International Interdisciplinary Congress on Renewable Energies, Industrial Maintenance, Mechatronics and Informatics
Booklets

1702902 CONACYT

RENIECYT - LATINDEX - Research Gate - DULCINEA - CLASE - Sudoc - HISPANA - SHERPA UNIVERSIA - Google Scholar DOI - REDIB - Mendeley - DIALNET - ROAD - ORCID

Title: Spur Gears with Contact Ratio Less Than Unity mechanical Engineering

Authors: JIMÉNEZ-RABIELA, Homero, VÁZQUEZ-GONZÁLEZ, Benjamín, RAMÍREZ-CRUZ, José Luis and BRAVO-ACOSTA, Adrian Gustavo

Editorial label ECORFAN: 607-8695
BCIERMMI Control Number: 2022-01
BCIERMMI Classification (2022): 261022-0001

Pages: 14
RNA: 03-2010-032610115700-14

ECORFAN-México, S.C.
143 – 50 Itzopan Street
La Florida, Ecatepec Municipality
Mexico State, 55120 Zipcode
Phone: +52 1 55 6159 2296
Skype: ecorfan-mexico.s.c.
E-mail: contacto@ecorfan.org
Facebook: ECORFAN-México S. C.
Twitter: @EcorfanC

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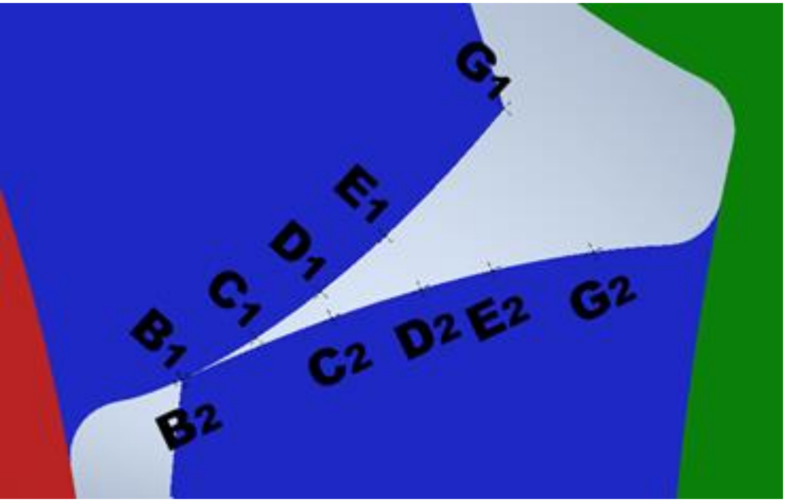
Introduction

