Faculty of Power and Aeronautical Engineering

Institute of Aeronautical Technology and Applied Mechanics Department of Fundamentals of Construction

Fundamentals of Mechanical Engineering VI

Fan Drive Transmission System

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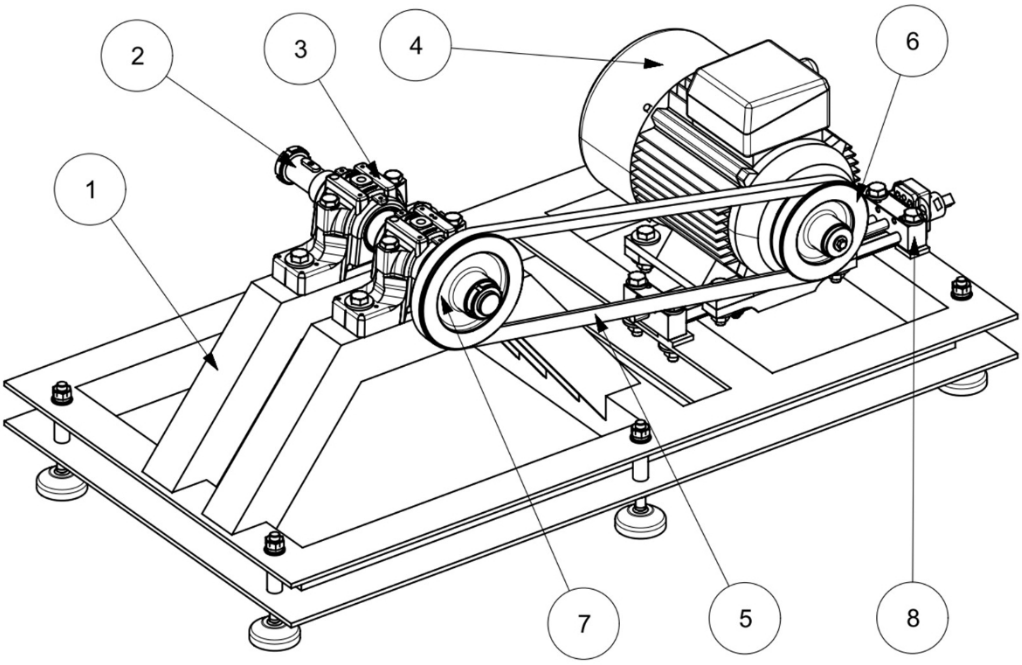
1 Introduction

The purpose of this work is to design a transmission system for a fan, driven by an electric motor, from which torque is transferred to the shaft by means of a belt transmission, equipped with a belt tensioning system. All components of the device are mounted on a welded frame. The shaft is bearing between the fan and the passive pulley.

1.1 Drawing illustrative

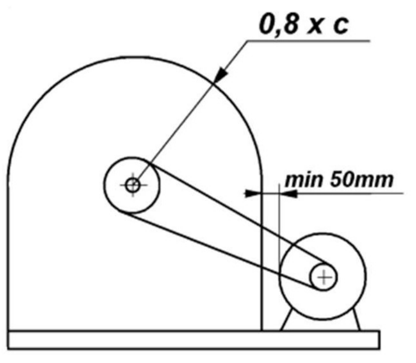


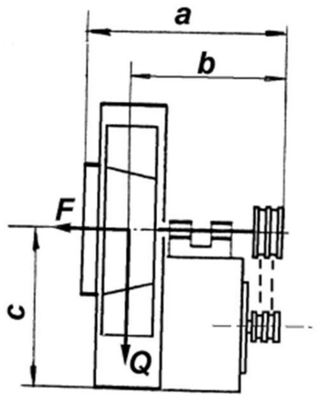
Main components:

1. Frame
2. Drive shaft
3. Bearing housing with bearing
4. Electric motor
5. V-belt
6. Small pulley
7. Large pulley
8. Linear table

1.2 Assumptions design

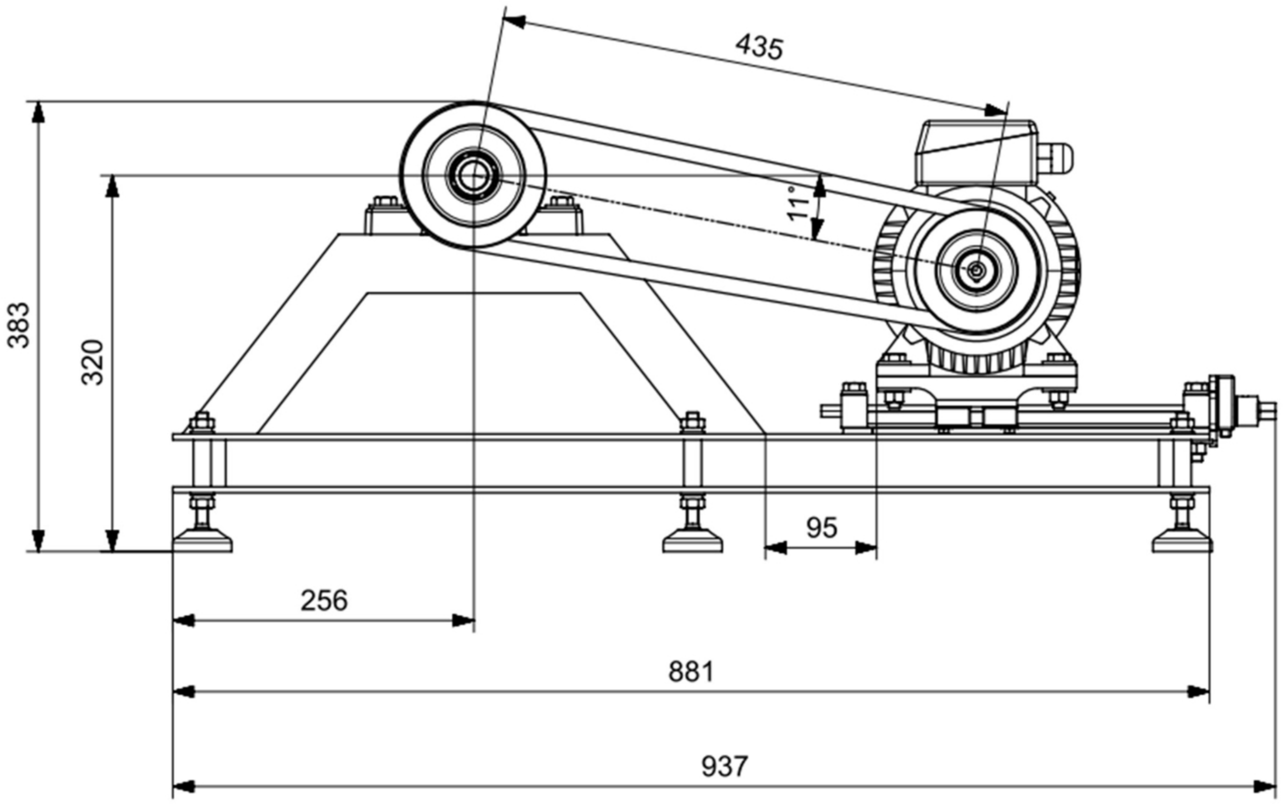
Typ napędu 2:

