**VIET NAM NATIONAL UNIVERSITY HO CHI MINH CITY**

**HO CHI MINH CITY UNIVERSITY OF TECHNOLOGY**

**FACULTY OF MECHANICAL ENGINEERING**

**DEPARTMENT OF MACHINE DESIGN**

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**TRANSMISSION SYSTEM PROJECT**

**Topic:**

**Designing mixing drum transmission system**

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Student: Phan Đăng Khôi Nguyên – 1952366

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**TRANSMISSION SYSTEM PROJECT**

**(ME3145)**

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Student: Phan Đăng Khôi Nguyên Student ID: 1952366

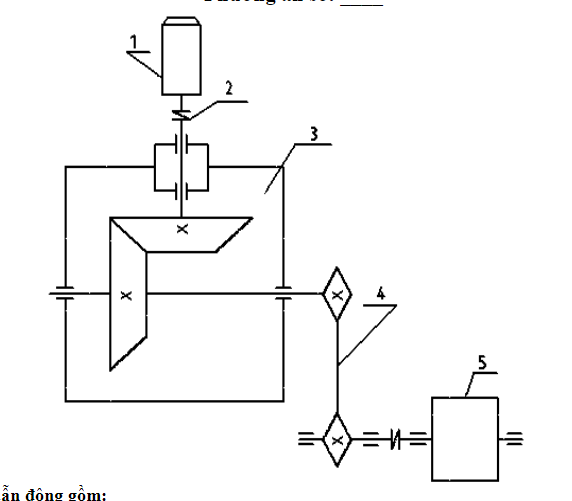
Instructor: Ph.D Nguyễn Tấn Tiến Sign:

Start date: 01/09/2021 End date: Defend date:

**Topic:**

**Designing mixing drum transmission system**

**Plan 9**

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**The transmission system includes:**

1. Motor
2. Flexible coupling
3. Single stage bevel gear reducer
4. Roller chain sprocket system
5. Mixing drum

**Design information:**

* Power of working shaft, P (kW): 2.5
* Number of revolutions per minute of working shaft, n (rpm): 160
* Service life, Lh (hours): 10000
* One – way rotation, works 2 shifts

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***PART 1: MOTOR SELECTION AND TRANSMISSION RATIO***

Power necessary for motor:

Wherein:

* ηsys : general efficiency of the system.
* ηbg : efficiency of bevel gears.
* ηrb : efficiency of roller bearings.
* ηc : efficiency of couplings.
* ηrc : efficiency of roller chain sys.

*(Assuming lubricated working environment, calculated with smallest efficiency)*

General system transmission ratio is:

Wherein:

* usys : transmission ratio of system.
* ubg : transmission ratio of bevel gear single stage reducer (in range 2÷4).
* urc : transmission ratio of roller chain system (in range 2÷5).

We choose: and

Based on table **P1.3** in reference **[1]**, we choose motors with so as to satisfy the condition

**Table 1.1 Motors and Transmision ratios**

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Motor type | Power (*kW*) | Rotational speed (*rpm*) |  |  |  |  |
| 4A100S4Y3 | 3.0 | 1420 | 0.83 | 82 | 2.2 | 2.0 |

Transmission ratio of reducer and roller chain system:

Working power on each shaft:

Rotational speed on each shaft:

Torque on each shaft:

**Table 1.2 Technical specifications of transmission system table**

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **Shaft**  **Parameters** | Motor | Shaft 1 | | Shaft 2 | | Shaft 3 | | Working shaft |
| Transmission ratio | 1 | | 2.958 | | 3 | | 1 | |
| Rotational speed (*rpm*) | 1420 | 1420 | | 480.054 | | 160.018 | | 160 |
| Power (*kW*) | 2.941 | 2.913 | | 2.712 | | 2.551 | | 2.5 |
| Torque (*Nmm*) |  |  | |  | |  | | 149218.750 |

***PART 2: DESIGNING CHAIN DRIVE***

**2.1 Initial Information**

|  |  |
| --- | --- |
| Transmission Power (*kW*) | 2.551 |
| Rotational speed (*rpm*) | 160.018 |
| Transmission Ratio | 3 |

**2.2 Chain Sprocket Calculations**

Calculate number of teeth for driving sprocket gear:

Calculate number of teeth for driven sprocket gear:

(so that )

Calculating the power:

With:

: gear tooth factor.

: rotational factor.

