UDC 621
DOI: 10.15587/1729-4061.2022.263585

A NUMERICAL IMPLEMENTATION STUDY OF THE STRESSES AND NOISE GENERATED BY THE GEAR ENGAGEMENT IN THE GEARBOX

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With the development of automotive technology and the increase in the performance of the moving parts in the car, the most important of which is the engine and gearbox. Where companies are working to increase the sound insulation of the movement of these parts outside and reduce the noise and stress generated because of movement, to study these noises and stresses, and to know the vibrations that give a high sound rate and wave pressure. Where it worked on a standard gearbox and simulated the movement of gears. To find out the stresses that are calculated because of this movement, the external stress caused by this movement of the gears, as well as the speed of the gears in the case of rotation, is also known. The results proved the contact areas of the main lip on the outer cover are greatly affected during the gearbox rotation process and also at high vibrations. The effect of vibrations and the Hertzian rate is significantly affected by it. At frequencies that reached 1500 Hz the value of stresses and deformation was relatively large. The acceleration and due to the different gear teeth, the acceleration value in this case was 3000 m/s^2. The maximum value of shear stresses reached 9.5·10^4 Pa at the frequency between 1500 and 2000 Hz. The value of the vibration 1500 Hz is the highest value that achieved high noise as it was 112 dB, which is a 112 dB higher than the rest of the vibrations. The achievement of the condition of the presence of large noise when vibration 1500 Hz is reached is achieved by analyzing the noise produced by a car's gearbox at that level. The amount of noise pressure and its wave through the outside air of the gearbox, where the amount of wave pressure of the noise reached a maximum value of 400 Pa and the lowest value – 500 Pa

Keyword: COMSOL multiphysics, simulation, gearbox engagement, stresses, noise, sound pressure level

Received date 27.06.2022
Accepted date 22.08.2022
Published date 31.08.2022

How to Cite: *Khleif, A. A. (2022). A numerical implementation study of the stresses and noise generated by the gear engagement in the gearbox. Eastern-European Journal of Enterprise Technologies, 4 (1 (118)), 47–55. doi: https://doi.org/10.15587/1729-4061.2022.263585*

1. Introduction

The power-coupling component, a crucial aspect of a half breed electric vehicle (HEV), was discovered to have a significant influence on the vibration and agitation of the whole vehicle. A tribo-dynamic model for planetary stuff sets of half-and-half electric vehicle designs is presented in this work. The review clarifies the relationship between the hypoid gear pair and propeller shaft and may be used to enhance the design of the driveline system. In especially for the bearing reactions, the combined effects of stuff shaft communication and gyroscopic behavior have a significant influence on the strong responses including stuff-twisting resonances. By adjusting the pinion bearing rotational solidness and the pinion bowing snapshot of inactivity for the model under consideration, NVH (clamor, vibration, and cruelty) refinement may be achieved. For the purpose of analyzing gear issues, this effort is planned to provide a specific time-recurrence research approach coupled with vibration and acoustic estimates. Through previous research and comparing their work with the fields of gearbox, which will be worked on in this research paper to complete conclusions that have not been addressed.

Therefore, research on investigate how the gearboxes produce these sounds and stresses is relevant. It also examines the impact of velocity and the frequency of the gearbox.

2. Literature review and problem statement

A computerized methodology of spur gears reconstruction and feature extraction using image processing technique is proposed [1]. The proposed system has been tested on different spur gears, and the extracted data shows reliable spur gear modeling. It generates accurate and efficient model and provides substantial saving in time and cost of production. The critical part of a half breed electric vehicle (HEV), the power-coupling component was found to have a massive impact upon the vibration and commotion of the entire vehicle. In this paper, the attributes of the bearing compelling powers in time space and recurrence area were recreated and investigated. The transmitting commotion characteristics of the lodging and the acoustic commitment of each board are dissected. At long last, the free damping structure and new stiffener structure are taken on to enhance the backside front of the lodging [2]. Further developed commotion, vibration and brutality refinement would be normal in the electric engine drive mode, while better transmission effectiveness is accomplished in the gas-powered motor drive mode. The paper presents a tribo-dynamic model for planetary stuff sets of halves-and-half electric-vehicle designs. Investigation is performed for a mixture electric C-section vehicle [3]. The review gives an understanding of the connection between hypoid gear pair and propeller shaft, and can be utilized to improve driveline system plan. The joined impacts of stuff shaft communication and gyroscopic way of behaving have impressive effect on the powerful reactions encompassing stuff-twisting resonances, particularly for the bearing reactions. NVH (clamor, vibration, and cruelty) refinement can be accomplished by tuning pinion bearing rotational solidness and pinion bowing snapshot of inactivity for the model considered [4]. Transmission proficiency and refinement of planetary wheel center point outfitting framework are key plan credits for weighty and rough terrain vehicles. Decrease of force misfortune straightforwardly prompting the advancement of new age ECO-axles requires investigation of stuff reaching conditions for greased up conjunctions. This is additionally impacted by gear elements, which is an essential for evaluation of commotion, vibration and cruelty execution [5]. This work is expected to create a defined time-recurrence investigation technique joined with vibration and acoustic estimations for gear issue analysis. The consolidated general straight chirp let change strategy gave more exact issue seriousness appraisal contrasted with other regularly used diagnostic techniques, for example, consistent wavelet changes and pseudo-Wigner-Ville dispersion strategies. It permits a precise assurance of the rakish area of a stuff issue and abettor portrayal of sidebands related with the seriousness level of stuff shortcoming [6]. This paper presents a clever starter gauge technique for gear minimizer commotion radiation and vibration attributes. In the commotion examination process, the impact of time-fluctuating cross section solidness, mistake excitation, and tooth flank contact include are exhaustively thought of. With the increment of burden, the commotion of minimizer changes with load as the variation of logarithmic capacity is noticed [7]. In this paper, impacts of lightweight stuff clear on static and dynamic way of behaving for electric drive framework in electric vehicles are considered. The outcomes show that switching gear web and stuff edge thickness can fundamentally diminish dynamic lattice power and dynamic reaction. Commotion, vibration, and brutality issue of an electric drive framework prepared on electric vehicles is likewise tended to [8]. The percent deviation of lattice solidness expanded with the expansion in contact proportion, however diminished slowly with expansion in load. The thunderous recurrence of vibration speed increase with pitch deviation decreases contrasting with the recurrence with standard pitch. This study gives proof that it is fundamental to think about the impact of pitch deviation when redesign the boundaries and anticipate vibration of round and hollow helical stuff transmission framework [9]. The most widely recognized and successful approach to turning apparatus analytic is to remove issue sway highlights from the vibration signals and to do additionally handling. Because of concurrent lattice of multiple gears and impact of the transporter pivot, the issue highlights are lowered in the solid consonant signs and other clamor. Let's give the earlier data of a chipped planetary gear set by elements recreation, and use the data in time area to further develop analysis execution [10–14].

