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ANALYSIS AND OPTIMISATION OF TORQUE ARM BRACKET

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Abstract: In this study, the failure (fracture) of Torque arm bracket was selected as investigation topic. The focus of this project is to investigate how a crack propagates and grows in a typical direction of Torque arm bracket However; this study is intended for introducing fracture mechanics from an application viewpoint. It essentially focuses on both stress and fatigue analyses.WAG-9 type electric locomotives, of Indian Railways' fleet used for Goods train and passenger train maintained at Ajani, loco shade of central railway has the history of frequent failure of the torque arm bracket .The study of failures revealed that, the exact time and instance of failure cannot be as ascertained as it does not halt the loco and it is noticed only when the loco comes for maintenance. The problem identified here is the crack failure of Torque arm bracket that has been solved by analytical and FE analysis. The cracks observed near the corner of bracket failed due to the varying load which cause for the fatigue failure. This problem has been analyzed by considering the two cases, static load and dynamic load.

Introduction

Torque Arm Bracket have heavy amount of stresses, vibration and loads they are prone to failure since they have to withstand heavy dynamic loads in the operating conditions. To ensure that the design is foolproof, it is necessary to bring down the stress levels to a permissible limit. In the presented analysis, Engine mounting bracket is to be analyze evaluate its structural performance and design optimization ,the bracket serves as the connector for traction motor and engine.

Fig 1.3 Torque Arm Bracket

Problem Statement

Cracks often develop in the corners of a structural member due to high stress concentration factor in those areas. If one can calculate the rate of crack growth, an Engineer can schedule inspection accordingly and repair or replace the part before failure happens. Moreover, being able to predict the path of a crack helps a designer to incorporate adequate geometric tolerance in structural design to increase the part life. While producing durable, reliable and safe structures are the goals of every component manufacturer, there are technical challenges that are not easy to be solved.

Given limited engine design space, engineers strive to optimize bracket geometry to produce high efficient and high performance engines that will operate at minimum weight and cost. Engineers materials from bracket and design the thinnest possible brackets. Benefits from this approach include reduced weight, and smaller probability of encountering brittleness inducing microstructure defects. The focus is to investigate the corner crack growth in a titanium-based alloy bracket. This paper will examine the stresses near the crack tip, compute the stress intensity factors and compare it against material toughness to determine the influence of the crack on the bracket.

Literaure Review:

'Priscilla L. Chin' says that, how to investigate a crack propagates and grows in a typical bracket. The finite element program

ANSYS and the crack growth program FRANC3D were used to simulate crack growth and to compute the stresses and the stress-intensity factor. A specific bracket design was selected and a corner crack was investigated. The stresses near the crack tip are compared against the yield strength of the material. The Mode I stress-intensity factor is compared against the material's fracture toughness. The results show that the bracket can tolerate small cracks in the structure. [3]

'Sameer U. Kolte, David Neihguk, Abhinav Prasad' says that, Structural analysis is performed to check durability of specified part for a given load and support conditions. For the component to be safe structurally, in any domain, the stresses generated should not exceed the yield strength of the material, considering possibility of fatigue failure, the component is optimized such that stresses generated do not exceed the endurance strength of the material. [4]

Design modeling of a torque Arm Bracket:
First of all, let us find the distance of centre of gravity of the section at X–X.
Let y = Distance of centre of gravity (G) from the top of the flange.

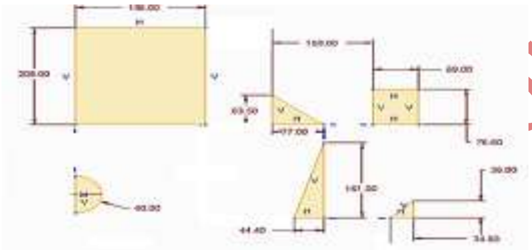

Fig 3.2 Centre of gravity

3.2 Available data

For Steel Material,
Ultimate strength, Sut = 540 MPa
Yield Strength, Syt = 0.6 Sut = 324 MPa
Considering Factor of Safety = 2
Hence, Allowable Design Stress= 324/2 = 162 MPa [13]
W = 10 kN = 10 × 10^3 N

