TALLINN UNIVERSITY OF TECHNOLOGY

SCHOOL OF ENGINEERING

Department of Mechanical and Industrial Engineering

PROJECT DOCUMENTATION

MES0225

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INITIAL CONDITIONS

Name

Electric winch (EST. *Elektriline vints*, GER. *elektrische Winde*)

Purpose, field of use

An electric winch is a versatile tool used for tasks such as lifting, lowering, and transporting heavy loads. This project aims to design a device that can be utilized across multiple industries, functioning both as a standalone unit and as an integrated component within larger systems or technologies. The primary focus of its development is to streamline and automate production processes, improving both efficiency and user convenience.

Basic functions, parameters and general technical requirements

General technical requirements

- load capacity – 1100 kg,
- rope velocity – 0,14 m/s,
- connection between gear motor and drum – chain drive
- during the design to minimize sizes and mass of the device.

Base functions:

- lifting of hard weight components in assembling shop
- simplification of the assembly

As an engine is used spur-or worm gear-motor that is connected to the drum by help of chain drive.

Base materials:

Drum is welded with steel components. Type of steel – S235J2G3 EN 10025. Drum by help of two hub units is supported on the shaft. Shaft is on full length of the drum. Shaft material – steel C45E EN10083. Torque is transmitted from drum to shaft by help of keys in both of hub units. The shaft is supported by self-aligning bearings. Bearing units are joined to the frame by bolts. The frame is made of steel pipes (material – S355J2H) or UNP-components (components – S235JRG2).

Mechanical properties of materials [1]:

Steel S235

- yield stress – ReH (σY) = 235 MPa;
- ultimate stress – Rm (σU) = 370 – 470 MPa.

Steel S355

- yield stress – ReH (σY) = 355 MPa;
- ultimate stress – Rm (σU) = 490 – 610 MPa.

Steel C45E

- yield stress – ReH (σY) = 370 MPa;
- ultimate stress – Rm (σU) = 630 MPa;
- fatigue limit – σ-1 = 275 MPa, σ-1 = 165 MPa.

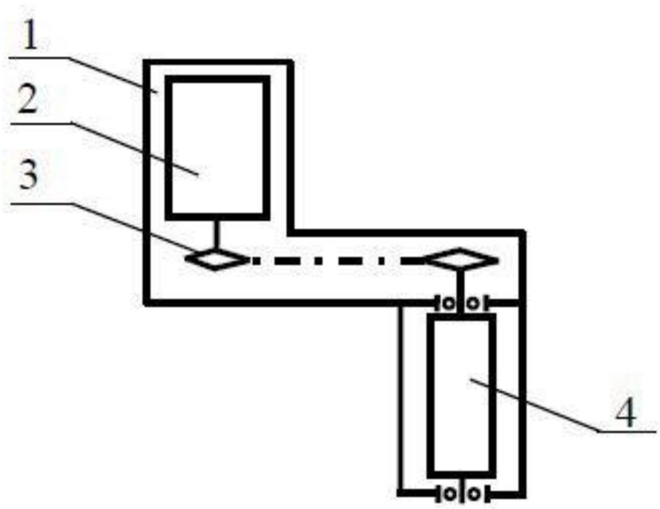
Modulus of elasticity – E = 2,1.105 MPa.

Shear modulus – G = 8,1.104 MPa.

Compliance with special conditions

The device must meet the following requirements:

- reliability,
- environmentally friendly; lubricants must not enter the environment,
- safety; once a year, the rope is tested for strength,
- climate resistance; operating temperature -10°C… + 40°C,
- aesthetics and ergonomics; the product has a commercial appearance.



1 – frame, 2 – gear motor, 3 – chain drive, 4 – drum.

Figure 1: Kinematic Diagram

DRUM DIAMETER CALCULATION AND ROPE SELECTION:

Maximum internal force of the rope should satisfy the following condition:

$$F_{max} \leq [F] = \frac{F_{cr}}{S}$$

The maximum tension force:

$$F_{max} = mg = 1100 \times 9.81 = 10791\ N$$

where g – gravitational acceleration, m – maximum mass

Design factor $[S]$ = 5.5

Then,

$$F_{cr} = F_{max} \times [S] = 10791 \times 5.5 = 59350 N$$

Taking into account that the rope can be wound in more than one layer let us use the rope TEK 13310 [3], that has Ft = 59,7 kN.

Figure 2: Rope TEK13310

Then,

$$F_{max} = 10{,}7kN \ < [F] = \frac{F_t}{S} = \frac{59{,}7}{5{,}5} \approx 10{,}9kN$$

Rope size d = 10 mm. Then the drum diameter

$$D = \ e \times d = 20 \times 10 = 200mm$$

where e – ratio depending on work conditions (in range 20 … 30), in our case e = 20

Let us use D = 200 mm

from the range 80, 90, 100, 110, 120, 140, 160; 200; 250; 320; 400; 450; 560; 630; 710; 800; 900; 1000 mm.

GEAR MOTOR CALCULATIONS:

Power for drum rotation,

$$P_t \ = T \ \times \omega_T$$

where T – torque, Nm

ω_T – angular velocity, rad/s.

