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

Sem. **II**, Year **2022-2023**

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N = 20

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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.

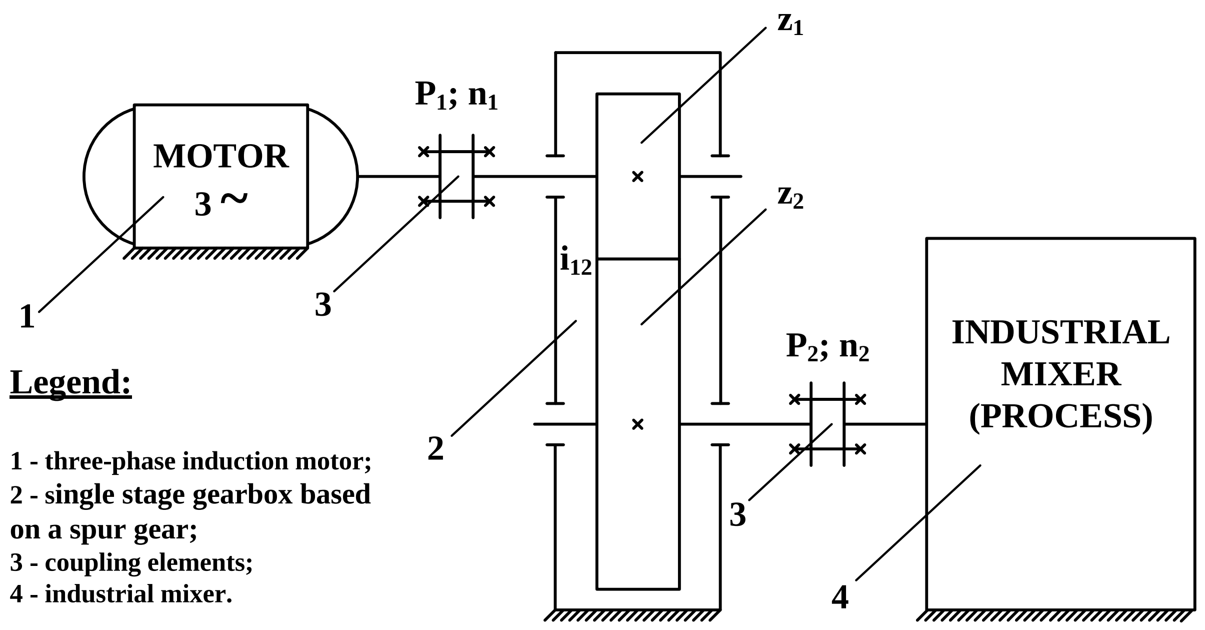


Fig. 1.1. *Kinematic diagram of the mechanical power transmission.*

Input data (30321):

$P_2 = 5 + 0.15*n$ [kW] – output power
$n_s = 1500$ [RPM] – synchronous speed of motor
$i_{12} = 2 + 0.1*n$ - transmission ratio
$z_1 = 25$ - number of teeth for the first gear
$L_h = 10.000$ [hours] - number of running hours

1.2 Gear speed reducers

1.2.1. What Are Speed Reducers?

Speed reducers, also known as gear reducers or gearboxes, are mechanical devices used to reduce the rotational speed (RPM) of a prime mover (such as an electric motor) and increase the torque output. They are commonly used in various applications where the input speed of a power source needs to be decreased while generating higher torque.

1.2.2. Why Are Transmissions and Speed Reducers Necessary?

Transmissions and speed reducers are indispensable components in mechanical systems due to their significant role in controlling speed, amplifying torque, and adapting to various operating conditions. One real-life example that highlights their necessity is the automotive industry.

In an automobile, the transmission and speed reducer, commonly known as the gearbox, play a crucial role in translating power from the engine to the wheels. The engine generates high-speed rotation but relatively low torque, while the wheels require lower speeds but higher torque for efficient acceleration and operation. The transmission allows the driver to select different gear ratios, altering the speed and torque output accordingly. In lower gears, the transmission increases the torque while reducing the speed, providing the necessary force for tasks like climbing steep inclines or accelerating from a standstill. As the vehicle gains momentum, higher gears are engaged to increase the speed while sacrificing some torque.

Furthermore, the transmission enables adaptability to various driving conditions. For instance, when driving uphill, a lower gear is selected to maintain sufficient torque for power delivery. On the other hand, when driving on a highway, a higher gear is engaged to achieve a higher top speed while optimizing fuel efficiency.

The speed reducer aspect of the gearbox further enhances torque output. The gears within the transmission, combined with the differential, allow the engine's power to be distributed to the wheels with appropriate speed reduction and torque multiplication. This ensures that the vehicle can effectively overcome resistance, such as air drag and friction, and operate optimally in different road conditions.

Without transmissions and speed reducers, automobiles would lack the necessary speed control, torque amplification, and adaptability to perform efficiently and smoothly. They enable the engine's power to be harnessed and tailored to meet the specific requirements of acceleration, maintaining speed, and navigating varying terrains.

1.2.3. Principles of Changing and Decelerating Speed

We stated that speed reducers are devices that obtain the required rotational speed. However, they have one more important role. That is that they obtain a rotational force (torque) proportional to the deceleration ratio.

For example, if we halve the rotational speed, that rotational force (torque) will double. This is because the mechanism of changing and decelerating speed uses the principle of leverage.

