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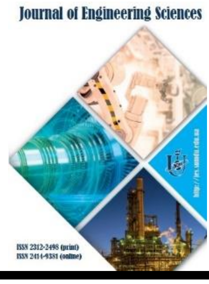

Design Optimization of the Modified Planetary Carrier

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Abstract. This paper aims to design a new model of the third-stage carrier assembly used in a planetary gearbox as a single part component with improved strength and fatigue life properties and lower production costs. First, the mounting carrier assembly is subjected to static, fatigue, and modal analysis, and based on obtained results, the operating conditions that ensure its trouble-free operation are proposed. In the next step, new designs of the carrier as a single piece component are proposed and subjected to similar analyses. The proper numerical analysis method is chosen to evaluate the fatigue life, total deformation, and von Misses stress for each new model. Based on these results, the best design is chosen and submitted to further improvement, ensuring a weight reduction of 5 %. This last model of the carrier assembly is the most optimal solution since the maximum deformation values decreased by more than 55 %, and the maximum von Misses stresses decreased by almost 38 %, which increased fatigue life. A more comprehensive range of operating conditions for the optimized carrier is proposed to ensure its suitability for use in each gearbox. The finite element method analysis is performed in ANSYS.

Keywords: planetary carrier, planetary gearbox, finite elements method, numerical analysis.

1 Introduction

Finite element method (FEM) analysis is a computer-based analytical tool used to simulate and analyze systems and products, especially those used in mechanical engineering. Its ability to replace physical tests and predict the material behavior more effectively based on the given input parameters has become an integral part of the production process. One of the many applications of FEM is the numerical analysis of gearboxes – design improvement, an increase of the fatigue life, damage prediction, etc. [1].

The main task of the gearboxes, in general, is the transformation of the mechanical energy and rotary motion between the driveshaft and the driven shaft. There are numerous advantages of the planetary gears compared to the parallel gears, especially in terms of their power density, relatively low weight, compact structure, kinematic flexibility, or self-centering ability [2].

Planetary gears are used in systems where high torque production is necessary because their construction enables the uniform distribution of load between several planet gears, which makes them also more resistant to damage. They are part of the systems and machines used in the automotive and aerospace industry, and in general, they are usually applied in the final stages of major transmissions [3].

The tasks are to perform numerical analysis of a mounted planetary carrier used as a crusher machine drive. The carrier assembly is subjected to the prescribed acting forces. After a detailed evaluation of the results, it will be possible to determine whether the carrier can transfer the prescribed load without failure. Both static and fatigue life analyses are performed in ANSYS.

In the next step, new carrier designs as a single part component are modeled and submitted to numerical analysis. Based on the obtained results, the most suitable variant is recommended to replace the mounted carrier. Subsequently, the primary operating conditions are defined based on the fatigue life and modal analysis.

2 Literature Review

Planetary gearboxes are the most complex types of gear arrangements. They are applied in systems where it is necessary to use gears with low weight or small dimensions because of the lack of space and reduce high speed and torque. This requirement applies to a wide range of industries, including turbine engines, tractors, automatic transmissions, construction equipment, and even electric screwdrivers [4].

These systems basically consist of 4 main parts (Figure 1): sun gear, carrier, planet gears, ring gear [3]:

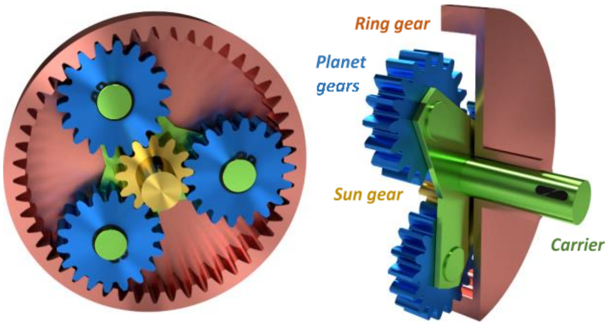


Figure 1 – Planetary gear parts

By fixing one of the coaxial elements, it is possible to achieve different gear ratios depending on which members are fixed. The remaining two elements are used as input and output, so we get three types of planetary gears summarized in Table 1 [5].

