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Reconstitution of the geometric elements of a conic gear

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Abstract. Conical gears are used if the axes of the shafts between which the rotation movement is transmitted are concurrent. The technology of execution of these gears is more demanding, they are more sensitive to the deviations of execution and / or assembly and introduce large axial forces, which complicates, to some extent, the construction of the supports of the wheel support shafts.
The straight-toothed conical gears, with relatively simple wheels for technological purposes, are used only at low peripheral speeds ($v < 3\ m/s$), when step deviations and those of the tooth profiles do not yet produce high dynamic demands and noise. These gears, however, are sensitive to less precise mountings and to deformations, under load, of the support shafts.
The calculation of the resistance of the conical gears, in contact and bending, is performed accepting the same calculation assumptions as in the calculation of cylindrical gears with straight teeth. It is, therefore, necessary to make a transition from the conical gear to an imaginary cylindrical gear, called virtual gear, and to find equivalence relations between the two gears - real and virtual.

1. Introduction

Conical gears are used if the axes of the shafts between which the rotation movement is transmitted are concurrent. The technology of execution of these gears is more demanding, they are more sensitive to the deviations of execution and / or assembly and introduce large axial forces, which complicates, to some extent, the construction of the supports of the wheel support shafts. As a rule, the conical pinion (the driving wheel) is mounted in the console, which leads to the increase of the elastic deformations of the shaft that supports it, negatively influencing the gear. Therefore, the input shaft must be oversized in terms of strength but not in terms of rigidity.

Constructively, it is necessary to create possibilities for "adjusting" the conical gear, so that the tips of the cones of the two wheels overlap and the contact of the teeth takes place in the central area of the tooth flanks.

The conical wheels, depending on the position of the tooth (relative to the cone generator) and its shape, can be executed with straight, inclined or curved teeth, defining conical gears with straight, inclined or curved teeth. Use one or other of these gears is linked to conditions imposed kinematic and dynamic transmission noise and execution technology.

2. General aspects about the replacement (virtual) cylindrical gear

The calculation of the resistance of the conical gears, on contact and bending, is performed by accepting the same calculation assumptions as with the calculation of cylindrical gears with straight teeth. It is, therefore, necessary to make a transition from the conical gear to an imaginary cylindrical gear, called *virtual gear*, and to find equivalence relations between the two gears - real and virtual.

The geometry of the cylindrical gear has been studied in a frontal plane section, and that of the conical gear can be performed in a front spherical section with the center at the point of intersection of the gear wheels axes. However, the study of the geometry of the conical wheels on such a spherical surface is complicated, which justifies the approximation of the front spherical surface by a flat (frontal) surface.

Since the sphere is not planar, the spherical areas containing the tooth profiles are approximated by cone trunks, tangent to the sphere after the splitting circles (Tredgold approximation).

The cones to which these cone trunks belong are called frontal cones. The front cone with the tip in O_1 is attached to the conical wheel 1 (with the angle of the cone δ_1), and the one with the tip in O2, the conical wheel 2 (with the angle of the cone δ_2), according to fig.1.a. Through the Tredgold approximation, the profile of the teeth on the sphere is projected onto the surface of the cone trunks.

