Influence of Tooth Pitting and Cracking on Gear Meshing Stiffness and Dynamic Response of Wind Turbine Gearbox

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Abstract- Gearboxes are widely used in wind turbine applications. Planet gears can operate in several conditions such as excessive applied torque, bad lubrication and manufacturing or installation problems. Tooth crack and surface failure occur due to excessive stress conditions. This can cause crack or removal and/or plastic deformation of the contacting tooth surfaces such as spalling and may lead to tooth breakage. The meshing stiffness of the affected tooth is found to be reduced proportionally to the severity of the defect. In this article an analytical method is proposed to quantify the reduction of meshing stiffness due to three common tooth faults: cracking, spalling and breakage. Bending, shear and contact stiffness are taken into account. Moreover, the dynamic response of wind turbine gearbox model which comprises 12-degree-of-freedom is computed by using analytical meshing stiffness issued from analytical modeling and the dynamic responses of each tooth fault is identified. The results indicate that the proposed analytical method allows quantifying the reduction of the meshing stiffness usually in these cases of faults and increase of the dynamic response, consequently give valuable information to diagnosis these faults.

Keywords- Meshing stiffness; spalling; analytical method; breakage; cracking; pitting; diagnosis; bending; shear; contact; dynamic response

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$R_p$, $R_s$</td>
<td>Planet and sun radius.</td>
</tr>
<tr>
<td>$F_{km1}$</td>
<td>Mesh stiffness force for helical gears.</td>
</tr>
<tr>
<td>$F_{km2}$</td>
<td>Mesh stiffness force for planetary gears.</td>
</tr>
<tr>
<td>$F_{cm1}$</td>
<td>Mesh damping force for helical gears.</td>
</tr>
<tr>
<td>$F_{cm2}$</td>
<td>Mesh damping force for planetary gears.</td>
</tr>
<tr>
<td>$C_{T1}$</td>
<td>Equivalent gear mesh damping.</td>
</tr>
<tr>
<td>$K_{T1}$</td>
<td>Equivalent gear mesh stiffness.</td>
</tr>
<tr>
<td>$C_{T2}$</td>
<td>Equivalent pinion, sun and carrier mesh damping.</td>
</tr>
<tr>
<td>$K_{T2}$</td>
<td>Equivalent pinion, sun and carrier mesh stiffness.</td>
</tr>
<tr>
<td>$M_D$</td>
<td>Driving moment.</td>
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<tr>
<td>$M_L$</td>
<td>Load moment.</td>
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<tr>
<td>$m_p$</td>
<td>Mass of planet gear</td>
</tr>
<tr>
<td>$m_s$</td>
<td>Mass of sun gear</td>
</tr>
<tr>
<td>$Y_p$</td>
<td>Planet vertical displacement</td>
</tr>
<tr>
<td>$Y_s$</td>
<td>Sun vertical displacement</td>
</tr>
<tr>
<td>$Y_{T1}$</td>
<td>Equivalent gear mass $m_{T1}$ vertical displacement</td>
</tr>
<tr>
<td>$Y_{T2}$</td>
<td>Equivalent gear mass $m_{T2}$ vertical displacement</td>
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Scale constant

<table>
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<tr>
<th>Symbol</th>
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<tr>
<td>$\mu$</td>
<td>Transmission ratio of planetary and helical gearbox</td>
</tr>
<tr>
<td>$R_p$, $R_H$</td>
<td>Overall gearbox transmission ratio</td>
</tr>
<tr>
<td>$R_O$</td>
<td>Total mesh stiffness of helical gear: two pairs</td>
</tr>
<tr>
<td>$K_{m1}$, $K_{m2}$</td>
<td>Total mesh stiffness of planetary gear: sun and planet: planet and ring</td>
</tr>
<tr>
<td>$K_{m3}$, $K_{m4}$</td>
<td>Total mesh damping of helical gear: two pairs</td>
</tr>
<tr>
<td>$C_{m1}$, $C_{m2}$</td>
<td>Total mesh damping of planetary gear: sun and planet: planet and ring</td>
</tr>
<tr>
<td>$C_{m3}$, $C_{m4}$</td>
<td>Helical gears radius</td>
</tr>
<tr>
<td>$m_{g1}$, $m_{g2}$</td>
<td>Mass of the helical gears</td>
</tr>
<tr>
<td>$K_{g1}$, $K_{g2}$</td>
<td>Gears shaft torsion stiffness.</td>
</tr>
<tr>
<td>$m_{gb}$, $m_{g4}$</td>
<td>Mass of the helical gears.</td>
</tr>
<tr>
<td>$K_{gb}$, $K_{g4}$</td>
<td>Gears shaft torsion stiffness.</td>
</tr>
<tr>
<td>$C_{gb}$, $C_{g2}$</td>
<td>Gears shaft torsion viscous damping coefficient.</td>
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Gearboxes are critical elements of most wind turbine designs. Cost effective turbine operation requires that gearboxes achieve low maintenance operation for the design life of the turbine. The recently introduced AGMA/AWEA (921-97) Recommended Practices for Gearbox Design and Specification of Gearboxes for Wind Turbine Generator Systems [1-2] are intended to help wind turbine manufacturers to determine the specifications necessary to ensure adequate gearbox designs. This study attempts to determine the degree to which current design guidelines can be used to maximize the likelihood that the gearbox will actually perform as desired. Recommended gearbox design practices are used to evaluate the adequacy of the integrated, planetary gearbox used in the ESI-80 250 kW teetering wind turbine. In turn, the Recommended Practices are evaluated to determine whether the recommendations properly address failure modes observed in these gearboxes in the field. The examination focuses on the recent failure of the gearbox on the research turbine at the University of Massachusetts (UMass).

