HCM UNIVERSITY OF TECHNOLOGY

MACHINE ELEMENTS

ME2007

Project Report

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Design Problem

D_{bc}	pulley diameter, mm
F_t	tangential force, N
L	service life, years
T	working torque, $\mathrm{N \cdot mm}$
t	working time, s
v_{bc}	conveyor belt speed, m/s
δ_u	error of speed ratio, %

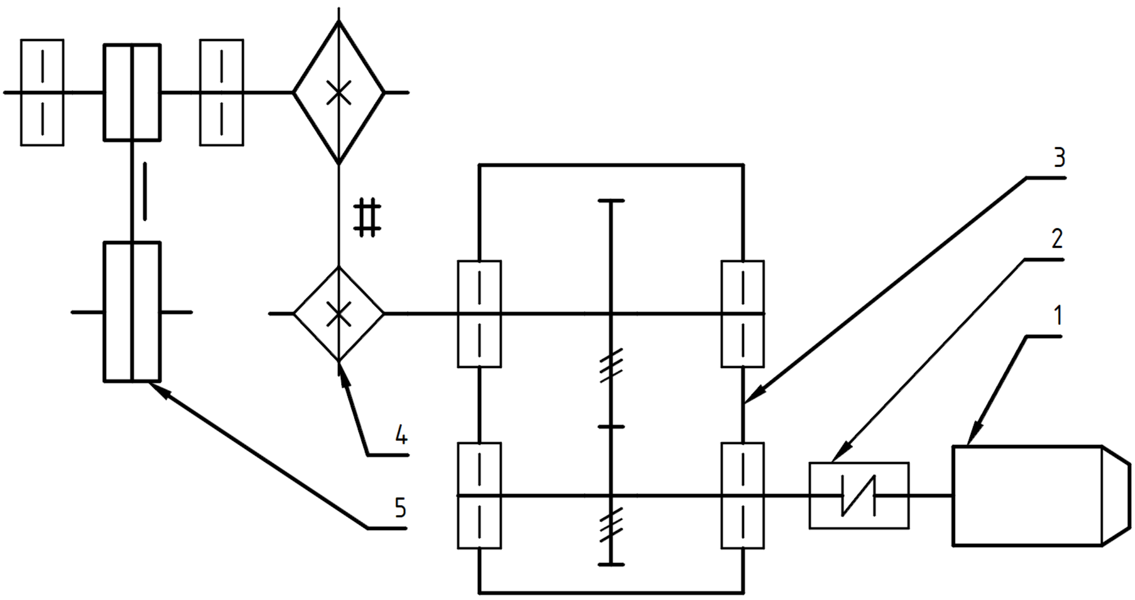

Figure 1: Mechanical transmission system of a belt conveyor

Given the mechanical transmission system in figure 1, determine the specifications for each machine element.

1. Electric motor
2. Elastic coupling
3. Gearbox
4. Chain drive
5. Belt conveyor

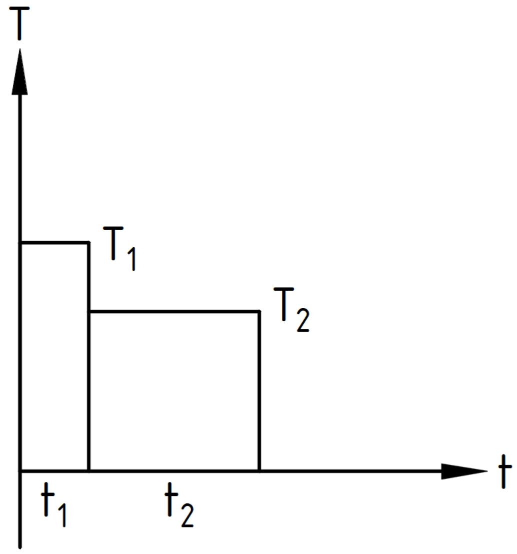

Figure 2: Input load diagram

Design parameters The chosen parameters are given in column 8:

- $F_t = 4500$ (N)
- $v_{bc} = 3.05$ (m/s)
- $D_{bc} = 500$ (mm)
- $L = 4$ (years)
- $T_1 = T$ (N · mm), $t_1 = 12$ (s)
- $T_2 = 0.7T$ (N · mm), $t_2 = 60$ (s)
- $\delta_u \leq \pm 5\%$

The machine works in one direction for 300 days, 2 shifts per day, 8 hours each and the load is low. Also, for the rest of this report, we will consider shaft 1 is the one connected to the electric motor, while shaft 2 is the connection between the chain drive and gearbox.

Chapter 1

Motor Design

1.1 Nomenclature

n_{bc}	rotational speed of belt conveyor, rpm
n_{sh}	rotational speed of shaft, rpm
P_m	maximum operating power of belt conveyor, kW
P_{motor}	calculated motor power to drive the system, kW
P_{sh}	operating power of shaft, kW
P_w	operating power of the belt conveyor given a workload, kW
u_{ch}	transmission ratio of chain drive
u_{hg}	transmission ratio of helical gear
u_{sys}	transmission ratio of the system
T_{motor}	motor torque, $\mathrm{N \cdot mm}$
T_{sh}	shaft torque, $\mathrm{N \cdot mm}$
η_b	bearing efficiency
η_c	coupling efficiency
η_{ch}	chain drive efficiency
η_{hg}	helical gear efficiency
η_{sys}	efficiency of the system
$_1$	shaft 1
$_2$	shaft 2

1.2 Calculate η_{sys}

From table (2.3):

$\eta_c = 1$

$\eta_b = 0.99$

$\eta_{hg} = 0.96$

$\eta_{ch} = 0.95$

$\eta_{sys} = \eta_c \eta_b^3 \eta_{hg} \eta_{ch} \approx 0.88$

1.3 Calculate P_{motor}

$P_m = \dfrac{F_t v_{bc}}{1000} \approx 13.73$ (kW)

From equation (2.13) :

$$P_w = P_m \sqrt{\frac{\left(\frac{T_1}{T}\right)^2 t_1 + \left(\frac{T_2}{T}\right)^2 t_2}{t_1 + t_2}} \approx 10.41 \text{ (kW)}$$

$P_{motor} = \dfrac{P_w}{\eta_{sys}} \approx 11.76$ (kW) $< P_m$

1.4 Calculate n_{motor}

$n_{bc} = \dfrac{6 \times 10^4 v}{\pi D_{bc}} \approx 116.5$ (rpm)

$u_{ch} = 5$ (table (2.4))

$u_{hg} = 5$ (table (2.4))

$u_{sys} = u_{ch} u_{hg} = 25$

$n_{motor} = u_{sys} n_{bc} \approx 2912.54$ (rpm)

1.5 Choose motor

To work normally, the maximum operating power of the chosen motor must be no smaller than both estimated P_{motor} and P_m. Since $P_{motor} < P_m$ for our case, the minimum operating power of choice is P_m. In similar fashion, its rotational speed must also be no smaller than estimated n_{motor}.

Thus, from table (P1.3), we choose motor 4A160M2Y3 which operates at 18.5 kW and 2930 rpm

$\Rightarrow P_{motor} = 18.5\,\text{kW}, n_{motor} = 2930\,(\text{rpm})$

Recalculating u_{sys} with the new P_{motor} and n_{motor}, we obtain:

$u_{sys} = \dfrac{n_{motor}}{n_{bc}} \approx 25.15$

Assuming $u_{hg} = const$:

$u_{ch} = \dfrac{u_{sys}}{u_{hg}} \approx 5.03$

1.6 Calculate power, rotational speed and torque

Let us denote P_{sh1}, n_{sh1} and T_{sh1} be the transmitted power, rotational speed and torque onto shaft 1, respectively. Similarly, P_{sh2}, n_{sh2} and T_{sh2} will be the transmitted parameters onto shaft 2. These notations will be used throughout the next chapters.