Torque,

$$T = F\frac{D}{2}$$

where F – load capacity.

Then,

$$T = F\frac{D}{2} = 10791\frac{0,2}{2} = 1079Nm$$

Angular velocity,

$$\omega_T = \frac{2v}{D} = \frac{2 \times 0,14}{0,2} = 1,4\frac{rad}{s}$$

Power for drum rotation,

$$P_t = T \times \omega_T = 1079 \times 1,1 \approx 1510W$$

Minimum necessary power of gear motor,

$$P_{M\,min} = \frac{P_T}{\eta_1 \eta_2 \eta_3}$$

where η_1 – efficiency of gear motor, let us use η_1 = 0,94 (for worm gears η_1 = 0,75),

η_2 = 0,92 – efficiency of chain drive,

η_3= 0,99 – efficiency of bearings.

Then,

$$P_{M\,min} = \frac{P_T}{\eta_1 \eta_2 \eta_3} = \frac{1187}{0,94 \times 0,92 \times 0,99} \approx 1764W$$

Frequency of drum rotation

$$n_T = \frac{30\omega_T}{\pi} = \frac{30 \times 1,4}{\pi} = 13,36min^{-1}$$

Then the frequency of gear motor rotation,

$$n_R = n_T \times u_k$$

where u_k– ratio of chain drive.

Let us use u_k = 1,6 (this ratio is changeable in range 1≤ u ≤7) then,

$$n_R = n_T \times u_k = 13,36 \times 1,6 = 21,39\,min^{-1}$$

According to minimum power and frequency of gear motor rotation let us use the R97 DV 100M4.

$$P_M=2.2$$

M= 970

$$n_R= 22$$

$$u_R= 65.21$$

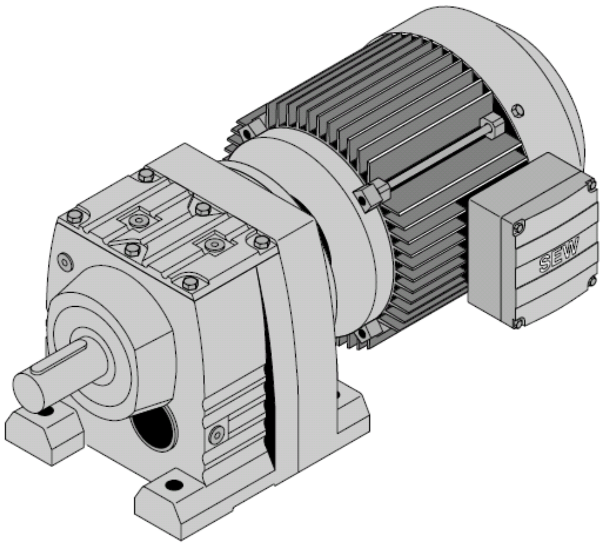


Figure 3: R-Type Gear Motor

Angular velocity,

$$\omega_R = \frac{\pi \times n_R}{30} \approx 2{,}30\frac{rad}{s}$$

Real frequency of the drum rotation,

$$n_T = \frac{n_R}{u_k} = \frac{22}{1{,}6} = 13{,}75 min^{-1}$$

Real velocity,

$$v = \frac{\pi \times n_T \times D}{60} = \frac{\pi \times 13{,}75 \times 0{,}2}{60} \approx \frac{0{,}14m}{s}$$

Let us check real power at the drum,

$$P = P_M \eta_1 \eta_2 \eta_3 = 2{,}2 \times 0{,}94 \times 0{,}92 \times 0{,}99 = 1{,}9kW$$

CALCULATION OF CHAIN DRIVE:

Twisting moment on the driving chainwheel

$$T_K = \frac{T}{u_k \eta_2 \eta_3} = 740Nm$$

Tario of chain drive u_k = 1,6

The minimum number of teeth of the driving chain wheel depends on frequency of wheel rotation: at large frequencies $z_{1\,min} = 19 \ldots 23$ *, at middle* $z_{1\,min} = 17 \ldots 19$*, at pure* $z_{1\,min} = 13 \ldots 15$*. It is recommended to use odd numbers.*