Calculating operational factor :

Wherein:

* Factor taking into account the position of the system (line connecting two centers of the gear comparing to horizontal line, up to ).
* Factor taking into account shaft distance and chain length (shaft distance ).
* Factor taking into account adjustments of the tensile force on chain .
* Factor taking into account load conditions (static load).
* Factor taking into account working conditions (working 2 shifts).
* Factor taking into account lubrication

Calculating power:

Based on table **5.5** in reference **[1]**, choose:

Chain pitch

Shackle stud diameter

Tube length

Mean chain velocity:

Calculating shaft distance:

Number of chain link:

Choose

Recalculating shaft distance:

To reduce the tensile force on chain, the shaft distance is reduced by

Choose

Checking times of impact on chain hinge in 1 second:

với (Corresponding to chain pitch 19.05)

35

**2.3 Durability Check**

Safety factor:

Wherein:

* Destroying load (based on table **5.2** in reference **[1]**).
* Dynamic load factor (average working conditions).
* Useful peripheral force: .
* Tensile force caused by centrifugal force: (: weight of 1 meter chain, based on table **5.2** in reference **[1]**).
* Tensile force caused by passive chain link
* Allowable safety factor (based on table **5.10** in reference **[1]**).

We have:

Conclusion: chain drive system has passed the durability check.

**2.4 Durability of Contact Check**

Wherein:

* Impact force on chain:
* Factor taking into account uneven load: (1 chain)
* Factor taking into account dynamic load: (tải trọng tĩnh)
* Factor taking into account number of teeth on sprocket gears:
* Elastic modulus:
* Area of hinge: (based on table **5.12** in reference **[1]**).

We have:

Choose material gray cast iron C4 28-48 (chilled).

**2.5 Chain Drive Design Summary**

Effective diameter of sprocket gear:

With

Force acting on shaft:

With for all chain drive systems up to .

***PART 3: DESIGNING BEVEL GEAR DRIVE***

**3.1 Initial Information**

|  |  |
| --- | --- |
| Torque on driving shaft 1 (*Nmm*) | 19590.951 |
| Rotational speed of driving gear (*rpm*) | 1420 |
| Rotational speed of driven gear (*rpm*) | 480.054 |
| Transmission Ratio | 2.958 |

**3.2 Reducer Calculations**

1. Choice of material

Since there is no special requirement and reducer works in normal conditions, we choose material for the bevel gear according to table **6.1** in reference **[1]**.

* For the driving gear, we will pick refined C45 steel with stiffness as our material.
* For the driven gear, we will use refined C40 steel with Brinell hardness of since it satisfies the following condition .

We are able to extract information about the fatigue contact limits and fatigue stress limit for driving gear:

and for driven gear:

1. Determining allowable contact stress and allowable bending stress for preliminary design

We first determine the number of cycles corresponding to the change of the stress-cycle diagram and , also called base number of loading cycles on the driving and driven gears:

Then we proceed to calculate the cyclic service time of the two gears:

Wherein

* is the number of tooth matings of the wheel per revolution.
* and : each gear’s respective rotational speed.
* Lh: lifetime action period in hours given.

Because , we will take

From table **6.2** in reference **[1],** we extract factors of safety and . With all the calculated values, we can determine the allowable contact and bending stresses:

When overloaded:

For well-lubricated, encased gear drive such as this one, we will do design calculation according to contact stress so as to prevent potential pitting.

1. Selecting allowable contact stress

Since we are designing a bevel gear reducer, allowable contact stress is given by:

When overloaded:

1. Pitch cone length

(For straight bevel gearing made of steel )

Based on table **6.21** in reference **[1]** and

Gear shaft is installed on roller bearings, so that we have:

Torque acting on driving gear shaft:

1. Gear mating parameters

Preliminary reference diameter for driving gear:

Based on table **6.22** in reference **[1]**, we choose

Since the stiffness of material used to make gears < 350 HB:

Tooth reference angle:

Number of teeth on equivalent cylindrical gear:

Mean diameter:

Mean modulus:

Modulus of bevel gear drive:

1. Recalculating parameters

Choose according to table **6.8** in reference **[1]**, we recalculate:

Therefore:

Number of teeth on driving gear:

Recalculating transmission ratio:

Recalculating tooth reference angle:

Mean diameter:

1. Bevel gear drive geometry parameters summary:

|  |  |  |
| --- | --- | --- |
| **Parameter** | **Driving gear** | **Driven gear** |
| Pitch cone length |  | |
| Width of face |  | |
| Outside ring modulus |  | |
| Transmission ratio |  | |
| Reference diameter |  |  |
| Tooth reference angle |  |  |
| Tooth height |  | |
| Tip tooth height |  |  |
| Root tooth height |  | |
| Outside tip tooth diameter |  |  |
| Mean velocity |  | |

1. Contact durability check

Contact durability must satisfies:

Based on table **6.12** in reference **[1]**, we have:

Based on table **6.5** in reference **[1]**, we have:

Horizontal mating factor:

Load factor:

Wherein:

Based on table **6.13** in reference **[1]**, using precision level 8.

