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Bevel Gears Strength Calculation: Comparison ISO, AGMA, DIN, KISSsoft and ANSYS FEM Methods

Gurkan IRSEL*

Keywords : root bending stress, contact stress, ansys finite element analysis, KISSsoft, DIN 3991:1988.

ABSTRACT

Gear strength calculation must be done in an appropriate and productive way to develop a strong and compact movement transmissions systems efficiently. In this research, the advantages and disadvantages of the calculation methods and the results of the calculation on a sample application are compared by referencing DIN 3991 method solution. Manual solution is an impractical and error-prone method. A max 27.5 % difference was found between ISO and AGMA analytical results. There is also a 17.3 % difference between KISSsoft solutions of the same methods. The KISSsoft solution gives immediate results in the event of changing different modules and revolutions. Modeling and mesh studies are required in ANSYS solution. Mesh affects the Ansys solution directly. There is 7 % difference between the Ansys FEM and KISSsoft (DIN) solutions. However, ANSYS enables the calculation in non-standard gears and modified gears. In this study, current gear calculation methods are presented, compared and evaluated in detail on an application.

INTRODUCTION

Gear is one of the most critical component in a mechanical power transmission system, and most industrial rotating machinery. Bevel gear is cylindrical in form and has teeth, which are of involute form in most cases (Karaveer et al, 2013). A pair of bevel gear teeth in action is generally subjected to two types of stresses: bending stresses and contact stress. Various research methods such as Theoretical, Numerical and Experimental have been done throughout the years.

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* *Assistant Professor, Department of Mechanical Engineering, Trakya University, Faculty of Engineering, Edirne, 22100 TURKEY.*

While the motion is being transmitted, repeated contact pressure and tooth bending stress occurs in gear pairs. This repetitive loading pattern frequently causes cracks in the tooth base and pitting damage to the tooth surface (Maršálek and Moravec, 2013, Silori et al., 2015). Therefore, the calculation of the amount of power that a gear can transmit for its intended operating time is significant (Doğan and Karpat, 2019). The basis of the power calculation that a gear can transmit is based on the mathematical model developed by Wilfred Lewis in 1892. While Lewis was developing this equation, he considered the tooth as a built-in object and calculated the maximum stress value of the force affecting on the tooth. Today, the AGMA method is still based on the Lewis equation. AGMA 2003 B97, ISO 10300: 2001 Method B and DIN 3991: 1988 presented in this study will be briefly referred to as AGMA, ISO, DIN respectively. Today, AGMA, DIN and ISO standards are analytical methods grounded on built-in beam bending stress. These methods calculate the bottom bending stress and contact stress by including the factors affecting the gear (loading method, stress correction factor, dynamic factor, etc.) to the mathematical model. They compare the results of the calculations not only with the strength values of the material, but also with the limit values calculated by their own factors such as permitted contact stress.

The factors that are used in the mathematical model of analytical methods are obtained with the help of calculations and tables. In formulations, manual application of many factors such as moment, rotation speed, lubrication form, material, surface properties, mounting type, loading type is an exhausting and time consuming process. Therefore, MATLAB applications were made (Singh and Dewangan, 2015). Applications like KISSsoft were developed. KISSsoft is a software that can carry out sizing, optimization and strength calculations for gears in a short time by using standards such as DIN, AGMA, ISO. Profile and length modifications can be defined and CAD data can be created in KISSsoft.

Finite element method is an effective and accepted solution method in defining complex geometries (Wei et al., 2018). Wilcox and Coleman (1973) compared the results of analytical calculations of gear with the numerical finite element method in 1973. The use of finite element method for gears has continued since the 1970's and has become increasingly popular (Aslantaş and Taşgetiren, 2004, Sun et al., 2018). ANSYS is the most common FEM software in gear calculation (Celik and Akinci, 2016, Zhan et al., 2015, Qin and Guan, 2014, Fetvaci and Erdem, 2004). It was seen that the ANSYS results were accordant with analytical results (Jadeja et al., 2013, Wen et al., 2018, Doğan et al., 2018, Sahu et al., 2017). However, generating the gear model and the mesh structure in ANSYS software requires expertness and iterative solutions. In particular, generating 3D gear mesh structure, the use of simple gear model, mistaken limiting conditions and poor mesh quality lead to false and misleading results (Lisle et al., 2017, Noaman, 2017). Therefore, experimental verification for structural stress analysis, especially the use of strain gauge, has become widespread.