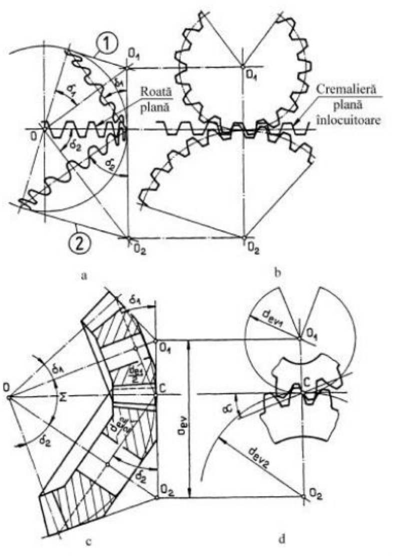

Figure 1 - Spherical areas of tooth profiles

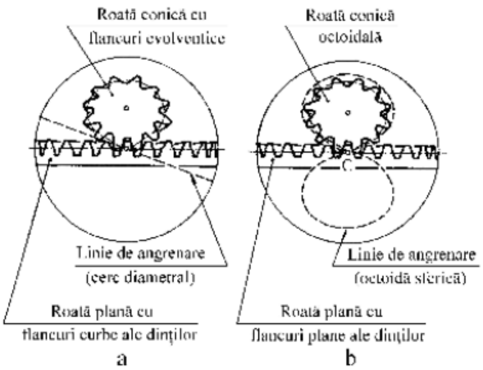

Figure 2 – Lines flanks

The unfolding of the outer front cones in plan, shown in fig.1.a, results in a cylindrical gear (replacement) of the real conical gear (see fig.1. b). The transition from the real conical gear to the replacement cylindrical gear is shown in fig.1. c and d. Because there is an infinity of concentric spheres, with the center in O (see fig.1. a), it is necessary to specify to which spheres the frontal cones are attached, to determine in which area the replacement cylindrical gear has been made.

The front cones used are exterior, middle and interior, whose characteristic elements will be noted with the indices *e, m, i*, and for the virtual gear the characteristic index will be v ($a_{ev}, d_{ev\,1}, d_{ev\,2}$).

By the approximation made, the flat wheel is bordered on the outside by a cylindrical surface, called the outer front cylinder. By carrying out in plan this cylinder, which contains the profiles of the flat wheel teeth, a *planar replacement rack* is obtained (fig.1. b). The profile of the replacement rack is approximated from a curve (sinusoidal segment) to a straight segment (fig. 2.), so the replacement cylindrical gear is an evolutionary gear and, therefore, the conclusions resulting from the cylindrical gear study apply to this gear.

The calculation of resistance is performed in the middle section, using, for this purpose, the middle replacement gear.

Because in the conical wheels, in most cases (except those with constant height teeth), the radii of the characteristic circles, the thickness of the tooth and the module are variable along the cone generator, it is necessary to specify the position, on the width of the wheel, where they are established these elements, the usual ones being on the outside, in the middle section, and possibly in the interior.

For the straight conical tooth, a cylindrical gear with a straight tooth is obtained, and for those with a inclined and curved tooth, a cylindrical gear with a inclined tooth is obtained - the angle of inclination being defined by the position on the length of the tooth where the frontal cones are applied.

3. Gear replacement for conical gear with straight teeth

At these gears, the geometric elements are determined on the outer frontal cone, because at its level the (outer) module is standardized. To establish the connections between the elements of the replacement gear and the real conical gear, will be followed the figures 1 c and d.

- The diameters of the splitting circles of the corresponding replacement gear wheels

the outer frontal cones are:

$$d_{ev1} = \frac{d_{e1}}{cos\delta_1}$$

$$d_{ev2} = \frac{d_{e2}}{cos\delta_2}$$

$d_{e1,2}$ - the diameters of the splitting circles of the real wheels with conical teeth, on the outside

- The tooth numbers of the replacement gear wheels are based on the fact that module m_e is the same as the real conical gear and the cylindrical replacement gear. We can write the relationships:

$$d_{ev1} = m_e \cdot z_{v1} = \frac{d_{e1}}{cos\delta_1} = \frac{m_e \cdot z_1}{cos\delta_1}$$

$$d_{ev2} = m_e \cdot z_{v2} = \frac{d_{e2}}{cos\delta_2} = \frac{m_e \cdot z_2}{cos\delta_2}$$

results

$$z_{v1} = \frac{z_1}{cos\delta_1}$$

$$z_{v2} = \frac{z_2}{cos\delta_2}$$

where

z_1, z_2 – the tooth numbers of the conical gear wheels

z_{v1}, z_{v2} – the gear tooth numbers of the replacement cylindrical gear

- The distance between the axes of the replacement gear which is a gear zero (zero or moved), as previously discussed, is determined by the relation