1.6.1 Power

$P_{ch} = P_m \approx 13.73\,(\text{kW})$

$P_{sh2} = \dfrac{P_{ch}}{\eta_b \eta_{ch}} \approx 14.59\,(\text{kW})$

$P_{sh1} = \dfrac{P_{sh2}}{\eta_b \eta_{hg}} \approx 15.35\,(\text{kW})$

1.6.2 Rotational speed

$n_{sh1} = n_{motor} = 2930$ (rpm)

$n_{sh2} = \dfrac{n_{sh1}}{u_{hg}} = 586$ (rpm)

1.6.3 Torque

$T_{motor} = 9.55 \times 10^6 \dfrac{P_{motor}}{n_{motor}} \approx 60298.63$ (N · mm)

$T_{sh1} = 9.55 \times 10^6 \dfrac{P_{sh1}}{n_{sh1}} \approx 50047.56$ (N · mm)

$T_{sh2} = 9.55 \times 10^6 \dfrac{P_{sh2}}{n_{sh2}} \approx 237825.99$ (N · mm)

In summary, we obtain the following table:

	Motor	Shaft 1	Shaft 2
P (kW)	18.5	14.59	15.35
u	5	5.03	
n (rpm)	2930	2930	586
T (N · mm)	60298.63	50047.56	237825.99

Table 1.1: System overall specifications

Chapter 2

Chain Drive Design

2.1 Nomenclature

$[i]$	permissible impact times per second
$[s]$	permissible safety factor
$[P]$	permissible power, kW
a	center distance, mm
a_{max}	maximum center distance, mm
a_{min}	minimum center distance, mm
B	bush length, mm
d	driving sprocket diameter, mm
d_c	pin diameter, mm
F_0	sagging force, N
F_1	tight side tension force, N
F_2	slack side tension force, N
F_r	force on the shaft, N
F_t	effective peripheral force, N
F_v	centrifugal force, N
i	impact times per second
k	overall factor
k_0	arrangement of drive factor
k_a	center distance and chain's length factor
k_{bt}	lubrication factor
k_c	rating factor
k_d	dynamic loads factor
k_{dc}	chain tension factor
k_f	loosing factor
k_n	coefficient of rotational speed
k_x	chain weight factor
k_z	coefficient of number of teeth

n_{01}	experimental rotational speed, rpm	v	instantaneous velocity along the chain, m/s
n_{ch}	rotational speed of a sprocket, rpm	x	chain length in pitches, the number of links
P_t	calculated power, kW	x_c	an even number of links
p	pitch, mm	z	number of teeth of a sprocket
p_{max}	permissible sprocket pitch, mm	z_{max}	maximum number of teeth of the driven sprocket
Q	permissible load, N		
q	mass per meter of chain, kg/m	$_1$	subscript for driving sprocket
s	safety factor	$_2$	subscript for driven sprocket

2.2 Find p

Since the driving sprocket is connected to shaft 1, $n_1 = n_{sh2} = 586$ (rpm).

Find z Since z_1 and z_2 is preferably an odd number according to p.80:

$z_1 = 29 - 2u_{ch} = 18.94 \approx 19$

$z_2 = u_{ch}z_1 = 95.57 \approx 97 \leq z_{max} = 120$

Because $z_1 \geq 15$, we use table (5.8) and interpolation to approximate p_{max}. Therefore, $p_{max} \approx 33.58$ (mm).

Find k Since $n_{ch} = 586 \approx 600$ (rpm), choose $n_{01} = 600$ (rpm), which is obtained from table (5.5). Then, we calculate k_z and k_n.

$k_z = \dfrac{25}{z_1} \approx 1.32$, $k_n = \dfrac{n_{01}}{n_{ch}} \approx 1.02$

Specifying the chain drive's working condition and ultizing table (5.6) , we find out that $k_0 = k_a = k_{dc} = k_{bt} = 1$, $k_d = 1.25$, $k_c = 1.3$.

$\Rightarrow k = k_0 k_a k_{dc} k_{bt} k_d k_c = 1.625$

Find p From table (5.5):

$P_t = P_{ch} k k_z k_n \approx 30.05\ (\text{kW}) \leq 42\ (\text{kW}) \Rightarrow [P] = 42\ (\text{kW})$

Using the table, we also get the other parameters:

$p = 31.75\ (\text{mm}),\ d_c = 9.55\ (\text{mm}),\ B = 27.46\ (\text{mm}),$

$$d_1 = \frac{p}{\sin\frac{180^\circ}{z_1}} \approx 192.9\ (\text{mm}),\ d_2 = \frac{p}{\sin\frac{180^\circ}{z_2}} \approx 980.49\ (\text{mm})$$

Having $p = 31.75\ (\text{mm}) \leq \text{p}_{\text{max}} \approx 33.58\ (\text{mm})$, we can safely choose the number of chains as 1, which is in agreement with the given figure. Hence, from table (5.2), we obtain the parameters in the section for 1 strand chain drive:

$Q = 56.7 \times 10^3\ (\text{N}),\ q = 2.6\ (\text{kg/m})$

By comparison to the conditions in the sub-table, the choice of B is satisfactory.

2.3 Find a, x_c, and i

Find x_c $a_{min} = 30p = 952.5\ (\text{mm}),\ a_{max} = 50p = 1587.5\ (\text{mm})$. Limiting the range of choice for a in $[a_{min}, a_{max}]$, we can approximate $a = 1000\ (\text{mm})$.

$$x = \frac{2a}{p} + \frac{z_1 + z_2}{2} + \frac{(z_2 - z_1)^2 p}{4\pi^2 a} \approx 125.89 \Rightarrow x_c = 126$$

Find a From equation (5.13) , we calculate a again with x_c:

$$a = \frac{p}{4}\left(x_c - \frac{z_2 + z_1}{2} + \sqrt{\left(x_c - \frac{z_2 + z_1}{2}\right)^2 - 2\frac{(z_2 - z_1)^2}{\pi^2}}\right) - 0.003a \approx 998.98\ (\text{mm})$$

Find i From table (5.9):

$$i = \frac{z_1 n_{sh2}}{15x} \approx 6 < [i] = 25$$

2.4 Strength of chain drive

For moderate workload, choose $k_d = 1.2$. Let the chain drive be angled 30° with respect to ground, we obtain $k_f = 4$.

$v_1 = \dfrac{n_{ch} p z_1}{6 \times 10^4} \approx 5.89$ (m/s)

Find F_t, F_v, F_0 We also need to calculate F_t, F_v and F_0:
$F_t = \dfrac{10^3 P_{ch}}{v_1} \approx 2329.53$ (N)
$F_v = q v_1^2 \approx 90.25$ (N)
$F_0 = 9.81 \times 10^{-3} k_f q a \approx 101.92$ (N)

Validate s This value must be larger than the permissible safety factor to operate properly. From equation (5.15):
$s = \dfrac{Q}{k_d F_t + F_0 + F_v} \approx 18.98 \geq [s] = 13.2$, where $[s]$ is chosen from table (5.10).

