Fatigue Life Prediction of 180 Ton Lifting Spreader Construction Material ST43 Using S-N Curve

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Abstract. Lifting spreader construction with a maximum capability of 180 tons every day works by carrying transformer loads that vary up to maximum loads. The loads tested were 105 tons, 125 tons, 150 tons and 180 tons. The tool not only experiences vertical tensile load but also experiences horizontal load due to acceleration or deceleration of the overhead crane. To prevent fracture due to fatigue, the lifting spreader needs to be analyzed its service life. Obtained data of mass transformers per year and construction specification will be dynamically analyzed using SAP2000 software to gain information maximum and minimum stress from each of its shell nodes. Data Collection of stress will be processed through the S-N Curve mechanism and Palmgren Miner rules to determine the cyclic load of the fatigue life plate of spreader. This analysis found that maximum capacity 180 tons value Sf above from endurance limit state. And the assessment decline fatigue life of the lifting spreader based data within 45 years. The researcher recommends a decrease in the value of the capacity till to 10 percent for things that are not expected that result in a decrease in the quality of construction.

1. Introduction

Lifting Speader is additional connection from twin hooked overhead cranes. The function of the lifting spreader makes four lifting ropes have an upright direction for transfer of power transformers. So that the four lifting ropes are safe from touching the transformer accessories such as bushings and arresters. The shape of the lifting spreader is like capital H shape with the center of the gravity being used as a fulcrum to twin hooked the overhead crane, while the 4 ends of the H point connect Kevlar cord to the four lift points of the transformer (Figure 1). In such shape model, lifting spreader experiences dynamic and subjected slow swinging while to be lifted and moved.

This work sought on focus finding high stress and analyze the influence of fatigue on critical weld connection location. It is not the entire field that is important, but only the stress field close to the crack tip. Traditionally, fatigue crack initiation has been treated by empirically determining the nominal stress amplitude [1].

3D model lifting spreader is made with meshing shell type by software SAP2000 that has capability static and dynamic analysis. Then results of the stress output are used to determine the damage to the factor D index. Group stress data will be processed through the S-N Curve mechanism and Palmgren Miner rules to determine the cyclic shell fatigue life of the tool plates.
2. Method

Prior to this analysis lifting spreader designed with manual calculation. A 180 tons load is used as a reference calculation. Static calculation part is focused on the arm's length to the point of load. The total stress is sum root square of quadratic bending stresses and three times quadratic shear stresses.

\[ \sigma = \sqrt{\sigma_b^2 + 3\tau_s^2} \]  

(1)

If the total load is 180 tons divided by 3 the result is 130 MPa and if divided by 4 is 97 MPa. These stress number is safe according to allowable stress 175Mpa. Beside that, it carefully tested lifting 180 tons load has done and static hang load for one hour.

A year after manufacture of lifting spreader, the need working life of this tool to be considered because decline fatigue strength. Fatigue life can be modeled as directed and idealized curve S-N, where S is the effective stress range, which can be developed using Miner’s linear fatigue damage relationship. N is related to applied stress range. Assessment of fatigue life on spreader lifting is difficult job, because several mechanisms interact in the process [2]. In dynamic analysis finding critical stresses can be obtained from numerical approach.

2.1. Modal and Time History Dynamic Analysis

Modal analysis is dynamic properties of structure under vibration excitation to seek mode of shape and natural frequency by ignoring damping [3]. Next applied sinusoidal time history process with 1g horizontal direction (Figure 2).
Figure 3. Numerical process with SAP2000 and manual calculation.

Lifting spreader is made of St42 plate material with variety of plate variations. On the lifting strap the shape of the arm is 15mm thick rectangular square hollow plate and add with plate radius at top. Kevlar cords are guarded by several reinforcing plates so they are not detached from the stand. The thickness main plates are 60 mm, others reinforcing plates are 15 mm and 30 mm thick. The properties strength St42 equal to ASM steel grade (a) 1015 with 320 MPa yield strength and 415 MPa tensile strength. A 2.05 x 105 MPa Young modulus was adopted for the steel beams. The weld material used was E70XX with 415 MPa yield strength and 495 MPa tensile strength.