This study focuses on comparing gear tooth bending stress calculation methods analytical, numerical and FEM. Thus, the KISSsoft (DIN) result is assumed that the calculation is correct. Manual solutions has been developed for bevel gear pairs with AGMA, ISO, DIN methods and the numerical solutions has been developed with KISSsoft and ANSYS.

DESIGN of GEARS

Bevel gears are conical and they are used to transmit the rotating power over shafts with 90 degree angle. As they are conical their modules are variable and they are defined by external module and mean module. In this study a pair of single-stage standard bevel gears were handled. Aim of the system is transmitting 160 Nm torque with a speed of 500 rpm. General geometric properties of gears are shown in Table 1.

Table 1. Geometric properties

Tooth geometry		Pinion		Gear
Shaft angle (°)	θ		90°	
Mean normal module	[mmn]		3.332	
Pressure angle (°)	[alfn]		14.5	
Number of teeth	[z]	19		17
Face width	[b]	17		17

The bevel gears material was selected to be as 16mnCr5 steel. CAD models of the gears were made with SolidWorks parametric solid modeling software. The gears were arranged with CATIA. Gear hubs were opened. The model for mounting the gears is shown in Figure 1.

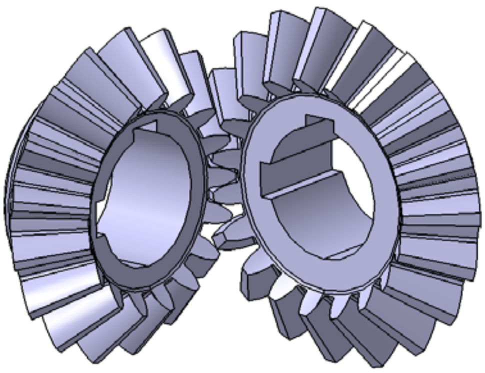

Fig. 1. Contact view of bevel gears

Analytical Calculation

Analytical calculation of the bevel gear set was made with ISO, AGMA and DIN methods These methods have been developed with data obtained from gear tests that lasted for decades. In analytical calculation, bending stress at the root of the tooth and contact stress of the tooth are calculated and compared. Comparison stress (allowable stress) is calculated by including strength values of materials and many factors such as dynamic factor and stress-intensity factor. The gears are damaged after a certain time when the calculated admissible stress values (effective contact stress (σ_{H0}, N/mm^2) and/or tooth root stress (σ_{F0}, N/mm^2) are exceeded or instantly in the form of tooth breakage. Table 2 shows the properties of the gear material, 16MnCr5.

Calculation of Bevel Gear Pair with DIN Standard

Bending strengths of the gear were calculated according to DIN 3990 standard. DIN 3990 calculation method are based on the comparison of the generated stress with the allowable stress. Here, the permitted voltage value is not yield strength. It can be seen from Table 3 that the allowable voltage is higher than the yield strength and this value is different for each standard. The calculation is presented as Equation 1 (Lisle et al., 2019) (A. Singh, 2018).

$$\sigma_{F0} = \frac{F_t}{b^{*}.m_n} . Y_F . Y_S . Y_{\in} . Y_{\beta} \qquad (1)$$

σ_{F0} nominal bending stress number, Y_F tooth form factor, Y_S stress correction factor, Y contact ratio factor, Y_β helix angle factor, b^* assumed and measured contact pattern width (mm).

$$b^{*} = 0.85.b, Y_F = 2.82, Y_S = 1.56, Y_{\in} = 0.65, Y_{\beta} = 1$$

Table 2. Pinion and gear material properties

Property		Pinion	Gear
Tooth material		16MnCr5	16MnCr5
Surface hardness (HRC)		63	63
Fatigue strength tooth root stress (N/mm^2)	$[\sigma_{Flim}]$/[sat]	430.00	430.00
Fatigue strength hertzian pressure (N/mm^2)	$[\sigma_{Hlim}]$/[sac]	1500.00	1500.00
Tensile stress (N/mm^2)	$[\sigma_B]$	1000.00	1000.00
Yield point (N/mm^2)	$[\sigma_S]$	695.00	695.00
Young modulus (N/mm^2)	[E]	206000	206000
Poission's Ratio	[v]	0.3	0.3

$\sigma_{F0} = 292\ MPa$

Bending strength of the gear was calculated according to ISO standard. The calculation is presented as Equation 2.

$$\sigma_{F0} = \frac{F_t}{b.m_n}.Y_F.Y_S.Y_\in.Y_\beta$$

$\sigma_{F0} = 249.05\ MPa$

Bending strength of the gear was calculated according to DIN Static standard. The calculation is presented as Equation 3.

