeth of the reduction gears.

This equation defines the transmission ratio i_{i4} between the input angular velocity ω_1 and the output angular velocity ω_4 of the simple gear mechanism (SGM).

The parameter i_{ob} represents the overall transmission ratio of the system. In this case, $b = 2$, as there are two pairs of meshing gears.

To determine the angular velocity ω_4 in the analyzed kinematic configuration, the following equation is used:

$$\omega_{out} = \omega_4 = \frac{\omega_1}{i_{ob}} , \tag{5.3}$$

This equation relates the input velocity ω_1 for the simple gear mechanism (SGM). i_{ob} represents the overall transmission ratio of the mechanism.

The relationship illustrates how the input angular velocity ω_1, to the output velocity ω_4 through the overall transmission ratio i_{ob} of the mechanism. The equation demonstrates how the input velocity ω_1 is transformed into the output velocity ω_4.

5.1.2 Transmission ratio in the planetary gear mechanism (PGM)

The transmission ratio of the planetary gear mechanism (PGM) is calculated using Willis's equation, which establishes the relationship between the input angular velocity, the output angular velocity, and the carrier velocity. The formula is expressed as follows:

$$i_{14}^{(H)} = \frac{\omega_1 - \omega_H}{-\omega_H} = \omega_{ob}, \quad 1 - i_{1H} = i_{ob}, \quad i_{14}^{(H)} = \frac{\omega_1}{\omega_H}. \tag{5.4}$$

Equations (5.4) defines the relationship between the input angular velocity ω_1, the carrier angular velocity ω_H, and the overall transmission ratio i_{ob} of the mechanism. The PGM ensures a link between the velocities of the different elements.

$$\omega_{out}^{H} = \omega_H = \frac{\omega_1}{1 - i_{ob}}. \tag{5.5}$$

Equation (5.5) provides a method for calculating the carrier angular velocity for a given input velocity ω_1. This relationship is useful for analyzing the speed in the planetary gear mechanism (PGM) when the carrier is the output component.

5.1.3 Transmission ratio in the differential gear mechanism (DGM)

The transmission ratio for the differential gear mechanism (DGM) is determined using Willis's method, which expresses the relationship between the input angular velocity ω_1, the output angular velocity ω_4, and the carrier angular velocity ω_H. The formula is given as:

$$i_{14}^{(H)} = \frac{\omega_1 - \omega_H}{\omega_4 - \omega_H} = i_{ob}. \tag{5.6}$$

Equation (5.6) describes the overall transmission ratio i_{ob} of the mechanism. i_{ob} is a fundamental parameter that characterizes the kinematic dependencies in the system. It establishes the relationship between the output angular velocity ω_4 and the carrier angular velocity ω_H representing the general transmission ratio of the mechanism.

For a predefined output component 4 in the mechanism, the relationship for $\omega_{\max}$ is expressed as:

$$\omega_{out} = \omega_4 = \frac{\omega_1 - \omega_H}{i_{ob}} + \omega_H. \tag{5.7}$$

Equation (5.7) provides the dependence between the input angular velocity ω_1, the carrier angular velocity ω_H, and the output angular velocity ω_4. This relationship is particularly useful for calculating cases where component 4 serves as the primary output.
When the output link is the carrier ω_H, the following formula is used:

$$\omega_{out} = \omega_H = \frac{i_{ob}\omega_4 - \omega_1}{(i_{ob} - 1)}. \tag{5.8}$$

Equation (5.8) illustrates how the angular velocity of the carrier ω_H depends on the other kinematic parameters of the system. This relationship is particularly important for analyzing cases where the carrier serves as the output component.

6. Results of the study on jerk

Jerk, which represents the rate of change of acceleration over time, is a key parameter for assessing motion smoothness and dynamic loads in mechanical systems. This parameter plays a critical role in determining the stability and efficiency of epicyclic mechanisms, especially in applications requiring precision and minimal excessive vibration increase.

In this study, jerk was analyzed for different kinematic configurations of the epicyclic mechanism, including a simple gear mechanism (SGM), a planetary gear mechanism (PGM), and two types of differential gear mechanisms (DGM1 and DGM2). The analysis involved determining the input and output angular velocities for each configuration using mathematical dependencies based on the kinematic laws of mechanics. By investigating these configurations, it was possible to assess how jerk influences motion smoothness at different stages of the mechanism's operational process. The results highlight the importance of uniform load distribution in mechanisms with mutually engaged satellites. This leads to a reduction in localized loads and smoother operation of the mechanism, which is particularly crucial for industrial applications where vibrations and noise must be minimized.

The dynamic analysis shows that mechanisms with mutually engaged satellites (DGM1 and DGM2) exhibit significantly lower jerk values compared to other configurations (SGM and PGM). This demonstrates the advantages of the proposed design, including improved stability and smoother operation, making it suitable for applications with high demands for reliability and durability.

The output angular velocity ω_{out} is calculated by multiplying the input angular velocity ω_{in} and is determined based on the transmission ratio of the mechanism:

$$\omega_{out} = \omega_{in} \times \text{Transmission ration.} \tag{5.9}$$

The acceleration (a) for each configuration is calculated as:

$$a = \omega_{out} - \omega_{in}\,. \tag{5.10}$$

The jerk (j) is defined as the rate of change of acceleration between two consecutive configurations of the mechanism. It reflects the dynamic variations in the system and is calculated as the ratio of the difference in acceleration to the time interval, which is considered constant.

$$j = \frac{\Delta a}{\Delta t} \tag{5.11}$$

where Δt is assumed to be constant.

This approach provides in-depth information on the dynamic behavior of the mechanism, allowing for an assessment of motion smoothness and stability under different kinematic configurations. These characteristics are crucial for analyzing mechanisms with high precision and stability requirements.

Table 6 presents the results of the jerk (j) analysis for different kinematic configurations, including the input and output angular velocities, acceleration, and the obtained jerk values. The data illustrate how each configuration affects the dynamic parameters of the mechanism, emphasizing the importance of proper design to ensure optimal functionality.

