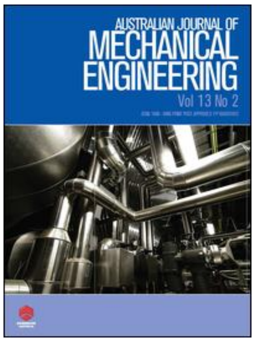


Taylor & Francis
Taylor & Francis Group

Australian Journal of Mechanical Engineering

ISSN: 1448-4846 (Print) 2204-2253 (Online) Journal homepage: https://www.tandfonline.com/loi/tmec20

Tooth contact analysis of straight bevel gears in the function of the modification of number of teeth of the driving gear

Sándor Bodzás

AUSTRALIAN JOURNAL OF MECHANICAL ENGINEERING
https://doi.org/10.1080/14484846.2019.1654964

Taylor & Francis
Taylor & Francis Group

ARTICLE

Tooth contact analysis of straight bevel gears in the function of the modification of number of teeth of the driving gear

Sándor Bodzás

Department of Mechanical Engineering, University of Debrecen, Debrecen, Hungary

ABSTRACT
The bevel gear is widely used in mechanical structures if we want to change the shaft positions, the transmission ratio and the rotation direction. The aim of the publication is the analysis of the TCA (Tooth Contact Analysis) parameters in the function of the modification of the number of teeth of the driving gear. This analysis is actually a finite element analysis with which the developing normal stress, normal strain and normal deformations could be determined on the contact zone of the gear pairs. This analysis is important for the purpose of the judgement of the goodness of the gear drive. Previously the exact CAD (Computer Aided Designing) modelling of the tested gear drive is needed and after that the exact assembly is necessary. In this publication, we want to determine the correlations among the TCA parameters and the number of teeth of the driven gear.

ARTICLE HISTORY
Received 10 May 2019
Accepted 7 August 2019

KEYWORDS
TCA; CAD; bevel gear; number of teeth

1. Introduction

Straight bevel gears are applied widely in machinery (in vehicles, tool machines, robots, for medical tools, etc.). They are used to connect shafts whose axes intersect in some angle; thus, the meshing surfaces form a cone on which teeth are shaped (Figures 1 & 2) (Argyris, Fuentes, and Litvin 2002; Dudás 2011; Dudás 1991; Erney 1983; Fuentes and Iserte 2011; Goldfarb, Trubachev, and Barmina 2018; Litvin and Fuentes 2004; Litvin 1972; Rohonyi 1980; Terplán 1975).

Based on Figure 3 it is visible that many parameters are needed for the exact description of the bevel gear. In case of this type, the teeth are situated parallel with the rotation axis of the cone body (Dudás 2011; Litvin and Fuentes 2004; Litvin 1972; Rohonyi 1980; Terplán 1975).

The aim of the TCA is the analysis of the developed mechanical parameters which have to be analysed on the gear connection. The main TCA parameters are the normal stress normal strain and normal deformation values which are determined perpendicularly to the gear surface (Argyris, Fuentes, and Litvin 2002; Fuentes and Iserte 2011; Kozák and Szeidl 0000; Litvin and Fuentes 2004; Páczelt, Szabó, and Baksa 11).

The production cost is very expensive because of the complex geometry, the tool costs, device cost, etc. That is why the real production has to be followed after the designing, the CAD modelling and the TCA analysis.

2. Designing of gear pairs

We have worked out a computer program because of the simplification of the geometric designing process. Previously the suggestions of the references have to be read (Dudley 1962; Litvin and Fuentes 2004; Litvin 1972; Rohonyi 1980; Terplán 1975). Knowing of the designing formulas the computer program could be written.

Input data for designing the drive pair are: m module, z_1 number of teeth of the driver bevel gear wheel, z_2 number of teeth of the driven gear wheel, c^* root clearance factor and the α_0 angle of contact (Dudley 1962; Litvin and Fuentes 2004; Litvin 1972; Rohonyi 1980; Terplán 1975). Knowing of them the program is calculated all necessary geometric parameters of the gear pairs and drawn the profile curves on the highest and the lowest diameters (Figure 3).

After the geometric calculation, the CAD models of the bevel gears could be created by SolidWorks software (Figure 4). Interpolation B-spline has to be fit on the profile points. Knowing of the necessary profile curves the teeth could be created by extrusion.

