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

SEMESTER PROJECT

SINGLE-STAGE SPEED REDUCTION 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.

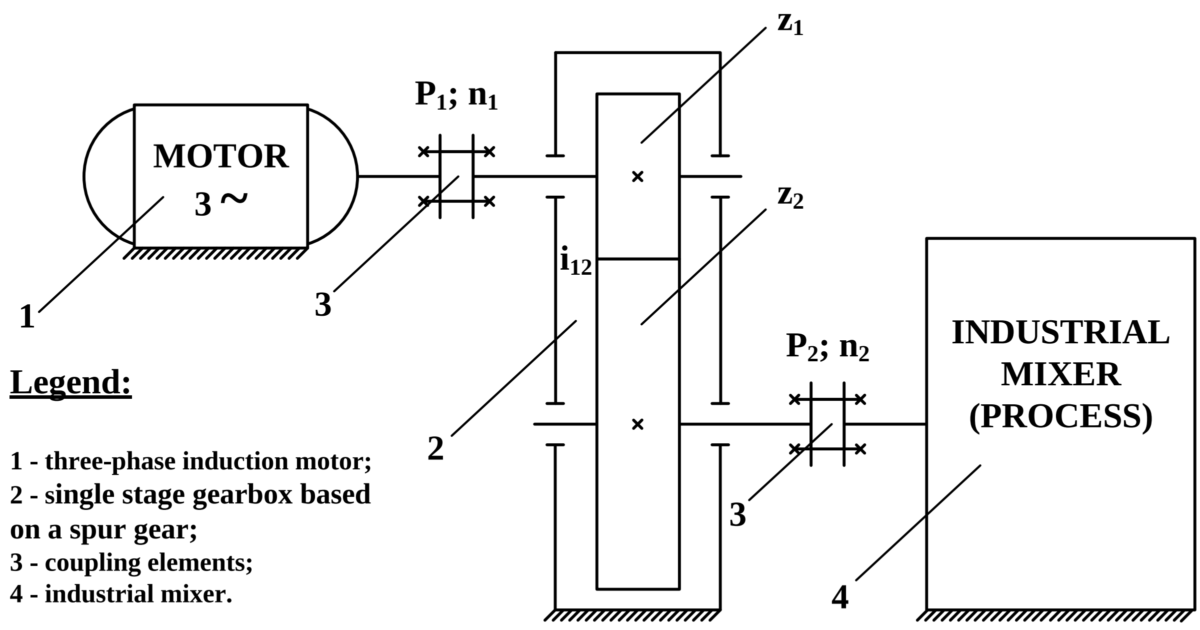


Fig. 1.1. *Kinematic diagram of the mechanical power transmission.*

<u>Input data:</u>

P_2 = 3.4 [kW] – output power

n_s = 3000 [RPM] – synchronous speed of the motor

i_{12} = 3.1 (transmission ratio)

z_1 = 22 (number of teeth of the input gear)

L_h = 10000 [hours] - number of running hours

1.2 Gear speed reducers

Speed reducers are independent subassemblies, consisting of gears mounted in closed housings. Due to their widespread, some types of reducers are standardized. [1]

Speed reducers are fairly simple pieces of machinery. A speed reducer is simply a gear train between the motor and the machinery that is used to reduce the speed with which power is transmitted. Speed reducers, also called gear reducers, are mechanical gadgets by and large utilized for two purposes. Gear reducers essential use is to duplicate the measure of torque produced by an information power source to expand the measure of usable work. [2]

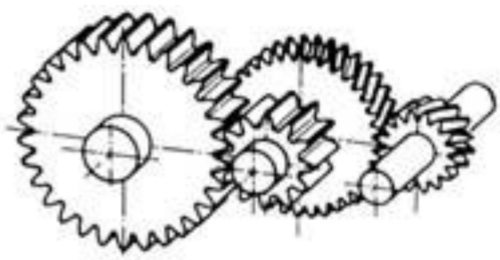


Fig. 1.2. *Industrial gear, speed reducer [3]*

How does a speed reducer increase the torque while decreasing the speed? The output gear of a speed reducer has more teeth than the input gear. So, while the output gear might rotate more slowly, reducing the speed of the input, the torque is increased.

So, to sum up, they take an input power source and increase the torque while decreasing the speed.

Speed reducers come in many shapes and sizes, but some of the most commonly found speed reducers are gearboxes. The output gear of a speed reducer has more teeth than the input gear. So, while the output gear might rotate more slowly, reducing the speed of the input, the torque is increased.

Speed reducers come in many shapes and sizes, but some of the most commonly found speed reducers are gearboxes.

For general purpose reducers, their main dimensions are standardized (STAS 6850-69 for cylindrical gearboxes, Stat 7026 -69 for cylindrical worm gearboxes).

In STAT 6848-63, it is given the symbolization of the general speed reducers with 1 to 3 stages formed by cylindrical: cylindrical-conical, cylindrical-worm and planetary gears.

One-stage cylindrical speed reducers achieve a transmission ratio i = 2: 6.3; those with two steps i = 7.1: 40; those with three stages i = 45: 280. A deviation from the nominal transmission ratio of $\pm$2.5% is allowed for those with one step, respectively $\pm$3% for those with two or three steps.

One-stage worm gearboxes achieve transmission ratios i = 8: 63, which allows a deviation of $\pm$3%.

The following figure (Fig.1.3) shows some constructive types of reducers:

a. with a step with cylindrical wheels with straight or inclined teeth
b. with a tapered gear with straight, inclined or curved teeth
c. with a single gear with cylindrical worm gear
d. with two steps with cylindrical wheels with straight or inclined teeth
e. with two steps with conical and cylindrical gears