$$\sigma_{F0} = \frac{F_t}{b^*.m_n}.Y_F.Y_\in. \quad (3)$$

$\sigma_{F0} = 188.77\ MPa$

AGMA Calculation

σ_F the bending stress at the root of the tooth (N/mm^2 , MPa), met is the metric outer transverse module (mm), Y_x is the size factor, Y_β is the tooth lengthwise curvature factor, Y_J is the bending strength geometric 80 factor, Tq1 is operating pinion torque (Nm), KA is overload factor . K_v is dynamic factor, $K_{H\beta}$ is load distribution factor, met is outer traverse module (mm) (Sekercioglu and Kovan, 2007, Zhan et al., 2015).

$$\sigma_F = \frac{2000.T_{q1}}{F.d_{e1}}.\frac{K_A.K_\vartheta}{m_{et}}.\frac{Y_x.K_{H\beta}}{Y_\beta.Y_j} \quad (4)$$

$T_{q1} = 160\ (Nm), K_A = 1, K_\vartheta = 1, Y_x = 0.520,$

$K_{H\beta} = 1, F = 17, d_{e1} = 68, m_{et} = 4, Y_\beta = 1,$

$Y_j = 0.17$

$\sigma_F = 211.68\ MPa$

KISSSOFT CALCULATION

KISSsoft is a software to calculate gear sizing, optimization and strength. The program is used to determine the strengths of the components, review the calculations, optimize sizing and report the safety coefficients and parameters of product life on the basis of components or complete system by including the dimensions and technical specifications in the program. All calculations in the program are made in accordance with the preferred standard such as DIN, ISO, AGMA. The calculation screen for 160 Nm and 500 1/min 100 h operation is shown in Figure 2a and the gear design is shown in Figure 2b.

Minimum service life of gears is generally accepted as 100 hours in the tables. KISSsoft calculation results of the system for 100 hours are given in Table 3. SF is safety factor (Bending), SH is safety factor (Pitting), σ_{FP} [sat] is allowable bending stress number, σ_H is contact stress number, σ_{HP} [sac] is allowable contact stress number, R_{eq} SF is required safety factor for bending, R_{eq} SH is required safety factor for pitting.

Calculation values for 1000 and 10000 hours operating time of gear pair obtained with KISSsoft in accordance with DIN, ISO and AGMA are given in Table 4. Load factor is 1.25. When the safety factors are considered, it is seen that there is no problem for the tooth root strength of the gear pair but the loading is at the limit for pitting formation.

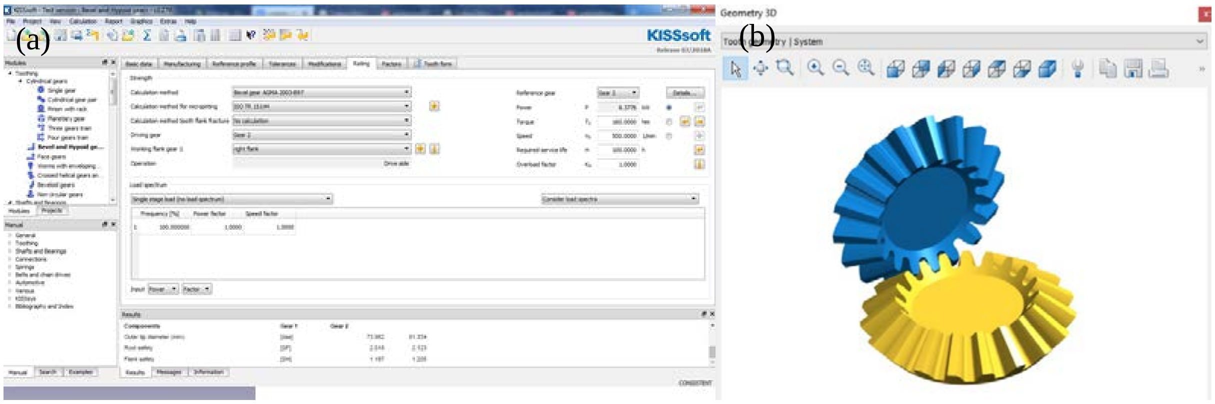

Fig. 2. a) KISSsoft gear strength calculation screen b) KISSsoft tooth geometry system

Table 3. KISSsoft nominal stress calculations

Method	H[hr]	K_A	SF	SH	σ_{F0} (MPa)	σ_F (MPa)	σ_{FP} (MPa)	R_{eq} SF	σ_{H0} (MPa)	σ_H (MPa)	σ_{HP} (MPa)	Req SH
AGMA	100	1.0	2.018	1.197	221.11	-	446.18	1.0	-	1766.34	2114.05	1.0
DIN	100	1.0	2.315	1.104	352.85	667.22	816.12	1.4	1126.61	1549.20	1730.97	1.4
ISO	100	1.0	2.714	1.069	299.93	567.87	814.30	1.4	1160.97	1597.48	1722.70	1.4
StaticDIN	Norm	1.0	3.010	-	230.88	-	695.0	1.0	-	-	-	1.4