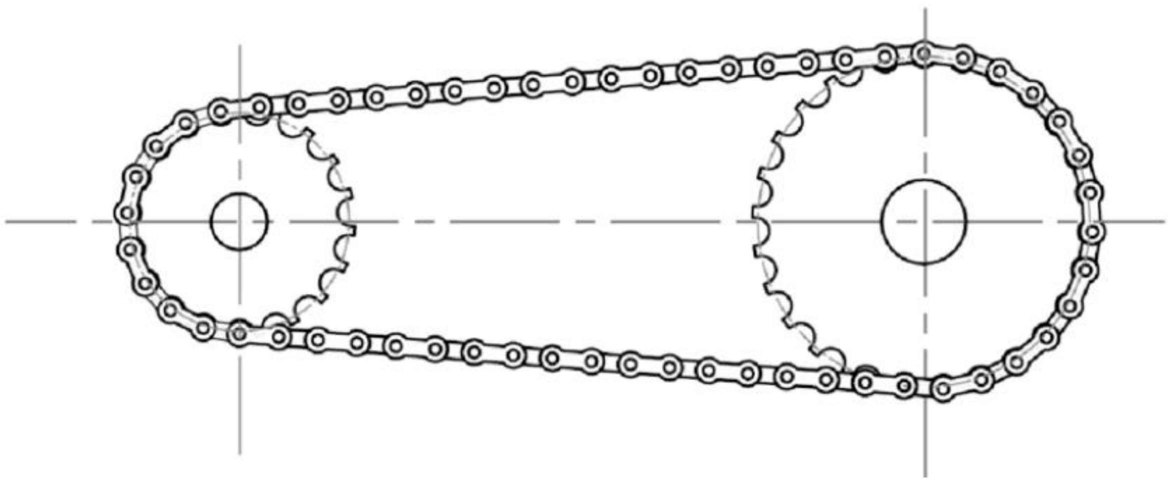


Figure 4: Image of the Chain Drive

Recommended number of teeth,

$$z_1 = 29 - 2u = 29 - 2 \times 1{,}6 \approx 26$$

Taken into account minimizing device size let us select,

$$z_1 = 19[5{,}6]$$

Then the number of teeth of the driven wheel can be found as follows,

$$z_2 = z_1 u = 19 \times 1{,}6 \approx 31$$

Load coefficient,

$$K = k_d k_a k_n k_r k_m$$

where k_d – dynamic coefficient (k_d = 1-calm load, k_d = 1,25 ... 1,5- variable or shock load)

k_a – coefficient taken into account distance between wheels centers

$$K = k_d k_a k_n k_r k_m = 1{,}25 \times 1 \times 1 \times 1{,}25 \times 1{,}3 \approx 2{,}03$$

The pitch of a single strand roller chain

$$t \geq 2{,}8 \times \sqrt[3]{\frac{KT_K}{z_1 p}}$$

Where p- overage pressure.

Let us select $p = 46MPa$

$$p = 46 \times k_z = 46[1 + 0{,}01(z_1 - 17)]$$

$$46[1 + 0{,}01(19 - 17)] = 47MPa$$

$$t \geq 2{,}8 \times \sqrt[3]{\frac{2{,}03 \times 740}{19 \times 47 \times 10^6}} = 33mm$$

According to the available information [5, 6], it can be used the chain drive 20B1W that has t = 31,75 mm.

The higher the chain pitch the greater the wheel diameter for the same teeth number. Decreasing of device size can be achieved by using smaller pitches of chain. In this case the chain strength can be achieved by using of double strand or triple strand chains.

Thus, let us select the double strand chain 20B2W [5, 6], that has the pitch t = 31,75 mm, the mass q = 5,91 kg/m, critical load F_{kr}= 170 kN, roller width b = 19,56 mm, roller diameter d = 19,05 mm.

Velocity of the chain,

$$v = \frac{z_1 t \times n_R}{60 \times 10^3} = \frac{19 \times 31{,}75 \times 22}{60 \times 10^3} = 0.22\frac{m}{s}$$

Circular Force

$$F_R = \frac{T_K \omega_R}{v} = \frac{T_K \times \pi \times n_R}{30v} = 7{,}7kN$$

Pressure on the roller

$$p = \frac{KF_R}{2bd} = \frac{2{,}03 \times 7700}{2 \times 19{,}56 \times 19{,}05} = 21MPa < [p] = 36 \times k_z$$

$$= 36[1 + 1 + 0.01(19 - 17)] = 36{,}7MPa$$

Strength conditions are satisfied. In the opposite case, it is necessary to use a triple strand chain.

Thus, it is selected the double strand chain 20B2W(DIN8187).

FORCE IN THE CHAIN

From bending

$$F_f = 9{,}81 k_f q a$$

Where k_f – coefficient taken into account angle of the chain location (k_f = *6* – at horizontal location, k_f = 1 – at vertical location, k_f = *1,5 – at angle of 45°* [7]),

a – distance between wheels center (let us use *a* = 800 mm).

The minimum distance between wheels centers $a_{min} = 0{,}6(D_{01} + D_{02}) + 30 \ldots 50$mm, where D_{01} and D_{02} external diameter of the wheel. Recommended distance is *a= 30t…50t,* where t – chain pitch. The maximum distance between wheels center $a_{max} \leq 80t$.

Then,

$$F_f = 9{,}81 k_f q a = 9{,}81 \times 1{,}5 \times 5{,}91 \times 0{,}8 = 70N$$

Centrifugal force,

$$F_t = qv^2 = 5{,}91 \times 0{,}22^2 = 0{,}2N$$

Load on the shaft,

$$F_V = F_R + 2F_f = 7700 + 2 \times 70 = 7840$$

Safety factor of the chain,

$$S = \frac{F_{kr}}{F_v + F_t + F_f} = \frac{170000}{7840 + 0{,}2 + 70} = 21 > [S] = 7{,}8$$

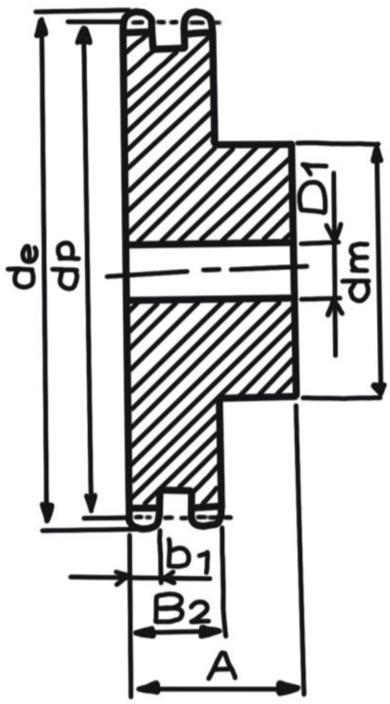


Figure 5: Chain Wheel

CHAIN WHEEL

Driving wheel – D114-19, driven wheel D114-31 [5, 6].

