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Stress-strain analysis of cycloidal gear tooth segment using FEM

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In robotics and industrial automation, cycloidal reducers are widely used due to their high transmission ratios, compact design, and low backlash. However, their specific kinematics and multiple simultaneous contacts result in complex stress and deformation states, particularly in the cycloidal gear-tooth region. Most previous studies analyzed the stress–strain state only at isolated angular positions of the cycloidal gear, while the behavior of a single tooth during the complete meshing cycle has remained insufficiently investigated. For this reason, this paper presents a combined analytical–numerical approach based on the finite element method (FEA) for the analysis of a single cycloidal gear tooth throughout the entire meshing cycle with the rollers of the ring gear. The results showed that higher Von Mises stress values occur in contacts between convex and concave surfaces than in contacts between two convex surfaces. In addition, the analysis enabled the identification of the most critical angular positions during operation and provided a more detailed insight into the evolution of stress distribution during meshing.

1. Introduction

High positioning accuracy, low backlash, compact design, and reliable operation under dynamic loads are among the main requirements imposed on modern power transmissions used in robotics and industrial automation. Due to their high transmission ratios and high-power density, cycloidal reducers are widely used in industrial robots, CNC machines, and automated systems for transport, storage,

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and material handling. However, their specific kinematics and multiple simultaneous contacts between the transmission elements lead to the occurrence of complex stresses and deformations, particularly in the region of the cycloidal gear.

To analyze the stress–strain state of cycloidal reducer elements, various software tools based on the Finite Element Method (FEA) are most commonly used, such as *Ansys*, *Abaqus*, *Adams*, *Algor*, *COSMOS*, and *MSC Patran/Nastran*. Among the first researchers who analyzed the stress–strain state and contact forces on cycloidal gear teeth with modified and unmodified profiles were Ishida [1] and Chmurawa [2]. Thube and Bobak [3] identified critical locations on vital elements of the cycloidal reducer from the perspective of load capacity; Gao et al. [4] conducted a stress–strain analysis of an optimized model, including the influence of friction and contact forces; Tariq et al. [5] carried out a comparative analysis of stress distribution in cycloidal and involute gearing; Bao et al. [6] reduced contact stresses on the cycloidal gear through optimization of geometric parameters such as the eccentricity size and the radius of the central rollers; He [7] presented a quasi-static model for determining the load distribution in a cycloidal gear; while Qi et al. [8] developed a numerical–analytical method that includes the analysis of contact forces and the stress–strain state of cycloidal reducers.

For precise modeling of the stress-strain state of cycloidal reducer elements, it is essential to determine the load intensity and type. Contact loads, as the primary input parameter in such analyses, have traditionally been determined using analytical models, while more recent studies also employ numerical methods. Among the most well-known analytical approaches are the models of Kudryavtsev [9] and Lehmann [10], which are based on simplified assumptions, such as the theoretical meshing case in which all rollers (ring gear and output) are simultaneously in contact with the cycloidal gear, while only half of them transmit the load. However, if a more realistic estimate of the number of rollers actually participating in load transmission is made, the results of these models can be aligned with those of more complex calculations that explicitly account for tooth profile modifications, contact deformations, and internal clearances between the elements of the cycloidal reducer, as presented by Zhang et al. [11], Li et al. [12], Zhang et al. [13], and Efremkov et al. [14].

Regarding numerical determination of contact loads, relatively few models have been defined to date. Vasić [15], together with collaborators, developed a numerical–analytical methodology capable of predicting the distribution of temperature fields and quantifying their influence on contact forces; Wu et al. [16] developed a model for determining contact loads, based on which tooth profile corrections of the cycloidal gear were performed; Wang [17] presented a numerical–analytical method for calculating contact forces and stresses, including an evaluation of cycloidal gear tooth wear. Wu et al. [18] proposed a multi-factor coupled optimization approach for cycloidal gear profile modification, considering contact stress, meshing stiffness, transmission accuracy, and meshing range simultaneously. Xu et al. [19] developed a generalized dynamic contact model

for bearing-cycloid-pinwheel transmission mechanisms used in RV reducers, including multi-tooth contact analysis and internal load transmission characteristics under dynamic operating conditions. Wang et al. [20] developed a multi-tooth contact model for RV reducers by dividing the contact zone into differential contact elements, enabling a more accurate analysis of contact forces, transmission error, contact ratio, and transmission efficiency under different profile modification methods and load conditions. Hsieh [21] investigated the influence of transmission design on the dynamic response and stress behavior of eccentric speed reducers using a system dynamics and FEM-based approach. Gao et al. [22] investigated the service lifespan of RV reducers through fatigue-strength-based modeling, ANSYS simulations, and accelerated experimental testing, emphasizing the influence of crankshaft bearing fatigue on the overall reducer durability.

