TECHNICAL UNIVERSITY OF CLUJ-NAPOCA
FACULTY OF AUTOMOTIVE, MECHATRONICS AND MECHANICAL ENGINEERING
DEPARTMENT OF MECHATRONICS AND MACHINE DYNAMICS

SEMESTER PROJECT

SINGLE-STAGE INDUSTRIAL GEARBOX

at

ELEMENTS OF MECHANICAL ENGINEERING

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Chapter 1. Introduction

1.1 Project theme

Design a single stage speed reducer gearbox based on a spur gear (an elementary gearing) that can be used for an industrial mixer. The conceptual design of the mechanical power transmission is presented in Fig. 1.1. It consists of a three-phase induction motor (1), a single stage spur gear reducer (2), two coupling element (3) and the industrial process (4). The material used for the gears is OLC 45 quality carbon steel and for the shafts OL 50 carbon steel. Each student will use their given input data to design the mechanical power transmission.

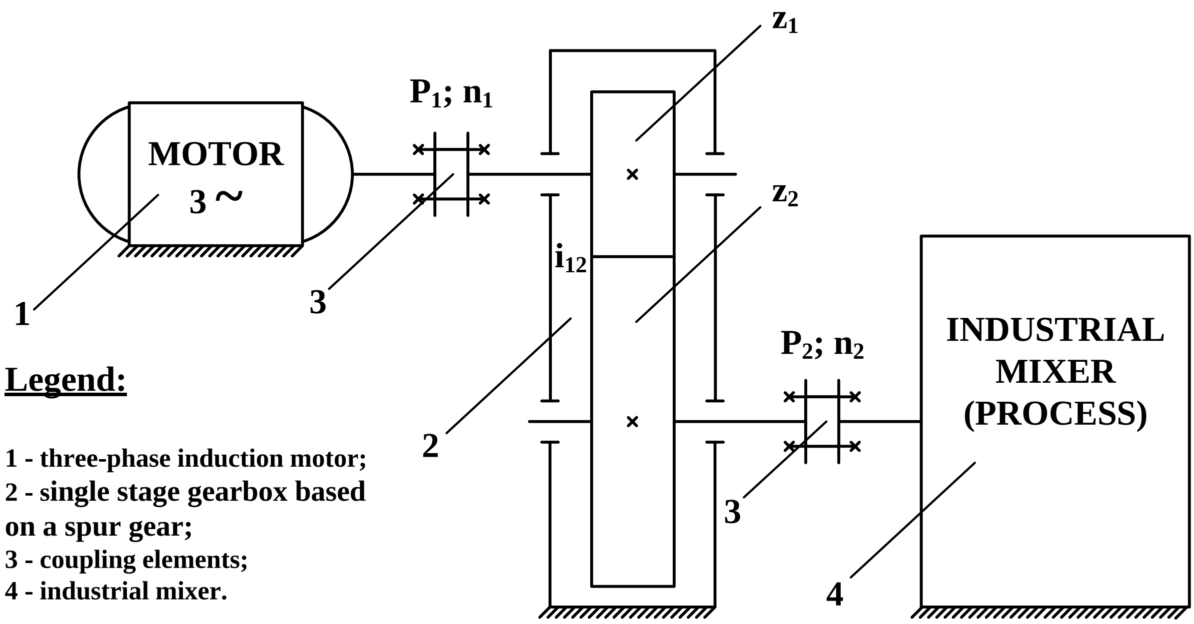

Fig. 1.1. *Kinematic diagram of the mechanical power transmission.*

<u>Input data:</u>

$n = 25 - student\ number$

$P_2 = 6{,}25\ [kw] - output\ power$

$n_s = 3000\ [RPM]$

$- \ synchronous\ speed\ of\ the\ motor$

$i_{12} = 5 - transmission\ ratio$

$z_1 = 25 - number\ of\ teeth\ for\ the\ first\ gear$

$L_h = 20000\ [hours] - \ number\ of\ running\ hours$

1.2 Gear speed reducers

The mechanical transmissions between the engine and the working machine do the following:

- increase or decrease the speed, respectively, the transmitted torque
- modify the trajectory or nature of the motion
- change the direction or plane of motion
- continuously adjust and modify the speed[1]
- combine the motion and transmitted moments from multiple motors or distribute the motion to multiple machines or working organs
- protect the machine's organs against overload.

A gear reducer is a mechanical system of gears in an arrangement such that input speed can be lowered to a slower output speed but have the same or more output torque. The operation of a gear reducer involves a set of rotating gears that are connected to a shaft with a high incoming speed, which is sent to a set of rotating gears where the speed or torque is changed. How many gears are in a gear reducer assembly is dependent on the speed requirements of the application.

Mechanical transmissions can be achieved through gearing and friction. Gear transmissions (gear wheels) with constant transmission ratio mounted in closed cases are called reducers when they reduce the speed ($i > 1$) and amplifiers when they increase the speed ($i < 1$). When they allow speed variation in steps, they are called gearboxes.

Reducers can have one, two, or more stages of reduction, built either as isolated subassemblies or as part of a machine assembly.

Depending on the relative positions of the driving and driven shafts, reducers are constructed with cylindrical gears with bevel gears and pseudoconical gears or in combinations of bevel gears or helical gears with cylindrical gears (at high transmission ratios).

