International Journal of Marine Engineering Innovation and Research, Vol. 10(3), Sept. 2025. 939-943
(pISSN: 2541-5972, eISSN: 2548-1479)

Design of Planetary Gear for Vertical Axis Marine Current Turbine

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(Received: 14 July 2025 / Revised: 25 July 2025 /Accepted: 28 August 2025 / Available Online: 25 September 2025)

***Abstract*— Indonesia as the archipelagic country in the world offers an abundant resource of ocean energy and one of them is marine current energy. One of the key concept of designing marine current energy converter is the gearbox. This paper discusses the design of gearbox using planetary gear to transmit power from 27RPM current turbine to a 5kW generator with 200RPM and 269Nm input torque. This paper presented the comparison of single stage planetary gear and double stage planetary gear from the aspect of maximum stress and the weight design. From the aspect the maximum stress, both of the gear design still under the limitation of the material stress, while based on the weight perspective the single stage has lower weight compared to double stage of planetary gear but from the size perspective the single stage has a larger diameter compared to the double stage planetary gear.**

Keywords— Marine current turbine, strength, planetary gear

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I. INTRODUCTION

Indonesia is one of the several countries with enormous marine renewable energy since its area is covered by the sea in more than 70%. Furthermore, there are more than 17,000 islands in Indonesia that connected by the sea which created many straits and provide huge potential for marine current energy. There are several in-depth research have been conducted to investigate the utilization of ocean current energy [1].

One of the character of current of the ocean is unidirectional of the current itself, it requires a turbine that able to adapt wherever the direction of the current, the rotation direction of the turbine must be in one way. Vertical axis turbine is one of a solution for this problem which connected to the gearbox [2]. A gearbox in marine which transmit energy from a turbine to a generator has different characteristic compared to ordinary gearbox, an ocean renewable enrgy turbine characteristic is a low speed but high torque. The design of a marine current turbine should be an increasing gearbox. This paper discusses an increasing gearbox which transmit power from a vertical axis which produces 1906Nm Torque and 29RPM which connected to a 5kW Permanent magnet generator with requires 269Nm torque and 200RPM rotation. Since the system is deployed in the ocean, the design of the gearbox must able to withstand the dynamic load of the ocean environment [3][4]. One of the feasible solution for this condition is the usage of planetary gearbox since its design is provides large torque to weight ratio compared to spur gear [5].

The design of the planetary gear should include the strength, and its weight must be considered also, this paper compares the strength and weight of single stage and two stage of planetary gear set for a 5kW marine current turbine.

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TABLE 1.
ORE POTENTIAL OF INDONESIA

Type	Theoretical (MW)		Practical (MW)
Wave	141,472	7,985	1,995
Tidal Current	287,822	71,955	17,989
Ocean Thermal	4,247,389	136,669	41,001
	4,676,683	216,609	60,985

II. METHOD

A. Stages of the Design

In order to design a planetary gearbox, there are several stages that needs to be carefully completed as shown by Figure 2. The design principles were based on the [7].

The overall design stages are complex process, in this paper only presents the strength and weight consideration even though the other stages or components were carried out. The detail of the design also summarised due to the limitation of the publication.

B. Stages of the Design

To harness the energy from the turbine, the gear set must be able to increase the rotation and reduce the torque generated from the turbine. The system designed the carrier as the input, sun gear as output and ring gear for the stationary to produce increasing rotational speed and decrease torque as shown by Figure 3.

C. Finite Element Method

The Finite Element Method (FEM) is a numerical technique used to approximate the solution of complex engineering problems involving structures, fluids, heat transfer, and other physical phenomena. It works by dividing a large and complex domain into smaller, simpler parts called finite elements, which are connected at discrete points known as nodes. Within each element, governing equations based on physical laws—such as equilibrium, compatibility, and material behaviour—are formulated and assembled into a global system of equations. By solving this system, FEM provides approximate values for quantities such as displacement, stress, temperature, or velocity throughout the domain. FEM is widely used in structural analysis, mechanical design, and simulation because of its ability to handle complex geometries, varying material properties, and boundary conditions with high accuracy [6] [7].

D. Von Mises Stress

Von Mises stress, also referred to as equivalent stress, is a scalar quantity used to represent the onset of plastic failure in materials subjected to multiaxial stress states [7]. It does not correspond to a physical stress component in a specific direction but rather serves as an equivalent value that facilitates direct comparison with the yield strength obtained from a uniaxial tensile test.