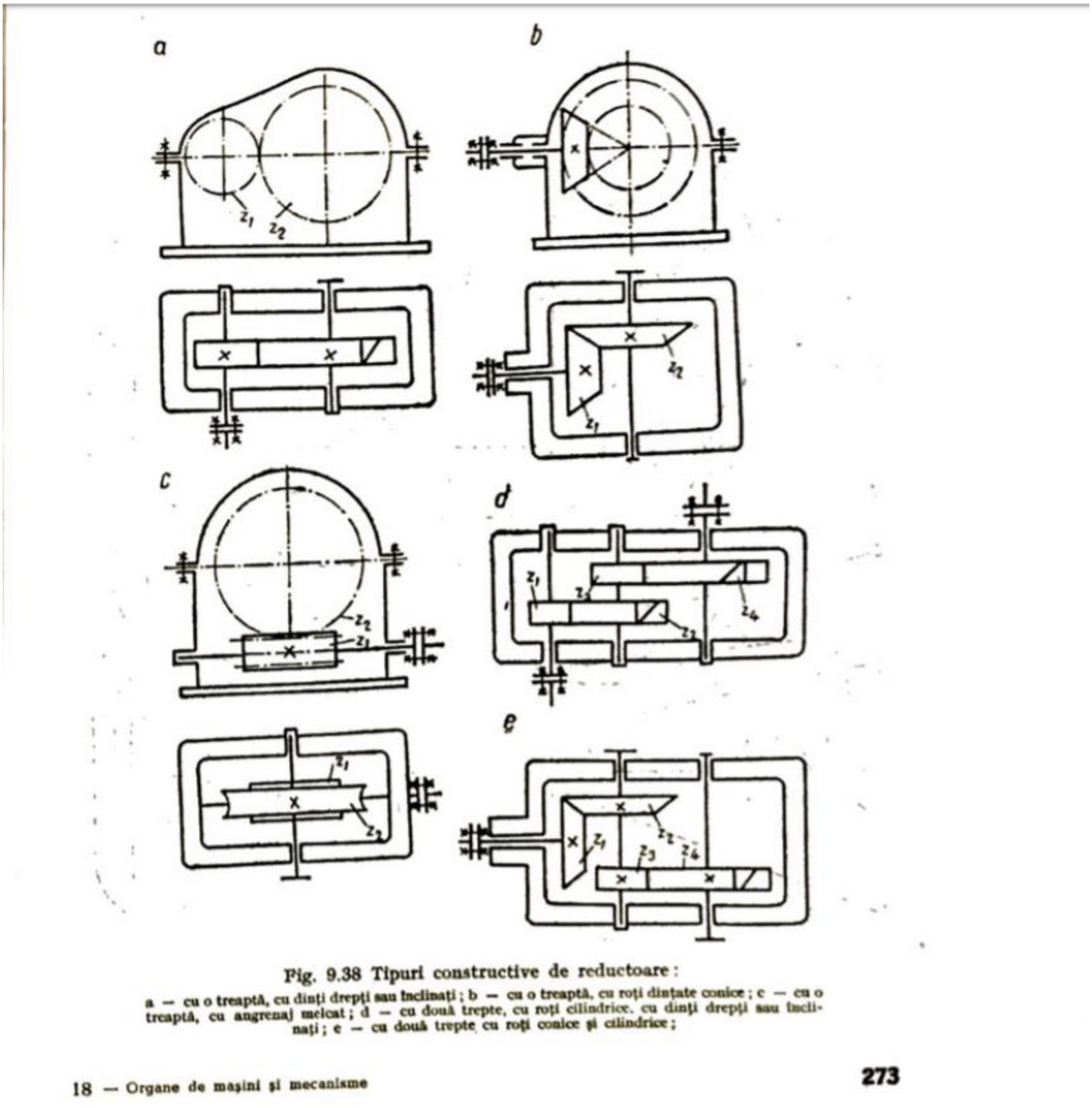


Fig. 1.3. *Constructive types of speed reducers [4]*

1.2.1 The elements of a speed reducer

The main elements of a speed reducer are: body of reducer (1), reducer cover (2), gears (3), gearbox bearings (4), the shafts on which the gears are mounted (5), the bearing caps (6), the wheel mounting pins on the shafts (7), the lifting rings (8), the inspection cover (9), the spacer bushes (10), the hole oil drain (11), oil level indicator (12), clamping screws (13), centering pins (14), seals (15), bearing collection and lubrication channel (16). [5]

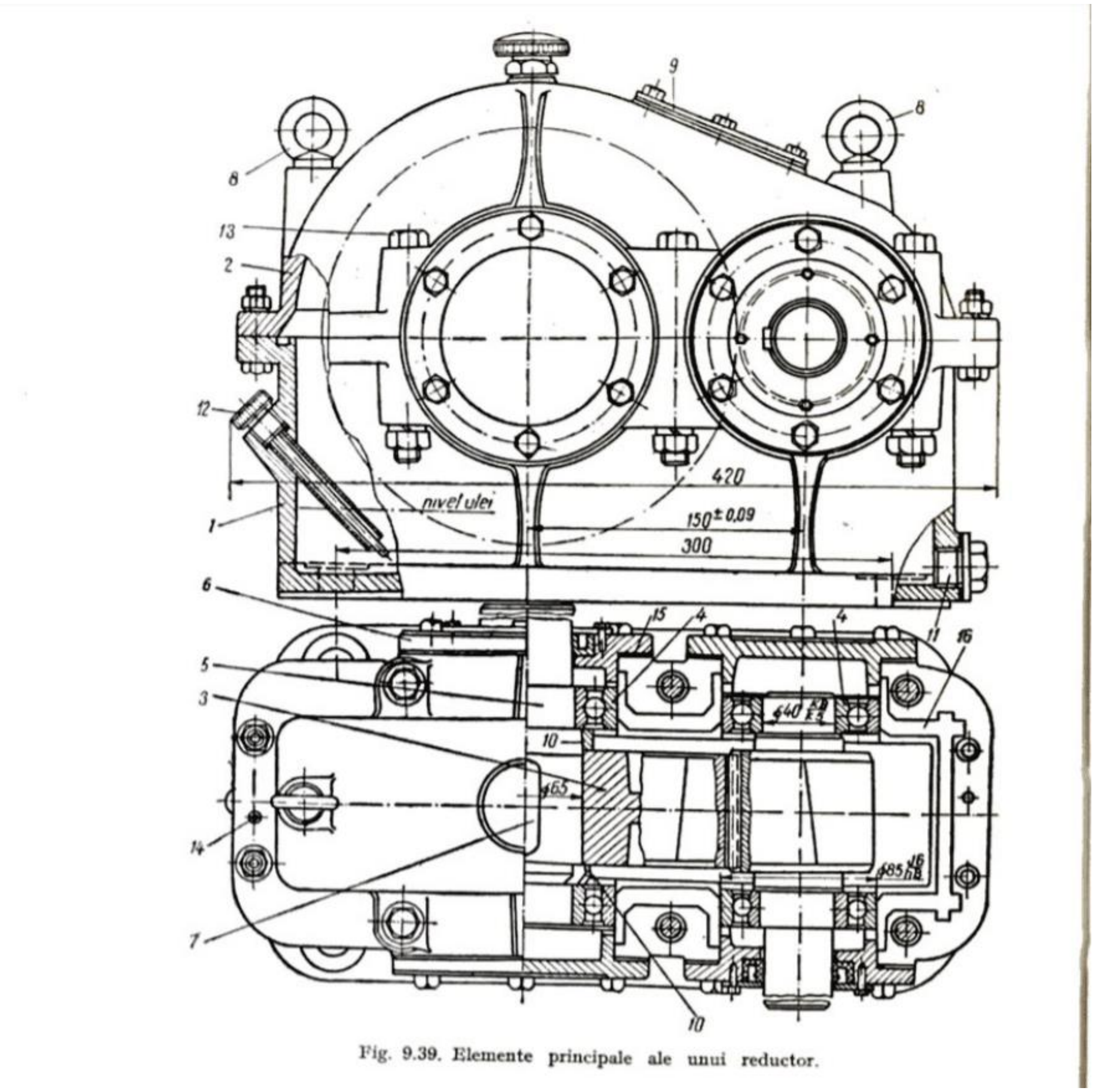


Fig. 1.4. *Main elements of a reducer [6]*

1.2.2 Type of Gears

There are many types of gears such as spur gears, helical gears, bevel gears, worm gears, gear rack, etc. These can be broadly classified by looking at the positions of axes such as parallel shafts, intersecting shafts and non-intersecting shafts. [7]

Spur Gear
Gears having cylindrical pitch surfaces are called cylindrical gears. Spur gears belong to the parallel shaft gear group and are cylindrical gears with a tooth line which is straight and parallel to the shaft. Spur gears are the most widely used gears that can achieve high accuracy with relatively easy production processes.

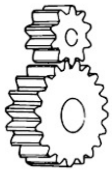


Fig. 1.5. *Example of spur gear [8]*

Helical Gear
Helical gears are used with parallel shafts similar to spur gears and are cylindrical gears with winding tooth lines. They have better teeth meshing than spur gears and have superior quietness and can transmit higher loads, making them suitable for high speed applications. When using helical gears, they create thrust force in the axial direction, necessitating the use of thrust bearings.

Fig. 1.6. *Example of helical gear [9]*

Gear Rack
Same sized and shaped teeth cut at equal distances along a flat surface or a straight rod is called a gear rack. A gear rack is a cylindrical gear with the radius of the pitch cylinder being infinite. By meshing with a cylindrical gear pinion, it converts rotational motion into linear motion.

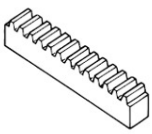


Fig. 1.7. *Example of gear rack [10]*

Bevel Gear
Bevel gears have a cone shaped appearance and are used to transmit force between two shafts which intersect at one point (intersecting shafts). A bevel gear has a cone as its pitch surface and its teeth are cut along the cone.

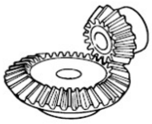


Fig. 1.8. *Example of bevel gear [11]*

Screw Gear
Screw gears are a pair of same hand helical gears with the twist angle of 45° on non-parallel, non-intersecting shafts. Because the tooth contact is a point, their load carrying capacity is low and they are not suitable for large power transmission.

Fig. 1.9. *Example of screw gear [12]*

Worm Gear
A screw shape cut on a shaft is the worm, the mating gear is the worm wheel, and together on non-intersecting shafts is called a worm gear. Worms and worm wheels are not limited to cylindrical shapes. There is the hour-glass type which can increase the contact ratio, but production becomes more difficult. Due to the sliding contact of the gear surfaces, it is necessary to reduce friction. For this reason, generally a hard material is used for the worm, and a soft material is used for worm wheel.

