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EFFECTS OF ROTATIONAL SPEED ON THE NATURAL FREQUENCY OF THE DIFFERENTIAL BEVEL GEAR

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With the development of techniques for the use of gears and their many shapes and types, the mechanical need for them has become great, especially in the use of them in the field of cars, and the most important of these types are bevel gears, as these gears are considered essential in differential gears.

The use of differential gears in mechanics in general helps to reduce noise in the movement, but there must be vibration resulting from this movement, and accordingly, bevel gears and the effect of rotating gears on free vibration have been studied.

Variable gears were used according to a simulation program and the study of the free vibrations that occur to them. The effect of rotational speed on the natural vibration greatly affects the transmission of movement in the car and increases the fault distortions that occur in the differential gearbox.

The result shows the natural vibration reached at the speed of 5000 rpm, and the value of the vibration reached 3564.5 Hz, which is the highest value compared to the remaining speed. The distortion at a speed of 1,000 rpm. The process of rotation and natural vibration affects the deformations and stresses that get on the gears themselves. The natural vibration is greatly reduced when the number of clamping places for the differential gear is increased. Compared to the presence of two spinning tires, the vibration value reduced. At a rotating speed of 5000 rpm, it is known that increasing the rotational speed raises the value 3015.9 Hz with one tire revolution with one tire rotation, the huge strains influence the little gears in the differential gearbox. The greatest value of distortion is 0.00067 m at 5000 rpm, which is the largest value of deformation compared to the rest of the employed rotational rates with one tire revolution

Keywords: bevel gear, differential gears, natural frequency, rotational speed, vibration for bevel gear

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1. Introduction

Cutting spiral bevel gears using double-flank machining, a revolutionary method requires specially designed milling tools. The route-planning algorithm considers the shape of the tool in addition to the machine's path. It is advised to use extraordinary methods to evaluate the time-varying contact firmness of rotating slant gears. Researchers have shown how altered contact direction significantly influences the distinctive responsiveness and lattice efficacy of material frameworks. Exhaustion crack failures were discovered during testing with a standard turboshaft motor. The purposeful decrease of the underlying load in aircraft motors may have impaired their ability to remain unyielding. Exhaustion fractures may have been caused by a coupling vibration between the gas generator rotor and the motor's focal driven slant gear (CDBG).

Therefore, studies that are devoted development of different models of bevel gear, frequency and the rotational speed of bevel gear are scientific relevance.

2. Literature review and problem statement

A novel damping model was introduced to defeat its disadvantages: the extent of damping force is connected with relative speed as opposed to outright speed. The recommended technique's precision and productivity were displayed at low velocities; dynamic properties and vibration rules were approved at high paces [1]. In gear applications, nature of configuration fundamentally impacts transmission, machine execution, and size and weight of the pinion wheels. The point of this exploration is to enhance weight, pitch cone distance, and effectiveness. Non-ruled Sorting Genetic Algorithm produces Pareto limits (NSGA-II) [2]. Double-flank machining is a novel technique for cutting spiral bevel gears using custom-shaped milling tools. The route-planning algorithm looks not just at the machine's path but also at the tool's form. It outperforms standard ball end milling in the semi-finishing stage of gear production [3]. Extraordinary methodology for assessing the time-changing contact firmness of twisting slant gears is recommended. Most of greased up transmission parts work in the blended oil region, proposing that the scouring surfaces have both ill temper contact and film oil. The transient blended contact stiffener and hosing discoveries show a significant distinction from the Hertz (dry) contact stiffeners and hoses [4]. A team of researchers has shown that shifted contact course considerably affects the unique responsiveness and lattice effectiveness of stuff frameworks. The determined transmission effectiveness under shifted contact pathways is as per genuine outcomes, rather than just the outcome of a blended EHL model for winding slant gears [5]. A winding slope stuff's exhaustion life is surveyed to give information to plan enhancement. It was put through some serious hardship under motor evaluated, furrow same, and incorporated comparable burden conditions. The discoveries might be useful in advancing the plan of farm hauler transmission twisting slope gears [6]. A paper proposes a model for spotting early disappointments brought about by mechanical imperfections. During creation and use, mathematical defects like pitting might arise on the stuff surfaces. For twisting slant equips, a transient blended oil model and a two-dof-torsional dynamic model are proposed [7]. Exhaustion crack disappointments were found in testing utilizing a customary turboshaft motor. The deliberate reduction of underlying load in airplane motors might have compromised help unbending nature. A coupling vibration between the gas generator rotor and the motor's focal driven slant gear (CDBG) might have led to exhaustion cracks [8]. Special reenactment model has been planned and executed into a business CAD stage. Movement and force are communicated across non-equal shafts. Because of their extraordinary efficiency, face processing and face stumbling are among the main methodology for the machining of hypoid and winding angle gears [9]. The lumped-mass approach was used to generate spiral bevel gears theoretically. The lateral vibration generated by gear eccentricity would amplify the torsional vibration of the oblique tail shaft. This is due to the bending-torsional coupled action on the rotor blades [10]. A study into the quantitative values and dynamics of physical factors in premises and workplaces of stationary and portable computers. Membrane-type protective panels configured for maximum resonant frequencies of low-frequency sound and infrasound have been proposed [11]. Knowing the consequences of movement in the differential gearbox and the quantity of emitted vibrations and how to address them through the two upcoming investigations was the paper's usefulness. The significance of the issue is understanding the state of the gears when the natural vibration grows and how to manage and prevent it, furthermore to address the effect of the gears on the cover that surrounds it. Therefore, work was done on the differential gearbox to install one side of the frame and to understand the circumstances of the two movement cases' natural vibration.