Table 4. Results of KISSsoft dynamic gear analysis

Method	H [hr]	K_A	SF	SH	σ_{F0} (MPa)	σ_F (MPa)	σ_{FP} (MPa)	σ_{H0} (MPa)	σ_H (MPa)	σ_{HP} (MPa)
AGMA	1000	1.25	1.550	0.932	-	276.39	428.26	-	1974.83	1840.41
DIN	1000	1.25	0.979	0.830	352.85	833.27	816.12	1126.61	1731.28	1453.77
ISO	1000	1.25	1.097	0.804	299.93	709.01	777.59	1160.97	1785.00	1435.25
AGMA	10000	1.25	1.487	0.81	-	276.39	411.07	-	1974.83	1602.19
DIN	10000	1.25	0.979	0.805	352.85	833.27	816.12	1126.61	1731.28	1410.49
ISO	10000	1.25	1.047	0.736	299.93	709.01	742.53	1160.97	1785.00	1353.57

FINITE ELEMENT ANALYSIS

ANSYS provides more accurate and precise results than analytical methods, but it requires comprehensive solution tools. Capacity of the processor, volume of the model and selected mesh method and its intensity affects obtaining the accurate result. The varying results of the finite element method depending on the mesh structure require experience based on experimental studies.

Since ANSYS is an analysis-oriented software, it is necessary to perform modeling processes with professional modeling software such as SolidWorks and CATIA and compulsory for complicated geometries. Since FEA analyzes require iterative solutions, the analysis model should focus on the part of interest. For gears with symmetrical geometry, a model with several teeth is sufficient for analysis. This process provides the skill to examine a part of the model, requires lesser processor capacity, lesser time and provides the skill to get accurate results.

ANSYS Gear Analysis Procedure

Gear sets, solid models and assemblies of which are above were arranged in accordance with CATIA analysis. These solid model files with stp extension were imported in ANSYS static structural model. The material of the model is defined as 16MnCr5 case-hardening steel in the ANSYS static structural analysis module. Joints were defined for gears. Gears, Contacts, joints, frictionless contact, momentum defined over joints and rotation are seen in Figure 3a. Frictionless Contacts properties were determined as; Behavior: Asymmetric, Formulation: Augmented Lagrange, Geometric Modification Interface Treatment: Adjust to touch. Rotation is defined as 0-60 degrees, torque value is fixed and defined as 160 Nm. Progressive analyses were made.

Mesh quality fine, mesh metric skewness average value is 0.26. Mesh statistics values are: 70027 nodes, 47712 elements (quadratic). Skewness value and number of elements comply with ANSYS mesh evaluation criteria. Mesh convergence change is % 0.018. Stress analyses of the gear pairs were made with Ansys (Guddad and Venkataram, 2017) Figure 3b shows the contact of the gears and the spherical mesh configuration of 26 mm diameter. Max element size is 1 mm. The surface local mesh size is 0.175 mm.

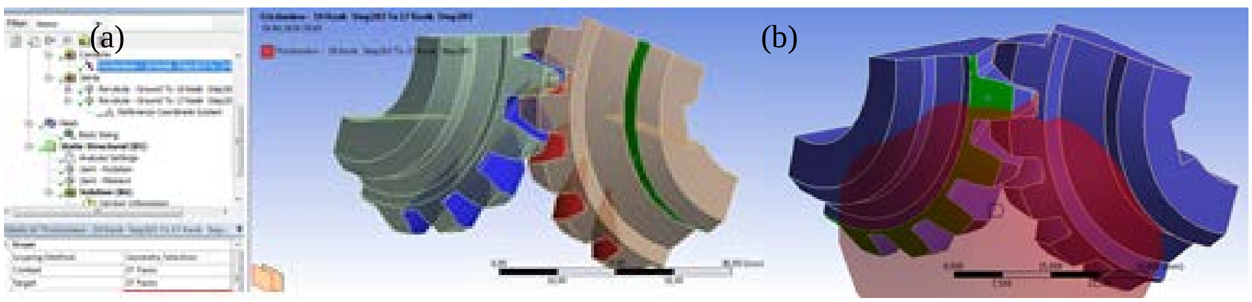

Fig. 3. a) Analysis procedures b) Spherical mesh definition

The analyses were solved by cumulative iteration. Large deflection was selected as on and solution output as force convergence and the graph of the solution was obtained (Figure 4). Solutions were performed for both pairs with similar procedures. The analysis solution for one gear pair lasted approximately for 2.5 hours. There are no singularity problems in the solution.