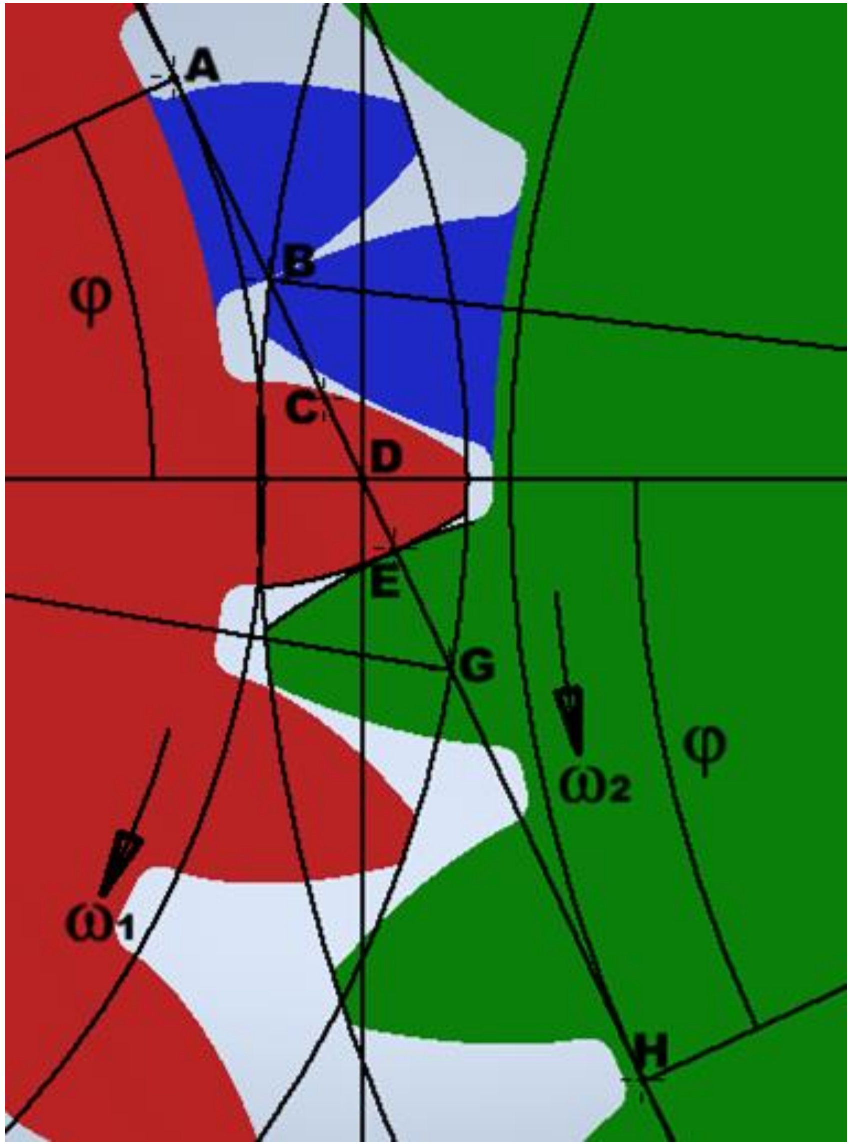
The movement transmission mechanism through spur gears, as well as the corresponding induced efforts, have aroused the interest of the scientific community. Contact relationship has been considered a critical factor for the qualitative and quantitative evaluation of the general behaviour of such gears.

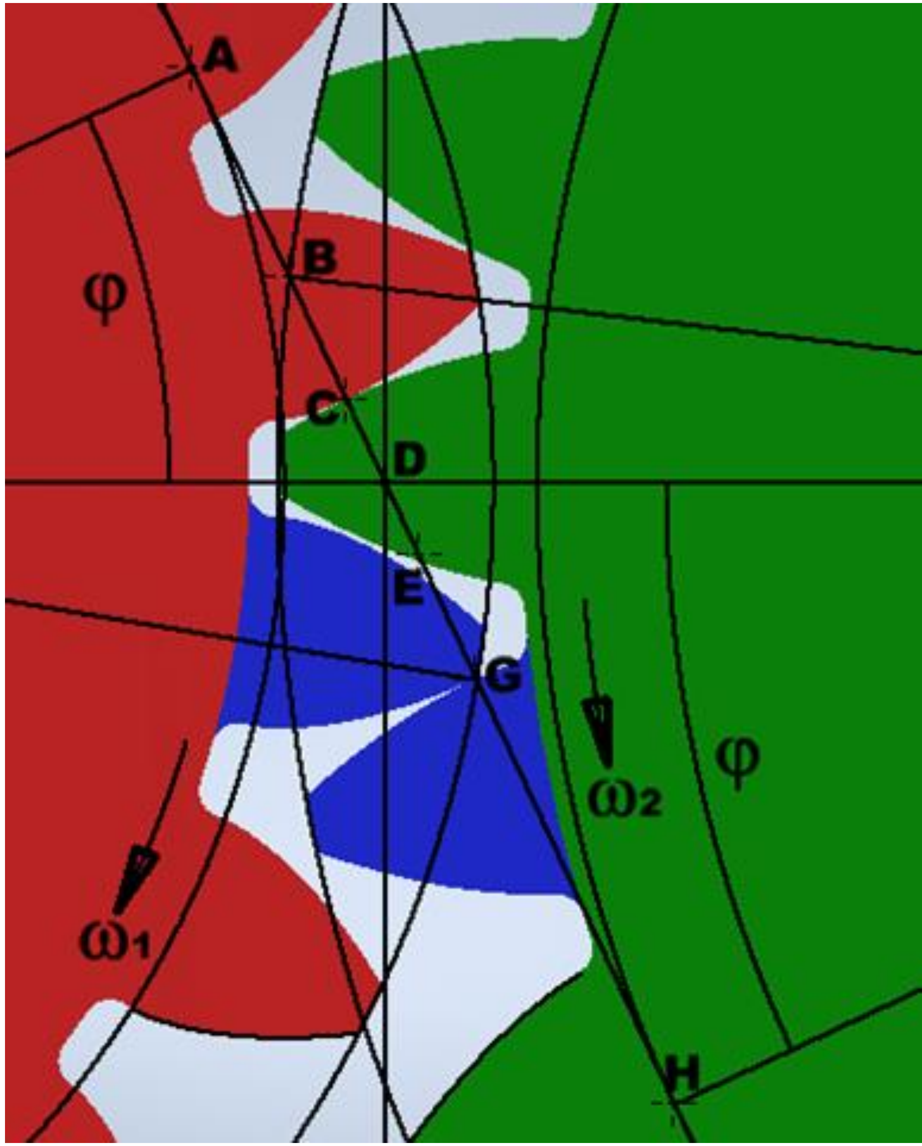
On the page https://vsip.info/cap-10-modos-de-falla-comunes-en-engranajes-pdf-free.html the most common failure modes in gears are described, stating that the cracking of the teeth can initiate on the face and spread to the opposite flank.