$$a_{ev} = \frac{1}{2}(d_{ev1} + d_{ev2}) = \frac{m_e}{2}(z_{v1} + z_{v2})$$

- The replacement gear ratio of the gear is determined by the known relationship

$$u_v = \frac{z_{v2}}{z_{v1}} = \frac{z_2 \cdot cos\delta_1}{z_1 \cdot cos\delta_2} = u \cdot \frac{cos\delta_1}{cos\delta_2}$$

where *u* is the gear ratio of the conical gear. In the case of orthogonal conical gear ($\Sigma = \delta_1 + \delta_2 = 90°$)

$$tg\delta_2 = \frac{d_2}{d_1} = u$$

$$\frac{cos\delta_1}{cos\delta_2} = \frac{sin\delta_2}{sin\delta_1} = tg\delta_2 u$$

$$=> u_v = u^2$$

The minimum number of teeth of the replacement gear pinion, at which there is no interference and no profile displacement, according to the geometry of the cylindrical gears, is:

$$z_{v1min} = \frac{2 \cdot h_a^*}{sin^2\alpha}$$

where h_a^* represents the coefficient of the reference head of the tooth.

Considering the minimum number of teeth of the conical sprocket it is:

$$z_{1min} = z_{v1min} \cdot cos\delta_1$$

For the flat reference wheel of a conical gear with a straight gear, the following parameters are standardized:

- ❖ the module m_e,
- ❖ the coefficients h_a^* and c^*
- ❖ the pressure angle α,

defined on the reference rack.

In contrast to cylindrical gears, tangential profile shifts are used for conical gears. The use of tangential displacements ensures equalization of the bending strength of the teeth of the two wheels. The tangential displacement represents the dimension with which the full increases (decreases) and, respectively, decreases (increases) the hollow of the rack, on a certain line, parallel to the reference right. As a result, the tangential displacement represents the size with which the tooth of one wheel is thickened and, respectively, the tooth of the other wheel is thinned. For the calculation of the geometric elements of a conical gear wheel, these displacements are established in the front section.

Considering that the conical gears are zero or zero-displaced gears, it follows that for the coefficients of the tangential displacements of the profile, the relation is always respected:

$$x_{\tau 2} = -x_{\tau 1}$$

4. Determination of the geometric elements of tooth of the conical wheels with straight teeth

The conical gears with straight teeth are executed with proportional decreasing teeth and with constant radial play over the entire width of the tooth.

Geometrically, a conical gear is defined by:

- ★ reference profile,
- ★ tooth module (at the right teeth has the same value both in the font and in the normal plane $m = m_f = m_n$),
- ★ tooth numbers,
- ★ the angle between the axes,
- ★ profile movements.

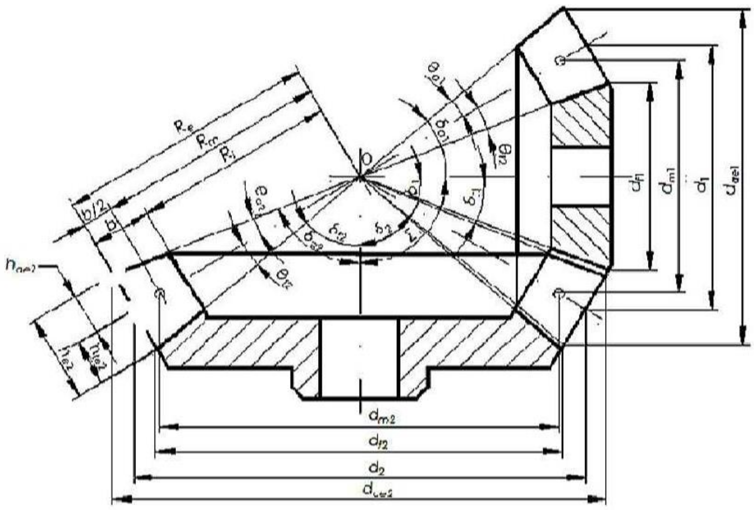

Figure 3 - The geometric elements of the tooth of the orthogonal gear with straight teeth

