DESIGN OF PITCH GEARBOX FOR 4MW-CLASS WIND TURBINE

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ABSTRACT

The pitch gearbox for 4MW-class wind turbine is comprised of a triple-gear planetary system. It is a component that keeps wind turbine blades rotating at a constant speed by adjusting their angles, and should have a life span of at least 20 years.

This study analyzed the design stability and vibrational characteristics of pitch gearbox components for 4MW-class wind turbine. Design stability was secured by performing stress analysis of the housing and carriers under actual loads and extreme loads, specified in Load Duration Distribution (LDD), through finite element analysis. The natural frequencies and mode shapes of the pitch gearbox were also analyzed. There was found to be significant deformation of the third planetary gear for the first and second mode shapes, of the second planetary gear for the third and fourth mode shapes, and of the first planetary gear for the fifth and sixth mode shapes. Critical speed was assessed based on analysis of excitation source in relation to mass imbalance and excitation source. In the range of operation of the pitch gearbox, that is, 260rpm – 320rpm, critical speed did not exist.

Keywords: wind turbine, Load Duration Distribution (LDD), pitch gear box, planetary gear, critical speed.

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1. INTRODUCTION

Recently, the world has seen a heightened interest in the development of renewable energy due to global warming, depletion of fossil fuels, and environmental pollution. In particular, wind energy is being actively studied as it is more technologically compatible and affordable compared to other sources of renewable energy.⁽¹⁾

A wind turbine system is composed of a blade, tower, gearbox, power generator, coupling, and various controllers. Only products with high reliability can enter the market as a life span

of at least 20 years is expected. With wind turbine systems growing larger, there is a need to develop lightweight, reliable gearboxes that produce less vibration and noise.⁽²⁻⁴⁾

Among the various elements of a wind turbine, the pitch system maintains an appropriate output level by sending signals to the blade control device when power output measurements indicate an increase or decrease is appropriate. Through blade angle control, the system starts up for the rotor to gain sufficient torque at wind speeds above the cut-in wind speed, maintains a constant rated output at wind speeds above the rated wind speed, and stops during high or low wind speeds. The pitch drive adjusts blade angles so that blades rotate at a constant speed according to wind speed. Among pitch drive components, the gearbox must meet the same requirements as that of the gear train: life span of at least 20 years, small size, and light weight. The volume and weight of planetary gears in the pitch drive should be reduced to satisfy performance requirements pertaining to dynamic properties.⁽⁵⁻⁶⁾

To evaluate the design safety of a pitch gearbox for 4MW-class wind turbines, this study used finite element analysis on pitch gearbox components to calculate the housing and carrier stress under actual loads and extreme loads specified in LLD. Vibration analysis was performed on the pitch gearbox, designed for 4MW-class wind turbines driven by triple-gear planetary systems. The commercial software MASTA was used to design a vibration model of the pitch gearbox. The natural frequency of the pitch gearbox was analyzed, and excitation source was examined while taking into account mass imbalance and gear mesh frequency of the planetary gear system. Critical speed analysis was performed for the excitation source at 260rpm – 320rpm, which was the range of operation of the pitch gearbox.

2. STRUCTURAL ANALYSIS OF PITCH GEARBOX COMPONENTS

2.1. Finite Element Analysis of Housing and Carrier

2.1.1 Finite element analysis of housing

The housing of the pitch gearbox for wind turbines is composed of a motor flange, ring gears of a triple-gear planetary system, flanges, and main housing. The main housing is mounted to the hub of the wind turbine, and is subject to the load of the bearing that supports the output shaft. The flange of the housing is joined to the ring gear using bolts, and plays the role of fixing the ring gear and receiving torque delivered through the ring gear. Finite element analysis was performed in ANSYS 19 to analyze the stress of the housing while taking into account torque and bearing load.

Figure 1 Structure of gear train housing for blade control

Figure 2 shows the FE model of the housing used in finite element analysis. Bolts used to join components were omitted to simplify the model and to minimize problems that may arise

from contact. A hex-dominant mesh having 6,090,630 nodes and 1,699,703 elements was generated for the FE model.

Figure 2 FE model of housing

The materials used in the pitch gearbox housing were structural steel alloy SCM440 and cast iron FCD450 and FCD600. The mechanical properties of each material involved in finite element analysis are presented in Table 1.