On Table 1 the calculated parameters of the bevel gears could be seen. After the geometric designing, the TCA could be followed.

3. TCA analysis of connecting bevel gear pairs

3.1. The adoption of the finite element mesh

The appropriate selection of the finite element mesh is the basis of the calculation process (Kozák and Szeidl 0000; Páczelt, Szabó, and

CONTACT Sándor Bodzás bodzassandor@eng.unideb.hu

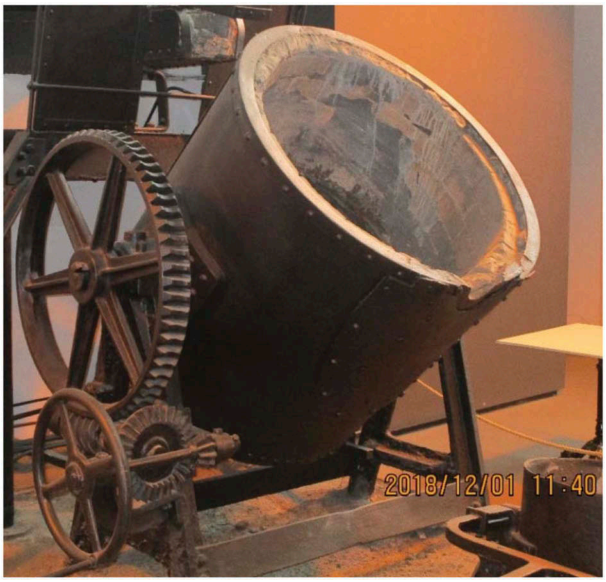


Figure 1. The application of straight bevel gear in case of moving of moulding vat.

Baksa 11). One of the main property of the bevel gear the tooth thickness is continuously changing along the tooth length from the highest cone's diameters to the lowest cone's diameters. Because of this property the equable mesh subdivision could not be applied.

We have adopted a coordinate system in the middle of the teeth connection zone (Figure 5).

The x axis of the adopted coordinate system is shown in the normal direction of the connecting surfaces (Figure 5).

Sphere of influence meshing has been applied on the contact zone. The mesh shape is dense triangles. The density is 1.5 mm and the sphere radius is 38 mm. Automatic meshing has been applied on the outside of the contact zone (Figure 6).

The type of the applied material has been structural steel (Table 2).

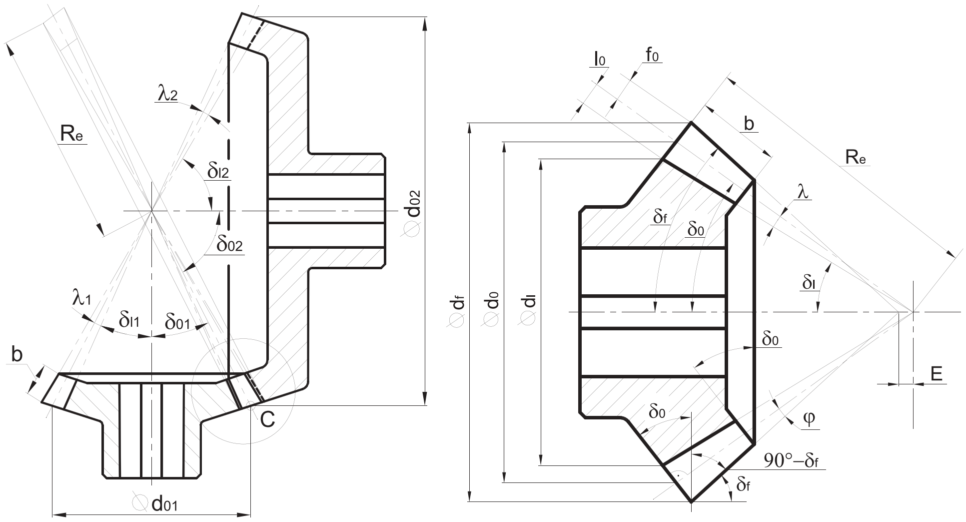


Figure 2. The main parameters of bevel gear pairs.