CODE	z	D_p	D_e	D_1	d_m	A
D114-19	19	192,91	208,1	25	120	80
D114-31	31	313,85	329	25	150	80

SHAFT DESIGN

Calculations

Minimum diameter of the shaft

$$d_v \geq \sqrt[3]{\frac{16T}{\pi[\tau]}} = \sqrt[3]{\frac{16 \times 1079}{\pi \times 30 \times 10^6}} = 0{,}05m$$

Where $[\tau] = 20 \ldots 30MPa$;

Let us use $d_v = 50mm$

Diameter under bearing d_t = 55 mm.

Internal diameter of the hub $d_r = 60$ mm.

Other sizes to be selected at design.

Hub length,

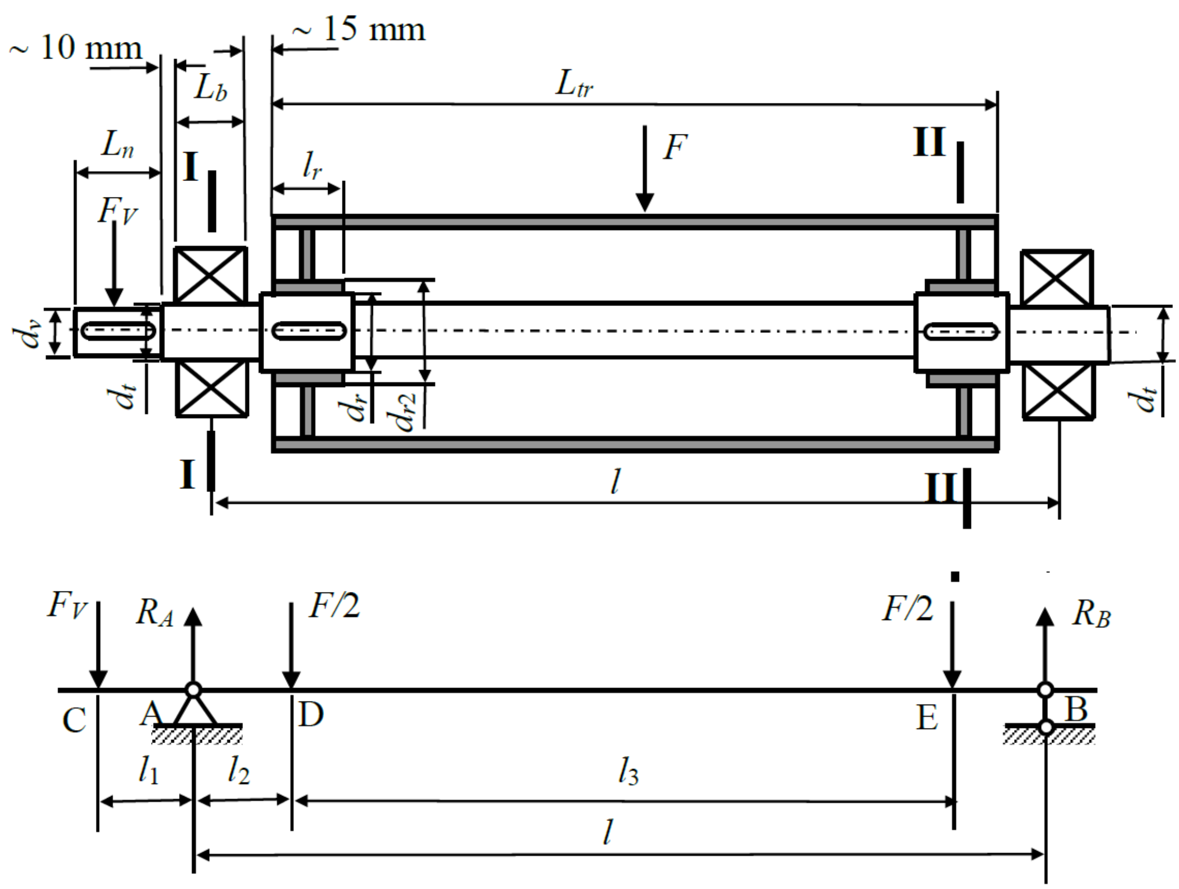
$$l_r = (1{,}2 \dots 1{,}8)d_r = (1{,}2 \dots 1{,}8)60 = 72 \dots 108\ mm$$

Let us use $l_r = 90mm$

External diameter of the hub,

$$d_{r2} = (1{,}6 \dots 1{,}8)d_r = (1{,}6 \dots 1{,}8)60 = 96 \dots 108mm$$

Let us use $d_{r2} = 100mm$



Let us select approximately distance between bearings according to drum length and width of bearing units and taken into account necessary distance between frame and drum.