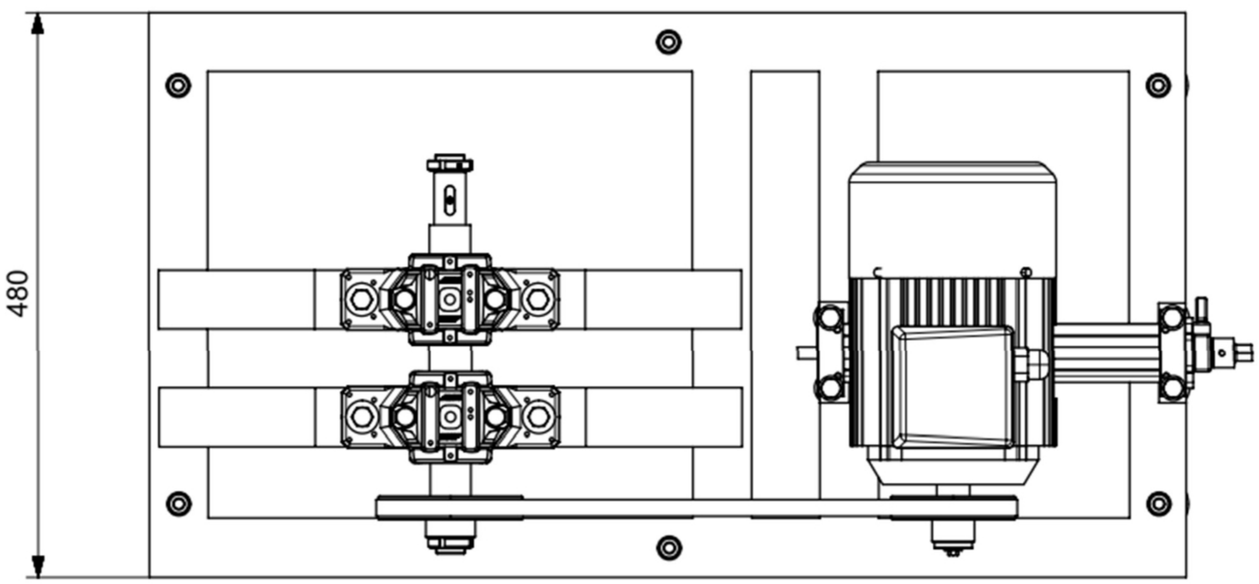




- two shaft supports are located between the pulley and the rotor
- power P= 1.2kW
- rotational speed n= 1200 rpm
- axial force F= *310N*
- rotor weight Q= *126N*
- hourly life L_{10h}= *130000h*
- width of the device along the shaft axis a= *505mm*
- distance from the center of gravity of the rotor to the opposite end of the shaft b=*440mm* *(later changed to b= 383)*
- height of the shaft axis from the ground c= *320mm*
- spacing between motor and rotor housing - min *50mm*

1.2.1 Gabaryty





1.3 Materials used in device

- shaft, disc protection disc on motor shaft: **C35** (R_e= 270MPa, R_m= 520MPa, Z_{sj}= 244MPa, Z_{gj}= 364MPa, Z_{so}= 150MPa, Z_{go}= 250MPa)

- Frame, engine mounting table: **S235JR** (R_e= 235MPa, R_m= 410MPa)

- pulleys: **St3SX** (R_m= 360MPa)

- keys: **C45** (R_e= 360MPa, R_m= 610MPa, Z_{sj}= 340MPa, Z_{gj}= 480MPa, Z_{so}= 170MPa, Z_{go}= 280MPa)

- Bushings for pulley, rotor: **AISI 303 (**R_e= 190MPa**,** R_m= 500MPa**)**

- shims under the guide: **SBR rubber**

The selection of materials was based on an analysis of similar components available on the market. All other parts are standardized and made by third-party manufacturers.

1.4 Factor safety

The designed device is to operate most of the day, under moderate loads. The adopted safety factor is x=4.

2 Selection of motor

$$\frac{P_{wy}}{5_{upn}} = \frac{1{,}2\ kW}{0{,}945} = 1{,}269\ kW$$

Assuming $5_{upn} = 5_{p.belt} \bullet 5_{bearings} = 0.95 \bullet 0.995 = 0.94525$ [1]. **SH 90 L-4** motor

with parameters was selected:

- Rated power $P_{se} = 1.5\ kW$
- Speed $n = 1410\ rpm$

The motor meets the design requirements by a margin. In addition, reduce the speed and power of the motor.

3 Transmission belt transmission

The designation and selection of belt transmission components were made based on [2]. A SPA V-belt was decided upon. Calculations were made on pulley diameters$d_p = 100$ and$D_p = 118$ (normalized values, selected on the basis of [2]).

