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Spur gears with contact ratio less than unity

Engranes rectos con relación de contacto menor a la unidad

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Abstract

The objective of this research is to evaluate the design of a pair of spur gears and another analogous pair with a contact ratio of less than unity. Considering two pairs of normalized gears with equal diametral pitches and pressure angles, as well as equal pitch radii in their driving gears and equal pitch radii in their driven gears. For the first pair of gears with normalized tooth numbers, the contact ratio greater than unity es obtained. For the second pair of gears, the number of teeth is proportionally reduced, consequently obtaining a contact ratio of less than one. For both pairs, the maximum von Mises stresses are obtained using the finite element method. The pairs are compared qualitatively and quantitatively. This work contributes with novel elements of judgment for a better decision making of the industrialists interested in reducing the problems of normalized spur gears such as noise, abrasive wear, adhesive wear, temperature, and efforts induced by the overlapping relationship between coupled teeth; proposing them a practical solution that will open new avenues of research.

Noise, Temperature, Abrasion

Resumen

El objetivo de esta investigación es evaluar el diseño de un par de engranes rectos y además otro par análogo con relación de contacto menor a la unidad. Considerando dos pares de engranes normalizados con pasos diametrales y ángulos de presión iguales además con radios de paso iguales en sus engranes motrices y con radios de paso iguales en sus engranes movidos. Para el primer par de engranes con números de dientes normalizados se obtiene la relación de contacto mayor a la unidad. Para el segundo par de engranes se reducen proporcionalmente los números de dientes obteniéndose consecuentemente una relación de contacto menor a la unidad. Para ambos pares se obtienen los esfuerzos máximos de Von Mises usando el método de los elementos finitos. Los pares se comparan cualitativa y cuantitativamente. Este trabajo contribuye con elementos de juicio novedosos para una mejor toma de decisiones de los industriales interesados en reducir los problemas de los engranes rectos normalizados como ruido, desgaste abrasivo, desgaste adhesivo, temperatura y esfuerzos inducidos por la relación de sobreposición entre dientes acoplados; proponiéndoles una solución práctica que abrirá nuevas vías de investigación.

Ruido, Temperatura, Abrasión

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Introduction

The movement transmission mechanism through spur gears, as well as the corresponding induced efforts, have aroused the interest of the scientific community. Contact relationship has been considered a critical factor for the qualitative and quantitative evaluation of the general behaviour of such gears.

Due to the importance of the subject, Díaz González, D. J. (2022) designed and manufactured a transportation vehicle for people with limited mobility by implementing Theo Jansen mechanism proportions and a mechanical transmission.

Badel Iriarte, A. A. (2022) Improved and characterized a pinion pump test bench for viscous fluids.

Cleveland Poo, B. (2022) Studed the torsional and radial vibrations of a spur gear transmission.

Nunura Dávila, L. A. (2022) designed and simulated a mechanical ventilator to treat respiratory failure in critical care for adults in Lambayeque.

Barroso-Molero, J. M. (2022) designed a speed reducer for a conveyor belt industrial.

ANSI/AGMA 1010-F14 (2020) Standard. Appearance of Gear Teeth – Terminology of Wear and Failure. Identifies and describes the classes of common gear failures and illustrates their degrees deterioration.

AGMA Standard 915-1-A02. Inspection Practices – Part 1: Cylindrical Gears – Tangential Measurements. It provides a Code of Practice dealing with the relevant inspection of the tangential component and runouts of gears (measurements referred to single flank contact). Supplement to ANSI/AGMA 2015-1-A01. ISBN: 1-55589-798-3.

On the page https://vsip.info/cap-10-modos-de-falla-comunes-en-engranajes-pdf-free.html the most common failure modes in gears are described, stating that the cracking of the teeth can initiate on the face and spread to the opposite flank.

Maras, S. *et al*. (2021) used vibration analysis and statistical process control to analyse failures in spur gears.

El Anouar, B. A. *et al*. (2016) they studied tooth wear on helical gears and stated: Vibration analyses in the time and frequency domain will enable wear diagnosis.

Herrera, A. *et al.* (2015) presented a study base on monitoring for early detection of possible tooth breakage in spur gears.

Pleguezuelos, M. & Pedrero, J. I. (2009) studied the performance of conventional spur gears, with non-uniform load distribution and variable friction along the line of action.

