

Review

Contributions to the Synthesis of Fixed Axle Gears by Avoiding the Interference Phenomenon

Relly Victoria Virgil Petrescu

ARoTMM-IFTtoMM, Bucharest Polytechnic University, Bucharest, (CE), Romania

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Email: rvvpetrescu@gmail.com

Abstract: It can be seen that the minimum number of teeth required to avoid interference for the standard pressure angle, normally on the $\alpha_0 = 20^\circ$ decomposition circle, is 13, corresponding to a transmission ratio $i = 1$ and increases with the transmission ratio i reaching to the maximum value of 18 teeth for $i > 100$. For normal transmission ratios, values ranging from 13 to 17 teeth for standard pressure angle. If α_0 decreases to 4° , z_{\min} varies between 275 and 410 teeth. When α_0 increases to 35° , z_{\min} varies between 5 and 6 teeth. By lowering the number of teeth of the drive wheel 1, the coverage and gearing decreases as well as increases the pressure angle, increases effort, wear and reduces the life span of the gear. If we increase in turn, the minimum number of input wheel teeth increases the coverage, increases the gear efficiency, reduces the pressure angles and effort in the coupling, increases gear reliability and operates with much lower vibrations and noise, with high yields and for a longer time. The minimum number of teeth required to avoid interference is basically a function of the transmission ratio $i = |i_{12}| = z_2/z_1$ and the normal pressure angle of the alpha circle and the angle of inclination of the beta teeth. This is practically also maintained in internal gear gears, where there are still two additional types of interference. It is noted that z_{\min} decreases when i decreasing and when α_0 and/or beta increases.

Keywords: Gears, Gearing, Avoid Interference, Robots, Mechatronic Systems, Mechanical Transmissions

Introduction

According to the standards in force (see Standard 915/2-81), the gear is defined as an elementary mechanism consisting of two gears (wheels, sectors, or toothed racks) in absolute/relative rotation/translation, in which one of the elements trains the other through the action of the teeth in successive and continuous contact.

Gears, or gears with gears, are basically upper couplings (generally C4th grade), which have the function of transmitting and/or turning the movement by reducing the speed (with the torque increase) or by increasing the angular velocity (with lowering the load), from the input to the exit, with an almost constant power holding (with very small losses, mechanical and friction due to the large and very high yields of the gears).

The oldest, most used (more widespread), more reliable and performing better, are fixed axle gears that will be presented in this chapter.

There are also movable axles (subject to a separate chapter), or mixed, which, although lighter and more

compact, work in return with lower yields than fixed axes and are less rigid and reliable.

From a structural-geometric-cinematic (and constructive) point of view, fixed axle gears are classified into three major categories, depending on the relative position of the axles of the two wheels making up the gear:

- A-parallel (cylindrical),
- B-competing conical)
- C-crosses (spindle-worm, spatial, toroidal).

The cylindrical gears (A) may be external (between two external gears) or inside (between an external gear and one with internal tothing).

They can also be combined, an element with rotating motion (a toothed wheel with external teeth) and the other one (rack).

The geometric elements of a toothed wheel and a gear can be seen in Fig. 1 and 2 (according to international standards).