The value of the actual ratio of such a gearbox is:

$$i_{rz} = \frac{D_p}{d_p} = \frac{118}{100} = 1.18$$

While the necessary gear ratio, resulting from the characteristics of the selected engine and from the project's assumptions, is:

$$i = 1.175$$

The ratio error is:

$$\frac{i_{rz} - i}{i} \cdot 100\% = 0.425\%$$

According to [1], this is an acceptable error.

Then calculated / selected (based on [2]):

- WheelbaseA_0:

$$A_{0min} = 0.7 \cdot (D_{(p)} + d_{(p)}) = 0.1526$$
$$A_{0ma} = 2 \cdot (D_{(p)} + d_{(p)}) = 0.436$$

Adopted$A_1 = 0.435\ m$

- Lane lengthL_p:

$$L_{p\_calculated} = 2 \cdot A + 1.57 \cdot (D_{(p)} + d_{(p)}) + \frac{(D_{(p)} - d_p)^2}{4A} = 1{,}212\ m$$

The normalized lane length$L_p = 1{,}207 m$, according to [2], was assumed.

- Exact wheelbase :A

$$A \approx p + \sqrt{p^2 - q} = 0.432\ m$$

Where:

$$p = 0.25 L_{(p)} - 0.393 (D_{(p)} + d_p) = 0{,}216$$

$$q = 0.125 (D_{(p)} - d_p)^2 = 3.83 \cdot 10^{-5}$$

- Operating conditions factor:$k_t = 1.2$, assuming an operating time of more than 16 hours per day for light drive
- Banding factor:$k_\varphi = 0.99$

$$\cos\frac{\varphi}{2} = \frac{D_p - d_p}{2A} \rightarrow \varphi = 177.68°$$

- Lane length ratio: $k_L = 0.88$
- Rated power:$N_1 = 2.54\ kW$

- Required number of transmission belts:

$$z = \frac{N \cdot k_t}{N_1 \cdot k_\varphi \cdot k_L} = 0{,}813$$

Rounding up, a single belt was used in the gearbox.

The forces acting on the belt, were also calculated according to [2]. In the design, the case that is most "dangerous" for the system should be checked. Therefore, it was decided to take into account the dynamic force.

- Dynamic axle load from forces in the passive tendons:

$$T_{(b)} \approx \frac{1020 \cdot (1.02 - k_\varphi) \cdot N \cdot k_t}{k_\varphi \cdot v} = 7{,}536\,N$$

Where the lane movement speed: $v = 7.38\ m/s$

- Dynamic axle load from forces in active tendons:

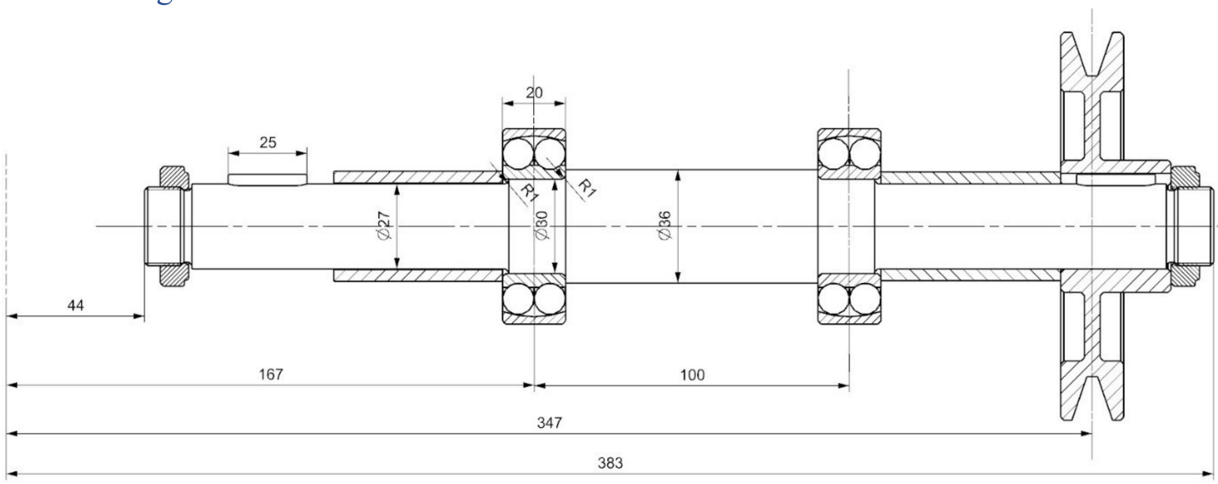
$$T_{(c)} \approx \frac{1020 \cdot N \cdot k_t}{k_\varphi \cdot v} = 251{,}2\,N$$

- Accidental dynamic force:

$$N_{s\,dyn} \approx \sqrt{T_c^2 + T_b^2 + 2 \cdot T_c \cdot T_b \cdot cos\varphi} = 258.73\,N$$

