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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

Nomenclature

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

1.1 Problem

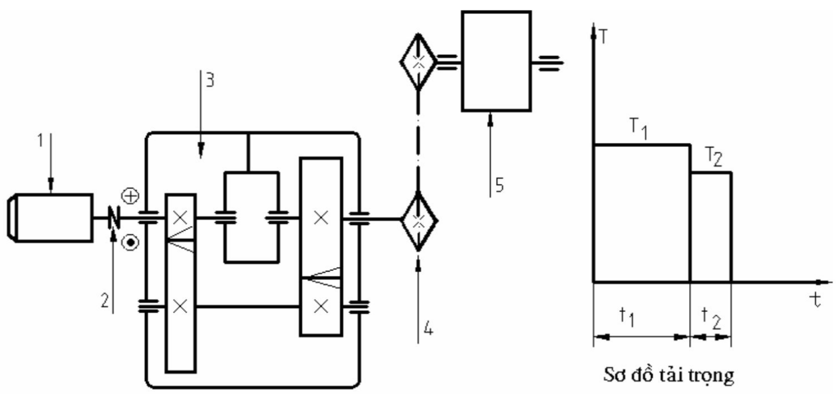

Figure 1.1: Working principle diagram and workload of the mixing machine: 1) electric motor, 2) elastic coupling, 3) two-stage coaxial helical speed reducer, 4) roller chain drive, 5) mixing tank (one-directional, light duty, operate 1 shift, 8 hours each)

1.2 Mixing machine parameters

From the parameters given in the document, we have:

$P = 7$ (kW)	$t_1 = 15$ (s)
$n = 65$ (rpm)	$t_2 = 11$ (s)
$L = 8$ (years)	$T_1 = T$ (N · m)
$K_{ng} = 260$ (days)	$T_2 = 0.7T$ (N · m)
$Ca = 1$ (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
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, $\mathrm{N \cdot mm}$
η_b	bearing efficiency
η_c	coupling efficiency
η_{ch}	chain drive efficiency
η_{hg}	helical gear efficiency
η_{sys}	efficiency of the system
$_1$	subscript for shaft 1
$_2$	subscript for shaft 2
$_3$	subscript for shaft 3

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

2.1 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 [1]:

- 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 [1]:

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

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

Calculate n_{mo} The purpose is to Using Table 2.4 [1]:

- 2-level transmission speed reducer, spur gear type. $u_h = 11$

- 1 chain drive, roller type. $u_{ch} = 2$

Multiplying all transmission ratio yields:
$u_{sys} = u_h u_{ch} = 11 \times 2 = 22$
$n_{mo} = u_{sys} n = 22 \times 65 = 1430$ (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 [1], we choose motor 4A132S4Y3 which operates at 7.5 kW maximum and 1455 rpm, which makes $n_{mo} = 1455$ rpm.
Recalculating u_{sys} with the new P_{mo} and n_{mo} derived from the chosen motor, we obtain:

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

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

$$u_{ch} = \frac{u_{sys}}{u_h} = \frac{22.38}{11} = 2.03$$

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.
$P_{ch} = \dfrac{P_w}{\eta_b} = \dfrac{6.2}{0.99} = 6.26$ (kW)
Shaft 3 power is affected by the chain drive.
$P_{sh3} = \dfrac{P_{ch}}{\eta_{ch}} = \dfrac{6.26}{0.96} = 6.52$ (kW)
Shaft 2 power is affected by the bearings and gear drives on shaft 3.
$P_{sh2} = \dfrac{P_{sh3}}{\eta_b \eta_{hg}} = \dfrac{6.52}{0.99 \times 0.97} = 6.79$ (kW)
Shaft 1 power is affected by the bearings and gear drives on shaft 2.
$P_{sh1} = \dfrac{P_{sh2}}{\eta_b \eta_{hg}} = \dfrac{6.79}{0.99 \times 0.97} = 7.07$ (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 [1]:

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

Then,
from motor to shaft 1: $n_{sh1} = n_{mo} = 1455$ (rpm)
from shaft 1 to shaft 2: $n_{sh2} = n_{sh1}/u_1 = 1455/3.32 = 438.70$ (rpm)
from shaft 2 to shaft 3: $n_{sh3} = n_{sh2}/u_2 = 438.70/3.32 = 132.27$ (rpm)

Torque Subsequently, the torque is calculated as follows:
$T_{mo} = 9.55 \times 10^6 \times P_{mo}/n_{mo} = 9.55 \times 10^6 \times 7.14/1455 = 46892.66$ (N · mm)
$T_{sh1} = 9.55 \times 10^6 \times P_{sh1}/n_{sh1} = 9.55 \times 10^6 \times 7.07/1455 = 46423.73$ (N · mm)
$T_{sh2} = 9.55 \times 10^6 \times P_{sh2}/n_{sh2} = 9.55 \times 10^6 \times 6.79/438.70 = 147857.49$ (N · mm)
$T_{sh3} = 9.55 \times 10^6 \times P_{sh3}/n_{sh3} = 9.55 \times 10^6 \times 6.52/132.27 = 470919.44$ (N · 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
$[s]$	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	σ_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.
$n_{sh3} = 132.27$ rpm
$P_{ch} = 6.26$ kW, $u_{ch} = 4.5$

3.2 Find the chain drive pitch

The driving sprocket is connected to shaft 3, $n_1 = n_{sh3} = 132.27$ rpm.