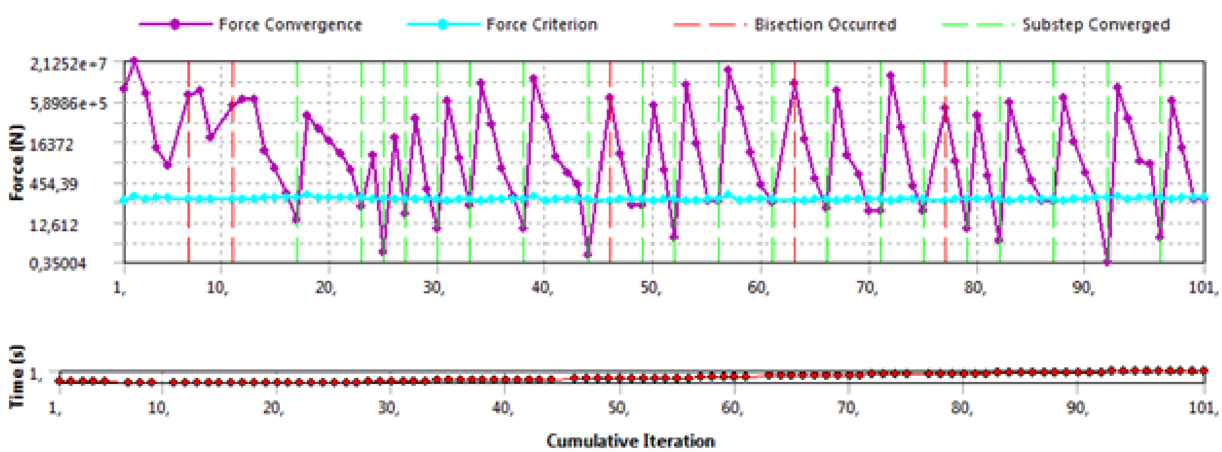

Fig. 4. Force convergence solution

Maximum bending stress of tooth root for the analysis is 328.13 MPa (Figure 5). Maximum tooth contact stress for the analysis is 1426.9 MPa (Figure 5). Stress distribution is not homogeneous

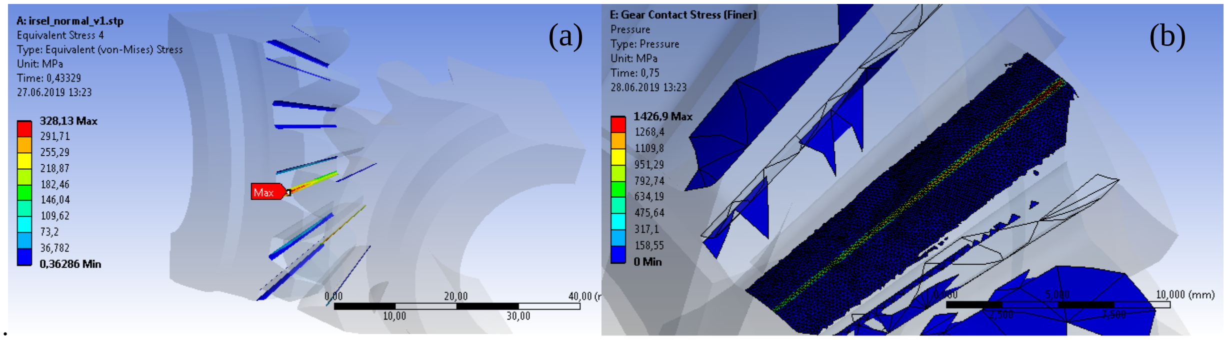

Fig. 5. a) Tooth root stress analysis, b) Contact tooth stress analysis

RESULT AND DISCUSSION

Analytical calculations of the gears were first made in 1890s and gear calculation by finite element method was first made in 1970s. Comparison of analytical and FEA results still continues at the present time. (Lin and Fong, 2015, Li et al., 2011, Wang et al., 2017, Mu et al., 2018, Guan et al., 2019, He and Lin, 2017). The aim of commercial companies and researchers is to develop an automated solution method with FEA software.

The finite element method comes up with successful results in comprehensive and versatile analysis of engineering problems. It is possible to obtain coherent results with analytical methods and experimental data by selecting the right mesh, determining the proper workstation, selecting appropriate solution method.