Table 6. Jerk analysis results for different gear configurations

Configuration	Input angular speed ω_{in}	Output angular speed ω_{out}	Acceleration a	Jerk (j)
OGM	$152 rad/s$	$75.88 rad/s$	$-76.12 rad/s^2$	$0.00 rad/s^3$
PGM	$152 rad/s$	$-152 rad/s$	$-304.00 rad/s^2$	$-227.88 rad/s^3$
DGM1	$152 rad/s$	$350 rad/s$	$198.00 rad/s^2$	$502.00 rad/s^3$
DGM2	$152 rad/s$	$548 rad/s$	$396.00 rad/s^2$	$198.00 rad/s^3$

Figure 11 presents the jerk analysis for different kinematic configurations of the epicyclic mechanism. The graph illustrates the jerk values for OGM, PGM, DGM1, and DGM2, highlighting the dynamic behavior of each configuration. The highest jerk values are observed in DGM1, indicating significant dynamic loads and instability. This suggests that DGM1 is suitable for applications with high load requirements but requires additional optimization to improve stability.

On the other hand, OGM demonstrates zero jerk, making it ideal for smooth and stable motion. DGM2 exhibits lower jerk values compared to DGM1, making it a more stable choice for applications requiring a balance between dynamic response and stability. PGM, with its negative jerk values, shows high dynamic potential but necessitates careful engineering to minimize dynamic loads.

The different kinematic configurations of the epicyclic mechanism exhibit significant variations in their dynamic behavior, highlighting their suitability for specific applications.

Simple gear mechanism (SGM or OGM): This configuration demonstrates moderate changes in acceleration $-76.12 rad / s^2$ and zero jerk $0.00 rad / s^3$, making it ideal for applications requiring smooth and stable motion. Its predictable behavior ensures consistent performance in systems where gradual motion transitions are critical.

Planetary gear mechanism (PGM): The PGM exhibits sharp negative changes in acceleration $-304.00 rad / s^2$ and high negative jerk $-227.88 rad / s^3$ This behavior can result in significant dynamic loads, suggesting its potential use in systems where such abrupt changes are permissible but requiring careful design to minimize the effects of these loads.

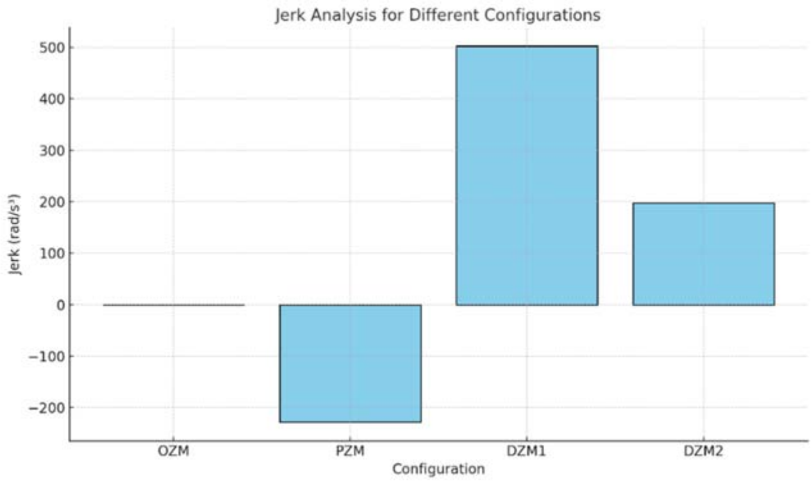


Fig.11. Jerk analysis for different kinematic configurations of the epicyclic mechanism.

Differential gear mechanism 1 (DGM1): This configuration demonstrates the highest values of acceleration $198.00 rad / s^2$ and jerk $502.00 rad / s^3$. These results indicate the presence of significant dynamic peaks, suggesting that DGM1 is suitable for applications with high load requirements. However, additional measures are necessary to enhance the mechanism's resilience and effectively address these extreme conditions.

Differential gear mechanism 2 (DGM2): Compared to DGM1, DGM2 exhibits lower values of acceleration $396.00 rad / s^2$ and jerk $198.00 rad / s^3$. This indicates greater stability during operation, making it a more balanced choice for applications requiring reliability and smoother performance under dynamic conditions.

The unique dynamic characteristics of each configuration make it suitable for specific engineering applications, depending on the requirements for motion smoothness, load management, and operational stability.

The analysis shows that different kinematic configurations of the epicyclic mechanism have a significant impact on jerk. DGM1 exhibits the highest dynamic instability, which is crucial for designing robust mechanisms. Including jerk as an evaluation criterion is essential for optimizing mechanical systems.

7. Results from the SolidWorks-simulation study

The stress analysis, performed using the finite element method (FEM), reveals that the maximum stresses occur in the contact zone between the teeth. Under applied torques of *50*, *100*, and *150 Nm*, the maximum von Mises stress values are *315 MPa*, *348 MPa*, and *413 MPa*, respectively. These values are significantly below the yield strength ($\sigma_S = 520 MPa$) and ultimate tensile strength ($\sigma_B = 625 MPa$) of the selected AISI 1045 steel material, proving that the design is optimized and capable of withstanding the specified loads without the risk of structural failure.

The uniform load distribution between the mutually engaged satellites significantly reduces the stress concentration in individual tooth pairs. This not only enhances the mechanism's durability under high loads but also significantly decreases the likelihood of microcrack formation or material fatigue. The results confirm that the system is efficiently designed for long-term operation under high torques.

Numerical analyses allow for detailed visualization and tabular organization of the results from the model. Studies conducted at input torques of *50*, *100*, and *150 Nm* demonstrate that the maximum equivalent stress occurs on the contact surface of the satellite gear, as illustrated in Figs 12, 13, and 14. Multiple teeth are engaged simultaneously, further distributing the load and reducing stress concentration.

Figure 12 presents the stress distribution based on the von Mises yield criterion under an applied torque of *50 Nm*. The calculated stresses reach *315.233 MPa*, which remains significantly below the allowable limits for the selected material. These results confirm that the design ensures safety and reliability under the specified conditions. For AISI 1045 steel, the yield strength and ultimate tensile strength are $\sigma_S = 520 MPa$ and $\sigma_B = 625 MPa$, respectively, guaranteeing a high safety margin for the mechanism's operation.