Width of chain wheel L_n= 80 mm;

Width of the bearing unit $L_b \approx 60\ mm$;

Length of the drum hub L_r = 90 mm;

Drum lengthL_{tr} = 400 mm.

$l = L_{tr} + 2 \cdot \frac{L_b}{2} + 2 \cdot 15 = 400 + 2 \cdot \frac{60}{2} + 2 \cdot 15 = 490$mm

$l_1 = \frac{L_n}{2} + \frac{L_b}{2} + 10 = \frac{80}{2} + \frac{60}{2} + 10 = 80$ mm;

$l_2 = \frac{l_r}{2} + \frac{L_b}{2} + 15 = \frac{90}{2} + \frac{60}{2} + 10 = 85$ mm;

$l_3 = l - 2l_2 = 490 - 2 \cdot 85 = 320$ mm;

Loads:

F= 10,7 kN; $F_v = 7{,}8$ kN;

$\sum m_A = 0$

$$R_B l - \frac{F}{2}(l_2 + l_3) - \frac{F}{2} l_2 + F_V l_1 = 0$$

Then

$$R_B = \frac{\frac{F}{2}(l_2 + l_3) + \frac{F}{2} l_2 - F_V l_1}{l} = \frac{5{,}35\ \times 0{,}405 + 5{,}35 \times 0{,}085 - 7{,}8 \times 0{,}080}{0{,}490} \approx 4{,}07\ kN$$

$\sum m_B = 0$

$$\frac{F}{2}(l - l_2 - l_3) + \frac{F}{2}(l + l_2) - R_A l + F_V(l + l_1) = 0$$

Then

$$R_A = \frac{\frac{F}{2}(l - l_2 - l_3) + \frac{F}{2}(l - l_2) + F_V(l + l_1)}{l} =$$

$$= \frac{5{,}35(0{,}490 - 0{,}085 - 0{,}320) + 5{,}35(0{,}490 - 0{,}085) + 7{,}8(0{,}490 - 0{,}080)}{0{,}490}$$

$$\approx 11{,}87\ kN$$

Let us design the diagrams of bending moments and torque:

$M_A = -7800 \cdot 0{,}080 \approx -624\ Nm$

$M_D = -F_V \cdot (l_1 + l_2) + R_A \cdot l_2 = -7800 \cdot (0{,}080 + 0{,}085) + 1187 \cdot 0{,}085 \approx -1186\ Nm$

$M_E = F_V \cdot (l - l_2 - l_3) = 7800 \cdot (0{,}490 - 0{,}085 - 0{,}320) \approx -663\ Nm$

Equivalent moment in the critical section (according to IV theory):

$M_{ekv}^{IV} = \sqrt{M^2 + 0{,}75T^2} = \sqrt{-624^2 + 0{,}75 \times 1079^2} \approx 695$ Nm.

$\sigma_{ekv}^{IV} = \frac{M_{ekv}^{IV}}{W} = \frac{32M_{ekv}^{IV}}{\pi d_t^3} = \frac{32 \cdot 695}{3{,}14 \cdot 0{,}055^3} \approx 49\ MPa < [\sigma] = \frac{R_{p0,2}}{S} = \frac{370}{1{,}5} \approx 247$ MPa

CONTROL:

Let us consider stress concentration between two diameters of the shaft.

Let us take the values of stress concentration factors K_σ and K_τ from Table 3 (Annex 1), and scale factors $K_{d\sigma}$and $K_{d\tau}$ from Table 4. Surface treatment factor $K_F = 0{,}97 \ldots 0{,}90$. Empirical factors $\varphi_\sigma = 0{,}1$ – for alloy and carbon steels and $\varphi_\tau = 0{,}25 \ldots 0{,}3$ for alloy steel and $\varphi_\tau = 0{,}20$ for carbon steel.

Let us use $R = 1\ mm$,

Then, $K_\sigma = 1{,}96$, $K_\tau = 1{,}3$, $K_{d\sigma} = 0{,}83$, $K_{d\tau} = 0{,}69$, $K_F = 0{,}95$, $\varphi_\sigma = 0{,}2$, $\varphi_\tau = 0{,}1$.

Safety factor for bending,

$$S_\sigma = \frac{\sigma_{-1}}{\frac{K_\sigma}{K_F K_{d\sigma}}\sigma_a + \psi_\sigma \sigma_m}$$

Where the amplitude stress,

$$\sigma_a = \frac{M}{W} = \frac{32M}{\pi d_t^3} = \frac{32 \cdot 624}{3{,}14 \cdot 0{,}055^3} \approx 38\ MPa$$

And the middle stress $\sigma_m = 0$.