3. The aim and objectives of the study

The aim of the study is to investigate the natural vibration and the effect of rotational speed on bevel gears. This will make it possible to predict the deformations and stresses that can be avoid it.

To achieve this aim, the following objectives are accomplished:

– using two wheels rotation, the rotational speed effect on natural vibration was studied;

– using one wheels rotation, the rotational speed effect on natural vibration was studied.

4. Materials and methods

4. 1. The object of the study

The use of differential gears in mechanics in general helps to minimize noise in movement, but this movement must cause vibration, therefore bevel gears and the influence of rotating gears on free vibration were investigated. Variable gears were employed in accordance with a modeling software and a study of the free vibrations that they generate. The influence of rotational speed on natural vibration has a substantial impact on automobile transmission and raises the probability of problems in the differential gearbox.

4. 2. The main hypotheses of the research

The process of studying bevel gears in a differential gearbox requires a program to design this domain, as it was designed with the SolidWorks program, an engineering program that has a library for all the real dimensions of gears as in Fig. 1.

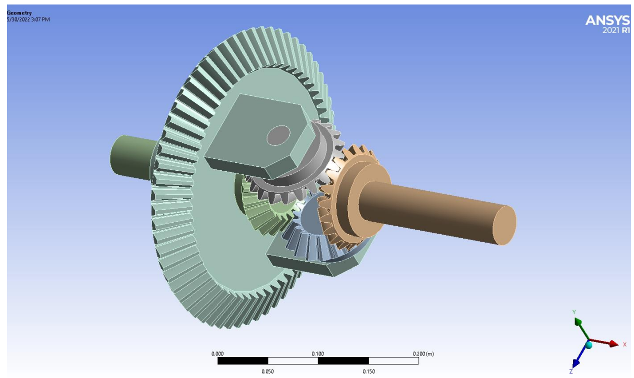

Fig. 1. Geometry design

After the design process, a mesh must be made so that the simulation process is carried out in an accurate manner. During which the best reliable mesh was identified to show accurate results, the number of the elements (tetrahedron) was increased to reach no change in the extracted results as in Fig. 2.

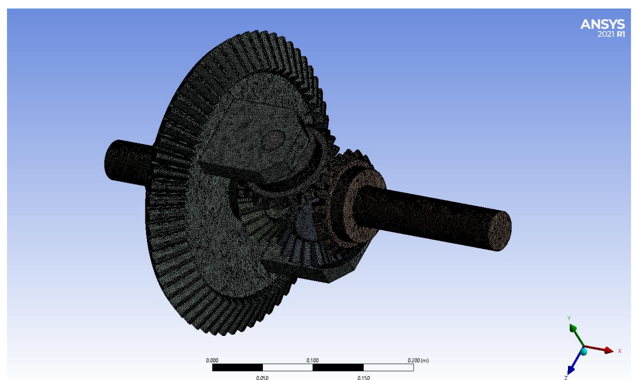

Fig. 2. Geometry mesh

It was noted that the best number for the element was 1384585, and the results showed that the vibration value reached 3564.5 Hz as Table 1.

Table 1

Mesh independency

Case	Element	Node	Maximum frequency (Hz)
1	453108	1045644	3586.1
2	745342	1423808	3566.4
3	1043564	1853443	3564.7
4	1384585	2365466	3564.5

The conditions that control the cases must be logical and keep pace with the truth, where a variable rotation speed was used for the main gear that comes through the main gearbox of the car, where the values of the speeds used were (1000, 2000, 3000, 4000 and 5000) rpm and took two possibilities for the rotation of the differential gearbox, the material of differential gear was steel where the first case represents the rotation of the two wheels and the second case is to install one wheel and another to have a rotation case. To study the natural vibration of gears, the ANSYS MODEL program was used, and 6 modes were taken for each case to see the natural vibration gradient with the change of conditions.

Equation of motion:

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}=\{f(t)\}, \quad (1)$$

M – mass; $\ddot{u}$ – acceleration; C – damping; $\dot{u}$ – velocity; K – stiffness; u – displacement; $f(t)$ – load.

