Load distribution analysis in hull and turret interface bearing of armoured fighting vehicles

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Abstract. There are simple and well established formulae and data sheets for load calculations and design of uniaxial and biaxial loaded bearings. The load calculations and bearing design are complex and challenging, when the bearing is subjected to multi-axial loads along with moment and shock load. In case of Armoured Fighting Vehicles, the hull & turret interface bearing is subjected to combined heavy axial, radial load coupled with heavy shock load and tilting moments. This may cause significant changes in bearing deflections, contact stress and fatigue endurance compared to simpler load distributions. The empirical formulae for calculation of load distribution on each rolling element lead to indeterminate equations with many unknown variables. In the present study, a comprehensive procedure for determining the loading pattern and also numerical estimation of maximum load, maximum contact stress and the load distribution on each rolling element of the bearing are analysed.

1. Introduction

The Hull Turret Interface Bearing (HTIB) enables to rotate the turret of an Armoured Fighting Vehicle (AFV) by 360° through the turret control drive. Figure 1 shows the Hull, Turret and the Interface Bearing of a typical AFV.
The firing force is transmitted from the turret to the hull through the bearing in addition to the weight of the turret and weapon system (W). Firing force originates from recoiling during main gun firing (F_recoil) and the direction of the force depends on the gun position in the Elevation (El) plane.

The load distribution of the slewing bearing under different operating conditions are mainly investigated.

2. Loads acting on the bearing
The various loads acting at different locations about a local coordinate system with origin at the centre of the HTIB are shown in figure 2.
2.1. Effect of gun elevation

In order to meet longer range requirements and engaging targets from negative slope, AFVs have provision for elevating / depressing the main gun through a specific range of angles in the El plane.

The forces and moments acting on the bearing due to the recoil force, depends upon the instantaneous gun position angle in El plane.

The effect of gun position angle in the El plane on the recoil force components is depicted in figure 3.

![Figure 3. Recoil force components.](image)

The resultant force components for the typical system of forces acting on the bearing are as follows:

Axial force experienced by the bearing,

\[ F_a = W + F_{recoil} \sin \varphi \]  

(1)

where, W is the dead load of the weapon system (kN) and \( \varphi \) is the gun elevation angle (radian)

Radial force experienced by the bearing,

\[ F_r = F_{recoil} \cos \varphi \]  

(2)

Due to the offset distance between C.G. of the turret & the bearing and also due to the recoil force offset (H₁), the corresponding moments act on the bearing. These moments are depicted in figure 4.

![Figure 4. Loads on raceways.](image)
Considering the forces acting on the HTIB and taking into account the geometry of the HTIB with respect to the main gun in straight and elevated conditions, the moments acting on HTIB can be formulated. The resultant moment components for the system of forces acting on the bearing are as follows:

Moment about x-axis,

\[ M_x = W_y \]  

Moment about y-axis,

\[ M_y = W x - F_{recoil} \cos \phi (H_1 - x_1 \tan \phi) \]  

Resultant Moment,

\[ M = \sqrt{(M_x^2 + M_y^2)} \]

2.2. Loads acting on a rolling element and contact stresses

The load supported by the bearing is distributed on the individual rolling elements. The loads are distributed among the rolling elements in a pattern depending on the gun position. It is required to find the maximum rolling element load to know the stresses developed.

For finding the rolling element loads, the two row of rolling elements are considered as non linear springs [1] that connect the inner and outer raceways and experiencing compression when the race ways approach. Rolling elements represented as springs are depicted in figure 5.

Figure 5. Loads on rolling elements.

A typical condition of the outer race deforming while the inner race being fixed is considered for analysis. The deformed bearing under the axial, radial and moment loads is depicted in figure 6.
Figure 6. Loads on raceways.

Considering the geometry of compression, the rolling elements may be represented as the nonlinear springs for the purpose of analysis. The nonlinear spring representation is made through making the spring stiffness as the value of contact stiffness between the rolling element and the raceway [2]. The compression and change of initial contact angle are depicted in figure 7.

Figure 7. Change in length and initial contact angle.

In the above figure, ‘A’ denotes the diametrical length of the rolling element at a mid-section, taken for the study of deformation. $\theta_T$ and $\theta_L$ represents the initial contact angles of top and bottom rolling elements respectively. $\theta_T = \theta_L = 0$. $\theta'_T$ and $\theta'_L$ are the change in the contact angles of the top and bottom rows respectively after deformation.