Reducers, according to the type of gear engagement, can be cylindrical, bevel, helical.

Based on the positions of the gear axes, we distinguish reducers with fixed axes and reducers with movable axes (differential reducers and planetary reducers).

Gear reducers have a wide range of applications due to the advantages they offer: constant transmission ratio, the ability to handle loads from a few Newtons to very high loads, compact size and high efficiency, simple and inexpensive maintenance, etc.

As disadvantages we need to mention that they are relatively high cost, precision manufacturing and assembly, noise generation, shocks, and vibrations, etc.

Cylindrical Gearing reducers

Cylindrical gear reducers are the most widespread due to their wide range of power and transmission ratios that can be achieved with them, as well as the possibility of standardization and production in specialized factories. In practice, reducers are encountered for powers up to 100,000 kW, with peripheral speeds of the gears reaching up to 200 m/s.

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[1] I. Ştefănescu, I. Crudu, D. Panţiru, L. Palaghian, “Atlas reductoare cu roti dintate”, Editura Didactică şi Pedagogică Bucureşti 1981.

Cylindrical gear reducers are standardized and typified both in our country and in countries with developed industries. The distance between axes, transmission ratio, and main dimensions are standardized, allowing for the mass production of housings and their use in reducers with various powers and transmission ratios.

Cylindrical gear reducers can be constructed with cylindrical gears with straight, helical, or herringbone teeth, with external tooth profile, and very rarely with internal tooth profile. The choice of tooth profile depends on the peripheral speed of the gear and the purpose of the transmission.

Cylindrical gears with straight teeth are recommended for low peripheral speeds when there are no shocks and noise, in cases where axial forces in shafts and bearings are not allowed, and in gearboxes with movable gears, etc.

Cylindrical gears with helical and herringbone teeth are recommended for quiet operation and high peripheral speeds. Gears with herringbone teeth are preferably used in large-sized reducers, while gears with straight and helical teeth are used in small and medium-sized reducers. The number of stages in reducers depends on the transmission ratio "i." For single-stage reducers, r ranges from 1.2 to 6.3 (maximum 8); for two-stage reducers, i ranges from 7.1 to 56 (maximum 60); for three-stage reducers, i ranges from 40 to 180 (maximum 200).

Single-stage reducers are used for powers up to 515 kW when splash lubrication is used and up to 950 kW when forced lubrication is applied. The efficiency is 0.98 to 0.99 in the case of gears with straight or helical teeth and 0.97 to 0.98 for gears with herringbone teeth.

Gear Reducer Torque

Torque is a rotational force that is received by the gear reducer and changed into a different force and speed with the amount of power remaining the same. Gear reducers are a gear or series of gears designed to reduce the torque of a motor, which increases in direct proportion to the reduction of rotations per unit of time or revolutions per minute. This is accomplished by base mounted or shaft mounted gear reducers.

Gears are used to multiply or divide torque, which is determined by the size of the gears. The ratio of the gear sizes increases or decreases torque, which is the foundational aspect of the operation of a gear box.

1.3 Power transmission design analysis and justification

Basic Statement of the Problem

We will design a power transmission for an industrial saw that will be used to cut tubing for vehicle exhaust pipes to length prior to the forming processes. The saw will receive 25 hp from the shaft of an electric motor rotating at 1750 rpm. The drive shaft for the saw should rotate at approximately 500 rpm. Functions, Design Requirements, and Selection Criteria for the Power Transmission.

Functions

The functions of the power transmission are as follows:
1. To receive power from an electric motor through a rotating shaft.
2. To transmit the power through machine elements that reduce the rotational speed to a desired value.
3. To deliver the power at the lower speed to an output shaft which ultimately drives the saw.

Selection Criteria

There are certain factors that have to be evaluated before deciding to purchase a gear reducer. The main purpose of a gear reducer is to adapt the characteristics of torque and speed of the input and output axis of a mechanism. It is for this reason that it is necessary to understand the torque and rotational speed of the application.

Torque of a Motor

A gear reducer increases the torque of a motor and creates a new torque for the receiving application. As an assist to customers, manufacturers express maximum and minimum torque in newton meters (Nm) for their products with the torque density varying between different gear reducers.

Speed of a Motor

The second purpose of a gear reducer is to reduce motor speed, which is expressed in terms of the reduction ratio. The rotational speed of a motor is changed by the rotational ratio to produce the output rotational speed, which is described in revolutions per minute.

Gear Reducer Selection

At this stage of the selection process, an expert, engineer, or designer that is knowledgeable in gear reducers is necessary since there are so many types of gear reducers designed to meet the requirements of a wide assortment of functions and parameters. The configuration of the input and output shafts are the first criteria related to choosing the type of gear reducer.

Gear Reducer Dimensions

To decide on the dimensions of a gear reducer, it is important to choose the right shaft, which can be orthogonal, coaxial, and parallel. Each of these shaft types have a different orientation in regard to the gear reducer, which are perpendicular, aligned and parallel.

Motor Performance

In some applications, motors experience shock or cyclic loads. When choosing a gear reducer, it is important to factor in these conditions in order to allow the gear reducer to be able to deal with the increased torque.

Operational Efficiency

Choosing a gear reducer that operates efficiently is necessary regarding its cost. The correct one for an application can be beneficial in regard to long term costs.