Planetary gears are effective power transmission elements where high torque to weight ratios, high reliability and superior efficiency are required. Example applications are automotive transmissions, tractors, wind turbines, helicopters, and aircraft engines. Gear vibrations are primary concerns in most planetary gear transmission applications, where the manifest problem may be noise or dynamic forces. Noise levels exceeding 110 dB observed in a helicopter cabin are attributed largely to vibration of the planetary gear. Large dynamic forces increase the risk of gear tooth or bearing failure. In [3-4], experiments are performed on a spur gear pair and observed various nonlinear phenomena including gear tooth contact loss, period-doubling and chaos. Tooth separations at large vibrations, which are common in spur–gear pairs, occur even in planetary gears as evident from the experiments by in [5].

Planetary gear researchers have developed lumped-parameter models and deformable gear models to analyze gear dynamics. The literature mainly addresses static analysis, natural frequencies and vibration modes, modeling to estimate dynamic forces and responses, and cancellation of mesh forces using the planetary gear symmetry through mesh phasing. Studies presented in [6-7] involve planetary gear models to estimate natural frequencies, vibration modes and dynamic forces. A 2D rotational–translational degree of freedom spur gear model and mathematically show the unique modal properties of equally spaced and diametrically opposed planet systems. All modes can be classified as one of rotational, translational, or planet modes. Mesh stiffness-induced parametric instability is studied in [8-9]. A helical planetary gear model is formulated and the effect of mesh phasing on the dynamics of equally spaced planet systems is investigated [10]. The effectiveness of mesh phasing in suppressing certain harmonics of planetary gear vibration modes based on self-equilibration of the dynamic mesh forces at sun–planet and ring–planet meshes. A thorough description of the relative mesh phasing between the sun–planet and ring–planet meshes in a planetary gear system is given, where a nonlinear dynamic planetary gear model is introduced by and the effects of various design parameters on the dynamic load sharing of the planets are examined. Accurate analytical modeling, including proper mesh phasing relations and detailed characterization of the nonlinear dynamics of planetary gears, is needed to estimate relative gear noise and predict dynamic forces in industrial applications. Little work has been done to characterize the nonlinear effects of tooth separation on planetary gear dynamics. The lack of experimental studies to understand the complex dynamics of planetary gears and the availability of finite element software specialized for gear dynamics motivated the present study.

Mesh stiffness variation as the number of teeth in contact changes is a primary excitation of gear vibration and noise. This excitation exists even when the gears are perfectly machined and assembled. In analytical gear vibration models, it is
represented by time-varying mesh stiffness’s that parametrically excite the system. This parametric excitation causes instability under certain operating conditions. The ensuing vibration creates noise, increases dynamic loads, and potentially damages the gear teeth and bearings [11-12]. Parametric instability in single-pair gears governed by a single-degree-of-freedom Mathieu equation has been extensively investigated. For multi-stage gear systems, there are surprisingly few studies on parametric instabilities from multiple meshes. The analyzed the instabilities of two-stage gear systems with a mesh phasing between the two mesh stiffnesses. However, their instability conclusions are contradictory. This was recently clarified using perturbation and numerical analyses. In addition, a derived simple formula that allows designers to suppress particular instabilities by properly selecting contact ratios and mesh phasing. For planetary gears, which have multiple time-varying mesh stiffnesses, no systematic study on their parametric instability has been found in the literature. A numerically computed dynamic response to mesh stiffnesses variations for planetary gears with three sequentially phased planets.

