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ME3011

# Design Project Report

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# Abstract

In machine design, every machine element must be calculated in a systematic matter. In this course, students are provided with essential skills to formulate almost every dimension manually, thus further improving their engineering skills before engaging the high-energy, fast-paced workforce.

When a machine element is being developed, it must satisfy some key engineering specifications such as being able to operate under designated lifespan, low cost and high efficiency. Other aspects are less important but also determined the overall design of the element include compactness, noise emission, appearance, etc.

To optimize the process of machine design, the general principles are considered as follows:

1. Identify the working principle and workload of the machine.

2. Formulate the overall working principle to satisfy the problem. Proposing feasible solutions and evaluating them to find the optimal design specifications.

3. Find force and moment diagram exerting on machine parts and characteristics of the workload.

4. Choose appropriate materials to make use of their properties and improve efficiency as well as reliability of individual elements.

5. Calculate dynamics, strength, safety factor, etc. to specify dimensions.

6. Design machine structure, parts to satisfy working condition and assembly.

7. Create presentation, instruction manual and maintenance.

In this report, I will design a fairly simple system to provide a concrete example of finishing all the tasks above.

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# Chapter 1

# Design Problem

## 1.1 Nomenclature

| | | | |
|---|---|---|---|
| $C_a$ | number of shift daily, shifts | $P$ | design power of the mixing tank, kW |
| $K_{ng}$ | working days/year, days | $T_1$ | working torque 1, $N \cdot m$ |
| $L$ | service life, years | $T_2$ | working torque 2, $N \cdot m$ |
| $n$ | rotational velocity of the mixing tank, rpm | $t_1$ | working time 1, s |
| | | $t_2$ | working time 2, s |

## 1.2 Problem

The problem is downloaded from E-learning website, designated number 8, see Figure 1.1.

## 1.3 Mixing machine parameters

From the parameters given in the document, we have:

| | |
|---|---|
| $P = 7$ (kW) | $t_1 = 15$ (s) |
| $n = 65$ (rpm) | $t_2 = 11$ (s) |
| $L = 8$ (years) | $T_1 = T$ $(N \cdot m)$ |
| $K_{ng} = 260$ (days) | $T_2 = 0.7T$ $(N \cdot m)$ |
| $Ca = 1$ (shifts) | |

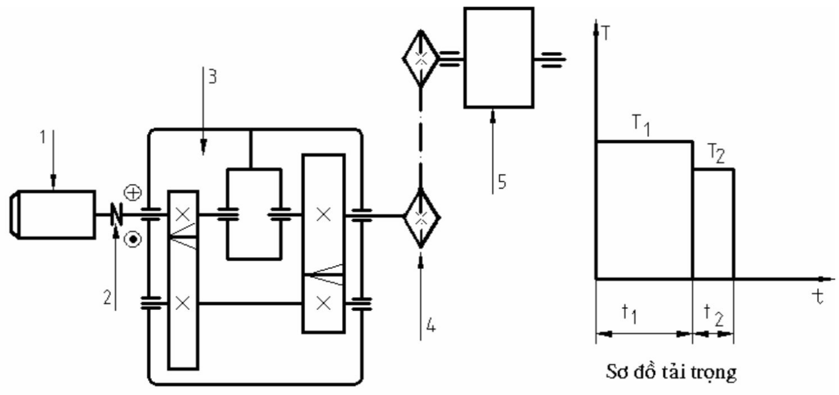


Figure 1.1: Working principle diagram and workload of the mixing machine: 1) electric motor, 2) coupling, 3) two-stage coaxial helical speed reducer, 4) roller chain drive, 5) mixing tank (one-directional, light duty, operate 1 shift, 8 hours each)

## 1.4 Requirements

- 01 report.
- 01 assembly drawing.
- 01 detailed drawing.

## 1.5 Design problem

1. Decide the working power of the electric motor and transmission ratio of the system.
2. Calculate and design machine elements:
   (a) Calculate system drives (belt, chain or gear).
   (b) Calculate the elements in speed reducers (gears, lead screws).
   (c) Draw and calculate force diagram exerting on the transmission elements.
   (d) Calculate, design shafts and keys.
   (e) Choose bearings and couplings.
   (f) Choose machine bodies, fasteners and other elements.

