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**FACULITY OF MECHANICAL OF ENGINEERING**



**TRANSMISSION SYSTEM PROJECT**

**Project 3 – Data no 6**

**Semester 231**

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|  |  |  |
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Ho Chi Minh City, February 21st 2024

# INTRODUCTION

Our nation is progressing along the path of development, with science and technology playing a crucial role in enhancing human life. The increasing application of science and technology is not only boosting labor productivity but also efficiently replacing manual labor, ensuring the safety of workers. Mechanized systems prove to be effective substitutes for human effort in automating production processes, thereby elevating overall labor productivity. By exercising control over these systems, we contribute to the ongoing modernization of automation initiatives in Vietnam.

The Transmission System Project serves as a pivotal component for students specializing in Mechatronics and Mechanical Engineering, providing fundamental insights into the design of mechanical transmission systems. This project enables us to comprehend the intricacies of automated systems in factories, plants, and workshops. The project's scope encompasses the application of knowledge acquired from foundational subjects such as Kinematics and Dynamics of Machines, Machine Elements, and Mechanical Engineering Drawing. Implementation of the project significantly enhances proficiency in AutoCAD and SolidWorks. This newfound skill set, combined with specialized knowledge, allows us to engage with practical systems and develop a comprehensive understanding in preparation for future projects and graduation theses.

We extend our sincere gratitude to Dr. Le Thuy Anh for his dedicated guidance in completing the transmission system project. Additionally, we express our appreciation to the instructors of this semester, as their feedback during project review sessions has been instrumental in expanding our knowledge and refining our drawing skills.

Being our first mechanical system project, some shortcomings in calculations and component selection are inevitable. We look forward to further guidance from our teachers to consolidate our knowledge and gain valuable experiences for our future careers.

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# CHAPTER 1: MOTOR AND TRANSMISSION RATIO SELECTION

## General efficiency of transmission system

According to table 2.3 in reference we select the efficiency of each transmission system parts:

The efficiency of helical gear (x2):

The efficiency of V-belt (x1):

The efficiency of sliding-contact bearing (x1):

The efficiency of couplings (x1):

The efficiency of rolling-contact bearings (x2):

The general efficiency of transmission system:

## Calculate required power

Working power of system, based on formula (2.11):

Rotational speed of screw impeller, based on formula (2.16):

Required power of motor, based on formula (2.8):

in which is the preliminary power for the motor.

## Motor selection

According to the calculated required power, we choose the standard motor power as With that standard motor power, we have the following options to choose our motor as well as its rotational speed and transmission ratio.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | **Rotational speed rpm** | **General transmission u** | **V-belt ub** | **Spur gear ug** |
| Motor 1 | 2830 | 30.16 | 3.24 | 8 |
| Motor 2 | 1430 | 15.24 | 3.0 | 5 |
| **Motor 3** | **945** | **10.07** | **2.52** | **4** |
| Motor 4 | 715 | 7.62 | 1.52 | 3.15 |

Based on table 3.1 Motor SGA (Company CMG, Australia). We choose motor 3 with a rotational speed of 945 rpm (112M).

## Working power on each shaft

Required power of motor:

Working power on shaft II:

Working power on shaft I:

Working power of motor:

## Rotational speed on each shaft

Rotational speed of motor:

Rotational speed of shaft I:

Rotational speed of shaft II:

Rotational speed of spiral screw:

## Torque on each shaft

Torque of motor:

Torque of shaft I:

Torque of shaft II:

Torque of spiral screw:

## 1.7 Parameters of transmission system

Table 1‑3: Transmission ratio calculation and distribution

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| **Parameters** | **Shaft** | | | | | |
| Motor | I | | II | | Working |
| **Power P (kW)** |  |  | |  | |  |
| **Ratio u** |  | |  | |  | |
| **Rotational speed n (rpm)** |  |  | |  | |  |
| **Torque T (Nm)** |  |  | |  | | 180.3 |

Where,

Shaft motor: connecting the motor to the V-belt drive.

Shaft I: connecting the V-belt drive to the 1-stage helical spur gear reducer.

Shaft II: connecting the 1-stage helical spur gear reducer to the couplings.

Shaft working: connecting the couplings to the spiral screw.

# CHAPTER 2: DESIGN BELT CONVEYOR

## 2.1 Select the belt type

According to initial parameters ( and ) and figure 4.1a, page 59 in references [1] we choose belt type B.

