CZECH TECHNICAL UNIVERSITY IN PRAGUE

FACULTY
OF MECHANICAL
ENGINEERING

Preliminary Calculations Report

Design

2022

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Given parameters:

Task type : C

Power on the drum, $P_b = 2.2$ kW

Tangential velocity on the drum, $v_b = 0.7$ m/s

Drum diameter, $D_b = 450$ mm $= 0.45$ m

(Reduced mass moment of inertia of driven belt conveyor, I_r)
Mass moment of inertia of powered conveyor, $I_{rb} = 54$ kgm^2

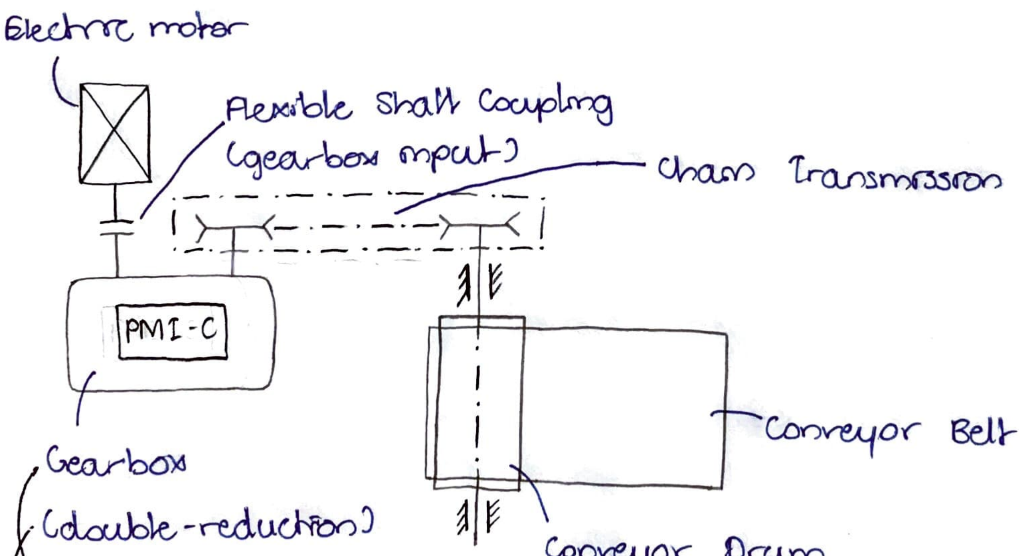
Bearing lifetime, $L_h = 20000$ hours

Type C:

double-reduction

Three-phase asynchronous motor

Electric motor → Flexible shaft coupling → Gearbox → Roller Chain transmission → Conveyor drum

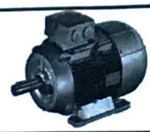


made of grey cast iron

good for large reduction of speed

necessary for slow moving machinery

Data sheet for three-phase Squirrel-Cage-Motors SIMOTICS



Motor type : 1CV2130C SIMOTICS SD - 132 S - IM B3 - 6p

Client order no.	Item-No.	Offer no.
Order no.	Consignment no.	Project
Remarks		

Electrical data

Safe Area

U [V]	Δ / Y	f [Hz]	P [kW]	P [hp]	I [A]	n [1/min]	M [Nm]	η [3] 4/4	η 3/4	η 2/4	cosφ [3] 4/4	cosφ 3/4	cosφ 2/4	I_A/I_N I_l/I_N	M_A/M_N T_l/T_N	M_K/M_N T_B/T_N	IE-CL
DOL duty (S1) - 155(F) to 130(B)																	
230	Δ	50	3.00	-/-	12.60	970	29.5	83.3	83.4	81.0	0.72	0.63	0.50	5.0	1.6	2.5	IE2
400	Y	50	3.00	-/-	7.20	970	29.5	83.3	83.4	81.0	0.72	0.63	0.50	5.0	1.6	2.5	IE2
460	Y	60	3.45	-/-	6.90	1170	28.0	87.5	87.9	86.2	0.72	0.64	0.51	5.2	1.6	2.6	IE2
460	Y	60	3.00	-/-	6.20	1175	24.5	87.5	87.2	84.7	0.69	0.60	0.47	6.0	1.8	2.9	IE2