Most of the problems encountered in the researches are the mathematical equations relating stresses, vibrations, noises and gear movements with each other. It requires highly efficient engineering software and computers capable of solving complex matrices and physics, which is done in this research paper.

3. The aim and objectives of the study

The aim of the study is to identify regularities of the stresses and noise generated by the gear engagement in the gearbox. This will make it possible by knowing the vibrations that give a high sound rate and wave pressure.

To achieve this aim, the following objectives are accomplished:

– to find out velocity and acceleration of gearbox;
– to discuss the effect of velocity on stresses;
– to evaluate the frequency of gearbox;
– to show shear stresses of gearbox;
– to predict sound pressure of gearbox.

4. Materials and methods

4. 1. Object of research

With the advancement of automotive technology, the efficiency of the car's moving components, especially the engine and transmission, has increased. It is important for businesses to examine the sounds and strains caused by the movement of these components, as well as the vibrations that result in a high sound rate and wave pressure, in order to improve sound insulation and decrease the resulting noise and stress.

4. 2. The main hypothesis of the study

It is the multibody dynamics module, which simulated the workings of a conventional gearbox and get the job done. For this analysis, the external stress due to this motion of the gears and, in the case of rotation, the speed of the gears, are both computed.

4. 3. Assumptions made in the work

The gearbox was designed using Dassault Systems SolidWorks Corp (France), a program specialized in designing precise geometric models, where the cover and space were designed traditionally. As for the gears, they were taken from the program laboratory, which depends on global dimensions ISO. Through the following Table 1, the dimensions and measurements of the gear and its box are shown.

Fig. 1 represents the final design that will be simulated in a practical application.

After the process of designing the gearbox Fig. 1, 2, the simulation process was carried out using the COMSOL Multiphysics program, and in order for the simulation process to be accurate, a mesh must be made that covers the requirements of the simulation process, as the reliability of the mesh was made by increasing the number of the element in order to reach a state of stability with the results, where it was found that the best number of element was reached Its mechanism is 130065 to reach an inductive state in noise Table 2.

After the mesh configuration process, Fig. 3, the simulation settings were applied, where two processors were used, the first (Multibody dynamics) which will be used for the movement of the solid parts.

Where stainless steel was used, where the gears' gears were identified with each other and their connection to the shafts controlling the movement, and the bearing areas and installation areas were defined. Where the rotational speed of the gears is 83.3333 rad/s. As for the second processor, it is (Pressure Acoustics), which specializes in simulating the noise and pressures that are formed as a result of the movement of the gearbox, where the air material is used in this processor and describe the outer ball that represents the woven range and defines the acceleration obtained from the first processor to detect noise and pressure.

Table 1

Dimensions gearbox

Symbols	Dimensions	Description
n1i	20	Number of teeth, first gear (counter shaft)
n2i	20	Number of teeth, second gear (counter shaft)
n3i	20	Number of teeth, third gear (counter shaft)
n4i	20	Number of teeth, fourth gear (counter shaft)
n5i	28	Number of teeth, fifth gear (counter shaft)
n1o	76	Number of teeth, first gear (main shaft)
n2o	44	Number of teeth, second gear (main shaft)
n3o	28	Number of teeth, third gear (main shaft)
n4o	20	Number of teeth, fourth gear (main shaft)
n5o	20	Number of teeth, fifth gear (main shaft)
gear_ratio1	3.8	Gear ratio, first gear
gear_ratio2	2.2	Gear ratio, second gear
gear_ratio3	1.4	Gear ratio, third gear
gear_ratio4	1	Gear ratio, fourth gear
gear_ratio5	0.71429	Gear ratio, fifth gear
cd	0.1 m	Center distance
d1i	0.041667 m	Pitch diameter, first gear (counter shaft)
d1o	0.15833 m	Pitch diameter, first gear (main shaft)
d2i	0.0625 m	Pitch diameter, second gear (counter shaft)
d2o	0.1375 m	Pitch diameter, second gear (main shaft)
d3i	0.083333 m	Pitch diameter, third gear (counter shaft)
d3o	0.11667 m	Pitch diameter, third gear (main shaft)
d4i	0.1 m	Pitch diameter, fourth gear (counter shaft)
d4o	0.1 m	Pitch diameter, fourth gear (main shaft)
d5i	0.11667 m	Pitch diameter, fifth gear (counter shaft)
d5o	0.083333 m	Pitch diameter, fifth gear (main shaft)
alpha	0.43633 rad	Pressure angle
beta	0.5236 rad	Helix angle
omega	83.333 rad/s	Engine or main shaft speed