The analysis of the parametric instability excited by multiple time-varying mesh stiffnesses in planetary gears is presented. The torsional vibration model used here considers the deferent contact ratios and planet phasing among multiple meshes, which are critical design parameters in planetary gears. The well-defined modal properties of planetary gears are used to derive simple expressions for instability boundaries separating the stable and unstable regions. From these expressions, the effects of contact ratios and mesh phasing are analytically determined. These results provide insight into planetary gear designs that avoid parametric instability. In practice, planet mesh-phasing schemes are often applied to cancel or neutralize the resonant response at speeds where the mesh frequency is near a natural frequency [13-14]. In this same spirit, this study shows that particular parametric instabilities can be eliminated under certain phasing conditions that can be achieved by proper selection of design parameters. Tooth separation non-linearity induced by parametric instability is numerically simulated and shown to have great impact on the unstable system responses.

However, this article presents proposed analytical method to quantify the reduction of meshing stiffness due to three common tooth faults: cracking, spalling and breakage. Bending, shear and contact stiffness are taken into account. Moreover, the dynamic response of wind turbine gearbox model which comprises 12-degree-of-freedom is computed by using analytical meshing stiffness issued from analytical modeling and the vibration signatures of each tooth fault is identified.

II. MODELING OF GEAR MESHING STIFFNESS

A. Total Effective Meshing Stiffness

The gear mesh stiffness model described in this study was based on the work presented in [15, 16]. The potential energy method was used to theoretically model the effective mesh stiffness. The total potential energy stored in the meshing gear system was assumed to include four components: Hertzian energy, bending energy, axial compressive energy and shear energy. Thus, for the single-tooth-pair meshing duration, the total effective mesh stiffness \( K_e \) can be expressed as:

\[
K_e = \frac{1}{1/K_0 + 1/K_1 + 1/K_2 + 1/K_3 + 1/K_4 + 1/K_5 + K_6}
\]

(1)

Where \( K_0, K_1, K_2, K_3 \) and \( K_4 \) represent total, Hertzian, bending, shear and axial compressive mesh stiffness respectively.

B. Meshing Gear Tooth Potential Energy

The potential energy stored in a meshing gear tooth can be calculated by

\[
U_t = \frac{\alpha}{2E_1} \int_0^L \frac{1}{2} F_1 (d-x)^2 \, dx
\]

(2)

\[
U_t = \frac{\alpha}{2E_A} \int_0^L \frac{1}{2} F_2 \cos \alpha \, dx
\]

(3)

\[
U_h = \int_0^L \frac{\pi}{4(1-\nu^2)} E \, dx
\]

(4)

Where \( I_c \) and \( A_c \) represent the area moment of inertia and area of the section where the distance from the tooth's root is \( x \), and G represents the shear modulus. They can be obtained by

\[
I_c = \begin{cases} 
\frac{1}{12} (h_1 + h_y) L & \text{if } x \leq g_c \\
\frac{1}{12} (2h_y)^2 L & \text{if } x > g_c 
\end{cases}
\]

(5)

\[
A_c = \begin{cases} 
(h_1 + h_y) L & \text{if } x \leq g_c \\
2h_y L & \text{if } x > g_c 
\end{cases}
\]

\[
G = \frac{E}{2(1+\nu)}
\]

Where \( h_y \) represents the distance between the point on the tooth's curve and the tooth's central line where the horizontal distance from the tooth's x.

III. TOOTH CRACKING AND PITTING FAILURES FORMULATION

A. Tooth Cracking Model formulation

On crack development in a gear, Ref. [17] considers that a crack is developing at the root of a single tooth of the pinion. A tooth root crack typically starts at the point of the largest stress in
the material. The computational model which applies the principles of linear elastic fracture mechanics is used to simulate gear tooth root crack propagation. Based on the computational results, the crack propagation path shows a slight curve extending from the tooth root. Also, indicates that crack propagation paths are smooth, continuous and in the most cases, rather straight with only a slight curvature. In this paper, based on the results shown in [16], the crack model was further simplified to consider the crack path to be straight line. The crack starts at the root of the pinion and then proceeds. However, the expression of the components in equations (2) to (4) when cracks are introduced.

Referring to Figures 1 to 2, the intersection angle, \( \alpha \), between the crack and the central line of the tooth is set at a constant 45°. The crack length, \( a_{\text{cr}} \), grows from zero with an increment size of \( \Delta a = 1.0 \text{ mm} \) until the crack reaches 3.0 mm.

With the crack introduced as described above, all components of the total mesh stiffness need to be calculated, that is Hertzian stiffness, axial compressive stiffness, bending stiffness and shear stiffness. Based on the work documented in [18], the Hertzian and axial compressive stiffness will not change due to the appearance of the crack, and their derivation are proved and presented in the following expressions. When \( h_{s1} < h_s \), or when \( h_{s1} \geq h_s \), \( \alpha_1 \leq \alpha_g \)

