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ME3011

# Design Project Report

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HCMC, October 5, 2020

# Acknowledgements

I would like to give a special thank to Prof. Nguyen Tan Tien, my supervisor to instruct me and fix my report to meet industrial standard.

# 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.

# Contents

<b>1</b>	<b>Design Problem</b>	<b>8</b>
1.1	Problem . . . . .	8
1.2	Mixing machine parameters . . . . .	9
1.3	Requirements . . . . .	9
1.4	Design problem . . . . .	9
<b>2</b>	<b>Choose Motor</b>	<b>10</b>
2.1	Choose motor for the mixing tank . . . . .	10
2.2	Calculate power, rotational speed and torque . . . . .	11
<b>3</b>	<b>Chain Drive Design</b>	<b>13</b>
3.1	Nomenclature . . . . .	13
3.2	Find the chain drive pitch . . . . .	14
3.3	Determine basic parameters of the chain drive . . . . .	15
3.3.1	Find number of links and center distance . . . . .	15
3.4	Strength of chain drive . . . . .	16
3.4.1	Impact frequency analysis . . . . .	16
3.4.2	Safety factor analysis . . . . .	16
3.4.3	Contact stress analysis . . . . .	17
3.5	Force on shaft . . . . .	17
<b>4</b>	<b>Gear Design (shaft 1-2)</b>	<b>19</b>
4.1	Choose material . . . . .	20
4.2	Estimate the permissible contact stress and bending stress . . . . .	20
4.3	Determine basic specifications of the Transmission system . . . . .	22
4.3.1	Determine basic parameters . . . . .	22
4.3.2	Other parameters . . . . .	23
4.4	Stress analysis . . . . .	23
4.4.1	Correct $[\sigma_H]$ , $[\sigma_{F1}]$ and $[\sigma_{F2}]$ . . . . .	23
4.4.2	Contact stress analysis . . . . .	24
4.4.3	Bending stress analysis . . . . .	25

CONTENTS	5
4.4.4 Overloading analysis . . . . .	26
<b>References</b>	<b>27</b>

# List of Figures

1.1 Working principle diagram and workload of the mixing machine: 1) electric motor, 2) elastic 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) . . . . . 8

# List of Tables

2.1	Output specifications . . . . .	12
3.1	Chain drive specifications . . . . .	18
4.1	Gear drive specifications. Material of choice is 45 steel . . . . .	26

# Chapter 1

## Design Problem

### 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 · m
$L$	service life, years	$T_2$	working torque 2, N · m
$n$	rotational velocity of the mixing tank, rpm	$t_1$	working time 1, s
		$t_2$	working time 2, s

### 1.1 Problem

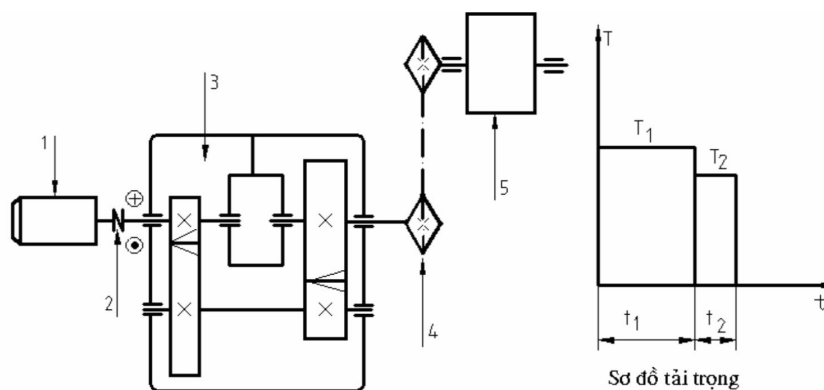


Figure 1.1: Working principle diagram and workload of the mixing machine: 1) electric motor, 2) elastic coupling, 3) two-stage coaxial helical speed reducer, 4) roller chain drive, 5) mixing tank (one-directional, light duty, operate 1 shift, 8 hours each)



## 1.2 Mixing machine parameters

From the parameters given in the document, we have:

$$P = 7 \text{ (kW)}$$

$$t_1 = 15 \text{ (s)}$$

$$n = 65 \text{ (rpm)}$$

$$t_2 = 11 \text{ (s)}$$

$$L = 8 \text{ (years)}$$

$$T_1 = T \text{ (N} \cdot \text{m)}$$

$$K_{ng} = 260 \text{ (days)}$$

$$T_2 = 0.7T \text{ (N} \cdot \text{m)}$$

$$Ca = 1 \text{ (shifts)}$$

## 1.3 Requirements

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

## 1.4 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

### Nomenclature

$n_{sh}$	rotational speed of shaft, rpm	$u_{sys}$	transmission ratio of the system
$P_{mo}$	calculated motor power to drive the system, kW	$T_{mo}$	motor torque, N · mm
$P_{sh}$	operating power of shaft, kW	$T_{sh}$	shaft torque, N · mm
$P_w$	operating power of the belt conveyor given a workload, kW	$\eta_b$	bearing efficiency
		$\eta_c$	coupling efficiency
		$\eta_{ch}$	chain drive efficiency
$u_1$	transmission ratio of quick stage	$\eta_{hg}$	helical gear efficiency
$u_2$	transmission ratio of slow stage	$\eta_{sys}$	efficiency of the system
$u_{ch}$	transmission ratio of chain drive	1	subscript for shaft 1
$u_h$	transmission ratio of speed reducer	2	subscript for shaft 2
		3	subscript for shaft 3

**Known parameters** From Chapter 1, we know that:

$$P = 7 \text{ kW}, n = 65 \text{ rpm}$$

$$T_1 = T, T_2 = 0.7T$$

$$t_1 = 15 \text{ s}, t_2 = 11 \text{ s}$$

### 2.1 Choose motor for the mixing tank

The choice of motor will affect the entire system, so it is necessary to pick the right one.

