HCM UNIVERSITY OF TECHNOLOGY

MACHINE ELEMENTS

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# Project Report


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# Contents

# List of Tables

# List of Figures

# Design Problem

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

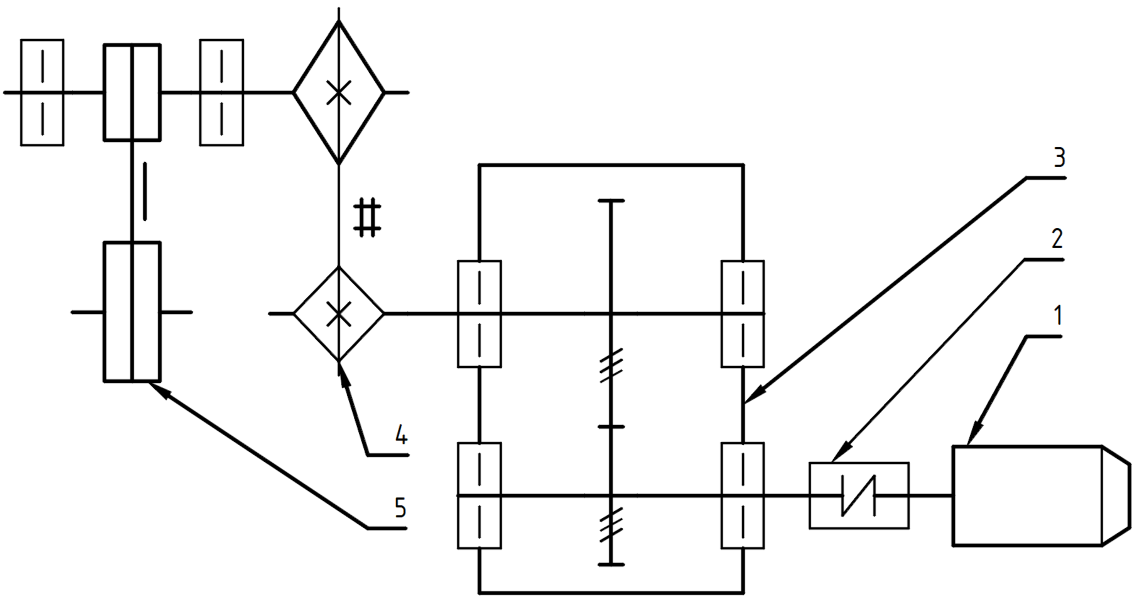


Figure 1: Mechanical transmission system of a belt conveyor

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

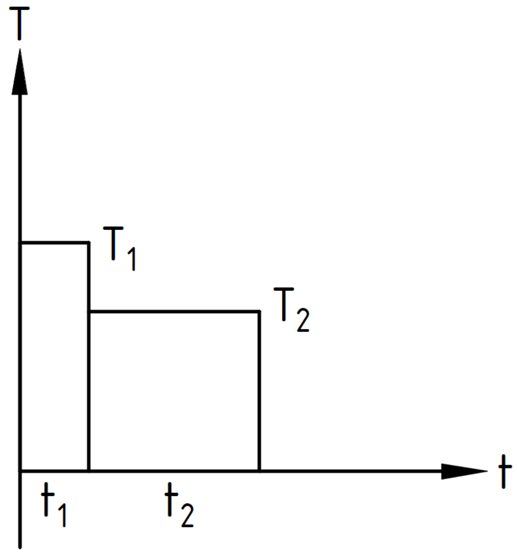


Figure 2: Input load diagram

**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, $\mathrm{N \cdot mm}$ |
| $\eta_b$ | bearing efficiency |
| $\eta_c$ | coupling efficiency |
| $\eta_{ch}$ | chain drive efficiency |
| $\eta_{hg}$ | helical gear efficiency |
| $\eta_{sys}$ | efficiency of the system |
| $_1$ | shaft 1 |
| $_2$ | shaft 2 |