Comparison of ISO, AGMA, DIN, KISSsoft and ANSYS Results

The gears are the most challenging machine elements in terms of calculation. Gears are dynamic elements 145 and exposed to fatigue. The materials used in the manufacture of gears have fatigue strength tooth stress and the fatigue strength for hertzian pressure endurance values are shown in Table 2.

Analytical calculations are made with a large number of parameters but if made manually, they are long processes requiring effort. Manual analytical calculations table reading, acceptances and therefore calculation errors may occur. The best way to make an accurate iterative analytical calculation in a short time and faultlessly is using software products. In this context, KISSsoft is a reliable and continuously updated software. Analytical methods can be performed with all software products that can calculate the formulations. MATLAB is one of them (Tunalioğlu and Tuç, 2014, Singh and Dewangan, 2015) and it is very popular. According to DIN method, there is a 17.3% difference between KISSsoft calculations and manual (analytical) calculations. When compared to the manual calculation, the software evaluates the calculations and tables more effectively. Analytical result of KISSsoft (DIN) is accepted as accurate. Analytical and KISSsoft differences and results are shown in Table 5. The differences are due to calculation methods of scale factors and table reading process.

Table 5. Analytical and KISSsoft results

Method	KISSsoft	Analytical	Difference %
AGMA	221.11	211.68	4.2
DIN	352.85	292	17.3
ISO	299.93	249.05	16.9
Static/DIN norm	230.88	188.77	18.2

Although analytical methods use similar methodology, they present different tooth root and contact stress values for the same problem. The calculated stress values and the allowable limit values for the gear are 160 different for each method. While AGMA tooth root stress is 221 MPa, the result in DIN method is 352 MPa. However, while the allowable limit stress value for AGMA is 446.18 MPa it is 816.12 MPa in DIN method (Table 6).

ANSYS is not a software specially developed for gear calculation. The impact of dynamic factors cannot be calculated with a static stress analysis. Therefore, ANSYS analysis results can be compared with analytical nominal stress values, where dynamic impacts are not taken into account (Ka=1, Kv=1, Keps=1). The tooth root stress calculated with ANSYS is von mises stress. ANSYS analysis was presented as a procedure and the analysis was made for 0-60 degrees rotation in steps. FEA contact stress "contact tool pressure" was used to calculate the nominal stress on the surface of the tooth. There is a 7% difference between ANSYS results and DIN tooth root stress result and the relationship between other analytical methods and FEA result is shown in Table 6. The slightly higher FEA results are caused by the non-homogeneous stress distribution as seen in Figure 5.

Table 6. ISO, AGMA, DIN, KISSsoft and ANSYS results

Method	KA	SF	SH	σ_F (MPa)	σFP (MPa)	Root Bending Error (%)	σ_H (MPa)	σHP (MPa)	Contact Stress Error (%)
FEA	1.0	2.11	1.051	328.13	695	7.00	1426.9	1500	7.89
AGMA (KISSsoft)	1.0	2.01	1.197	220.11	446.18	37.61	1520.06	2114.05	1.88
DIN (KISSsoft)	1.0	2.31	1.104	352.85	816.12	0.00	1549.20 1126.61	1730.97	0.0
ISO (KISSsoft)	1.0	2.71	1.069	299.93	814.30	14.99	1597.48 1160.97	1722.70	-3.11
Static/DIN norm (KISSsoft)	1.0	3.01	-	230.88	695	34.56	-	-	-

Productivity of Calculation Methods

Calculation of gears is mostly made with analytical methods and basing on experience at the present time. Computer aided engineering calculations and strength calculations can be made by scientific organizations and enterprises that have R&D departments. While analytical methods do not require expertise, gear calculation with ANSYS requires time and experience in processes such as solid modeling, assembly, creation of analysis solution procedure, repetitive mesh configuration. Approximate solution times for analytical, numerical and FEA are seen in Table 7.

Table 7. Calculation times

Method	KISSsoft	Manual	ANSYS
Calculation time	~5 mins	~1.4 hours	~24 hours

CONCLUSIONS

This paper described the advantages and disadvantages of different calculation methods and the results of the calculations methods of tooth bending stress of bevel gears.

