Proceedings of Engineering and Technology Innovation, vol. 22, 2022, pp. 50-59

Effect of Gear Materials on the Surface Contact Strength of Spur Gears

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Received 14 October 2021; received in revised form 21 January 2022; accepted 22 January 2022

DOI: https://doi.org/10.46604/peti.2022.8695

Abstract

In gear applications, the initiation of cracks at or near the contact surfaces of gear mates occurs mostly due to the surface contact mode of gear failure. This study aims to investigate the influence of gear materials on the contact strength of spur gears. In this study, four different gear materials are selected and the contact stress on spur gear mates is analyzed. Hertz's contact stress equation and ANSYS 16 are used for the theoretical analysis and finite element (FE) method, respectively. The results of Hertz's equation are compared with the results of ANSYS 16. The results show that the contact stress on spur gear mates varies when different gear materials are used in both methods. This indicates that the surface contact strength of spur gears is greatly influenced by the type of gear materials.

Keywords: spur gear, contact stress, analytical method, Hertz's equation, finite element method

1. Introduction

Gears are the basic machine elements in a mechanical power transmission system and most industrial rotating machines. Humans have used them for different applications for several years. However, some problems still exist. For instance, gear tooth surface damage, which is caused by wear and surface fatigue, can result in the change of material properties in different working conditions [1-3]. Scholars have investigated the solution to the problems by using various methods. Among them, theoretical and numerical methods have been used frequently due to low research expenses.

Contact stress is the main cause of gear failure. It initiates the crack propagation around the contact surfaces of gear mates, and stress concentration may develop based on it [2, 4-5]. To increase gears' load-carrying capacity and component reliability, it is important to improve the surface strength of gears. Surface strength is the factor that determines the gear life rather than bending strength due to the large enough teeth profile of gears.

Different techniques were adopted by researchers to increase the working ability, efficiency, and working life of gears. Furthermore, the surface life of gears can be improved by subjecting the gear materials to different surface treatment processes. Gear parameters also influence the gear performance greatly [3-6].

Mechanical properties of a material are the properties that encompass a response to an applied load. They reveal the elastic and plastic behavior of the material upon applied forces. These properties are used to classify and identify metals. Strength, ductility, hardness, toughness, impact strength, etc. are some mechanical properties of metals. Strength is the ability of metals to withstand deformation due to external loads, whereas ductility is the tendency of metals to be drawn into wire before fracturing under tensile loads [4-5].

The main purpose of this study is to investigate the influence of gear materials on the contact strength of spur gears by using the theoretical method (Hertz's contact stress equation) and finite element (FE) method (ANSYS Workbench). Both gear designers and manufacturers could take advantages of the findings in this study.

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2. Materials, Methods, and Conditions

The contact stress on spur gears is the main concern in the research field. The analytical, experimental, and FE methods are the basic tools to compute and predict the contact stress on gears. Any method out of the three can support the other one for validating the results [3].

2.1. Analytical methods and conditions

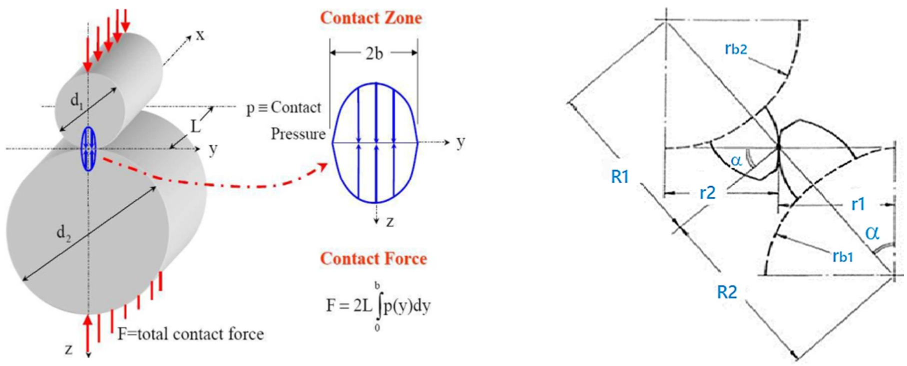
Gear failure is categorized into two general groups: bending and contact failure. Bending failure occurs in the root of gear teeth, whereas contact failure occurs on the gear surface when the gear strength is inadequate. The bending stress can be calculated by the Lewis formula, whereas the Hertz's equation is used for the contact stress. The design procedures developed by American Gear Manufacturing Association (AGMA) and International Organization for Standard (ISO) were also used by the researchers to design spur gears for different applications [5-8].