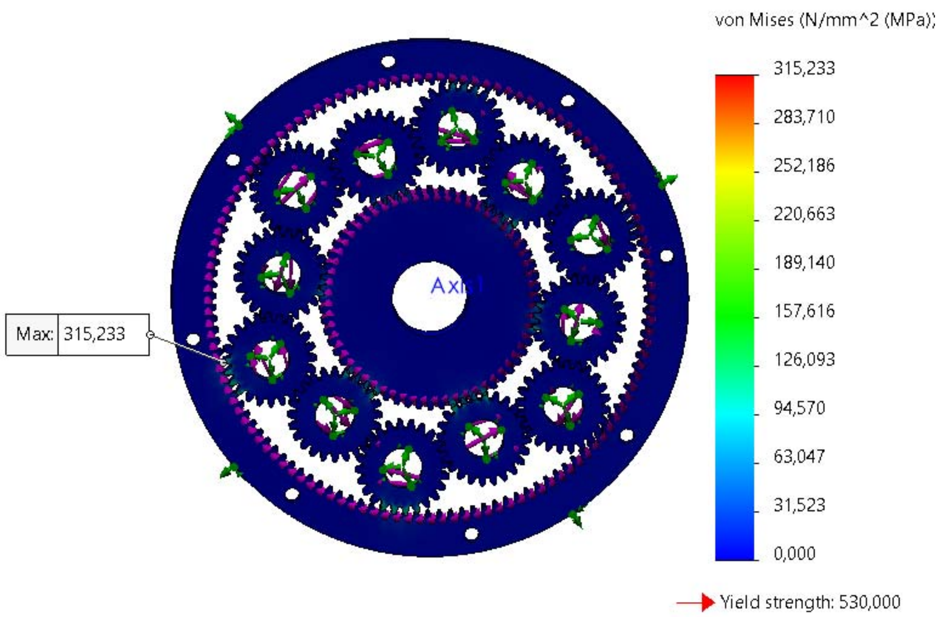


Fig.12. Maximum equivalent stress on a gear mechanism at a specified torque of *50 Nm*.

In Fig.13, the stress distribution in the gear mechanism is presented, calculated according to the von Mises yield criterion under an applied torque of *100 Nm*. The maximum stress values reach *348.912 MPa*, which is significantly below the yield strength limit for the selected material ($\sigma_S = 520 MPa$ for AISI 1045 steel). These results confirm that the design is capable of withstanding the applied load without risk of structural failure.

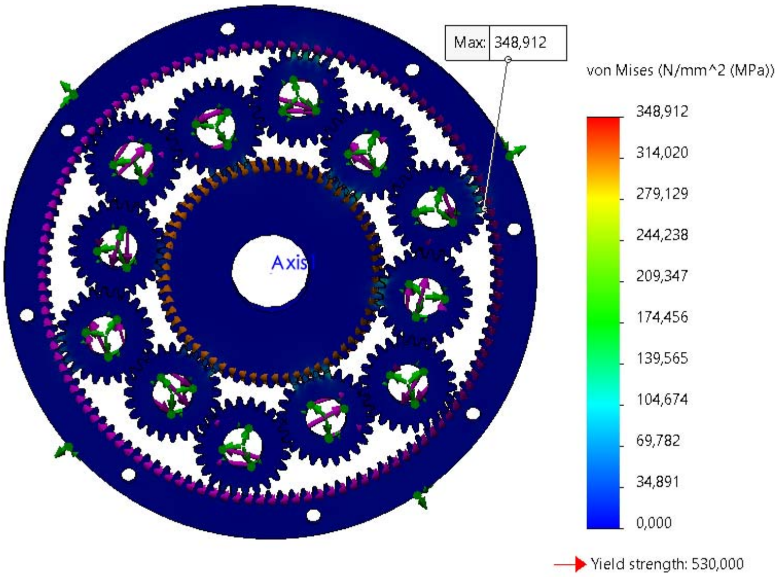


Fig.13. Maximum equivalent stress on a gear mechanism at a specified torque of *100 Nm.*

The uniform stress distribution on the contact surfaces of the gear teeth is attributed to the precise geometry of the gear profile and the mutual meshing of the satellites. This ensures the reliable operation of the mechanism and minimizes local stress concentrations, which could lead to wear or damage. The presented data demonstrate that the mechanism meets the allowable stress requirements and is designed with a sufficient safety margin for long-term operation.

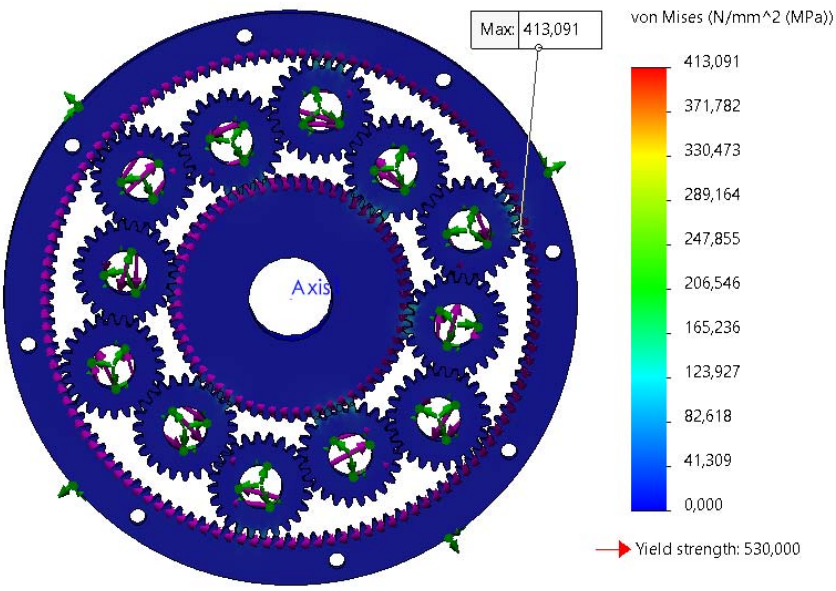


Fig.14. Maximum equivalent stress on a gear mechanism at a specified torque of *150 Nm.*

In Fig.14, the distribution of maximum stresses in the gear mechanism is presented, calculated according to the von Mises yield criterion under an applied torque of *150 Nm*. The maximum stress value reaches *413.091 MPa*, which is significantly below the yield strength limit of the material ($\sigma_S = 520MPa$ for

AISI 1045 steel). This confirms that the mechanism can reliably withstand the applied load without risk of plastic deformation or structural failure.