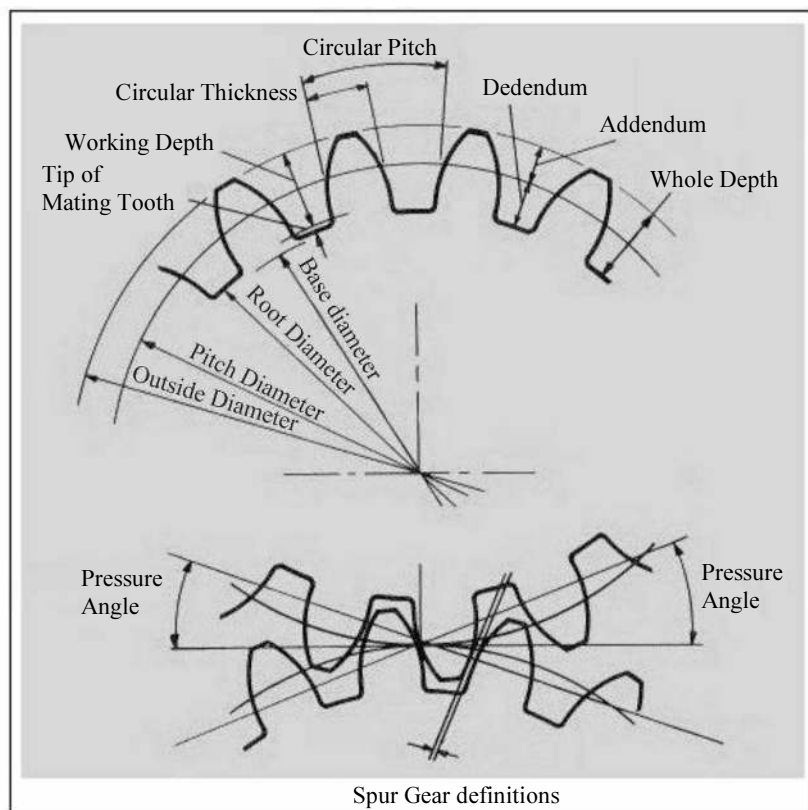
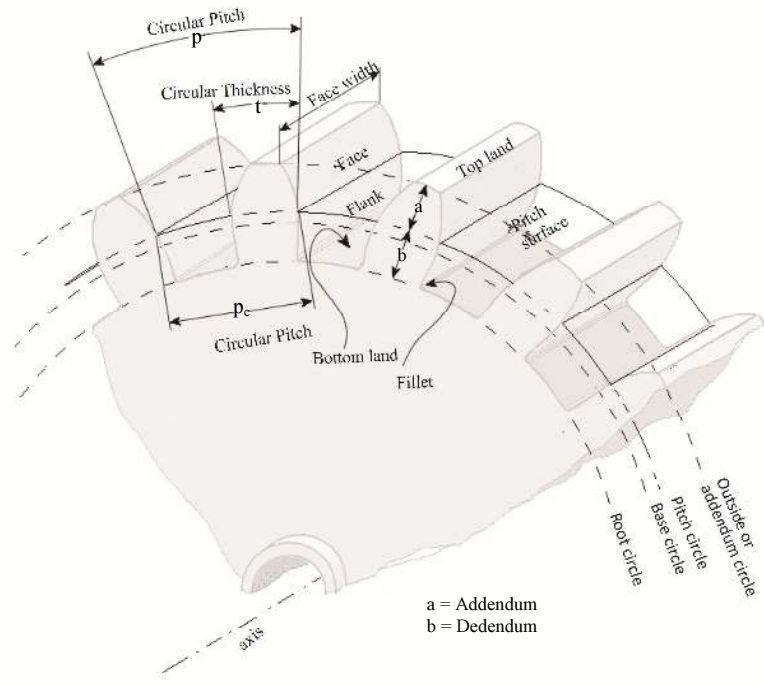


Fig. 1: The geometrical elements of a straight toothed gear; head, root and division circles; circular step

If the rotation axes are parallel, the gear unit is called cylindrical. When the tooth line has the same

direction as the axis of rotation, it is said that the gear has straight teeth.