Regarding a pair of gears with contact ratio greater than unity an analogous pair with said ratio less than unity; eliminates the possibility of overlapping; reduces the number of impacts between teeth, noise, system temperature, abrasive wear, and adhesive wear.

Additional to the added value indicated in the previous paragraph; the pair of gears with a contact ratio less than unity also makes it possible to reduce the time of application of the load that generates efforts and work, the plastic deformation, the removal of material, the rolling of the profile, the grooves parallel to the direction of slip, wear in the support areas of the gears on their axes, the variation in the distance between centres, the radial pressure, the transverse vibrations of the entire rotor system. All the above with a considerable decrease in the efforts due to bending in the teeth; since decreasing their number increases their width, increases to the cube the moment of inertia, and decreases the bending effort.

Base on the assumption that it is always possible to optimize the interface between two components of a system, the proposed reduction of the contact ratio solves the problem caused by abrasive and adhesive wear. In the second section, the parameters, their units, and functional relationships between them are defined. A pair of gears with a standard number of teeth is described in the third section. The fourth section describes the pair of gears with a reduced number of teeth compared to the normalized one. The results are presented, analysed, and compared in the fifth, sixth and seventh sections.

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Nomenclature

The meanings of some variables are shown in Figures 1, 2 and 3, all of them are explained below. P_d = 4 = diametral pitch in teeth per inch of pitch diameter; $P_{c1} = P_{c2}$ = circular pitch in inches, in gears 1 or 2; $P_{c3} = P_{c4}$ = circular pitch in inches, in gears 3 or 4; $P_{b1} = P_{b2}$ = base pitch in inches, in gears 1 or 2; $P_{b3} = P_{b4}$ = base pitch in inches, in gears 3 or 4; N_i with $i = 1, 2, 3, 4$ = number of teeth in gear i; $r_1 = r_3$ = 2.5 = pitch radius in inches, on gears 1 or 3; $r_2 = r_4$ = 3.75 = pitch radius in inches, on gears 2 or 4; $r_{a1} = r_{a3}$ = addendum radius in inches, on gears 1 or 3 = $r_1 + \frac{1}{P_d} = r_3 + \frac{1}{P_d}$; $r_{a2} = r_{a4}$ = addendum radius in inches, on gears 2 or 4 = $r_2 + \frac{1}{P_d} = r_4 + \frac{1}{P_d}$; φ = 25 = pressure angle in degrees; $r_{b1} = r_{b3}$ = base radius in inches, on gears 1 or 3 = $r_1 cos\varphi = r_3 cos\varphi$; $r_{b2} = r_{b4}$ = base radius in inches, on gears 2 or 4 = $r_2 cos\varphi = r_4 cos\varphi$. α_i = angle subtended by half a tooth on the pitch circumference in degrees, in gear i = $\frac{90°}{N_i} - \frac{0.2}{P_d}$; β_i = angle subtended by half a gap on the pitch circumference in degrees, in gear i = $\frac{90°}{N_i} + \frac{0.2}{P_d}$. $\omega_1 = \omega_3$ = angular velocity in radians per second clockwise, on gears 1 or 3 = $1.5\omega_2 = 1.5\omega_4$; $\omega_2 = \omega_4$ = angular velocity in radians per second counter clockwise, on gears 2 or 4. Of the pair of teeth to monitor A, B, C, D, E, G are loci on the pressure line; A is the tangency with the base circumference of gears 1 or 3; B is the start of contact and is also the intersection with the addendum circumference of gears 2 or 4; C is the flank tangency at 1 with face at 2; D is the tangency of the pitch circumferences of gears 1 and 2 and also the tangency of the pitch circumferences of gears 3 and 4; E is the tangency of face at 1 with flank at 2; G is the end of the contact and is also the intersection with the addendum circumference of gears 2 or 4; z = distance from B to G; $O_1 = O_3$ = geometric centre of gears 1 or 3; $O_2 = O_4$ = geometric centre of gears 2 or 4; R_j with j = A, B, C, D, E, G, H is the position vector j, with respect to O_1 or O_3; $R_{jO2} = R_{jO4}$ is the position vector j, with respect to O_2 or O_4; B, C, D, E and G with subscript 1 or 2 are the material contact lines, on gears 1 or 2, at loci B, C, D, E and G.

