Mechanical Control Methods in Wind Turbine Operations for Power Generation

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***Abstract*—This paper investigates various mechanical control methods used for controlling wind turbine (WT), which play an important role in WT power generating stations. The gear control, blade pitch angle control and yaw control methods are the major content of this paper. A simplified methodology is used for gear design to obtain the speed of driven gear in a practical case. Some important issues related to gear system are investigated such as, scoring, corrosion, spalling, gear noise etc. The classifications of wind turbines and its blade pitch angle control regions are identified for fluctuating wind speed. Furthermore, the importance of yaw control to extract wind power is explained and wind speed/ directions sensor is discussed for yaw controlling.**

***Index Terms*—wind turbines, gear ratio, pitch angle, yaw system, angular velocity**

I. Introduction

Since last two decade, there have been many energy related crises experienced due to depletion of fossil fuels e.g. coal and oil; and environmental concerns resulting from the fuel consumption. The most effective utilization of renewable energies, such as wind and solar energy etc. are highly expected instead of the fossil fuels [1]. For generating the electric power from wind power, some mechanical controls are necessary to adjust the wind turbine structure, which includes gear ratio, pitch angle of turbine blades to control the fluctuations in the generated electric power output and movement control of yaw with the wind potential variation. When gear teeth absolutely fit together or interlock in systematic manner, they are said to be in mesh. Gears in mesh are capable of transmitting force and motion from one gear to another. A gear transmitting the force or motion is called the driver gear and the gear connected to the driver gear is called driven gear [2]-[3].

Wind energy potential is not always constant and therefore the generated electric power of a wind turbine generator (WTG) is fluctuate [4]-[6]. The aim of this paper is to study the behaviour of the WTG operated under variable speed with pitch angle controlling system, under turbulent winds also [7].

Another control mechanism is yaw movement ability for directing the wind turbine with the direction of the wind. It is found by a wind direction sensor, which helps to actuate the yaw control. An important function of the pitch control is to prevent the WT from over speeding in order not to exceed its mechanical limits and generator power rating [8]-[9].

In this paper, the endeavors have been made to present the comprehensive mechanical control methods used for controlling the wind turbine. The gear control methods have been identified and presented [10]-[14].

II. Wind Energy Conversion System

According to axis alignment, the wind energy conversion system (WECS) can be broadly categorized in two types as:

(a) Horizontal axis wind turbines (HAWT) can be sub-classified into following categories as:
- Dutch type WT
- Multi blade WT
- High speed propeller WT

(b) Vertical axis wind turbines (VAWT) can be sub-classified into following categories as:
- Savonius rotor type WT
- Darrieus rotor type WT

The complete wind turbine structure is shown in Fig. 1. Generated power of WT is calculated by the following equation as:

$$P_m = \frac{1}{2}\rho\pi R^2 C_p(\lambda,\beta)V_w{}^3 \tag{1}$$

where ρ is air density, A is turbine swept area and V_w is wind speed. The power coefficient is represented by C_p which is non linear in nature. it depends on tip speed ratio λ and blade pitch angle β as $C_p(\lambda,\beta)$.

The tip speed ratio λ is defined as:

$$\lambda = \frac{\Omega_t R}{V_w} \tag{2}$$

where Ω_t is the rotor speed, R is the radius and V_w is wind velocity and produced torque T_m of the wind turbine can be calculated as:

$$T_m = \frac{P_m}{\Omega_t} \tag{3}$$

The C_p can be calculated as:

$$C_p(\lambda,\beta) = c_1\left(\frac{c_2}{\lambda_i} - c_3\beta - c_4\right)e^{-\frac{c_5}{\lambda_i}} + c_6\lambda \tag{4}$$

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where,

$$\frac{1}{\lambda_i} = \frac{1}{\lambda + 0.08\beta} - \frac{0.035}{1+\beta^2} \quad (5)$$

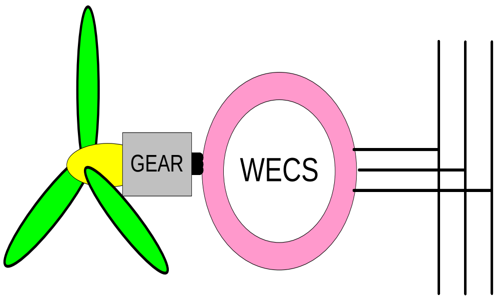

Figure 1. Wind energy conversion system.

