MAS416-G 23H

Modelling and Simulation of Mechatronic Systems

Designing of Wind Turbine Control Mechanism and Drive Train Modelling

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Submitted By

Group 11

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INTRODUCTION

Wind energy stands out as an important sector in the realm of renewable power, drawing upon the kinetic energy inherent in moving air to generate electricity through the use of wind turbines. These turbines feature blades connected to a generator, with the wind's force driving their rotation and facilitating the conversion of mechanical energy into a clean and sustainable electrical power source. This environmentally friendly energy alternative has gained substantial recognition as a key element in the global effort to reduce dependence on fossil fuels and combat climate change.The significance of wind energy in the contemporary era cannot be overstated, primarily owing to its renewable and sustainable attributes. In stark contrast to finite fossil fuels, wind power taps into the Earth's natural wind currents, offering a renewable source of electricity that is both clean and environmentally conscious. The process of electricity generation from wind does not release greenhouse gases or pollutants, thereby playing a crucial role in endeavours to mitigate climate change and enhance air quality. As the world confronts escalating environmental challenges, the imperative of transitioning to cleaner energy sources, exemplified by wind energy, becomes increasingly evident. However, the intermittent nature of wind poses a considerable challenge to harnessing its energy effectively. The control system of wind turbines emerges as a key battleground in addressing this challenge, relying predominantly on two controls: torque control and pitch control. Torque control comes into play during wind conditions below the rated speed. Its primary goal is to regulate generator torque and fine-tune rotor speed to optimize power production. Conversely, pitch control takes over in conditions surpassing the rated wind speed. This system adjusts the angle of the turbine blades to manage the aerodynamic forces acting upon them. Through modulation of the pitch angle, the blades can sustain a consistent rotational speed and cap the power output at the rated capacity [1].

The undertaking proposed involves the modelling and simulation of a wind turbine drivetrain coupled with a control loop designed to regulate the rotor speed of the turbine. By delving into these control mechanisms, the project aims to enhance the efficiency and reliability of wind energy systems, ultimately contributing to the broader mission of transitioning to sustainable and eco-friendly energy sources.

PROBLEM STATEMENT

This project is a part of the course MAS416 Modeling and Simulation at the University of Agder. This project report will consist of methodology to model and simulate a wind turbine drive train and a control loop to control the rotor speed of the turbine. The rotor speed of the wind turbine should be controlled in the following way as mentioned in TABLE 1.

TABLE 1. ROTOR SPEED CONTROL INTERVALS.

Interval A	If the wind speed is so small that the available power is less than 10 % of the nominal power, then the wind turbine should be at rest.
Interval B	The power factor should be kept at its optimum value, i.e., keep l = 6 and q = 0°. This interval is valid until the optimal tip speed ratio cannot be fulfilled without violating the maximum allowable tip speed. Use the generator to provide controller effort.
Interval C	Maintain maximum allowable tip speed, i.e., constant rotor speed and θ = 0°. This interval is valid until the obtainable air power reaches the nominal power value of the wind turbine. Use generator to provide controller effort.
Interval D	Maintain maximum allowable tip speed and nominal air power, i.e., constant rotor speed and constant generator torque. This interval is valid until the maximum acceptable wind speed is reached, $umax = 25\ m/s$. Use the pitch to provide controller

Interval E	Finally, with the wind speed larger than its maximum acceptable value the wind turbine should be at rest.

The different intervals are illustrated in Figure 1 below,

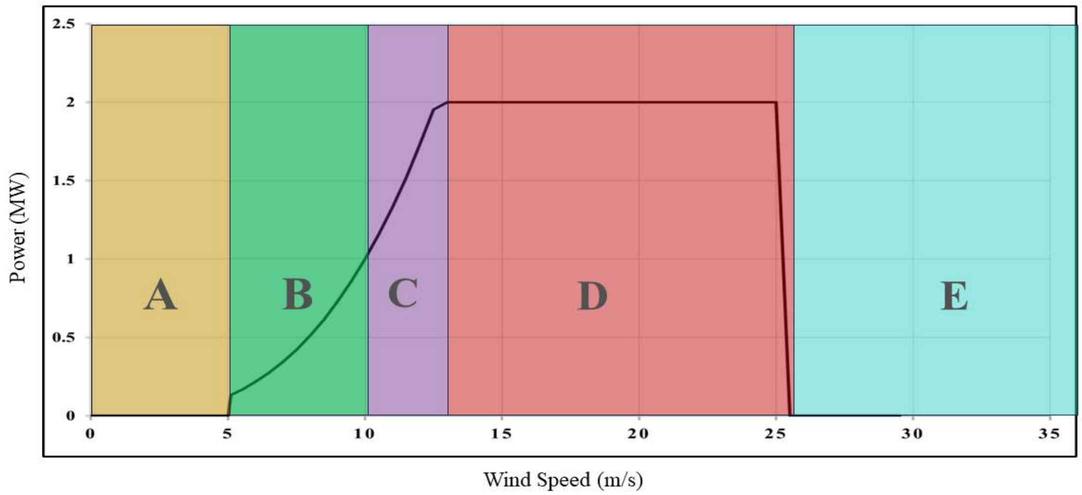

Figure 1. Power of the turbine at different intervals.

. The performance of the control strategies is verified by means of simulation. The simulated situations should at least include:

For this the performance of the control strategies is verified through simulation of the following four cases:

I. Wind speed increasing from 0 m⁄s to 25 m⁄s in 60 seconds.
II. Constant wind speeds within each of the previously mentioned interval's A ... D.
III. A jump in wind speeds (5 m⁄s) in a short time (1 second) from interval C to D or from interval D to C.
IV. Observe the wind turbine characteristics under portraying real wind conditions.

ASSUMPTION

- The wind turbine has a nominal power of P = 2 MW
- The gearbox consists of a planetary state and two regular stages. There are series of mass of inertia in drive train. These are listed in below.

Component	Mass moment of inertia, J ($kg.m^2$)
Rotor and blades.	2000000
Main shaft, planet carrier and planet gears.	200
Sun wheel, sun shaft and the first gear.	20
First pinion intermediate shaft and the second gear.	10
Second pinion, high-speed-shaft, coupling, brake and generator rotor.	220

- Blade radius of R = 35 m
- The maximum allowable tip speed is V_{tip_max} = 60 m/s

- Allowable Cut-in speed= 5 m/s; Cut-out speed= 25 m/s
- The generator has a nominal power of $P_{g,nom}$ = 2.1 MW
- Nominal speed of Generator $n_{g,nom}$ = 1550 (rev/min)
- For the generator torque control loop, the natural frequency, fn= 8 Hz , and the damping ratio, ξ=0.8, of its 2nd order system .
- For the pitch control loop, the natural frequency, fn= 1 Hz , and the damping ratio, ξ=1.0, of its 2nd order system .
- The generator torque limit is : $0 \leq \tau g \leq 1.8\ \tau g$
- The pitch angle limit is : $0 \leq \theta \leq 30^0$The main shaft is viewed as a torsional spring-damper system with a spring coefficient, $k\varphi$= 600 000 Nm/rad and a damper coefficient, $b\varphi$=400 000 $Nm \cdot s/rad$.

BACKGROUND AND DESIGN PROCEDURES

1.1 WIND POWER

A wind turbine has a limited amount of wind energy that can be converted to electrical energy, given from a power formula. Calculation of wind power in a wind turbine involves considering the wind speed, blade length, air density, and swept area to estimate the power output of the turbine. The power of the wind utilized by the turbine, P_{air} is calculated as equation (1) [2],

$$P_{air} = \frac{1}{2}\rho v^3 \pi R^2 C_p(\lambda, \theta) \qquad (1)$$

Air density (ρ) represents the mass of air per unit volume and is typically around 1.23 kg/m^3 at standard conditions, R is radius of the rotor to find the swept area of turbine, Wind speed (V) is the velocity of the wind.

Power coefficient (Cp) is ratio between total wind power and the power turbine can convert to mechanical energy. It requires the tip-speed ratio, λ, which is the ratio between the wind's incoming velocity and the blade tip velocity, and the pitch angle, θ, in degrees. These are determined by equation (2) [2],

$$C_p(\lambda, \theta) \approx (0.44 - 0.012\theta)\sin\left(\frac{\pi\lambda}{12 - 0.3\,\theta}\right) - 0.0015\lambda\theta \qquad (2)$$

Tip speed ratio (λ) can be defined as the ratio of a turbine's tangential tip speed to the speed of the incoming wind. This can be expressed by equation 3 [2],

$$\lambda = \frac{Wrot.R}{v} \qquad (3)$$

1.2 GEAR SYSTEM

The teeth of a gear mesh to transfer rotational force from one gear to another. In wind turbines, Gears are quite important, particularly in the gearbox system where they assist in controlling and transmitting rotational energy from the rotor to the generator. The three primary types of gear are worm gears, spur gears, and helical gears. For this assignment, helical gears will be used.

Rotor and generator angular velocities are used to calculate the overall gear ratio, which can be calculated using the equation (4) below [3],

$$i_{gear} = \frac{n_{g,nom}}{n_{r,nom}} \qquad (4)$$

The total gearing system, as shown in Figure 2, can be divided into two subsystems, the planetary gear and the single-stage gear.