Table 1 – Planetary gear ratios determined for different fixed elements [5]

Input	Output	Fixed part	Calculation	Gear ratio
Sun gear	Carrier	Ring gear (*R*)	$1 + K/C$	3.4:1
Carrier	Ring gear	Sun gear (*S*)	$1/(1 + C/K)$	0.71:1
Sun gear	Ring gear	Carrier (*C*)	$-K/C$	–2.4:1

The third stage carrier analyzed in this paper is a part of a gearbox with an arrangement where the fixed member is the ring gear, the input is on the sun gear, and the output then passes through the carrier. This arrangement provides the highest gear ratio but based on other researches focusing on the analysis of a similar carrier assembly, the effect of high load causes the pins to protrude from the body of the carrier what leads to their failure, formation of the plastic deformation area and eventually cranks [6].

When designing a planetary gear, several conditions must be met to avoid interference between the ring and planet gears. Interference can result in unworkable conditions caused by the teeth of gears cutting into each other because of the insufficient number of planet gears teeth or inaccurate difference in the number of teeth between planet and ring gears [6].

$$\frac{z_1 + z_2}{x} = integer\ number\,; \tag{1}$$

$$z_2 = (z_1 + 2) \cdot z_3\,, \tag{2}$$

where z_1 represents the number of the sun gear teeth, z_2 is the number of the ring gear teeth, z_3 is the number of the planet gear teeth, and x is the number of planet gears [5].

A complete dynamic analysis of planetary gears is demanding even under ideal geometric conditions because of the singularities that occur at the points of contact between the teeth of the sun gear and planet gears as well as between the teeth of planet gears and the ring gear, leading to very different values of contact stress [7].

In the planetary gearbox, the torque is transmitted from the central wheel via satellites to the carrier and the main rotor shaft. One of the tools for assessing the condition of these components is vibration analysis. The vibrations of planetary gears are difficult to analyze. It is necessary to consider the influence of several factors, e.g., similar vibrations of the planet wheels, multiple and time-varying transmission paths from the transmission to the transducers, which are usually attached to the transmission housing. When these factors are combined, they reduce the efficiency of conventional fault detection algorithms, so these data sets have been adapted specifically for the case of planetary gearboxes and applied to the time-synchronous averages of the planetary carrier vibrations [8].

Today an increasing number of gear designers are using loaded tooth contact analysis (LTCA) methods to get precise information on the load distribution in both dimensions of the flank (lead and profile direction) on the entire gear flank. The use of the algorithm is a good solution to get proper values for the face load factor $K_{H\beta}$ of spur and helical gears, but it has also proved to be efficient when adapted for planetary systems [9].

The current trend is to minimize the weight and the cost, and since the structures become lighter, their flexibility is more and more critical. Static calculations of stresses in the wind gearbox have been widely performed with only rigid assumptions. Therefore it is necessary to update the calculation methods. A gear element is developed and tested in FEM software to be linked to flexible components. In typical methods, the gear points and the local loads are calculated separately, but with the gear elements, the input of the calculation is only the torque and the speed [10].

The present studies also address having more planet gears, which significantly increases the input torque density while using flexible pins. In this type of design, the pin stiffness and position tolerances are essential parameters as they affect the dynamic performances significantly. This issue is solved by modeling a double cantilevered flexible pin and analyzing the contributions of pin stiffness and misalignment applying the lumped parameter approach [11].

3 Research Methodology

The analyzed carrier assembly is a part of the planetary gearbox manufactured by the Slovak company. It is used as a part of a driving mechanism in the crusher machine with an axial distance of 650 mm between the crushing rollers. Two front-planet towers drive the gearbox while each of them has two hydraulic motors. The output transmission ratio is 1:1, so the torque is distributed from the planet gearbox equally between two output shafts connected to the crushing rollers mounted in a common bearing.

The carrier is made of low-alloy stainless chrome-molybdenum steel 42CrMo4+QT characterized by resistance to abrasive wear and medium shock of dynamic forces, higher hardenability for stressed machine parts and components such as shafts and couplings. After simplifying the geometry of the mounted planetary carrier and applying the appropriate adjustments, the resulting model has been subjected to analysis (Figure 2).