In machine elements, which work under pressure loading and have rolling-sliding motion, the value of contact stress depends on the size of contact areas. The contact stress can be calculated based on elastic theory. The damage of gear teeth due to the contact stress commonly occurs either on the pitch line (only rolling movement) or away from the pitch line (both sliding and rolling movements) [9].

2.1.1. Contact problem modeling using parallel cylinders

The contact problem modeling is created by using parallel cylinders in order to apply the Hertz's equation to spur gear mates. The mating gears are considered equivalent to the contact cylinders with the same radius of curvature at contact point [9]. Fig. 1 below illustrates the contact modeling.

As shown in Fig. 2 below, the shape of the contact area of the two cylinders is rectangular (2B × L). However, the stress is distributed along the width (2B) with an elliptical shape. The maximum pressure on this contact area occurs along the midway of the width (B). The value of this maximum pressure is calculated by Eq. (1).



(a) Cylindrical contact in compression [10] (b) Contacted spur gear teeth [11]

Fig. 1 Contact problem modeling using parallel cylinders

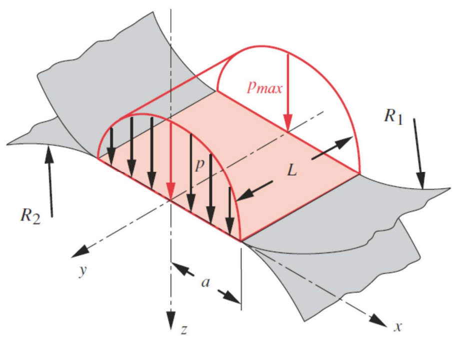


Fig. 2 Ellipsoidal-prism pressure distribution [9]

From the above figure, the maximum pressure, P_{max}, can be formulated as:

$$P_{max} = 2F\pi BL \tag{1}$$

where F is the total contact force. Half of the contact width of the pressure distribution can be obtained by:

$$B = \sqrt{\frac{4FR_e}{\pi LE_e}} \tag{2}$$

where E_e is the equivalent modulus of gear material and R_e is the equivalent radius of curvature.

$$\frac{1}{E_e} = \frac{1}{E_1} + \frac{1}{E_2} \tag{3}$$

$$\frac{1}{R_e} = \frac{1}{R_1} + \frac{1}{R_2} \tag{4}$$

At pitch point, $R_1 = R_p$ and $R_2 = R_g$. This is given by:

$$R_p = d_p \sin\frac{\alpha}{2} \tag{5}$$

$$R_g = d_g \sin\frac{\alpha}{2} \tag{6}$$

where d refers to pitch diameter, p refers to pinion, g refers to gear, and α refers to pressure angle. The equivalent radius will be:

$$R_e = \frac{1}{\frac{1}{R_p} + \frac{1}{R_g}} = \frac{1}{\frac{2}{\sin\alpha}(\frac{1}{d_p} + \frac{1}{d_g})} \tag{7}$$

The tangential and normal force components (F_T and F_N) and the torque (T_p) can be related to the pinion and gear.

$$F = F_N = \frac{F_T}{\cos\alpha} = \frac{T_p}{R_p \cos\alpha} \tag{8}$$

Substituting Eqs. (2), (7), and (8) into Eq. (1), the maximum contact pressure or Hertz stress can be obtained.

$$P_{max} = \sqrt{\frac{T_p E_e}{\pi L R_e R_p \cos\alpha}} \tag{9}$$

In the gear contact kinematics, there are three steps as identified in Fig. 3. In the beginning, the contact occurs through a combination of rolling and sliding movements between the teeth. Then, the sliding movement has the frictional effect, and there is simple rolling contact in the pitch diameter region. After this point, both sliding and rolling movements will occur again [12].

In this study, the authors have considered some assumptions. The materials in contact are considered homogeneous. The load which is normal to the contact tangent plane causes the contact stress. In other words, there are no tangential forces between the surfaces as the contact area is considered very small. Thus, the frictional effect is totally neglected.