Then,

$$S_\sigma = \frac{\sigma_{-1}}{\frac{K_\sigma}{K_F K_{d\sigma}}\sigma_a + \psi_\sigma \sigma_m} = \frac{275}{\frac{1{,}96}{0{,}95 \cdot 0{,}83} \cdot 38 + 0{,}2 \cdot 0} \approx 2{,}9$$

Safety factor for torsion,

$$S_\tau = \frac{\tau_{-1}}{\frac{K_\tau}{K_F K_{d\tau}}\tau_a + \psi_\tau \tau_m},$$

Where middle and amplitude stress

$$\tau_m = \tau_m = \frac{\tau_{max}}{2} = \frac{T}{2W_p} = \frac{16T}{2\pi d_t^3} = \frac{16 \cdot 1079}{2 \cdot 3{,}14 \cdot 0{,}055^3} \approx 16{,}5\ MPa$$

Then

$$S_\tau = \frac{\tau_{-1}}{\frac{K_\tau}{K_F K_{d\tau}}\tau_a + \psi_\tau \tau_m} = \frac{165}{\frac{1{,}3}{0{,}95 \times 0{,}69}16{,}5 + 0{,}1 \times 16{,}5} = 4{,}8$$

Thus the common safety factor

$$S = \frac{S_\sigma S_\tau}{\sqrt{S_\sigma^2 + S_\tau^2}} = \frac{2{,}9 \cdot 4{,}8}{\sqrt{2{,}9^2 + 4{,}8^2}} \approx 2{,}5$$

The common factor of safety should be [S] = 2.5 ... 3. Thus, the designed diameters of the shaft can be used.

BEARINGS:

Let us use bearing unit SY [8] with self-aligning bearing.

Shaft diameter that will be in contact with the bearing inner ring is dt = 55 mm.

Let us select bearing unit *SY 55 TF*, that consists of the bearing *YAR 211-2F* and housing *SY 511 M*.

Force to be applied on the bearings: $R_A = 18{,}40kN$ and $R_B = 4{,}07kN$. Let us calculate bearing life based on the maximum load. Frequency of the bearing inner ring rotation is $n = n_t \approx 14 min^{-1}$.

Y-bearing plummer block units, cast housing, grub screw locking

Dimensions					Basic load ratings		Limiting speed	Mass	Designations		
					dynamic	static	with shaft tolerance h6		Bearing unit	Housing	Bearing
d	A	H	H_1	L	C	C_0					
mm					kN		r/min	kg	-		
55	60	126	63,5	219	43,6	29	3600	3,6	**SY 55 TF**	SY 511 M	YAR 211-2F

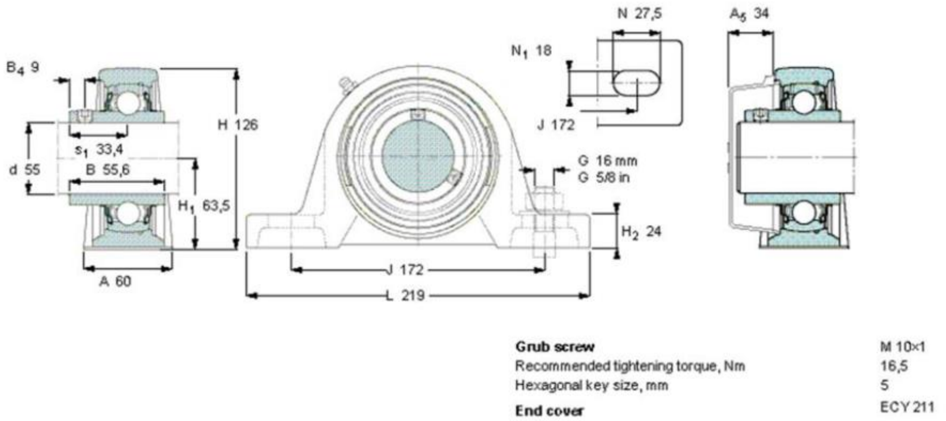


Figure 6:Bearing unit SY 55 TF

KEY CALCULATIONS:

Let us choose the keys from the Table 5 (Annex 1).

Shaft diameter d_V= 50 mm, then b = 16 mm, h= 10 mm, t_1= 6 mm, t_2 = 4.3 mm.

Hub diameter d_r = 60 mm. To optimize shaft manufacturing technology let us choose the same key for this shaft diameter as for d_V = 50 mm. Then b = 16 mm, h = 10 mm, t_1 = 6 mm, t_2= 4.3 mm.

Hub length $l_r \approx$ 90 mm. Key length should be 8…10 mm shorter. Thus, let us choose the length of the key 16x10 equal to 80 mm (Table 5, Annex 1).

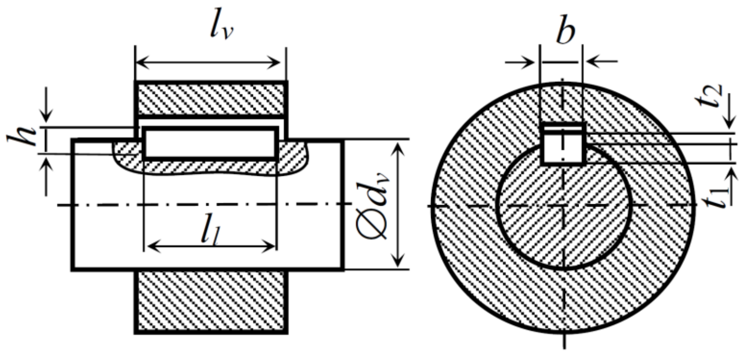


Figure 7: Prismatic key

SHAFT DRUM JOINING:

Load from the shaft to the drum is applied through two hubs and two keys.