IM B3 / IM 1001 | FS 132 S | IP65 | IEC/EN 60034 | IEC, DIN, ISO, VDE, EN

Environmental conditions : -20 °C - +40 °C / 1000 m | Locked rotor time (hot / cold) : 27.1 s | 41.5 s

Mechanical data

Sound level (SPL / SWL) at 50Hz|60Hz	63 / 75 dB(A) [2]	67 / 79 dB(A) [2]	Vibration severity grade		A
Moment of inertia	0.0240 kg m²		Thermal class		F
Bearing DE | NDE	6308 2Z C3	6308 2Z C3	Duty type		S1
bearing lifetime			Direction of rotation		bidirectional
L_{10mh} $F_{Rad\ min}$ for coupling operation 50|60Hz [1]	40000 h	32000 h	Frame material		cast iron
Lubricants	Unirex N3		Net weight of the motor (IM B3)		56 kg
Regreasing device	No		Coating (paint finish)		Special paint finish C3
Grease nipple	-/-		Color, paint shade		RAL7030
Type of bearing	Preloaded bearing DE		Motor protection		(B) 3 PTC thermistors - for tripping (standard) (2 terminals)
Condensate drainage holes	Yes (standard)		Method of cooling		IC411 - self ventilated, surface cooled
External earthing terminal	No				

Terminal box

Terminal box position	top	Max. cross-sectional area	6 mm²
Material of terminal box	cast iron	Cable diameter from ... to ...	11 mm - 21 mm
Type of terminal box	TB1 H01	Cable entry	2xM32x1,5-1xM16x1,5
Contact screw thread	M4	Cable gland	3 plugs

Notes:

I_A/I_N = locked rotor current / current nominal
M_A/M_N = locked rotor torque / torque nominal
M_K/M_N = break down torque / nominal torque

1) L10mh according to DIN ISO 281 10/2010
2) at rated power / at full load

3) Value is valid only for DOL operation with motor design IC411

responsible dep. DI MC LVM	technical reference	created by DT Configurator	approved by	*Technical data are subject to change! There may be discrepancies between calculated and rating plate values.*	Link documents
SIEMENS	document type datasheet			document status released	
	title 1LE1601-1CC02-2AB4-Z H01+H20			document number	
© Siemens AG 2022				rev. 01 | creation date 2022-01-13	language en | Page 1/2

→ RPM of driving drum, $n_b = \frac{60 \cdot V_b}{\pi \cdot D_b} = \frac{60 \cdot 0.7}{\pi \cdot 0.45} = 29.71\ min^{-1}$ / rpm
(revolutions per minute

→ Power of motor, $P_m' = \frac{P_b}{\eta_c}$

Total mechanical efficiency, $\eta_c = \eta_{12} \cdot \eta_{34} \cdot \eta_{at}$

Efficiency of pair of mating gears (including bearings), η_{12}, η_{34}

↳ helical mating gears: $\eta_{12} = \eta_{34} = 0.98$

Additional transmission (belt or chain transmission), η_{at}

↳ Type C ⇒ efficiency of chain transmission, $\eta_{chain} = 0.94$

$\Rightarrow \eta_c = \eta_{12} \cdot \eta_{34} \cdot \eta_{chain} = 0.98 \cdot 0.98 \cdot 0.94 = 0.90$