For modal analysis:

$$[M]\{\ddot{u}\}+[K]\{u\}=\{0\}, \quad (2)$$

free and undamped system:

$$\{\ddot{u}(t)\}=0. \quad (3)$$

Moving with constant velocity:

$$x=A\sin(\omega t+\theta), \quad (4)$$

A is amplitude; ω is angular frequency; θ is phase angle.

The overall equilibrium equations for linear structural static analysis are:

$$[K]\{u\}=\{F\}, \quad (5)$$

or

$$[K]\{u\}=\{F^a\}+\{F^r\}, \quad (6)$$

where $[K]=\sum_{m=1}^{N}[K_e]$ – total stiffness matrix; $\{u\}$ – nodal displacement vector; N – number of elements; K_e – element stiffness matrix (described in Element Library) (may include the element stress stiffness matrix (described in Stress Stiffening)); $\{F^r\}$ – reaction load vector; $\{F^a\}$ – the total applied load vector, is defined by:

$$\{F^a\}=\{F^{nd}\}+\{F^{ac}\}+\sum_{m=1}^{N}\left(\{F_e^{th}\}+\{F_e^{pr}\}\right), \quad (7)$$

where $\{F^{nd}\}$ – applied nodal load vector; $\{F^{ac}\}=[M]\{a_c\}$ – acceleration load vector; $[M]=\sum_{m=1}^{N}[M_e]$ – total mass matrix; $[M_e]$ – element mass matrix (described in Derivation of Structural Matrices); $\{a_c\}$ – total acceleration vector (defined in Acceleration Effect); $\{F_e^{th}\}$ – element thermal load vector (described in Derivation of Structural Matrices); $\{F_e^{pr}\}$ – element pressure load vector (described in Derivation of Structural Matrices).

4. 3. Assumptions and reductions accepted in the work

The most important hypotheses that were worked on to form the simulation program start from the design process, which was in a simplified form for the differential gearbox. As there was no need to design the outer cover of the differential gearbox to complete the simulation process. And also, with regard to the parts of the ball bearing that have been compensated for by the limitations in the simulation program.

5. Results of research effects of rotational speed on the natural frequency

5. 1. Effect of rotational speed on natural vibration and keeping two wheels rotation

The effect of rotational speed on the natural vibration greatly affects the transmission of movement in the car and increases the fault distortions that occur in the differential gearbox. Through the following results, it will be noticed that with an increase in the rotational speed, the natural vibration increases, as the natural vibration reached at the speed of 5000 rpm, and the value of the vibration reached 3564.5 Hz, which is the highest value of the natural vibration compared to the remaining speed.as Fig. 3.

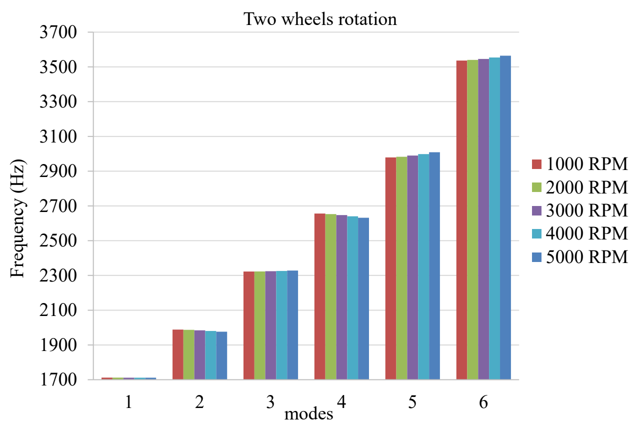

Fig. 3. The value of normal vibration with different rotational speed

The number of modes shows the nature of gear deformation with different values of natural vibration. Through Fig. 4, the difference in distortions from mode to mode at a rotational speed of 5000 rpm is observed.

The process of rotation and natural vibration affects the deformations and stresses that act on the gears themselves. It is noted from Fig. 5 that the deformation value increases with the increase in gear rotation, where the deformation value at the speed of 5000 rpm reached 0.0006 m, which is the highest value of deformation compared to the rest of the used rotational speeds.

The effect of the rotational speed is not limited to creating deformations on the gears but also on the stresses applied to the gears as well. It is noticed that the large stresses affect the small gears in the differential gearbox as shown in Fig. 6.

The higher the gears' rotational speed, the higher the stress values, reaching $2.6 \cdot 10^9$ pa at a speed of 5000 rpm, which is the highest value of stress compared to the rest of the cases.

5. 2. Effect of rotational speed on natural vibration and keeping one wheels rotation

The increase in the number of clamping areas for the differential gear significantly reduces the natural vibration. It is noticed from Fig. 7 that the value of the vibration decreased compared to the presence of two rotating tires. It is known that the increase in the rotation speed increases the value of the natural vibration, as it is noted that the value of the natural vibration was 3015.9 Hz at a rotational speed of 5000RPM.