4.1. Measuring elements

1. **The tooth numbers**

$$z_1 = 13$$
$$z_2 = 40$$

where
z_1 - the number of teeth of wheel 1
z_2 - the number of teeth of wheel 2

2. **The diameter of the outer head circle**

$$d_{ae1} = d_1 + 2 \cdot h_{ae1} \cdot cos\delta_1 = m_e \cdot z_1 + 2 \cdot m_e(h_a^* + x_{hm1}) \cdot cos\delta_1 = 63{,}5\ mm$$
$$d_{ae2} = d_2 + 2 \cdot h_{ae2} \cdot cos\delta_2 = m_e \cdot z_2 + 2 \cdot m_e(h_a^* + x_{hm2}) \cdot cos\delta_2 = 161{,}2\ mm$$

where
$d_{1,2}$- the diameters of the outer partition
$h_{ae1,2}$- the outer height of the tooth's head
$\delta_{1,2}$- the semicircles of the splitting cones
m_e- the external module
h_a^* - the height coefficient of the reference head of the tooth
$x_{hm1,2}$ – coefficients of radial profile displacements

3. **The width of the tooth**

$$b = \psi_R \cdot R_e = 25\ mm$$

where
$\psi_R = 0{,}3$
$R_e = 83{,}34\ mm$

4.2. Reference profile elements

4. Normal pressure angle of division α, in degrees

$$\alpha = 20°$$

5. The height coefficient of the reference head of the tooth

$$h_a^* = 1$$

6. The coefficient of the reference

$$c_0^* = 0{,}2$$

7. The coefficient of the radius of connection at the foot of the tooth

$$\rho_f^* \leq 0{,}3$$

4.3. Calculated elements

8. Transmission report

$$i = \frac{n_1}{n_2} = \frac{z_2}{z_1} = \frac{sin\delta_2}{sin\delta_1} = \frac{d_{ae2}}{d_{ae1}} = \frac{40}{13} = 3{,}076$$

9. The ratio of tooth numbers

$$u = \frac{z_{mare}}{z_{mic}} \geq 1$$
$$u = i_{12} = 3{,}076$$

10. Rolling cone seams

$$\delta_1 = arctg\left(\frac{1}{i}\right) = arctg\frac{1}{3{,}076} = arctg0{,}325 = 18°$$
$$\delta_2 = \Sigma - \delta_1 = 90° - 18° = 72°$$

where

$\Sigma = 90°$ - angle between axes

11. The external module

$$m_e = \frac{d_{ae1}}{z_1 + 2 \cdot cos\delta_1} = 4{,}26\ mm$$

12. Standardized external module

Table 1 – Standard values for the nomal module [mm]

Modulul, mm (după STAS 822-82)	Mecanică fină	**0,05**; 0,055; **0,06**; 0,07; **0,08**; 0,09; 0,1; 0,11;**0,12**; 0,14; **0,15**; 0,18; 0,2; 0,22 ; **0,25**; 0,28;**0,3**; 0,35; **0,4**; 0,45; **0,5**; 0,55; **0,6**; 0,7; **0,8**; 0,9; 1,0.
	Mecanică generală și grea	**1**; 1,125; **1,25**; 1,375; **1,5**; 1,75; **2**; 2,25; **2,5**; 2,75; **3**; 3,5; **4**; 4,5; **5**; 5,5; **6**; 7; **8**; 9; **10**; 11; **12**; 14; **16**; 18; **20**; 22; **25**; 28; **32**; 36; **40**; 45; **50**; 55; **60**; 70; **80**; 90; **100**.

$$=> m_{e\,STAS} = 4\ mm$$

13. The coefficients of radial displacements of profile

$$x_{hm1} = \frac{d_{ae1} - (m_{e\,STAS} \cdot z_1 + 2 \cdot m_{e\,STAS} \cdot cos\delta_1)}{2 \cdot m_{e\,STAS} \cdot cos\delta_1} = \frac{63{,}6 - (4 \cdot 13 + 2 \cdot 4 \cdot cos18°)}{2 \cdot 4 \cdot cos18°}$$
$$= \frac{63{,}6 - (52 + 8 \cdot 0{,}951)}{8 \cdot 0{,}951} = \frac{63{,}6 - 59{,}608}{7{,}608} = \frac{3{,}992}{7{,}608} = 0{,}5247$$