**Calculate system overall efficiency** From Table 2.3 [3]:

- 1 elastic coupling which connects the motor and the speed reducer.  $\eta_c = 1$
- 4 sealed rolling bearings. 3 of which belong to the speed reducer and the last one is used for the shaft of the mixing tank.  $\eta_b = 0.99$

- 2 sealed pairs of helical gear drive which connect the shafts inside the speed reducer.  $\eta_{hg} = 0.97$
- 1 sealed roller chain drive connecting the speed reduce and the mixing tank.  $\eta_{ch} = 0.96$

Aggregate all efficiencies yields:

$$\eta_{sys} = \eta_c \eta_b^4 \eta_{hg}^2 \eta_{ch} = 1 \times 0.99^4 \times 0.97^2 \times 0.96 = 0.87$$

**Calculate required power for operation** The power  $P$  from Chapter 1 applies for systems with single loading input. In case of varying load each cycle, the equivalent power is calculated using Equation 2.13 [3]:

$$P_w = P \sqrt{\frac{\left(\frac{T_1}{T}\right)^2 t_1 + \left(\frac{T_2}{T}\right)^2 t_2}{t_1 + t_2}} = 7 \times \sqrt{\frac{\left(\frac{T}{T}\right)^2 \times 15 + \left(\frac{0.7T}{T}\right)^2 \times 11}{15 + 11}} = 6.2 \text{ (kW)}$$

$$P_{mo} = \frac{P_w}{\eta_{sys}} = \frac{6.2}{0.87} = 7.14 \text{ (kW)}$$

**Calculate  $n_{mo}$**  The purpose is to Using Table 2.4 [3]:

- 2-level transmission speed reducer, spur gear type.  $u_h = 11$
- 1 chain drive, roller type.  $u_{ch} = 2$

Multiplying all transmission ratio yields:

$$u_{sys} = u_h u_{ch} = 11 \times 2 = 22$$

$$n_{mo} = u_{sys} n = 22 \times 65 = 1430 \text{ (rpm)}$$

**Choose motor** To work normally, the maximum operating power of the chosen motor must be no smaller than  $P_{mo}$ . In similar fashion, its rotational speed must also be no smaller than  $n_{mo}$ . Thus, from Table P1.3 [3], we choose motor 4A132S4Y3 which operates at 7.5 kW maximum and 1455 rpm, which makes  $n_{mo} = 1455 \text{ rpm}$ .

Recalculating  $u_{sys}$  with the new  $n_{mo}$  derived from the chosen motor:

$$u_{sys} = n_{mo} / n = 1455 / 65 = 22.38$$

Retaining the transmission ratio of the speed reducer (i.e. let  $u_h = \text{const} = 11$ ), the new transmission ratio of the chain drive is then:

$$u_{ch} = u_{sys} / u_h = 22.38 / 11 = 2.03$$

## 2.2 Calculate power, rotational speed and torque

Let  $P_{sh1}$ ,  $n_{sh1}$  and  $T_{sh1}$  be the transmitted power, rotational speed and torque onto shaft 1, respectively. Similarly,  $P_{sh2}$ ,  $n_{sh2}$  and  $T_{sh2}$  are the transmitted parameters onto shaft 2 and

$P_{sh3}$ ,  $n_{sh3}$  and  $T_{sh3}$  are used for shaft 3. Unless otherwise stated, these notations will be used throughout the next chapters.

**Power** The entire system is described followed by calculation as follows:

Chain drive power is affected by the bearings on the shaft of the mixing tank.

$$P_{ch} = \frac{P_w}{\eta_b} = \frac{6.2}{0.99} = 6.26 \text{ (kW)}$$

Shaft 3 power is affected by the chain drive.

$$P_{sh3} = \frac{P_{ch}}{\eta_{ch}} = \frac{6.26}{0.96} = 6.52 \text{ (kW)}$$

Shaft 2 power is affected by the bearings and gear drives on shaft 3.

$$P_{sh2} = \frac{P_{sh3}}{\eta_b \eta_{hg}} = \frac{6.52}{0.99 \times 0.97} = 6.79 \text{ (kW)}$$

Shaft 1 power is affected by the bearings and gear drives on shaft 2.

$$P_{sh1} = \frac{P_{sh2}}{\eta_b \eta_{hg}} = \frac{6.79}{0.99 \times 0.97} = 7.07 \text{ (kW)}$$

**Rotational speed** The design goal of the speed reducer is to lubricate both driven gears equally, which has a size disadvantage. Therefore, the transmission ratio of each pair of gears is calculated using Equation 3.12 [3]:

$$u_1 = u_2 = \sqrt{u_h} = \sqrt{11} = 3.32$$

Then,

from motor to shaft 1:  $n_{sh1} = n_{mo} = 1455 \text{ (rpm)}$

from shaft 1 to shaft 2:  $n_{sh2} = n_{sh1}/u_1 = 1455/3.32 = 438.70 \text{ (rpm)}$

from shaft 2 to shaft 3:  $n_{sh3} = n_{sh2}/u_2 = 438.70/3.32 = 132.27 \text{ (rpm)}$

**Torque** Subsequently, the torque is calculated as follows:

$$T_{mo} = 9.55 \times 10^6 \times P_{mo}/n_{mo} = 9.55 \times 10^6 \times 7.14/1455 = 46892.66 \text{ (N} \cdot \text{mm)}$$

$$T_{sh1} = 9.55 \times 10^6 \times P_{sh1}/n_{sh1} = 9.55 \times 10^6 \times 7.07/1455 = 46423.73 \text{ (N} \cdot \text{mm)}$$

$$T_{sh2} = 9.55 \times 10^6 \times P_{sh2}/n_{sh2} = 9.55 \times 10^6 \times 6.79/438.70 = 147857.49 \text{ (N} \cdot \text{mm)}$$

$$T_{sh3} = 9.55 \times 10^6 \times P_{sh3}/n_{sh3} = 9.55 \times 10^6 \times 6.52/132.27 = 470919.44 \text{ (N} \cdot \text{mm)}$$

In summary, we obtain the following table:

	Motor	Shaft 1	Shaft 2	Shaft 3
$P$ (kW)	7.14	7.07	6.79	6.52
$u$	-	1	3.32	3.32
$n$ (rpm)	1455	1455	438.70	132.27
$T$ (N · mm)	46892.66	46423.73	147857.49	470919.44