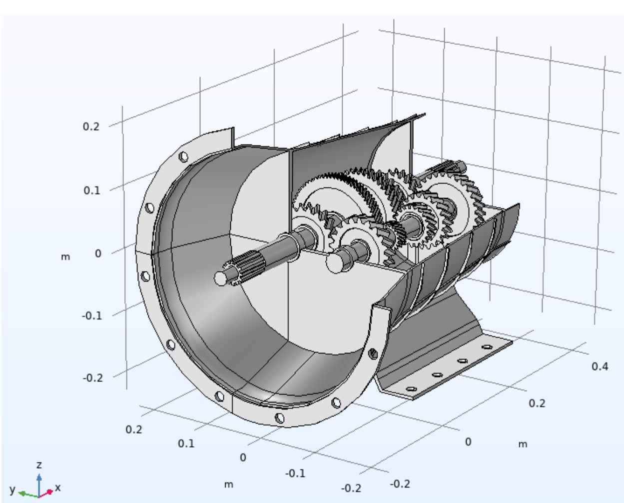


Fig. 1. Gearbox geometry

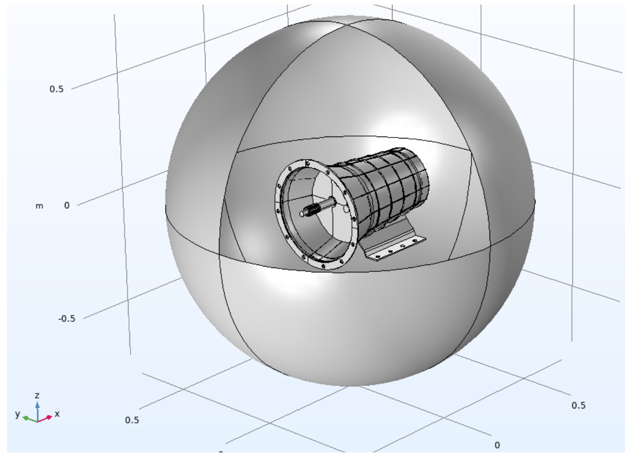


Fig. 2. Gearbox with space sphere geometry

Table 2

Mesh independency

Case	Element	Sound pressure level at 1500 Hz
1	52321	123.9
2	84532	114.2
3	112566	112.7
4	130065	112.6

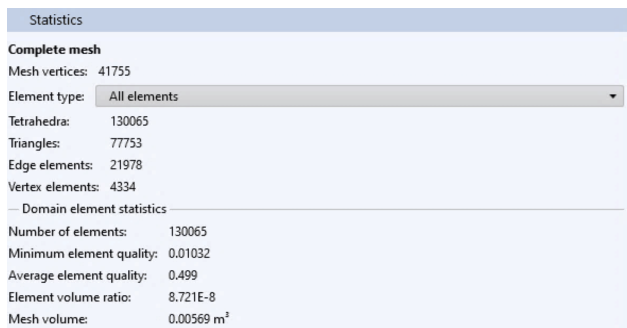


Fig. 3. Statistics of mesh

Multibody Dynamics Module:

$$\rho\frac{\partial^2 u}{\partial t^2}=\nabla\cdot(FS)^T+Fv,\ \ F=I+\nabla u, \tag{1}$$

$$S=S_{ad}+JF_{inel}^{(-1)}(C:C_{el})F_{inel}^{(-T)},$$

$$C_{el}=\frac{1}{2}\left(F_{el}^{\ T}F_{el}-I\right),\ \ F_{el}=F_{inel}^{\ (-1)}, \tag{2}$$

$$S_{ad}=S_o+S_{ext}+S_q, \tag{3}$$

$$E=\frac{1}{2}\left[(\nabla u)^T+\nabla u+(\nabla u)^T\,\nabla u\right], \tag{4}$$

$$C=C(E,V). \tag{5}$$

Pressure Acoustics, Frequency Domain:

$$\nabla\cdot\left(\frac{-1}{\rho_c}(\nabla p_t-q_d)\right)-\frac{k_e q^2 p_t}{\rho_c}=Q_m, \tag{6}$$

$$p_t=\rho+p_b, \tag{7}$$

$$k_{eq}^2 = \left(\frac{\omega}{c_c}\right)^2. \tag{8}$$

Two studies were added, the first related to the first processor, where the total time to solve the gears, rotation is a complete cycle. As for the second study, frequencies from 1000 Hz to 3000 Hz were used, with a difference of 500 Hz.

4. 4. Simplifications adopted in the work

The difficulties that have been overcome it in the simulation process are to simplify some pieces that are not useful for the work by inserting it in the simulation cases like to replace the ball bearing with suitable parameters found in the simulation software.

5. Results of the stresses and noise generated by the gear engagement

5. 1. Velocity and acceleration of gearbox

Where note from the previous Fig. 4 that the linear velocity was 4.8 m/s at the time that completes a full rotation of the gear.

As for the acceleration and due to the different gear teeth, the acceleration varies from one place to another, so the acceleration value in this case was 3000 m/s^2.

5. 2. The effect of velocity on stresses

As a result of the rotational movement and vibrations that occur in the gearbox, note through Fig. 5 the number of stresses generated from this movement, where the amount of stress reached 2·10^7 Pa.