$P_t$ and $P_o$ denote the contact points of the top row rolling element, with the inner race and outer races respectively. The similar points of the bottom row rolling element are denoted by $Q_t$ and $Q_o$ respectively. $P_t'$, $P_o'$, $Q_t'$ and $Q_o'$ denote the deformed contact points, accordingly.

Deformations $\delta_a$, $\delta_r$, and $\alpha$ represent the axial, radial and angular deformations respectively. The deformation of raceway causes compression of the rolling elements that leads to change in initial contact angle, $\theta$. The Rolling Element Load and deformation are related by,

$$Q \propto \delta^n$$  \hfill (6)

where, $n$ is the load deflection exponent $= 1.11$ for rolling elements.

Introducing the contact stiffness as the constant of proportionality in the relation between the rolling element and raceway [1],

$$Q = K_n \delta^{1.11}$$  \hfill (7)
K_n denotes the non-linear stiffness of the springs. where, the total sum of deformations of inner and outer raceway contacts is given by [1],

$$\delta_s = \delta_i + \delta_o$$  \hspace{1cm} (8)

The equivalent stiffness of the inner and outer rolling element-raceway contacts is given by,

$$K_s = \left[ \frac{1}{K_i} + \frac{1}{K_o} \right]^{1.11}$$  \hspace{1cm} (9)

where, $K_i$ & $K_o$ represents the stiffness of the inner and outer raceway rolling elements respectively. $K_s$ denotes the equivalent stiffness of the inner and outer raceway rolling elements. Hence the rolling element load is given by,

$$Q = K_s \delta_s^{1.11}$$  \hspace{1cm} (10)

3. Deformation formulation & solution
3.1. Formulation for deformation
Based on the applied loads, the bearing raceway deforms and that leads to different ways of sharing the load by the rolling elements. Therefore by determining the load distribution, it is possible to know the individual rolling element loads.

The following formulation is used to find the load distribution of the bearing, by relating the deformations of the whole bearing to the individual rolling element deformation [3],

$$A + \delta_1 = A_1$$  \hspace{1cm} (11)
$$A + \delta_2 = A_2$$  \hspace{1cm} (12)

*(subscript 1 denotes the top row rolling element parameters and 2 denotes the bottom row rolling element parameters)*

Hence the total deformation in the top row and bottom row rolling element are respectively given by,

$$\delta_1 = A_1 - A$$  \hspace{1cm} (13)
$$\delta_2 = A_2 - A$$  \hspace{1cm} (14)

where, $A_1$ & $A_2$ are the deformed diametrical lengths of the top and bottom row rolling elements respectively. Since $A_1$ & $A_2$ are functions of the angular location of the rolling element ($\psi$), henceforth they may be referred as $A_{1\psi}$ & $A_{2\psi}$ respectively. $A_{1\psi}$ & $A_{2\psi}$ are given by,

$$A_{1\psi} = \left( (A \sin \theta + \delta_a + R_i \alpha \cos \psi)^2 + (A \cos \theta + \delta_r \cos \psi + \frac{c}{2} \alpha \cos \psi)^2 \right)^{1/2}$$  \hspace{1cm} (15)
$$A_{2\psi} = \left( (A \sin \theta - \delta_a - R_i \alpha \cos \psi)^2 + (A \cos \theta + \delta_r \cos \psi - \frac{c}{2} \alpha \cos \psi)^2 \right)^{1/2}$$  \hspace{1cm} (16)

Inserting the eqns. (15) & (16) in (13) & (14)

$$\delta_{1\psi} = A \left\{ \left( (\sin \theta + \delta_a + R_i \alpha \cos \psi)^2 + (\cos \theta + \delta_r \cos \psi + \frac{c}{2} \alpha \cos \psi)^2 \right)^{1/2} - 1 \right\}$$  \hspace{1cm} (17)
\[ \delta_{2\psi} = A \left\{ \left[ \left( \sin \theta - \delta_a - R_l \bar{a} \cos \psi \right)^2 + \left( \cos \theta + \delta_r \cos \psi + \frac{c}{2} \bar{a} \cos \psi \right)^2 \right]^{1/2} - 1 \right\} \]  
(18)

Here \( \delta_a = \frac{\delta_a}{A}, \delta_r = \frac{\delta_r}{A} \).