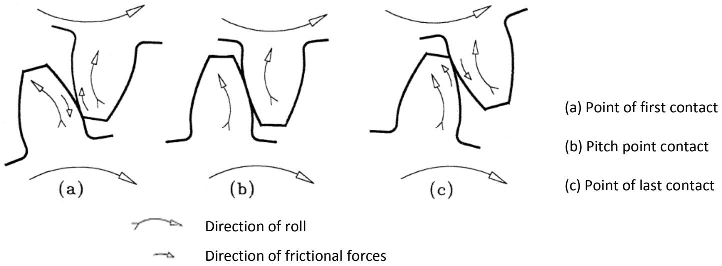


Fig. 3 Mechanics of gear tooth contact [12]

2.1.2. Material requirements

The crucial aspect to consider in gear engineering and design is the selection of gear materials. Strength, durability, and cost (both material and manufacturing cost) are the most important factors when selecting the gear materials. The purpose of material selection is to find the right combination of physical properties that satisfy the requirements of certain applications at the lowest cost. Depending on the shape, nature, and final application of gears, designers and manufacturers choose one from an unlimited number of gear materials to create the finished product. The most commonly used materials include many types of steel, brass, bronze, cast iron, ductile iron, aluminum alloys, powder metals, and plastics [13-14]. Table 1 below displays the mechanical properties of four engineering gear materials selected for this study.

Table 1 Properties of gear materials selected for this study

Selected gear materials	Modulus of elasticity (GPa)	Poisson's ratio	Density (kg/m^3)
Case-hardened wrought steel	206	0.3	7800
ASTM class 35 cast iron	114	0.29	7150
Gray cast iron (GG-30)	91	0.25	7050
Beryllium-aluminum alloy (AlBeMet 162)	179	0.165	2070

2.1.3. Geometrical design of spur gear mesh

Fig. 4 below shows the standard meshing spur gears. The standard meshing spur gears are the reference circles of two gears that contact and roll each other. Based on some set values, all the geometrical specifications of the gears are calculated [15]. In this study, the gear and pinion are considered to have identical geometrical parameters which are presented in Table 2. The input parameters of the mating gear pair (gear train) are specified in Table 3.

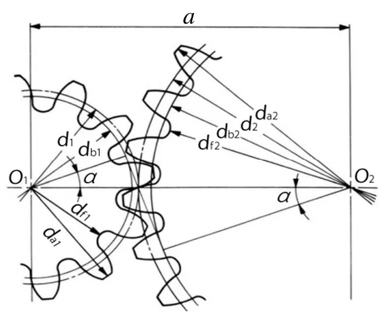


Fig. 4 Geometry of standard meshing spur gears [15]

Table 2 Specification of the meshing spur gears

No.	Description of gear geometry	Symbol	Formula	Value
1	Module	m	Set value	2 mm
2	Number of teeth	z		30
3	Pressure angle	α		20°
4	Face width	b		12 mm
5	Pitch circle diameter	d	$d = m \times z$	60 mm
6	Base circle diameter	d_b	$d_b = d \times cos\varphi$	56.38 mm

Table 2 Specification of the meshing spur gears (continued)

No.	Description of gear geometry	Symbol	Formula	Value
7	Outside circle diameter	d_o	$d_o = m \times (z+2)$	64 mm
8	Addendum	h_a	$h_a = m$	2 mm
9	Center distance	a	$a = \frac{1}{2}(d+d) = d$	60 mm
10	Pitch co-efficient	R_e	$R_e = \frac{1}{\frac{2}{sin\varphi}(\frac{1}{d_p}+\frac{1}{d_g})}$	5.13

Table 3 Input parameters for the gear train

No.	Input parameters	Value
1	Nominal Input power (P)	10 KW
2	Pinion speed (N)	100 rpm
3	Nominal torque on the pinion shaft $T_p = 9550 \times \frac{P}{N}$	955 Nm

2.1.4. *Model calculation of contact stress analysis by Hertz's equation*

Now, let's take grey cast iron (GG-30) material for both gears. By using Eq. (10), the material co-efficient (equivalent modulus) can be obtained as shown in Eq. (11).