R_v = speed ratio = $\frac{\omega_{1,3}}{\omega_{2,4}} = 1.5$; R_c = contact ratio = $\frac{z}{P_{bi}}$; σ_{max} = maximum Von Mises effort in pounds per square inch; F = force in pounds; T = torque in inch pounds; t = time in seconds for the pair of teeth to be monitored to pass their initial contact to their final contact; σ_{ta} = effort for tolerances and adjustments = effort due to mechanical errors.

Figure 1 corresponds to the initial contact of the teeth to be monitored. Figure 2 corresponds to the final contact of the teeth to be monitored. From Figures 1 and 2 the distance from B to E equals the distance from C to G.

Figure 3 shows the material lines at the teeth to be monitored, which contact at the corresponding loci on the pressure line at locus B.

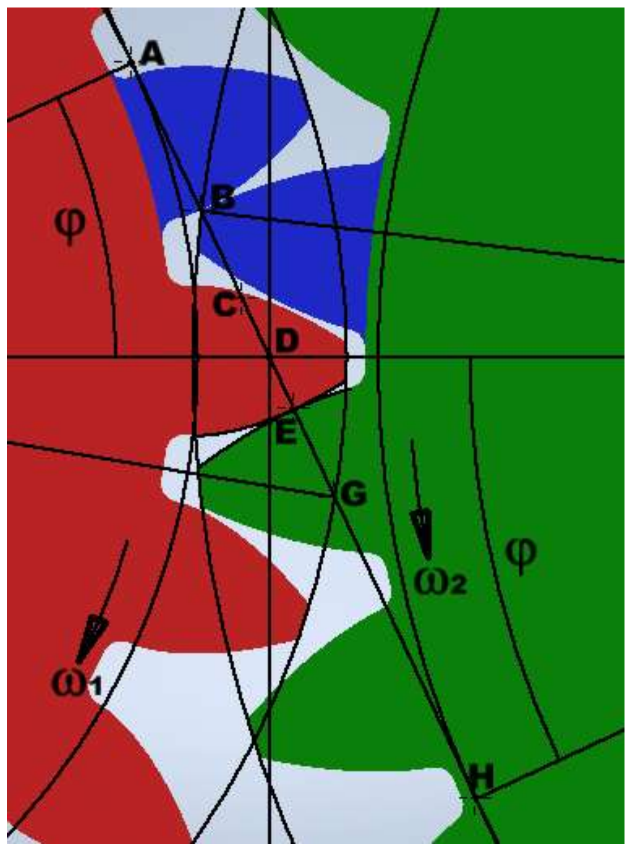

Figure 1 Contact on B
Own Elaboration

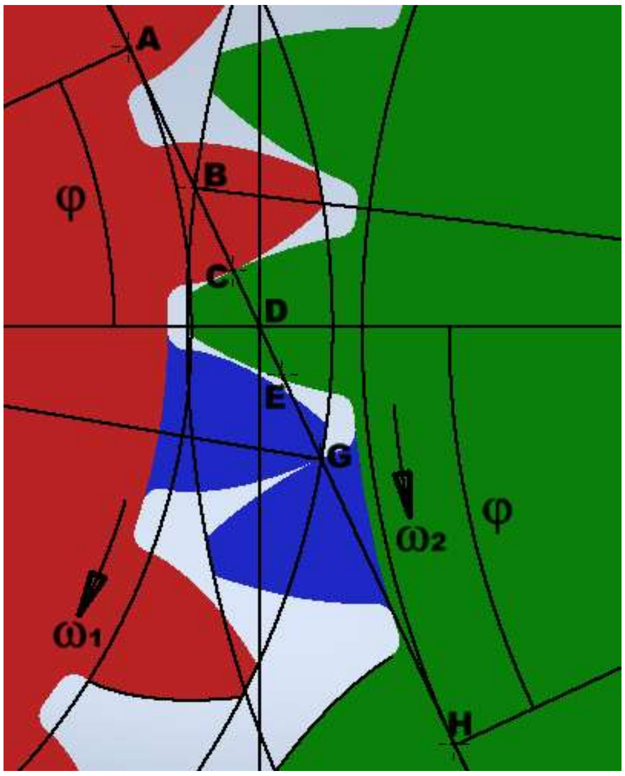

Figure 2 Contact on G
Own Elaboration

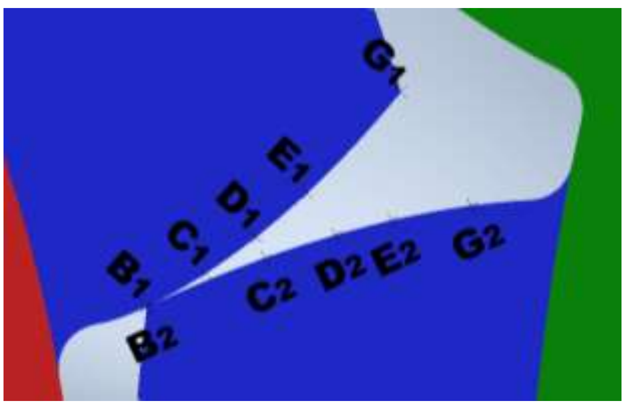

Figure 3 Material lines
Own Elaboration

Pair of Gears with Standard Number of Teeth

It is formed by gears 1 and 2, with 20 and 30 teeth respectively. From Figure 1 or 3, the distance from B to G from the beginning to the end of the contact of the gears to be monitored, is equal to:

$$z = \overline{AG} + \overline{BH} - \overline{AD} - \overline{DH} \tag{1}$$

$$\overline{AG} = \sqrt{(r_{a1})^2 - (r_{b1})^2} \tag{2}$$

$$\overline{BH} = \sqrt{(r_{a2})^2 - (r_{b2})^2} \tag{3}$$

$$\overline{AD} = r_1 sen\varphi \tag{4}$$

$$\overline{DH} = r_2 sen\varphi \tag{5}$$

Dividing z between P_{bi} we obtain:

$$R_c = \frac{zN_1}{2\pi r_1 cos\varphi} = \frac{zN_2}{2\pi r_2 cos\varphi} \tag{6}$$

Making equations of the of addendum circumference of gears 1 and 2 simultaneous with the equation of the pressure line, we obtain:

$$y_B = \overline{BH} cos\varphi - r_2 sen\varphi cos\varphi \tag{7}$$

$$x_B = r_1 - \overline{BH} sen\varphi + r_2 (sen\varphi)^2 \tag{8}$$

$$y_G = cos\varphi (r_1 sen\varphi - \overline{AG}) \tag{9}$$

$$x_G = r_1 + \overline{AG} sen\varphi - r_1 (sen\varphi)^2 \tag{10}$$

Equations (7) to (10) will be used to obtain the magnitude, direction, sense and point of application of the corresponding position vectors.

The distance from B to G divided between distance from B to E is equal to the contact ratio, this condition was used to define the position vector of point E.

If the graphic, structural and mechanical errors for both gears are zero and the contact ratio is greater than unity and less than 2; of the time t a percentage will correspond to contact between two pair of teeth and another to contact between a pair of teeth. During that same time t there will be 2 impacts between the teeth of gears 1 and 2.

Of the contact between two pairs of teeth; according to Figure 1, be first the one that does it in B and second the one that does it in E. If the graphic and structural errors are zero in both gears, with mechanical errors equal to zero for gear 1 and with mechanical errors different from zero for gear 2; takings as reference the two teeth of the first and tooth of gear 1 of the second, the following scenarios can be generated.

a) The separation of the teeth of gear 2 is greater than the design, the contact of the second pair of teeth, in the position shown in Figure 4, will be impossible and the energy will be transmitted only through the first pair.

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A contact ratio greater than unity in theory cannot be guaranteed in practice. It should not be overlooked that the second pair of teeth will contact each other for a short time prior to the initial contact of the first pair of teeth.