Table 2.1: Output specifications

# Chapter 3

## Chain Drive Design

### 3.1 Nomenclature

$[i]$	permissible impact times per second	$F_1$	tight side tension force, N
$[s]$	permissible safety factor	$F_2$	slack side tension force, N
$[P]$	permissible power, kW	$F_r$	force on the shaft, N
$[\sigma_H]$	permissible contact stress, MPa	$F_t$	effective peripheral force, N
$A$	cross sectional area of chain hinge, mm <sup>2</sup>	$F_v$	centrifugal force, N
$a$	real center distance, mm	$F_{vd}$	contact force, N
$a_i$	estimated center distance, mm	$i$	impact times per second
$a_{max}$	maximum center distance, mm	$K_d$	weight distribution factor
$a_{min}$	minimum center distance, mm	$k$	overall factor
$B$	width between inner link plate, mm	$k_a$	center distance and chain's length factor
$d$	chordal diameter, mm	$k_{bt}$	lubrication factor
$d_a$	addendum diameter, mm	$k_c$	rating factor
$d_f$	dedendum diameter, mm	$k_d$	dynamic load factor
$d_l$	roller diameter, mm	$k_{dc}$	chain tension factor
$d_O$	pin diameter, mm	$k_f$	loosing factor
$E$	modulus of elasticity, MPa	$k_n$	coefficient of rotational speed
$F_0$	sagging force, N	$k_r$	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.

$$n_{sh3} = 132.27 \text{ rpm}$$

$$P_{ch} = 6.26 \text{ kW}, u_{ch} = 4.5$$

## 3.2 Find the chain drive pitch

Chain hinge wear is one of the main failure modes, which poses a risk of damaging the entire system. In this section, the method to find the right pitch is due to the strength criterion of the chain hinge, derived into Equation 5.3 [3]. The driving sprocket is connected to shaft 3,  $n_1 = n_{sh3} = 132.27 \text{ rpm}$ .

**Calculate  $z$**  The number of teeth determines the how stable is the rotational speed, which relates to the impact intensity and service life:

$$z_1 = 29 - 2u_{ch} = 29 - 2 \times 2.03 = 25 \geq 19 \text{ (rounded to the nearest odd number)}$$

$$z_2 = u_{ch}z_1 = 2.03 \times 25 = 51 \leq 120 \text{ (Equation 5.1 [3])}$$

**Calculate  $[P]$**  An experiment is conducted to find the optimal pitch given the permissible power and angular rotational speed. A roller chain drive having 25 teeth on the driving sprocket is tested in 8 different cases of  $n_{01}$  in somewhat similar condition with our design purpose, p.80 [3]. In this problem,  $z_1 = 25$ ;  $n_1 = 132.27 \text{ rpm}$ , which is close to  $n_{01} = 200 \text{ rpm}$ , which yields  $k_z = 25/z_1 = 25/25 = 1$  and  $k_n = n_{01}/n_1 = 200/132.27 = 1.51$ .

Another step is to specify the working condition of the chain, Table 5.6:

- The centerline between 2 sprockets is parallel with the ground.  $k_0 = 1$
- Center distance  $a = (30 \div 50)p$ , which is similar to the experiment.  $k_a = 1$
- Center distance is modifiable through displacing the sprockets.  $k_{dc} = 1$
- Moderate impact load.  $k_d = 1.5$

- 1 shift.  $k_c = 1$
- Dusty condition with moderate lubrication quality.  $k_{bt} = 1.3$

Then, we can obtain the value  $[P]$ , Equation 5.3 [3]:

$$P_t = P_{ch} k k_z k_n = P_{ch} k_0 k_a k_{dc} k_d k_c k_{bt} k_z k_n$$

$$= 6.26 \times 1 \times 1 \times 1 \times 1.5 \times 1 \times 1.3 \times 1 \times 1.51 = 18.46 \text{ (kW)} \leq [P]$$

Inspecting Table 5.5 [3] at column  $n_{01} = 200 \text{ rpm}$ , the closet value is  $[P] = 19.3 \text{ kW}$ . Knowing  $[P]$ , the pitch is  $p = 31.75 \text{ mm}$ , Table 5.5 [3]. Consequently,  $d_c = 9.55 \text{ mm}$ ,  $B = 27.46 \text{ mm}$ . Because minimizing damage from impact onto the drive is essential, Table 5.8 [3] is consulted. In this case, the pitch is indeed suitable.

### 3.3 Determine basic parameters of the chain drive

#### 3.3.1 Find number of links and center distance

The parameters are found:

**Find  $x_c$**  The  $a_{min} = 30p = 30 \times 31.75 = 952.50 \text{ (mm)}$ ,  $a_{max} = 50p = 50 \times 31.75 = 1587.50 \text{ (mm)}$ . Limiting the range of choice for  $a$  in  $[a_{min}, a_{max}]$  to reduce the effect from chain weight, we can approximate  $a = 1300 \text{ mm}$  and find  $x_c$ :

$$x = \frac{2a}{p} + \frac{z_1 + z_2}{2} + \frac{(z_2 - z_1)^2 p}{4\pi^2 a} = \frac{2 \times 1300}{31.75} + \frac{25 + 51}{2} + \frac{(51 - 25)^2 \times 31.75}{4\pi^2 \times 1300} = 120.31$$

$x$  is rounded up to the nearest even number.  $x_c = 122$

**Find  $a$**  Using  $x_c$  to find the correct center distance, Equation 5.13 [3]. In addition, it is recommended to loose the chain an amount of  $0.002 \div 0.004a$  to reduce tension. Choosing the amount of  $0.003a$ , the coefficient 0.997 is included in the formula below:

$$a = \frac{0.997p}{4} \left[ x_c - \frac{z_2 + z_1}{2} + \sqrt{\left( x_c - \frac{z_2 + z_1}{2} \right)^2 - 2 \left( \frac{z_2 - z_1}{\pi} \right)^2} \right]$$

$$= \frac{0.997 \times 31.75}{4} \left[ 122 - \frac{51 + 25}{2} + \sqrt{\left( 122 - \frac{51 + 25}{2} \right)^2 - 2 \left( \frac{51 - 25}{\pi} \right)^2} \right] = 1019.99 \text{ (mm)}$$