$P_m' = \frac{P_b}{\eta_c} = \frac{2200}{0.90} = 2444.44\ W = 2.44\ kW$

→ Total speed ratio, $r_c = 15 - 45$ for type C

$$r_c = \frac{n_m}{n_b}$$

Number of asynchronous revolutions [min^{-1}], $n_m \leq n_b \cdot r_{cmax}$

$n_m \leq 29.71 \cdot 45$

$n_m \leq 1336.95\ min^{-1}$

⇒ Chosen motor: 1LE1601 - 1CC02 - 2AB4 - Z

↳ Power of electric motor, $P_m = 3\ kW$

↳ Revolution of electric motor, $n_m = 970\ min^{-1}$

↳ Dimension of shaft end, $D\ (d_m) = 38\ mm$

Czech electrical distribution values: 230 V × 50 Hz

→ Distribution of total speed ratio, i_c:

Type C: $i_p = \dfrac{i_c}{i_{chain}}$

$i_{chain} \approx 1.5$

speed ratio, $i_p = u_p$

transference number (gear ratio), u_p

$i_c = \dfrac{n_m}{n_b} = \dfrac{970}{29.71} = 32.65$

$i_p = \dfrac{i_c}{i_{chain}} = \dfrac{32.65}{1.5} = 21.77 = u_p$

→ Partial transference numbers, u_{12}, u_{34}

↳ maximum partial transference number, $u = 5$

recommended conditions: $u_{12} > u_{34}$ (shouldn't be an integer)

⇒ I choose: $\cancel{u_{34} = 4.5}$ ~~u_{12} = [illegible]~~ $u_{12} = 4.7$

$\cancel{u_{12} = 4.9}$ $\cancel{u_{34} = 4.6}$ $u_{34} = 4.6$

→ Determination of number of teeth:

number of teeth of pinion, z_1, z_3 — the smaller gear of the 2 mating gears

recommendation: $z_1, z_3 = 17, 18, 19$

$z_1 > z_3$ ⇒ I choose $z_1 = 19$ and $z_3 = 17$

$u_{12} = \dfrac{z_2}{z_1}$, $u_{34} = \dfrac{z_4}{z_3}$

$z_2' = z_1 \cdot u_{12} = 19 \cdot 4.7 = 89.3$ ⇒ I choose $z_2 = 90$

$z_4' = z_3 \cdot u_{34} = 17 \cdot 4.6 = 78.2$ ⇒ I choose $z_4 = 79$

→ New transference number, $u_p = u_{14} = \frac{z_2}{z_1} \cdot \frac{z_4}{z_3} = \frac{90}{19} \cdot \frac{79}{17} = 22.01$

→ Torques:

Torque of electric motor, $M_{km} = \frac{P_m}{\omega_m}$, $\omega_m = \frac{2 \cdot \pi \cdot n_m}{60}$

$$\omega_m = \frac{2 \cdot \pi \cdot 970}{60} = 101.58 \ rad \cdot s^{-1}$$

$$M_{km} = \frac{3000}{101.58} = 29.53 \ Nm = 29\,530 \ Nmm$$

→ Driving mechanism of Type C:

$$M_{kI} = M_{km} = 29.53 \ Nm = 29\,530 \ Nmm$$

$$M_{kII} = M_{kI} \cdot u_{12} \cdot \eta_{12} = 29.53 \cdot 4.7 \cdot 0.98 = 136.02 \ Nm = 136\,020 \ Nmm$$

$$M_{kIII} = M_{kII} \cdot u_{34} \cdot \eta_{34} = 136.02 \cdot 4.6 \cdot 0.98 = 613.18 \ Nm = 613\,180 \ Nmm$$

$$M_{kb} = M_{kIII} \cdot i_{chain} \cdot \eta_{chain} = 613.18 \cdot 1.5 \cdot 0.94 = 864.58 \ Nm = 864\,580 \ Nmm$$

M_{kI} → torque on input shaft

M_{kII} → torque on countershaft

M_{kIII} → torque on output shaft

M_{kb} → torque on conveyor (drum)

(minimum)

→ Diameter of shafts:

Torsion, $\tau = \frac{M_k}{W_k} = \frac{16 \cdot M_k}{\pi \cdot d^3} \leq \tau_D$

$$\Rightarrow d \geq \sqrt[3]{\frac{16 \cdot M_k}{\pi \cdot \tau_D}}$$

Chosen allowable torsional stress, τ_D:

- input shaft I: $\tau_{DI} = 25\ Nmm^{-2}$

- countershaft II: $\tau_{DII} = 35\ Nmm^{-2}$

- output shaft III: $\tau_{DIII} = 50\ Nmm^{-2}$

Diameter of input shaft, d_I:

$$d_{I\,mm} \geq \sqrt[3]{\frac{16 \cdot M_{kI}}{\pi \cdot \tau_{DI}}} = \sqrt[3]{\frac{16 \cdot 29\,530}{\pi \cdot 25}} = 18.19\ mm \Rightarrow \text{I choose } d_I = 20\ mm$$

Diameter of countershaft, d_{II}:

$$d_{II\,mm} \geq \sqrt[3]{\frac{16 \cdot M_{kII}}{\pi \cdot \tau_{DII}}} = \sqrt[3]{\frac{16 \cdot 136\,020}{\pi \cdot 35}} = 27.05\ mm \Rightarrow \text{I choose } d_{II} = 30\ mm$$

Diameter of output shaft, d_{III}:

$$d_{III\,mm} \geq \sqrt[3]{\frac{16 \cdot M_{kIII}}{\pi \cdot \tau_{DIII}}} = \sqrt[3]{\frac{16 \cdot 613\,180}{\pi \cdot 50}} = 39.68\ mm \Rightarrow \text{I choose } d_{III} = 40\ mm$$

→ Flexible shaft coupling:

Torque of coupling: $M_{ks} = k \cdot M_{km}$

Torque on shaft of electric motor [Nm], M_{km}

Operation factor (standardized)

↳ for this type of coupling and application, $k = 1.5 - 1.7$

(I choose $k = 1.6$)

$\Rightarrow M_{ks} = 1.6 \cdot 29.53 = 47.25\ Nm$

→ Materials of gears:

material X   material Y

I choose to use different materials for pinion and gear wheel

↳ X has to be harder than Y

Chosen material X: 14 NiCr 14 (16 420)

Thermal treatment: carburizing and hardening

Hardness: in tooth core, $\sigma_{HV} = 300$ HV

on tooth face, $V_{HV} = 650 - 720$ HV

↳ > 350 HV

Fatigue limit (base values):

in bending, $\sigma^{o}_{Flimb} = 700 \ Nmm^{-2}$

in contact, $\sigma^{o}_{Hlimb} = 1270 \ Nmm^{-2}$

Chosen material Y: 41 Cr 4 (14 140)

Thermal treatment: tooth face hardening

Hardness: in tooth core: $\sigma_{HV} = 250$ HV

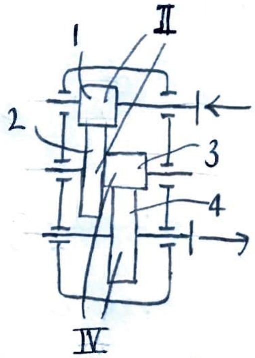
on tooth surface: $V_{HV} = 600 - 675$ HV

Fatigue limit (base values):

in bending, $\sigma^{o}_{Flimb} = 450 \ Nmm^{-2}$

in contact, $\sigma^{o}_{Hlimb} = 1140 \ Nmm^{-2}$

→ Design of Helical Gears (normal module):



$$m_n' \geq f_p \cdot \sqrt[3]{\frac{K_F \cdot M_{kn}}{\left(\frac{b_{wF}}{m_n}\right) \cdot z_n \cdot \sigma_{FP}}}$$

m_n → calculated module (must be rounded off to standardized value)

f_p → coefficient of the hardened gears ($f_p = 18$)