Chapter 2. Selecting the Actuator

2.1 Selecting the AC motor

$$n_s = \frac{60 * f}{p} \ [RPM] => f = \frac{n_s * p}{60} = \frac{3000 * 1}{60} = 50 \ [Hz] \quad (1)$$

$$s = \frac{n_s - n_n}{n_s} = \frac{3000 - 2860}{3000} = 0{,}0467 \ [\mathrm{kW}] \quad (2)$$

$$P_m = \frac{P_2}{\eta} = \frac{6{,}25}{0{,}9601} = 6{,}5099 [kW] \quad (3)$$

$$\eta = \eta_g * \eta_b^2 * \eta_l = 0{,}9796 * 0{,}9950^2 * 0{,}99 = 0{,}9601 \quad (4)$$

$$P_n > P_m => P_n = 7.5 \ [kW] \quad (5)$$

$$n_n = 2860 \ [RPM] \quad (6)$$

Motoare asincrone de uz general – PERFORMANTE

Tip motor	Putere nominala [kW]	Turatie nominala [rpm]	Curent nominal (400V) A	η %	cos φ	Ip In	Mp Mn	Mmax Mn	Masa [kg]
2 poles									
ASU 63a-2	0,18	2640	0,59	58,5	0,75	4,5	2,8	3,1	9,1
ASU 63b-2	0,25	2650	0,73	66,1	0,75	4,5	3,5	3,4	10,0
ASU 71a-2	0,37	2650	0,99	66,5	0,81	4,5	2,4	2,4	12,6
ASU 71b-2	0,55	2660	1,41	68,0	0,83	4,5	2,4	2,4	14,4
ASU 80a-2	0,75	2675	1,81	73,0	0,82	4,5	2,7	2,7	17,8
ASU 80b-2	1,1	2680	2,51	76,3	0,83	4,8	2,7	2,7	19,8
ASU 90S-2	1,5	2680	3,28	78,5	0,84	5,0	2,4	2,5	26,0
ASU 90L-2	2,2	2700	4,64	81,5	0,84	5,0	2,4	2,8	28,8
ASU 100LW-2	3,0	2825	6,14	83,0	0,85	5,0	3,0	2,9	38,0
ASU 112M-2	4	2850	8,06	84,3	0,85	6,0	3,1	3,1	49,0
ASU 132Sa-2	5,5	2860	10,8	85,9	0,86	6,5	3,1	2,6	75,0
ASU 132Sb-2	7,5	2860	14,6	87,1	0,85	6,5	3,1	2,5	80,0
ASU 160Ma-2	11	2900	21,1	88,6	0,85	6,3	2,4	2,4	105
ASU 160Mb-2	15	2900	28,1	89,5	0,86	6,3	2,4	2,4	120
ASU 160L-2	18,5	2910	34,2	90,3	0,86	6,5	2,4	2,5	135
ASU 180M-2	22	2920	40,2	90,8	0,87	6,5	2,5	2,8	175
ASU 200La-2	30	2930	54,4	91,5	0,87	7,0	2,5	2,7	235
ASU 200Lb-2	37	2930	67,1	92,5	0,86	7,0	2,7	2,9	265
ASU 225M-2	45	2940	80,5	92,8	0,87	7,0	2,2	2,5	330
ASU 250M-2	55	2940	97,9	93,2	0,87	7,8	2,2	2,45	430
ASU 280S-2	75	2945	134	94,0	0,86	7,1	2,3	3,0	520
ASU 280M-2	90	2960	155	94,0	0,89	7,8	2,4	2,4	600
ASU 315S-2	110	2965	187	94,5	0,90	7,7	2,3	2,6	710
ASU 315M-2	132	2970	219	94,5	0,92	7,9	2,1	2,8	810

Figure 1. Table for choosing the actuator

2.2 Power transmission kinematics

- Determining the number of teeth of the output gear:

$$z_2 = i_{12} * z_1 = 5 * 25 = 125 \tag{7}$$

$$u_{12} = \frac{z_2}{z_1} = \frac{125}{25} = 5;\ -2.5\% \leq \frac{u_{12} - i_{12}}{i_{12}} * 100 = 0 \leq 2.5\% \tag{8}$$

- Determining the input/output shaft speed:

$$n_1 = n_n = 2860[RPM] \tag{9}$$

$$n_2 = \frac{n_1}{u_{12}} = \frac{2860}{5} = 572\ [RPM] \tag{10}$$

- Determining the power transmitted by input shaft:

$$P_1 = P_m = 6{,}5099\ [kW] \tag{11}$$

- Determining the input/output shaft torque:

$$T_1 = 95500 * \frac{P_1}{n_1} = 95500 * \frac{6{,}5099}{2860} = 217{,}3766\ [\mathrm{daN/cm^2}] \tag{12}$$

$$T_2 = 95500 * \frac{P_2}{n_2} = 95500 * \frac{6{,}25}{572} = 1043{,}4878\ [\mathrm{daN/cm^2}] \tag{13}$$

Chapter 3. Spur Gear Design

3.1 Gears tooth strength analysis and verification

Spur Gears Design

Materials:

1. **Cast irons** = fonte
2. **Brass alloys** = bronzuri (roți melcate în general)
3. **Plastic materials** = materiale plastice
4. **Cast steels** = oțeluri turnate (cele mai utilizate) – aliaje de fier cu un conținut de carbon de până la 2% și cu alte elemente de aliere
   - **Carbon steels**
     - **Plain carbon steels** – oțeluri carbon obișnuite notate cu OL plus un număr care arată rezistența minimă la rupere exprimată în [Kgf/mm2] ~ [daN/mm2].
     - **Quality carbon steels** – oțel carbon de calitate tratat termic notat cu OLC urmat de un număr care indică conținutul mediu de carbon exprimat în sutimi de procent.
   - **Alloy steels** – oțeluri de cementare (conținut redus de carbon) și oțeluri de îmbunătățire (cu conținut mediu de carbon). Simbolizarea se face cu inițialele elementelor de aliere urmate de un unmăr ce indică conținutul mediu în carbon al oțelului exprimat în sutimi de procente.

Gears Materials

1. Plain carbon steel – OL 50, OL 60, OL 70
2. Quality carbon steel a) OLC 10, OLC 15, OLC 20 (oțeluri de cementare) b) OLC 45, OLC 60 (oțeluri de îmbunătățire)
3. Alloy steel – 18 MoCN 13, 18 MoCN 06, 21 MoMC 12, 18 MC 10, 40 C 10, 35 CN 15, AISI 4140 etc.

Hardness is a characteristic of a material, not a fundamental physical property. It is defined as the resistance to indentation, and it is determined by measuring the permanent depth of the indentation. More simply put, when using a fixed force (load) and a given indenter, the smaller the indentation, the harder the material. Indentation hardness value is obtained by measuring the depth or the area of the indentation using one of over 12 different test methods.

AISI 4140 alloy steel is a chromium-, molybdenum-, and manganese-containing low alloy steel. Alloy steels are designated by AISI four-digit numbers and comprise different kinds of steels, each with a composition which exceeds the limitations of B, C, Mn, Mo, Ni, Si, Cr, and Va set for carbon steels. It has high fatigue strength, abrasion and impact resistance, toughness, and torsional strength.

Pitch Line Velocity

$$v \cong 0{,}1 * \sqrt[4]{n_1^2 * \frac{P_1}{u_{12}}} \cong 0{,}1 * \sqrt[4]{2860^2 * \frac{8{,}723288504}{5}} \cong 6{,}1463 \left[\frac{m}{s}\right] \tag{14}$$

$n_1 - pinion\ speed\ [RPM]; P_1 - transmitted\ power\ [HP](1\ kW = 1{,}34\ HP);$
$u_{12} - transmission\ ratio$

Load (service) factor

$$k = k_c * k_d = 0{,}8197 * 1{,}2000 = 0{,}9836 \tag{15}$$

$k_c - load\ distribution\ factor; k_d - load\ distribution\ factor$

Equivalent Torque

$$T_{e1} = k * T_1 = 0{,}9836 * 217{,}3766 = 213{,}8130\ [daN * cm] \tag{16}$$

Contact stress (dimensioning center/axial distance A)

$\propto = 20° - pressure\ angle$

$$\begin{aligned} A_{min} &= (u_{12} + 1) * \sqrt{\frac{T_{e1}}{\psi_A * u_{12} * sin2\alpha} * \left(\frac{865}{\sigma_{ak}}\right)^2} \\ &= (5+1) * \sqrt{\frac{213{,}8130}{0{,}5 * 5 * \sin(2 * 0{,}3490658504)} * \left(\frac{865}{\sigma_{ak}}\right)^2} \\ &= 7{,}9434 \end{aligned} \tag{17}$$

$$A_w > A_{min} => 8 > 7{,}9434 \tag{18}$$

$$m_{min} = \frac{2 * A_w}{z_1 + z_2} * \frac{cos\alpha}{cos\alpha_0} = \frac{2 * 8}{25 + 125} * \frac{\cos(0{,}3490658504)}{\cos(0{,}3490658504)} = 0{,}1067 \tag{19}$$

$$m \geq m_{min} => 0{,}12 \geq 0{,}1067 \tag{20}$$

$$A = \frac{m(z_1 + z_2)}{2} = \frac{0{,}12(25 + 125)}{2} = 9 \tag{21}$$

Axial coefficient of the gear

$B - gear\ face\ width;\ \psi_A = 0{,}3 \div 0{,}6\ (medium\ speed\ reducer)$

$$\psi_A = \frac{B}{A} = 0{,}5 \tag{22}$$

Bending stress (verification)

$$\sigma = \frac{2T_{e1}}{\pi * m^3 * z_1 * \psi_m * C_f * cos\alpha_0} = \frac{2 * 213{,}8130}{3{,}1415 * 0{,}12^3 * 25 * 37{,}5 * 0{,}1355 * \cos 20°} = 659{,}894013 \le \sigma_{ai}\ [\frac{daN}{cm^2}] \tag{23}$$

$$m_b = \sqrt[3]{\frac{0{,}68 * T_{e1}}{z_1 * \psi_m * C_f * \sigma_{ai}}} = \sqrt[3]{\frac{0{,}68 * 213{,}8130}{25 * 37{,}5 * 0{,}1355 * 1405{,}2288}} = 0{,}0934 \le m \tag{24}$$