Table 1 Mechanical properties of materials for finite element analysis of housing

Component	Material	Young's Modulus [GPa]	Poisson's Ratio	Yield Strength [MPa]
Motor Flange	FCD450	173	0.3	250
1st Ring Gear	SCM440 (265HV,MQ)	207	0.3	835
2nd Ring Gear	SCM440 (265HV,MQ)	207	0.3	835
3rd Ring Gear	SCM440 (265HV,MQ)	207	0.3	835
1st Flange	FCD450	173	0.3	250
2nd Flange	FCD450	173	0.3	250
Main Housing	FCD600	173	0.3	370

Figure 3 shows the boundary conditions for finite element analysis of the housing. The gear train housing for blade control is installed at the hub of the wind turbine. The main housing is inserted into the mount at which the housing is installed, and fastened using bolts. The inner surface of the mount is in contact with the main housing. To ensure boundary conditions similar to the actual conditions of housing installation, fixed support was applied to bolt holes connecting the main housing, and compression only support to the contact area between the inner side of the mount and main housing.

Figure 3 Boundary conditions of finite element analysis of housing

The load conditions shown in Figure 4 were applied to calculate torque acting on the ring gear and stress due to bearing loads acting on the main housing from bearings supporting the output shaft. Torque acting on the ring gear is produced when it meshes and rotates with the planetary gear. As such, the method of applying torque on the ring gear teeth may be the most realistic, but this can cause excessive stress on tooth roots. To minimize stress concentration on tooth roots, torque was applied to bolt holes of bolts connecting the ring gear to the flange. Using the remote point function in ANSYS, torque was applied to various bolt holes about the center of each ring gear. In addition, bearing loads were applied using the load function. Finite element analysis was performed for extreme loads and active times specified in LDD, and the load conditions are presented in Table 2. The equivalent stress and displacement distribution of each component in finite element analysis are shown in Figure 5 and Figure 6. Table 3 presents the equivalent stress and yield strength of each component.

Figure 4 Load conditions of finite element analysis of housing

The results of finite element analysis revealed that the main housing with an equivalent stress of 196.04MPa had the highest stress among components. The main housing was made of cast iron FCD600, and had a yield strength and safety factor of approximately 370MPa and 1.88, respectively.

Table 2 Extreme load acting on housing

Load Condition	Value
1st Ring Gear Torque	1,186.1136 [Nm]
2nd Ring Gear Torque	7,726.908 [Nm]
3rd Ring Gear Torque	36,067.3418 [Nm]
Bearing Load 1	264,770 [N]
Bearing Load 2	668,440 [N]

Table 3 Equivalent stress and safety factor of each housing component

Component	Equivalent Stress [MPa]	Yield Strength [MPa]	Safety Factor
Motor Flange	0.83	250	301.2
1st Ring Gear	8.56	835	97.5
2nd Ring Gear	25.36	835	32.9
3rd Ring Gear	109.23	835	7.6
1st Flange	6.03	250	41.4
2nd Flange	18.95	250	13.1
Main Housing	196.04	370	1.88

(a) Motor Flange

(b) 1st Ring Gear

(c) 2nd Ring Gear

(d) 3rd Ring Gear

Figure 5 Equivalent stress distribution based on finite element analysis of housing

2.1.2. Finite element analysis of carriers

The pitch gearbox for wind turbines is composed of a triple-gear planetary system, which consists of a sun gear, ring gear, planetary gear, and a carrier supporting the planetary gear. The rotational force of the motor is given as input to the first sun gear with all ring gears in the fixed state, and this is delivered as output from the first carrier. The first carrier is connected to the second sun gear, and the second carrier to the third sun gear. The final output is produced through the third carrier. In the case of sun gear input, fixed ring gears, and carrier output, the carrier rotates in the same direction as the sun gear, and decreases in speed. Due to the reaction of the planetary gear, the carrier is subject to a heavy load that places it at a higher risk of damage [6].

This study conducted finite element analysis on carriers using ANSYS 19. Figure 6 shows the FE models of each carrier for finite element analysis. The carriers are connected to splines via higher-order sun gears and output shaft. The splines were omitted to simplify the carrier models and to eliminate stress concentration. A hex-dominant mesh was generated, with the first carrier having 3,148,478 nodes and 822,298 elements, the second carrier having 9,092,536 nodes and 2,475,709 elements, and the third carrier having 25,477,355 nodes and 6,850,776 elements.

(a) 1st Carrier

(b) 2nd Carrier

(c) 3rd Carrier

Figure 6 FE model of carriers

The materials used in carriers were carbon steel S45C and cast iron FCD450 and FCD600. The mechanical properties of each material involved in finite element analysis are presented in Table 4.