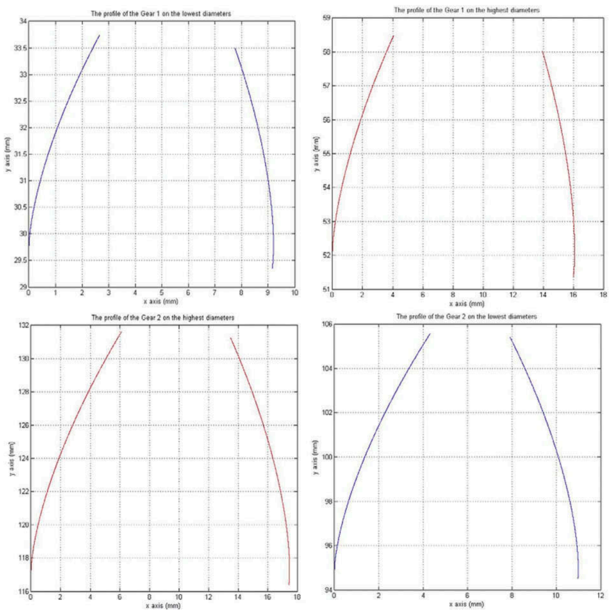
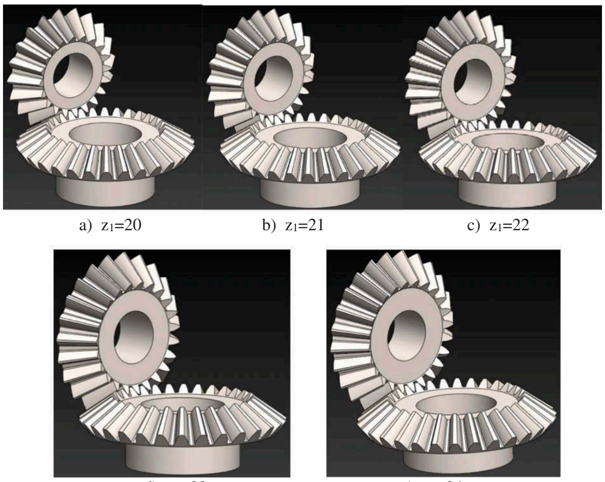


Figure 3. The profiles of bevel gear pairs on the main diameters (m = 10, z_1 = 20, z_2 = 30).



a) z_1=20 b) z_1=21 c) z_1=22

d) z_1=23 e) z_1=24

Figure 4. The type of the CAD models of the designed gears.

Table 1. The calculated parameters of the designed bevel gear pairs.

The main parameters of the bevel gear pairs	Gear drive I.	Gear drive II.	Gear drive III.	Gear drive IV.	Gear drive V.
Number of teeth of the driving gear (z_1)	20	21	22	23	24
Number of teeth of the driven gear (z_2)			30		
Module (m) [mm]			10		
The largest pitch circle diameter of the driving gear (d_{01}) [mm]	200	210	220	230	240
The largest pitch circle diameter of the driving gear (d_{02}) [mm]			300		
Half pitch angle of the pitch circle of the driving gear (δ_{01}) [°]	56.309	55.008	53.746	52.523	51.340
Half pitch angle of the pitch circle of the driven gear (δ_{02}) [°]	33.69	34.992	36.253	37.476	38.659
Effective pitch surface radius (R_e) [mm]	180.277	183.098	186.01	189.01	192.093
Addendum on the largest diameter (f_0) [mm]			10		
Dedendum on the largest diameter (l_0) [mm]			12		
The largest tip circle diameter of the driving gear (d_{f1}) [mm]	211.094	221.469	231.827	242.168	252.493
The largest tip circle diameter of the driven gear (d_{f2}) [mm]	316.641	316.384	316.128	315.872	315.617
The largest root circle diameter of the driving gear (d_{a1}) [mm]	186.687	196.236	205.807	215.397	225.007
The largest root circle diameter of the driven gear (d_{a2}) [mm]	280.03	280.338	280.646	280.953	281.259
Face width (b) [mm]	51.507	52.313	53.145	54.003	54.883
Dedendum angle (λ) [°]	3.808	3.749	3.691	3.632	3.574
Tip cone angle of the driving gear (δ_{f1}) [°]	60.118	58.757	57.437	56.156	54.914
Tip cone angle of the driven gear (δ_{f2}) [°]	37.498	38.741	39.945	41.108	42.234
Root cone angle of the driving gear (δ_{l1}) [°]	52.501	51.258	50.055	48.891	47.765
Root cone angle of the driven gear (δ_{l2}) [°]	29.881	31.242	32.562	33.843	35.085
Circular pitch on the largest pitch circle diameter (t) [mm]			31.415		
Clearance at flank (j_s) [mm]			1.570		
Pitch circle tooth thickness on the largest diameters (S_{ax}) [mm]			17.278		
Transmission ratio (i)	1.5	1.428	1.363	1.304	1.25