Contact (bearing) stresses:

$$\sigma_C = \frac{2T}{2d(h - t_1)(\, l_r - b)} = \frac{2 \times 1079}{2 \times 0{,}06(0{,}01 - 0{,}0055)(\,0{,}09 - 0{,}016)} = 54MPa$$

Design stress $[\sigma_C]$ *= 130 MPa for steel hub at variable or shock loads.*

Shearing stress:

$$\tau = \frac{2T}{2d \times b(l_r - b)} = \frac{2 \times 1079}{2 \times 0{,}06 \times 0{,}016(0{,}09 - 0{,}016)} = 15{,}1MPa$$

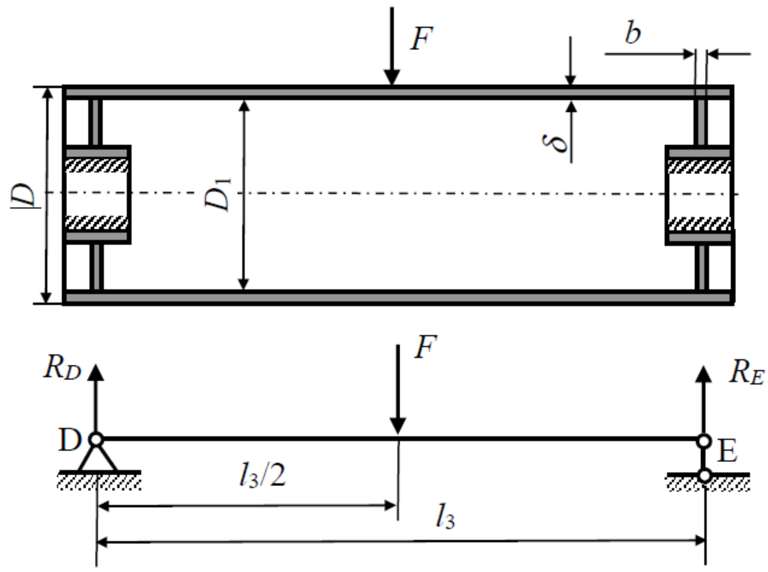


Figure 8: Drum

SHAFT-CHAIN WHEEL JOINING

Width of the chain wheel $l_n \approx 80\ mm$. Let us choose the length of the key 16x10 equal to 63 mm (Table 5, Annex 1).

Bearing stresses:

$$\sigma_C = \frac{2T}{d(h - t_1)(l_n - b)} = \frac{2 \times 1079}{0{,}06(0{,}01 - 0{,}0055)(0{,}08 - 0{,}016)} = 124MPa$$

Since the key 16x10 does not satisfy the strength condition, let us choose two keys.

Then,

$$\sigma_C = \frac{2T}{2d(h-t_1)(l_n-b)} = \frac{2 \times 1079}{2 \times 0{,}06(0{,}01-0{,}0055)(0{,}08-0{,}016)} = 62MPa$$

Strength conditions are satisfied.

Shearing stress:

$$\tau = \frac{2T}{2d \times b(l_n-b)} = \frac{2 \times 1079}{2 \times 0{,}06 \times 0{,}016(0{,}08-0{,}016)} = 17MPa$$

DRUM CALCULATIONS

Drum is made of steel *S235J2G3 EN 10025*. The design stress $\sigma \approx 120\ MPa$. Rope size $d = 10\ mm$. The thickness of the drum wall:

$$\delta \geq \frac{F_{max}}{t[\sigma]} = \frac{10791}{0.01 \cdot 120 \cdot 10^6} \approx 0{,}0089$$

$$t \approx d = 10$$

Let us $\delta = 7$ mm and $b \approx \delta$.

Drum Diameter $D = 200$ mm. Then $D_1 = D - 2\delta = 200 - 2 \times 7 = 186\ mm$

Reaction forces,

$$R_D = R_E = \frac{F}{2} = \frac{10791}{2} = 5395\ N$$

Maximum bending moment,

$$M = R_D \frac{l_3}{2} = 5395 \frac{0{,}32}{2} = 862\ Nm$$

Compressing stress,

$$\sigma_S = \frac{F_{max}}{\delta \times t} = \frac{10791}{0{,}007 \times 0{,}01} = 154\ MPa$$

Bending Stress,

$$\sigma_M = \frac{M}{W} = \frac{M}{0{,}1\frac{D^4 - D_1^4}{D}} = \frac{970}{0{,}1\frac{0{,}2^4 - 0{,}186^4}{0{,}2}} = 4{,}8\ MPa$$

Torsional stress,

$$\tau = \frac{T}{W_0} = \frac{F_{max}\frac{D}{2}}{0{,}2\frac{D^4 - D_1^4}{D}} = \frac{10791\frac{0{,}2}{2}}{0{,}2\frac{0{,}2^4 - 0{,}186^4}{0{,}2}} = 2{,}6\ MPa$$

Equivalent stress,

$$\sigma_{ekv} = \sqrt{(\sigma_s + \sigma_M)^2 + 4\tau^2} = \sqrt{(154 + 4{,}8)^2 + 4 \times 2{,}6^2} = 158\ MPa$$

The found value is bigger than 120MPa so we should change the delta value.