In Fig. 6, the effect of the gear rotation angle on the stresses that occurs in it is shown.

Notice the disturbance in the number of stresses during a complete rotation of the gear, where the maximum value of the stresses obtained was 3.5·10^8 Pa for the time 0.0051 s of rotation.

5. 3. Frequency of gearbox

As for the effect of vibrations and the Hertzian rate, the value of the stresses is significantly affected by it, as it is known in Fig. 7 that at frequencies that reached 1500 Hz the value of the stresses and deformation was relatively large. As notice from Fig. 7 that the value of the stresses with the change in frequency increased to 3·10^6 Pa at the frequency 1500 Hz.

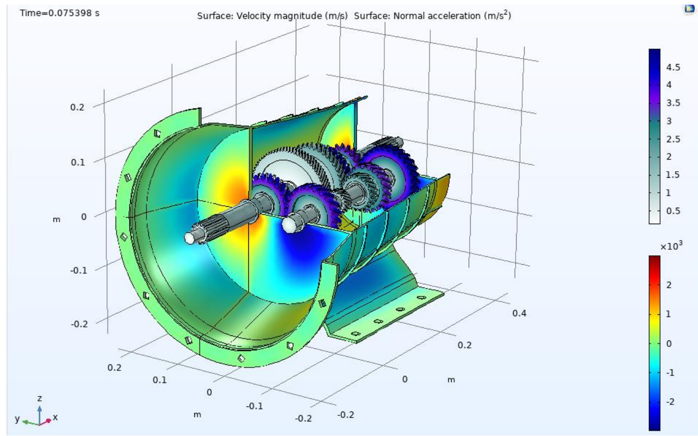


Fig. 4. Velocity and acceleration contour

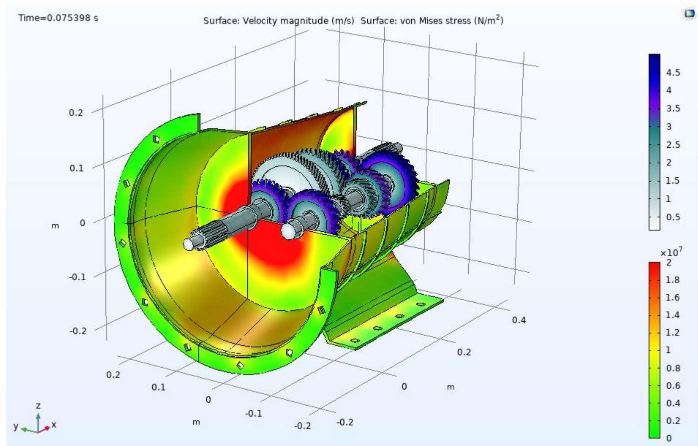


Fig. 5. Velocity and Von Mises stress contour

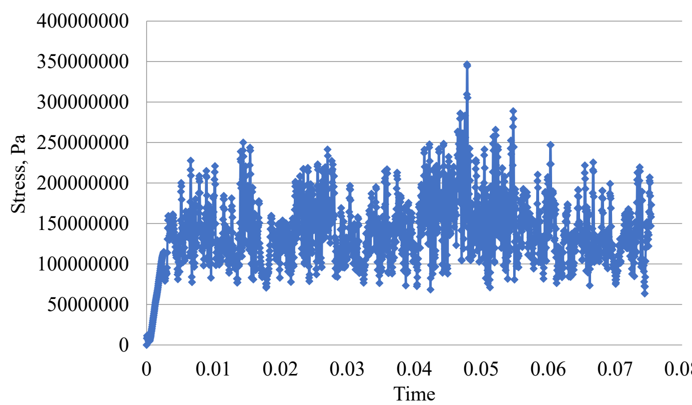


Fig. 6. Von Mises stress with time (s)

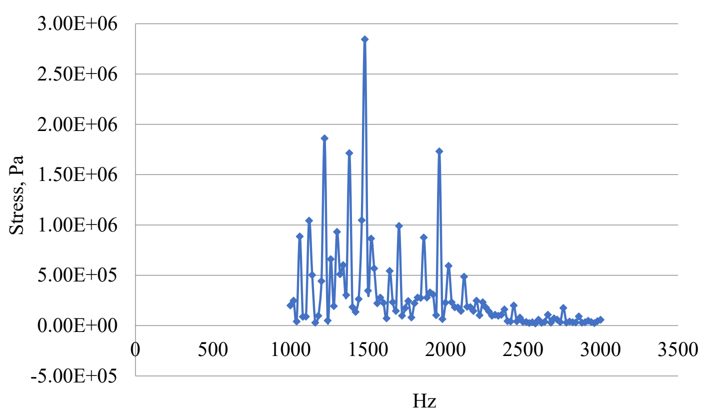


Fig. 7. Von Mises stress with frequency

5. 4. Shear stresses of gearbox

The contact areas of the main lip on the outer cover are greatly affected during the gearbox rotation process and at high vibrations, where in Fig. 8 notice the effect of frequency on the shear stress areas, as the maximum value of shear stresses reached $9.5 \cdot 10^4$ Pa at the frequency between 1500 and 2000 Hz.

As for the time, the shear stresses when the gears rotate for a full revolution, the value of the shear stresses reached $9.27 \cdot 10^7$ Pa during the time 0.075398 s, as in Fig. 9 shows the gradation of shear stresses through the different regions.

From Fig. 10, it is possible to see the value of acceleration with the rotation of the main shaft during one cycle, which is equivalent to 360 degrees.