2.5 Force on shaft

From p.87:

$F_2 = F_0 + F_v \approx 192.17$ (N)

$F_1 = F_t + F_2 \approx 2521.7$ (N)

Choose $k_x = 1.15$ and follow equation (5.20) :

$F_r = k_x F_t \approx 2678.96$ (N)

In summary, we have the following table:

	driving	driven
$[P]$ (kW)	42	
Q (N)	56700	
p (mm)	31.75	
i	6	
a (mm)	998.98	
z	19	97
d (mm)	192.9	980.49
d_c (mm)	9.55	
B (mm)	27.46	
v (m/s)	5.01	
u_{ch}	5	

Table 2.1: Chain drive specifications

Chapter 3

Gearbox Design (Helix gears)

3.1 Nomenclature

$[\sigma_H]$	permissible contact stress, MPa
$[\sigma_F]$	permissible bending stress, MPa
$[\sigma_H]_{max}$	permissible contact stress due to overload, MPa
$[\sigma_F]_{max}$	permissible bending stress due to overload, MPa
AG	accuracy grade of gear
a	center distance, mm
b	face width, mm
c	gear meshing rate
d	pitch circle, mm
d_a	addendum diameter, mm
d_b	base diameter, mm
d_f	deddendum diameter, mm
F_a	axial force, N
F_r	radial force, N
F_t	tangential force, N
H	surface roughness, HB
K_d	coefficient of gear material
K_F	load factor from bending stress
K_{FC}	load placement factor
K_{FL}	aging factor due to bending stress
K_{Fv}	factor of dynamic load from bending stress at meshing area
$K_{F\alpha}$	factor ofload distribution from bending stress on gear teeth
$K_{F\beta}$	factor of load distribution from bending stress on top land

K_H	load factor of contact stress
K_{HL}	aging factor due to contact stress
K_{Hv}	factor of dynamic load from contact stress at meshing area
$K_{H\alpha}$	factor of load distribution from contact stress on gear teeth
$K_{H\beta}$	factor of load distribution from contact stress on top land
k_x	a coefficient
k_y	a coefficient
m	traverse module, mm
m_F	root of fatigue curve in bending stress test
m_H	root of fatigue curve in contact stress test
m_n	normal module, mm
N_{FE}	working cycle of equivalent tensile stress corresponding to $[\sigma_F]$
N_{FO}	working cycle of bearing stress corresponding to $[\sigma_F]$
N_{HE}	working cycle of equivalent tensile stress corresponding to $[\sigma_H]$
N_{HO}	working cycle of bearing stress corresponding to $[\sigma_H]$
S	length, mm
S_F	safety factor of bending stress
S_H	safety factor of contact stress
v	rotational velocity, m/s
x	gear correction factor
Y_F	tooth shape factor
Y_β	helix angle factor
Y_ϵ	contact ratio factor
y	center displacement factor
z_H	contact surface's shape factor
z_M	material's mechanical properties factor
z_{min}	minimum number of teeth corresponding to β
z_v	virtual number of teeth
z_ϵ	meshing condition factor
α	normal pressure angle, following Vietnam standard (TCVN 1065-71), i.e. $\alpha = 20°$
α_t	traverse pressure angle, °
ϵ_α	traverse contact ratio
ϵ_β	face contact ratio
β	helix angle, °
β_b	base circle helix angle, °
ψ_{ba}	width to shaft distance ratio
ψ_{bd}	face width factor
σ_b	ultimate strength, MPa
σ_{ch}	yield limit, MPa

σ^o_{Flim}	permissible bending stress corresponding to working cycle, MPa	$_1$	subscript for pinion
σ^o_{Hlim}	permissible contact stress corresponding to working cycle, MPa	$_2$	subscript for driven gear
		$_w$	subscript for variable value after correction

3.2 Choose material

From table (6.1) , the material of choice for both gears is steel 40X with $S \leq 100\,(\text{mm})$, HB250, $\sigma_b = 850\,(\text{MPa})$, $\sigma_{ch} = 550\,(\text{MPa})$.
Table (6.2) also gives $\sigma^o_{Hlim} = 2\text{HB} + 70$, $S_H = 1.1$, $\sigma^o_{Flim} = 1.8\text{HB}$, $S_F = 1.75$
Therefore, they have the same properties except for their surface roughness H.
For the pinion, $H_1 = \text{HB}250 \Rightarrow \sigma^o_{Hlim1} = 570\,(\text{MPa})$, $\sigma^o_{Flim1} = 450\,(\text{MPa})$
For the driven gear, $H_2 = \text{HB}240 \Rightarrow \sigma^o_{Hlim2} = 550\,(\text{MPa})$, $\sigma^o_{Flim2} = 432\,(\text{MPa})$

3.3 Calculate $[\sigma_H]$ and $[\sigma_F]$

3.3.1 Working cycle of bearing stress

Using equation (6.5) :
$N_{HO1} = 30H_1^{2.4} = 17.07 \times 10^6$ (cycles)
$N_{HO2} = 30H_2^{2.4} = 15.4749 \times 10^6$ (cycles)

3.3.2 Working cycle of equivalent tensile stress

Since $H_1, H_2 \leq \text{HB}350$, $m_H = 6$, $m_F = 6$.
Both gears meshed indefinitely, thus $c = 1$.
Applying equation (6.7) and T_1, T_2, t_1, t_2 in the initial parameters:

$$N_{HE1} = 60c\left[\left(\frac{T_1}{T}\right)^3 n_1t_1 + \left(\frac{T_2}{T}\right)^3 n_2t_2\right] \approx 5.73\times 10^6 \text{ cycles } (n_1 = n_2 = n_{sh1})$$
$$N_{HE2} = 60c\left[\left(\frac{T_1}{T}\right)^3 n_1t_1 + \left(\frac{T_2}{T}\right)^3 n_2t_2\right] \approx 1.15\times 10^6 \text{ cycles } (n_1 = n_2 = n_{sh2})$$
$$N_{FE1} = 60c\left[\left(\frac{T_1}{T}\right)^{m_F} n_1t_1 + \left(\frac{T_2}{T}\right)^{m_F} n_2t_2\right] \approx 3.35\times 10^6 \text{ cycles } (n_1 = n_2 = n_{sh1})$$
$$N_{FE2} = 60c\left[\left(\frac{T_1}{T}\right)^{m_F} n_1t_1 + \left(\frac{T_2}{T}\right)^{m_F} n_2t_2\right] \approx 0.67\times 10^6 \text{ cycles } (n_1 = n_2 = n_{sh2})$$

3.3.3 Aging factor

For steel, $N_{FO1} = N_{FO2} = 4\times 10^6$ (MPa). Applying equations (6.3) and (6.4) yield:

$K_{HL1} = \sqrt[m_H]{N_{HO1}/N_{HE1}} \approx 1.2$
$K_{HL2} = \sqrt[m_H]{N_{HO2}/N_{HE2}} \approx 1.54$
$K_{FL1} = \sqrt[m_F]{N_{FO1}/N_{FE1}} \approx 1.03$
$K_{FL2} = \sqrt[m_F]{N_{FO2}/N_{FE2}} \approx 1.35$