The stress distribution indicates that the contact area between the gear teeth is designed to minimize local stress concentrations. This is attributed to the precise geometry of the gear profile and the optimized meshing between the satellites. The results validate that the mechanism meets the safety and performance requirements, while also providing a sufficient margin of durability under high operational loads.

The simulations conducted in SolidWorks-Simulation reveal that the maximum deformations of the gear teeth occur at the upper part of the profile and remain within permissible limits (*0.014 mm* for all analyzed torque values). This minimal deformation demonstrates the precision of the design and the mechanism's ability to maintain functionality even under high loads.

The optimized geometry of the gear profile, combined with the high-quality material AISI 1045, ensures resistance against plastic deformations and prevents improper meshing of the gear teeth. This guarantees that the mechanism will continue to operate efficiently without loss of accuracy, regardless of the applied loads. The high durability of the mechanism results from both the proper distribution of contact forces between the gear teeth and the efficient transmission of torque.

In Fig.15, the displacement distribution under an applied torque of 50 N·m is presented. The maximum displacements are localized in the upper part of the gear tooth profile, reaching values up to 0.014 mm, which are significantly below permissible limits. These results confirm that the mechanism meets the requirements for allowable displacements, which is a key indicator of its reliability and longevity.

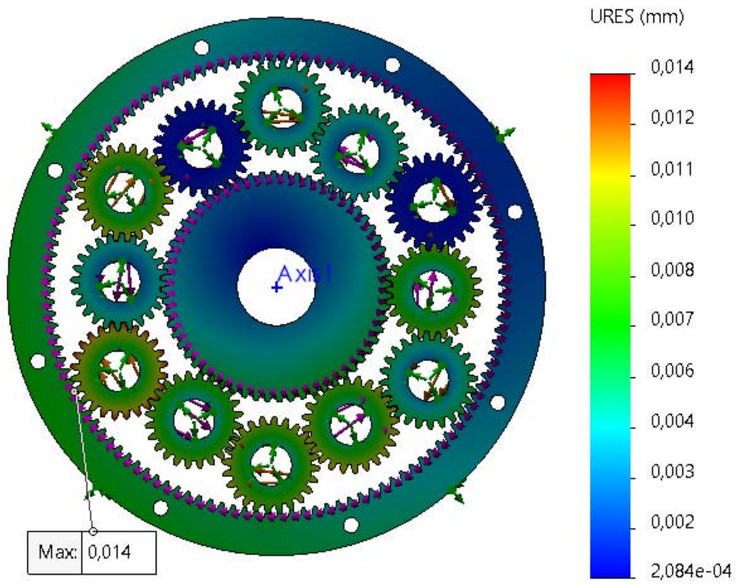


Fig.15. Displacement of the components of the gear mechanism at a specified torque of *50 Nm.*

Figure 16 presents the displacement distribution in the gear mechanism under an applied torque of *100 Nm*. The maximum displacements are localized in the upper part of the gear tooth profile, reaching a value of *0.014 mm*. This value is significantly below the permissible limits, confirming that the design meets engineering requirements for mechanical stability and reliability.

The results indicate that the minimal deformations generated at this torque level do not negatively affect meshing accuracy or the efficiency of the mechanism. This is a direct result of the optimized gear design, including the proper geometry of the involute profile and the selection of a high-strength material.

The presented data confirm that the mechanism is suitable for operation under high loads, ensuring long-term functionality and precision. The resistance to displacement contributes to wear prevention and extends the operational lifespan of the system.

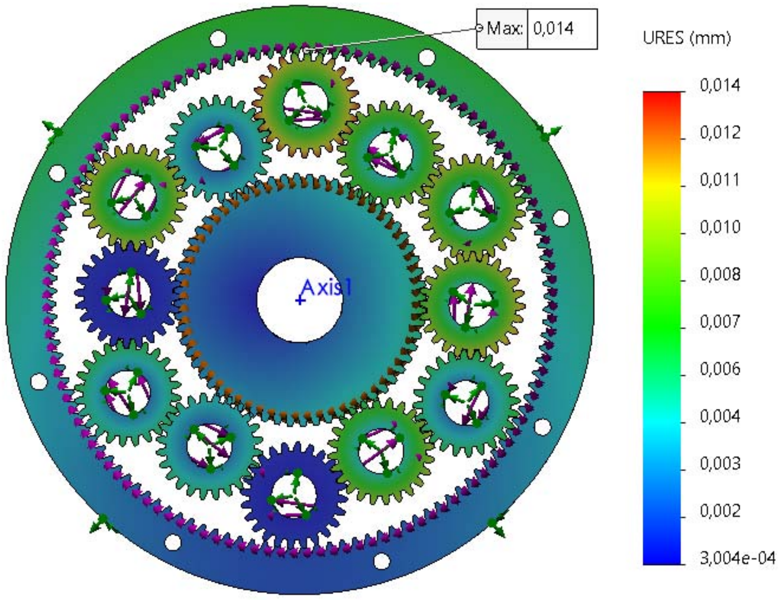


Fig.16. Displacement of the components of the gear mechanism at a specified torque of *100 Nm.*

Figure 17 presents the displacement distribution in the gear mechanism under an applied torque of *150 Nm*. The maximum displacements, observed in the upper part of the gear tooth profile, reach a value of *0.014 mm*. This value is significantly below the permissible limits, demonstrating that the mechanism meets the technical requirements for displacement and stability.