$$E_e = \frac{1}{\frac{1-v^2}{E_1}+\frac{1-v^2}{E_2}} \tag{10}$$

$$E_e = \frac{1}{\frac{1-0.25^2}{9.1\times10^4}+\frac{1-0.25^2}{9.1\times10^4}} = 48513.3 \text{ MPa} \tag{11}$$

The Hertz contact stress can be calculated using Eq. (9).

$$\sigma_H = P_{max} = \sqrt{\frac{955000\times48513.3}{\pi\times20\times30\times5.13\times\cos20}} = 2258.03 \text{ MPa} \tag{12}$$

$$\sigma_{vonmiss} = 0.56\times2258.03 = 1264.41 \text{ MPa} \tag{13}$$

By using the same approach, the maximum Hertzian contact stresses of the other gear materials selected for this study are calculated. The maximum contact stresses calculated based on the theoretical analysis (Hertz's equation) are presented in Table 4.

Table 4 Maximum theoretical contact stress for the selected gear materials by Hertz's equation

Selected gear materials	Maximum contact stress by Hertz's equation (MPa)
Case-hardened wrought steel	1194.56
ASTM class 35 cast iron	1177.75
Grey cast iron (GG-30)	1264.41
Beryllium-aluminum alloy (AlBeMet 162)	1325.82

2.2. *Finite element method*

The FE method is a numerical technique used for solving the physical problems encountered in engineering and mathematical modeling. Finite element analysis (FEA) is the simulation of any given physical phenomenon by applying the FE method. FEA is used to mathematically model the physical phenomenon and solve the complex structural, thermal transfer, fluid flow, and multi-physics problems. Engineers use FEA to reduce the number of experiments and/or physical prototypes and optimize the machine elements in their design phase for faster and better product developments while saving expenses. FEA uses mesh generation techniques to divide a complex physical problem into finite small elements coupled with the FE algorithm [16-19].

Many contact problems of rotational machine components can be solved by using the FE method. ANSYS Workbench, one of the frequently used FE software, provides a computerized approach for predicting the response of the object to the applied loads [6, 19]. In the present study, ANSYS 16.0 Workbench package is used to determine the maximum allowable contact stress on spur gears. The steps in ANSYS 16 Workbench solution procedures are explained in the following sub-sections. The FE results will be compared and discussed with the theoretical results in section 3.

2.2.1. Constructing the geometric model of the matting spur gears

The geometric model of the gear teeth profile is constructed by using the SolidWorks modeling software. The geometrical data of the gears are taken from Table 2. The generated model is saved in the ANSYS supported geometry file formats such as IGES and ACIS. Fig. 5 shows a three-dimensional (3D) geometrical model of a single spur gear done by the SolidWorks software.

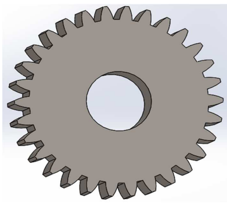


Fig. 5 Three-dimensional model of a spur gear by using SolidWorks

2.2.2. Importing the model to ANSYS Workbench

The generated gear model is imported to the ANSYS Workbench interface for the analysis purpose. All the dimensions and geometries of the gear mesh are calculated and presented in Table 2 above. The imported model of assembled gears is presented in Fig. 6 below.