3. Choose assembly tolerance.

4. Bibliography

# Chapter 2

# Choose Motor

## 2.1 Nomenclature

| | |
|---|---|
| $n_{sh}$ | rotational speed of shaft, rpm |
| $P_{mo}$ | 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 |
| $u_1$ | transmission ratio of quick stage |
| $u_2$ | transmission ratio of slow stage |
| $u_{ch}$ | transmission ratio of chain drive |
| $u_h$ | transmission ratio of speed reducer |
| $u_{sys}$ | transmission ratio of the system |
| $T_{mo}$ | 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 |
| $_3$ | shaft 3 |

**Known parameters** From Chapter 1, we know that:
$P = 7\,\mathrm{kW}$, $n = 65\,\mathrm{rpm}$
$T_1 = T$, $T_2 = 0.7T$
$t_1 = 15\,\mathrm{s}$, $t_2 = 11\,\mathrm{s}$

## 2.2 Calculate $\eta_{sys}$

From Table 2.3:
$\eta_c = 1$
$\eta_b = 0.99$
$\eta_{hg} = 0.97$
$\eta_{ch} = 0.96$
$\eta_{sys} = \eta_c \eta_b^4 \eta_{hg}^2 \eta_{ch} = 0.87$

## 2.3 Calculate $P$

From Equation 2.13 [1]:

$$P_w = P\sqrt{\frac{\left(\frac{T_1}{T}\right)^2 t_1 + \left(\frac{T_2}{T}\right)^2 t_2}{t_1 + t_2}} = 6.2\ (\text{kW})$$

$$P_{mo} = \frac{P_w}{\eta_{sys}} = 7.14\ (\text{kW}) > P$$

## 2.4 Calculate $n_{mo}$

Using Table 2.4 [1]:
$u_{ch} = 4$, $u_h = 10$
$u_{sys} = u_{ch} u_h = 40$
$n_{mo} = u_{sys} n = 2600\ (\text{rpm})$

## 2.5 Choose motor

To work normally, the maximum operating power of the chosen motor must be no smaller than both estimated $P_{mo}$ and $P$. Since $P_{mo} > P$ for our case, the minimum operating power of choice is $P_{mo} = 7.14\ (\text{kW})$. In similar fashion, its rotational speed must also be no smaller than estimated $n_{mo}$.

Thus, from Table P1.3 [1], we choose motor 4A112M2Y3 which operates at 7.5 kW and 2922 rpm.
$\Rightarrow P_{mo} = 7.5\ (\text{kW}), n_{mo} = 2922\ (\text{rpm})$

Recalculating $u_{sys}$ with the new $P_{mo}$ and $n_{mo}$, we obtain:
$u_{sys} = \frac{n_{mo}}{n} = 44.95$
Assuming $u_h = const$:
$u_{ch} = \frac{u_{sys}}{u_h} = 4.5$

## 2.6 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 and $P_{sh3}$, $n_{sh3}$ and $T_{sh3}$ will be the transmitted parameters onto shaft 3. These notations will be used throughout the next chapters.

### 2.6.1 Power

$P_{ch} = \frac{P_w}{\eta_b} = 6.3$ (kW)
$P_{sh3} = \frac{P_{ch}}{\eta_{ch}} = 6.52$ (kW)
$P_{sh2} = \frac{P_{sh3}}{\eta_b \eta_{hg}} = 6.79$ (kW)
$P_{sh1} = \frac{P_{sh2}}{\eta_b \eta_{hg}} = 7.07$ (kW)

### 2.6.2 Rotational speed

To minimize the speed reducer weight, we can use the following formula, see Equation 3.12 [1]:

$$u_1 = 0.7332 u_h^{0.6438}$$

Then,
$u_1 = 3.23$ and $u_2 = u_h/u_1 = 3.1$
$n_{sh1} = n_{mo} = 2922$ (rpm)
$n_{sh2} = n_{sh1}/u_1 = 905.02$ (rpm)
$n_{sh3} = n_{sh2}/u_2 = 292.20$ (rpm)