The small displacement values highlight the efficiency of the design, which is optimized for high loads. The precise geometry of the involute profile, combined with the durability of the selected material, ensures reliable gear meshing and prevents deviations that could lead to functional issues or increased wear.

The simulation results confirm that the structure is designed to maintain its operational characteristics even under such load levels, ensuring long-term functionality and system reliability.

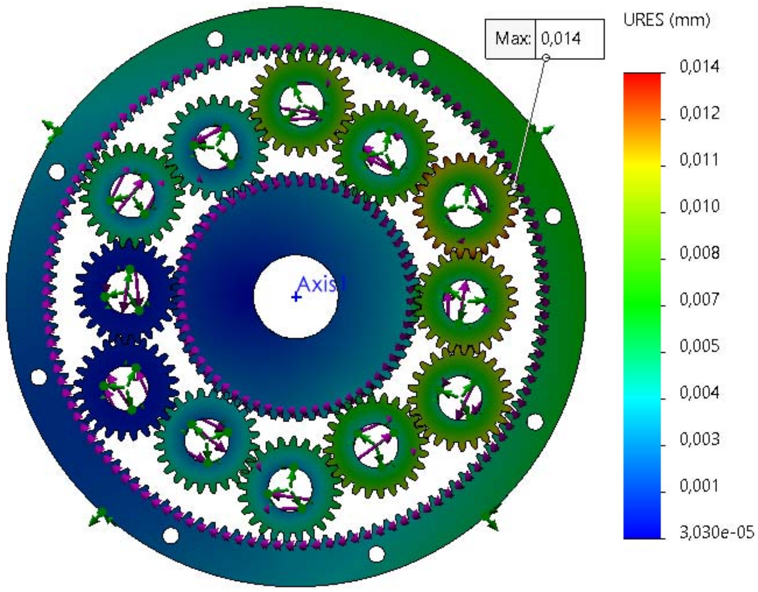


Fig.17. Displacement of the components of the gear mechanism at a specified torque of *150 Nm.*

Within the study, the influence of torque on the stability and performance of the mechanism was analyzed to assess its efficiency under different load conditions. Simulations conducted with torques of *50*, *100*, and *150 Nm* demonstrated that the mechanism maintains its stability even under high loads. A linear relationship was established between the applied load and the resultant stresses, proving that the mechanism operates within the elastic range of the selected material. This predictable linear behavior is a crucial indicator of the structural reliability, especially for applications requiring high load capacity and durability.

The study also examined contact interactions between the gear teeth, which play a fundamental role in load distribution. Simulations revealed that the larger contact area between the gear pairs, ensured by the mutual meshing of the satellites, significantly reduces local stresses. This minimizes the risk of mechanical failures, such as wear or microcracks, and extends the operational lifespan of the mechanism. The uniform distribution of contact pressure enhances the efficiency of torque transmission and contributes to the long-term reliability of the system.

To gain a deeper understanding of the mechanism's behavior, simulations were conducted to examine displacement distribution under different conditions. The analysis included scenarios considering local contact between the gear teeth while neglecting backlash in meshing. These conditions allowed for an evaluation of mechanical clearances and their impact on the mechanism's stability. The results showed that displacement values remain within permissible limits, confirming the structural reliability and resilience of the design. The maximum displacements, recorded in the upper part of the gear profile, reached *0.014 mm*, regardless of the applied load. This minimal deformation ensures that the mechanism retains meshing accuracy and efficiency, even under increased loads.

Tests conducted at input torques of *50*, *100*, and *150 Nm* confirmed that the mechanism operates stably and predictably. The contact surfaces were modeled as touching, ensuring realistic working conditions. The obtained results emphasize the importance of optimized contact characteristics for the durability and reliability of the system. These features make the mechanism suitable for industrial applications, including automotive engineering, the energy sector, and aerospace technology, where longevity and resilience are critical factors.

Figures 18 and 19 present the stress analysis results under "local contact" conditions, with backlash ignored for different values of input torque (*50*, *100*, and *150 Nm*). The graphs illustrate the stress distribution among the satellites, which plays a crucial role in ensuring the mechanism's stable operation.

The pie chart in Fig.18 visualizes the comparison of stress values at different input torques for each satellite. It demonstrates that stress is evenly distributed, with no significant peaks that could cause localized overloads. The linear graph in Fig.19 provides additional insight into stress values per satellite, emphasizing that an increase in torque leads to a proportional rise in stress, without exceeding the allowable limits.

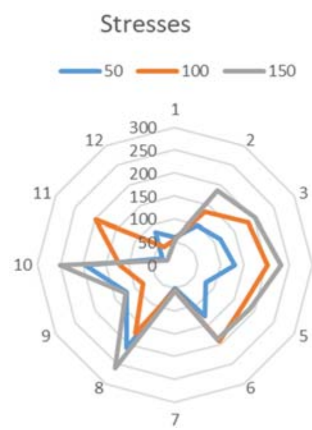


Fig.18. Circular diagram of stresses obtained from the analysis under the condition of 'local contact' and ignoring clearances.

These results highlight that the mechanism maintains stability and efficiency under various operating conditions. The even distribution of loads reduces the risk of localized stress concentrations, which in turn extends the system's service life and enhances its reliability. The presented data demonstrate that the mechanism is designed with high precision and is capable of operating effectively under significant loads, making it suitable for industrial applications that require durability and reliability.

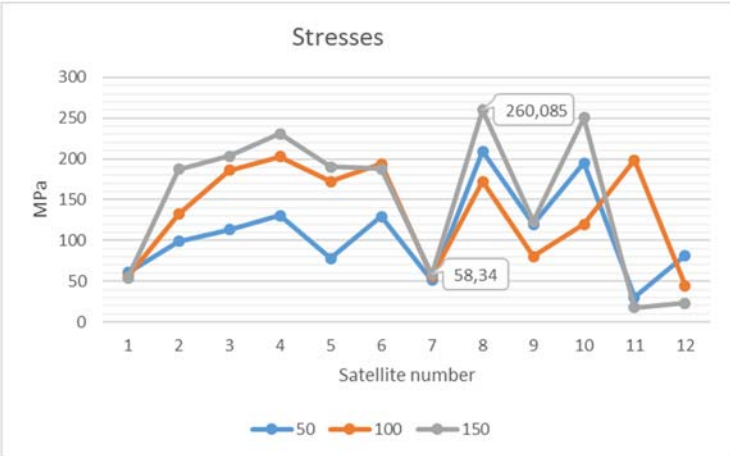


Fig.19. Graph of stresses obtained from the analysis under the condition of 'local contact' and ignoring clearances.