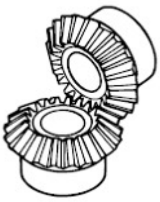


Fig. 1.10. *Example of worm gear [13]*

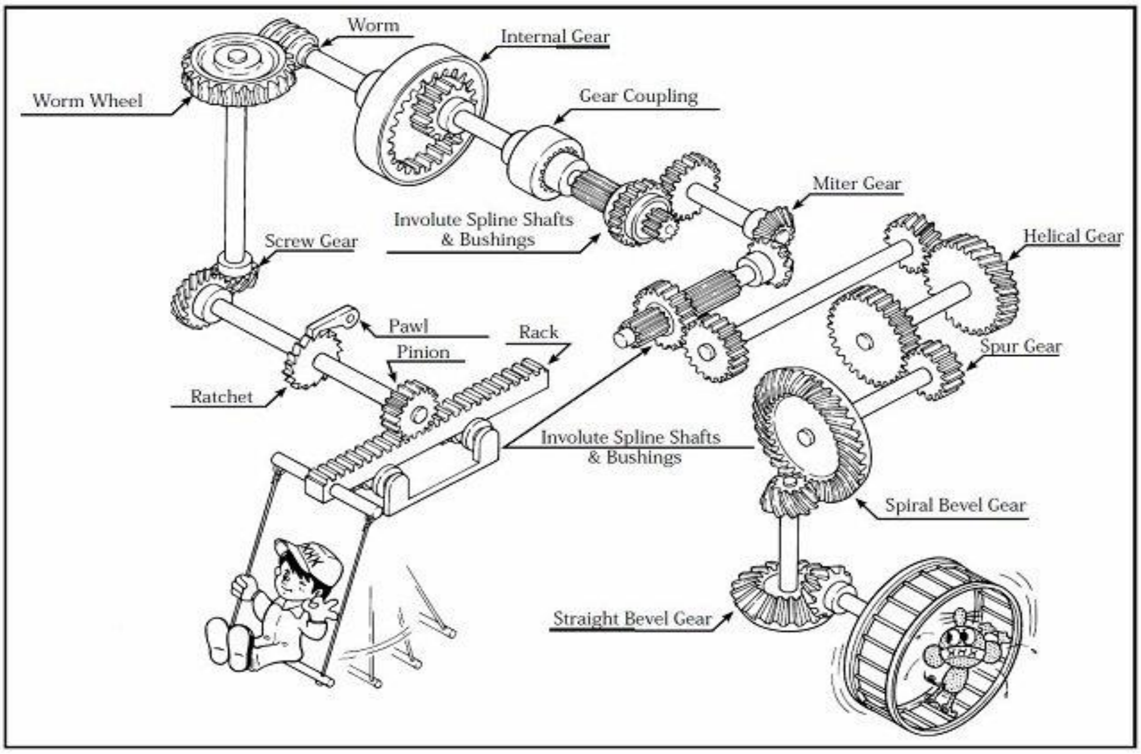


Fig. 1.11. *An overview of gears (Important Gear Terminology and Gear Nomenclature) [14]*

1.3 Power transmission design analysis and justification

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 gear.

Selection Criteria

The list of criteria should be produced by an interdisciplinary team composed of people having broad experience with the market for and use of such equipment. The details will vary according to the specific design. As an illustration of the process, the following criteria are suggester for the present design:

1. Safety: The speed reducer should operate safely and provide a safe environment for people near the machine.
2. Cost: Low cost is desirable so that the saw appeals to a large set of customers.
3. High reliability
4. Low maintenance
5. Small size
6. Smooth operation, low noise, low vibration.

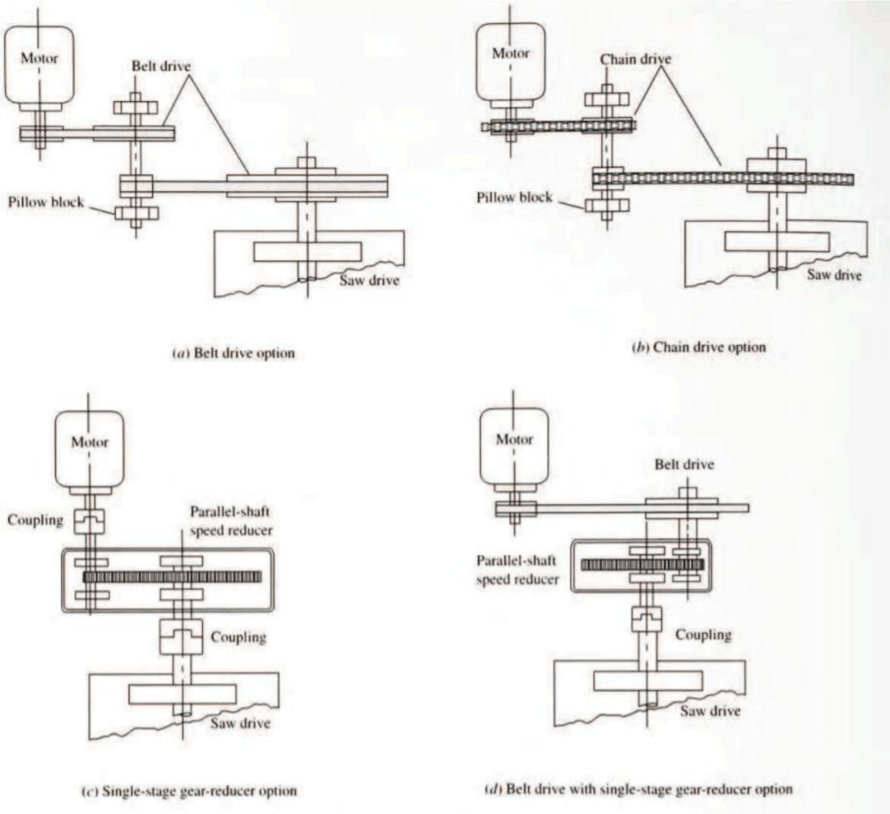


Fig. 1.12. *Options for speed reduction for saw drive design project [15]*

The are many ways that the speed reduction for the saw can be accomplished. Fig. 1.12. show four possibilities : (a) belt drive, (b) chain drive, (c) gear-type drive connected through flexible couplings, and (d) gear-type drive with a belt drive on the input side and connected to the saw with a flexible coupling.

Design Alternatives for the gear-type reducer:

1. **Single-reduction spur gears**: The nominal ratio of 3.50:1 is reasonable for a single pair of gears. Spur gears produce only radial loads which simplify selection of the bearings that support the shafts. Efficiency should be greater than 95% with reasonable precision of the gears, bearing and seals.
   Spur gears are relatively in-expensive to produce. Shafts would be parallel and should be fairly easy to align with the motor and the drive shaft for the saw.

2. **Single-reduction helical gears**: These gears are equally practical as spur gears. Shaft alignment is similar. A smaller size should be possible because of the greater capacity of helical gears. Axial thrust loads would be created which must be accommodated by the bearings and the housing. Cost is likely to be somewhat higher.

3. **Bevel gears**: These gears produce a right-angle drive which may be desirable. They are somewhat more difficult to design and assemble to achieve adequate precision.

4. **Worm and worm gear drive:** This drive produces a right-angle drive. It is typically used to achieve a higher reduction ratio than 3.5:1. Efficiently is usually much lower than the 95% called for in the design requirements. [16]

Chapter 2. Selecting the Actuator

2.1 Selecting the AC motor

To choose the right engine, we will need to calculate synchronous speed, slip, actuator power and mechanical transmission efficiency.

Synchronous speed

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

$n_s = 3000\ [RPM]$
f - motor supply's frequency in hertz
p - number of pairs of magnetic poles

$$f = \frac{n_s * p}{60} = \frac{3000*1}{60} = 50\ [Hz] \quad (2)$$

Slip

$$s = \frac{n_s - n_n}{n_s}\ , where\ n_n < n_s \quad (3)$$

n_s = synchronous speed [RPM]
n_n = rotor nominal speed [RPM]

Actuator power

$$P_m = \frac{P_2}{\eta} = \frac{3.4}{0.93} = 3.65\ [kW] \quad (4)$$

$P_m - required\ motor\ power\ [kW]$ $P_2 - output\ power\ [kW]$
$\eta -$ mechanical transmission efficiency

Mechanical transmission efficiency

$\eta_g - spur\ gear\ efficiency = 0.96$ $\eta_b = 0.99$ $\eta_l = 0.99$

$\eta_b - one\ pair\ of\ bearings\ efficiency = 0.99$

$\eta_l - lubrication\ efficiency = 0.99$

$$\eta = \eta_g * \eta_b^2 * \eta_l = 0.93 \quad (5)$$

Table 2.1. *Asynchronous Engine performance [17]*

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

From the Table 2.1., we choose $P_n = 4[KW]$ $n_n = 2850[RPM]$ $\rightarrow$ $ASU - 112\,M - 2$

Motoare asincrone seria ASU - IM 1001 (IMB3)