III. Wind Aerodynamics and Blade Pitch Control

At a cross section of WT blade, the wind velocity components are shown in Fig. 2, velocity components e.g. wind velocity, flow factors and rotor speed etc. and also these components are helpful to determine angle of attack.

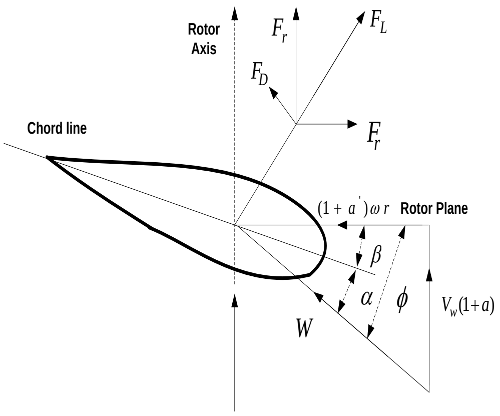

Figure 2. Blade element velocity components and forces at a blade cross-section.

As shown in Fig. 2, inflow wind velocity is perpendicular to the rotor plane. When the wind passes through the rotor plane, wind velocity reduces, due to axial interference and blade rotates with the angular velocity ω. Blade element which is at the distance r from the rotor axis will be rotating at a speed of ωr in the rotor plane. When wind passes through the rotor plane and interacts with the moving rotor plane, a tangential velocity $a'\omega r$ is introduced and the net tangential flow velocity of blade element becomes $(1+a')\omega r$. Hence the resultant velocity of blade can be determined as,

$$W = \sqrt{V_w^2(1-a^2) + \omega^2 r^2 (1+a')^2} \quad (6)$$

where W is the resultant relative velocity of the blade, a and a' are flow factors and r is distance of blade element from the rotor axis. The resultant relative velocity gives rise to aerodynamic forces on the blade, therefore a lift force can be calculated as,

$$F_L = \frac{1}{2}\rho C W^2 C_L \quad (7)$$

And also a drag force can be expressed as,

$$F_D = \frac{1}{2}\rho C W^2 C_D \quad (8)$$

where C_L is lift coefficient, C_D is drag coefficient and C is the chord length of the WT blade.

The lift force F_L decreases and due to this the mechanical power of WT also reduces. In the pitch adjusting variable speed WT system, the angle of attack α decreases as the pitch angle β increases, the curve between the pitch angle and wind speed is shown in Fig. 3 as,

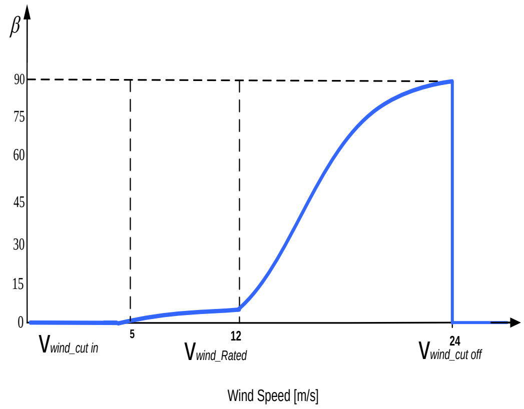

Figure 3. Blade pitch angle-wind speed curve.

The WT generated power varies as the wind speed varies. WT must be protected against mechanical overloads and possible risk of damages at high wind potential. This is achieved by pitching the blades into the position where a part of incoming wind will pass through WT.

In the wind power generation, it is important to keep WT at the optimal speed for the stabilization of the power generation regardless of the wind speed variations [21]-[22]. In order to reduce the fluctuations in generated power caused by the wind speed variations, the blade pitch angle plays an important role [20].

Aerodynamic torque and bending moment depends nonlinearly on wind speed, pitch angle, tower and blade deflections. Further, there is variation as rotor turns to position the blades in a turbulent wind profile that varies spatially with respect to the rotor disk even in constant wind conditions [22]-[23]. Blade pitch angle control of WTGs is providing appropriate adjustment of the amount of wind received by the WTGs to suppress the output fluctuation by varying the pitch angle of the WT blade [21]-[23].

The pitch angle ranges manipulate from10 degree to 90 degree. V_w, V_{wi}, V_{wr}, and V_{wo} which represents wind velocity, cut-in velocity, rated velocity, and cut-out velocity, respectively. The following regions are suggested for setting the pitch angle depending upon wind speed as,

(a) $V_w \leq V_{wi}$ (5 m/s or less)

Set pitch angle to 90 degree. In this region, WTG does not generate power.