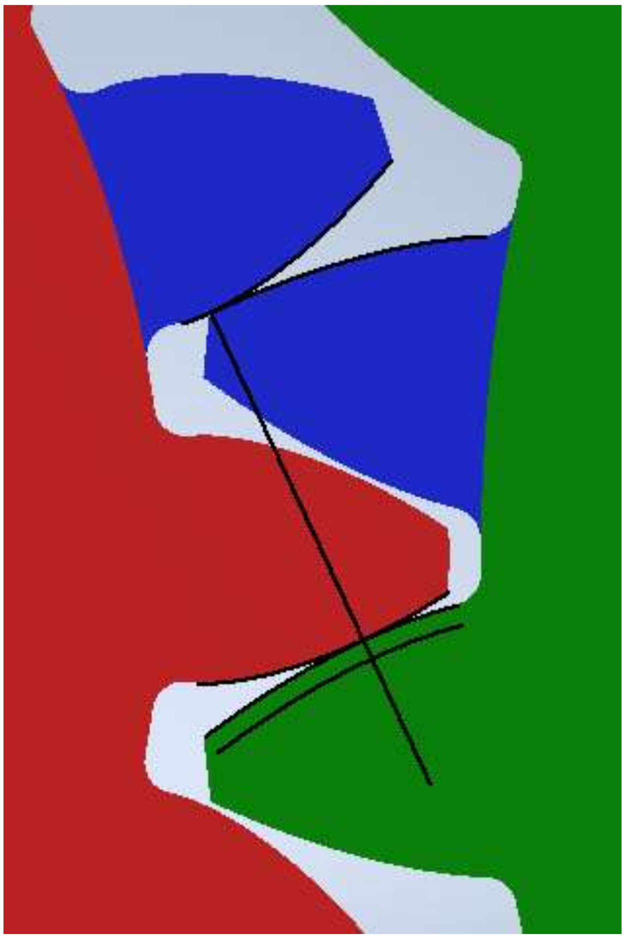

Figure 4 Contact only in B
Own Elaboration

With force F_B equal to 200 and T equal to 453.1538935183, the maximum von Mises stress equal to 20010 was determined using the finite element method, see Figures 5 and 6.

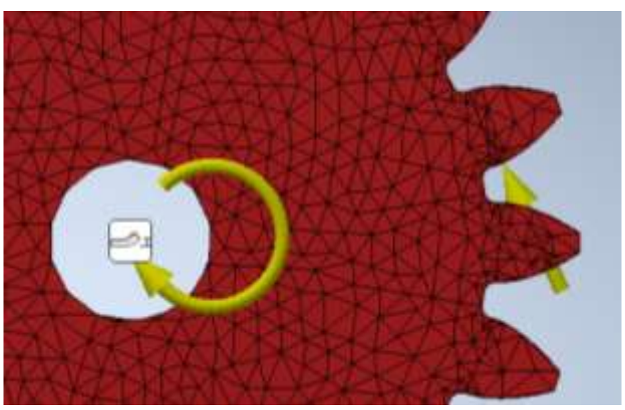

Figure 5 Meshed 1,2
Own Elaboration

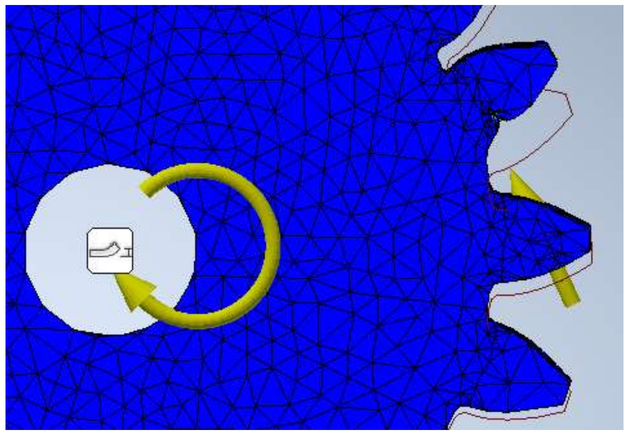

Figure 6 Numerical simulation 1,2
Own Elaboration

b) The separation of the teeth of gear 2 is equal to the of design, the contact of the second pair of teeth will be possible and the energy will be transmitted through the two pair of teeth, see Figure 7. Note that the contact in B will be more efficient than in E, because the radial component is smaller. Considering the mechanical errors in gear 2 with their tolerances and fits, the probability of this scenario is minimal.