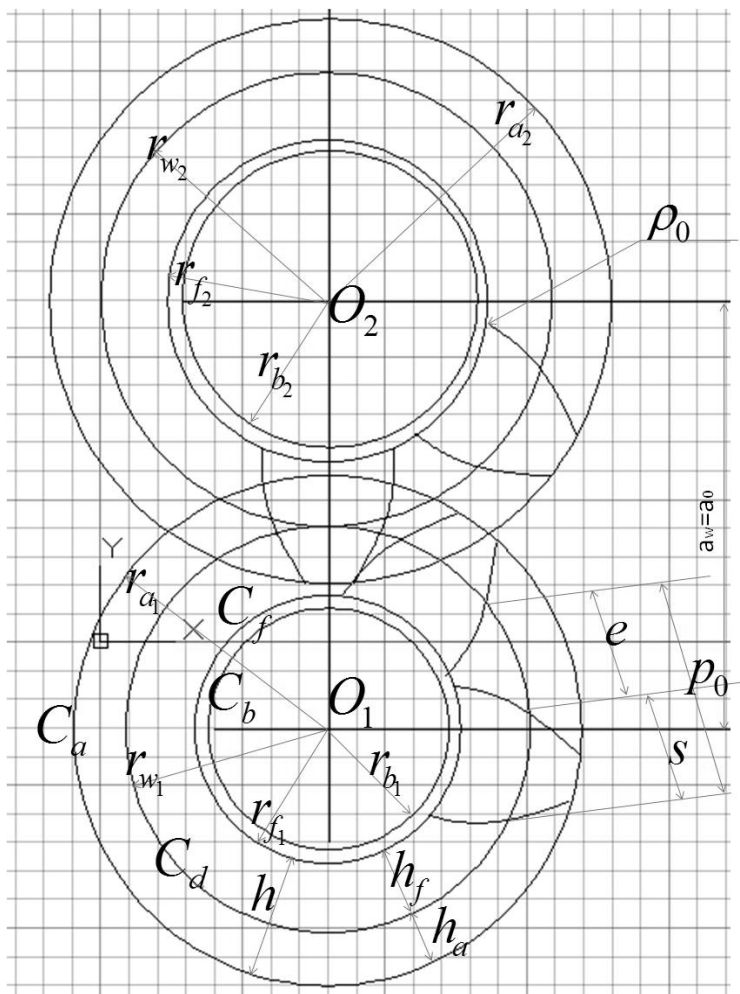


Fig. 2: The geometrical elements of a straight toothed gear; head, root and division circles; circular step

The main parameters of such gear are shown in Fig. 1, in which the teeth of an unpainted wheel are represented in an outer cylindrical gear not rigid with straight teeth.

The starting element of a wheel is the splitting circle (or step - on which the pitch is measured), a circle that defines the position of the other circles of the wheel. The diameter of the splitting circle is one of the first elements to be calculated on a wheel and on a gear (at a gear we will have two wheels, so two division diameters).

Rulkov *et al.* (2016; Agarwala, 2016; Babayemi, 2016; Gusti and Semin, 2016; Mohamed *et al.*, 2016; Wessels and Raad, 2016; Maraveas *et al.*, 2015; Khalil, 2015; Rhode-Barbarigos *et al.*, 2015; Takeuchi *et al.*, 2015; Li *et al.*, 2015; Vernardos and Gantes, 2015; Bourahla and Blakeborough, 2015; Stavridou *et al.*, 2015; Ong *et al.*, 2015; Dixit and Pal, 2015; Rajput *et al.*, 2016; Rea and Ottaviano, 2016; Zurfi and Zhang, 2016a; 2016b; Zheng and Li, 2016; Buonomano *et al.*, 2016a; 2016b; Faizal *et al.*, 2016; Cataldo, 2006; Ascione *et al.*, 2016; Elmeddahi *et al.*, 2016; Calise *et al.*, 2016;

Morse *et al.*, 2016; Abouobaida, 2016; Rohit and Dixit, 2016; Kazakov *et al.*, 2016; Alwetaishi, 2016; Riccio *et al.*, 2016a; 2016b; Iqbal, 2016; Hasan and El-Naas, 2016; Al-Hasan and Al-Ghamdi, 2016; Jiang *et al.*, 2016; Sepúlveda, 2016; Martins *et al.*, 2016; Pisello *et al.*, 2016; Jarahi, 2016; Mondal *et al.*, 2016; Mansour, 2016; Al Qadi *et al.*, 2016b; Campo *et al.*, 2016; Samantaray *et al.*, 2016; Malomar *et al.*, 2016; Rich and Badar, 2016; Hirun, 2016; Bucinell, 2016; Nabilou, 2016b; Barone *et al.*, 2016; Chisari and Bedon, 2016; Bedon and Louter, 2016; Santos and Bedon, 2016; Minghini *et al.*, 2016; Bedon, 2016; Jafari *et al.*, 2016; Chiozzi *et al.*, 2016; Orlando and Benvenuti, 2016; Wang and Yagi, 2016; Obaiys *et al.*, 2016; Ahmed *et al.*, 2016; Jauhari *et al.*, 2016; Syahrullah and Sinaga, 2016; Shanmugam, 2016; Jaber and Bicker, 2016; Wang *et al.*, 2016; Moubarek and Gharsallah, 2016; Amani, 2016; Shruti, 2016; Pérez-de León *et al.*, 2016; Mohseni and Tsavdaridis, 2016; Abu-Lebdeh *et al.*, 2016; Serebrennikov *et al.*, 2016; Budak *et al.*, 2016;