3.3.4 Calculate $[\sigma_H]$ and $[\sigma_F]$

Since the motor works in one direction, $K_{FC} = 1$
$[\sigma_{H1}] = \sigma^o_{Hlim1}K_{HL1}/S_{H1} \approx 621.61$ (MPa)
$[\sigma_{H2}] = \sigma^o_{Hlim2}K_{HL2}/S_{H2} \approx 771.63$ (MPa)
$[\sigma_{F1}] = \sigma^o_{Flim1}K_{FC1}K_{FL1}/S_{F1} \approx 264.85$ (MPa)
$[\sigma_{F2}] = \sigma^o_{Flim2}K_{FC2}K_{FL2}/S_{F2} \approx 332.48$ (MPa)
The permissible contact stress due to overload must be lower than 1.25 times of either $[\sigma_{H1}]$ or $[\sigma_{H2}]$, whichever is smaller. For permissible bending stress, it is equal to either $[\sigma_{F1}]$ or $[\sigma_{F2}]$, whichever is larger:
$[\sigma_H]_{max} = \frac{1}{2}\left([\sigma_{H1}] + [\sigma_{H2}]\right) \approx 696.62 \text{ (MPa)} \leq 1.25[\sigma_H]_{min} = 1.25[\sigma_{H1}]$
$[\sigma_F]_{max} = [\sigma_{F2}] = 0.8\sigma_{ch} \approx 440$ (MPa)

3.4 Transmission Design

3.4.1 Determine basic parameters

Examine table (6.5) gives $K_a = 43$

Assuming symmetrical design, table (6.6) also gives $\psi_{ba} = 0.5$

$\Rightarrow \psi_{bd} = 0.53\psi_{ba}(u_{hg} + 1) = 1.59$

From table (6.7) , using interpolation, we approximate $K_{H\beta} \approx 1.108$, $K_{F\beta} \approx 1.2558$

Since the gear system only consists of involute gears and it is also a speed reducer gearbox, we estimate a using equation (6.15a) gives:

$$a = K_a(u_{hg} + 1)\sqrt[3]{\frac{T_{sh1}K_{H\beta}}{[\sigma_H]^2 u_{hg}\psi_{ba}}} \approx 91.94\,(\text{mm})$$

According to SEV229-75 standard, we choose $a_w = 100\,\text{mm}$

3.4.2 Determine gear meshing parameters

Find m Applying equation (6.17) and choose m from table (6.8) :

$m = (0.01 \div 0.02)a_w \approx (0.92 \div 1.84)\,(\text{mm}) \Rightarrow m = 1.5\,(\text{mm})$

Find z_1, z_2, b_w Let $\beta = 15°$. Combining equation (6.18) and (6.20), we come up with the formula to calculate z_1. From the result, z_1 is rounded up to the nearest odd number (preferably a prime number).

$$z_1 = \frac{2a_w \cos\beta}{m(u_{hg} + 1)} \approx 19.73 \Rightarrow z_1 = 21$$

$z_2 = u_{hg}z_1 = 105$

$\Rightarrow b_w = \psi_{ba}a_w = 50\,(\text{mm})$

Recalculate β There are 2 approaches for correction involving the change of either α or β. Because altering α leads to many other corrections (d_1, d_2 and a), β will be used instead.

Since z_1 is rounded, we must find β to obtain the correct angle, ensuring that $\beta \in (8°, 20°)$. Using equation (6.32):
$\beta = \arccos \dfrac{m(z_1 + z_2)}{2a_w} \approx 19.09°$

Find x_1, x_2 To find x_1 and x_2, we will follow the calculation scheme provided in p.103. Since $\beta \approx 19.09° \in (17, 21]$, $z_{min} = 15$, which leads to z_1 satisfying condition $z_1 \geq z_{min} + 2 > 10$, according to table (6.9). Combined with $u_{hg} = 5 \geq 3.5$, we obtain $x_1 = 0.3$, $x_2 = -0.3$, disregarding the calculation of y.

3.4.3 Other parameters

$$d_1 = d_{w1} = \frac{mz_1}{\cos\beta} \approx 33.33\ \text{(mm)}$$
$$d_2 = d_{w2} = \frac{mz_2}{\cos\beta} \approx 166.67\ \text{(mm)}$$
$$d_{a1} = d_1 + 2(1 + x_1)m \approx 37.23\ \text{(mm)}$$
$$d_{a2} = d_2 + 2(1 + x_2)m \approx 168.77\ \text{(mm)}$$
$$d_{f1} = d_1 - (2.5 - 2x_1)m \approx 30.48\ \text{(mm)}$$
$$d_{f2} = d_2 - (2.5 - 2x_2)m \approx 162.02\ \text{(mm)}$$
$$d_{b1} = d_1 \cos\alpha \approx 31.32\ \text{(mm)}$$
$$d_{b2} = d_2 \cos\alpha \approx 156.62\ \text{(mm)}$$
$$\alpha_t = \alpha_{tw} = \arctan\frac{\tan\alpha}{\cos\beta} \approx 20.65°$$
$$v = \frac{\pi d_1 n_{sh1}}{6 \times 10^4} \approx 5\ \text{(m/s)}$$

3.4.4 Contact stress analysis

From section 6.3.3. in the text, contact stress applied on a gear surface must satisfy the condition below:

$$\sigma_H = z_M z_H z_\epsilon \sqrt{2T_{sh1} K_H \frac{u_{hg} + 1}{b_w u_{hg} d_{w1}^2}} \leq [\sigma_H]$$

Find z_M $z_M = 274$, according to table (6.5)

Find z_H $\beta_b = \arctan(\cos\alpha_t \tan\beta) \approx 17.94° \Rightarrow z_H = \sqrt{2\dfrac{\cos\beta_b}{\sin(2\alpha_t)}} \approx 1.7$

Find z_ϵ Obtaining z_ϵ through calculations:

$$\epsilon_\alpha = \frac{\sqrt{d_{a1}^2 - d_{b1}^2} + \sqrt{d_{a2}^2 - d_{b2}^2} - 2a_w \sin\alpha_t}{2\pi m \dfrac{\cos\alpha_t}{\cos\beta}} \approx 1.34$$

$$\epsilon_\beta = b_w \frac{\sin\beta}{m\pi} \approx 3.47 > 1 \Rightarrow z_\epsilon = \epsilon_\alpha^{-0.5} \approx 0.86$$

Find K_H We find K_H using equation $K_H = K_{H\beta} K_{H\alpha} K_{Hv}$

From table (6.13), $v \le 6\,(\mathrm{m/s}) \Rightarrow \mathrm{AG} = 8$

From table (P2.3), using interpolation, we approximate:

$K_{Hv} \approx 1.05$, $K_{Fv} \approx 1.14$

From table (6.14), using interpolation, we approximate:

$K_{H\alpha} \approx 1.09$, $K_{F\alpha} \approx 1.27$

$\Rightarrow K_H \approx 1.27$

Find σ_H After calculating z_M, z_H, z_ϵ, K_H, we get the following result:

$$\sigma_H \approx 663.86\,\mathrm{MPa} \le [\sigma_H] \approx 696.62\,\mathrm{MPa}$$

3.4.5 Bending stress analysis

For safety reasons, the following conditions must be met:

$$\begin{aligned} \sigma_{F1} &= 2\frac{T_{sh1} K_F Y_\epsilon Y_\beta Y_{F1}}{b_w d_{w1} m_n} && \le [\sigma_{F1}] \\ \sigma_{F2} &= \frac{\sigma_{F1} Y_{F2}}{Y_{F1}} && \le [\sigma_{F2}] \end{aligned}$$

Find Y_ϵ Knowing that $\epsilon_\alpha \approx 1.64$, we can calculate $Y_\epsilon = \epsilon_\alpha^{-1} \approx 0.75$

