School of Science, Engineering and Technology

MIET2510 - MECHANICAL DESIGN

Design Project

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Created Date: December 11th, 2023.

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1. Executive Summary

This document concentrates on calculating, designing, drawing, and selecting the components for a wind turbine. For the shaft design, the main shaft is determined to have a diameter of $30\ (mm)$ while the high-speed shaft diameter will be $10\ (mm)$, the length of both of which is assumed to be $0.3\ (mm)$ and their material is decided to be an AISI 1045 Cold-Drawn Steel. Regarding the bearing selection, a tapered roller bearing with a 30-mm bore and a dynamic loading rating of $21357\ (N)$ will be brought into the implementation. With an account for the gear design, the spur type is selected, and the four gears put into the gearbox have the teeth number of 90, 10, 50, and 25 with their respective pitch diameters of 135, 15, 100, and 50. The last component will be the disc brake, whose diameters are $15\ (mm)$ as the inner radius and $40\ (mm)$ as the outside radius.

2. Introduction

As the world is running short on fossil fuel consumption, other forms of energy, many of which are renewable, begin to emerge as potential candidates for future replacement. Those energy types are gradually proving their efficiency and usefulness in various aspects of human societies, such as transportation or farming. Among those types, wind energy has already appeared in many energy solutions as it is clean and renewable. The work of converting this type of energy into electricity is often performed by a wind turbine, which leads the wind kinetic energy through some mechanical components to end up in the generator to produce the electrical energy for usage [1]. To scientifically confirm that the turbine system operates expectedly and exhibits a long operation life, a standard formal design procedure must be carried out. Therefore, this report will guide the reader through calculating and choosing each component in the wind turbine in terms of sizes, materials, and standards… to ensure that the final product meets the client's expectations.

The sample design of the final product can be viewed in Figure 1 below [2].

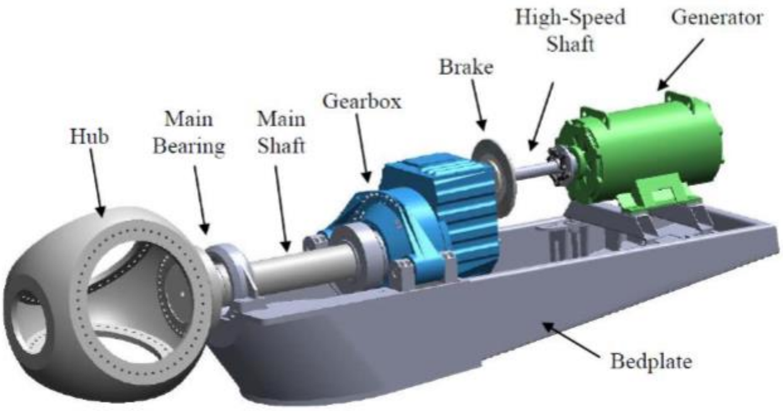

Figure 1. Sample design of the wind turbine [2]

3. Design Specification

- The weight and size of the design should be as optimal as possible.
- The material is not restricted to one specific type, which means the designer may choose the desired type based on careful consideration.
- The gearbox must have a ratio of 18:1, whose type and arrangement are determined by the designer.
- A proper alignment must exist between the main shaft and the high-speed shaft. The shaft length must be smaller than 0.5 metres. The designer is not restricted in any specifications to select the bearing and key for the shaft.
- The turbine blades are predicted to rotate between 350 RPM and 500 RPM under the nominal condition, from which the main shaft experiences the largest torque of 100 Nm, the 1500 N radial loading, and the 800 N axial loading.
- The wind turbine is expected to last eight years of continuous operation.
- All mechanical components are calculated and selected with a reliability of 98%.
- The safety factor of 2 is applied for all the parts.
- The user can assume some factors and coefficients in light of uncertainties.

4. Design Primary Parameters

Figure 2 below depicts the main components of the wind turbine system that will be discussed thoroughly throughout this report and their essential design parameters that need to be calculated and carefully considered.

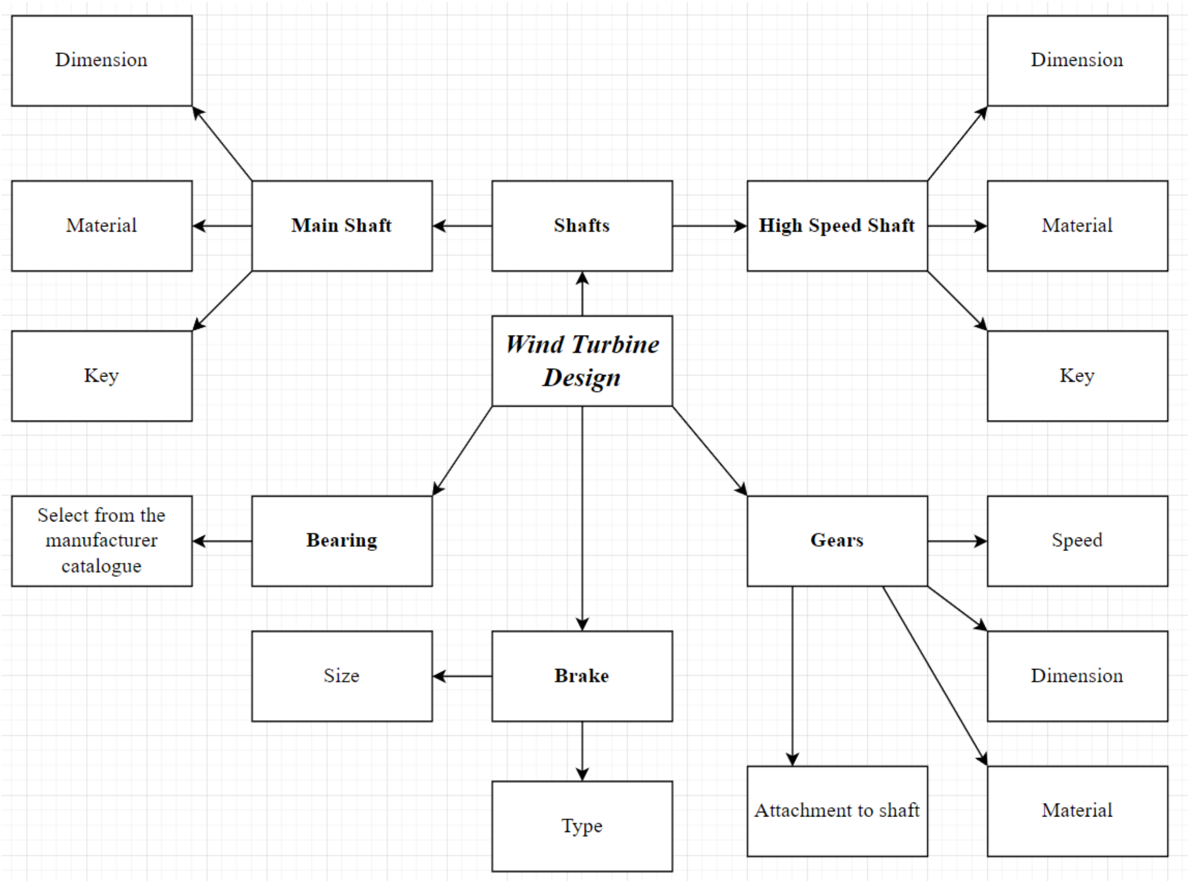

Figure 2. Main components and their design parameters of the turbine system diagram