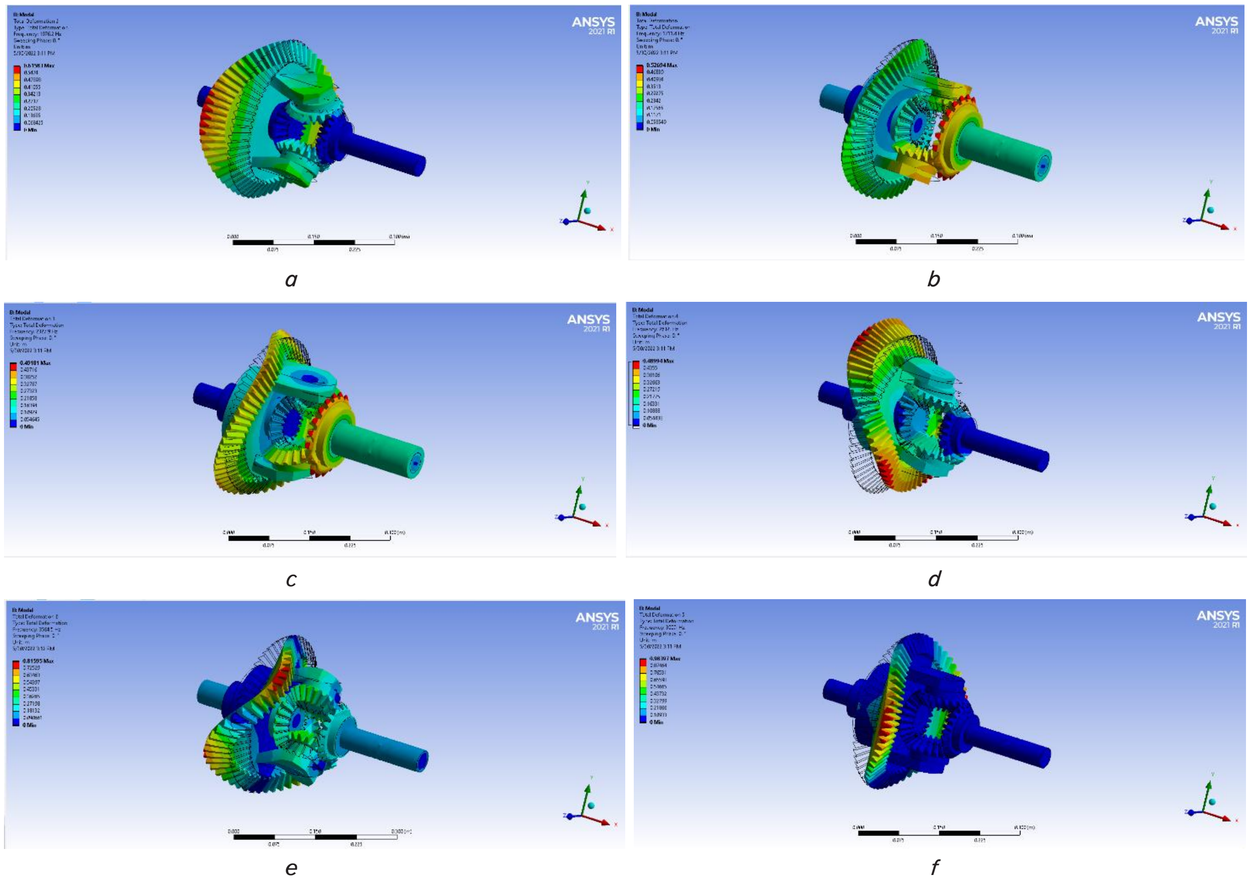

Fig. 4. Contour of deformation at: *a* — Mode 1; *b* — Mode 2; *c* — Mode 3; *d* — Mode 4; *e* — Mode 5; *f* — Mode 6