Find Y_β $Y_\beta = 1 - \dfrac{\beta}{140} \approx 0.86$

Find Y_F Using formula $z_v = z\cos^{-3}(\beta)$ and table (6.18):

$z_{v1} = z_1\cos^{-3}(\beta) \approx 24.88 \Rightarrow Y_{F1} \approx 3.6$

$z_{v2} = z_2\cos^{-3}(\beta) \approx 124.42 \Rightarrow Y_{F2} \approx 3.64$

Find K_F Using $K_{F\beta}$, $K_{F\alpha}$, K_{Fv} calculated from the sections above, we derive:

$K_F = K_{F\beta}K_{F\alpha}K_{Fv} \approx 1.82$

Find σ_F Since $m_n = m\cos\beta \approx 1.42$, substituting all the values, we find out that:

$$\sigma_{F1} \approx 179.07\,(\text{MPa}) \leq [\sigma_{F1}] \approx 264.85\,(\text{MPa})$$

$$\sigma_{F2} \approx 181.06\,(\text{MPa}) \leq [\sigma_{F2}] \approx 332.48\,(\text{MPa})$$

3.4.6 Force on shafts

$F_t = \dfrac{2T_{sh1}}{d_1} \approx 3003.15\,(\text{N})$

$F_r = F_t\tan\alpha_t \approx 1131.8\,(\text{N})$

$F_a = F_t\tan\beta \approx 1039.35\,(\text{N})$

In summary, we have the following table:

	pinion	driving gear
H (HB)	250	240
$[\sigma_H]$ (MPa)	621.61	771.63
$[\sigma_F]$ (MPa)	264.85	332.48
$[\sigma_H]_{max}$ (MPa)	696.62	
$[\sigma_F]_{max}$ (MPa)	440	
σ_H (MPa)	621.61	771.63
σ_F (MPa)	179.07	181.06
σ_H (MPa)	663.86	
α_{tw} (°)	20.65	
β (°)	19.09	
a_w (mm)	100	
b_w (mm)	50	
m (mm)	1.5	
z	21	105
d (mm)	33.33	166.67
d_a (mm)	37.23	168.77
d_f (mm)	30.48	162.02
d_b (mm)	31.32	156.62
v (m/s)	5	
u_{hg}	5	

Table 3.1: Gearbox specifications

Chapter 4

Shaft Design

4.1 Nomenclature

- $[\tau]$ permissible torsion, MPa
- r position of applied force on the shaft, mm
- hr tooth direction
- cb role of gear on the shaft (active or passive)
- cq rotational direction of the shaft
- σ_b ultimate strength, MPa
- σ_{ch} yield limit, MPa
- S safety factor
- F_x applied force, N
- F_t tangential force, N
- F_r radial force, N
- F_a axial force, N
- a_w shaft distance, mm
- d shaft diameter, mm
- d_w gear diameter, mm
- $_x$ subscript for x-axis
- q standardized coefficient of shaft diameter
- b_O rolling bearing width, mm
- l_m hub diameter, mm
- k_1 distance between elements, mm
- k_2 distance between bearing surface and inner walls of the gearbox, mm
- k_3 distance between element surface and bearing lid, mm
- h_n distance between bearing lid and bolt, mm
- T torque on shaft
- α_{tw} meshing profile angle, $^\circ$
- β helix angle, $^\circ$
- $_1$ subscript for shaft 1
- $_2$ subscript for shaft 2

4.2 Choose material

For moderate load, we will use quenched steel 40X to design the shafts. From table 6.1, the specifications are as follows: $S \leq 100\,(\text{mm})$, HB260, $\sigma_b = 850\,(\text{MPa})$, $\sigma_{ch} = 550\,(\text{MPa})$.

4.3 Tranmission Design

4.3.1 Load on shafts

Applied forces from Gears

Following p.186, the subscript convention of the book will be used in this chapter. If a symbol has 2 numeric subscripts, the first one is the ordinal number of shafts while the second one is used for machine elements.
On shaft 1, the motor is labeled 1 and the pinion is labeled 2. On shaft 2, the driven gear is labeled 1 and the driving sprocket is labeled 2. Therefore, we obtain:
$r_{12} = -d_{w12}/2 \approx -13.57\,(\text{mm})$, $hr_{12} = +1$, $cb_{12} = +1$, $cq_1 = +1$
$r_{21} = +d_{w21}/2 \approx +67.84\,(\text{mm})$, $hr_{21} = -1$, $cb_{21} = -1$, $cq_2 = -1$

Find magnitude of F_t, F_r, F_a Using the results from the previous chapter: $\alpha_{tw} \approx 21.17°$, $\beta = 20°$, $d_{w12} \approx 27.14\,(\text{mm})$

$$\begin{cases} F_{t12} = F_{t21} = \dfrac{2T_{sh1}}{d_{w12}} \approx 2769.03\,(\text{N}) \\ F_{r12} = F_{r21} = \dfrac{F_{t12}\tan\alpha_{tw}}{\cos\beta} \approx 1141.36\,(\text{N}) \\ F_{a12} = F_{a21} = F_{t12}\tan\beta \approx 1007.84\,(\text{N}) \end{cases}$$

Find direction of F_t, F_r, F_a Following the sign convention, we obtain the forces:

$$\begin{cases} F_{x12} = \dfrac{r_{12}}{|r_{12}|} cq_1 cb_{12} F_{t12} \approx -2769.03\,(\text{N}) \\ F_{y12} = -\dfrac{r_{12}}{|r_{12}|} \dfrac{\tan\alpha_{tw}}{\cos\beta} F_{t12} \approx 1141.36\,(\text{N}) \\ F_{z12} = cq_1 cb_{12} hr_{12} F_{t12} \tan\beta \approx 1007.84\,(\text{N}) \end{cases}$$

$$\begin{cases} F_{x21} = \dfrac{r_{21}}{|r_{21}|} cq_2 cb_{21} F_{t21} \approx 2769.03\,(\text{N}) \\ F_{y21} = -\dfrac{r_{21}}{|r_{21}|} \dfrac{\tan\alpha_{tw}}{\cos\beta} F_{t21} \approx -1141.36\,(\text{N}) \\ F_{z21} = cq_2 cb_{21} hr_{21} F_{t21} \tan\beta \approx -1007.84\,(\text{N}) \end{cases}$$

Applied forces from Chain drives

Assuming the angle between x-axis and F_r is 210° and $F_r \approx 2539.28$ (N) (chapter 2), we get the direction of F_r on shaft 2:

$$\begin{cases} F_{x22} = F_{r22} \cos 210^\circ \approx -1269.64\,(\text{N}) \\ F_{y22} = F_{r22} \sin 210^\circ \approx -2199.08\,(\text{N}) \end{cases}$$

4.3.2 Preliminary calculations

Since shaft 1 receives input torque T_{sh1} and shaft 2 produces output torque T_{sh2}, $[\tau_1] = 15\,(\text{MPa})$ and $[\tau_2] = 30\,(\text{MPa})$. Using equation (10.9), we can approximate d_1 and d_2:

$d_1 \geq \sqrt[3]{\dfrac{T_{sh1}}{0.2[\tau_1]}} \approx 23.22\,(\text{mm}) \Rightarrow d_1 = 25\,(\text{mm})$

$d_2 \geq \sqrt[3]{\dfrac{T_{sh2}}{0.2[\tau_2]}} \approx 30.99\,(\text{mm}) \Rightarrow d_2 = 35\,(\text{mm})$

4.3.3 Identify the distance between bearings and applied forces

In this section, we will find all the parameters in Figure 4.1. However, if a parameter has 2 numeric subscripts, the first one will denote the ordinal number of shafts.