5. Shafts Design

5.1. Design Assumption

- **Shaft Length:** the design requirement only restricts the length of the shaft not to be longer than 0.5 meters, so that the length of 0.3 meters is used for design and calculation.
- **Shaft Material:** the turbine shaft is commonly made of a material that has high strength, durability, and corrosion resistance, which is why steel is one of the most popular choices for this component [3][4]. This report will use AISI 1045 Cold-Drawn Steel as the material for the shaft since it is also regarded as the common type of steel used for this application. The main properties of AISI 1045 Cold-Drawn Steel are in Figure 3 below [5]:

Name:	AISI 1045 Steel, cold drawn
Model type	Linear Elastic Isotropic
Yield strength	5.3e+08 N/m^2
Tensile strength	6.25e+08 N/m^2
Elastic modulus	2.05e+11 N/m^2
Poisson's ratio	0.29
Mass density	7850 kg/m^3
Shear modulus	8e+10 N/m^2
Thermal expansion coefficient	1.2e-05 /Kelvin

Figure 3. The material properties of AISI 1045 Cold-Drawn Steel [5]

- **Shaft Key:** is determined to be a rectangular sunk key due to the ease of installation and reasonable cost. The key material will be the same as the shaft. Since the key belongs to the parallel type, the fit between the key and the hub can be clearance fit [6].

5.2. Theoretical Background

For this analysis, the bearing reaction forces acting on the shaft are assumed to be negligible. Since the main shaft will experience torsion, bending moment, and axial loading. The primary formulae used for designing the shafts are [7]:

- For shaft under bending moment, torsion, and axial loading
  - Distortion-Energy Theory (DET):

$$\frac{4}{\pi d^3}\sqrt{(8M+Pd)^2+48T^2}=\frac{S_y}{n_s} \quad (5.1)$$

  - Maximum-Shear-Stress Theory (MSST):

$$\frac{4}{\pi d^3}\sqrt{(8M+Pd)^2+64T^2}=\frac{S_y}{n_s} \quad (5.2)$$

- For shaft under bending moment, and torsion
  - Distortion-Energy Theory (DET):

$$d=\left(\frac{32n_s}{\pi S_y}\sqrt{M^2+\frac{3}{4}T^2}\right)^{\frac{1}{3}} \quad (5.3)$$

  - Maximum-Shear-Stress Theory (MSST):

$$d=\left(\frac{32n_s}{\pi S_y}\sqrt{M^2+T^2}\right)^{\frac{1}{3}} \quad (5.4)$$

where:

$$S_y: Yield\ Strength\ of\ the\ material;\ n_s: Safety\ Factor;\ d: Shaft\ Diameter$$

$M: Maximum\ Bending\ Moment$; P: Axial Loading; T: Maximum Torque

- Shaft Key Stress [7]:

$$T = Fr \ (5.3)$$

where:

$$T: Carried\ Torque;\ F: Tangential\ Force\ at\ the\ shaft\ surface; r: shaft\ radius$$

- Rectangular Key Length [7]:

$$L = \frac{2Fn}{S_y w} \ (5.4)$$

where:

$$F: Tangential\ Force\ at\ the\ shaft\ surface; S_y: Yield\ Strength\ of\ the\ material$$

$$n: Safety\ Factor; S_y: Width\ of\ the\ square\ key$$

- Rayleigh Equation [7]:

$$\omega_{cr} = \sqrt{\frac{g \sum_{i=1,\dots,n} W_i \delta_{i,max}}{\sum_{i=1,\dots,n} W_i (\delta_{i,max})^2}} \ (5.5)$$

where:

$$W_i: the\ i^{th}\ weight\ placed\ on\ the\ shaft; \delta: deflection$$

- Dunkerley Equation [7]:

$$\frac{1}{{\omega_{cr}}^2} = \frac{1}{{\omega_1}^2} + \frac{1}{{\omega_2}^2} + \cdots + \frac{1}{{\omega_n}^2} \ (5.6)$$

where:

$$\omega_1: critical\ speed\ if\ only\ mass\ 1\ exists$$

$$\omega_2: critical\ speed\ if\ only\ mass\ 2\ exists$$

$$\omega_n: critical\ speed\ if\ only\ the\ nth\ mass\ exists = \sqrt{\frac{g}{\delta_n}} \ (5.7)$$

5.3. Main Shaft Design Calculation

Figure 4 below illustrates the main shaft's free-body diagram, shear and moment diagrams, and torque diagram.

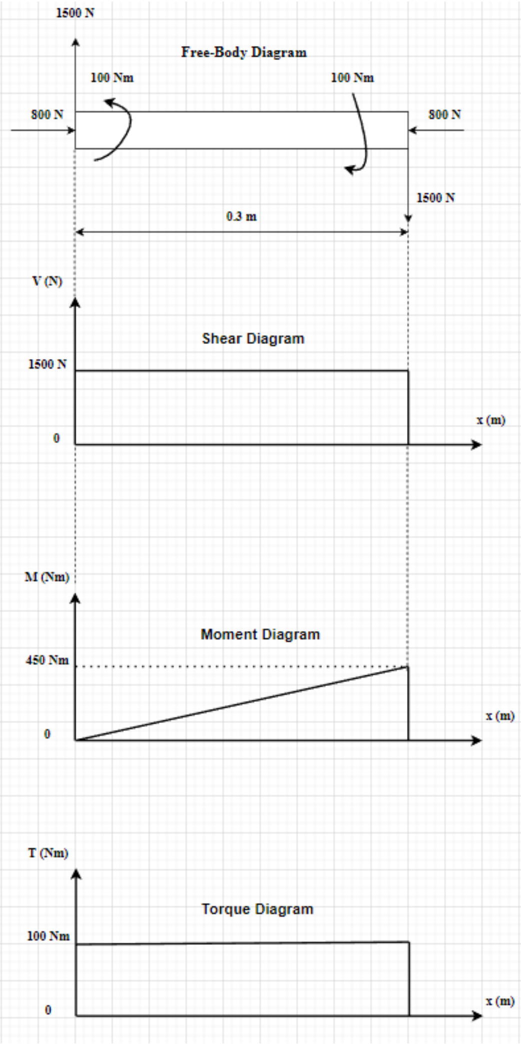

Figure 4. Free-body, Shear, Moment, and Torque Diagrams of the main shaft

From those diagrams and previous sections, we can deduce that:

- $M = 450$ (Nm)
- $T = 100$ (Nm)
- $S_y = 530$ (MPa)
- $n_s = 2$

Plugging the known values into the equation 5.1, we have:

$$\frac{4}{\pi d^3}\sqrt{(8\times 450+800\times d)^2+48(100)^2}=\frac{530\times 10^6}{2}\Rightarrow d\approx 26\ (mm)$$

Plugging the known values into the equation 5.2, we have:

$$\frac{4}{\pi d^3}\sqrt{(8\times 450+800\times d)^2+64(100)^2}=\frac{530\times 10^6}{2}\Rightarrow d\approx 27\ (mm)$$

The largest diameter out of those two results is 27 mm, but the chosen value for shaft design is rounded to 30 mm for convenience and following the shaft's typical dimension.

To calculate the shaft critical speed, the free-body diagram has to be modified to an equivalent version (shown in Figure 5 below) to facilitate the deflection calculation. Since, each tip of the shaft all bears the axial, radial, and moment loadings, one tip will be transformed into a fixed support while the other end will retain all the loading's information. This modification will transform an equilibrium-balanced shaft into a typical cantilever beam but not changing the working principles.

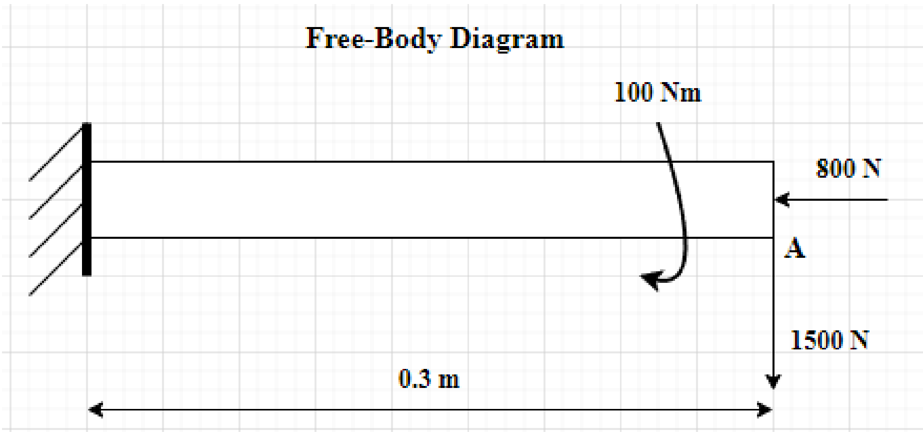

Figure 5. Modified Free-Body Diagram of the main shaft.

The deflection and moment of inertia formulae table is included in Appendix A. By applying those equations, some conclusions, based on the assumption that the gear weight is negligible compared to the magnitude of the radial load, can be reached:

$$\delta_A=\frac{PL^3}{3EI}=\frac{(1500)(0.3)^3}{3(2.05\times 10^{11})\frac{\pi}{4}(0.03)^4}\approx 0.0001(m)$$

Using the Rayleigh Equation (5.5):

$$\omega_{cr}=\sqrt{\frac{gW_A\delta_{A,max}}{W_A(\delta_{A,max})^2}}=\sqrt{\frac{(9.81)(1500)(0.0001)}{(1500)(0.0001)^2}}\approx 313.21\left(\frac{rad}{s}\right)=29519.35(rpm)$$

Applying the Dunkerley Equation with (5.6) and (5.7):

$$\omega_A = \sqrt{\frac{g}{\delta_A}} = \sqrt{\frac{9.81}{0.0001}} = 313.21\left(\frac{rad}{s}\right)$$

$$\frac{1}{{\omega_{cr}}^2} = \frac{1}{{\omega_A}^2} \Rightarrow \omega_{cr} = \omega_A = 313.21\left(\frac{rad}{s}\right) = 29519.35(rpm)$$

After drawing on two different formulae to find the critical speed, it can be observed that the lower and upper limits of this design parameter stay the same, whose value is $29519.35(rpm)$. As stated in the problem statement, the shaft is assumed to operate at $350\ (rpm)\ to\ 500\ (rpm)$, whose condition is well below where the critical speed draw the boundaries. Therefore, it can be concluded that the working conditions provided in section 3 are feasible and not causing any noticeable problems.

5.4. High-Speed Shaft Design Calculation

The free-body and torque diagrams of the high-speed shaft are shown in Figure 6 below. In general, it only experiences torque 18 times less than the main shaft due to the gearbox transmission ratio of 18:1.

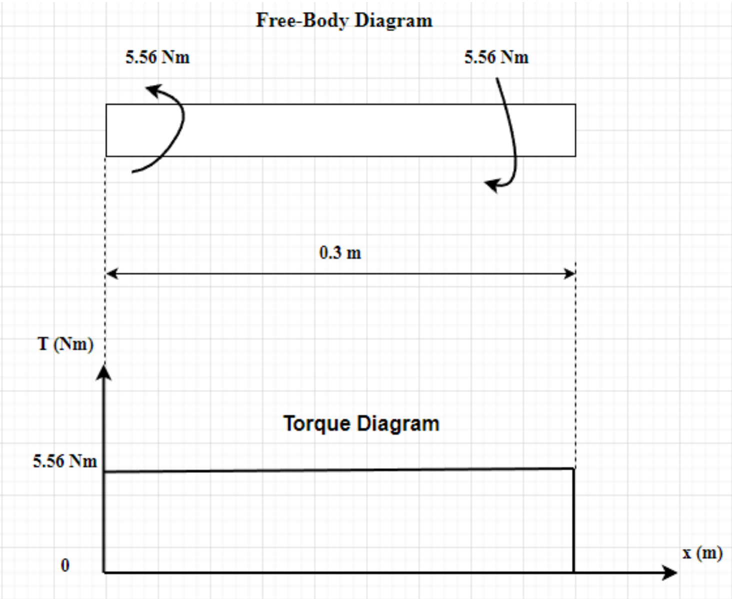

Figure 6. Free-body and torque diagrams of the high-speed shaft

Since the high-speed shaft does not experience any axial load, the formulae used for determining its diameter will differ from those for the main shaft.