3.3 The Centre of gravity of various sections.

Table 3.1Distance from Centre of gravity of various sections

Part	Area (mm^2)	$\bar{x}$	$\bar{y}$	$\bar{x}$ A	$\bar{y}$ A
1	40560	97.5	104	3.95×10^6	4.22×10^6
2	-2444.75	25.66	142.66	-0.062×10^6	-0.348×10^6
3	-5278.5	34.5	83.25	-0.182×10^6	-0.439×10^6
4	-2513.3	86	85	-0.216×10^6	-0.213×10^6
5	-3574.2	180.2	53.66	-0.644×10^6	-0.192×10^6
6	-622.8	139	12	-0.0865×10^6	-0.0075×10^6
TOTAL	**26126.45**			**2.75×10^6**	**3.02×10^6**

3.4 Calculations

$$\bar{X} = \frac{\sum A\bar{x}}{\sum A} = \frac{2.75\times10^6}{26126.45} = 105.25mm$$

$$\bar{Y} = \frac{\sum A\bar{y}}{\sum A} = \frac{3.02\times10^6}{26126.45} = 115.6mm$$

Distance of C.G. from the left of the flange,
x_1 = 105.25 mm

and distance of C.G. from the right of the flange,
x_2= 195 – 105.25 = 89.75 mm

Moment of inertia about an axis passing through the centre of gravity of the section,
I_{GG}= 3.02 × 10^6 mm^4

Due to the tilting action of the load W, the cross-section of the bracket X-X will be under bending stress. The upper fibres of the left flange will be under maximum tension and the lower fibres of the web will be under maximum compression.
∴Section modulus for the maximum tensile stress,

$$Z_1 = \frac{I_{GG}}{x_1} = \frac{3.02\times10^6}{105.25} = 28.69\times10^3 mm^3$$

and section modulus for the maximum compressive stress,

$$Z_2 = \frac{I_{GG}}{x_2} = \frac{3.02\times10^6}{89.75} = 33.65\times10^3 mm^3$$

We know that bending moment exerted on the section,

M = 10 × 10^3 × 105.25 = 1052 × 10^3 N-mm

∴ Maximum bending stress (tensile) in the flange,

$$\sigma_{b1} = \frac{M}{Z_1} = \frac{1052.2\times10^3}{28.69\times10^3} = 36.68 N/mm^2$$

and maximum bending stress (compressive) in the web,

$$\sigma_{b2} = \frac{M}{Z_2} = \frac{1052.2\times10^3}{33.65\times10^3} = 31.27 N/mm^2$$

The eccentric load also induces direct tensile stress in the bracket. We know that direct tensile stress,

$$\sigma_{t1} = \frac{Load}{Cross - Sectionalareaofbracketat C.G.}$$

$$\sigma_{t1} = \frac{10\times10^3}{26126.45} = 0.3827\ N/mm^2$$

∴ Maximum tensile stress produced in the section at X–X (i.e. in the flange),

$\sigma_t = \sigma_{b1} + \sigma_{t1}$= 38.68 +0.3827 =37.06 N/mm^2< 162 N/mm^2

and maximum compressive stress produced in the section at X–X (i.e. in the web),

$\sigma_t = \sigma_{b2} - \sigma_{t1}$ = 31.27 -0.3827 =30.88 N/mm

[14]

Table 3.2 Analytical Analysis of Bracket

Sr. No.	Force (N)	Tensile Stress (MPa)	Compressive Stress (MPa)
1	10000	37.07	30.88
2	20000	74.14	61.79
3	30000	111.20	92.69
4	40000	148.27	123.58
5	45000	166.81	139.03

FATIGUE ANALYSIS OF BRACKET:

For Steel Material,

Ultimate strength, Sut = 540 MPa

Yield Strength, Syt = 0.6 Sut = 324 MPa

Endurance Strength, Se = 0.5 Sut = 270 MPa

Considering Factor of Safety = 2

CASE-I:

σ_{min} = 37 MPa

σ_{max} = 167 MPa

$$\sigma_{mean} = \frac{\sigma_{max}+\sigma_{min}}{2} = \frac{167+37}{2} = 102 MPa$$

$$\sigma_{variable} = \frac{\sigma_{max}-\sigma_{min}}{2} = \frac{167-37}{2} = 65 MPa$$

Stress Concentration Factor $= \frac{MaximumStress}{NominalStress} = \frac{37}{30} = 1.23$

Using Soderberg Criteria

$$\frac{1}{F.S.} = \frac{\sigma_{mean}}{\sigma_y} + \frac{\sigma_{variable}K_s}{\sigma_e}$$

$$\frac{1}{2} = \frac{102}{324} + \frac{65\times1.23}{\sigma_e}$$

$$\sigma_e = 431.7 MPa$$

Number of stress cycle, N = 8 × 10^3 cycles (High cycle finite life)

CASE-II:

σ_{min} = 39.58 MPa

σ_{max} = 178.12 MPa

$$\sigma_{mean} = \frac{\sigma_{max}+\sigma_{min}}{2} = \frac{178.12+39.58}{2} = 108.85 MPa$$

$$\sigma_{variable} = \frac{\sigma_{max}-\sigma_{min}}{2} = \frac{178.12-39.58}{2} = 69.27 MPa$$

Stress Concentration Factor $= \frac{MaximumStress}{NominalStress} = \frac{37}{30} = 1.23$

Using Soderberg Criteria

$$\frac{1}{F.S.} = \frac{\sigma_{mean}}{\sigma_y} + \frac{\sigma_{variable}K_s}{\sigma_e}$$

$$\frac{1}{2} = \frac{108.85}{324} + \frac{69.27\times1.23}{\sigma_e}$$

$$\sigma_e = 519.5 MPa$$

Number of stress cycle, N = 4.5 × 10^1 cycles (Low cycle finite life)

CASE-III:

σ_{min} = 33.03 MPa

σ_{max} = 148.64 MPa

$$\sigma_{mean} = \frac{\sigma_{max}+\sigma_{min}}{2} = \frac{148.64+33.03}{2} = 90.8 MPa$$

$$\sigma_{variable} = \frac{\sigma_{max}-\sigma_{min}}{2} = \frac{148.64-33.03}{2} = 57.8 MPa$$

Stress Concentration Factor $= \frac{MaximumStress}{NominalStress} = \frac{37}{30} = 1.23$

Using Soderberg Criteria

$$\frac{1}{F.S.} = \frac{\sigma_{mean}}{\sigma_y} + \frac{\sigma_{variable} K_s}{\sigma_e}$$

$$\frac{1}{2} = \frac{90.8}{324} + \frac{57.8 \times 1.23}{\sigma_e}$$

$$\sigma_e = 323.51 MPa$$

Number of stress cycle, N = 3.4×10^5 cycles (High Cycle finite life)

CASE-IV:

σ_{min} = 30.3 MPa

σ_{max} = 135.6 MPa

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{135.6 + 30.3}{2} = 82.95 MPa$$

$$\sigma_{variable} = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{135.6 - 30.3}{2} = 52.65 MPa$$

Stress Concentration Factor $= \frac{MaximumStress}{NominalStress} = \frac{37}{30} = 1.23$

Using Soderberg Criteria

$$\frac{1}{F.S.} = \frac{\sigma_{mean}}{\sigma_y} + \frac{\sigma_{variable} K_s}{\sigma_e}$$

$$\frac{1}{2} = \frac{82.95}{324} + \frac{52.65 \times 1.23}{\sigma_e}$$

$$\sigma_e = 265.4 MPa$$

Number of stress cycle, N = 1×10^7 cycles (Infinite life) [15]