**Find other parameters** The values below are necessary for modeling the chain drive (the last two values use  $d_l = 19.05 \text{ mm}$  from Table 5.2 [3]):

$$d_1 = p / \sin \left( \frac{\pi}{z_1} \right) = 31.75 / \sin \left( \frac{180}{25} \right) = 253.32 \text{ (mm)}$$

$$d_2 = p / \sin \left( \frac{\pi}{z_2} \right) = 31.75 / \sin \left( \frac{180}{51} \right) = 515.75 \text{ (mm)}$$

$$\begin{aligned}
d_{a1} &= p \left( 0.5 + \cot \frac{180}{z_1} \right) = 31.75 \left( 0.5 + \cot \frac{180}{25} \right) = 267.20 \text{ (mm)} \\
d_{a2} &= p \left( 0.5 + \cot \frac{180}{z_2} \right) = 31.75 \left( 0.5 + \cot \frac{180}{51} \right) = 530.65 \text{ (mm)} \\
d_{f1} &= d_1 - 2(0.502d_l + 0.05) = 253.32 - 2(0.502 \times 19.05 + 0.05) = 234.08 \text{ (mm)} \\
d_{f2} &= d_2 - 2(0.502d_l + 0.05) = 515.75 - 2(0.502 \times 19.05 + 0.05) = 496.50 \text{ (mm)}
\end{aligned}$$

### 3.4 Strength of chain drive

#### 3.4.1 Impact frequency analysis

After determine the center distance, it is necessary to compare the impact frequency with its permissible value, which is  $[i] = 25$ , Table 5.9 [3]. Replacing all the variables gives:

$$i = \frac{z_1 n_1}{15x} = \frac{25 \times 132.27}{15 \times 120.31} = 1.83 < [i]$$

which satisfies the condition.

#### 3.4.2 Safety factor analysis

Overloading often occurs at the beginning of operation or due to large load, which could damage the chain drive. In order to operate safely, the chain drive's safety factor must satisfy the following condition:

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

**Find  $v$**  Calculated for other variables. The rotational speed of the smaller sprocket is determined using the formula:  $v_1 = n_1 p z_1 / 60000 = 132.27 \times 31.75 \times 25 / 60000 = 2.6 \text{ (m/s)}$

**Find  $Q$  and  $q$**  Table 5.2 [3] and  $p = 31.75$  are used.  $Q = 88500 \text{ N}$  and  $q = 3.8 \text{ kg/m}$

**Find  $k_d$** : The workload is moderate. When turning on the machine, it is around 1.5 times the amount of the nominal load.  $k_d = 1.2$

**Find  $k_f$** : The chain drive is parallel to the ground.  $k_f = 6$

**Find  $F_t$ ,  $F_v$  and  $F_0$**  Substituting the values above to obtain the values:

$$F_t = 1000 P_{ch} / v_1 = 1000 \times 6.26 / 2.6 = 3578.34 \text{ (N)}$$

$$F_v = q v_1^2 = 3.8 \times 6.26^2 = 11.64 \text{ (N)}$$

$$F_0 = 9.81 \times 10^{-3} k_f q a = 9.81 \times 10^{-3} \times 6 \times 3.8 \times 19.05 = 295.92 \text{ (N)}$$

**Find  $[s]$** : The limit is found using interpolation, Table 5.10 [3].  $[s] = 7.66$

Replacing all the variables gives:

$$s = \frac{88500}{1.2 \times 3578.34 + 295.92 + 11.64} = 19.23 \geq 7.66$$

which satisfies the condition.



### 3.4.3 Contact stress analysis

The following condition must be met, Equation 5.18 [3]:

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

**Find  $[\sigma_H]$**  The chain drive has  $z_2 = 51 > 30$  and  $v_1 = 2.6 \text{ mm} < 5 \text{ mm}$ . Thus the suitable material is quenched 45 steel, which has  $[\sigma_H] = 500 \text{ MPa}$ , Table 5.11 [3] and  $E_{sprocket} = 205000 \text{ MPa}$  [1].

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

**Find  $F_{vd}$**  For 1-strand chain,  $F_{vd} = 13 \times 10^{-7} n_1 p^3 = 13 \times 10^{-7} \times 132.27 \times 31.75^3 = 6.22 \text{ (N)}$

**Find  $k_r$**  Since  $z_1$  is used to estimate  $k_r$  on p.87 [3],  $k_r = 0.42$ .

**Find  $E$**  Assuming the sprockets and chain are made up from the same material (steel). The chain is cast iron ASTM-A48 No. 30A, which has  $E_{chain} = 117000 \text{ MPa}$  [2]. The equivalent modulus of elasticity is then:

$$E = \frac{2E_{sprocket}E_{chain}}{E_{sprocket} + E_{chain}} = \frac{2 \times 205000 \times 117000}{205000 + 117000} = 148975.16 \text{ (MPa)}$$

**Find  $A$**  The projected area of the hinge depends on the pitch  $p = 31.75 \text{ mm}$ , Table 5.12 [3].  $A = 262 \text{ mm}^2$ .

The value  $k_d = 1.5$  and  $F_t = 3578.34 \text{ N}$  are found above. Replacing all the variables gives:

$$\sigma_H = 0.47 \sqrt{\frac{0.42 \times (3578.34 \times 1.5 + 6.22) \times 148975.16}{262 \times 1}} = 476.25 \text{ (MPa)} \leq 500 \text{ MPa}$$

which satisfies the condition.