- The analytical gear calculation methods such as AGMA 2003 B97, ISO 10300: 2001 Method B and DIN 3991: 1988 make the calculations based on tooth root bending and contact stress. However, analytical methods are based on similar calculations, it was determined that the numerical variable approach adopted in the safety factor and characteristic parameters causes differences in the results.
- ANSYS does not provide permissible gear fatigue data. But ISO, DIN and AGMA, albeit with their differences, provide material data that has been established experimentally using their respective procedures.
- With this study, the KISSsoft software company removed the situation of switching to static calculations in low lifetimes applied in DIN calculation method. The bending stress value will not change as of the 2019 version for the static case.
- It was revealed that the sensitivity of the KISSsoft software for determining the value of the parameter in the calculation method for gear and gear systems was higher than other manuel solution methods.
- It was determined that it is not possible to include some factors such as dynamic loads, loading type factors, lubrication type, surface hardening in the calculations as required in ANSYS. In this context, it can be stated that the results of standard analytical solution methods and

Ansys results in gear calculations can only be compared for the static case.

• This study revealed that the calculation methods and their comparison criteria are different. For example, in Table 3, AGMA tooth root stress value is 221 MPa and DIN standard tooth root stress value is 352 MPa. But the safety factors are surprising, while the safety factor of AGMA is 2.01, the DIN safety factor is 2.30. It has been determined that it is more accurate to consider and compare the safety coefficients, not the calculated stress values.

• ANSYS FEM analysis calculates the stress value in the assembly condition of the system. Thus, it is possible to see the stress value at any specific location in the system. It was determined that in the tooth contact area, the stress changes during the contact. For this reason, It was determined that the FEM solution result was higher than the analytical solution results. Analytical methods assume that gears are making good contact.

• In software such as ANSYS where FEM methods are used, it has been revealed that while there are time consuming and demanding procedures for a standard gear calculation, KISSsoft software customized for gears provides the calculation results with high accuracy and with less procedures. The difference in safety factor between the two methods is about 8% and they are compatible.

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REFERENCES

Aslantaş, K. and Taşgetiren, S., ‘’A Study of Spur Gear Pitting Formation and Life Prediction,’’ Wear, Vol. 257, No.11, pp.1167–1175 (2004).

Doğan, O., and Karpat, F., ‘’Crack Detection for Spur Gears With Asymmetric Teeth Based on the Dynamic Transmission Error,’’ Mechanism and Machine Theory, Vol.133, pp.417–431 (2019).

Doğan, O., Yılmaz, T. G., and Karpat, F., ‘’Stress Analysis of Involute Spur Gears with Different Parameters by Finite Element and Graphical Method, ’’ Journal of the Faculty of Engineering and Architecture of Gazi University, Vol.33, No.4, pp.1493–1504 (2018).

Fetvaci, M. C., and Erdem, C., ‘’Finite Element Modeling of Spur Gears for Analysing Tooth,’’ Vol.19, No.2, pp.199–203 (2004).

Guan, Y., Yang, X., Fang, Z., and Chen, G., ‘’Comparative Analysis of Three Geometric Models for Crown Gear Coupling. Mechanism and Machine Theory,’’ Vol.136, pp.269–283 (2019).

Guddad, R., and Venkataram, N., ‘’Effect of Shaft and Bearing Flexibilities on Gear Tooth Contact Analysis,’’ Materials Today: Proceedings, Vol.4, No.10, pp.10823–10829 (2017).

He, C., and Lin, C., ‘’Analysis of Loaded Characteristics of Helical Curve Face Gear,’’ Mechanism and Machine Theory, Vol.115, pp.267–282 (2017).

Jadeja, R. M., Chauhan, D. M., and Lakhani, J. D., ‘’Bending Stress Analysis of Bevel Gears”. International Journal of Innovative Research in Science, Engineering and Technology, Vol.2, No.7, pp.3041–3046 (2013).

Karaveer, V., Mogrekar, A., Preman, T., and Joseph, R., ‘’Modeling and Finite Element Analysis of Spur Gear,’’ International Journal of Current Engineering and Technology, Vol.3, No.5, pp.2104–2107 (2013).

Celik, K. H., and Akinci, I., ‘’Analytical and Finite Element Method Based Stress Analysis of Rotary Elements: Case Study for the Motion Transmission Gears of a Rotary Drum Mower,’’ Journal of Failure Analysis and Prevention, Vol.16, No.2, pp.293–301 (2016).

Li, W., Wood, A., Weidig, R., and Mao, K., ‘’An Investigation on the Wear Behaviour of Dissimilar Polymer Gear Engagements,’’ Wear, Vol.271, No.9–10, pp.2176–2183 (2011).

Lin, C. H., and Fong, Z. H., ‘’Numerical Tooth Contact Analysis of a Bevel Gear Set by Using Measured Tooth Geometry Data,’’ Mechanism and Machine Theory, Vol.84, No.168, pp.1–24 (2015).