According to table 4.5, page 76 references [1], we choose belt type B with:

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Type | A ( | (mm) | (mm) | (mm) | h (mm) | (mm) |
| B |  |  |  |  |  |  |

## Calculate the diameter of the small pulley

Calculate the diameter of the small pulley:

So, the standard value of is .

## Calculate the belt velocity

Velocity of driving pulley:

In which:

diameter of driving pulley

rotation of driving pulley

If the velocity bigger than , it can lead to lower efficiency, higher temperature, shorten the lifespan of the machine.

The belt velocity is (satisfied)

## Calculate the diameter of the big pulley

Choose the creep factor:

Diameter of the big pulley:

So, the standard value of is

The real transmission ratio of belt drive:

Error of transmission ratio:

If the error , the machine will not operate as well as we want.

## Calculate the center distance and the belt length L

Calculation for center distance:

According to table , page in references we can choose .

The belt length L:

According to the standard value, we choose the belt length .

## Check the number of turns in 1 second

Check for number of turns:

## Recalculate center distance

The center distance:

In which: k and can be determined by these formulas:

Thus:

The value of a is still within the acceptable range

## Calculate the contact angle

The contact angle:

The contact angle needs to bigger than to ensure efficiency, lifespan and avoid reducing transmission capacity as well as friction.

⟹ Satisfy the condition

## Calculate the number of belts

According to the table 4.19, page 62 in references [1], we select:

Contact angle correction factor:

Velocity correction factor:

Belt length correction factor and allowable effective power:

Transmission ratio correction factor according to Table 4.17 page 61 in reference:

Factor considering the influence of uneven load distribution between belts according to Table 4.18 page 61 in reference:

Factor considering the influence of the load conditions:

(working two shifts, static)

The number of belts:

We choose .

## Width and outer diameter d of pulley

According to table 4.21, page 63 in references , we have:

Width of pulley:

Outer diameter of smaller pulley:

Outer diameter of larger pulley:

## 2.11 Determine the force on each shaft

Initial tension force:

Tension force on each belt:

Circumferential force:

Circumferential force on each belt:

Shaft load:

Force on the tight side:

Force on the loose (slack) side:

Centrifugal tension according to formula page 64 in reference:

Based on the type B belt drive, we have

in which is the length density of belt

## Stress in belt drive and service life

Maximum belt stress in belt drive:

⇒ (satisfied)

If maximum belt stress exceeds the threshold of , the belt can become damaged and torn, shaft also can be misaligned.

Service life:

In which:

Fatigue strength of belt (V-belt)

Exponent of fatigue curve (V-belt)

## Summary

Table 2‑1: Parameter for V-belt drivers

|  |  |
| --- | --- |
| **Parameter** | **Value** |
| Belt type | B-type V-belt |
| Number of the belts z |  |
| Center distance (mm) |  |
| Belt length (mm) |  |
| Warp angle () |  |
| Pulley’s width B (mm) |  |
| The number of Belt's turn per second ( |  |
| Driving pulley’s diameter (mm) |  |
| Driven pulley’s diameter (mm) |  |
| Maximum belt stress |  |
| Initial tension (N) |  |
| Tangential force(N) |  |
| Shaft load (N) |  |
| The force on the tight side |  |
| The force on the loose (slack) side |  |
| Service life of belt (hours) |  |

# CHAPTER 3: DESIGN HELICAL GEAR DRIVE IN SINGLE STAGE REDUCER

## 3.1 Initial parameters

|  |  |
| --- | --- |
| Name | Value |
| Power transmitting, |  |
| Rotational speed, |  |
| Torque, |  |
| Transmission ratio, |  |

## Chose the material

According to table 6.1 in reference , we choose:

Driving gear: choose C45 steel with hardness , through induction hardening, structural improvement, .

To increase teeth wear, the hardness of driven gear needs to follow the formula:

Driven gear: choose C45 steel with hardness , through structural improvement,

## Determine all kinds of the stress on gears:

According to table 6.2\*, we have:

Limitation of contact stress :

Limitation off bending stress :

Safety coefficient of contact stress:

Safety coefficient of bending stress:

Coefficient considering the impact of loading:

(𝑜𝑛𝑒 − 𝑤𝑎𝑦 𝑡𝑟𝑎𝑛𝑠𝑚𝑖𝑠𝑠𝑡𝑖𝑜𝑛)

Number of cycles of change in contact stress :

Number of cycles of change in bending stress:

Working time:

The equivalent number of cycles if the gear is working with working load and number of revolution n is constant :

Because

Allowable contact stress and bending stress:

Because of helical gear, we calculate allowable stress by the equation below:

Checking with the condition: (satisfied)

Allowable contact stress when overloading:

Allowable bending stress when overloading:

## Calculating geometry parameters of helical gears system:

According to table 6.6\*, we have coefficient of gear width:

From here infer:

According to table 6.7\*, we have coefficient of unequal load distribution on the gear:

;

Calculation for the center distance of both gears:

We choose the standard center distance, (𝑆𝑡𝑎𝑛𝑑𝑎𝑟𝑑 𝑆𝐸𝑉229 − 75).