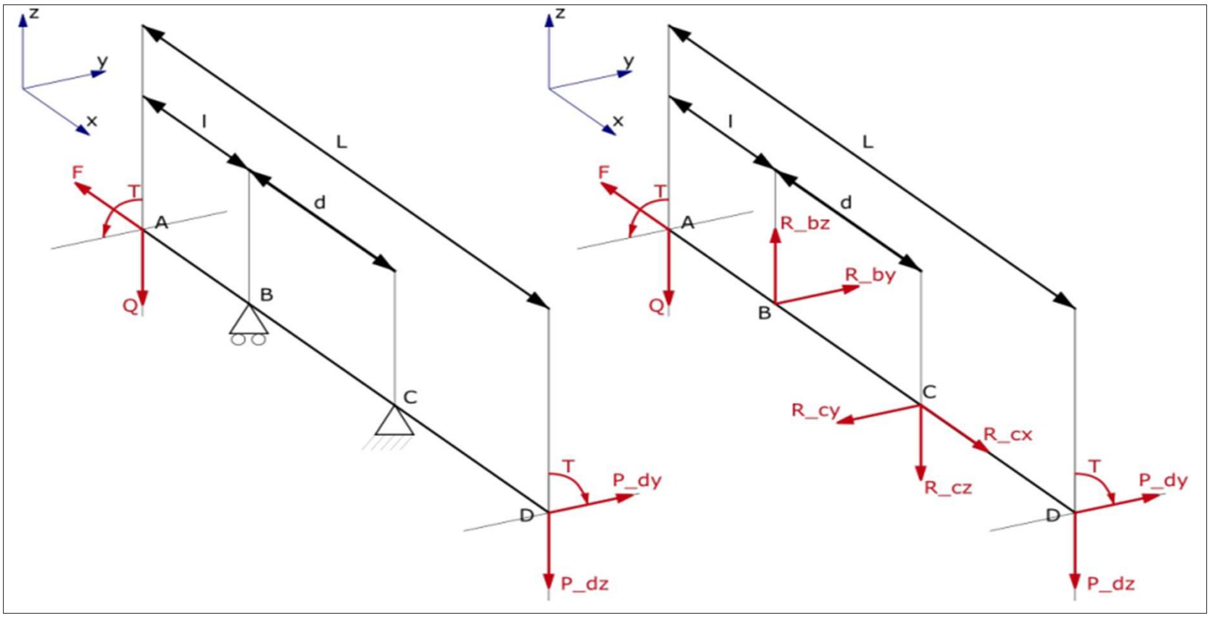
4 Loads shaft

4.1 Arrangement of diameters of the shaft



The above figure shows the arrangement of the various elements on the shaft, the distances between them and the lengths of the various segments of the shaft.

4.2 Statics of the shaft



Data adopted for calculations:

Designation	Value	Unit
L	0,347	m
l	0,167	m
d	0,1	m
F	310	N
Q	126	N
T	11,94	Nm
P_dy	255,32	N
P_dz	41,81	N

Where the torsional moment from the belt: $T = \frac{(N \bullet D)\ (p)}{2v}$

Dynamic force component on y-direction: $P_{dy} = N_{s_{dyn}} \bullet \sin\alpha$

Dynamic force component on z direction: $P_{dy} = N_{s_{dyn}} \bullet \cos\alpha$

Where alpha is the angle between the vertical and the dynamic force from the chord

Freeing the system from ties and calculating the reaction forces in the supports, we get:

Designation	Value	Unit
R_{bx}	0	N
R_{by}	204,3	N
R_{bz}	303	N
R_{bw}	365,69	N
R_{cx}	310	N
R_{cy}	-459,6	N
R_{cz}	135,2	N
R_{cw}	479	N

Where

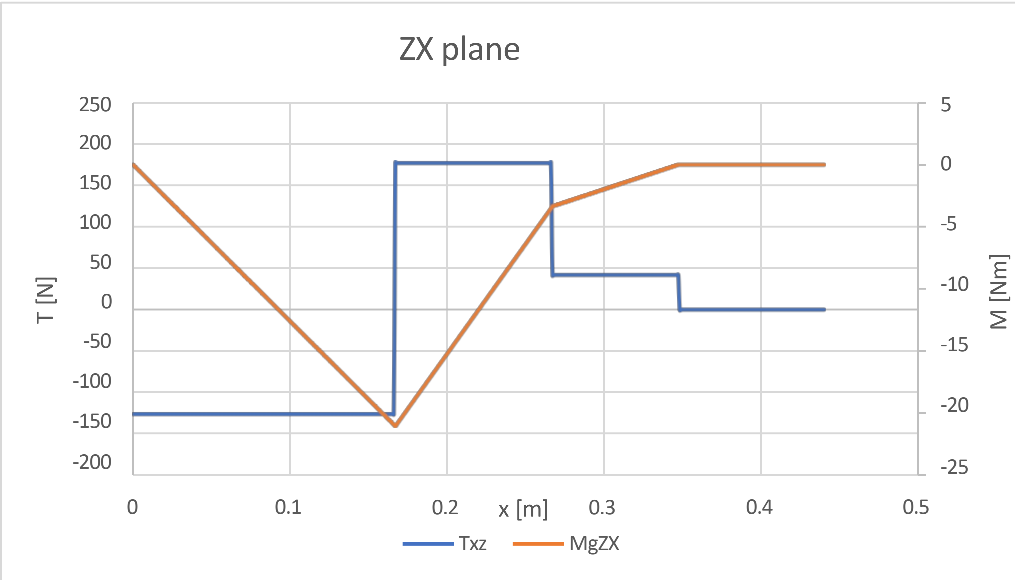
$$R_{bw} = \sqrt{{R_{by}}^2 + {R_{bx}}^2}$$

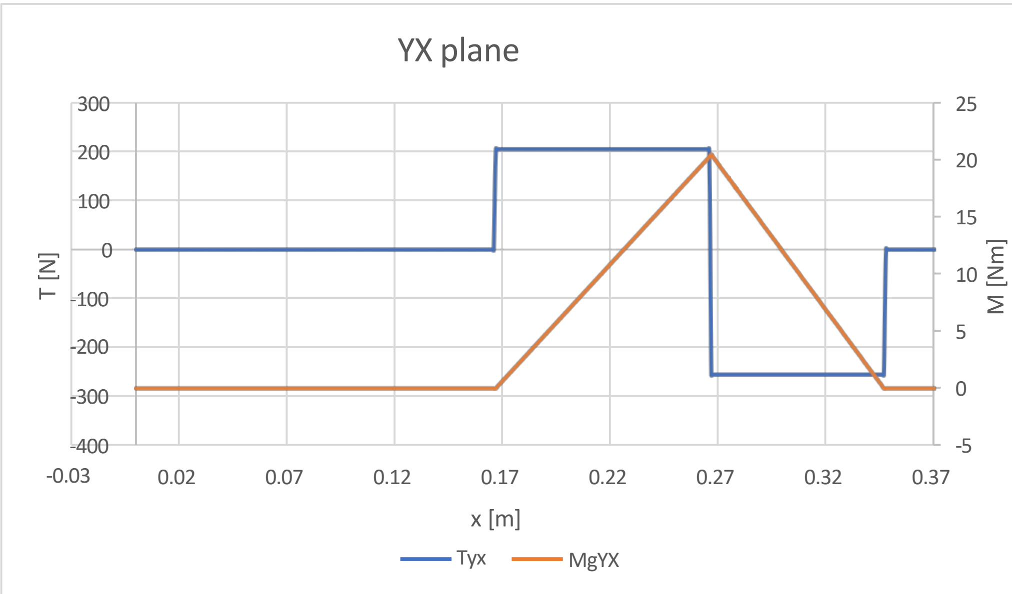
$$R_{cw} = \sqrt{{R_{cy}}^2 + {R_{cx}}^2}$$

The non-sliding base was placed in a more stressed position.