Augustine *et al.*, 2016; Jarahi and Seifilaleh, 2016; Nabilou, 2016a; You *et al.*, 2016; AL Qadi *et al.*, 2016a; Rama *et al.*, 2016; Sallami *et al.*, 2016; Huang *et al.*, 2016; Ali *et al.*, 2016; Kamble and Kumar, 2016; Saikia and Karak, 2016; Zeferino *et al.*, 2016; Pravettoni *et al.*, 2016; Bedon and Amadio, 2016; Chen and Xu, 2016; Mavukkandy *et al.*, 2016; Gruener, 2006; Yeargin *et al.*, 2016; Madani and Dababneh, 2016; Alhasanat *et al.*, 2016; Elliott *et al.*, 2016; Suarez *et al.*, 2016; Kuli *et al.*, 2016; Waters *et al.*, 2016; Montgomery *et al.*, 2016; Lamarre *et al.*, 2016; Daud *et al.*, 2008; Taher *et al.*, 2008; Zulkifli *et al.*, 2008; Pourmahmoud, 2008; Pannirselvam *et al.*, 2008; Ng *et al.*, 2008; El-Tous, 2008; Akhesmeh *et al.*, 2008; Nachientai *et al.*, 2008; Moezi *et al.*, 2008; Boucetta, 2008; Darabi *et al.*, 2008; Semin and Bakar, 2008; Al-Abbas, 2009; Abdullah *et al.*, 2009; Abu-Ein, 2009; Opafunso *et al.*, 2009; Semin *et al.*, 2009a; 2009b; 2009c; Zulkifli *et al.*, 2009; Marzuki *et al.*, 2015; Bier and Mostafavi, 2015; Momta *et al.*, 2015; Farokhi and Gordini, 2015; Khalifa *et al.*, 2015; Yang and Lin, 2015; Chang *et al.*, 2015; Demetriou *et al.*, 2015; Rajupillai *et al.*, 2015; Sylvester *et al.*, 2015; Ab-Rahman *et al.*, 2009; Abdullah and Halim, 2009; Zotos and Costopoulos, 2009; Feraga *et al.*, 2009; Bakar *et al.*, 2009; Cardu *et al.*, 2009; Bolonkin, 2009a; 2009b; Nandhakumar *et al.*, 2009; Odeh *et al.*, 2009; Lubis *et al.*, 2009; Fathallah and Bakar, 2009; Marghany and Hashim, 2009; Kwon *et al.*, 2010; Aly and Abuelnasr, 2010; Farahani *et al.*, 2010; Ahmed *et al.*, 2010; Kunanoppadon, 2010; Helmy and El-Taweel, 2010; Qutbodin, 2010; Pattanasethanon, 2010; Fen *et al.*, 2011; Thongwan *et al.*, 2011; Theansuwan and Triratanasirichai, 2011; Al Smadi, 2011; Tourab *et al.*, 2011; Raptis *et al.*, 2011; Momani *et al.*, 2011; Ismail *et al.*, 2011; Anizan *et al.*, 2011; Tsolakis and Raptis, 2011; Abdullah *et al.*, 2011; Kechiche *et al.*, 2011; Ho *et al.*, 2011; Rajbhandari *et al.*, 2011; Aleksic and Lovric, 2011; Kaewnai and Wongwises, 2011; Idarwazeh, 2011; Ebrahim *et al.*, 2012; Abdelkrim *et al.*, 2012; Mohan *et al.*, 2012; Abam *et al.*, 2012; Hassan *et al.*, 2012; Jalil and Sampe, 2013; Jaoude and El-Tawil, 2013; Ali and Shumaker, 2013; Zhao, 2013; El-Labban *et al.*, 2013; Djalel *et al.*, 2013; Nahas and Kozaitis, 2013; Petrescu and Petrescu, 2014a; 2014b; 2014c; 2014d; 2014e; 2014f; 2014g; 2014h; 2014i; 2015a; 2015b; 2015c; 2015d; 2015e; 2016a; 2016b; 2016c; 2016d; Fu *et al.*, 2015; Al-Nasra *et al.*, 2015; Amer *et al.*, 2015; Sylvester *et al.*, 2015b; Kumar *et al.*, 2015; Gupta *et al.*, 2015; Stavridou *et al.*, 2015b; Casadei, 2015; Ge and Xu, 2015; Moretti, 2015; Wang *et al.*, 2015; Antonescu and Petrescu, 1985; 1989; Antonescu *et al.*, 1985a; 1985b; 1986; 1987; 1988; 1994; 1997; 2000a; 2000b; 2001; Aversa *et al.*, 2017a; 2017b; 2017c; 2017d; 2017e; 2016a; 2016b; 2016c; 2016d; 2016e; 2016f; 2016g;