### 2.6.3 Torque

$T_{mo} = 9.55 \times 10^6 \times P_{mo}/n_{mo} = 24512.32\ (\mathrm{N \cdot mm})$
$T_{sh1} = 9.55 \times 10^6 \times P_{sh1}/n_{sh1} = 23106.95\ (\mathrm{N \cdot mm})$
$T_{sh2} = 9.55 \times 10^6 \times P_{sh2}/n_{sh2} = 71672.45\ (\mathrm{N \cdot mm})$
$T_{sh3} = 9.55 \times 10^6 \times P_{sh3}/n_{sh3} = 213175.22\ (\mathrm{N \cdot mm})$

In summary, we obtain the following table:

| | Motor | Shaft 1 | Shaft 2 | Shaft 3 |
|---|---|---|---|---|
| $P$ (kW) | 7.5 | 7.07 | 6.79 | 6.52 |
| $u$ | - | 1 | 3.23 | 3.1 |
| $n$ (rpm) | 2922 | 2922 | 905.02 | 292.20 |
| $T$ ($\mathrm{N \cdot mm}$) | 24512.32 | 23116.54 | 71672.45 | 213175.22 |

Table 2.1: Output specifications

# Chapter 3

# Chain Drive Design

## 3.1 Nomenclature

| | |
|---|---|
| $[i]$ | permissible impact times per second |
| $[s]$ | permissible safety factor |
| $[P]$ | permissible power, kW |
| $[\sigma_H]$ | permissible contact stress, MPa |
| $A$ | cross sectional area of chain hinge, $mm^2$ |
| $a$ | real center distance, mm |
| $a_i$ | estimated center distance, mm |
| $a_{max}$ | maximum center distance, mm |
| $a_{min}$ | minimum center distance, mm |
| $B$ | width between inner link plate, mm |
| $d$ | chordal diameter, mm |
| $d_a$ | addendum diameter, mm |
| $d_f$ | dedendum diameter, mm |
| $d_l$ | roller diameter, mm |
| $d_O$ | pin diameter, mm |
| $E$ | modulus of elasticity, MPa |
| $F_0$ | 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 |
| $F_{vd}$ | contact force, N |
| $i$ | impact times per second |
| $K_d$ | weight distribution factor |
| $k$ | overall factor |
| $k_a$ | center distance and chain's length factor |
| $k_{bt}$ | lubrication factor |
| $k_c$ | rating factor |
| $k_d$ | dynamic load factor |
| $k_{dc}$ | chain tension factor |
| $k_f$ | loosing factor |
| $k_n$ | coefficient of rotational speed |
| $k_r$ | number of tooth factor |
| $k_x$ | chain weight factor |
| $k_z$ | coefficient of number of teeth |

| | | | |
|---|---|---|---|
| $k_0$ | arrangement of drive factor | $v$ | instantaneous velocity along the chain, m/s |
| $n$ | angular rotational speed, rpm | $x$ | chain length in pitches, the number of links |
| $n_{01}$ | experimental angular rotational speed, rpm | $x_c$ | an even number of links |
| $P_t$ | calculated power, kW | $z$ | number of teeth of a sprocket |
| $p$ | pitch, mm | $z_{max}$ | maximum number of teeth of the driven sprocket |
| $p_{max}$ | permissible sprocket pitch, mm | $\sigma_H$ | contact stress, MPa |
| $Q$ | permissible load, N | $_1$ | subscript for driving sprocket |
| $q$ | mass per unit length, kg/m | $_2$ | subscript for driven sprocket |
| $s$ | safety factor | | |

**Known parameters** From Chapter 1, we know that:
The chain type is roller.
$n_{sh3} = 292.2$ rpm.
$P_{ch} = 6.3$ kW, $u_{ch} = 4.5$.

## 3.2 Find $p$

The driving sprocket is connected to shaft 3, $n_1 = n_{sh3} = 292.2$ (rpm).