The module m can be calculated according to the center distance:

We choose

The condition of the tooth angle:

Based on equation (6.31)\*:

We choose the number of teeth on the driving gear:

The number of teeth on the driven gear:

Lead angle of helical gear :

Pitch circle diameter of the driving gear:

Pitch circle diameter of the driven gear:

Rolling circle diameter:

Outside diameter of driving gear:

Outside diameter of driven gear:

Root diameter of driving gear:

Root diameter of driven gear:

Driven gear width:

Driving gear width:

Pressure angle:

Pressure angle in helical gear:

## Peripheral speed v and the angle of helical gear

According to table 6.13, page 106 in reference , we choose the grade of accuracy is

## The value of the forces acting on gear mesh

Peripheral force Ft:

Axial force Fa:

Radial force Fr:

The normal force :

Figure 3.2 Forces acting on gears

## 3.7 Dynamic load factors

Based on the table 6.14, we choose level of accuracy is 9, so we have ;

Calculating the factor :

The factor of kinematic load appears on meshing zone:

We can calculate the total load factor when contacting and bending:

## Checking stress

Where:

According to formula 6.38b, page 105 in reference we have:

According to table 6.5, page 96 in reference , we have:

According to formula 6.39, page 106 in reference , we have:

As (), the gear has enough contact strength.

## Tooth shape factor

Equivalent gear teeth:

Tooth shape factor:

## 3.10 Check the bending strength

When:

Because so we choose driving gear to calculate:

As , the gear has enough bending strength.

## Summary

Table 3.1 Helical gear final specification

|  |  |  |  |
| --- | --- | --- | --- |
| **Parameter** | **Sign** | **Driving wheel** | **Driven wheel** |
| Center distance (mm) |  |  | |
| Module |  |  | |
| Number of teeth (teeth) |  |  |  |
| Peripheral speed (m/s) |  |  | |
| The pitch diameter (mm) |  |  |  |
| The pitch outside diameter (mm) |  |  |  |
| The root diameter (mm) |  |  |  |
| Lead angle of helical gear |  |  | |
| Peripheral force (N) |  |  | |
| Radial force (N) |  |  | |
| Axial force (N) |  |  | |
| Normal force (N) |  |  | |

# CHAPTER 4: DESIGN TWO SHAFTS IN THE SPEED REDUCER

## Choose material

Given that we are in the process of designing a gearbox, it is advisable to opt for C45 steel, characterized as a medium carbon steel. The selected heat treatment method is soft hardening. The benefit of selecting this material lies in its cost-effectiveness and the wide range of applications it offers.

According to table 6.1 in references , we choose:

Material: C45 Steel

Method: Normalization, structural improvement

Hardness: HB

Ultimate strength:

Yield limit:

## Preliminary selecting diameter of shaft

Preliminary choosing allowable torsional stress for shaft I and for shaft II because shaft ΙΙ suffer from bigger torsion than shaft Ι.

Preliminary diameter of shaft:

Shaft I:

According to standard, we select preliminary diameter of shaft I:

Shaft II:

According to standard, we select preliminary diameter of shaft II:

## Distance between bearings and force application point

**Shaft I:**

According to table 10.2 in reference we choose the width of bearing: mm

According to table 10.3 in reference we choose:

(mm): Distance from driving gear’s face to gearbox inner surface or between rotating elements.

(mm): Distance from bearing face to inner gearbox-wall.

(mm): Distance from element face to gearbox cover.

(mm): Length from bearing cover to bolt.

Mayo length of helical spur gear:

Mayo length of belt drive:

Because we choose so it has the length as the gear’s width.

Distance between elements on shaft:

**Shaft II:**

According to table 10.2 in reference [1], we choose the width of bearing:

According to table 10.3 in reference [1], we have:

mm: Distance from driving gear’s face to gearbox inner surface or between rotating elements

mm: Distance from bearing face to inner gearbox-wall.

mm: Distance from element face to gearbox cover.

mm: Length from bearing cover to bolt.