## 1.2 Calculate $\eta_{sys}$

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) :

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

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

## 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

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

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

### 1.6.3 Torque

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

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

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

In summary, we obtain the following table:

| | Motor | Shaft 1 | Shaft 2 |
|---|---|---|---|
| $P$ (kW) | 18.5 | 15.35 | 14.59 |
| $u$ | 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

# Gearbox Design (Helix gears)

## 2.1 Nomenclature

| | |
|---|---|
| $[\sigma_H]$ | permissible contact stress, MPa |
| $[\sigma_H]_{max}$ | permissible contact stress due to overload, MPa |
| $[\sigma_F]$ | permissible bending stress, 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 of load 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 $[\sigma_H]$ |
| $S$ | specific length, mm |
| $S_F$ | safety factor of bending stress |
| $S_H$ | safety factor of contact stress |

| | |
|---|---|
| $T$ | input torque, $\text{N} \cdot \text{mm}$ |
| $v$ | rotational velocity, m/s |
| $x$ | gear correction factor |
| $Y_F$ | tooth shape factor |
| $Y_\beta$ | helix angle factor |
| $Y_\varepsilon$ | 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 $\beta$ |
| $z_v$ | virtual number of teeth |
| $z_\varepsilon$ | meshing condition factor |
| $\alpha$ | normal pressure angle, following Vietnam standard (TCVN 1065-71), i.e. $\alpha = 20°$ |
| $\alpha_t$ | traverse pressure angle, ° |
| $\varepsilon_\alpha$ | traverse contact ratio |
| $\varepsilon_\beta$ | face contact ratio |
| $\beta$ | helix angle, ° |
| $\beta_b$ | base circle helix angle, ° |
| $\psi_{ba}$ | width to shaft distance ratio |
| $\psi_{bd}$ | face width factor |
| $\sigma_b$ | ultimate strength, MPa |
| $\sigma_{ch}$ | yield limit, MPa |

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

## 2.2 Choose material

From table (6.1) , the material of choice for both gears is steel 40X with $S \leq 100$ (mm), HB250, $\sigma_b = 850$ (MPa), $\sigma_{ch} = 550$ (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 $H$.
The reasoning is given on p.91, where $H_2 = H_1 - 10 \div 15$
For the pinion, $H_1 = \text{HB250} \Rightarrow \sigma^o_{Hlim1} = 570$ (MPa), $\sigma^o_{Flim1} = 450$ (MPa)
For the driven gear, $H_2 = \text{HB240} \Rightarrow \sigma^o_{Hlim2} = 550$ (MPa), $\sigma^o_{Flim2} = 432$ (MPa)

## 2.3 Calculate $[\sigma_H]$ and $[\sigma_F]$

### 2.3.1 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)

### 2.3.2 Working cycle of equivalent tensile stress

Since $H_1, H_2 \leq \text{HB350}$, $m_H = 6$, $m_F = 6$.
Both gears meshed indefinitely, thus $c = 1$.

From working condition, we calculate:
$L_h = 8\left(\frac{\text{hours}}{\text{shift}}\right) \times 2\left(\frac{\text{shifts}}{\text{day}}\right) \times 300\left(\frac{\text{days}}{\text{year}}\right) \times 4\,(\text{years}) = 19200\,(\text{hours})$
Applying equation (6.7) and $T_1, T_2, t_1, t_2$ in the initial parameters:

$$N_{HE1} = 60n_{sh1}cL_h\left[\left(\frac{T_1}{T}\right)^3\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^3\frac{t_2}{t_1+t_2}\right] \approx 1.53\times 10^9\,(\text{cycles})$$

$$N_{HE2} = 60n_{sh2}cL_h\left[\left(\frac{T_1}{T}\right)^3\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^3\frac{t_2}{t_1+t_2}\right] \approx 0.31\times 10^9\,(\text{cycles})$$

$$N_{FE1} = 60n_{sh1}cL_h\left[\left(\frac{T_1}{T}\right)^{m_F}\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^{m_F}\frac{t_2}{t_1+t_2}\right] \approx 0.89\times 10^9\,(\text{cycles})$$

$$N_{FE2} = 60n_{sh2}cL_h\left[\left(\frac{T_1}{T}\right)^{m_F}\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^{m_F}\frac{t_2}{t_1+t_2}\right] \approx 0.18\times 10^9\,(\text{cycles})$$