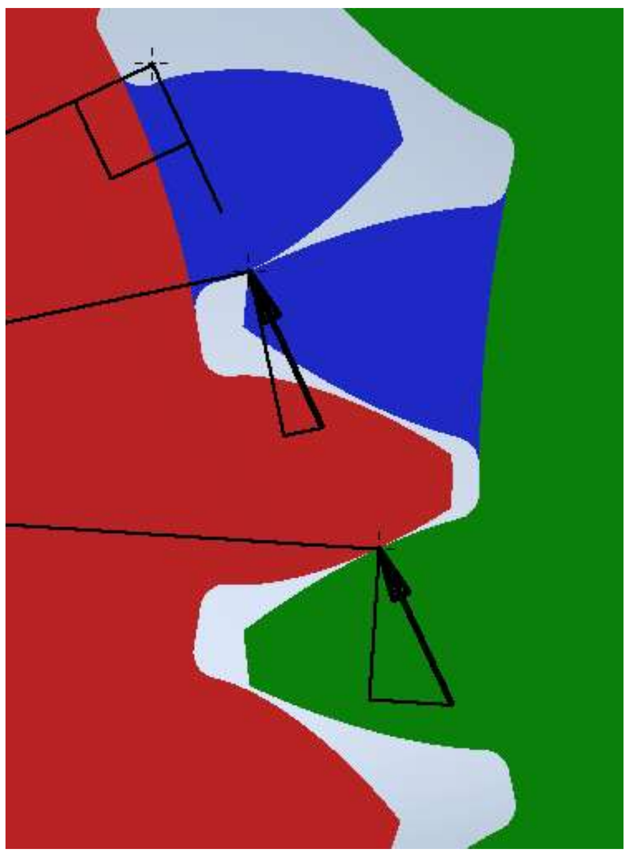

Figure 7 Contact in B and E
Own Elaboration

With force F_B equal to 100, force F_E equal to 100 and T equal a 453.1538935183, the maximum von Mises stress equal to 10020 was determined using the finite element method.

c) The separation of the teeth of gear 2 is less than the of design, there will be contact between the first and the second pair of teeth, see Figure 8. When the teeth of gear 2 engage with the teeth of gear 1; the scenario b will be generated; also forces caused by the mechanical errors of gear 2 will be generated which are not shown due to their random nature, which oppose the work for which the system was designed. The contact efforts induce energy losses, those and these will increase for gear teeth separation minor.

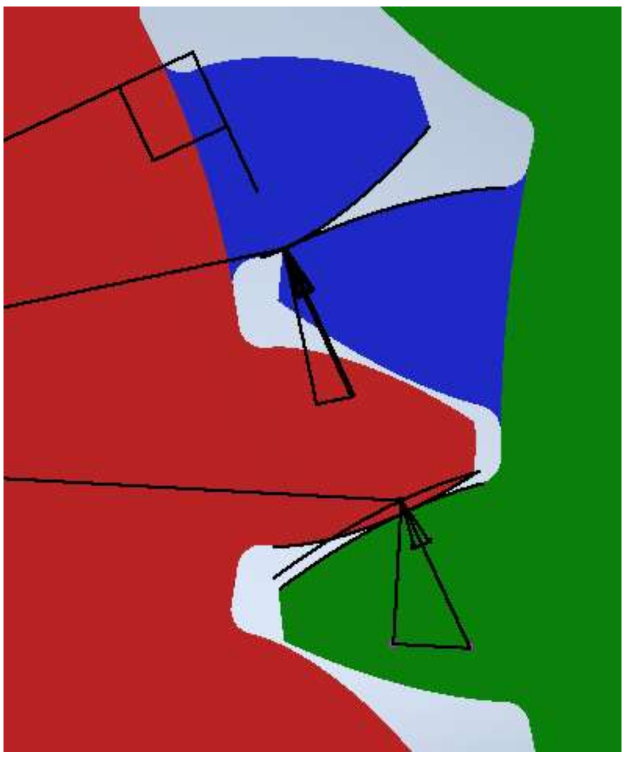

Figure 8 Abrasion and adhesion
Own Elaboration

The forces shown in Figures 5 through 8 are on gear teeth 1 and collinear with the line of pressure.

Pair of Gears with Smaller Number of Teeth

This pair of gears, 3 and 4, have 12 and 18 teeth respectively to keep the velocity ratio constant. Since z is the distance between locus B and G, It can be calculated with (1).

Dividing z between P_{bi} we obtain:

$$R_c = \frac{zN_3}{2\pi r_3 cos\varphi} = \frac{zN_4}{2\pi r_4 cos\varphi} \tag{11}$$

Making the equations of the of addendum circumference of gears 3 and 4 simultaneous with the equation of the line of pressure, analogous equations to (7) to (10) are obtained.

The position vectors of A, B, D, G and H are invariant with respect to those defined in the pair of gears 1 and 2.

Since the contact of the first pair of teeth will have ended when initiate contact the second, the position vectors of C and E will not exist.

For this pair of gears, the contact ratio is less than unity and, therefore, there will be contact between only one pair of teeth, see Figure 9.