The stress-strain state of key cycloidal reducer elements can also be assessed experimentally, using appropriate testing equipment that includes force sensors and strain gauges. Blagojević [23, 24] carried out experimental tests under static loading conditions and compared them with numerically obtained results, while Matejić [25] conducted similar studies under dynamic conditions. The results of these investigations demonstrated that FEA-based software, with well-defined boundary conditions, can provide a fairly realistic assessment of the stress-strain state.

Although significant progress has been achieved in the analysis of cycloidal reducers, most previous studies mainly considered isolated angular positions of the cycloidal gear or simplified contact conditions. In addition, the majority of available investigations focused on global load distribution and overall stress levels, while the evolution of the stress-strain state of a single tooth during the complete meshing cycle has remained insufficiently investigated. For this reason, the aim of this paper is to perform a detailed stress-strain analysis of a single cycloidal gear tooth throughout the entire meshing cycle with the rollers of the ring gear. A combined analytical-numerical approach based on the finite element method (FEA) was applied in order to determine the variation of contact stresses and displacements, as well as to identify the most critical angular positions during operation.

2. Force determining methodology

To determine the stress-strain state of the cycloidal gear, it is necessary to know the intensity and type of load. For this reason, the analysis begins with the consideration of the input torque.

When the input shaft of the cycloidal reducer is subjected to the driving torque T_{in} , the cycloidal gear, which meshes with the rollers of the ring gear, begins to perform a complex motion, both eccentric and rotational.

The transmission ratio is determined by the following equation:

$$u_{cr} = \frac{z_1}{z_2 - z_1}, \quad (1)$$

where: z_1 – number of teeth of the cycloidal gear; z_2 – number of rollers of the ring gear.

The complex motion of the cycloidal gear is then transmitted to the output rollers that pass through the circular holes in the cycloidal gear, with the output roller carrier rigidly connected to the output shaft.

If the torque on the ring gear, originating from both cycloidal gears, is denoted as $2T_2$, and the torque on the output roller carrier, which is rigidly connected to the output shaft, as T_{ou} , the relationship between the torques can be expressed by the following equation:

$$T_{in} - 2T_2 + T_{ou} = 0. \quad (2)$$

If the torques acting on only one cycloidal gear are considered, and mechanical power losses are neglected ($\eta_{CR} \approx 1$), the torque equilibrium equation can be written in the following form:

$$T_1 - T_2 + T_3 = 0, \quad (3)$$

where: T_1 – driving torque of one cycloidal gear, $T_1 = T_{in}/2$; T_2 – torque on the ring gear originating from one cycloidal gear; T_3 – output torque originating from one cycloidal gear, $T_3 = T_{ou}/2$.

The action of torques on the cycloidal reducer components results in three contact forces—reactions, namely:

- contact force F_{Ni} at the current contact surface between the cycloidal gear tooth and the rollers of the ring gear (normal contact force).
- contact force F_{Ki} at the current contact surface between the output rollers and the holes in the cycloidal gear (output contact force).
- eccentric force F_E considered as a concentrated force acting at the centre of the cycloidal gear C_1 forming an angle ε with x -axis.

The distribution of these contact forces on one cycloidal gear at the driving angle 0° is shown in Fig. 1. On the other cycloidal gear, which is not shown and is rotated by 180° relative to the first, the same contact forces act (also rotated by 180°); therefore, in further analysis, it is sufficient to consider only one cycloidal gear.

Since the values of contact forces depend on the angular displacement of the cycloidal gear β , it is crucial to determine which rollers of the ring gear are in contact at a given moment and transmit the corresponding load. The eccentricity direction plays an important role in identifying the rollers currently engaged in load transmission. This direction divides the cycloidal reducer into two zones – one containing rollers transmitting load in contact with cycloidal gear 1, and the other containing rollers transmitting load in contact with cycloidal gear 2.

However, not all rollers participate in load transmission simultaneously; only a certain number are engaged. This selective distribution results from internal clearances, which serve multiple purposes: compensating for manufacturing errors, enabling proper assembly and disassembly, and ensuring conditions for effective lubrication.