4.3 Distribution of components of effort

All calculations were carried out assuming, the theoretical case in which the shaft starts at the center of gravity of the fan.



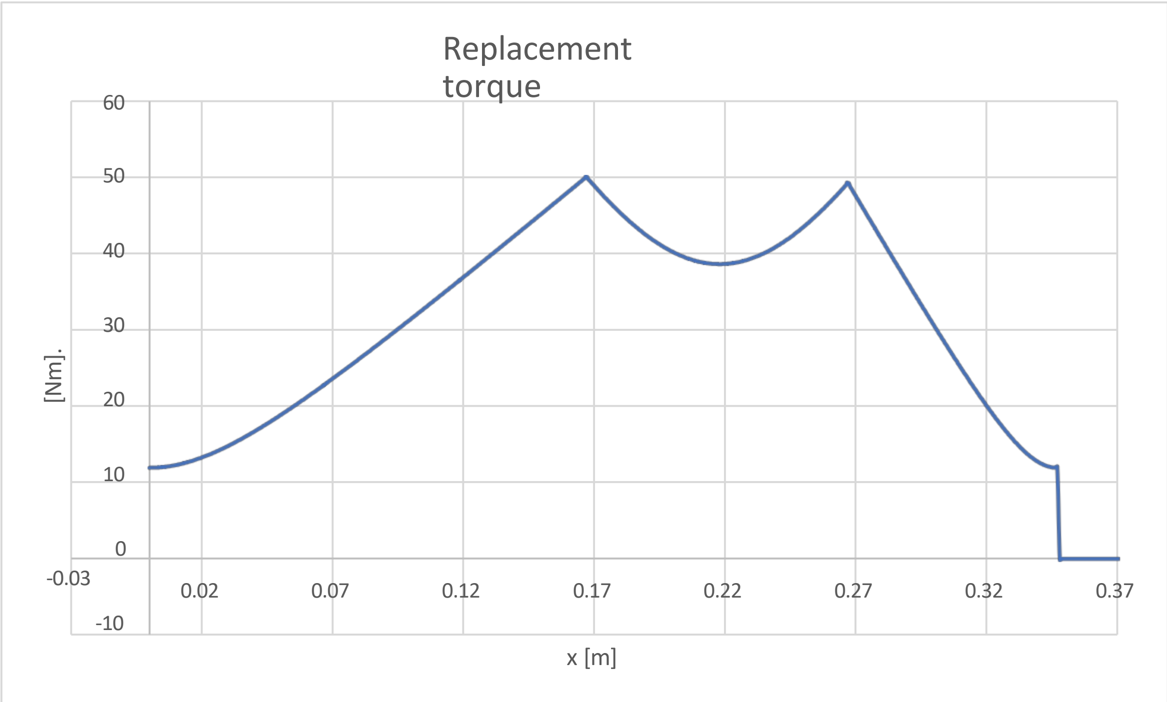


After determining the basic load distributions on the shaft, the resultant and equivalent bending moment were calculated:

$$M_{gw} = \sqrt{M_{yx}^{2} + M_{zx}^{2}}$$

$$M_{zast} = \sqrt{M_{(gw)}^{(2)} + (\alpha \bullet T)^{2}}$$

Where $\alpha = \frac{\sqrt{3}}{4}$



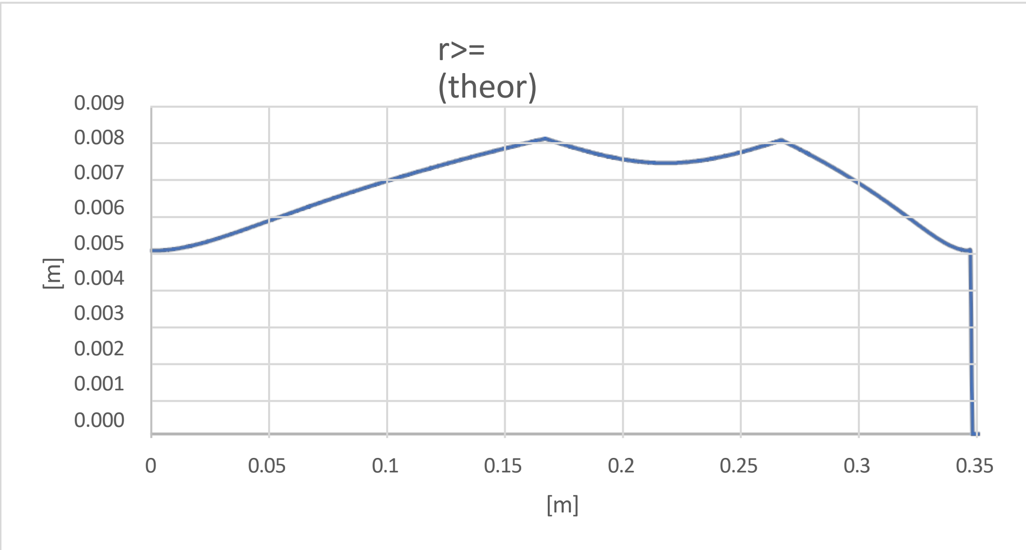
5 Determination of shaft diameters

5.1 Strength ad hoc

When determining the theoretical diameter, axial loads were ignored - they have a marginal effect on the obtained results.