Based on table **6.15** and **6.16** in reference **[1]**:

Tooth width

Therefore

Since the difference is negligible, we can adjust width of face:

We have:

Conclusion: bevel gear drive has passed the contact durability check.

1. Bending durability check

Bending durability must satisfies:

Load factor:

Wherein:

Based on table **6.21** in reference **[1]** and , we have

Based on table **6.15** in reference **[1],**

Based on table **6.16** in reference **[1],**

Therefore:

We have:

Equivalent number of teeth:

Factor of tooth type:

Therefore:

Conclusion: bending stress condition is satisfied, we can adjust the width of tooth to 30 *mm.*

1. Calculate force acting on gear shaft

Centripetal force:

Centrifugal force:

Axial force:

*For the bevel gear drive, we don’t have to do overload check since the system is under static load. Rotational speed is constant and the probability of overloading is low.*

*Therefore, in the scope of this project, we don’t take into account the overload conditions of bevel gear drive.*

***PART 4: DESIGNING SHAFTS AND KEYS***

**4.1 Choose material**

We use refined C45 steel as material.

* Stiffness: 200HB
* Tensile strength:
* Yield strength:
* Allowable stress:

**4.2 Preliminary shaft design**

Torque parameters:

|  |  |
| --- | --- |
| **Torque** | **Magnitude (*Nmm*)** |
|  |  |
|  |  |
|  |  |

With this information we proceed to calculate each of the shafts’ preliminary diameter:

Choose

Choose

Choose

**4.3 Shaft Length Calculations**

Based on table **10.3** in reference **[1]**, we choose lengths:

* Distance from rotating element to wall:
* Distance from bearing to wall
* Distance from rotating element to cover plate
* Height of cover plate plus screw:
* Bearing width, chosen according to preliminary diameter (table **10.2** in reference **[1]**)

Parameters of coupling are based on table **16-10a** in reference **[2]**

**4.3.1 Shaft 1**

Length of coupling:

Length of driving gear hub:

Width of bearing:

Distance between two bearings:

Length of shaft 1:

**4.3.2 Shaft 2**

Length of driven gear hub:

Length of chain sprocket hub:

Width of bearing:

Diameter of tip ring of driven gear:

Length of shaft 2:

**4.4 Force caused by Coupling and Chain Sprocket Calculations**

**4.4.1 Force caused by Flexible Coupling**

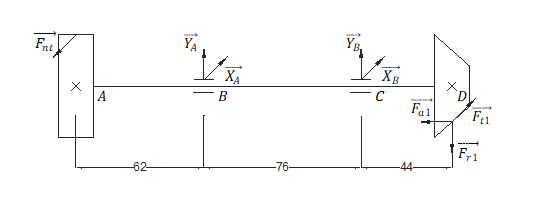
Force caused by flexible coupling is centrifugal. Direction of this force has the tendency to increase stress and deformation caused by centripetal force of driving bevel gear acting on shaft.

**4.4.2 Force caused by Chain Sprocket**

Force acting on shaft caused by chain drive system is tensile force towards center, with direction from driving chain sprocket to driven chain sprocket.