From table (10.2), we can estimate b_O. On shaft 1, $b_{O1} = 15$ (mm). On shaft 2, $b_{O2} = 21$ (mm). Using equation (10.10), the gear hubs are $l_{m13} = l_{m12} = 1.5d_1 \approx 34.83$ (mm), $l_{m23} = l_{m22} = 1.5d_2 \approx 46.48$ (mm) (l_{m22} is the chain hub)

From table (10.3), we choose $k_1 = 10$ (mm), $k_2 = 8$ (mm), $k_3 = 15$ (mm), $h_n = 18$ (mm). This parameters apply for both shafts in the system.

Table (10.4) introduce the formulas for several types of gearbox. Since our system only concerns about 1-level gear reducer, the formulas below are used:

On shaft 1:

$l_{12} = -l_{c12} = -\left[0.5(l_{m12} + b_{O1}) + k_3 + h_n\right] \approx -57.92$ (mm)

$l_{13} = 0.5(l_{m13} + b_{O1}) + k_1 + k_2 \approx 42.92$ (mm)

$l_{11} = 2l_{13} \approx 85.83$ (mm)

On shaft 2:

$l_{22} = -l_{c22} = -\left[0.5(l_{m22} + b_{O2}) + k_3 + h_n\right] \approx -66.74$ (mm)

$l_{23} = 0.5(l_{m23} + b_{O2}) + k_1 + k_2 \approx 51.74$ (mm)

$l_{21} = 2l_{23} \approx 103.48$ (mm)

4.3.4 Determine shaft diameters and lengths

From the diagram, we derive the reaction forces at A_1, A_2, B_1, B_2, which are R_{A1x}, R_{A1y}, R_{B1x}, R_{B1y}, R_{A2x}, R_{A2y}, R_{B2x}, R_{B2y}. Using equilibrium conditions:

$$\begin{cases} \sum_i \mathbf{F_i} &= 0 \\ \sum_i \mathbf{r_i} \times \mathbf{F_i} &= 0 \end{cases}$$

We obtain the results as follows:

From the reaction forces, we can easily draw shear force-bending moment diagram

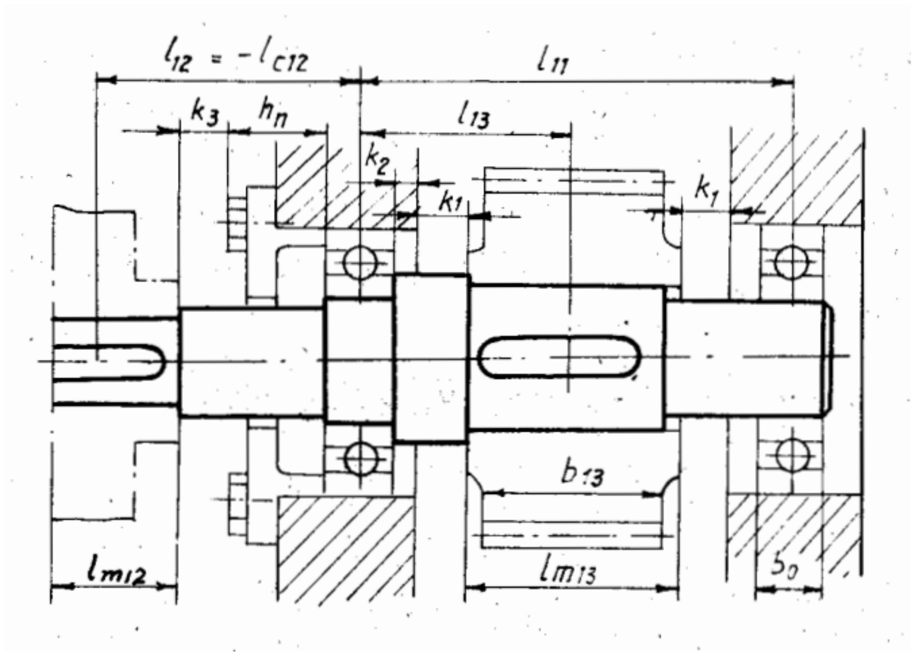

Figure 4.1: Shaft design and its dimensions

$$\begin{cases} R_{A1x} \approx 1384.51 \text{ (N)} \\ R_{A1y} \approx -570.68 \text{ (N)} \\ R_{B1x} \approx 1384.51 \text{ (N)} \\ R_{B1y} \approx -570.68 \text{ (N)} \end{cases} \qquad \begin{cases} R_{A2x} \approx 703.98 \text{ (N)} \\ R_{A2y} \approx 4188.06 \text{ (N)} \\ R_{B2x} \approx -2203.37 \text{ (N)} \\ R_{B2y} \approx -847.62 \text{ (N)} \end{cases}$$

for both shafts on 2 major planes (xOz) and (yOz).

From equation (10.15), we calculate the total bending moment at point C_2, A_2, D_2, B_2, A_1, D_1, B_1, C_1

