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A study of mechanical behaviour of the double-row tapered ring bearing for the main shaft of a direct-drive wind turbine

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Certificate of Original Authorship

I, Jingyang Zheng certify that the research work in this thesis, submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy, in the School of Mechanical and Mechatronic Engineering, Faculty of Engineering and Information Technology in the University of Technology Sydney.

I also certify that the thesis has been written by me. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

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Nomenclature

Symbols	Definitions
a	Semi-major axes of the elliptical contact area
a^*	Values of dimensionless quantity
a_1	Bearing reliability factor
a_2	Bearing material factor
a_3	Bearing application factor
a_{ISO}	Combined bearing life modification factor
$\ddot{a}(t)$	Node acceleration vectors
$\dot{a}(t)$	Node velocity vectors
A	Mass matrix vector
A	Material constant parameters
b	Semi-minor axes of the elliptical contact area
b^*	Values of dimensionless quantity
B	Damping matrix vector
c	Contact stiffness dependent on the elastic modulus and contact geometry of rollers and rings
c_f	Contact stiffness dependent on the material properties of bearing elements
C	Stiffness matrix vector
C	Basic dynamic load rating of bearings
d_m	Pitch diameter of the bearing
D_m	Mean roller diameter of the bearing
D_{max}	Diameter of roller large-end

D_{min}	Diameter of roller small-end
$D_{we,M,k}$	k^{th} slice diameter of a tapered roller in right and left row
D_{ij}	Coefficients
$D_{s,ij}$	Coefficients with friction force
e	Weibull slope
E	Second type complete elliptic integrals
E_M	Elastic moduli of two contact bodies
E'	Reduced elastic modulus
f_M	Equivalent force of each raceway
$f(x,y)$	the initial separation of roller and raceway
F	First type complete elliptic integrals
F	External force
$F_{M,r}$	Resultant force and moment on a roller
$F_{t,M,N}$	Thrust contact friction
F_g	Resultant force of inner ring in the global coordinate system
G_M	Shear moduli of two contact bodies
h	Material constant parameters
h_k	Crowned drop at k^{th} lamina of a roller
$J_{M,r}$	Jacobian matrix
K_M	Transformation matrices
K_N	Contact constant
$KD_{M,r}$	Transformation matrices
$KD_{M,i}$	Transformation matrices
l_f	Straight-line distance between P_M and intersection points between

	flange forces and roller centre line
l_k	Position of k^{th} slice
l_{we}	Effective contact length between roller and raceway
L_{10}	Basic rating life of bearings (in millions of revolutions)
m	Height of rectangular contact area
m_r	Mass of roller
$M_{M,N}$	Resultant normal moment along the contact of roller
n	Width of rectangular contact area
n'	total number of rectangular segments with $n' = n \times m$
N	Millions of stress cycles
$o_{M,i}$	Origin of local raceways cylinder coordinate systems
$o_{M,r}$	Origin of local roller cylinder coordinate systems
p	Contact pressure
p_L	Load-life exponent
p_{max}	Maximum contact pressure
p_j	Contact stress over segment j and is assumed to be constant over the segment's area
$p(x, y)$	Contact pressure at point (x, y)
P	Dynamic equivalent load of bearings
P_M	Origin of local roller inclined coordinate systems
$q_{M,N,k}$	Contact force at the k^{th} slice
$Q_{M,N}$	Resultant normal load along the contact of roller
$Q_{M,f}$	Flange contact force between roller big end and inner ring
Q_{Nq}	Rolling element loads among the contact areas

$Q_n(t)$	Node load vector
r_i	Inner raceway radius in the middle of contact length
r_m	Pitch radius of bearing
r_o	Radius of outer raceway in the middle of contact length
R_1	Radius of elastic body 1
R_2	Radius of elastic body 2
R_s	sphere radius of roller big end
R_M	Transformation matrices
RD_M	Transformation matrices
S	Survival probability
$u_{M,k}$	Inner ring displacement and rotation vectors in inclined local coordinate system
$u_{M,R}$	Inner ring displacement and rotation vector in local roller cylinder coordinate system
$v_{M,k}$	Roller displacement and rotation vectors in inclined local coordinate system
$v_{M,R}$	Roller displacement and rotation vector in local roller cylinder coordinate system
V	Volume of the material
$w(x,y)$	Contact deformation between two elastic bodies at point (x,y) without the friction force
$w_s(x,y)$	Contact deformation between two elastic bodies at point (x,y) with friction force
x	$x-$ axis in global inertial coordinate system