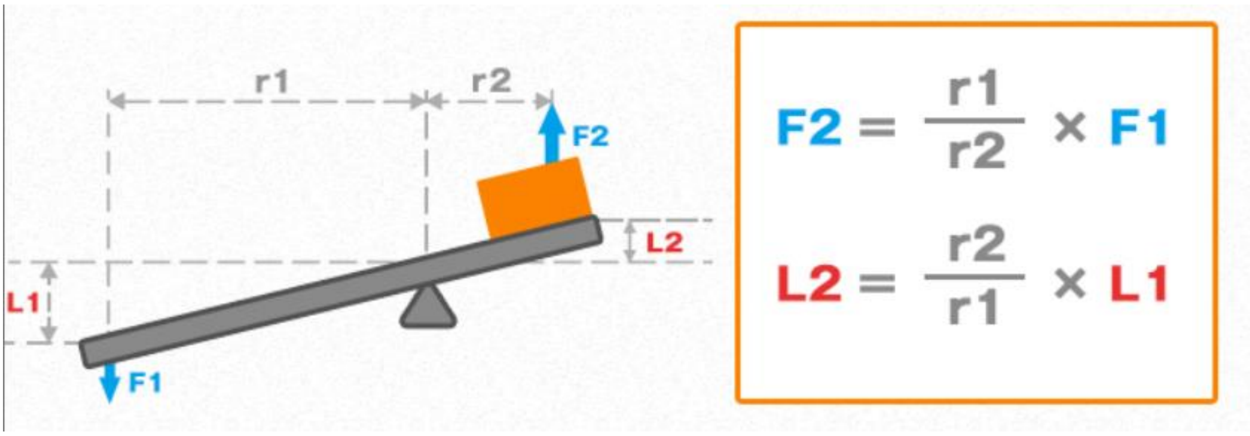


Fig. 1.2.1. Principle of Leverage

The above figure shows the principle of leverage. The formula in the above figure demonstrates that a large force called F2 can be obtained with the force of F1.

In addition, we can see that the movement distance of L2 is shorter than the movement distance of L1. We get the following if we illustrate this by replacing this principle with an easy-to-understand rotating body. Accordingly, it is possible to increase the rotational force by lowering the rotational speed.

1.2.4 Types and Structures of Speed Reducers

Table 1.1. *Speed Reducers*

Parallel Axis Gear Speed Reducer	This is a speed reducer combined with a spur gear. It is available in one to four stages depending on the deceleration ratio. Power is transmitted with rolling. Therefore, it is possible to secure a transmission efficiency of about 98% per stage. The deceleration ratio is about 1/5 to 1/2,500 and can cover a wide range of deceleration ratios. There are differences in the number of gear stages combined with the deceleration ratio. This means that the dimensions of the speed reducer change depending on the adaptive motor output.
Helical Speed Reducer	This is the same as a parallel axis gear speed reducer, but it uses helical gears as its gears. Helical gears are those with a twisted tooth trace. As a result, the tooth engagement rate improves. This means that it is possible to obtain power transmission with a high degree of smoothness and quietness. However, they have a weak point. They generate thrust in proportion to the magnitude of the power transmission. This is because the tooth trace is smooth. It is possible to overcome this weak point with a double helical gear configured with an opposite helical gear to counteract this thrust force.

Bevel Gear Speed Reducer	This is a speed reducer combined with a bevel gear. A typical gear angle is 90°. Combining small and large gears leads to a deceleration in speed. However, these are often used to output with a 90° swing. To that end, it is also called a miter gear if the deceleration ratio is 1:1 among bevel speed reducers. Those that improve the engagement ratio by twisting the tooth trace of the bevel gears like helical gears are called spiral bevel gears
Hypoid Speed Reducer	This is a gear laid out to offset the center of a pinion gear axis just be a certain amount with respect to the center of the gear axis. These add slip to the power transmission of spiral bevel gears like the worm speed reducer we describe later. This makes smoother power transmission possible. Hypoid speed reducers can obtain a large deceleration ratio. This is because the gear ratio is large like that of a worm speed reducer. Nevertheless, the engagement is complicated. Therefore, it is necessary to precisely adjust the engagement position.
Worm Speed Reducer	This is a speed reducer that combines a worm gear and a worm wheel. It is possible to obtain a large deceleration ratio of about 1/10 to 1/60 just with this combination. If the lead angle of the worm gear becomes smaller (higher deceleration ratio), it is called a self-lock. It is possible to use this to prevent lifting devices falling by utilizing the fact it is difficult to rotate from the worm wheel side (output side). In addition, reducing the thrust backlash of the input axis makes it possible to suppress the backlash. Accordingly, this is a speed reducer with various advantages. Nevertheless, power is transmitted according to the sliding of the tooth surface compared to spur gears. Consequently, heat is easily generated and the transfer efficiency is not very good at about 50%.

[1]

1.3 Power transmission design analysis and justification

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 mixer.

We will consider several design requirements:

• The nominal speed of the motor needs to be smaller than the synchronous speed of themotor

- We need to choose materials than have adequate strength values
- When using gears, we need to make sure that the gear coverage is higher than 1.2 but lower than 2
- Values for axial distance, gear module, different lengths of shaft parts and certain diameters need to be standardized
- Diameter values for the output shaft parts need to respect a certain inequality which will be presented later
- The output shaft will use 3 keys: one where the gear is mounted and 2 at the shaft end
- All the parts need to be resistant to contact and bending stress The decision to use a single-reduction spur gear reducer design for this project comes down to its simplicity and to the final cost that is very likely to be lower than other potential design such as helical gears or bevel gears. Also, spur gears have great power transmission efficiency and are easy to manufacture and install on the reducer.