Table 4 Mechanical properties of materials for finite element analysis of carriers

Component	Material	Young's Modulus [GPa]	Poisson's Ratio	Yield Strength [MPa]
1st Carrier	S45C	207	0.3	490
2nd Carrier	FCD450	173	0.3	250
3rd Carrier	FCD600	173	0.3	370

As boundary conditions for finite element analysis of carriers, fixed support was applied to the part connecting the first and second carriers to higher-order sun gears, and to the part connecting the third carrier to the output shaft. No other contact conditions were applied to the planetary pin since it was integrated with the first carrier. On the other hand, the bonded contact condition was applied to contact between planetary pins and the second and third carriers. The load conditions of finite element analysis are shown in Figure 7. The carriers of the planetary gear system support torque delivered through planetary pins from planetary gears. By setting the planetary pins as remote points, torque was applied at the center of rotation of carriers.

Figure 7 Load conditions in finite element analysis of carriers

Finite element analysis was performed for extreme loads specified in the LDD of the gear train for blade control, and the load conditions applied to each carrier are shown in Table 5.

Table 5 Extreme loads acting on carriers

Load Condition	Value
1st Carrier Torque	1,350.852 [Nm]
2nd Carrier Torque	8,899.733 [Nm]
3rd Carrier Torque	44,967.075 [Nm]

Figure 8 shows the equivalent stress and displacement distribution based on the results of finite element analysis, while Table 6 presents the safety factor in relation to equivalent stress and yield strength. The equivalent stress was about 94.5 MPa for the first carrier, 157.41 MPa for the second, and 234.1 MPa for the third. The safety factor was about 5.18 for the first carrier, 1.58 for the second, and 1.58 for the third.

Table 6 Equivalent stress and safety factor from finite element analysis of carriers

Component	Equivalent Stress [MPa]	Yield Strength [MPa]	Safety Factor
1st Carrier	94.5	490	5.18
2nd Carrier	157.41	250	1.58
3rd Carrier	234.1	370	1.58

Figure 8 Result of finite element analysis of carriers

3. VIBRATION ANALYSIS OF PITCH GEARBOX

3.1. Planetary Gear System

The pitch gearbox for 4MW-class wind turbines was composed of a triple-gear planetary system. With the ring gear in the fixed state, the power from the electric motor is entered into the first sun gear and sent out through the first carrier. The first carrier is connected to the second sun gear, and the second carrier to the third sun gear. The output shaft is driven by the output produced through the third carrier.

Table 7 shows the specifications of the planetary gear system of the pitch gearbox. The reduction ratio was 1/8.2 for the first gear, 1/7.588 for the second, and 1/5.05 for the third. The total reduction ratio was approximately 1/314.22, and the rotating speed of 288.61 rpm of the input axis is delivered through the planetary gears and reduced to 0.918 rpm at the output shaft.

Table 7 Specifications of planetary gear system

Stage		1st	2nd	3rd
Number of Teeth	Sun	15	17	19
	Planet	46	47	28
	Ring	108	112	77
Face Width	Sun	35	45	65
	Planet	20	40	60
	Ring	28	52	70
Module		2	2.5	4.5
Pressure Angle		20°	27°	27°
Reduction Ratio		8.2	7.588	5.05
Total Reduction Ratio		314.22		

Table 8 shows the materials of each gear of the planetary gear system. SCM420 was used for the sun gear and planetary gear was, SCM440 for the ring gear, S45C for the first carrier, and FCD materials for the second and third carriers. Table 9 gives the mechanical properties of each material.

Table 8 Materials of planetary gear system

Stage		1st	2nd	3rd
Material	Sun	SCM420 25 HRC	SCM420 25 HRC	SCM420 25 HRC
	Planet	SCM420 25 HRC	SCM420 25 HRC	SCM420 25 HRC
	Ring	SCM440 265 HV	SCM440 265 HV	SCM440 265 HV
	Carrier	S45C	FCD450	FCD600

Table 9 Materials of planetary gear system

	Young's Modulus [GPa]	Yield Strength [MPa]	Poisson's Ratio	Density []
SCM420 25 HRC	207	790	0.3	7800
SCM440 265 HV	207	835	0.3	7800
S45C	207	490	0.3	7800
FCD450	173	250	0.3	7200
FCD600	173	370	0.3	7200

3.2. Vibration Modeling of Pitch Gearbox

Figure 9 shows the vibration model of the pitch gearbox. The vibration model was designed using the commercial software MASTA 10.2.