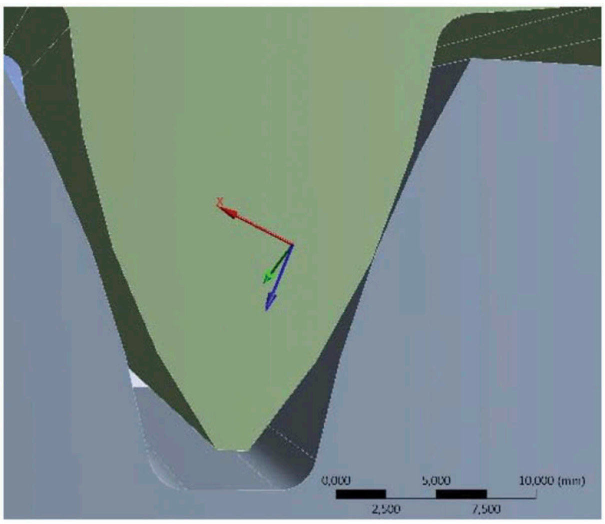


Figure 5. The adoption of the coordinate system into the middle of the connection zone.

The driving bevel gear has been loaded by 600 Nm moment. Five degrees of freedom have been fixed on the driving gear only the turning motion around the axis of rotation has been let. The driven gear has been totally fixed.

3.2. Analysis of the normal stress

The normal stress is interpreted on perpendicular direction of the tooth surface (Kozák and Szeidl 0000; Litvin and Fuentes 2004; Páczelt, Szabó, and Baksa 11). This direction is the most determinative in the aspect of tooth deformation. This normal stress has been calculated for the tooth contact zone (Figure 7).

Based on the calculations the normal stresses could be seen in the function of the number of teeth of the driving bevel gear on Figure 8.

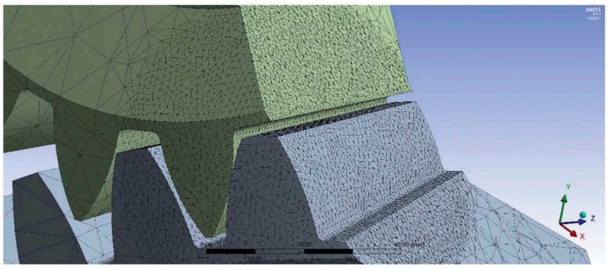


Figure 6. The adoption of the finite element mesh.

Table 2. The properties of the applied material.

Density	7850 kg/m^3
Yield limit	250 MPa
Ultimate strength	460 MPa

Based on the results the minimum normal stress of the surface of the driving bevel gear is developed in case of z_1 = 24 number of teeth in absolute value. The minimum normal stress of the surface of the driven bevel gear is developed in case of z_1 = 21 number of teeth in absolute value (Figure 8).

3.3. Analysis of the normal elastic strain

The normal elastic strain has been calculated for the tooth contact zone (Figure 9). It is defined on the perpendicular direction for the tooth surface (Kozák and Szeidl 0000; Litvin and Fuentes 2004; Páczelt, Szabó, and Baksa 11).