Drawing on the equation 5.3 will lead to:

$$d = \left(\frac{32(2)}{\pi(530 \times 10^6)} \sqrt{0^2 + \frac{3}{4}(5.56)^2} \right)^{\frac{1}{3}} \approx 5.7\ (mm)$$

Applying the equation 5.4 will lead to:

$$d = \left(\frac{32(2)}{\pi(530 \times 10^6)} \sqrt{0^2 + (5.56)^2} \right)^{\frac{1}{3}} \approx 6\ (mm)$$

From those two results, it can be concluded that the larger value is 6 mm. However, by following the same procedure as choosing the diameter for the main shaft, the final value is rounded to 10 mm for design convenience and commonality adherence.

The procedure of finding the critical speed for the high-speed shaft is similar to what has been implemented for the main shaft. However, since this section assumes that the high-speed shaft only experiences the torque and no axial and moment loadings, it can be ideally deduced that the component will have little to no deflections. Because the deflection effect can be ignored, the critical speed can be expected to be so high that the operating speed $(500\ \times 18 = 9000\ rpm)$ lies much lower than the required limit. Therefore, the high-speed shaft functioning condition is acceptable.

5.5. Shaft Key Design

Since the keys only serve to attach the shaft and the gear, most of their design parameters are based on the selected gear (discussed in section 7). More specifically, the gear model fetched from the supplier has been designed with a built-in keyway. Therefore, the keyways of both shafts must bear the same width and height as those in the gears. The shaft key dimensions will multiply the width and height of the keyway in the gear by two.

- For the main shaft key:
    - Width: $\frac{D}{4} = \frac{30}{4} \approx 8\ (mm)$
    - Height: $7.8\ (mm)$
    - Tangential force: $F = \frac{T}{r} = \frac{100\ (Nm)}{15\ (mm)} = 6.67\ (kN)(applying\ equation\ 5.3)$
    - Length: $L = \frac{2Fn}{S_y w} = \frac{2(6.67 \times 10^3)(2)}{530 \times 10^6 \times 8 \times 10^{-3}} \approx 7\ (mm)$

The discovered length for the key from the theoretical calculation above is $7\ (mm)$. However, the built-in keyway in the hub gear from the supplier spans about $30\ (mm)$ in length, resulting in the 30-mm long key spreading along the gear keyway.

- For the high-speed shaft key:
    - Width: $\frac{D}{4} = \frac{10}{4} \approx 4\ (mm)$
    - Height: $5\ (mm)$
    - Tangential force: $F = \frac{T}{r} = \frac{\frac{100}{18}(Nm)}{5\ (mm)} = 1.11\ (kN)(applying\ equation\ 5.3)$
    - Length: $L = \frac{2Fn}{S_y w} = \frac{2(1.11\times10^3)(2)}{530\times10^6\times8\times10^{-3}} \approx 1\ (mm)$

The deduction from the calculation for the high-speed shaft is similar to that of the main shaft, meaning that the design dimensions, even though derived from the formula, are primarily designed from the gear pre-provided keyway. The measured length of that gear keyway is $30\ (mm)$, which is also determined to be the shaft key length.

6. Main Bearing Design

6.1. Design Assumptions

- Bearing Type: since the main shaft experiences both axial and radial loadings, the bearing chosen type must have the capability to withstand the combined loading condition. For this reason, a tapered roller bearing is selected since it is designed to handle heavy loads of any type without deformation. Furthermore, it proves to be more stable and durable, as well as deal with misalignment problems better than ball bearings [8].

- Bearing Material: the bearing material is usually Chrome Steel SAE 52100 as it has an outstanding performance of resistance against wearing out, being able to function at a very high temperature, and possessing a high hardness grade [9]. However, in this project, the manufacturer determines the material for the bearing.

6.2. Theoretical Background

- Reliability requirement [10]:

$$L_n = \frac{10^6 K_r}{60n}\left(\frac{C}{F_e}\right)^a \quad (6.1)$$

where:

$$L_n: Life\ of\ the\ desried\ reliability\ (hours);\ n: Rotational\ speed\ (rpm)$$

$$C: Basic\ dynamic\ loading\ rating\ (N);\ F_e: Equivalent\ Radial\ Load\ (N)$$

$$a = \frac{10}{3}\ for\ roller\ bearings; K_r: Adjustment\ factor$$

- Adjustment factor deduction [10]:

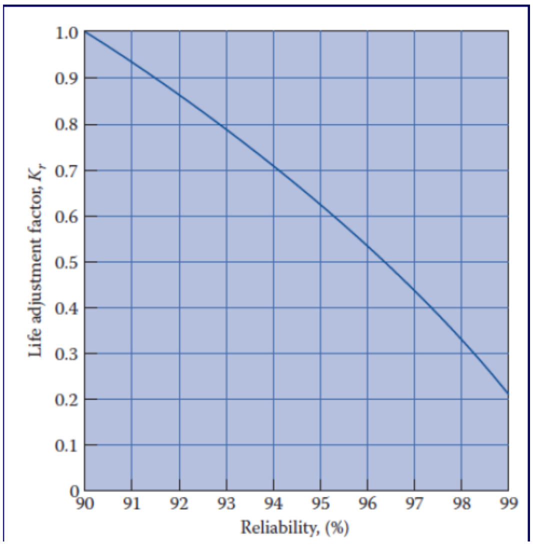

Figure 7. Life Adjustment Factor with respect to the Reliability graph [10]

Based on the Figure 7 and the required reliability of 98%, the adjustment factor (K_r) is estimated to be 0.32.

- Equivalent radial load of roller bearings:

$$F_e = 0.4F_r + KF_a \quad (6.2)$$

where:

F_r: $Radial\ loads\ (N)$; F_a: $Axial\ loads\ (N)$; $K\ (or\ Y)$: $provided\ in\ the\ manufacturer\ catalogue$

6.3. Design Calculations

To ease the burden of the design process, this project relies on the bearing manufacturer HIGHTEMP, whose website enables the user to enter the shaft diameter and desired bearing type and suggests available options. The combination of 30-mm shaft diameter and tapered roller bearing type will result in the model, named 32006, shown in Figure 6.3 below [6.3]. The fundamental parameters for this model are listed as:

- Reference Speed: 7500 (RPM)
- Limiting Speed: 10000 (RPM)
- Dynamic Load Rating (C): 21357 (N)
- Static Load Rating (C0): 19109 (N)
- Limiting value (e): 0.31

- Calculation factor (Y): 1.1
- Material: 52100 Steel

After selecting a suitable model, the only work is to check whether the dynamic load (C) calculated from the project requirements is less than the rating value stated by the manufacturer.