Calculate z The number of teeth determines the how stable is the rotational speed, which relates to the impact intensity and service life:
$z_1 = 29 - 2u_{ch} = 29 - 2 \times 2.03 = 25 \geq 19$ (rounded to the nearest odd number)
$z_2 = u_{ch} z_1 = 2.03 \times 25 = 51 \leq 120$ (Equation 5.1 [1])

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 [1]. In this problem, $z_1 = 25$; $n_1 = 132.27$ rpm, which is close to $n_{01} = 200$ 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 [1]:

$$P_t = P_{ch}k_0k_ak_{dc}k_dk_ck_{bt}k_zk_n$$
$$= 6.26 \times 1 \times 1 \times 1 \times 1.5 \times 1 \times 1.3 \times 1 \times 1.51 = 18.46\,(\text{kW}) \leq [P]$$

Inspecting Table 5.5 [1] at column $n_{01} = 200$ rpm, the closet value is $[P] = 19.3$ kW. Knowing $[P]$, we have $p = 31.75$ mm, Table 5.5 [1]. Consequently, $d_c = 9.55$ mm, $B = 27.46$ mm. Consulting Table 5.8 [1], 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 $a_{min} = 30p = 30 \times 31.75 = 952.50$ (mm), $a_{max} = 50p = 50 \times 31.75 = 1587.50$ (mm). Limiting the range of choice for a in $[a_{min}, a_{max}]$, we can approximate $a = 1300$ mm and find x_c:

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

Then, round x up to the nearest even number gives $x_c = 122$.

Find a Using x_c to find the correct center distance, Equation 5.13 [1]. 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 multiplied in the formula below:

$$a = \frac{0.997p}{4}\left[x_c - \frac{z_2 + z_1}{2} + \sqrt{\left(x_c - \frac{z_2 + z_1}{2}\right)^2 - 2\left(\frac{z_2 - z_1}{\pi}\right)^2}\right]$$
$$= \frac{0.997 \times 31.75}{4}\left[122 - \frac{51 + 25}{2} + \sqrt{\left(122 - \frac{51 + 25}{2}\right)^2 - 2\left(\frac{51 - 25}{\pi}\right)^2}\right] = 1019.99\,(\text{mm})$$

Other parameters The values below are necessary for modeling the chain drive:

$$d_1 = p/\sin\left(\frac{\pi}{z_1}\right) = 31.75/\sin\left(\frac{180}{25}\right) = 253.32\,(\text{mm})$$
$$d_2 = p/\sin\left(\frac{\pi}{z_2}\right) = 31.75/\sin\left(\frac{180}{51}\right) = 515.75\,(\text{mm})$$
$$d_{a1} = p\left(0.5 + \cot\frac{180}{z_1}\right) = 31.75\left(0.5 + \cot\frac{180}{25}\right) = 267.20\,(\text{mm})$$
$$d_{a2} = p\left(0.5 + \cot\frac{180}{z_2}\right) = 31.75\left(0.5 + \cot\frac{180}{51}\right) = 530.65\,(\text{mm})$$

Look up to find $d_l = 19.05$ (mm), see Table 5.2 [1]:

$d_{f1} = d_1 - 2(0.502d_l + 0.05) = 253.32 - 2(0.502 \times 19.05 + 0.05) = 234.08$ (mm)
$d_{f2} = d_2 - 2(0.502d_l + 0.05) = 515.75 - 2(0.502 \times 19.05 + 0.05) = 496.50$ (mm)

3.4 Strength of chain drive

3.4.1 Impact frequency analysis

After determine the center distance, it is necessary to validate the impact frequency. From Table 5.9 [1], it is $[i] = 25$. Calculating i gives:

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

3.4.2 Safety factor analysis

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]:
$F_t = 10^3 P_{ch}/v_1 = 2410.48$ (N)
$F_v = q v_1^2 = 17.54$ (N)
$F_0 = 9.81 \times 10^{-3} k_f q a = 156.11$ (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.

3.4.3 Contact stress analysis

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 $A = 180\,\text{mm}^2$, Table 5.12 [1].

Knowing k_d and F_t, we get the result:

$$\sigma_H = 591.29\ (\text{MPa}) \leq 600\,\text{MPa}$$

which is satisfactory.

3.5 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)

3.6 Other parameters

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

References

[1] Chat Trinh and Uyen Van Le. *Thiet Ke He Dan Dong Co Khi*. 6th ed. Vol. 1. Vietnam Education Publishing House Limited, 2006.