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

Nomenclature







Pair of Gears with Standard Number of Teeth

$$z = \overline{AG} + \overline{BH} - \overline{AD} - \overline{DH} \qquad (1)$$

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

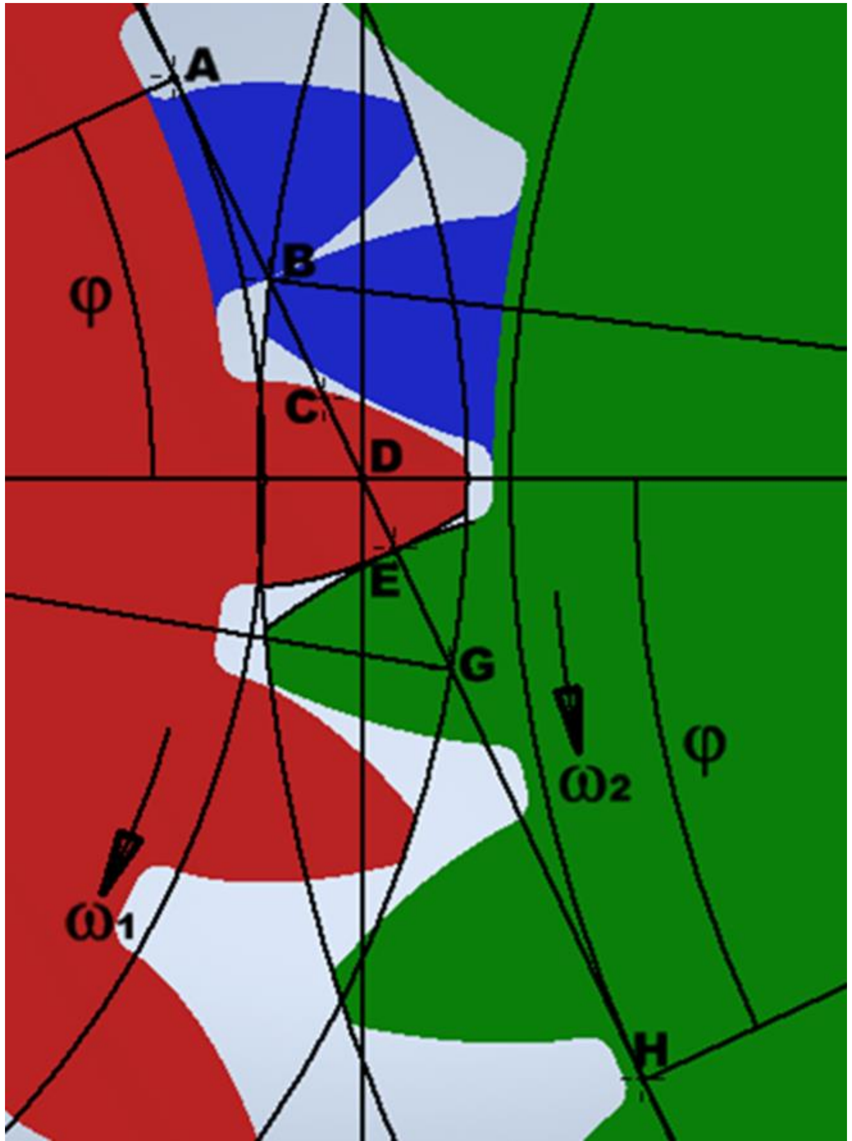
$$\overline{BH} = \sqrt{(r_{a2})^2 - (r_{b2})^2} \qquad (3)$$