 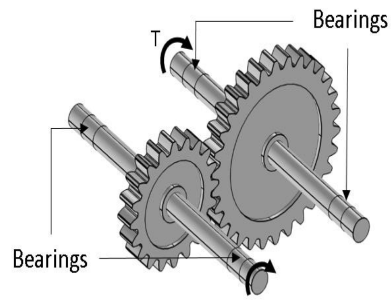

Figure 2. (a) Planetary Gear (b)Single Stage Gear.

Planatary Gear

A planetary gear system, also referred to as an epicyclic gear system, comprises three primary components: a sun gear, planet gears, and a ring gear. This configuration facilitates the transmission of rotational motion and torque in a unique manner.

Positioned at the system's center, the sun gear is typically driven by an external source such as an electric motor or an engine. Encircling the sun gear, the smaller planet gears are evenly spaced and connected to a carrier or arm. These planet gears engage with both the sun gear and the outermost ring gear, which features internal teeth. The rotation of the sun gear initiates a dual motion in the planet gears: they orbit around the sun gear while simultaneously rotating on their own axes. This complex movement results in a fascinating interaction where the planet gears not only revolve around the central sun gear but also spin individually on their axes. This intricate interplay among the sun gear, planet gears, and ring gear provides the planetary gear system with the capability to generate diverse gear ratios and torque outputs [3].

Single Stage Gear

A single-stage gear system is characterized by its simplicity, involving only one gear pair to transmit power from the input shaft to the output shaft. This configuration is commonly employed in a variety of applications, such as gearboxes, pumps, and other mechanical systems.In this setup, the transmission of power occurs directly through the engagement of a singular set of gears. The primary function of a single gear system is to either decrease or increase the rotational speed of the rotor, a parameter determined by the gear ratio. In this system, a driver gear is linked to an input shaft and transfers torque to the driven gear, which is attached to the output shaft. When the goal is to decrease speed, the driver gear requires a greater number of teeth compared to the driven gear. This fundamental principle of gear mechanics allows for the effective manipulation of speed in mechanical systems.

Helical Gear

Helical gears are a type of cylindrical gear with teeth that are cut at an angle to the axis of rotation. They are similar to spur gears but offer advantages in terms of torque capacity and smoothness of operation. Helical gears are cylindrical gears with teeth bent into a helix shape; these teeth are positioned at an angle to the gear axis called the helix angle. Helical gears feature stronger teeth and a higher load-carrying capability than spur gears.

The main purpose of helical gears as power transmission devices is to enhance torque and decrease speed between rotating shafts. Special teeth in helical gears are positioned at a specific angle to the shaft and the gear face. When two teeth in a helical gear system make contact, the initial point of touch is at one end of the tooth, and as the gears turn, the contact gradually expands until the two teeth are fully engaged. Since more than one tooth makes contact during the action, the gear can withstand a greater load. Due to the high torque and relatively low speeds, a helixangle of 15° is selected for the planetary gearing system. Gears with parallel shafts have opposite hands, while gears with perpendicular shafts have same hands are shown at Figure 3 The planetary gear system consists of helical gears with intermeshing parts. All sun, planet and ring gears must, therefore, have the same helix angle, while having different hand. A single helical gear for planetary gear was chosen as there is no need for a complex double-stage gear. Due to the high torque and relativly low speeds, a helix angle of 15° is selected for the plantery gearing system.

The single-staged gearing systems have high speeds and less torque, which makes it necessary to decrease rotational friction losses. Increasing the helix angle smooths operation with less friction. For the single-stage gears, a higher helix angle can be applied with the usage of double helical gear, also known as a herringbone. Herringbone gears have the advantage of having a net zero force with increased helical angle, due to both gears being mounted side by side, and taking up opposite directional force applied to the single gears. a helical angle of 30 degrees is chosen for the two single stage gears.

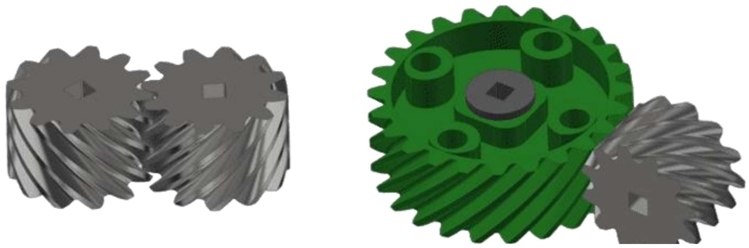

Figure 3. 3D representation of Helical Gear.

1.3 CALCULATION OF $W_{rot,max}$

The angular rotational speed of the turine given by the specifications can be calculated as,

$$\text{Wrot, max} = \frac{V_{tip,max}}{R} = \frac{60\ m/s}{35m} = 1.714\frac{rad}{s}$$

$$\text{Wrot, max} = 1{,}714\ \times\frac{60}{2\pi} = 16.37\ rev/min$$

1.4 GEAR SYSTEM CALCULATIONS AND SIMSCAPE MODELLING

The gear ratio based on the design specification for the whole system should be,

$$i_{gear} = \frac{n_{g,nom}}{n_{r,nom}} = \frac{1550}{16.37} = 94.68$$

The selection of the gear is determined by the specified requirements outlined in the task description, as well as industry citation [4]. The complete configuration of the gear systems is documented in TABLE 2. The specific design dimensions can be found in Appendix B: Gear System Designing, accompanied by a 3D model created in Simscape. In this particular design, the resulting gear ratio of the overall system is determined to be 94.8 as mentioned in TABLE 2, which closely aligns with our desired target.

TABLE 2 OVERVIEW OF THE GEAR SYSTEM DESIGN

Stage	Cog	Number of Teeth	Module (mm)	Diameter (mm)	Gear Ratio
Planetary Gear	Ring	124	16	1984	5.13
	Sun	30		480	
	Planet	47		752	
Single Stage Gear 1	Driver	136	8	1088	4.53
	Driven	30		240	
Single Stage Gear 2	Driver	102	5	510	4.08
	Driven	25		125	

A Simscape-based 3D model has been designed to visually represent the entire system. To simplify the complexitites of 3D design, we have exclusively incorporated essential components, including the rotor, blades, and gear specifications, forming the foundational model. This model serves as a visual representation illustrating the system's behaviour in response to input from the control system.

For a detailed examination of the model's architecture, the Simulink schematic is provided in Appendix J-3D-model. This 3D design facilitates a practical understanding of how the control system operates within the system. It offers insights into the dynamic interactions between the control system and key physical components, contributing to a comprehensive understanding of the system's functionality as presented in Figure 4.

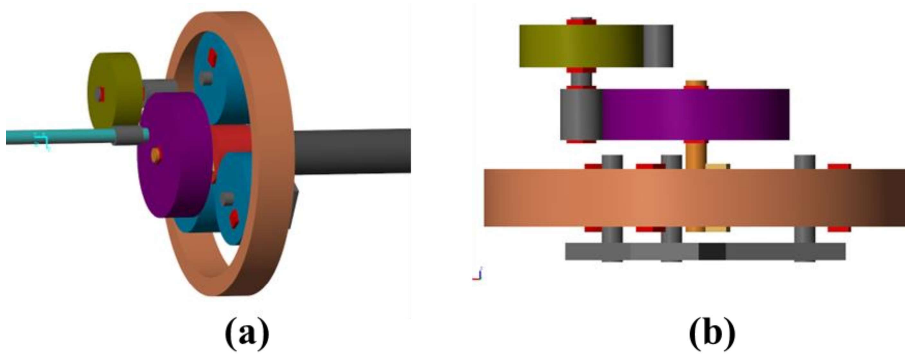

Figure 4. Simscape modelling of the gearing system (a) Isometric view, (b) side view.

1.5 CONTROL SYSTEMS

A wind turbine's control system is an essential part responsible for optimizing the turbine's operation and ensuring its efficiency and safety. Depending on the wind conditions, it employs multiple control techniques and algorithms to alter factors such as blade pitch, generator torque, and yaw angle. In this project, we will attempt to control the rotation speed of the rotor using the $W_{rot(ref)}$. The $W_{rot(ref)}$ will be determined by the wind speed extracted from the environment [5]. In the second section, the error between the required rotor speed and the rotor speed obtained from drive train flexibility will be controlled by two mechanisms. According to the problem statement in PROBLEM STATEMENT, the first control system's task is to control the generator's torque, and the second part's task is to handle the blade's pitch angle. These two controllers will work together to accomplish the specified rotor speed at a given wind speed. Figure 5 illustrates the whole control system as a block diagram.

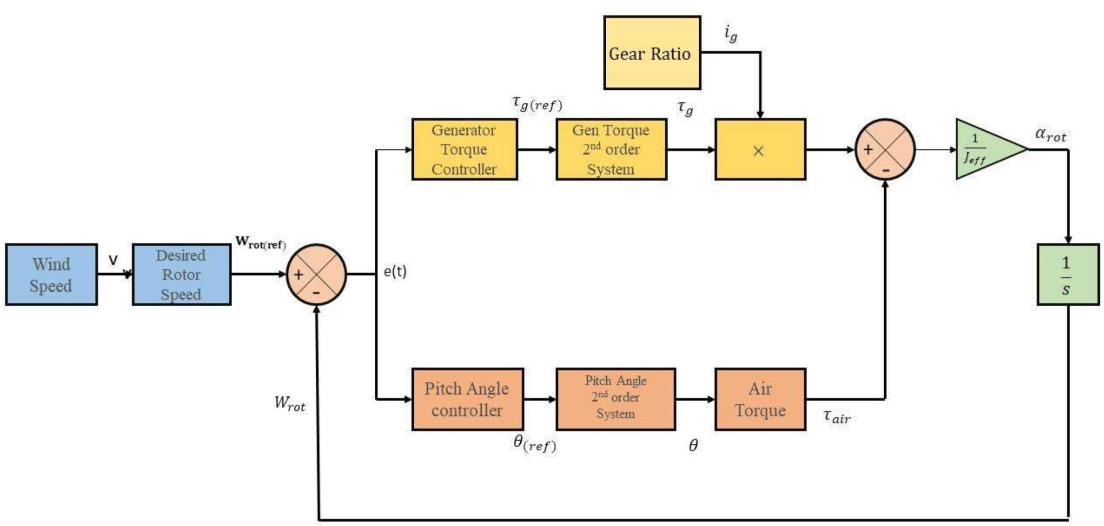

Figure 5. Control Systems of whole model.