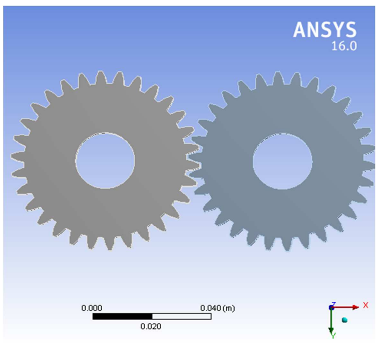


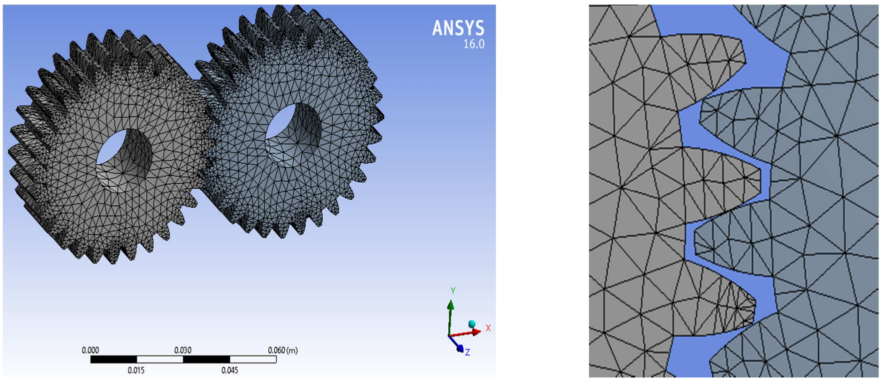
Fig. 6 Assembled spur gear set

2.2.3. Specifying the material properties

Engineering materials may be specified after the geometry is imported. The specific properties of gear materials under analysis, such as Young's modulus, Poisson's ratio, and density, are used for FEA. In the ANSYS Workbench interface, the engineering data can be selected from the analysis tab and the corresponding values for each material can be inserted.

2.2.4. Conducting the mesh refinement

Meshing is the process of dividing the entire model into small cells so that at every cell the equations are solved. It gives an accurate solution and improves the quality of the solution. Mesh refinement is conducted in order to generate fine mesh elements. Fig. 7 shows the mesh model of the gear mates.



(a) Refined mesh of gear assembly (b) Close-up view of tooth flanks

Fig. 7 Meshed assembly in ANSYS Workbench simulation

2.2.5. *Setting the boundary condition*

Boundary condition refers to the external load on the border of the model structure. This is normally done by considering that the tooth contact is at the pitch point as illustrated in Fig. 3(b). In the lower gear, the fixed support boundary condition is introduced. Besides, the other gear is subjected to frictionless support and torque/moment in a clockwise direction as illustrated in Fig. 8 below.

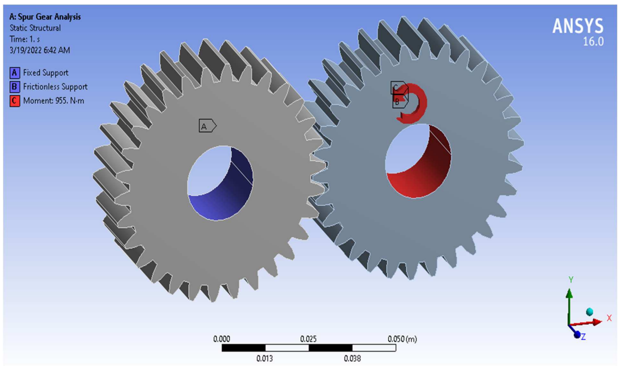
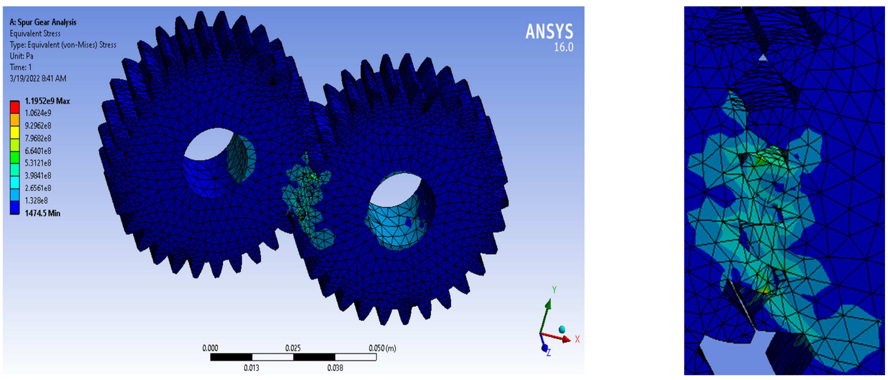