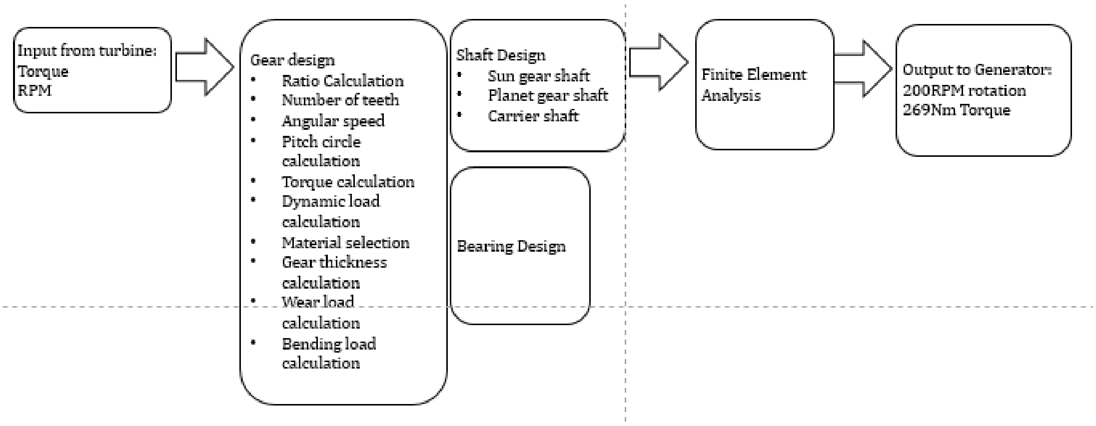

Figure. 2. Planetary Gearbox Design Stages

TABLE 2.
CHARACTERISTIC OF PLANETARY GEAR

Sun Gear	Carrier	Ring Gear	Speed	Torque	Rotation
Input	Output	Stationary	Reduction	Increase	Same with input
Stationary	Output	Input	Slight reduction	Increase	Same with input
Out put	Input	Stationary	Increase	Decrease	Same with input
Stationary	Input	Output	Slight Increase	Decrease	Same with input
Input	Stationary	Output	reduction	Increase	Contrary with input
Out put	Stationary	Input	Increase	Decrease	Contrary with input

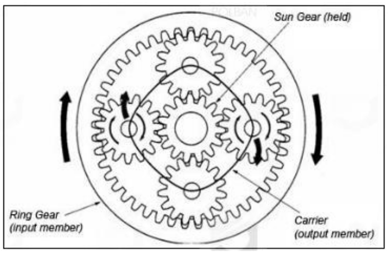

Figure. 1. Input and output Gear

The concept is derived from the distortion energy theory, which postulates that yielding occurs when the distortion energy in the material exceeds the distortion energy at yield under uniaxial loading conditions [8].

$$\frac{1}{6}[(\tau_{11}-\tau_{22})^2+(\tau_{22}-\tau_{33})^2+(\tau_{33}-\tau_{11})^2 + 6(\tau_{12}^2+\tau_{23}^2+\tau_{13}^2)] = k^2 \quad (1)$$

The constant is determined experimentally, and the stress tensor represents the state of stress in the material. Typically, this constant is obtained through uniaxial stress tests, where the equation simplifies accordingly

$$\frac{\tau_y^2}{3} = k^2 \quad (2)$$

If τ_y reaches the value of tension elastic limit, S_y, equation (2) becomes

$$\frac{S_y^2}{3} = k^2 \tag{3}$$

By substitutes equation (2) to equation (1) we get the common formula for Von Misses stress that we were used in FEM calculations:

$$\frac{1}{2}[(\tau_{11} - \tau_{22})^2 + (\tau_{22} - \tau_{33})^2 + (\tau_{33} - \tau_{11})^2 + 6(\tau_{12}^2 + \tau_{23}^2 + \tau_{13}^2)] = \tau_v^2 \tag{4}$$

The von Mises yield criterion rewritten as:

$$\tau_v \geq S_y \tag{5}$$

When, the von Mises Stress is greater than the value of simple yield limit stress, the material will expect to yield.