[2]

Chapter 2. Selecting The Actuator

2.1 Selecting the AC motor

In order to select the AC motor, we need to calculate the following:

Syncronous speed:

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

Where :
f - the motor supply's frequency in Hertz, and p is the number of pairs of magnetic poles
p – number of pairs of magnetic poles
ns = 1500 $[RPM]$ (synchronous speed of the motor)

Table 2.1. *Number of pairs of magnetic poles for different values of synchronous speed [3]*

p	1	2	3	4		…
n_s	3000	1500	1000	750		…

ns = 1500[RPM], so we have two pairs of magnetic poles (p = 2).

From equation (1) we can obtain the frequency f

$$f=50\ [Hz] \quad (2)$$

Slip:

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

$$n_n < n_s \quad (4)$$

Where:
n_s – synchronous speed [RPM]
n_n – rotor nominal speed [RPM]

The actuator power:

$$P_m = \frac{P_2}{\eta} = 5.36\ [KW] \quad (5)$$

Where:
P_m – required motor power$[kW]$
P_2 – 8 $[kW]$(output power)
η – mechanical transmission efficiency

Mechanical transmission efficiency:

$$\eta = \eta_g * \eta_b^2 * \eta_l \quad (6)$$

η_g = 0.98 (spur gear efficiency)
η_b = 0.995 (one pair of bearings efficiency)
η_l = 0.99 (lubrication efficiency)

$$\eta = 0.9605 \quad (7)$$

So the actuator power is:

$$P_m = 8.32880 \quad (8)$$

For choosing the actuator, we must respect the conditions:

$$\begin{cases} P_n < P_m \\ n_n < n_s \end{cases} \quad (9)$$

Taking the conditions at (9) in consideration, we have chosen the ASU-160M-4 actuator, with the specifications in the figure:

ASU 160M-4	11	1440	20,3	89,0	0,88	6,5	2,2	2,4	115

[4]

Fig. 2.1. *ASU 160M-4 specifications*

2.2 Power transmission kinematics

2.2.1 Determining the number of teeth of the output gear

$$z_2 = i_{12} \cdot z_1 = 100 \quad (10)$$

where z1 and i12 are given as input data.

The real transmission ratio:
u12 = z2/z1 = 4

2.2.2 Determining the input/output shaft speed

n_1 =1440 [RPM] – input shaft speed (11)
n_2 = 360 [RPM] – output shaft speed (12)

2.2.3 Determining the power transmitted by the input shaft

$$P_1 = P_m = 8.3288 \text{ [kW]} \quad (13)$$

2.2.4 Determining the input/output shaft torque

T_1 = 95500 * P_1/n_1 = 552.361 [daN * cm]
T_2 = 95500 * P_2/n_2 = 2122.2 [daN * cm]

(14)

Chapter 3. Spur Gear Design

3.1 Gears tooth strength analysis and verification

Chosen material : OLC45 – carbon steel with flanking
Hardness :

$$HB = 0.102 * \left(2 * F\right) / \left(\pi * D * \left(D - \sqrt{D^2 - d^2}\right)\right) = 197$$

3.1.1 Equivalent load

The pitch line velocity:

$$v = 0.1 \cdot \sqrt[4]{n_1^2 \cdot \frac{P_1}{u_{12}}} = 4.55839 \text{ [m/s]}$$

(15)

The load (service) factor:

$$k = k_c \cdot k_d = 1.12 \cdot 1.2 = 1.3440 \quad (16)$$

The equivalent torque:

$$T_{e1} = k \cdot T_1 = 1.3440 * 552.361 = 742.373$$

(17)

3.1.2 Contact stress (dimensioning center/axial distance A)

$$A_{min} = (u_{12}+1) * \sqrt[3]{\frac{T_{e1}}{\Psi_A * u_{12} * sin2\alpha} * \left(\frac{865}{\sigma_{ak}}\right)^2} \; [cm] \quad (18)$$

Where:
α – pressure angle = 20^o
Ψ_A – axial coefficient of the gear = 0.3÷0.6. We have chosen 0.4.
Therefore,

$$A_{min} = 13.7035$$

And,

$$\sigma_{ak} = 26 \cdot HB = 26 * 197 = 5122 \; [daN/cm^2]$$

Choose
$A_w > A_{min} \Rightarrow A_w = 14\,[cm] = 140[mm]$

(19)

$$m_{min} = \frac{2 \cdot A_w}{z_1+z_2} \cdot \frac{\cos\alpha}{\cos\alpha_0} = 0.2240[cm]$$

(20)

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	

Fig. 2.2 *Standardized values for the module m [5]*

And from fig. 2.2 I chose the value of m so that:
$m >= m_{min} => m = 0.25$ [cm]
(21)

And the final value of A is :

$$A = \frac{m \cdot (z_1+z_2)}{2} = 15.6250\;[cm]$$

(22)

3.1.3 Bending Stress (verification)

The gear wheel was dimensioned at contact stress => verify its resistance at bending stress