M_{kn} → torque on shaft of pinion [Nm]

K_F → coefficient of additional working loads

$$K_F \approx K_A \cdot K_{F\beta} \approx K_A \cdot K_{H\beta}$$

K_A → coefficient of dynamical external force ($K_A = 1.0$ for the conveyor belt)

$K_{F\beta}$ ($K_{H\beta}$) → coefficient of load distribution along tooth width, depends on the ~~reg~~rigidity of shaft (supports of shaft)

$$\psi_d = \frac{b_{wH}}{d_1}$$

$\frac{b_{wF}}{m_n}$ → relative tooth face width

depends on if it is two-sided (symmetrical / asymmetrical) supported mating gears (I will use symmetrical)

z_n → number of teeth of pinion

σ_{FP} → allowable bending stress [$N mm^{-2}$]

preliminary choice of this parameter: $\sigma_{FP} \approx 0.6 \cdot \sigma_{Flimb1}$

σ_{Flimb1} → base bending fatigue stress of pinion

$$\sigma_{Flimb1} \approx \sigma^0_{Flimb}$$

σ^0_{Flimb} → bending fatigue strength corresponding to the base number of loading cycles, N_{Flimb}

Calculation for normal module of first mating gears: (1, 2)

Since both of the mating gears 1 and 2 are surface hardened:

$\hookrightarrow \psi_d = \left(\frac{b_{wH}}{d_1}\right)_{max} = 1.1$

$\hookrightarrow u = 2 \Rightarrow$ I choose $\frac{b_{wF}}{m_n} = 19$

From the graphs of $V_{HV} > 350$ HV:

$V_{HV} \rightarrow$ surface hardness of the pinion ~~or~~ or the mating gear

Case II: $K_{H\beta} = 1.42$, $K_{F\beta} = 1.65$ } average ≈ 1.5

$\Rightarrow K_{F_1} \approx 1 \cdot 1.5 = 1.5$

$\sigma_{FP_1} \approx 0.6 \cdot \sigma_{Flimb1} \approx 0.6 \cdot \sigma^{o}_{Flimb} \approx 0.6 \cdot 700 = 420 \text{ Nmm}^{-2}$

$$m_{12}' \geq f_p \cdot \sqrt[3]{\frac{K_{F_1} \cdot M_{t1}}{\left(\frac{b_{wF}}{m_n}\right) \cdot z_1 \cdot \sigma_{FP_1}}} = 18 \cdot \sqrt[3]{\frac{1.5 \cdot 29.53}{19 \cdot 19 \cdot 420}} = 1.19 \text{ mm}$$

$\Rightarrow$ I choose $m_{12} = 2$ mm

Calculation for normal module of second mating gears: (3, 4)

Since both mating gears 3 and 4 are surfaced hardened:

$\hookrightarrow \psi_d = \left(\frac{b_{wH}}{d_1}\right)_{max} = 1.1$

$\hookrightarrow u = 2 \Rightarrow$ I choose $\frac{b_{wF}}{m_n} = 19$

From the graphs of $V_{HV} > 350$ HV:

case IV: $K_{H\beta} = 1.21$, $K_{F\beta} = 1.35$ } average ≈ 1.3

$\Rightarrow K_{F_3} \approx 1 \cdot 1.3 = 1.3$

$\sigma_{FP_3} \approx 0.6 \cdot \sigma^{o}_{Flimb} = 0.6 \cdot 700 = 420 \ Nmm^{-2}$

$$m'_{34} \geq k_p \cdot \sqrt[3]{\frac{K_{F_3} \cdot M_{k\,II}}{\left(\frac{b_{wF}}{m_n}\right) \cdot z_3 \cdot \sigma_{FP_3}}} = 18 \cdot \sqrt[3]{\frac{1.3 \cdot 136.02}{19 \cdot 17 \cdot 420}} = 1.97 \text{ mm}$$