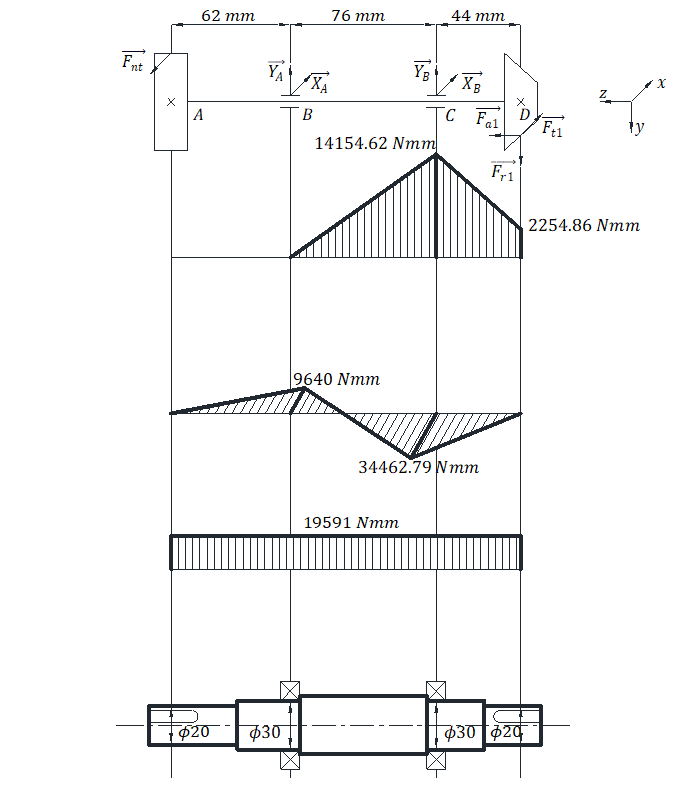
**4.5 Shaft Calculations**

**4.5.1 Shaft 1**

****

Moment caused by axial force:

Force equations:

****

*Figure 4.5.1 Bending Moment and Torque diagram for shaft 1*

Allowable stress

Equivalent moment at cross section j:

Shaft diameter at cross section j:

Calculating shaft diameter at cross section j:

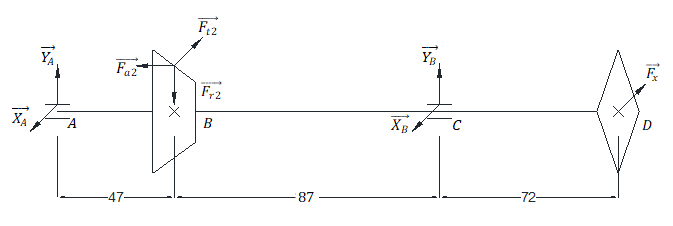
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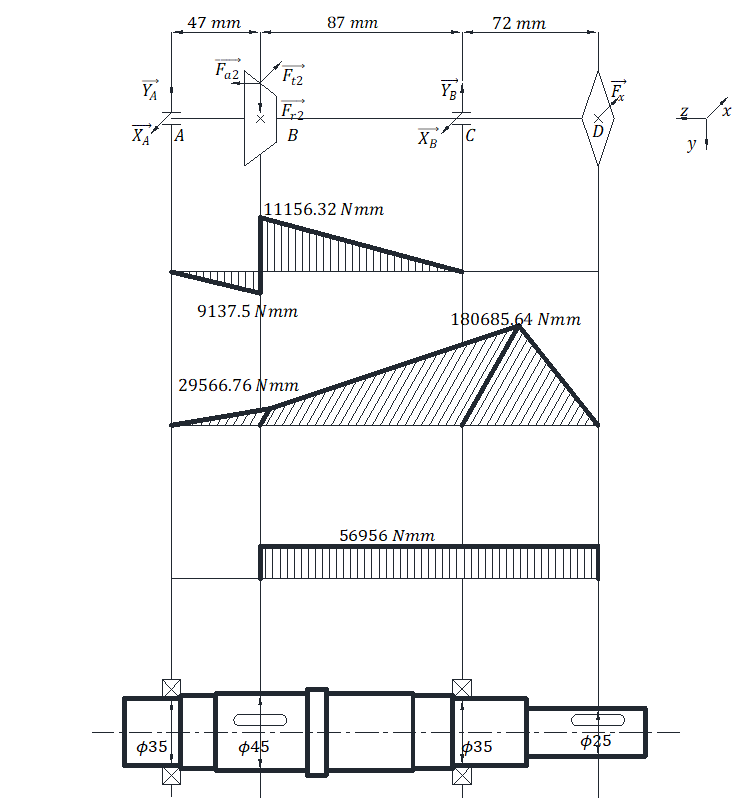
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**4.5.2 Shaft 2**

****

Moment caused by axial force:

Force equations:



*Figure 4.5.2 Bending Moment and Torque diagram for shaft 2*

Allowable stress

Choose

Choose

Choose

**4.6 Fatigue Strength Check**

We will conduct check on safety factor:

Wherein:

is the value for allowable safety factor, choosing it to be 3 so that we don’t need to check on shaft stiffness.