The bending tension s:

$$\sigma = \frac{2 \cdot T_{e1}}{\pi \cdot m^3 \cdot z_1 \cdot \psi_{m \cdot C_f} \cdot cos\alpha} = 381.490 \left[\frac{daN}{cm^2}\right]$$

(23)

Where:

Cf – shape coefficient of the gears (for z1 = 25 it is equal to 0.1355)

Ψm – modular coefficient of the gears which should be in the interval [8, 40] and is calculated by the formula:

$$\Psi\text{m} = \frac{A * \Psi\text{A}}{m} \qquad (24)$$

$$=> \psi_m = 25$$

Now we verify if

$$\sigma < \sigma_{ai} \qquad (25)$$

Where

$$\sigma_{ai} = \frac{\sigma_0}{k_\sigma \cdot C} = 1327 \left[\frac{daN}{cm^2}\right] \quad (26)$$

σ0= 43 [daN / mm^2] = 4300 [daN/ cm^2] for OLC45

$k\sigma$ – stress concentration coefficient = 1.2÷2

C – safety coefficient = 1.5 ÷2

It results that the inequality at (25) is true;

$$m_b = \sqrt[3]{\frac{(0.68 \cdot T_{e1})}{z_1 \cdot \psi_m \cdot C_f \cdot \sigma_{ai}}} = 0.1652[cm]$$

Which satisfies

$$m_b < m$$

(27)

3.2 Final geometrical elements of the gears

We have:

m = 1.5 [mm]

z1 = 25

z2 = 98

a_0 = 20

f_0 = 1

w_0 = 0.25

And we compute:

D_{d1} = m/10 * z1 = 0.6250 (28)

D_{d2} = m/10 * z2 = 2.5000 (29)

D_{e1} = m * (z1 + 2 * f0) = 6.75 (30)

D_{e2} = m * (z2 + 2 * f0) = 25.5 (31)

D_{i1} = m * (z1 - 2 * f0 - 2 * w0) = 5.6250 (32)

Di_2 = m * (z2 - 2 * f0 - 2 * w0) = 143.25 (33)

B_2 = A * 10 * Ψ_A = 62.50 (34)

B_1 = B2 + m = 62.750 (35)

R_{e1} = 3.3750
R_{e2} = 12.750

The radius of the base circles of the 2 gears:

$R_{b1} = \frac{m * z_1}{2} * cos\alpha_0$ (36)

$$R_{b2} = \frac{m * z_2}{2} * cos\alpha_0$$

Rb1 = 2.9365 (37)

Rb2 = 11.7462 (38)

$$N_1 = z_1 * \frac{\alpha_0}{180} + 0.5$$
$$N_2 = z_2 * \frac{\alpha_0}{180} + 0.5$$

N1 = 3 (39)
N2 = 12 (40)

The grade of coverage of the gear:

$$\varepsilon = \frac{\sqrt{R_{e1}^2 - R_{b1}^2} + \sqrt{R_{e2}^2 - R_{b2}^2}}{\pi * m * cos\alpha_0}$$

=> ε = -63.436538894546686 (41)

$$L_{n1} = m[(N_1 - 0.5)\pi + z_1 inv\alpha_0]cos\alpha_0$$
$$L_{n2} = m[(N_2 - 0.5)\pi + z_2 inv\alpha_0]cos\alpha_0$$

=> L_{n1} = 1.9273 (42)
=> L_{n2} = 8.8163 (43)

Chapter 4. Output Shaft Design

4.1 Pre-dimensioning

4.1.1 Pre-dimensioning the output shaft at torsional stress

Pre-dimensioning diameter :

$$T_2 = 95500 \times \frac{P_2}{n2} = 2122.2[\text{daN} * \text{cm}] \quad (44)$$

$$d_{pmin} = \sqrt[3]{\frac{16 * T_2}{\pi * \tau_{at}}} = 4.16\ [cm] = 41.6\ [mm] \quad (45)$$

where $\tau_{at} = 150$

$$d_p \geq d_{pmin} => d_p = 42\ [mm] = 4.2\ [cm] \quad (46)$$

4.1.2 Preliminary length of the output shaft

$$b = B_2 = 62.5\ [mm] \quad (47)$$

$$l = b + 2 * l_1 + l_2 = 124.5\ [mm] \quad (48)$$

Where
l_1 = 10 [mm]
l_2 = 42 [mm]

4.1.3 Forces acting on a spur gear mesh

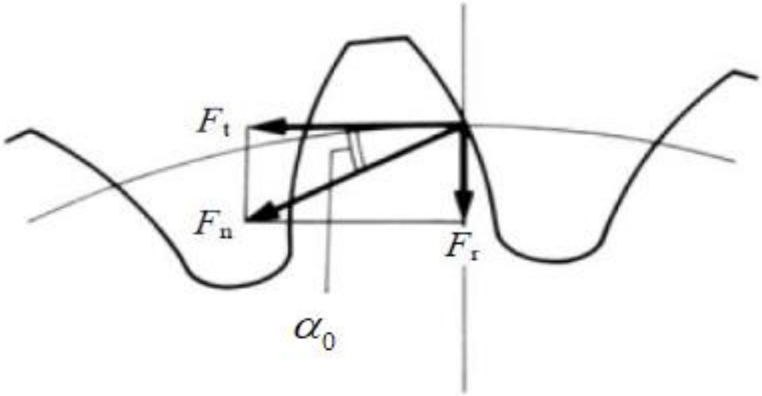
The tangential force :

$$F_{t2} = \frac{2 * T_2}{D_{d2}} = 169.7\ [\text{daN}] \quad (49)$$

Where
D_{d2} = m * z_2 = 25 [mm] = 2.5 [cm]