### 3.2.1 Calculate $z$

Since $z_1$ and $z_2$ is preferably an odd number, Equation 5.1 [1]:
$z_1 = 29 - 2u_{ch} = 21$
$z_2 = u_{ch} z_1 = 95 \leq 120$

### 3.2.2 Calculate $[P]$

Since 200 is the nearest to $n_{ch} = 292.2$ rpm, we choose $n_{01} = 200$ rpm, which is one of the experimental angular rotational speed values. Then, we can obtain the value $[P]$, Equation 5.3 [1]:

$$P_t = P_{ch} k_0 k_a k_{dc} k_{bt} k_d k_c k_z k_n \leq [P]$$

where $k_z = 25/z_1 = 1.32$ and $k_n = n_{01}/n_1 = 1.02$.

Assuming good operational condition, $k_0 = k_a = k_{dc} = k_{bt} = 1$, $k_d = 1.25$, $k_c = 1$, see Table 5.6 [1]. Calculation yields $P_t = 6.38$ (kW) $\leq [P]$. Inspecting Table 5.5 [1] at column $n_{01} = 200$ rpm, choose the closet value $[P] = 11$ kW.

### 3.2.3 Determine p

Knowing $[P]$, we have $p = 25.4$ mm, Table 5.5. Consequently, $d_c = 7.95$ mm, $B = 22.61$ mm. Consulting Table 5.8, the pitch is indeed suitable.

## 3.3 Find $a$, $x_c$, and $i$

### 3.3.1 Find $x_c$

$a_{min} = 30p = 762$ (mm), $a_{max} = 50p = 1270$ (mm). Limiting the range of choice for $a$ in $[a_{min}, a_{max}]$, we can approximate $a_i = 1000$ mm and find $x_c$:

$$x = \frac{2a_i}{p} + \frac{z_1 + z_2}{2} + \frac{(z_2 - z_1)^2 p}{4\pi^2 a_i} = 140.26$$

Then, round $x$ up to the nearest even number gives $x_c = 142$.

### 3.3.2 Find $a$

Using $x_c$ to find the correct center distance, see Equation 5.13 [1]. In addition, it is recommended to loosing the chain an amount of $0.002 \div 0.004a$, which explains the coefficient 0.097 in the formula below:

$$a = \frac{0.097p}{4}\left[x_c - \frac{z_2 + z_1}{2} + \sqrt{\left(x_c - \frac{z_2 + z_1}{2}\right)^2 - 2\left(\frac{z_2 - z_1}{\pi}\right)^2}\right] = 1019.99 \text{ (mm)}$$

### 3.3.3 Find $i$

The permissible impact frequency is $[i] = 30$, Table 5.9 [1]. Calculating $i$ gives:

$$i = \frac{z_1 n_1}{15x} = 2.92 < [i]$$

## 3.4 Strength of chain drive

In order to operate safely, the chain drive's safety factor must satisfy the following condition:

$$s = \frac{Q}{k_d F_t + F_0 + F_v} \geq [s]$$

Rotational speed of the smaller sprocket is determined using the formula below:

$$v_1 = \frac{n_1 p z_1}{60000} = 2.6\,(\text{m/s})$$

**Find** $k_d$: Assuming moderate workload, choose $k_d = 1.2$.

**Find** $F_t$**,** $F_v$ **and** $F_0$: Knowing $p$, it is easy to look up the values $Q = 56700\,\text{N}$ and $q = 2.6\,\text{kg/m}$ from Table 5.2 [1]:

$F_t = 10^3 P_{ch}/v_1 = 2410.48\ (\text{N})$

$F_v = qv_1^2 = 17.54\ (\text{N})$

$F_0 = 9.81 \times 10^{-3} k_f q a = 156.11\ (\text{N})$

**Find** $k_f$: Let the chain drive be parallel to the ground, we get $k_f = 6$.

**Find** $[s]$: The limit $[s] = 8.71$ is found using interpolation, Table 5.10 [1].

Replacing all the variables gives:

$$s = 18.49 \geq 8.71$$

which satisfies the condition.