The bending mesh stiffness of the cracked tooth is

\[
\frac{1}{K_b} = \frac{1}{\pi} \int_{-\sqrt{\frac{2W}{k}}(\cos(\alpha_1) + \cos(\alpha_2))}^{\sqrt{\frac{2W}{k}}(\cos(\alpha_1) + \cos(\alpha_2))} \left( \frac{2}{\sqrt{x^2 - a_1} + \sqrt{x^2 - a_2} + \sqrt{x^2 - a_3}} \right) dx
\]

and the shear mesh stiffness of the cracked tooth is

\[
\frac{1}{K_s} = \frac{1}{\pi} \int_{-\sqrt{\frac{2W}{k}}(\cos(\alpha_1) + \cos(\alpha_2))}^{\sqrt{\frac{2W}{k}}(\cos(\alpha_1) + \cos(\alpha_2))} \left( \frac{2}{\sqrt{x^2 - a_1} + \sqrt{x^2 - a_2} + \sqrt{x^2 - a_3}} \right) dx
\]

where \( \alpha \) is the pressure angle = +20° for external meshing

\[
\alpha = -20° \text{ for internal meshing}
\]

From the results derived in [19], the stiffness of Hertzian contact of two meshing teeth (commonly nonlinear) is practically a constant along the entire line of action independent to both the position of contact and the depth of interpenetration. Based on equation (4), \( K_b \) can be approximated by a constant value depending on the tooth width and the mechanical properties of the gear material:

\[
K_b = \frac{\pi EW}{4(1 - \nu^2)}
\]

B. Tooth Pitting Models formulation

In this section, tooth pitting faults are modeled: tooth spalling and tooth breakage. Spalling on a gear tooth surface causes the contact area between meshing tooth pair to decrease. It is produced by a combination of high surface stresses and relatively high sliding velocities between teeth in contact. In the initial stages of spalling, cracks appear in the tooth surface and spread from the failure origin in a fan like manner in the direction of the sliding. A piece of the material is removed from the surface, giving the appearance of a great deal of destructive pitting in which the pits have run together forming a spalled area.

Breakage is the fracture of a whole tooth or substantial part of a tooth. Common causes include overload and cyclic stressing of the gear tooth material beyond its endurance limit. Bending fatigue breakage starts with a crack in the root section and progresses until the tooth or part of it breaks off. Overload breakage appears as a stringy, fibrous break that has been rapidly pulled or torn apart. Random fracture can occur in areas such as the top or the end of a tooth, rather than the usual root fillet section.

These two tooth faults result in a material removal which causes a decrease of the contact zone between teeth in contact and an increase in deflection. In the absence of the damage on a tooth’s surface, the contact between teeth in contact is in a straight line along the width \( W \). However, when there is spalling damage, the width of contact changes at the defect location because of the removal of material from the surface of the tooth. It is assumed here that no tooth contact occurs in the spalled area. On the other hand, the area moment of inertia and the surface of the cross sectional areas of the tooth in the affected area are diminished. Spalling is modeled as a rectangular indentation having the dimensions \( a_1 \) and \( h_s \) as shown in Figure 3. \( W \) is the width of contact of two meshing teeth for the healthy case. In the presence of the spalling this width becomes:

\[
W_s = W - W_s
\]

\( \{s_1, \ldots, s_t\} \) are the reduced tooth thickness in the affected zone (sections \( S_s \), in Figure 3). All these parameters are introduced in equation (8) by modifying area moment of inertia, width and cross sectional areas. the computation of the summation have to be done taking into account the changed shape of the elementary cross sectional areas \( A_i \) and area moment of inertia \( I_i \) as shown in Figure 5 (A). Tooth gear meshing stiffness is then computed.

A broken tooth on the addendum circle is also considered. To simplify the modeling of this type of damage, at each section \( s_b \) of the tooth, the shape of the breakage is approximated with straight line as
shown in Figure 4 defined by the height $h_b$, the thickness $t_b$, the width $w_b$ and the thickness $[t_1, t_2, t_3, \ldots, \text{etc.}]$ of the tooth. The width of contact changes as the line of contact moves and is evaluated instantaneously by:

$$W_i = W - w_b$$  \hspace{1cm} (10)

The tooth and gear meshing stiffness of the gear pair is also calculated according to equation (8) by taking into account the geometric changes due to the tooth breakage and updating the values of width cross sectional area and area moment of inertia (Figure 5 (B)).

### IV. DYNAMIC SIMULATION OF GEARBOX VIBRATION RESPONSE

The mathematical model be will adopted with torsional and lateral vibration. The model represents the wind turbine gearbox stages (one planetary and two helical stages) gearbox system is given in Figure 6. It is a two-parameter (stiffness and damping) model with torsional and lateral vibration, which means that it includes both the linear and rotational equations of the system’s motion. This model represents a system with twelve degrees of freedom, which is driven by electric motor moment, $M_D$ and loaded with external moment, $M_L$. This model is simple enough to enable to focus on the effects of tooth faults on the vibration response of the system. Thus, in this paper, it is assumed that all gears are perfectly mounted rigid bodies with ideal geometries. Inter-tooth friction is ignored here for simplicity.