Mayo length of screw drive:

We chose

Mayo length of belt drive:

Because we choose so it has the length as the gear’s width Distance between elements on shaft:

We choose

## Calculate the reaction forces acting on shaft I:

Force acting from helical gear:

Peripheral force: (N)

Centrifugal force: (N)

Axial force: (N)

Moment caused by axial force:

The torque acting on the shaft:

According to free body diagram in figure, we have force equation of equilibrium and moment equation of equilibrium in plane:

According to free body diagram in figure, we have force equation of equilibrium and moment equation of equilibrium in plane:

)

## Diagram of bending moment and torque diagram of shaft I:



Figure 4.1 Bending moment and torque diagram of shaft I

According to moment distribution diagram, the most critical position is section C:

Equivalent moment at A, B, D:

According to table 10.5 in reference [1] with , we choose allowable bending stress

Shaft diameters are determined:

According to standard, we choose

We choose as it is the other bearing of this shaft, so they have to be the same size.

Because at position A there is a keyway, according to the standard we can choose

Because at position C there is a keyway, according to the standard we can choose

## Calculate the reaction forces acting on shaft II:

Force acting on shaft II:

Force acting from helical gear:

Tangential force:

Centrifugal force:

Axial force: N

Force acting on coupling:

The radial force acting on the shaft from the coupling:

We choose

Moment caused by axial force:

The torque acting on the shaft:

According to free body diagram in figure, we have force equation of equilibrium and moment equation of equilibrium in plane:

According to free body diagram in figure, we have force equation of equilibrium and moment equation of equilibrium in plane:

## Diagram of bending moment and torque diagram of shaft II:



Figure 4.2 Bending moment and torque diagram of shaft II

According to moment distribution diagram, the most critical position is section C:

Equivalent moment at A, B, D:

According to table 10.5 in reference with , we choose allowable bending stress

Shaft diameters are determined:

According to standard, we choose mm.

We choose as it is the other bearing of this shaft, so they have to be the same size.

Because at position C there is a keyway, according to the standard we can choose .

Because at position A there is a keyway, according to the standard we can choose .

## Choose the key for the shaft:

We choose the same material with the gear C45, heat treatment is normalization, structural improvement.

According to table 9.1a in reference we choose key length following the standard

, choose key parameter following largest cross-section on shaft.

Check the condition for the volume stress:

With:

: Volume stress

: Shear stress

𝑇: Torque on shaft at special position

𝑑: Diameter of the special position

, ℎ, : Parameter of the key

[: Allowable volume stress (according to table 9.5 in reference , the key is C45 and the system is static load)

: Allowable shear stress

Table 4.1 Selection of key dimensions

|  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **Shaft** | **Cross-section** | **d** |  |  |  |  |  |  | **(MPa)** |
| **I** | A |  |  |  |  |  |  |  |  |
| C |  |  |  |  |  |  |  |  |
| **II** | A |  |  |  |  |  |  |  |  |
| C |  |  |  |  |  |  |  |  |

All the feather keys are satisfied.

## 4.9 Checking for the integral gear shaft condition.

With:

: The distance between the root diameter and the top of the keyway.

: The module of the gear.

Checking for the shaft I:

The shaft I is the driving shaft with the integral gear, so we do not need the key at the gear position.

Checking for the shaft II:

At the gear in the reducer:

At the external gear:

The shaft II is the shaft without the integral gear.

## Checking the shaft according to the safety factor

The structure of the shaft has been designed to ensure durability if the safety factor at critical sections meets the following conditions:

With:

: Allowable safety factor

: Safety factor is only considered for bending and torsional stresses, determined by the following the formula:

Where:

: Ultimate bending stress corresponding to each symmetrical cycle for carbon steel

: Ultimate torsional stress corresponding to each symmetrical cycle for carbon steel

: According to table 10.12 in reference we have:

;

: According to table 10.7 in reference we have:

: According to table 10.10 in reference we have:

: Surface fatigue strength factor. According to table 10.9 in reference we choose:

: Due to the rotation shaft, the bending stress changes according to the symmetric cycle:

: When the shaft rotates in one direction, the torsional stress changes according to the cycle:

With:

Table 4.2 Parameters for strength check

|  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **Shaft** | **Cross-section** | **Key** |  |  |  |  |  |  |  |  |  |
| **I** | **A**  **(20)** |  | 642.47 | 1428.86 | 0.88 | 0.81 | 0 | 19 | - | 7.6 | - |
| **B**  **(30)** | - | 2650.72 | 5301.43 | 0.88 | 0.81 | 13.4 | 4.6 | 20.2 | 31.6 | 17 |
| **C**  **(40)** |  | 6114.44 | 11647.62 | 0.88 | 0.81 | 10.9 | 2.1 | 24.8 | 69.2 | 23 |
| **D**  **(30)** | - | 2650.72 | 5301.43 | 0.88 | 0.81 | 0 | 4.6 | - | 31.6 | - |
| **II** | **A**  **(32)** |  | 2025.72 | 4780.6 | 0.85 | 0.78 | 0 | 19.6 | - | 7.2 | - |
| **B**  **(40)** | - | 6283.19 | 12566.37 | 0.85 | 0.78 | 0 | 7.5 | - | 18.7 | - |
| **C**  **(50)** |  | 10375.28 | 23189 | 0.85 | 0.78 | 5.6 | 4 | 46.4 | 25 | 22 |
| **D**  **(40)** | - | 6283.19 | 12566.37 | 0.85 | 0.78 | 4 | 7.5 | 65.3 | 18.7 | 18 |

The shafts satisfy fatigue conditions.

**Test for static durability**

To prevent large elasticity deformation on shafts or break down when being subjected to sudden load (such as when starting the motor). Testing each shaft static durability at the section where the largest moment is located, the result must satisfy the equation below:

Where:

**Shaft I:**

The most critical position is section C:

The shaft satisfies the static durability test.

**Shaft II:**

The most critical position is section B:

The shaft satisfies the static durability test.

# CHAPTER 5: CALCULATE AND DESIGN THE BEARING

## Calculate the reaction force acting on the bearing in shaft I

Calculate the reaction force acting on the bearing in shaft I

Centrifugal force at B:

Centrifugal force at D:

Due to so we choose

This satisfies the condition that Angular contact ball bearing is suggested:

Select Angular contact ball bearing.

According to **Table P2.12\*** in reference we have information about the bearing:

Table 5.1 Parameter of bearing on shaft I

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
|  |  | **)** |  |  |  |  |
|  |  |  |  |  |  |  |

## Choose accuracy grade for the bearing in shaft I

Choose grade 0 as it both is cheap and suitable with the system.

## Calculate the basic dynamic load in shaft I

According to table 11.4 in reference with:

In which:

: Coefficient of which ring rotate

We calculate the load at A with the formula:

In which:

: Coefficient of load specificity (based on table 11.3 in reference [1] for static load)

: Coefficient of temperature

Longevity of bearing in million rounds:

The dynamic load sustainability:

In which:

: Grade of bending line for ball bearing

The condition is satisfied.

Re-calculate the longevity of bearing:

Service life of the bearing:

## Test for static load in shaft I

Where:

: Static notational load.

: Allowable static notational load.

: Coefficient of centripetal load (based on table 11.6 in reference )

: Coefficient of axial load (based on table 11.6 in reference )

Because , condition is satisfied.

## Limited number of revolutions of bearing in shaft I

Limited number of revolutions of bearing:

Where: : Center diameter of rollers:

: Number of revolutions

Based on Table 11.7 in reference [1] we have:

With angular contact bearing lubricated by grease:

Because , condition is satisfied.

## Calculate the reaction force acting on the bearing in shaft II

Centrifugal force at A:

Centrifugal force at B:

Due to so we choose

This satisfies the condition that deep grove ball bearing is suggested:

Select deep grove ball bearing.

According to Table P2.7\* in reference we have:

Table 5.2 Parameter of bearing on shaft II

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
|  |  | **)** |  |  |  |  |
|  |  |  |  |  |  |  |

## Choose accuracy grade for the bearing in shaft II

Choose grade 0 as it both is cheap and suitable with the system.

## Calculate the basic dynamic load in shaft II

According to table 11.4 in reference with

In which:

: Coefficient of which ring rotate

We calculate the load at A with the formula:

In which:

: Coefficient of load specificity (based on table in reference for static load)

: Coefficient of temperature

Longevity of bearing in million rounds:

The dynamic load sustainability:

In which:

: Grade of bending line for ball bearing

The condition is satisfied.

Re-calculate the longevity of bearing:

Service life of the bearing:

## Test for static load in shaft II

Because

Where:

: Static notational load.

: Allowable static notational load.

: Coefficient of centripetal load (based on table 9.6 in reference)

: Coefficient of axial load (based on table 9.6 in reference )

Because , condition is satisfied.