On the other hand, we have:

Fatigue strength of Cacbon steel (material of shaft) is:

Torsional strength is:

Since shaft rotates, bending stress changes in symmetrical cycle:

Mean value of normal stress at cross section j:

Magnitude of normal stress at cross section j:

Where is bending moment

is induced bending moment, according to table **10.6** in reference **[1]**, for shaft with one keyway:

The system is designed for one-way rotation:

Mean value of stress at cross section j:

Magnitude of normal stress at cross section j:

Where is torsional moment at cross section j

is torsional moment, according to table **10.6** in reference **[1]**, for shaft with one keyway:

Factors take into account mean stress value on fatigue strength, based on table **10.7** in reference **[1]**, we have:

Factors are calculated by formuli **10.25** and **10.26** in reference **[1]**:

Factor taking into concentrated stress caused by surface condition since shaft is manufactured by lathe machine with roughness factor , correlates to strength factor

Factor taking into incremental strength , shaft surface is case-hardened.

Dimension parameters: according to table **10.10** in reference **[1]**

Parameteres according to table **10.12** in reference **[1]**, correlates to keyway produced by milling cutter:

We can summarize parameters of fatigue strength check on shafts:

|  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| Shaft | Position (cross section) | Key |  |  |  |  |  |  |  |  |  |
| **1** | A (20) |  | 655.35 | 1440.75 | 0.92 | 0.89 | 0 | 6.8 | x | 24.93 | x |
| D (20) |  | 655.35 | 1440.75 | 0.92 | 0.89 | 3.44 | 6.8 | 77.06 | 24.93 | 23.72 |
| B, C (30) | x | 1533.98 | 3067.96 | 0.9 | 0.85 | 23.33 | 3.19 | 11.13 | 30.82 | 10.47 |
| **2** | A (35) | x | 4209.24 | 8418.49 | 0.865 | 0.795 | 0 | 0 | x | x | x |
| C (35) | x | 4209.24 | 8418.49 | 0.865 | 0.795 | 42.93 | 6.77 | 5.82 | 22.44 | 5.63 |
| B (45) |  | 3566.39 | 7775.63 | 0.865 | 0.795 | 8.86 | 3.66 | 28.19 | 41.52 | 23.32 |
| D (25) |  | 1133.98 | 2667.96 | 0.9 | 0.85 | 0 | 10.67 | x | 15.19 | x |

Conclusion: all safety factors on table is larger than . Therefore all shafts have passed fatigue strength check.

**4.7 Static Strength Check**

Wherein:

Shaft 1:

Shaft 2:

Conclusion: all shafts have passed static strength check.

**4.8 Key Check**

Parameteres of keys are based on table **9.1a** in reference **[1]**.

Conditions of impact and shearing strength:

Wherein:

* : Torsional moment on shaft (*Nmm*).
* : diameter of shaft at cross section with key.
* : length of key.
* : height of key.
* : depth of keyway

We have key checking table:

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Shaft | Diameter d |  |  |  |  |
| **1** | 20 |  | 35 | 32.65 | 16.33 |
| 20 |  | 20 | 32.65 | 16.33 |
| **2** | 45 |  | 30 | 30.14 | 9.04 |
| 25 |  | 35 | 30.38 | 9.11 |

Conclusion: all stress values on key satisfy.

***PART 5: BEARING AND COUPLING SELECTION***

**5.1 Bearing Selection**

**5.1.1 On Shaft 1**

Working time:

Inner diameter of bearing:

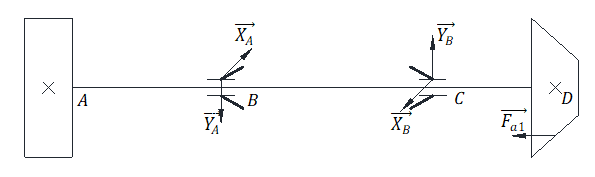
Since driving bevel gear is installed on input shaft, axial force acting on shaft is . Therefore, we choose roller bearing on input shaft, distributed in round shape.

Choose bearing with respect to dynamic load:

Based on table **P 2.11** in reference **[1]**

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
| Designation |  |  |  |  |  |  |  |  |
| 7206 | 30 | 62 | 16 | 17.25 | 1.5 | 13.67 | 29.8 | 22.3 |

Force distribution:



Centrifugal force at roller bearings:

Where:

Axial force at B:

Since

Axial force at C:

Since

We have:

Based on table **11.4** in reference **[1]**:

Convention load on bearing:

Wherein:

rotating inner ring

factor taking into account heat ()

applied to minimal impact conditions, short-term overload according to table **11.3** in reference **[1]**

Since we recalculate with respect to parameters at C.