Fig. 8 Setting the boundary condition

3. Results and Discussion

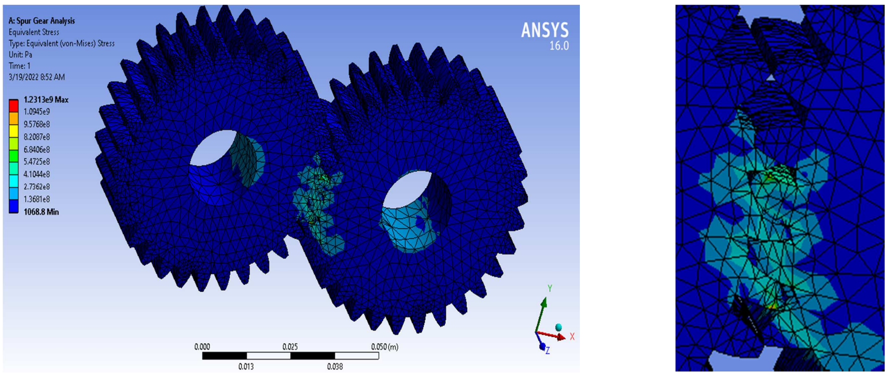
3.1. *Results*

The Von-Mises stress at the contact region for the contact model of mating spur gears is displayed on the solution processor of ANSYS 16 Workbench. The images below (Figs. 9-12) show the values for four different materials considered in this study.



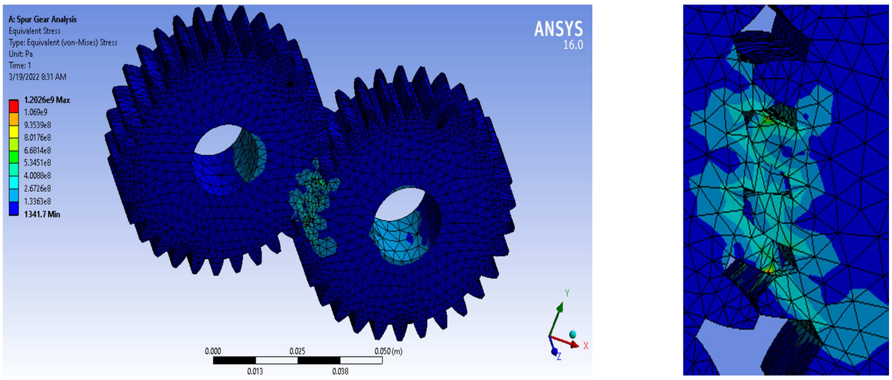
(a) Equivalent stress for case-hardened wrought steel (b) Close-up view of tooth

Fig. 9 Von-Mises stress for case-hardened wrought steel



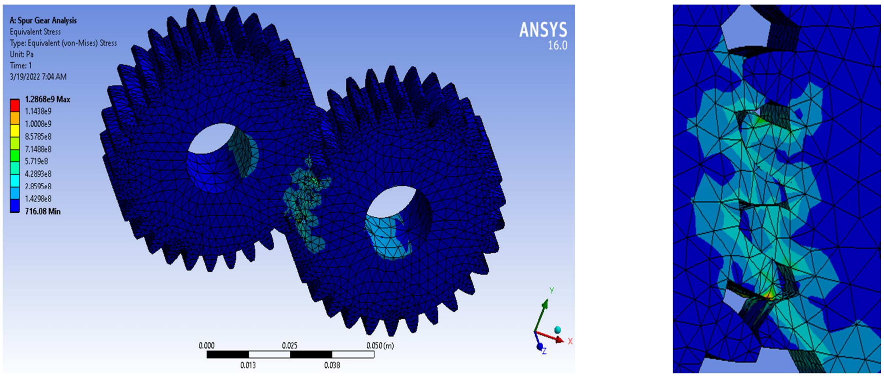
(a) Equivalent stress for grey cast iron GG-30 (b) Close-up view of tooth

Fig. 10 Von-Mises stress for grey cast iron GG-30



(a) Equivalent stress for ASTM class 35 cast iron (b) Close-up view of tooth

Fig. 11 Von-Mises stress for ASTM class 35 cast iron



(a) Equivalent stress for AlBeMet 162 (b) Close-up view at tooth

Fig. 12 Von-Mises stress for Beryllium-aluminum alloy (AlBeMet 162)

3.2. Discussion

The main aim of this study is to analyze the change of the contact stress on spur gears with different gear materials. The analysis is done by varying the gear materials under constant loading conditions and geometric parameters by using the Hertz theory and FE methods (ANSYS 16). The results of the analysis by these two methods are compared and presented in Table 5.

The selection of appropriate gear materials for different applications is the key task in gear design and/or manufacturing processes. The use of these different gear materials provides a range of contact stress which is useful in material selection [9]. As illustrated in Table 4, the results of contact stress in spur gear mates using the theoretical (Hertz's equation) and FE method (ANSYS 16) agree with each other with a maximum error of 2.9%. This consequently indicates that the FE simulation (ANSYS Workbench) is compatible and almost identical with the theoretical analysis (Hertz's equation) for the materials under analysis. This fact is demonstrated in Fig. 13.