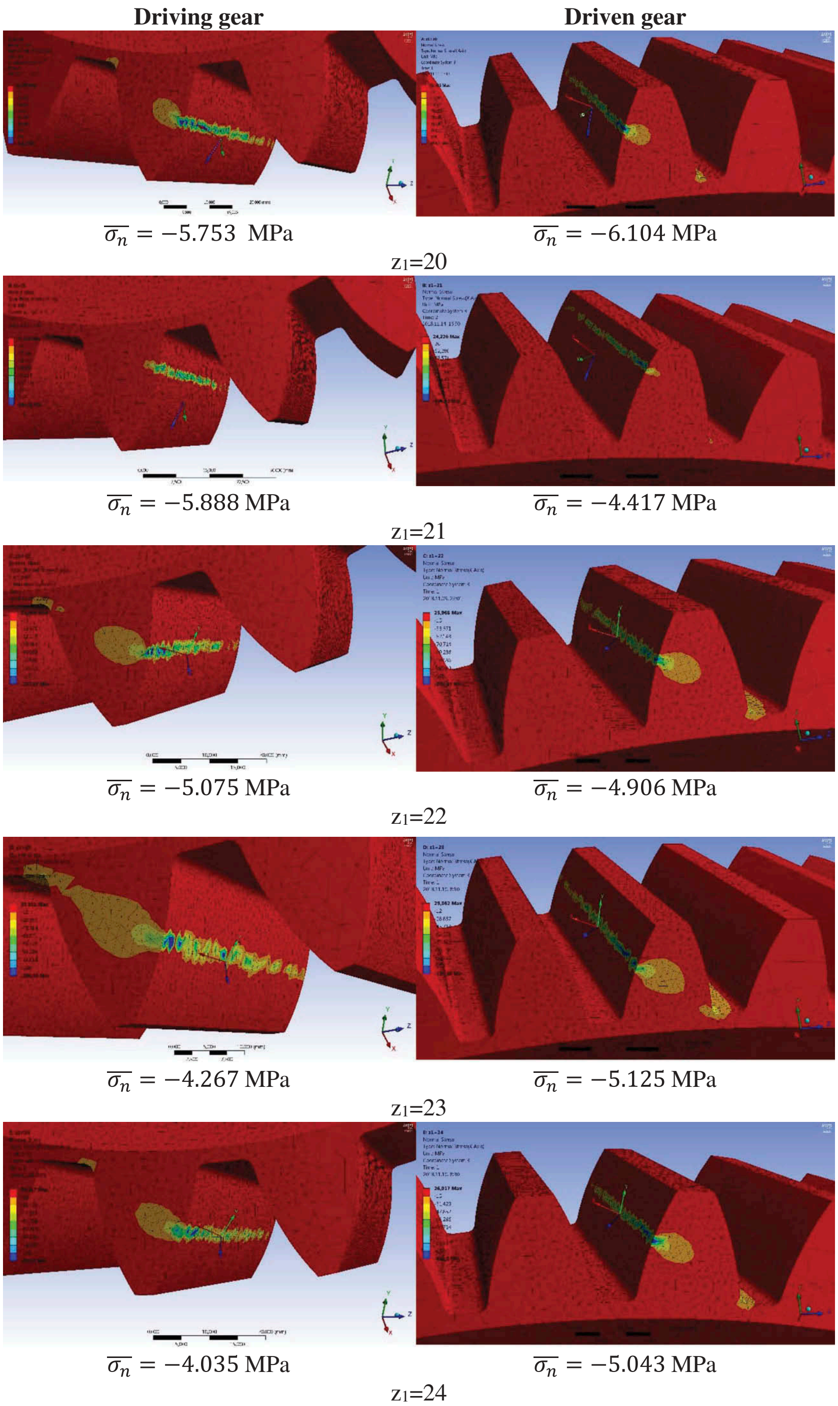


Figure 7. Normal stress results for every bevel gear pairs.

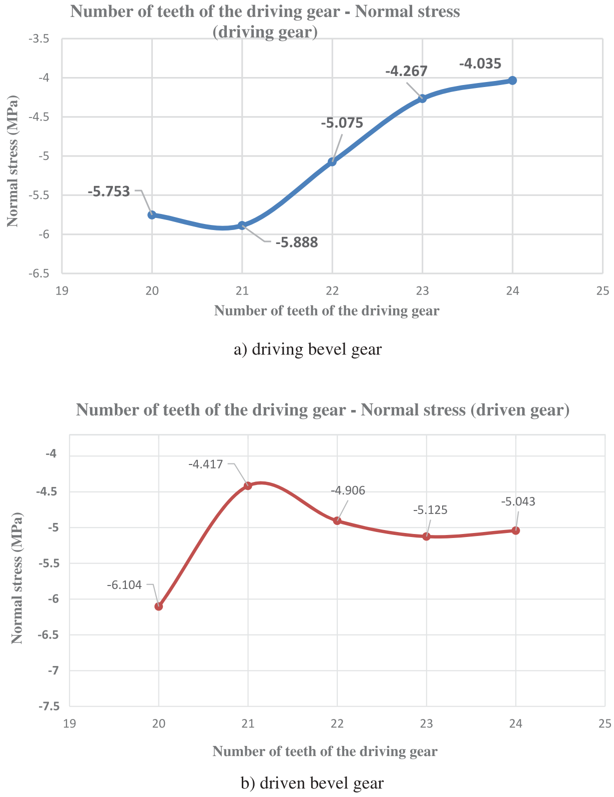


Figure 8. The developed normal stresses in the function of the number of teeth.