Figure 9 Analysis model of pitch drive

3.3. Analysis of Vibrational Properties of Pitch Gearbox

3.3.1 Analysis of natural frequency of pitch gearbox

The vibration system of the pitch gearbox for wind turbines can be expressed as follows.

$$[M]\ddot{q} + [C]\dot{q} + [K]q = 0 \quad (1)$$

Here, $[M]$ is the system mass matrix, $[C]$ is the system damping matrix, and $[K]$ is the system stiffness matrix. q is the degree of freedom of the system.

To create conditions similar to those of actual rotation, the vibration analysis was performed under a rotating speed of approximately 288.61 rpm and torque of 56.55 kNm at the input axis. Figure 10 shows the natural frequency of the pitch gearbox. The first natural frequency occurred at 231.75Hz, and the second to sixth at 235.73Hz, 566.89Hz, 586.65Hz, 642.81Hz, and 643.70Hz, and the second to sixth at 235.73Hz, 566.89Hz, 586.65Hz, 642.81Hz, and 643.70Hz.

Figure 10 Natural frequency of pitch drive

Figure 11 Mode shape of pitch drive

Figure 11 shows the mode shapes of the pitch gearbox. There was significant deformation of the third planetary gear for the first and second mode shapes, of the second planetary gear for the third and fourth mode shapes, and of the first planetary gear for the fifth and sixth mode shapes.

3.3.2 Analysis of excitation source of pitch gearbox

The pitch gearbox has excitation sources due to factors such as mass imbalance and gear transmission error. The main causes of gear transmission error include machining error at the gear tooth surface and system deformation under excessive loads. The gear mesh frequency of the gear system is obtained by multiplying the rotating speed of the gear body with the number of teeth. Mass imbalance indicates the presence of eccentric mass due to machining error, and is the same as the rotating speed of the axis.⁽⁴⁾

A wind turbine system produces vibration and noise when the excitation frequency matches the natural frequency. With excitation frequency expressed as $\gamma_i, i = 1, 2, \dots, N$ and natural frequency of the wind turbine system as $\lambda_i, i = 1, 2, \dots, N$, resonance occurs at

$$\gamma_i = \lambda_i \quad (2)$$

If $\gamma_i = c_i \omega_{cr}$, the critical speed becomes

$$\omega_{cr} = \lambda_i / c_i \quad (3)$$

Here, c_i is the coefficient of excitation frequency.⁽²⁾

Table 10 shows the excitation frequency ratio in relation to mass imbalance and gear mesh frequency of the pitch gearbox. Here, X is the input rotating speed, which was set as 288.61 rpm for the first sun gear.

Table 10 Excitation frequency ratio of pitch drive

Excitation Source	Excitation Frequency Ratio		
		Sun	Planetary
Mass Imbalance	1st Stage	1X	0.122X
	2nd Stage	0.122X	0.016X
	3rd Stage	0.016X	0.0032X
Gear Mesh Frequency	1st Stage	15X	
	2nd Stage	2.073X	
	3rd Stage	0.305X	

3.3.3 Critical speed analysis of pitch gearbox

The range of operating speed of the pitch gearbox was set as 260 – 320 rpm, and critical speed analysis was performed while taking into account the mass imbalance and gear mesh frequency presented in Table 10. Figure 12 and Figure 13 show the results of critical speed analysis in relation to mass imbalance and gear mesh frequency, respectively. According to the results, resonance due to mass imbalance and gear mesh frequency did not occur in the range of operating speed of 260 – 320 rpm.

Figure 12 Campbell diagram of mass imbalance

Figure 13 Campbell diagram of gear mesh frequency

4. CONCLUSION

This study examined the design stability and vibrational characteristics of components of a pitch gearbox, composed of a triple-gear planetary system, for 4MW-class wind turbines. The following conclusions were derived.

- Through finite element analysis of pitch gearbox components, design stability was secured by calculating and analyzing stress of housing and carriers under the extreme loads and active times specified in LDD.
- The natural frequencies and mode shapes of the pitch gearbox were analyzed. It was found that there was significant deformation of the third planetary gear for the first and second mode shapes, of the second planetary gear for the third and fourth mode shapes, and of the first planetary gear for the fifth and sixth mode shapes.
- Excitation sources due to mass imbalance and gear mesh frequency of the pitch gearbox were analyzed, and critical speed analysis was performed. Critical speed did not exist in the range of operating speed of 260rpm – 320rpm.

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