Lisle, T. J., Shaw, B. A., and Frazer, R. C., ‘’External Spur Gear Root Bending Stress: A Comparison of ISO 6336:2006, AGMA 2101-D04, ANSYS Finite Element Analysis and Strain Gauge Techniques,’’ Mechanism and Machine Theory, Vol.111, pp.1–9 (2017).

Lisle, T. J., Shaw, B. A., and Frazer, R. C., ‘’Internal Spur Gear Root Bending Stress: A Comparison of ISO 6336:1996, ISO 6336:2006, VDI 2737:2005, AGMA, ANSYS Finite Element Analysis and Strain Gauge Techniques,’’ Proceedings of the Institution of Mechanical Engineers, Part C:

Journal of Mechanical Engineering Science, Vol.233, No.5, pp.1713–1720 (2019).

Maršálek, P., and Moravec, V., ‘’A Methodology for Gear Fatigue Tests and Their Evaluation (part 2),’’ Journal of Middle European Construction and Design of Cars, Vol.9, No.3, pp.18–22 (2013).

Mu, Y., Li, W., Fang, Z., and Zhang, X., ‘’A Novel Tooth Surface Modification Method for Spiral Bevel Gears with Higher-Order Transmission Error,’’ Mechanism and Machine Theory, Vol.126, pp.49–60 (2018).

Noaman, B., ‘’Involute Gear Tooth Stresses Analysis Using Finite Element Modeling,’’ American Scientific Research Journal for Engineering, Technology and Sciences, August, Vol.34, No 1, pp.269-284 (2017).

Qin, W. J., and Guan, C. Y., ‘’An Investigation of Contact Stresses and Crack Initiation in Spur Gears Based on Finite Element Dynamics Analysis,’’ International Journal of Mechanical Sciences, Vol.83, pp.96–103 (2014).

Sahu, R., Singh, S., and Bhoi, A., ‘’Parametric Stress Analysis of Spur and Helical Gear,’’ International Journal of Mechanical And Production Engineering, No.11, pp.63–69 (2017).

Sekercioglu, T., and Kovan, V., ‘’Pitting Failure of Truck Spiral Bevel Gear,’’ Engineering Failure Analysis, Vol.14, No.4, pp.614–619 (2007).

Silori, P., Shaikh, A., Nithin Kumar, K. C., and Tandon, T., ‘’Finite Element Analysis of Traction Gear Using ANSYS,’’ Materials Today: Proceedings, Vol.2, No.4–5, pp.2236–2245 (2015).

Singh, A., Bevel Gears. In Fundamentals of Machine Design, Cambridge (2018).

Singh, A. K., and Dewangan, M., ‘’Bending Stress and Contact Stress Analysis of Spur Gear Tooth Using ‘ MATLAB .’’’ Vol.3, No.20, pp.1–5 (2015).

Sun, Y., Ma, H., Huangfu, Y., Chen, K., Che, L. Y., and Wen, B., ‘’A Revised Time-Varying Mesh Stiffness Model of Spur Gear Pairs with Tooth Modifications,’’ Mechanism and Machine Theory, Vol.129, pp.261–278 (2018).

Tunalioğlu, M. Ş., and Tuç, B., ‘’Theoretical and Experimental Investigation of Wear in Internal Gears,’’ Wear, Vol.309, No.1–2, pp.208–215 (2014).

Wang, Y., Lan, Z., Hou, L., Chu, X., and Yin, Y., ‘’An Efficient Honing Method For Face Gear With Tooth Profile Modification,’’ International Journal of Advanced Manufacturing Technology, Vol.90, No.1–4, pp.1155–1163 (2017).

Wei, J., Zhang, A., Wang, G., Qin, D., Lim, T. C., Wang, Y., and Lin, T., ‘’A Study Of Nonlinear Excitation Modeling Of Helical Gears with Modification: Theoretical Analysis and Experiments,’’ Mechanism and Machine Theory, Vol.128, pp.314–335 (2018).

Wen, Q., Du, Q., and Zhai, X. ‘’A New Analytical Model to Calculate the Maximum Tooth Root Stress and Critical Section Location of Spur Gear,’’ Mechanism and Machine Theory, Vol.128, pp.275–286 (2018).

Wilcox, L., and Coleman, W. ‘’Application of Finite Elements to the Analysis of Gear Tooth Stresses,’’Journal of Engineering for Industry, (1973).

Zhan, J., Fard, M., and Jazar, R. ‘’A Quasi-Static FEM for Estimating Gear Load Capacity,’’ Measurement: Journal of the International Measurement Confederation, Vol.75, pp.40–49 (2015).