To account for the existence of internal clearances, this paper adopts the widely accepted assumption that one-third of the rollers participate in load transmission

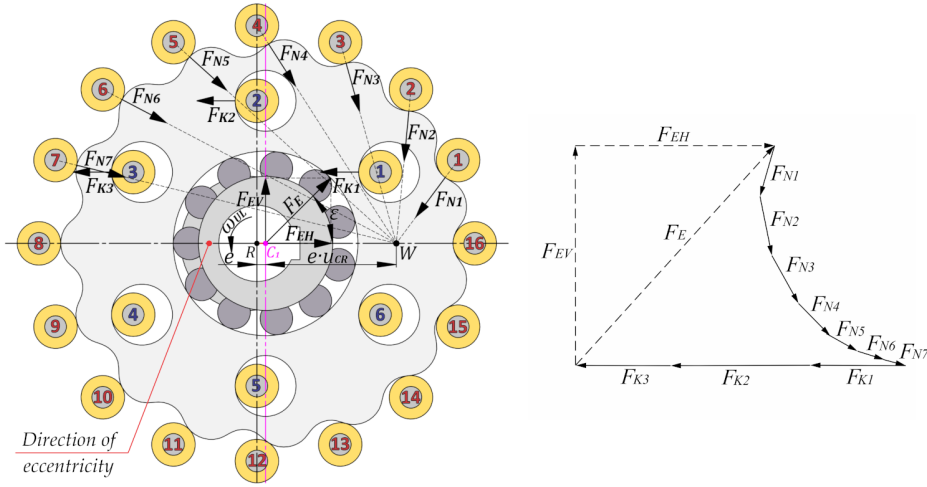


Fig. 1. Distribution of contact forces on a single cycloidal gear at driving angle $\beta = 0^\circ$

for each cycloidal gear [18]. Consequently, the rollers closest to the eccentricity direction are excluded from the load transmission process.

To calculate the values of the normal contact forces F_{Ni} at the current contact surface between the cycloidal gear teeth and the loaded rollers of the ring gear, the Lehmann model [10] was used (this model is described in detail in [18], and only the basic equation is presented here):

$$F_{Ni}(\beta) = (c_N \cdot \Delta\beta) \cdot r_{Ni}(\beta) \cdot \sin \psi_{Ni}(\beta), \quad (4)$$

where: β – angular displacement of the cycloidal gear; c_N – stiffness of the ring gear rollers; $\Delta\beta$ – small angular displacement of the cycloidal gear; $r_{Ni}(\beta)$ – distance between the contact line of the i -th central roller and the central axis of the cycloidal gear; $\psi_{Ni}(\beta)$ – angle between the direction of the normal contact force of the i -th central roller and the line $r_{Ni}(\beta)$ connecting the contact line with the central axis of the cycloidal gear.

It should be noted that the applied Lehmann model is based on quasi-static assumptions and does not include velocity-dependent or inertial effects. Therefore, phenomena such as vibrations and time-varying dynamic loads were not considered in the present analysis. The contact forces were determined exclusively based on contact geometry, local deformations, and variable contact stiffness.

3. Numerical analysis assumptions

The stress-strain state of the cycloidal gear was analyzed using the Finite Element Method (FEA) in the commercial software SolidWorks 2017. The selected software provides an integrated environment for geometric modeling, mesh generation, boundary condition definition, load application, and finite element analysis

within a single platform. This approach simplifies the simulation procedure, improves consistency between the geometric and numerical models, enables easier modification of the model during different stages of the analysis, and reduces the possibility of errors during data transfer between different software packages.

Since the main objective of this study is to investigate the behavior of a single cycloidal gear tooth throughout the entire meshing cycle with the rollers of the ring gear, the process was discretized into 26 precisely defined angular positions, with an angular step of 2° between two consecutive positions. In this way, a separate numerical analysis was carried out for each angular position of the cycloidal gear, and the obtained results were compared with one another to identify the critical positions.

All relevant settings used in the performed numerical analyses are presented in the following subsection.

3.1. Geometric model

The stress–strain analysis was carried out on a cycloidal gear that is a component of a single-stage cycloidal reducer of conventional design, with a nominal power of 5.5 kW and a transmission ratio of 11. Its basic geometric parameters are presented in Table 1.