1.6 GENERATOR TORQUE CONTROL AND PITCH ANGLE CONTROL

To get the proper power effort from this wind turbine the two control methods must be worked simultaneously. The torque that will be generated from the torque system will be multiplied by gear ratio to get the desired speed at the generation and the pitch angle control will work after threshold wind speed which is 12.5 m/s in our case to maintain the rated power at the generation end. Pitch angle regulation is subsequently employed to maintain the rotation speed at its maximal tip speed, thereby optimizing power conversion, even when wind speeds surpass the threshold speed, until the cut-out wind speed (25 m/s) is attained. The generator torque acts as a brake on the rotor speed, W_{rot}, in the opposite direction of the wind velocity [6]. This is used to control the rotor speed until the wind speed that allows the turbine's nominal power, P_{nom}, while respecting the maximum tip speed requirement, (V_{tip_max}) is reached. The air torque can be calculated from the Air Torque block using the equation (5) and equation(6). So the dynamics of the passive system will work as the Figure 6 below.

$$\alpha_{rot} = \frac{1}{j_{eff}}(\tau_{air} - \tau_g * i_{gear}) \quad (5)$$

$$\tau_{air} = \frac{P_{air}}{\mathrm{Wrot}} \quad (6)$$

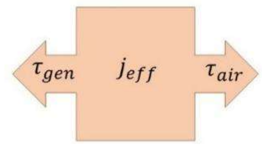

Figure 6. Simplified Free body diagram.

1.7 SECOND ORDER SYSTEM

The term "order" in the context of a control system refers to the system's level or complexity. The number of independent energy storage elements or the greatest order of the linear differential equation that describes the system determines it. A control system's order is significant because it influences the system's behavior, stability, and performance. In the context of this project, wherein double time-derivatives are employed to determine rotational acceleration, it is necessary to employ a second order transfer function to independently convert the gains associated with the P, I, and D control elements. The transfer function of a second order system is represented by equation (7-8), wherein K denotes the gain and ξ represents the damping ratio.

$$G(s) = \frac{k}{\frac{s^2}{W_{sys}^2} + \frac{2\xi s}{w_{sys}} + 1} \quad (7)$$

$$w_{sys} = 2\pi f_n \quad (8)$$

Where, f_n is the natural frequency of the system.

1.8 PID CONTROLLERS

A PID controller, which stands for "Proportional-Integral-Derivative controller," is an important piece of equipment for controlling different process factors like speed, temperature, flow, and pressure. The PID controller is known for being very flexible. It uses a feedback loop to make sure that the system's real output stays close to a target or setpoint [7].

The PID processor is made up of three separate parts:

Proportional Control: This part makes an output that is directly related to the difference between the real output and the setpoint that has already been set. Proportional control is effective at quickly resolving any deviation from the desired setpoint because it responds instantly. The P term decrease the risetime.

Integral Control: This part of the control system figures out and takes into account the mistake that has accumulated up over time, then changes the control signal. Its job is very important for reducing steady-state mistakes and making sure the system reaches and stays at the setpoint over time. The I term minimize the steady state error.

Derivative Control: This part of the system measures the rate at which the mistake is changing and then changes the control signal based on that rate. Its job is to guess what changes will happen in the system and stop overshooting or oscillates before they happen. This makes the system more stable. The D term minimize stabilizes the system.

The control system of PID controller is presented by a block below:

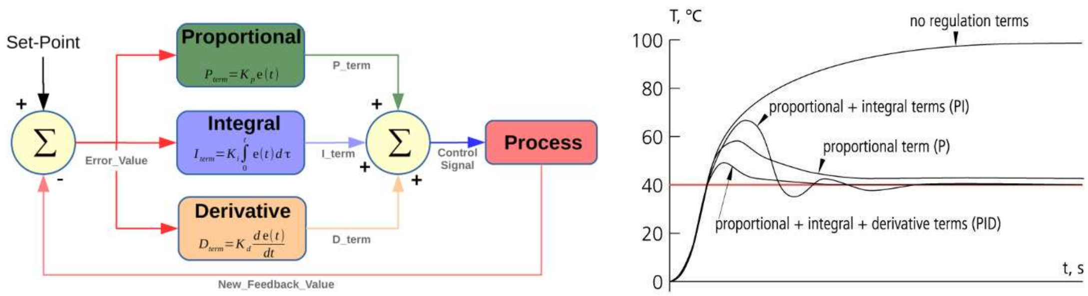

Figure 7. (a) PID controllers block diagram; (b) PID controlling Mechanism.

1.9 FILTERING

To obtain the desired values from the controllers, we must apply filters to the output. The pitch of the blade must not exceed 30°, and the generator's torque must not exceed $1.8\tau_g$ and either value should not fall below 0. Filters are used in the PID controller with output saturation after the transfer function.

1.10 SATURATION

When it comes to a PID controller, saturation is when the output of the controller hits its highest or lowest point because of the system's physical limits. When these limits are reached, the control signal can't be raised or lowered any further, which is called saturation. Depending on what the system is controlling, the response can be either higher or lower. Saturation for our system is maximum and minimum breaking power for torque control (generator maximum torque of $1.8\tau_g$ and maximum and minimum pitch angle of the blades of (0-30)°.

1.11 ANTI WIND UP

In a PID (Proportional-Integral-Derivative) controller, the anti-windup approach is a strategy used to address and minimize the consequences of integrator wind-up. When the system under control impacts its limit, an integral wind-up happens. This means that even when the actuator or process cannot react appropriately, the integral term of the PID controller keeps adding errors. Implementing anti-windup measures will restrict the integration term in this case. This can be achieved through two approaches: incorporating an internal "anti-windup feedback circuit" into the controller that has a gain known as the "Back-calculation" coefficient, or completely disabling the integrator when the error it receives has the same sign as the output of the controller. The respective names of these two techniques are the back-calculation method and the clamping method [8].

DRIVE TRAIN MODELLING AND CONTROLLER DESIGN

Tuning of the system was conducted by subjecting it to testing under a consistent wind speed in each of the regions dedicated to torque control and pitch control. For the torque control region, a constants wind velocity of 6 m/s was chosen, while a threshold of wind velocity 20 m/s was selected for the pitch control region. Through a process of trial and error, a steady state behaviour was attained. The final parameters for both controllers, as well as the associated transfer functions operating frequency, damping ratio, as determined and are presented in TABLE 3.

TABLE 3 CONTROLLER AND TRANSFER FUNCTION PARAMETERS

Control Type	**Operating Frequency f_n**	**Damping ratio ξ**	**P**	**I**	**D**	**Filter Coefficient**
Torque Control	8	0.8	-600000	-300000	- -	800
Pitch Control	1	1	-250	-100	-100	800

The drivetrain of a wind turbine consists of various components, including structural elements like the main shaft, mechanical constituents like gears and bearings, and electrical components such as the generator and converter (not considered in this report). Wind turbine drivetrains can be either geared or direct drive and can be mounted on fixed base or floating wind turbines. In direct-drive systems, the rotor of the wind turbine is directly connected to the generator, resulting in a large torque and inertia coupling. Non-torque loads can also be transmitted directly to the support structure.

To observe the impact of the practical system's mass on the designed control system (as shown in Appendix C: Drive Train Felxibility Modelling), a drivetrain flexibility model is introduced in the simulation. This model helps analyze how the mass of the system (spring and damping coefficients) affects the behavior of the control system. The results of the simulation are presented in Figure 8, which illustrates the behavior of the simulated practical system on the control system.

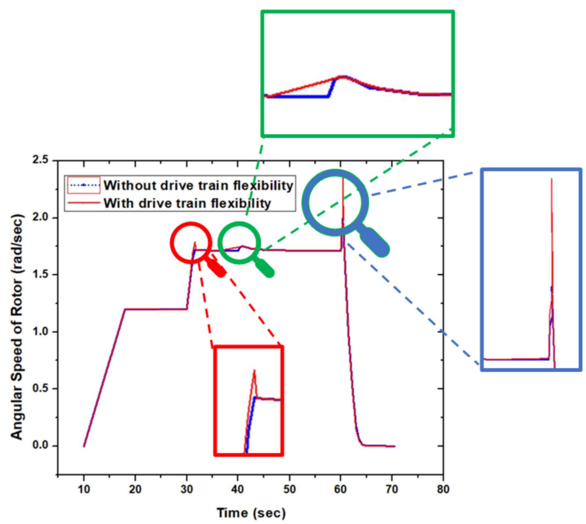

Figure 8. Simulated characteristics of the angular speed of rotor for case 2 illustrating the impact of drive train

In the Figure, it is worth noting that the inclusion of the drivetrain introduces a minor fractional overshoot during the transitions for the initial set of gains in the behaviour of the practical system. Therefore, further optimization of the gain seemed necessary and reduced as per the requirement. The final gains of the PID controllers are listed in TABLE 3, and their behaviour is reported in Sections 2.1-2.4 for various cases.