Let us $\delta = 10$ mm and $b \approx \delta$

Drum Diameter D = 200 mm. Then $D_1 = D - 2\delta = 200 - 2 \times 10 = 180\ mm$

Reaction forces,

$$R_D = R_E = \frac{F}{2} = \frac{10791}{2} = 5395\ N$$

Maximum bending moment,

$$M = R_D\frac{I_3}{2} = 5395\frac{0{,}32}{2} = 862\ Nm$$

Compressing stress,

$$\sigma_S = \frac{F_{max}}{\delta \times t} = \frac{10791}{0{,}01 \times 0{,}01} = 107\ MPa$$

Bending Stress,

$$\sigma_M = \frac{M}{W} = \frac{M}{0{,}1\frac{D^4 - D_1^4}{D}} = \frac{970}{0{,}1\frac{0{,}2^4 - 0{,}180^4}{0{,}2}} = 3{,}52\ MPa$$

Torsional stress,

$$\tau = \frac{T}{W_0} = \frac{F_{max}\frac{D}{2}}{0,2\frac{D^4 - D_1^4}{D}} = \frac{10791\frac{0,2}{2}}{0,2\frac{0,2^4 - 0,180^4}{0,2}} = 1,96\ MPa$$

Equivalent stress,

$$\sigma_{ekv} = \sqrt{(\sigma_s + \sigma_M)^2 + 4\tau^2} = \sqrt{(107 + 3,52)^2 + 4 \times 1,96^2} = 110\ MPa$$

REFERENCES

1- http://www.ruukki.com
2- http://www.forankra.ee/
3- http://www.sew-eurodrive.com
4- http://www.alas-kuul.ee
5- http://www.sks.fi
6- http://www.skf.com

n_1, min^{-1}	Chain pitch, mm							
	12,7	15,875	19,05	25,4	31,75	38,1	44,45	50,8
50	46	43	39	36	34	31	29	27
100	37	34	31	29	27	25	23	22
200	29	27	25	23	22	19	18	17
300	26	24	22	20	19	17	16	15
500	22	20	18	17	16	14	13	12
750	19	17	16	15	14	13	-	-
1000	17	16	14	13	13	-	-	-
1250	16	15	13	12	-	-	-	-

Note. 1) If $z_1 \neq 17$, then the number from the table should be multiplied by coefficient $k_z = 1+0,01(z_1\text{-}17)$.
2) The design pressure should be decreased 15% for double strand chain.

Figure 9: Design Pressure

n_1, min^{-1}	Chain pitch, mm							
	12,7	15,875	19,05	25,4	31,75	38,1	44,45	50,8
50	7,1	7,2	7,2	7,3	7,4	7,5	7,6	7,6
100	7,3	7,4	7,5	7,6	7,8	8,0	8,1	8,3
300	7,9	8,2	8,4	8,9	9,4	9,8	10,3	10,8
500	8,5	8,9	9,4	10,2	11,0	11,8	12,5	
750	9,3	10,0	10,7	12,0	13,0	14,0	-	-
1000	10,0	10,8	11,7	13,3	15,0	-	-	-
1250	10,6	11,6	12,7	14,5	-	-	-	-

Figure 10: Chain Safety Factor

Steel		Shaft diameter d, mm						
		20	30	40	50	70	100	120
carbon	$K_{d\sigma}$	0,92	0,88	0,85	0,82	0,76	0,70	0,61
	$K_{d\tau}$	0,83	0,77	0,73	0,70	0,65	0,59	0,52
alloy	$K_{d\sigma}$ja $K_{d\tau}$	0,83	0,77	0,73	0,70	0,65	0,59	0,52

Figure 11: Scale Factor

Shaft diameter d, mm	key cross- section, mm		hole depth, mm	
	width b, mm	height h, mm	shaft, t_1	hub, t_2
$12 < d \leq 17$	5	5	3	2,3
$17 < d \leq 22$	6	6	3,5	2,8
$22 < d \leq 30$	8	7	4	3,3
$30 < d \leq 38$	10	8	5	3,3
$38 < d \leq 44$	12	8	5	3,3
$44 < d \leq 50$	14	9	5,5	3,8
$50 < d \leq 58$	16	10	6	4,3
$58 < d \leq 65$	18	11	7	4,4
$65 < d \leq 75$	20	12	7,5	4,9
$75 < d \leq 85$	22	14	9	5,4
$85 < d \leq 95$	25	14	9	5,4
$95 < d \leq 110$	28	16	10	6,4
$110 < d \leq 130$	32	18	11	7,4

Note. Standard length of keys: 6; 8; 10; 14; 16; 18; 20; 22; 25; 28; 32; 36; 40; 45; 50; 56; 63; 70; 80; 90; 100; 110; 125; 140; 160; 180; 200; 250; 280; 315; 335; 400; 450.

Figure 12: Key Size and size of key holes