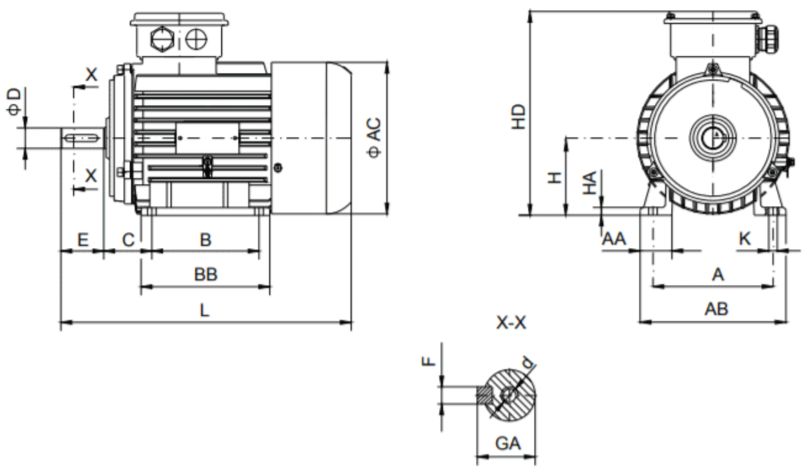


Fig. 2.2. *ASU- IM 1001 (IMB3) asynchronous Engine [18]*

Table 2.3. *Type of engine to choose [19]*

Gabarit	A	B	C	H	K	D		E	F h9	GA	d	AA	AB	BB	HA	AC	HD	L
						Nom.	Tol.											
63	100	80	40	63	7	11	j6	23	4	12.5	M4	31	131	104	9	125	162	258
71	112	90	45	71	7	14	j6	30	5	16	M5	37	141	125	9	140	182	295
80	125	100	50	80	10	19	j6	40	6	21.5	M6	35	155	125	9	158	216	287
90S	140	100	56	90	10	24	j6	50	8	27	M8	37	170	150	9	177	238	339
90L	140	125	56	90	10	24	j6	50	8	27	M8	37	170	150	9	177	238	339
100LW	160	140	63	100	12	28	j6	60	8	31	M10	47	200	176	10	199	257	387
100LX	160	140	63	100	12	28	j6	60	8	31	M10	47	200	176	10	199	257	387
112M	190	140	70	112	12	28	j6	60	8	31	M10	55	224	176	12	221	284	406
132S	216	140	89	132	12	38	k6	80	10	41	M12	68	264	220	14	263	333	496
132M	216	178	89	132	12	38	k6	80	10	41	M12	68	264	220	14	263	333	496

From Tabel 2.3., we can observe the dimensions for our engine 112M.

2.2 Power transmission kinematics

Determining the number of teeth of the output gear:

$$z_2 = i_{12} * z_1 = 3.1 * 22 = 68\ [number\ of\ teeths] \qquad (6)$$

Therefore, a new transmission ratio u with the new z2 value can be calculated as:

$$u = \frac{z_2}{z_1} = \frac{68}{22} = 3.09 \qquad (7)$$

In this situation the following condition must be meet:

$$-3\% \leq \frac{u - i_{12}}{i_{12}} * 100 = \frac{3.09 - 3.1}{4.2} * 100 = -0.322 \leq 3\% \qquad => i_{12} = u = 3.09 \qquad (8)$$

Determining the I/O shaft speed

$$n_1 = n_n = 2850\ [RPM] \qquad (9)$$

$$n2 = \frac{n_1}{i_{12}} = \frac{2850}{3.09} = 922.33\ [RPM] \qquad (10)$$

Determining the power transmitted by Input Shaft

$$P_1 = P_m = 3.65\ \ [KW] \qquad (11)$$

Determining the I/O Shaft Torque

$$M_{t_1} = 95500 * \frac{P_1}{n_1} = 95500 * \frac{3.65}{2850} = 122.3 \qquad [daN * cm] \qquad (12)$$

$$M_{t_2} = 95500 * \frac{P_2}{n_2} = 95500 * \frac{3.4}{922.33} = 352.04 \quad [daN * cm] \qquad (13)$$

Chapter 3. Spur Gear Design

3.1 Gears tooth strength analysis and verification

The two primary failure modes for gears are:

(1) Tooth Breakage - from excessive bending stress,

(2) Surface Pitting/Wear - from excessive contact stress. [20]

Hardness

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.

$$HB = 0.102 * \frac{2*F}{\pi * D(D - \sqrt{D^2 - d^2})} \tag{14}$$

Table 3.1. *The values of the hardness of some materials used in the construction of the gears and the resistances to the contact pressure [21]*

OLC 45	HB = 197	26 HB	Normalizat
	HRC = 45 ÷ 55 (călit)	190 HRC	Îmbunătățit

We choose HB = 197 for OLC 45 (quality carbon steel - gears material) from ***Table 3.1.[21]***

Computational load - F_c (force), M_{tc} (torque), P_c (power)

It can be a force, torque or power which depends on the accuracy of the gears.

Load factor

$$k_c = distribution\ load\ factor = 1.1$$

$$k_d = dynamic\ load\ factor = 1.2$$

$$k = k_c * k_d = 1.1 * 1.2 = 1.32 \tag{15}$$

Pinion peripheral speed

$$v = 0.1 * \sqrt[4]{n_1^2 * \frac{P_1}{i_{12}}} = 0.1 * \sqrt[4]{2850^2 * \frac{4.89}{3.09}} = 5.98 \qquad [\frac{m}{s}] \tag{16}$$

where
$n_1 - pinion\ speed\ [RPM]$
$P_1 - transmitted\ power\ in\ horse\ power\ [HP]\ (1kW = 1.34\ HP)$
$i_{12} - transmission\ ratio$

Computational load

$$M_{t_{c1}} = k * M_{t_1} = 1.32 * 122.3 = 161.436 \qquad [daN * cm] \tag{17}$$

Contact Stress (Dimensioning center/axial distance A)

Axial coefficient of the gear

$$\psi_A = \frac{B}{A} = \ (small\ speed\ reducer) \qquad \psi_A = 0.45\ (0.3{:}\,0.6) \qquad HB = 197 \quad [OLC\ 45] \tag{18}$$

α (pressure angle) = 20°

$$\sigma_{ak} = 26 * HB = 26 * 197 = 5122 \left[\frac{daN}{cm^2}\right] \tag{19}$$

$$A = (i_{12} + 1) * \sqrt[3]{\frac{M_{t_{c1}}}{\psi_A * i_{12} * \sin(2 * \alpha)} * \left(\frac{865}{\sigma_{ak}}\right)^2} \tag{20}$$

$$= (3.09 + 1) * \sqrt[3]{\frac{161.436}{0.45 * 3.09 * 0.643} * \left(\frac{865}{5122}\right)^2} = 6.99 \approx 7 \qquad [cm]$$

$A = 70 \qquad [mm]$

Module m [mm]

$$m = 2 * \frac{A}{z1 + z2} * \frac{\cos(\alpha)}{\cos(\alpha_0)} = 2 * \frac{70}{22 + 68} = 1.55 \ \approx 1.75^* \qquad [mm] \tag{21}$$

$m =^* 1.75 \quad [mm]$ We choose m from Annex 2.

Centre distance A[mm]

$$A = m * \frac{z_1 + z_2}{2} = 1.75 * \frac{22 + 68}{2} = 80 \qquad [mm] \tag{22}$$

Modular coefficient of the gear

$$\psi_m = \frac{A * \psi_A}{m} = \frac{80 * 0.45}{1.75} = 20.57 \;\; (8 \div 40 - for\ gearbox) \tag{23}$$

Shape coefficient C_f of the gears

$$C_f = 0.13 \tag{24}$$

Bending Stress (verification)

$$\sigma = \frac{2 * M_{t_{c1}}}{\pi * m^3 * z_1 * \psi_m * C_f * \cos(\alpha_0)}$$

$$= \frac{2 * 161.43}{\pi * (0.175)^3 * 22 * 20.57 * 0.13 * 0.93} = 350.5 \qquad [\frac{daN}{cm^2}] \tag{25}$$