Table 1. Basic geometric parameters of the analyzed cycloidal gear

Parameter	Desc.	Unit	Value
Number of cycloidal gear teeth	z_1	–	11
Number of ring gear rollers	z_2	–	12
Eccentricity	e	mm	5
Radius of ring gear roller	q	mm	8
Pitch circle radius of ring gear	r	mm	100
Radius of rolling (moving) circle	R_a	mm	8.33
Radius of base (fixed) circle	R_b	mm	91.67
Cycloidal gear width	b_{cz}	mm	13
Diameter of output roller hole in cycloidal gear	D_{ocz}	mm	42
Number of output rollers	u	–	6

For the material of the cycloidal gear, quenched-and-tempered steel 42CrMo4 was used, and its basic mechanical properties are shown in Table 2.

Since the profile of the cycloidal gear tooth is most often an equidistant of a shortened epitrochoid, convex and concave surfaces can be distinguished on the tooth profile. On the other hand, the central rollers are convex surfaces, meaning that in the meshing process, contact occurs either between two convex surfaces or between convex and concave surfaces.

Table 2. Basic mechanical properties of the cycloidal gear material (42CrMo4)

Property	Value	Units
Elastic Modulus, E	210000	N/mm ²
Poisson's Ratio, ν	0.28	N/A
Shear Modulus, G	79000	N/mm ²
Mass Density, ρ	7800	kg/m ³
Tensile Strength, R_m	1000	N/mm ²
Yield Strength, R_e	750	N/mm ²

To enable a reliable assessment of the stress–strain state under these meshing conditions, appropriate surfaces with dimensions of 0.5×13 mm were formed on the analyzed tooth flanks of the cycloidal gear to simulate the contact between the cycloidal gear tooth and the ring gear roller. These surfaces were used for the application of the normal contact load in the subsequent stages of the analysis.

3.2. Finite elements mesh

The mesh density was chosen to provide an optimal balance between calculation accuracy and analysis time. Accordingly, a curvature-based tetrahedral mesh was used, allowing better adaptation to the complex geometry of the cycloidal gear. To verify the adequacy of the selected mesh density, a convergence study was performed using three mesh refinement levels, and the results are presented in Table 3. The maximum von Mises stress obtained with a 4 mm mesh was 116 MPa, while the 3 mm mesh yielded a value of 114.6 MPa, corresponding to a difference of 1.21%. Based on these results, a maximum element size of 4 mm was adopted for all analyses, as it provided sufficiently accurate results while reducing computational time compared to finer mesh configurations. The final mesh consisted of 207 891 nodes and 134 801 interconnected finite elements. To improve accuracy in the contact regions, local mesh refinement was applied, with the minimum element size reduced to 0.8 mm.

Table 3. Mesh convergence study results

Analysis	Max. element size (mm)	Min. element size (mm)	Total elements	Max von Mises (MPa)	Change (%)
1	5	1	71 350	72.85	–
2	4	0.8	134 801	116	59.23
3	3	0.6	295 354	114.6	1.21

3.3. Constraints and boundary conditions

To ensure that the contact conditions correspond to real operating conditions, the normal contact force $F_{Ni}(\beta)$ (Fig. 2) was applied to the current contact surface between the cycloidal gear tooth and the ring gear roller. The magnitude of this force was determined analytically, based on the expressions presented in Section 2, while its direction was defined as the line connecting the center of the ring gear roller with the instantaneous meshing pole W .

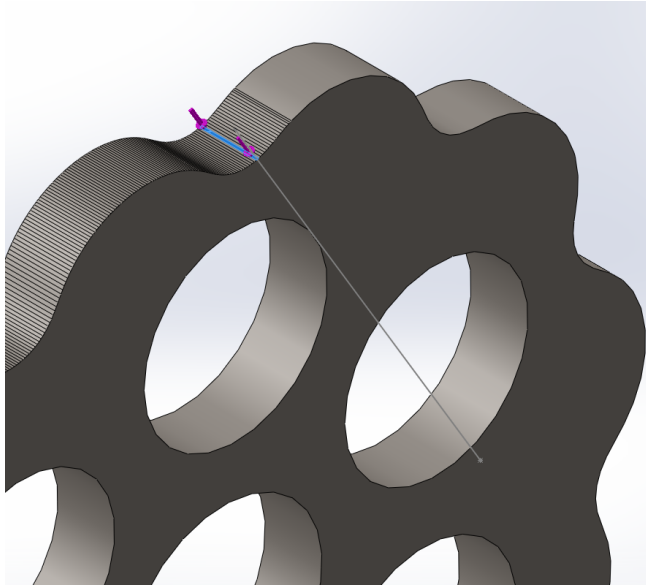


Fig. 2. Load on the analyzed tooth flank

It should be emphasized that the position of the center of the ring gear roller remains constant during reducer operation, while the position of the instantaneous meshing pole W varies and directly depends on the angular displacement of the cycloidal gear β . Accordingly, its coordinates (x_w, y_w) can be determined using the following equations:

$$x_w(\beta) = (z_1 + 1) \cdot e \cdot \cos \beta, \quad (5)$$

$$y_w(\beta) = (z_1 + 1) \cdot e \cdot \sin \beta, \quad (6)$$

where: z_1 – number of cycloidal gear teeth, e – eccentricity, β – angular displacement of the cycloidal gear.

In the next stage of the analysis, the boundary conditions and contact interactions were defined. The input shaft, rigidly connected to the eccentric sleeve, and the output shaft were fixed in order to provide the necessary reaction forces in the static analysis. A cylindrical face constraint was applied to the inner surface

of the cycloidal gear bore, allowing free rotation of the cycloidal gear around the eccentric sleeve. In addition, an auxiliary constraint was introduced on the rear face of the cycloidal gear to prevent axial displacement. To ensure realistic load transfer, no-penetration contacts were defined between the cycloidal gear and the eccentric sleeve, as well as between the cycloidal gear and the output rollers. This boundary-condition setup more accurately represents the operating conditions of the cycloidal reducer and enables reliable evaluation of the stress–strain state of the cycloidal gear tooth during the complete meshing cycle. Fig. 3 shows the final numerical model with the defined boundary conditions, contact interactions, load distribution, and finite element mesh.

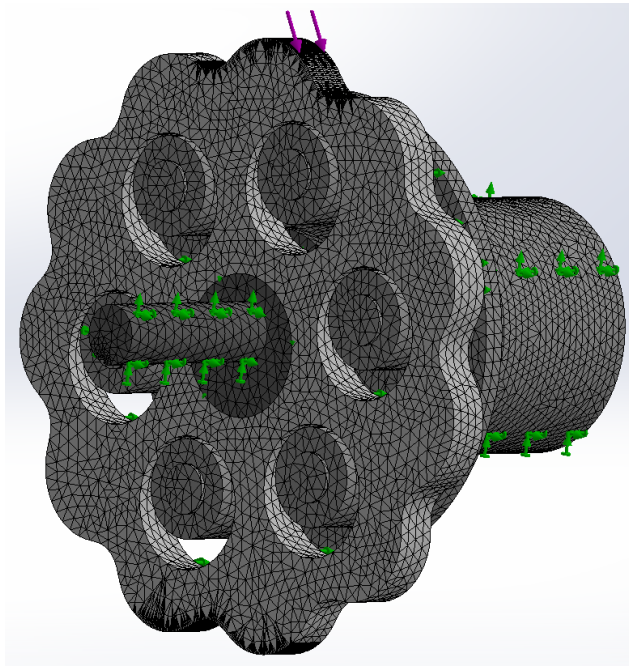


Fig. 3. Final model used in one of the analyses, with defined boundary conditions, contact interactions, load distribution, and applied finite element mesh

This setup enabled a reliable assessment of the tooth flank's behavior under load across different meshing positions.

4. Results analysis

4.1. Results display

The variation of contact force with cycloidal gear angular position throughout the analyzed meshing cycle is shown in Fig. 4. It can be observed that the calculated contact force increases up to the cycloidal gear rotation angle $\beta = 19^\circ$, after which

it gradually decreases. This variation is caused by the shifting of the contact zone along the tooth profile of the cycloidal gear, which influences the load distribution in different meshing positions.