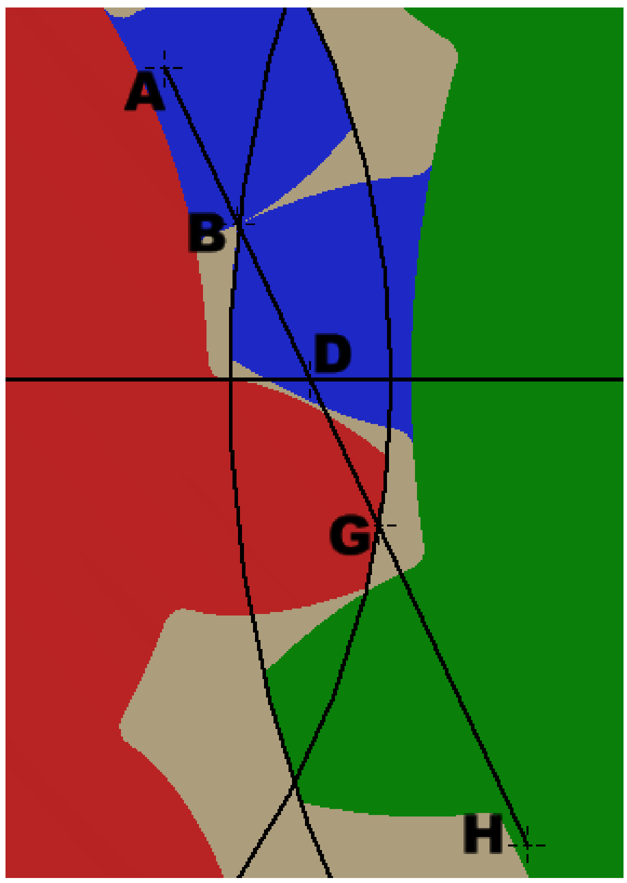

Figure 9 Initial contact between 3 and 4
Own Elaboration

The action and reaction forces along the pressure line will generate effort and work. When tooth 4 engages with tooth 3, the forces will correspond to direct contact along the pressure line or rolling contact at D.

Energy losses will be lower. When the number of teeth decreased, the angle subtended by a tooth increase; consequently, the moment of inertia increases to the cube due to the increase in width, therefore decrease the efforts induced by a load. Prior to the beginning of the contact between a pair of teeth and having finished the contact of the pair of teeth that goes ahead, there will be a short time without contact between the teeth, which will not imply that the system no longer works. During time t there will be only one pair of teeth in contact and only one impact.

With force at B equal to 200 and torque equal to 453.1538935183, the maximum Von Mises stress equal to 19100 was determined using the finite element method; see Figures 10 and 11.

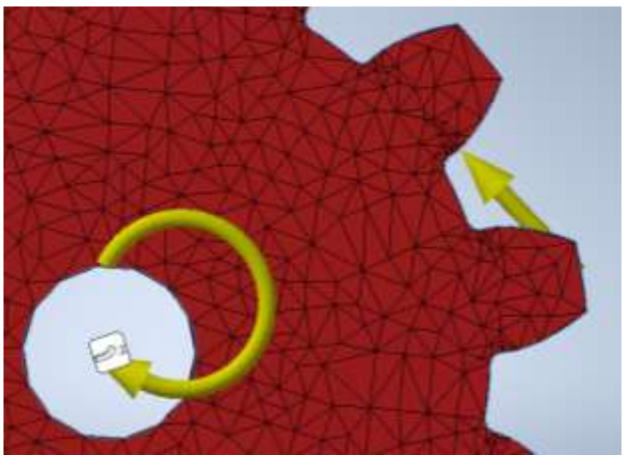

Figure 10 Meshed 3,4
Own Elaboration

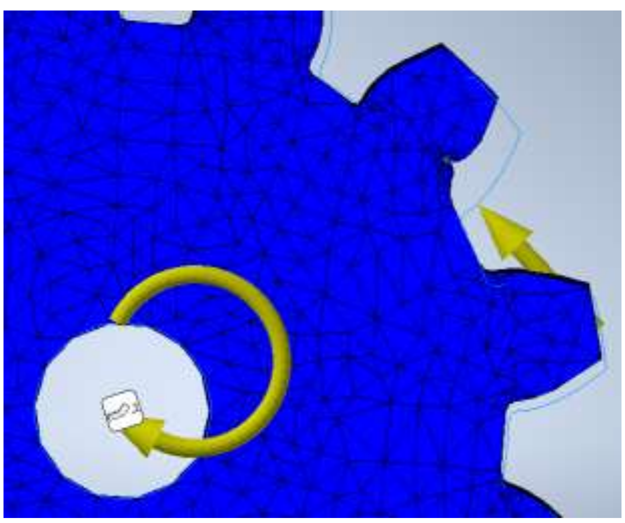

Figure 11 Numerical simulation 3,4
Own Elaboration

Results

The input data of the two systems are indicated in Table 1.