Based on the calculations the normal elastic strain could be seen in the function of the number of teeth of the driving bevel gear on Figure 10.

Based on the results the minimum normal elastic strain of the surface of the driving bevel gear is developed in case of $z_1 = 24$ number of teeth in absolute value. The minimum normal elastic strain of the surface of the driven bevel gear is developed in case of $z_1 = 24$ number of teeth in absolute value (Figure 10). That is why the increasing of the number of teeth of the driving gear the normal elastic strain will be increased in case of both connecting tooth surfaces.

3.4. Analysis of the normal deformation

The normal deformation has been analysed into the x direction which is perpendicular for the contact surfaces (Kozák and Szeidl 0000; Litvin and Fuentes 2004; Páczelt, Szabó, and Baksa 11). This direction is the most determinative because the main deformation is applied into the perpendicular direction of the contact surfaces (Figure 11).

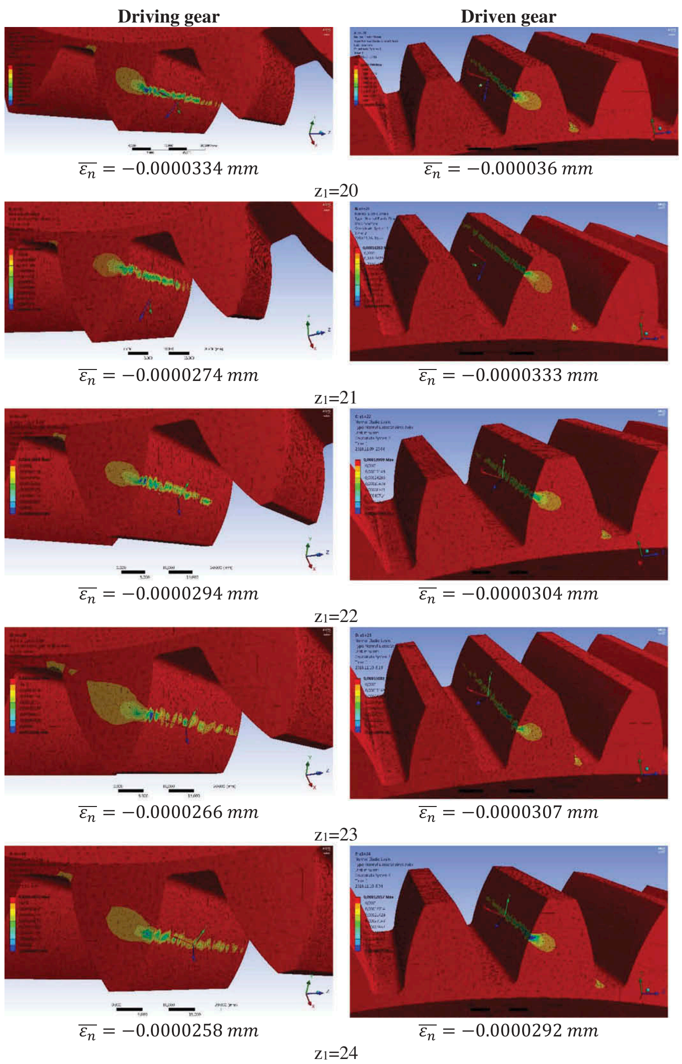
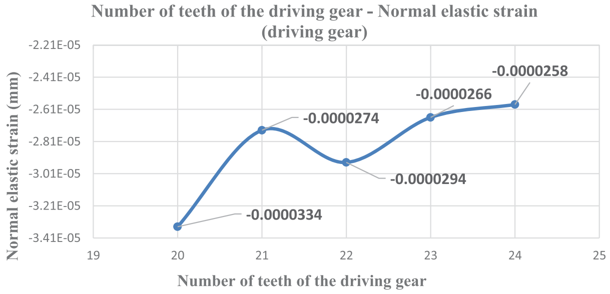
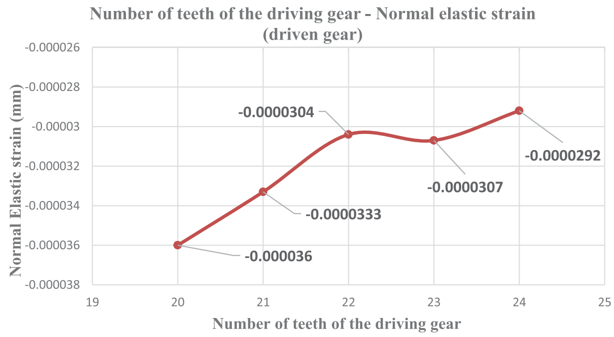


Figure 9. Normal elastic strain results for every bevel gear pairs.