Fatigue Analysis of Bracket

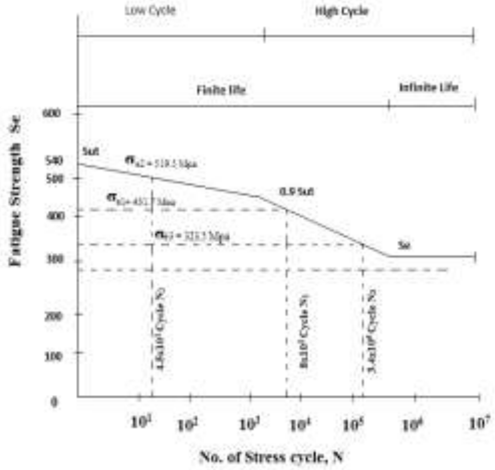

Fig 4.1 Fatigue Analysis of Bracket

TABLE 4.1 Fatigue analysis of bracket

Sr No	Fatigue Strength	Ultimate tensile strength (Mpa)	Endurance strength (Mpa)	Types of Stress Cycle
1	4.5×10^1	540	519.5	Low cycle
2	8×10^3	486	431	High cycle
3	3.4×10^5	270	323.5	High cycle

The analytical and FE analysis of torque arm bracket are compared for all cases. The analytical analysis for the static and fatigue analysis indicates the stresses increases during the variable loads. The tabular representation of the analytical analysis directly shows stresses occurred on the shaft. The calculated load has been applied at the FE analysis and the stresses are compared which gives idea for the analysis of the modification in design. The tabular representation of the FE analysis of current and modified design shows the stresses has been reduced by applying fillets near the corner surface.The FE analysis is also shows the region of high cycle & low cycle near the notch surface from which the failure is known to happen frequently. Hence analysis proves that the bracket is heavily loaded and on some instances like wagons loaded with wet coal responsible for the extra tractive effort developed on the Engine. Further the actual loading condition would have the additional load or shocks as gap in track, alignment errors, jumping of wheel, breaking loads etc. All these factor shall contribute to increase in the load and hence the stress.

Therefore it can be concluded that, there is an urgent need to reduce this stresses to the safer limits. One way to do this would be reduction in pre-stresses by removing the corner of notch into fillet. This shall reduce the pre-stressing of torque arm bracket and traction motor assembly increasing the overall displacement of composite assembly of interference fit. This shall lead to decrease in pre-stressing of torque arm bracket, the life of bracket shall definitely improve.

SUMMERY AND CONCLUSION

The studies executed in the scope of both stress and fatigue analyses of the torque arm bracket have revealed the followings.

1. Stress analyses performed by both analytical and Ansys program show that the shear stress near the corner under the yield strength of the

bracket material. This indicates that a fatigue crack-initiation is maylikely to happen near the corner.

2. Fatigue failure is a tensile stress phenomenon. The elevated tensile stress due to the corner effect creates resolved shear stress. Slip in the crystals occurs when the resolved shear stress attains a critical value, which is characteristic of the material. Thus, the plastic flow resulting in crack-initiation commences at the corner.

3. The static loading is safe and develops tensile stress up to 39.58 N/mm^2 which is increases to the static loading of about 162 N/mm^2.

4. The combined stresses i.e. for static and fatigue analysis reveals the stresses are more than the ultimate shear stresses which cause for the crack propagations.

5. The stress concentration is more at the corner surface of bracket which can be reduces by applying fillet to the bracket and increasing thickness from 57mm to 65mm.

6. Although the nominal stresses in the bracket are well below the yield strength, the elevated stresses near the corner, as stated above, may attain the yield strength due to sharp severity. For this reason, the making of the bracket should be done with great care and all unwanted irregularities such as cracks, notches, corners scratches, cavities, …etc. should never be allowed to occur during manufacturing. Also, as seen from fatigue analysis, presence of a sharp corner shortens the useful life of bracket considerably, relating to lowered endurance limit comparing to that of modified bracket.

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