From equation (7),

\[ Q_\psi = K_n \delta_\psi^{1.11} \]  
(19)

Inserting equations (17) & (18) in (19),

\[ Q_{1\psi} = K_n \times A^{1.11} \left\{ \left( \sin \bar{\theta} + \delta_a + R_l \bar{a} \cos \psi \right)^2 + \left( \cos \bar{\theta} + \delta_r \cos \psi + \frac{c}{2} \bar{a} \cos \psi \right)^2 \right\}^{1/2} \]  
(20)

\[ Q_{2\psi} = K_n \times A^{1.11} \left\{ \left( \sin \bar{\theta} - \delta_a - R_l \bar{a} \cos \psi \right)^2 + \left( \cos \bar{\theta} + \delta_r \cos \psi - \frac{c}{2} \bar{a} \cos \psi \right)^2 \right\}^{1/2} \]  
(21)

At any azimuth position of the rolling element, the changed contact angles \( \theta_1 \) & \( \theta_2 \) can be determined by, [4]

\[ \sin \theta_1 = \frac{\left( \sin \theta + \delta_a + R_l \bar{a} \cos \psi \right)}{\left( \sin \bar{\theta} - \delta_a - R_l \bar{a} \cos \psi \right)^2 + \left( \cos \bar{\theta} + \delta_r \cos \psi + \frac{c}{2} \bar{a} \cos \psi \right)^2}^{0.5} \]  
(22)

\[ \cos \theta_1 = \frac{\left( \cos \theta + \delta_r \cos \psi - \frac{c}{2} \bar{a} \cos \psi \right)}{\left( \sin \bar{\theta} - \delta_a - R_l \bar{a} \cos \psi \right)^2 + \left( \cos \bar{\theta} + \delta_r \cos \psi + \frac{c}{2} \bar{a} \cos \psi \right)^2}^{0.5} \]  
(23)

Since, \( F_a, F_r \) and \( M \) are acting on the bearing, for static equilibrium to exist,

\[ F_a - \sum_{\psi=1}^{Z} (Q_{1\psi} \sin \theta_1 - Q_{2\psi} \sin \theta_2) = 0 \]  
(24)

\[ F_r - \sum_{\psi=1}^{Z} (Q_{1\psi} \cos \theta_1 + Q_{2\psi} \cos \theta_2) = 0 \]  
(25)

\[ M - \sum_{\psi=1}^{Z} (Q_{1\psi} \sin \theta_1 - Q_{2\psi} \sin \theta_2) \cos \psi + \frac{c}{2} \sum_{\psi=1}^{Z} (-Q_{1\psi} \sin \theta_1 - Q_{2\psi} \sin \theta_2) = 0 \]  
(26)

where, \( F_a, F_r \) and \( M \) are obtained from (1), (2) & (5) respectively.

### 3.2. Solution for rolling element loads and deformation

The above simultaneous non-linear equations having variables \( \delta_a, \delta_r \) and \( \alpha \) are solved using a code in MATLAB 8.0. By knowing the deformations \( \delta_a, \delta_r \) and \( \alpha \), individual rolling element loads \( Q_{1\psi}, Q_{2\psi} \) and changed contact angles \( \theta_{a1\psi}, \theta_{a2\psi} \) are found.

A flowchart depicting the methodology of solution based on Hooke Jeeves Pattern Search method [4] is furnished in figure 8.
Figure 8. Methodology of solution
4. Calculations for a typical HTIB
The deformations $\delta_a, \delta_r$ and $\alpha$, individual rolling element loads $Q_{1\Phi}, Q_{2\Phi}$ and changed contact angles $\theta_{1\Phi}, \theta_{2\Phi}$ are calculated, considering a double row roller bearing with the following parameters,

Dead load of turret and weapon system, $W = 220$ kN
Recoil force, $F_{\text{recoil}} = 800$ kN (Firing)
Initial contact angle $\theta = 45^0$
Number of rolling elements per row, $z = 403$
Rolling element diameter = 12 mm
Bearing pitch diameter, $d_m = 2016$ mm
Effective length, $l_{\text{eff outer}} = 6.25$ mm
Distance between two races, $C = 12.5$ mm
Sum of raceway-rolling element, $\Sigma \rho = 1/6$ mm$^{-1}$
Contact stiffness, $k_{\rho} = 32.86 \times 10^5$ N/mm

4.1. Variation of maximum rolling element load with gun elevation
Analysis was carried out to find out the load variation of each rolling element at different firing angles along with static load acting on the bearing. Accordingly, graphs depicting the variation of each rolling elements in top and bottom rows at maximum depression, zero degree and maximum elevation firing angles (-10, 0, 20) were plotted.

The X-Y graph depicting the load variation of top and bottom row rolling elements at maximum depression angle of firing is shown in figure 9.