Modular coefficient of the gear

$$\psi_m = \frac{A * \psi_A}{m} = \frac{9 * 0{,}5}{0{,}12} = 37{,}5 \tag{25}$$

Allowable bending stress level

$$\sigma_{ai} = \frac{\sigma_0}{k_\sigma * C} = \frac{4300}{1{,}7 * 1{,}8} = 1405{,}2288\ [\frac{daN}{cm^2}] \tag{26}$$

$$k_\sigma = 1{,}2 \div 2 \tag{27}$$

$$C = 1{,}5 \div 2 \tag{28}$$

$$\sigma_0 = 4300 \tag{29}$$

3.2 Final geometrical elements of the gears

$$D_{e1} = 2R_{e1} = m(z_1 + 2f_0) = 0{,}12(25 + 2 * 1) = 3{,}24 \tag{30}$$

$$D_{e2} = 2R_{e2} = m(z_2 + 2f_0) = 0{,}12(125 + 2 * 1) = 15{,}24 \tag{31}$$

$$h = m(2f_0 + \omega_0) = 0{,}12(2 * 1 + 0{,}25) = 0{,}2700 \tag{32}$$

$$D_{i1} = 2R_{i1} = m(z_1 - 2f_0 - 1\omega_0) = 0{,}12(25 - 2 * 1 - 1 * 0{,}25) = 2{,}7000 \tag{33}$$

$$D_{i2} = 2R_{i2} = m(z_2 - 2f_0 - 1\omega_0) = 0{,}12(125 - 2 * 1 - 1 * 0{,}25) = 14{,}7 \tag{34}$$

$$B_2 = A * \psi_A = 9 * 0{,}5 = 4{,}5000 \tag{35}$$

$$B_1 = B_2 + m = 4{,}500 + 0{,}12 = 4{,}6200 \tag{36}$$

$$D_{d1} = m * z_1 = 0{,}12 * 25 = 3{,}0000 \tag{37}$$

$$D_{d2} = m * z_2 = 0{,}12 * 125 = 15{,}0000 \tag{38}$$

$$R_{b1} = R_{d1} * cos \propto_0 = \frac{m * z_1}{2} * cos \propto_0 = \frac{0{,}12 * 25}{2} * \cos(0{,}3490658504) = 1{,}4095 \tag{39}$$

$$R_{b2} = R_{d2} * cos \propto_0 = \frac{m * z_2}{2} * cos \propto_0 = \frac{0{,}12 * 125}{2} * \cos(0{,}3490658504) = 7{,}04769465 \tag{40}$$

$$\varepsilon = \frac{\sqrt{R_{e1}^2 - R_{b1}^2} + \sqrt{R_{e2}^2 - R_{b2}^2} - A\cos\propto}{\pi m \cos\propto_0} = \frac{\sqrt{(1{,}6200)^2 - (1{,}4095)^2} + \sqrt{(7{,}6200)^2 - (7{,}04769465)^2} - 9\cos(0{,}3490658504)}{\pi * 0{,}12 * \cos(0{,}3490658504)} = 1{,}743447683 \tag{41}$$

$$N_1 = z_1 * \frac{\alpha_0}{180}{}^\circ + 0{,}5 = 25 * \frac{0{,}3490658504}{180} + 0{,}5 = 4{,}0000 \tag{42}$$

$$N_2 = z_2 * \frac{\alpha_0}{180}{}^\circ + 0{,}5 = 125 * \frac{0{,}3490658504}{180} + 0{,}5 = 15{,}0000 \tag{43}$$

$$inv\alpha_0 = th\alpha_0 - \alpha_0 = \tan(0{,}3490658504) - 0{,}3490658504 = 0{,}0149 \tag{44}$$

Chapter 4. Output Shaft Design

4.1 Pre-dimensioning

Pre-dimensioning the output shaft at torsional stress

$$d_{pmin} = \sqrt[3]{\frac{16 * T_2}{\pi * \tau_{at}}} = \sqrt[3]{\frac{16 * 1043{,}4878}{\pi * 150}} = 3{,}2844[cm] \qquad (45)$$

Torsional shear stress

$$\tau_{at} = 120 \div 250 \left[\frac{daN}{cm^2}\right] \qquad (46)$$

Choosing a final value for preliminary diameter d_p [mm]

$$d_p \geq d_{pmin} => d_p = 35\ [mm] \qquad (47)$$

Preliminary length of the output shaft

$$b = B_2[mm] \qquad (48)$$

$$l = b + 2 * l_1 + l_2 = 4{,}5000 + 2 * 10 + 21 = 46{,}0000[mm] \qquad (49)$$

Forces acting on a spur gear mesh

$$\eta \approx 1 => F_{t1} \approx F_{t2}; F_{r1} \approx F_{r2} \qquad (50)$$

Tangential force

$$F_{t2} = \frac{2T_2}{D_{d2}} = \frac{2 * 1043{,}4878}{3{,}000} = 695{,}6585[daN] \qquad (51)$$

Radial force

$$F_{r2} = F_{t2} tan\alpha_0 = 695{,}6585 * \tan(0{,}3490658504) = 253{,}1990 \qquad (52)$$