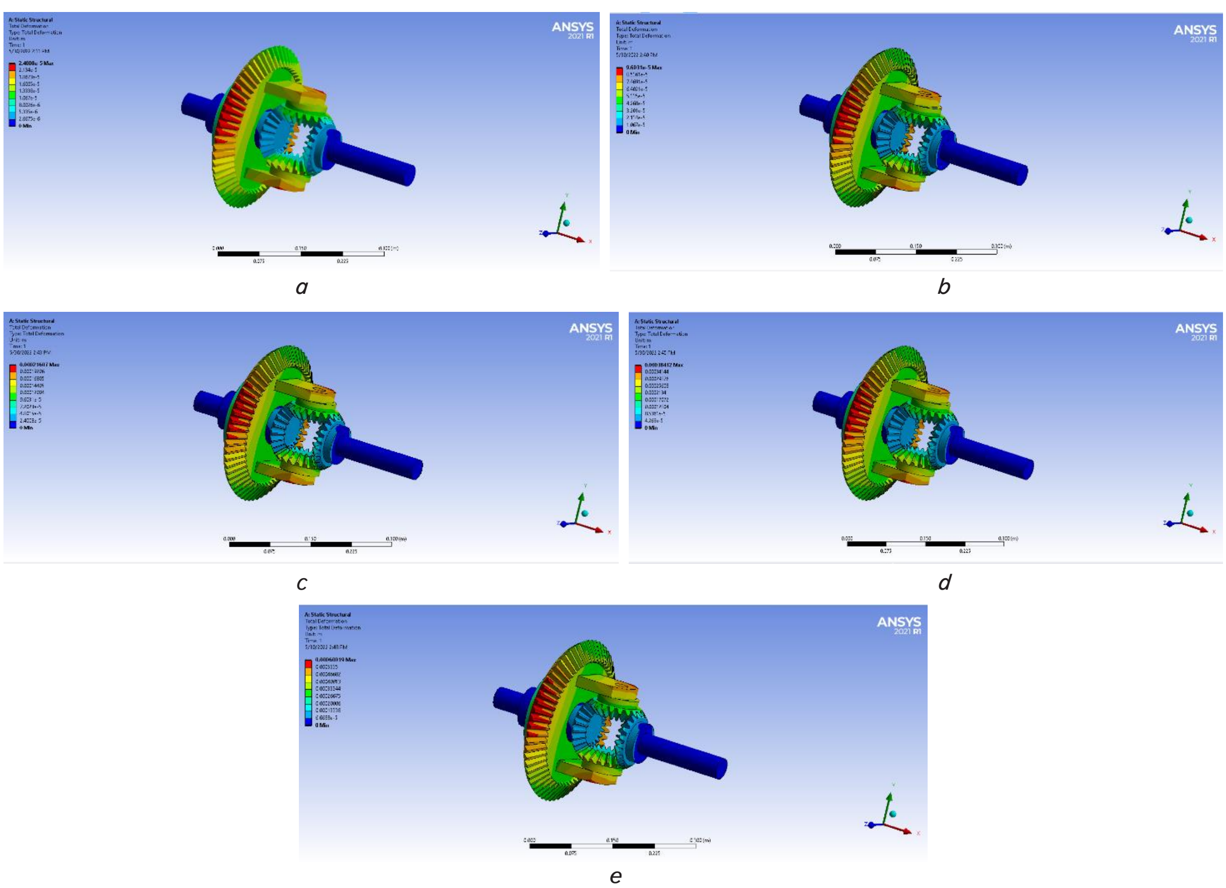

Fig. 5. Contour of deformation at: *a* — 1000 rpm; *b* — 2000 rpm; *c* — 3000 rpm; *d* — 4000 rpm; *e* — 5000 rpm