Lifting spreader modeling is built using the shell model. A shell is a material property with the thickness of the cross section geometry and the combined behavior of the membrane and the bending plate. In dynamic analysis, the mass of the structure is used to calculate the force of inertia. The mass contributed by the Shell element is likened to the element’s joint. No inertia effect is considered in the element itself. Dead load can be activated with the own weight of all elements in the model. For a Shell element, the weight itself is evenly distributed through the plane of the element. The amount of weight alone is the same as the density multiplied by the surface area and thickness. When an element of stress occurs in plane xy, only x and y experience stress while \( \sigma_z = 0 \) dan \( \tau_{zx} = \tau_{xz} = \tau_{yz} = \tau_{yz} = 0 \).
Then the transformation of the maximum and minimum Principal Stress equations is as follows:

\[
S_{\text{Max}} = \frac{S_{11} + S_{22}}{2} + \sqrt{\left(\frac{S_{11} - S_{22}}{2}\right)^2 + S_{12}^2} \tag{2}
\]

\[
S_{\text{Min}} = \frac{S_{11} + S_{22}}{2} - \sqrt{\left(\frac{S_{11} - S_{22}}{2}\right)^2 + S_{12}^2} \tag{3}
\]

Benchmarking using FEM SAP2000 with a dynamic method of 90 percent mass participation ratio, a one-way sinusoidal time history of 1g acceleration with damping 2 percent and load 180 tons divide to 4 point at cord location. It found critical points as shown below:

The biggest number $S_{11}$ is 168 MPa was found (Figure 5). The stress is area number 4125 located right in the weld area. Lifting spreader experiences fluctuation stress without reverse stress, hence only $S_{\text{Max}}[2]$ and $S_{\text{Min}}[3]$ to be taken as reference data. A 180 tons data on $S_{11}$, $S_{22}$, $S_{12}$ and $S_{\text{Max}}$, for critical shell areas can be seen at Table 1.

Comparing result of static analysis from equation [1] of dynamic stresses analysis is much greater 169.1MPa. Furthermore, the data taken to calculate fatigue life is data obtained from FEM software, assuming variant stress as alternating value stress are considered a cyclical process. Also benchmarking analysis to be applied with load 105 tons, 125 tons, 150 tons.

**Table 1.** Shell Node $S_{\text{Max}}$ for load 180 tons  

| Area | ShellType | M11Top | M22Top | M12Top | Smax |
|------|-----------|--------|--------|--------|------|
| 4125 | Shell-Thin| 79.26  | -9.5   | -8.76  | 31.1 |
| 4125 | Shell-Thin| 79.26  | -9.5   | -8.76  | 31.1 |
| 4125 | Shell-Thin| 168.85 | 71.8   | 4.59   | 169.1|
| 4125 | Shell-Thin| -2.05  | 3.4    | -2.83  | 5.7  |
| 4125 | Shell-Thin| 29.26  | -6.4   | -6.76  | 31.1 |
| 4125 | Shell-Thin| 70.62  | -9.5   | -8.52  | 71.8 |
| 4125 | Shell-Thin| 168.85 | 71.8   | 4.59   | 169.1|
| 4125 | Shell-Thin| -2.06  | 3.4    | -2.83  | 5.7  |

**Table 2.** $S_{\text{Max}}, S_{\text{Min}}$ half cycle

| Load Type | Smax | Smin | Half Cycle |
|-----------|------|------|------------|
| 180       | 31.1 | 71.8 | 169.1      |
| 150       | 25.9 | 59.9 | 141.4      |
| 125       | 21.6 | 50   | 118.4      |
| 105       | 18.2 | 42.1 | 100        |

2.2. *Rainflow counting method*

In most laboratory fatigue testing, the specimen is loaded so that stress is cycled either between a maximum and minimum tensile stress and specified level of compressive stress. The latter of the two,
considered a negative tensile stress. Applied stress $\sigma_m$ is the algebraic average of the maximum and minimum stress in one cycle. The stress amplitude, $\sigma_a$ is one-half the range stress\[4].

$$\sigma_m = \frac{\sigma_{\text{max}} + \sigma_{\text{min}}}{2} \quad (4)$$

$$\sigma_a = \frac{|\sigma_{\text{max}} - \sigma_{\text{min}}|}{2} \quad (5)$$

Rainflow counting has become the most widely accepted method for processing of random signals for fatigue analysis, and testing has demonstrated good agreement with measured fatigue lives when compared to other counting algorithms. The concept was first developed by Matsuishi and Endo, where the identification of cycles was likened to the path by rain running down a pagoda roof\[5].