Table 2 – Recommended values for normal profile radial displacement coefficients for orthogonal conical gears with straight teeth

Z_1	La rapoarte de transmitere i_{12}=										
	1	1,05	1,10	1,20	1,30	1,40	1,60	2,00	3,00	5,00	6,00
12	-	-	-	-	-	-	-	-	0,52	0,55	0,57
13	-	-	-	-	-	-	-	0,46	0,50	0,53	0,54
14	-	-	-	-	-	0,35	0,38	0,43	0,48	0,51	0,52
15	-	-	-	0,18	0,24	0,30	0,36	0,41	0,47	0,49	0,50
16	-	-	0,10	0,16	0,20	0,28	0,34	0,40	0,45	0,48	0,48
18	0	0,05	0,09	0,15	0,18	0,26	0,32	0,37	0,43	0,45	0,46
20	0	0,05	0,08	0,13	0,16	0,23	0,30	0,35	0,40	0,43	0,44
22	0	0,04	0,07	0,12	0,14	0,22	0,28	0,32	0,38	0,40	0,42
24	0	0,04	0,06	0,11	0,13	0,20	0,26	0,30	0,35	0,37	0,38
30	0	0,04	0,06	0,10	0,12	0,18	0,22	0,26	0,31	0,33	0,35
40	0	0,03	0,05	0,08	0,09	0,14	0,18	0,21	0,25	0,28	0,30

$$=> x_{hm1} = 0{,}50$$

For zero shifted gears, it is considered

$$=> x_{hm2} = -x_{hm1} = -0{,}50$$

14. Recalculation of head diameters on the outside

$$d_{ae1} = d_1 + 2 \cdot h_{ae1} \cdot cos\delta_1 = m_e \cdot z_1 + 2 \cdot m_e(h_a^* + x_{hm1}) \cdot cos\delta_1 = 63{,}41\ mm$$

The 0,09 mm difference between the calculated d_{ae1} and the measured d_{ae1} represents a measurement error.

$$d_{ae2} = d_2 + 2 \cdot h_{ae2} \cdot cos\delta_2 = m_e \cdot z_2 + 2 \cdot m_e(h_a^* + x_{hm2}) \cdot cos\delta_2 = 161{,}23\ mm$$

The -0,03 mm difference between the calculated d_{ae2} and the measured d_{ae2} represents a measurement error.

15. The coefficients of radial profile displacements

They are chosen from the table 3

Table 3 – Recommended values for the coefficients of tangential displacement to conical gears

β_m		La rapoarte de transmitere i_{12}=									
		1 .. 1,3	1,3 ... 1,6	1,6 ... 1,9	1,9 ... 2,25	2,25 ... 2,75	2,75 ... 3,5	3,5 ... 4,5	4,5 ... 6	6 ... 8	8 ... 10
0°– 15°		-	0,01	0,02	0,03	0,05	0,06	0,08	0,09	0,10	0,12

$$x_{sm1} = 0{,}06$$
$$x_{sm2} = -0{,}06$$

16. The diameters of the outer partition

$$d_1 = m_{e\,STAS} \cdot z_1 = 4 \cdot 13 = 52\ mm$$
$$d_2 = m_{e\,STAS} \cdot z_2 = 4 \cdot 40 = 160\ mm$$

17. The outer length of the splitter cone generator

$$R_e = \frac{d_1}{2 \cdot sin\delta_1} = \frac{52}{2 \cdot sin18°} = \frac{52}{2 \cdot 0{,}309} = \frac{52}{0{,}618} = 84{,}14\ mm$$