4.2 Shaft loading diagram

Reaction forces and bending moments in both planes

- Vertical plane

$$R_{V5} = R_{v7} = \frac{F_{t2}}{2} = \frac{695,6585}{2} = 347,8293[daN] \quad (53)$$

$$M_{iVmax} = \frac{1}{4} * F_{t2} * l = \frac{695,6585}{4} * 46,0000 = 8000,0728[daN * cm] \quad (54)$$

- Horizontal plane

$$R_{H5} = R_{H7} = \frac{F_{r2}}{2} = \frac{253,1990}{2} = 126,5995[daN] \quad (55)$$

$$M_{iHmax} = \frac{1}{4} * F_{r2} * l = \frac{253,1990}{4} * 46,0000 = 2911,7884[daN * cm] \quad (56)$$

- Resulting reaction forces

$$R_5 = R_7 = \sqrt{R_{V5}^2 + R_{H5}^2} = \sqrt{(347,8293)^2 + (126,5995)^2} = 370,1522[daN] \quad (57)$$

- Resulting bending moment

$$M_{imax} = \sqrt{M_{iVmax}^2 + M_{iHmax}^2} = \sqrt{(8000,0728)^2 + (2911,7884)^2} = 8513,4997[daN * cm] \quad (58)$$

Equivalent bending moment

$$M_{emax} = \sqrt{M_{imax}^2 + (\alpha * T_2)^2} = \sqrt{(8513,4997)^2 + (0,6 * 1043,4878)^2} = 8536,490446[daN * cm] \quad (59)$$

$$\alpha = \frac{\sigma_{aiIII}}{\sigma_{aiII}} = 0,6 \quad (60)$$

Diameter in critical section of the shaft

$$d_{min} = \frac{104}{100} * \sqrt[3]{\frac{32 * M_{emax}}{\pi * \sigma_{aiII}}} = \frac{104}{100} * \sqrt[3]{\frac{32 * 8536,490446}{\pi * 450}} = 6,0125[cm] \quad (61)$$

Final value of diameter d in critical section

$$d \geq d_{min} => d = 32 \ [mm] \quad (62)$$

4.3 Final geometry of the shaft

Diameter of the end shaft

$$d_{cmin} = \sqrt[3]{\frac{16 * T_2}{\pi * \tau_{at}}} = \sqrt[3]{\frac{16 * 1043{,}4878}{\pi * 750}} = 1{,}9207[cm] \tag{63}$$

$$\tau_{at} = (0.6 \div 0.65)\sigma_{aII} \left[\frac{daN}{cm^2}\right]; \sigma_{aII} = 1200 \left[\frac{daN}{cm^2}\right] \tag{64}$$

Choosing a final value of diameter d_c and then the length l_c

$$d_c \geq d_{cmin} => d_c = 20\ [mm] \tag{65}$$

Length of the end of the shaft

$$l_c = 50\ [mm] \tag{66}$$

Seal diameter

$$d_e = d_c + (3..5) = 20 + 5 = 25[mm] \tag{62}$$

$$D_m = 40\ [mm]; B_m = 7\ [mm] \tag{63}$$

$$l_e = (0{,}5 \dots 0{,}8) * d_p = 0{,}7 * 35 = 24{,}5\ [mm] \tag{64}$$

Bearings diameter

$$d_r = d_e + (3..5) = 30[mm] \tag{65}$$

$$D_r = 90\ [\ mm];\ B = 23\ [mm]; C = 42{,}5\ [kN]; Simbol = 6406 \tag{66}$$

Shoulder diameter and length

$$d_u = d + (5..7) = 38[mm] \tag{67}$$

$$l_u = 4..7 = 4\ [mm] \tag{68}$$

4.4 Choosing longitudinal parallel key

Required key length at contact stress

$$d = 32 => b = 10\ [mm]; h = 8\ [mm]; t_1 = 5\ [mm]; t_2 = 3{,}3\ [mm] \tag{69}$$

Dimensioning at contact stress

$$l_{min} = \frac{4 * T_2}{h * d * p_a} = \frac{4 * 1043{,}4878}{8 * 32 * 800} = 2{,}0381[cm] \tag{70}$$

$$p_a = 800 \left[\frac{daN}{cm^2}\right] \tag{71}$$

$$l_p = (0.8 \dots 0.9) * B_2 = 0.8 * 4{,}5000 = 3{,}6000[mm] \geq l_{min} \tag{72}$$

$$l_{pa} = 36 \tag{73}$$

Verification of shaft key shear stress

$$\tau_f = \frac{2 * T_2}{b * d * l_{pa}} = \frac{2 * 1043{,}4878}{10 * 32 * 36} = 0{,}1812 < \tau_{af} \left[\frac{daN}{cm^2}\right] \tag{74}$$

$$\tau_{af} = 960 \left[\frac{daN}{cm^2}\right] \tag{75}$$

Observations about l_{pb}

$$d_c = 20 => b = 6\ [mm]; h = 6\ [mm]; t_1 = 3{,}5\ [mm]; t_2 = 2{,}8\ [mm] \tag{76}$$

$$l_{min} = \frac{4 * T_2}{1.5 * h * d_c * p_a} = \frac{4 * 1043{,}4878}{1{,}5 * 6 * 20 * 800} = 2{,}8986\ [cm] \tag{77}$$

$$p_a = 800 \left[\frac{daN}{cm^2}\right] \tag{78}$$

$$l_{pb} = 28 \tag{79}$$

$$\tau_f = \frac{2 * T_2}{1.5 * b * d_c * l_{pb}} = \frac{2 * 1043{,}4878}{1{,}5 * 6 * 20 * 28} = 0{,}6211 < \tau_{af} \left[\frac{daN}{cm^2}\right] \tag{80}$$

$$\tau_{af} = 960 \left[\frac{daN}{cm^2}\right] \tag{80}$$

4.5 Verification of shaft deflection and critical speed

Check results of calculation (deflection)