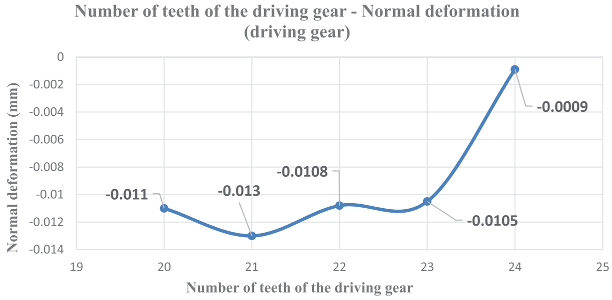


a) driving bevel gear

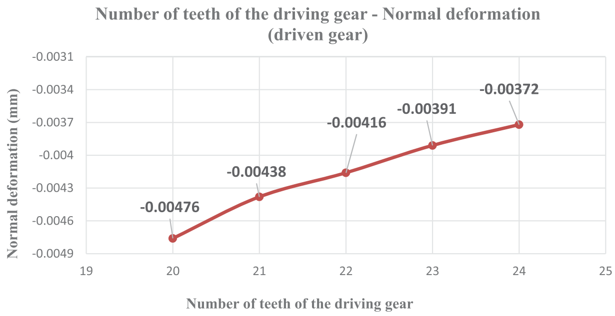


b) driven bevel gear

Figure 10. The developed normal elastic strain in the function of the number of teeth.



a) driving bevel gear



b) driven bevel gear

Figure 12. The developed normal elastic strain in the function of the number of teeth.

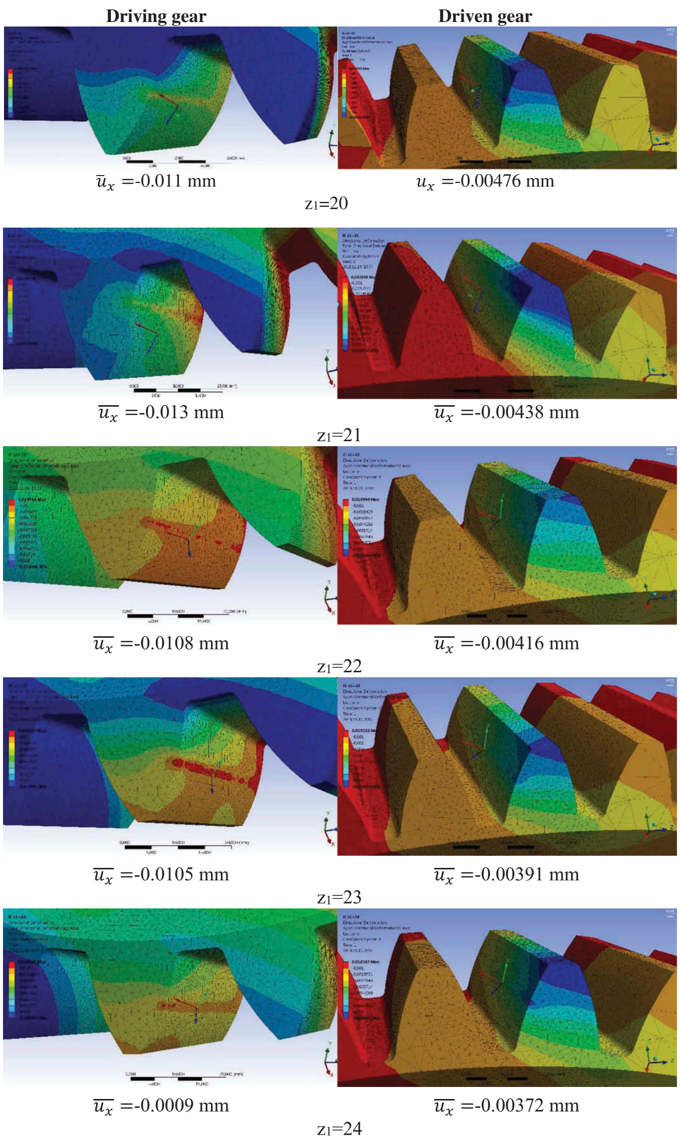


Figure 11. Normal deformation results for every bevel gear pairs.