(b) $V_{wi} \leq V_w \leq V_{wr}$ (5 to 12 m/s)

Set the pitch angle to 10 degree so that WTG can receive maximum wind energy.

(c) $V_{wr} \leq V_w \leq V_{wo}$ (12 to 24 m/s)

Adjust pitch angle within the specified range (from 10 degree to 90 degree) so that the power output of WTG is constant (rated power output).

(d) $V_{wo} \leq V_w$ (more than 24 m/s ~)

Set the pitch angle to 90 degree for safety as well as in region (a).

The blade pitch angle controls the output of WTGs from exceeding the rated capacity. Therefore, the pitch angle is conventionally changes in this region actively [21]-[24].

In given Fig. 4, wind speed variation range is shown with their range as already discussed. According to the wind speed variation the generated power is shown as:

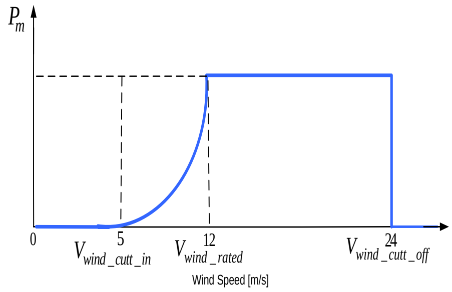

Figure 4. Power- wind speed curve.

IV. Gear System Technology

Gears are particularly useful for changing the shaft speed and motion transmission also. The gear connected to the rotor shaft has more teeth than the gear connected to generator for power generating. The rotational speed is transferring from a big size gear (driver gear) to a small size gear (driven gear), the rotational speed increases but the torques decreases, while total power *P* remains the same. This concept can be explained as:

$$P = \rho_1 r_1 = \rho_2 r_2 \tag{9}$$

The ratio of shaft speed is equal to the inverse ratio of the number of teeth on the connecting gears [10]-[12].

$$\frac{r_1}{r_2} = \frac{N_2}{N_1} \tag{10}$$

where:

r_1/r_2 = Ratio of shaft speed

N_2/N_1 = Ratio of number of teeth.

A. Gear Design Methodology

The equations governing gear design are summarized as:

Diametral pitch *(DP)*, is defined as the ratio of number of gear teeth and pitch diameter as,

$$\text{Diametral Pitch} = \frac{\text{Number of Gear Teeth}}{\text{Pitch Diameter}} \tag{11}$$

The diametral pitch is represented as the number of teeth per inch of pitch diameter [1].

A gear system shows gear teeth, pitch diameter and diametral pitch in Fig. 5. Big driver gear have less number of teeth per inch of diametral pitch. Another method is to describe this, a gear teeth size varies inversely with the diametral pitch.

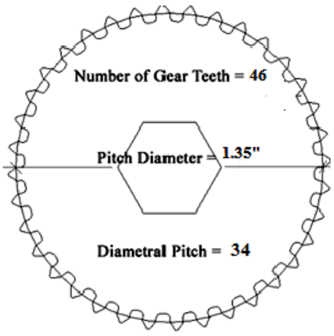

Figure 5. A gear system shows gear teeth, pitch diameter and diametral pitch.

The gear ratio is defined as,

$$\text{Gear Ratio} = \frac{\text{Driven Gear Teeth}}{\text{Driver Gear Teeth}} \tag{12}$$

Gear ratio is used for some mechanical advantages e.g. torque or high rotational speed. The gear ratio of a given pair of spur gear system is determined by dividing number of teeth on the driven gear, by the number of teeth on the driver gear [1]. As shown in Fig. 6, the driver gear and driven gear are having 70 and 46 number of teeth respectively. The gear ratio can be determined as,

$$\text{Gear Ratio} = \frac{46}{70} = 1:1.52$$

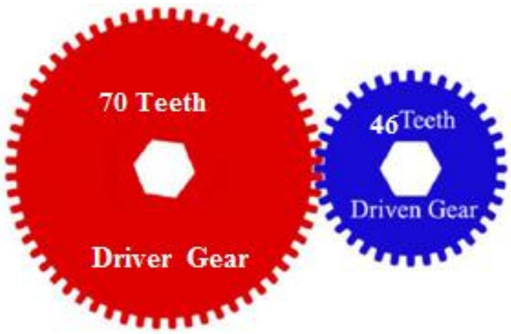

Figure 6. Meshing of gear system.