18. The longitudinal coefficient of the tooth width

$$\psi_R = \frac{b}{R_e} = \frac{25}{84{,}14} = 0{,}297$$

19. The median length of the splitting cone generator

$$R_m = R_e - \frac{b}{2} = 84{,}14 - \frac{25}{2} = 71{,}64\ mm$$

20. Height of the splitting head of the tooth on the two outer wheels

$$h_{ae1} = m_{e\,STAS}(h_a^* + x_{hm1}) = 4 \cdot (1 + 0{,}5) = 6\ mm$$
$$h_{ae2} = m_{e\,STAS}(h_a^* + x_{hm2}) = 4 \cdot (1 - 0{,}5) = 2\ mm$$

21. Height of the splitting foot of the tooth on the two wheels on the outside

$$h_{fe1} = m_{e\,STAS}(h_a^* + c_0^* - x_{hm1}) = 4 \cdot (1 + 0{,}2 - 0{,}5) = 2{,}80\ mm$$
$$h_{fe2} = m_{e\,STAS}(h_a^* + c_0^* - x_{hm2}) = 4 \cdot (1 + 0{,}2 + 0{,}5) = 6{,}80\ mm$$

22. The height of the tooth on the outside

$$h_{e1} = h_{e2} = m_{e\,STAS} \cdot (2 \cdot h_a^* + c_0^*) = 4 \cdot (2 \cdot 1 + 0.2) = 8{,}80\ mm$$

23. The diameters of the foot circles

$$d_{fe1} = d_1 - 2 \cdot h_{fe1} \cdot cos\delta_1 = 52 - 2 \cdot 2{,}80 \cdot cos18° = 52 - 5{,}60 \cdot 0{,}951 = 52 - 5{,}326 = 46{,}67\ mm$$
$$d_{fe2} = d_2 - 2 \cdot h_{fe2} \cdot cos\delta_2 = 160 - 2 \cdot 6{,}80 \cdot cos72° = 160 - 13{,}60 \cdot 0{,}309 = 160 - 4{,}202 = 155{,}80\ mm$$

24. The angle of the head of the tooth

For descending play

$$\theta_{a1} = arctg\left(\frac{h_{ae1}}{R_e}\right) = arctg\frac{6}{84,14} = arctg0,0713 = 4,08°$$
$$\theta_{a2} = arctg\left(\frac{h_{ae2}}{R_e}\right) = arctg\frac{2}{84,14} = arctg0,024 = 1,36°$$

25. Angle at the foot of the tooth

For descending play

$$\theta_{f1} = arctg\left(\frac{h_{fe1}}{R_e}\right) = arctg\frac{2,80}{84,14} = arctg0,0332 = 1,90°$$
$$\theta_{f2} = arctg\left(\frac{h_{fe2}}{R_e}\right) = arctg\frac{6,80}{84,14} = arctg0,081 = 4,62°$$

26. The head cone semicircle

$$\delta_{a1} = \delta_1 + \theta_{a1} = 18 + 4,08 = 22,08°$$
$$\delta_{a2} = \delta_2 + \theta_{a2} = 72 + 1,36 = 73,36°$$

27. The half cone of the foot cones

$$\delta_{f1} = \delta_1 - \theta_{f1} = 18 - 1,90 = 16,10°$$
$$\delta_{f2} = \delta_2 - \theta_{f2} = 72 - 4,62 = 67,38°$$

5. Determination of the geometric elements of the replacement cylindrical gear (virtual)

► *The tooth numbers of the wheels of the replacement cylindrical gear*

$$z_{e1} = \frac{z_1}{cos\delta_1} = \frac{13}{cos18°} = \frac{13}{0,951} = 13,66$$
$$z_{e2} = \frac{z_2}{cos\delta_2} = \frac{40}{cos72°} = \frac{40}{0,309} = 129,44$$