The radial force:

$$F_{r2} = F_{t2} * tan\alpha_0 = 61.79\ [daN] \quad (50)$$



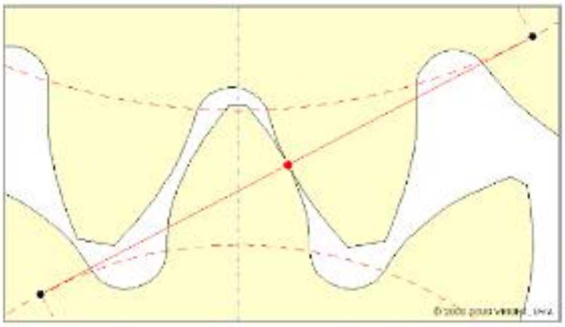


Fig 4.1. *Forces acting on a spur gear mesh [6]*

4.2 Shaft Loading Diagram

4.2.1 Reaction forces and bending moments in both planes

The vertical plane:

$$R_{V5} = R_{V7} = \frac{F_{t2}}{2} = 84.88\ [daN] \quad (51)$$

$$M_{iVmax} = \frac{1}{4} \cdot F_{t2} \cdot l = 528.43\ [daN * cm] \quad (52)$$

The horizontal plane:

$$R_{H5} = R_{H7} = \frac{F_{r2}}{2} = 30.89\ [daN] \quad (53)$$

$$M_{iHmax} = \frac{1}{4} \cdot F_{r2} \cdot l = 192.33\ [daN * cm] \quad (54)$$

Resulting reaction forces:

$$R_5 = R_7 = \sqrt{R_{V5}^2 + R_{H5}^2} = 90.33\ [daN\ ] \quad (55)$$

Resulting bending moment:

$$M_{imax} = \sqrt{M_{iVmax}^2 + M_{iHmax}^2} = 562.35\ [daN * cm] \ (56)$$

4.2.2 Equivalent bending moment

$$M_{emax} = \sqrt{M_{imax}^2 + (\alpha T_2)^2} = 576.58[daN \cdot cm\ ] \ (57)$$

Where

$\alpha = \frac{\sigma_{aiIII}}{\sigma_{aiII}} = 0.6$ (58)

$\sigma_{aiIII} = 450 \ [daN/cm^2]$,

$\sigma_{aiII} = 750 [daN/cm^2]$
for OL50

The minimum diameter of the critical section of the shaft :

$$d_{min} = \sqrt[3]{\frac{32 \cdot M_{emax}}{\pi \cdot \sigma_{aiIII}}} = 5.07 \ [cm] = 50.7 \ [mm] \ (59)$$

Because a key will be used to assembly the gear with the shaft, the value of dmin must be increased by 4%. We will round up the value to a standardized value.