The pitch diameter is given by following equation.

$$D = \frac{N}{P} \tag{13}$$

where:

D = Pitch diameter
N = Number of teeth on the gear
P = Diametral pitch

The pitch diameter refers to the diameter of the pitch circle. If the gear pitch is known then pitch diameter can be calculated easily.

Addendum (A) is expressed as:

$$A = \frac{1}{P} \tag{14}$$

The radial distance from the pitch circle to the top of the gear tooth is called addendum.

Deddendum (B) is expressed as:

$$B = \frac{\pi}{2P} \tag{15}$$

The redial distance from the pitch circle to the bottom of the tooth is called deddendum.

Outside diameter (OD) is given as:

$$OD = \frac{N+2}{P} \tag{16}$$

The overall diameter of the gear is called outside diameter.

Circular pitch (CP) is expressed as,

$$CP = \frac{\pi D}{N} \tag{17}$$

The measured distance along with the circumference of pitch diameter (PD) from the point of one tooth to the corresponding point on an adjacent tooth is called circular pitch.

Circular thickness (T) of gear designed is expressed as,

$$T = \frac{\pi D}{2N} \tag{18}$$

Thickness of a tooth measure the circumference of the pitch circle is called circular thickness. The gear design methodology has been explained mathematically using Eq. (9) to (18) [10-15].

V. ESTIMATION OF GEAR VELOCITY AND RPM

In given example; the velocity is calculated inches per minute but the gear industries often use it in feet per minute. The velocity can be determined by using following expression as:

Velocity = Pitch circle circumference $\times$RPM

Example: The 34 diametral pitch driver gear, shown in Fig. 6 is turning at 110 rpm. The velocity of the driver gear can be calculated using following steps as,

Step 1. Calculate the pitch diameter (D)

$$D = \frac{Teeth}{Pitch} = \frac{N}{P} = \frac{70}{34} = 2.058 \text{ Inches}$$

Step 2. Obtain the circumference of the pitch circle using the pitch diameter.

Circumference = $\pi \times D$
=3.1416$\times$2.058= 6.46541 Inches

Step 3. Determine the velocity using the gear velocity formula.

Velocity= Circumference $\times$RPM
= 6.46541$\times$RPM

= 6.4651$\times$110 = 711.1951 Inches/ minutes

$$= \frac{711.1957}{12} = 59.266 \text{ Feet/ minute}$$

Calculate the velocity of the driven gear in the above case.

The 46 tooth driven gear in the above example is being driven by a 70 tooth driver gear. In order to calculate the driven gear velocity, first calculate the driven gear RPM using the gear ratio.

Step-1. Determine the driven gear RPM using the gear ratio as,

Driven gear RPM = Drive gear RPM $\times$Ratio
= 110$\times$1.52 =167.2 RPM

Step- 2. Determine the pitch diameter as,

$$D = \frac{Teeth}{Pitch} = \frac{46}{34} = 1.35294 \text{ Inches}$$

Step-3. Determine the circumference of the pitch circle using the pitch.

Diametric circumference = $\pi \times D$
= 3.1416 $\times$1.35294 = 4.2504 Inches

Step-4. Calculate the gear velocity using the gear velocity expression as,

Velocity = Circumference $\times$RPM
= 4.2504 $\times$167.2 RPM

= 710.66688 Inches per minute

$$= \frac{710.66688}{12} = 59.22224 \text{ Feet/ minute}$$

The velocity of the 70 tooth driver gear is 59 feet per minute approximately and the velocity of the 46 tooth driven gear is 59 feet per minute. Gears in mesh rotate at different RPM but always at equal velocity.

VI. ISSUES OF GEAR SYSTEM IN WIND TURBINE

Increasing the revolution or speed (RPM) of gear shaft is necessary in designing process of WT for generating the electric power. For efficient design, the gear designer should have a thorough knowledge of the possible causes and issues of gear failure. It is easy to predict the service life of the gear set with reasonable accuracy after assessing the following relevant factors.

A. *Normal and Abrasive Wear*

When metal slides against the metal, an inevitable consequence is the gradual loss of material from the surface of the teeth in mesh. This can be termed as normal wear; proper lubrication is one of the best and effective options to minimize this type of loss. Abrasive wear takes place due to surface injury or damaged caused by particles trapped in between the tooth surfaces [13]-[15].