### 2.3.3 Aging factor

For steel, $N_{FO1} = N_{FO2} = 4\times 10^6$ (MPa). Applying equations (6.3) and (6.4) yield (if $K_{HL}, K_{FL} < 1$, $K_{HL} = 1$ and $K_{FL} = 1$ according to the properties given on p.94):

$K_{HL1} = \sqrt[m_H]{N_{HO1}/N_{HE1}} \approx 0.47 < 1 \Rightarrow K_{HL1} = 1$

$K_{HL2} = \sqrt[m_H]{N_{HO2}/N_{HE2}} \approx 0.61 < 1 \Rightarrow K_{HL2} = 1$

$K_{FL1} = \sqrt[m_F]{N_{FO1}/N_{FE1}} \approx 0.41 < 1 \Rightarrow K_{FL1} = 1$

$K_{FL2} = \sqrt[m_F]{N_{FO2}/N_{FE2}} \approx 0.53 < 1 \Rightarrow K_{FL2} = 1$

### 2.3.4 Calculate $[\sigma_H]$, $[\sigma_{F1}]$, $[\sigma_{F2}]$

Since the motor works in one direction, $K_{FC} = 1$. In ideal conditions, we assume $Z_R Z_V K_{xH} = 1$ and $Y_R Y_s K_{xF} = 1$ according to p.92:

$[\sigma_{H1}] = \sigma^o_{Hlim1}K_{HL1}/S_{H1} \approx 518.18$ (MPa)

$[\sigma_{H2}] = \sigma^o_{Hlim2}K_{HL2}/S_{H2} \approx 500$ (MPa)

$[\sigma_{F1}] = \sigma^o_{Flim1}K_{FC1}K_{FL1}/S_{F1} \approx 257.14$ (MPa)

$[\sigma_{F2}] = \sigma^o_{Flim2}K_{FC2}K_{FL2}/S_{F2} \approx 246.86$ (MPa)

The mean permissible contact stress must be lower than 1.25 times of either

$[\sigma_{H1}]$ or $[\sigma_{H2}]$, whichever is smaller:
$[\sigma_H] = \frac{1}{2}([\sigma_{H1}] + [\sigma_{H2}]) \approx 509.09\,(\text{MPa}) \le 1.25[\sigma_H]_{min} = 1.25[\sigma_{H2}]$
The permissible contact stress and bending stress due to overload are calculated as follows:
$[\sigma_H]_{max} = 2.8\sigma_{ch} = 1540\,(\text{MPa})$
$[\sigma_F]_{max} = 0.8\sigma_{ch} = 440\,(\text{MPa})$

## 2.4 Transmission Design

### 2.4.1 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:
$a = K_a(u_{hg} + 1)\sqrt[3]{\frac{T_{sh1}K_{H\beta}}{[\sigma_H]^2 u_{hg}\psi_{ba}}} \approx 113.7\,(\text{mm})$
According to SEV229-75 standard, we choose $a_w = 125$ mm

### 2.4.2 Determine gear meshing parameters

**Find** $m$ Applying equation (6.17) and choose $m$ from table (6.8) :
$m = (0.01 \div 0.02)a_w = 1.25 \div 2.5\,(\text{mm}) \Rightarrow m = 1.5\,(\text{mm})$