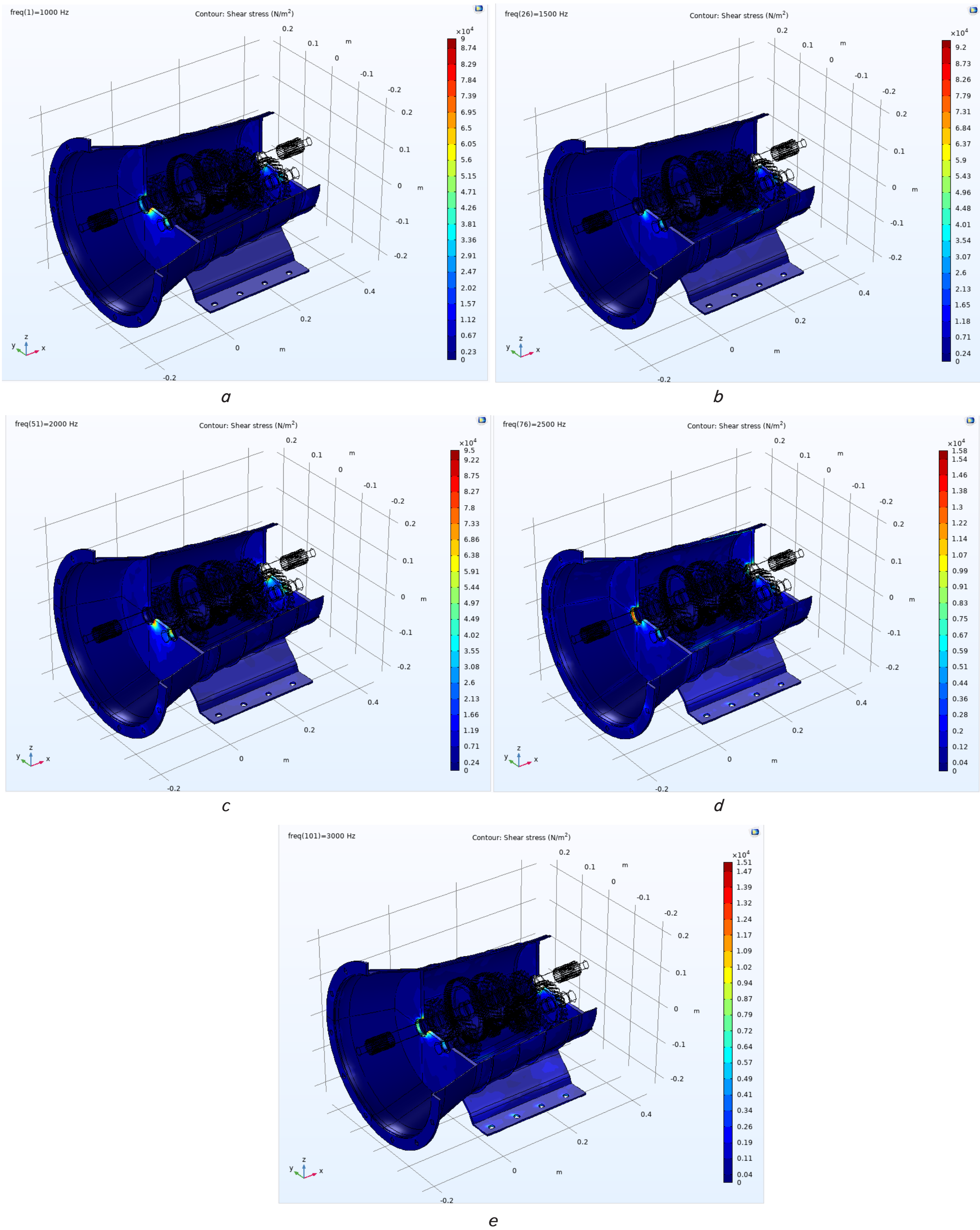


Fig. 8. Shear stress at: *a* — 1000 Hz; *b* — 1500 Hz; *c* — 2000 Hz; *d* — 2500 Hz; *e* — 3000 Hz