$\Rightarrow$ I choose $m_{34} = 3$ mm

→ Design of Helical Gears (dimensions):

Pitch diameter, $d_n = \frac{m_n \cdot z_n}{\cos \beta_n}$ [mm]

z_n → number of teeth

β → helix angle on pitch cylinder

↳ This value is chosen according to the standards (ČSN 01 4610)

It is recommended: $\beta_{12} > \beta_{34}$

↳ I choose $\beta_{12} = 10°$ and $\beta_{34} = 8°$

$$d_1 = \frac{m_{12} \cdot z_1}{\cos \beta_{12}} = \frac{2 \cdot 19}{\cos 10} = 38.59 \text{ mm}$$

$$d_2 = \frac{m_{12} \cdot z_2}{\cos \beta_{12}} = \frac{2 \cdot 90}{\cos 10} = 182.78 \text{ mm}$$

$$d_3 = \frac{m_{34} \cdot z_3}{\cos \beta_{34}} = \frac{3 \cdot 17}{\cos 8} = 51.50 \text{ mm}$$

$$d_4 = \frac{m_{34} \cdot z_4}{\cos \beta_{34}} = \frac{3 \cdot 79}{\cos 8} = 239.33 \text{ mm}$$

Tooth face width of gears, $b = \left(\frac{b_{WF}}{m_n}\right) \cdot m_n$ [mm]

$b_1 = \left(\frac{b_{WF}}{m_n}\right) \cdot (> m_{12}) = 19 \cdot 2.5 = 47.5$ mm $\Rightarrow$ I choose $b_1 = 48$ mm

$b_2 = \left(\frac{b_{WF}}{m_n}\right) \cdot m_{12} = 19 \cdot 2 = 38$ mm

$b_3 = \left(\frac{b_{WF}}{m_n}\right) \cdot (> m_{34}) = 19 \cdot 4 = 76$ mm

$b_4 = \left(\frac{b_{WF}}{m_n}\right) \cdot m_{34} = 19 \cdot 3 = 57$ mm $\Rightarrow$ I choose $b_4 = 58$ mm

→ Adaptation of Center distance:

Calculated center distance of helical gears, $a_{12} = \frac{m_{12} \cdot (z_1 + z_2)}{2 \cdot \cos \beta_{12}}$

↳ Practical application: round off to an integer

$$a_{12} = \frac{m_{12} \cdot (z_1 + z_2)}{2 \cdot \cos \beta_{12}} = \frac{2 \cdot (19 + 90)}{2 \cdot \cos 10} = 110.68 \text{ mm} \Rightarrow \text{I choose } \cancel{111 \text{ mm}}$$

$a_{12} = 111$ mm

$$a_{34} = \frac{m_{34} \cdot (z_3 + z_4)}{2 \cdot \cos \beta_{34}} = \frac{3 \cdot (17 + 79)}{2 \cdot \cos 8} = 145.42 \text{ mm} \Rightarrow \text{I choose}$$

$a_{34} = 146$ mm

$$a_{tw} = a_t \frac{\cos \alpha_t}{\cos \alpha_{tw}}, \quad \tan \alpha_t = \frac{\tan \alpha_n}{\cos \beta}$$

a_{tw} → chosen rolling center distance [mm]

a_t → calculated center distance of helical gears [mm]

α_t → transverse pressure angle (corresponding to the uncorrected center distance, a_t)

α_n → normal profile angle (profile angle of the basic rack tooth profile), $\alpha_n = 20°$