RESULTS AND ANALYSIS

2.1 CASE 1

In Case 1, there is a wind velocity profile in which the velocity consistently increases at a fixed rate from 0 m/s to 25 m/s over a duration of 60 seconds. The velocity surge at this constant rate is extended for 80 seconds to assess the turbine's braking capability beyond the cut-out speed. This scenario is depicted in Figure 9(a), where the wind speed continuously rises at a rate of 0.41667 m/s^2. It is important to highlight that a threshold magnitude of 12.622 m/s is defined as the point at which the control mechanisms shift. Below this threshold, the torque control mechanism generates sufficient torque for the turbine, and the blade angle remains fixed at 0°. However, beyond this threshold value, the control mechanism begins to adjust the pitch angle while maintaining a constant magnitude of generated torque to ensure consistent power generation.

As previously mentioned, the turbine's cut-in and cut-out speeds are set at 5 m/s and 25 m/s, respectively. Beyond the cut-in speed, the reference rotor speed increases linearly from 1 rad/s in relation to the wind speed until it reaches the maximum tip speed of 60 m/s. However, in practical cases, there is a slight delay and a nonlinear response in the initial wind velocities. Eventually, the practical case aligns with the reference rotor speed as shown in Figure 9(b). At around 32 seconds, a slight anomaly in the rotor speed occurs, indicating that the controller has switched to pitch control, and the pitch control mechanism begins to regulate. The wind speed results in a maximum angular velocity of 1.680 rad/s, starting from the threshold velocity and continuing until the cut-out windspeed of 25 m/s is reached. After reaching the cut-out windspeed, the torque-brake system is activated to assist in keeping the turbine at rest, causing the generated torque to become zero.

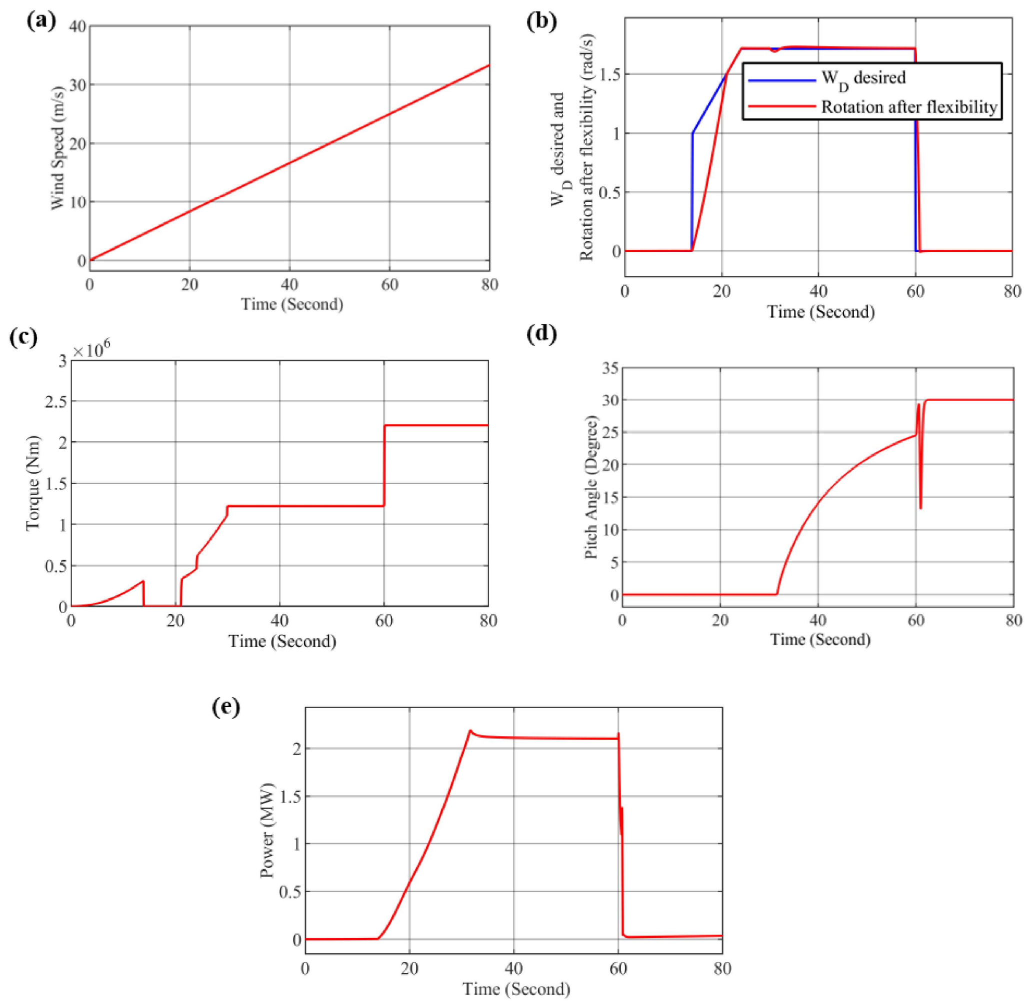

Figure 9. Simulated characteristics of the turbine for case 1; (a) Artificially created wind speed behavior, (b) Reference and practically obtained rotational speed of rotor with drive-train flexibility, (c) Generated Torque (d) Pitch angle of the rotor blades (e) Power curve.

Figure 9(c) presents the generated torque obtained from the gear settings output. Consistent with the typical behaviour of wind turbines, the generated torque increases from the cut-in speed. However, there is an abrupt drop to zero observed between 14 to 21 seconds, which corresponds to a velocity range of 5.83 m/s to 8.75 m/s because of the sudden rise of speed that in turn responded through decreasing the torque. Moreover, after initially reaching a certain level of torque, another noticeable change occurs around 30 seconds when the control mechanism shifts at the threshold velocity of 12.622 m/s. Subsequently, the generated torque remains constant at the nominal generator torque value. Beyond the cut-out speed, when the braking mechanism is engaged, the generated torque is maintained at a constant value of 1.8 times the nominal generator torque.

In Figure 9(d), the pitch angle of the blade is illustrated for both operating regions of wind speed. Up to 30 seconds, when the wind velocity is below the threshold velocity, the pitch control mechanism remains inactive, and the torque control mechanism supplies the required torque, resulting in a constant pitch angle of 0°. However, once the threshold limit is exceeded, the pitch angle undergoes continuous adjustments to maintain a constant extracted output from the turbine. Subsequently, after reaching the cut-out limit, the pitch angle remains unchanged at 30°.

The power extracted from the turbine is depicted in Figure 9(e), and it is evident that the turbine initiates power generation starting from the cut-in speed limit. Up to the threshold limit for pitch control, the generated power increases at a rate proportional to the cube of the wind velocity. Once the control mechanism shifts, a slight overshoot is observed in the power curve, followed by a constant generation of power at the nominal rotor value. Finally, after reaching the cut-out speed, the power generated from the turbine becomes zero.

2.2 CASE 2

In Case 2, there is a wind velocity profile characterized by constant wind speeds within each of the intervals specified in Table 1, which outlines the use cases and operating intervals, shown in Figure 10(a). Specifically, in the region from 1 to 5, the wind velocity remains constant at 4, 7, 11, 15, and 27 m/s. It is crucial to note that the threshold magnitude for shifting the control mechanisms is maintained at the same value as in Case 1.