$$d_{min} + 4\% * d_{min} \Rightarrow d = 52 \ [\text{mm}] \ (60)$$

4.3 Final geometry of the output shaft

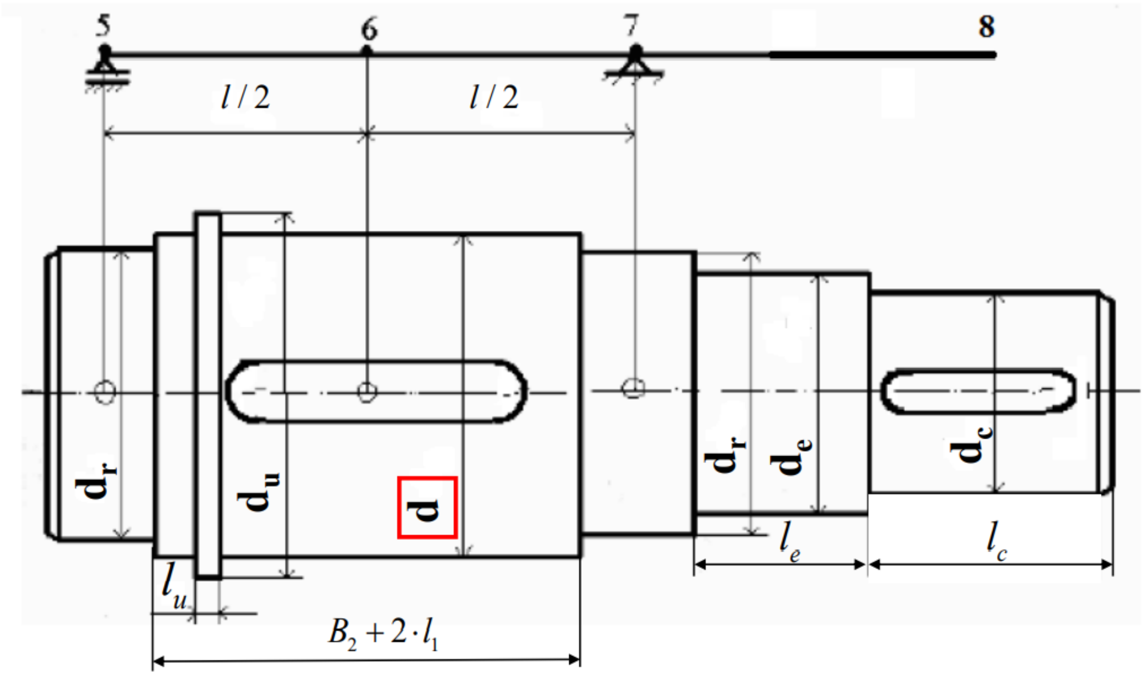


Fig 4.2 *Final geometry of the output shaft [7]*

4.3.1 Diameters in other points of the shaft

$$d_u > d > d_r > d_e > d_c \quad (61)$$

Where:
d_u – shoulder diameter (to be chosen)
d – diameter where the output gear will be mounted (computed before)
d_r – diameter of the shaft where the bearings will be mounted (to be chosen)
d_e – diameter of the shaft where the seal will be mounted (to be chose)
d_c – diameter of the end of the shaft; end of the shaft will be connected to the process, in this case the industrial mixer through a coupling element (to be chosen)

4.3.2 Diameter of the end shaft

$$d_{cmin} = \sqrt[3]{\frac{16 * T_2}{\pi * \tau_{at}}} = 2.41\ [cm] \rightarrow 24.1\ [mm] \quad (62)$$

Where
$\tau_{at} = 0.6 * \sigma_{aiII}$
$\sigma_{aiII} = 1200\ [\frac{daN}{cm^2}]$

We have the condition

$$d_c > d_{cmin} \quad (63)$$
$$=> d_c = 25$$

4.3.3. Length of the end shaft

d_c [mm]	l_c [mm] seria		d_c [mm]	l_c [mm] seria	
	lungă	scurtă		lungă	scurtă
10	23	20	38	80	58
11			40	110	82
12	30	25	42		
14			45		
16	40	28	48		
18			50		
19			55		
20	50	36	56		
22			60	140	105
24			63		
25	60	42	65		
28			70		
30	80	58	71		
32			75		
35			80	170	130

***Fig. 4.3** Length of the end shaft [8]*

From the table above, we extract the value for l_c

$$l_c = 42\ [mm] \quad (64)$$

Seal diameter :

$$\begin{cases} d_e = d_c + 3\ [mm] = 21\ [mm] \\ l_e = 0.5 * d_p = 14\ [mm] \end{cases} \quad (65)$$

4.3.4 Bearings diameter

d_r must be a multiple of 5

$$d_r = d_e + 4\ [mm] = 30\ [mm] \quad (66)$$

4.3.5 Shoulder diameter and length

$$\begin{cases} du = d + 5\ [mm] = 57\ [mm] \\ l_u = 5\ [mm] \end{cases} \quad (67)$$

4.4 Choosing longitudinal parallel key

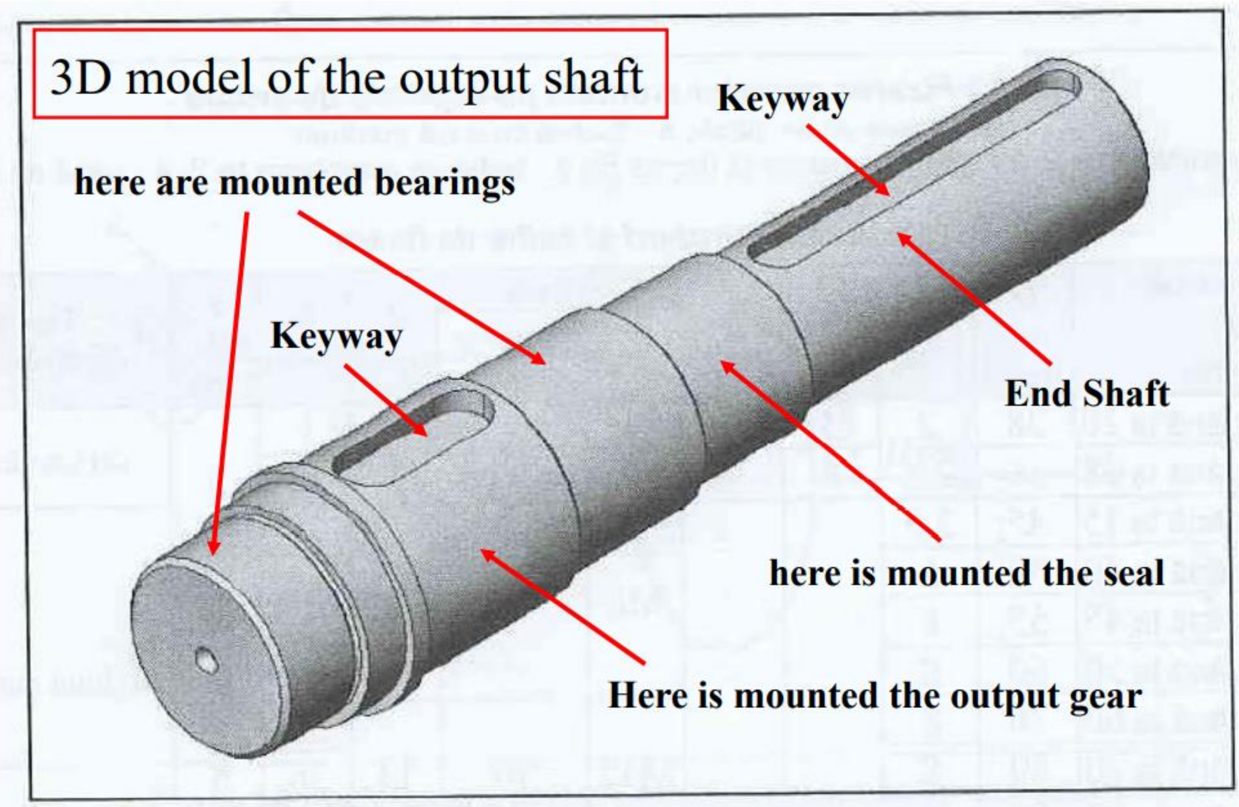


Fig. 4.4 *3D model of the output shaft [9]*

4.4.1 Stress analysis of parallel keys

A key has 2 failure mechanisms :