β → helix angle

$$\alpha_t = \tan^{-1}\left(\frac{\tan \alpha_n}{\cos \beta}\right)$$

$$\alpha_{tw} = \cos^{-1}\left(\frac{a_t \cdot \cos \alpha_t}{a_{tw}}\right)$$

$$\alpha_{t12} = \tan^{-1}\left(\frac{\tan \alpha_n}{\cos \beta_{12}}\right) = \tan^{-1}\left(\frac{\tan 20}{\cos 10}\right) = 20.28°$$

$$\alpha_{tw_{12}} = \cos^{-1}\left(\frac{a_{t12} \cdot \cos \alpha_{t12}}{a_{tw_{12}}}\right) = \cos^{-1}\left(\frac{\cancel{111}\ 110.68 \cdot \cos(20.28)}{\cancel{110.68}\ 111}\right) = \cancel{19.83°}\ 20.72°$$

$$\alpha_{t34} = \tan^{-1}\left(\frac{\tan \alpha_n}{\cos \beta_{12}}\right) = \tan^{-1}\left(\frac{\tan 20}{\cos 8}\right) = 20.18°$$

$$\alpha_{tw34} = \cos^{-1}\left(\frac{a_{t34} \cdot \cos \alpha_{t34}}{a_{tw34}}\right) = \cos^{-1}\left(\frac{145.42 \cdot \cos(20.18)}{146}\right) = 20.79°$$

→ The sum of the basic rack tooth profile displacement, $x_\Sigma = x_1 + x_2$

$$x_\Sigma = x_1 + x_2 = \frac{z_1 + z_2}{2 \tan \alpha_n} \cdot (\text{inv}\ \alpha_{tw} - \text{inv}\ \alpha_t)$$

involute function, $\text{inv}\ \alpha = \tan \alpha - \frac{\pi}{180} \cdot \alpha$

For $x_\Sigma = x_1 + x_2$:

$$\text{inv}\ \alpha_{tw_{12}} = \tan \alpha_{tw_{12}} - \frac{\pi}{180} \cdot \alpha_{tw_{12}} = \tan(20.72) - \frac{\pi}{180} \cdot 20.72 =$$

$$= \cancel{0.13679}\ 0.016645$$

$$\text{inv}\ \alpha_{t12} = \tan \alpha_{t12} - \frac{\pi}{180} \cdot \alpha_{t12} = \tan(20.28) - \frac{\pi}{180} \cdot 20.28 =$$

$$= \cancel{0.13079}\ 0.01556$$

$$x_\Sigma = x_1 + x_2 = \frac{z_1 + z_2}{2 \cdot \tan \alpha_n} \cdot (\text{inv}\ \alpha_{tw_{12}} - \text{inv}\ \alpha_{t12}) =$$

$$= \frac{19 + 90}{2 \cdot \tan 20} \cdot (\cancel{0.13679}\ 0.01664 - \cancel{0.13079}\ 0.01556) = \cancel{0.91190}\ 0.16172 < 0.3$$

$\Rightarrow x_1 = \cancel{x_\Sigma}\, x_\Sigma$ → only pinion is corrected $\Rightarrow x_1 = 0.16172$

$x_2 = 0$ → wheel is without correction

For $x_\Sigma = x_3 + x_4$:

$$inv\, \alpha_{tw34} = \tan \alpha_{tw34} - \frac{\pi}{180} \cdot \alpha_{tw34} = \tan(20.79) - \frac{\pi}{180} \cdot 20.79 =$$

$$= 0.01681$$

$$inv\, \alpha_{t34} = \tan \alpha_{t34} - \frac{\pi}{180} \cdot \alpha_{t34} = \tan(20.18) - \frac{\pi}{180} \cdot 20.18 =$$

$$= 0.01532$$

$$x_\Sigma = x_3 + x_4 = \frac{z_3 + z_4}{2 \cdot \tan \alpha_n} \cdot (inv\, \alpha_{tw34} - inv\, \alpha_{t34}) =$$

$$= \frac{17 + 79}{2 \cdot \tan 20} \cdot (0.01681 - 0.01532) = 0.19650 < 0.3$$

$\Rightarrow x_3 = x_\Sigma = 0.19650$

$x_4 = 0$

→ layout drawing:

It is drawn ~~o~~ using a scale of 1:2