## 3.5 Determine sprocket specifications and output force on the shaft

The following condition must be met, Equation 5.18 [1]:

$$\sigma_H = 0.47\sqrt{\frac{k_r(F_t k_d + F_{vd})E}{AK_d}} \leq [\sigma_H]$$

Since the chain drive only has one strand, $K_d = 1$.

**Find** $[\sigma_H]$ Quenched 45 steel is the material of use for the chain drive, which has HB210, $[\sigma_H] = 600$ (MPa) and $E = 2.1 \times 10^5$ (MPa), see Table 5.11 [1].

**Find** $F_{vd}$ For 1-strand chain, $F_{vd} = 13 \times 10^{-7} n_1 p^3 = 6.22$ (N)

**Find** $k_r$ Since $z_1$ is used to estimate $k_r$, $k_r = 0.47$.

**Find E** Assuming the sprockets and chain are made up from the same material (steel), $E = 2.1 \times 10^5$ MPa

**Find** $A$ From the given parameters and value $p$, the area $A = 180\,\mathrm{mm}^2$, Table 5.12 [1].

Knowing $k_d$ and $F_t$, we get the result:

$$\sigma = 591.29\ (\mathrm{MPa}) \leq [\sigma_H]$$

which is satisfactory.

## 3.6 Force on shaft

Applying the following equations, see p.87 [1]:
$F_2 = F_0 + F_v = 173.65$ (N)
$F_1 = F_t + F_2 = 2584.13$ (N)
Choose $k_x = 1.15$ to obtain $F_r$, Equation 5.20 [1]:
$F_r = k_x F_t = 2772.05$ (N)

## 3.7 Other parameters

$d_1 = p/\sin\left(\dfrac{\pi}{z_1}\right) = 170.42$ (mm)
$d_2 = p/\sin\left(\dfrac{\pi}{z_2}\right) = 768.22$ (mm)
$d_{a1} = p\left(0.5 + \cot\dfrac{180}{z_1}\right) \approx 181.22$ (mm)
$d_{a2} = p\left(0.5 + \cot\dfrac{180}{z_2}\right) \approx 780.50$ (mm)
Look up to find $d_l = 15.88$ (mm), see Table 5.2 [1]:
$d_{f1} = d_1 - 2(0.502 d_l + 0.05) \approx 154.36$ (mm)
$d_{f2} = d_2 - 2(0.502 d_l + 0.05) \approx 752.16$ (mm)

In summary, we have the following table:

| | driving | driven |
|---|---|---|
| $[P]$ (kW) | 11 | |
| $a$ (mm) | 1019.99 | |
| $B$ (mm) | 22.61 | |
| $d$ (mm) | 170.92 | 768.22 |
| $d_a$ (mm) | 181.22 | 780.50 |
| $d_f$ (mm) | 154.36 | 752.16 |
| $d_l$ (mm) | 15.88 | |
| $d_O$ (mm) | 7.95 | |
| $i$ | 2.92 | |
| $p$ (mm) | 25.4 | |
| $Q$ (N) | 56700 | |
| $u_{ch}$ | 4.5 | |
| $v$ (m/s) | 2.6 | |
| $z$ | 21 | 95 |

Table 3.1: Chain drive specifications

# Chapter 4

# Gearbox Design (Helix gears)

## 4.1 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, $\mathrm{N \cdot mm}$
- $v$ rotational velocity, m/s
- $x$ gear correction factor
- $Y_F$ tooth shape factor
- $Y_R$ surface roughness factor of the gear's face
- $Y_s$ sensitivity to stress concentration factor
- $Y_\beta$ helix angle factor
- $Y_\varepsilon$ contact ratio factor
- $y$ center displacement factor
- $Z_R$ surface roughness factor of the working's area
- $Z_v$ speed 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
- $\sigma^o_{Hlim}$ permissible contact stress corresponding to working cycle, MPa
- $_1$ subscript for the pinion
- $_2$ subscript for the driven gear
- $_q$ subscript for the quick transmission stage
- $_s$ subscript for the slow transmission stage
- $_w$ subscript for the value after correction

**Known parameters** From Chapter 1 and 2, we know that:
$L = 8$ years, $K_{ng} = 260$ days, $Ca = 1$ shift
$T_1 = T$, $T_2 = 0.7T$, $t_1 = 15$ s, $t_2 = 11$ s
$n_{sh1} = 2922$ rpm, $n_{sh2} = 905.02$ rpm, $n_{sh3} = 292.20$ rpm
$u_h = 10$, $u_1 = u_q = 3.23$, $u_2 = u_s = 3.1$

This chapter will increase readability by calculating both stages at the same time with the first one being quick stage and the latter is slow stage.