Applying equation 6.3, we will have:

$$P = F_e = 0.4(1500) + 1.1(800) = 1480\ (N)$$

Applying equation 6.2 results in:

$$L_2 = 8 \times 365 \times 24 = \frac{10^6 \times 0.32}{60 \times 500} \left(\frac{C}{1480}\right)^{\frac{10}{3}} \Rightarrow C \approx 20.68\ (kN)$$

From the calculated dynamic load rating whose calculation is performed above, it can be concluded that the resulting value is less than the value of C in the catalogue (21357 N). Therefore, the evidence is strong enough to clarify that the wind turbine will not suffer from the operation failure. In other words, the chosen tapered roller bearing model (32306), which will be installed on two ends of the main shaft, is suitable for this project. The actual model image on the manufacturer's website can be viewed in Figure 8.

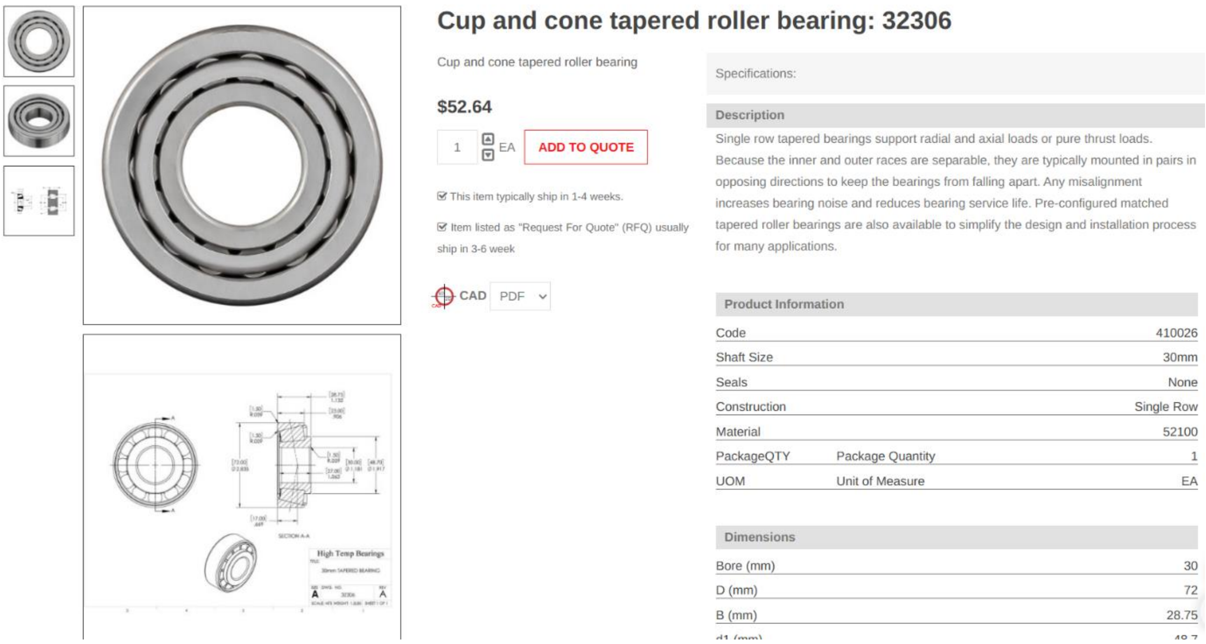

Figure 8. Tapered Roler Bearing Model suggested for design [11]

7. Gear Design

7.1. Design Assumptions

- Gear Type: spur gears are selected in this project because they can promise the greatest efficiency of all types while promising the simplest design and purchasing processes. They are also considered the most common among other gear types [12].

- Gear Material: the most common material for spur gear manufacturing is claimed to be carbon steel [13] since it is economical, durable, and offers machining simplicity. Therefore, this type of steel will be used for spur gear in this project.

- Number of Teeth: this value will vary throughout each gear and be assumed throughout the calculation process.

- Pressure Angle: is assumed to be 20 degrees instead of 14.5 degrees since 20 is the more commonly used value in the modern days.

- Gear and Shaft attachment: key will be used to attach these two components as it prevents slipping from rotation and promise efficient torque transmission.

7.2. Theoretical Background

- Gear Ratio [14]:

$$g_r = \frac{d_{p1}}{d_{p2}} = \frac{N_1}{N_2} = \frac{\omega_2}{\omega_1} \quad (7.1)$$

- Gear module [14]:

$$m = \frac{d_p}{N} \quad (7.2)$$

- Centre distance [14]:

$$c_d = \frac{d_{p1} + d_{p2}}{2} \quad (7.3)$$

where:

$$N: the\ number\ of\ teeth;\ \omega: angular\ velocity;\ d_p: pitch\ diameter$$

7.3. Design Calculations

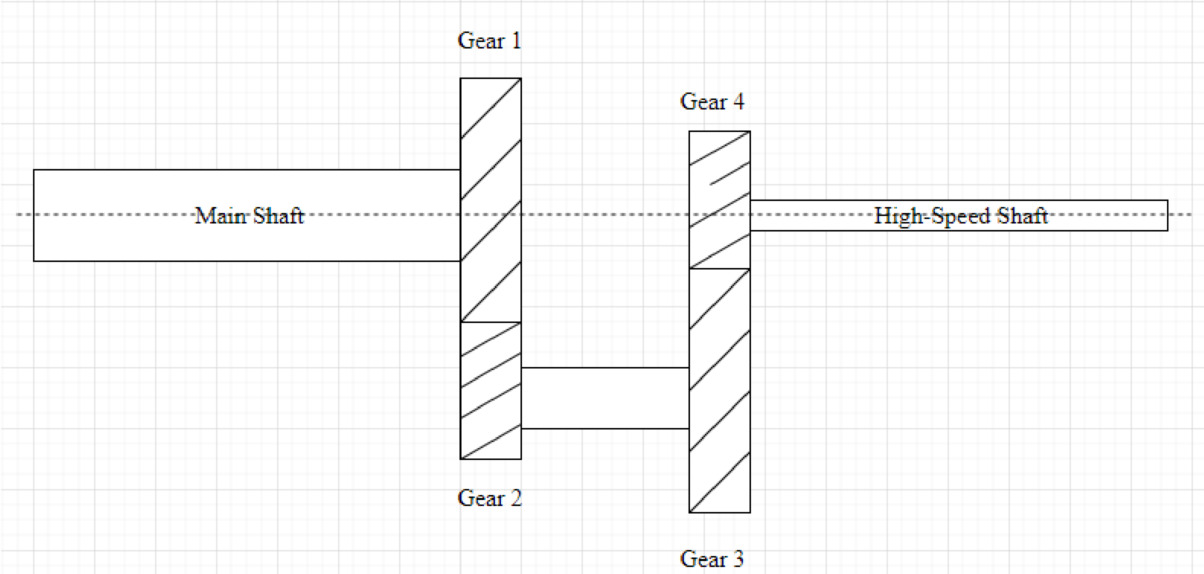

Figure 9. Simplified Gearbox Internal Layout Drawing

Figure 9 above depicts the simplified model of the gearbox used in this project. For this gearbox to satisfy all the design requirements, it must follow that:

- The transmission ratio is 18:1, indicating that the high-speed shaft must rotate 18 times faster and deliver the torque 18 times lower than the main shaft.
- The main shaft and the high-speed shaft must be in line, which means they must lie on the same centre line as shown in the drawing.
- To allow for workable meshing, the gears (1 and 2, 3 and 4) must share the same module (m).
- The bore diameters of gear 1 and gear 4 must be following the calculated shaft diameter mentioned in section 1.

Drawing on the equation 7.1 along with the given transmission ratio, we can deduce that:

$$\frac{T_{main\ shaft}}{T_{high-speed\ shaft}} = \frac{\omega_4}{\omega_1} = \frac{N_1}{N_2} \times \frac{N_3}{N_4} = \frac{18}{1} \Rightarrow \frac{N_1}{N_2}\frac{N_3}{N_4} = \frac{6}{1} \times \frac{3}{1} = \frac{9}{1} \times \frac{2}{1}$$

Based on the calculations above, it can be discovered that the design ratio can be broken into two cases (6 and 3, or 9 and 2). From that discovery, the module, and the number of teeth of a gear can be chosen freely of any values as long as they adhere to the constraints listed and produce the expected transmission ratio mentioned above. The gear manufacturer selected for this application is KHK, whose website provides a variety of gear types along with diverse standardized component parameters for the client to choose from. The four gears opted for design will have the specifications as follows:

Table 1. Gear Selection Table [15]

Gear Number	**Model**	**Number of teeth (teeth)**	**Bore diameter (mm)**	**Pitch diameter (mm)**	**Module**	**Material**
Gear 1	SSG1.5-90J30	$N_1 = 90$	30	135	1.5	S45C
Gear 2	SSGS1.5-10	$N_2 = 10$	12.2	15	1.5	S45C
Gear 3	PSA2-50	$N_3 = 50$	12	100	2	MC901
Gear 4	PS2-25J10	$N_4 = 25$	10	50	2	MC901

For the main and high-speed shafts to satisfy the in-line requirement, the centre distance between gears 1 and 2 should be equal to that of gears 3 and 4. Reflecting upon the table of gear selection above, we can clarify that:

$$\frac{135+15}{2} = 75; \frac{100+50}{2} = 75 \Rightarrow \frac{135+15}{2} = \frac{100+50}{2}$$

Therefore, the main shaft can stay on the same centre line as the high-speed shaft.

8. Brake Design

8.1 Design Assumption

- Brake Type: this project will look into implementing a thrust disk brake system, whose benefits are high durability, effortless maintenance, and high resistance to wearing and environmental factors such as dirt or other contaminants [16].
- Material: the brake material is assumed to follow the currently most popular option, which is cast iron [17]. This material offers excellent capability in heat dissipation.
- Model: the uniform wear model design procedure will be followed
- Operating condition: it is assumed that the disc brake will operate in a dry environment.

8.2 Theoretical Background

- For the uniform wear model:

$$T_\omega = \pi\gamma\mu r_i p_{max}({r_o}^2 - {r_i}^2) \ (8.1)$$

$$P_\omega = 2\pi\gamma p_{max} r_i (r_o - r_i) \ (8.2)$$

$where\ \mu: coefficient\ of\ friction; p_{max}: maximum\ contact\ pressure;\ \gamma: extend\ of\ brake\ pad$

$$r_o: outside\ radius; r_i: inside\ radius$$

8.3 Design Calculation

Since the brake will be installed on the high-speed shaft:

$$T_\omega = T_{high-speed\ shaft} = 5.56\ (Nm)$$

Friction material	Coefficient of friction, μ	Maximum contact pressure,[a] p_{max} kPa	Maximum bulk temperature, $t_{m,\ max}$ °C
Molded	0.25–0.45	1030–2070	204–260
Woven	0.25–0.45	345–690	204–260
Sintered metal	0.15–0.45	1030–2070	204–677
Cork	0.30–0.50	55–95	82
Wood	0.20–0.30	345–620	93
Cast iron; hard steel	0.15–0.25	690–1720	260

[a] Use of lower values will give longer life.

Figure 10. Disc Brake Design Parameters Table (Operating Dry) [18]

Based on the Figure 10 above with the assumptions made in section 8.1:

$$Coefficient\ of\ friction\ (\mu) = 0.25$$

$$Maximum\ contact\ pressure (p_{max}) = 690\ (kPa)$$

$$Extend\ of\ brake\ pad\ (\gamma)\ is\ assumed\ to\ be\ 1$$

Applying the equation 8.1 yields:

$$r_i({r_o}^2 - {r_i}^2) = \frac{n_s T_\omega}{\pi \mu p_{max}} = \frac{2(5.56)}{\pi(0.25)(690\ \times 10^3)} = 2.05 \times 10^{-5} (m^3)$$

$$\Rightarrow r_o = \sqrt{\frac{2.05 \times 10^{-5}}{r_i} + {r_i}^2}$$

The inner and minimum outside radii are:

$$\frac{dr_o}{dr_i} = \frac{0.5}{\sqrt{\frac{2.05 \times 10^{-5}}{r_i} + {r_i}^2}} \left(-\frac{2.05 \times 10^{-5}}{{r_i}^2} + 2r_i \right) = 0 \Rightarrow r_i = 21.72\ (mm)$$

$$\Rightarrow r_o = \sqrt{\frac{2.05 \times 10^{-5}}{0.02172} + (0.02172)^2} \approx 37.62\ (mm)$$

From the two radius values, the radius ratio can also be deduced:

$$\beta = \frac{r_i}{r_o} = \frac{21.72}{37.62} \approx 0.577$$

The greatest acceptable normal force applied to the brake is:

$$P = \frac{2n_s T_\omega}{\mu(r_o + r_i)} = \frac{2 \times 2 \times 5.56}{0.25 \times (0.03762 + 0.02172)} \approx 1500\ (N)$$

Based on the calculated results above, for ease of manufacturing, the inner radius (r_i) will be rounded to $25\ (mm)$ and the outside radius (r_o) will take $40\ (mm)$ as its design value.