- it can be sheared off
- it can be crushed

4.4.2 Calculate the required key length at contact stress

A key with rounded edges will be used for mounting the gear

Dimensions of the first gear based on d = 52 [mm] = 5.2 [cm]

$$b = 16 \ [mm] = 0.6 \ [cm]$$
$$h = 10 \ [mm] = 0.6 \ [cm]$$
$$t1 = 6 \ [mm] = 0.6 \ [cm]$$
$$t2 = 4.3 \ [mm] = 0.43 \ [cm]$$

$$l_{min} = \frac{4 * T_2}{h * d * p_a} = 2.04 \ [cm] = 20.4 \ [mm] \quad (68)$$

The final length of the key:

$$l_p = 0.8 * \ l_c = 50 \ and \ is > l_{min} \quad (69)$$

$l_{pa} \geq \ l_p => l_{pa} = 56 \ [mm] = 5.6$ [cm]

Verification of the shaft key at shear stress

$$\begin{cases} \tau_f = \frac{2 * T_2}{b * d * l_{pa}} = 910.9 \left[\frac{daN}{cm^2}\right] < \tau_{af} \left[\frac{daN}{cm^2}\right] \\ \tau_{af} = 960 \left[\frac{daN}{cm^2}\right] - OL50 \end{cases} \quad (70)$$

Dimensions for the second key – 2 identical keys placed at a difference of 180 degrees:

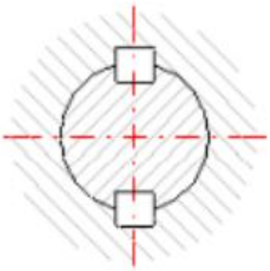


***Fig. 4.5**. Struct with 2 keys at diff of 180 degrees [10]*

Dimensions of the second gear based on d_c = 18:

$$b = 8 \ [mm] = 0.8 \ [cm]$$
$$h = 7 \ [mm] = 0.7 \ [cm]$$
$$t_1 = 4 \ [mm] = 0.4 \ [cm]$$
$$t_2 = 3.3 \ [mm] = 0.33 \ [cm]$$

Dimensioning at contact:

$$\begin{cases} l_{min} = \dfrac{4 * T_2}{1.5 * h * d_c * p_a} = 4.04 \ [cm] = 40.4 \ [mm]. \\ p_a = 800 \left[\dfrac{daN}{cm^2}\right] - OL50 \end{cases} \quad (71)$$

$$l_p = 50\ [mm]$$

After checking the chart and confirming that l_p (23) agrees with the interval (14 – 70), we choose the final standardized length of the keys:

$$l_{pb} = 56[mm]$$

$$\begin{cases} \tau_f = \dfrac{2 * T_2}{1.5 * b * d_c * l_{pb}} = 252.64 \left[\dfrac{daN}{cm^2}\right] < \tau_{af} \left[\dfrac{daN}{cm^2}\right] \\ \tau_{af} = 960 \left[\dfrac{daN}{cm^2}\right] - OL50 \end{cases} \quad (72)$$

4.5 Verification of shaft deflection and critical speed

4.5.1 Check results of calculation - deflection

Deflection in vertical plane :

$$\begin{cases} f_V = \dfrac{F_{t2} * l^3}{48 * E * I} [cm] = 0.009\ [cm] \\ E = 2.1 * 10^6 \left[\dfrac{daN}{cm^2}\right] \\ I = \pi * \dfrac{d^4}{64} = 3.58\ [cm] \end{cases} \quad (73)$$

Deflection in horizontal plane:

$$\begin{cases} f_H = \frac{F_{r2} * l^3}{48 * E * I} [cm] = 0.0004\ [cm] \\ E = 2.1 * 10^6 \left[\frac{daN}{cm^2}\right] \\ I = \pi * \frac{d^4}{64}\ [cm] \end{cases} \quad (74)$$

The final value:

$$f = \sqrt{f_V^2 + f_H^2} = 0.00096[cm] \quad (75)$$

4.5.2 Check results of calculation - vibration

Weight of the output gear:

$$\begin{cases} G = \gamma * V = \gamma * \frac{\pi * D_d^2}{4} * B_2 = 239.3\ [daN] \\ \gamma = 7.8 * 10^{-3} \left[\frac{daN}{cm^3}\right] - for\ steel \end{cases} \quad (76)$$

Static deflection:

$$\begin{cases} f_{st} = \frac{G*l^3}{48*E*I}\ [cm] = 0.00013\ [cm] \\ E = 2.1 * 10^6 \left[\frac{daN}{cm^2}\right] \\ I = \pi * \frac{d^4}{64}\ [cm] \end{cases} \tag{77}$$