Life of bearing in million revolutions:

Static load rating:

using roller bearing

Real life cycle of bearing:

For roller bearing, we have:

Based on formula **11.19** in reference **[1]**

Conclusion: bearing satisfies static load ratings.

**5.1.2 On Shaft 2**

Working time:

Inner diameter of bearing:

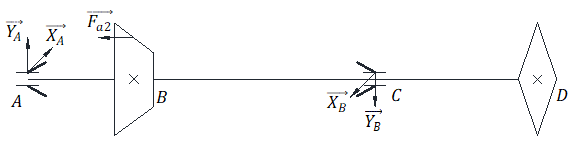
Since driving bevel gear is installed on input shaft, axial force acting on shaft is . Therefore, we choose roller bearing on input shaft, distributed in round shape.

Choose bearing with respect to dynamic load:

Based on table **P 2.11** in reference **[1]**

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
| Designation |  |  |  |  |  |  |  |  |
| 7207 | 35 | 72 | 17 | 18.25 | 2.0 | 13.83 | 35.2 | 26.3 |

Force distribution:



Centrifugal force at roller bearings:

Where:

Axial force at A:

Since

Axial force at C:

Since

We have:

Based on table **11.4** in reference **[1]**:

Convention load on bearing:

Wherein:

rotating inner ring

factor taking into account heat ()

applied to minimal impact conditions, short-term overload according to table **11.3** in reference **[1]**

Since we recalculate with respect to parameters at C.

Life of bearing in million revolutions:

Static load rating:

using roller bearing

Real life cycle of bearing:

For roller bearing, we have:

Based on formula **11.19** in reference **[1]**

Conclusion: bearing satisfies static load ratings.

**5.2 Coupling Selection**

We will use flexible coupling since its advantages are simple mechanism, easy to manufacture, easy to replace and reliability.

Torsional moment at input shaft:

Based on table **16.10a** in reference **[2]**, we have parameters for coupling:

|  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  |  |  |  |  |  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |  |  |  |  |  |
| 250 | 20 | 125 | 65 | 168 | 83 | 40 | 90 | 4 | 65000 | 14 | 75 |

Checking impact strength:

Where , allowable impact stress of rubber

: factor taking into account working condition, choose 1.35 for mixing drum system

Conclusion: shaft has passed impact strength check.

Checking strength of pin:

With , allowable stress of pin

We have:

Conclusion: pin has passed strength check.

***PART 6: REDUCER CASE DESIGN***

Reducer case takes the responsibility of maintaining relative position between components, receiving load from elements installed on the outside, containing lubrication oil and covering inside components.

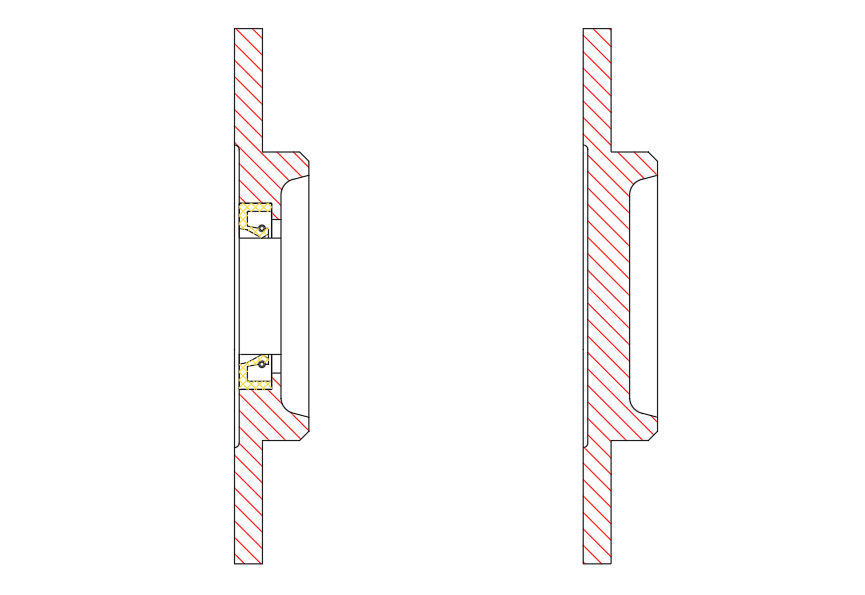
The material used to produce this case is cast iron GX15-32.