9. Design Assembly

This section puts everything from the beginning together and displays images of the final product, which is a wind turbine model.

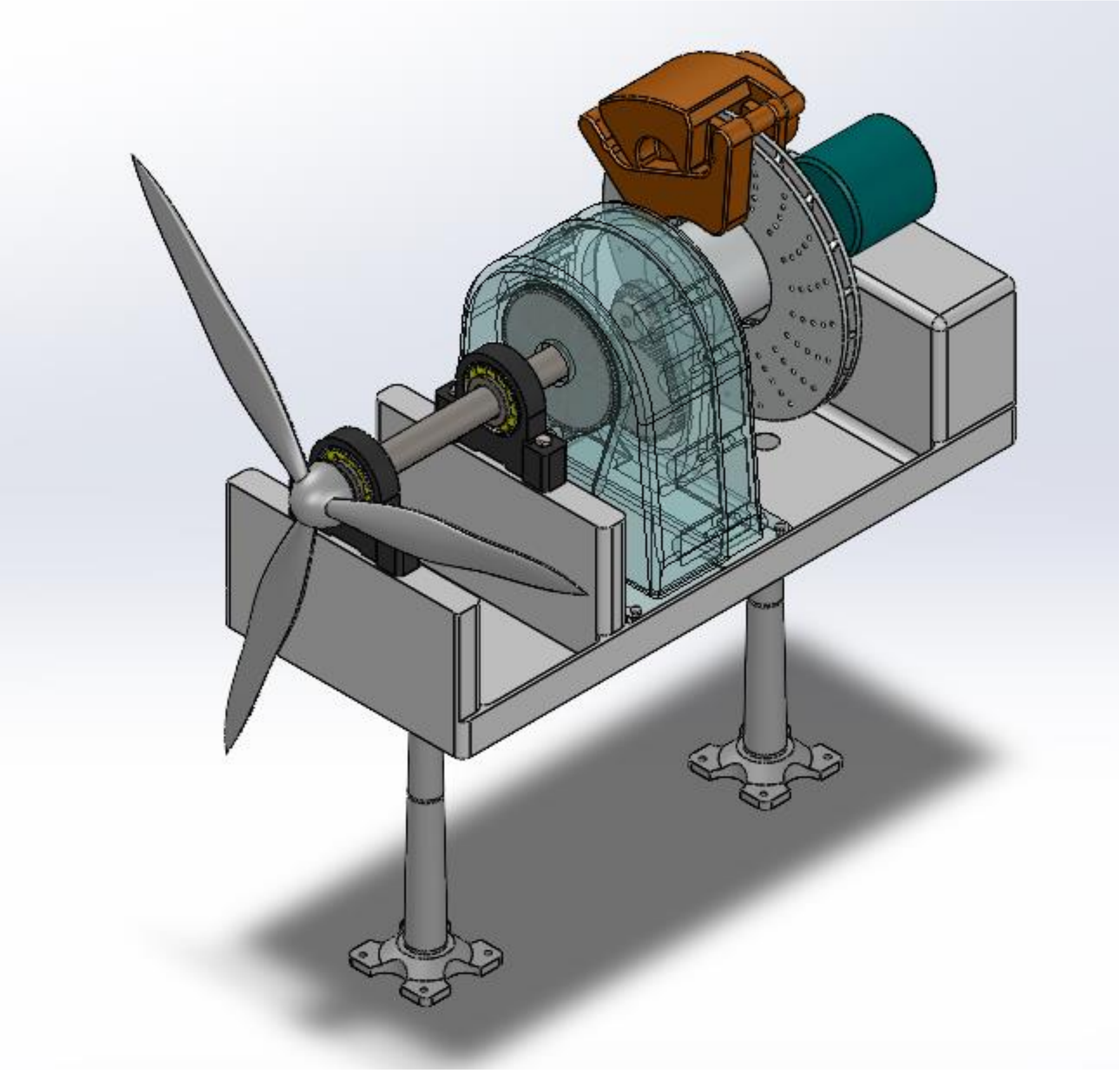

Figure 11. 3D drawing of the wind turbine model

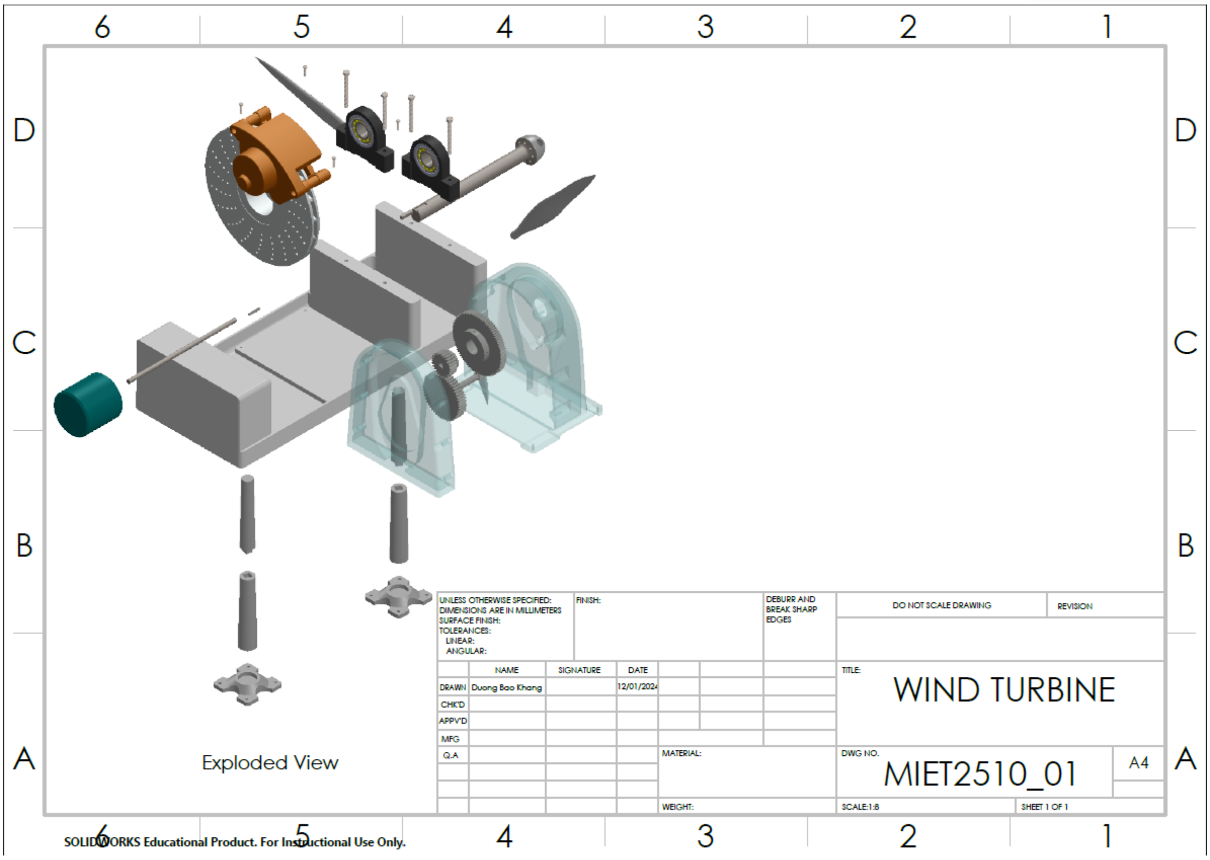

Figure 12. 2D drawing of the wind turbine.

10.Conclusion

This document aims to guide the reader through the process of calculating and selecting the suitable components (shaft, bearing, gear, and brake) to design a wind turbine model. Overall, each calculation section, diving into the detail of speculating a specific component, has been executed based on the scientific method presented in a certified textbook so that the final product is bound to function expectedly with the operating conditions provided in the problem statement.

11. Reference List

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12.Appendix A: Additional Materials

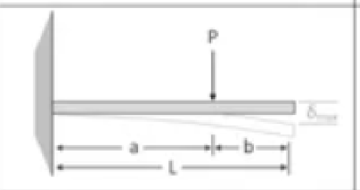
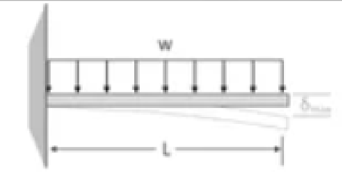
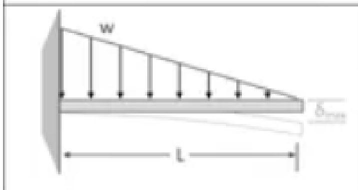
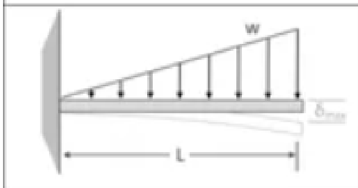
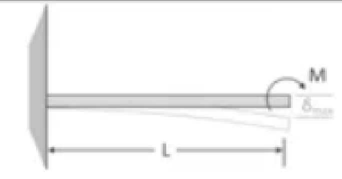
Beam and load cases	Maximum Beam Deflection
	$\delta_{max} = \dfrac{PL^3}{3EI}$
	$\delta_{max} = \dfrac{Pa^2(3L-a)}{6EI}$