## 3.5 Force on shaft

Apply the following equations, p.87 [3]:

$$F_2 = F_0 + F_v = 295.92 + 11.64 = 307.55 \text{ (N)}$$

$$F_1 = F_t + F_2 = 3578.34 + 307.55 = 3885.89 \text{ (N)}$$

The chain drive is parallel to the ground corresponding to  $k_x = 1.15$ . Apply Equation 5.20 [3]:

$$F_r = k_x F_t = 1.15 \times 3578.34 = 2772.05 \text{ (N)}$$

In summary, we have the following table:

	Chain drive	Driving sprocket	Driven sprocket
$a$ (mm)	1323.02	-	-
$B$ (mm)	27.46	-	-
$d$ (mm)	-	253.32	515.75
$d_a$ (mm)	-	267.20	530.65
$d_c$ (mm)	9.55	-	-
$d_f$ (mm)	-	234.08	496.50
$d_l$ (mm)	19.05	-	-
$z$		25	51

Table 3.1: Chain drive specifications

# Chapter 4

## Gear Design (shaft 1-2)

### Nomenclature

$[\sigma]$	permissible stress, MPa	$m_n$	normal module, mm
$[\sigma]_{max}$	permissible stress due to overloading, MPa	$N_E$	working cycle of equivalent tensile stress
AG	accuracy grade of gear	$N_O$	working cycle of bearing stress
$a$	center distance, mm	$S$	safety factor
$b$	face width, mm	$v$	rotational velocity, m/s
$c$	gear meshing rate	$Y_F$	tooth shape factor
$d$	pitch circle, mm	$Y_R$	surface roughness factor
$d_a$	addendum diameter, mm	$Y_s$	stress concentration factor
$d_b$	base diameter, mm	$Y_\beta$	helix angle factor
$d_f$	deddendum diameter, mm	$Y_\varepsilon$	contact ratio factor
$H$	surface roughness, HB	$Z_R$	surface roughness factor of the working's area
$K_d$	coefficient of gear material	$Z_v$	speed factor
$K$	overall load factor	$z_H$	contact surface's shape factor
$K_C$	load placement factor	$z_M$	material's mechanical properties factor
$K_L$	aging factor		
$K_v$	dynamic load factor at meshing area	$z_v$	virtual number of teeth
$K_\alpha$	load distribution factor on gear teeth	$z_\varepsilon$	meshing condition factor
$K_\beta$	load distribution factor on top land	$\alpha$	normal pressure angle. In TCVN 1065-71, $\alpha = 20^\circ$
$K_{qt}$	overloading factor	$\alpha_t$	traverse pressure angle, $^\circ$
$m$	root of fatigue curve in stress test	$\varepsilon_\alpha$	traverse contact ratio
$m_t$	traverse module, mm	$\varepsilon_\beta$	face contact ratio
		$\beta$	helix angle, $^\circ$

$\beta_b$	base circle helix angle, °	$\sigma_{max}$	stress due to overloading, MPa
$\psi_{ba}$	width to shaft distance ratio	1	subscript for the pinion
$\psi_{bd}$	face width factor	2	subscript for the driven gear
$\sigma_b$	ultimate strength, MPa	$F$	subscript relating to bending stress
$\sigma_{ch}$	yield limit, MPa	$H$	subscript relating to contact stress
$\sigma_{lim}^o$	permissible stress corresponding to working cycle, MPa		

**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

$u_h = 10$ ,  $u_1 = u = 3.23$ ,  $u_2 = u_s = 3.1$

## 4.1 Choose material

For small scale machines, type I is sufficient. Materials of this type are often quenched and have low hardness value ( $HB \leq 350$ ), which make it easy to manufacture precisely. However, their permissible stresses are low. In this design problem, 45 steel is used to manufacture both gears. The specifications of the material are HB205,  $\sigma_b = 600$  MPa,  $\sigma_{ch} = 340$  MPa, Table 6.1 [3].

## 4.2 Estimate the permissible contact stress and bending stress

The permissible stresses are calculated as follows, Equation 6.1 and 6.2 [3]:

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

$$[\sigma_F] = \frac{\sigma_{Flim}^o}{S_F} Y_R Y_s K_{xF} K_{FC} K_{FL}$$

Initial estimation of these values assumes  $Z_R Z_v K_{xH} = 1$  and  $Y_R Y_s K_{xF} = 1$ . The factors will be accounted for in the next sections.

**Find**  $\sigma_{lim}^o$  The value depends on hardness. It is also recommended to have the hardness of the pinion slightly higher than the driving gear from 10 to 15 units. Using Table 6.2 [3], the formulas below are used for quenched 45 steel:

$$\sigma_{Hlim}^o = 2HB + 70$$

$$\sigma_{Flim}^o = 1.8HB$$

For the pinion with  $H_1 = HB205$ :

$$\sigma_{Hlim1}^o = 2 \times 205 + 70 = 480 \text{ (MPa)}$$

$$\sigma_{Flim1}^o = 1.8 \times 205 = 369 \text{ (MPa)}.$$

For the driven gear with  $H_2 = H_1 - 15 = 205 - 15 = \text{HB190}$ :

$$\sigma_{Hlim2}^o = 2 \times 190 + 70 = 450 \text{ (MPa)}$$

$$\sigma_{Flim2}^o = 1.8 \times 190 = 342 \text{ (MPa)}.$$

**Find  $S$**  The safety factors are found using Table 6.2 [3].  $S_H = 1.1$ ,  $S_F = 1.75$

**Find  $K_{FC}$**  The motor works in one direction and the load is placed in one way.  $K_{FC} = 1$

**Find  $K_L$**  The aging factor is determined by Equation 6.3 and 6.4 [3] for contact stress and bending stress, respectively. The surface hardness of the gears is smaller than 350, which makes  $m_H = 6$ ,  $m_F = 6$ . Also, both pairs of gears are meshed indefinitely which makes  $c = 1$ . From working condition, the service life in hours are found:

$$L_h = 8 \left( \frac{\text{hours}}{\text{shift}} \right) \times CaK_{ng}L = 8 \times 1 \times 260 \times 8 = 16640 \text{ (hours)}$$

$N_{HE}$  and  $N_{FE}$  are found using Equation 6.7 and 6.8 [3]:

$$\begin{aligned} N_{HE1} &= 60n_{sh1}cL_h \left[ \left( \frac{T_1}{T} \right)^3 \frac{t_1}{t_1 + t_2} + \left( \frac{T_2}{T} \right)^3 \frac{t_2}{t_1 + t_2} \right] \\ &= 60 \times 1455 \times 1 \times 16640 \left[ \left( \frac{T}{T} \right)^3 \frac{15}{15 + 11} + \left( \frac{0.7T}{T} \right)^3 \frac{11}{15 + 11} \right] = 1.05 \times 10^9 \text{ (cycles)} \end{aligned}$$