The reducer’s basic dimensions are given in the table below:

|  |  |
| --- | --- |
| **Parameters meaning** | **Parameters** |
| Thickness:  Case body  Case top |  |
| Stiffening rib:  Thickness |  |
| Diameter:  Foundation bolt  Bearing side bolt  Connecting top and body bolt  Cover plate screw  Observation cap screw |  |
| Top and body flange:  Body thickness  Top thickness  Width of top and body |  |
| Width of flange at bearing side  Center of bolt at bearing side và  Distance from bolt center to hole |  |
| Bottom flange:  Thickness  Width |  |
| Cleft between elements:  Gear and wall  Larger gear and bottom wall |  |
| Number of foundation bolt |  |

**Table 6.1 Reducer’s basic dimensions**

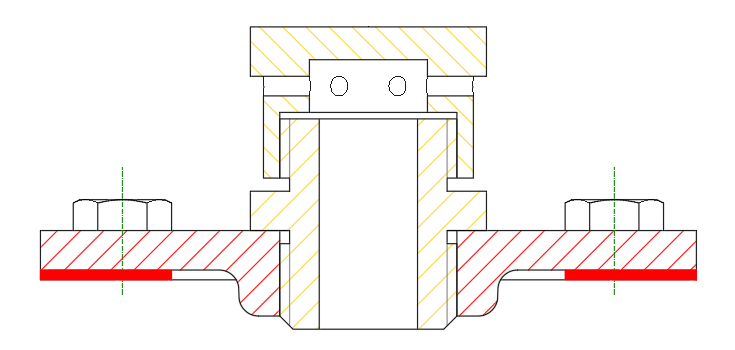
There are a total of 3 bearing cover plates, all of which are made of cast iron GX15-32. In figure 6.1 below, the cover plate on the left is a through hole cover plate which has an oil seal attached to prevent outside contaminants such as dust from coming in and inside lubricant from leaking out. The one on the right is a blind hole cover plate. Basic dimensions of these plates are given in table 6.2.

*Figure 6.1 Bearing Cover Plates (through hole – left; blind hole – right)*

**Table 6.2 Bearing Cover Plates**

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| **Shaft** |  |  |  |  |  |  |  |
| 1 | 74 | 90 | 115 | 65 | 10 | M8 | 4 |
| 2 | 74 | 90 | 115 | 65 | 10 | M8 | 4 |

Observation cap is a plate/lid on top of the reducer through which can inspect the inside of the machine. In this model, the ventilation plug is mounted onto the observation cap. Figure 6.2 below illustrates a set of observation cap – ventilation plug and table 6.3 – 6.4 give dimensions of these parts. The cap is made of cast iron GX15-32 (shown in red hatch) while the vent plug is made of steel CT3 (shown in yellow hatch).



*Figure 6.2 Set of observation cap – ventilation plug*

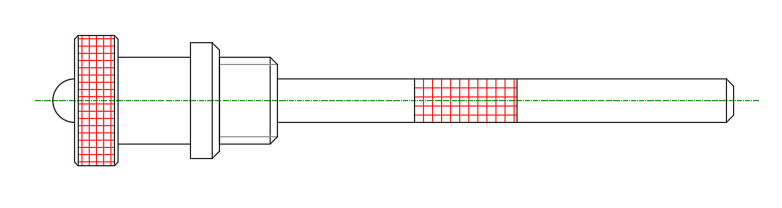
**Table 6.3 Observation Cap**

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| A1 | B1 | C | K | R | Screw | Quantity |
| 100 | 80 | 75 | 30 | 3 | M8x12 | 4 |

**Table 6.4 Ventilation Plug**

|  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| A | B | C | D | E | G | H | J |  | K | L | M | N | O | Q | P | R |
| M272 | 15 | 30 | 15 | 45 | 36 | 32 | 6 | 4 | 10 | 8 | 22 | 6 | 32 | 18 | 36 | 32 |

The oil dipstick is used to check the oil level inside the reducer. The highest oil level and the lowest oil level are 14 (*mm*) apart. Note that the oil level does not go above the center of the roller/ball of the ball bearing; consequently, we install oil splasher to help splash the lubricant on to the working gears.