$$d_o = \sqrt[3]{\frac{M_{zast} \cdot 10^3 \cdot x}{0,1 \cdot Z_{go}}}$$

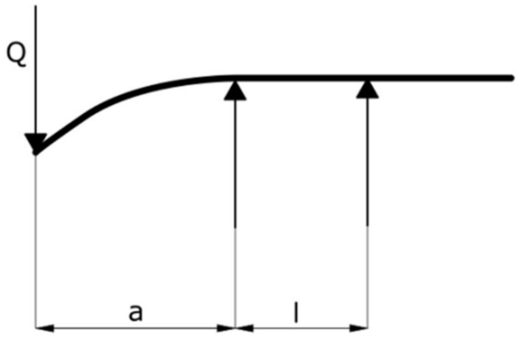
Where: $x = 4$, while $Z_{go} = 250\,MPa$



5.2 Static stiffness of the shaft

A diameter that is the arithmetic average of the two extreme diameters on the shaft was used to calculate the deflection line:

$$d = \frac{D_{\min} + D_{\max}}{2} = \frac{0.027 + 0.36}{2} = 0,0315$$



In addition to this, the load from the gearbox was ignored - only the force is responsible for the deflection . To calculate the deflection line, the formula was used:

$$f = \frac{Q \bullet a^2}{3EJ} \bullet (l + a) = 3.08 \bullet 10^5 m$$

Designation	Value	Unit
Q	126	N
E	210	GPa
J	$4.83 \cdot 10^{-8}$	m^4
l	0,1	m
a	0,167	m

Where, E is the Young's modulus of C35 steel, and the bending moment of inertia: $J = \frac{\pi \cdot d^4}{64}$.

According to [3], the deflection arrow must meet the condition: $f = (2 \div 3) \cdot 10^4 \cdot l = (2 \div 3) \cdot 10^5\ m$

The requirement was not met. After analysis of the critical rotation and consultation with the instructor, the deviation was found to be acceptable.

5.3 Critical rotation of the shaft

The critical speed was obtained from the relationship: :

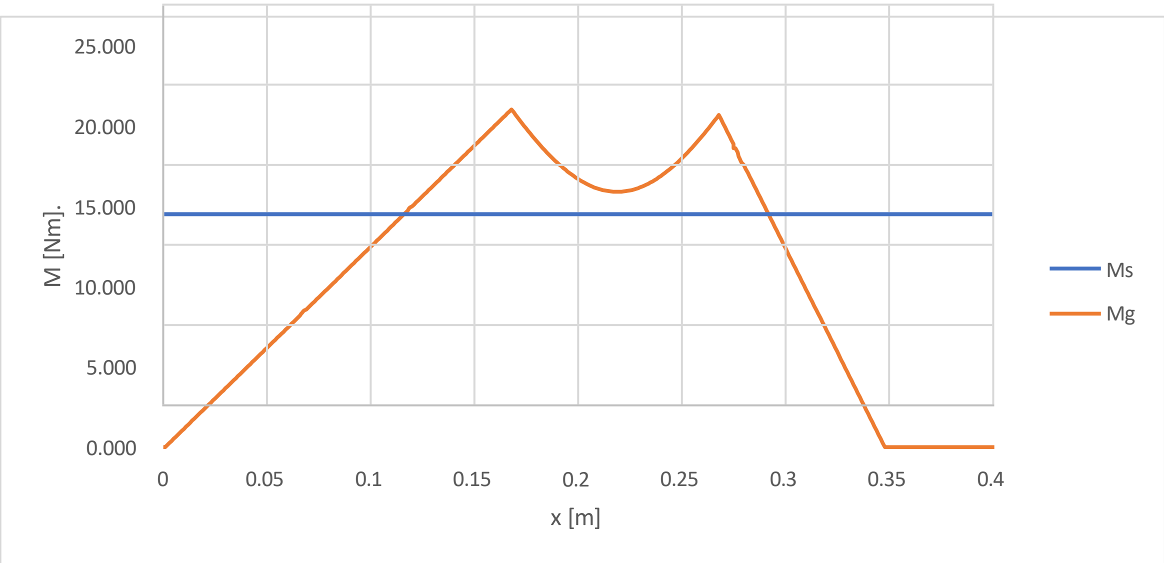
$$\omega_{kr} = \sqrt{\frac{g}{f}} = \sqrt{\frac{9{,}81}{3.08 \cdot 10^5}} = 564\ \frac{rad}{s} = 5388\ rpm$$

The critical speed is more than 4 times the required speed - the shaft meets the requirements.

5.4 Fatigue strength analysis of the shaft

For the fatigue strength analysis of the shaft, 2 cross-sections were selected that may prove to be "weak links". These are the location of the diameter change from the largest diameter to the diameter of the journal on which the bearings are mounted, and the location of the keyhole.

The diagram below shows the moments that were taken into account in the calculations:



These moments for the counted points are respectively:

x [mm]	M [Nm]	Ms [Nm]
90	11.34	11.93
177	19.38	11.93

Coefficients for calculations:

$\alpha_{1s} = 1{,}62$ $\quad \dfrac{1}{\varepsilon_{1s}} = 1{,}28$ $\quad \beta_{k1s} = 1{,}41$ $\quad \beta_{p1s} = 1{,}06$ $\quad \beta_{1s} = 1{,}47$

$\alpha_{1g} = 1{,}94$ $\quad \dfrac{1}{\varepsilon_{1g}} = 1{,}305$ $\quad \beta_{k1g} = 1{,}71$ $\quad \beta_{p1g} = 1{,}06$ $\quad \beta_{1g} = 2{,}77$

$\alpha_{2s} = 1{,}9$ $\quad \dfrac{1}{\varepsilon_{2s}} = 1{,}34$ $\quad \beta_{k2s} = 1{,}61$ $\quad \beta_{p2s} = 1{,}02$ $\quad \beta_{2s} = 1{,}63$

$\alpha_{2g} = 2{,}37$ $\quad \dfrac{1}{\varepsilon_{2g}} = 1{,}38$ $\quad \beta_{k2g} = 2{,}01$ $\quad \beta_{p2g} = 1{,}02$ $\quad \beta_{2g} = 3{,}03$

Tensile indices and stresses in successive sections:

W_{1s}	254	W_{1g}	127	W_{2s}	176.7	W_{1g}	88.3
r1sa	23.44	σ_1	89	r2sa	33.7	σ_2	219
r1sm	23.44	-	-	r2sm	33.7	-	-

Coefficient psi = 0.05 for steels with medium carbon content. Then fatigue safety stocks:

σ_{1s}	151.47	σ_{2s}	195.4
σ_{1g}	23.39	σ_{2g}	17.65

Accidental safety stocks:

σ_1	24.76
σ_2	17.34

All received results are many times greater than adopted safety factor x=4.