$$\begin{aligned} N_{HE2} &= 60n_{sh2}cL_h \left[ \left( \frac{T_1}{T} \right)^3 \frac{t_1}{t_1 + t_2} + \left( \frac{T_2}{T} \right)^3 \frac{t_2}{t_1 + t_2} \right] \\ &= 60 \times 438.70 \times 1 \times 16640 \left[ \left( \frac{T}{T} \right)^6 \frac{15}{15 + 11} + \left( \frac{0.7T}{T} \right)^6 \frac{11}{15 + 11} \right] = 0.32 \times 10^9 \text{ (cycles)} \end{aligned}$$

$$\begin{aligned} N_{FE1} &= 60n_{sh1}cL_h \left[ \left( \frac{T_1}{T} \right)^{m_F} \frac{t_1}{t_1 + t_2} + \left( \frac{T_2}{T} \right)^{m_F} \frac{t_2}{t_1 + t_2} \right] \\ &= 60 \times 1455 \times 1 \times 16640 \left[ \left( \frac{T}{T} \right)^6 \frac{15}{15 + 11} + \left( \frac{0.7T}{T} \right)^6 \frac{11}{15 + 11} \right] = 0.91 \times 10^9 \text{ (cycles)} \end{aligned}$$

$$\begin{aligned} N_{FE2} &= 60n_{sh2}cL_h \left[ \left( \frac{T_1}{T} \right)^{m_F} \frac{t_1}{t_1 + t_2} + \left( \frac{T_2}{T} \right)^{m_F} \frac{t_2}{t_1 + t_2} \right] \\ &= 60 \times 438.70 \times 1 \times 16640 \left[ \left( \frac{T}{T} \right)^6 \frac{15}{15 + 11} + \left( \frac{0.7T}{T} \right)^6 \frac{11}{15 + 11} \right] = 0.27 \times 10^9 \text{ (cycles)} \end{aligned}$$

The working cycle of bearing stress is found using the formula below, Equation 6.5 [3]:

$$N_{HO1} = 30H_1^{2.4} = 30 \times 205^{2.4} = 10600601.81 \text{ (cycles)}.$$

$$N_{HO2} = 30H_2^{2.4} = 30 \times 190^{2.4} = 8833440.68 \text{ (cycles)}.$$

For steel,  $N_{FO1} = N_{FO2} = 4000000$  (MPa). The factors  $K_L$  are rounded up to 1 if it is less than 1, p.94 [3]. Apply Equation 6.3 and 6.4 [3]:

$$K_{HL1} = \sqrt[6]{N_{HO1}/N_{HE1}} = \sqrt[6]{10600601.81/1.05 \times 10^9} = 0.46 < 1 \Rightarrow K_{HL1} = 1$$

$$K_{HL2} = \sqrt[6]{N_{HO2}/N_{HE2}} = \sqrt[6]{8833440.68/0.32 \times 10^9} = 0.55 < 1 \Rightarrow K_{HL2} = 1$$

$$K_{FL1} = \sqrt[6]{N_{FO1}/N_{FE1}} = \sqrt[6]{4000000/0.91 \times 10^9} = 0.40 < 1 \Rightarrow K_{FL1} = 1$$

$$K_{FL2} = \sqrt[6]{N_{FO2}/N_{FE2}} = \sqrt[6]{4000000/0.27 \times 10^9} = 0.49 < 1 \Rightarrow K_{FL2} = 1$$

**Calculate**  $[\sigma_H]$ ,  $[\sigma_{F1}]$ ,  $[\sigma_{F2}]$  Replacing all the values:

$$[\sigma_{H1}] = \sigma_{Hlim1}^o K_{HL1}/S_H = 480 \times 1/1.1 = 436.36 \text{ (MPa)}$$

$$[\sigma_{H2}] = \sigma_{Hlim2}^o K_{HL2}/S_H = 450 \times 1/1.1 = 409.09 \text{ (MPa)}$$

$$[\sigma_{F1}] = \sigma_{Flim1}^o K_{FC} K_{FL1}/S_F = 369 \times 1 \times 1/1.75 = 210.86 \text{ (MPa)}$$

$$[\sigma_{F2}] = \sigma_{Flim2}^o K_{FC} K_{FL2}/S_F = 342 \times 1 \times 1/1.75 = 195.43 \text{ (MPa)}$$

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  $1.25[\sigma_{H2}]$ . Replacing all the variables gives:

$$[\sigma_H] = \frac{[\sigma_{H1}] + [\sigma_{H2}]}{2} = \frac{436.36 + 409.09}{2} = 422.73 \text{ (MPa)}$$

$$\leq 1.25[\sigma_{H2}] = 1.25 \times 409.09 = 511.36 \text{ (MPa)}$$

which satisfy the condition.

In case of overloading, the permissible contact and bending stresses are calculated as follows:

For quenched material:  $[\sigma_H]_{max} = 2.8 \times 600 = 952 \text{ (MPa)}$

For material with hardness smaller than HB350:  $[\sigma_F]_{max} = 0.8\sigma_{ch} = 0.8 \times 340 = 272 \text{ (MPa)}$

## 4.3 Determine basic specifications of the Transmission system

### 4.3.1 Determine basic parameters

Figure 1.1 shows both pairs are helical, which gives  $K_a = 43$ , Table 6.5 [3]. Also, the pinion is symmetrical about the bearings and both gears surface have hardness smaller than HB350, resulting in  $\psi_{ba} = 0.3$ , Table 6.6 [3]. Then,  $\psi_{bd}$  is calculated using Equation 6.16 [3]:

$\psi_{bd} = 0.53\psi_{ba}(u + 1) = 0.53 \times 0.3 \times (3.32 + 1) = 0.69$ , which is smaller than the permissible ratio  $\psi_{bdmax} = 1.2$ , Table 6.6 [3].

The gear placement in the speed reducer is similar to diagram 5 of Table 6.7 [3].  $K_{H\beta} = 1.04$ ,  $K_{F\beta} = 1.1$

**Find a** Since the gear drive only has involute gears,  $a$  is estimated using Equation 6.15a [3]. It is also rounded up to the nearest multiple of 5 (small production type, p.99 [3]):

$$a = K_a(u + 1) \sqrt[3]{\frac{T_{sh1} K_{H\beta}}{[\sigma_H]^2 u \psi_{ba}}} = 43 \times (3.32 + 1) \times \sqrt[3]{\frac{46423.73 \times 1.04}{422.73^2 \times 3.32 \times 0.3}} = 120.14 \text{ (mm)}$$

$$\Rightarrow a = 180 \text{ mm}$$

Due to complexity and maintenance, the undercutting process will be omitted throughout the calculations, that is, it is not factored in any variables of the gear drives.