$\bar{x}$	Distances of the centre of segment j relative to the centre of segment i, $\bar{x} = x_j - x_i$
$x_{M,i}$	$x -$ axis in local raceways cylinder coordinate systems
$x_{M,r}$	$x -$ axis in local roller cylinder coordinate systems
y	$y -$ axis in global inertial coordinate system
$\bar{y}$	Distances of the centre of segment j relative to the centre of segment i, $\bar{y} = y_j - y_i$
$y_{M,i}$	$y -$ axis in local raceways cylinder coordinate systems
$y_{M,r}$	$y -$ axis in local roller cylinder coordinate systems
z	$z -$ axis in global inertial coordinate system
$z_{M,i}$	$z -$ axis in local raceways cylinder coordinate systems
$z_{M,r}$	$z -$ axis in local roller cylinder coordinate systems
Z	Number of rollers in a row
Z_0	Depth at which the largest orthogonal shear stress appears
α	Inclined angle of main shaft
α_i	Contact angle of inner ring
α_m	Mean contact angle
α_o	Contact angle of outer ring
α_f	Inner ring flange angle
$\beta_{M,N}$	Relative misalignment
γ_x	Angle displacement of inner ring in $x -$ axis
γ_y	Angle displacement of inner ring in $y -$ axis
δ	Displacement of the inner ring
δ_x	Displacement of the inner ring in $x -$ axis

δ_y	Displacement of the inner ring in $y-$ axis
δ_z	Displacement of the inner ring in $z-$ axis
δ^*	Values of the dimensionless quantity
δ_e	Relative deformation between the contacting elastic bodies
$\delta_{M,f}$	Flange contact deformation between roller big end and inner ring
$\delta_{M,N}$	Contact deformations between rollers and raceways caused by translation motion
$\delta_{M,N,k}$	Contact deformations of the k^{th} slice on right and left rows
δ_{Nq}	Deformation of contact area
ε	Roller semi-cone angle
ε_δ	Tolerances for inner race displacement
ε_v	Tolerances for roller displacement
ξ	$\xi-$ axis in coordinate system $(P_M, \xi_M, \eta_M, \zeta_M)$
η	$\eta-$ axis in coordinate system $(P_M, \xi_M, \eta_M, \zeta_M)$
ζ	$\zeta-$ axis in coordinate system $(P_M, \xi_M, \eta_M, \zeta_M)$
θ_M	Rotation angle of inner ring
λ	Half angular extent of spherical roller end
μ_o	Nominal angle between the centre line of a roller and the flange contact line of inner ring
μ_M	Angle transformed by μ_o to account for the shift of contact points between the big-end of roller and the flange of raceway under low-speed and heavy-load conditions
μ_z	Friction coefficient between roller and races
ρ	Radius of curvature

$\sum\rho$	Sum of curvature radius
σ_i	Value of fatigue criterion related stress
σ_{ui}	Critical value of the fatigue criterion
τ_0	Largest orthogonal shear stress in contact zones
ν_M	Poisson's ratios of two contact bodies
$\psi_{M,i}$	Revolution angle of inner raceway around the $z_{M,i}-$axis in the coordinate system $\left(o_{M,I}, x_{M,I}, y_{M,I}, z_{M,i}\right)$
$\psi_{M,r}$	Revolution angle of roller around the $z_{M,r}-$axis in the coordinate system $\left(o_{M,r}, x_{M,r}, y_{M,r}, z_{M,r}\right)$
ϕ_M	Rotation angle of a roller
$\omega_{M,cage}$	Angular velocity of cages
$\omega_{M,r}$	Rotational velocity of rollers
$\omega_{M,or}$	Orbital revolution velocity of rollers
ω_o	Velocity of outer ring
Δl_k	Width of the slice
ΔS_i	Survival probability
Δt	Time increment
ΔV_i	Volume element
Δ_x	Radial clearance
Δ_z	Axial clearance
superscript	Definitions
j	The j^{th} roller
f	Friction force
Subscript	Definitions

i	Inner ring
M	$M = 1, 2$ indicate the right row and left row, respectively
N	$N = i, o$ indicate the inner and outer rings, respectively
o	Outer ring
r	Roller