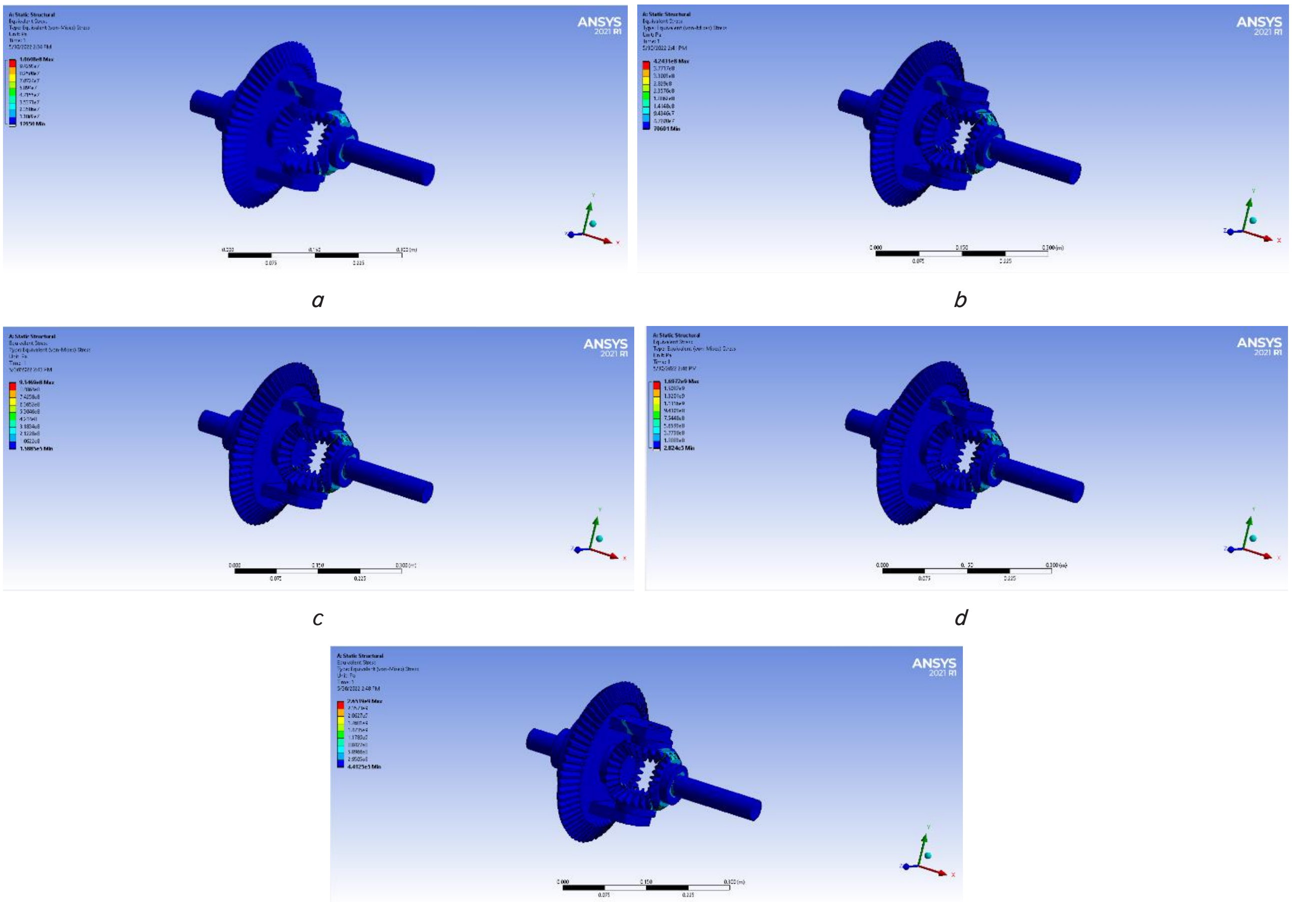

Fig. 6. Contour of stress at: *a* – 1000 rpm; *b* – 2000 rpm; *c* – 3000 rpm; *d* – 4000 rpm; *e* – 5000 rpm

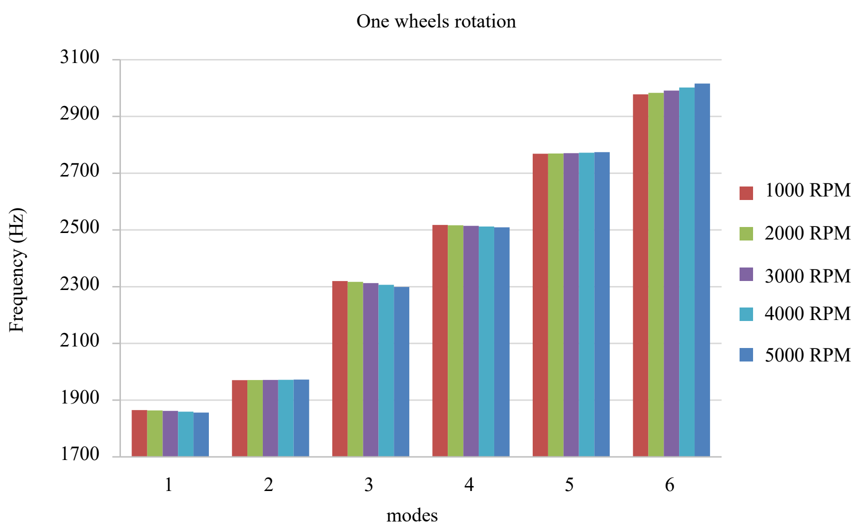

Fig. 7. The value of normal vibration with different rotational speed

The process of rotation and natural vibration affects the deformations and stresses that act on the gears themselves. It is noted from Fig. 8 that the deformation value increases with the increase in gear rotation, where the deformation value at the speed of 5000 rpm reached 0.00067 m, which is the highest value of deformation compared to the rest of the used rotational speeds. It is noticed that the large stresses affect the small gears in the differential gearbox, and that the higher the gears' rotational speed, the higher the stress values, reaching $2.76 \cdot 10^9$ Pa at a speed of Fig. 9. The rotation is 5000 rpm, which is the highest value of stress compared to the rest of the cases.

From the above contours it can be seen that the effect of the rotational speed is not limited to creating deformations on the gears but also on the stresses applied to the gears as well.