![Figure 9](image)

**Figure 9.** Load variation at maximum depression angle of firing

The surface plot wherein XY plane represents the spatial distribution of rolling elements in the azimuth plane and Z represents the rolling element load, depicting the load variation of top and bottom row rolling elements at maximum depression angle of firing is shown in figure 10. The direction of the recoil force acting on the bearing is also indicated.
A polar diagram is plotted for the rolling element load, $Q$ against the angular location of the rolling elements, $\psi$ depicting the load variation of top and bottom row rolling elements at maximum depression angle of firing as shown in figure 11.

The X-Y graph depicting the load variation of top and bottom row rolling elements at 0° angle of firing is shown in figure 12.
The surface plot wherein XY plane represents the spatial distribution of rolling elements in the azimuth plane and Z represents the rolling element load, depicting the load variation of top and bottom row rolling elements at 0° angle of firing is shown in figure 13. The direction of the recoil force acting on the bearing is also indicated.

![Figure 13. Load variation at 0° angle of firing.](image)

A polar diagram is plotted for the rolling element load, $Q$ against the angular location of the rolling elements, $\psi$ depicting the load variation of top and bottom row rolling elements at 0° angle of firing is shown in figure 14.

![Figure 14. Load variation at 0° angle of firing.](image)

The X-Y graph depicting the load variation of top and bottom row rolling elements at maximum elevation angle of firing is shown in figure 15.

![Figure 15. Load variation at maximum elevation angle of firing.](image)
The surface plot wherein XY plane represents the spatial distribution of rolling elements in the azimuth plane and Z represents the rolling element load, depicting the load variation of top and bottom row rolling elements at maximum elevation angle of firing is shown in figure 16. The direction of the recoil force acting on the bearing is also indicated.

![Figure 16](image)

**Figure 16.** Load variation at maximum elevation angle of firing.

A polar diagram is plotted for the rolling element load, Q against the angular location of the rolling elements, \( \psi \) depicting the load variation of top and bottom row rolling elements at maximum elevation angle of firing is shown in figure 17.

![Figure 17](image)

**Figure 17.** Load variation at maximum elevation angle of firing.

In general, the rolling element load depends on the relative location of the top and bottom rows of rolling elements. In the event of the raceways coming together i.e., raceways approaching, the rolling element load increases and if they move apart, the load tends to decrease. Any downward resulting force causes the top row rolling element-raceway contact to compress more than the bottom row. This is the reason for higher contribution of top row in the load distribution of such loading conditions [5].

It is evident from the plots of rolling element load under various conditions that the rolling element load acting on the top row rolling elements is higher compared to the bottom row. This is due to the loading pattern of the bearing, considering the forces due to gravity and recoil in positive elevation angles.
A graph depicting the variation of maximum rolling element load in top and bottom rows with firing angles is shown in figure 18.

![Graph](image)

**Figure 18.** Variation of maximum rolling element load with firing angles.

5. Contact stress and safety factor

5.1. Hertz Contact Stress

Point contact and line contacts under load become area contact with dimensions that are very small compared to that of contacting bodies [1] and hence limiting the loads at the contact to be finite and the values are given as

\[
b = 3.35 \times 10^{-3} \left( \frac{Q}{l_{eff} \Sigma \rho} \right)^{1/2}
\]

(27)

\[
\sigma_{max} = \frac{2Q}{\pi lb}
\]

(28)

Where,

\( \Sigma \rho \) - Sum of raceway rolling element curvatures (mm)

\( l_{eff} \) - Effective contact length (mm)

\( Q \) - Rolling element load (N)

\( b \) - Contact surface width (mm)

\( \sigma_{max} \) - Maximum Hertzian contact stress (Pa)

5.2. Safety Factor

Safety factor (SF) of the bearing is the ratio of allowable rolling element load to the actual maximum rolling element load. The SF is required to be greater than one to have margin in static load capacity [6]. The SF is expressed in terms of maximum Hertz contact stress as given by,

\[
SF = \left( \frac{8000}{\sigma_{max}} \right)^2
\]

(29)

The SF for various load cases considered is depicted in figure 19.
6. Conclusion
By analyzing the loading pattern, the load distribution on the rolling elements of the bearing has been determined. Accordingly, the maximum load, maximum contact stress have been calculated. Individual rolling element loads are thus used to find the bearing design parameters such as the subsurface stresses and safety factor.

7. References
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Figure 19. Safety factor for various load cases