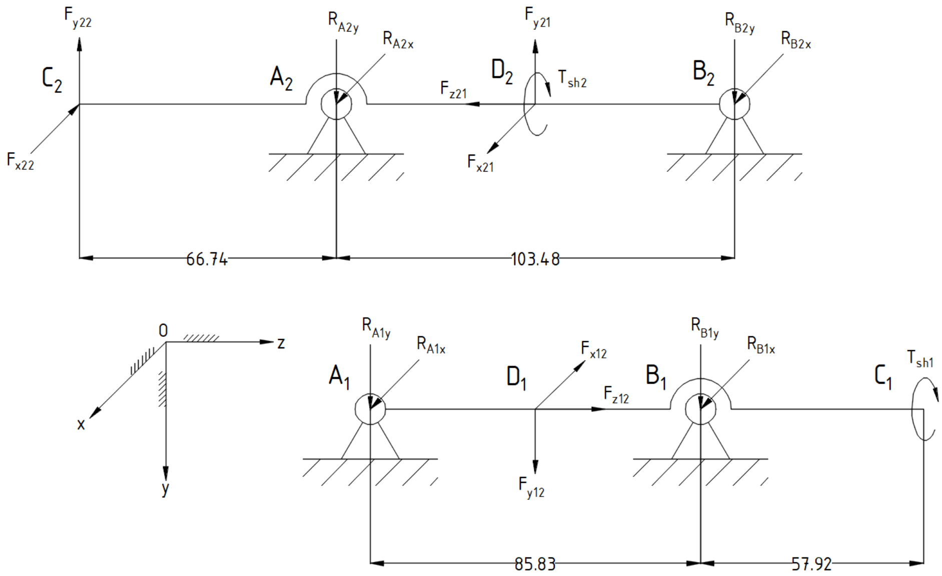

Figure 4.2: Force analysis of shafts

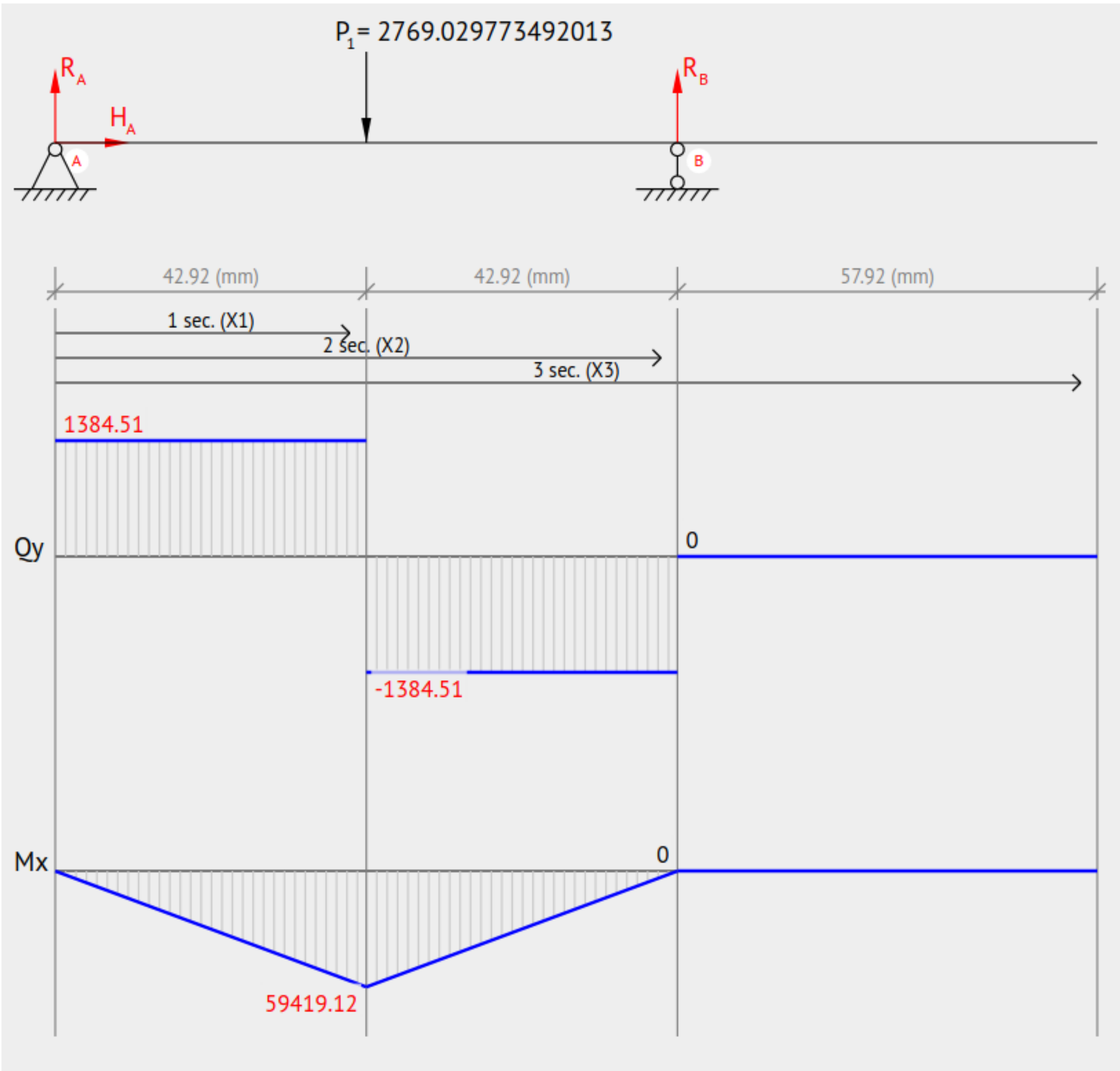

Figure 4.3: Shear force - Bending moment diagram on (xOz) of shaft 1

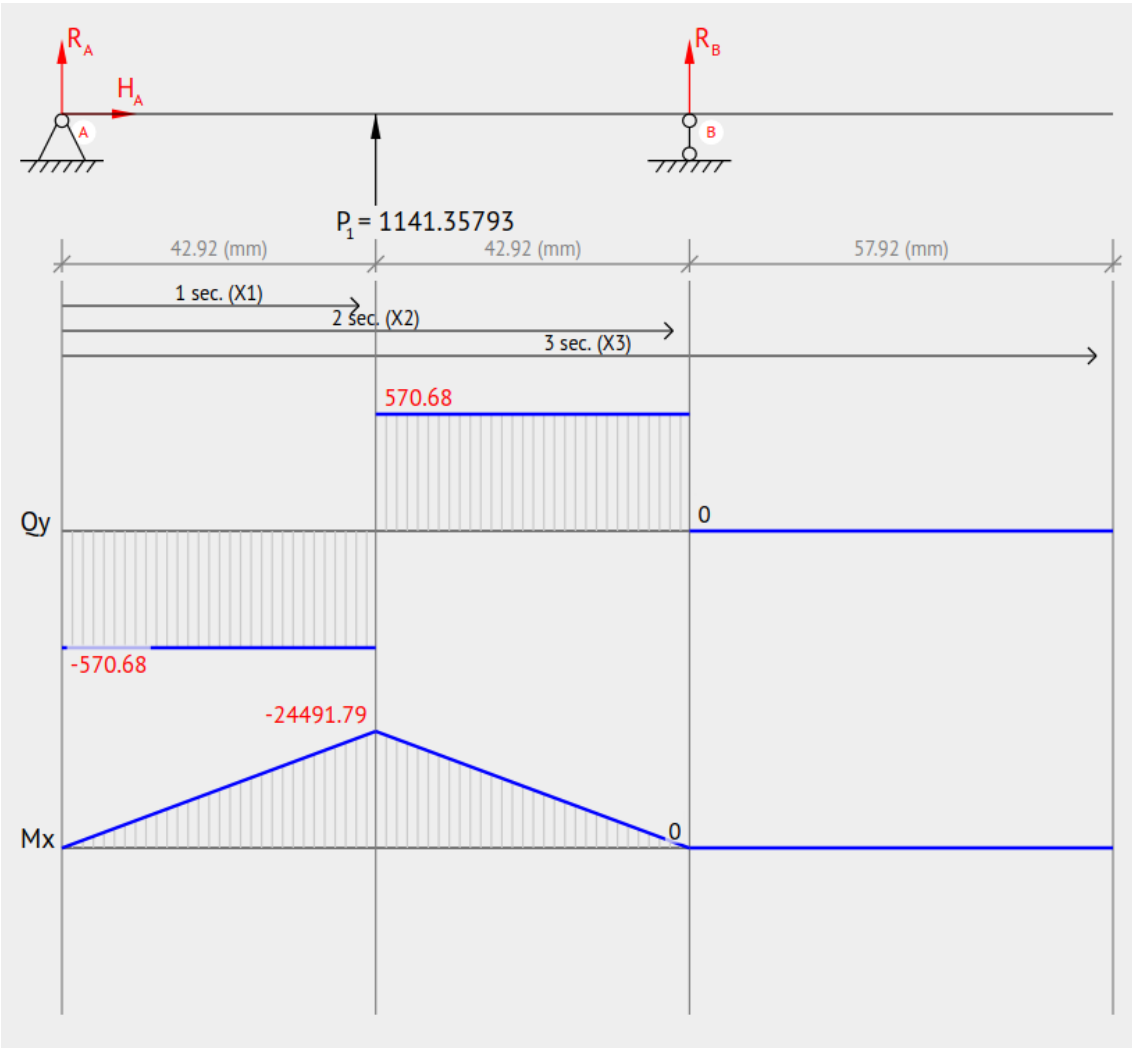