The $\sigma_{\text{max}}$ and $\sigma_{\text{min}}$ stress data obtained from the Rainflow process. Because the critical high stress on weld position, $\sigma_{\text{max}}$ and $\sigma_{\text{min}}$ are multiplied by the concentration factor and sensitivity Notch (Kf) 1.06. Then the amplitude stress ($\sigma_A$) and midrange stress ($\sigma_m$) data are gained. This data used as reference in S-N Curve.

Figure 5. Stress graph of $\frac{1}{2} \sin$ for 4 type

3. Analysis and Discussion

3.1. S-N Curve and Gerber Method

The specification of the St42 material is equivalent to ASM grade (a) 1015 with ultimate tensile stress (Sut) 415 MPa and the endurance limit ($S_e'$) is 40% Sut, 166 MPa. Because this lifting spreader experiences repetitive dynamic load, the calculated voltage endurance cycle $S_e = k_a k_b k_c k_d k_f e' S_e$ $S_e$ is 70.8 MPa.
With the S-N curve method, the number of N cycles is obtained by

$$N = \left( \frac{S_f}{a} \right)^{1/b} \quad (6)$$

$$S_f = \text{Failure Stress}$$

$$a = \frac{(f \sigma_{ut})^2}{\sigma_e} \quad (7)$$

$$b = -\frac{1}{3} \log(f \sigma_{ut}/\sigma_e) \quad (8)$$

Based on the above formulation, the value of a is 1968.7 MPa and the value of b is -0.2406. The failure criterion approach uses the Gerber formulation, which is by finding the amplitude stress of $S_a$:

$$S_a = s^{2\sigma_{ut}} \left\{ -1 + \sqrt{1 + \left( \frac{2\sigma_e}{\sigma_{ut}} \right)^2} \right\} \quad (9)$$

$$r = \frac{s_{a}}{\sigma_{m}} \quad (10)$$

From this formulation with the value of r is 0.9607 then the $S_a$ value is 68.75 MPa. With the maximum amplitude stress value ($\sigma_a$) in the cycle process 1-8-1 ‘is 88.08 MPa is greater than $S_a$ 68.75 MPa, the failure cycle stress ($S_f$) can be obtained from the formulation below:

$$S_f = \frac{\sigma_a}{(1-r^{2\sigma_{ut}})} \quad (11)$$

Then the value of $S_f$ is 92.59 Mpa above value endurance limit 70.8 MPa. So that the number of N values from equation [6] is 329564 times or 3.29x10^5 times. Whereas in the cycle process 2-3-2 ‘, 4-5-4’ and 6-7-6 ‘because each $\sigma_a$ is smaller than $S_a$, then the value of $S_f$ is equal to $\sigma_e$. So that the number of N values is very large or infinite because it is under endurance limit state.

3.2. Number of N cycles of swing Lifting Spreader and Palmgren-Miner rule.

Information from site, transformer displacement for a weight of 180 tons ranges in a year about 5 or 6 times. And the transfer process using spreader lifting is done in a super careful manner, even the height of the transformer lift from the floor is only around 10 cm. And fast swings are avoided at the time of transfer. Based on this information, it can be assumed that light swings occur 1 times per second or 1 Hz swing frequency. If the transfer process takes around 20 minutes, the swing cycle is 1200 times. And in a year the number of swing cycles with a total of 6 transformers is 7200 swing cycles ($n_i$).

Palmgren Miner formulation data is a comparison of the number of cycles $n_i$ occurs with the number of $N_i$ permits. With i is the variation of each load that occurs. The total sum must be less than 1. Near value 1 tend to failure.
\[ \sum_{i=1}^{l} D = \sum_{i=1}^{l} \frac{n_i}{N_i} < 1 \quad (12) \]

4. Conclusions and Recommendations

Based on the data above, the lifting spreader lifespan can be determined by comparing the calculation of N fatigue cycles from equation [6]. \( N \) total 329564 divided by the swing frequency in a year from Lifting Spreader in the case of 180 tons load transfer, which is 7200 swing cycles. \( Y_i = \frac{N_i}{n_i} = \frac{329564}{7200} = 45.7 \) years. The researcher recommends using Spreader Lifting, the capacity is reduced by 10% so that the Sf value is below the Endurance Limit. So it can be used for a longer time.

5. References

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