He displacements obtained and their impact on the stability of the mechanism are presented in Fig.20 and Fig.21. The tests were conducted at input torques of *50*, *100*, and *150 Nm* under conditions of "local contact" and ignoring backlash. These conditions allow for a realistic assessment of the mechanism's behavior and the effects of different loads on its performance.

Figure 20 presents a circular displacement graph, illustrating how the applied torques affect the position of the planetary gears. The graph clearly demonstrates that the mechanism maintains its stability even under increased loads, with displacements remaining evenly distributed among the satellites.

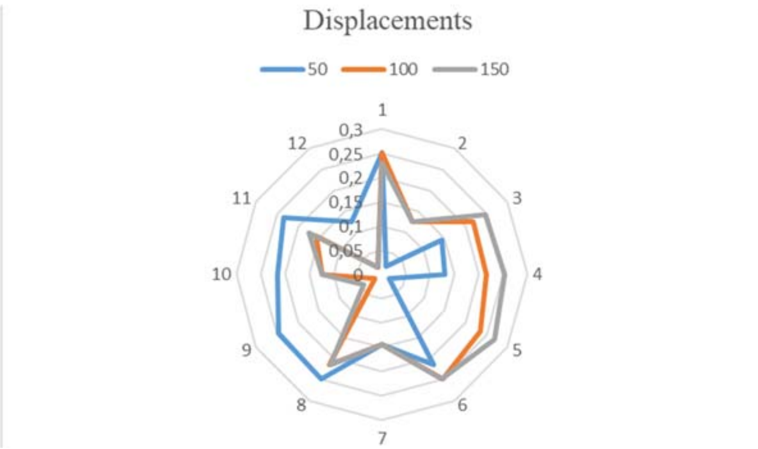


Fig.20. Circular diagram of displacements obtained from the analysis under the condition of 'local contact' and ignoring clearances.

Figure 21 presents a linear graph illustrating the relationship between the applied torques and the resulting displacements of each planetary gear. The results indicate that the mechanism operates within the permissible displacement limits, maintaining the precision and efficiency of the gear meshing. Regardless of the increase in torque, the displacements remain stable and predictable, demonstrating the high resilience and reliability of the design.

These findings confirm that the mechanism is designed with high precision and is capable of stable operation under various working conditions. Its characteristics make it particularly suitable for industrial applications requiring durability and reliability, such as automotive engineering, aerospace industry, and energy systems.

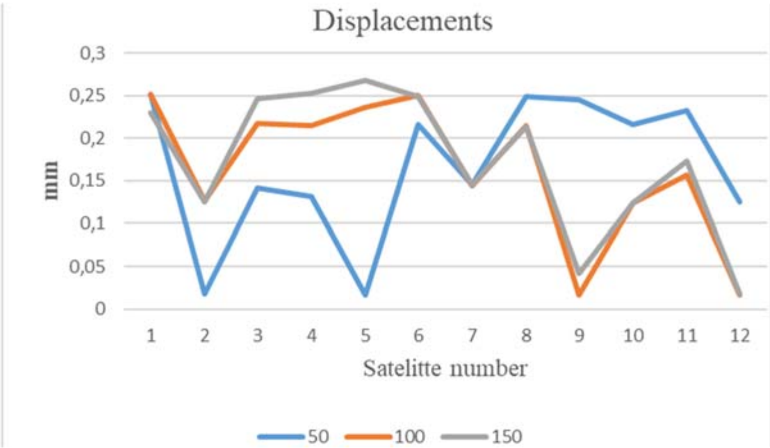


Fig.21. Graph of displacements obtained from the analysis under the condition of 'local contact' and ignoring clearances.

Additional stress analyses and interactions between the gear teeth highlight the key advantages of the mechanism with mutually engaged satellites. The obtained results demonstrate that this mechanism significantly improves the load distribution across individual gears, effectively eliminating localized stress concentrations that are typical for conventional gear mechanisms. In standard designs, localized loads in the contact zone often lead to accelerated wear and reduced efficiency. However, in the mutually engaged satellite mechanism, the interactions between the gears provide a significantly larger contact area, which reduces local stresses and enhances the overall efficiency of the system.

By applying the finite element method (FEM), it has been established that the maximum stresses, calculated according to the fourth strength theory (von Mises), remain well below the permissible values for the material used (AISI 1045 steel). The yield strength and ultimate strength of the material, *530 MPa* and *625 MPa*, respectively, were not exceeded under any operational conditions, proving that the mechanism is designed for high durability and reliability, even under significant loads.

One of the key physical characteristics of the mechanism is the uniform distribution of torque among the satellites. This ensures more stable and smoother motion, which is critical for applications requiring high dynamic stability. The simulations conducted for various torque values (*50*, *100*, and *150 Nm*) indicate that deformations of the gear teeth remain within permissible limits, ensuring the long-term reliability and functionality of the mechanism.

Furthermore, the mutual engagement of the satellites and the ring gear significantly reduces backlash in the system, thereby improving the kinematic accuracy of the mechanism. This is particularly important for industrial applications requiring high precision, such as automotive engineering, the energy sector, and aerospace industry. The uniform load distribution not only extends the service life of the mechanism but also reduces noise and vibrations during operation, thereby enhancing comfort and efficiency in various applications.

The study results highlight the advantages of the mutually engaged satellite mechanism, both theoretically and practically. Its wear resistance, torque transmission efficiency, and ability to operate under high loads make it highly suitable for a wide range of industrial applications, where reliability, efficiency, and durability are of paramount importance. These features position it as an ideal choice for modern engineering solutions that combine innovation with practical applicability.