2016h; 2016i; 2016j; 2016k; 2016l; 2016m; 2016n; 2016o; Cao *et al.*, 2013; Dong *et al.*, 2013; Comanescu, 2010; Franklin, 1930; He *et al.*, 2013; Lee, 2013; Lin *et al.*, 2013; Liu *et al.*, 2013; Padula and Perdereau, 2013; Perumaal and Jawahar, 2013; Petrescu, 2011; 2015a; 2015b; Petrescu and Petrescu, 1995a; 1995b; 1997a; 1997b; 1997c; 2000a; 2000b; 2002a; 2002b; 2003; 2005a; 2005b; 2005c; 2005d; 2005e; 2011a; 2011b; 2012a; 2012b; 2013a; 2013b; 2013c; 2013d; 2013e; 2016a; 2016b; 2016c; Petrescu *et al.*, 2009; 2016; 2017a; 2017b; 2017c; 2017d; 2017e; 2017f; 2017g; 2017h; 2017i; 2017j; 2017k; 2017l; 2017m; 2017n; 2017o; 2017p; 2017q; 2017r; 2017s; 2017t; 2017u; 2017v; 2017w; 2017x; 2017y; 2017z; 2017aa; 2017ab; 2017ac; 2017ad; 2017ae; 2018a; 2018b; 2018c; 2018d; 2018e; 2018f; 2018g; 2018h; 2018i; 2018j; 2018k; 2018l; 2018m; 2018n).

Materials and Methods

The number of gear pairs engaged simultaneously (for good gearing) is the degree of coverage. So the gear ratio ("contact ratio" in English) marked with \square (shows how many pairs of teeth are engaged at the same time; Fig. 3).

In order for the engagement to take place without shocks, without sliding, no noise and no play, the gear is designed so that when a pair of teeth is out of engagement, the next pair is already engaged.

The circle on the base circle shows how long a pair is engaged.

Whenever it encompasses in the actual AE engagement segment, so many pairs of engagement will fit simultaneously into the AE segment on which the actual engagement is made. Practically, the degree of coverage will be the ratio of AE to p_b .

It must be overhead to have multiple pairs in simultaneous engagement so that no "dead times", interruptions of engagement, gaming and gambling collisions occur due to gaming, which also produces vibrations and noises.

A higher degree of coverage also brings increased mechanical efficiency.