## Limited number of revolutions of bearing in shaft II

Limited number of revolutions of bearing:

Where: : Center diameter of rollers:

: Number of revolutions

Based on Table 11.7 in reference we have:

With angular contact bearing lubricated by grease.

Because , condition is satisfied.

## Summary

Table 5.3 Bearings final specification

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| **Shaft** |  |  | **)** |  |  |  |  |
| **I** |  |  |  |  |  |  |  |
| **II** |  |  |  |  |  |  |  |

# CHAPTER 6: CALCULATE AND DESIGN THE COUPLING

## 6.1 Initial parameters

Torque acting on shaft I:

The rotational speed of the chain coupling:

To decrease the impact of shaft, we select elastic coupling.

We have information about elastic coupling based on table 16-10a\*:

Table 6.1 Coupling specification

|  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **[T] (Nm)** | **(rpm)** | **(mm)** | **(mm)** | **l (mm)** | **D (mm)** | **(mm)** | **L (mm)** |  | **Z** |
|  |  |  | 65 | 110 |  | 105 |  | 56 | 6 |

Table 6.2 Dimensions of the elastic ring

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **[T] (Nm)** | **(rpm)** | **(mm)** | **(mm)** | **(mm)** | **(mm)** | **(mm)** | **(mm)** | **(mm)** |
|  |  |  |  |  |  |  |  | 1.5 |

According to table 16-1\*, we select the coefficient of working mode:

(the machine is belt conveyor)

We can calculate the moment torque of coupling:

## Calculate and check the safety factor

CHAPTER 7: DESIGNING GEARBOX AND OTHER AUXILIARY ELEMENTS

7.1 Initial parameters

The center distance between shafts:

The length of shaft I and shaft II: and

* + 1. **Choose the material for gearbox cover**

The gear housing is crucial in gearbox design, ensuring the machine's components stay secure, distributing loads evenly, and safeguarding against dust.

The critical standard for the gearbox is to have high stiffness and low weight, so cast iron GX15-32 is used as material. To facilitate installation, cap and housing unit are cast separately.

* + 1. **Calculation and designing gearbox cover**

Table 7.1 Table of parameters and calculation of gearbox cover

|  |  |  |
| --- | --- | --- |
|  | **Parameters** | **Calculation** |
| **Width** | Gearbox body, (mm) |  |
| Gearbox cover, (mm) |  |
| **Rib** | Width, (mm) | Choose |
| Height, (mm) |  |
| Slope | Approximately |
| **Diameters** | Based bolt, |  |
| Bearing side bolt, |  |
| Body-cover-assembly bolt, |  |
| Screw for oil caps, |  |
| Screw for top oil cap, |  |
| **The part between the cover and body**  **(mm)** | Body’s top thickness, |  |
| Cover’s top thickness, |  |
| Widths (mm) |  |
|  |
|  |
|  |
| **The bearing seat areas** | Bolt center’s diameter of shafts, (mm) | Shaft I: |
| Shaft II: |
| Outer diameter of shafts, (mm) | Shaft I: |
| Shaft II: |
| Diameter of screws, | Shaft I and II: |
| Height, (mm) | Shaft I and II: |
| Distance from bolt’s center to edge of hole, (mm) |  |
| **The gearbox body’s bottom**  **(mm)** | Thickness, |  |
| Width, , |  |
|  |
| **Gap between elements**  **(mm)** | Between gears and gearbox inner wall |  |
| Between bigger gear and gearbox body’s bottom |  |
| Between gear faces |  |
|  | Number of based bolts |  |

7.2. Other auxiliary machine elements

* + 1. **Ring bolt and hook**

To lift and move the gearbox (when manufacturing, assembling…), we usually add ring that C20 steel is chosen as material on its body and lid.

According to reference , the center distance of our gearbox is so the weight selected is , we choose eye bolt M8, C-shaped hook with lifting weight of .

Table 7.2 Ring bolt and hook parameter

|  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **Thread d** |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| **M8** |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |

* + 1. **Dowel pin**

To ensure the relative position of the body and the lid, we use conical dowel pins.

Here, we use the bevel locating bolt because it ensures good positioning due to no gaps, while the cylindrical pin easy to machine but there will be a gap between the pin and the box wall after several times using. According to table 18.4b in the reference , the parameters for the pin:

* + 1. **Gearbox cap**

To seal the bearing, prevent dust, and secure the outer ring of the bearing on the housing. We use 2 types of caps, which are:

Solid gearbox Cap: a cover that does not have a hole or an opening for a shaft to pass through.