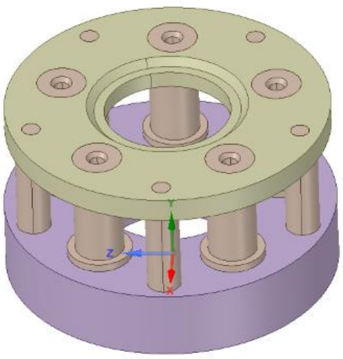


Figure 2 – Modification of the mounted planetary carrier geometry for static analysis

The automatic meshing method called MultiZone meshing was used for the model because of its complexity. A finer mesh was applied to the pins and other active load-bearing components using the mesh size definition function to achieve the most accurate results. This unevenly distributed mesh of the finite elements is shown in Figure 3.

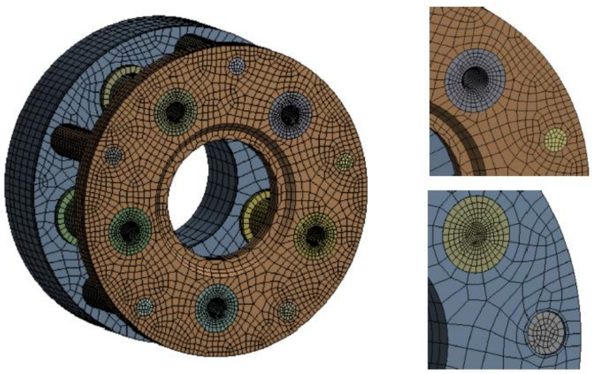


Figure 3 – Assembly model with finite element mesh

In the last step, the prescribed load was applied by a circumferential force perpendicular to each pin. This is the force transmitted to the carrier assembly via planet gears and pins from the sun gear.

4 Results

4.1 Numerical analysis of the mounted carrier assembly

We subjected the carrier assembly to static, modal, and fatigue life analysis to predict its service life under a given load.

The most considerable deformations reach a value of 0.73 mm at the outer edge of the carrier plate. The deformations on the main body of the carrier are compared to that much smaller as the main body is significantly stiffer. The deformation of the screw joint is approximately 20 % greater than the deformation of the pins, and they are both more deformed at the point of contact with the carrier plate (Figure 4).

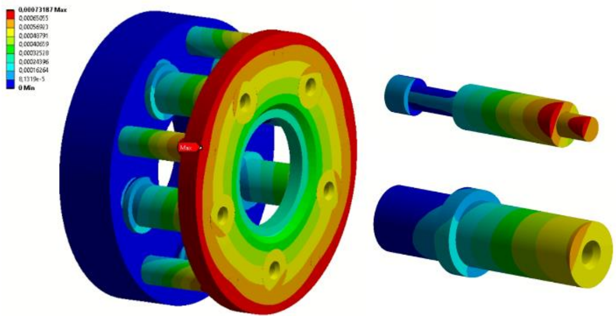


Figure 4 – Total deformation of the mounted carrier assembly

The stress is concentrated at the edge of the pinhole on the main body of the carrier, wherein practice the most frequently the zone of wear and subsequent failure occurs. (Figure 5). Since the pin is pressed into the main body of the carrier with a more extensive interference, the stress concentration is lower at the hole in the carrier plate. The maximum von Misses stress value exceeds the prescribed value of the yield stress for the given material by almost 4 % at the edge of the hole in the main body of the carrier. This is the place where the highest probability of failure due to fatigue processes is expected.

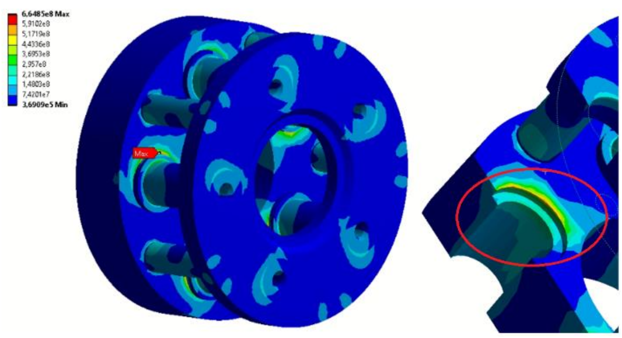
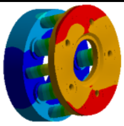
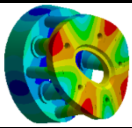
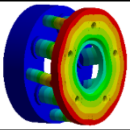
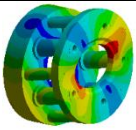
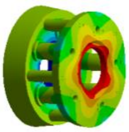
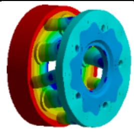


Figure 5 – Von Misses stress distribution on the mounted carrier assembly

The mounted carrier can be used in the crusher machines designed for 8 hours of continuous trouble-free daily operation. Based on a fatigue life analysis, the carrier is suitable for this basic operation. However, repeated switching on and off the crusher machine would lead to premature failure (formation of the plastic deformation areas, cracks, etc.).