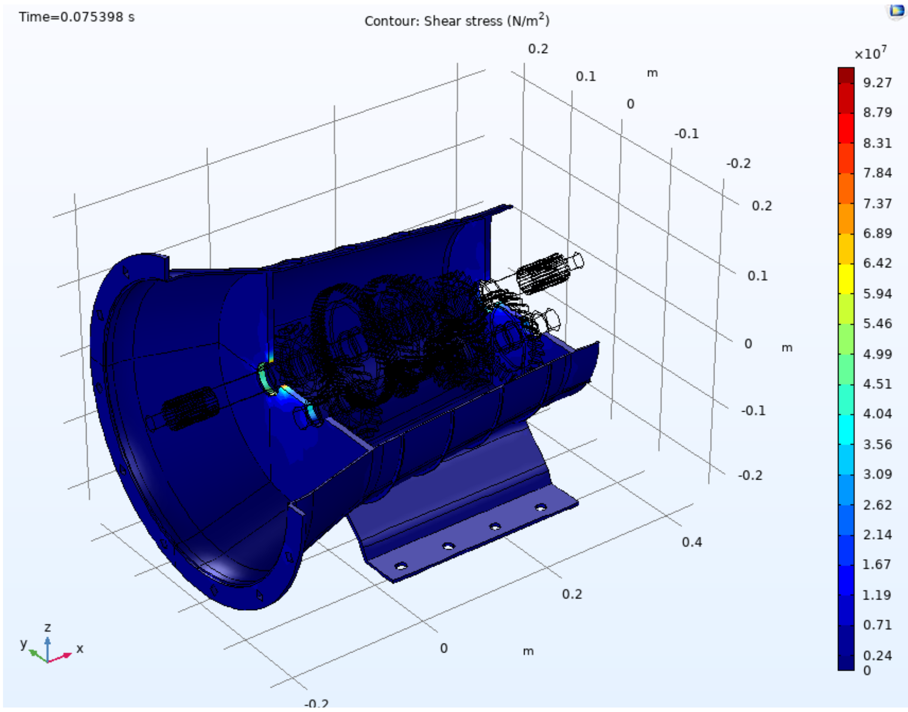


Fig. 9. Shear stress contour

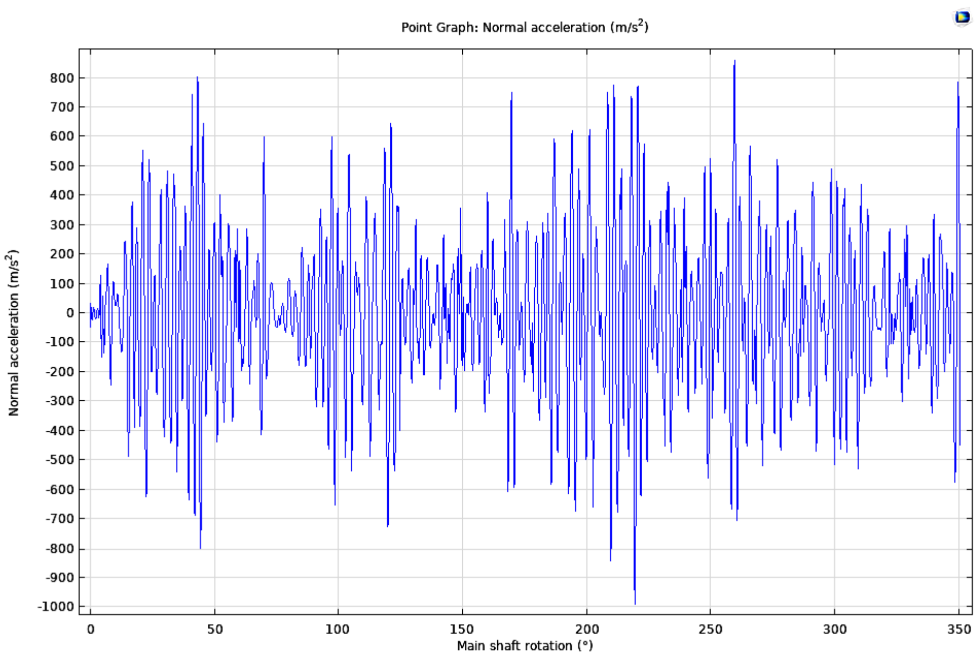


Fig. 10. Acceleration with shaft rotation

As the difference in the amount of acceleration is due to the vibration generated during the rotation of the gears.

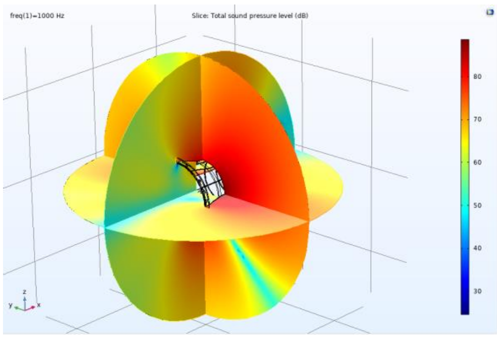
5. 5. Sound pressure of gearbox

As the Fig. 11 represent the outer space of the gearbox, and through movement, vibration works, and the value of the vibration affects the amount of noise generated. Note through the previous Fig. 11 that the value of the vibration 1500 Hz is the highest value that achieved high noise as it was 112 dB, which is the highest value for noise compared to the rest of the vibrations.

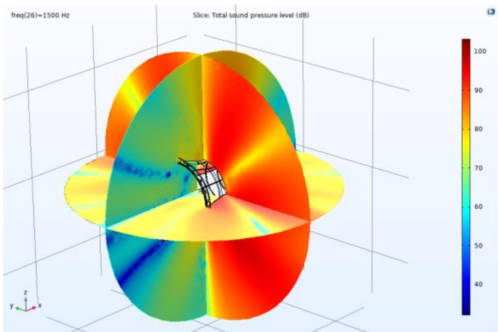
Where confirm the value of the noise coming out of the gearbox when its vibrations, as note, as in the Fig. 12, the achievement of the condition of the presence of large noise when vibration 1500 Hz.

Through Fig. 13, notice that the amount of noise pressure and its wave through the outside air of the gearbox.