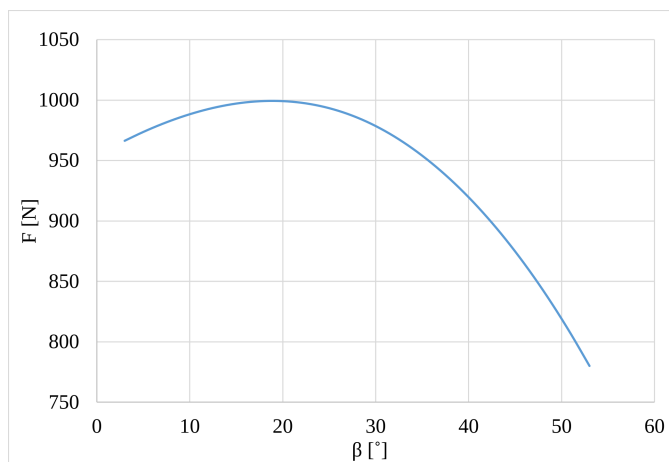


Fig. 4. Variation of contact force with cycloidal gear angular position

Based on the defined coordinates of the instantaneous meshing pole and the corresponding contact forces, the values of maximum Von Mises stresses and total displacements were determined for each analyzed position. In this way, the entire meshing cycle of a single tooth flank was covered, allowing assessment of the variation of load and deformations in the cycloidal gear. The Von Mises stress results are shown diagrammatically in Fig. 5, while the results of total displacements are shown in Fig. 6. In both cases, variations were recorded as a consequence of the shift in the contact location between the cycloidal gear tooth and the ring gear

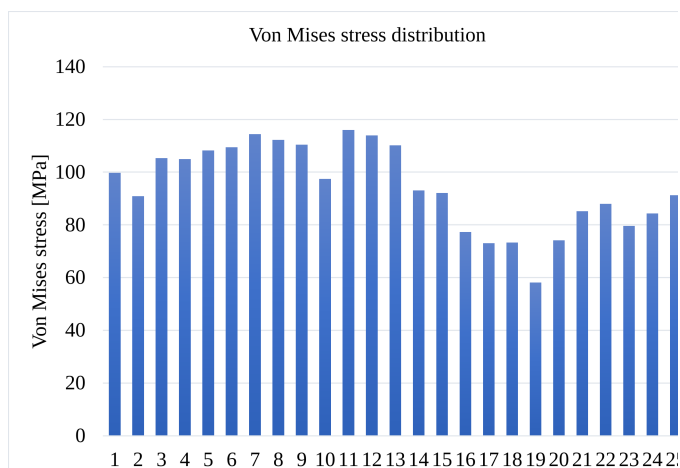


Fig. 5. Distribution of Von Mises stresses (1–26)

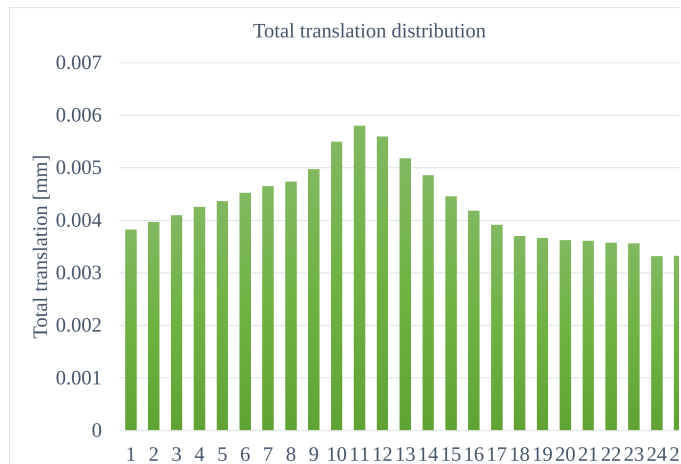


Fig. 6. Distribution of total displacements (1–26)

roller. In Figs. 5 and 6, the X-axis represents positions 1–26, corresponding to the successive meshing positions of a single tooth flank.

Fig. 7 shows the distribution of Von Mises stresses for analysis position 11, corresponding to the cycloidal gear angular position $\beta = 23^\circ$, where the maximum stress value of 116 MPa was recorded. The highest stress concentrations occur in the contact zone between the cycloidal gear tooth and the ring gear roller due to localized load transfer over a relatively small effective contact area. As the contact zone shifts along the tooth profile during meshing, the local contact stiffness and load distribution continuously change, resulting in non-uniform stress evolution throughout the operating cycle. The stress values in the remaining regions of the