In the last step, the natural frequencies, natural shapes of oscillations, and deformations are obtained through the modal analysis. The main task is to evaluate the results to determine whether the resonance may occur in the system in conjunction with the operating frequencies. The results of the modal analysis for the first six natural shapes are summarized in Table 2.

Table 2 – Modal parameters of the mounted carrier assembly

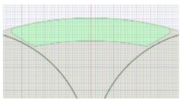
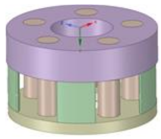
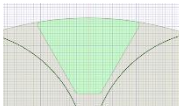
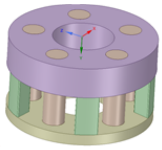
ω, Hz	Natural shapes	ω, Hz	Natural shapes
902		2118	
1091		2262	
2091		2738	

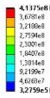
The input angular velocity is 70 Hz, and the output velocity is only 1.5 Hz, so based on the results, it is safe to say that resonance will not occur in this assembly because the mounted carrier is sufficiently rigid and characterized by high natural frequencies.

4.2 Design and numerical analysis of the carrier assembly as a single part component

Based on the evaluation of the graphical representation of the natural shapes, it can be assumed that the outer and inner edge of the carrier plate is prone to deformation because of their lower stiffness. These and several other aspects like the available information about the options for the carrier construction are considered when designing the model as a single part component. The main research objective is to improve strength properties, fatigue life and provide a broader operating condition. At the same time, there must be no significant increase in weight, which would lead to an increase in production costs. The material of the product does not change. Two new designs are created with the screw connections replaced by the supports at the outer edge of the carrier plate, while the cross-sectional area of the screw joint is used as a reference value for the design of the new supports to prevent excessive weight gain. Their 3D shape and the resulting assembly are shown in Table 3.

Table 3 – Two new designs of the carrier with supports

A = 24.7 cm^2 		
A = 34.1 cm^2 		



The basic parameters of the analysis like the material, model simplification, contact types, and meshing methods remain unchanged. A new type of contact is used between the supports and the carrier plate to prevent their displacement and separation. The results of these analyzes are shown in Figures 6–7.

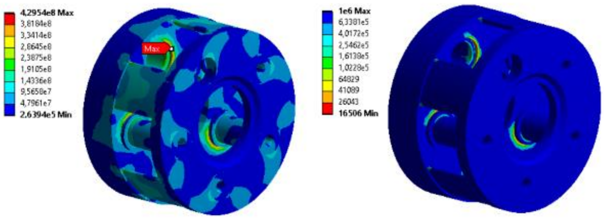


Figure 6 – Von Misses stress distribution and fatigue life of model no. 1

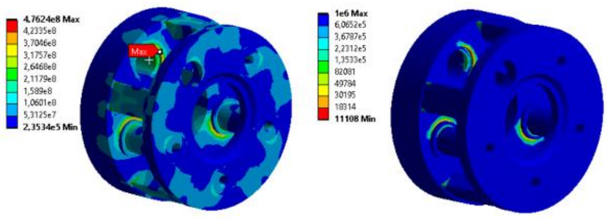


Figure 7 – Von Misses stress distribution and fatigue life of model no. 2

In the next step, new designs were subjected to the fatigue life analysis to determine the new range of operation conditions. The modal analysis was not performed because the carrier proved to be sufficiently rigid. The main results for all models submitted to the static and fatigue life analysis are shown for better comparison in Table 4.