Where the amount of wave pressure of the noise reached a maximum value of 400 Pa and the lowest value – 500 Pa, which gives a clear concept about the formation of noise through the movement of the gear inside the gearbox.



a



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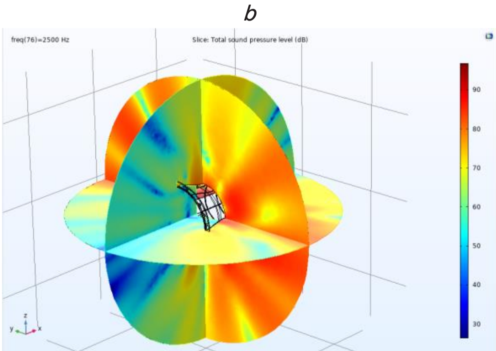
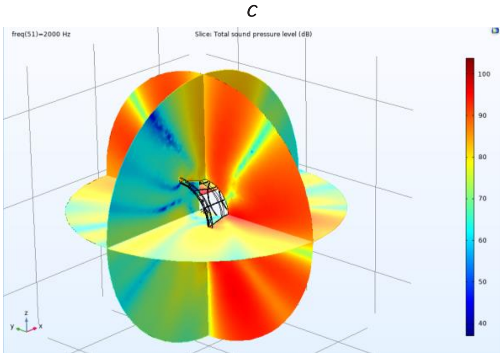
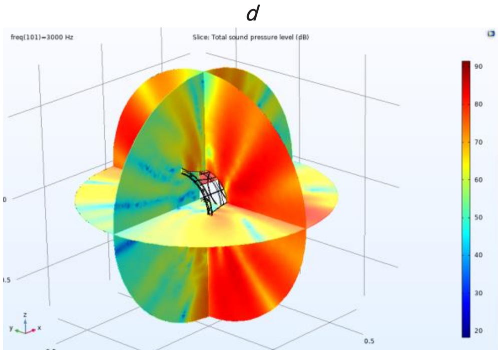


c

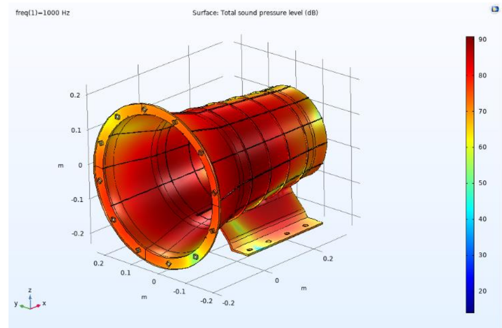


d

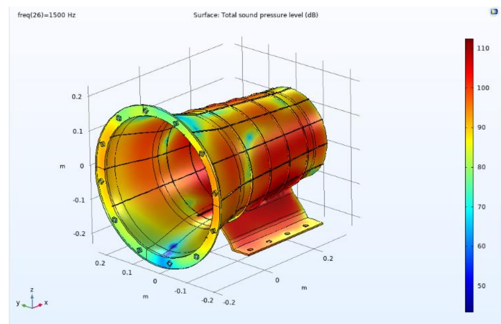


e

Fig. 11. Sound pressure level at: *a* – 1000 Hz; *b* – 1500 Hz; *c* – 2000 Hz; *d* – 2500 Hz; *e* – 3000 Hz



a



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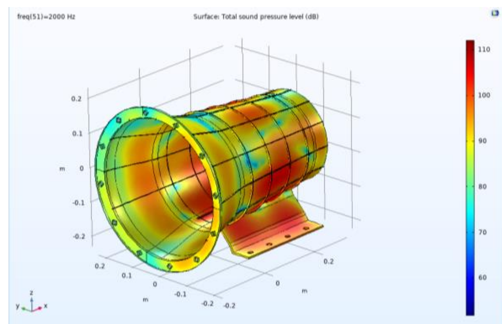

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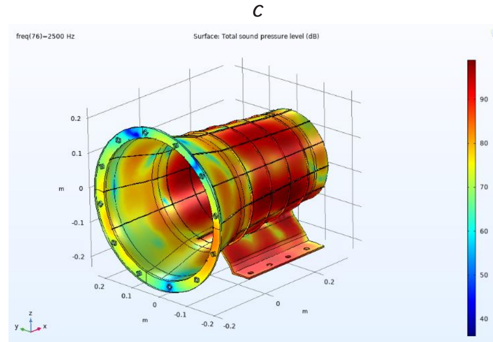

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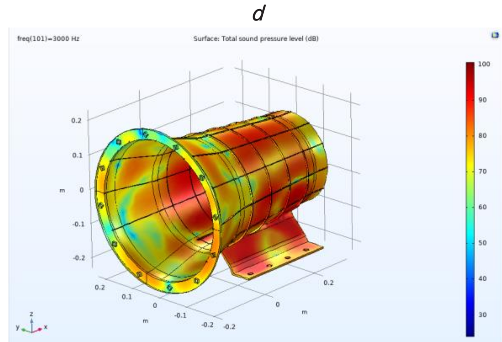

e

Fig. 12. Sound pressure level of casing at: *a* – 1000 Hz; *b* – 1500 Hz; *c* – 2000 Hz; *d* – 2500 Hz; *e* – 3000 Hz