Allowable Bending Stress

$$\sigma_{ai} = \frac{\sigma_0}{k_\sigma * C} = \frac{4300}{1.8 * 1.8} = 1327.2 \qquad [\frac{daN}{cm^2}] \tag{26}$$

$$\sigma_0 = 43 \qquad \left[\frac{daN}{mm^2}\right]$$

Concentration coefficient

$$k_\sigma = 1.8\ (1.2 \div 2)$$

Safety Coefficient

$$C = 1.8\ (1.5 \div 2)$$

3.2 Final geometrical elements of the gears

m = 1.75 [mm]

$$D_{e1} = 2 * R_{e1} = m * (z_1 + 2 * f_0) = 1.75 * (22 + 2 * 1) = 42 \qquad [mm] \tag{27}$$

$$D_{e2} = 2 * R_{e2} = m * (z_2 + 2 * f_0) = 1.75 * (68 + 2 * 1) = 122.5 \qquad [mm] \tag{28}$$

$$h = m * (2 * f_0 + \omega_0) = 1.75 * (2 * 1 + 0.25) = 4 \qquad [mm] \tag{29}$$

$$D_{i1} = 2 * R_{i1} = m * (z_1 - 2 * f_0 - 2 * \omega_0) = 1.75 * (22 - 2 * 1 - 2 * 0.25) = 34.125 \qquad [mm] \tag{30}$$

$$D_{i2} = 2 * R_{i2} = m * (z_2 - 2 * f_0 - 2 * \omega_0) = 1.75 * (68 - 2 * 1 - 2 * 0.25) = 114.625 \qquad [mm] \tag{31}$$

$$B = A * \psi_A = 80 * 0.45 = 36 \quad [mm] \tag{32}$$

$$B_2 = A * \psi_A = 36 \quad (output\ gear) \quad [mm] \tag{33}$$

$$B_1 = B_2 + m = 36 + 1{,}75 \approx 38 \qquad [mm] \tag{34}$$

The radius of the basic circles of the two wheels

$$R_{b1} = R_{d1} * \cos(\alpha_0) = \frac{m * z_1}{2} * \cos(\alpha_0) = 17.9 \qquad [mm] \tag{35}$$

$$R_{b2} = R_{d2} * \cos(\alpha_0) = \frac{m * z_2}{2} * \cos(\alpha_0) = 55.3 \qquad [mm] \tag{36}$$

$$R_{e1} = \frac{D_{e1}}{2} = 21 \qquad [mm] \tag{37}$$

$$R_{e2} = \frac{D_{e2}}{2} = 61.25 \qquad [mm] \tag{38}$$

Gear coverage

$$\varepsilon = \frac{\sqrt{R_{e1}^2 - R_{b1}^2} + \sqrt{R_{e2}^2 - R_{b2}^2} - A * \sin(\alpha)}{\pi * m * \cos(\alpha_0)} \tag{39}$$

$$\varepsilon = \frac{\sqrt{21^2 - (17.9)^2} + \sqrt{(61.25)^2 - (55.3)^2} - 80 * 0.34}{\pi * 1.75 * 0.93} = 1.97$$

The gear coverage is bigger than 1.2. It ensures a good operation of the gear.

The tooth control

For the pinion $z_1 = 22 => n_1 = 3$

$$L_{n1} = m * \left((n_1 - 0.5) * \pi + z_1 * inv(\alpha_0)\right) * \cos(\alpha_0) \quad (40)$$
$$= 1.75 * \left((3 - 0.5) * \pi + 22 * 0.014\right) * 0.93 = 13.3 \qquad [mm]$$

For the driven wheel $z_2 = 68 => n_2 = 8$

$$L_{n2} = m * \left((n_2 - 0.5) * \pi + z_2 * inv(\alpha_0)\right) * \cos(\alpha_0) \quad (41)$$
$$= 2 * \left((8 - 0.5) * \pi + 68 * 0.014\right) * 0.93 = 39.9 \approx 40 \qquad [mm]$$

Table 3.2. *Number of teeth [22]*

Numărnl de dinți n peste care se măsoară cota L_n se ia astfel: pentru $z = 18 \div 26$, $n = 3$; $z = 27 \div 35$, $n = 4$; $z = 36 \div 44$, $n = 5$; $z = 45 \div$ $\div 53$, $n = 6$; $z = 54 \div 62$, $n = 7$; $z = 63 \div 71$, $n = 8$; $z = 72 \div 80$, $n = 9$; $z = 81 \div 89$, $n = 10$; $z = 90 \div 98$, $n = 11$, $z = 99 \div 107$, $n = 12$; $z = 108 \div 116$, $n = 13$.

From Table 2.3., we choose $n_2 = 8, for\ z_2 = 68$.

Chapter 4. Output Shaft Design

4.1 Pre-dimensioning

We will use OL50 Material for output shaft.
Shafts are subjected to torsion and bending stress.

Pre-dimensioning the output shaft at torsional stress

$$d_p = \sqrt[3]{\frac{16 * M_{t2}}{\pi * \tau_{at}}}\ [cm] \tag{42}$$

Output shaft torque

$M_{t2} = 352\ [daN * cm]$

Torsional shear stress

$$\tau_{at} = 185\ \ [\frac{daN}{cm^2}] \tag{43}$$

We choose the average value for τ_{at}.

We calculate d_p:

$$d_p = \sqrt[3]{\frac{16 * M_{t2}}{\pi * \tau_{at}}}\ [cm] = \ d_p = \sqrt[3]{\frac{16 * 352}{\pi * 185}} = 2.13\ [cm] = 22\ [mm] \tag{44}$$

$$b = B2\ [mm] = 36\ [mm] \tag{45}$$

The length of the sections separating organs in relative motion inside the housing

$$l1 = 10\ [mm] \tag{46}$$

The length of bearing spindles

$$l2 = 0.5 * dp\ [mm] = 11\ [mm] \tag{47}$$

Preliminary length of the output shaft

$$l = b + 2 * l1 + l2\ [mm] = 36 + 20 + 11 = 67\ [mm] \tag{48}$$

Forces acting on a spur gear mesh

$\eta = 1 \Rightarrow F_{t1} \approx F_{t2}\ and\ F_{r1} \approx F_{r2}$

Tangential Force

$$F_{t2} = \frac{2 * M_{t2}}{D_{d2}} = \frac{2 * 352}{12} = 59\ [daN][daN] \quad (49)$$

$$D_{d2} = m * z2 = 1.75 * 68 = 119\ [mm] \approx 12\ [cm] \quad (50)$$

Radial Force

$$F_{r2} = F_{t2} * tan\ \alpha_0\ [daN] = 59 * 0.36 = 22\ [daN] \quad (51)$$

4.2 Shaft loading diagram

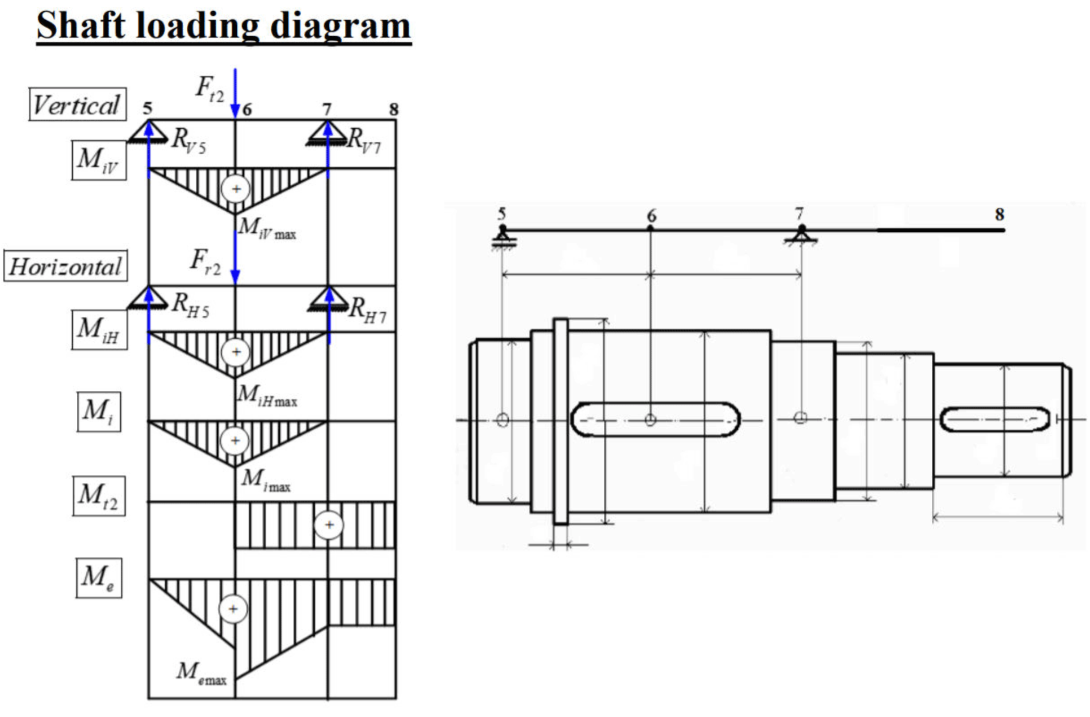


Fig.4.1. *Shaft loading diagram [23]*

4.2.1 Reaction forces and bending moments in both planes

Vertical plane

$$R_{V5} = R_{V7} = \frac{F_{t2}}{2} = \frac{59}{2} = 29.5\ [daN] \tag{52}$$

$$M_{IV_max} = \frac{1}{4} * F_{t2} * l\ [daN * cm] = 0.25 * 59 * 6.7 = 99\ [daN * cm] \tag{53}$$