The uniform load distribution, minimal deformations, and optimized gear geometry play a key role in extending the service life of the mechanism. This is particularly important for industries such as automotive engineering, aerospace technology, and the energy sector, where reliability, durability, and precision are critical factors. The improvement in load distribution, combined with the reduction of vibrations, ensures smoother and quieter operation of the mechanism. In automotive transmissions, this contributes to high efficiency and long lifespan under high loads. In the aerospace industry, the mechanism demonstrates excellent resistance to extreme conditions, making it ideal for flight control systems and robotic manipulators. In the energy sector, particularly in wind turbines, the mechanism improves energy transmission and significantly extends service life, while reducing maintenance costs.

The results of the simulation analysis, conducted in SolidWorks-Simulation, underscore the superiority of mutually engaged satellites compared to traditional mechanisms in terms of efficiency, durability, and reliability. The mechanism exhibits high adaptability to various loads and is exceptionally suitable for applications requiring precision and endurance, making it a preferred choice in critical industries such as automotive engineering, aerospace technology, and energy production.

Its practical applications are extensive and demonstrate significant potential. In automotive engineering, the mechanism can be utilized in hybrid and electric vehicles, where low weight, high efficiency, and minimal vibrations are key factors. In the energy sector, the mechanism can be used to optimize power transmission in wind turbines, leading to extended system lifespan and reduced maintenance costs. In the aerospace industry, its stability under extreme conditions makes it suitable for aviation and space system control.

The study demonstrates the importance of the mutually engaged satellite mechanism for various industrial applications. The obtained results emphasize its reliability, efficiency, and adaptability under different operational conditions. Future research could focus on further design optimization and expanding its applicability to new technological and industrial fields.

8. Discussion

The development of the new structural model for the synthesis of epicyclic chains underwent several stages, incorporating both theoretical and practical aspects. Throughout this process, key challenges were identified that influenced the methodology and design of the model. One of the main difficulties was the complexity of structural synthesis and the systematization of possible kinematic schemes (Chen *et al.* [18]; Hsu and Hsu, [19]). This required the development of an algorithm for automated generation and verification of configurations, utilizing graph-based methods to avoid duplication and ensure completeness.

Another critical aspect was the uniform load distribution among the components of the mechanism. This issue was addressed by integrating force balance criteria into the optimization process, which included analyzing the geometric parameters and their influence on dynamic stability (Cheng *et al.* [20]. Additional validation of the model was confirmed through previous experimental studies conducted by Ivanov *et al.* [23], which provided a basis for comparison with numerical results. This highlighted the strong correlation between theoretical and experimental data, demonstrating the reliability of the proposed methodology.

The concept of similarity plays a key role in the design and synthesis of epicyclic chains within this model. It enables grouping and classification of kinematic schemes, reducing the number of unique configurations that require analysis. The application of similarity as a fundamental principle in synthesis eliminates duplication and accelerates the design process, while simultaneously facilitating parameter optimization for balanced load distribution and stability of dynamic characteristics.

The integration of the model into CAD environments created opportunities for automated generation of parametric gear models. This approach significantly simplified the design and simulation processes, allowing for rapid and efficient adaptation to different applications. Using specialized simulation tools, such as SolidWorks, numerical analyses were conducted, further optimizing the mechanism's design (Litvin *et al.* [21]; Wang *et al.* [22]).

The effectiveness of the proposed synthesis method for epicyclic chains was validated through previous experimental studies, detailed in (Ivanov *et al.* [23]). These studies focused on the impact of external forces on load distribution in full planetary gear engagement systems.

A custom-designed experimental test rig with a closed-loop power flow system was used to apply external loads parallel to the applied torque. Strain gauge sensors were employed to collect stress data, analyzing various loading scenarios and carrier positions. The bending stresses in the planet pin connections were calculated, along with the load sharing factor (LSF). The findings demonstrated the reliability and stability of the mechanism under different operating conditions, providing graphical analysis of load distribution.

The current research builds upon and extends these experimental studies, utilizing their results for further mechanism optimization. Through numerical analyses and simulation methods, an in-depth evaluation of the kinematic and dynamic characteristics of the system was conducted, ensuring a better understanding and enhanced efficiency of the mechanism under various operating conditions.

The discussion emphasizes the importance of a systematic approach in engineering synthesis and validation of epicyclic chains. These results lay the groundwork for future developments and the expansion of the model's applicability in various industrial sectors. The integration of the similarity concept and algorithmic approach in this model provides flexibility and efficiency, distinguishing it from traditional approaches. These principles and innovations open new possibilities for the application of epicyclic mechanisms in modern engineering systems.

9. Conclusion

The study highlights the key advantages of the mutually meshed satellite mechanism, which include uniform load distribution, minimal deformations, and resistance to vibrations. These characteristics make the mechanism highly suitable for industrial applications with strict requirements for reliability, efficiency, and durability. Its advantages position it as a reliable solution for industries where operational efficiency and system longevity are of critical importance.

The design and analysis of the EGM mechanism encompass all aspects of its structure and functionality, relying on innovative engineering approaches. The EGM construction is fully developed, incorporating all key components that have been designed and optimized in accordance with theoretical and practical requirements. The mechanism successfully performs all intended motions, demonstrating high functionality and achieving optimal output parameters, as outlined in the previous analyses.

The simulation analysis, conducted using the finite element method (FEM), examines the mechanism under various input torque values exceeding standard operational parameters. This ensures reliable and stable performance, even when using more powerful motors or in applications requiring higher output parameters. The study achieved the following results:

Results at different input torque levels:

- *50 Nm*: *315.233 MPa*, maximum displacement: *0.014 mm*;
- *100 Nm*: *348.912 MPa*, maximum displacement: *0.014 mm*;
- *150 Nm*: *413.091 MPa*, maximum displacement: *0.014 mm*.

The analysis reveals that even with a twofold increase in input torque, stress levels remain within permissible limits. This underscores the structural robustness of the mechanism and its ability to withstand high loads without compromising functionality or stability. The low values of maximum displacements further confirm the high precision and resilience of the system.