Table 5 Comparison of maximum contact stress for the selected gear materials

Selected gear materials	Maximum contact stress by ANSYS 16 (MPa)	Maximum contact stress by Hertz's equation (MPa)	Error (%)
Case-hardened wrought steel	1195.2	1194.56	-0.05%
ASTM class 35 cast iron	1202.6	1177.75	-2.1%
Grey cast iron (GG-30)	1231.3	1264.41	2.6%
Beryllium-aluminum alloy (AlBeMet 162)	1286.8	1325.82	2.9%

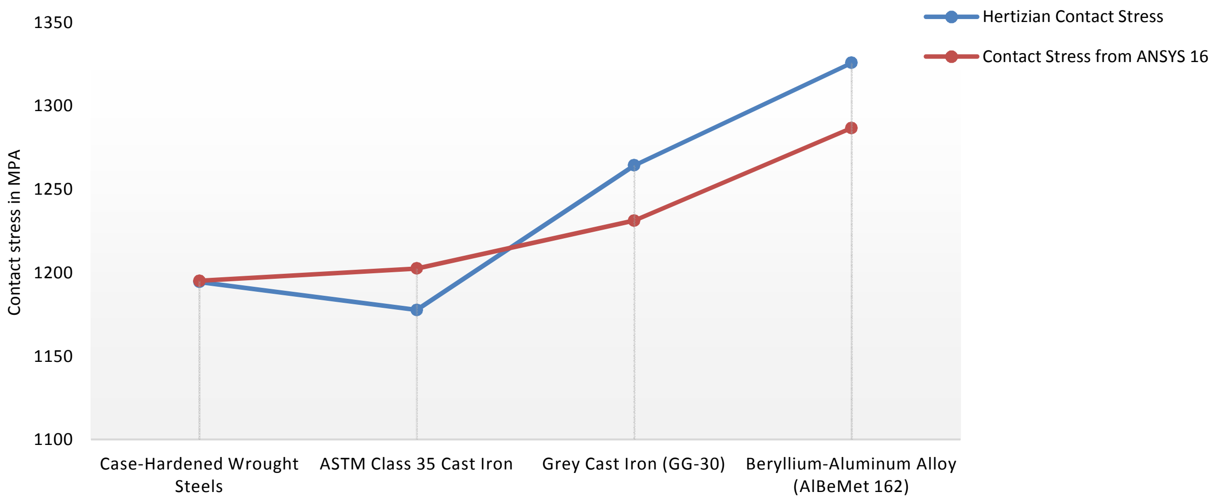


Fig. 13 Comparison of the contact stresses obtained by theoretical analysis and ANSYS simulation

Fig. 13 shows that the contact stress from each material is different, though the analysis is done by keeping all conditions constant except the type of materials. This indicates that the type of gear materials has a great influence on gear failure due to the contact stress.

4. Conclusions

The surface strength of the gear tooth is a crucial parameter to prevent the failure of mating gears. In this study, the theoretical approach is used to predict the surface contact stress on the spur gear mates, and is done on different gear materials to study their influence on the surface contact strength of spur gears. Moreover, the study shows that the effective way of predicting the surface contact stress is to use the 3D geometrical contact model of spur gears under the FE method (ANSYS 16). The accuracy of the FE results for the selected materials is verified by comparing the FE results with the analytical results using Hertz's standard formula.

The meshed in-volute spur gear is properly modeled by rotating cylindrical contacts for the theoretical investigation of surface contact stresses under the applied torque/moment. The 3D spur gear model is done in the modeling software (SolidWorks) and imported to ANSYS 16 Workbench for FEA for the selected gear materials. The correlation between the theoretical formulation and FE results is compared with tolerable accuracy. The comparison shows that the results are well-matched with a maximum of 2.9% accuracy error.

Both the results of FE and theoretical methods show that the value of contact stress varies as the gear materials are changed. This indicates that the type of gear materials has a direct influence on the surface contact strength of spur gear mates. Thus, gear designers and/or gear manufacturers have to consider gear materials as a factor for the surface contact failure of gears.

Conflicts of Interest

The authors declare no conflicts of interest.

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