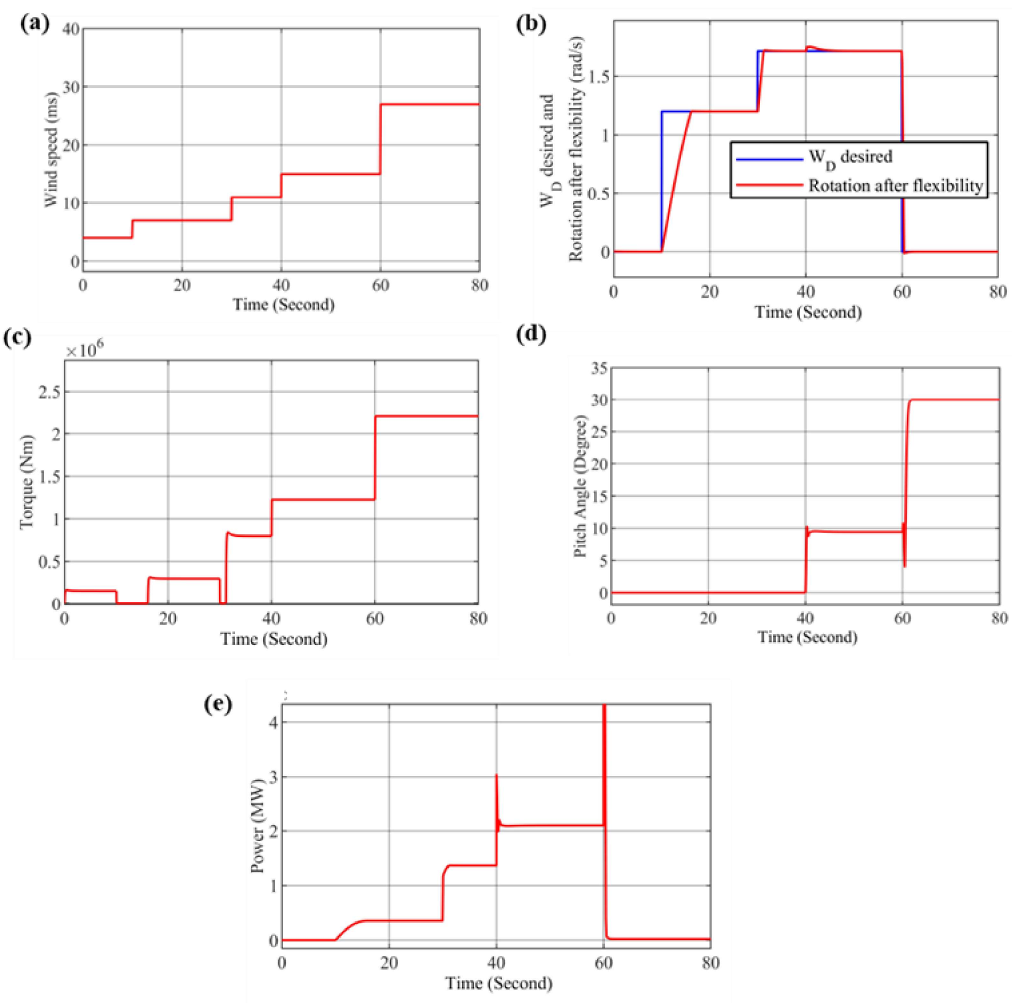

Figure 10. Simulated characteristics of the turbine for case 2; (a) Artificially created wind speed behavior, (b) Reference and practically obtained rotational speed of rotor with drive-train flexibility, (c) Generated Torque (d) Pitch angle of the rotor blades, (e) Power curve.

The rotor speed profile for Case 2 is presented in Figure 10(b). In Region 1, where the wind velocity remains below the cut-in speed limit, the rotor speed is visualized as zero, as shown in the figure. In the subsequent three consecutive regions, the wind velocity remains constant within the boundaries of the cut-in and cut-out speeds. As a result, the generated torque is maintained at constant magnitudes in these regions. In Region 5, the wind velocity exceeds the cut-out limit, triggering the activation of the torque-brake system to keep the turbine at rest, causing the generated torque to drop to zero. Notably, in Region 4, a slight peak is observed in the generated torque, indicating the shift in the control mechanism.

Figure 10(c) displays the generated torque obtained from the gear settings output. Within the torque control region, after the initial torque startup, there are a few moments in which the torque drops to zero at the instant of a sudden rise in wind velocity, before quickly jumping back up to the designated torque magnitude. This phenomenon is clearly observed at the beginning of both Region 2 and Region 3, when a sudden rise is speed occurs. Additionally, after initially reaching a specific torque level, another noticeable change occurs around 40 seconds when the control mechanism shifts at the threshold velocity of 12.622 m/s. Subsequently, the generated torque remains constant at the nominal generator torque value. Beyond the cut-out speed, when the braking mechanism is engaged, the generated torque is maintained at a constant value of 1.8 times the nominal generator torque.

Figure 10(d) illustrates the pitch angle of the turbine blade for different wind speed operating regions. Up until 40 seconds, when the wind velocity is below the threshold velocity, the pitch control mechanism remains inactive, and the required torque is supplied by the torque control mechanism, resulting in a constant pitch angle of 0°. However, once the threshold limit is surpassed, the pitch angle undergoes continuous adjustments to maintain a constant extracted output from the turbine, with a small overshoot observed initially. Subsequently, after reaching the cut-out limit, the pitch angle remains unchanged at 30°.

The power extracted from the turbine is depicted in Figure 10(e). It is evident that the turbine starts generating power from the cut-in speed limit, and the power output varies depending on the achievable wind speed velocity. When the control mechanism shifts, a slight overshoot is observed in the power curve, followed by a steady generation of power at the nominal rotor value. Finally, after reaching the cut-out speed, the power generated from the turbine drops to zero, accompanied by a significant abrupt overshoot in the turbine's behaviour.

2.3 CASE 3

In Case 3, there is a sudden increase in wind speed by 5 m/s within 1 second during the transition from C to D or D to C. This situation is illustrated in Figure 11(a), where the wind speed abruptly rises from 10 m/s to 15 m/s at 25 seconds and remains constant thereafter. It should be noted that while the wind speed is at 10 m/s, the torque control mechanism is active, and the pitch angle is maintained at 0°. However, after the sudden increase in speed, the torque is maintained at a specific magnitude, and the pitch angle is adjusted according to the speed.

Figure 11(b) displays the reference rotational speed and the actual rotational speed of the turbine. It can be observed that the reference rotational speed remains constant since the blade tip speed consistently exceeds the maximum allowable tip speed. However, in practical situations, it takes some time for the turbine to reach the reference speed from a stalled position, so a slight delay and a small amount of overshoot are acceptable. The sudden speed increase occurs at 25 seconds, surpassing the threshold for the control mechanism. At this moment, the control mechanism shifts from torque control to pitch control, resulting in a slight deviation. From that point onwards, the turbine's rotational speed follows the reference speed.

Figure 11(c) illustrates the generated torque obtained from the gear settings output. Following the general characteristics of the wind turbine, the generated torque starts to increase as the actual rotational speed aligns with the reference rotational speed. Additionally, based on the wind velocity extracted after initially reaching a certain torque level, another noticeable change is observed around 25 seconds when the speed changes, while the generated torque remains constant when the velocity remains unchanged.

Figure 11(d), the pitch angle of the blade is depicted for both operating regions of wind speed. Until 25 seconds, when the wind velocity is constant at 10 m/s, the pitch control mechanism remains stalled, and the torque control mechanism maintains a constant torque, resulting in a constant pitch angle of 0°. However, after the sudden change in wind velocity, the pitch angle abruptly changes, exhibiting a small amount of overshoot due to the pitch control mechanism, and then remains constant as the wind velocity remains unchanged.

The power extracted from the turbine is shown in Figure 11(e), where it can be observed that the turbine takes some time to generate the optimum extractable power from a specific wind speed. This delay is due to the time required for the rotor speed to reach a certain magnitude and for the torque control mechanism to operate. As expected, the extracted power gradually increases until it stabilizes. The sudden change in speed causes another jump in extracted power, but with an overshoot resulting from the pitch control mechanism.

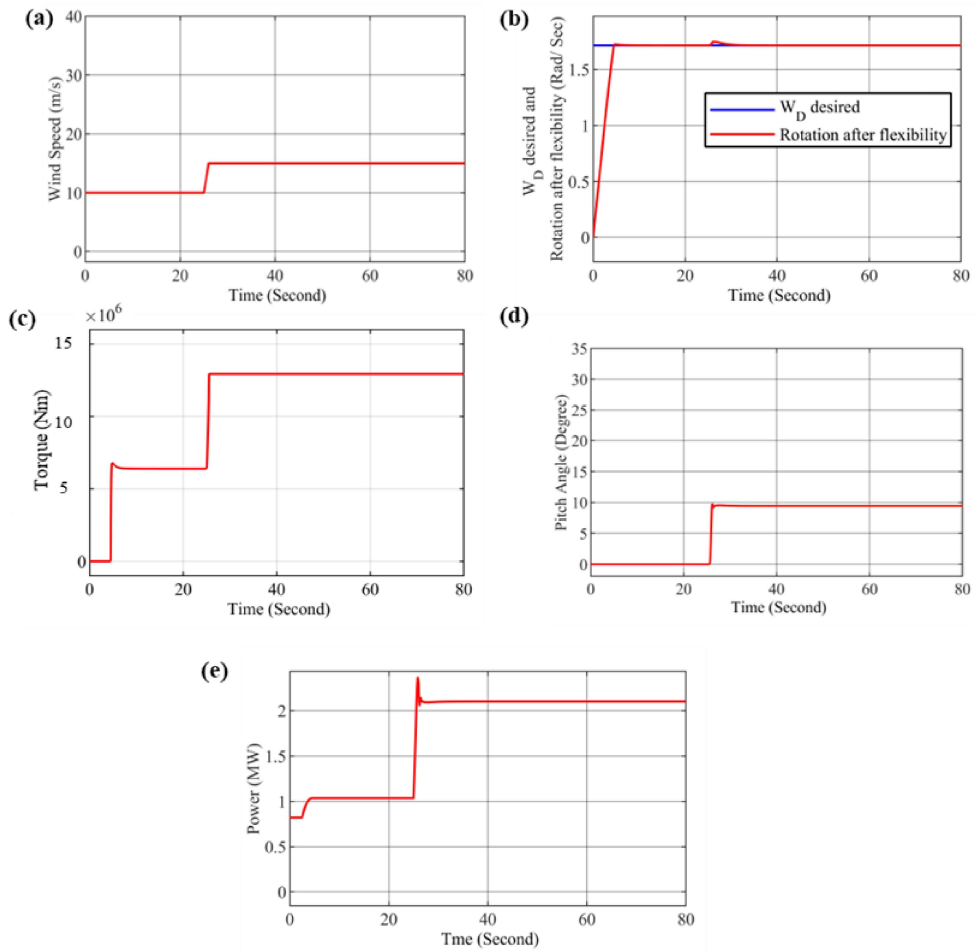

Figure 11. Simulated characteristics of the turbine for case 3; (a) Artificially created wind speed behavior, (b) Reference and practically obtained rotational speed of rotor with drive-train flexibility, (c) Generated Torque (d) Pitch angle of the rotor blades, (e) Power curve.