**Find** $z_1$**,** $z_2$**,** $b_w$ Let $\beta = 14°$. 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).

$z_1 = \dfrac{2a_w \cos\beta}{m(u_{hg} + 1)} \approx 26.95 \Rightarrow z_1 = 27$
$z_2 = u_{hg} z_1 = 135$
$\Rightarrow b = \psi_{ba} a_w = 62.5$ (mm)

**Correct $\beta$** There are 2 approaches for correction involving the change of either $\alpha$ or $\beta$. Because altering $\alpha$ leads to many other corrections ($d_1$, $d_2$ and $a_w$), $\beta$ will be used instead.
Since $z_1$ is rounded, we must find $\beta$ to obtain the correct angle, ensuring that $\beta \in (8°, 20°)$. Using equation (6.32):
$\beta_w = \arccos \dfrac{m(z_1 + z_2)}{2a_w} \approx 13.59°$

**Find $x_1$, $x_2$** To find $x_1$ and $x_2$, we will follow the calculation scheme provided in p.103. Since $\beta_w \approx 13.59° \in (12, 17]$, $z_{min} = 16$, 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 $y$.

### 2.4.3 Basic parameters

$d_1 = d_{w1} = \dfrac{mz_1}{\cos\beta} \approx 41.67$ (mm)
$d_2 = d_{w2} = \dfrac{mz_2}{\cos\beta} \approx 208.33$ (mm)
$d_{a1} = d_1 + 2(1 + x_1)m \approx 45.57$ (mm)
$d_{a2} = d_2 + 2(1 + x_2)m \approx 210.43$ (mm)
$d_{f1} = d_1 - (2.5 - 2x_1)m \approx 38.82$ (mm)
$d_{f2} = d_2 - (2.5 - 2x_2)m \approx 203.68$ (mm)

$d_{b1} = d_1 \cos\alpha \approx 39.15$ (mm)
$d_{b2} = d_2 \cos\alpha \approx 195.77$ (mm)
$\alpha_t = \alpha_{tw} = \arctan \dfrac{\tan\alpha}{\cos\beta_w} \approx 20.53°$
$v = \dfrac{\pi d_1 n_{sh1}}{6 \times 10^4} \approx 6.39$ (m/s)

### 2.4.4 Find $[\sigma_{Hw}]$, $[\sigma_{Fw1}]$ and $[\sigma_{Fw2}]$

In this section, we will try to approximate these parameters based on the factors $Z_R$, $Z_V$, $K_{xH}$ and $Y_R$, $Y_s$, $K_{xF}$ to substitute to equation (6.1) and (6.2):

$$
\begin{aligned}
[\sigma_{Hw}] &= [\sigma_H] Z_R Z_V K_{xH} \\
[\sigma_{Fw}] &= [\sigma_F] Y_R Y_S K_{xF}
\end{aligned}
$$

Assuming smooth surface condition, $Z_R = 1$.

$Z_V = 0.85v^{0.1} \approx 1.02$ with $H \leq 350$.

In case of $v > 5$ (m/s), $K_{xH} = 1$.

The pair of gears are properly polished, which makes $Y_R = 1.1$

$Y_s = 1.08 - 0.0695 \ln(m) \approx 1.05$

Since $d_{a1}, d_{a2} \leq 400$ (mm), $K_{xF} = 1$, which leads to:

$[\sigma_{Hw}] = 520.93$ (MPa)

$[\sigma_{Fw1}] = 297.51$ (MPa)

$[\sigma_{Fw2}] = 285.61$ (MPa)

### 2.4.5 Contact stress analysis

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

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

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

**Find $z_H$** $\beta_b = \arctan(\cos\alpha_t \tan\beta_w) \approx 12.76° \Rightarrow z_H = \sqrt{2\frac{\cos\beta_b}{\sin(2\alpha_{tw})}} \approx 1.72$