**Find  $m$**  Using Equation 6.17 [3] and Table 6.8 [3], we determine  $m$  for each pair of gears:  
 $m_t = (0.01 \div 0.02)a = (0.01 \div 0.02) \times 180 = 1.8 \div 3.6 \text{ (mm)} \Rightarrow m_t = 2 \text{ mm}$

**Find  $z_1, z_2, b$**  Arbitrarily choose  $\beta = 20^\circ$  (permissible value is  $8 \div 20^\circ$ ). Combining Equation 6.18 and 6.20 [3] to calculate  $z_1$ :

$$z_1 = \frac{2a \cos \beta}{m_t(u+1)} = \frac{2 \times 180 \cos 20^\circ}{2 \times (3.32 + 1)} = 39.18 \Rightarrow z_1 = 40$$

Then, find  $z_2$  and  $b$ :

$$z_2 = uz_1 = 3.32 \times 40 = 132.66 \Rightarrow z_2 = 133$$

$$b = \psi_{ba}a = 0.3 \times 180 = 54 \text{ (mm)}$$

**Correct  $\beta$**  The helix angles are corrected to compensate for rounding center distances and number of teeth, Equation 6.32 [3]:

$$\beta = \arccos \frac{m_t(z_2 + z_1)}{2a} = \arccos \frac{2 \times (133 + 40)}{2 \times 180} = 16.03^\circ$$

### 4.3.2 Other parameters

$$d_1 = m_t z_1 / \cos \beta = 2 \times 40 / \cos 16.03^\circ = 83.24 \text{ (mm)}$$

$$d_2 = m_t z_2 / \cos \beta = 2 \times 133 / \cos 16.03^\circ = 276.76 \text{ (mm)}$$

$$d_{a1} = d_1 + 2m_t = 83.24 + 2 \times 2 = 87.24 \text{ (mm)}$$

$$d_{a2} = d_2 + 2m_t = 276.76 + 2 \times 2 = 280.76 \text{ (mm)}$$

$$d_{f1} = d_1 - 2.5m_t = 83.24 - 2.5 \times 2 = 78.24 \text{ (mm)}$$

$$d_{f2} = d_2 - 2.5m_t = 276.76 - 2.5 \times 2 = 271.76 \text{ (mm)}$$

$$d_{b1} = d_1 \cos \alpha = 83.24 \times \cos 20^\circ = 78.22 \text{ (mm)}$$

$$d_{b2} = d_2 \cos \alpha = 276.76 \times \cos 20^\circ = 260.07 \text{ (mm)}$$

$$\alpha_t = \arctan (\tan \alpha / \cos \beta) = \arctan (\tan 20^\circ / \cos 16.03^\circ) = 20.74^\circ$$

$$v_1 = \pi d_1 n_{sh1} / 60000 = \pi \times 83.24 \times 1455 / 60000 = 6.34 \text{ (m/s)}$$

## 4.4 Stress analysis

### 4.4.1 Correct $[\sigma_H]$ , $[\sigma_{F1}]$ and $[\sigma_{F2}]$

In reality, the factors  $Z_R Z_v K_{xH}$  and  $Y_R Y_s K_{xF}$  do not equal to 1. This part will correct them to obtain the correct permissible stresses.

- The roughness deviation is less than  $1.25 \mu\text{m}$ .  $Z_R = 1$

- The surface hardness of the gear drive is less than HB350.  $Z_v = 0.85v_1^{0.1} = 0.85 \times 6.34^{0.1} = 1.02$
- The gears have  $d_{a1}, d_{a2} \leq 700$  mm.  $K_{xH} = 1$
- The gears are not polished.  $Y_R = 1$
- $Y_s = 1.08 - 0.0695 \ln(m_t) = 1.03$
- $d_{a1}, d_{a2} \leq 400$  mm.  $K_{xF} = 1$

Replacing all the variables gives:

$$[\sigma_H] = 422.73 \times 1 \times 1.02 \times 1 = 432.21 \text{ MPa}$$

$$[\sigma_{F1}] = 210.86 \times 1 \times 1.03 \times 1 = 217.57 \text{ MPa}$$

$$[\sigma_{F2}] = 195.43 \times 1 \times 1.03 \times 1 = 201.65 \text{ MPa}$$

#### 4.4.2 Contact stress analysis

The contact stress applied on a gear surface must satisfy Equation 6.33 [3]:

$$\sigma_H = z_M z_H z_\varepsilon \sqrt{2T_{sh1} K_H \frac{u+1}{b u d_1^2}} \leq [\sigma_H]$$

**Find**  $z_M$  According to Table 6.5 [3],  $z_M = 274$ .

**Find**  $z_H$  Applying Equation 6.34 [3] and 6.35 [3]:

$$\beta_b = \arctan(\cos \alpha_t \tan \beta) = \arctan(\cos 20.47^\circ \tan 16.03^\circ) = 15.04^\circ$$

$$z_H = \sqrt{\frac{2 \cos \beta_b}{\sin(2\alpha_t)}} = \sqrt{\frac{2 \times \cos 15.04^\circ}{\sin(2 \times 20.74^\circ)}} = 1.71$$

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

$$\begin{aligned} \varepsilon_\alpha &= \frac{\sqrt{d_{a1}^2 - d_{b1}^2} + \sqrt{d_{a2}^2 - d_{b2}^2} - 2a \sin \alpha_t}{2\pi m_t \cos \alpha_t / \cos \beta} \\ &= \frac{\sqrt{87.24^2 - 78.22^2} + \sqrt{280.76^2 - 260.07^2} - 2 \times 180 \times \sin 20.74^\circ}{2\pi \times 2 \times \cos 20.74^\circ / \cos 16.03^\circ} = 1.38 \end{aligned}$$

$$\varepsilon_\beta = b \frac{\sin \beta}{m_t \pi} = 54 \times \frac{\sin 16.03^\circ}{2 \times \pi} = 2.37 > 1 \Rightarrow z_\varepsilon = \varepsilon_\alpha^{-0.5} = 1.38^{-0.5} = 0.85$$