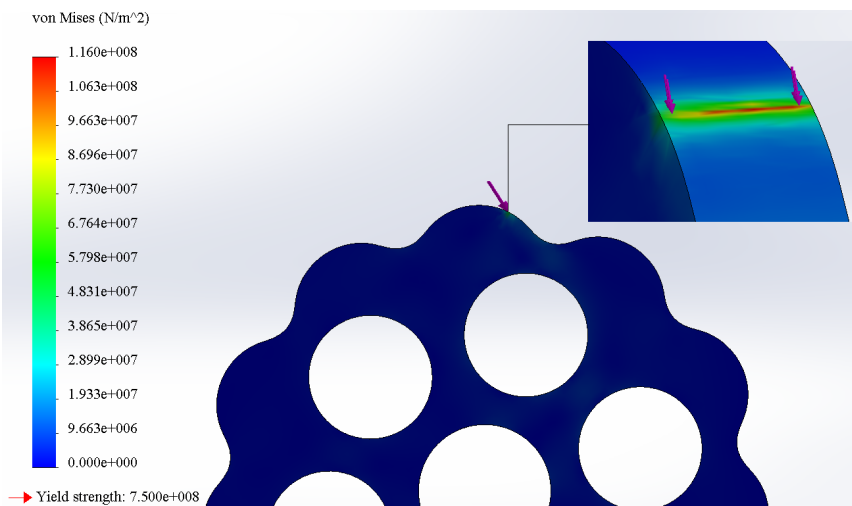


Fig. 7. Distribution of maximum Von Mises stresses on the cycloidal gear tooth

cycloidal gear are significantly lower, indicating that the contact region represents the dominant load-carrying zone. The purple arrow indicates the location of the applied contact force in the simulation.

Fig. 8 shows the distribution of total displacements for the same analysis position, corresponding to $\beta = 23^\circ$, where the maximum total displacement reached 0.005804 mm. The largest displacements occur in the contact region between the cycloidal gear tooth and the ring gear roller, where local compliance and contact deformation are most pronounced. As the meshing position changes during operation, the location and magnitude of the displacement field vary due to changes in load transfer conditions. The displacement distribution confirms that the deformation behavior of the cycloidal gear is strongly influenced by the instantaneous meshing position and the corresponding contact load.

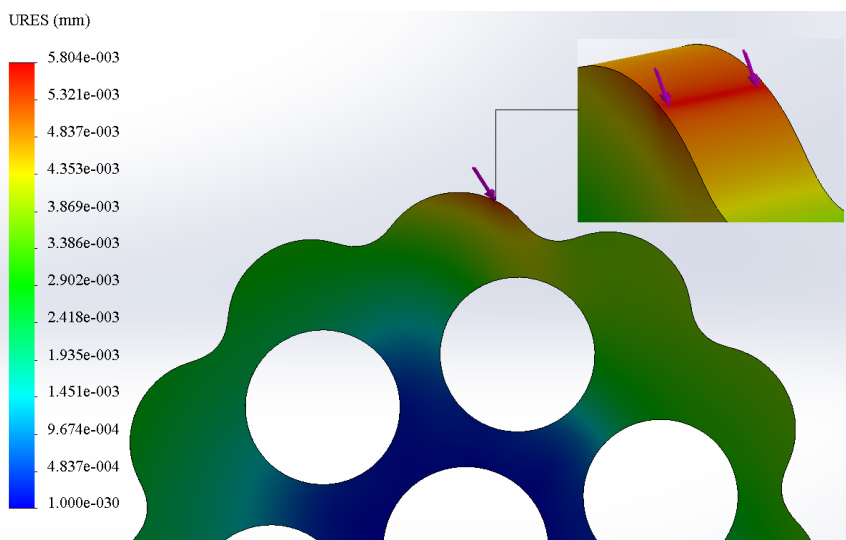


Fig. 8. Distribution of maximum total displacements on the cycloidal gear tooth

The conducted analysis confirmed that the highest stress and displacement values occur in the contact region between the cycloidal gear tooth and the ring gear roller, while the values in other regions of the cycloidal gear are considerably lower. The obtained results indicate that the contact zone represents the most critical region from the standpoint of stress concentration and deformation behavior under contact loading.

4.2. Comparison with literature

To confirm the validity of the obtained results, a comparison was made with available studies from the literature. Since no studies employing an identical modeling approach were identified, works involving numerical and experimental inves-

tigations of the stress–strain state of cycloidal gears under static or dynamic loading conditions were selected for comparison.

Blagojević [24] carried out a stress–strain analysis of a cycloidal gear for the case of single meshing, where a concentrated force was applied at the contact between the cycloidal gear tooth and the ring gear roller. The reported difference between numerical and experimental values along the tooth flank did not exceed 7.3%.

Matejić [25] investigated the dynamic behavior of a cycloidal reducer by combining experimental measurements using strain gauges with numerical simulations for single, double, and triple-meshing cases. The deviation between numerical and experimental results at the position of maximum stress ranged from 3% to 15%. Load levels corresponding to 50%, 100%, and 150% of the maximum contact force were analyzed, while the comparison presented in this study is based on the results obtained for the highest loading case.