Horizontal plane

$$R_{H5} = R_{H7} = \frac{F_{r2}}{2} = \frac{22}{2} = 11\ [daN] \tag{54}$$

$$M_{IH_max} = \frac{1}{4} * F_{r2} * l\ [daN * cm] = 0.25 * 22 * 6.7 = 36.85\ [daN * cm] \tag{55}$$

Resulting reaction forces

$$F_{R5} = F_{R7} = \sqrt{R_{V5}^2 + R_{H5}^2}\ [daN] \tag{56}$$

$$= \sqrt{(29.5)^2 + 11^2} = 31.5\ [daN]$$

$$M_{i_max} = \sqrt{M_{IV_max+}^2 + M_{IH_max}^2}\ [daN * cm] = 105\ [daN * cm] \tag{57}$$

Equivalent bending moment

$$M_{emax} = \sqrt{M_{i_max}^2 + (\alpha * M_{t2})^2}\ [daN * cm] = \sqrt{105^2 + (0.6 * 352)^2} = 236\ [daN * cm] \tag{58}$$

$$\alpha = \frac{\sigma_{ai_{iii}}}{\sigma_{ai_{ii}}} = \frac{450}{750} = 0.6 \tag{59}$$

$$d_{min} = \sqrt[3]{\frac{32 * M_{emax}}{\pi * \sigma_{ai_{iii}}}}\ [cm] \tag{60}$$

$$d_{min} = \sqrt[3]{\frac{32 * 236}{\pi * 450}}\ [cm] = 1{,}74\ [cm]$$

$$1{,}74 * \frac{104}{100} = 18\ [mm] \Rightarrow\ d_{min} = 18\ [mm] \tag{61}$$

4.3 Final geometry of the output shaft

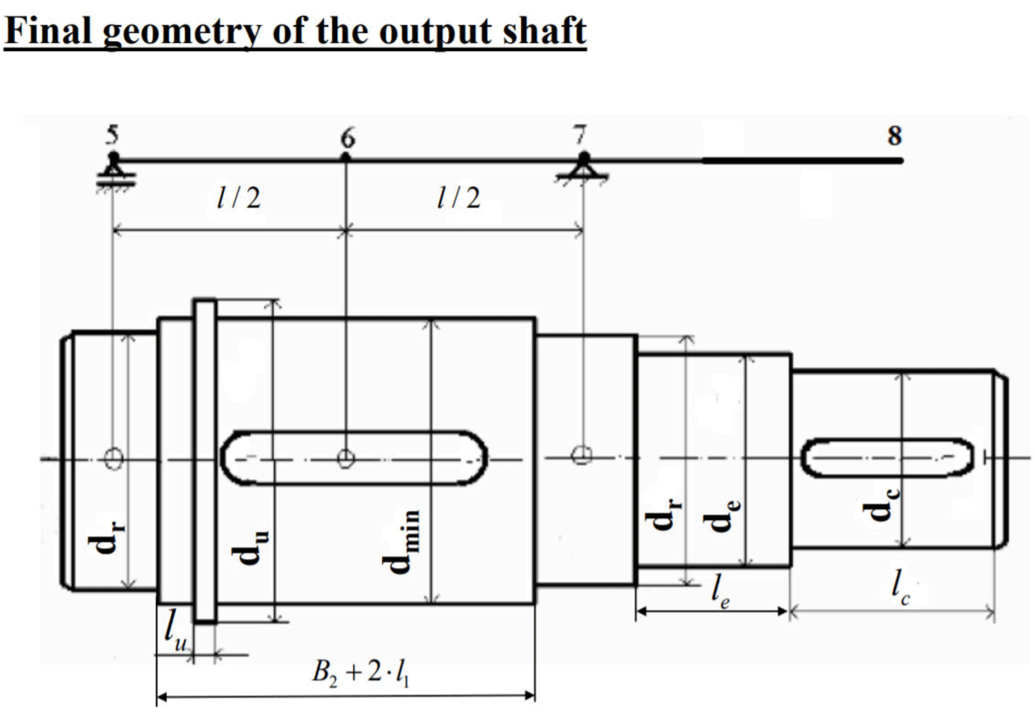


Fig. 4.2. *Drawing for output shaft [24]*

$$d_u = d_{min} + 6 = 18 + 6 = 24\ [mm] \tag{62}$$

$$l_u = 4\ [mm] \tag{63}$$

Bearing Diameter

$$d_r < d_{min} \Rightarrow d_r = 15\ [mm] \Rightarrow D_r = 35\ [mm] \tag{64}$$

Table 4.3. *Bearings diameter [25]*

Bearings diameter

$$d_r < d_{\min}$$

d_r [mm]	D_r [mm]	B [mm]	Sarcina dinamică de bază C [kN]	Simbol
15	32	9	5,6	6002
	35	11	7,8	6202
	42	13	11,4	6302
17	35	10	6	6003

$d_r = 15\ [mm] \quad D_r = 35\ [mm]$ (65)

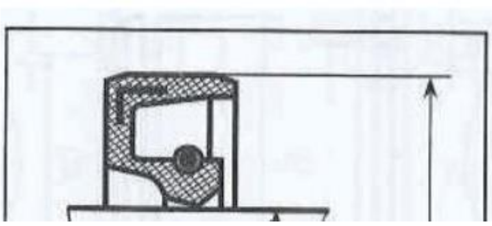
Seal Diameter

$d_e = 12\ [mm]$ (66)

Table 4.4. *Asynchronous Engine performance [26]*

Seal diameter

$$d_e \leq d_r$$



Diametrul nominal al arborelui d_e [mm]	Diametrul exterior al manşetei D_m [mm]	Lăţimea manşetei B_m [mm]
10	22	7
	25	
12	24	7
	25	
	30	

Table 4.5. *Asynchronous Engine performance [27]*

l_e →

Nr.crt.	Lungimea aproximată	Valoarea
1.	Lungimea tronsoanelor pe care se montează butuci de roţi dinţate, roţi de curea sau de fricţiune	$(1,2..2)d_p$
2.	Lungimea fusurilor lagărelor de alunecare	$(1...2)d_p$
3.	Lungimea fusurilor lagărelor cu rulmenţi	$(0,3...1)d_p$
4.	Lungimea tronsoanelor pe care se efectuează etanşarea	$(0,5...0,8)d_p$
5.	Lungimea tronsoanelor care separă organe aflate în mişcare relativă în interiorul carcasei	10 mm
6.	Lungimea tronsoanelor care separă organe aflate în mişcare relativă în exteriorul carcasei	20 mm

We choose

$$l_e = 0.6 * d_p = 0.6 * 22 = 13\ [mm] \tag{67}$$

Shaft end diameter

$$d_c = 10\ [mm\ ] \rightarrow l_c = 23\ [mm] \tag{68}$$

Table 4.6. *Asynchronous Engine performance [28]*

Shaft end diameter

$d_c < d_e$

d_c [mm]	l_c [mm] seria	
	lungă	scurtă
10	23	20
11		

4.4 Choosing longitudinal parallel key

Keys are used to transmit torque from a rotating machine element to the shaft.

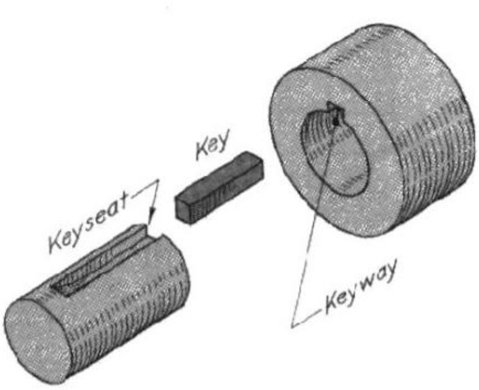


Fig. 4.7. *Longitudinal parallel key [29]*

Stress analysis of parallel keys

A key has two failure mechanisms:

• it can be sheared off.
• it can be crushed due to the compressive bearing forces.