Shaft - passing Gearbox Cap: a cover designed with a hole or opening that allows a shaft to pass through.

The bearing cap is molded from gray cast iron GX15-32. From table 18-2 [1]:

Table 7.3 Gearbox cap parameter

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| **Gearbox cap** |  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |  |

* + 1. **Access door**

We make an access door on top of the housing, on which there is an air vent, to check the inside the gearbox and for pouring oil. The parameters based on table in reference :

Table 7.4 parameter of access door

|  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  |  |  |  |  |  |  |  |  |  |
| 150 | 100 | 190 | 140 | 175 | - | 120 | 12 | M8 x 22 | 4 |

* + 1. **Oil drain plug**

After a period of working, the oil will be contaminated (because of dust and other factors), we need to replace the oil. To drain the contaminated oil, an oil discharge is designed at the bottom of the gearbox.

According to table 18.7 in the reference [2], the parameters for the plug are:

Table 7.6 Oil drain plug parameter

|  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **d** | **b** | **m** | **f** | **L** | **c** | **q** | **D** | **S** |  |
|  |  |  |  |  |  |  |  |  |  |

* + 1. **Oil dipstick**

To check the oil level, we use the level indicator. To avoid interference from the oil wave, the indicator has a cover.

Table 7.7 Oil level indicator parameter

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  |  |  |  |  |  |  |  |  |
|  | 5 | 6 | 18 | 12 | 30 | 12 | 6 | 3 |

**CHAPTER 8. THE TOLERANCE**

8.1. Bearing’s tolerance

The inner ring of the bearing experiences cyclic loads, so it is chosen to be assembled using an intermediate shaft system to ensure that the inner ring does not slip on the shaft surface during operation. Hence, we must opt for a k6 fit class to ensure even bearing wear during rotation.

The outer ring of the roller bearing does not rotate and thus experiences localized loads. It is assembled using a hole system. To allow the bearing to move axially as the temperature increases during operation, we choose an intermediate fit type of H7.

8.2. Assembly elements

**8.2.1 Assembly of the gears and the shafts.**

For the gears mounted on the shaft, the fitting type chosen is H7/k6.

**8.2.2 Assembly of the bearing cap and the housing body.**

To facilitate disassembly and adjustment, we choose a loose fit type of H7/h6.

**8.2.3 Assembling of the oil shield on the shaft**

To facilitate disassembly, we choose an intermediate fit type of H7/k6.

**8.2.4 Assembly of the Positioning Pins**

To ensure concentricity and prevent loosening, we choose a fit type of H7/p6.

**8.2.5 Assembly of the key**

In the axial direction, the fit type on the shaft is P9, and the fit type on the bearing is D10.

**8.2.6 Assembly of Positioning Pins**

To ensure concentricity and prevent loosening, we choose a fit type of H7/p6.

**8.2.7 Assembly of Threads**

In the axial direction, the fit type on the shaft is P9, and the fit type on the bearing is D10.

8.3. Summary of assembly tolerance

Table 7‑1: Assembly tolerance

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| **Machine elements** | **Assembly** | **ES (µm)** | **es (µm)** | **EI (µm)** | **ei (µm)** | **(µm)** | **(µm)** |
| **Gear in shaft II** | ∅50H7/k6 | +30 | +21 | 0 | +2 | 21 | 28 |
| **Grease seal on shaft I** | ∅35H7/js6 | +21 | +8 | 0 | -8 | 8 | 29 |
| **Grease seal on shaft II** | ∅45H7/js6 | +25 | +8 | 0 | -8 | 8 | 33 |
| **Spacer on shaft I** | ∅25H8/js6 | +33 | +6.5 | 0 | -6.5 | 6.5 | 39.5 |
| **Spacer on shaft II** | ∅35H8/js6 | +39 | +8 | 0 | -8 | 8 | 47 |
| **V-belt drive in shaft I** | ∅20k6 | - | +15 | - | +2 | - | - |
| **Coupling in shaft II** | ∅32k6 | - | +18 | - | +2 | - | - |

Table 7‑2: Tolerance for Angular contact ball bearing

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Inner ring** | | | | | |
| **Element** | **Assembly** | **ES** | **es** | **EI** | **ei** |
| **Shaft I** | ∅30k6 | - | +15 | - | +2 |
| **Shaft II** | ∅40k6 | - | +18 | - | +2 |
| **Outer ring** | | | | | |
| **Element** | **Assembly** | **ES** | **es** | **EI** | **ei** |
| **Shaft I** | ∅72H7 | +30 | - | +0 | - |
| **Shaft II** | ∅90H7 | +30 | - | +0 | - |