	$\delta_{max} = \dfrac{wL^4}{8EI}$
	$\delta_{max} = \dfrac{wL^4}{30EI}$
	$\delta_{max} = \dfrac{11wL^4}{120EI}$
	$\delta_{max} = \dfrac{ML^2}{2EI}$

Figure 13. Deflection formulae list [19]

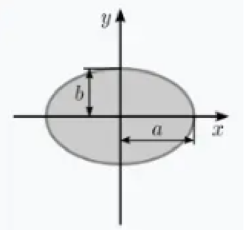
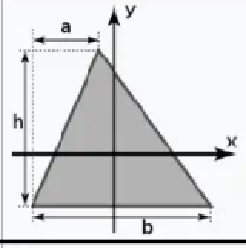
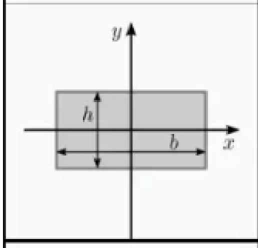
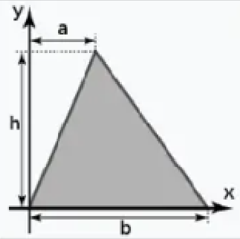
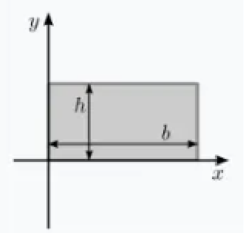
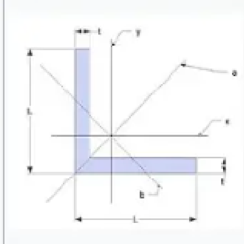
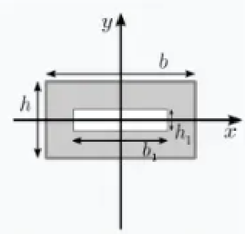
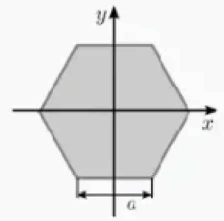
Shape	Moment of Inertia	Shape	Moment of Inertia
	$I_x = \frac{\pi}{4}ab^3$ $I_y = \frac{\pi}{4}a^3b$		$I_x = \frac{bh^3}{36}$ $I_y = \frac{b^3h - b^2ha - bha^2}{36}$ [5]
	$I_x = \frac{bh^3}{12}$ $I_y = \frac{b^3h}{12}$		$I_x = \frac{bh^3}{12}$ $I_y = \frac{b^3h + b^2ha - bha^2}{12}$ [5]
	$I_x = \frac{bh^3}{3}$ $I_y = \frac{b^3h}{3}$		$I_x = I_y = \frac{t(5L^2 - 5Lt + t^2)(L^2 - Lt + t^2)}{12(2L - t)}$ $I_{(xy)} = \frac{L^2t(L-t)^2}{4(t-2L)}$ $I_a = \frac{t(2L-t)(2L^2 - 2Lt + t^2)}{12}$ $I_b = \frac{t(2L^4 - 4L^3t + 8L^2t^2 - 6Lt^3 + t^4)}{12(2L-t)}$
	$I_x = \frac{bh^3 - b_1h_1^{\ 3}}{12}$ $I_y = \frac{b^3h - b_1^{\ 3}h_1}{12}$		$I_x = \frac{5\sqrt{3}}{16}a^4$ $I_y = \frac{5\sqrt{3}}{16}a^4$

Figure 14. Moment of inertia formulae list [20]