The engagement segment AE is calculated directly with relation (1), Fig. 4:

$$\begin{cases} AE = K_1E + K_2A - K_1K_2 \\ K_1E = \sqrt{r_{a_1}^2 - r_{b_1}^2} \\ K_2A = \sqrt{r_{a_2}^2 - r_{b_2}^2} \\ K_1K_2 = K_1C + K_2C = r_1 \cdot \sin \alpha_0 + r_2 \cdot \sin \alpha_0 \\ = (r_1 + r_2) \cdot \sin \alpha_0 = a_0 \cdot \sin \alpha_0 \\ AE = \sqrt{r_{a_1}^2 - r_{b_1}^2} + \sqrt{r_{a_2}^2 - r_{b_2}^2} - a_0 \cdot \sin \alpha_0 \end{cases} \quad (1)$$

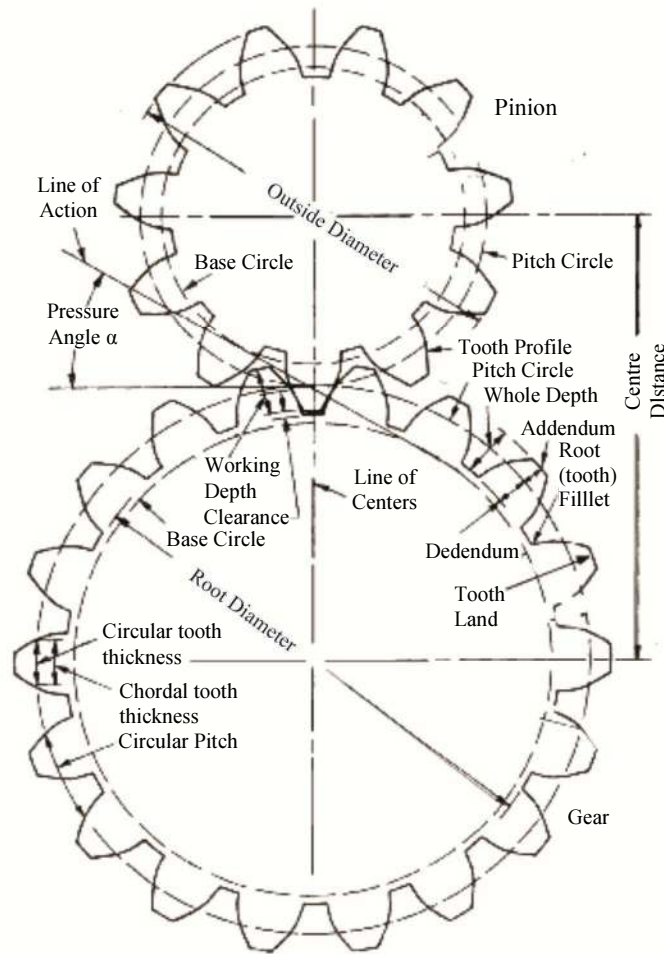


Fig. 3: The geometrical elements of a straight toothed gear; the right-hand drive (or line of action) or the pressure line

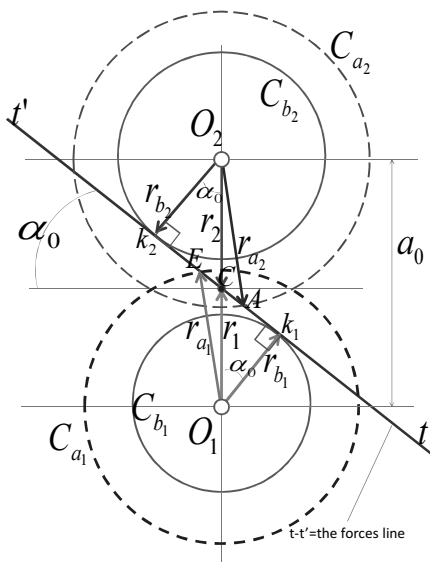


Fig. 4: The pressure or forces line

The degree of coverage ε is determined by dividing AE to the step p_b (relation 2):

$$\varepsilon \equiv \varepsilon_{12} = \frac{\sqrt{r_{a1}^2 - r_{b1}^2} + \sqrt{r_{a2}^2 - r_{b2}^2} - a_0 \cdot \sin \alpha_0}{m \cdot \pi \cdot \cos \alpha_0} \quad (2)$$