Table 7‑3: Tolerance for Thread Assembly

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Thread Cross-Sectional Dimension bxh** | **Allowance for the Width of the Thread Groove on the Shaft** | **Allowance for the Width of the Thread Groove on the Bearing** | **Allowance Limit on the Shaft,** | **Allowance Limit on the Bearing,** |
|  | P9 | D10 |  |  |
| **8x7** | -0.015 | +0.098  +0.040 | +0.2 | +0.1 |
| **12x8** | -0.018 | +0,120  +0,050 | +0.2 | +0.2 |
| **14x9** | -0.061 | +0,120  +0,050 | +0.2 | +0.2 |
| **18x11** | -0.061 | +0,120  +0,050 | +0.2 | +0.2 |

**CHAPTER 9. DESIGN OF TRANSMISSION SYSTEM**

* 1. Extrusion machine

Extrusion machine uses compression extrusion technology to create plastic pipes, plastic sheets, plastic floors, plastic films, etc. The main function of the machine is to make a molten plastic mixture in a brief time to shape products. After that, the plastic will immediately cool down and maintain its solid shape according to the product extrusion mold.

Extrusion machines play a key role in the plastic production line, specifically as follows:

* As equipment used to shape and produce plastic products, the adjusting machine is the most important in a complete plastic production line.
* Can manage a variety of thermoplastics such as PP, PE, ABS, PVC, PS, PET, ...
* Quality extrusion machines also help the plastic to be shaped quickly, creating beautiful, uniform finished products.

Regardless of the application, a gear driver drives the extrusion machine. In the extrusion machine, there will be a shaft connected to the output of the gearbox. When the motor rotates, this shaft will rotate, and the mixing tank system will also rotate to perform its function.

The transmission systems used in extrusion machines are truly diverse, such as: worm gear drive, coaxial gear drive with spur gears, helical gears, bevel gears, ...

In the scope of this project, the transmission is designated as a 1-stage helical spur gear reducer, and the output shaft is connected to a V – belt drive before reaching the working shaft.

A machine with blue and yellow parts

Description automatically generated with medium confidence

Figure 8‑1: Extrusion machine

![A large blue machine with a rope attached to it

Description automatically generated]()

Figure 8‑2: Holton Crest Continuous Rotary Extrusion Machine

* 1. Spiral screw

Diagram of a machine with text

Description automatically generated with medium confidence

*Figure 8‑3: Single screw extruder*

Single screw extrusion is one of the main types of polymer processing operation. The principal function of a single screw extruder is to build up pressure in the polymer melt so that the polymer can be extruded through the die. Most of the single screw extruders are plasticating which mean that the solid resin balls or powders melt in the screw due to the pressure. However, some single screw system can be used for mixing also. Single screw extruder is useful when pure polymer like HDPE is used.

Single screw extruder consists of a screw, barrel, drive mechanism, resin feed arrangement and controls. The constantly turning screw moves the resin through the heated barrel where it is heated to proper temperature and blended into a homogeneous melt. A turbulent back pressure is built up which pushes the melt out of the extruder in the shape of the die. The resin sometimes is not completely melted in the basic extrusion screw. The barrier screw in designed to counter this problem. Sometimes, additional flights are attached to the transition section to separate molten and solid plastic to different channels. As the solid pellet moves forward it melts due to shear against the wall and thus melts and flows into the liquid channel. Thus, the solid channel narrows gradually and the liquid channel widens.

The ability of a screw to manufacture products of superior quality with high productivity and low cost is called its performance. At the design stage of an extrusion Single Screw Extruder process, evaluation of screw performance is especially important. The deformation measure or stretching which materials undergo due to the regular flows inside a conventional screw channel increases linearly with the extruder channel length.

* 1. **Design for spiral screw**

The pitch of the screw blade not only determines the rising angle of the screw, but also determines the sliding surface of the material under a certain filling coefficient, so the pitch directly affects the conveying process of the material.

For solid blade, D is the diameter of blade, the pitch of screw blade can be determined:

A diagram of a function

Description automatically generated

Figure 8‑4: Structural parameters of Screw blade diameter and pitch

# REFERENCES

Trịnh Chất, Lê Văn Uyển (2007)*, Tính toán thiết kế hệ dẫn động cơ khí, Tập 1&2*, NXB Giáo dục, Hà Nội.