Based on the calculations the normal deformation could be seen in the function of the number of teeth of the driving bevel gear on Figure 11.

Based on the results the minimum normal deformation of the surface of the driving bevel gear is developed in case of $z_1 = 24$ number of teeth in absolute value. The minimum normal deformation of the surface of the driven bevel gear is developed in case of $z_1 = 24$ number of teeth in absolute value (Figure 12). That is why the increase of the number of teeth of the driving gear the normal deformation will be increased in case of both connecting tooth surfaces.

4. Conclusion

The aim of the research is the analysis of the modification of the number of teeth of the driving gear in the function of the TCA parameters (normal stress, normal, elastic strain and normal deformation).

Before the TCA the determination of the geometric parameters is needed. We have worked out a new-type computer-aided software which is helped us for the facilitation of the calculation process. This software is determined the necessary geometric parameters and the profile curves of the bevel gear pairs. After the process, the CAD models of the designed gears have to be created. Knowing of the profile curves an interpolation B-spline is fitted into the calculated points.

Based on the CAD models of the analysed gears the TCA analysis could be done. We have designed five types of bevel gear pairs. We have done the TCA analysis for every gear pairs.

The effect of the normal stress, normal elastic strain and the normal deformation has been analysed between the contact surfaces. After the analysis, we have evaluated the received results and created the functions separately for the driving gear and the driven gear. We could determine the consequences of these functions.

Acknowledgments

This research was supported by the János Bolyai Research Scholarship of the Hungarian Academy of Sciences.

Disclosure statement

No potential conflict of interest was reported by the author.

References

Argyris, J., A. Fuentes, and F. L. Litvin. 2002. “Computerized Integrated Approach for Design and Stress Analysis of Spiral Bevel Gears.” *Computer Methods in Applied Mechanics and Engineering* 191: 1057–1095. Elsevier. doi:10.1016/S0045-7825(01)00316-4.

Dudás, I. 2011. *Gépgyártástechnológia III., A. Megmunkáló Eljárások És Szerszámaik, B. Fogazott Alkatrészek Gyártása És Szerszámaik*. Budapest: Műszaki Kiadó.

Dudás, L. 1991. “Kapcsolódó Felületpárok Gyártásgeometriai Feladatainak Megoldása Az Elérés Modell Alapján.” Kandidátusi értekezés, Budapest, TMB. p.144, 2005.

Dudley, D. W. 1962. „*Gear Handbook*”. New York-Toronto -London: MC Graw Hill Book Co.

Erney, G. 1983. *Fogaskerekek*. Budapest: Műszaki Könyvkiadó. p. 460.

Fuentes, A., and J. L. Iserte. 2011. “Gonzalez – Perez, I., Sanchez – Marin, F. T.: Computerized Design of Advanced Straight and Skew Bevel Gears Produced by Precision Forging.” *Computer Methods in Applied Mechanics and Engineering* 2363–2377. Elsevier. doi:10.1016/j.cma.2011.04.006.

Goldfarb, V., E. Trubachev, and N. Barmina. 2018. *Advanced Gear Engineering*. Springer. p. 197. ISBN 978-3-319-60398-8.

Kozák, I., and G. Szeidl. 0000. *Fejezetek a Szilárdságtanból, Kézirat, 2008 – 2012*. Miskolci Egyetem. p.284. elektronikus jegyzet.

Litvin, F. L. 1972. *A Fogaskerékkapcsolás Elmélete*. Budapest: Műszaki Könyvkiadó.

Litvin, F. L., and A. Fuentes. 2004. *Gear Geometry and Applied Theory*. United Kingdom: Cambridge University Press. ISBN 978 0 521 81517 8.

Páczelt, I., T. Szabó, and A. Baksa. *A Végeselem Módszer Alapjai*. Miskolci Egyetem. p.243.

Rohonyi, V. 1980. *Fogaskerékhajtások*. Budapest: Műszaki Könyvkiadó.

Terplán, Z. 1975. *Gépelemek IV., Kézirat*, 220. Budapest: Tankönyvkiadó.