Because friction is ignored, the vibration in the $x$ direction is free response and will disappear due to inherent damping. It is focused only on the motion in the $y$ direction. However, the equations of motion are:

$$I_y \ddot{\theta}_y + C_y R_y(\dot{\theta}_y - \dot{\theta}_g) + K_y R_y(\dot{\theta}_y - \dot{\theta}_g) = M_D$$  \hspace{1cm} (11)

$$I_y \ddot{\theta}_y - C_y R_y(\dot{\theta}_y - \dot{\theta}_g) + K_y R_y(\dot{\theta}_y - \dot{\theta}_g) = R_y(F_{\text{int}} + F_{\text{ad}})$$  \hspace{1cm} (12)

$$I_{g1} \ddot{\theta}_{g1} - C_{g1} R_{g1}(\dot{\theta}_{g1} - \dot{\theta}_g) - K_{g1} R_{g1}(\dot{\theta}_{g1} - \dot{\theta}_g) = R_y(F_{\text{int}} + F_{\text{ad}})$$  \hspace{1cm} (13)

$$I_{g2} \ddot{\theta}_{g2} - C_{g2} R_{g2}(\dot{\theta}_{g2} - \dot{\theta}_g) - K_{g2} R_{g2}(\dot{\theta}_{g2} - \dot{\theta}_g) = R_y(F_{\text{int}} + F_{\text{ad}})$$  \hspace{1cm} (14)

$$I_y \ddot{\theta}_y - C_y R_y(\dot{\theta}_y - \dot{\theta}_g) + K_y R_y(\dot{\theta}_y - \dot{\theta}_g) = R_y(F_{\text{int}} + F_{\text{ad}})$$  \hspace{1cm} (15)

$$I_{g1} \ddot{\theta}_{g1} - C_{g1} R_{g1}(\dot{\theta}_{g1} - \dot{\theta}_g) + K_{g1} R_{g1}(\dot{\theta}_{g1} - \dot{\theta}_g) = R_y(F_{\text{int}} + F_{\text{ad}})$$  \hspace{1cm} (16)

$$I_{g2} \ddot{\theta}_{g2} - C_{g2} R_{g2}(\dot{\theta}_{g2} - \dot{\theta}_g) + K_{g2} R_{g2}(\dot{\theta}_{g2} - \dot{\theta}_g) = R_y(F_{\text{int}} + F_{\text{ad}})$$  \hspace{1cm} (17)

$$I_y \ddot{\theta}_y - 2C_y R_y(\dot{\theta}_y - \dot{\theta}_g) - 2K_y R_y(\dot{\theta}_y - \dot{\theta}_g) = -M_L$$  \hspace{1cm} (18)

1) **Meshing Forces- Helical**

$$F_{km1} = K_{T1}(R_{g1}\dot{\theta}_1 - R_{g2}\dot{\theta}_2 + Y_1 - Y_2)$$  \hspace{1cm} (19)

$$F_{km1} = C_{T1}(R_{g1}\dot{\theta}_1 - R_{g2}\dot{\theta}_2 + \dot{Y}_1 - \dot{Y}_2)$$  \hspace{1cm} (20)

$$F_{km2} = K_{T2}(R_{g3}\dot{\theta}_3 - R_{g4}\dot{\theta}_4 + Y_3 - Y_4)$$  \hspace{1cm} (21)

$$F_{cm2} = C_{T2}(R_{g3}\dot{\theta}_3 - R_{g4}\dot{\theta}_4 + \dot{Y}_3 - \dot{Y}_4)$$  \hspace{1cm} (22)

$$F_{KT1} = K_{T1}(R_{g1}\dot{\theta}_1 - R_{g4}\dot{\theta}_4 + Y_1 - Y_4)$$  \hspace{1cm} (23)

$$F_{CT1} = C_{T1}(R_{g1}\dot{\theta}_1 - R_{g4}\dot{\theta}_4 + \dot{Y}_1 - \dot{Y}_4)$$  \hspace{1cm} (24)

2) **Transverse Equations of Motion**

Based on [20], the vertical motion (in the $y$ direction) equations are:

$$m_{T1} \ddot{Y}_1 = F_{KT1} + F_{CT1} - F_U - F_{UC}$$  \hspace{1cm} (25)

$$m_{T2} \ddot{Y}_4 = -F_{KT1} - F_{CT1}$$  \hspace{1cm} (26)

$$m_{SP} \ddot{Y}_S = F_{KT2} + F_{CT2}$$  \hspace{1cm} (27)

$$m_{SP} \ddot{Y}_P = -F_{KT2} - F_{CT2} + F_I + F_{IC}$$  \hspace{1cm} (28)

$$F_U = K_1 Y_1$$  \hspace{1cm} (29)

$$F_I = K_3 Y_P$$  \hspace{1cm} (30)

$$F_{UC} = C_1 \dot{Y}_1$$  \hspace{1cm} (31)