Computing the critical speed:

$$\begin{cases} n_{cr} = \frac{30}{\pi} * \sqrt{\frac{g}{f_{st}}}\ [RPM] = 8371.5\ [RPM] \\ g = 981 \left[\frac{cm}{s^2}\right] \end{cases} \tag{78}$$

Chapter 5. Rolling bearing selection

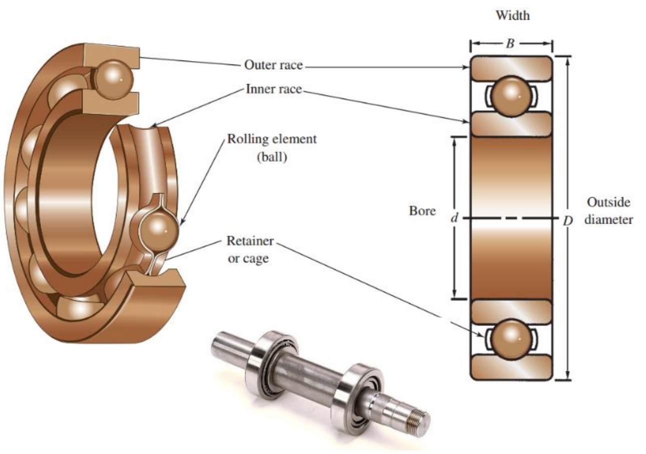


Fig. 5.1 *Deep groove ball bearings, single row [11]*

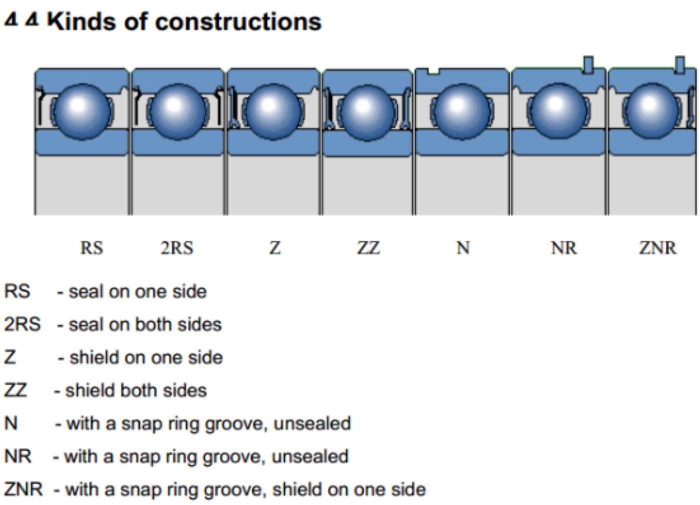


Fig. 5.2 *Types of constructions[12]*

5.1. Fundamental Calculation

The rated life L of the rolling bearing, is the life that is achieved or exceeded by 90% of identical bearings under the same operational conditions. It will be computed using the following formula [14] :

$$L = \frac{60 * n_2 * L_h}{10^6} = 2160\ [million\ of\ rotation] \tag{79}$$

$$F_e = F_r = R_5 = R_7 = \sqrt{R_{V5}^2 + R_{H5}^2} = 90{,}3\ [daN] \tag{80}$$

The minimum value for the basic dynamic load rating:

$$C_{min} = L^{0.33} * F_e = 544.79\ [daN] = 5.44\ [kN] \tag{81}$$

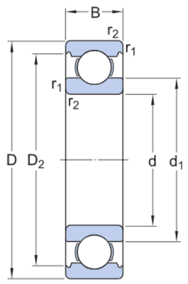
5.2 Bearing selection from SKF catalogue

Select from the SKF catalogue the bearing that suits best

W 61906	30	47	9	6.24	5	30 000	19 000

Fig. 5.3. *Deep Groove Ball Bearings[13]*

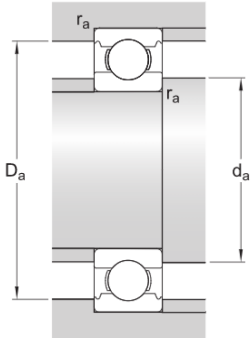
5.3 Technical specification of the selected bearing



Dimensions

d	30 mm	Bore diameter
D	47 mm	Outside diameter
B	9 mm	Width
d_1	≈ 35.1 mm	Shoulder diameter
D_2	≈ 44.08 mm	Recess diameter
$r_{1,2}$	min. 0.3 mm	Chamfer dimension

Fig. 5.4. *Dimensions [14]*



Abutment dimensions

d_a	min. 32 mm	Diameter of shaft abutment
D_a	max. 45 mm	Diameter of housing abutment
r_a	max. 0.3 mm	Radius of shaft or housing fillet

Fig. 5.5 *Abutment Dimensions [15]*

REFERENCES

[1] What Are Transmissions and Speed Reducers? Their Types and StructuresWhat Are Transmissions and Speed Reducers? Their Types and Structures| Miki Pulley
[2] Completion of the Design of a Power Transmission
[3], [4], [5], [6], [7], [8], [9], [10], [11], [12] Project Guide | Ph.D Radu Ciprian Rad
[13] -Deep groove ball bearings | SKF
[14] W 61906 - Stainless steel deep groove ball bearing
[15] W 61906 - Stainless steel deep groove ball bearing

ANNEXES

A1 - Technical drawing of the output shaft