6 Selection of bearings

Using the SKF website, self-aligning ball bearings 2206 ETN9 and housings SNL 206-305. bearing parameters can be found in the table below:

Designation	Value	Unit
d_w	30	mm
D_z	62	mm
B	20	mm
C	23,8	kN
C_0	6,7	kN
P_u	0,345	kN
e	0,33	-
X	0,65	-
Y_2	1,9	-

In this subsection, the bearing life is checked. Only the more exposed bearing, locked in the x-axis and loaded longitudinally, is checked. Calculations were carried out according to the scheme in the SKF bearing catalog [4].

Forces acting on the bearing:

- $R_{cw} = 479N$ - lateral force
- $R_{cx} = 310N$ - axial force

$$\frac{R_{cx}}{R_{cw}} = 0.647 > 0.33$$

Thus:

$$F_{zast} = X \cdot R_{cw} + Y_2 \cdot R_{cx} = 900.4\,N$$

Hourly bearing operation time to destroy 10% of the sample:

$$L_{10h} = \frac{10^6}{60 \cdot n}\left(\frac{C}{F_{zast}}\right)^p = 256515\,h$$

Where $p = 3$ for ball bearings.

The bearing will last the required 130,000 hours of operation.

7 Grooves

Based on [1], a prismatic keyway was selected with dimensions: 8x7x25 mm. Calculations were carried out for the data:

Designation	Value	Unit
d_{shaft}	27	mm
T	11,94	Nm
h	7	mm
t_1	4	mm
L	25	mm

The parameters of the grooves are chosen so that the decisive stresses are surface pressures:

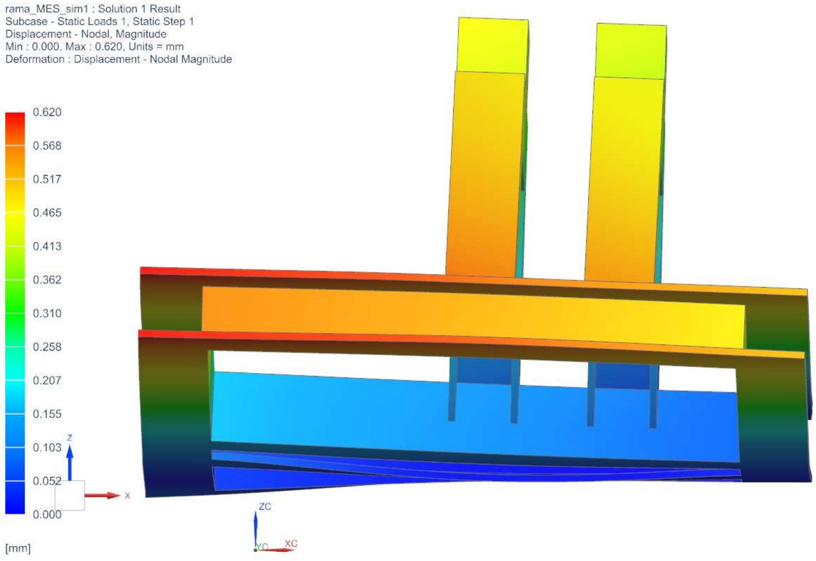
$$p_{dop} = \frac{2T \cdot 10^3}{(h - t_1) L \cdot d_{shaft}} = 14{,}74\ MPa$$

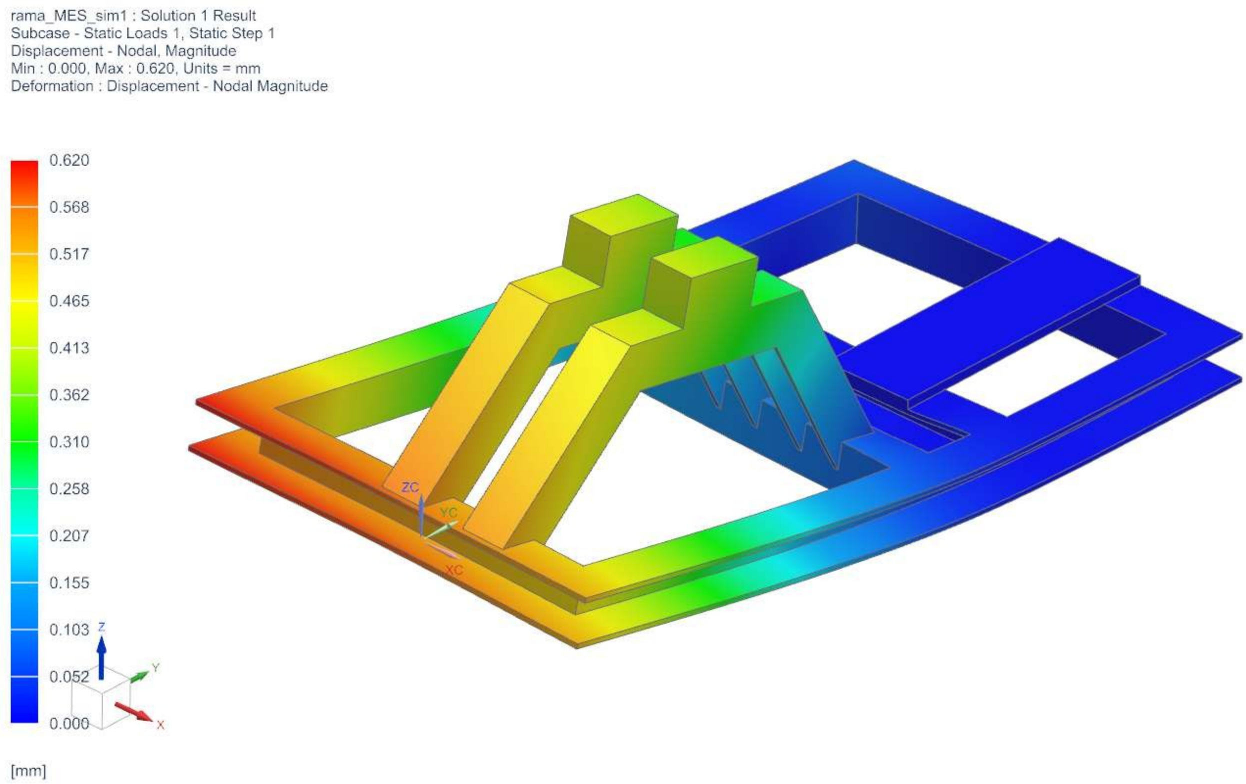
Thus, the keyway meets the strength requirements [1].