E. Von Mises Stress

Once the strain tensor [ε] is obtained, the stress tensor [σ], which includes shear stress components such as $\tau_{xy}, \tau_{yz}, \tau_{zx}$ is computed using the constitutive material law. For linear elastic materials, this relationship typically follows Hooke's Law in three dimensions and is expressed in matrix form as:

$$[\sigma] = [D].[\varepsilon] \tag{6}$$

Where:

$[\sigma]$: Stress tensor
$[\varepsilon]$: Strain tensor from displacement between node
$[D]$: Stiffness material matrix

Here, [D] is the material stiffness matrix, which incorporates mechanical properties such as the Young's modulus and Poisson's ratio. This matrix defines how the material responds to strain and enables the FEM model to compute all components of stress, including normal and shear stresses, under complex loading conditions. This formulation allows FEM to simulate realistic material behavior beyond what is possible with basic analytical expressions.

F. Planetary Gear

A planetary gear, also known as an epicyclic gear train, is a gear system consisting of one or more outer gears (planet gears) revolving around a central gear (sun gear). It is typically used in mechanical systems where high torque and compact design are required. The planetary gear system allows for multiple configurations that can achieve gear reduction, torque multiplication, or even reversal of output direction [9].

A planetary gear set consists of four main components: a sun gear, planet gear, a ring gear, and a planet carrier. The sun gear is the central gear that drives the planet's gears, which rotate around it and mesh with the internal teeth of the ring gear. These planet gears are supported by the carrier, which holds them in position and transfer's motion. This compact, coaxial arrangement enables efficient torque transmission and smooth load distribution.

The gear ratio of a planetary gear system depends on the number of teeth on the sun gear (N_S) and ring gear (N_R).One commonly used configuration is where the sun is the input, the ring is fixed, and the carrier is the output.

$$Gear\ Ratio = 1 + \frac{N_R}{N_S} \tag{7}$$

This implies that for every full revolution of the sun gear, the carrier will rotate a fraction depending on the gear ratio. The higher the ring-to-sun ratio, the lower the output speed and the higher the torque.

For the system to mesh properly, the gear teeth relation must satisfy:

$$N_R = N_S + 2N_P \tag{7}$$

Where:

N_P : Number of planetary gear teeth

This relationship ensures that the planet gears mesh simultaneously with both the sun and ring gears, maintaining geometrical symmetry and even load distribution.

This section will discuss the comparison of single stage and double stage of planetary gear. The design of the single stage and double stage planetary gear can be shown by Figure 4

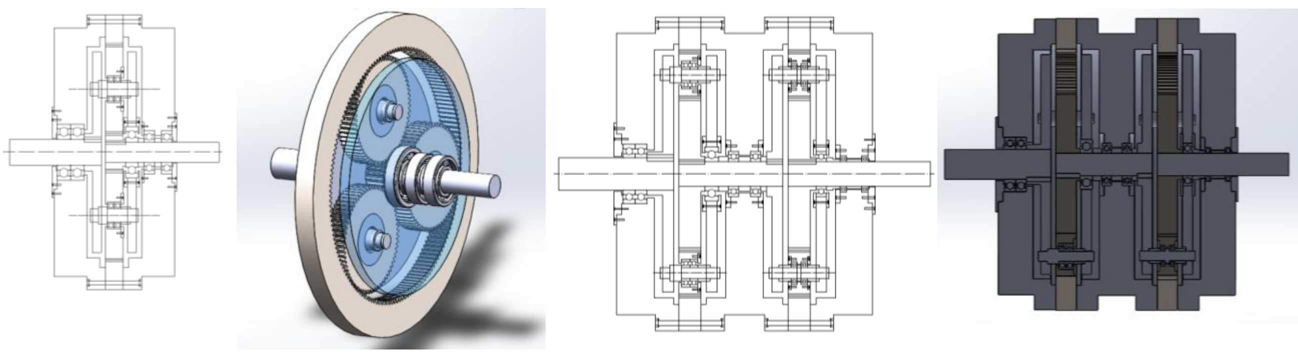

Figure 2. Single Stage and Double stage planetary gear

TABLE 3.
MATERIAL USED FOR THE PLANETARY GEAR

Part	Material	AISI/SAE	Treatment	Strength (Mpa)		Hardness
				Yield	tensile	(BHN)
Ring	Alloy Steel	4140	normalized ± 1600F	675	1020	302
Sun	Alloy Steel	4340	normalized ± 1600F	860	1,280	363
Planet	Alloy Steel	4150	normalized ± 1600F	896	1,340	321
Carrier	Alloy Steel	4340	normalized ± 1600F	860	1,280	363

A. *Single Stage Planetary Gear*

The specification of the design results of single stage planetary gear can be shown by Table 4. In order to make sure the strength of the gear, a finite element analysis was carried out and the results can be shown by Table 5 where the highest stress is the contact feed teeth of planet gear 37.728 Mpa, it can be concluded that the maximum stress is still under allowable stress.