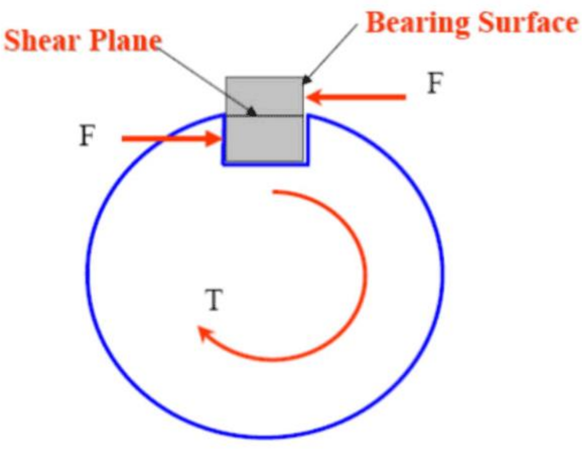


Fig. 4.8. *Stress analysis of parallel keys [30]*

Calculate the required key length at contact stress

We choose the dimensions b, h, t_1, t_2 based on d_{min} from the tabel.

$$d_{min} = 18\ [mm] \tag{69}$$

$$b = 6\ [mm]\ \ h = 6\ [mm] \tag{70}$$

$$t_1 = 3.5\ [mm] \quad t_2 = 2.8\ [mm] \tag{71}$$

Dimensioning at contact stress

$$l_{min} = \text{key minimum length} = \frac{4 * M_{t2}}{h * d_{min} * p_a} = \frac{4 * 352}{0.6 * 1.8 * 800} = 1.6\ [cm] \tag{72}$$

$$p_a = 800\ [\frac{daN}{cm^2}] \tag{73}$$

$$l_p = 0.8 * B_2 = 0.8 * \ 3.6 = 2.9\ [cm] > l_{min} = 1.6\ [cm] \tag{74}$$

$$l_p = 32\ [mm]\ based\ on\ tabel\ \rightarrow\ l_{min} = 16\ [mm] \tag{75}$$

4.5 Verification of shaft deflection and critical speed

Deflection in vertical plane

$$f_v = \frac{F_{t2} * l^3}{48 * E * I}\ [cm] \tag{76}$$

$$E = 2.1 * 10^6\ \left[\frac{daN}{cm^2}\right] \tag{77}$$

$$I = \pi * \frac{d_{min}^4}{64}\ [cm] = \ \pi * \frac{1.8^4}{64} = 0.5\ [cm] \tag{78}$$

$$f_v = \frac{59 * (6.7)^3}{48 * 2.1 * 10^6 * 0.5} = 352 * 10^{-6}\ [cm] \tag{79}$$

Deflection in horizontal plane

$$f_H = \frac{F_{r2} * l^3}{48 * E * I}\ [cm] = \frac{22 * (6.7)^3}{48 * 2.1 * 10^6 * 0.5} = 132 * 10^{-6}\ [cm] \quad (80)$$

Check result of calculation (deflection)

$$f = \sqrt{f_V^2 + f_H^2} < f_{adm}\ [cm] \quad (81)$$

$$f_{adm} = 5 * 10^{-3} * m = 3 * 10^{-4} * l\ [cm] \quad (82)$$

$$f = \sqrt{132^2 + 352^2} = 376 * 10^{-6} [cm] \quad (83)$$

$$f_{adm1} = 5 * 10^{-3} * 0.175 = 0.875 * 10^{-4}\ [cm] \quad (84)$$

$$f_{adm2} = 3 * 10^{-4} * 67 = 201 * 10^{-5}\ [cm] \quad (85)$$

$$f < f_{adm} => f = 376 * 10^{-6}\ [cm] \quad (86)$$

Check results of calculation (vibrations)

Weight of the output gear

$$G = \gamma * V = \gamma * \frac{\pi * D_d^2}{4} * B_2\ [daN] \quad (87)$$

$$\gamma = 7.8 * 10^{-3} \left[\frac{daN}{cm^3}\right] - for\ steel \quad (88)$$

$$D_{d2} = m * z2 = 1.75 * 68 = 12\ [cm] = 120\ [mm] \quad (89)$$

$$B_2 = 3.6\ [cm] \quad (90)$$

$$G = 7.8 * 10^{-3} * \frac{\pi * 12^2}{4} * 3.6 = 3176 * 10^{-3}\ [daN] = 3.2\ [daN] \quad (91)$$

Static Deflection

$$f_{st} = \frac{G * l^3}{48 * E * I}\ [cm] \quad (92)$$

$$f_{st} = \frac{3.2 * (6.7)^3}{48 * 2.1 * 10^6 * 0.5} = 16.7 * 10^{-6}\ [cm]$$

Verification of shaft deflection and critical speed

- **Check results of calculation (deflection, vibrations)**

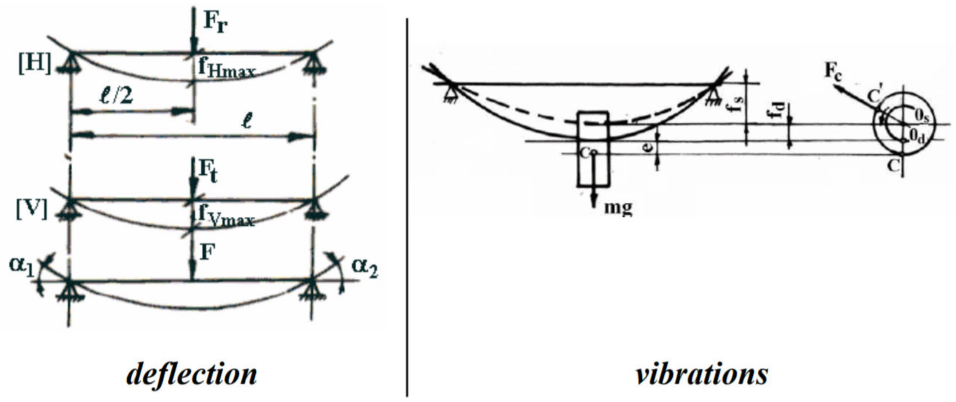


Fig. 1.2. *Deflection and vibrations of shaft [31]*

Critical Speed

$$n_{cr} = \frac{30}{\pi} * \sqrt{\frac{g}{f_{st}}}\ [RPM] \tag{93}$$

$$g = 9.81 \frac{m}{s^2} = 9.81 * 10^2\ cm/s^2 \tag{94}$$

$$n_{cr} = \frac{30}{\pi} * \sqrt{\frac{9.81 * 10^2}{16.7 * 10^{-6}}} = 73100\ [RPM] \tag{95}$$

4.5.1 The total length of output shaft

$L_t = 114\ [mm]$

Chapter 5. Rolling Bearing Selection

5.1 Calculate dynamic basic bearing load rating C

Life's equation (Lundberg & Palmgren)

The basic life L of rolling bearings is the life that is achieved or exceeded by 90% of identical bearings under the same operational conditions provided that commonly used materials were used, usual production quality achieved, and bearings are operated under normal operational conditions. The basic life is defined by the equation [32] :

$$L = (\frac{C}{F})^p \tag{96}$$

where **C** = dynamic basic bearing load rating [daN]

$$L = \frac{60 * n_2 * L_h}{10^6} \ [million\ of\ rotation] \tag{97}$$

$L_h = 10000\ [hours]$
$n_2 = 922.33\ [RPM]$
$n_1 = n_n = 2850\ [RPM]$

Resulting reaction forces

$$R_5 = R_7 = F_r = \sqrt{R_{V5}^2 + R_{H5}^2}\ [daN] = 31.5\ [daN] \tag{98}$$

$F_r - radial\ force$ will be used for choosing bearings.

Equivalent bearing load

It is a radial load (with radial bearings) or axial load (with axial bearings), at which all bearings of the same type show the same life as reached under conditions of a real load.

$$\mathrm{F_e} = 1.2 * (X * V * Fr + Y * F_a)\ [daN] \tag{99}$$

where

$\mathrm{F_r} -$ radial component of the real load [daN]
$\mathrm{F_a} -$ axial component of the real load [daN]
$\mathrm{X, Y} -$ coefficient of $radial\ load (single - row\ radial\ ball\ bearings: X = 1, Y = 0, V = 1)$

Selection of ball bearing

Output Shaft

$$F_e = 1.2 * F_r = 1.2 * 31.5 = 37.8\ [daN] \tag{100}$$

$$L = \frac{60 * 922.33 * 10000}{10^6} = 553\ \ [milion\ of\ rotations] \tag{101}$$

$$\left(\frac{C_{min}}{F_e}\right) = L^{0.33}\ \rightarrow C_{min} = 37.8 * 7.94 = 300\ [daN] = 3\ [KN] \tag{102}$$

$$C_{\min} = 3[KN]\ \ d_r = 15\ [mm]\ D_r = 35\ [mm] \tag{103}$$

$$C = 7.8\ [KN] \tag{104}$$

$$C \geq C_{min}$$

We choose the bear $62202 - 2RS1$ due to equation (104).

5.2 Bearing selection from SKF catalogue

• From SKF catalogue we have selected a bearing of 62202 - 2RS1.