**Find**  $K_H$  and  $K_F$  We find  $K_H$ ,  $K_F$  using Equation 6.39 and 6.45 [3].

The gear drive velocity determines the accuracy grade, Table 6.13 [3]. In case of helical gear, it is:

$$v_1 = 6.34 \text{ m/s} \leq 10 \text{ m/s} \Rightarrow \text{AG} = 8$$

From Table P2.3 [3], using interpolation, we approximate:

$$K_{Hv} = 1.04, K_{Fv} = 1.18$$



From Table 6.14 [3], using interpolation, we approximate:

$$K_{H\alpha} = 1.06, K_{F\alpha} = 1.3$$

Knowing  $K_{H\beta}$  and  $K_{H\gamma}$  from previous section, multiply all the values to obtain  $K_H$  and  $K_F$ :

$$K_H = K_{H\beta}K_{H\gamma}K_{H\alpha} = 1.04 \times 1.1 \times 1.06 = 1.21$$

$$K_F = K_{F\beta}K_{F\gamma}K_{F\alpha} = 1.1 \times 1.18 \times 1.3 = 1.68$$

**Find  $\sigma_H$**  Replacing all the variables gives:

$$\begin{aligned}\sigma_H &= 274 \times 1.71 \times 1.55 \times \sqrt{2 \times 46423.73 \times 1.21 \times \frac{3.32 + 1}{54 \times 3.32 \times 83.24^2}} \\ &= 249.04 \text{ (MPa)} \leq 432.21 \text{ MPa}\end{aligned}$$

which satisfies the condition.

#### 4.4.3 Bending stress analysis

For safety reasons, Equation 6.43 and 6.44 [3] must be met for both pairs of gears:

$$\begin{aligned}\sigma_{F1} &= \frac{2T_{sh1}K_F Y_\varepsilon Y_\beta Y_{F1}}{b d_1 m_t \cos \beta} \leq [\sigma_{F1}] \\ \sigma_{F2} &= \sigma_{F1} Y_{F2} / Y_{F1} \leq [\sigma_{F2}]\end{aligned}$$

**Find  $Y_\varepsilon$**  Using  $\varepsilon_\alpha$  calculated in the previous section,  $Y_\varepsilon = \varepsilon_\alpha^{-1} = 1.38^{-1} = 0.72$

**Find  $Y_\beta$**  The value of  $Y_\beta$  is calculated using the equation on p.108 [3]:

$$Y_\beta = 1 - \beta / 140^\circ = 1 - 16.03^\circ / 140^\circ = 0.89$$

**Find  $Y_F$**  Using the formula  $z_v = z \cos^{-3}(\beta)$  and Table 6.18 [3]:

$$z_{v1} = z_1 \cos^{-3}(\beta) = 40 \times \cos^{-3}(16.03^\circ) = 45.05 \Rightarrow Y_{F1} = 3.67$$

$$z_{v2} = z_2 \cos^{-3}(\beta) = 133 \times \cos^{-3}(16.03^\circ) = 149.81 \Rightarrow Y_{F2} = 3.60$$

**Find  $K_F$**  The value of  $K_F$  has already been found in the previous section.  $K_F = 1.68$

**Find  $\sigma_F$**  The module in Equation 6.43 [3] is  $m_n = m_t \cos \beta$ . Replacing all the values gives:

$$\begin{aligned}\sigma_{F1} &= \frac{2 \times 46423.73 \times 1.68 \times 0.72 \times 0.89 \times 3.67}{54 \times 83.24 \times 2 \times \cos 16.03^\circ} = 42.44 \text{ (MPa)} \leq 217.57 \text{ MPa} \\ \sigma_{F2} &= 42.44 \times 3.60 / 3.66 = 41.57 \text{ (MPa)} \leq 201.65 \text{ MPa}\end{aligned}$$

which satisfies the conditions.

#### 4.4.4 Overloading analysis

From the load diagram, we determine the overloading factor:

$$K_{qt} = \sqrt{\frac{\left(\frac{T_1}{T}\right)^2 t_1 + \left(\frac{T_2}{T}\right)^2 t_2}{t_1 + t_2}} = \sqrt{\frac{\left(\frac{T}{T}\right)^2 \times 15 + \left(\frac{0.7T}{T}\right)^2 \times 11}{15 + 11}} = 1.13$$

Using the values of  $[\sigma_H]_{max}$  and  $[\sigma_F]_{max}$  calculated in previous section combined with Equation 6.48 and 6.49 [3], we are able to verify the stresses are below overloading limits. Replacing all the variables gives

$$\sigma_{Hmax} = \sigma_H \sqrt{K_{qt}} = 249.04 \times \sqrt{1.13} = 264.64 \text{ (MPa)} \leq 952.00 \text{ MPa}$$

$$\sigma_{F1max} = \sigma_{F1s} K_{qt} = 42.44 \times 1.13 = 47.92 \text{ (MPa)} \leq 440 \text{ MPa}$$

$$\sigma_{F2max} = \sigma_{F2} K_{qt} = 41.57 \times 1.13 = 46.95 \text{ (MPa)} \leq 272.00 \text{ MPa}$$

which satisfy the conditions.

In summary, we have the following table:

	Gear drive	Pinion	Driven gear
$a$ (mm)	180	-	-
$d$ (mm)	-	83.24	276.76
$d_a$ (mm)	-	87.24	280.76
$d_b$ (mm)	-	78.22	260.07
$d_f$ (mm)	-	78.24	271.76
$m$ (mm)	2	-	-
$u$	3.32	-	-
$\alpha_t$ (°)	20.74	-	-
$\beta$ (°)	16.03	-	-

Table 4.1: Gear drive specifications. Material of choice is 45 steel

# References

- [1] Charles Mareau, Daniel Cuillerier, and Franck Morel. *Experimental and numerical study of the evolution of stored and dissipated energies in a medium carbon steel under cyclic loading C*. 2013.
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- [3] Chat Trinh and Uyen Van Le. *Thiet Ke He Dan Dong Co Khi*. 6th ed. Vol. 1. Vietnam Education Publishing House Limited, 2006.