i	1	2	3	4
P_d	4	4	4	4
φ	25	25	25	25
r_i	2.5	3.75	2.5	3.75
N_i	20	30	12	18

Table 1 Gears data
Own Elaboration

The numbers of teeth on the gears are such that the same speeds ratio between 1,2 and 3,4 is preserved.

For better comparison, the contact ratio for 1,2 and 3,4 are listed in Table 2.

i	1	2	3	4
z	1.026393766		1.026393766	
P_{bi}	0.7118124714		1.186354119	
R_c	1.4419440614		0.8651664368	

Table 2 Contact ratios
Own Elaboration

The maximum von Mises stress are recorded in Table 3.

Scenary	σ_{max} at 1	σ_{max} at 3
A	20010	------------------
B	10020	------------------
C	10020 + σ_{ta}	------------------
unique	----------------	19100

Table 3 Efforts maximum
Own Elaboration

Table 4 shown distances between loci, see Figures 1 and 2, as well as vectors of position of input and output teeth, see Figure 3.

Distance	In 1	In 3
BE	0.7118124714	------------------
BC = EG	0.3145812946	------------------
R_B	2.3274025727	2.3274025727
R_G	2.75	2.75
R_E	2.5847511397	------------------
R_{CO2}	3.8434188264	------------------
R_{EO2}	3.6747549369	------------------
R_{GO2}	3.5670068543	3.5670068543

Table 4 Distances
Own Elaboration

Analysis of Results

In gear pair 1 and 2, for 30.6491821169 percent of t there are two pairs of teeth in contact and for 69.3508178831 percent the contact is between a unique pair of teeth. During the same time t there Will be two couplings (shocks) between the teeth of both gears.

In the pair of gears 3 and 4, during 100 percent of t there is a pair of teeth in contact. During 13.48335632 percent of t there are no teeth in contact. During 113.48335632 percent of t there will be a coupling (shock) between of both gears.

Results Comparison

From the results obtained for the pair of gears with normalized number of teeth (20 and 30), and for the pair of gears with reduced number of teeth (12 and 18), it can be asserted:

For a contact ratio equal to 1; when any pair of teeth begins their contact, the pair of teeth ahead will be ending their contact; the loads will act, in any pair of teeth, on the tip of the tooth of gear 2 and on the flank of the tooth of gear 1; the loads will act, in the pair of teeth ahead, on the tip of the tooth of gear 1 and on the flank of the tooth of gear 2. Mechanical errors in the gears make it impossible, in practice, to obtain the unity contact ratio. Contact ratio greater or less than unity? is the logical question.

To answers it, it is necessary to consider that currently the contact ratios of spur gears are greater than unity and have the following disadvantages: abrasive wear, adhesive wear, pitting, variable distance between centre, noise, high temperatures, among others; all of them caused by mechanical errors.

For contact ratios less than unity. Even with mechanical errors, the disadvantages mentioned in the previous paragraph will be considerably minors. It will always be more convenient, considering tolerances and fits, to engage one pair of teeth than to engage two pairs of teeth. By reducing the number of teeth, the circular pitch increases, increasing the width of the tooth as a cantilever beam, reducing consequently the efforts at the root of the tooth.

Even when the presented case study modifies the normalizes number of teeth (20 and 30) reducing it to (12 and 18), it is possible to reduced it to 10 and 15, 8 and 12, 6 and 9; if the passage of one tooth of each gear at a time is ensured.

Gratitude

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Financing

This work has been financed by the Department of Energy.

Conclusions

The proposed spur gear pair design will reduce temperature, noise, abrasive wear, adhesive wear, and pitting, contributing to more sustainable industrial processes. The present work does not significantly modify the possibility of interference between teeth of the pair of spur gear, constituting a niche of opportunity to improve the behaviour of the system.

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