8 Statics frame

The frame is made of S235JR structural steel. The component consists of channels welded together. The legs of the device, made on the basis of Kipp's K0742 feet, are arranged by means of welded, unthreaded bushings. The movement of the feet is blocked by a set of nuts and washers, and their design allows easy reduction of unevenness in the ground. Bolted connections with washers were also used to attach the belt tension system and bearing housings.

The stiffness of the structure was examined using the FEA module built into the NX system. The elements directly receiving the loads, i.e. the bearing housings and the motor together with the shaft, were replaced by simple solids. A mesh of dimension was superimposed on such a frame. The calculation results are shown below. The image itself is the deformation scaled by the characteristic dimension.

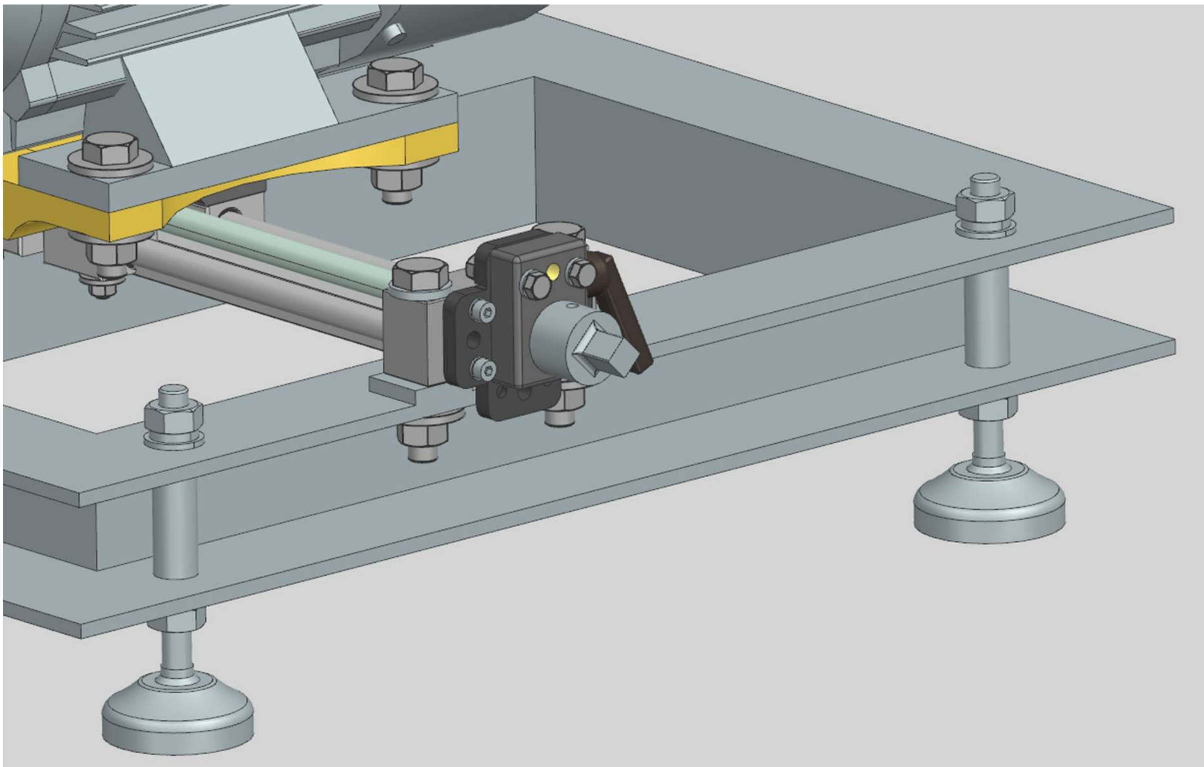




The maximum deformation is about 0.62 mm - a small and acceptable value in view of the dimensions of the whole system and the presence of the belt tension system, which can compensate for any changes.

9 Description of the tensioning system belt

The device uses a belt tensioning system in the form of a sliding motor on a table - SLW- 1040 from igus.



The maximum axial load capacity of the system is 700N, radial 2800N. The guides provide high rigidity and prevent twisting of the component. A screw in the axis of symmetry of the system is responsible for adjusting the position of the motor. With the help of a grub screw, an adapter for a 10 key is attached to it. The rotation itself is blocked by the SHT-HK-12 manual clamp, connected through an adapter. Both parts were made by the same company.

10 Sources

[1] Kurmaz L. W., Kurmaz O. L. **Fundamentals of Node and Machine Parts Design.** 2011
[2] Starego W., **Constructor's Guide (for the design of belt gears) Sanok Rubber.** 2017
[3] Mazanek E., **Examples of calculations from the fundamentals of amshin design vol. 2**
[4] SKF, **Catalog of rolling bearings.** 2014
[5] Catalog data from websites:
  [5.1] https://www.skf.com/pl/productinfo/productid-2209%20ETN9 (as of as of 15/04/2023)