$$\overline{AD} = r_1 sen\varphi \qquad (4)$$

$$\overline{DH} = r_2 sen\varphi \qquad (5)$$

Dividing z between P_{bi} we obtain:

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



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

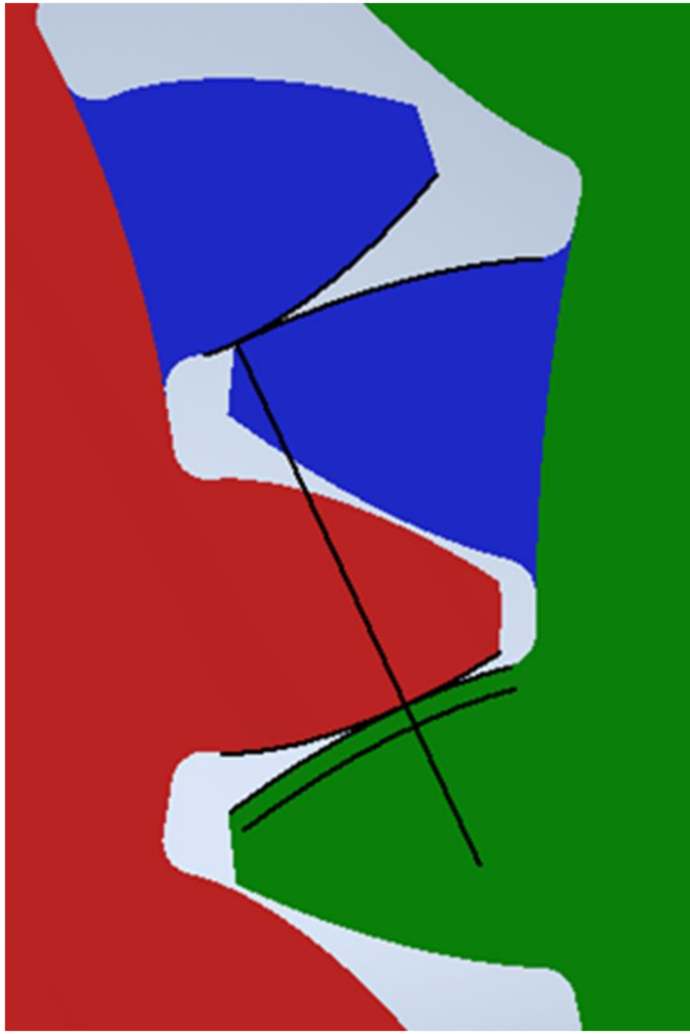
$$y_B = \overline{BH}cos\varphi - r_2 sen\varphi cos\varphi \qquad (7)$$