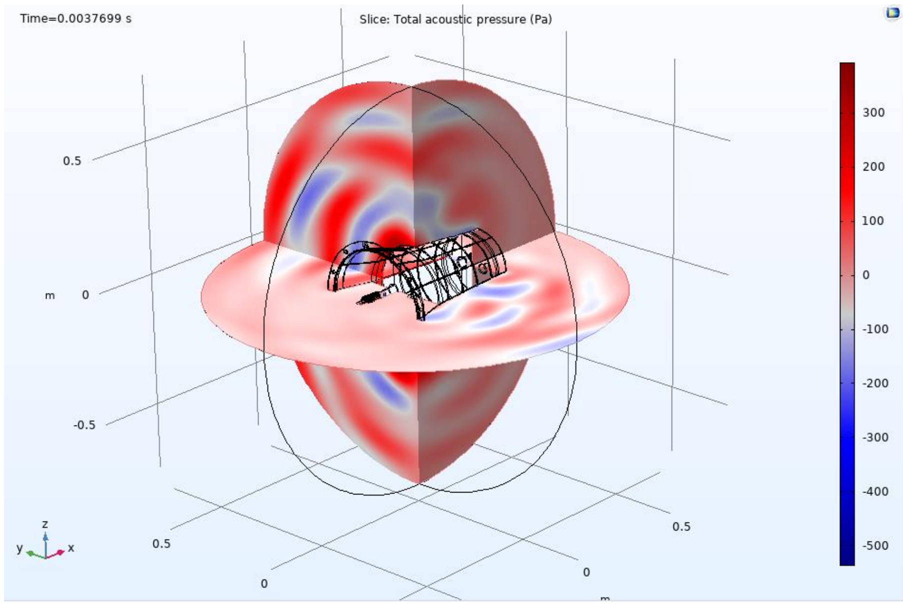

Fig. 13. Acoustic pressure contour

6. Discussion the results of stresses and noise generated by the gear engagement

Whereas the preceding Fig. 4 shows that the linear velocity was 4.8 m/s when the gear completed a full turn. Because of the varied gear teeth, the acceleration changes from place to place, hence the acceleration value in this example was 3000 m/s^2.

Fig. 5 shows the number of stresses created by this movement as a consequence of the rotational movement and vibrations in the gearbox, where the amount of stress reached $2{\cdot}10^7$ Pa. Fig. 6 depicts the influence of the gear rotation angle on the stresses that arise in it. Take note of the variation in the number of stresses acquired throughout a full revolution of the gear, with the greatest value of stresses obtained being $3.5{\cdot}10^8$ Pa for the duration 0.0051 s of rotation.

The effect of vibrations and the Hertzian rate on the value of the stresses is significant, as it is known that at frequencies reaching 1500 Hz, the value of the stresses and deformation was relatively large, as shown in Fig. 7, where the value of the stresses increased to $3{\cdot}10^6$ Pa with frequency change.

The contact areas of the major lip on the outer cover are strongly influenced throughout the gearbox rotation process and also at high vibrations, as shown in Fig. 8, where the highest value of shear stresses reached $9.5{\cdot}10^4$ Pa at frequencies between 1500 and 2000 Hz. When the gears rotated for a complete rotation, the shear stresses reached $9.27{\cdot}10^7$ Pa in 0.075398 s, as shown in Fig. 9, which depicts the gradation of shear stresses across the various locations. Fig. 10 for the value of acceleration with the rotation of the main shaft throughout one cycle, which is comparable to 360 degrees, since the difference in the amount of acceleration is caused by the vibration created by the gears' revolution.

The Fig. 11 depicts the gearbox's exterior space, and vibration operates via movement, and the degree of the vibration determines the quantity of noise created. The prior statistics show that the value of the vibration 1500 Hz reached high noise since it was 112 dB, which is the greatest value for noise compared to the other vibrations. Where to validate the value of the noise coming out of the gearbox when it vibrates, as shown in Fig. 12, the attainment of the condition of the existence of big noise during vibration 1500 Hz. Fig. 13, note the quantity of noise pressure and its wave across the gearbox's outside air. Where the quantity of noise wave pressure reached a maximum of 400 Pa and a minimum of - 500 Pa, this provides a clear idea about the generation of noise via the movement of the gear within the gearbox.

One of the limitations encountered is the method of meshing gears, which are many and have a high mesh. It requires large computers and is able to solve these limits that were encountered.

One of the most important defects that were encountered is that the gears were engaged in a way that instructed a link between the gears and not linking a domain to another domain capable of giving more accurate distortion results on the gears.

The way of meshing gears, which are many and have a high mesh, is one of the restrictions faced. It requires huge computers and is capable of overcoming the limitations faced. One of the most significant flaws discovered was that the gears were engaged in such a manner that a connection between the gears was commanded rather than joining a domain to another domain capable of providing more precise distortion results on the gears.

7. Conclusions

1. The acceleration and due to the different gear teeth, the acceleration value in this case was 3000 m/s^2 that the linear velocity was 4.8 m/s at the time that completes a full rotation of the gear. The acceleration varies from one place to another, so the values in this example are slightly different.

2. The effect of the gear rotation angle on the stresses that occurs in the disturbance in the number of stresses during a complete rotation, where the maximum value of the stresses obtained was $3.5{\times}10^8$ Pa for the time 0.0051 s of rotation.

3. The effect of vibrations and the Hertzian rate is significantly affected by it. At frequencies that reached 1500 Hz the value of stresses and deformation was relatively large. The value of the stresses with the change in frequency increased to $3{\cdot}10^6$ at the frequency 1500 Hz.

4. The contact areas of the main lip on the outer cover are greatly affected during the gearbox rotation process and at high vibrations. The effect of frequency on the shear stress areas, as the maximum value of shear stresses reached $9.5 \cdot 10^4$ Pa at the frequency between 1500 and 2000 Hz.

5. The rotational movement and vibrations that occur in the gearbox, from this movement, which reaches 20000000 Pa. The value of the vibration 1500 Hz is the highest value that achieved high noise, as it was 112 dB, which is a 112 dB higher than the rest of the vibrations. As the previous represent the outer space of the gearbox, and through movement, vibration works, and the value of vibration affects the amount of noise generated. This is where confirm the value of the noise coming out of the gearbox when its vibrations, are measured. The achievement of the condition of the presence of large noise when vibration 1500 Hz is reached is achieved by analyzing the noise produced by a car's gearbox at that level. The amount of noise pressure and its wave through the outside air of the gearbox, where the amount of wave pressure of the noise reached a maximum value of 400 Pa and the lowest value 500 Pa, gives a clear concept about the formation of noise through the movement of gear.

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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