3) **Meshing Forces- Planetary**

$$F_{KT2} = K_{T4}(R_{g5}\dot{\theta}_5 - R_{g6}\dot{\theta}_6 + Y_5 - Y_6) - K_{T3} Y_P$$  \hspace{1cm} (32)

$$F_{CT2} = C_{T4}(R_{g5}\dot{\theta}_5 - R_{g6}\dot{\theta}_6 + \dot{Y}_5 - \dot{Y}_6) - C_{T3} \dot{Y}_P$$

With the established model, the next step is to use these sets of equations for computer simulation when the gear is new or faulty (cracked or breakage or spalling) to various degrees. To focus on the effects of faults size thorough the total mesh stiffness, it is assumed further that the vertical radial stiffness of the input bearings, $k_1$, and that of the output bearings, $k_5$, are identical and constant, that is, $k_1 = k_5 = k_S$; the damping coefficient of the input bearings, $c_1$, and that of the output bearings, $c_5$, are equal to a constant $c_S$; the torsional stiffness of the input flexible coupling, $k_p$, and that of the output flexible coupling, $k_p$, are equal to a constant $k_1$; the damping coefficient of the input flexible coupling, $c_1$, and that of the output flexible coupling, $c_5$, are equal to $c_S$. Also, the mesh
damping coefficient, $c_m$, is set to be proportional to the total mesh stiffness, $k_m$, that is,

$$C_i = \mu i K_i,$$

where $\mu$ is the scale constant measured in seconds, and its value has been selected in this simulation as $3.99 \times 10^{-3}$ (s). Furthermore, the other parameters of the whole gearbox system are listed in TABLES 1 and 2. Using Matlab’s ODE15s function, the acceleration plots can be derived for the perfect (healthy) gear teeth and faulty gear tooth with increasing deterioration levels.

It has been pointed out in [21] that the severity, extent and age of damage can be better represented by pulses. The height of the pulse also has a significant influence on the amplitude. This also changes with the severity and age of the defect and may even decrease with the advancement of the defect. Rectangular pulses could be considered as the simplest impulsive loading, but in practice the shape of the signal will be controlled by the nature of the system in addition to the type of exciting force. In a gear pair system due to the deformation and elasticity of the contacting components, neither the force nor the response time history will have the shape of rectangular pulses. To model the signal more representatively, half-sine pulses are considered. The response of the practical system can be best represented by decaying sinusoid

$$x(t) = \left( \frac{K}{\sqrt{1 - \xi^2}} \right) e^{-\xi \omega_n t} \sin^2 \left( \omega_n t \right)$$

(34)

Where

$$K = \frac{A_x}{\sqrt{1 - \xi^2}}$$

$A_x$ is the height of the pulse $= 1$

$\xi$ is the damping ratio

$\omega_n = \frac{NZ}{60}$ is the frequency of generated pulse

$$K_N = K \hat{x}(t) \quad C_N = C \hat{x}(t) \quad i = 1, 2$$

(35)

$K_N, C_N$ are the total effective meshing stiffness and damping

$\hat{K}, \hat{C}$ are the amplitudes of the total effective meshing stiffness and damping

V. RESULTS AND DISCUSSION

The technical parameters for the whole studied model are given in Tables 1 and 2

A. Gear Meshing Stiffness for the Healthy Case

For typical gear parameters given in Tables 1 and 2, simple Matlab programs were written and obtained numerical values of the total effective mesh stiffness as a function of the gear rotation angle. Based on equation (1), the total effective mesh stiffness within one shaft period of the helical gears meshes, planet-ring gear mesh and planet-sun gears mesh are plotted in Figures 7, 8 and 9 respectively when the gears teeth are perfect (that is, have no cracks or pitting). The total effective mesh stiffness fluctuation is around a mean value of $7.830 \times 10^9$ N/m for helical gears mesh (external), $8.534 \times 10^9$ N/m for planet-ring gears mesh (internal) and $9.764 \times 10^9$ N/m for planet-sun (external) gears mesh.

B. Gear Meshing Stiffness for a Cracked Tooth

The crack is considered to be developed at the root of a single tooth for one of the planet gears. A tooth root crack typically starts at the point of the largest stress in the material. In Ref. [21], a computational model which applies the principles of linear elastic fracture mechanics is used to simulate gear tooth root crack propagation. Based on the computational results, the crack propagation path shows a slight curve extending from the tooth root and that crack propagation paths are smooth, continuous, and in most cases, rather straight with only a slight curvature.