Fig. 9 presents the distribution of Von Mises stresses along the tooth flank. The X-axis is shown as a relative dimensionless coordinate in the interval 0–1, enabling comparison with previously published numerical and experimental studies. Compared to the results reported by Blagojević [24] and Matejić [25], the stress values obtained in the present study are generally higher, particularly in the contact region.

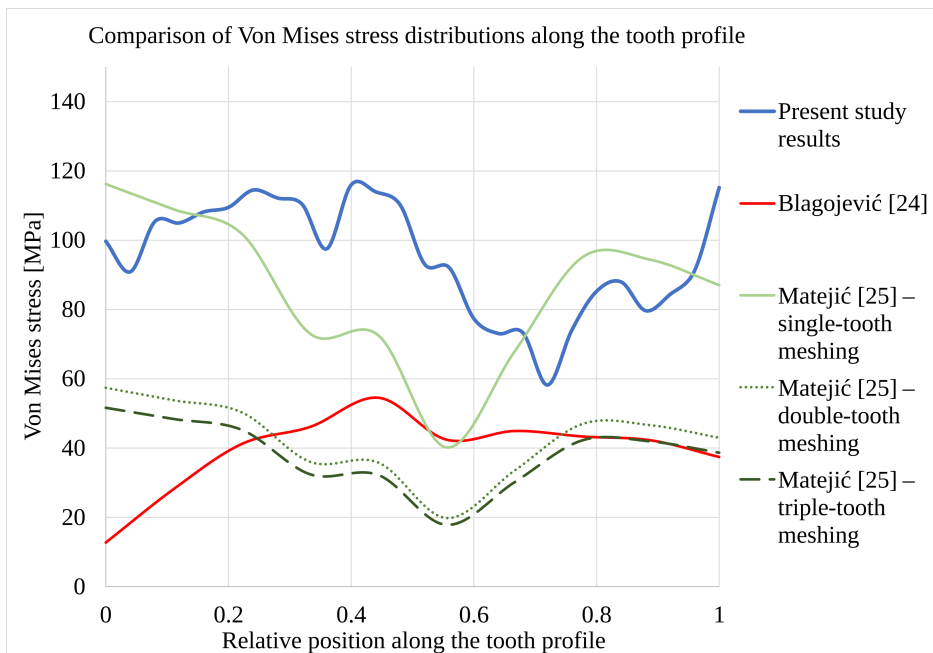


Fig. 9. Comparison of Von Mises stress distribution along the cycloidal gear tooth profile: results of this study, Blagojević [24], and Matejić [25] for single, double, and triple meshing

The observed differences in stress distribution can be attributed to the applied three-dimensional numerical model, revised boundary conditions that allow rotation of the cycloidal gear around the eccentric sleeve, and the use of distributed contact loading over finite contact surfaces, which enabled a more detailed representation of the contact conditions during meshing. In addition, the obtained stress trends are qualitatively consistent with previously reported analyses of RV reducers and eccentric cycloidal transmissions under different operating conditions [19–22]. Unlike many earlier studies that focused on isolated meshing positions or simplified loading conditions, the proposed approach enables a more comprehensive analysis of stress evolution throughout the complete meshing cycle and facilitates identification of the most critical regions along the cycloidal gear tooth profile.

5. Conclusion

A numerical analysis of the stress–strain state of a cycloidal gear was performed using the Finite Element Method, covering the complete meshing cycle of a single tooth flank. The conducted analysis enabled a detailed evaluation of the variation in contact stresses and displacements at different meshing positions between the cycloidal gear tooth and the ring gear roller. The obtained results showed that both the maximum Von Mises stress and the maximum total displacement occurred at the cycloidal gear angular position $\beta = 23^\circ$. The highest stress concentrations and displacements were identified in the contact region, confirming that this zone represents the most critical region from the standpoint of load transfer and deformation behavior during operation.

Comparison with previously published numerical and experimental investigations demonstrated qualitative agreement in the trends of the stress distribution. In contrast, the higher stress values observed in the present study are mainly due to the applied three-dimensional numerical model, more realistic boundary conditions, and distributed contact loading. Unlike many earlier investigations focused on isolated meshing positions, the proposed approach enabled analysis of stress evolution throughout the entire meshing cycle and identification of the most critical operating positions of the cycloidal gear.

Future research will focus on investigating different materials, corrected tooth profiles, and experimental validation to reduce stress concentrations and improve the operational reliability of cycloidal reducers.

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