2.4 CASE 4

In Case 4, the wind velocity profile is characterized by variable wind speeds throughout the entire interval, aiming to simulate a practical scenario that a wind turbine may encounter. To model this variability, the wind velocity has been represented as a function of time,

$$v = 4\sin\left(2\frac{\pi}{6}t\right) + 9 + \sin\left(2\frac{\pi}{10}t\right) + \sin\left(2\frac{\pi}{12}t\right)$$

The depicted equation represents the overall behavior of the wind speed, as illustrated in Figure 12(a). Similar to the previous cases, it is important to emphasize that the threshold magnitude for shifting the control mechanisms remains consistent with the value observed in Case 1.

Figure 12(b) showcases the reference rotational speed and the actual rotational speed of the turbine. It is evident that the reference rotational speed undergoes abrupt variations, briefly exceeding the maximum permissible tip speed before returning below the limit. However, in practical scenarios, it is expected that the turbine requires some time to reach the reference speed, allowing for a slight delay and a minor degree of overshoot. This behavior is observed in the figure, where the turbine endeavors to replicate the theoretical behavior and minimize the error, but the changes occur too rapidly for the turbine to fully adjust.

Figure 12(c) depicts the generated torque acquired from the gear settings output. Consistent with the overall characteristics of the wind turbine, the generated torque begins to rise as the actual rotational speed aligns with the reference rotational speed. However, when a sudden surge in wind velocity occurs, the rotor adjusts to synchronize with the reference speed by reducing the torque to zero. As a result of the persistent discontinuity in wind velocity, a somewhat fluctuating behavior in generated torque becomes apparent.

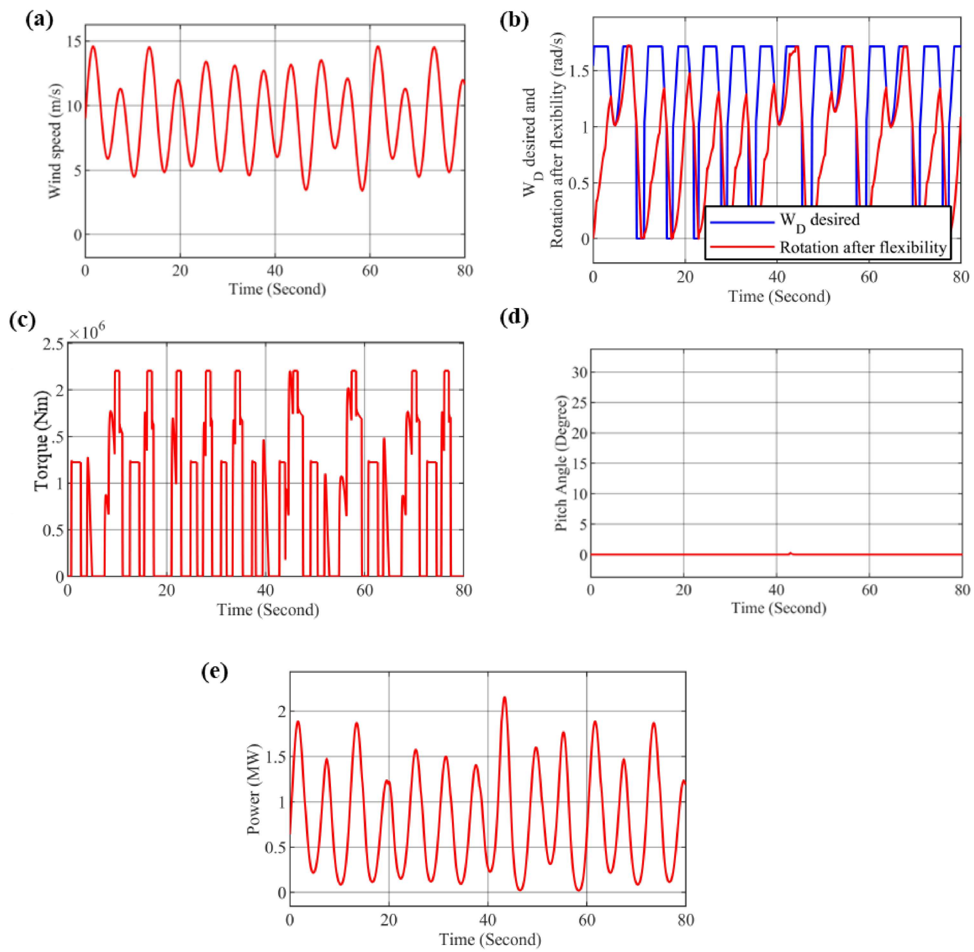

Figure 12. Simulated characteristics of the turbine for case 4; (a) Artificially created wind speed behavior, (b) Reference and practically obtained rotational speed of rotor with drive-train flexibility, (c) Generated Torque (d) Pitch angle of the rotor blades (e) Power curve.

In Figure 12(d), the pitch angle of the blade is illustrated for different operating regions of wind speed. However, it is observed that the pitch control mechanism does not effectively respond to the continuous anomalies in wind velocity. The pitch control mechanism requires a certain amount of time to adapt and react accordingly, also for a very short period of time wind goes beyond the threshold value, and it appears to never become activated in response to the wind velocity variations. Consequently, an erroneous behavior of consistently remaining at zero pitch angle is observed.

In Figure 12(e), the power extracted from the turbine is depicted, showcasing the fluctuating nature of the extracted power depending on the wind velocity. Furthermore, due to the fact that the wind speed never exceeds the cut-out speed, the braking system remains inactive and does not engage.

CONCLUSION

The objective of the assignment is to simulate a control system for pitch and torque control in a wind turbine, specifically to regulate the angular velocity of its rotor. Based on the wind conditions predefined in the problem descriptions, the control mechanism is regulated to obtain optimum power generation from the turbine. Additionally, the evaluation of the gearing system focused on its internal components and their impact on the drivetrain.

By comparing the graphs obtained from the lookup table with those generated by MATLAB-Simulink for pitch angle control, generator torque, wind speed, and power generated, it is evident that the estimated lookup table closely aligns with the theoretical predictions. The simulations accurately reflect the behavior of a real wind turbine in practical wind conditions. This demonstrates the efficacy of the control system in effectively regulating the wind turbine.

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Appendix

Appendix A: Look Up Table For The Steady State Values

Wind Speed (m/s)	Angular velocity (rad/s)	Pitch angle (in °)	Air Power (MW)
0-5.5	0	0	0
6	1.03	0	0.21
7	1.2	0	0.38
8	1.37	0	0.49
10	1.71	0	0.94
12	1.71	0	1.65
12.5	1.71	0	1.91
13	1.71	1.71	2
14	1.71	6.37	2
16	1.71	12.82	2
18	1.71	16.69	2
20	1.71	19.12	2
22	1.71	21.8	2
24	1.71	23.77	2
25	1.71	24.52	2
25 and onwards	0	30	0

The turbine's cut-in speed, which is the minimum wind speed required for power generation, is 5.5 m/s. Therefore, the turbine starts generating power once the wind speed reaches this threshold. The tip speed, which is the speed at the end of the turbine blade, reaches its maximum value at a wind velocity of 10 m/s. As a result, the angular velocity of the rotor remains constant beyond this wind speed boundary. At a wind velocity of 12.622 m/s, the control mechanism transitions from torque control to pitch control. Prior to this transition, the pitch angle remains at 0, and it starts functioning afterwards. Additionally, at the same wind velocity, the turbine achieves its maximum power output and maintains it by adjusting the pitch angle. The braking system comes into operation when the wind speed exceeds 25 m/s.

Appendix B: Gear System Designing

For the planetary gear must be calculated so the gears fit inside the system. As number of teeth and diameter for each part is related, the number of teeth and diameter for the planetary gear, and diameters for both regular stages.

$$N_R = N_s + 2 \times N_p$$
$$d = N \times Module$$

There is also a relation between calculated gear ratios (i) from the number of teeth with a difference between the planetary gear with its two stages, and the regular gears with only single stage. The gear ratios are calculated as,

$$i_{planet} = \frac{N_R}{N_s} + 1$$
$$i_{Single\ stage} = \frac{N_{Driver}}{N_{Driven}}$$

$$i_{total} = i_{planet} \times i_{single\ stage\ 1} \times i_{single\ stage\ 2}$$

1. For Planetary Gear System:

$$N_s = 30,$$
$$N_p = 47$$
$$N_R = N_s + 2 \times N_p = 124$$

Gear ratio at the planetary stage,

$$i_{planet} = \frac{124}{30} + 1 = 5.13$$

Diameters of the cogs ($Module = 16mm$),

$$d = N \times Module$$
$$d_s = 30 \times 16 = 480\ mm$$
$$d_p = 47 \times 16 = 752\ mm$$
$$d_R = 124 \times 16 = 1984\ mm$$