## 4.2 Choose material

Because all gears are the same in material and working hours except for their angular rotational speed, this section applies for both pairs.

The material of choice for the 2 pair of gears is steel 40X. The specifications are $S \leq 100$ mm, HB250, $\sigma_b = 850$ MPa, $\sigma_{ch} = 550$ MPa, see Table 6.1 [1].

From Table 6.2 [1], $\sigma^o_{Hlim} = 2\text{HB}+70$, $S_H = 1.1$, $\sigma^o_{Flim} = 1.8\text{HB}$, $S_F = 1.75$.

Therefore, they have the same properties except for their surface roughness $H$, since $H_2 = H_1 - 10 \div 15$.

For the pinion, $H_1 = \text{HB}250 \Rightarrow \sigma^o_{Hlim1} = 570$ MPa, $\sigma^o_{Flim1} = 450$ MPa.

For the driven gear, $H_2 = \text{HB}240 \Rightarrow \sigma^o_{Hlim2} = 550$ MPa, $\sigma^o_{Flim2} = 432$ MPa.

In this part, distinguishing between 2 stages is unnecessary since the variables are material-dependent, which in this case all the gears have identical choice of material. Therefore, unless otherwise specified, a single subscript 1 or 2 indicates the variable applies for both stages

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

The permissible stresses are calculated as follows, see Equation 6.1 and 6.2 [1]:

$$[\sigma_H] = \frac{\sigma^o_{Hlim}}{S_H} Z_R Z_v K_{xH} K_{HL}$$

$$[\sigma_F] = \frac{\sigma^o_{Flim}}{S_F} Y_R Y_s K_{xF} K_{FL}$$

**Calculate working cycle of bearing stress** The stress is found using the formula below, see Equation 6.5 [1]:
$N_{HO1} = 30H_1^{2.4} = 17067789.40$ (cycles).
$N_{HO2} = 30H_2^{2.4} = 15474913.67$ (cycles).

**Calculate working cycle of equivalent tensile stress** Since $H_1, H_2 \leq$ HB350, $m_H = 6$, $m_F = 6$. Also, both pairs of gears are meshed indefinitely, which makes $c = 1$. From working condition, we calculate:
$L_h = 8\left(\frac{\text{hours}}{\text{shift}}\right) \times CaK_{ng}L = 16640$ (hours)
Find $N_{HE}$ and $N_{FE}$ using equation 6.7 and 6.8 [1]:
$$N_{HE1q} = 60n_{sh1}cL_h\left[\left(\frac{T_1}{T}\right)^3\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^3\frac{t_2}{t_1+t_2}\right] = 2.11\times 10^9 \text{ (cycles)}$$
$$N_{HE2q} = 60n_{sh2}cL_h\left[\left(\frac{T_1}{T}\right)^3\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^3\frac{t_2}{t_1+t_2}\right] = 0.65\times 10^9 \text{ (cycles)}$$
$$N_{FE1q} = 60n_{sh1}cL_h\left[\left(\frac{T_1}{T}\right)^{m_F}\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^{m_F}\frac{t_2}{t_1+t_2}\right] = 1.83\times 10^9 \text{ (cycles)}$$
$$N_{FE2q} = 60n_{sh2}cL_h\left[\left(\frac{T_1}{T}\right)^{m_F}\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^{m_F}\frac{t_2}{t_1+t_2}\right] = 0.57\times 10^9 \text{ (cycles)}$$
$$N_{HE1s} = 60n_{sh2}cL_h\left[\left(\frac{T_1}{T}\right)^3\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^3\frac{t_2}{t_1+t_2}\right] = 0.65\times 10^9 \text{ (cycles)}$$
$$N_{HE2s} = 60n_{sh3}cL_h\left[\left(\frac{T_1}{T}\right)^3\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^3\frac{t_2}{t_1+t_2}\right] = 0.21\times 10^9 \text{ (cycles)}$$
$$N_{FE1s} = 60n_{sh2}cL_h\left[\left(\frac{T_1}{T}\right)^{m_F}\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^{m_F}\frac{t_2}{t_1+t_2}\right] = 0.57\times 10^9 \text{ (cycles)}$$
$$N_{FE2s} = 60n_{sh3}cL_h\left[\left(\frac{T_1}{T}\right)^{m_F}\frac{t_1}{t_1+t_2} + \left(\frac{T_2}{T}\right)^{m_F}\frac{t_2}{t_1+t_2}\right] = 0.18\times 10^9 \text{ (cycles)}$$