B. *Scoring*

Scoring is essentially attributed to the lubrication failure. Course rides and radial scratch lines are formed from the tip of the teeth drawn to the pitch circle. Lack of adequate lubricant may cause metal to metal contact, resulting in momentary welding between contacting surfaces due to molecular adhesion [16], [17]. Extreme pressure (EP) lubricants are used to prevent or minimize scoring. Besides improper lubrication, aspects such as misalignment, interference, tooth spacing error and poor surface finish are also contributory factors to scoring failures.

C. *Tooth Breakage*

This kind of failure occurs due to fatigue, and sudden overload or shock. The fatigue crack at gear tooth is shown in Fig. 7.This kind of failure starts with a crack which progressively widens till a portion or a whole tooth breaks away.

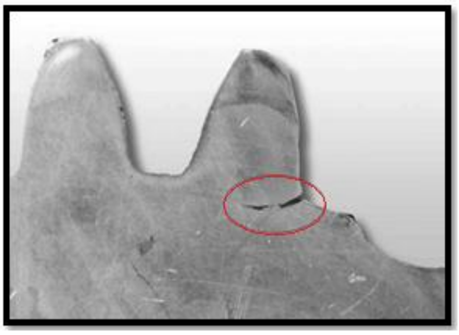

Figure 7. Fatigue crack at gear tooth

D. *Corrosion*

This is caused by chemical action by using the wrong kinds of lubricants or it may be due to agents prevailing in the surrounding atmosphere which may be of corrosive nature.

E. *Spalling*

This is also a surface fatigue failure similar to pitting. The damage to tooth surface may be extensive. Chunks of tooth break away as small or large flakes. Unlike pitting, the damage is not confined to the pitch line area, but may occur at the tip area.

F. *Gear Noise*

The vibrations created by sound waves are very disturbing for the normal functioning of machine tools, automobiles and marine engine drives [17].

VII. LOAD ASSESSMENT OF GEAR SYSTEM

There is no ideal measurement method in place allowing accurate, reliable and durable detection of operating loads of the gearbox in WTs. Based on challenging environment condition in wind turbines, the installation of load monitoring sensors does currently not provide perfects results [18].

Due to the experiences in the field of unexpected overload situation, there is still an urgent need for turbine manufacturer of getting an accurate load detection system implemented which must be able to cover all critical parameters [19], [20], in order to ensure long life of turbine system.

VIII. WIND SPEED MEASUREMENT AND YAW DIRECTION CONTROL

Various types of instruments are used to measure the wind speed and direction for WT power generation e.g. wind vane, anemometer. In which the wind vane is indicated by prevailing wind indicates the direction from which the wind is blowing. Other modern instrument is anemometer, it is utilizing for wind speed measurement as well as wind direction. It is mounted on the top of WT hub system; generally these types of instruments are used in wind energy industries for wind resource assessment and WT control also. There is no moving part inside the anemometer. With the help of the wind speed and direction measurement method, the movement of yaw system towards the wind direction is easy to harvest maximum wind for power generation, when the direction of wind in a certain direction, the flag would point in that direction. The schematic diagram of anemometer is shown in Fig. 8 as,

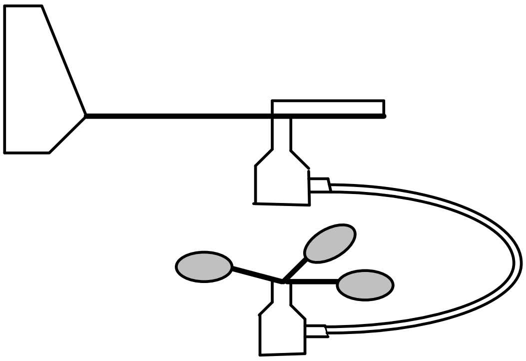

Figure 8. Anemometer for wind speed/ direction measurement

A cup anemometer conventionally consists of three hemispherical or conical cups, arranged in horizontal rotor configuration around a central vertical shaft that drives a signal generation device, shown in Fig. 9 as:

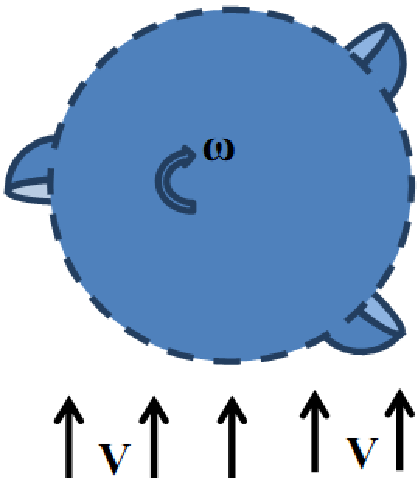

Figure 9. Cup- anemometer for wind speed measurement.