2. For Single Stage Gear System 1:

$$N_{driver} = 136,$$
$$N_{driven} = 30$$

Gear ratio at the planetary stage,

$$i_{single\ stage\ 1} = \frac{136}{30} = 4.53$$

Diameters of the cogs ($Module = 8\ mm$)

$$d = N \times Module$$
$$d_{driver} = 136 \times 8 = 1088\ mm$$
$$d_{driven} = 30 \times 8 = 240\ mm$$

3. For Single Stage Gear System 2:

$$N_{driver} = 102,$$
$$N_{driven} = 25$$

Gear ratio at the planetary stage,

$$i_{single\ stage\ 1} = \frac{102}{25} = 4.08$$

Diameters of the cogs ($Module = 5\ mm$)

$$d = N \times Module$$
$$d_{driver} = 102 \times 5 = 510\ mm$$
$$d_{driven} = 25 \times 5 = 125\ mm$$

Appendix C: Drive Train Felxibility Modelling

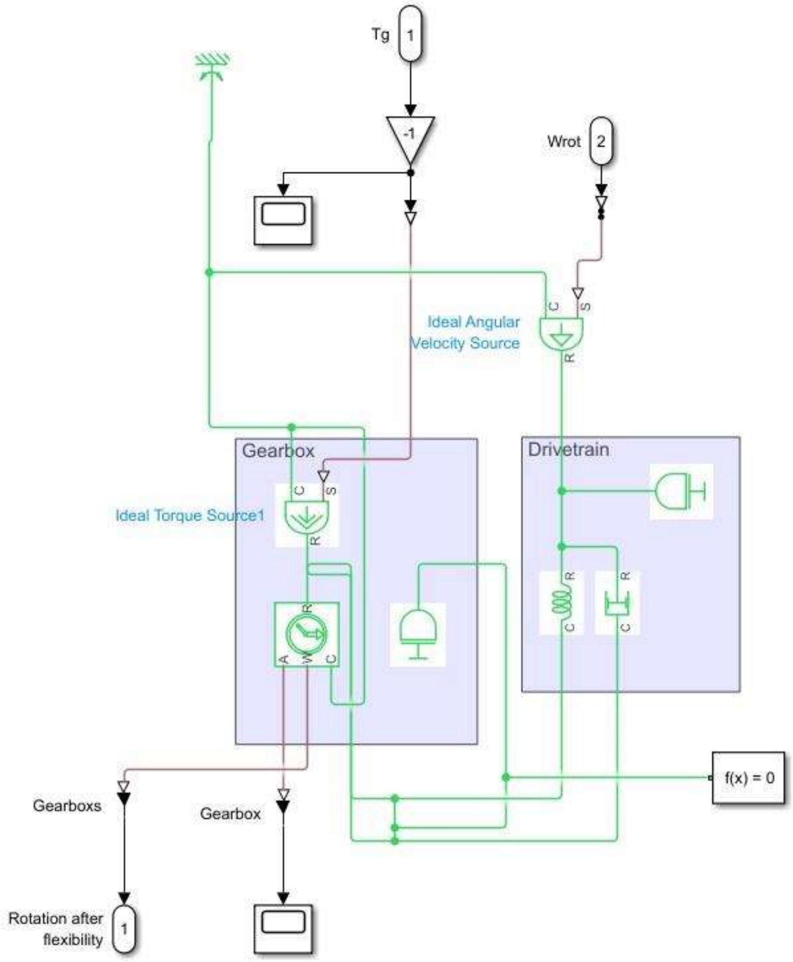
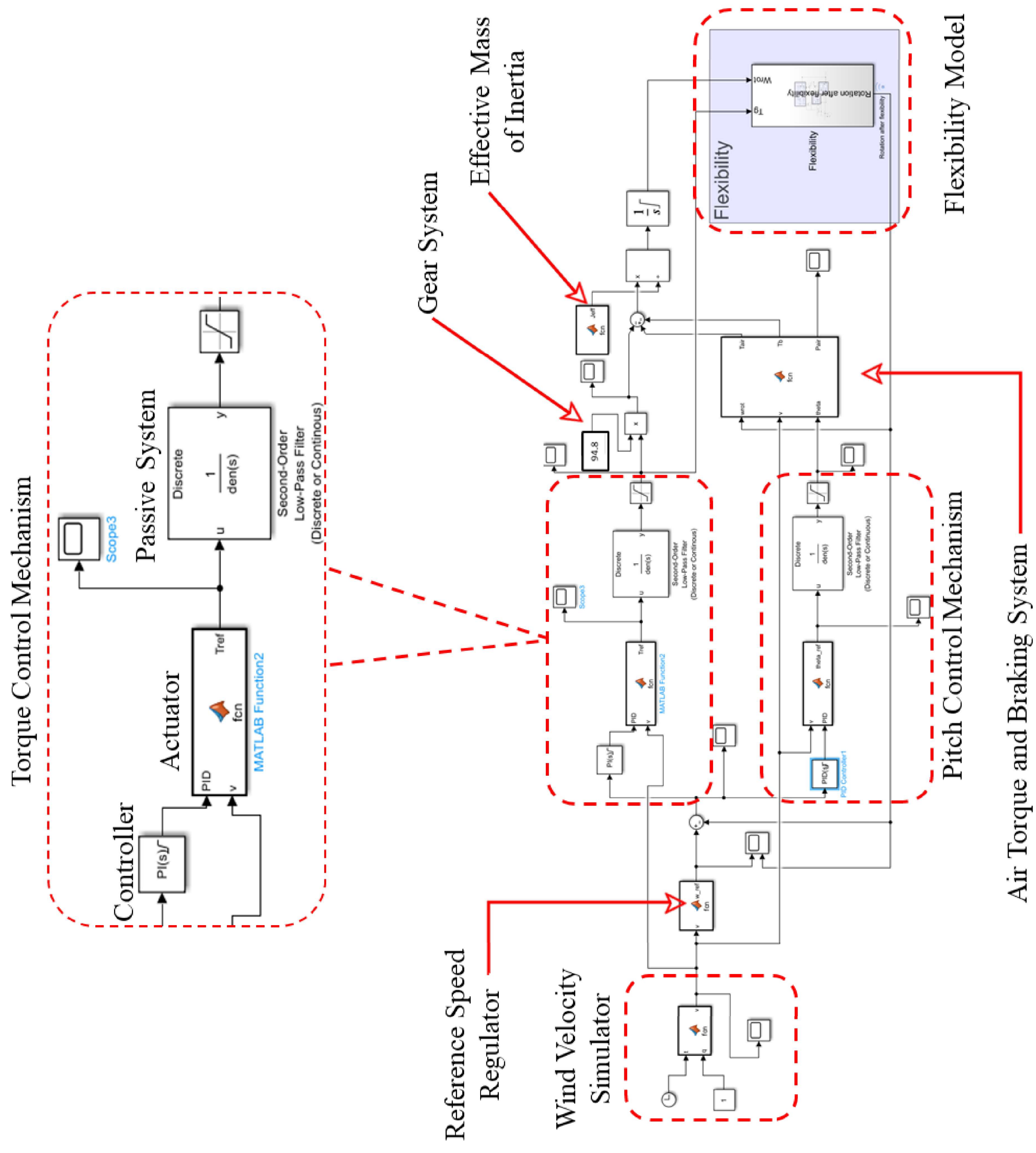


Figure 13. Drive train flexibility modelling for the system.

Appendix D: Overall Simulink/ Simscape model



Appendix E: MATLAB Coding

Wind Velocity Simulator

```
function v = fcn(t,q)

if q == 1 % Case 1: Ramping wind speed from 0 [m/s] to 25 [m/s]
v = 1e-100 + (25/60)*t;

elseif q == 2 % Case 2: Constant wind speed within each interval (A,B,C,D and E)
if t <= 10
v = 4;
elseif 10 < t <= 30
v = 7;
elseif 30 < t <= 40
v = 11;
elseif 40 < t <= 60
v = 15;
else
v = 27;
end

elseif q == 3 % Case 3: Jump in wind speed of 5 [m/s] in a short t (1 [s])
if t <= 25
v = 10;
elseif t > 25 && t <= 26
v = 5*(t-25)+10;
else
v = 15;
end

elseif q==4 % Case 4: Case to mimic practical condition
v=4*sin(2*pi/6*t)+9+sin(2*pi/10*t)+1*sin(2*pi/12*t);
else
v=0;
end
end
```

Reference Speed Regulator

```
function w_ref  = fcn(v)

R = 35; % Radius of rotor blades in meters
lambda = 6; % Optimal tip speed ratio
v_tmax = 60; % Maximum tip speed in m/s
v_max = 25; % Maximum allowed wind speed in m/s
theta = 0; % Pitch of the rotor blade in rad
rho = 1.2254; % Density of air in kg/m^3
Pmax = 2e6; % Max power of the windmill in W
pi = 3.14;

v_tip = lambda*v; % Tip speed
```

```
Cp = (0.44-0.012*theta) *sin((pi*lambda)/(12-0.3*theta)) -0.0015*lambda*theta;
Pair = 0.5*rho*v^3*pi*R^2*Cp;

% For Interval B, C and D
if v_tip > v_tmax
w_ref = v_tmax/R;
else
w_ref = (lambda*v)/R;
end

% For Interval E
if v > v_max
w_ref = 0;
end

% For Interval A
if Pair <= Pmax*0.1
w_ref = 0;
end
end
```