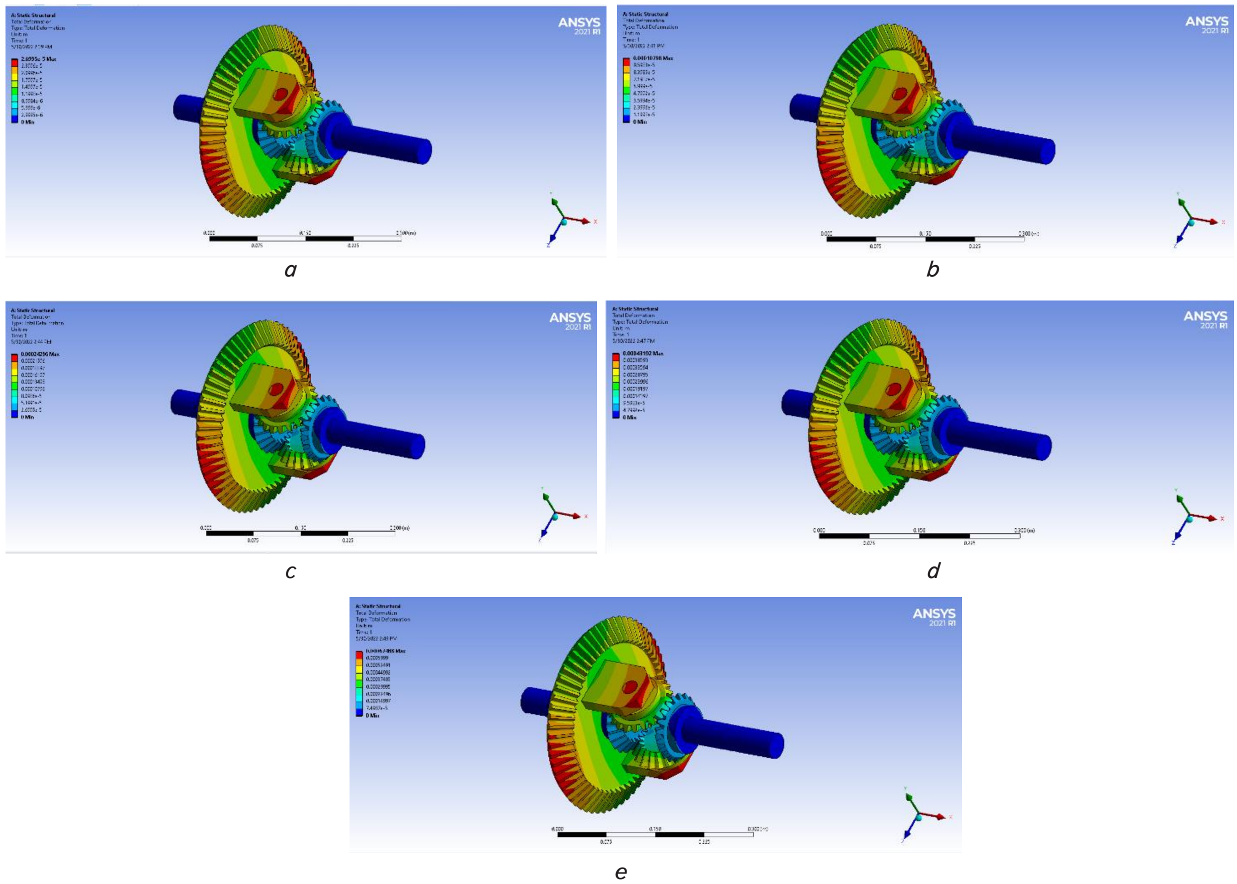

Fig. 8. Contour of deformation at: *a* — 1000 rpm; *b* — 2000 rpm; *c* — 3000 rpm; *d* — 4000 rpm; *e* — 5000 rpm