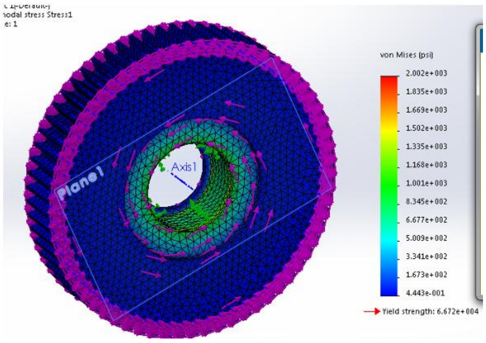

Figure 3. Maximum stress of planet Gear single stage

TABLE 5.
DOUBLE STAGE PLANETARY GEAR SPECIFICATION

Part	Mechanism	RPM	Torque (Nm)	Nt	d (mm)	P (modul)	b (mm)
Carrier	Turbine input	29	1906				
Planet	Rotating	253	74	27	85.73	8	40.6
Ring		Fix Stationary		143	454	8	40.6
Sun	Output to Generator	77	731	89	282.6	8	40.6

B. *Double Stage Planetary Gear*

As part of the comparison, the same procedure undertaken for the double stage of planetary gear. A FEM analysis carried out to reveal the maximum stress occurred in the design, and the maximum stress was 100.112Mpa which occurred in the second stage of contact feed teeth of the sun gear as shown by Table 6 for stage 1 and Table 7 for stage 2.

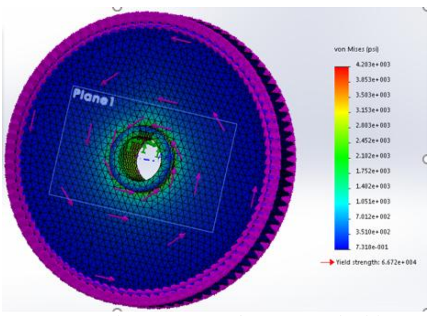

Figure 4. Maximum stress of Sun Gear double stage

C, *Weight Comparison*

Based on the 3D design and weight calculation along with all the component of the system both single stage and double stage of planetary gear, the weight of the single stage planetary gear is 65.65kg while the double stage of planetary gear is 130.33kg, it can be concluded that the weight of the single stage of planetary gear is lower compared to the double stage gear.

TABLE 6.
DOUBLE STAGE PLANETARY GEAR STRESS

	Double Stage (Stage 1)				
	Shaft & Planet gear tooth (Mpa)	Status	Shaft & sun gear tooth (Mpa)	Allow Stress (Mpa)	Status
Contact Feed shaft	9.467	Pass	28.979	603	Pass
Contact feed teeth	29.089	Pass	100.112	603	Pass

TABLE 7.
DOUBLE STAGE PLANETARY GEAR STRESS

	Double Stage (Stage 2)				
	Shaft & Planet gear tooth (Mpa)	Status	Shaft & sun gear tooth (Mpa)	Allow Stress (Mpa)	Status
Contact Feed shaft	14.596	Pass	13.61	603	Pass
Contact feed teeth	30.564	Pass	46.154	603	Pass

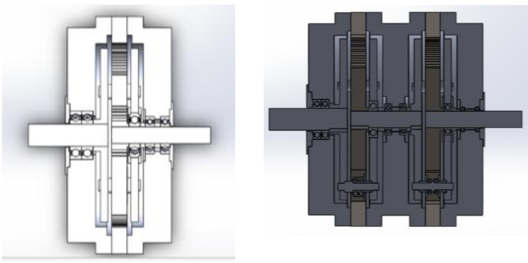

Figure. 5. 3D model of single stage (left) and double stage (right)

IV. CONCLUSION

Based on analysis which have been presented in the previous chapter, it can be concluded that the design of single stage planetary gear is sufficient to withstand the 1906Nm Input torque, 29 RPM input and transmit to a generator with 200RPM and 269Nm minimal torque, and the weight of the single stage is 65.65kg compared to 130.33kg of double stage which is lighter. Both of the comparison lead to a conclusion to single stage planetary gear for marine current energy generator.

ACKNOWLEDGEMENTS

The authors would like to thanks to Sepuluh Nopember Institute of Technology for the Software

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