**Find** $z_\varepsilon$ Obtaining $z_\varepsilon$ through calculations:

$$\varepsilon_\alpha = \frac{\sqrt{d_{a1}^2 - d_{b1}^2} + \sqrt{d_{a2}^2 - d_{b2}^2} - 2a_w \sin\alpha_{tw}}{2\pi m \dfrac{\cos\alpha_t}{\cos\beta_w}} \approx 1.41$$

$$\varepsilon_\beta = b\frac{\sin\beta_w}{m\pi} \approx 3.12 > 1 \Rightarrow z_\varepsilon = \varepsilon_\alpha^{-0.5} \approx 0.86$$

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

From table (6.13), $v \leq 10\,(\mathrm{m/s}) \Rightarrow \mathrm{AG} = 8$

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

$K_{Hv} \approx 1.07$, $K_{Fv} \approx 1.18$

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

$K_{H\alpha} \approx 1.1$, $K_{F\alpha} \approx 1.29$

$\Rightarrow K_H \approx 1.3$

**Find** $\sigma_H$ After calculating $z_M$, $z_H$, $z_\varepsilon$, $K_H$, we get the following result:

$$\sigma_H \approx 477.51\,(\mathrm{MPa}) \leq [\sigma_{Hw}] \approx 509.09\,(\mathrm{MPa})$$

### 2.4.6 Bending stress analysis

For safety reasons, the following conditions must be met:

$$\begin{aligned} \sigma_{F1} &= 2\frac{T_{sh1}K_FY_\varepsilon Y_\beta Y_{F1}}{bd_{w1}m_n} && \leq [\sigma_{Fw1}] \\ \sigma_{F2} &= \frac{\sigma_{F1}Y_{F2}}{Y_{F1}} && \leq [\sigma_{Fw2}] \end{aligned}$$

**Find** $Y_\varepsilon$ Knowing that $\varepsilon_\alpha \approx 1.41$, we can calculate $Y_\varepsilon = \varepsilon_\alpha^{-1} \approx 0.71$

**Find** $Y_\beta$ $Y_\beta = 1 - \dfrac{\beta_w}{140} \approx 0.9$

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

$z_{v1} = z_1\cos^{-3}(\beta_w) \approx 29.4 \Rightarrow Y_{F1} \approx 3.54$

$z_{v2} = z_2\cos^{-3}(\beta_w) \approx 147.01 \Rightarrow Y_{F2} \approx 3.63$

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

$K_F = K_{F\beta}K_{F\alpha}K_{Fv} \approx 1.91$

**Find $\sigma_F$** Since $m_n = m\cos\beta_w \approx 1.46$, substituting all the values, we find out that:

$$\sigma_{F1} \approx 130.83\ (\text{MPa}) \le [\sigma_{Fw1}] \approx 297.51\ (\text{MPa})$$

$$\sigma_{F2} \approx 115.98\ (\text{MPa}) \le [\sigma_{Fw2}] \approx 285.61\ (\text{MPa})$$

### 2.4.7 Force on shafts

$F_t = \dfrac{2T_{sh1}}{d_{w1}} \approx 2402.28\ (\text{N})$

$F_r = F_t\tan\alpha_{tw} \approx 899.55\ (\text{N})$

$F_a = F_t\tan\beta_w \approx 580.75\ (\text{N})$

In summary, we have the following table:

| | pinion | driving gear |
|---|---|---|
| $H$ (HB) | 250 | 240 |
| $[\sigma_F]$ (MPa) | 257.14 | 246.86 |
| $[\sigma_H]$ (MPa) | 509.09 | |
| $[\sigma_H]_{max}$ (MPa) | 1540 | |
| $[\sigma_F]_{max}$ (MPa) | 440 | |
| $a_w$ (mm) | 100 | |
| $b$ (mm) | 50 | |
| $m$ (mm) | 1.5 | |
| $d_w$ (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 |
| $u_{hg}$ | 5 | |
| $v$ (m/s) | 5 | |
| $x$ (mm) | 0.3 | -0.3 |
| $z$ | 21 | 105 |
| $\alpha_{tw}$ (°) | 20.65 | |
| $\beta_w$ (°) | 19.09 | |