Table 5.1. *Choice of bearing depending on C (dynamic basic bearing load rating) [33]*

■ SKF Explorer ▶ Popular item

Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Designations	
			dynamic	static		Reference speed	Limiting speed	Bearing	Snap ring
d [mm]	D [mm]	B [mm]	C [kN]	C_0 [kN]	P_u [kN]	[r/min]	[r/min]		
15	35	11	8.06	3.75	0.16		13000	▶ 6202-2RSH	
15	35	11	8.06	3.75	0.16		13000	6202-RSH	
15	35	14	7.8	3.75	0.16		13000	62202-2RS1	
15	35	14	11.9	7.5	0.32	32000	17000	4202 ATN9	
15	42	13	9.95	5.4	0.232	40000	20000	W 6302-2Z	

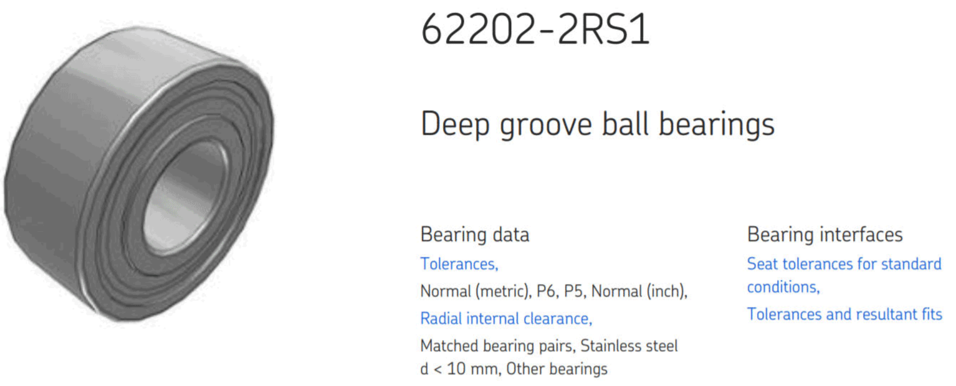


Fig. 5.2. *62202-2RS1 Ball Bearing [34]*

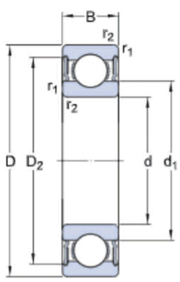
Its basic load rating (dynamic) is 7.8 [KN].

5.3 Technical specification of the selected bearing

Dimensions of the bearing are as follow:

Table 1.1. *Technical specification of 62202-2RS1 bearing [35]*

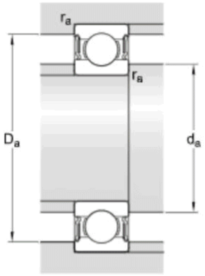
Technical specification



DIMENSIONS

d	15 mm
D	35 mm
B	14 mm
d_1	≈ 21.7 mm
D_2	≈ 30.35 mm
$r_{1,2}$	min. 0.6 mm

Table 1.1. *Abutment dimensions of 62202-2RS1 bearing [36]*



ABUTMENT DIMENSIONS

d_a	min. 19.2 mm
d_a	max. 21.6 mm
D_a	max. 30.8 mm
r_a	max. 0.6 mm

Table 1.1. *Calculation data and mass of bearing [37]*

CALCULATION DATA

Basic dynamic load rating	C	7.8 kN
Basic static load rating	C_0	3.75 kN
Fatigue load limit	P_u	0.16 kN
Limiting speed		13000 r/min
Calculation factor	k_r	0.025
Calculation factor	f_0	13

MASS

Mass bearing	0.054 kg

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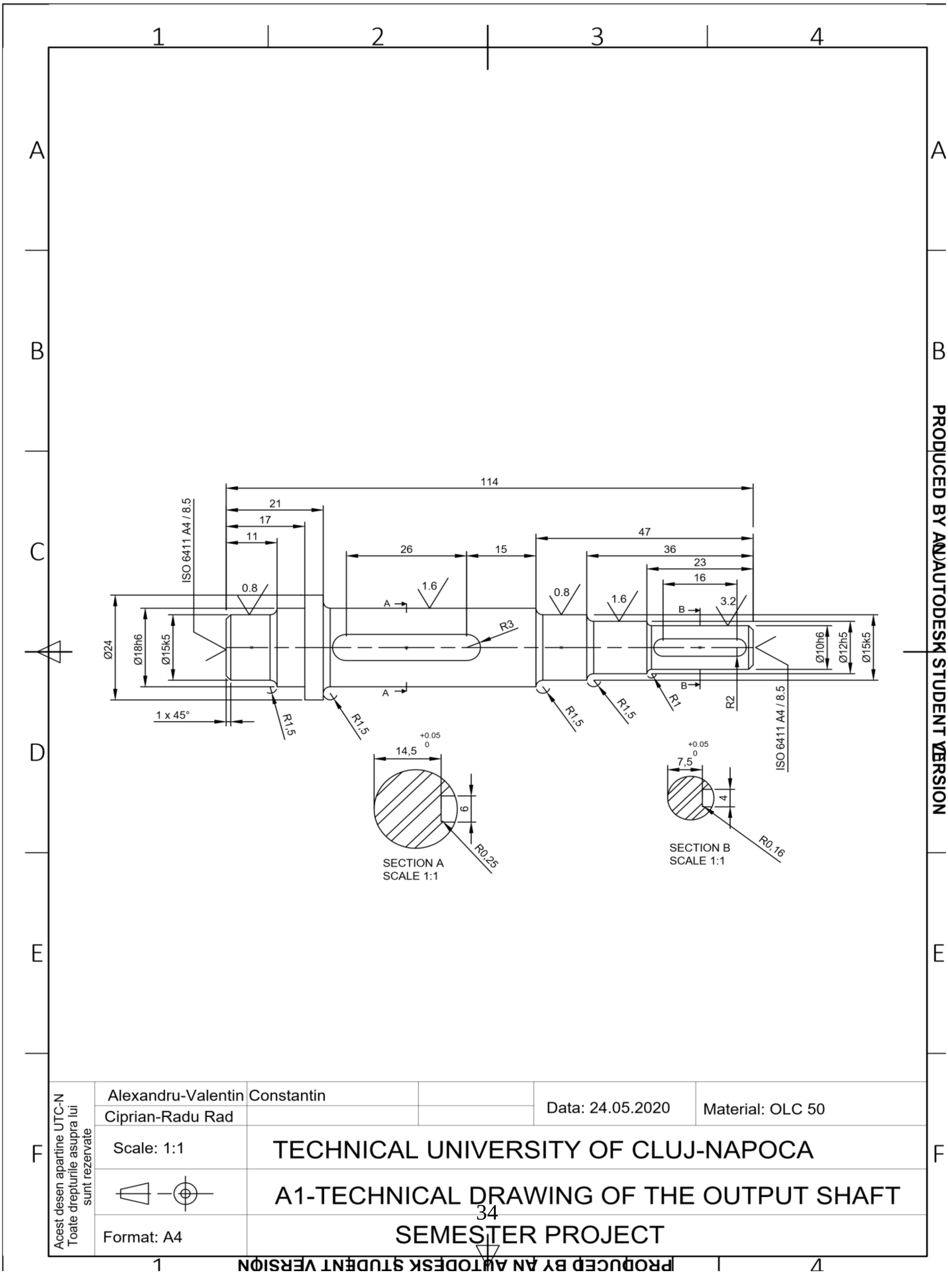
ANNEXES

Annex 1 – Technical drawing of the output shaft (AutoCAD sketch)



Annex 2 - Choosing the module m [mm] depending on centre distance A [mm]

Centre distance *A* [mm]	
40	400
50	*450
63	500
*71	*560
80	630
*90	*710
100	800
*112	*900
125	1000
*140	*1120
160	1250
*180	*1400
200	1600
*224	*1800
250	2000
*280	*2240
315	2500
*355	

* *second recommendation*

$$m = \frac{2A}{z_1 + z_2} \frac{\cos\alpha}{\cos\alpha_0} [mm]$$

$$\alpha_0 = 20°$$

$$\boxed{m_{STAS} > m}$$

Module *m* [mm]			
0.05	*0.45	4	*36
*0.055	0.5	*4.5	40
0.06	*0.55	5	*45
*0.07	0.6	*5.5	50
0.08	*0.7	6	*55
*0.09	0.8	*7	60
0.1	*0.9	8	*70
*0.11	1	*9	80
0.12	*1.125	10	*90
*0.14	1.25	*11	100
0.15	*1.375	12	
*0.18	1.5	*14	
0.2	*1.75	16	
*0.22	2	*18	
0.25	*2.25	20	
*0.28	2.5	*22	
0.3	*2.75	25	
*0.35	3	*28	
0.4	*3.5	32	