**Calculate aging factor** For steel, $N_{FO1} = N_{FO2} = 4\times 10^6$ (MPa). Equation 6.3 and 6.4 [1] gives (if the factors are smaller than 1, round them up to 1, see p.94 [1]):
$K_{HL1q} = \sqrt[m_H]{N_{HO1}/N_{HE1q}} = 0.45 < 1 \Rightarrow K_{HL1q} = 1$
$K_{HL2q} = \sqrt[m_H]{N_{HO2}/N_{HE2q}} = 0.54 < 1 \Rightarrow K_{HL2q} = 1$
$K_{FL1q} = \sqrt[m_F]{N_{FO1}/N_{FE1q}} = 0.36 < 1 \Rightarrow K_{FL1q} = 1$
$K_{FL2q} = \sqrt[m_F]{N_{FO2}/N_{FE2q}} = 0.44 < 1 \Rightarrow K_{FL2q} = 1$

$K_{HL1s} = \sqrt[m_H]{N_{HO1}/N_{HE1s}} = 0.54 < 1 \Rightarrow K_{HL1s} = 1$
$K_{HL2s} = \sqrt[m_H]{N_{HO2}/N_{HE2s}} = 0.65 < 1 \Rightarrow K_{HL2s} = 1$
$K_{FL1s} = \sqrt[m_F]{N_{FO1}/N_{FE1s}} = 0.44 < 1 \Rightarrow K_{FL1s} = 1$
$K_{FL2s} = \sqrt[m_F]{N_{FO2}/N_{FE2s}} = 0.53 < 1 \Rightarrow K_{FL2s} = 1$

**Calculate** $[\sigma_H]$**,** $[\sigma_{F1}]$**,** $[\sigma_{F2}]$ Since the motor works in one direction, $K_{FC} = 1$, which means all $K$ factors are equal to 1 and we can safely skip them.

For initial estimation, assume $Z_R Z_v K_{xH} = 1$ and $Y_R Y_s K_{xF} = 1$, we obtain the permissible stresses (notice they are equal in both quick and slow transmission stages):
$[\sigma_{H1}] = \sigma^o_{Hlim1}/S_H = 518.18$ (MPa)
$[\sigma_{H2}] = \sigma^o_{Hlim2}/S_H = 500$ (MPa)
$[\sigma_{F1}] = \sigma^o_{Flim1}/S_F = 257.14$ (MPa)
$[\sigma_{F2}] = \sigma^o_{Flim2}/S_F = 246.86$ (MPa)

The mean permissible contact stress must be lower than 1.25 times of either $[\sigma_{H1}]$ or $[\sigma_{H2}]$, whichever is smaller. In this case, it is $1.25[\sigma_{H2}]$ or 625 MPa:

$$[\sigma_H] = \frac{1}{2}\left([\sigma_{H1}] + [\sigma_{H2}]\right) = 509.09 \text{ (MPa)} \leq 625$$

which satisfy the condition.