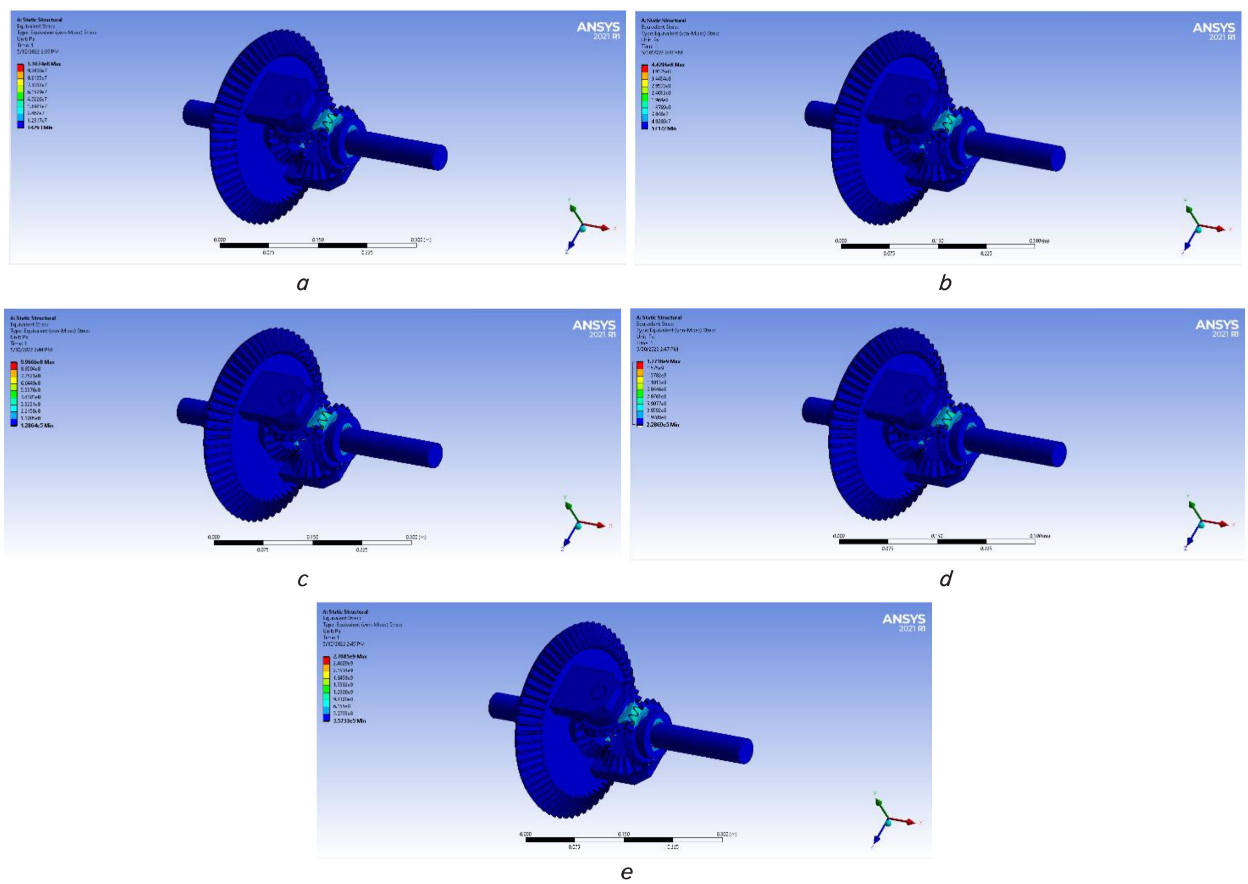

Fig. 9. Contour of stress at: *a* — 1000 rpm; *b* — 2000 rpm; *c* — 3000 rpm; *d* — 4000 rpm; *e* — 5000 rpm

6. Discussion of research effects of rotational speed on the natural frequency

The influence of rotational speed on natural vibration has a significant impact on vehicle movement transmission and causes fault distortions in the differential gearbox.

The following data show that as the rotational speed grows, so does the natural vibration, as the natural vibration reached 5000 rpm and the value of the vibration reached 3564.5 Hz, which is the maximum value of the natural vibration compared to the remaining speed. as seen in Fig. 3.

The number of modes demonstrates the nature of gear deformation with varying levels of natural vibration. Fig. 4 depicts the variation in distortions from mode to mode at a rotating speed of 5000 rpm. The rotational and natural vibrational processes influence the deformations and strains that act on the gears themselves.

Fig. 5 shows that the deformation value rises with increasing gear rotation, with the deformation value reaching 0.0006 m at 5000 rpm, the greatest value of deformation compared to the other rotational speeds utilized.

The rotational speed has an influence not only on the deformations of the gears but also on the stresses imposed on the gears. The significant stresses influence the tiny gears in the differential gearbox, and the greater the rotational speed of the gears, the higher the stress values, reaching $2.6 \cdot 10^9$ Pa at a speed of Fig. 6. The rotation is 5000 rpm, which has the largest stress value compared to the other situations.

The increased number of clamping surfaces for the differential gear decreases natural vibration substantially. Fig. 7 shows that the value of the vibration reduced when two spinning tires were present. It is well known that increasing the rotational speed increases the value of the natural vibration, as the value of the natural vibration was 3015.9 Hz at 5000 rpm. The rotational and natural vibrational processes influence the deformations and strains that act on the gears themselves.

Fig. 8 shows that the deformation value rises with increasing gear rotation, with the deformation value reaching 0.00067 m at 5000 rpm, the greatest amount of deformation compared to the other rotational speeds utilized. The rotational speed has an influence not only on the deformations of the gears but also on the stresses imposed on the gears. The significant stresses influence the tiny gears in the differential gearbox, and the greater the rotational speed of the gears, the higher the stress values, reaching $2.76 \cdot 10^9$ Pa at a speed of Fig. 9. The rotation is 5000 rpm, which has the largest stress value compared to the other situations.

The design phase, which was simplified for the differential gearbox, is where the most significant hypotheses that went into creating the simulation program begin. Since the differential gearbox's outside cover did not need to be designed, the simulation process could be considered complete. Additionally, with regard to the components of the ball bearing that the simulation program's restrictions have replaced. The disadvantage of this study is not to use very modern differential gearbox and it could be developed by using more complex driving conditions.

7. Conclusions

1. The effect of rotational speed on the natural vibration greatly affects the transmission of movement in the car and increases the fault distortions that occur in the differential gearbox. The higher the gears' rotational speed, the higher the stress values, reaching $2.6 \cdot 10^9$ Pa at a speed of 5000 rpm, which is the highest value of stress compared to the rest of the cases.

2. The natural vibration is greatly reduced when the number of clamping places for the differential gear is increased. The greater the rotational speed of the gears, the higher the stress values, which reach $2.76 \cdot 10^9$ Pa at 5000 rpm, the greatest amount of stress compared to the other examples.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

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