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

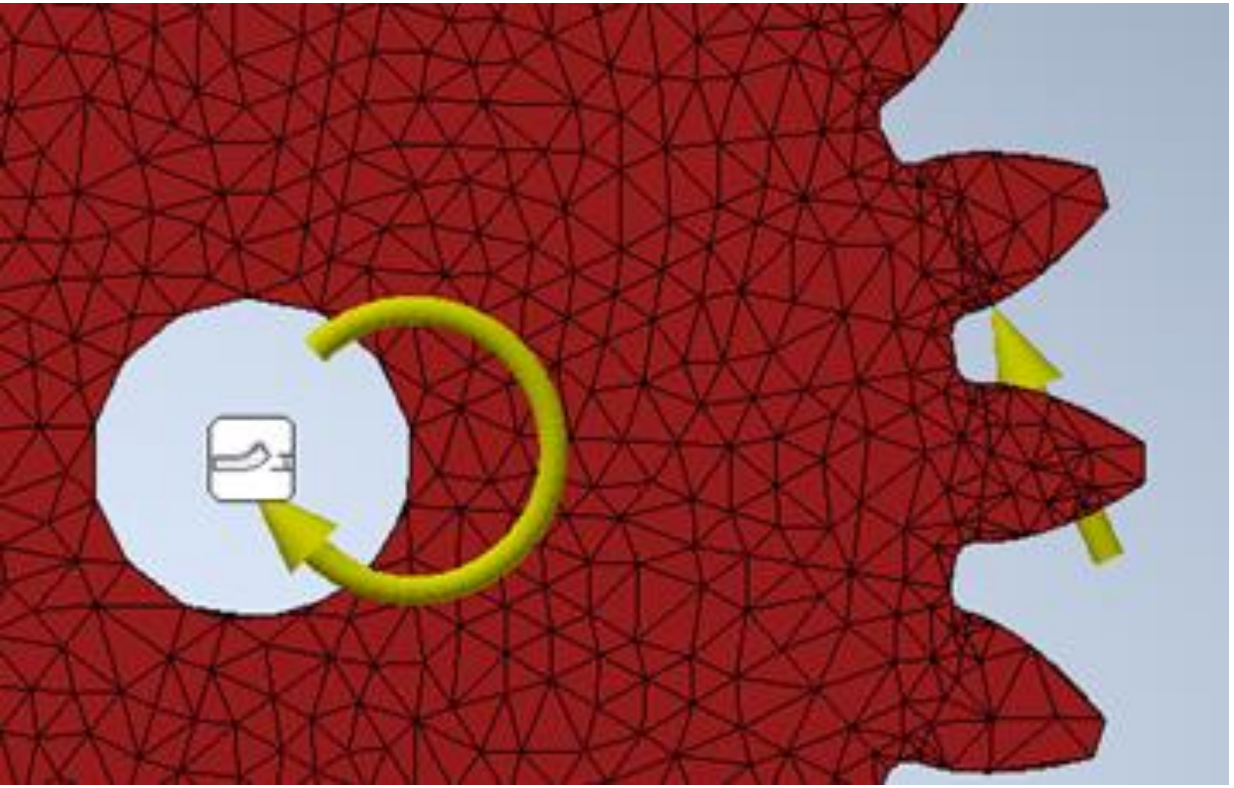
$$y_G = cos\varphi(r_1 sen\varphi - \overline{AG}) \qquad (9)$$

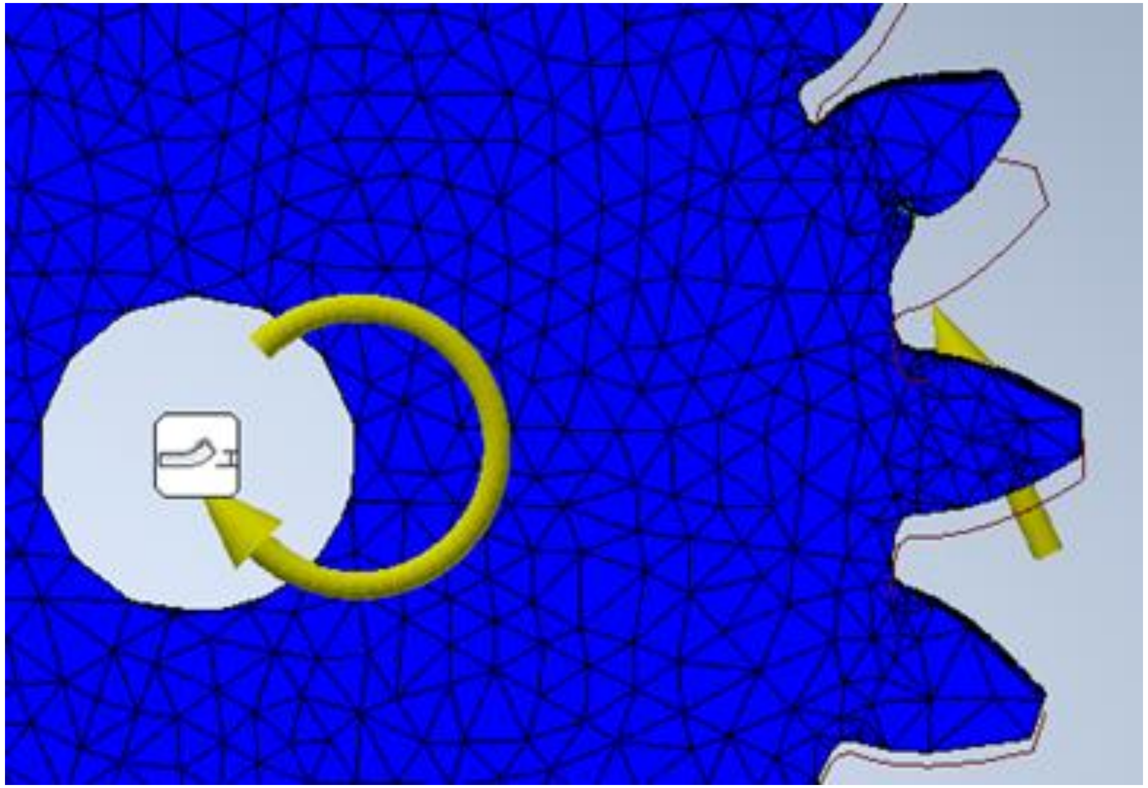
$$x_G = r_1 + \overline{AG}sen\varphi - r_1(sen\varphi)^2 \qquad (10)$$