Torque Control Mechanism (Actuator)

```
function Tref = fcn(PID,v)

P_g_nominal = 2.1e6;
W_g_nominal = (1550*2*pi/60);
T_g_nominal = P_g_nominal/W_g_nominal;
Pmax = 2e+6;
R = 35;
ang = 0;
rho = 1.2254;
lamda = 6;
Tref = 0;

Cp = (0.44-0.012*ang) *sin((pi*lamda)/(12-0.3*ang)) - 0.0015 * lamda * ang;

if Cp <= 0
    Cp = 0;
end
Pair = (1/2)*rho * (v^3) * pi * (R^2) *Cp;

Tref = PID;

if Pair >= Pmax
    Tref = T_g_nominal;
end
if Tref >= 1.8*T_g_nominal
   Tref = 1.8*T_g_nominal;
end
if v >= 25
    Tref = 1.8*T_g_nominal;
end
if Tref <= 0
    Tref = 0;
```

```
end
end
```

Pitch Control Mechanism (Actuator)

```
function theta_ref = fcn(v,PID)

Pmax = 2e+6;
lambda = 6;
theta = 0;
rho = 1.225;
pi = 3.14;
R = 35;

Cp = (0.44-0.012*theta) *sin((pi*lambda)/(12-0.3*theta)) - 0.0015 * lambda *theta;

Pair = (1/2)*rho * (v^3) * pi * (R^2) *Cp;

if Pair >= Pmax
theta_ref = PID;
else
theta_ref = 0;
end
if theta_ref >= 30
theta_ref = 30;
end
if theta_ref <= 0
theta_ref = 0;
end
end
```

Effective Mass of Inertia (J_{eff})

```
function Jeff = fcn()

% Mass moment of inertia [kg*m^2]
J_rotor_blades = 2*10^6;
J_mainshaft_planetcarrier_planetgear = 200;
J_sunwheel_sunshaft_firstgear = 20;
J_firstpinion_intermediateshaft_secondgear = 10;
J_secondpinion_highspeedshaft_coupling_brage_generatorrotor = 220;

% Gearing
ig_tot = 94.684;
ig_planetary = 4.258;
ig_1gear =  4.615;

%Effective Mass moment of inertia
Jrot = J_rotor_blades + J_mainshaft_planetcarrier_planetgear;
J1gear = J_sunwheel_sunshaft_firstgear * ig_planetary^2;
J2gear = J_firstpinion_intermediateshaft_secondgear * ig_planetary^2 * ig_1gear^2;
Jgen = J_secondpinion_highspeedshaft_coupling_brage_generatorrotor * ig_tot;

Jeff = Jrot + J1gear + J2gear +Jgen;
```

Air Torque and Braking System

```
function [Tair,Tb, Pair] = fcn(wrot,v,theta)
% System constants
R = 35;
rho = 1.225;
if wrot <= 0.01
wrot = 0.01;
end
%theta = theta*180/pi;
% Calculate tip speed ratio
lambda = wrot*R/v;
% Calculate Power Factor
Cp = (0.44-(0.012*theta))*sin((pi*lambda)/(12-(0.3*theta))) - 0.0015*lambda*theta;
% Calculate wind power
Pair = (1/2)*rho*v^3*pi*R^2*Cp;
% Brake
if v >= 25
Tb = Pair/(wrot);
else
Tb = 0;
end
% Calculate torque generated by air
Tair = Pair/(wrot)-Tb;
end
```

Appendix F: 3D Modelling of Gear system and Rotor

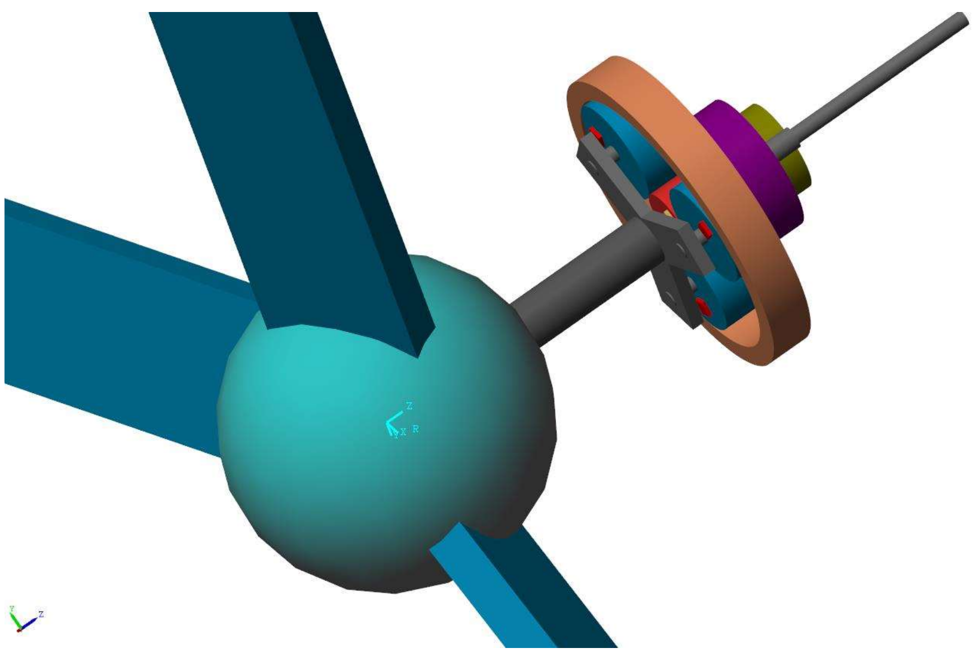


Figure 14. Simscape Modelling of entire gear system and rotor.

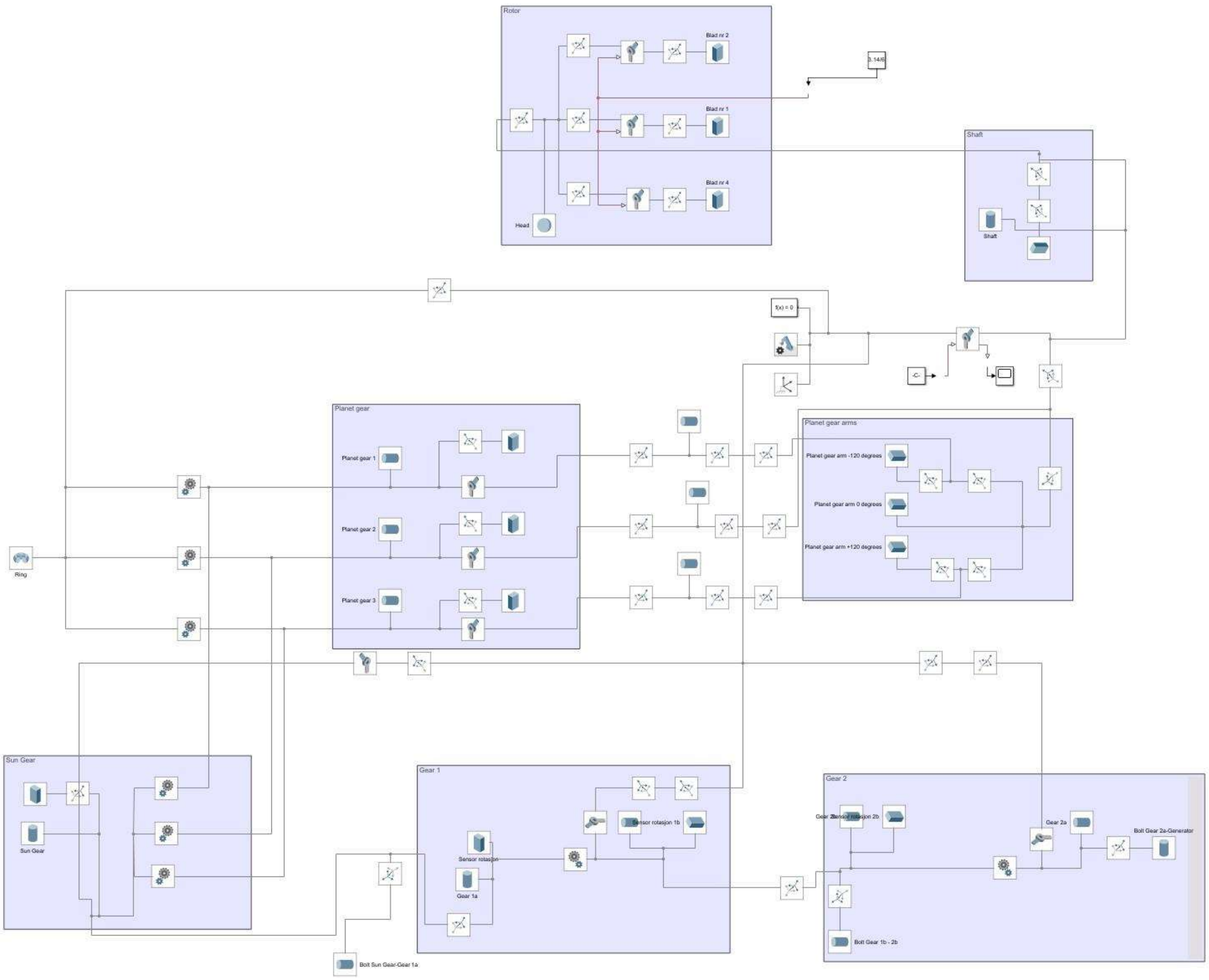


Figure 15. Gear system and rotor 3D modelling Circuit.