Table 4 – Results summarization

Model	Deformation, mm	von Mises stress, MPa	Fatigue life, 10^3 cycles
Original	0.732	665	3.2
no. 1	0.354	430	16.5
no. 2	0.478	476	11.1

In both models, there is a significant improvement in all properties. Even though model no. 2 has a larger cross-sectional area of supports, the most significant improvement can be observed in model no. 1. The main body diameter is then reduced so that it is equal to the diameter of the carrier plate. This reduction ensures a weight loss of 5%, which can positively affect production costs. The results of static and fatigue life analysis for the final model with reduced diameter are shown in Figure 8.

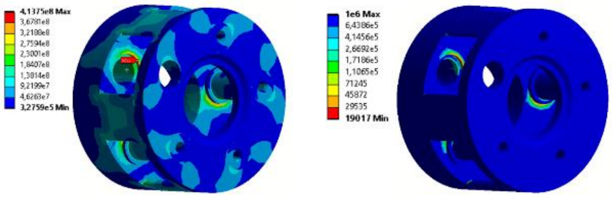


Figure 8 – Von Misses stress distribution and fatigue life of the optimized model with reduced diameter

5 Discussion

This paper represents a specific use of computer-aided FEM in practice, which requires the solution of very complex problems. FEM enables testing and optimization of machine parts, while the costs are significantly lower than performing physical tests. On the other hand, it is also beneficial to supplement this method with the results obtained from experimental measurements if possible. The experimental methods, when applied correctly, represent the actual state of the part in specific operating conditions.

New operating conditions are proposed based on the results of numerical analysis (Table 5). Compared to the original model, they include a more comprehensive range of applications and the use of the crusher machine in practice.

Table 5 – New operating conditions proposal for the optimized carrier assembly

Daily operation time	No. of switch on per hour	Cycles until failure, 10^3	Suitability
8 hrs	0	0.73	Suitable
up to 8 hrs	1	5.84	Suitable
8–12 hrs	1	8.76	Suitable
12 - 24 hrs	1	17.5	Suitable
up to 8 hrs	2–3	17.5	Suitable
8–12 hrs	2	17.5	Suitable

Planetary gearboxes need to have a large load capacity, compact size, and high power density. Therefore, they are vulnerable to fatigue crack even when they are well designed. Fatigue crack caused mainly by a harsh working environment may eventually cause failures of planetary gearboxes if not detected early. Many studies also investigate the behavior of a planetary gear system with a crack. The crack can increase the relative displacement between the carrier and ring gear and induce more difference in response frequency with gravity and clearance [12]. This may lead to a smaller range of operating conditions.

In another study, vibration data from a number of US Army UH-60A Black Hawk helicopter transmissions were used to test two new methods of detecting a fatigue crack in a planet carrier. Vibration measurements of faulted and un-faulted transmissions over a range of torque levels in controlled test-cell and on-aircraft conditions showed that new methods are reliable under test-cell conditions but less effective under low-torque on-aircraft conditions [13].

These studies could be used for further analysis of the optimized carrier assembly to test its reliability in less-than-ideal conditions when the fatigue crack evolves.

6 Conclusions

In this paper, a numerical analysis of the planetary carrier assembly was performed in ANSYS. The main aim was to evaluate the obtained results and design a new carrier model as a single part component with improved strength and fatigue life properties.

The results for equivalent stresses of the mounted carrier assembly showed that the stress exceeded the yield strength of the material by almost 5 % at the point of contact between the pin with the main body of the carrier. This is an area where the formation of a plastic deformation zone and subsequent failure can be expected. The fatigue analysis proved that the maximum number of cycles until failure is sufficient only for the continuous operation (eight hours a day), and the repeated switching on and off would lead to premature failure due to fatigue processes. The modal parameters obtained from the modal analysis showed that the carrier assembly has high natural frequencies and is sufficiently rigid. It is also safe to say that resonance will not occur as the operating frequencies are significantly lower.

The new designs were created based on the previous results and subjected to analysis. Model no. 1 proved to be the most efficient one since the maximum equivalent stress was reduced significantly, resulting in an increase in the number of cycles until failure. Furthermore, model no. 1 was modified by reducing its main body diameter to minimize its weight. This modification ensured a weight reduction of 5 %.