Figure 4.4: Shear force - bending moment diagram on (yOz) of shaft 1

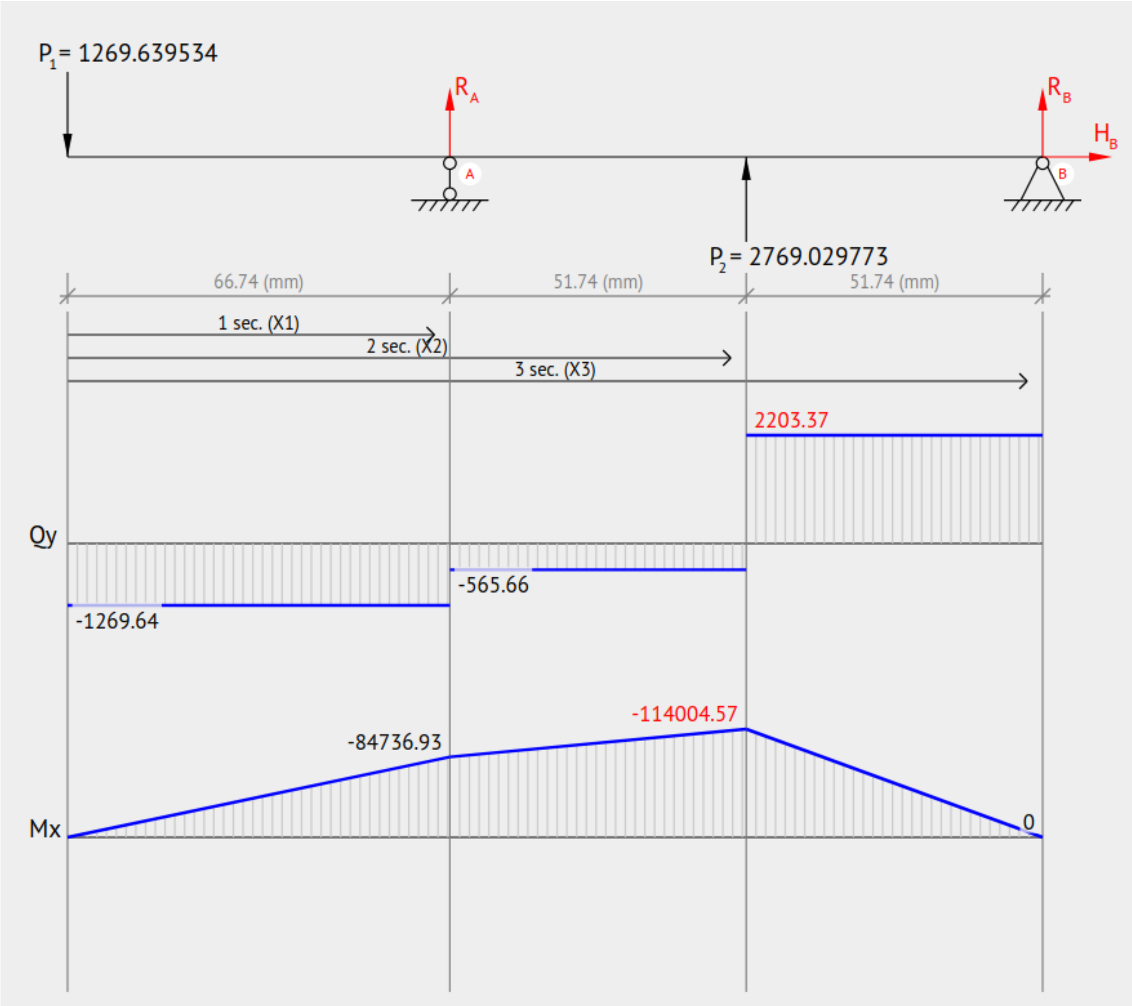

Figure 4.5: Shear force - Bending moment diagram on (xOz) of shaft 2

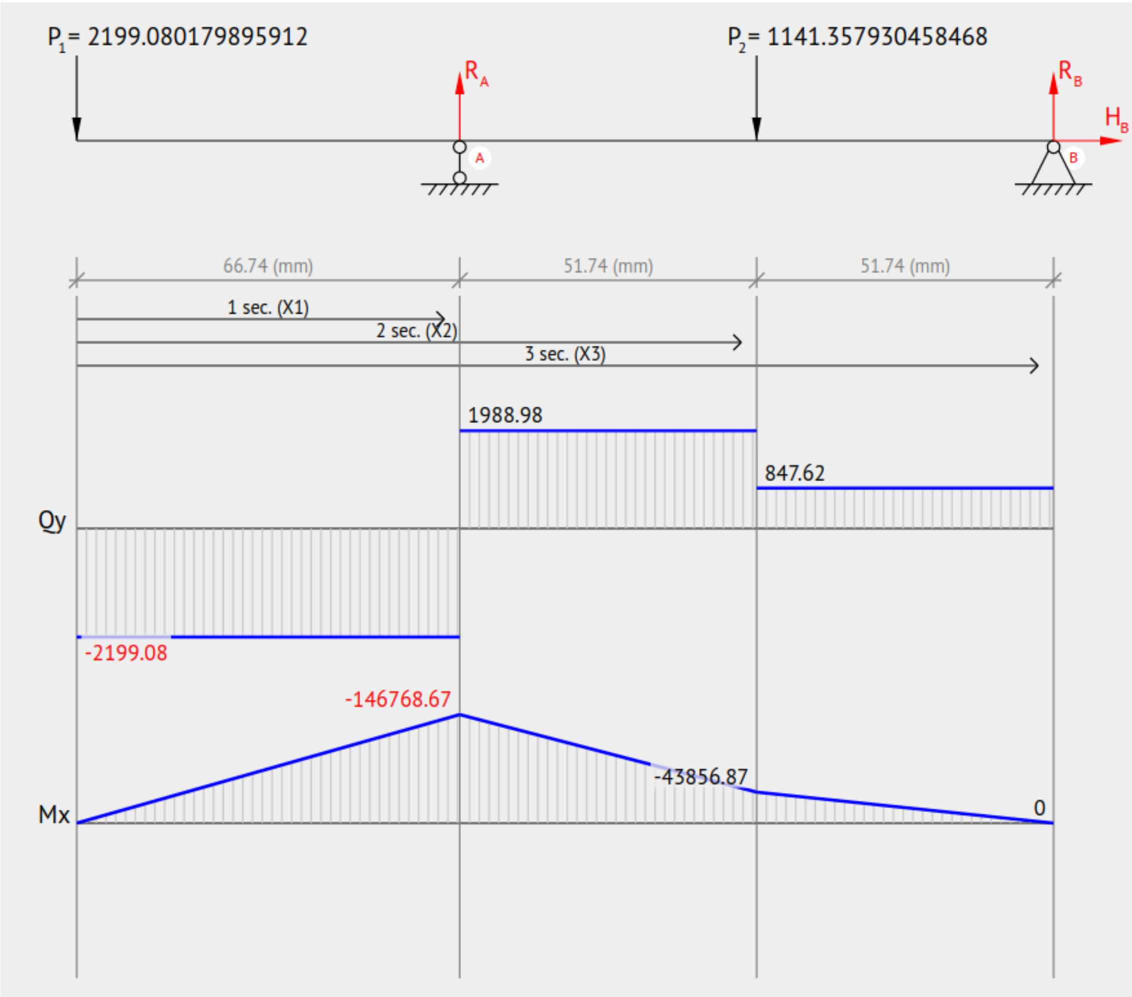

Figure 4.6: Shear force - Bending moment diagram on (yOz) of shaft 2