- 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, will be impossible and the energy will be transmitted only through the first pair. 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.

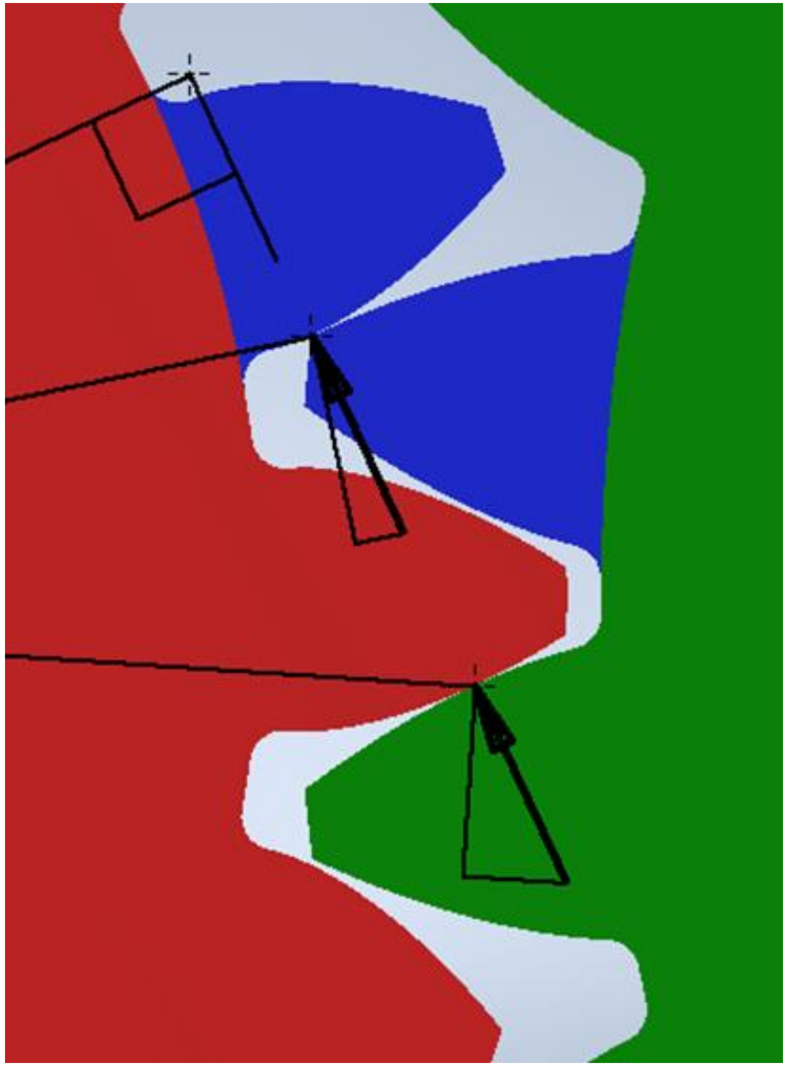


With force F_B equal to 200 and T equal to 453.1538935183, the maximum Von Mises stress equal to 20010 was determined using the finite element method.