Table 2.1: Gearbox specifications

# Chapter 3

# Bearing Design

## 3.1 Nomenclature

| | |
|---|---|
| $[s]$ | permissible safety factor |
| $[\sigma]$ | permissible static strength, MPa |
| $[\tau]$ | permissible torsion, MPa |
| $a_w$ | shaft distance, mm |
| $b_O$ | rolling bearing width, mm |
| $C_d$ | basic dynamic load rating, N |
| $cb$ | role of gear on the shaft (active or passive) |
| $cq$ | rotational direction of the shaft |
| $d$ | base shaft diameter, mm |
| $d_w$ | gear diameter, mm |
| $F_a$ | axial force, N |
| $F_r$ | radial force, N |
| $F_t$ | tangential force, N |
| $F_x$ | applied force, N |
| $h_n$ | distance between bearing lid and bolt, mm |
| $hr$ | tooth direction |
| $K_x$ | surface tension concentration factor |
| $K_y$ | diminish factor |
| $K_\sigma$ | combined influence factor in tension |
| $K_\tau$ | combined influence factor in shear |

| | |
|---|---|
| $k_d$ | temperature factor |
| $k_t$ | load condition factor |
| $k_\sigma$ | fatigue stress concentration factor in tension |
| $k_\tau$ | fatigue stress concentration factor in shear |
| $L$ | rated life in million revolutions, million rev |
| $L_h$ | rated life in hours, h |
| $l$ | length (general), mm |
| $l_m$ | hub length (general), mm |
| $M$ | moment, $\text{N} \cdot \text{mm}$ |
| $M_e$ | equivalent moment, $\text{N} \cdot \text{mm}$ |
| $M_{max}$ | maximum moment at the cross section, $\text{N} \cdot \text{mm}$ |
| $m$ | load-life exponent |
| $Q$ | equivalent dynamic load, kN |
| $q$ | standardized coefficient of shaft diameter |
| $R$ | reaction force, N |
| $r$ | shoulder fillet radius, mm |
| $\bar{r}$ | position of applied force on the shaft, mm |
| $S$ | length defined by table (6.1), mm |
| $s$ | calculated safety factor |
| $s_\sigma$ | safety factor in tensile stress |
| $s_\tau$ | safety factor in shear stress |
| $T$ | torque at the cross section, $\text{N} \cdot \text{mm}$ |
| $W$ | section modulus, $\text{mm}^3$ |
| $W_O$ | polar section modulus, $\text{mm}^3$ |
| $X$ | dynamic radial load factor |
| $Y$ | dynamic axial load factor |
| $\alpha$ | contact angle, $^\circ$ |
| $\alpha_{tw}$ | traverse meshing angle, $^\circ$ |
| $\beta$ | helix angle, $^\circ$ |
| $\psi_\sigma$ | mean stress influence factor |
| $\psi_\tau$ | mean shear influence factor |
| $\sigma_{-1}$ | endurance limit at stress ratio of -1, MPa |
| $\sigma_a$ | tensile stress amplitude, MPa |
| $\sigma_b$ | ultimate strength, MPa |
| $\sigma_{ch}$ | yield limit, MPa |
| $\sigma_m$ | mean tensile stress, MPa |
| $\sigma_{td}$ | static strength, MPa |
| $\tau_{-1}$ | endurance limit at shear ratio of -1, MPa |
| $\tau_a$ | shear stress amplitude, MPa |
| $\tau_m$ | mean shear stress, MPa |
| $_{sh1}$ | subscript for shaft 1 |
| $_{sh2}$ | subscript for shaft 2 |
| $_x$ | subscript for x-axis |
| $_y$ | subscript for y-axis |
| $_z$ | subscript for z-axis |