The last model of the carrier assembly designed as a single part component is the most optimal solution for the application in the given gearbox. The maximum values of deformation decreased by more than 55%. The maximum von Misses stresses decreased by almost 38%, while the most important is that their value in critical places fell below the value of the yield strength of the material. Fatigue life has increased almost six times compared to the original model, which resulted in a new proposal of the operating conditions for the optimized carrier, including a broader range of the crusher machines applications in practice. The overall summary of the results from the mounted carrier to its most optimal shape as a single part component is shown in Table 6.

Table 6 – Final comparison between the mounted carrier assembly, model no. 1, and optimized model

	Deformation, mm	von Mises stress, MPa	Fatigue life 10^3 cycles
Original	0.732	665	3.2
no. 1	0.354	430	16.5
Optimized model	0.325	414	19.0

7 Acknowledgments

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References

1. Nielsen, C. V., Zhang, W., Alves, L. M., Bay, N., Martins, P. A. F. (2013). *Finite Element Formulations. Modeling of Thermo-Electro-Mechanical Manufacturing Processes*. Springer Briefs in Applied Sciences and Technology, https://doi/org/10.1007/978-1-4471-4643-8_2.
2. Talbot, D., Kahraman, A. (3014). A methodology to predict power losses of planetary gear sets. *International Gear Conference 2014, 26–28 August 2014, Lyon, France*, pp. 625–635, https://doi.org/10.1533/9781782421955.625.
3. Fan, L., Wang, S. H., Wang, X., Han, F., Lyu, H. (2016). Nonlinear dynamic modeling of a helicopter planetary gear train for carrier plate crack fault diagnosis. *Chinese Journal of Aeronautics*, Vol. 29(3), pp. 675–687, https://doi.org/10.1016/j.cja.2016.04.008.
4. Boyce, M. P. (2012). *Gears. Gas Turbine Engineering Handbook*. Elsevier, pp. 605-625, https://doi.org/10.1016/B978-0-12-383842-1.00014-7.
5. Browell, R., Hancq, A. (2006). *Calculating and Displaying Fatigue Results*. ANSYS® Online White Papers. Available online: https://support. ansys.com/staticassets/ANSYS/staticassets/resourcelibrary/whitepaper/fatigue.pdf.
6. Hd el Morsy, M. (2011). Dynamic analysis of wind turbine gearbox planet gear carrier using finite elements approach. *International Journal of Current Research*, Vol. 12, pp. 220–225.
7. Hao, D., Wang, D. (2013). Finite-element modeling of the failure of interference-fit planet carrier and shaft assembly. *Engineering Failure Analysis*, Vol. 33, pp. 184–196, https://doi.org/10.1016/j.engfailanal.2013.04.029.
8. August, R., et al. (1984). *Dynamics of Planetary Gear Trains*. NASA Contractor Report. Available online: https://ntrs.nasa.gov/archive/nasa/casi.ntrs.nasa.gov/19840017959.pdf.
9. Kissling, U., Dinner, H. (2014). A procedure to determine the optimum flank line modifications for planetary gear configurations. *International Gear Conference 2014, 26–28 August 2014, Lyon, France*, pp. 65–75, https://doi.org/10.1533/9781782421955.65.
10. Moreau, R., Mevel, B. (2014). A gear element in a wind-turbine gearbox. *International Gear Conference 2014, 26–28 August 2014, Lyon, France*, pp. 495–505, https://doi.org/10.1533/9781782421955.495.
11. Zhu, C. C., Xu, X. Y., Lim, T. C., Du, X. S., Liu, M. Y. (2013). Effect of flexible pin on the dynamic behaviors of wind turbine planetary gear drives. P*roceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, Vol. 227(1), pp. 74–86, https://doi.org/10.1177/0954406212447029.
12. Chen, X., Chen, Y., Zuo, M. J. (2020). Dynamic modelling of a planetary gear system with sun gear crack under gravity and carrier-ring clearance. *Procedia Manufacturing*, Vol. 49, pp. 55–60, https://doi.org/10.1016/j.promfg.2020.06.011.
13. Blunt, D. M., Keller, J. A. (2006). Detection of a fatigue crack in a UH-60A planet gear carrier using vibration analysis. *Mechanical Systems and Signal Processing*, Vol. 20(8), pp. 2095–2111, https://doi.org/10.1016/j.ymssp.2006.05.010.