An additional indicator of the mechanism's quality is its low irregularity factor, which should be further validated through real-world testing using appropriate measuring equipment. These tests will provide additional insights into the mechanism's behavior in industrial applications.

The EGM mechanism exhibits exceptional stability, reliability, and efficiency, making it an ideal solution for industries such as automotive engineering, energy systems, and aerospace technology. The findings from this study provide a solid foundation for further optimization and integration of the mechanism into various industrial systems. This highlights its potential for application in modern high-tech manufacturing and operational environments.

Acknowledgements

The author/s would like to thank the Research and Development Sector at the Technical University of Sofia for the financial support.

Nomenclature

DGM1 and *DGM2*	– differential gear mechanisms
EGM	– epicyclic gear mechanism
OGM	– ordinary gear mechanism
PGM	– planetary gear mechanism

References

[1] He Z., Lin T., Luo T., Deng T. and Hu Q. (2013): *Parametric modeling and contact analysis of helical gears with modifications*.– Journal of Mechanical Science and Technology, vol.30, No.11, pp.4859-4867.

[2] Fuentes A., Nagamoto H. and Litvin F. L. (2010): *Computerized design of modified helical gears finished by plunge shaving*.– Computer Methods in Applied Mechanics and Engineering, vol.199, No.25-28, pp.1677-1690.

[3] Souza M.B.D., Vieira R.D.S. and Martins D. (2017): *Enumeration of kinematic chains with zero variety for epicyclic gear trains with one and two degrees of freedom*.– International Symposium on Multibody Systems and Mechatronics, Springer, Cham, vol.54, pp.15-24.

[4] Shanmukhasundaram V.R., Rao Y.V.D. and Regalla S.P. (2019): *Structural synthesis and classification of epicyclic gear trains: an acyclic graph-based approach*.– Mechanism and Machine Theory, vol.134, pp.393-424.

[5] Yang W., Ding H., Zi B. and Zhang D. (2018): *New graph representation for planetary gear trains*.– Journal of Mechanical Design, vol.140, No.1, pp.1-10.

[6] Pennestrì E. and Valentini P.P. (2015): *Kinematics and enumeration of combined harmonic drive gearing*.– Journal of Mechanical Design, vol.137, No.12, pp.1-11.

[7] Pennestrì E. and Valentini P.P. (2002): *Dynamic analysis of epicyclic gear trains by means of computer algebra*.– Multibody System Dynamics, vol.7, No.3, pp.249-264.

[8] Kahraman A. and Kharazi A.A.M. (2003): *A deformable body dynamic analysis of planetary gears with thin rims*.– *Journal of Sound and Vibration*, vol.262, pp.752-768.

[9] Yang W. and Ding H. (2018): *Automatic detection of degenerate planetary gear trains with different degree of freedoms*.– *Applied Mathematical Modelling*, vol.64, pp.320-332.

[10] Wang Y., Liu Y., Tang W. and Liu P. (2017): *Parametric finite element modeling and tooth contact analysis of spur and helical gears including profile and lead modifications*.– Engineering Computations, vol.34, No.8, pp.2877-2898.

[11] Yang W.J. and Ding H.F. (2018): *The complete set of one-degree-of-freedom planetary gear trains with up to nine links*.– Journal of Mechanical Design, vol.141, No.4, pp.1-22.

[12] Wu Y.J., Wang J.J. and Han Q.K. (2011): *Tooth profile modification of helical gears and experimental study based on finite element contact analysis*.– Journal of Aerospace Power, vol.26, No.2, pp.409-415.

[13] Yan P., Zhong Y.F. and Zhou J. (2003): *A systematic method for the design and evaluation of the output characteristics of epicyclic gear mechanisms*.– Mechanism and Machine Theory, vol.38, No.1, pp.43-52.

[14] Ye Z., Huang X., Liang S., Wu C. and Cui R. (2024): *A synthesis method for epicyclic chains based on new topological model and similarity*.– Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science, vol.238, No.15, pp.7463-7477.

[15] Gao A., Li J., Zhang Y., Zhou M. and Wang K. (2014): *Finite element analysis and optimization of meshing spur gears in a mining reducer*.– Hoisting and Conveying Machinery, vol.4, pp.83-85.

[16] Hsu C.H. and Lam K.T. (1992): *A new graph representation for the automatic kinematic analysis of planetary spur-gear trains*.– ASME Journal of Mechanical Design, vol.114, No.1, pp.196-200.

[17] Hsu R.H. and Fong Z.H. (2010): *Analysis of auxiliary crowning in parallel gear shaving*.– Mechanism and Machine Theory, vol.45, No.9, pp.1298-1313.

[18] Chen D.Z., Shieh W.B. and Yeh Y.C. (2008): *Kinematic characteristics and classification of geared mechanisms using the concept of kinematic fractionation*.– Journal of Mechanical Design, vol.130, No.8, pp.1-7.

[19] Hsu C.H. and Hsu J.J. (1997): *An efficient methodology for the structural synthesis of geared kinematic chains*.– Mechanism and Machine Theory, vol.32, No.8, pp.957-973.

[20] Cheng B.K., Fang T.T. and Li C.Y. (2008): *Gear geometry of cycloid drives*.– Science in China, Series E: Technological Sciences, vol.51, No.5, pp.598-610.

[21] Litvin F.L., Fuentes A. and Gonzalez-Perez I. (2003): *Modified involute helical gears: computerized design, simulation of meshing and stress analysis*.– Computer Methods in Applied Mechanics and Engineering, vol.192, No.33-34, pp.3619-3655.

[22] Wang S.Y., Yu T. and Sun J.Y. (2005): *Analysis of profiling plunge shaving cutter on shaving machine*.– Chinese Journal of Mechanical Engineering, vol.41, No.4, pp.163-167.

[23] Ivanov V., Aleksandrov A., Tsonev V., Kuzmanov N., Troha S. and Dimitrov L. (2022): *The effect of external forces on the load sharing of a full planet engagement planetary gear train*.– 2022 International Conference on Communications, Information, Electronic and Energy Systems (CIEES), Veliko Tarnovo, Bulgaria, pp.1-6.

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