In this paper, based on the results shown in Ref. [21], the crack model is further simplify and consider the crack path to be a straight line. Further referring to Figures 1 and 2, the intersection angle, $u$, between the crack and the central line of the tooth is set at a constant $45^\circ$. The crack length, $q_1$, grows from zero to 3 mm with an increment size of $Dq_1 = 1$ mm. With the expressions of the components of the total mesh stiffness provided above, the total mesh stiffness value can be given each shaft rotation angle and each crack size. For a pair of standard steel involute spur teeth whose main parameters are given in Tables 1 and 2, take three specific crack sizes given above as an example. The total mesh stiffness under each of the two crack sizes has been calculated as a function of the shaft rotation angle and plotted in Figures 10 and 11. From Figure 12 and in terms of RMS values, it can be observed that as the size of the crack grows, the total mesh stiffness in terms of RMS when the cracked tooth is in meshing becomes much lower. In addition, the RMS of the total mesh stiffness for the external mesh is higher than that for internal mesh in terms of either shaft rotation angle or grows of the size. This is important information for fault detection and assessment.

C. Gear Meshing Stiffness for a Spalling Tooth

A spalling is modeled around the pitch circle of one planet gear tooth as described in Section 3.2. Referring to Figures 3 and 5 which represent the evolution of the meshing stiffness for a spalling and put the width $w_v = 4.6$ m, length $a_v = 0.9$ mm and
height as \( h = 1 \text{ mm} \), the total effective meshing stiffness is calculated in terms of shaft rotation angle as shown in Figure 13 and have a mean value of \( 7.01 \times 10^9 \text{ N/m} \) for external gears meshing.

In Figure 14, shows the comparison between external and internal gears meshing at different spalling widths. The spalling is being have height as \( h = 1 \text{ mm} \), length \( a = 0.9 \text{ mm} \) with a variable width \( w = 1.35 \text{ mm} \), 2.3 mm, 4.6 mm and 6.9 mm. A stiffness reduction is observed as the increase of the removed material width \( w \).

D. Gear Meshing Stiffness for a Breakage Tooth

A breakage affecting a tooth of the planet is introduced. It is modeled as described in Section 3.2. Referring to Figure 4 which represents the evolution of the meshing stiffness for a breakage and put with the width \( w_b = 4.6 \text{ mm} \), a height \( h_b = 1.35 \text{ mm} \) and thickness \( t_b = 0.6 \text{ mm} \), the total effective meshing stiffness is calculated in terms of shaft rotation angle as shown in Figure 15 and have a mean value of \( 8.376 \times 10^9 \text{ N/m} \) for external meshing.

Figure 16 shows the comparison between external and internal gears meshing. A broken tooth is now modeled with breakage thickness \( t_b = 0.6 \text{ mm} \), width \( w_b = 0.67 \text{ mm} \) and with variable height \( h_b = 1.35 \text{ mm} \), 2.25 mm, 3.25 mm and 4.6 mm. A reduction in the total meshing stiffness is observed. This reduction is more accentuated when the height of the breakage increases. This is explained by the fact that when the breakage affects the whole height of the tooth, the bending, shear and contact energies increase simultaneously.

E. Dynamic Response of the System

The dynamic response is implemented by using Matlab’s ODE15s function with the parameters given in Tables 1 and 2. The dynamic response (in time history and frequency domain) registered on the planet and planet gears carrier for a healthy gearbox. The computed total meshing stiffness in the healthy and defected cases is injected in the equations of motion (equations 11 to 35).

(1) Healthy case

Figures 17 to 18 show the dynamic responses registered on planet gears carrier \( \theta \) for healthy case in time history and frequency domain at speed 40 rpm, 40 Nm torque load. These figures indicate the influence of speed and torque load on the dynamic response, where the dynamic response increases as the increase of either speed or torque load. More information can be extracted from the figures.

The comparison between the registered dynamic responses theoretically and those measured and present in Ref. [22] is shown in Figure 19, where a good agreement is obtained, despite some discrepancies are existed. These discrepancies are due to the following reasons:

- Only multi-rigid bodies with ideal geometries are considered.
- Inter-tooth friction is ignored.
- Bearings and other gearboxes structure components are ignored to limit the number of - degree - of - freedom considered
- Flexibility in the gearbox components are ignored

(2) Presence of cracking

In Figure 20, the fault influences are not very obvious to visual observation; the signals look very similar compared to the perfect gearbox dynamic response signal (healthy). The dynamic response signals generated with crack levels lower than without crack and look quite similar to one another. When the tooth root crack depth \( q_r \) changes of higher levels produces obvious changes in the gear dynamic response signals. The obvious periodical impulses caused by the cracked tooth appear in the time domain signal as the time increases; this carries diagnostic information that is important for extracting features of tooth defects. Because of the influence caused by the cracked tooth repeats only once in a revolution, the duration between every two impulses is equal to one shaft period. However, the simulated dynamic responses under various depths of crack enable us to compare the performance of different fault growth indicators. These indicators are interested because it is reflected to the presence of crack.