► *The transmission ratio of the replacement cylindrical gear*

$$i_{12} = \frac{z_{e2}}{z_{e1}} = \frac{129,44}{13,66} = 9,47$$

► *Replacement diameters of cylindrical replacement wheels*

$$d_{v1} = \frac{d_1}{cos\delta_1} \cdot \frac{R_m}{R_e} = \frac{52}{cos18°} \cdot \frac{71,64}{84,14} = \frac{52}{0,951} \cdot 0,851 = 54,68 \cdot 0,851 = 46,53\ mm$$
$$d_{v2} = \frac{d_2}{cos\delta_2} \cdot \frac{R_m}{R_e} = \frac{160}{cos72°} \cdot \frac{71,64}{84,14} = \frac{160}{0,309} \cdot 0,851 = 517,80 \cdot 0,851 = 440,65\ mm$$

► *The head diameters of the replacement cylindrical wheels*

$$d_{va1} = d_{v1} + 2 \cdot h_{am1} = 46{,}53 + 2 \cdot 5{,}11 = 46{,}53 + 10{,}22 = 56{,}75\, mm$$

where

$$h_{am1} = h_{ae1} - \frac{b}{2} \cdot tg\theta_1 = 6 - \frac{25}{2} \cdot tg4{,}08° = 6 - 12{,}5 \cdot 0{,}071 = 6 - 0{,}8875 = 5{,}11\, mm$$

$$d_{va2} = d_{v2} + 2 \cdot h_{am2} = 440{,}65 + 2 \cdot 1{,}70 = 440{,}65 + 3{,}40 = 444{,}05\, mm$$

where

$$h_{am2} = h_{ae2} - \frac{b}{2} \cdot tg\theta_2 = 2 - \frac{25}{2} \cdot tg1{,}36° = 2 - 12{,}5 \cdot 0{,}024 = 2 - 0{,}3 = 1{,}70\, mm$$

► *The diameters of the base circles of the replacement cylindrical wheels*

$$d_{vb1} = d_{v1} \cdot cos\alpha = 46{,}53 \cdot cos20° = 46{,}53 \cdot 0{,}939 = 43{,}72\, mm$$
$$d_{vb2} = d_{v2} \cdot cos\alpha = 440{,}65 \cdot cos20° = 440{,}65 \cdot 0{,}939 = 413{,}77\, mm$$

► *The distance between the axles of the replacement gear*

$$a_w = \frac{d_{v1} + d_{v2}}{2} = \frac{46{,}53 + 440{,}65}{2} = \frac{487{,}18}{2} = 243{,}59\, mm$$

► *The degree of coverage of the replacement gear*

$$\varepsilon_{v\alpha} = \frac{\sqrt{d_{va1}^2 - d_{vb1}^2} + \sqrt{d_{va2}^2 - d_{vb2}^2} - 2 \cdot a_w \cdot sin\alpha}{2\pi \cdot m_n \cdot cos\alpha}$$
$$= \frac{\sqrt{56{,}75^2 - 43{,}72^2} + \sqrt{444{,}05^2 - 413{,}77^2} - 2 \cdot 243{,}59 \cdot sin20°}{2 \cdot 3{,}14 \cdot 3{,}40 \cdot cos20°}$$
$$= \frac{\sqrt{3220{,}56 - 1911{,}44} + \sqrt{197180{,}40 - 171205{,}61} - 487{,}18 \cdot 0{,}342}{21{,}352 \cdot 0{,}939}$$
$$= \frac{\sqrt{1309{,}12} + \sqrt{25974{,}79} - 166{,}62}{20{,}05} = \frac{36{,}18 + 161{,}17 - 166{,}62}{20{,}05} = \frac{30{,}73}{20{,}05}$$
$$= 1{,}53$$

where

$$m_m = m_{e\,STAS} \cdot \frac{R_m}{R_e} = 4 \cdot \frac{71{,}64}{84{,}14} = 4 \cdot 0{,}851 = 3{,}40$$

6. Conclusions

The calculation of the resistance of the conical gears, in contact and bending, is performed by accepting the same calculation assumptions as in the calculation of the cylindrical gears with a straight tooth. virtual gear, and to find equivalence relations between the two gears - real and virtual.

The geometry of the cylindrical gear has been studied in a frontal plane section, and that of the conical gear can be performed in a front spherical section (front spherical gear), with the center at the point of intersection of the gear wheels axes.

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