- Deflection in vertical plane

$$f_V = \frac{F_{t2} * l^3}{48 * E * I} = \frac{695,6585 * 46^3}{48 * 2100000,0000 * 51471,8540} = 0,0000131[cm] \quad (81)$$

$$E = 2.1 * 10^6 \left[\frac{daN}{cm^2}\right] \quad (82)$$

$$I = \pi * \frac{d^4}{64} = \pi * \frac{32^4}{64} = 51471,8540[cm] \quad (83)$$

- Deflection in horizontal plane

$$f_H = \frac{F_{r2} * l^3}{48 * E * I} = \frac{253,1990 * 46^3}{48 * 2100000,0000 * 51471,8540} = 0,0000048[cm] \quad (84)$$

$$E = 2.1 * 10^6 \left[\frac{daN}{cm^2}\right] \quad (85)$$

$$I = \pi * \frac{d^4}{64}[cm] \quad (86)$$

$$f = \sqrt{f_v^2 + f_h^2} = \sqrt{(0,0000131)^2 + (0,0000048)^2} = 0,000013888 < f_{adm}[cm] \quad (87)$$

$$f_{adm} = 5 * 10^{-3} * m = 0,0006[cm] \quad (88)$$

Check results of calculation (vibrations)

- Weight of the output gear

$$G = \gamma * V = \gamma * \frac{\pi * D_d^2}{4} * B_2 = 0,0078 * 795,2156 = 6,2027[daN] \quad (89)$$

$$\gamma = 7.8 * 10^{-3} \left[\frac{daN}{cm^3}\right] \quad (90)$$

- Static deflection

$$f_{st} = \frac{G * l^3}{48 * E * I} = \frac{6,2027 * 46^3}{48 * 2100000,0000 * 51471,8540} = 0,0000[cm] \quad (91)$$

$$E = 2.1 * 10^6 \left[\frac{daN}{cm^2}\right] \tag{92}$$

$$I = \pi * \frac{d^4}{64} = 51471{,}8540 \text{ [cm]} \tag{93}$$

Critical speed

$$n_{cr} = \frac{30}{\pi} * \sqrt{\frac{g}{f_{st}}} = \frac{30}{\pi} * \sqrt{\frac{981{,}0000}{0{,}0000}} = 876787{,}9091[RPM] \tag{94}$$

$$g = 981 \left[\frac{cm}{s^2}\right] \tag{95}$$

$$n_2 < n_{cr} \tag{96}$$

Chapter 5. Rolling Bearing selection

5.1 Dynamic basic bearing loas rating C

Rated life L equation (Lundberg & Palmgren)

$$L = \left(\frac{C}{F_e}\right)^p = \frac{60 * n_2 * L_h}{10^6} = \frac{60 * 572{,}0000 * 20000}{10^6} = 686{,}4000[\text{million of rotations}] \tag{97}$$

Equivalent bearing load F_e

$$F_e = X * V * F_r + Y * F_a [daN] \tag{98}$$

$$F_e = F_r [daN] \tag{99}$$

Equivalent bearing load

$$F_e = F_r = R_5 = R_7 = \sqrt{R_{V5}^2 + R_{H5}^2} = 370{,}1522[\text{daN}] \tag{100}$$

Selection of ball bearing

$$L = \frac{60 * n_2 * L_h}{10^6} [million\ of\ rotations];\ F_e = F_r [daN] \tag{101}$$

$$\frac{C_{min}}{F_e} = L^{0.33} => C_{min} [daN] \tag{102}$$

$$C_{min} = F_e * L^{0.33} = (686{,}4000)^{0{,}33} * 370{,}1522 = 31{,}9485[\text{kN}] \tag{103}$$

$$C = 42{,}5\ kN \geq C_{min} \tag{104}$$

5.2 Bearing selection from SKF catalogue

$$F_e = F_r = R_5 = R_7 = \sqrt{R_{V5}^2 + R_{H5}^2} = 370{,}1522[\text{daN}] \tag{105}$$