List of figures

List of tables

Table of Contents

Abstract

Double-row tapered roller bearings (TRBs) are one of the main components of a modern direct-drive wind turbines, TRBs are commonly used to support the main shafts of wind turbines since faults can lead to the malfunctions and downtime of wind turbines. In recent decades, some numerical approaches have been proposed for calculating the contact force and pressure distribution of double-row TRBs. Nevertheless, most of the existing studies failed to consider angular misalignment between the inner and outer rings as well as the friction force between the rollers and raceways. A fatigue life analysis of roller bearings is typically performed for bearings under constant rotating speed and invariant loading conditions. The bearings used in floating direct-drive wind turbines, they often experience oscillating motions with varying loading patterns; thus, for which the standard fatigue life analysis is not valid in this case due to the presence of fluctuating loads.

Notably, a quasi-static state does not exist for bearing in the actual operating condition. Since the dynamic model is unable to show the detailed dynamic mechanical behaviour of double-row TRBs such as the contact area stress, total displacement of bearing components of bearings, velocity and acceleration of rolling elements (by considering the combined radial and axial load), the angular misalignment of roller and inner ring, roller skewing conditions, components vibration characteristics and roller-end flange friction a new general dynamic model was proposed in this section based on the previous studies. This is because most of the previous studies are unsuitable for real working conditions.

However, to verify the proposed dynamic model, a simplified finite element analysis (FEA) model was also established using the commercial software ANSYS Workbench.

Hence, the data obtained from this dynamic behaviour analysis can be used to implement the fatigue life prediction for the double-row TRBs, which can significantly benefit their design and manufacturing.

This thesis presents a comprehensive quasi-static model to investigate the internal load and contact pressure distribution in a double-row TRB by considering the angular misalignment, the combined external loads and friction force. Most importantly, the presented numerical model was validated by other published references and the simplified FEA model. First of all, it was found that a small misalignment angle between the inner and outer rings can cause a significant change in the magnitude and distribution of the contact force and pressure. A double-row TRB with a crowned roller profile exhibits a substantial improvement in contact pressure distribution by eliminating the contact pressure. Peak contact pressure can be significantly reduced on a roller with crowned profile, even with a misaligned bearing. Comparisons of the simulated contact loads and pressure distributions demonstrate that need to consider angular misalignment and friction force in the modelling of large size and heavy-load double-row TRBs.

Furthermore, this thesis presents a fatigue life analysis for double-row TRBs under oscillating external load and speed conditions in which the double-row TRB was used to support the main shaft of a large modern direct-drive wind turbine. Meanwhile, the proposed comprehensive and quasi-static model of the double-row TRB outlines the internal load distribution of rollers. The contact pressure of rollers is then provided based on an iterative scheme using the elastic contact model. Thereafter, the basic rating life of the double-row TRB under an oscillating external load and speed is provided to calculate the fatigue life. Numerical simulations were also performed to investigate the effects of the oscillating load and speed, angular misalignment, and internal clearance on the fatigue life of this bearing.

Finally, the simulation results of the dynamic model analysis indicated that the combined radial load and pure radial load have a significant effect on the vibration of rollers and the inner ring of double-row TRBs in a floating direct-drive wind turbine. Meanwhile, the angular misalignment of the inner ring also affects the vibration of the rollers and the inner ring itself. With an increase in the misalignment angle, the vibration of roller elements became increasingly apparent. The vibration frequency of rollers and the inner ring gradually decreased with an increasing misalignment angle. Additionally, the vibration of components in the double-row TRBs is sensitive to the initial axial preload.

Keywords: Direct-drive wind turbine, Double-row tapered roller bearing, Angular misalignment, Contact pressure, Load distribution, Deformation, Roller profile, Fatigue life, Preload, Clearance, Oscillating load, Rotating speed, Dynamic behaviour, Vibration, Displacement.