Results and Discussion

In order to avoid the interference phenomenon (Fig. 4), point A must be between C and K_1 (i.e., the wheel head wheel 2, Ca_2 must cut the engagement segment between the points C and K_1 and in no shape not exceed K_1). Similarly, the circle Ca_1 must cut the right-hand drive between points C and K_2 , determining the point E, which in no way must pass K_2 . These conditions of avoidance of interference are written with relations (3):

$$\left\{ \begin{array}{l}
 CA < K_1 C \quad \text{si} \quad CE < K_2 C \\
 CA = K_2 A - K_2 C = \sqrt{r_{a_2}^2 - r_{b_2}^2} - r_2 \cdot \sin \alpha_0; \quad CA < K_1 C \\
 \Rightarrow \sqrt{r_{a_2}^2 - r_{b_2}^2} - r_2 \cdot \sin \alpha_0 < r_1 \cdot \sin \alpha_0 \Rightarrow \sqrt{r_{a_2}^2 - r_{b_2}^2} < (r_1 + r_2) \cdot \sin \alpha_0 \\
 \Rightarrow d_{a_2}^2 - d_{b_2}^2 < (d_1 + d_2)^2 \cdot \sin^2 \alpha_0 \Rightarrow \\
 \Rightarrow m^2 \cdot (z_2 + 2)^2 - m^2 \cdot z_2^2 \cdot \cos^2 \alpha_0 < m^2 \cdot (z_1 + z_2)^2 \cdot \sin^2 \alpha_0 \\
 \Rightarrow z_2^2 + 4 \cdot z_2 + 4 - z_2^2 < z_1^2 \cdot \sin^2 \alpha_0 + 2 \cdot z_1 \cdot z_2 \cdot \sin^2 \alpha_0 \\
 \Rightarrow 4 \cdot z_2 + 4 < z_1^2 \cdot \sin^2 \alpha_0 + 2 \cdot z_1 \cdot z_2 \cdot \sin^2 \alpha_0 \\
 \text{from } CE < K_2 C \Rightarrow 4 \cdot z_1 + 4 < z_2^2 \cdot \sin^2 \alpha_0 + 2 \cdot z_1 \cdot z_2 \cdot \sin^2 \alpha_0 \\
 \text{obtains sistem } \begin{cases} 4 \cdot z_2 + 4 < z_1^2 \cdot \sin^2 \alpha_0 + 2 \cdot z_1 \cdot z_2 \cdot \sin^2 \alpha_0 \\ 4 \cdot z_1 + 4 < z_2^2 \cdot \sin^2 \alpha_0 + 2 \cdot z_1 \cdot z_2 \cdot \sin^2 \alpha_0 \end{cases} \\
 \text{for } i \equiv |i_{12}| = \frac{z_2}{z_1} \Rightarrow z_2 = i \cdot z_1; \Rightarrow \\
 \begin{cases} \sin^2 \alpha_0 \cdot (1 + 2 \cdot i) \cdot z_1^2 - 2 \cdot 2 \cdot i \cdot z_1 - 4 > 0 \\ \sin^2 \alpha_0 \cdot (i^2 + 2 \cdot i) \cdot z_1^2 - 2 \cdot 2 \cdot z_1 - 4 > 0 \end{cases} \quad \text{with solutions :} \\
 \begin{cases} z_{1,2} = \frac{2 \cdot i \pm 2 \cdot \sqrt{i^2 + \sin^2 \alpha_0 + 2 \cdot i \cdot \sin^2 \alpha_0}}{(2 \cdot i + 1) \cdot \sin^2 \alpha_0} \\ z_{1,3,4} = \frac{2 \pm 2 \cdot \sqrt{1 + i^2 \cdot \sin^2 \alpha_0 + 2 \cdot i \cdot \sin^2 \alpha_0}}{(2 \cdot i + i^2) \cdot \sin^2 \alpha_0} \end{cases} \quad \text{keeps solutions +} \\
 \begin{cases} z_{1_2} = 2 \cdot \frac{i + \sqrt{i^2 + \sin^2 \alpha_0 + 2 \cdot i \cdot \sin^2 \alpha_0}}{(2 \cdot i + 1) \cdot \sin^2 \alpha_0} \\ z_{1_4} = 2 \cdot \frac{1 + \sqrt{1 + i^2 \cdot \sin^2 \alpha_0 + 2 \cdot i \cdot \sin^2 \alpha_0}}{(2 \cdot i + i^2) \cdot \sin^2 \alpha_0} \end{cases} \quad (3)
 \end{array} \right.$$