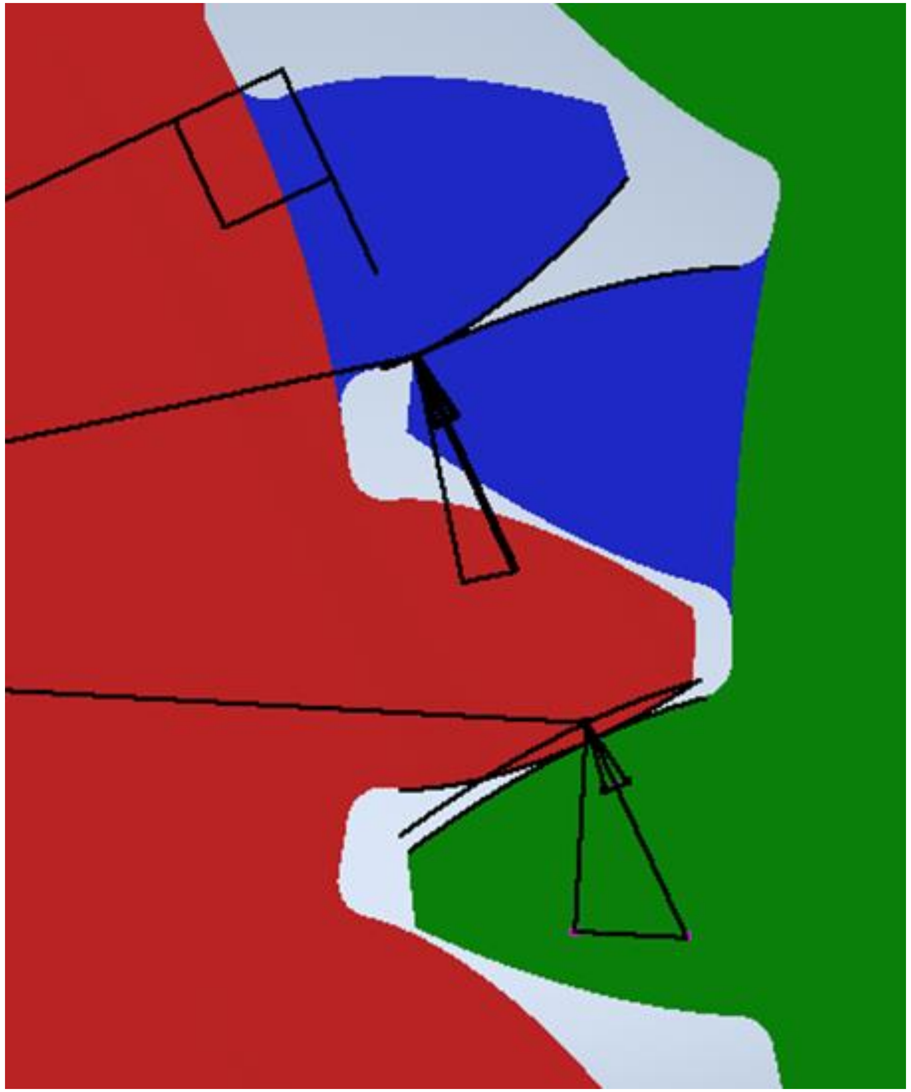


- The separation of the teeth of gear 2 is equal to the of design, the contact of the second pair of teeth will be possible and the energy will be transmitted through the two pair of teeth. Note that the contact in B will be more efficient than in E, because the radial component is smaller. Considering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



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.

- 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. 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.



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,

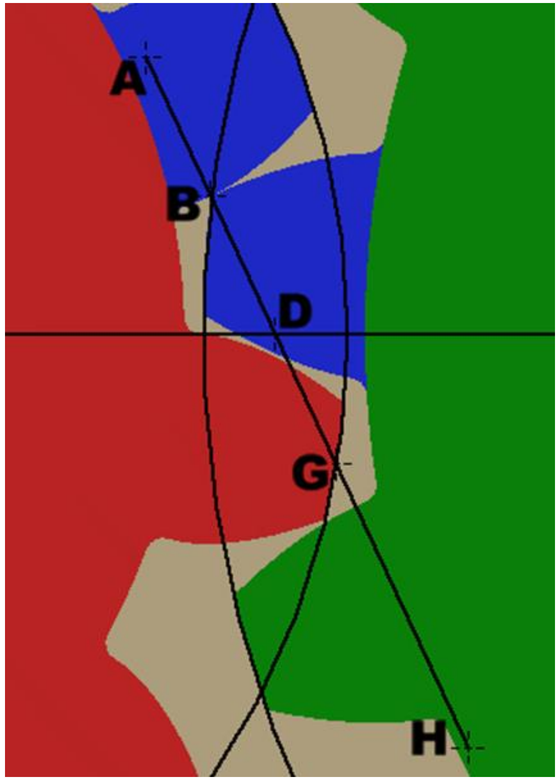
It can be calculated with (1).