Figure 21 shows the comparison between the dynamic response of planet gear and planet gear carrier at different crack depths, which indicate that the increase of the crack depth is accomplished by an increase in the dynamic response for both planet gear and planet gears carrier. Moreover, the dynamic responses recorded for the planet carrier are higher than that recorded for the planet gear in all the crack depths considered.

(3) Presence of breakage

Figure 22 represents the dynamic response of the planet gears carrier in the case of a broken planet tooth (with dimensions \( t_b = 0.6 \text{ mm} \), \( h_b = 1.35 \text{ mm} \) and \( w_b = 4.6 \text{ mm} \)). The breakage fault effects are not very obvious to visual observation; the signals look very similar compared to the perfect gearbox dynamic response signal (healthy). The dynamic response signals generated with breakage levels lower than without breakage and look quite similar to one another.

As shown in Figure 23, tooth breakage height \( h_b \) change of higher levels produces obvious changes in the gear dynamic response signals in terms of RMS values. The obvious periodical impulses caused by the cracked tooth appear in the
time domain signal as the time increases. Moreover, it is observed that on a gearbox with a broken tooth the apparition of impulses on the time signals having the periodicity of the defected gear rotational period which will lead to increase the dynamic response.

*(d) Presence of spalling*

Figure 24 represents the dynamic response of the planet gears carrier for a spalled tooth of the pinion (with \( a_e = 0.9 \) mm, \( w_e = 4.6 \) mm and \( b_n = 1.0 \) mm). The same phenomenon as stated for breakage tooth is observed.

In Figure 25, tooth spalling width \((w_s)\) change of higher levels produces obvious changes in the gear dynamic response signals in terms of RMS values. The obvious periodical impulses caused by the cracked tooth appear in the time domain signal as the time increases. Moreover, it is observed that on a gearbox with a spalling tooth the apparition of impulses on the time signals having the periodicity of the defected gear rotational period which will lead to increase the dynamic response.

VI. CONCLUSIONS

1- An analytic model for calculating total mesh stiffness with a gear tooth cracked at different levels is developed. The numeric values of the calculated total mesh stiffness are input into a 12-degree-of-freedom lumped parameter model to simulate the dynamic response of the pair of meshing gears under different deterioration levels of the tooth crack on the planet gear.

2- The effect of spalling and tooth breakage on this stiffness was modelled and computed. The proposed analytical method allows quantifying the reduction of the total meshing stiffness usually observed in these cases of faults. It is a simple method that can give valuable information on such changes in stiffness.

3- Through computer simulation, the effects of tooth crack, breakage and spalling on the dynamic response of wind turbine gearbox with spur gears. A pair of meshing spur gears consisting of a perfect gears (sun and ring) and a planet with a cracked, breakage and spalling tooth have been analyzed.

VII. ACKNOWLEDGEMENTS

The authors acknowledge the support of Science and Technology Development Fund (STDF), Egypt, through the awarded grant No. ID 1484, on monitoring of wind turbine gearbox. The authors would like to thank University of Helwan, which made this study possible.

REFERENCES


Table 1 Technical Parameters for Helical Gearbox (2-Gear Wheel and 2-Pinion Wheel)

<table>
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<th>Remarks</th>
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Table 2 Technical Parameters for Planetary Gearbox (3 Planet Gears)

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Figure 1 The cracked tooth

Figure 2 The cracked tooth model
Figure 3  Schematic graph of spalling

Figure 4  Schematic graph of breakage

Fig. 5 Shape change of the cross sections Si due to defects: (A) spalling, (B) breakage

Figure 6  Wind turbine gearbox system
Figure 7 Total effective meshing stiffness for helical gears

Figure 8 Total effective meshing stiffness for internal meshing

Figure 9 Total effective meshing stiffness for external meshing

Figure 10 Total meshing stiffness for external gears meshing at crack depth of $q_1=1$ mm

Figure 11 Total meshing stiffness for external gears meshing at crack depth of $q_1=3$ mm

Figure 12 Comparison between external and internal gears meshing at different crack depths

Figure 13 Total meshing stiffness for external gears meshing

Figure 14 Comparison between external and internal gears meshing at different spalling widths
Theor Experi

Figure 19 Frequency-domain of the dynamic response of the planet carrier
Cracked gearbox, Speed 40 rpm, Torque load 40 Nm

Figure 20 Time history of the dynamic response of the planet gear

Figure 21 Comparison between the dynamic response of planet gear and carrier

Figure 22 Time history of the dynamic response of the planet gear

Figure 23 Comparison between the dynamic response of planet gear and carrier

Figure 24 Time history of the dynamic response of the planet gear

Figure 25 Comparison between the dynamic response of planet gear and carrier
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