Table 1: Z_{min} for avoiding the interferences

α_0	20 [deg]									
i	1	1.25	1.6	2	2.5	3.15	4	5	6.3	8
z_{1_2}	12.32	12.96	13.62	14.16	14.64	15.07	15.44	15.74	15.99	16.22
α_0	20 [deg]									
i	10	12.5	16	20	25	31.5	40	50	63	80
z_{1_2}	16.38	16.52	16.64	16.73	16.80	16.86	16.91	16.95	16.98	17.00
α_0	4 [deg]									
i	1	1.25	1.6	2	2.5	3.15	4	5	6.3	8
z_{1_2}	275.	294.4	313.8	329.3	342.9	355.	365.6	373.9	380.9	387.0
α_0	35 [deg]									
i	1	1.25	1.6	2	2.5	3.15	4	5	6.3	8
z_{1_2}	4.88	5.03	5.19	5.32	5.44	5.55	5.64	5.72	5.79	5.84

The relation that generates it always gives lower values than the relationship that generates it, so that the condition (4) is sufficient to find the minimum number of teeth required to avoid the interference of the gear teeth; in other words, the initial condition is that point A should be between points C and K:

$$z_{1_2} = 2 \cdot \frac{i + \sqrt{i^2 + \sin^2 \alpha_0 + 2 \cdot i \cdot \sin^2 \alpha_0}}{(2 \cdot i + 1) \cdot \sin^2 \alpha_0} \quad (4)$$

Table 1 shows the values obtained with the relationship (4) for different standardized values of the

transmission ratio i and for three different values assigned to the pressure angle.

Conclusion

It can be seen that the minimum number of teeth required to avoid interference for the standard pressure angle, normally on the $\alpha_0 = 20^\circ$ decomposition circle, is 13, corresponding to a transmission ratio $i = 1$ and increases with the transmission ratio i reaching to the maximum value of 18 teeth for $i > 100$. For normal transmission ratios, values ranging from 13 to 17 teeth for standard pressure angle. If α_0 decreases to 4° , z_{\min} varies between 275 and 410 teeth.

When α_0 increases to 35° , z_{\min} varies between 5 and 6 teeth.

By lowering the number of teeth of the drive wheel 1, the coverage and gearing decreases as well as increases the pressure angle, increases effort, wear and reduces the life span of the gear.

If we increase in turn, the minimum number of input wheel teeth increases the coverage, increases the gear efficiency, reduces the pressure angles and effort in the coupling, increases gear reliability and operates with much lower vibrations and noise, with high yields and for a longer time.

The minimum number of teeth required to avoid interference is basically a function of the transmission ratio $i = |i_{12}| = z_2/z_1$ and the normal pressure angle of the alpha circle and the angle of inclination of the beta teeth.

This is practically also maintained in internal gear gears, where there are still two additional types of interference.

It is noted that z_{\min} decreases when i decreasing and when α_0 and/or β increases.

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2-Contract research integration. 19-91-3 from 29.03.1991; Beneficiary: MIS; TOPIC: Research on designing mechanisms with bars, cams and gears, with application in industrial robots.

3-Contract research. GR 69/10.05.2007: NURC in 2762; theme 8: Dynamic analysis of mechanisms and manipulators with bars and gears.

4-Labor contract, no. 35/22.01.2013, the UPB, "Stand for reading performance parameters of kinematics and dynamic mechanisms, using inductive and incremental encoders, to a Mitsubishi Mechatronic System" "PN-II-IN-CI-2012-1-0389".

All these matters are copyrighted! Copyrights: 394-qodGnhhtej, from 17-02-2010 13:42:18; 463-vpstuCGsiy, from 20-03-2010 12:45:30; 631-sqfsgqvutm, from 24-05-2010 16:15:22; 933-CrDztEfqow, from 07-01-2011 13:37:52.

Ethics

This article is original and contains unpublished material. Authors declare that are not ethical issues and no conflict of interest that may arise after the publication of this manuscript.

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