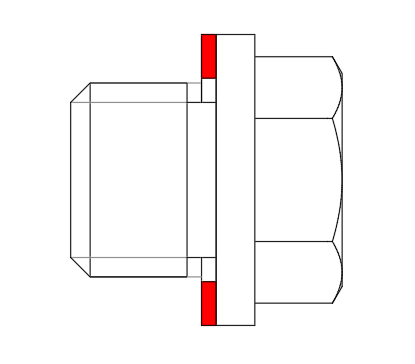


*Figure 6.3 Oil Dipstick*

**Table 6.5 Oil Dipstick**

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  |  |  |  |  |  |  |  |  |
| M121.25 | 5 | 6 | 18 | 12 | 30 | 12 | 6 | 3 |

The drain plug empties out the oil bath. Its dimension is given in table 6.6

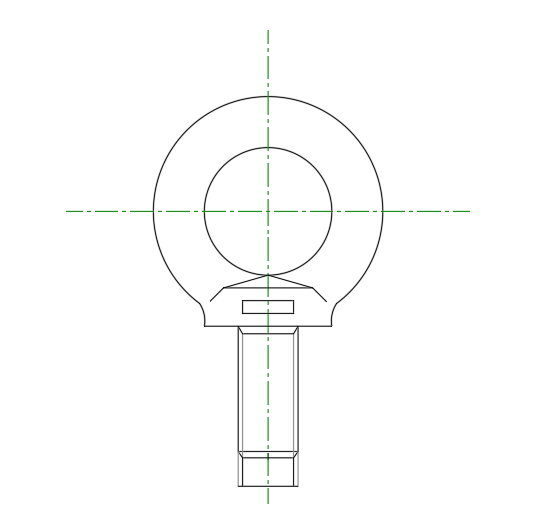


*Figure 6.4 Drain Plug*

**Table 6.6 Drain Plug**

|  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  |  |  |  |  |  |  |  |  |  |  |
| M201.5 | 25 | 25.4 | 30 | 15 | 4 | 22 | 2.5 | 20 | 32 | 3 |

Eye bolt for lifting and transportation

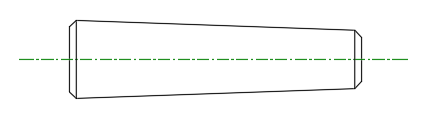


*Figure 6.5 Eye Bolt*

**Table 6.7 Eye Bolt**

|  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  |  |  |  |  |  |  |  |  |  |
| M8 | 36 | 20 | 8 | 20 | 10 | 12 | 6 | 18 | 20 |

Alignment pins are used to ensure relative positions of top and body cases so as to reduce the chance of deformation at the outside ring of bearings



*Figure 6.6 Alignment Pin*

**Table 6.8 Alignment Pin**

|  |  |  |
| --- | --- | --- |
|  |  |  |
| 6 | 1 | 30 |

***PART 7: TOLERANCES***

All fits on the two shafts are hole – basis. In a hole-basis system, the size of the hole remains constant and the diameter of the shaft is varied to determine the fit; conversely, in a shaft-basis system the size of shaft remains constant and the hole diameter is varied to determine the fit. The ISO system uses an alpha-numeric code to illustrate the tolerance ranges for the fit, with the upper-case representing the hole tolerance and lower-case representing the shaft.

**Table 7.1 Tolerances in Gear Assembly**

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| Parts | Code allocated with fit |  |  |  |  | Max excess | Max clearance |
| Driving gear |  |  |  |  |  |  |  |
| Driven gear |  |  |  |  |  |  |  |

**Table 7.2 Tolerances in Bearing Assembly**

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Code allocated with fit |  |  |  |  | Max excess | Max clearance |
|  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |
|  |  |  |  |  |  |  |

**Table 7.3 Tolerances in Key Assembly**

|  |  |  |
| --- | --- | --- |
| Key dimensions | Keyway width P9 | Depth of keyway |
|  |  |  |
|  |  |  |
|  |  |  |

***CONCLUSION:***

Through the time of doing this project on designing a mechanical transmission system, I now have a better understanding of how to analyze the problem of designing.

Since the characteristics of this project is to calculate the transmission system, it has encouraged students to be more observant and apply previous theoretical knowledge to find optimal solution for the design.

Although I have finished this project on my own in due time, with thorough guidance and helps of **Mr. Nguyễn Tấn Tiến** from the mechanical engineering faculty, there are undoubtedly still many shortcomings and mistakes due to my naivety and lack of practical knowledge. Therefore, I’m looking forward to your corrections and comments so that I can gain experience and add them to my knowledge.

I would like to sincerely thank **Mr. Nguyễn Tấn Tiến** for your help and dedicated guidance.

Student:



Phan Đăng Khôi Nguyên

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