Let's consider the relative motion of the cup with respect to the wind velocity, in the left half of the above Fig. 9, the velocity of anemometer cup and wind, both are in same direction [25]. Hence the torque produced by the left half is:

$$K_{D1}\left(V - R\omega\right)^2 \frac{\rho}{2}\frac{\pi d^2}{4}R \tag{19}$$

In the right of the Fig. 9, the velocity of anemometer cup and wind, both are in opposite direction. Hence the torque produced by the right half is,

$$K_{D2}\left(V + R\omega\right)^2 \frac{\rho}{2}\frac{\pi d^2}{4}R \tag{20}$$

where V is the horizontal component of wind velocity, ω is the cup wheel angular velocity, R is the cup arm length (length between center of the wheel and center of the cup), d is the cup diameter, ρ is the air density and K_{D1} and K_{D2} are the aerodynamics drag coefficients of the cup in regressive and progressive phases respectively [26].

In the steady state the accelerating torque balances the frictional torque of the shaft.

$$\text{Frictional torque} = K_{D1}\left(V - R\omega\right)^2 \frac{\rho}{2}\frac{\pi d^2}{4}R$$

$$-K_{D2}\left(V + R\omega\right)^2 \frac{\rho}{2}\frac{\pi d^2}{4}R \tag{21}$$

Normally the frictional torque is very small or zero, compared with the aerodynamics torque and can be determined as,

$$K_{D1}\left(V - R\omega\right)^2 \frac{\rho}{2}\frac{\pi d^2}{4}R$$

$$-K_{D2}\left(V + R\omega\right)^2 \frac{\rho}{2}\frac{\pi d^2}{4}R \tag{22}$$

Eq. (22) can be rewrite as:

$$\frac{K_{D2}}{K_{D1}} = \left(\frac{1-\frac{R\omega}{V}}{1+\frac{R\omega}{V}}\right)^2 \tag{23}$$

$$f_0 = \frac{R\omega}{V} = \frac{1-K}{1+K} \tag{24}$$

where f_0 is the anemometer factor and C is the square root of the ratio between regressive and profressive aerodynamic drag coefficients as,

$$K = \sqrt{\frac{K_{D2}}{K_{D1}}} \tag{25}$$

$$V = \frac{R\omega}{f_0} \tag{26}$$

The yaw control mechanism is embedded based system in which the wind speed/ direction are sensed and this signal is controlled by the microcontroller based electronic system. The wind velocity is measured in m/s, shown in Eq. (27) as,

$$\text{Velocity} = \text{Time} \times \text{Acceleration} \tag{27}$$

Yaw control is responsible to give direction for wind turbine with the same direction of the wind potential. This is achieved by a wind direction sensor, which actuates the yaw control towards the wind direction [23]-[26].

Sometimes the WT undergoes a yaw error, if the rotor is not perpendicular to the wind. The existence of a yaw error indicates that a lower fraction of the energy in the wind will be flowing through the rotor area and available for extraction. The lost power fraction is proportional to the cosine of the yaw error angle as [26],

$$\Delta P = \lambda Cos\varphi \tag{28}$$

where:

φ = yaw error angle

ΔP = power loss caused by the yaw error

λ = proportionality constant

If the yaw error would only lead to reduce in the power output, it would be an acceptable method of power control. The rotor blades under a yaw error would be bending back and forth in a flap wise fashion for each turn of the rotor. Running a WT with a yaw error subjects it to a large fatigue load that could lead to its eventual fatigue failure. It becomes necessary to equip wind machines with yaw mechanism placing them in a direction that is perpendicular to the wind direction [26].

IX. Conclusion

In this paper, a detailed study is carried out of WECS and wind aerodynamics. A suitable range of blade pitch angle is identified, which are based on wind speed variations to control generated output power of WT system. A simplified gear design methodology has been proposed and velocity has been calculated for one practical case. Mechanical control of WT system is achieved by gear box controlling. Some related issues with gear system e.g. scoring, tooth breakage, corrosion, spalling and gear noise are discussed. In this paper, a theory is also investigated on wind direction sensor, which actuates the yaw control towards the wind direction. The yaw control which is important for extracting the wind power has been explained.

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