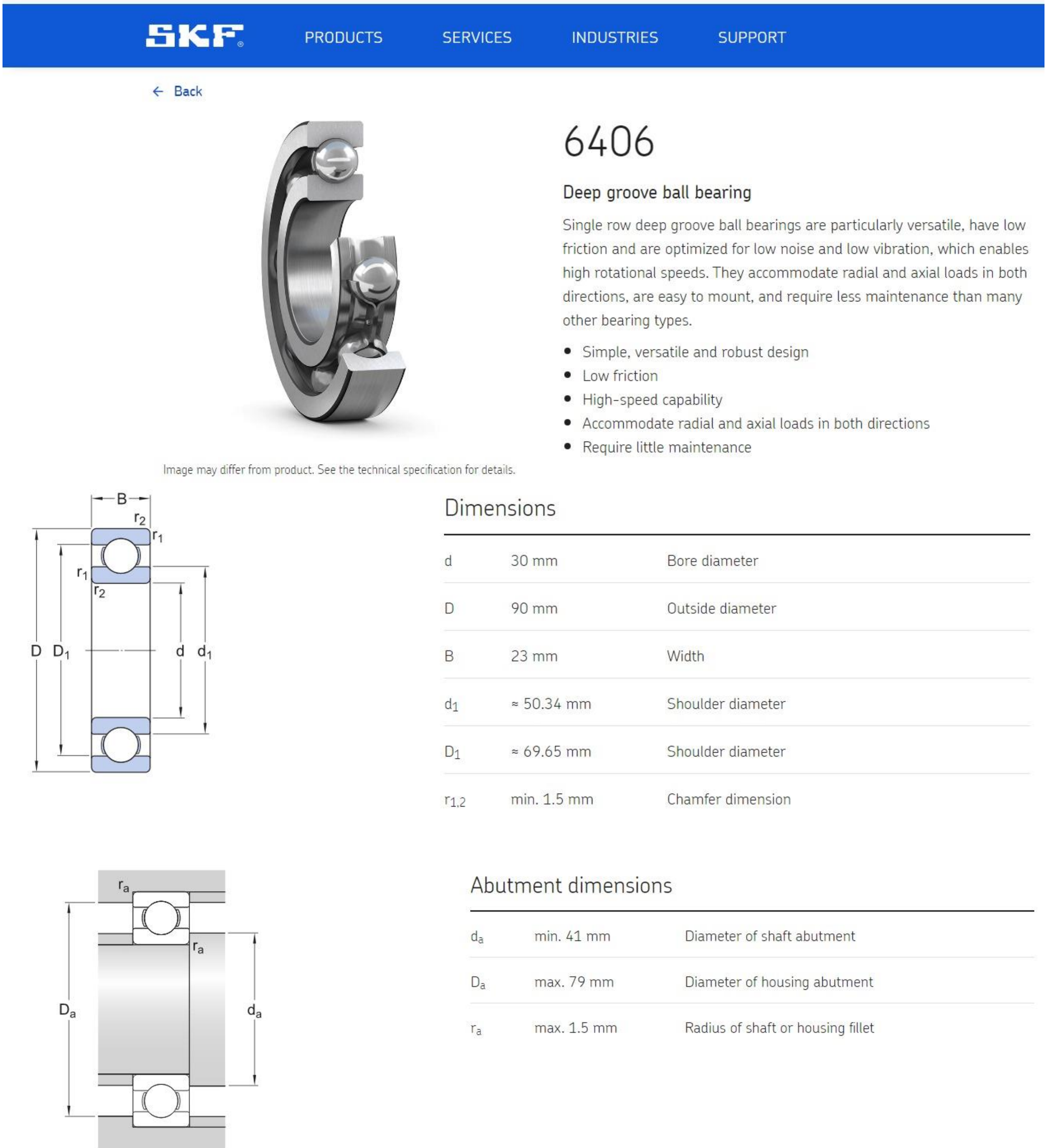
$$n = 3000[RPM] \tag{106}$$

$$L_h = 20.000[hours] \tag{107}$$

$$L = \frac{60 * n_2 * L_h}{10^6} = 686{,}4000[million\ of\ rotations] \tag{108}$$

$$C_{min} = L^{0{,}33} * F_e = 31{,}9485[kN] \tag{109}$$

5.3 Technical specification of the selected bearing



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6406

Deep groove ball bearing

Single row deep groove ball bearings are particularly versatile, have low friction and are optimized for low noise and low vibration, which enables high rotational speeds. They accommodate radial and axial loads in both directions, are easy to mount, and require less maintenance than many other bearing types.

- Simple, versatile and robust design
- Low friction
- High-speed capability
- Accommodate radial and axial loads in both directions
- Require little maintenance

Image may differ from product. See the technical specification for details.

Dimensions

d	30 mm	Bore diameter
D	90 mm	Outside diameter
B	23 mm	Width
d_1	≈ 50.34 mm	Shoulder diameter
D_1	≈ 69.65 mm	Shoulder diameter
$r_{1,2}$	min. 1.5 mm	Chamfer dimension

Abutment dimensions

d_a	min. 41 mm	Diameter of shaft abutment
D_a	max. 79 mm	Diameter of housing abutment
r_a	max. 1.5 mm	Radius of shaft or housing fillet

Figure 2. Table for choosing the bearing 6406

Calculation data

Basic dynamic load rating	C	43.6 kN
Basic static load rating	C_0	23.6 kN
Fatigue load limit	P_u	1 kN
Reference speed		18 000 r/min
Limiting speed		11 000 r/min
Minimum load factor	k_r	0.035
Calculation factor	f_0	12.1

Mass

Mass bearing	0.75 kg

Figure 3. Table for choosing the bearing 6406

REFERENCES

[1] I. Ștefănescu, I. Crudu, D. Panțiru, L. Palaghian, "Atlas reductoare cu roti dintate", Editura Didactică și Pedagogică București 1981.
[2] Industrial Quick Search, https://www.iqsdirectory.com/articles/gearbox/gear-reducers.html.

ANNEXES

A1 - Technical drawing of the output shaft