Dividing z between P_{bi} we obtain:

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

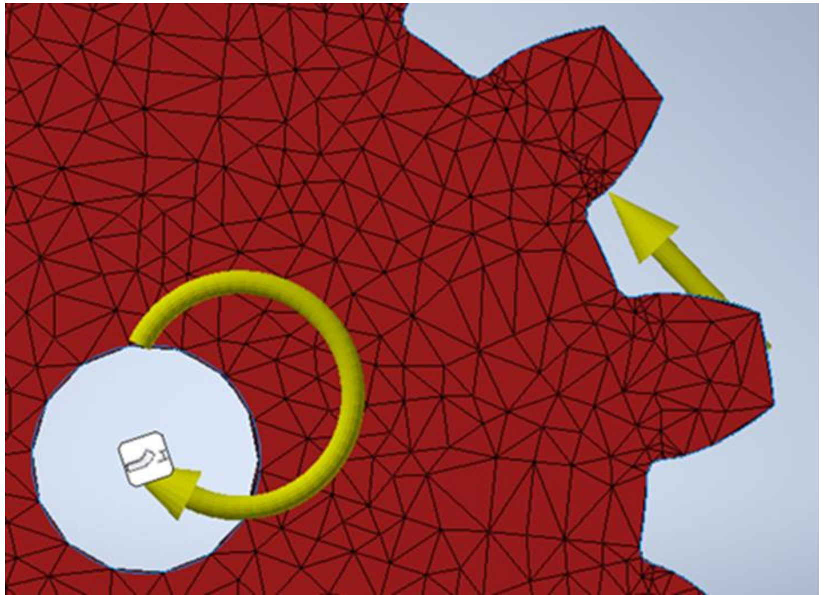
The position vectors of A, B, D, G and H are invariant with respect to those defined in the pair of gears 1 and 2.

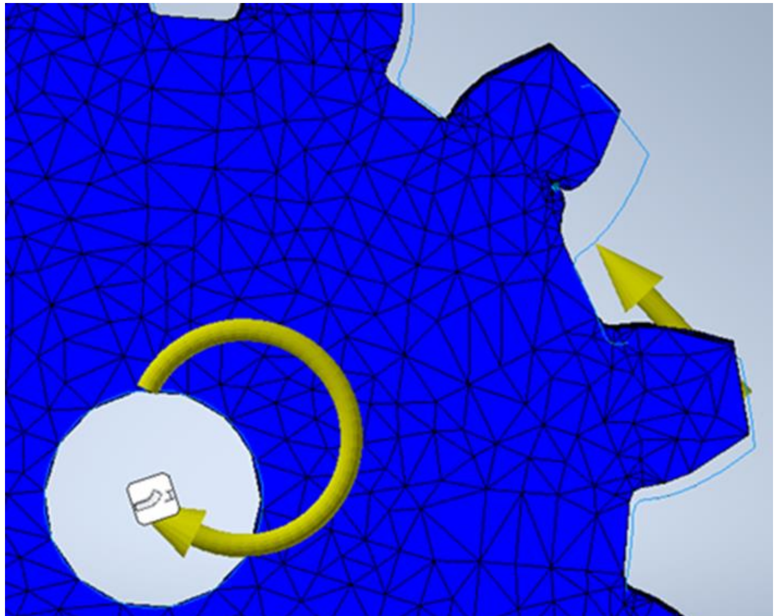
Since the contact of the first pair of teeth will have ended when initiate contact the second, the position vectors of C and E will not exist.



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

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.





Results

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

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

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

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

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.

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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