## 3.2 Choose bearing type

As for the types, we will examine $\dfrac{F_a}{F_r}$ at $A_1$, $B_1$, $A_2$ and $B_2$ in the 2 shafts from the previous chapter, where $F_a$ are $|F_{z12}|$ in case of shaft 1 and $|F_{z21}|$ in case of shaft 2, which are axial loads; $F_r$ is the magnitude of reaction force $R_y$ from the shaft onto a bearing along y-axis, which is essentially a radial load. The larger ratio in a shaft will be our ratio of choice to determine the bearing type.

Taking our results from the shaft design chapter:

$$\begin{cases} R_{A1y} \approx -341.74 \text{ (N)} \\ R_{B1y} \approx -583.72 \text{ (N)} \\ F_{z12} \approx 580.75 \text{ (N)} \end{cases} \quad \text{and} \quad \begin{cases} R_{A2y} \approx 3708.15 \text{ (N)} \\ R_{B2y} \approx -462.64 \text{ (N)} \\ F_{z21} \approx -580.75 \text{ (N)} \end{cases}$$

yields

$$\begin{cases} \left|\dfrac{F_{z12}}{R_{A1y}}\right| \approx 1.7 \\ \left|\dfrac{F_{z12}}{R_{B1y}}\right| \approx 0.99 \end{cases} \qquad \begin{cases} \left|\dfrac{F_{z21}}{R_{A2y}}\right| \approx 0.16 \\ \left|\dfrac{F_{z21}}{R_{B2y}}\right| \approx 1.26 \end{cases}$$

Since $1.7 > 1$ and $1.26 > 1$, the 2 pairs of bearings are double row angular contact ball bearings with $\alpha_{sh1} = \alpha_{sh2} = 36°$; AG = 0 according to the recommendations on p.212 and p.213.

## 3.3 Bearing dimensions

### 3.3.1 Calculate basic dynamic load rating

$$C_d = Q \sqrt[m]{L}$$

**Find equivalent dynamic load**

Since we only use angular contact ball bearings, the following formula applies:

$$Q = (XVF_r + YF_a)k_t k_d$$

Since the inner ring rotates, $V = 1$. The design problem also does not give any further information about operating temperature, which gives $k_t = 1$. In addition, we get $k_d = 1$ from table (11.3) based on the machine's condition (low load and power rating).

From the previous section, $\alpha_{sh1} = 36°, \alpha_{sh2} = 36°$. Inspecting table (11.4), $e_{sh1} = e_{sh2} = 0.95$. These values are then compared with $\left|\frac{F_a}{VF_r}\right|$ to look up the correct column.

For shaft 1, $\left|\frac{F_a}{VF_r}\right| = \left|\frac{F_{z12}}{VR_{A1y}}\right| \approx 1.7 > e_1 \Rightarrow X_1 = 0.6, Y_1 = 1.07$.

For shaft 2, $\left|\frac{F_a}{VF_r}\right| = \left|\frac{F_{z21}}{VR_{B2y}}\right| \approx 1.26 > e_2 \Rightarrow X_2 = 0.6, Y_2 = 1.07$.

Recall that the forces are in absolute values, the parameters are substituted to the formula to obtain:

$Q_1 \approx 826.45$ (kN)

$Q_2 \approx 1389.39$ (kN)

**Find rated life**

Equation (11.2) is rearranged to calculate $L$:

$$L = L_h 60 n_{sh} \times 10^{-6}$$

In our transmission system, since the gearbox is a speed reducer working 2 shifts daily, we approximate $L_h \approx 30000$ (hours) according to table (11.2), which gives:

$L_1 \approx 5274$ (million rev)

$L_2 \approx 1054.8$ (million rev)

Combining the results and letting $m = 3$ (ball bearings are used in this case) yield:

$C_{d1} = 14385.63$ (N)