In case of overloading, the permissible contact and bending stresses are calculated as follows:
$[\sigma_H]_{max} = 2.8\sigma_{ch} = 1540$ (MPa)
$[\sigma_F]_{max} = 0.8\sigma_{ch} = 440$ (MPa)

## 4.4 Determine basic specifications of the Transmission system

### 4.4.1 Determine basic parameters

Figure 1.1 shows both pairs are helical, which gives $K_a = 43$, Table 6.5 [1]. Also, the entire speed reducer has asymmetrical design, resulting in $\psi_{ba} = 0.3$, Table 6.6 [1]. This value is then used in Equation 6.16 [1] to find $\psi_{bd}$:

$\psi_{bdq} = 0.53\psi_{ba}(u_q + 1) = 0.67$
$\psi_{bds} = 0.53\psi_{ba}(u_s + 1) = 0.65$

Using interpolation, we approximate the factors, Table 6.7 [1]:
$K_{H\beta q} = 1.04$, $K_{F\beta q} = 1.09$
$K_{H\beta s} = 1.06$, $K_{F\beta s} = 1.13$
Since the gear system only consists of involute gears and it is also a speed reducer gearbox, we estimate $a$, Equation 6.15a [1]:

$$a_q = K_a(u_q + 1)\sqrt[3]{\frac{T_{sh1}K_{H\beta q}}{[\sigma_H]^2 u_q \psi_{ba}}} = 83.12\ (\mathrm{mm})$$

$$a_s = K_a(u_s + 1)\sqrt[3]{\frac{T_{sh2}K_{H\beta s}}{[\sigma_H]^2 u_s \psi_{ba}}} = 119.85\ (\mathrm{mm})$$

It is recommended to round up the center distances to the nearest multiple of 5 for small production, p.99 [1]. Thus, $a_{wq} = 85$ mm and $a_{ws} = 120$ mm.

### 4.4.2 Determine gear meshing parameters

**Find** $m$ Using Equation 6.17 [1] and Table 6.8 [1], we determine $m$ for each pair of gears:
$m_q = (0.01 \div 0.02)a_{wq} = 0.85 \div 1.7\ (\mathrm{mm}) \Rightarrow m_q = 1.5\ (\mathrm{mm})$
$m_s = (0.01 \div 0.02)a_{ws} = 1.2 \div 2.4\ (\mathrm{mm}) \Rightarrow m_s = 2\ (\mathrm{mm})$

**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 $z_1$. Then, find $z_2$ and $b$.

$$z_{1q} = \frac{2a_{wq}\cos\beta}{m_q(u_q + 1)} = 26.01 \Rightarrow z_{1q} = 27$$

$z_{2q} = u_q z_{1q} = 87.17 \Rightarrow z_{2q} = 88$
$b_q = \psi_{ba}a_{wq} = 25.50\ (\mathrm{mm})$

$$z_{1s} = \frac{2a_{ws}\cos\beta}{m_s(u_s + 1)} = 28.42 \Rightarrow z_{1s} = 29$$

$z_{2s} = u_s z_{1s} = 89.82 \Rightarrow z_{2s} = 90$
$b_s = \psi_{ba}a_{ws} = 36.00\ (\mathrm{mm})$

**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 $\beta_w \approx 13.59° \in (10, 15]$, $z_{min} = 11$, 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 $y$.

### 4.4.3 Basic parameters

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

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

### 4.4.4 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):

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

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)

### 4.4.5 Contact stress analysis

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

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

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

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

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

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

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

**Find** $K_H$ We find $K_H$ using equation $K_H = K_{H\beta}K_{H\alpha}K_{Hv}$
From table (6.13), $v \leq 10\,(\mathrm{m/s}) \Rightarrow$ 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:

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

### 4.4.6 Bending stress analysis

For safety reasons, the following conditions must be met:

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

**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:

$$\sigma_{F1} \approx 114.11\,(\text{MPa}) \leq [\sigma_{Fw1}] \approx 297.51\,(\text{MPa})$$
$$\sigma_{F2} \approx 117.01\,(\text{MPa}) \leq [\sigma_{Fw2}] \approx 285.61\,(\text{MPa})$$

### 4.4.7 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 4.1: Gearbox specifications

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